ISSN: 2277-9655
Scientific Journal Impact Factor: 3.449
(ISRA), Impact Factor: 1.852

IJESRT

INTERNATIONAL JOURNAL OF ENGINEERING SCIENCES & RESEARCH TECHNOLOGY

Design of LPG Refrigeration System and Comparative Energy Analysis with Domestic Refrigerator

Ibrahim Hussain Shah [1], Kundan Gupta*[2]

[1] Assistant Professor, Mechanical Engineering Department,Institute of Engineering and Technology, Devi Ahilya Vishvavidhyalaya,Indore (M.P.)

[*2] M.E. Student (Design and Thermal), Mechanical Engineering Department, Institute of Engineering and Technology,Devi Ahilya Vishvavidhyalaya, Indore (M.P)

gupta.kundan660@gmail.com

Abstract

Supply of continuous electricity is still not available in several areas of the country and the world. At such places, this work will be helpful for refrigreration of food, medicines, etc... In this work we have investigated the performance of a refrigerator based on liquefied petroleum gas (LPG) refrigerant since LPG is locally available and is easy to transport anywhere. LPG is a byproduct in petroleum refineries and comprises of 24.4% propane, 56.4% butane and 17.2% isobutene which have very low boiling point (lower than 0 ºC). The use of LPG for refrigeration purpose can be environment friendly since it has no ozone depletion potential (ODP). Usually LPG is used as a fuel for cooking food in houses, restraurants, hotels, etc.. and the combustion products of LPG are CO_2 and H_2O.

In this project we have designed and analysed a refrigerator using LPG as refrigerant. LPG is available in cylinders at high pressure. When this high pressure LPG is passed through the capillary tube of small internal diameter, the pressure of LPG is dropped due to expansion and phase change of LPG occurs in an isoenthalpic process. Due to phase change from liquid to gas latent heat is gained by the liquid refrigerant and the temperature drops. In this way LPG can produce refrigerating effect for a confined space.

From experimental investigations, we have found that the COP of a refrigerator which uses LPG is higher than a domestic refrigerator.

Keywords: LPG Refrigeration, LPG, Capillary tube, Evaporator, COP, VCR, Refrigerating Effect.

Introduction

Although government agencies are not able to continuously supply a major portion of electricity in both the urban as well as in rural areas. Still the people in these regions require refrigeration for a variety of socially relevant purposes such as cold storage or storing medical supplies and domestic kitchens this project has the novelty of using LPG instead of electricity for refrigeration. This solution is convenient for refrigeration in regions having scares in electricity.

It works on the principle that during the conversion of LPG into gaseous form, expansion of LPG takes place. Due to this expansion there is a pressure drop and increase in volume of LPG that results in the drop of temperature and a refrigerating effect is produced. This refrigerating effect can be used for cooling purposes. So this work provides refrigeration for socially relevant needs as well as replaces global warming creator refrigerants.

While going through the literature review in LPG refrigeration system, Conventional VCR(Vapour Compression Refrigeration System) uses LPG as refrigerant and produced the refrigerating effect . But in our proposed very simple type of refrigeration system in which the high pressure LPG is passing through a capillary tube and expands. After expansion the phase of LPG is changed and converted from liquid to gas and then it passes through the evaporator where it absorbs the heat and produces the refrigerating effect. After evaporator it passes through the gas burner where it burns.

Design of LPG Refigeration System

There are main four parts in this system

1. Copper Tubes (For carrying LPG cylinder to filter before capillary)
2. Capillary tube
3. Valves (Gas supply control valves)

ISSN: 2277-9655
Scientific Journal Impact Factor: 3.449
(ISRA), Impact Factor: 1.852

4. Evaporator

1. Copper Tubes

Air-Conditioning and Refrigeration Systems—**Copper** is the preferred material for use with most refrigerants. Because of its good heat transfer capacity as well as corrosion resistance and cheaper in cost.As for all materials, the allowable internal pressure for any copper tube in service is based on the formula used in the American Society of Mechanical Engineers Code for Pressure Piping (ASME B31): [**10**]

$$P = 2S\,(t_{min} - C)/\,D_{max} - 0.8\,(t_{min} - C)$$

Where:

P = allowable pressure, bar

S = maximum allowable stress in tension, bar

t_{min} = wall thickness (min.), in mm

D_{max} = outside diameter (max.), in mm

C = a constant for copper tube, because of copper's superior corrosion resistance, the B31 code permits the factor C to be zero. Thus the formula becomes:

$$P = 2St_{min}/D_{ma} - 0.8t_{min}$$

According to the pressure 100 psi the tube outside diameter is become = 7 mm and the thickness of the tube is = 1.5 mm.

2. Capillary tube

An analytical computation of length of capillary tube

The fundamental equations applicable to the control volume bounded by points 1and 2 in fig. are

1. Conservation of mass
2. Conservation of energy
3. Conservation of momentum

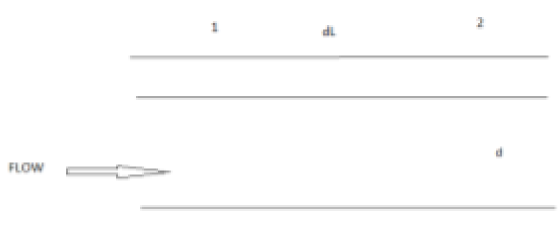

Fig.1 Small elementary length of capillary tube

The equation relating state and conditions at points 1and 2 in a very short length of capillary tube in the figure is written using following notions [4].

A: Cross sectional area of inside of tube, m²

D: ID of tube, m.

f: friction factor, dimensionless

h: enthalpy, kJ/kg.

h_f : enthalpy of saturated liquid , kJ/kg

h_g : enthalpy of saturated vapour, kJ/kg

ΔL: length of increment, m.

P: pressure, Pa

Re: Reynolds No., VD/Ʊ

v: specific volume of m³/kg

v_f : specific volume of saturated liquid, m³/kg

v_g: specific volume of saturated vapour, m³/kg

V: velocity of refrigerant, m/s

w: mass flow rate, kg/s

x: dryness friction

μ: Viscosity, p_a×s

μf: viscosity of liquid, pa×S

μ g: viscosity of Vapour, p_a×s

For calculation of length of capillary tube we have used the following relations and find out the length. The equation of conservation of mass is as follows

$w = V_1A/\,v_1 = V_2A/\,v_2$... (1)

or

$w=V_1/\,v_1 = V_2/\,v_2$ ... (2)

The conservation of energy gives

$1000\,h_1 + V^2_1/\,2 = 1000\,h_2 + V^2_2/\,2$... (3)

This assumes negligible heat transfer in and out of system. The momentum equation in words states that the difference in forces applied to the element because of drag and pressure difference on opposite ends of the element equals that is needed to accelerate the fluid [6].

$[(p_1-p_2) - f\Delta\ L/D\ V_2/\,2v]\ A = w\,(V_1-V_2)$ ..... (4)

As the refrigerant flows through the tube, its pressure and saturation temperature progressively drop and the fraction of vapour .x. continuously increases. At any point

$h = h_f\,(1-x) + x\,h_g$.... (5)

And

$v = v_f\,(1-x) + x\,v_g$..... (6)

The quantities of equation (4) V, v and f all change as refrigerant flows from point 1 to 2. Simplifying using equation (2)

$f\,\Delta L/D.\ V_2/\,2v = f\,\Delta L/D\ V/\,2\ w/A$...... (7)

In the calculation to follow, V used in equation (7) will be mean velocity

$V_m = V_1 + V_2\,/\,2$..... (8)

The friction factor with turbulence is

$F= 0.33/Re^{\,0.25} = 0.33/\,(VD/\,\mu\ v)^{\,0.25}$... (9)

The viscosity in two phase flow is given by

$\mu = \mu_f\,(1-x) + x\,\mu_g$.... (10)

The mean friction factor fm applicable to incremental length 1-2 is

$f_m = f_1+f_2/2 = [0.33/Re_1^{0.25} + 0.33/Re_2^{0.25}]/\,2$.. (11)

ISSN: 2277-9655
Scientific Journal Impact Factor: 3.449
(ISRA), Impact Factor: 1.852

The essence of the analytical calculation is to determine the length ΔL between points 1-2 as shown in fig. for a given reduction in saturation temperature of the refrigerant. The flow rate and other conditions at point 1 are known and for a required selected temperature at point 2, The Remaining conditions at point 2 and ΔL would be computed in the following steps:

1. Temperature t_2 selected
2. p_2, h_{f2}, h_{g2}, v_{f2}, and v_{g2} are computed, all being function of temperature (or pressure).
3. Combination of equation (2) and (3) gives
$1000\ h_2 + v^2{}_2/\ 2\ (w/A)2 = 1000\ h_1 + v^2/\ 2$... (12)

Substituting equations (5) and (6) into (12)
$1000\ h_{f2} + 1000(h_{g2} - h_{f2})\ x + [\{v_{f2} + (v_{g2} - v_{f2})\ x\}^2 (w/A)^2] = 1000\ h_1 + V_1{}^2/\ 2$...... (13)

In equation, all quantities being knows except x, which could be solved by quadratic equation,
$X = [-b + \sqrt{b2-4ac}]/2a$.... (14)

Where,
$a = (v_{g2} - v_{f2})2\ (w/A)^2 \times 1/2$
$b = 1000(h_{g2} - h_{f2}) + v_{f2}\ (v_{g2} - v_{f2})\ (w/A)$ and
$c = 1000(h_{f2} - h_1) + (w/A)^2\ 1/2\ v_{f2}\ 2 - V_1{}^2/2$
Properties of LPG at 10.27 bars [16]
h_{f1} = enthalpy of saturated liquid = 169.1kJ/kg
h_{g1} = enthalpy of saturated vapour = 498.0kJ/kg
v_{f1} = specific volume of saturated liquid = 2.050×10^{-3} m³/kg
v_{g1}= specific volume of saturated vapour = 0.0448m³/kg
Properties of LPG at 1.67 bars
h_{f2} = enthalpy of saturated liquid = 22.9kJ/kg
h_{g2} = enthalpy of saturated vapour = 435.0kJ/kg
v_{f2} = specific volume of saturated liquid = 1.763×10^{-3} m³/kg
v_g= specific volume of saturated vapour = 0.2585m³/kg
w = V/v
V= volume flow rate = 1.1liter/ hr
w =9.45×10^{-4} Kg/sec
From this calculations the length of capillary tube is = 2.97m

3. Valves
In this system we have used two flow control valves of globe type of 4 mm of internal diameter.

4. Evaporator
Evaporators are heat exchangers with fairly uniform wall temperature employed in a wide range of HVAC-R products, spanning from household to industrial applications. In general, they are designed aiming at accomplishing a heat transfer duty at the penalty of pumping power. There are two well-established methods available for the thermal heat exchanger design, the log-mean temperature difference (LMTD) and the effectiveness/number of transfer units (e-Ntu) approach (Kakaç and Liu, 2002; Shah and Sekulic, 2003). The second has been preferred to the former as the effectiveness, defined as the ratio between the actual heat transfer rate and the maximum amount that can be transferred, provides a 1st-law criterion to rank the heat exchanger performance, whereas the number of transfer units compares the thermal size of the heat exchanger with its capacity of heating or cooling material. Furthermore, the e-Ntu approach avoids the cumbersome iterative solution required by the LMTD for outlet temperature calculations. [14]
In general, evaporators for refrigeration applications are designed considering the coil flooded with two-phase refrigerant, and also a wall temperature close to the refrigerant temperature (Barbosa and Hermes, 2012), so that the temperature profiles along the streams are not constant, in these cases, the heat transfer rate if it is calculated from: [13]
$Q = m.c_p\ (T_o - T_i) = \varepsilon.m.c_p\ (T_s - T_i)$

Where m is the mass flow rate, T_i, T_o and T_s are the inlet, outlet and surface temperatures, respectively, $Q=h \times A_s\ (T_s-T_m)$ is the heat transfer rate, T_m is the mean flow temperature over the heat transfer area, A_s, and ε is the heat exchanger effectiveness, calculated from (Kays and London, 1984):
$e = 1 - \exp(-NTU)$

Where NTU is the number of transfer units. We have selected the plate and tube type evaporator because it provides a gentle type of evaporation with low residence time. It also preserves the food and othe products from bacterial attack. It requires low installation cost.

Design calculations for evaporator
The evaporator has following dimentions:
Length = 325 mm, Bredth = 265 mm and Height = 135 mm
The evaporator is made from six plywood sheets of 3mm thickness which enclose six thermocol sheets of 10 mm thickness. The areas for these sheets are as follows:
Area1 = 265×135 = 0.03578 m^2,
Area2 = 265×325 = 0.08612 m^2,

ISSN: 2277-9655
Scientific Journal Impact Factor: 3.449
(ISRA), Impact Factor: 1.852

Area3 = 265×135 = 0.03578 m^2,
Area4 = 265×325 = 0.08612 m^2,
Area5 = 325×135 = 0.04388 m^2,
Area6 = 325×135 = 0.04388 m^2,
Thermal conductivity of plywood k_p = 0.12 W/m.k
Thermal conductivity of thermo coal k_t = 0.02 W/m.k
Thickness of plywood = 3 mm
Thickness of thermo coal = 10 mm
Temperature of atmosphere = 35 ^{0}C = 298 K
Temperature of evaporator = -9 ^{0}C = 264 K

Heat flow from area 1 due to conduction
$Q_1 = (T_a-T_e)/ (Rth_p + Rth_t)$
$= (T_a-T_e)/ ((L_p/K_P.A) + (L_t/K_t.A))$
$= (294-264)/ (0.698+13.97)$
$= 2.317W$

Heat flow from area 2 due to conduction
Q_2 = 5.58 W, Q_3 = 2.32 W, Q_4 = 5.58 W, Q_5 = 2.84 W
Q_6 = 2.84 W
Total heat flow from all areas due to conduction = 21.47 W

Heat flow from evaporator due to convection
Inside heat transfer coefficient = 30 W/m^2.K
Outside heat transfer coefficient = 10 W/m^2.K
Rate of heat transfer Q [12]
$Q = U.A. (T_a-T_e)$
The overall heat transfer coefficient
$1/U = (1/U_o) + (L_p/k_p) + (L_t/k_t) + (1/U_i)$
$1/U = 0.649$
U = 1.54 W/m^2.K
Rate of heat transfer from area 1
$Q_1 = 1.54×0.03578(298-264)$
$= 1.873W$
Q_2 = 4.50 W, Q_3 = 1.873 W, Q_4 = 4.50 W, Q_5 = 2.29 W
Q_6 = 2.29 W
Total heat flow from all areas due to convection = 17.326 W
Heat transfer due to radiation Q
$Q = \sigma T^4$
$= 5.67× 10^{-8}(35-(-9.3))^4$
$= 0.21W$
Total heat flow from evaporator due to conduction, convection and radiation Q_t
$Q_t = 21.47+17.326+0.21$
$=39.006W$

The LPG Refrigeration Cycle
LPG Gas Cylinder:
From the LPG gas cylinder of 14.5 kg, LPG flows through the pipe and reaches to the capillary tube. LPG gas pressure is approximate 12.41 bars.

Capillary Tube:
As the capillary tube, capillary tube downs the pressure up to less than 1.2 bars.

Evaporator:
In the evaporator LPG is converted into the vapor from with low pressure. After passing through the evaporator low pressure and temperature LPG vapor absorbs heat from the chamber system.

Gas Burner:
After performing the cooling effect, low pressure LPG gas goes into the burner where the burns. As we know whenever the fluid flow through the narrow pipe there is a pressure drop. The amount of pressure drop in our system is calculated. [10]

From the Darcey-Weisbach equation, the pressure drop in the refrigerant piping is calculated for 13 feet length tube is 0.23 in terms of equivalent length.

Basic Experimental Setup of LPG refrigeration system
The basic components in this system are shown in set up diagram and the changes in thermodynamics properties of the fluid flowing (LPG) is shown in the systems line diagram.


Fig.2 Experimental set up


Fig. 3 Experimental set up

ISSN: 2277-9655
Scientific Journal Impact Factor: 3.449
(ISRA), Impact Factor: 1.852

The experiment of this project was done on 3 April, 2014 at 1:00 p.m. and readings were taken at 10 minute's interval, for 1 hour which is as shown in table 1 below:

Table.1: Experimental Readings

Time (min)	Inlet pressure (bar)	Outlet pressure (bar)	Water temp (ºC)	Evaporator temp (ºC)
10	5.516	1.45	25.2	18.0
20	5.415	1.43	17.2	12.2
30	5.310	1.36	14.2	11.7
40	5.210	1.35	10.1	6.3
50	5.120	1.30	5.4	-3.9
60	5.020	1.30	0.3	-9.3

Again we were taken reading on this project on second day on 4 April, 2014 at 1:00 p.m. and readings were taken at 10 minute's interval, with same cylinder for 1 hour which is as shown in table 2 below:

Table.2: Experimental Readings

Time (min)	Inlet pressure (bar)	Outlet pressure (bar)	Water temp (ºC)	Evaporator temp (ºC)
10	5.019	1.30	25.10	18
20	5.000	1.28	17.24	12.4
30	4.910	1.26	14.25	11.3
40	4.820	1.23	10.11	5.3
50	4.690	1.20	5.44	-3.4
60	4.520	1.20	0.33	-9.1

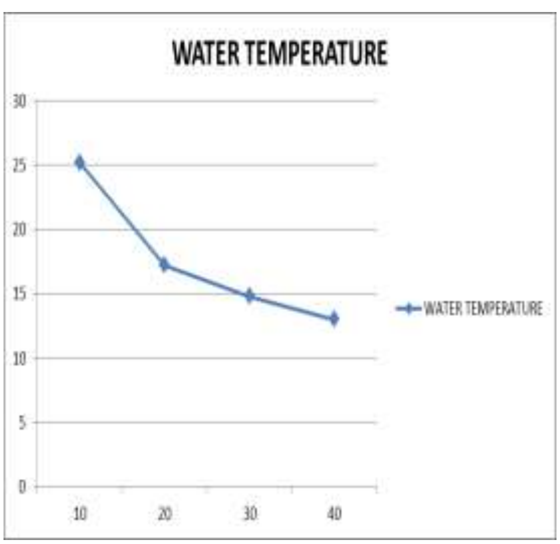

Chart.1:Water temperature v/s time (min)

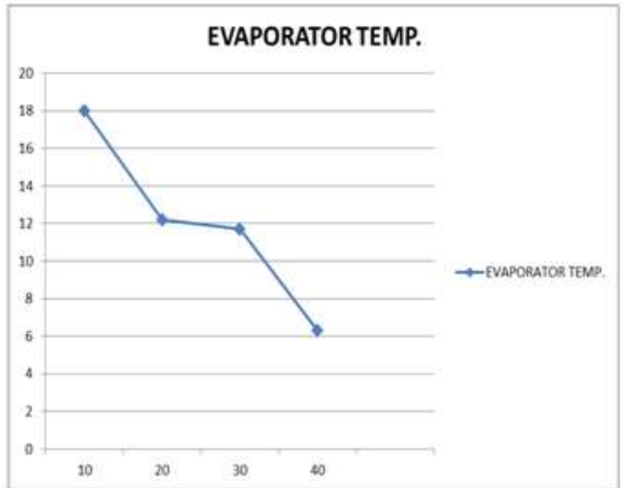

Chart 2 Evaporator temp. v/s time(min)

This is the p-h diagram of LPG refrigeration system

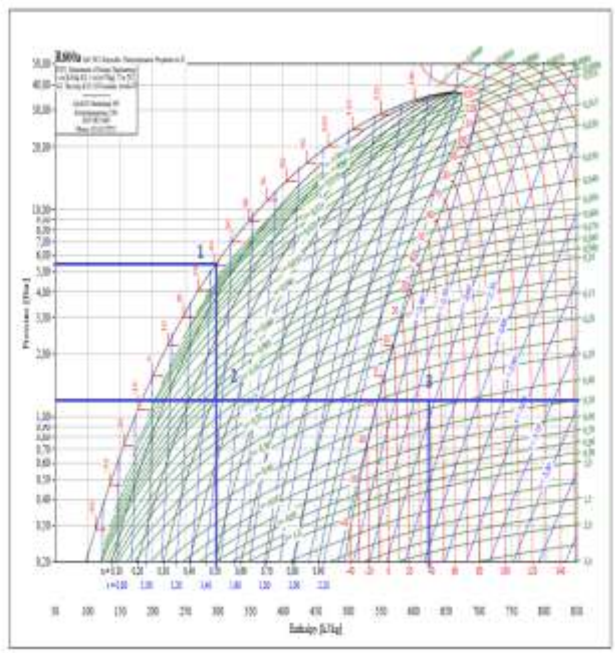

Fig.4 p-h diagram of LPG Refrigerator [17]

Size of refrigerator: - 335×265×135 mm³
Initial temperature of water: - 30^{0}C
Initial temperature of evaporator: - 33^{0}C
Specific heat of LPG vapor is 1.495kJ/KgK

Refrigerating effect [1]
The properties of LPG at 5.516 bars are
Enthalpy h_1 = 430.3 kJ/Kg
Temp. t_1= 4 ^{0}C
The properties of LPG at 1.316 bars are
Enthalpy h_3 = 107.3 kJ/Kg

ISSN: 2277-9655
Scientific Journal Impact Factor: 3.449
(ISRA), Impact Factor: 1.852

Temp. t_3= -30 ^{0}C
Heat extracted from evaporator in 1 hour (Q_{eva}) = Heat gained by LPG (Q_{LPG})
(Q_{eva}) = Heat extracted from (water + surrounding air inside of evaporator +container + leakage)
m_w = mass of water =6.5kg
c_{pw} = specific heat of water=4180J/kg.K
$(\Delta T)_W$ =28.3 ^{0}C
m_c =mass of container =1.30kg
c_{pc}= specific heat of aluminium container = 903 J/kg.K
$(\Delta T)_c$ =28.3 ^{0}C
x_{LPG} = Dryness fraction of LPG from graph =0.5

$$(Q_{eva}) = Q_{evap} + Q_{air} + Q_{cont} + Q_L$$
$$= m_w c_{pw}(\Delta T) + m_a c_{pa}(\Delta T) + m_c c_{pc}(\Delta T) + Q_L$$

We have taken 6.5 kg of water in an aluminium container of weight 1.30 kg.
Since there is very less amount of air so it is neglected.

$$= 6.5\times4180\times28.7 + 0 + 1.3007\times903\times28.7$$
$$= 0.81348 \text{ MJ}$$

Heat gained by LPG (Q_{LPG}) = Latent heat gain $(Q_L)_{LPG}$ +Sensible heat gain$(Q_{Sen})_{LPG}$
$= m_{LPG} .x_{LPG} .h_{fg} + m_{LPG} .c_{pLPG} . (T_{sup}-T_{sat})$
$=9.45\times10^{-4}\times0.5\times375\times10^3\times3600+9.45\times10^{-4}\times1.67\times(-9.3-30)$
= 861151.662J/hr = 0.862MJ/hr

So the refrigerating effect is = h_3-h_2
= 630.3-307.3
= 323kJ/Kg

For work input we have a LPG cylinder of 14.5 Kg. so the work input is amount of energy required for filling of 1 cylinder. A typical LPG bottling plant has the following major energy consuming [8].
Equipment:-
1. LPG pumps
2. LPG compressors
3. Conveyors
4. Blowers
5. Cold repair facilities including painting
6. Air compressors and air drying units.
7. Transformer, MCC & DG sets
8. Fire fighting facilities
9. Loading and unloading facilities

Some of the LPG bottling plants use a comprehensive monitoring technique for
Keeping track of energy / fuel Consumption on per ton basis. PCRA Energy Audit [8]
1. Consumption = 40×4200=168000kWh
2. For lighting energy consumption= 227340kWh
3. LPG compressor consumption= 153360 kWh

1. Total consumption for LPG pumps
One pump having 40 kW motor and 96 m head or 150cubic meter /hour discharge
Annual operating = 4200 hrs
Annual energy 6 hrs /day in 350 days
= 168000+227340+153360
= 548700kWh
Per day consumption
= 548700/350
=1567.71 kWh
500 cylinders are refilled every day, so per cylinder electricity consumption.
=1567.71/500
=3.1354kWh
For filling of 1 LPG cylinder of 14.5 kg the power input is
= 3.1354kWh
So 1 kg of LPG is
= 3.1354/14.5
=0.2162 kWh
We run the set up for 1 hr
= 0.2162×1000/ (9.45/10000) ×3600
= 63.55W

COP OF THE LPG REFRIGERATION SYSTEM

COP = (h_3-h_2)/w
= (630.3-307.3)/63.55
= 5.08

After finding out the COP of the LPG refrigerator we found out the heat librated by LPG after burning in the burner with the burner efficiency of 92 %.
Heat liberated by LPG Q_L= m×c_v
We have the volume flow rate of LPG is 0.1 liter per min. and the specific volume of LPG at 1.56 bar pressure is 1.763×10^{-3} m^3/Kg.
So mass flow rate of LPG is = $0.0001/1.763\times10^{-3}$
= 0.0567 Kg/min
m = 9.45×10^{-4} Kg/sec
c_v = 46.1 MJ/Kg
Q_L= $9.45\times10^{-4}\times 46.1\times10^3$
= 43.56 W

ISSN: 2277-9655
Scientific Journal Impact Factor: 3.449
(ISRA), Impact Factor: 1.852

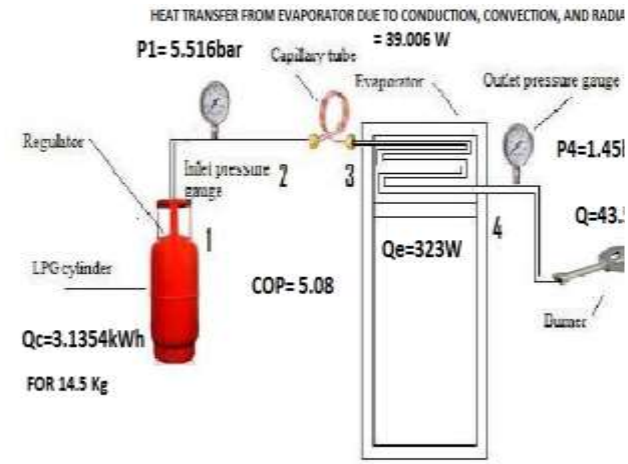

Fig.5 LPG Refrigerator cycle

We have seen in these calculations that the input for the LPG filling is 3.1354 kWh for one 14.5 Kg of cylinder which is equal to 1 unit of electricity.

So we have run the set up for 1 hour and got refrigerating effect as well as the heat from LPG.

Compare with Domestic Refrigerator

COP for a domestic refrigerator using the R134a refrigerant of capacity of 165 litres and a compressor pressure of 10 bars and evaporator pressure of 1.4 bars.

The work done on the compressor = -54 kJ/kg.

The heat absorbed by the evaporator [137 kJ/kg], and that rejected by the condenser [-191 kJ/kg].

The Coefficient of Performance of the refrigerator (COP_R) (defined as the heat absorbed in the evaporator divided by the work done on the compressor - always presented as a positive value even though the work done w_c is negative)

[COP_R = 2.53].

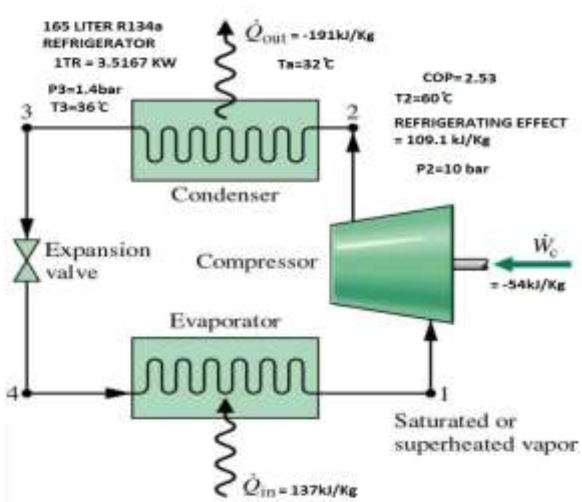

Fig.6 Domestic refrigerator cycle

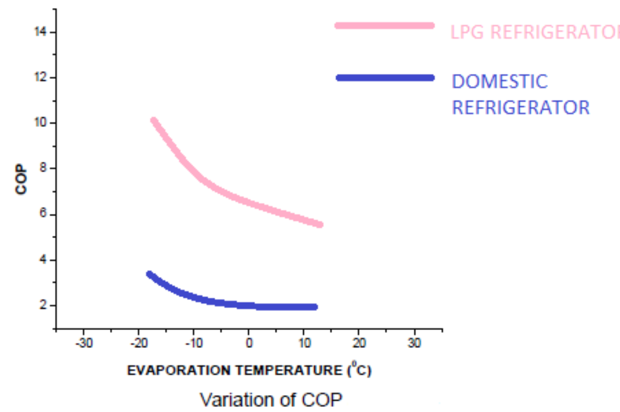

Fig.7

Conclusion

The aim of the LPG refrigerator was to use LPG as a refrigerant and utilising the energy of the high pressure in the cylinder for producing the refrigerating effect. We have the LPG at a pressure of 12.41 bar in Domestic 14.5 kg cylinder equipped with a high pressure regulator and this pressure has reduced up to 1.41 bar with the help of capillary tube. But if we use a low pressure regulator as is the practice in conventional domestic LPG gas stove, the pressure of LPG after the expansion device and before the burner would be different. So we have calculated the refrigerating effect with the help of changes in properties of LPG (pressure, temperature, and enthalpy) before and after the evaporator using high pressure regulator and the amount of refrigerating effect is 323kJ/Kg.

Since we don't have the actual amount of energy that will be consumed in producing 1 Kg of LPG in the refinery and were not available in any of the Energy Audit Report of Refinery, that's why we have taken the energy input from refilling plant only. For energy input we have taken the amount of energy required for refilling 1 Kg of LPG in the bottling plant (PCRA energy audit report, HPCL LPG bottling plant Asauda Bahadurgarh (Haryana) Dec. 2006.) is 0.216 kWh. With this energy input the COP of the LPG refrigerator is 5.08 and it is greater than the domestic refrigerator. But in the future scope the

ISSN: 2277-9655
Scientific Journal Impact Factor: 3.449
(ISRA), Impact Factor: 1.852

result may differ if energy input for 1Kg of LPG production, would be taken from the energy audit report of any refinery.

References

1. *Shank K. Wang, Handbook of air conditioning and refrigeration" Edition.*
2. *A. Baskaran, P. Koshy Mathews, International Journal of Scientific and Research Publications", Volume 2, Issue 9, 1 ISSN 2250-3153, September 2012*
3. *B. O. Bolaji, Investigating the performance of some environment-friendly refrigerants as alternative to R12 in vapour compression refrigeration system", PhD Thesis in the Department of Mechanical Engineering, Federal University of Technology Akure, Nigeria (2008).*
4. *Prashant Sharma, Rahul Sharma, "International Journal of Latest Research in Science and Technology" ISSN (Online):2278-5299 Vol.1,Issue 1 :45-48,May-June(2012)*
5. *ASHRAE, "Thermo physical Properties of Refrigerants", Chapter 20, ASHRAE Fundamental, Inc. Atlanta 20 (2001) 1-67.*
6. *W. F Stoecker., and J. W. Jones, "Refrigeration and Air conditioning", TATA McGraw-Hill pub. Co. Ltd.pp. 264.*
7. *ASHRAE, 2002, "Adiabatic capillary tube selection", Refrigeration Handbook, chapter. 45, pp.45.26-45.30, ASHRAE.*
8. *"PCRA energy audit report", HPCL LPG bottling plant Asauda Bahadurgarh (Haryana) Dec. 2006.*
9. *"Basic statics on Indian petroleum and natural gas" 2006-07.*
10. *Shank K. Wang, "Handbook of air conditioning and refrigeration" page no. 11.14 chapter 11.*
11. *ASHRAE handbook 1998.*
12. *C.P. ARORA, "Hand book of Refrigeration and air conditioning", by page no. 425*
13. *A. Bejan, "The thermodynamic design of heat and mass transfer processes and devices", Heat and Fluid Flow pp.258-276, 1987*
14. *Hermes CJL, "Conflation of e-Ntu and EGM design methods for heat exchangers with uniform wall Temperature", Int. J. Heat and Mass Transfer, pp.3812-3817, 2012.*
15. *J R Barbosa, C Melo, CJL Hermes, PJ Waltrich, "A Study of the Air-Side Heat Transfer and Pressure Drop Characteristics of Tube-Fin 'No-Frost' Evaporators", Applied Energy 86, pp.1484-1491, 2009.*
16. *MICHAEL J. MORAN, "Properties of LPG from fundamental of engineering thermodynamics".*
17. *W. C. Reynolds, M.J. Skovrup & H.J.H Knudsen, "Thermodynamics properties in SI DTU Department of energy engineering" .*

Author Biblography

	**Ibrahim Hussain Shah** Assistant Professor, Mechanical Engineering Department, Institute of Engineering and Technology, Devi Ahilya Vishvavidhyalaya, Indore (M.P.), India his_hussain@rediffmail.com
	**Kundan Gupta** M.E. Student (Design and Thermal), Mechanical Engineering Department, Institute of Engineering and Technology, Devi Ahilya Vishvavidhyalaya, Indore (M.P), India gupta.kundan660@gmail.com