Design and Development of Portable Vapour Compression Refrigerator

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Abstract— Today we can preserve the food in refrigerator by keeping it at desired low temperature. But when we went for a picnic or for a long journey we require to preserve the food and maintain the beverages at low temperature. In order to maintain the optimal temperature range of 2 to 8^o Celsius for food preservation, these region need reliable refrigeration system. The most of the refrigerator products in the market are not suitable because they are large in size and bulky. In this attention is made on mini refrigerator which are smaller in size and portable. so, we can use this unit to transport the beverages over long distances in remote areas by storing them at low temperature. The aim of this study is to design the components of the mini VCRC one by one and to analyse the cycle. The effect of heat transfer in condensation and evaporation processes has been investigated. Moreover, an isentropic, rotary compressor has been selected for the system. R-134a has been considered as the refrigerant throughout the study. Firstly, the evaporator part will be presented in the report. The design procedure and the dimensions of the evaporator will be mentioned in the other chapter of the report. Also, the design of cabinet is done and the heat losses from each side of the cabinet are calculated to decide the power input required for the stated system.

Key words: Refrigerator, Vapour Compression Cycle, R-134a

I. Introduction

A refrigeration is a cooling appliance comprising a thermally insulated compartment and mechanism to transfer heat from it to the external environment, cooling the contents to a temperature below ambient. Refrigerators are extensively used to store foods which deteriorate at ambient temperatures; spoilage from bacterial growth and other processes is much slower at low temperature. Before the invention of the refrigerator, icehouses are used to provide cold storage for most of the year. After that, the first artificial refrigeration system was demonstrated by William Cullen at the University of Glasgow, Scotland in 1748. However, no reports are available for the proof of any practical use of his findings. In 1805, Oliver Evance designed refrigerator based on a closed cycle of compressed ether, represented the first effort to used simple vapour instead of vaporizing a liquid. In 1834, the practical refrigerating machine was built by Jacob Perkins using a vapour compression cycle. After that, in 1857, James Harrison introduced vapour compression refrigeration to the brewing and meat packing industries. A first fully hermetic refrigerating unit was invented by Adde Audffren in 1903. The invention of the Electrolux refrigeration system without any moving parts was done by Baitzar von Platen and Carl Munters in year 1922. Start of the 20th century, all refrigerators use the vapour compression refrigeration cycle until now.

There are several basic refrigeration techniques:
1) Ice box
2) Cold air system
3) Magnetic cooling
4) Sterling refrigeration system
5) Pulse tube refrigeration
6) Vapour compression refrigeration
7) Vapour absorption refrigeration
8) Thermoelectric cooling
9) Thermo-acoustic refrigeration

Technology	Coefficient of Performance	Prospect for Competing with Vapour Compression Cycle
Vapour Compression Refrigeration	3.5-4	-
Vapour Absorption Refrigeration	0.6-0.7	Good
Thermoelectric	0.6	Fair
Magnetic	Up to 3	Good
Stirling Refrigerator	2-3	Fair
Pulse Tube Refrigertor	2-3	Moderate

Table 1: Comparison between Different Refrigeration Techniques

II. Literature Review

Hisham Ettouney (2006) [1] presented a comprehensive design model of the single effect mechanical vapour compression process. He found the evaporator dimensions, demister dimensions, dimensions of the non-condensable gases venting orifice and capacity of the vacuum system.

The behaviour of performance parameters of a simple vapour compression refrigeration system were studied by J. K. Dabas et al.(2011)[2] They also investigated the effect of different lengths of capillary tube. They concluded that larger capillary tube decreases the tendency of refilling of evaporator but offers less evaporator temperature effective in lower range of refrigeration temperature.

S. Mancin, C. Zilio and G. Righetti, L. Rossetto (2013)[3] presented the experimental analysis of small scale refrigeration system. They used water cooled miniature scale refrigeration system with R134a as a working fluid and implemented a new concept oil free linear compressor prototype.

J.P. Yadav and B.R. Singh (2011)[4] studied and fabricated a prototype model of ice plant. They analysed the model for its cooling capacity assumed per unit mass flow rate of refrigerant.

M. A. Akintunde (2011) [5] presented the validation of the design model for VCR system. He concluded that the model and the experimental data showed the same trend. He noted that the model predicted almost a linear relationship for both refrigeration capacity with the refrigerant charge and the COP with evaporator circulating water while the experimental data justified the same assertion.

K.T. Ooi (2014) [6] formulated a lumped mathematical model for the compressor to predict the overall performance of the machine. He focused on improving the COP of the compressor by improving its mechanical performance.

A. A. Sathe et al.(2008)[7] describe performance measurements on a prototype miniature rotary compressor with a refrigerant R-134a using a compressor load stand based on hot gas bypass design.

Later, Chriac and Chriac(2015) [8] designed a system for about 100 W and design the evaporator as well. Condensation temperature has been selected as 55°C and evaporation temperature was 10°C. R-134a has been usedcas the refrigerant.

Sahoo. K. C, Das. S. N (2014) [9] in this paper develops a more accurate theoretical

Procedure for the design of adiabatic capillary tube of a domestic refrigerator considering a rigorous pressure drop analysis on the refrigerator R-600a while expanding through that tube accompanied with phase change through flash vaporization.

P. K. Bansal, I. McGill and P. M. Lloyd [10] this paper present the effect of capillary tube variation has been investigate with respect to power consumption the stability of the compressor cycle and the refrigerant sub cooling in refrigerator.

Joel BOENG, Claudio MELO (2012) [11] in this paper the aim of this study was to develop a methodology to minimize the energy consumption of household refrigerator focused on the proper choice of the pair capillary tube refrigerant charge, the energy consumption was firstly mapped, by varying the expansion restriction and the refrigerant charge.

A. Design of Cabinet

In design of cabinet, main focus is made on the calculation of thermal losses.

1) Calculation of thermal losses:

According to the selected model, the standard, the climate class to be applied following formula can be applied to calculate thermal losses for each surface of the refrigerator-

$$P=\frac{1}{s}*k*A*\Delta t°$$

Before that calculation we have to mention the dimensions of the refrigerator:

Parameters	Dimensions
Outer Dimensions	270w*320d*470h mm
Inner Dimensions	200w*250d*400h mm
Compressor area	54*54*70 mm
Insulation of refrigerator	35 mm
Refrigerator temperature	2°c-5°c
Compressor temperature	85°c
Condenser Temperature	46°c
Evaporator temperature	0°c
Ambient temperature	40°c

Table 2: Refrigerator Dimensions

Co-efficient of transmission K are as following-

Polyurethane with cyclopentane in densities 30-35 kg/m3, K=0.02-0.025

Magnetic door gasket, K=0.06-0.08

2) Calculation of each part is mention as below:

Sr. no	Part of refrigerator	Total heat loss
1.	Refrigerator door	1.64571 kcal/hr
2.	Compressor upper	0.07776kcal/hr
3.	Refrigerator side	4.114 kcal/hr
4.	Refrigerator back	2.1028 kcal/hr
5.	Refrigerator door gasket	5.9294 kcal/hr
		Σ=13.8689kcal/hr

Table 3.2: Total Heat losses of Refrigerator

Total Transmission losses= 13.8689 kcal/hr = 16.11 W

Power necessary to Freeze 10 kg of water:

$$P=1/24\ (G*C*\Delta t1)+(G*CI)$$

$$P=\frac{1}{24\times 2}\ [(\mathbf{10}\times\mathbf{1}\times\mathbf{36})+(\mathbf{10}\times\mathbf{541.68})]$$

$$P=120.35\text{kcal/hr}$$

This freezing power has to be added to the 13.8689 kcal/hr thermal transmission losses to determine the needed compressor power which in this case is,

=120.35 kcal/hr +13.8689 kcal/hr

=134.2189 kcal/hr = 155 W

Our compressor refrigeration capacity is 350W which is greater than our required freezing power.

Therefore design of cabinet is valid.

B. Vapour Compression Refrigeration Cycle:

The vapour compression cycle of its high index of performance or efficiency, it is most widely used in commercial refrigeration system. A complete vapour compression cycle is shown on the p-v diagram. We can draw this cycle by using the coolpack software. In that we only have to give the condenser and evaporator temperature and the compressor heat losses then after we can directly got the vapour compression cycle on p-h diagram as well as t-s diagram. And get the values of mass flow rate, COP, volume flow rate, work done, heat losses, pressure ratio which we can take the reference for designing the refrigerator system. Designing of evaporator, capillary tube and the different component of refrigerator.

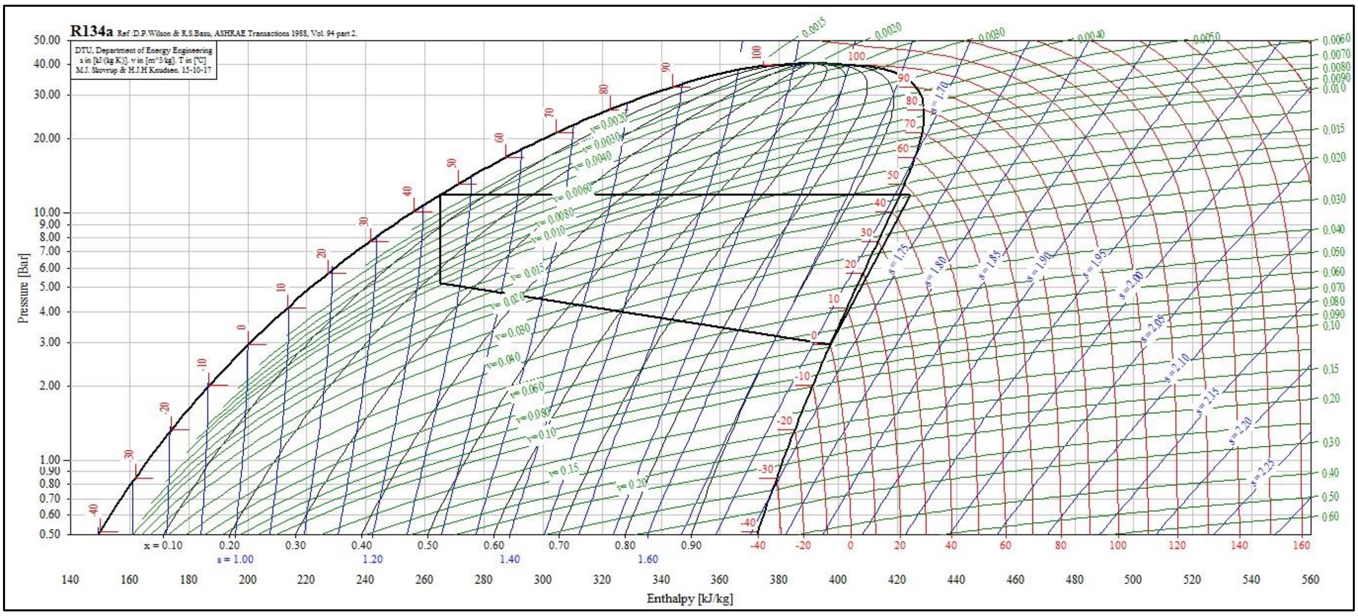

Fig. 1: P-h Diagram of Vapour Compression Refrigeration Cycle by Using Coolpack software

Figure shows the schematic of a standard, saturated single stage vapour compression refrigeration system and an operating cycle on a T-S diagram. The saturated vapour compression refrigeration system consist of mainly four processes that are as follows:

- Process 1-2 : Isentropic compression of saturated vapour in compressor
- Process 2-3 : Isobaric heat rejection in condenser
- Process 3-4 : Isenthalpic expansion of saturated liquid in expansion device
- Process 4-1 : Isobaric heat extraction in the evaporator
- Roll Bond Evaporator :

1) General :

An evaporator is basically a heat exchanger which absorbs the heat from the cabinet and keeps it at the desired low temperature. The refrigerant flowing through the evaporator evaporates after the heat is being absorbed. Now days, the conventional refrigerators use roll bond evaporators.

Roll bond evaporators deliver efficient thermal performance. This roll bond evaporator is fabricated by rolling together two sheets of aluminium applying heat and the pressure during the rolling process such that the two sheets are effectively welding together into a single sheet. By applying special coating called "weld stop" or a chemical ink between the sheets prior to the rolling/welding operation, it is possible to prevent two sheets from welding together in the areas where coating is applied.

Thus by applying the coating in a serpentine pattern, it is possible to create serpentine shaped un-welded region within this welded part. By subsequently applying hydraulic pressure to this un-welded region, it is possible to inflate the un-welded serpentine region to form a serpentine pattern through the plate. One end of the evaporator is connected to the exit of the expansion valve and other end to the compressor inlet. The refrigerant passes through the channel and produces desired refrigeration effect. This roll bond evaporator has an excellent U value in between 6-8 W/m2K.

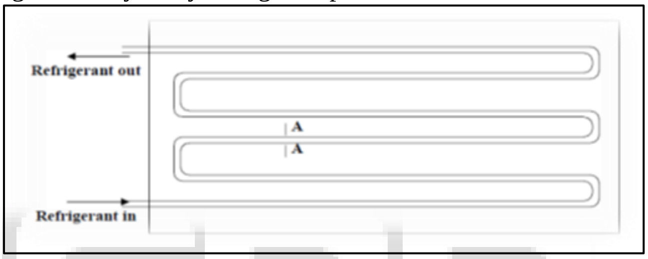

Fig. 2: Roll Bond Evaporator

2) Operational Condition:

Table Operational Conditions of Evaporator

Exit condition	Saturated vapour
Evaporator Temperature	0°C
Ambient temperature	40°C
Evaporator pressure	.2928 MPa
Total heat dissipation	300W

3) Design of Evaporator:

The three heat transfer resistances in evaporator are:

1) Refrigerant side for heat transfer from solid to liquid surface
2) Metal wall
3) Cooled medium side which could be due to air

4) Overall heat transfer Coefficient:

Evaluation of the overall heat transfer coefficient is an important step in the design of the evaporator. The overall heat transfer coefficient of aluminium roll bond evaporator is in between 6-8 W/m2K.

Hence, U= 8 W/m2K.

5) Internal Convection Heat Transfer Coefficient (hi)

Refrigerant Properties at 0°C

Table Refrigerant Properties at 0°C

μg	10.73 μPa.s	μf	271.1 μPa.s
kg	.01179 W/mK	kf	.0934 W/mK
Cpg	.897 kJ/kgK	Cpf	1.341 kJ/kgK
Vg	0.06935 m3/kg	Vf	7.7297*10-4 m3/kg

Refrigerating Capacity= 150 W

Thus, 150=ṁ*(h1-h4)

150 = ṁ*(398.6-256.41)

Therefore, ṁ = 1.126*10-3 kg/s

The commercially available inner and outer diameter of the roll bond evaporator is,

di = 5.7727*10-3 m

d0 = 6.35*10-3 m

Prandtl number = Cpf*µf /kf=3.8923

Reynold's number is given by the formula,

Re = 4*ṁ/Π*di*µf = 916.093

Rem = Re*(1+√Vg/Vf)

Thus, Rem = 9593.33

Then the equation for the Nusselt Number is given by the following correlation,

Nu = .023*Re.8 * Pr.4

Therefore Nu = 60.7275

Nu = hi * d0 /kf

By putting the obtained values in the above equation, we get,

hi = 879.83 W/m^2 K

Hence, the value of internal convection heat transfer coefficient is 879.83 W/m^2 K.

6) Air Side Heat Transfer Coefficient (ho)

The overall heat transfer coefficient is given by,

$1/U = 1/hi + 1/h_o + \Delta x/kw$

For aluminium kw = 202 W/m^2 K

$1/8 = 1/879.83 + 1/h_o + .5775*10\text{-}3/202$

Thus, h_o = 8.0736 W/m ^{2} K

7) Total Heat Transfer Area:

The required area of the evaporator is then given by the equation,

Qe = U*A*Δt

A0 = Qe / U*Δt

Thus A0 =0.4687 m^2

So the total heat transfer area for the evaporator is found to be 0.4687 m^2.

8) Face Area (Af) :

Total heat absorbed by the space is given by,

Qe = ṁa*Cpa*Δt

Therefore, the mass flow rate of the air = 3.7287*10-3 kg/s

As specific heat of the air = Cpa = 1.005 kJ/kgK

Hence face area is given by,

ṁa = ρ*Af * Vf

Thus, Af = 6.617*10-4 m^2 = 6.617 cm^2

C. Capillary Tube:

1) Operating Condition of Capillary Tube

The condition of the refrigerant at the inlet state is liquid state and at the exit is liquid vapour mixture. The operating condition are summarized in Table

Inlet temperature	46^0
Outlet temperature	0^0
Pressure difference	8.975 bar
hg	169.74 kJ/kg
hf	200 kJ/kg
$\mathbf{h_g}$	265.67kJ/kg
Diameter	0.8mm
Refrigerant	R134a

Table 4: Operating Conditions of Capillary Tube

D. Design Calculations:

In the capillary tube design there are two unknown viz. length of capillary tube and diameter of capillary.

Hence, from calculation of evaporator, mass flow rate is

$$\dot{\mathbf{m}} = \mathbf{1.126 \times 10^{-3} kg/s}$$

The cross section area of the capillary tube is

$$A=\frac{\pi}{4}(\mathbf{0.8 \times 10^{-3}})2$$

$$= \mathbf{5.026548 \times 10^{-7} m^2}$$

Let

$$\mathbf{G} = \frac{\dot{\mathbf{m}}}{\mathbf{A}}$$

$$= \frac{\mathbf{1.127 \times 10^{-3}}}{\mathbf{5.0265 \times 10^{-7}}}$$

$$= \mathbf{2.24035 \times 10^3\ {}^{kg\ s^{-1}}/_{m^2}}$$

$$= \frac{\mathbf{u}}{\mathbf{v}}$$

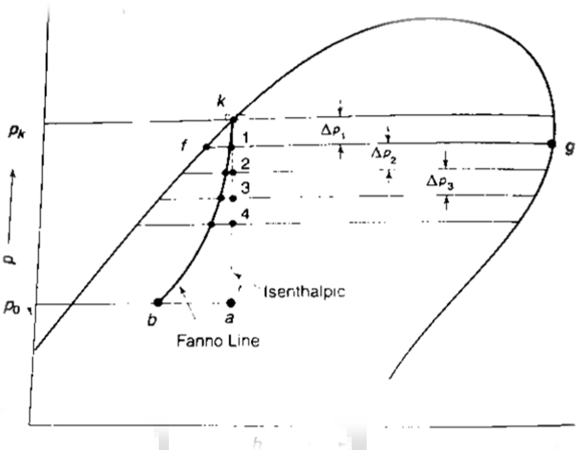

Figure Incremental pressure drops in capillary tube

Y=G/2D

=(2.24035× 〖10〗^3)/(2×(0.8× 〖10〗^(-3)))

= 1.400218 ×106 kgs-1m-3

Z = DG

= (0.8× 〖10〗^(-3)) × (2.24035×103)

=1.79228kgs-1m-1

Δh= h_k-h_1

=-Δ(K.E)

Calculation can now be done for actual fanno-line, starting from hk = 262.56 KJ/kg.

Point 1,

At 400C, from the table of properties for R134a

hf = 256.43 kJ/kg

hfg = 169.74 kJ/kg

vf1 = 0.878 × 10-3 m3/kg

vg1 = 19.9 × 10-3 m3/kg

Hence,

$$\mathbf{x} = \frac{\mathbf{h_k - h_1}}{\mathbf{h_g}}$$

$$\mathbf{x} = \frac{\mathbf{265.67 - 256.67}}{\mathbf{169.74}}$$

= 0.053020

V1 = Vf1 + x(Vg1 – Vf1)

= 0.878 × 10-3 + 0.053020 (19.9 – 0.878) × 10-3

= 1.886 × 10-3 m3/kg

u1 = GV1

= (2.24035 × 103) × (1.886× 10-3)

= 4.2253001 m/s

$$\mathbf{\Delta h} = \frac{\mathbf{(4.225)^2 - (2.2403)^2}}{\mathbf{2}}$$

= 0.006416 KJ/kg

Table Calculation for Fanno-line flow

Section	t °(C)	P (bar)	X	103 v m3/kg	u = GV	Δh kJ/kg
K	46	11.904	0	0.892	2.24	0
1	40	10.167	0.053	1.886	4.225	0.006416
2	30	7.703	0.134	4.28	9.88	0.03704
3	20	5.718	0.203	7.928	17.76	0.11175
4	10	4.147	0.264	13.54	30.244	0.2995
5	0	2.929	0.34	22.57	50.56	0.820

Friction factor calculations:

Point 1 (40 0 C)

Viscosities,

$$\mu_f = 0.16314 \text{ cP}$$
$$\mu_g = 0.01244 \text{ cP}$$
$$\mu_1 = (1 - x_1)\mu_f + x_1\mu_g$$
$$= (1 - 0.05302)0.16314 + (0.03611)(0.01244)$$
$$= 0.15514 \text{ cP}$$
$$\mathrm{Re} = \frac{Z}{\mu_1}$$
$$= \frac{1.79228}{0.15514 \times 10^{-3}}$$
$$= 11552.66$$
$$f_1 = \frac{0.32}{(\mathrm{Re})^{0.25}}$$
$$= \frac{0.32}{(11552.66)^{0.25}}$$
$$= 0.03$$

Table Friction Factor Calculations

Point	t (oC)	x	μ_f (cP)	μ_g (cP)	μ (cP)	Re	f
K	48	0	0.151	0.01263	0.156	11821.64	0.03
1	40	0.053	0.163	0.01244	0.155	11552.66	0.03
2	30	0.134	0.186	0.01197	0.162	11015.85	0.031
3	20	0.263	0.210	0.01154	0.170	10530.43	0.031
4	10	0.264	0.247	0.01102	0.184	9706.98	0.032
5	0	0.34	0.270	0.01074	0.187	9580.28	0.032

Length calculations

Consider section k-1.

Total pressure drop

kΔP1 = 11.904 – 10.167

= 1.737 bar

Acceleration pressure drop

$$\Delta_{P_A} = G\Delta u$$
$$= (2.24035 \times 10^3) \times (4.225 - 2.24)$$

= 4.447094 × 103 N/m2

Friction pressure drop

$$\Delta_{P_f} = \Delta_{P_1} - \Delta_{P_A}$$

=1.737×105 – 0.04470×105

=1.73655×105 N/m2

Mean friction factor

$$f = \frac{0.03 + 0.03}{2}$$

= 0.03

Mean velocity

$$u = \frac{4.225 + 2.25}{2}$$

= 3.23 m/s

Incremental length

$$\Delta L = \frac{\Delta_{P_f}}{Yfu}$$
$$= \frac{1.73655 \times 10^5}{(1.400218 \times 10^6)(0.03)(3.23)}$$
$$= 0.01279 \text{ m}$$

Table Capillary Tube Length Calculations

Section	ΔP (bar)	Δ_{P_A} (bar)	Δ_{P_f} (bar)	ΔL (m)
k-1	1.737	0.04470	1.736	0.0127
1-2	2.46	0.12017	2.339	2.076
2-3	1.985	0.18207	1.802	0.3134
3-4	1.571	0.2796	1.2913	0.4933
4-5	1.218	0.45509	7.5491	0.1769
Total length required = $\sum \Delta L$ = 3.07 m				

III. CONCLUSION

In this study, various parts of a vapour compression refrigeration cycle is designed for a mini refrigerator. The related studies in the literature are discussed. The outer dimensions of the cabinet are 270w*320d*447h mm and inner dimensions of cabinet are 200w*200d*400h mm. From that we calculate the capacity of cabinet is 155W. From condenser temperature at 46oC and evaporator temperature at 0oC we got the mass flow rate is 1.127*10-3 kg/s. Considering the above conditions we select the capillary tube of diameter 0.8 mm and from that we got the length of capillary tube is 3.07 m.

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