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Portable Domestic Refrigerator: A Design Approach

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ABSTRACT

This project work is designed mainly for the preservation of foodstuffs, storing of medicines and the cooling of water and drinking fluids in our homes and offices. In the choice of the necessary parameters for the design, Dichlorodifluoromethane, Freone (R12) is used as the refrigerant. An evaporating pressure of 1.237 bar and condensing pressure of 13.66 bar and corresponding temperatures of -25°c and 55°c was considered and the volume of the inside of the refrigerator was fixed at O.2m^3 (200litres). The power of the compressor of the refrigerator is 46W. The cooling load as estimated has been assumed and coefficient of performance was calculated arid the various design components such as the compressor, condenser, capillary tube were analyzed. The parameters have indeed facilitated the successful design of this refrigerator.

Keywords: Refrigerator, Compressor, Condenser, Evaporator, Expansion device

INTRODUCTION

Refrigeration refers to the process of lowering the temperature of an insulated enclosure by extracting heat from the space and rejecting it to the surroundings or some other external medium of higher temperature. Refrigerator is an insulated enclosure or apparatus for producing and maintaining a lower temperature and transferring heat from a cold chamber which is at a temperature lower than that of its surrounding [Eastop and Mcconkey, 1993]. Refrigerators are aimed at preserving food by keeping it cool. Storing and prolonging the life of perishables has been a concern throughout the history of refrigeration, hence, refrigeration strictly removes heat rather than the production of cold thereby inhibiting the growth of bacteria rather than killing the bacteria [Khurmi and Gupta, 2005]. In many ways the refrigerators seems little more than a glorified cupboard whose use, given a basic level of dexterity, requires minimal skill or technological understanding. The process brings about the interaction between the refrigerator and its immediate environment in which it is used. The design of a portable domestic refrigerator have in consideration the climatic conditions such as the ambient temperature and the atmospheric pressure of the environment in relations to the geographical area as well as the health needs and comfort of the users which differs across the globe.

A refrigerator operates on the reversed cannot cycle which is called refrigeration cycle. The working fluid (refrigerant) condenses and evaporate at temperatures and pressures close to the atmospheric conditions. Therefore, the boiling temperature decreases with a decreasing pressure.

In most refrigeration system of the horse power range such as the domestic refrigerator, the mechanical vapour compression of refrigeration system are used along with air cooled condensers.

The basis of the mechanical refrigeration system is a series of consecutive thermodynamics process occurring in a closed system in which the refrigerant used does not leave the system, but is circulated throughout the system alternately condensing and evaporating. In evaporating, the refrigerant absorbs its latent heat from the surrounding. While condensing, it gives out its heat to the atmospheric medium.

The vapour compression refrigeration system is therefore a latent heat pump. The system consist both mechanical and electrical component of which the principal mechanical component are as follows:

i. Compressor — is used for increasing the pressure
ii. Condenser — is for condensing
iii. Capillary tube — is for expansion and metering
iv. Evaporator — is for evaporation and establishment of the cooling effect.

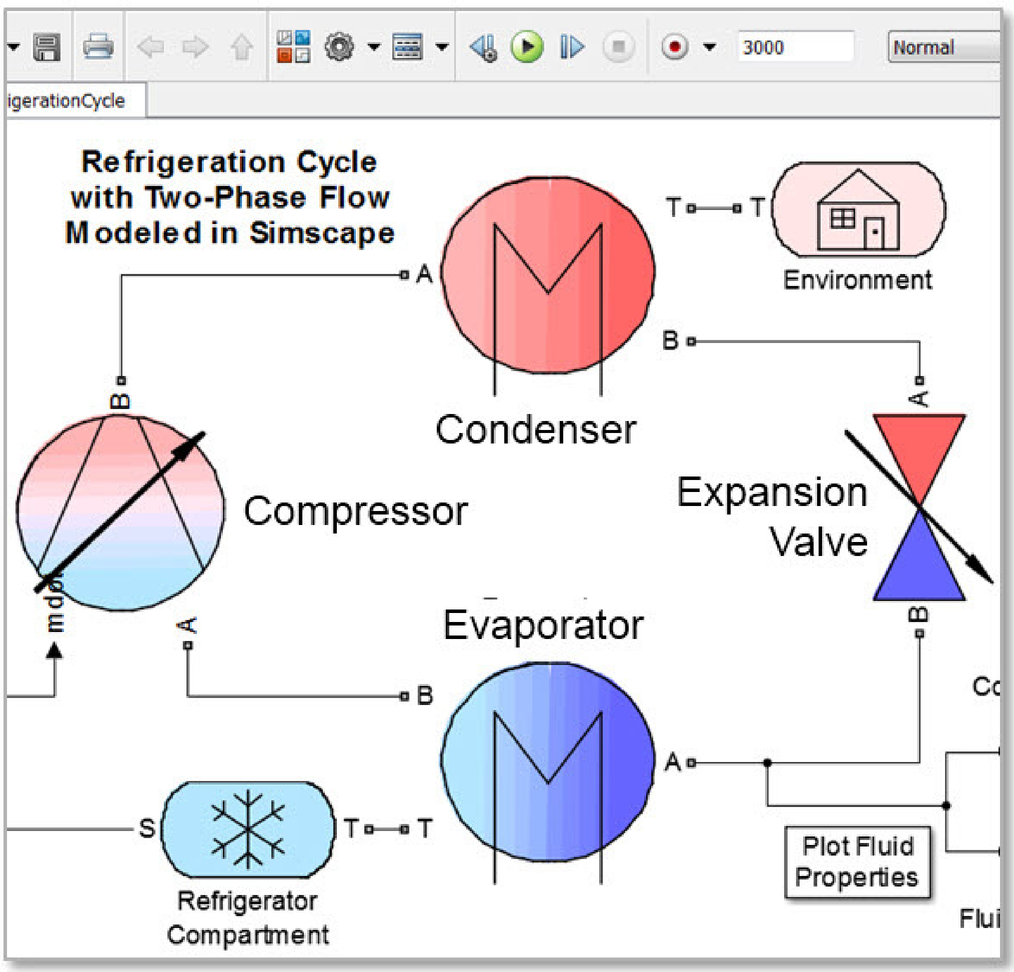
Also to be considered is the kind of foods and fluids to be refrigerated based on the choice of refrigerator and temperature. Therefore, refrigerator operates according to different temporal logic than its surrounding, and a site where the world in effect, turns more slowly.

Refrigerator design and manufactured away from our tropical areas cannot function effectively or at its maximum capacity without having in mind the climatic conditions and the kind of food to be refrigerated.

The successful consideration of this condition will no doubt brings about a satisfying output of refrigerators in our homes. Proper refrigeration is assured with the temperature kept always below 100^{O}C, which modern science agrees is the danger point. Unnecessary food spoilage is eliminated and the health of every member of the family is protected.

The design of a portable domestic refrigerator is treated systematically in the following ways:

1. Specification of component sizes such as the power of the compressor, the evaporator, and the condenser sizes.
2. Component design: Thermodynamics analysis, functions and properties as the use of mathematical calculations was considered in the design of the refrigerating components.
3. The complete design of the portable domestic refrigerator was developed from the components.
4. Working drawings, diagram illustrations for easy understanding of the principles with the components and the refrigerator were treated.



Source: http://www.arca53.dsl.pipex.com

Fig. 1 : Vapour Compression Refrigeration System

COMPRESSOR

The compressor is the device for work input into the system. It sucks either the wet vapour, saturated vapour or superheated vapour from the evaporator and raises the temperature and pressure of the refrigerant by compressing it to the condenser. It maintains a pressure difference between the high and low sides of the system.
Considering the basis on which processes are created it can be deduce that,

i. The pressure and temperature of the condenser are raised allowing the refrigerant to give up heat to the atmospheric air outside, and is used to absorb the heat.
ii. The pressure and the temperature of the refrigerant is used to boil and absorb heat from its surrounding.

Basically, compressors are classified into three general types. Reciprocal compressors, centrifugal compressors and axial compressors. Also field service accessibility includes open type, Hermetic and semi-hermetic compressors.
Base on the portability of this design only the open type and hermetic compressor will be discussed.
OPEN-TYPE COMPRESSORS OR SEMI-HERMETIC
The compressor can be opened or dismantle up in the field for repairs and are usually belt driven but may also be directly coupled to an external electrical motor.
HERMETIC COMPRESSORS
These are directly coupled to a motor by a means of a common shaft. These compressors are completely sealed by welding and therefor inaccessible for service.

THE CONDENSER

The condenser is for the removal of heat from the refrigeration cycle. It takes place by using atmospheric medium.
The condenser is simply a device for condensing a gas into liquid. It also transfers heat from the refrigerant to the atmosphere. It is described as the door through which unwarranted heat flows out of the refrigerators.
Superheated high-pressure refrigerant vapour is cooled to its boiling or condensing point by rejecting form, based on this evaporations it can be said that the rejected heat candidates of both heat vaporization and compression.

EXPANSION DEVICES

The expansion devices are important drive that divides the high— pressure side and the low pressure of refrigeration system. It is connected to the receiver and the evaporator. The main types of expansion used in industrial and commercial refrigeration and air-conditioning system are, capillary tube, automatic or constant pressure expansion valve, thermostatic expansion valve, low-side flow valve, high side float valve.

INSULATION

The insulation refers to the retarding of the flow of heat in to the refrigerated space. Insulation is made between the inner surface (cooling spare) and the outer surface.
In the above-discussed mechanical components the Hermetic compressors, Air-cooled condensers and capillary tubes are used in domestic refrigerators.

DESIGN CALCULATION AND ANALYSIS

The designing of a portable domestic refrigerator is based on the required cooling capacity. The condensing and evaporating temperatures are chosen accordingly.

REFRIGERATION CYCLE

The refrigerator is assumed to operate in a simple saturation practical vapour compression refrigerator cycle as shown in the p-h diagram of fig 2.

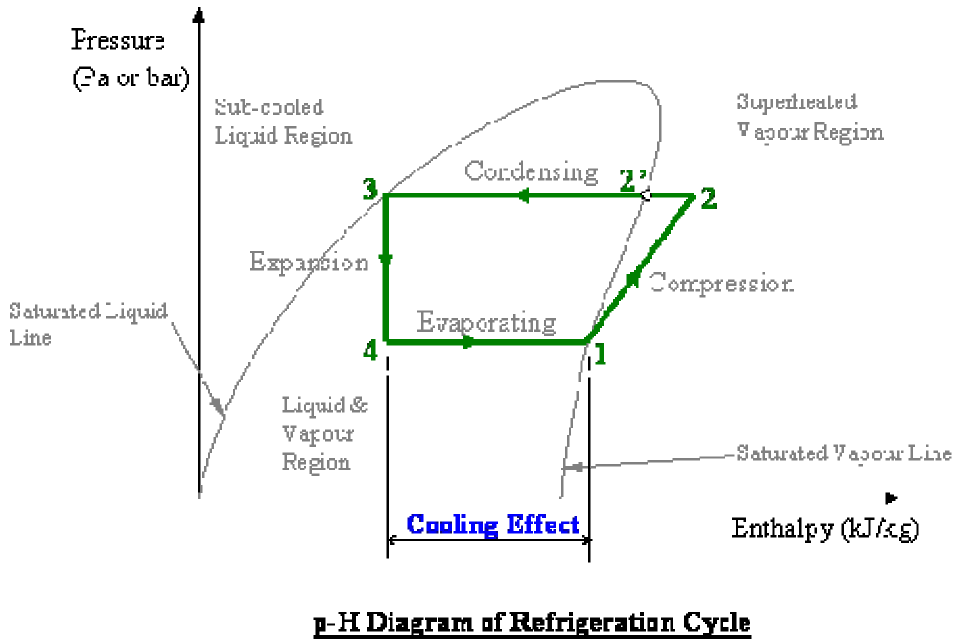


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Fig. 2: P-H DIAGRAM OF A SIMPLE SATURATION PRACTICAL VAPOUR

COMPRESSION REFRIGERATION CYCLE

The events of the cycle drawn on the p-h diagram is as follows:

Process 1-2: Saturated vapour refrigerant is compressed isentropically
From state 1 to state 2.

Process 2-3: The refrigerant is condensed at constant pressure to
Saturated liquid at state 3 as heat rejected at the condenser.

Process 3-4: Is a throttling process where enthalpy is constant.

Process 4-1: Is evaporation at constant pressure. The evaporation is what brings about the cooling of the object (space) or surrounding by extracting heat from it.

In a refrigerator cycle we assume that only unit mass of refrigerant flows round the cycle therefore values of enthalpy are specific, but the actual mass of refrigerant that flows per unit time is called the mass rate (kg/s).

Therefore, the parameters to be considered are chosen as follows:

Refrigerant used is Dichlorodifluoromethane --- cf_2Cl_2 (R_{12})

Working pressures

Evaporating Pressure 1.237 bar

Condensing Pressure = 13.66 bar

Corresponding temperatures are:

Evaporating temperature = -25°c = (273 — 25) 248°k

Evaporating temperature = 55^0 C (273 + 55) 328°k.

h_3 = 90.25 kj/kg = h4 (throttling process)

h_1 = 176.48 kj/kg

h_2 = 232.5kj/kg

Refrigerating effect Q_{41} = (hi — h_4) kj/kg ……………………………………………..................................1.0

= (176.48 90.25) = 86.23 kj/kg Q41 = 86.23 kj/kg

Work in put to the compressor (Wnet) = (h_1 — h_4) kj/kg ……………… 1.2

= (232.5-176.48) = 56.02 kj/kg

Wnet = 56.02 kj/kg

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COEFFICIENT OF PERFORMANCE OF THE REFRIGERATOR

COP_{ref} $\frac{\text{Refrigerating effect}}{\text{Net Work}} = \frac{Q_{41}}{Wnet} = \frac{h_1 — h_4}{h2—h_1}$ ……………………………………………………….1.1

$CO_{Pre} = \frac{h_1 - h_4}{h_2 — h_1} = \frac{176.48 — 90.25}{232.5 — 176.48} = \frac{86.23}{56.02} = 1.54$

$CO_{Pre} = 1.54$

THE POWER OF THE COMPRESSOR

The power of the compressor is chosen to be 46W. By conversion the compressor power in 1KW = $\frac{46}{1000}$ = 0.046KW

MASS FLOW RATE

The mass flow rate is get from the equation

P = ṁWnet. ………………………………………………………………………………………………1.2

Where P is the power of the compressor

Wnet is the net work

m is the mass flow rate

m p/Wnet ……………………………………………………………………………………………...1.3

Substituting the values into the equation.

$m = \frac{0.046}{56.02} = 8.2 \times {}^{10\text{-}3}$ kg/s

m = 8.2 x i03 kg/s

COOLING CAPACITY

Q_0 m q_0 ……………………………………………………………………………………………………1.4

$8.2 \times 10^{-3} \times 86.23 = 0.707$kw

Where m is the mass flow rate

q_0 is the refrigerating effect.

VOLUME FLOW RATE

V = mv ……………………………………………………………………………………………………1.5

Where V is the volume flow rate (m3/s)

V is the specific volume (m3/s)

V is the mass flow rate (kg/s)

V = 8.2×10^3 kg/s x 0.1312 (m3/kg)

V = 1.07584 (m3/s)

V = 10.7584x104m3/s

THE CONDENSER DESIGN ANALYSIS

The condenser capacity (Qc) is obtained mathematically from the expression

Qc:	=	$m(h_2—h_3)$ ……………………………………………………………………………1.6
m	=	8.2 x 103kg/s
h_2	=	232.5kj/kg
h_3	=	90.25 kj/kg
Qc	=	8.2×10^{-3} (232.5-90.25) (kg/s x kj/kg)
Qc	=	8.2 x 10 x 142.25
Qc	=	1.16645kw

HEAT TRANSFER CALCULATIONS

The rate of heat transfer from one medium to another can be as expressed as:

$Q = \alpha A \Delta t$ ……………………………………………………………………………1.7

Where α = Convective heat transfer coefficient

A = Surface area of component in m^2

Δt = Temperature difference between refrigerant and the surface.

HEAT TRANSFER COEFFICIENT

Fluid flow in tubes in refrigeration only, convective heat transfer is consider.

THE NUSSELT NUMBER, Nu

Is the needed dimensionless group used in practical applications as it involves the convective heat transfer coefficient which can be deduced from the expression:

$$Nu = \frac{\alpha\, di}{K} \quad \text{……1.8}$$

Where α = convective heat transfer coefficient

di - Internal diameter of pipe or tube in (m)

k = Thermal conductivity

In a turbulent flow for a convective heat transfer coefficient in tubes, the nusselts number can be approximated to:

$$Nu = 0.023\, Re^{0.8}\, P_r^{\,n} \quad \text{……1.9}$$

Where n = 0.3 for cooling fluids

N = 0.4 for heating fluids.

Re = Reynolds number

P_r= Prandtis number

Reynolds number (Re)

This is the needed dimensionless quantity relating to fluid flow in either a tube or pipe. It is expressed as

$$Re = \frac{u\, di\, o}{\mu}$$

For $3 \times 10^3 \leq Re \leq 10^5$ the flow is turbulent [3].

Where u = Velocity fluid flow in m/s

di = pipe internal diameter in m

p = Density of fluid In kg/m^3

μ = Absolute dynamic viscosity in kg/ms

Where p = 1232 kg/m^3

Diameter of condenser pipe dr = 0.005m

Absolute dynamic viscosity $\mu = 2.1582 \times 10^4$ kg/ms

From the equation discharge or volume flow rate (Q)

A=AU …………………………………… ………………………………………………………1.10

ḿ=bQ ……………………………………………………………………………………… ………1.10a

ḿ=bAU

MAKING u THE SUBJECT OF THE FORMULA.

U=ḿ/Ba

But $A = \pi d^2/4$

Substituting A into U

$U = 4ḿ/ b\pi d^2$

Velocity if refrigerant = $\frac{4\acute{m}}{\hat{} Pd^2}$ ………………………………………………………. ………………1.10b

U = $\frac{4 \times 8.2 \times 10^3 kg/s}{3.142 \times 1232\ kg/m^3 \times (0.005)^2 m^2}$

= $\frac{3.28 \times 10^2}{0.0967736}$ = 0.34m/s

Reynolds number Re

Re = $\frac{Udb}{\mu}$ …………………………………… ………………………………………. 1.11

But μ = 2.1582×10^4 kg/ms

u = 0.34m/s

dr = 0.005m

p = 1232 kg/m^3

Re = $\frac{0.34 m/s \times 0.005 m \times 1232 kg/m^3}{2.1582 \times 10^{-4}\ kg/ms}$ = 7904

PRANDTL NUMBER, P_r

This is also one of the dimensionless quantity needed to express fluid flow in heat transfer. It is presented as

Pr = ω/α ……………………………………………… ……………………1.12

Where α = $k/\rho\ C_\rho$

U = μ/ρ

Pr = $\frac{ux/\rho\ C_\rho/k}{/p}$ $= \frac{C_p\mu}{k}$

$\therefore$ Pr = $\frac{c_p u}{k}$ ………………………………………………………………………… .. 1.13

Where

α = thermal diffusity

u = Kinematic viscosity

m = Absolute dynamic viscosity of refrigerant

C_p = specific heat at constant pressure

K = Thermal conductivity

C_p = 103.6 kj/kg

μ = 2.1582×10^{-4} kg/ms

K = 0.0814 (W/MS)

P_r = $\frac{103.6 \times 2.1582 \times 10^{-2}}{0.0914}$ = 2.8

Pi = 2.8

The value of the Reynolds number calculated above shows that the flow is turbulent and therefore DITTUS-BOELTER equation holds as follows:

From equation 1.11

Nu = $0.023\ Re^{0.8}\ Pr^{0.3}$

Pr = 2.8

Re = 9704

Nu = $0.023(9704)^{0.8}\ (2.8)^{0.3}$

Nu = 0.023 x 1547.25 x 1.3619

Nu = 48.47

MEAN HEAT TRANSFER COEFFICIENT (hm)

Hm = $\frac{Nuk}{d}$ ………………………………………………………………… 1.14

hm = $\frac{48.37 \times 0.0814}{0.005}$

Hm = 789.09 W/m^{2}k

= 0.789 KW/M^{2}k

THE TOTAL HEAT TRANSFER SURFACE AREA (A) IS OBTAINED AS FOLLOWS:

From equation 1.7

Q_c: = $m(h_2 - h_3)$

$\dot{m}$ = 8.2×10^3 kg/s

h_2 = 323.5kj/kg

h_3 = 90.25 kj/kg

Q_c = 8.2×10^3 (323.5 – 90.25) (kg/s x kj/kg)

Q_c = $8.2 \times 10^3 \times 142.25$

Q_c = 1.16645kw

Q_c = hm A (tr – tr) …………………………………………………….....   ...... 1.15

Where t_r = Room temperature

t_r = Fluid temperature

hm = mean heat transfer coefficient

$$A = \frac{Qc}{Hm\,(tr - tr)}$$

$$A = \frac{1.16645}{0.789(55-28)} \qquad = \frac{1.16645}{21.303}$$

A = 0.0548, A = $5.48 \times 10^3 m^2$

LENGTH OF THE CONDENSER TUBES

The Length is obtained from the mathematical equation

A = $_{JI}$dl ……………………………………………………………………....   ......1.16

$$L = \frac{A}{JId} = \frac{0.0548}{3.142 \times 0.0005} = 3.49$$

∴ L = 3.49m

EVAPORATOR DESIGN CALCULATIONS AND ANALYSIS

The evaporator capacity (Q_E) is the same as the cooling load. The prandti is 0.68.

Density p is chosen as = 15.39 kg/m^3

Q_E = $m(h_1 = h_4)$ ………………………………………………………….1.17

Where m = Mass flow rate kg/s

m = 8.2×10^3 kg/s

h_1 176.48 kj/kg

Q_E = 8.2×10^3 (176.48 - 90.25)

707.086×10^3W

Q_E = 0.707KW

TUBE DIAMETER FOR THE EVAPORATOR

ID = Internal diameter = 0.005m

QD = Outer diameter = 8 internal diameter = 9.53mm)

$$\therefore 5 \text{ MMID} - \frac{9.53 \times 5\text{mm}}{8} = 6.0\text{mmoD}$$

$$\text{Mean diameter Dm} = \frac{ID + 0D}{2} = \frac{6 + 5}{2} = 5.5\text{m}$$

from equation 1.11b

$$\text{Velocity of refrigerant in the evaporator (u)} = \frac{4\dot{m}}{JIpD^2}$$

$$\Rightarrow \frac{4 \times 8.2 \times 10^3 \text{ kg/s}}{3.142 \times 15.39\text{kg/m}^3 \times 10.005^2\text{m}^2}$$

$= \frac{0.0328}{1.208882=45 \times 10^{-3}} = 27.13 m/s$

$\therefore$ u = 27.13m/s

From equation 1.12

Thus Reynolds number Re $= \frac{Udp}{M}$

Where u = 27.13m/s

d = 0.005m

p = 15.39kg/m^3

u = 11.281 x 10^4 kg/ms

$Re = \frac{27.13 \times 0.005 \times 15.39}{11.28 \times 10^4} = \frac{2.0876535}{11.28 \times 10^4}$

Re = 18507.76

The value of the Reynolds number shows that the flow is turbulent, therefore DITTUS BOLTER equations holds [3]

From equation 1.10

$Nu = 0.023\ Re^{0.8}\ Pr^{0.4}$

$Nu = 0.023 \times (18507.76)^{0.5}\ (0.68)^{0.4}$

$= 0.023 \times 2593.48 \times 1.904.98$

Nu = 51.12

From equation 1.15

Nusselt Number, Nu

$Nu = \frac{h_m d}{K}$

$h_m = \frac{N_u k}{d} = \frac{51.12\ x\ 9{,}28\ x\ 10^2\ W/m^2k}{0.005m}$

h_m = 94.88 W/m^{3}k = 0.094881 KW/m^{3}k

Total surface area for heat transfer

Q_E = hmA (tr – tr.s.t). ……………………………………………………………………1.18

$A = \frac{Q_E}{hm\ (t_r - t_{r.st})}$ …………………………………………………………….................. 1.18a

Where t_r = 28 and $t_{r.st}$ = 7

$A = \frac{0.070\ kw}{0.09488 KW/m^2\ K\ X\ (28.7)} = \frac{0.707}{1.99248}$

A = 0.35m^2

LENGTH OF EVAPORATOR TUBES

The length of the evaporator tube is determined from the mathematical expression as:

From equation 1.16

A = π dl

$L = \frac{A}{\pi\ d} = \frac{0.35}{3.142 \times 0.0055}$

L = 22.28m

Number of tubes $= \frac{L}{A} - \frac{22.28}{0.35} = 64$

= 64 tubes

CAPILLARY TUBE LENGTH

The prominent mathematical equation is applied to the control volume as indicated in point 1 which the condensing process and point of which more so the evaporator process of a capillary can be stated as the conservation of mass, conservation of momentum.

CONSERVATION OF MASS

The mathematical expression for conservation of mass is primarily stated as follows:

From continuity equation

Mass flow rate, $\dot{m} = \frac{UA}{V}$

Where m is the mass flow rate (kg/s)

U is the velocity (m/s)

V is the specific volume)m^3/kg)

But $A_1 - A_2 = A$

$$\dot{m} = \frac{U_1 A_1}{V_1} = \frac{U_2 A_2}{V_2} \quad \text{.......... 1.19}$$

Where V_1 is the specific volume at point o.

V_2 is the specific volume at point 1.

At point 0 the evaporating temperature is – 25^{0}c.

P_0 = 1.237bar = 123N/m^2

Absolute or dynamic viscosity µ1 = 1.88 x 10^4 NS/M

m = 8.2 x 10^3 kg/s

dc = 0.0015m

$\frac{m}{A} = \frac{U_1}{Vi} \qquad \underline{U_1 = \acute{m}xv_1} / A$

$$U1 = \frac{4 \times 8.2 \times 10^{-3} \times 0.1312}{3.142\,(0.0015)^2} = \frac{4.30336 \times 10^{-3}}{7.0695 \times 10^{-3}}$$

U1 = 608.72 m/s

Reynolds number at point o

$$Re = \frac{\rho_1 - dc\, u_1}{\mu_1}$$

where b1 = 15.39kg/m^3

dc = 0.0015m

u_1 = 608.72m/s

μ_1 = 1.88 X 10^4 Ns/m

$$Re = \frac{15.39 \times 0.0015 \times 698.72}{1.88 \times 10^{-4}}$$

Re = 74746.28

$$\text{Friction factor f1} = \frac{0.33}{Re^{0.25}} = \frac{0.33}{(74746.28)^{0.25}}$$

$$F1 = \frac{0.33}{15.353} = 0.02$$

Considering point 55^{0}C saturated liquid from the chart is obtained as

p = 13.66 bar = 1366000 N/m^3

Specific volume v2 = 0.0125m^3/kg

Capillary tube diameter dc = 0.0015m

Absolute or dynamic viscosity of refrigerant

μ_2 = 1.0987 x 10^{-4} NS/m

m = 8.2 x 10^{-3} kg/s

From the continuity

$$\frac{m}{A} = \frac{u_2}{V_2} \qquad u_2 = \frac{mx\, v_2}{A}$$

$$U_2 = \frac{8.2 \times 10^{-3} \times 0.00125X4}{3.142\,(0.0015)^2} = \frac{4.2 \times 10^{-4}}{7.0695X10^{-6}}$$

U_2 = 60m/s

$Um = \frac{u_1 + u_2}{2} = \frac{608.72 + 60}{2} = 334.36$ m/s

Reynolds number at point p

$Re = \frac{u_1\, dc\, u_2}{\mu_2} = \frac{15.39 \times 0.0015 = 12606072}{1.0987 \times 10^{-4}}$

$Re = 12606.72$

$F_2 = \frac{0.33}{Re^{0.25}} = \frac{0.33}{(12606.72)^{0.25}} = \frac{0.33}{10.6}$

$F_2 = 0.031$

$Fm = \frac{f_1 + f_2}{2} = \frac{0.02 + 0.031}{2} = \frac{0.051}{2} = 0.0255$

CONSERVATION OF MOMENTUM

$$\left[(P - Po) - \frac{\text{f.mL}}{\text{d}}\frac{U^2 m}{2_{v2}}\right] \text{A} = (\text{U}_2 - u_1) \ldots \ldots 2.0$$

For $3 \times 10^3 \leq Re \leq 10^5$

Where fm is the mean friction $= \frac{f_1 + f_2}{2}$ …………2.1

Um is t mean velocity $= \frac{U_1 U_2}{2}$ …………2.2

P and p^0 is the pressure at point (1) and (0)

U_1 and u_2 the velocity at point (1) and (2)

1 is the capillary tube length

D1 is the diameter of capillary tube

The above mathematical expression best defined the difference in forces applied to the element because of drag and pressure difference in opposite side of the element which equals the needed accelerated fluid which is commonly known as the momentum equation.

$p\text{-}p^0\text{-} \frac{fml}{d} \quad \frac{um}{2v_2} = (U_2 - u_1)\, \acute{m}/s$

where p = 13.66 bar = 1366000 N/m^2

p^0 = 1.237 bar = 123700N/m^2

$f_m = 0.0255$

$u_m = 334.36$m/s

$v_2 = 0.0125 m^3/kg$

$A = 3.142 \frac{(0.0015)^2}{4} = 1.767375 \times 10^{-6} m^2$

Substituting into the equation

$(136600 - 123700) - \frac{0.255L}{0.0015} \times \frac{(334.36)^2}{2x0.0015} = (60\text{-}608.72) \frac{(0.2 \times 10^{-3})}{1.76375 \times 10^{-6})}$

1243300-76021694.53L = 2548.8748

- 76021694.53L =- 25458.8748 - 1242300

- 7602194.53L =- 1496886.875

$L = \frac{-1496886.875}{-76021694.53}$

L = 1.97m

ISOTHERMAL EFFICIENCY

The isothermal efficiency of the cycle is given by:

$T_{iso} = \frac{\text{Isothermal work}}{\text{Indicated work}}$ …………2.3

I_{iso} $\left[\left[\frac{P}{P_0}\right]\frac{\gamma-1}{\gamma} \quad {}_{-1}\right]\frac{\gamma}{\gamma-1}$ …………………………………………………………………………………….2.4

Where P is the pressure at condenser (bar)

P_0 is the pressure at evaporator (bar)

The expected isothermal efficiency of the refrigeration system is 80 = 90%

I_{iso} $\left[\left[\frac{P}{P_0}\right]\frac{\gamma-1}{\gamma} \quad {}_{-1}\right]\frac{\gamma}{\gamma-1}$

Where P is the pressure at condenser (bar)

$\frac{T}{T_0} = \left[\frac{P}{P_0}\right] \quad \frac{Y-1}{Y}$

Considering Y = 1.137 = 1.14

$I_{iso} = \text{In}\left[\frac{13.66}{1.237}\right]$

$.\left[\left[\frac{13.66}{1.237}\right]\frac{1.14-1}{1.14}-1\right]\frac{1.14}{1.14-1}$

$I_{Iso} = \frac{\text{In}\ (11.043)}{[\ (11.043)^{0.1228}\ -1]\ \ 8.143}$

Considering Y = 1.137 = 1.14

$I_{Iso} = \text{In}\left[\frac{13.66}{1.237}\right]$

$.\left[\left[\frac{13.66}{1.237}\right]\frac{1.14-1}{1.14}-1\right]\frac{1.14}{1.14-1}$

$= \frac{\text{In}\ (11.043)}{[\ (11.043)^{0.1228}\ -1]\ \ 8.143}$

$I_{Is0} = \frac{2.4018}{2.739} = 0.86 \times 100$

$I_{Iso} = 86\%$

The 14% difference depends on the chosen parameters of this design, most important is the evaporating and condensing temperatures efficiency did not fall below the expected efficiency.

The mathematical calculation and possible solutions so far obtained can be over emphasized as it goes a long to determine the effectiveness of the design work if putting practice.

INSULATION

The outer corner is made of aluminum

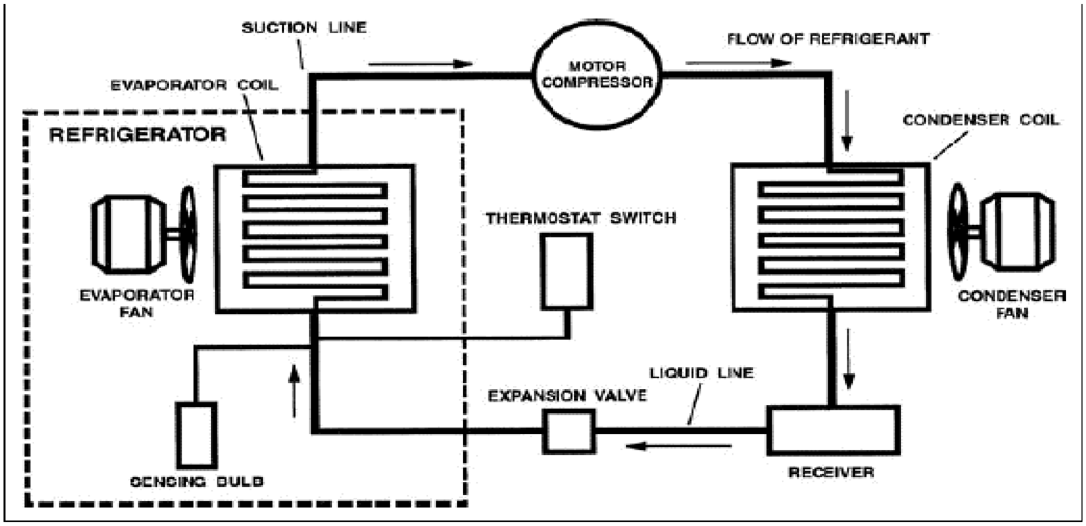
The inner corner is made of cellular plastic

The insulating material is fiber glass

DISCUSSION

The refrigerator has fiberglass as the insulting material and a coated white aluminum as the outer casing. The inner cabin is made of cellular plastic.

Below is a diagrammatical representation of the flow arrangement of the working components of the designed refrigerator.



Source: https://www.bartlettltd.co.uk

FIG. 3: THE FLOW DIAGRAM OF THE WORKING COMPONENTS OF THE DOMESTIC REFRIGERATOR

As shown above, in the evaporator at point 4, the regenerator enters the evaporator with a low pressure of 1.237 bar corresponding temperature of 25^0c.
The refrigerants equally leaves the evaporator with the same pressure and temperature with dry saturated vapour to point 1.

SUCTION LINE
The suction line transfer low pressure at dry saturated vapour to the compressor at 25^0c and 1.237 bar of point in the schematic diagram.

THE COMPRESSOR
The refrigerant is compressed isentropically in the compressor at point 2 in the flow diagram above; the pressure is raise to a value of 13.66 bar and corresponding temperature value of 55^0c.

THE CONDENSER
Condensation begins at point 2 with superheat vapour of 13.66 bar and 55^0c respectively. The feasibility of the design refrigerator can be actualized by comparing an already made domestic refrigerator to the given parameters in this design work.

THE EVAPORATOR
From the design calculation it is noted that the total surface area of the dry evaporation was calculated to the $0.42m^2$ which timing of as component parts and the system in general.

It is precautionary that one should ensure that the refrigerator is lived side by side, back front side level to avoid noise and unnecessary vibrations. It is prominent to note that one should avoid the content of external pipes or fitting with metallic, walls, more so, the cabinet should not be installed close to cookers or any heating equipment.
The compressor should be mounted on vibration dampers to avoid its vibration.

MAINTENANCE
To ensure an efficient performance of this design refrigerator the user must ensure a good maintenance culture to enhance durability and reliability are some of its typical characteristics.
To avoid distortion breaking of minor linings which damage the doors care and tenderness must be taken.

CONDENSERS

The condensers must be kept free from external obstruction to natural air flow and should be cleaned always to retain its maximum condensing efficiency.

RECOMMENDATION

It is generally observed that portable domestic refrigeration are usually kept mostly in the kitchen and dining rooms and offices where lots of heating for domestic homes is been carried out. This intense heat generated by heating equipment effect the condenser performance and its durability, and it is advisable to be kept in a cool or highly ventilated place.

CONCLUSION

From analytical point of view of this design work putting into consideration the calculation, theories of the various parameters from the integral component such as the compressor, condenser, evaporator and capillary tube length along other portable design refrigerator if putting into practice complete favourable with an already manufactured domestic refrigerator of the same choice of parameters.

More so, it has added advantages of being suitable for freezing food stuffs and preservation of perishable items taking into cognizance and humidity of the environment. The above deductions compliments a consolidated design work.

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