iJRASET
International Journal For Research in
Applied Science and Engineering Technology

INTERNATIONAL JOURNAL FOR RESEARCH

IN APPLIED SCIENCE & ENGINEERING TECHNOLOGY

Volume: 8 **Issue:** X **Month of publication:** October 2020

DOI: https://doi.org/10.22214/ijraset.2020.31814

Power Generation using Waste Heat from Condenser of Vapour Compression Refrigeration System

Faris V F[1], Dr. Jacob Elias[2],
[1, 2]*Department of Mechanical Engineering, School of Engineering, CUSAT, India*

Abstract: In this work, a detailed analytical study of the topic power generation using waste heat from condenser of vapour compression refrigeration system were done. Here, a phase change material (PCM) cycle is incorporated along with basic vapour compression refrigeration system. Enormous amount of waste heat energy is released from the conventional vapour compression refrigeration system to the atmosphere through condenser. This project conduct the work on 200 TR Chiller of R410A refrigerant. For 200 TR chiller, amount of 739.7 kW of heat energy is liberated to atmosphere with refrigerant mass flow rate of 5.22 kg/s. Here the waste heat that liberated from the condenser of vapour compression refrigeration system is absorbed by colourless phase change material (PCM), Trichlorouoromethane, also called freon-11, CFC-11, or R-11. It is non-flammable, noncorrosive, non-explosive and sweetish odor liquid that boils at room temperature. Mass flow rate of R11 fluid used in this project is 3.778 kg/s. First, liquid form of R11 PCM is converted into superheated gaseous form by absorbing the latent heat and sensible heat from the condenser of vapour compression refrigeration system. This gaseous PCM passes through turbine, hence turbine rotates and power is develops. The pressure of PCM reduces when it passes through turbine. The condition of PCM at the exit of turbine is of dryness fraction of 0.85. This PCM is pass through the condenser and get cooled by liberating heat energy to atmosphere. By liberating heat energy it become saturated liquid. Then PCM is pumped by external pump driven by part of power from the turbine. Then this PCM is again go to heat absorbing medium to absorb waste heat from vapour compression refrigeration system, hence cycle is completed. The ultimate goal of this project is analysis of the power develop by the turbine through this. Here, shell and tube heat exchanger is used for both heat absorbing medium and heat releasing medium for R11. For heat absorbing medium R410a is used as shell side fluid and R11 is used as tube side fluid. For heat releasing medium, water or air is used as shell side fluid and R11 is used as tube side fluid. In this project, it is used a centrifugal pump for pumping application.
Keywords: PCM, R410A, Trichlorofluoromethane, Heat exchanger, Turbine.

I. INTRODUCTION

There is an enormous amount of waste heat energy, that released from the condenser of vapor compression refrigeration system and this waste heat energy is simply dumped into the environment even though it could still be reused for some useful and economic purpose. Amount of waste heat energy liberated from condenser of vapor compression refrigeration system is 25% more than heat energy absorbed by refrigerant on evaporator at which cooling creates. Similarly large quantity of heat energy is generated from Boilers, Kilns, Ovens and Furnaces. If some of this waste heat could be recovered, a considerable amount of primary fuel could be saved. Waste heat recovery and utilization is the process of capturing and reusing waste heat for useful purposes. Not all waste heat is practically recoverable. However, much of the heat could be recovered and this waste heat energy can be utilized for power generation and other useful purpose such as preheating of combustion air, pre-heating boiler feed water or process water, sanitary hot water, preheating air for processes, space heating in winter, Reheat coils for humidity control, preheating water for washing purposes. In this project, an attempt has been made to power generation by utilizing waste heat energy from condenser of refrigerator. The main objective of this project is to generate power using waste heat recovery system of industrial chiller. Yinhai Zhu and Peixue Jiang present a paper on hybrid vapor compression refrigeration system with an integrated ejector cooling cycle in 2012. Laia Miro et al., presented a paper on Thermal energy storage (TES) for industrial waste heat (IWH) recovery: A review in 2016. Alexandre Bertrand et al., presented a paper on In-building waste water heat recovery: An urban-scale method for the characterization of water streams and the assessment of energy savings and costs in 2017. Waltteri Salmi et al., developed a concept of energy source for an absorption refrigeration system using waste heat of ship in 2017. Haoxin Xu et al., conduct a study on Application of material assessment methodology in latent heat thermal energy storage for waste heat recovery in 2017.

II. THE OBJECT OF STUDY

In this project, it is selected a chiller of 200 TR (7000 kW) capacity of refrigerant R410A. Thus the purpose of this project is to demonstrate the technical and economic feasibility of the power generation using heat recovery system that recover waste heat from R410A gas at the condenser equipment of basic vapor compression refrigeration system. The prime motto of this project is to increase in the COP of the process by reducing electrical energy consumption for the compressor.

A. Goals, Objectives and Motivation

The objective of this project is to determine the potential energy savings associated with improved utilization of waste heat from vapor compression refrigeration systems. Existing and advanced strategies for waste heat recovery in vapor compression refrigeration system are analyzed. More than 50% of total energy is used for air conditioning purposes in some buildings such as Shopping mall, Retail shop, Supermarket. Amount of waste heat is 25% more than the cooling that the process creates. Significant amount of waste heat is rejected by the condenser of supermarket refrigeration systems. But practical uses of waste heat from supermarket refrigeration systems are typically limited to space heating and water heating. Recently, several researchers have attempted to more effectively utilize the waste heat from refrigeration systems using various other techniques. Use of waste heat to drive heat pumps for space heating or cooling has been proposed. In addition, waste heat may be used to preheat the regeneration air used in solid desiccant adsorption dehumidification systems or to preheat the liquid desiccant in absorption dehumidification systems. In this project the waste heat is utilized for power generation by driving a turbine. By implementation of waste heat recovery we can increase the efficiency of the system. Thus the input energy, that is electrical energy consumption can be reduced and leads low running cost and reduction in equipment size.

III. CONCEPT OF THIS PROJECT

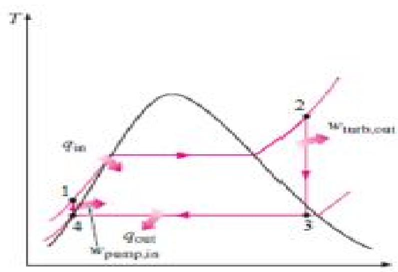

Fig. 1 T-S diagram of R11 cycle

In this project, power is develop by Rankine cycle that is driven by waste heat energy from condenser. Waste heat that is absorbed by R11 PCM and the Liquid R11 is converted into gaseous form. Then it passes through turbine. Then turbine rotates and power can be derived.

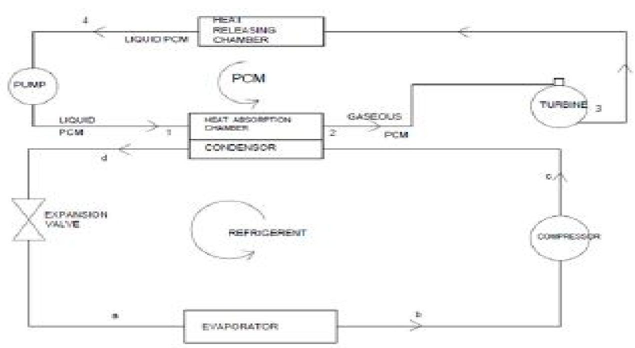

Fig. 2 Schematic diagram of this project

The minimum dryness fraction at turbine outlet is 0.85 and this vapor/liquid mixture goes through condenser and release the heat energy to form saturated liquid. From this it is pressurized by pump and returns the starting position and cycle continuous. In this project, it is planned to study the case of chiller of 200 TR capacity.

ISSN: 2321-9653; IC Value: 45.98; SJ Impact Factor: 7.429
Volume 8 Issue X Oct 2020- Available at www.ijraset.com

IV. TOTAL WASTE HEAT CALCULATION

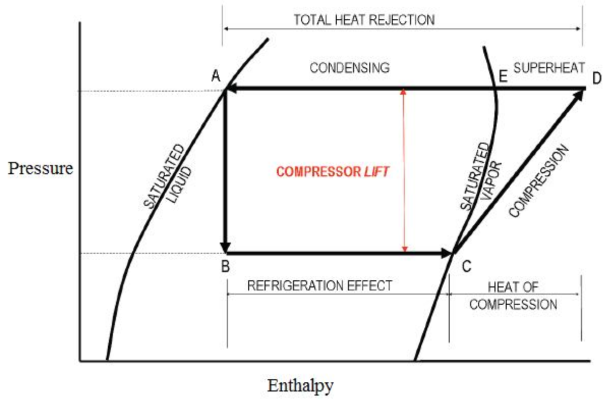

Fig. 3 P-H diagram of R410A refrigerent

Refrigerant (R410A) temperature at inlet of condenser=55^0 C
Refrigerant temperature at exit of condenser=50^0 C
Refrigerant temperature at Evaporator=-5^0 C
For 200 tonnage chiller, Evaporator heat load=Q=200 ton
Q=200X3.5=700 kW
Q_{evap} = m (h_C- h_B);
700=m X (421-286.9)
m= 5.22 kg/s
Condenser waste heat load, Q_{waste},
Q_{waste} =m (L + C_p dt) = 5.22 (137.7+.8 x 5) = 739.7 kW.

V. SELECTION OF PCM

	PCM	Boiling point at atmospheric pressure	Latent heat (kJ/kg) at atmospheric pressure
1	Carbon Di sulfide (CS_2)	46.2	351
2	Cyclopentane	49.3	478
3	Diethyl ether	34.7	390
4	Ethyl Bromide	38.4	250.74
5	Trichloro Fluro methane(R11)	23.8	180
6	Isopentane	27.8	339
7	Isoprene	34.1	360
8	Methyl Acetate	57.2	410.77
9	Dichloro methane	39.8	330
10	Methyl iodide	42.6	192.13
11	Pentane	36	357

Fig. 4 List of PCMs, Currespounding Boiling point and Latent heat at atmospheric pressure

From this list, R11 (Trichloro Fluro methane) is selected for the following properties. It is the widely used refrigerant, because of its high boiling point (compared to most refrigerants), it can be used in systems with a low operating pressure, making the mechanical design of such systems less demanding than that of higher-pressure refrigerants R-12 or R-22. It is nonvolatile and not flammable

A. Reason for rejecting other PCMs

1) *Carbon Di Sulfide:* Volatile, toxic, cancer causing agent liquid and adversely affect reproductive and development system
2) *Cyclopentane:* Highly volatile, flammable liquid, adversely affect central nervous system of animals
3) *Diethyl ether:* Volatile, Flammable liquid, breathing cause high irritation to the nose and throat
4) *Ethyl Bromide:* Volatile compound and effect to skin, Liver, Kidney and it is toxic
5) *Isopentane:* Extremely volatile and extremely flammable. It may affect Liver and heart.
6) *Isoprene:* Very hazardous in case of skin contact (irritant), of eye contact (irritant). Hazardous in case of ingestion, of inhalation (lung irritant). Slightly hazardous in case of skin contact (corrosive, permeator). In Inflammation of the eye is characterized by redness, watering, and itching. Skin in Inflammation is characterized by itching, scaling, reddening, or, occasionally, blistering.
7) *Methyl Acetate:* It is flammable liquid with a characteristically pleasant smell of glues and nail polish removers.
8) *Dichloro Methane:* High volatility. Very hazardous in case of eye contact (irritant), of ingestion, of inhalation. Hazardous in case of skin contact (irritant, permeator). Inflammation of the eye is characterized by redness, watering, and itching.
9) *Methyl Iodide:* Volatile Liquid, Hazardous in case of skin contact (irritant), of eye contact (irritant), of ingestion, of inhalation (lung irritant). Slightly hazardous in case of skin contact (permeator). Severe over-exposure can result in death.

B. Mass of R11 Refrigerant

We want to find out the mass flow rate of R11 refrigerant (m) flowing in this cycle

$Q = m(Cp_l \cdot dt_l + L + Cp_g \cdot dt_g)$

At 2 bar T1'= T2'=44^o C, let T1= 30^o C, T2= 50^o C

Cp_l = Specific Heat Capacity of R11 Liquid = 0.869 kJ/kg K

Cp_g = Specific Heat Capacity of R11 Gas = 0.988kJ/kg K

dt_l = T1'-T1 = 14^o C

dt_g = T2-T2' = 6^o C

L=175.34 kJ/kg at 2 bar pressure

m=3.778 kg/s

VI. DESIGN OF EACH COMPONENT

A. Design of Heat Exchanger for Exchanging heat from Condenser to heat Absorbing Chamber

In this project, Shell and Tube Heat Exchanger is used for transferring heat from refrigerant R410A to Phase Change Material R11. Here we take Shell side fluid as R410A and tube side fluid as R11.

1) *Tube Design (Standard sizes from Tables):* R11 fluid, Carbon Steel, Tube outside diameter, (d_o) =19.05 mm, Tube length (L)=4.877 m, Thickness= 0.889 mm(for Gauge 20), Tube Pitch=1.25 X do=23.81 mm

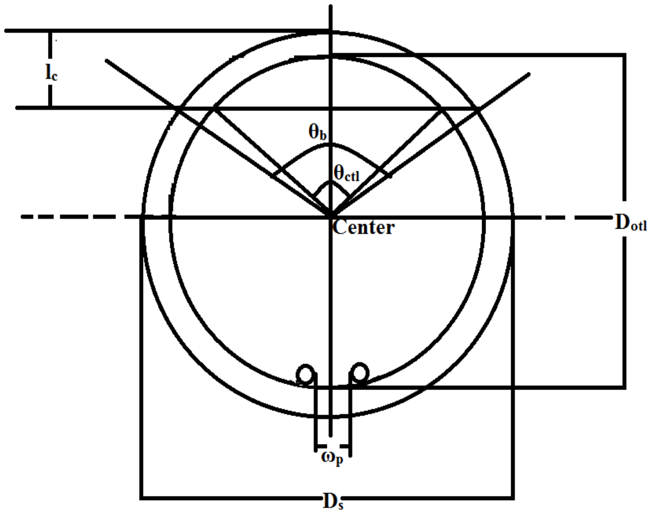

Fig. 5 Cross section of Shell and Tube heat exchanger

2) *Shell Design (Standard sizes from Tables):* R410A fluid, Carbon steel pipe, Nominal Shell diameter = 0.35 m, Thickness= 7.92mm for SCH 20 pipe, Shell side inside diameter, (D_s) =0.334 m, Tube to Baffle hole diametrical clearance, (d_{tb}) =0.8 mm, Shell to Baffle diametrical clearance, (d_{sb}) =3.2 mm (for D_s=0.35 m diameter) Baffle spacing(L_{bc}) =[Minimum: 1/5 D_s or 51 mm ,Maximum: from Table] = [0.0668 m, 1.524 m]=0.3 m, Baffle cut length(l_c) =0.0835 m , Inlet Baffle spacing(L_{bi})=0.32 m. Outlet Baffle spacing(L_{bo}) =0.32 m, Diameter of the outer tube limit(D_{otl})=0.32 m, Width of bypass lane (ω_p) =19 mm, Longitudinal Tube pitch(X_l) =17 mm, Transverse tube Pitch(X_t) =35 mm, Number of tubes (N_t) =125

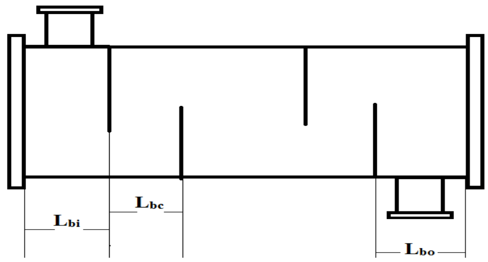

Fig. 6 Exchanger front view (cross section)

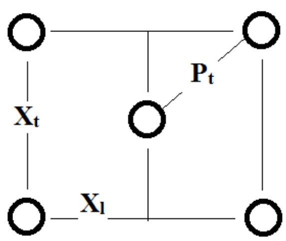

Fig. 7 Arrangement of tubes in Exchanger

For window section(θ_b) =2 COS$^-$ (1-2 l_c / D_s) =2.094$'$=120^0, Gross Window Area(A_{frw}) =D_s^2/4 (θ_b/2- (1-2 l_c / D_s) SIN(θ_b/2) = 0.0171 m^2, Baffle cut angle(θ_{ctl}) =2 COS$^-$ (D_s -2 l_c / D_{ctl}) =1.965$'$ =112.6^0, Fraction F_w of total tubes in the window section(F_w) = θ_{ctl}/(2Π)-SIN(θ_{ctl})/(2Π) =0.1658, Number of tubes in window section(N_{tw}) =Fw x N_t = 21 nos, The net flow area in one window section(A_{ow}) = A_{frw} -A_{frt} = 0.0171-5.878 x 10^{-3} = 0.01122 m^2, Hydraulic Diameter (D_{hw}) =4 A_{ow}/(Π d_o N_{tw} + Π D_s (θ_b/2 Π)) =0.0283 m
For Cross Flow, Number of Effective tube rows in cross flow in each window(N_{rcw}) = 0.8/X_l (l_c-1/2(D_s-D_{ctl}) = 3, Number of tube rows crossed during flow through one cross flow section between the baffle tips (N_{rcc}) =(D_s-2l_c)/X_l=(0.334-2 x 0.0835)/(17 x 10^{-3}) =10, Cross flow area for the 45^o tube layout bundle(A_{ocr})= L_{bc} [D_s – D_{otl} +2 D_{ctl} /X_t (P_t – d_o) =0.02877 m^2, Number of Baffles(N_b) = (L – L_{bi} –L_{bo})/L_{bc} +1 =(4.877-0.3-0.32)/0.3+1=15.12=15.
Tube to Baffle Leakage Area (A_{otb}) = Π d_o d_{tb} N_t /2 (1-F_w)=2.5 x 10^{-3} m^2, Shell to Baffle Leakage Area (A_{osb}) =Π D_s d_{sb} /2 (1-θ_b/2 Π)=1.11 x 10^{-3} m^2

3) *Shell side heat transfer Coefficient:* Thermal Conductivity of tube wall = 53.6 W/mK, Tube side fluid = R11, Shell Side fluid =R410 A, m_t =3.778 kg/s, m_s = 5.22 kg/s, Shell side mass velocity (G_s) = m_s /A_{ocr} = 181.43 kg/m^2 s, Shell side Reynolds number (Re_s) = G_s d_0 / μ_s = 181.43 x 0.01905/(92 x 10^{-6}) , Re_s =37567, Nu_s = 1.04 Re_s $^{0.4}$ $Pr_s^{0.36}$ (Pr_s / Pr_w)$^{0.25}$, Pr_s = 1.05, Here Pr_s =Pr_w, Nu_s = 71.58
Shell side fluid is condensing, So h_{id} =0.943 (k ^{3} ρ ^{2} g L/ (T_s - T_t) μ_s)$^{0.25}$ = 159. 5W/m^2 K, Baffle Cut and spacing effect correction factor(J_c)= 0.55+ 0.72 F_c = 0.55 + 0.72 x 0.6684 = 1.0312, where F_c = 1- 2F_w, r_s = A_{osb} / (A_{osb} + A_{otb}) =0.3074 , r_{lm} = (A_{osb} + A_{otb}) / A_{ocr} = 0.1254, Tube-to-baffle and baffle-to-shell leakage factor (J_l) = 0.44 (1-r_s) + (1-0.44 (1-r_s)) e$^{-(2.2\ rlm)}$ = 0.832, Unequal baffle spacing factor, J_s, L_i^+ = L_{bi}/L_{bc} $^=$ L_o^+= L_{bo}/L_{bc} =0.32/0.3 =1.0667, n=0.6 for turbulent flow, J_s = (N_b - 1 +(L_i^+)$^{1-n}$ +(L_o^+)$^{1-n}$) / (N_b - 1 +(L_i^+) +(L_o^+)), J_s = 1.060, Adverse temperature gradient factor , J_r =1 for Re>100, Shell side heat transfer coefficient h_s = J_c J_l J_s J_r h_{id} = 1.0312 X 0.832 X 1.060 X 1 X 159.5 = 145.055 W/m^2 K

4) *Tube side heat transfer Coefficient:* Number of tube (N_t) = 125, Tube-side flow area (A_{ot}) =Π/4 x d_i^2 N_t = 0.0292 m^2 , Re_t = m_t x d_i /(A_{ot} μ_t) = 196891, Nu = 0.024 Re$^{0.8}$ x Pr$^{0.4}$ = 657 , Heat transfer coefficient = h_t = Nu x K / d_i = 408 W/(m^2 K)

B. Analysis of Turbine

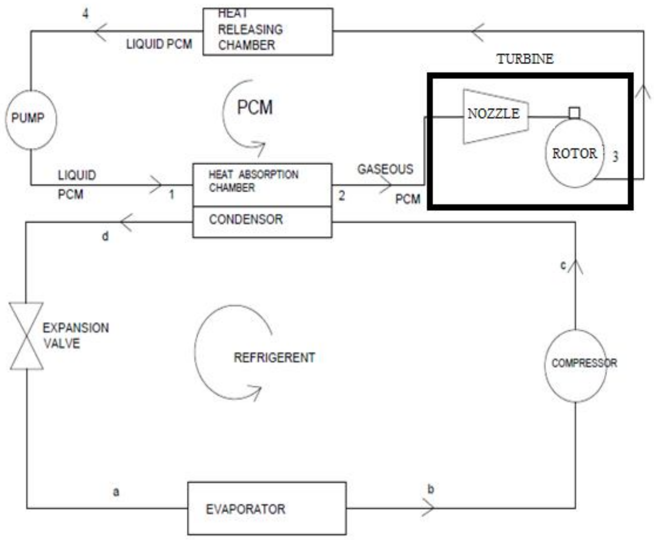

Fig. 8 Schematic diagram of idea and Turbine in box

Υ =1.1391, R=8314/137.368=60.5 J/kg, T_2=50^O C =323 K, P_2=2 bar =2 X 10^5 Pa , ρ_2= 10.23 kg/m^3, Take d_2=0.18 m, m= ρ R T = 3.778 kg/s. 3.778 = 10.23 X 0.18^2 X C, C= 11.4 m/s. M_2= C/a =11.4/ (1.1391 X 60.5 X 323)$^{.5}$ = 0.0764

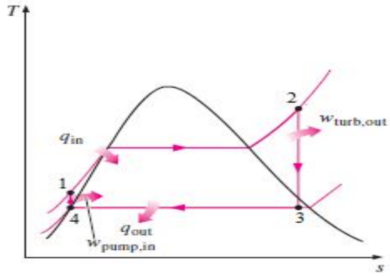

Fig. 9 T-S diagram of Idea

Stagnation Temperature(T_o) , Stagnation Pressure (P_o), Corresponding density (ρ_o), $T_o / T = 1+ (\Upsilon -1)/2\ M^2$, $P_o / P = (1+ (\Upsilon -1)/2\ M^2)^{\Upsilon/(\Upsilon-1)}$., $\rho_o / \rho = (1+ (\Upsilon -1)/2\ M^2)^{1/(\Upsilon-1)}$, Substitute condition at inlet of Nozzle
T_o =323.13 K, P_o=2.00704 bar, ρ_o = 10.4 kg/m^3, for the condition at exit is sonic ,Substitute M=1
We get $T_{2'}$=302.11 K, $P_{2'}$ =1.153 bar, $\rho_{2'}$=6.395 kg/m^3

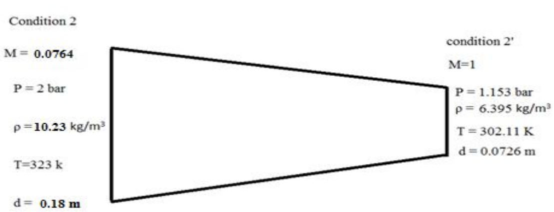

Fig. 10 Schematic diagram of turbine

m= $\rho_{2'}$ X $A_{2'}$ X $C_{2'}$ =3.778 kg, $C_{2'}$= (1.1391 X 60.5 X 302.11)$^{.5}$ = 144. 3 m/s, Substituting we get, $A_{2'}$= A^*= 4.09 X 10^{-3} m^2, $d_{2'}$= 7.22 cm

C. Analysis of rotating part

1) Tangential Velocity of blade

Let u is tangential velocity of blade, $u=u_1=u_2$.

For pelton wheel, Velocity of Jet at inlet is given by, $V_1= C_v\ u /\Phi$ (C_v is coefficient of velocity = 0.98 or 0.99), Φ = Speed ratio, Φ =0.45, V_1 = 144.3 m/s, u = V_1 x Φ / C_v = 144.3 x 0.45 /0.99 = 65.59 m/s

2) Diameter of pelton wheel (D)

Jet ratio, m = Diameter (D) of pelton wheel / Diameter of Jet (d)

d=Exit diameter of Nozzle = 0.0726 m, m = D/d (=12 for most cases), D =12 x 0.0726 = 0.8712 m

Number of Rotation of bucket (N)

u = Π D N/60, N = 23.96 RPS, Number of buckets (Z) = 15 nos, Width of bucket = 5 d = 0.363 m, Depth of bucket = 1.2 d = 0.08712 m

Work done by Turbine by enthalpy relation

For Rankine cycle, $h_2 = h_{2'} + Cp\ (T_2 - T_{2'})$ ($h_{2'}$ = 246.14 kJ /kg, C_p = 0.988 kJ/kgK), h_2 =246.14 + 0.988 (50-44) = 252.068 kJ/kg, $h_3 = h_4 + x\ h_{fg}$ = 210.345 kJ/kg, η= Isentropic turbine efficiency = 0.83 , Work done by Turbine = η m (h2-h3) = 133.98 kW

Magnitude of Velocity and Guide angles

$V_{r1} = V_1$=144.3 m/s, u =u_1= u_2 = 65.59 m/s, V_{r1} = 144.3 - 65.59 =78.71 m/s, $V_{w1} = V_1$ =144.3 m/s, $V_{r2} = V_{r1}$ (We neglect the velocity loss due to friction between inside surface of runner and R11).

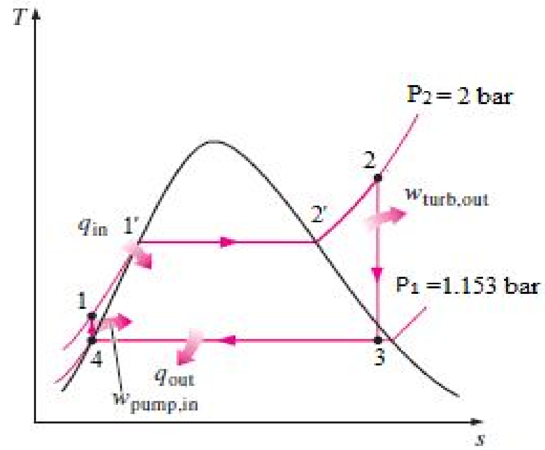

Fig. 11 T-S diagram of Idea

V_{r2} = 78.71 m/s , $V_{w2} = V_{r2}\cos(\Phi)- u_2$= 8.273 m/s, We assume material of bucket as stainless steel of grade 409, $V_{r2}\sin(\Phi)$= Vf2 =26.92 m/s, V_2 = Velocity of Jet at Outlet = $((V_{f2})^2+ (V_{w2})^2)^{0.5}$ = 28.162 m/s, tan(β) = V_{f2} / V_{w2} = 26.92 / 8.273, β = 72.91^0

D. Design of Condenser

Term and pictorial representations are same as that in section design of Heat exchanger for exchanging heat from condenser to heat absorbing chamber

1) Tube Design (Standard sizes from Tables)

R11 fluid, Carbon Steel, Tube outside diameter, (d_o) =15.89 mm, Tube length (L) =3.658 m, Thickness= 0.889 mm (for Gauge 20), Tube Pitch=1.25 x do=19.8625 mm

2) Shell Design (Standard sizes from Tables)

Water or Air, Carbon steel pipe, Nominal Shell diameter =0.33 m, Thickness= 7.92mm for SCH 20 pipe, Shell side inside diameter(D_s)=0.314 m, Tube to Baffle hole diametrical clearance(d_{tb})=0.8 mm, Shell to Baffle diametrical clearance(d_{sb})=3.2 mm (for D_s=0.33 m diameter)Baffle spacing (L_{bc})=[Minimum: 1/5 D_s or 51 mm ,Maximum: from Table] =0.25 m

Baffle cut length(l_c) =0.0785 m, L_{bi}=0.3 m, L_{bo}=0.3 m, D_{otl}=0.3 m, ω_p =17 mm, X_l=15 mm, X_t=27 mm, Number of tubes (N_t) =160, Gross Window Area(A_{frw}) =0.01514 m^2, Baffle cut angle(θ_{ctl}) =2 COS$^-$ (D_s -2 l_c / D_{ctl}) =1.9547 =112.9^0, Number of tubes in window section(N_{tw}) = Fw x N_t =0.1644 x 160 = 26.304 = 26 nos, Area occupied by tubes in window section (A_{frt}) = Π/4 d_o^2 F_w x N_t =5.216 x 10^{-3} m^2, The net flow area in one window section(A_{ow}) = A_{frw} -A_{frt} = 0.00992 m^2, Hydraulic Diameter (D_{hw}) =4 A_{ow}/(Π d_o N_{tw} + Π D_s (θ_b/2 Π)) =4 X 0.00992/(Π x 0.01589X26+ Π x 0.314 x 2/3 Π /(2 Π)) =0.0244 m, Number of Effective tube rows in cross flow in each window (N_{rcw}) = 0.8/X_l (l_c-1/2(D_s-D_{ctl}) =3, Number of Baffles (N_b) = (L – L_{bi} –L_{bo})/L_{bc} +1 =13. Bypass flow Area, A_{obp} =L_{bc} (D_s-D_{otl} +0.5 N_p ω_p) =5.625 x 10^{-3} m^2.

Fraction of cross flow area available for flow bypass (F_{bp}) = A_{obp}/A_{ocr} =0.2305, Tube to Baffle Leakage Area (A_{otb}) = Π d_o d_{tb} N_t /2 (1-F_w) = 2.67 x 10^{-3} m^2, Shell to Baffle Leakage Area (A_{osb}) =1.0523 x 10^{-3} m^2

3) Shell side heat transfer Coefficient

Thermal Conductivity of tube wall = 53.6 W/mK, m_t =3.778 kg/s, Let m_s =18.1 kg/s, Shell side mass velocity (G_s) = m_s /A_{ocr} = 741.803 kg/m^2 s, Shell side Reynolds number (Re_s) = G_s d_0 / μ_s, Re_s =16285.6, Nu_s =1.04 $Re_s^{0.4}$ $Pr_s^{0.36}$ (Pr_s/Pr_w)$^{0.25}$, Pr_s =4.77, Here Pr_s =Pr_w, Nu_s = 88.31, Shell side fluid is condensing, So h_{id} =0.943 ($k^3 \rho^2$ g L/ l (T_s - T_t) μ_s)$^{0.25}$ = 92.45 W/m^2 K, Baffle Cut and spacing effect correction factor (J_c)= 0.55+ 0.72 F_c = 0.55 + 0.72 x 0.6712 = 1.033 , where F_c = 1- 2F_w, Tube-to-baffle and baffle-to-shell leakage factor (J_l), r_s = A_{osb} / (A_{osb} + A_{otb}) =0.282 , r_{lm} = (A_{osb} + A_{otb}) / A_{ocr} =0.152, J_l = 0.44 (1- r_s) + (1-0.44 (1- r_s)) $e^{-(2.2\ rlm}$) = 0.832, Adverse temperature gradient factor , J_r =1 for Re>100, Shell side heat transfer coefficient, h_s = J_c J_l J_r h_{id} = 1.0312 x 0.832 x 1 x 159.5 = 79.32 W/m^2 K

4) Tube side heat transfer Coefficient

Number of tube, N_t = 125, Tube-side flow area = A_{ot} =Π/4 x d_i^2 N_t = 0.0292 m^2, Re_t =m_t x d_i /(A_{ot} μ_t) = 196891, Nu = 0.024 $Re^{0.8}$ x $Pr^{0.4}$ = 657 , Heat transfer coefficient = h_t = Nu x K / d_i = 408 W/(m^2 K)

E. Design of Pump

Work done by the centrifugal pump on liquid

In case of centrifugal pump, work is done by impeller on liquid. Work done impeller = m (h_1-h_4), m = 3.778 kg/s, h_1 = h_4 + C_p (T_1 - T_4), Work input to pump, W_p = m(h_1 - h_4) = m C_p (T_1 - T_4) = 8.207 kW

1) Design of Centrifugal Pump

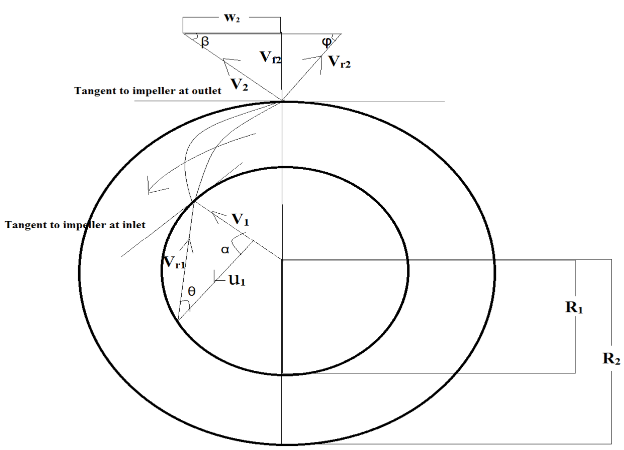

Fig. 12 Velocity triangles at Inlet and Outlet of Impeller

For best efficiency, the liquid enters the impeller radially at inlet, which means absolute velocity makes an angle of 90^0 with the direction of motion of impeller at inlet, u_1 = Tangential velocity of impeller at inlet = Π D_1 N /60

u_2 = Π D_2 N / 60, V_1 = Absolute velocity of liquid at inlet, V_{r1} = Relative velocity of liquid at inlet, α = Angle made by absolute velocity (V_1) at inlet with direction of motion of vane, θ = Angle made by relative velocity (V_{r1}) at inlet with the direction of vane. V_2, V_{r2}, β, Φ are corresponding values at outlet.

As the liquid enters the impeller radially, means absolute velocity of water at inlet in the radial direction, $\alpha = 90^0$, $V_{w1} = 0$. Work done by impeller on liquid per second per unit weight of liquid striking per second, W_p in m of liquid = -(Work done in case of turbine) = $W_p = 1/g\ (V_{w2}\, u_2 - V_{w1}\, u_1)$, d_1 = diameter of suction pipe = 7 cm =0.07 m, D_2 = diameter of impeller at outlet = 0.8 m, D_1 = diameter of impeller at inlet = 0.2 m, N = 1625.15 rpm , u_2 = 68.27 m/s, V_{w2} = 0.035 m/s, $\alpha= 90^0$, $\theta = 2.24^0$, $\beta = 86.97^{0,}$ Diameter of delivery pipe (d_2)= 6.9 cm

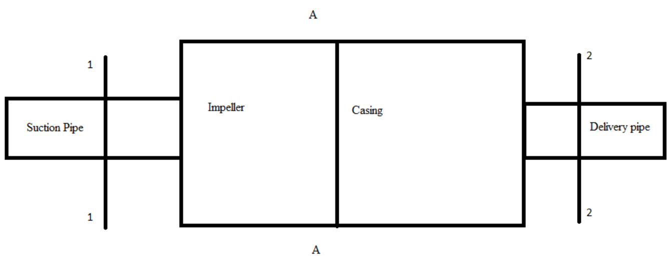

Fig. 13 Piping sections of Pump

Specific speed of a centrifugal pump(Ns) = $N(Q)^{0.5}/(H_m)^{3/4}$, Q = $A_1\ V_1 = 2.57 \times 10^{-3}\ m^3$, H_m = 221.43 m of liquid, N = 1630 rpm, Substitute these values in equation Ns = $N(Q)^{0.5}/(H_m)^{3/4}$. We get N_s = 1.44 rpm, Specific speed of pump = 1.44 rpm

VII. RESULTS, DISCUSSIONS, AND CONCLUSIONS

Large quantity of heat energy is wasted from the condenser of vapor compression refrigeration system to the environment even though it could still be reused for some useful and economic purpose. It is concluded that the power generation can be possible by using waste heat energy from the condenser of vapor compression refrigeration system. Here, a phase change material (PCM) cycle is incorporated with basic vapor compression refrigeration system. In this project, power output is 133.98 kW. 133.98 kW of power is developed by utilizing 739.7 kW of waste heat energy. So 18% of waste heat energy can be effectively utilized for power generation. In this project, a centrifugal pump is used for pumping R11. 8.207 kW is used for driving the centrifugal pump. This power is taken from output power of the turbine. So remaining 125.773 kW power can be developed effectively by incorporation of PCM cycle. This system has many advantages such as increase in the COP of the process, decrease in the electrical energy consumption for the compressor, reduction in the equipment size and low running cost. The disadvantages of this system are high capital cost, more space needed for additional setup and the maintenance required for additional equipment for power generation.

REFERENCES

[1] Yinhai Zhu and Peixue Jiang. Hybrid vapor compression refrigeration system with an integrated ejector cooling cycle, International journal of refrigeration, 2012.
[2] Laia Miro, Jaume Gasia, and Luisa F Cabeza. Thermal energy storage (tes) forNindustrial waste heat (iwh) recovery: a review, Applied Energy, 2016.
[3] Kevin Merlin, Jerome Soto, Didier Delaunay, and Luc Traonvouez. Industrial waste heat recovery using an enhanced conductivity latent heat thermal energy storage, Applied Energy, 2016.
[4] Alexandre Bertrand, Riad Aggoune, and Francois Marechal, In-building waste water heat recovery: An urban-scale method for the characterisation of water streams and the assessment of energy savings and costs, 192, 2017.
[5] Waltteri Salmi, Juha Vanttola, Mia Elg, Maunu Kuosa, and Risto Lahdelma, Using waste heat of ship as energy source for an absorption refrigeration system, Applied Thermal Engineering, 2017.
[6] Haoxin Xu, Alessandro Romagnoli, Jia Yin Sze, and Xavier Py, Application of material assessment methodology in latent heat thermal energy storage for waste heat recovery, Applied Energy, 2017.