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Design and Experimental Investigation of Solar Adsorption Refrigeration System Using Silica Gel - Water

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Abstract: ***Adsorption refrigeration systems are developed due to the need of replacing the conventional systems which utilize environmentally harmful refrigerants and consume high grade electrical power. Solar adsorption refrigeration devices are of significance to meet the needs for cooling requirements such as water chiller, air-conditioning, ice-making and medical or food preservation in remote areas. They are also noiseless, non-corrosive and environmentally friendly. Keeping this as my motivation, I decided to work on water chiller problem. This chiller would generate cooling water for required temperature utilizing solar as an energy source. Main part of my project would be on designing a compact sized system with silica gel-water working pair.***

Keywords: Solar Energy, Adsorption, Silica Gel-Water, Evaporator, Condenser, Adsorber bed

1. Introduction

The process of adhesion of atoms, molecules or ions from a fluid (liquid/gas) or dissolved solid to a surface is the adsorption process. It is a surface phenomenon and leads to development of film of adsorbate, on the surface of the adsorbent.

The solid surface on which adsorption occurs is called as ADSORBENT and the substance which adheres to adsorbent is ABSORBATE.

Figure 1: Adsorption process

It leads to enrichment of material or increase the density of fluid in the vicinity of an interface.

Desorption: On the application of heat the adsorbate molecules get removed from the surface of the adsorbent. In my system I am using solar energy as a source of heat.

1.1 Adsorption Refrigeration Cycle

Adsorption Refrigeration system consist of components same as vapour compression refrigeration system such as compressor, evaporator, throttle valve and condenser. It uses heat energy instead of mechanical energy. Hence in this system mechanical compressor is replaced by thermal compressor.

The refrigerant vapour is adsorbed on the surface of adsorbent bed. Hence the performance of the system is dependent on the adsorption/desorption characteristics of the particular adsorbent/refrigerant pair.

The major components of the system are as under:
1) Thermal Compressor
2) Condenser
3) Expansion Valve
4) Evaporator

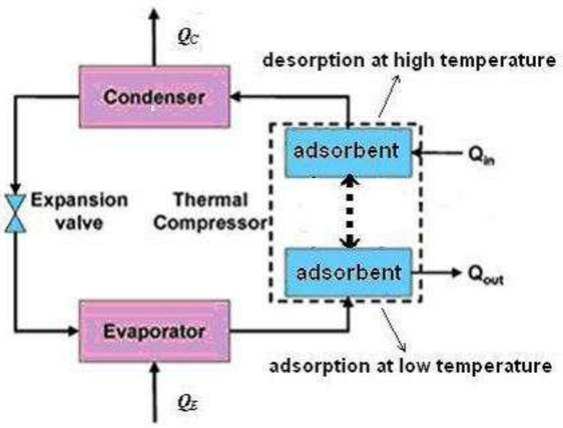

Figure 2: Vapour adsorption refrigeration system

The cycle consists of the following processes:

1) **Process 1: Cooling and adsorption:** During this the low pressure vapour enters the adsorber and it releases heat, hence the temperature of adsorbent decreases and it gets adsorbed on the adsorbent. This is equivalent to compression process in vapour expansion refrigeration cycle.

2) **Process 2: Heating and desorption:** During this process the adsorber receives heat due to which desorption of

molecules from the surface takes place and hence there is the pressure rise. This is equivalent to compression process in vapour compression refrigeration cycle.

3) **Process 3: Condensation:** During condensation the desorbed vapour is liquefied in the condenser. This is equivalent to condensation process in vapour compression refrigeration cycle.
4) **Process 4: Evaporation:** The liquefied refrigerant reaches evaporator where it takes heat and gets vaporize. This leads to cooling effect similar to evaporation as in vapour compression refrigeration cycle.

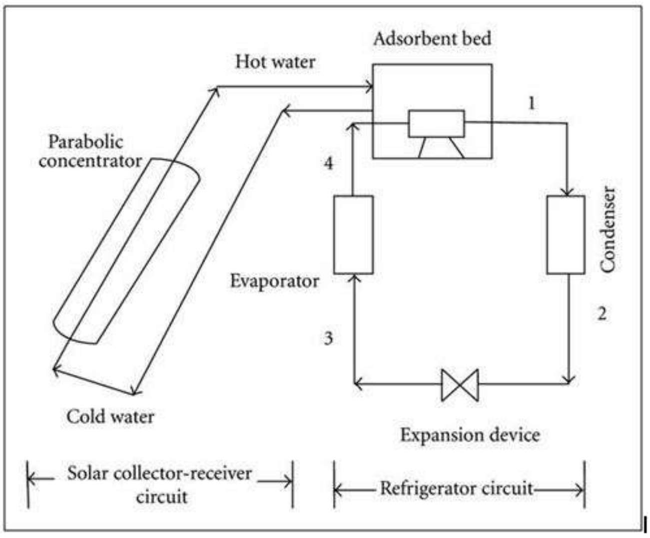

Figure 3: Solar powered vapor adsorption system.

A working cycle for an adsorption refrigeration system is shown in the above figure. It is shown for an adsorber bed, in which processes takes place as shown. At low temperature, bed adsorbs the refrigerant and at high temperature (particular desorption temperature) it desorbs the refrigerant with high pressure.

Thus an adsorber bed replaces mechanical compressor in its own style, and leads to different operating mechanisms for the system.

1.2 Necessities of Adsorption Refrigeration System

1) **No power consumption:** The adsorption system works on solar or any other waste heat. Hence it doesn't require electricity. This is a major advantage, as particularly in remote areas, it's quite difficult to get electricity. Also lesser the power consumption leads to higher efficiency and reduced cost.
2) **Compression mechanism:** Compression process is required to increase the pressure of the refrigerant in the system and hence make it flow. In the adsorption system, mechanical compressor is replaced by thermal compressor. An adsorber bed works as a thermal compressor. The compression of the refrigerant is done by adsorption of the refrigerant by the adsorbent.
3) **Direct utilization of heat:** This system utilizes the heat of solar to heat the adsorber bed. Thus direct utilization of the heat increases the efficiency instead of first converting it in electricity as in PV cell.
4) **Noiseless operation:** As there are no moving parts in the system, it leads to noise less operation of the system.
5) **Low maintenance cost:** In VCR system, compressor needs high attention regarding maintenance. Also in VAR system, pump needs effort in maintenance with respect to lubrication and other problems. While in this system, no such major maintenance cost is added. Only minor leakage should be considered. This is a great advantage.
6) **Running cost:** The conventional refrigeration systems require electricity. Thus running cost is increased. In VAR systems, waste heat or steam is utilized. Thus not that much of electricity consumed as in VCRs. But requirement of pumping power adds running cost; whereas adsorption refrigeration has the lowest running cost.
7) **Type of refrigerant used and its cost:** The refrigerants used in compression systems are quite expensive. While absorption systems uses ammonia-water or LiBr-water; In this case, leakage of ammonia leads to great damage, as it is poisonous. The adsorption system is quite beneficial in these circumstances because we use water as a refrigerant.

The above mentioned advantages of the adsorption system lead me to select it for the water chiller. The water chiller needs temperature maintained between 15^0 C and 20^0 C for drinking purpose. Also low operating and maintenance cost, applicable for small capacity, no necessity of electricity and eco-friendly system can be considered best for water chiller. That's why I selected this system as a water chiller.

2. Literature Review

1) Soon-Haeng Cho et al[1] theoretical and experimental studied and performed on the recovery of a low-grade waste heat using a silica gel-water adsorption cooling system. This system included four components: two absorbers, a condenser, and an evaporator. Its cold generation capacity was 1.2 RT to produce chilled water at 4-7°C. For this purpose they used numerical model which can predict thermal performance of the system. By this model they determine the effect of the heat-transfer rate of individual component on the cold generation capacity. They suggested that by modifying the heat-transfer rates of the condenser and adsorber, the thermal performance could be improved by about three times.
2) D.C. Wang et al. [2] built the prototype of a novel SI-Water adsorption chiller. From their experiment RC and COP were 7.15 kW and 0.38 respectively for heat source at 84.8^0C and cooling water temperature as 30.6^0C and chilled water outlet temperature as 11.7^0C. RC were reached 6 kW under the condition of 65^0C and cooling water temperature as 30.5^0C and chilled water outlet temperature as 17.6^0C.They concluded that the 900 s cycle time would be better choice. Mass recovery time were decreased with increase in heat source temperature. At last they concluded that chiller had a potential of improvement if the refrigeration output loss of the evaporator was avoided in the structural design.
3) Xiaolin Wang and H.T. Chua [3] used an effectual lumped parameter model investigation of the COP and how this COP can be increased. The COP of the system is always

main idea. So during the switching of cooling water firstly flows through the hot adsorber which is to be precooled for the desorption it is heated. On the other side silica gel-water system low temperature water heat is developed for good heat interaction. But this system COP is very low. From the Lumped parameter model they found that cooling water temperature is low as possible to get the maximum efficiency of the adsorber bed. Because as the low temp is there then the maximum adsorption can take place but the temperature is not less than evaporator temperature. So COP can increase upto 0.3-0.4. From the experiment they can conclude that water circulation system made significant role in the adsorption technology.

4) H.L. Luo et al. [4] built the novel solar-powered adsorption cooling system for low-temperature grain storage which consists of a 49.4 m^2 Glass tube solar collector, a silica gel–water adsorption chiller, a cooling tower and a fan coil unit. The adsorption chiller was composed of two identical adsorption units, each of them containing an adsorber, a condenser, and an evaporator/receiver. During the period from July to September of 2004, the system was put into experimental operation to cool the headspace of a grain bin. Three months of operation showed promising performance. The chiller had a cooling power between 66 and 90 W/m^2 of collector surface, with a daily solar cooling coefficient of performance (COP solar) ranging from 0.096 to 0.13. The electric cooling COP was between 2.6 and 3.4.
5) W.S. Chang et al. [5] designed and constructed the solar-powered compound system for heating and cooling in a golf course in Taiwan. They used Plate fin and tube heat exchangers as an adsorber, evaporator and condenser. They concluded that the under standard conditions of 80^0C hot water, 30^0 C cold water and 14^0 C chilled water , COP of 0.37 and cooling power of 9kW achieved. SCP of the system was 72W/kg. From the field test they achieved CO of 0.403 and SCP of 7.79 kW. Form the daily operation they found that efficiency of solar-heating as 28.4%, adsorption cooling as 45.2% and solar cooling as 12.8%. The fluctuation of chilled water outlet temperature could be reduced by installing a small buffer tank after the chilled water outlet of the adsorption chiller.
6) T.Miyazaki et al. [6] presented study of a new cycle time allocation in silica gel-water based adsorption chillers to enhance their performances. The new cycle time contributed in the reduction of delivered chilled water fluctuations. They used Freundlich mathematical modeling to describe the adsorption isotherm of RD type silica get-water pair. The proposed new cycle time allocation allowed the continuous cooling effect over the cycle without sacrificing the effect of pre-heating/pre-cooling. Their Simulation result showed that the new cycle time was effective for both RD type silica gel-water and CaCl2-in-silica gel water pairs, and the cooling capacity was increased as much as by 6%. Also, their new cycle allocation time improve the COP of the system.
7) C.J. Chen et al. [7] developed adsorption chiller which contained two adsorption and desorption chamber and one chilled water tank. Silica gel thermal conductivity is very less so heat transfer take place at very low speed and to improve COP of the adsorption system they found hygroscopic salt is pores of silica gel. For this finned tube heat exchanger and condenser and evaporator are shell and tube type heat exchangers are used. After this work from the observational table they can determine the optimal solution of the cycle time the average hot water temperature, cooling water temperature and chilled water temperature is 78.5, 31.5, and 10.5 respectively for 720s cycle timing. From this COP is 0.45.
8) G. Zhang et al. [8] developed Solar driven and Silica gel-water used adsorption chiller by lumped parameter model. In this work they investigated that for an open circulation of the hot water for short time and closed circulation of the hot water for the large time is better. From number of experiments scientist noted that for less than or equal to 65’C generation temperature the cycle timing is large for optimum output and give maximum COP. They made some assumption for getting result. And for the 10-15 ‘C of the evaporator and 1100s cycle timing the cop of the system varies between 0.30-0.45. When the stable heat source is there and closed circulation of the hot water the maximum COP is achieved. And if the stable source is not there then initial temperature influence on the COP is more.
9) Sourav Mitra et al. [9] presented a two-stage Si-Water adsorption desalination (AD) and chiller system. They formulated mathematical relationship based on conservation of mass and energy along with isotherm relation and kinetics for RD type silica gel + water pair. They predicted the dynamic characteristic of the adsorber bed with the help of LDF model. They modeled dynamics of heat exchanger using LMTD method. They introduced inter-stage pressure as a new parameter for optimizing the two-stage operation of AD chiller system. From their experiment they concluded that that the transient response of the high pressure stage-2 adsorber is slower than the low pressure stage-1 adsorber owing to its larger thermal inertia. The variation of Specific Cooling Capacity and Specific Daily Water Production with cycle time described similar trend. COP increased with increasing cycle time in their experiment.
10) S.W. Hong et al. [10] analyzed numerically the performance of fin-tube type adsorption chiller. They developed 2D ax symmetric transient model and investigated the effect of 10 different parameters i.e. fin thickness, fin height, fin pitch, diffusion coefficient, particle size, cycle time, cycle ratio, temperature of hot water, fluid velocity and porosity. They found an optimum condition of chiller through analysis of variance (ANOVA). Fin thickness and fin pitch were most dominant parameter affecting COP; levels of contribution were 48.67% and 26.94% respectively. Higher COP could be obtained by decreasing fin thickness because in thinner fin, thermal mass was reduced and amount of adsorbent increased, which increases Cooling energy. The larger fin pitch increases the COP. COP increased with increases in the porosity because higher porosity reduced intra-particle resistance, which resulted in increased intra-mass transfer capacity, hence adsorption was increased in bed. COP of chiller decreased if the inlet water temperature increases. But up to, an optimum temperature i.e. 80°C COP increased because as the hot water temperature increased, increase the heat supply to the adsorber bed is small compared to the cooling effect

obtained from the evaporator. As the temperature increased after 80°C COP decreased. From all 10 parameter, hot water temperature had the highest effect (74.02%) on the SCP of the chiller. SCP of chiller increased with increased in the hot water temperature. This happened due to when the hot water temperature increased, the desorption ability were improved by the enhanced heat transfer capacity. As Cycle time increased, the heat and mass transfer capacity were time for adsorption and desorption process, which resulted in higher COP of chiller. But as the cycle time increased, SCP of chiller was reduced. As the size of particle decreased the COP of chiller were increased because smaller size particle offered less intra-particle resistance. SCP decreased with higher fin height because the increased volume of adsorption bed by the extended fin height, which reduced the heat and mass transfer capacity. COP was not affected by the fin height. As Diffusion Coefficient increased, SCP increased due to enhanced intra-particle mass transfer capacity, which resulted in higher adsorption amount. The confirmation experiments with a combination of all the optimum levels gave COP 0.6782 and 217.68 W/kg SCP.

11) Ankush Kumar Jaiswal et al. [11] studied the Dynamic performance of a single-stage, two-bed, SI-Water adsorption chiller operating in Bangalore, India. They used Evacuated Tube Collector. They evaluated system performance based on daily averages of solar coefficient of performance (DACOPsol) and cooling capacity (DACC). Both of this parameter would maximize by using appropriate solar collector area and cycle time. They found that the CACC increases with increase in the solar collector area resulting from higher hot water inlet temperature. Similar effect was observed when the cycle time was increased for a fixed collector area. And also significantly large collector area or large cycle time, the CACC is nearly constant during mid-day as the hot water inlet temperature reaches its maximum allowable limit of 95^0C. The DACC increased with increase in solar collector area for a fixed cycle time. However, for a given solar collector area, it achieved an optimum at a certain cycle time beyond which it decreased due to hot water inlet temperature limitation. The optimum cycle time varied with change in collector area and the month of operation because hot water achieves 95^0C depends on the collector area and irradiance. The maximum DACOPsol decreased with increase in collector area irrespective of cycle time. The sizing of the collector area and choice of cycle time were key parameters for successful operation of a solar driven adsorption chiller directly coupled to the collector field without a thermal storage.

Table 1: Summary of literature review

Sr No	Author Name	Application	Input Energy	P kPa	Tg °C	Te °C	Tc °C	Cycle Time s	COP/ SCP	Capacity
1	Modeling of a silica gel/water by Soon-Haeng Cho et al	Chiller	Waste heat	9.4	80	4-7	20	600	-	1.2 RT
3	Study of novel silica gel water chiller by D.C. Wang et al	Chiller	Waste heat	-	84.8	11.7	30.6	900	0.38	9 Kw
5	Two bed silica gel-water adsorption chillers xiaolinwang et al	Chiller	-	0.9-20	85	14-15	30	1200	0.4	-
6	An efficient solar-powered adsorption chiller and its application in low-temperature grain storage by H.I.Luo	Grain storage	solar energy	-	65-80	10	-	600-900	0.45	-
8	Design & Performance of Si-water by W.S. Chang et al	Compound System for Heating & Coling	solar energy	-	80	14	30	-	0.37	7.15 kW
9	Theoretical research of a silica gel-water by L.Li et al.	Chiller	Engine waste heat	-	85	20	31.1	1000-1100	0.4	16 kW
11	Study On Compact Silica gel -water by C.J.Chen et al	Chiller	-	-	65	20	31	1000-1100	0.4	10 kW
13	Simulation Of Operating G.Zhang et al	Chiller	-	-	100	10-15	25	1100	-	-
16	Simulation study of a two-stage adsorber by SouravMitra et al	Adsorption desalination and Chilling	solar energy	0.9-20	85	-	30	1200	0.25	21 kW
19	Effect of ay and con by P.G.Youssef et al	Chiller	-	-	65-85	-	-	425	0.450	-
20	Performance Anaylysis of Four Types of By L.Q.Zhu et al	Refrigerator	solar energy	-	90	15-20	-	720	720	3.2 kW
23	Performance of silica gel-water solar adsorption cooling system by GhilenNasreh et al	Chiller	-	-	94	15	-	840	840	3-5 kW

3. Design of the Components

I used and assembled various components like adsorber bed, condenser, evaporator, expansion device, pressure gauge, charging valve and parabolic concentrator into the entire system. Out of these components the readymade components that I used were expansion device, pressure gauge, charging valve, parabolic concentrator and the components like adsorber bed, condenser and evaporator were manufactured as per the design obtained from calculations with reference to DQ kern book[12].

3.1. Design of Adsorber bed

Figure 4: Fabricated Adsorber Bed

For adsorber bed I have to assume hot water inlet outlet temperature and water as refrigerant inlet outlet temperature. For adsorber bed I have used shell and tube type heat exchanger.

T_{hi}=100 °C T_{ho}=80 °C ΔT_h=20 °C μ = 0.000307 m^2/s k= 0.676 W/mK

T_{ci}=10 °C T_{co}=70 °C ΔT_c=60 °C μ = 0.000628 m^2/s k=0.6327 W/mK $\dot{m}_{ref}$= 1kg/hr

Specific heat at mean temperature of hot water and refrigerant

Cph=4.208 kJ/kgK Cpc=4.179 kJ/kgK

Q=$\dot{m}_{ref}\,Cph\Delta T_c$= 69.68 W

From this by balancing the energy equations I get:

Q=$\dot{m}_{hot\,water}\Delta T$Cph $\dot{m}_{hot\,water}$=2.97 kg/hr

Now

Q=UA (LMTD) where A=πDL

Assume L=0.1 m, Di=0.0102 m, Do=0.0127 m

Therefore Ai=0.0032 m^2, Ao=0.0039m^2

$$\text{LMTD}\quad \frac{\theta_1-\theta_2}{ln\frac{\theta_1}{\theta_2}} \quad =49.5\ °C$$

Q=UA (LMTD)

From this:

Uassume=439.7 W/m^2°C

Now I have to find U using hi and ho

Tube side:

$\dot{m}/Ai$ =ρV(tube side) ρV= 0.08 kg/m^2s

Re = $\rho VDi/\mu$ Re=1.299

At this from the tube side graph of DQ kern[12]= 3

$$h_i = jH\left(\frac{k}{D}\right)N_{pr}^{\ 0.33}$$

From this hi=302.4 W/m^2°C

Shell Side design:

$\dot{m}/Ao$ = ρV(shell side) ρV= 0.2067 kg/m^2s

Re = ρVDo/ μ Do=0.0127 m and Re= 8.55

At this from the shell side graph of DQ kern[12]= 7

jH= $\frac{hoDe}{k}(\frac{\mu Cp}{k})$^(-0.33)

And ho=461.38 W/m^2°C **Uactual**=$\frac{h_i h_o}{h_i + h_o}$

Therefore **Uactual**= 200 W/m^2°C

So $\frac{U_{assume}-U_{actual}}{U_{assume}} \approx 0.5\ \%$

Hence the design is right.

3.2. Design of Evaporator

For designing evaporator, I took immersed helical coil heat exchanger because helical coil heat exchanger gave better effectiveness for laminar flow with low flow rate and also size of its also less compared to double pipe heat exchanger.

Table 2: Properties of refrigerating fluid (inside fluid property) and heat transfer fluid (outside fluid property)

Property	outside fluid property	inside fluid property
Viscosity µ kg/(m)(h)	3.12	4.9608
heat Capacity Cp	0.97282	0.9783
thermal Conductivity k	0.526836	0.501711
Prantal Number Npr	5.761182607	9.673199591
density	996.6	999.9
Mout (kg/h)	25	1
Tin	30	7
Tout	20	7

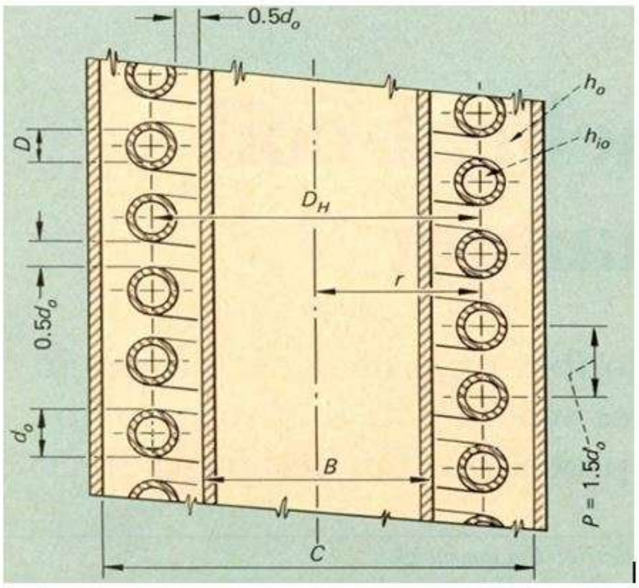

Figure 5: Immersed helical coil heat exchanger

Figure 6: Helical Coil Used in the System

I took standard ½" inch pipe for designing of evaporator. Pitch of helical coil is empirically taken as 1.5Do.

Table 3: Dimensions of Coil

Outside Diameter of Coil	Do (m)	0.0127
Inside Diameter of Coil	Di (m)	0.0102
Pitch	P (m)	0.01905

Figure 7: Developed Evaporator

For deigning of container I took hollow cylinder. Capacity of adsorber chiller is 25 liter/hour. From that for cylindrical container having height as 1.5 times diameter, I got diameter of cylinder as 0.29m and 0.435 m.

1) Diameter of helix Dh = C-(2Do) = 0.29-(2*0.127) = 0.2646 m
2) Radius of helix r= Dh/2 = 0.2646/2 = 0.1323 m
3) Inside diameter of hollow cylinder B=Dh-(2Do) = 0.2646-(2*0.127) = 0.2392 m

Table 4: Dimensions of evaporator

Outside Diameter of Container	C (m)	0.29
Inside Diameter of Container	B (m)	0.2392
Inside Diameter of Helix	Dh1 (m)	0.2519
Outside Diameter of Helix	Dh2 (m)	0.2773

Length of coil L, needed to make N turns:

$$L = N\sqrt{(2\pi r)^2 + p^2} = N\sqrt{(2\pi(0.1323))^2 + 0.01905^2} = 0.8310N\ m$$

Volume of annulus:

$$V_a = \left(\frac{\pi}{4}\right)(C^2 - B^2)pN = \left(\frac{\pi}{4}\right)(0.29^2 - 0.2392^2)0.01905N = 0.000402N\ m^3$$

Volume occupied by the coil:

$$V_a = \left(\frac{\pi}{4}\right)(\mathrm{Do}^2)L = 0.0001053N\ m^3$$

Volume available for the flow of fluid in the annulus:

$$V_f = V_c - V_a = 0.0002970\ m^3$$

Shell side equivalent Diameter (De):

$$D_e = \frac{4V_f}{\pi D_0 L} = 0.3582\ m$$

Mass Velocity of fluid:

$$G_s = \frac{M}{[(\pi/4)((C^2 - B^2) - ({D_{h2}}^2 - {D_{h1}}^2))} = 2367.12\ kg/m^2h$$

Reynolds number:

$$N_{re} = \frac{DG_s}{\mu} = 27.17$$

Heat transfer coefficient outside coil:

$$h_o = 0.6{N_{Re}}^{0.5}{N_{pr}}^{0.31}\left(\frac{k}{D_e}\right) = 79.16$$

Next step is to compute heat transfer coefficient inside coil.

For this fluid velocity:

$$u = {}^{q}/_{A_f}$$

$$where, q(volumetric\ flow\ rate) = {}^{M}/_{\rho} = {}^{1}/_{1000} = 0.001\frac{m^3}{h}$$

$$and\ A_f = \frac{\pi {D_i}^2}{4} = \frac{\pi 0.0102^2}{4} = 8.17 \times 10^{-5} m^2$$

So,

$$u = {}^{q}/_{A_f} = \frac{0.001}{8.17 \times 10^{-5}} = 12.32\frac{m}{h}$$

Reynold number tube side:

$$N_{re} = \frac{\rho u D_i}{\mu} = 25.15$$

Colburn factor (jH) for N_{re}=25.15 is 2. Now,

$$h_i = jH\left(\frac{k}{D}\right){N_{pr}}^{0.33} = 209.59$$

Corrected inside heat transfer coefficient:

$$h_{ic} = h_i\left[1 + 3.5\left(\frac{D}{D_h}\right)\right] = 237.19$$

Over all heat transfer coefficient U:

Coil wall thickness $x = \frac{D_o - D}{2} = 0.00125\ m$

Shell side fouling factor Ra = 0.0001 $hm^2{}^\circ C/kcal$

Tube side fouling factor Rt = 0.0001 $hm^2{}^\circ C/kcal$

Thermal conductivity of tube material k_c=344 kcal/h°Cm^2

Now, overall heat transfer is given by:

$$\frac{1}{U} = \frac{1}{h_o} + \frac{1}{h_{io}} + \frac{x}{k_c} + R_a + R_t$$

From this I got U=55.34 kcal/h°Cm^2

Log-mean temperature difference(LMTD):

$$\Delta T_m = \frac{[(T_{oi} - T_{ii}) - ((T_{oo} - T_{oi})]}{\ln[(T_{oi} - T_{ii}) - ((T_{oo} - T_{oi})]}$$

From this, $\Delta T_m = 17.52\,{}^\circ C$

Heat load:

$$Q = C_{pw}M_{out}(T_{in} - T_{out}) = 243.12\frac{kcal}{h}$$

I took heat load as 302 kcal/h

Area of Coil required:

$$Q = UA\Delta T_m$$

From this area is 0.2532 m^2

Number of turns required:

$$N = \frac{A}{\pi D_o L}$$

From this number of turns required as 7.63.

Length of Coil:
L=0.831N = 6.34 m
Total length is 6.34 m

3.3. Design of Condensor

Figure 8: Condenser

Min water = 1.13184 lbs/hr
Mout water = 88.183 lbs/hr
Condenser Inlet T_1 = Condenser Outlet T_2 = 158 °F
Cooling Water t_1 = 86 °F

Shell Side
Diameter = 5 in, Passes = 1
Tube Side
Diameter = 0.25 in, Passes = 4,
Number (n) and Pitch (P) = 1 and 1 in. square

Heat Balance
Tube side, Water Q = mL = 1.13184 * 970.4 = 1098.34 Btu/hr. (Latent heat of water = 970.4 Btu/lb)
Shell side, Water Q = MCw(t_2-t_1)
1098.34 = 88.183 * 1 * (t2 – 86)
t_2 = 98.45 °F(t_2 = Cooling water outlet)

LMTD :

$$\text{LMTD} = \frac{\Delta t1 - \Delta t2}{\ln(\Delta t1/\Delta t2)} = \frac{59.545}{72} = 65.5754$$

($t_1 = T_1 - t_2$), ($t_2 = T_2 - t_1$)

For Hot fluid - Tube Side Water:
Flow area, a' = $\left(\frac{\pi}{4}\right) D^2$ = 0.049107 in^2
Total Flow area a = (N*a')/(144*n) = 0.0000853 ft^2
Mass Velocity G = m/a = 13275.75 lbs / hr ft^2
Reynolds number Re = D*G/ μ = 280.1875
At Tavg = 158 °F:
D = 0.02083 ft, C = 0.97116 Btu / lbs°F, k = 0.38317 Btu /hrft°F
From Process Heat Transfer by D.Q.Kern , Table 24.
j_H = 1, hi = (j_H)*(k/D)*(Cμ/k)$^{1/3}$
hi = 24.96829 Btu / hr ft^2°F

For Cold fluid - Shell Side Water:
Flow area, a = D*C'B / 144P = 0.01736 ft^2
Mass Velocity G = m/a = 5079.34 lbs / hr ft^2
Equivalent Diameter De = $\frac{4*(P2-\frac{\pi}{4}D^2)}{12\pi D}$

De = 0.40341 ft
Reynolds number Re = D*G/ μ = 1135.62
At Tavg = 92.2275 °F;
C = 0.97166 lbs / fthr, De = 0.40341 ft,
k = 0.36086 Btu /hrft F
From Process Heat Transfer by D.Q.Kern , Table 28.
j_H = 12, ho = (j_H)*(k/De)*(Cμ/k)$^{1/3}$
ho = 18.1805 Btu / hr ft2 °F

Overall heat transfer co-efficient U
U = $\frac{hi*ho}{hi+ho}$ = 10.52026 Btu / hr ft^2 F
Heat Transfer Q = U*A*LMTD
Total Surface Area A = $\pi * D * L$
Length of Tube L = 2.0263 ft = 61.76 cm
So, **Approximate Length of Tube = 120 cm**

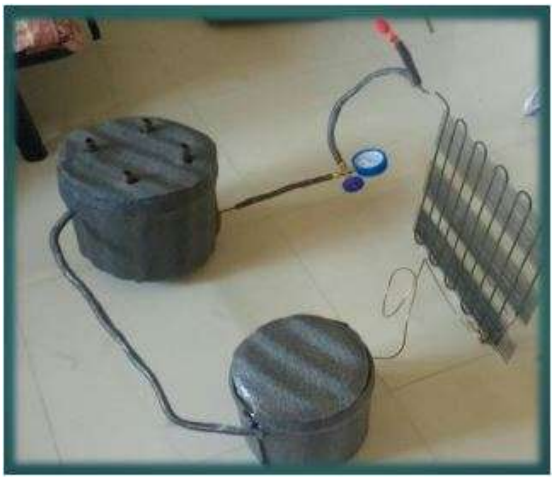

Figure 9: Developed Adsorption Refrigeration System

4. Conclusions and Future Plans

From the above design, it can be concluded that the system with required specifications can be developed in an efficient way and further it can be experimented.

As the adsorption refrigeration is a new concept, even though lots of research work is already carried out in this field, this field demands a lot of work yet and I am eager to contribute at my best to develop this field for the betterment of human welfare.

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