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DEVELOPMENT OF SEMI-CONTINUOUS SOLAR POWERED ADSORPTION WATER CHILLER FOR FOOD PRESERVATION

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ABSTRACT

Solar powered adsorption refrigeration systems have been preserved the food for the national requirement and also protected the environment. In this research article, the design and development of semi-continuous solar powered Adsorption water chiller for food preservation are presented. The design of the main components includes an adsorber bed, a condenser, an expansion device and an evaporator are performed by using heat transfer correlations. The outcomes of design are presented and discussed. The cooling produced in 10 kg of water was 554 kJ in 6 hours for the water flow of 170 kg/hour, 25° C condenser temperature and 65° C adsorber temperature.The fluctuation in system pressure is observed in the range of 30 kPa to 80 kPa for desorption and adsorption process during experimentation. The chiller performance was tested and compared with the earlier adsorption chiller. The comparison showed that proposed chiller has higher specific cooling power (SCP), low cycle time and low generation temperature due to activated carbon fiber-methanol pair and effective design of the system.

Keyword: *Solar Energy, Adsorption Refrigeration, Food Preservation, ACF-Methanol*

INTRODUCTION

Solar powered adsorption refrigeration system uses natural refrigerant and operates at low generation temperature which can be achieved by a flat plate collector. This system uses very low intrinsic parts which can be operated with no or little electricity. The main drawback of adsorption cooling is lower COP and higher thermal mass. Many researchers have made efforts for improvement in performance and reduction in overall mass of the system. Solar based cooling systems are intermittent due to nature of availability of solar energy. To develop a continuous cooling system, energy storage or double bed must be designed which ultimately adds cost and extra equipment. An adsorption chiller is thermally driven refrigeration system operated by solar energy or waste heat. The construction is the same to vapour compression refrigeration system except for thermal compressor. Other components like evaporator, condenser and expansion device are same. Due to the porous structure of adsorbent, refrigerant from the evaporator is adsorbed at low temperature and pressure which produces a cooling effect. Adsorbed mass of refrigerant is desorbed by supplying heat to adsorbent material and adsorbed by providing low temperature to the adsorbent. In this way, the intermittent cycle is operated, and cooling is produced by heating & cooling the adsorber bed periodically. Isobaric adsorption and desorption with temperature swing operation in adsorber bed produce refrigerating effect. Figure 1 shows the schematic diagram of adsorption chiller operation.

Adsorption chiller works on physiosorption phenomenon in which adsorbate gathers over the surface of the adsorbent. In this phenomenon, adsorption processes occur due to lower temperature of adsorber bed (20 – 35 °C) and desorption due to the higher temperature of adsorber bed (60 – 90 °C) which is attainable by solar energy. The refrigeration is produced by repeated heating & cooling of adsorber bed by hot & cold water. The solar-powered Adsorption chiller consists of ETC, water tank (hot/cold), adsorber bed, condenser, evaporator and capillary tube. In daytime solar energy is collected by ETC and subsequently converted into hot water. By keeping separate hot and cold water tanks, adsorber bed obtains heating and cooling for system process. Figure 2 represents the Clapeyron diagram for the thermodynamic cycle.

For refrigeration applications, ACF has a potential as an adsorbent due to its high adsorption capacity.ACF provides larger surface area and ease in packing which makes it favourable for adsorber bed (El-Sharkawy et al., 2016). Moreover, ACF performs the best in a cyclic adsorption/desorption in which time cycle is 1/5 to 1/10 of activated carbon (Wang et al., 1997).

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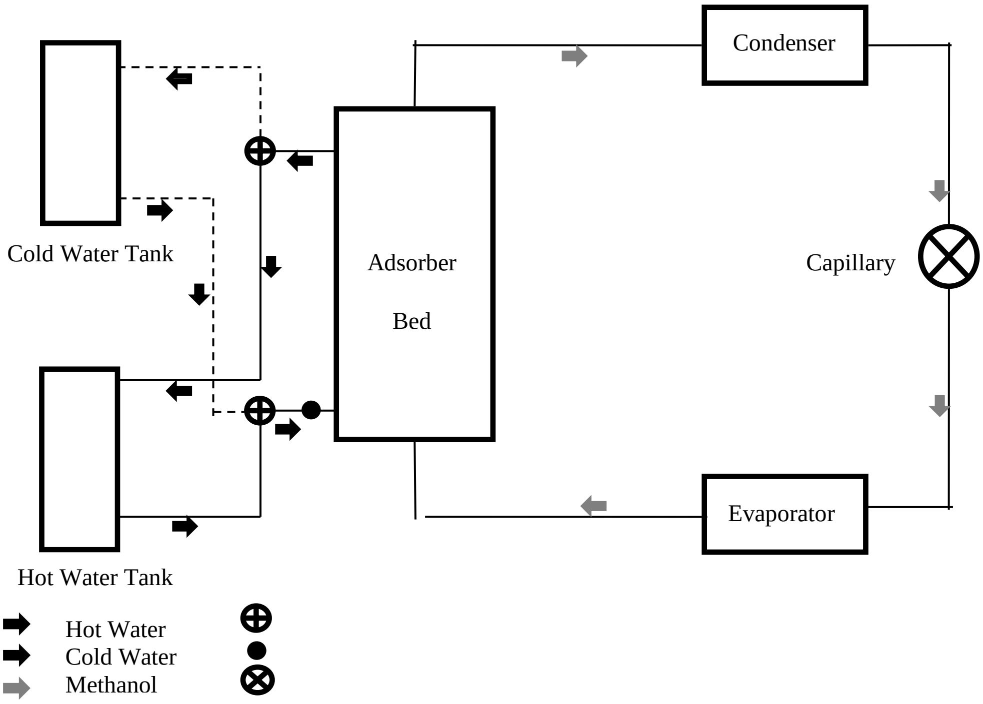

Figure 1. Schematic of adsorption chiller operation

Due to fast adsorption/desorption time and superior adsorption capacity of ACF, the heat & mass transfer are accelerates in the adsorber bed which in turn enhances the COP and the SCP of adsorption cooling system (Attan et al., 2011). The adsorption volume of ACF is found higher than activated carbon and silica gel when observed under nitrogen adsorption isotherm process. Additionally, ACF has a fully reversible isotherm in a whole range of relative pressure which supports better adsorption and desorption during cycle (Saha et al., 2006). ACF-Methanol pair is observed higher adsorption capacity (0.3406 kg/kg) than activated carbon-methanol (0.2565 kg/kg), activated carbon-ethanol (0.2008 kg/kg) and silica gel-water (0.1868 kg/kg) under analysis. Also, this pair has a lower mean adsorbent cost per kW of cooling and higher solar COP for food storage, medicine preservation and ice maker (Allouhi et al., 2015). ACF proved its better adsorptive properties then silica gel, zeolite and activated carbon regarding higher BET surface area (1.93 *106 m2/kg), higher total pore volume (10.28 *10-4 m3/kg) and lower average pore diameter (21.60 A°) in transient modelling. In a similar study, ACF-ethanol pair having 1.2 times more COP than silica-gel based chiller in performance evaluation (Saha et al., 2007).

Methanol as a refrigerant gives better performance due to its high vapour pressure, low boiling point and high latent heat (Nguyen et al., 2016). Wang et al. (2000) developed a solar-powered hybrid system of a water heater and ice maker which produced 30 kg hot water at 47.8 º C, and by night, it had a cooling capacity of 0.26 MJ/kg of adsorbent. Rivera et al. (2011) designed a prototype solar intermittent refrigeration system for SCP of 8 kg ice/day with NH_3- Lithium nitrate pair. Alghoul et al. (2009) analyzed a dual purpose solar continuous adsorption system for domestic refrigerator and water heating with Malaysian AC and methanol. From the study, they got (COP)dual system -ice 0.091, (COP)cycle-ice 0.44, (COP)dual system –domestic hot water 0.73, (COP)dual system 0.821 with cost and a payback period of this system.

From the literature survey, it was concluded that total cycle time for refrigeration is either 24 hours or few minutes', i.e. intermittent or continuous system. In this article, the design and development of semi-continuous solar powered adsorption refrigeration technology are discussed, and ACF-Methanol has been selected as working for present design. The purpose of the study was to develop a semi-continuous system (4 to 7 hours cycle time) with higher SCP (75 kJ/kg) operated by low generation temperature (65ºC). The proposed design of solar-powered adsorption chiller for food preservation has potential to adopt for local cooling at farms for reducing spoilage of food before its transfers to cold storage or market.

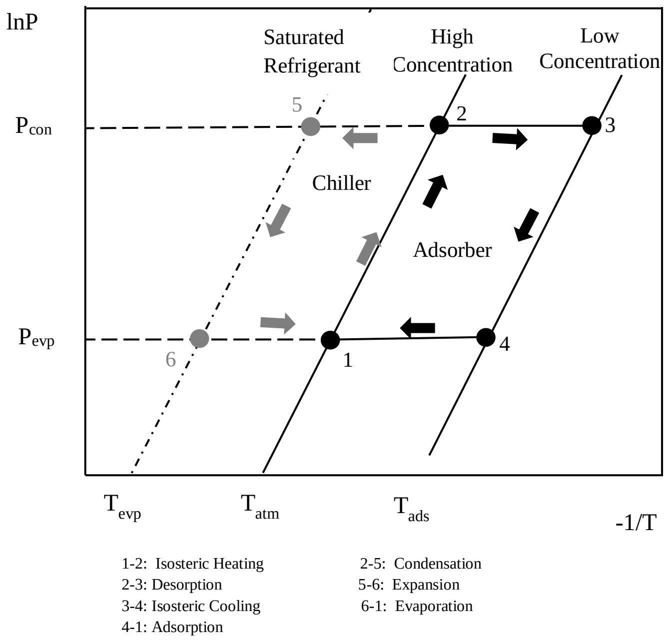

Figure 2. Clapeyron diagram for the adsorption chiller thermodynamic cycle

DESIGN PROCESS

The design of adsorption refrigeration system relies on knowledge of chemical science, heat and refrigeration technology. With the physiosorption principal and necessity of refrigerating effect, the design of the system was performed. For food preservation, the temperature of storage system should be maintained at 10° C (i.e. vegetables and fruits can be preserved at this temperature for one or two weeks) (Zhai et al., 2013). An Adsorption chiller was designed for producing water temperature at 8-10 ° C in 4 to 6 hour. The cycle time for this whole process was 360 minutes, and hence it works as a semi-continuous system. The size of the system is decided by the adsorption capacity of adsorbent. For the adsorption capacity of the working pair, the experimental setup was developed, and the value came out as 0.44 kg/kg.

The required mass of refrigerant is determined by the cooling effect and adsorption capacity. The gained mass decides the size of the chiller. In India as well as other parts of the world, the solar water heater is based on flat plate collector which can produce water temperature up to 55-70° C. Adsorption working pair is chosen in such a way that it will give satisfactory results at such low generation temperature. This system can easily be coupled with a solar water heater to give twin advantages of hot water and refrigerating effect (Sumathy et al., 2013).The system is designed in a way that is effective, reasonable, compact and eases in manufacture with readily available resources.

The design of adsorption chiller is based on the following assumption,

- Specific heat & density are constant
- Adsorbent bed is composed of uniform size
- particles and the bed porosity is constant
- Heat transfer in the heating /cooling fluids and the metal is one dimensional
- No environmental effect and steady state during operation

Mass of Methanol and ACF

The mass of methanol is achieved by cooling requirement of product, i.e. water,

$$\mathrm{Q}_{\mathrm{ref}} = m_w c_{pw} dT \tag{1}$$

$$m_{ref} = \frac{\mathrm{Q_{ref}}}{\mathrm{h_{fg}}} \tag{2}$$

Either by using Dubinin Astakhov correlation or physical measurement, the value of adsorption capacity is achieved.

$$x = \frac{m_{ref}}{m_{ads}} \tag{3}$$

From above equation, the mass of ACF is calculated.

Adsorber Bed Design

In this study, shell and tube heat exchanger is chosen for adsorber bed. From literature survey and heat transfer analysis, the heat exchanger dimensions are identified. In this research, diameter and length of shell and tube are given, and a number of the tubes is then calculated. Also, the mass flow rates, the temperatures of refrigerant and heat transfer fluid are identified. Using TEMA code and heat transfer correlation, the final dimensions of the heat exchanger are available in Table 1. The bed schematic and photograph is shown in Figure 3.

Table 1. Calculated dimensions for adsorber bed

Parameter	**Specification**
Heat Exchanger	Shell and tube type
Area	0.22 m^2
Shell	154 mm in diameter,750 mm length
Tube	9.5 mm in diameter, cu.,26 nos.

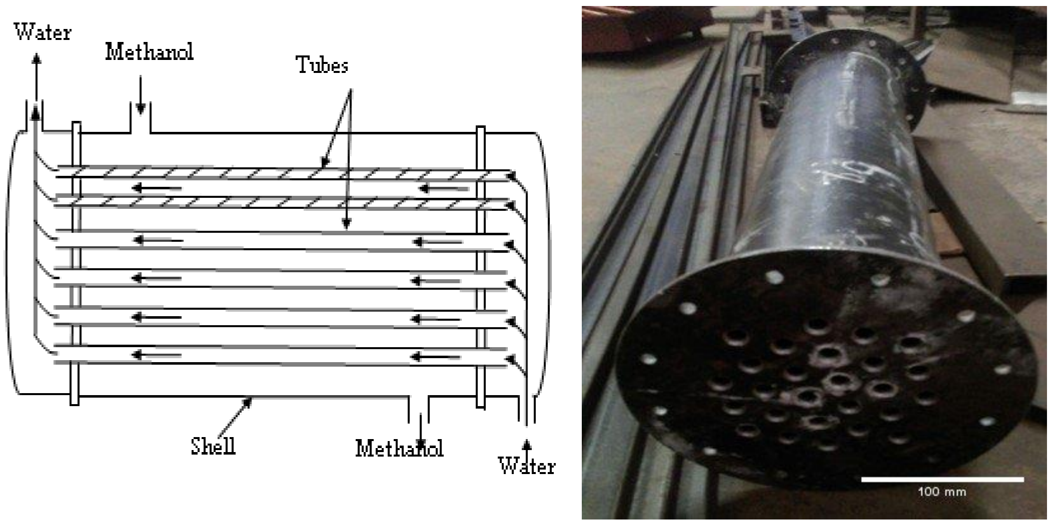

Figure 3. Shell and tube type adsorber bed

Heat Duty in adsorber bed

$$\mathrm{Q_{ads}} = m_m c_{pm} dT_m \tag{4}$$

Calculate hi (h clean) for tube side by using Nusselt correlation,

Reynolds number

$$Re = \frac{\rho u D}{\mu} \tag{5}$$

Apply for laminar flow

$$Nu = 0.332 Re^{0.5} \mathrm{Pr}^{0.33} \tag{6}$$

And

$$\mathrm{h_i} = \frac{\mathrm{Nuk}}{\mathrm{D}} \tag{7}$$

$$\mathrm{h_i} = \mathrm{h_{clean}} \tag{8}$$

Calculate h_{foul} by considering the effect of fouling factor,

$$R_f = \frac{1}{\mathrm{h_{foul}}} - \frac{1}{\mathrm{h_{clean}}}$$

The value of Rf is 0.001 for city water (Standards of the Tubular Exchanger Manufacturers Association, 2007).Using theory of adsorbent thickness (≤ Ycritical) for better flow of methanol and ease in penetration, three layers of ACF is taken in the experiment (Mitra, 2016). For heat transfer from the tube fluid to shell refrigerant, four thermal resistance are involved- inside, cu tube, ACF and outside (Holman, 2008)

$$R_i = \frac{1}{\mathrm{h_{foul} A_i}} \tag{9}$$

$$R_{cu} = \frac{\ln\left(\frac{\mathrm{R_o}}{\mathrm{R_i}}\right)}{2\pi \mathrm{k_{cu} L}} \tag{10}$$

$$R_{acf} = \frac{\ln\left(\frac{\mathrm{Ra_o}}{\mathrm{Ra_i}}\right)}{2\pi \mathrm{k_{acf} L}} \tag{11}$$

The overall heat transfer coefficient can then be expressed regarding these four resistances,

$$\mathrm{U_o} = \frac{1}{(\mathrm{R_i} + \mathrm{R_{cu}} + \mathrm{R_{acf}} + \mathrm{R_o})\mathrm{A_o}} \tag{12}$$

And the area of the heat exchanger is calculated to be

$$A = \frac{\mathrm{Q_{ads}}}{\mathrm{U_o\ F\ LMTD}} \tag{13}$$

Where F is a correction factor, and its value is unity (Holman, 2008). From the area obtained, one can find the number of tubes for the heat exchanger. The radius of ACF must be less than the critical radius for better heat transfer and smaller pressure drop inside the shell. The pressure drop must be less than allowable pressure drop (Holman, 2008 & Donald Q.Kern, 2009). Three layers of ACF is wrapped over the tubes, allowing the space between tubes for methanol vapour to flow in the shell during desorption, hence the pressure drop in the shell is less compared to the allowable drop.

Condenser Design

The Water cooled heat exchanger is chosen as the condenser for the present study. In this configuration, the condensing effect is efficient, and there is flexibility to vary condenser temperature for experimentation. The final dimensions of the condenser are listed in Table 2. The schematic and image of the condenser are shown in Figure 4.

Table 2. Calculated dimensions for condenser

Parameter	value
Heat Exchanger	Shell and Tube type
Area	0.043 m^2
Shell	Box type, 500 mm length, 500 mm width and 120 mm
Tube	12.7 mm diameter cu. ,1.5 m. length

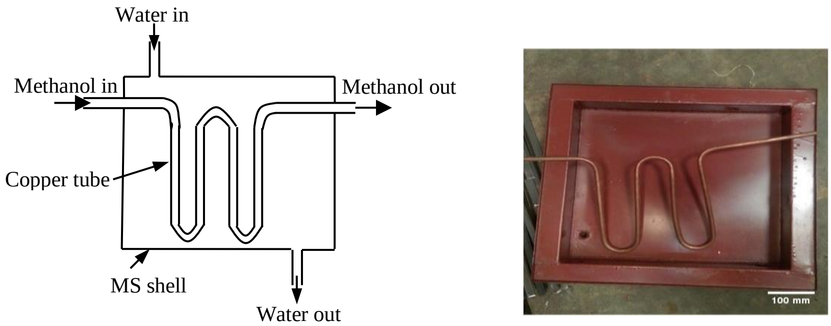

Figure 4. Water-cooled condenser

Methanol vapour is coming from adsorber bed after desorption and goes to a water-cooled condenser. The temperature and flow rate of methanol vapour depends on cycle time and generation temperature in the adsorber bed. Energy balance between methanol vapour and water in the condenser gives the area of tubes. In this work, quantity and flow rate of water in the condenser is given. The design calculation for the condenser is as follow (Donald Q.Kern, 2009),

Heat capacity in condenser

$$Q_c = m_w c_{pw} dT \tag{14}$$

Considering counter flow arrangement in the condenser, the LMTD can be found.

For the overall heat transfer coefficient, one first calculates the tube side and shell side heat transfer coefficients,

Flow area per pipe

$$A_t = \frac{\pi}{4} D^2 \tag{15}$$

Mass velocity

$$G_t = \frac{m_w}{A_t} \tag{16}$$

Reynolds number

$$Re_t = \frac{DG_t}{\mu} \tag{17}$$

Tube side heat transfer coefficient

$$h_i = j_h \left(\frac{k}{D}\right) (c_p\mu/k)^{0.33} (\mu/\mu_w)^{0.14} \tag{18}$$

Where $j_h = (h_i D/k) (c_p\mu/k)^{0.33} (\mu/\mu_w)^{0.14}$

Including the thickness of tube, the corrected heat transfer coefficient is given by

$$h_{io} = \frac{h_i d_i}{d_o} \tag{19}$$

Shell side heat transfer coefficient

We first assume a shell side heat transfer coefficient (ho) and by trial and error, fix it using the tube wall and condensate film temperatures.

The overall heat transfer coefficient (Uc) for a clean tube can be calculated by

$$U_c = \frac{h_i h_o}{h_i + h_o} \tag{20}$$

Consider the dust coefficient (hd),

$$h_d = \frac{1}{R_d} \tag{21}$$

The overall design Coefficient (Ud) is then given by

$$U_d = \frac{U_c h_d}{U_c + h_d} \tag{22}$$

Now the area required for the condenser tube (Ac) is given by,

$$A_c = \frac{Q_c}{U_d \text{ LMTD}} \tag{23}$$

The calculated area gives the total tube length required for the condenser.

Expansion Device

An adsorption refrigeration system with ACF-methanol as the working pair works under vacuum so that the capillary tube is sufficient to maintain the pressure difference in the system. The dimensions are calculated for the capillary tube are summarized in Table 3. In the capillary tube, the pressure drop is due to friction and flashing effect. This pressure drop is directly proportional to the length of the tube and inversely proportional to its diameter. In this chiller, total pressure drop observed in the capillary is 51 kPa, i.e. from condenser pressure 55 kPa to evaporator pressure 4 kPa.

The design of capillary tube implies selection of bore and calculation of length for maintaining the required flow at the given pressure difference between condenser and evaporator. Following is the design procedure for capillary tube:

For a given cooling load, identify the mass flow rate (mm) of methanol. Select the bore (Dcap) size from available standard capillary size.

Table 3. Calculated parameter for capillary tube

Parameter	**value**
Bore (D)	2.54 mm
Mass flow Rate (m°)	0.00087 kg/sec
Mass Velocity (G)	171.69 kg/sec.m^2
Length (L)	1142.52 mm

Assume methanol is entering the capillary tube is a saturated liquid. At the condenser pressure, the temperature is Tc, and at the evaporator pressure, the temperature is Te. Now divide the temperature drop from Tc to Te in some parts. The design steps based on isenthalpic flow are as follow (Arora, 2010)

Quality of methanol at the end of decrement,

$$x_1 = \frac{\mathrm{h'}_{\mathrm{c}} - \mathrm{h'}_{\mathrm{f1}}}{\mathrm{h'}_{\mathrm{fg1}}} \tag{24}$$

Calculate the specific volume

$$\mathrm{v}_1 = \mathrm{v}_{\mathrm{f1}} + \mathrm{x}_1\mathrm{v}_{\mathrm{fg1}} \tag{25}$$

Determine the cross-sectional area of capillary

$$\mathrm{A} = \frac{\pi}{4}(\mathrm{D}_{\mathrm{cap}})^2 \tag{26}$$

Determine the flow velocity by continuity equation

$$\frac{\mathrm{u}}{\mathrm{v}} = \frac{\mathrm{m_m}}{\mathrm{A}} = \mathrm{G} \quad (\mathit{Asuume\ mass\ velocity}) \tag{27}$$

$$\mathrm{u_c} = \mathrm{m_m}\frac{\mathrm{V_c}}{\mathrm{A}} \tag{28}$$

$$\mathrm{u_1} = \mathrm{m_m}\frac{\mathrm{V_1}}{\mathrm{A}}$$

And,

By iterations, one obtains h1 for the Fano line flow

$$\mathrm{h'}_1 = \mathrm{h'}_{\mathrm{c}} - \frac{\mathrm{u_1}^2}{2} \tag{29}$$

Calculate the pressure drop by momentum equation

$$\Delta P_a = \frac{m_m}{A}(u_c - u_1) = G\,(u_c - u_1) \tag{30}$$

Determine the pressure drop due to friction

$$\Delta p_f = \Delta p - \Delta p_a \tag{31}$$

Now relate the pressure drop to friction factor,

$$\Delta p_f = (\rho f L u^2)/(2D) \tag{32}$$

Simplify the equation to give

$$\Delta p_f = Y f u \Delta L \tag{33}$$

Where Y= (G/2D) and f = (0.324/Re0.25)

In this way, ΔL is calculated and summation of ΔL will give the total length of the capillary tube.

Evaporator

After reviewing literature for water chiller, it is found that immersion coil type heat exchanger is the best configuration. The mass of methanol and quantity of product decide the size of the heat exchanger. Thermal and mechanical design of coil type heat exchanger has performed accordingly. The final dimensions obtained are shown in Table 4.

Table 4. Calculated parameters for evaporator

Parameter	**value**
Heat Exchanger	Shell and coil type
Area	0.24 m^2
Shell	200 mm in diameter, 400 mm high
Tube	12.7 mm in diameter, 6 m. long

The schematic and photograph of the evaporator are shown in Figure 5. The helical coil heat exchanger is best suited for laminar flow and limited space. The design of helical coil and shell is determined by the mass velocities of the fluids. The following are the steps in the design of evaporator (Patil, 1982).Calculate the overall heat transfer coefficient,

For laminar flows, the shell side heat transfer coefficient (h_o) is given by,

$$(h_o \frac{D_e}{k}) = 0.6\,(Re)^{0.5}(Pr)^{0.31} \tag{34}$$

And the tube side heat transfer coefficient (hi),

$$h_i = j_h \left(\frac{k}{D}\right)(Npr)^{0.33} \tag{35}$$

Now, the corrected tube side coefficient (hio) is given by,

$$h_{io} = \frac{h_i D}{D_o} \tag{36}$$

The overall heat transfer coefficient

$$\frac{1}{U} = \frac{1}{h_o} + \frac{1}{h_{io}} + \frac{x}{k} + R_c + R_s \tag{37}$$

Where x is the coil thickness, k is the thermal conductivity of coil metal, Rc & Rs are the fouling factors for coil and shell respectively.

Now the area required for the helical coil is

$$A_{coil} = \frac{Q_{ref}}{U\ LMTD} \tag{38}$$

Where Qref is the cooling load in the evaporator. The calculated area gives the total tube length required for the evaporator.

The heat transfer fluid is water which is 250 litre in capacity. Two separate tanks are provided for hot and cold water which is supplied in a cycle to the bed as shown in Figure 6. Water tanks and other equipment are enclosed with insulation. The specifications of insulation are listed in Table 5. The final calculated dimensions with specifications of adsorption chiller are summarized in Table 6.

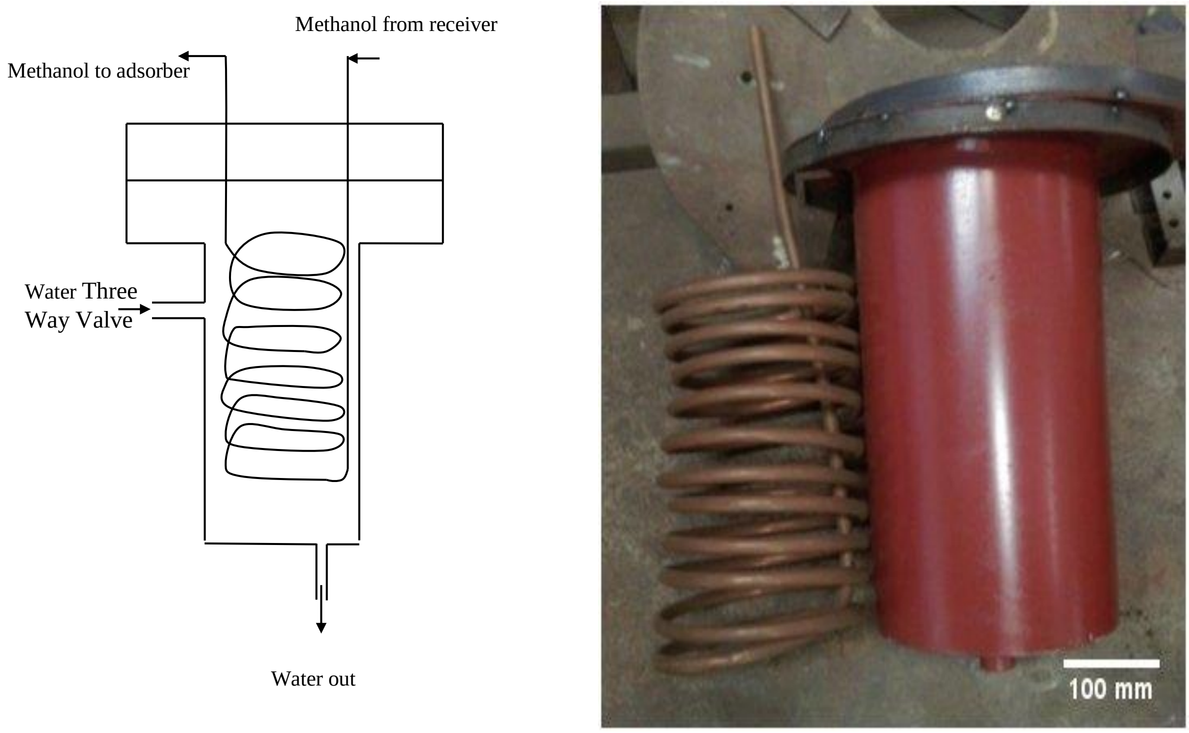

Figure 5. Immersion coil type evaporator

Figure 6. Hot and cold water tank

Table 5. Specifications of insulation

Insulation Cover	**Specification**
Water Tank	Rock Wool, density 48 kg/ m^3 , 100 mm thick
Evaporator	Puf, density 40 kg/ m^3, 50 mm thick
Condenser	Rock Wool, density 48 kg/ m^3 , 50 mm thick

Table 6. Final specification of adsorption chiller

Component	**Type**	**Specification**
Adsorber bed	Shell and tube	Shell: 154 mm in diameter, 750 mm long Tube: 3/8 inch cu. ,26 tubes
Evaporator	Helical coil immersion water cooled	Shell-12 L Tube: ½ inch cu.6 m long,11 turns, 19.05 mm in pitch
Condenser	Shell and tube Water cooled	Box type shell- 25 L Tube: ½ inch cu. , 1.5 m long
Hot/cold water tank	Cylindrical insulated tank	250 L capacity metal tank with insulation

EQUIPMENT DESCRIPTION

The schematic and photograph of semi-continuous solar powered adsorption chiller are shown in Figure 7 and Figure 8. This system comprises a hot water tank with temperature regulator to pretend solar water heater. With this arrangement, experimentation can be conveniently conducted at any time and any location for simulated conditions. For precise control, there was a thermostat with temperature relay attached to a water tank. With this arrangement, manual control in the mass flow rate of water and temperature control of hot water and cold water was possible.

Also, the frequency of water supply (hot water timing / cold water timing) is maintained. For measuring the temperature at different locations of the system, calibrated K type thermocouples were installed. Dial pressure gauge was used to provide system pressure during operation. The uncertainty analysis of instruments and set up are listed in Table 7. There was also temperature controller provided in the condenser to monitor the real conditions. The reduction in temperature of water kept in the evaporator shell gave cooling effect produced in each cycle. The overall system design has been developed for better cooling effect, and the best combination of parameters for efficient performance has been identified. To measure the drop in evaporator temperature, cyclic heating & cooling of adsorber bed were required for a specific time. Heating was observed in adsorber bed by hot water and cooling by tap water. For achieving the chilling effect in water, the temperature of hot & cold water, the frequency of water supply and mass flow rate of the same were fixed. The hot water is flowing through adsorber bed for 10 min, measure the temperature at adsorber bed as well as the evaporator and the system pressure. The same mentioned method is followed by supplying cold water for 30 min to the bed.

Table 7. Uncertainty analysis-list of instruments with accuracy, range and percentage of errors

Instrument	Accuracy	Range	% Error
K Type Thermocouple	± 0.1° C	0-400 °C	0.5
Dial Pressure Gauge	± 2 cm of hg	0-76 cm	1
Level Indicator	±0.5 mL	0-50 ml	0.5
Weighing Balance	±0.01 gm	0-500 gm	0.01
Hour Meter	±0.01 hour	0-9999.99 hour	0.01
Measuring Beaker	±0.5 mL	0-1000 ml	0.5

The total uncertainty found for adsorption capacity is 0.25%, Cooling Effect is 0.01 %, COP is 8% and SCP is 1.71 %.

EXPERIMENTAL RESULTS

The chiller is designed to achieve temperature drop of 10 ° C in the evaporator which can be beneficial for food preservation. The temperature drop in the evaporator is 9° C achieved by water flow rate of 170 kg/hour and at a condenser temperature 25°C. The variation in system pressure is observed from 230 mm of Hg. (30 kPa) to 600 mm of Hg. (80 kPa) for desorption and adsorption process during experimentation. The total time taken for a cooling effect of 554 kJ is 6 hours which can be reduced by maintaining the flow rate of water. The maximum COP of the system can be observed by electronics controls and time of water supply. The observed results are mentioned in Table 8 in which evaporator temperature is decreasing in each cycle. In the results, 9° C drop in evaporator attained in 6 hours. The variation in evaporator temperature and system pressure is shown in Figure 9 and Figure 10. The drop in temperature of evaporator reflects the refrigerating effect produced by semi-continuous adsorption water chiller. The fluctuation in system pressure is due to desorption and adsorption mechanism observed by heat transfer fluid (hot water/tap water) during the process. The test results are compared with previous studies as shown in Table 9. The comparison revealed that proposed adsorption system has lesser cycle time, the low mass of adsorbent and refrigerant for higher SCP and COP. Even generation temperature is less which can help in selecting the low power heat source and conventional solar collector for this chiller. The proposed system is produced desire cooling effect by keeping moderate adsorption temperature (i.e. by using atmospheric air or tap water) and moderate desorption temperature(Solar Water Heater). The presented adsorption chiller is better by selection of working pair, design and optimization of working parameters.

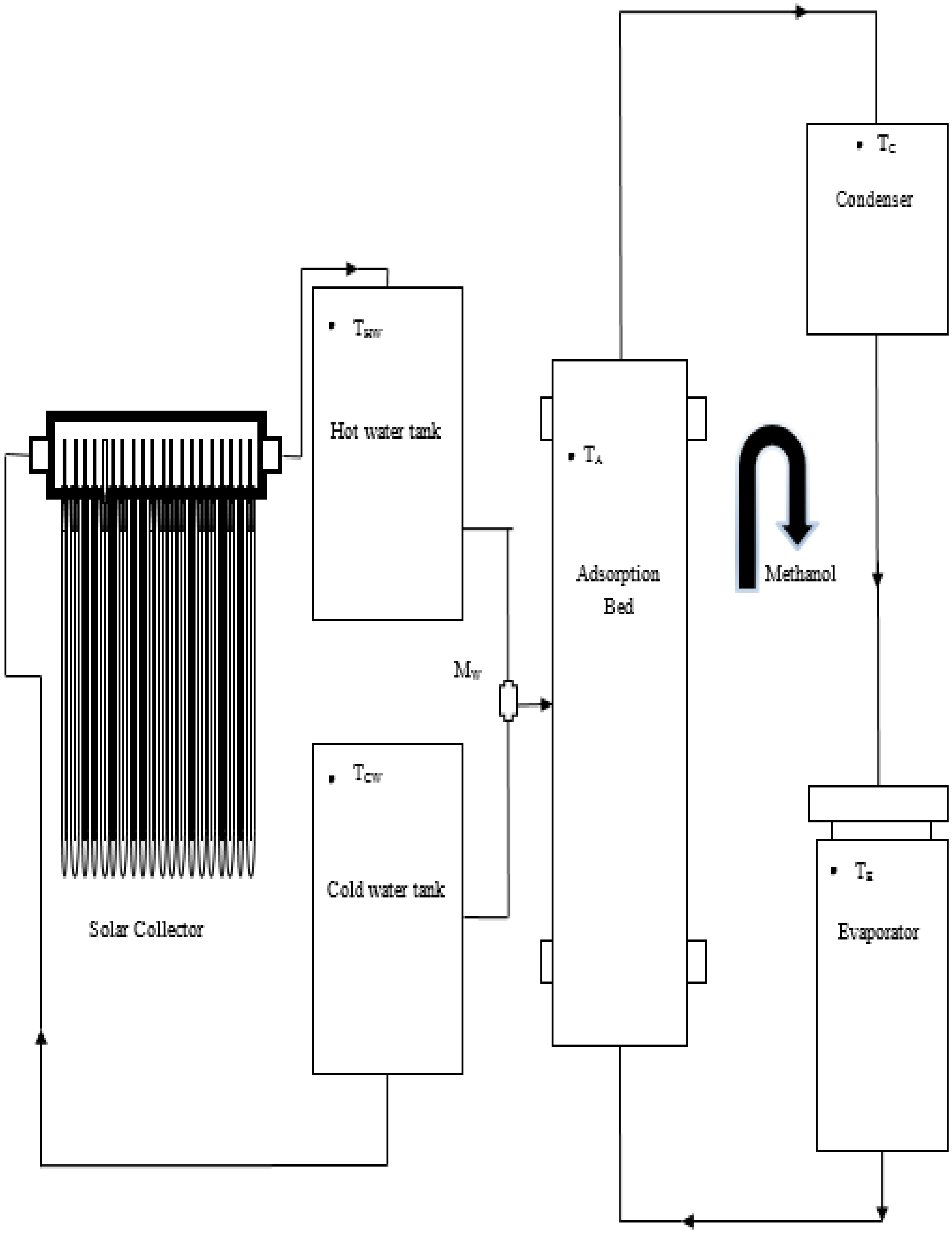

Figure 7. Schematic of Adsorption water chiller

Table 8. Testing results

TIME	PRESSURE (mm of Hg vacuum)	Water Flow Rate (kg/hour)	Avg. Hot Water Temp(Degree C)	Avg. tap Water Temp(Degree C)	Drop in Evaporator water Temp. (Degree C)
06.30pm	490.00		49.45	29.55	28.20
06.40	370.00	177.00	49.30	30.95	28.30
06.55	475.00	150.00	53.75	32.05	27.90
07.10	475.00		57.90	33.80	27.30
07.20	300.00	144.00	57.95	34.05	26.90
07.35	475.00	142.00	61.95	34.35	26.50
07.50	475.00		64.40	34.45	25.70
08.00	210.00	206.00	63.20	34.55	25.60
08.15	475.00	220.00	64.40	34.55	25.00
08.30	475.00		64.25	34.00	24.50
08.40	225.00	202.00	61.30	34.40	24.60
08.55	475.00	170.00	61.25	34.30	24.10
09.10	475.00		61.20	34.35	23.80
09.20	280.00	150.00	59.05	34.30	23.60
09.35	475.00	170.00	60.20	34.25	23.50
09.50	475.00		60.10	34.40	23.00
10.00	265.00	165.00	59.65	34.55	22.60
10.15	475.00	215.00	60.10	34.45	22.40
10.30	475.00		60.05	34.55	21.70
10.40	300.00	205.00	59.20	34.60	21.70
10.55	475.00	222.00	60.35	34.60	21.20
11.10	475.00		60.25	34.55	20.80
11.20	275.00	160.00	60.20	34.60	20.70
11.35	475.00	221.00	60.40	34.55	20.10
11.5	475.00		60.30	34.50	20.00
00.00	290.00	171.00	59.65	34.60	20.00
00.15 am	475.00	221.00	60.50	34.60	19.80
00.30	475.00		60.50	33.80	19.20

Table 9. Comparison of results [2, 21-26]

Parameter/ Author	A. Boubakri	L. R. Rodrı et.al.	H. Z. Hassan	R. Z. Wang et.al.	M. Pons	R.Suleiman et.al.	R. Z. Wang et.al.	E.E. Anyanwu	Bhargav et.al.
Product Load	5.2.0 kg of Ice/day	2.20 MJ/ sqm. Per day	12.15 MJ per cycle	8.00 kg of Ice/day	30.00kg of Ice/day	4814.83 KJ	10.00kg of Ice/day	3.00 kg of Ice/day	9.00° C drop in 10.00 kg of water
Cooling Effect (KJ)	2392.00	2200.00	12150.00	3680.00	13800.00	4814.83	4600.00	1380.00	554.00
Cooling Effect (W)	27.68	25.46	143.75	42.59	159.72	55.73	53.24	15.97	27.18
Generation Temp.(° C)	95.00	120.00	120.00	120.00	100.00-110.00	80.00	98.00	100.00	65.00
Adsorption Temp. (° C)	22.00	18.00	30.00	30.00	25.00	25.00	20.00	20.00	33.00
Cycle Time (hour)	24.00	24.00	24.00	24.00	24.00	-	24.00	20.00	6.00
Condenser Temp. (° C)	20.00	-	35.00	40.00	25.00-40.00	25.00	30.00	35.00	25.00
Evaporator Temp. (° C)	-10.00	0.70	-5.00	-10.00	-3.00	0.00	-2.00	-10.00	20.00
COP	0.14	0.09	0.62	0.55	0.12	0.601	0.07	0.02	0.43
SCP (W/kg)	1.38	3.53	3.19	85.18	1.22	2.13	1.90	1.90	60.40
Mass of AC (kg)	20.00	7.20	45.00	0.50 kg ACF	130.00	26.07	28.00	8.40	0.45 kg ACF
Mass of Methanol(kg)	-	2.20	10.79	-	-	8.10	8.00	-	0.65
Collector	FPC – 1.00 sqm.	CPC - 0.55 Aperture Area	-	-	FPC – 6.00 sqm	FPC – 2.00 sqm	Heat pipe ETC – 2.00 sqm	FPC – 1.20 sqm	Water Tank with Electrical Heater

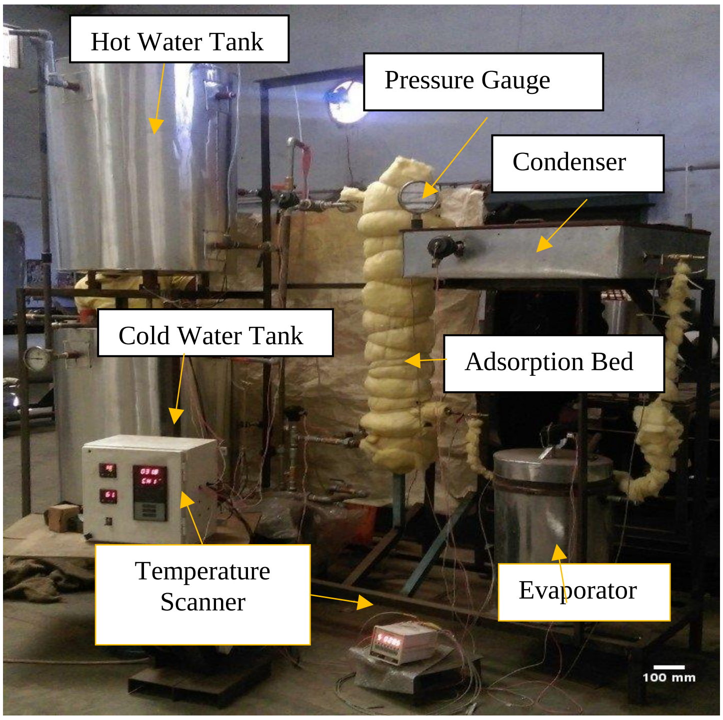

Figure 8. Photograph of Adsorption Chiller

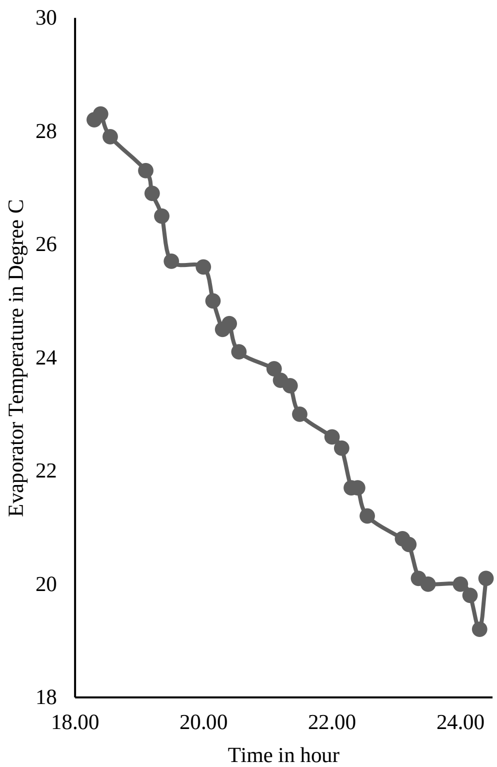

Figure 9. Variation in Evaporator temperature with time

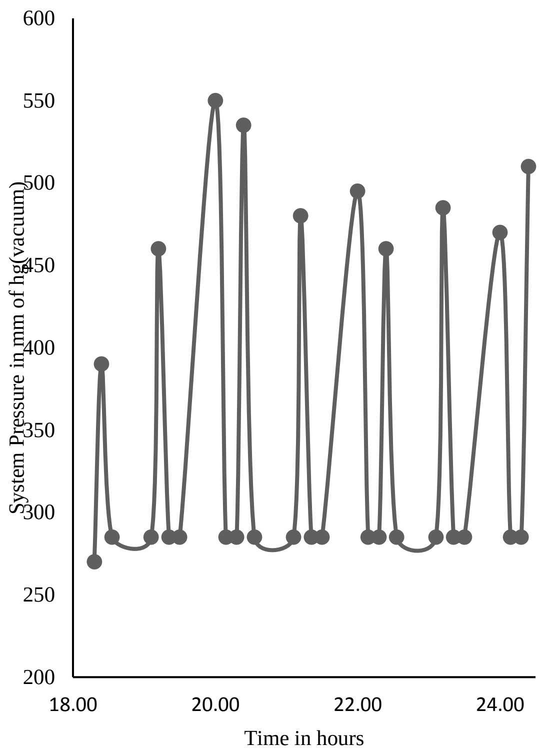

Figure 10. Variation in system pressure with time

CONCLUSION

The solar-powered adsorption refrigeration system is undoubtedly a better option than a conventional chiller because of its eco-friendly nature, low cost and simplicity. The objective of the study is to develop semi-continuous adsorption water chiller powered by the residential solar water heater. From the obtained results it is concluded that hybridizing of the solar water heater with adsorption refrigerator can satisfy water heating and food preservation requirement. The working environment in term of the temperature of hot water & cold water, the flow rate of water and temperature of the condenser can easily manage with available resources for the production of cooling effect through the developed chiller. This chiller is expected to be reasonable in INDIA in upcoming time for short-term storage of food at the farm. The present system can achieve high SCP at low generation temperature by adsorptive properties of working pair and efficient design of chiller. The off-site observation and control can be possible by proper electronic instrument and software with the system.

NOMENCLATURE

A	Heat transfer area (m^2)
ACF	Activated carbon fiber
BET	Brunauer–Emmett–Teller
Cp	Specific heat ($kJ\ kg^{-1}\ K^{-1}$)
COP	Coefficient of performance
Cu	Copper
D	Inside diameter of tube (m)
D_{cap}	Capillary bore
D_e	Shell side equivalent diameter
D_o	Outside diameter of tube
dT	Temperature drop in water
dT_m	Temperature difference in methanol
ETC	Evacuated tube collector
G	Mass velocity
h	Convective heat transfer coefficient
h’	Enthalpy
i	Inside
k	Thermal conductivity ($W\ m^{-1}\ K^{-1}$)
L	Length of pipe (m)
LMTD	Log mean temperature difference
m_{ref}	Mass of Methanol (kg)
m_w	Mass flow rate of water ($kg\ s^{-1}$)
h_{fg}	Latent heat of methanol
m_{ads}	Mass of ACF(kg)
m_m	Mass flow rate of methanol($kg\ s^{-1}$)
Nu	Nusselt number
o	Outside
Q	Heat flux (kW)
Pr	Prandtl number
R	Resistance to heat transfer
Re	Reynolds number

SCP	Specific cooling power (kJ/kg)
T	Temperature (K)
TEMA	Tubular exchanger manufacturers association
u	Free stream velocity of the fluid (m s^{-1})
v	Specific volume
x	Adsorption capacity (kg of methanol / kg of ACF)
ρ	Density (kg m^{-3})
ΔP	Pressure drop
ΔL	Incremental length of capillary tube
μ	Dynamic Viscosity

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