Proceedings World Geothermal Congress 2020+1
Reykjavik, Iceland, April - October 2021

Hybdrid Geothermal and Solar Setup for Low Enthalpy Geothermal Reservoirs: A Case Study from Gujarat, India

Namrata Bist and Anirbid Sircar

Mailing address, SPT, Pandit Deendayal Petroleum University, Raysan, Gandhinagar, Gujarat, India

E-mail address,Namrata.bist@spt.pdpu.ac.in

Keywords: Hybrid, Geothermal, Efficiency, Enthalpy, Solar, Carnot principle

ABSTRACT

The usage of non renewable energies has made our environment unsustainable. In order to minimize this threat we must use renewable energies which are self replenished & less environmentally damaging. Geothermal energy is continuous renewable energy which is advantageous over other high enthalpy but intermittent sources of energy such as solar, wind, tide etc. The thermal efficiency of a geothermal electric plant is around 10-23%. By hybridizing it, the average temperature of the working fluid and the efficiency and electric output can be significantly increased (according to Carnot principle). Such hybridization can also decelerate the depletion of heat content of the geothermal reservoir and hence extend its lifespan. This paper focuses on designing a hybrid Geothermal and Solar power plant by utilizing shortcomings of one as the advantages of other. Enthalpy improvement calculations are performed both using software (Aspenhysis+) and hand calculations. This study also attempted to perform thermal efficiency calculations of the system. This is attempted by simulating the enthalpy values of refrigerant in software (miniRefprop) and performing hand calculations for power generation.

1. INTRODUCTION

Mankind is dependent on fossil fuels from generations and is now aware of their drawbacks such as restricted access, depleting deposits, environmental consequences, etc. Thus, renewable energy sources are now naturally getting attention as solution to most of our problems. Zhao and Guo (2015) categorize wind, biomass and solar energy as ones with greatest socio-economic and eco-friendly benefits. These energy sources, when combined, would turn out to be more feasible for electricity generation than their stand-alone usage, as suggested by Baredar *et al.* (2010). When various renewable sources are integrated as a single hybrid system, the strengths of one source offsets the shortcomings of another one (Gonzalez *et al.*, 2015). As a result, we obtain better efficiency, higher total output and lowered overall cost. Rahman *et al.*, (2014) emphasizes that majority of villages in developing countries like India have abundant renewable sources, for example, solar radiation, biomass, biogas and wind energy. Most rural areas have already some kind of experience of using solar PV cells but recent investigations by various researchers have suggested integration of solar PV with biomass energy. They further establish the fact that use of separate renewable energy sources is often costly and unreliable. According to Eziyi and Krothapalli (2014), such integrations will empower local resources and at the same time, lower the dependency of food prices on oil prices. Similarly, villages in remote areas have increasingly been employing wind mills for off-grid usage. However, both the solar and wind energies are intermittent. The geothermal energy is available round the clock, 365 days a year, but is a low enthalpy source. On the other hand, Solar radiation is a high energy source, but is unavailable during night/rainy days/days without sunlight. Interestingly, we are presenting the qualities of one form of energy as the solution to the short comings of the other. Also, both of these have high potential in similar geological settings. GSI (Geological Survey of India) has identified about 340 geothermal hot springs which are characterized by tertiary and quaternary orogeny activity in Himalaya, Mesozoic and Tertiary block faulting and epiorogeny activity in shield area. GSI has identified ten geothermal provinces in India out of which a total 10,000MW could be generated (Sharma 2015). The thermal efficiency of geothermal electric plants is relatively low. By hybridizing a geothermal plant, the average temperature of the working fluid and thereby the thermal efficiency and electrical output of the plant can be significantly increased (according to the Carnot principle). The hybridization can decelerate the depletion of heat content of the geothermal reservoir overtime thus extending its longevity.

2. STUDY AREA

Dholera at 30 km in southwest direction from Dhandhuka village of Ahmedabad district and 60 km away in north direction from city of Bhavnagar. Dholera thermal springs are situated along the margin of Saurashtra Peninsula. They fall in the locale of Western Marginal fault of Cambay Basin. Land in Dholera is covered by recent to alluvium and mud flats. The area is also occupied by quaternary soils deposited in subsiding area by the side of Cambay Basin to an extend of about 100 m over Tertiary sediments resting on Deccan Traps at a depth of about 500–600 m. A total of four springs were seperated in a radius of 4 km Dholera, Uthan, Swaminarayan temple and Bhadiyad. Dholera springs have the highest geothermal flow rate in Gujarat. Geochemical Methods are crucial methods applied during various stages of exploration and exploitation to evaluate reservoir characteristics. It is used simulatenously with geological and hydro-geological appraisals to supplement available information regarding the prospectively of potential region of investigation at relatively low costs compared to geophysical methods or drilling. Studies from pre-existing wells and surveys have suggested the presence of a sizable low enthalpy geothermal resource base. Geochemical study shows that the geothermal waters in this region are rich with sodium, potassium chloride and sulphate. The temperatures estimated from the cross-plots and the Geo-thermometric analyses did show that the springs were a part of low enthalpy geothermal reservoir system. This helped increase the confidence in implementing a green-field project in this area for a district heating and cooling system in the future as the area is a part of the Delhi-Mumbai Economic Corridor and the Smart City project of the government. Dholera has been declared a Special Investment Region and the presence of these potential reservoirs helps increasing the chances of implementing projects with lesser carbon footprints when economic activity starts (Shah M. et al., 2017). Studies showed that buried granites have the potential to behave as enhanced geothermal resources because they are the substantial source of heat energy (Gupta M., 2009). The subsurface lithological information deduced from geophysical and geological data of NE part of the

Deccan basalt province revealed that there is high heat producing granites under basalt cover. Artificial thermal reservoir/reservoirs can be created at depth > 1 km in these granites. Future technological innovation in drilling and hydro-fracturing techniques will make these sites best suited for developing large scale EGS and reduce power deficit in the country (Chandrasekhar V. and Chandrasekharam D., 2008).Dholera site consists of space heating and cooling systems. At dholera, there are two geothermal wells located which are producing hot geo-water at 3 l/s and 4 l/s respectively. The temperature of the water produced is around 45°C. The combined flow rate of the inlet line is 7 l/s and it is stored in water tank. This water tank also takes inlet of fresh water and they combined are used to run hot loop. The tank water is pumped through hot circulation pump and directed to the heat exchanger where it will heat exchange with water coming out from heat pump. After heat exchange, water is sent to heat pump where it will send to condenser. In condenser, water will get heated during heat exchanging with refrigerant. This heated water will again sent back to the plate type heat exchanger where it will heat exchange with geo heater. After heat exchange water will be dumped if its temperature will rise above 75°C else it will sent back to water tank. In cold loop fresh water is pumped into heat pump. In heat pump refrigerant will absorb heat from the water thus reducing temperature of water to 24°C. The cold water is then sent to Air Handling Unit (AHU). In this AHU air is sucked from “Sabhamandap”. Cold water in this AHU unit passes in series of tube and comes in thermal contact with sucked air. Heat is absorbed by the cold water from air and cooled air is sent back to “Sabhamandap”. Water is directed to heat pump for cooling purpose and cycle remains continuous. The schematic of the plant is shown in Fig. 1 here.

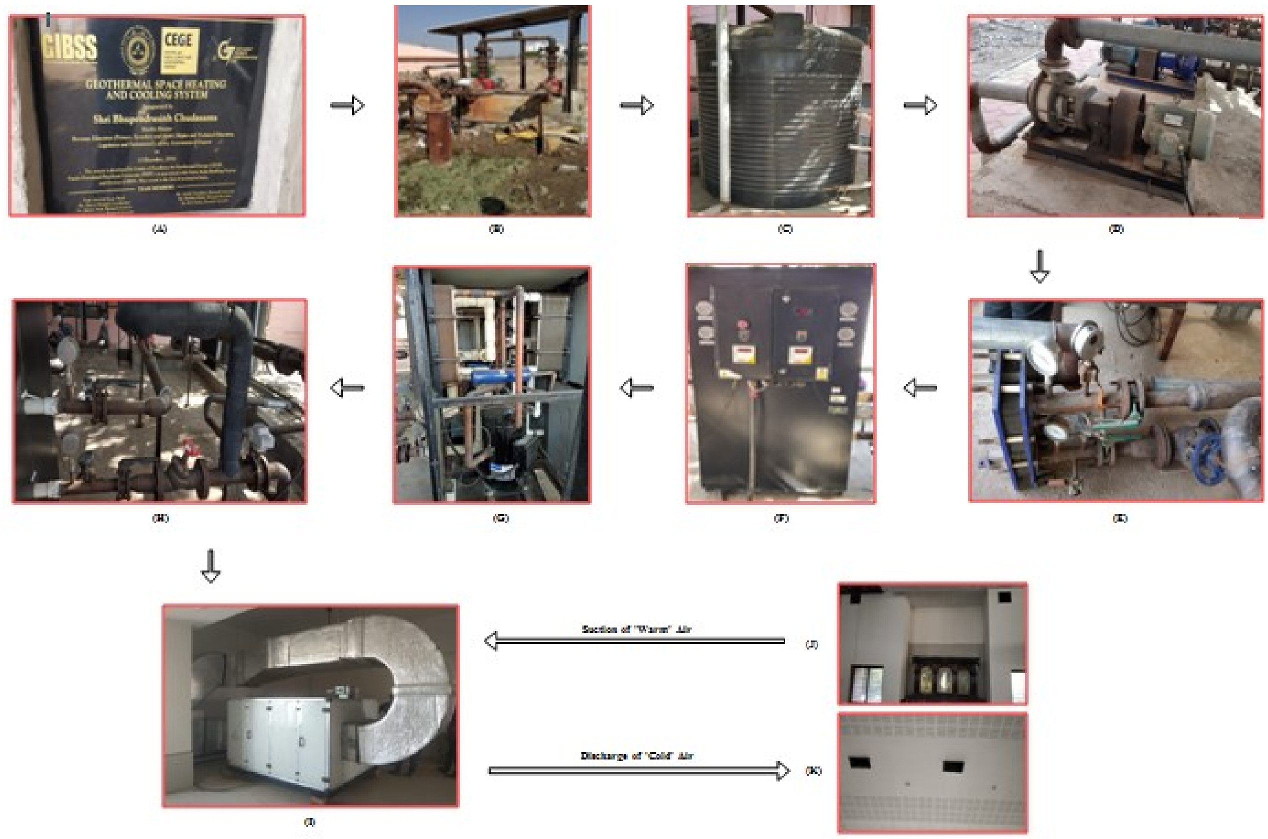

Figure 1: Space Heating and Cooling Setup at Dholera, Gujarat, India.(A) Dholera site plague stone,(B) Production Well, (C) Water Tank,(D) Hot Circulation Pump,(E) Plate Type Heat Exchanger,(F, G) Heat Pump,(H) Cold Loop, (I) Air Handling Unit (AHU), (J, K) "Sabhamandap"(picture courtesy : Centre of Excellence for Geothermal Energy, PDPU, Gujarat).

3. DESIGN OF HYBRID SETUP FOR DHOLERA PLANT

The objective of this study is to develop innovative solar-geothermal hybrid energy conversion systems for low enthalpy geothermal resource at Dholera, Gujarat. The hybrid setup takes advantage of the combination of solar thermal and geothermal power cycles. This combination may be related to the differing diurnal cycles of geothermal and solar energy. Many Solar collector choices are available depending on the desired working temperature. The typical designs for the ranges are : a) **greater than 400°C** - power tower or dish b) **350-390°C -** conventional parabolic trough or linear Fresnel c) **150-300°C -** parabolic trough with water and some linear Fresnel d) **100-200°C -** evacuated-tube flat-plate e) **less than 150°C -** flat-plate for home heating and hot water.

3.1 Power cycle configuration

Given a non-continuous flow of geothermal water from the well, the temperature obtained is roughly around 45°C. But, on ensuring a continuous flow of geothermal water, we can safely consider water's average temperature to be near 60°C. A power cycle is proposed in figure 2. Just after the production well, we employ a bypass valve whose function is to guide the flow differently under different scenarios. Precisely speaking, during night/rainy-day/no sunlight, the bypass valve will directly allow the water to flow to separator. However, during day-time, we are employing **Parabolic Solar Trough Collectors** to transfer the solar energy and hence raise the temperature of **Solar Heat Transfer Fluid (HTF),** which in our case happens to be Therminol-VP1. After careful literature review, we have chosen shell and tube heat exchanger, **Solar Geo Heat Exchanger (SGHX).** The HTF will be employed on shell side whereas geothermal water remains on tube-side so as to allow efficient heat transfer. Post heat-exchange, the temperature of geothermal water ought to rise to a temperature of ~200°C. After the exchange of heat in the heat exchanger, the Flash is deliberated for low-to-moderate geothermal resources. The solar energy is used to increase the temperature of the pressurized geothermal water to a desired high value (roughly above saturation) to allow flashing at a designed pressure. This produces steam and hot water for use in a steam turbine and for heating the Working fluid (WF) in the binary loop, respectively. The flash pressure is an adaptable quantity, restricted by the temperature limit on the WF. The pressure is set at 60 bar and dropped to 15.5 bar flashed after which we get some percentage of vapor and some percentage of condensed water is sent into the separator. The vapors are sent to the first turbine to power it, whereas the condensed water is throttled down to 15.5 bar to a temperature of 160-180 °C. It is then sent to the second heat exchanger where working fluid i.e. the refrigerant exchanges heat with the condensed water due to which the refrigerant heats up and turns into vapors after which the second turbine is powered. This cycle works on the basis of Organic rankine cycle (ORC).The essence of ORC lies in the selection of the working fluid, the physical and chemical attributes of the WF determine the power efficiency of system. In order to use low temperature heat source, working fluids of ORC should hold normal boiling temperature below 350°K.

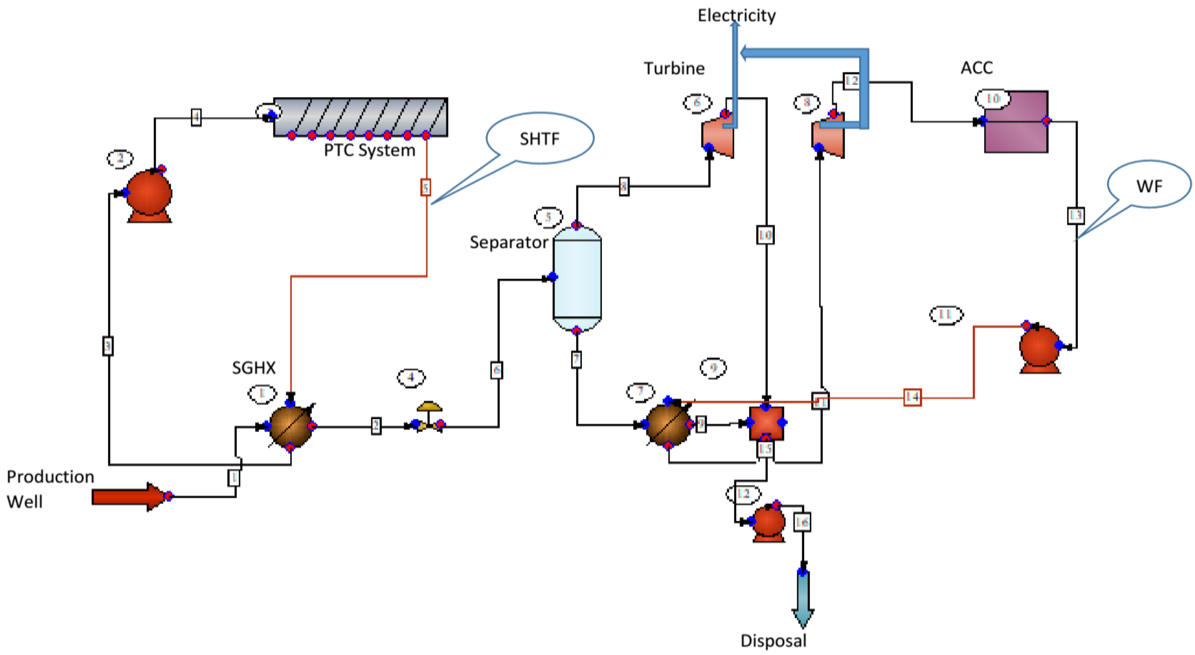

Figure 2: Proposed power cycle configuration.

<u>3.1.1 Selection of working fluid</u>

Deciding the working fluid for use in ORC cycles is major point, because depending on the source and use of heat level, the fluid must have excellent thermodynamic properties at lowest temperatures and pressures. The fluid should possess the following properties as well: economical, nontoxic, non-flammable, environment friendly, high utilization of the available energy from the heat source etc. For the study, the authors chose Therminol VP1 which is a eutectic mixture of 73.5% diphenyl oxide/26.5 diphenyl as the working fluid.

<u>Performance benefits of Therminol VP1(eutectic mixture of 73.5% diphenyl oxide/26.5 diphenyl)</u>

Solar Heat transfer fluid, Therminol VP-1, is a high temperature synthetic heat transfer fluid designed to operate at vapor phase systems or liquid phase systems. The expected performance benefits are:

Heat Transfer Properties – Therminol VP-1 has high thermal stability (highest among all organic heat transfer fluids) and low viscosity. It performs efficiently and uniformly in a wide use range of temperatures (12°C to 400°C).

Low Viscosity – It has a low viscosity upto temperature of 12°C (54°F) (crystallization point).

Vapor Phase Heat Transfer Fluid – It can be utilised as a liquid heat transfer fluid or as a boiling-condensing heat transfer medium up to its maximum use temperature.

Precise Temperature Control – It can be used as precise temperature control fluid due to its ability to operate as a vapor phase heat transfer fluid.

4. SOLAR GEOTHERMAL HEAT EXCHANGER CALCULATIONS

Design of the Heat exchanger is very important part of the hybridization. A schematic of Shell and Tube Heat Exchanger is drafted for this hybridization experiment as shown in fig 3. The authors performed **hand calculations** and compared them with the calculations on **ASPEN+** software. The process started with designing a Shell and tube heat exchanger. For efficient electricity production, the temperature of water needs to be high. We desire that the outlet temperature is 180°C. We consider counter-current flow as it is advantageous in extracting a higher proportion of the heat content of the heating fluid. The Logarithmic mean temperature difference (LMTD) value for counter-current flow is much larger than for co-current flow at the same end temperatures.

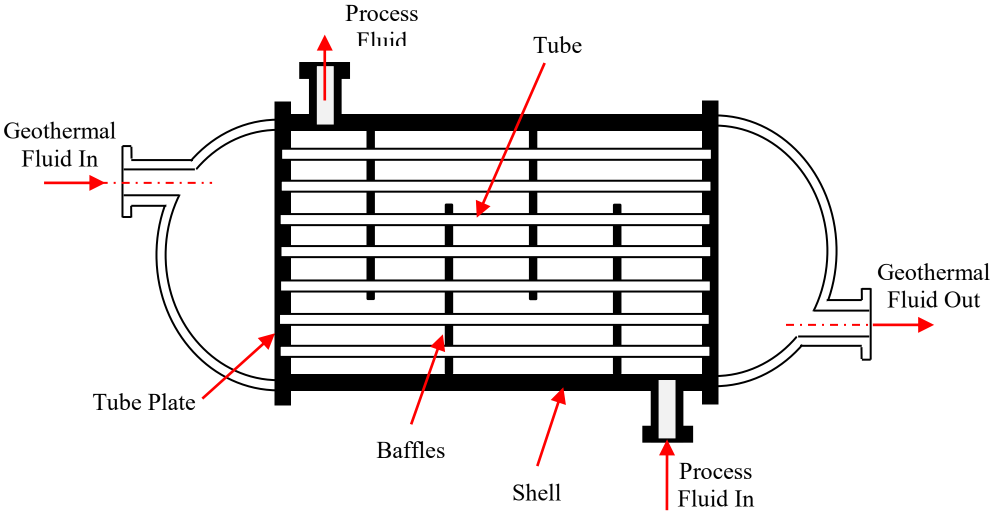

Figure 3: Schematic of Shell and Tube Heat exchanger.

4.1 Hand Calculations

Shell and Tube Heat Exchanger (SGHX) Design

We use a shell and tube heat exchanger as an evaporator, here the flow rate of water source is at 8 Kg/s and the inlet temperature at the tube side is 60°C and we need an output temperature at the tube side of 180°C. So considering the desired system we maintain a temperature of 350°C at the shell input and we obtain 275°C at the shell output. Heat transfer fluid (HTF) will be heated to 350°C using solar collector and will be passed through SGHX where HTF temperature will be reduced to 275°C at the outlet of the SGHX. Now this HTF will be again sent to the solar collector to get heated again and the process continues in closed loop.

Tube-side (Tin) = 60°C Tube-side (Tout) = 180°C

Shell-side (Tin) = 350°C Shell-side (Tout) = 275°C

Obtain the capacity in subsequent shell side stream by mass balance equation.

$$m_1 C_{water}\, \Delta T_1 + L\, m_1 + m_1 C_{steam}\, \Delta T_1 = m_2 C_2\, \Delta T_2 \qquad (1)$$

Where mc ΔT gives us the capacity of the evaporator in the inlet and outlet, m1 = mass flow rate of the geothermal water , m2 = mass flow rate of the therminolVp1 (Heat transfer fluid), C_{water} = 4.18 kJ/kg·°C, L = 2.3 * kJ/kg·°C, C_{steam} = 2 kJ/kg·°C , C_2 = 2.58 kJ/kg·°C(therminol VP1) .

-- 8*4.18*(100-60) + 2.3 * *8+ 8*2.09*(180-100) = m_2* 2.588*(350-275)

-- 1337.6+18400+1337.6 = m_2*194.16

-- m_2= 21075.2/194.16 = 108.57 ~ 108 kg/s

-- m_2= **108 kg/s**

4.1.1 Design mechanism formula

In order to design the heat transfer area required we need to calculate the flow rate and LMTD. The following steps are performed to deduce the formula.

-- Calculate the required heat transfer rate, Q in from specified information about fluid flow rates and temperatures

-- Make an initial estimate of the overall heat transfer coefficient, U, based on the fluids involved.

-- Calculate the log mean temperature difference, ΔTm, from the inlet and outlet temperatures of the two fluids.

-- Calculate the estimated heat transfer area required,

-- Select a preliminary heat exchanger configuration.

$$A = Q/(U.\Delta T_M) \tag{2}$$

Where A = Heat transfer area (m²), U = Overall heat transfer coefficient (kJ/h.m².°C), Q = Heat transfer rate (kJ/h) , ΔTm = Log mean temperature difference (°C)

LMTD (counter current flow)

$$LMTD = (\Delta T1 - \Delta T2) / \ln(\Delta T1/\Delta T2) \tag{3}$$

$\Delta T1 = T_{Hot,In} - T_{Cold,Out} = 350.00 - 180.00 = 170.00$

$\Delta T2 = T_{Hot,Out} - T_{Cold,In} = 275.00 - 60.00 = 215.00$

LMTD = 191.62

4.1.2. Shell Side data

4.1.2.1 Heat transfer rate

$$Q = m\, Cp\, (T_2 - T_1) \tag{4}$$

Where Q = heat transfer flow rate, Cp = specific heat capacity , m=Mass flow rate of the heat transfer fluid (VP1), T2 = shell side out temp, T1= Shell side in temp

Q = 108* 2.588*(350-275)

Q = 20962.8 KJ/s

4.1.2.2 Shell side cross flow area

Using standard values from Tubular Exchanger Manufacturers Association (**TEMA**) classification :

Shell id = 4.2 m, Pitch diameter= 0.075 m, Baffle spacing= 0.1 m, Pattern = triangle, Conductivity = 53 Kcal –m/hr-m2-k

The clearance is calculated as follows:

$$\text{Clearance } (c'') = P_t - \text{OD of tube}$$
$$\text{OD of tube} = (\text{ id} + (2 \times \text{thickness}))$$
$$\text{Cross flow area of shell} = a_s = ID(\text{shell}).\ C''.\ B/ P_t \tag{5}$$

C’’ = 0.075 – 0.06 = 0.015 m

$a_s = 4.2*0.015*0.1/0.075 = 0.084\ m^2$

4.1.3 Tube side data

4.1.3.1 U (Overall Heat transfer Coefficient)

Heat transfer coefficient = 875 W/m2.k. (using T.E.M.A standards for light oil and water)

4.1.3.2 Heat transfer area

Calculating the first four items allows determination of the required heat transfer rate, Q, and the inlet and outlet temperatures of both fluids, allowing calculation of the log mean temperature difference, ΔTm. With values now available for Q, U, and ΔTm , an initial estimate for the required heat transfer area can be calculated from the equation 1:

$A = 20962.8 \text{ x} / (875* 191.62) = 125\ m^2$

4.1.3.3 Tube sizing

Step 1: We use the continuity equation:

$$\dot{m} = PAV \tag{6}$$

where, ṁ is the flow rate, ρ is the density of the fluid, A is the cross sectional area of the pipe,V is the velocity of the fluid perpendicular to the cross section area of the pipe ,We have the data for the density; flow rate and velocity (determined by industry limitation criteria)

Using this we calculate the area of the pipe where mass flow rate of water = 8 Kg/s, velocity = 4 m/s (for liquids, T.E.M.A Standard)

Bist and Sircar

$A = 8/1000*4 = 0.002\ m^2$ (the cross sectional area of the pipe)

Step 2: Calculating the diameter of the pipe :

$$d = \sqrt{4A/\Pi} \tag{7}$$

$d= \sqrt{4*0.02/\Pi} = 0.05$ m or 1.98 inch ≈ 2 inch (As per standard industrial diameter value)

D. Number of tubes required:

$$Sa = \Pi dl \tag{8}$$

Sa = Surface Area per tube (m^2) , Length of the tube (L) = 10 m

$Sa = 3.14 * 0.05* 10= 1.57\ m^2$

Number of tubes required will be (n) = Heat transfer area / Surface area per tube (9)

$n = 125\ m^2/ 1.57\ m^2 = 80$ tubes

E. Pressure Drop Calculations:

$$\Delta P = 4Fl\rho v^2/2d \tag{10}$$

$$F = 0.78/ N_{Re}^{0.25} \tag{11}$$

$$N_{Re} = \rho vd/\mu \tag{12}$$

Where ΔP = pressure drop, F = friction factor, ρ = density of water, v = velocity, d = diameter, N_{Re} = Reynold's number, ρ = density of water, μ = viscosity of water

$N_{Re} = 1000*4*0.05 / 6.56 *10^{-4} = 304878$

$F = 0.78/ N_{Re}^{0.25} = 3.32*10^{-3}$

Now calculating

$\Delta P = 4*3.32*10^{-3}*10*1000*16 / 2*0.05 = 21.25$ KPa (Maximum allowable pressure drop = 10 psi, hence within constraints)

4.2 Software calculation

Simulation was performed using ASPEN+ software to replicate the results of temperature improvement of water via software as well.

4.2.1. ASPEN+ Simulation

ASPEN+ is process simulation software better suited for chemical process design like designing of a heat exchanger etc. We carried out seven basic steps to simulate the design and output of a Shell and Tube heat exchanger. In step 1 we select the two components of heat exchanger. In this case one is water and the other is TherminolVP1(eutectic mixture of Diphenyl oxide and Biphenyl). In step 2, we have to select the type of heat exchanger that is desired (in this case Shell and Tube Heat Exchanger) for the simulation and select the hot stream and cold stream, in and out. After selecting the component we select the method used for calcutaion, in step 3 globally used method NRTL-RK(non random two liquid model) is used. NRTL-RK is recommended for highly non-ideal chemical systems, and can be used for Vapor Liquid Equilibrium (VLE) and Liquid Level Equilibrium(LLE) applications. The model can also be used in the advanced equation of state mixing rules. In step 4, we assign the flow direction of the two liquids to be counter current flow, enter the heat transfer area already calculated using hand calculations(equation 2), set the desired hot stream outlet temperature(275 degree C), enter LMTD value (equation 3), enter pressure drop value(equation 9). In step 5 we enter the shell side data such as TEMA shell pass, number of tube passes, orientation of the heat exchanger (horizontal in this case), shell inside diameter, shell bundle clearance (equation 5). In step 6 we enter the tube side data such as the total number of tubes (equation 9), length of each tube, pitch, material (carbon steel) and other tube parameters such as internal diameter and outer diameter In the final step 7, the simulation is run. The final stream results obtained are satisfactory i.e. 184.7 °C. This value is near to the desired value of 180 degree C. The console is shown in the following figures.

Step 1 : Selection of components, one is Water and other is Therminol VP1

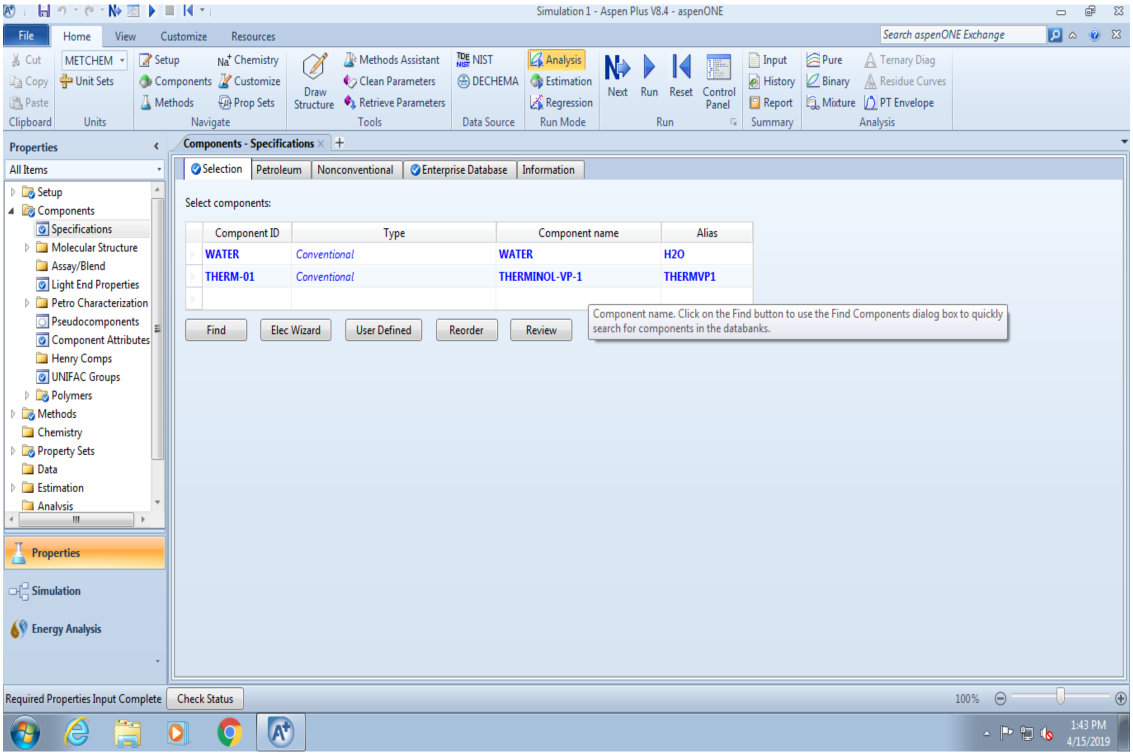

Figure 4: Components selection.

Step 2: Selection of heat exchanger

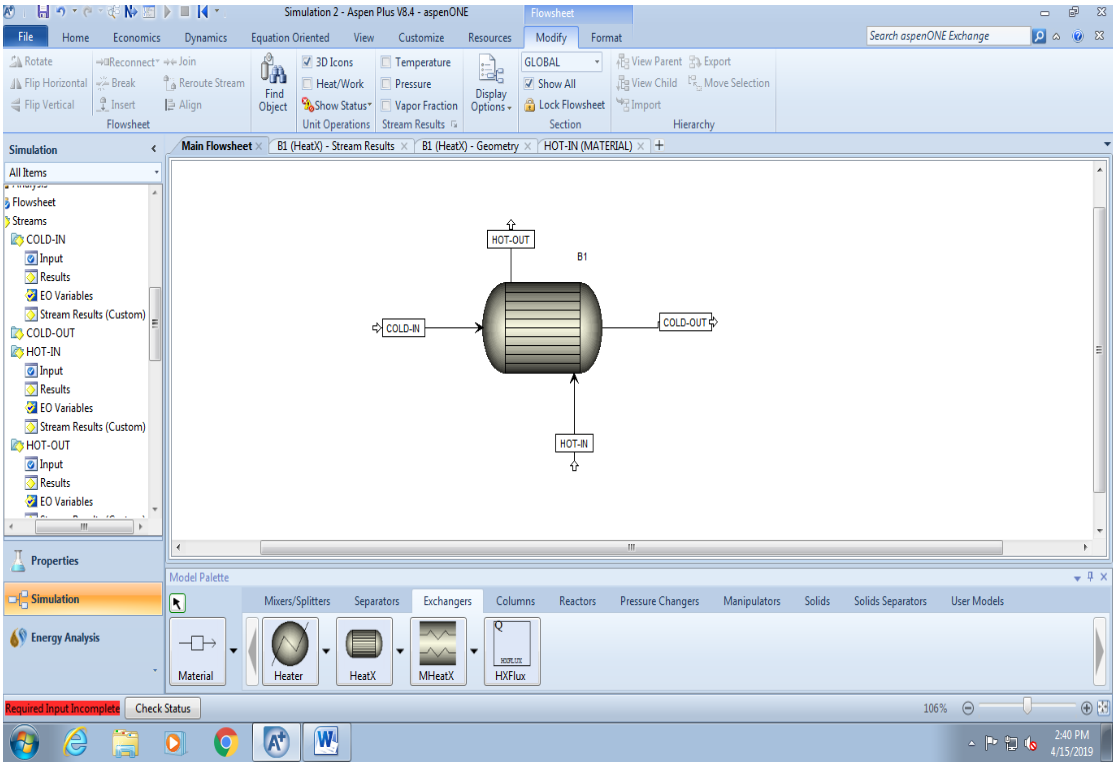

Figure 5: Selection of hot (therminol VP1 and cold stream (water).

Step 3 : Method to be used : Using industrially viable commonly used method known as NRTL-RK(Non random two liquid model)

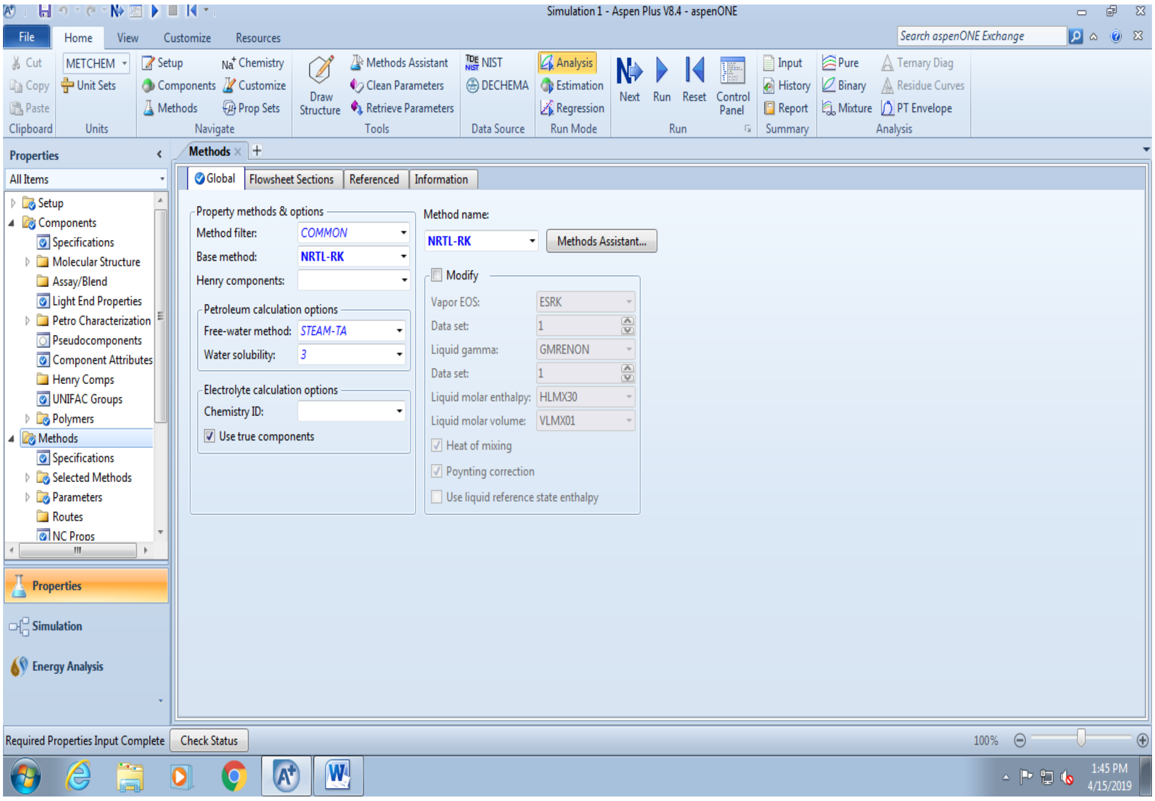

Figure 6: Selection_of method.

Step 4: Specification : This section is used to set the flow direction, heat exchanger area, temperature etc.

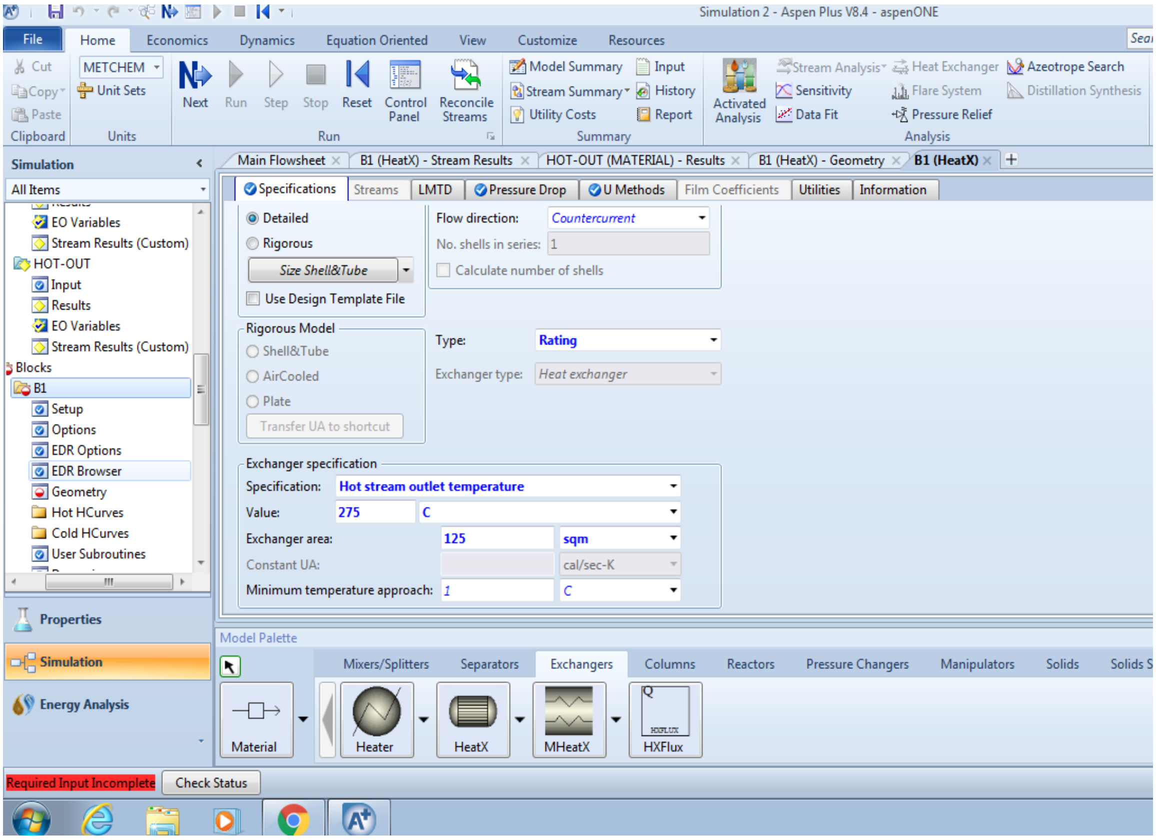

Figure 7: Selection of Specification.

Step 5 : Shell side data : entering the shell side data into the software

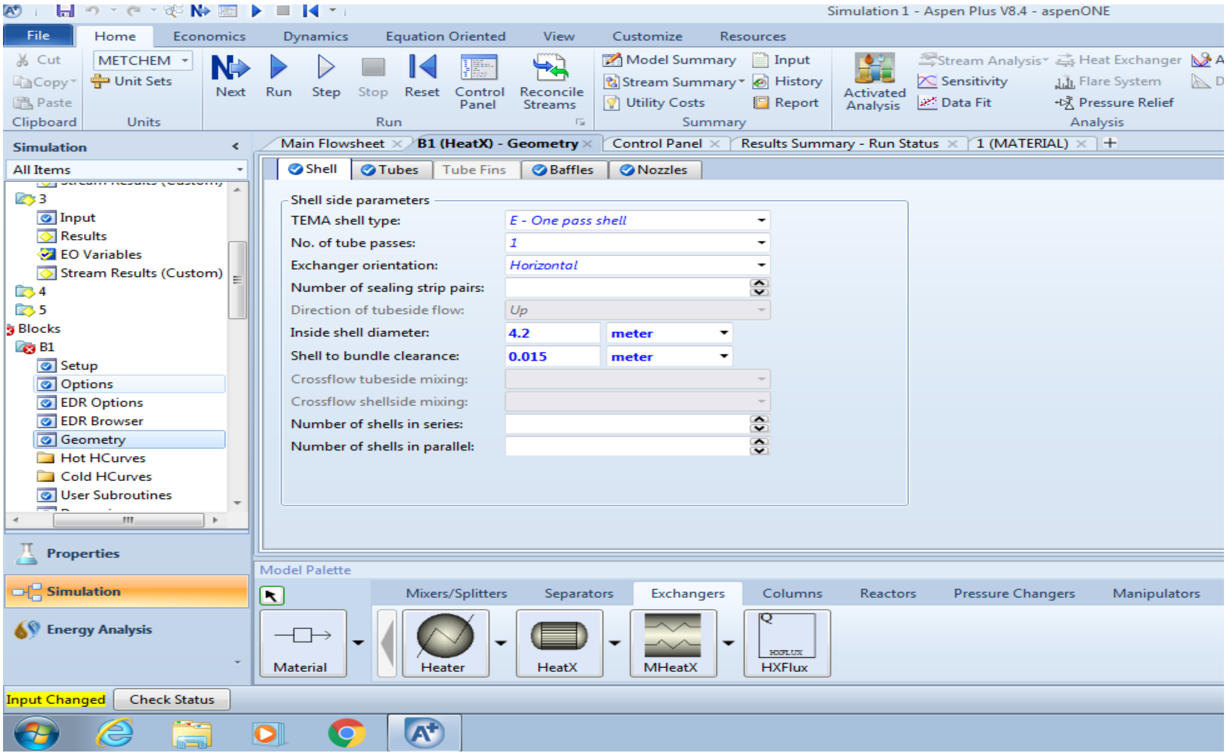

Figure 8: Entering tube side data.

Step 6 : Tube side data : entering the tube side data into the software

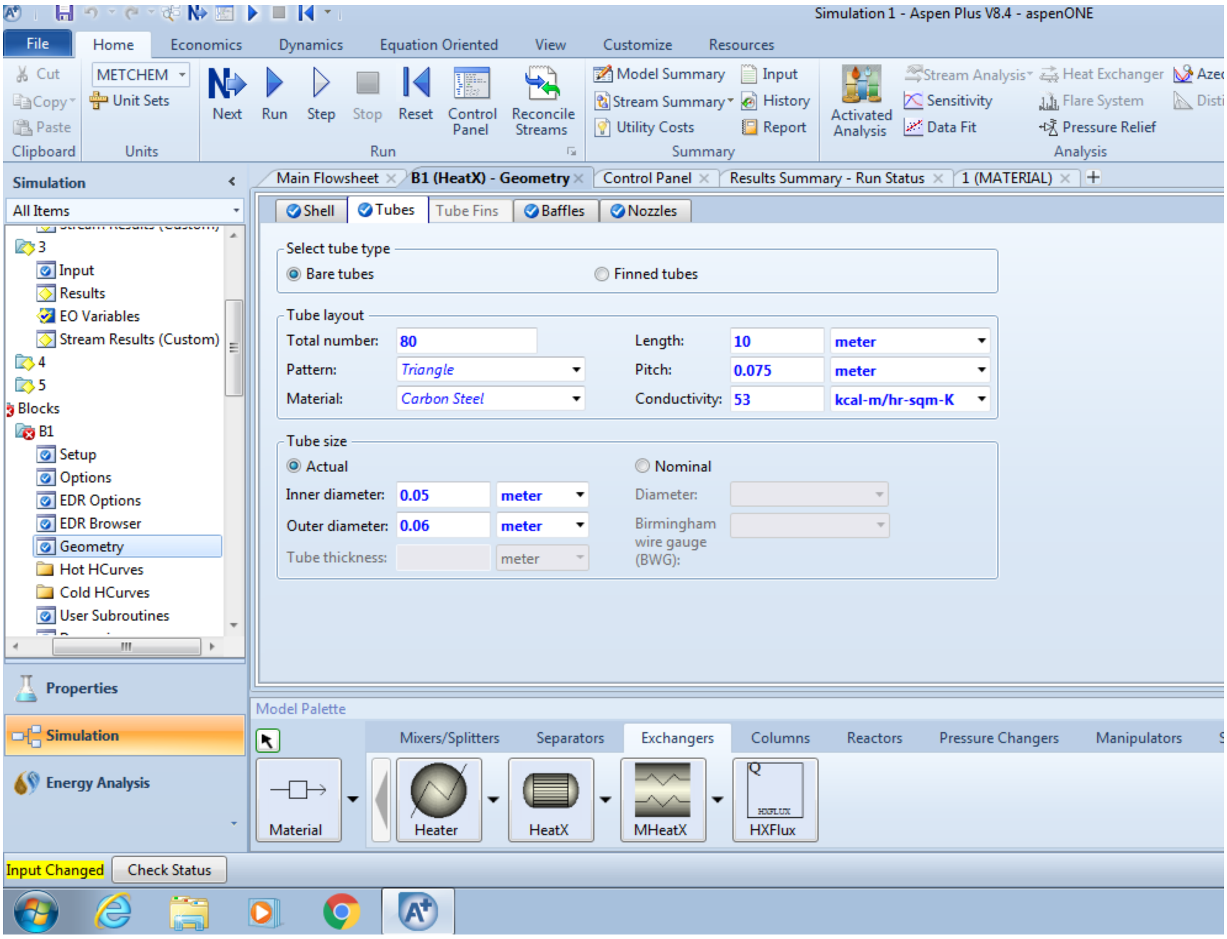

Figure 9: Entering shell side data.

Step 7: Final stream result: final result after running the simulation

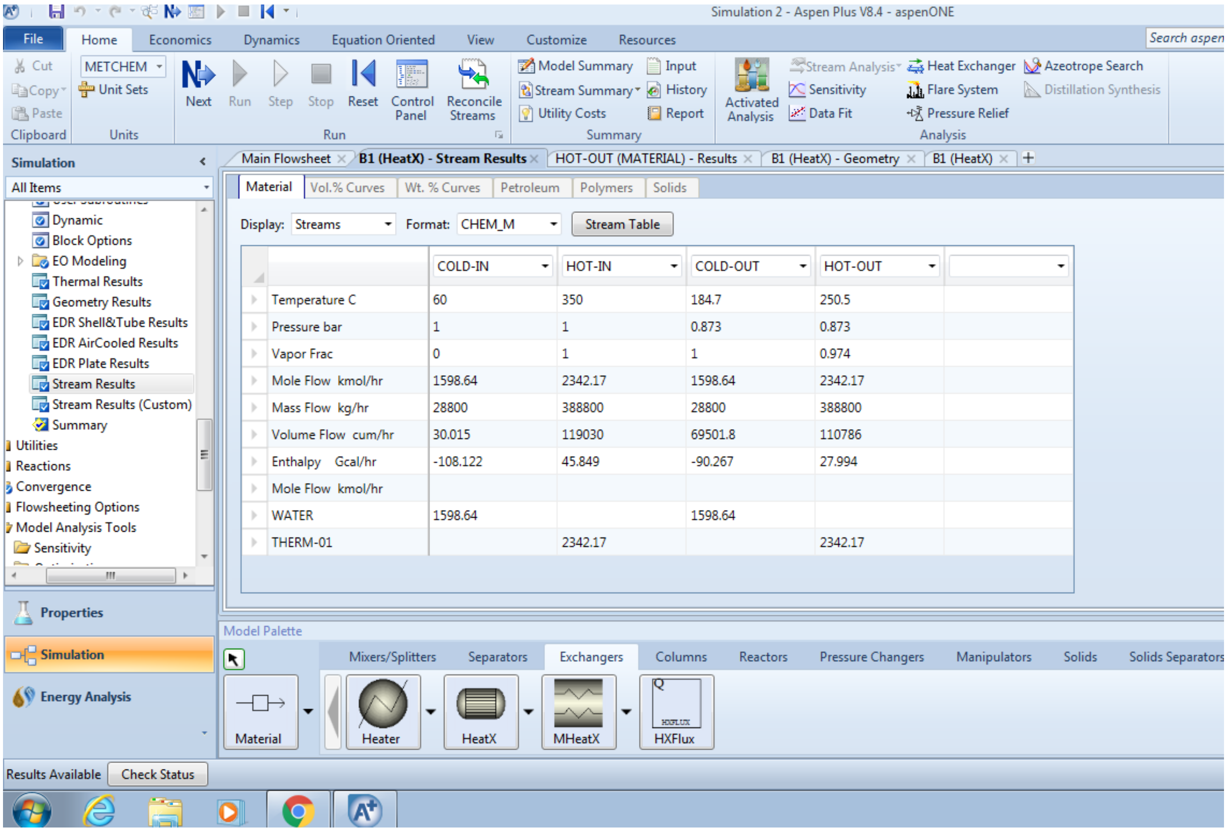

Figure 10: End stream results showing the outlet temperature of water increased to 184.7 ºC.

5. POWER GENERATION

For the power generation, we need to design an Organic Rankine Cycle(ORC) and perform calculations in order to understand the efficiency of the hybrid system and the amount of power that may be generated from the system. We would be using heat transfer fluid R-134a (1,1,1,2 Tetrafluoroethane) in a plate type heat exchanger(PHE)

Assumptions:

- The efficiency of heat exchanger is 0.95
- The pump compression factor is 2.8
- The process between pump, heat exchanger and turbine is considered isentropic in nature

Water side

Water in = 180°C, Water out = 110 °C, Mass flow rate = 8 Kg/s

Heat energy

$$Q = mc_p\Delta T \tag{13}$$

Where Q = heat energy, m = mass flow rate, ΔT = temperature difference between Water in and water out

Q = 8 x 2.09 x (180 – 110)

Q = 1170 KJ

R-134(a) (1,1,1,2 Tetrafluoroethane) side

R134(a) in = 25 °C, R134(a) out = T_2, Mass flow rate = m_2

Mass flow rate calculation

The efficiency of heat exchanger is assumed as 0.95

$$\eta = \frac{Q_{act}}{Q_{max}} \tag{14}$$

$$0.95 = \frac{1170}{m2\ xc_p x\ (\ Tmax - Tmin)}$$

$$0.95 = \frac{1170}{m2\ x\ 1.718\ \ x\ (\ Tmax - Tmin)}$$

$$m_2 = \frac{1170}{0.95\ x\ 1.718\ \ x\ (\ 180 - 25)}$$

m_2 = 4.62 Kg/s

<u>From heat balance equation</u>

By using software miniREFPROP, we can calculate the enthalpy value using iterative method based on one reference value(here enthalpy of refrigerant H1). The software is developed by National Institute of Standards and technology(NIST) that contains a limited number of pure fluids (water, CO2, R134a, nitrogen, oxygen, methane, propane etc), along with air as a pseudo-pure fluid.

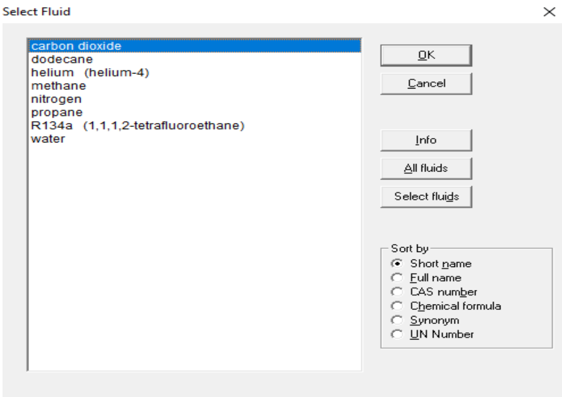

Figure 11: Fluid selected R-134a.

H_1 = enthalpy of the refrigerant leaving the condenser and entering the pump

H_2 = enthalpy of the refrigerant leaving the pump and entering the evaporator

H_3 = enthalpy of the refrigerant leaving the evaporator and entering the expander

H_4 = enthalpy of the refrigerant leaving the expander and entering the condenser

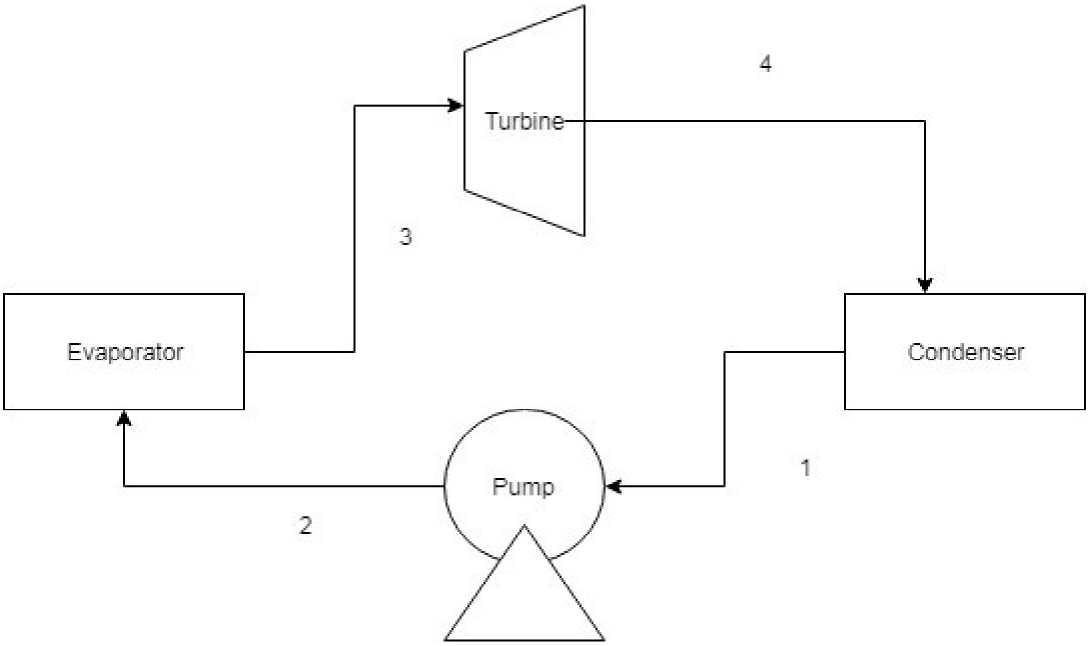

Figure 12: ORC Cycle.

Taking the reference value of H_1 = 231.8 KJ/Kg at temperature 23°C from Dupont table

HFC-134a Saturation Properties - Temperature Table										
Temp	Pressure	Volume		Density(Kg/m3)		Enthalpy(KJ/Kg)		Entropy(KJ/Kg.K)		Temp
°C	kPa	Liquid	Vapor	Liquid	Vapor	Liquid	Latent	Liquid	Vapor	°C
20	572.25	0.0008	0.036	1224.4	27.791	227.5	182.5	1.0964	1.7189	20
21	590.16	0.0008	0.0349	1220.7	28.659	228.9	181.6	1.1012	1.7185	21
22	608.49	0.0008	0.0338	1217	29.549	230.4	180.7	1.106	1.7182	22
23	627.25	0.0008	0.0328	1213.3	30.462	231.8	179.8	1.1107	1.7178	23
24	646.44	0.0008	0.0318	1209.6	31.399	233.2	178.9	1.1155	1.7175	24
25	666.06	0.0008	0.0309	1205.9	32.359	234.6	178	1.1202	1.7171	25
26	686.13	0.0008	0.03	1202.1	33.344	236.1	177	1.125	1.7168	26
27	706.66	0.0008	0.0291	1198.3	34.354	237.5	176.1	1.1297	1.7165	27

Figure 13: R-134(a) Property Table (Modified after Dupont., 2015).

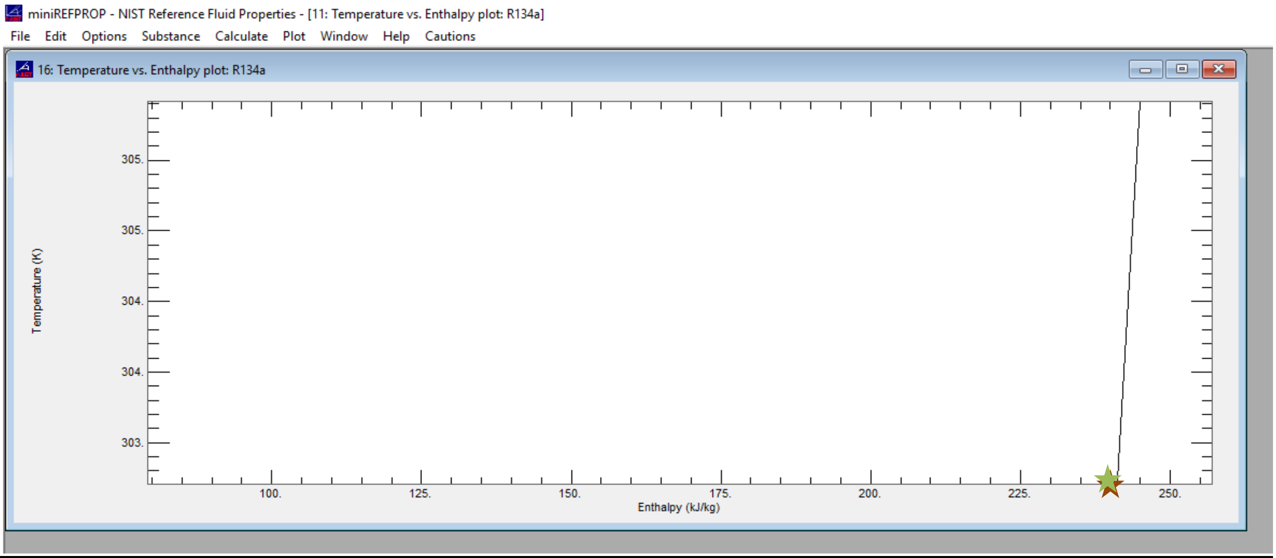

Figure 14: Depicting value of H2 = 244 KJ/Kg at temperature 32.1 °C which is the enthalpy of the refrigerant leaving the pump and entering the evaporator.

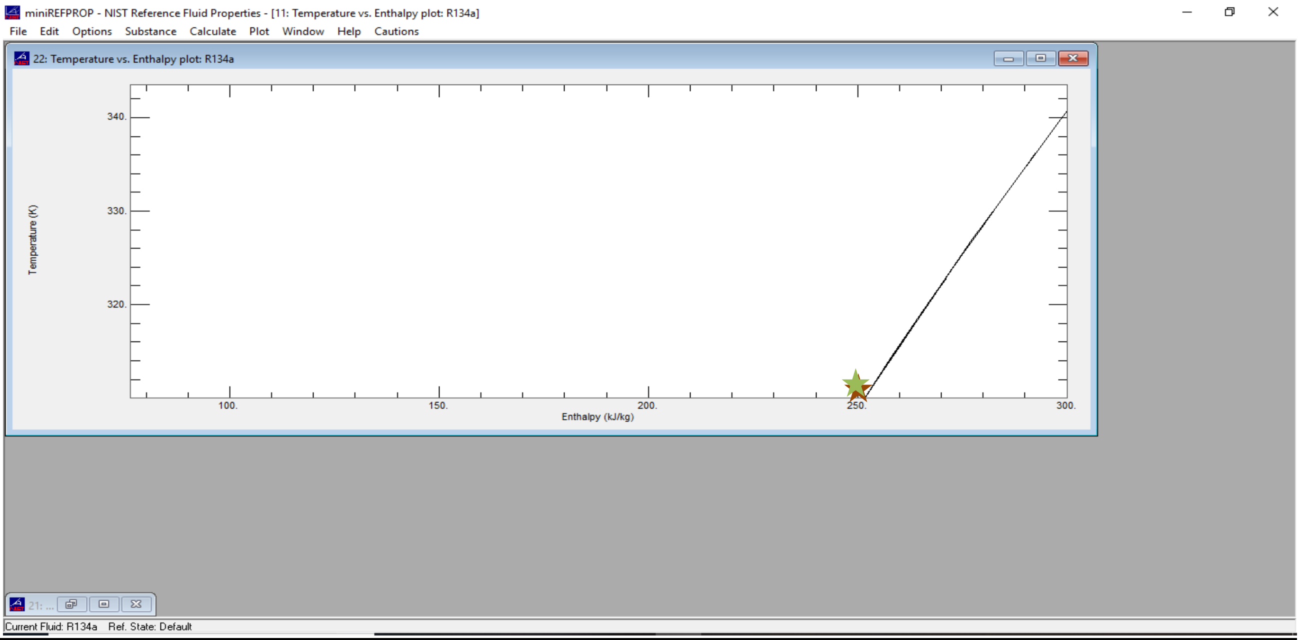

Figure 15: Depicting value of H3 = 298 KJ/Kg at temperature 67°C which is enthalpy of the refrigerant leaving the evaporator and entering the expander.

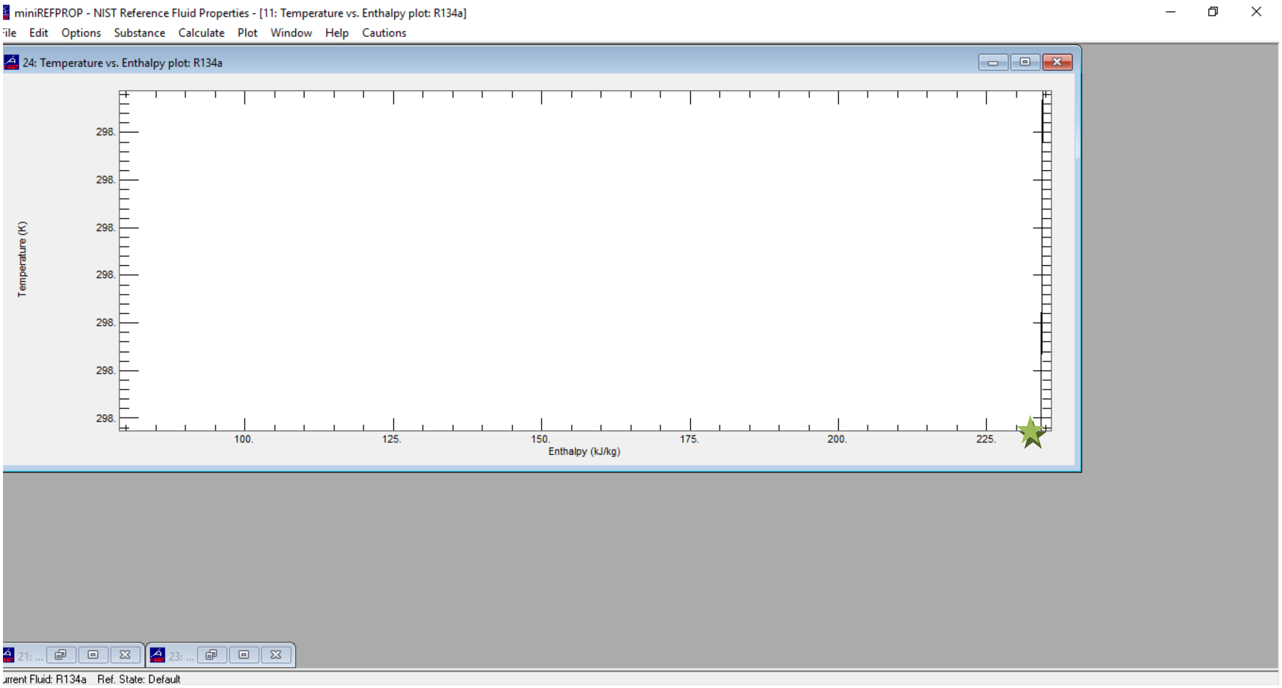

Figure 16: Depicting value of H4 = 234 KJ/Kg at 25°C which is enthalpy of the refrigerant leaving the expander and entering the condenser.

<u>Expander power (W_t)</u>

$$\text{Expander power } (W_t) = m(R134a) \text{ x } (h_3\text{-}h_4) \tag{15}$$

$$\mathbf{W_t = 4.62 \text{ x } (298\text{-}234) = 295.68 \text{ KW}}$$

<u>Thermal efficiency</u>

$$\text{Thermal efficiency} = w_t / [(h_3\text{-}h_1) \text{ x m } (r143a)] \tag{16}$$

$$\mathbf{\text{Thermal efficiency} = 291.68/[(298\text{-}231.80] \text{ x } 4.62 = 95.37\%}$$

<u>Theoretical expander power</u>

$$W_{th} = m(r134a) \text{ x } (h_3\text{-}h_1) \tag{17}$$

$$\mathbf{W_{th} = 4.62 \text{ x } (298\text{-}231.8) = 305.84 \text{ KW}}$$

<u>Net power produced s</u>

$$P_w = m.(r134a) \text{ x } [(h_3 - h_4) - (h_2\text{-}h_1)] \tag{18}$$

$$\mathbf{P_w = 4.62 \text{ x } [(298 - 234) - (244 - 231.8)] = 239.316 \text{ KW}}$$

6 RESULTS AND DISCUSSION

a) Using the simulation software (ASPEN+), we can conclude that a satisfactory increase in the enthalpy of water can be achieved (from 60ºC to 184.7 ºC) (Fig 10).

b) Based on this improved temperature we can design an ORC (Fig 12). The main parts of this ORC will be condenser, pump, turbine and evaporator.

c) This ORC can generate electricity, in this case up to 240 KW (Equation 18).

7 CONCLUSION

Solar and Geothermal energies fully complement each other in same geographical locations. Such hybridization setups can be used in large extends mainly in third world countries such as India as the low cost is prime factor here. An attempt is made in this paper to calculate the power generated through hybridization and is found to be satisfactory. The above study helps in attaining the following goals:

a) Enthalpy improvement: By using hybrid solar and geothermal energy, the temperature of water which is currently flowing at around 45-60 ºC can be improved **4.5 folds** or 124.7 ºC increase and can reach upto 184.7 °C. At this stage the water will be in the form of dry steam, which can be directly used to produce electricity via a gas turbine.

b) Electricity Generation: Theoretical calculations suggest that by hybridization , power of the scale approximately 240 KW(equation 18) can be generated using this project.

c) Societal benefits: The generated power can be utilized by the villagers around the project who are of poor socio economic background.

d) Hybridization of renewable energy projects the benefits of efficiency improvement hence economical feasibility.

The main challenge would come in the field integration. However the authors are hopeful that soon this setup would be integrated making it first on the map of India.

REFERENCES

Baredar, P., Sethi V. K. and Pandey M.: Correlation analysis of small wind–solar– biomass hybrid energy system installed at RGTU Bhopal,MP (India), Clean Technologies and Environmental Policy, vol. 12, nr. 3, Springer, (2010), 265 – 271.

Chandrasekhar, V. and Chandrasekharam, D. : Enhanced geothermal resources in NE Deccan province, India, Transactions , Geothermal Resources Council, 32, (2008), 59-61.

Eziyi, I. and Krothapalli, A.:Sustainable Rural Development: Solar/Biomass Hybrid Renewable Energy System, Energy Procedia, 57, (2014), 1492-1501.

Gonzalez, A., Jordi-Roger, R, and Rius, A.: Optimal Sizing of a Hybrid Grid-Connected Photovoltaic-Wind-Biomass Power System, Sustainability, vol. 7, nr. 9, (2015), 12787 – 12806.

Gupta, M.: Geothermal energy: A secure resource for development of Ladakh region, India, Transactions , Geothermal Resources Council, 33,(2009), 545-550.

Rahman, M., Hasan, M. M., Paatero J. V. and Lahdelma R.:Hybrid application of biogas and solar resources to fulfill household energy needs: A potentially viable option in rural areas of developing countries, Renewable Energy, (2014),1-11 .

Shah, M.,Sircar, A., Sahajpal, S., Sarkar, P., Sharma, D., Garg, S., and Shukla Y.: Geochemical analysis for understanding prospectivity of low enthalpy geothermal reservoirs of Dholera, Proceedings, 42nd workshop on geothermal reservoir engineering, Stanford University, Stanford, (2017),1-16.

Sharma, S. K.: Geothermal, A viable eco-friendly source of energy for India, Proceedings World Geothermal Congress, (2005), 24-29.

Zhao, H. and Guo, S.: External Benefit Evaluation of Renewable Energy Power in China for Sustainability, Sustainability, 7, (2015), 4783–4805.

http://www.ipieca.org/resources/energy-efficiency-solutions/power-and-heat-generation/solar-thermal/#reference-1

https://www.therminol.com/products/Therminol-VP1