February 6, 2020
Design of Thermal Systems

DHW: Parabolic Trough Power Plant

Objective: To design a solar thermal power plant.

Assumptions

Given:
- Temperature of oil before solar field is 240 ˚C[1]
- Exit temperature (T1) of water is 50 ˚C[1]
- Isentropic efficiency of turbine is.85[1]
- All components except the turbine are ideal so no work is lost due to inefficiencies[1]
- Heat Transfer fluid is used in solar field[1]

General:
- Assume steady state flow so that the mass flow is not changing in the system
- Fluids are incompressible so we don't have to worry about phase change of the liquid

Boiler:
- There is no loss in heat transferred from the oil to water
- Isobaric, so the pressure in state 2 and 3 are equal
- The water in state 3 is super-heated to have better efficiency [5]
- Assume ideal counter flow so that water and oil are in thermal equilibrium

Turbine:
- Assumed an adiabatic process, so Q3 = Q4 making the only contributing factor to the work on the turbine the heat entering.

Condenser:
- Isobaric, P1 = P4

Pump:
- Assume reversible adiabatic, S1 = S2

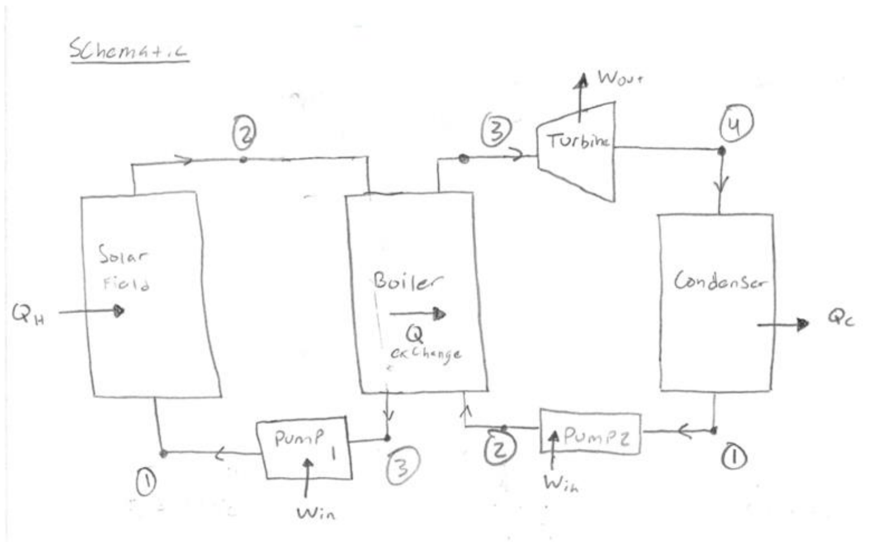


Fig. 1 Schematic of the Parabolic Trough Power Plant.

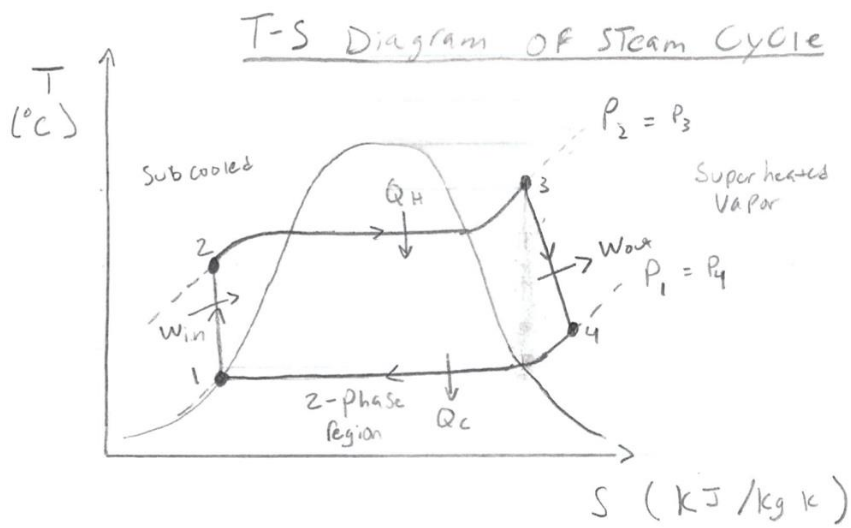


Fig. 2 T-S diagram of steam cycle.

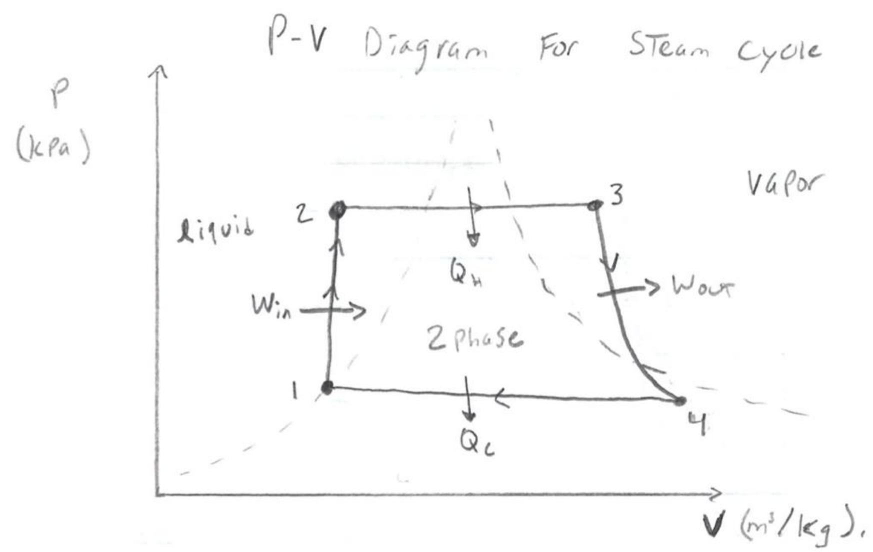


Fig. 3 P-V diagram of steam cycle.

Thermal Oil: Dynalene MS-2

When choosing a thermal fluid to use, several factors were considered. First, was finding a fluid with a high operating temperature and specific heat capacity. For this I found that most thermal fluids were around 400 ˚C with heat capacities ranged from .9-5 [kJ/kg K]. The reason why there was a limit on the operating temperature was because at high temperature the oil starts to decompose and cause the efficiency to go down. The second factor that I looked at was finding a thermal fluid that was non-toxic. As a result, I chose Dynalene MS-2 as my heat transfer fluid. Dynalene MS-2 is a non-toxic fluid with an operating temperature up to 485 ˚C and specific heat of 1.60 kJ/kg K[3]. Above this maximum operating temperature this thermal oil would start to become inert gas with lower pressures. Over a long period of time the fluid would start to solidify. Although Dynalene MS-2 has a high operating temperature and is safe, one property that I could have been weighted more heavily was viscosity. My fluid had a viscosity of 4 [cP], which compared to water which has a viscosity of 0.89 [cP][3]. I would want to look at this further because if my thermal oil had a lower viscosity it would require work from the pump to move the fluid.

Table of Optimized Parameters

Table 1 Parameters for Max

State	Phase	P (kPa)	T (˚C)	s (kJ/kg * K)	h (kJ/kg)
1	Saturated L	12.35	50	0.7037	209.31
2	Saturated L	3346	-	0.7037	212.6632
3	Superheated Vapor	3346	240	6.1427	2.80E+03
4	Saturated Vapor	12.35	50	6.5207	2.09E+03

Table 2 Temperatures of Dynalene MS-2

State	T (˚C)
1	240
2	480[4]
3	240

Table 3 Pipe and Dynalene Properties[6]

v (m^3/kg)	0.000568182
L (m)	11
d (m)	0.07
F[3]	0.038
ρ	1760
V (m/s)	2
Cp[4]	1.6 kJ/kg K

Table 4 Work of Components

Component	Work (kW)
Pump in solar field	0.0119
Pump in steam cycle	74.5612
Turbine	15720
Q_in	57600
Electrical Efficiency	0.27162199

Calculations

1. In order to calculate the efficiency in the system I first calculated the heat rate in the oil field. To do this I need to find what the heat rate and work are from the solar field and pump.

$$= v\left(f\frac{l}{D}\frac{1}{2}\rho V^2\right) = \left(5.68e^{-4}\frac{M^3}{kg}\right)(0.038)\frac{(11\ m)}{(0.07\ m)} * \frac{1}{2}\left(1760\ \frac{kg}{M^3}\right)\left(2\frac{m}{s}\right)^2$$

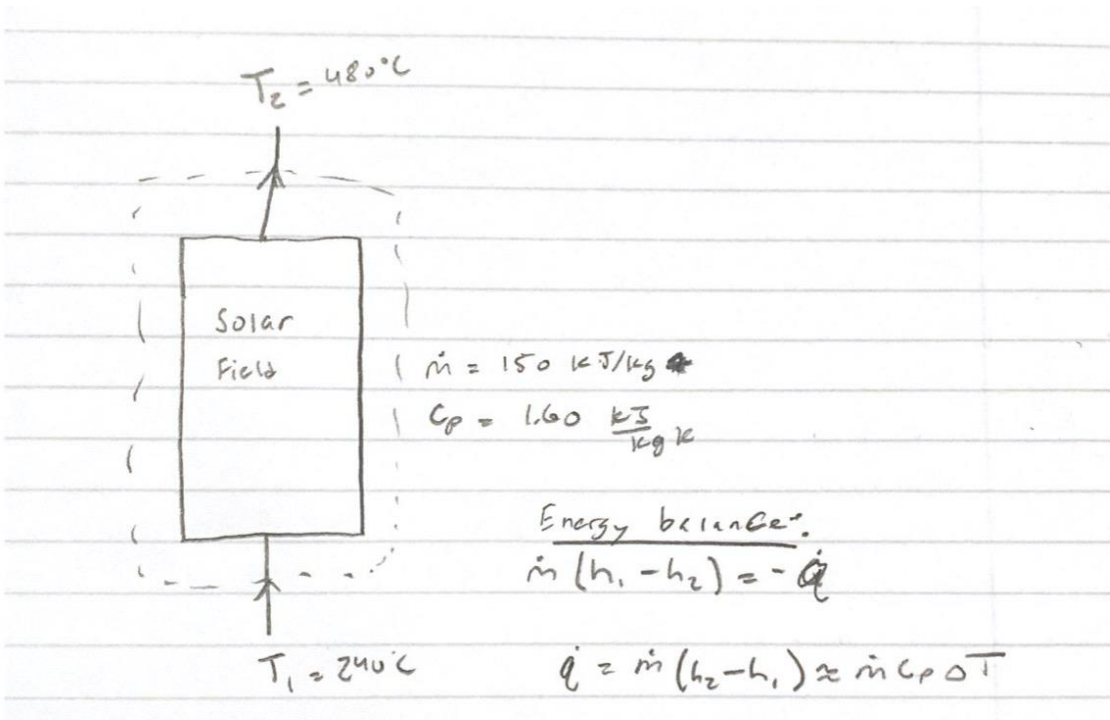
$$W_{p1} = 0.0119\ kW$$

$$\dot{q} = \dot{m}(h1 - h2) + W_{p1}$$

Since we do not know the enthalpy values, we can make the approximation that the change in enthalpy is about the same as the heat capacity times the difference in temperature.

$$\dot{q} \cong \dot{m} * cp * (t_2 - t_1) + W_{p1}$$

$$\dot{q} = \left(150\,\frac{kJ}{kg}\right)\left(1.60\,\frac{kJ}{kg * k}\right)(485 - 240)K + 0.0119\;kW = 57600\;kW$$



Schematic 1: example of a control volume for energy balance

Once we calculated the heat rate from the oil to the water, we now vary the pressure at 2 to optimize our electrical output power in the system. From this I found that a pressure of 3346 kW optimized the system. You can use this pressure value to help find the enthalpy at state 2 and 3. With that you can now calculate the mass flow rate:

$$\dot{m} = \frac{\dot{q}}{(H3 - H2)}$$

$$\dot{m} = \frac{(57600\;kW)}{(2800 - 212.66)kJ/kg} = 22.23\;\frac{kJ}{kg}$$

Now using the mass flow rate and the enthalpy from states 3 and 4, the work done in the turbine can be calculated as:

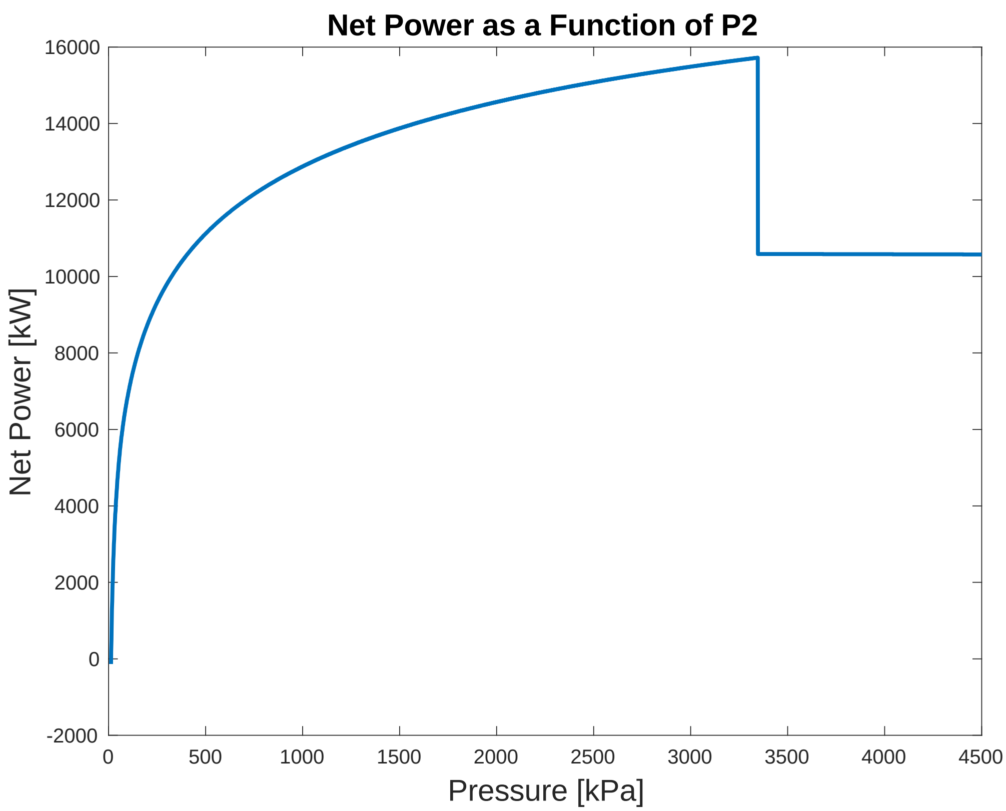
$$W_t = \dot{m} * (H3 - H4)$$

$$W_t = \left(22.23\frac{kJ}{kg}\right)(2800 - 2090)\frac{kJ}{kg} = 15.72\ MW$$

The work for the pumps can be calculated in a similar way with their values shown in table 4. Now the efficiency of the power plant can be calculated.

$$\eta = \frac{W_{net}}{Q_{in}} = \frac{W_t - W_{p,s} - W_{p,w}}{\dot{q}} = \frac{15.72 - \ 1.19e - 5 - 0.0745\ MW}{57.6\ MW} = .2731$$

<u>Plot</u>



From the plot above you can see that as the pressure at 2 is increased, the overall net power also increases until it gets to 3.346 MPa. At this point there is a spike down and then what looks to be a flat line. The line actually has a small slope, but I think the spike is caused by the steam quality decreasing which would hurt the efficiency of the turbine. I think this because you are at high pressure which could condense the steam.

Equations and Final Answer

Mass flow in Rankine cycle: $\dot{m} = \frac{\dot{q}}{(H3-H2)} = $ 22.23 kJ/kg

High Pressure level: 3.346 MPa (Took the pressure from the curve with maximum power output)

Net electric output:

$W_{net} = W_t - W_{p,s} - W_{p,w}$ = 15.72MW

Thermodynamic efficiency: $\eta = \frac{W_{net}}{Q_{in}} = 27.31\%$

Thermal efficiency: $\eta_t = 1 - \frac{T_c}{T_H} = 37.04\ \%$

Discussion

The result for the design of a parabolic trough power plant was not exactly what I expected. I computed an efficiency of 27% which is between the efficiency values of a parabolic trough power plant (15% and 39%)[7]. However, when looking at my thermal efficiency of 37.04% in the system you can see from *The Mechanics of Solar Power*[7] that this is well below the maximum efficiency seen in a thermal solar power plant. I think this could be because of my thermal oil I chose. If I had an oil with a higher heat capacity, I think the efficiency would go up. Fall the assumptions like the heat exchange in the boiler, having all the components besides the turbine as ideal and isentropic process I would have thought that the efficiencies to be higher.

One aspect that was not taken into account though was the cost. This could severally affect the parameters that can be made in the system[5]. With a pressure of 3.346 MPa the thickness of the pipes needed to counteract the pressure may cost too much and the pressure would then need to be lowered in order to improve cost.

References

[1] P. Weissensee, *1st Design Homework – Geothermal Power Plant,* MEMS 412 – Design of thermal systems, Washington University in Saint Louis.

[2] Sonntag, R. E., J., V. W. G., & Borgnakke, C. (2008). *Fundamentals of thermodynamics*. Hoboken, NJ: Wiley.

[3] Munson, B. R. (2014). Fundamentals of fluid mechanics (7th ed.). Hoboken, NJ: John Wiley & Sons, Inc.

[4] Paraschiv S., Paraschiv S. L., Ion I., Vatachi N., (2010). Design and Sizing Characteristics of a Solar Thermal Power Plant with Cylindrical Parabolic Concentrators in Dobrogeo Region, Dunarea De Jos" University of GalaTI, Romania, from https://www.dynalene.com/wp-content/uploads/2018/07/Dynalene_MS-2_Technical_Data_Sheet.pdf

[5] Kelly B., Kearney D., (2004), Parabolc Trough Solar System Piping Model, National Renewable Energy Laboratory, from https://www.nrel.gov/docs/fy06osti/40165.pdf

[6] Length/ Diameter of pipe https://www.researchgate.net/publication/301286435_Design_and_Sizing_Characteristics_of_a_Solar_Thermal_Power_Plant_with_Parabolic_Trough_Collectors_for_a_Typical_Site_in_Palestine

[7] Philip G. Jordan, (2014) The Mechanics of Solar Power, Solar Energy Market from

Used in discussion for comparing efficiency https://www.sciencedirect.com/topics/engineering/parabolic-trough

Appendix A

Matlab Code:

```
Febuary 6, 2020
% DOTS
% Professor Weisensee
% DHW 1

clc;
clear;
% This code calculates the efficiency of the max electrical power for a
% thermal solar power plant from some given assumption in the system.
% The process used to find this max output was to vary the Pressure at P2
% since it was equal P3 which would optimize the power coming from the
% turbine. Because we are varying P2 this caused some of the other variable
% to change throughout the calculations so they were put into a for loop
% and each iteration was recorded. In the end I found the optimized net
% power output to 3.3854 MW at a pressure for P2 of 33.4 Bars (3340 KPa)
% which made an efficiency of 60%.

% Known values in solar field
t1 = 240;     % [?C]
t2 = 480;     % [?C] source
t3 = 240;     % [?C]
m_dot = 150; % [KJ/Kg]
cp = 1.60;    % [KJ/Kg*k] source

% Values to find work in pump 1, which were all looked up.
v = 1/1760; % source specific volume [m^3/Kg]
l = 11;     % source length of pipe [m]
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```
d = 0.70;      % source diameter of pipe [m]
f = 0.38;  % Friction factor got from moody diagram.
rho = 1760; % Source [Kg/ m^3]
V = 2;      % Source value m/s

% Calc. Work lost from pump in soloar field.
wp1 = v*(f*l/d*.5*rho*V^2); % [kW]

% Calc. heat rate from solar field, this q_dot is now assumed to all go
%into the boiler and transfer to the water.
q_dot = m_dot*cp*(t2-t1) + wp1; % [KJ/s]

% Known values in the Rankine Cycle
T1 = 50;     % [?C]
P1 = .1235;  % [Bars] assumed that we are at saturated liquid.
S1 = 0.7037; % [KJ/Kg * k]
H1 = 209.31; % [KJ/ Kg]
T3 = 240;    % [?C]
P4 = P1;     % [KPa] since condenser assumed constant pressure.
S2 = S1;     % [KJ/Kg * k] assumed pump is reversible adiabatic.
n = .85;     % isentropic efficiency of the turbine.

% We are trying to vary P2 to get maximum work.
P2 = .12:0.01:45; % I have to have pressure in Bars b/c of the XSteam func.
P3 = P2;

for i = 1:length(P2)

    H2(i) = XSteam('h_ps', P2(i), S2);  % [KJ/ Kg]
    H3(i) = XSteam('h_pT', P3(i), T3);  % [KJ/ Kg]

    S3(i) = XSteam('s_pT', P3(i), T3);  % [KJ/Kg * k]

%S4(i) = XSteam('s_ph', P4, H4a(i));   % [KJ/Kg * k]
    H4t(i) = XSteam('h_ps', P4, S3(i)); % [KJ/ Kg]
    H4a(i) = H3(i) - n*(H3(i)-H4t(i));  % [KJ/ Kg]

    M_dot(i) = q_dot/(H3(i)-H2(i));     % [KJ/Kg]

    Wt(i) = M_dot(i)*(H3(i)-H4a(i));    % [kW]
end

% Calc. work lost in pump in rankine cycle.
for i = 1:length(P2)

WP2(i) = M_dot(i)*(H2(i) - H1);

end

% Total Power out of sytem
P = Wt - wp1 - WP2;         % [kW]
P_max = max(P);             % Maximum Electrical Power output. [kW]
Pressure = P2(P == P_max); % Gets pressure at max power. [Bars]
```

```
% eff = P_max/q_dot;
eff =P_max/q_dot;
TE = 1 - (T1+273)/(T3+ 273);
M = M_dot(P == P_max);

plot(P2*100, P,'LineWidth', 2)
xlabel('Pressure [kPa]','FontSize', 15)
ylabel('Net Power [kW]','FontSize', 15)
title('Net Power as a Function of P2','FontSize', 15)

% Calc. values when P is maximized for the table
H2_max = H2(P == P_max);
H3_max = H3(P == P_max);
H4a_max = H4a(P == P_max);
S3_max = S3(P == P_max);
S4 = XSteam('s_ph', .1235, 2092.3);
T4 = XSteam('t_ps', .1235, 6.5207);
T2 = XSteam('t_ps', 1000, 0.7037); % for some reason any value above 1000
WP2_max = WP2(P == P_max);         % will result in T2 become NAN.
```