

# Solar Parabolic Trough Simulation and Application for a Hybrid Power Plant in Thailand

Eron Jacobson<sup>a,b,c</sup>, Nipon Kejjoy<sup>a\*</sup>, Sukruedee Nathakaranakule<sup>a</sup> and Wattanapong Rakwichian<sup>a</sup>

<sup>a</sup> School of Renewable Energy Technology, Naresuan University, Phitsanulok 65000 Thailand.

<sup>b</sup> University of Washington Seattle, Washington 98101, USA.

<sup>c</sup> Seattle Engineer at the Boeing Company, P. O. Box 3707, Seattle, Washington 98124, USA.

\* Corresponding author, E-mail: niponk@nu.ac.th

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**ABSTRACT:** The purpose of this research project was to develop a software simulation of a solar parabolic trough for use in the design of a solar thermal power plant. In order to determine the optimum size of a parabolic trough to power a solar thermal power plant system in Thailand, a simulation is necessary to determine both long-term operating performance and short term local temperatures in the system. The simulation described in this paper was created for this purpose. It can evaluate both water and high temperature oil as the heat transfer fluid (HTF), in addition to the ability to vary many of the operating parameters of the system. The simulation was created to be a tool to determine the optimum parameters for the parabolic trough as the primary input to a power plant. It is necessary due to the many parameters that characterize a solar parabolic trough system, which must be considered when determining the output of the system. The simulation was constructed using methods from widely accepted reference materials and assumptions to the best ability.

**KEYWORDS:** System simulation, Solar parabolic trough, Solar thermal power plant, Optimum size, Heat transfer fluid.

## INTRODUCTION

The simulation described in this paper was created to be used as a design tool for the BioSun project. The goal of the BioSun project is to develop a concept design for a hybrid power plant system using a solar parabolic trough as the primary energy source with a backup system consisting of a biomass gasifier in

Thailand. These two energy sources will be used to provide heat to a thermal storage system, which will in turn be used to provide steam to drive a steam engine and 25 kW generators. This paper focuses on the concept design for the parabolic trough as the primary energy source. The very basic schematic of the proposed hybrid system is shown in Fig 1.

The energy output of the parabolic trough depends

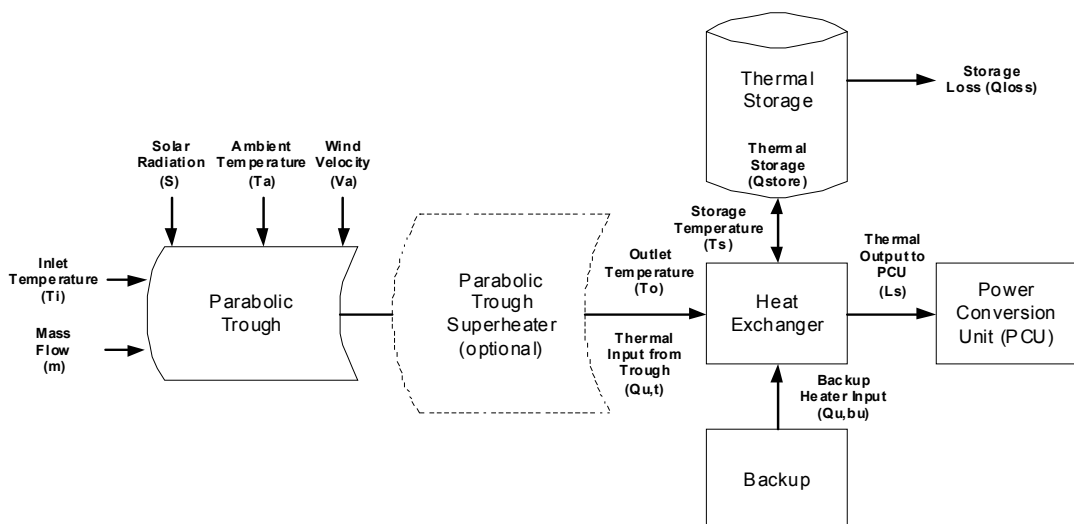


Fig 1. Simplified Block Diagram of Proposed Hybrid Power Plant.

**Table 1.** Monthly Average Daily Direct Radiation in Phitsanulok.

	Monthly Average Daily Direct (beam) Radiation, $H_{b,bar}$ (MJ/m <sup>2</sup> )												
	J	F	M	A	M	J	J	A	S	O	N	D	Avg
Phitsanulok	9.6	11.0	12.4	13.3	14.1	13.5	10.9	13.8	10.1	9.6	9.5	9.8	11.5

on many parameters including the solar radiation input, the size of the trough, and the heat transfer fluid (HTF). The equations describing the performance of a parabolic trough cannot be solved in a simple manner to find the optimum parameters of a trough for a desired energy output; rather the equations describing the parabolic trough must be solved for each given set of parameters and the resulting energy outputs compared to find the optimum parameters. This would be a tedious task if performed manually. Therefore it is necessary to have a simulation tool to easily compare results for given differing parameters. In order to find the optimum parameters for the parabolic trough to meet the demands of the BioSun system, a year simulation with hourly time step was constructed using an iterative process with Excel macro environment, which allows the parameters of the trough to be changed and the output HTF temperature and productive energy output to be known.

### Solar Radiation

Silapakorn University of Thailand performed a study to measure the total radiation for each province in Thailand<sup>1</sup>. For a parabolic trough, only the direct solar radiation is of practical use. To isolate the portion of the measured radiation that is diffuse and the portion of the radiation that is direct, the following steps were followed using Duffie and Beckman<sup>2</sup>. Given that the latitude ( $f$ ) of Phitsanulok is 16°, 47' N, the Monthly Average Extraterrestrial Radiation ( $H_{o,bar}$ ) was entered into the simulation as a lookup table using data from Duffie and Beckman<sup>2</sup>. Next the Monthly Average Clearness Index ( $K_{T,bar}$ ), the Sunset Hour ( $w_s$ ), and the Declination ( $d$ ) were calculated for each day using equations in Duffie and Beckman<sup>2</sup>. The fraction of the monthly average daily diffuse radiation and direct radiation were estimated using equations in Duffie and Beckman<sup>2</sup> (since  $w_s > 81.4^\circ$ , and  $0.3 \leq K_{T,bar} \leq 0.8$ ), and were entered into a lookup table in the simulation. The values for the Daily Direct Beam Radiation in Phitsanulok on a monthly average are listed in Table 1. The resulting hourly beam radiation data is the default input for the trough for a given day of the year.

### Optical Performance Angle of Incidence

For the case of the BioSun project in Thailand, a

single north-south axis rotational tracking trough was chosen to reduce the average angle of incidence over a yearly period. The Angle of Incidence ( $\theta$ ) is given by reference<sup>2</sup> in equation (1), while the Zenith Angle ( $\theta_z$ ) is found by equation (2) for an east-to-west tracking (north-south axis) trough:

$$\cos \theta = (\cos^2(\phi) + \cos^2(\delta) * \sin^2(w))^{1/2} \quad (1)$$

$$\cos \theta_z = \cos(\phi) * \cos(\delta) * \cos(w) + \sin(\phi) * \sin(\delta) \quad (2)$$

### Absorbance of Collector Pipe

The surface absorbance of a collector is dependant upon the angle of incidence of the solar ray to that surface. Since data is not typically available for describing the relationship of the ratio of angular absorbance to normal absorbance, it is estimated in reference<sup>2</sup> with equation (3) for angles of incidence between zero and 80°:

$$\frac{\alpha}{\alpha_n} = 1 + 2.0345e^{-3}\theta - 1.990e^{-4}\theta^2 + 5.324e^{-6}\theta^3 - 4.799e^{-8}\theta^4 \quad (3)$$

where  $\theta$  is in degrees

The final beam specular absorbance for the absorber pipe is then found by multiplying the ratio of equation (3) by the normal specular absorbance coefficient of the collector material itself.

### Transmittance, Reflection and Absorptance of a Single Cover System

The glass of the absorber cover is assumed to be a smooth surface of partially transparent material. Accordingly, the angle of refraction for the interface between the glass cover and the air can be calculated by equation (4)<sup>2</sup>:

$$\frac{n_1}{n_2} = \frac{\sin(\theta_1)}{\sin(\theta_2)} \quad (4)$$

where:  $n_1$  equals 1 (assumed refractive index of air),  $n_2$  is the refractive index of the glass,  $\theta_1$  is  $\theta$  (angle of incidence), and  $\theta_2$  is angle of refraction.

The perpendicular and parallel components of the unpolarized radiation passing from one medium (air) to the next (glass) are described by equations (5) and (6)<sup>2</sup>:

$$r_{\perp} = \frac{\sin^2(\theta_2 - \theta_1)}{\sin^2(\theta_2 + \theta_1)} \quad (5)$$

$$r_{\parallel} = \frac{\tan^2(\theta_2 - \theta_1)}{\tan^2(\theta_2 + \theta_1)} \quad (6)$$

For a one cover system as is assumed for the trough simulation, the transmittance of the initially unpolarized radiation is the average transmittance of the perpendicular and parallel components as described by equation (7)<sup>2</sup>:

$$\tau_r = \frac{1}{2} * \left( \frac{(1 - r_{\parallel})}{(1 + r_{\parallel})} + \frac{(1 - r_{\perp})}{(1 + r_{\perp})} \right) \quad (7)$$

**Absorption by Glazing**

The amount of radiation transmitted through a partially transparent medium (glass) is described by equation (8)<sup>2</sup>. The remaining portion of the radiation is that absorbed by the medium. This is described as glazing.

$$\tau_a = \frac{I_{transmitted}}{I_{incident}} = \exp\left(\frac{-K * L}{\cos\theta_2}\right) \quad (8)$$

where  $K$  is extinction coefficient of glass, and  $L$  is thickness of glass cover.

**Optical Properties of the Cover System**

Reference<sup>2</sup> suggests that the transmittance, reflectance and absorptance of a one cover system can be simplified into equation (9) for practical collector cover applications.

$$\tau = \tau_a * \tau_r \quad (9)$$

**Intercept Factor**

The intercept factor for a linear imaging concentrator (as is the case for the parabolic trough) is the fraction of the reflected radiation that is incident on the absorbing surface of the receiver. For a receiver of large enough diameter, the fraction of the reflected radiation is 1.0. For a perfect linear imaging concentrator, the diameter ( $D\gamma_{=1}$ ) of the cylindrical receiver for an Intercept Factor ( $\gamma$ ) equals to 1.0, which is described by equation (10)<sup>2</sup>. For a non-perfect imaging concentrator (defects in the surface of the reflector), the diameter ( $D\gamma_{=1}$ ) is described by (11)<sup>2</sup>:

$$D_{\gamma=1} = w * \frac{\sin(0.267)}{\sin(\phi_r)} \quad (10)$$

$$\phi_r = \tan^{-1}\left(\frac{8 * (f/w)}{16 * (f/w)^2 - 1}\right) \quad (10a)$$

$$D_{\gamma=1} = w * \frac{\sin(0.267 + \delta_d / 2)}{\sin(\phi_r)} \quad (11)$$

where  $w$  is width of the aperture,  $f$  is the focal length of the aperture, and  $\delta_d$  is the dispersion angle.

For the simulation, the trough is assumed to be imperfect, meaning that the dispersion angle is greater than zero. The intercept factor was thus chosen for the simulation as described in equations (12)<sup>2</sup>:

$$\gamma = \begin{cases} 1 & \text{if } D_o \geq D\gamma_{=1} \\ D_o / D\gamma_{=1} & \text{if } D_o < D\gamma_{=1} \end{cases} \quad (12)$$

where  $D\gamma_{=1}$  is given by equation (11).

It is worthwhile to note that using the proportion  $D_o / D\gamma_{=1}$  for the intercept factor is very conservative for the second case of equation (12) since the distribution of the radiation is theoretically a normal distribution.

**Overall Optical Efficiency**

The overall optical efficiency is the combination of the efficiencies from the cover transmission, collector absorptance, and the primary mirror reflectivity. This is shown in equation (13)<sup>2</sup>:

$$\eta_o(\theta) = \rho_e * \alpha * \tau * \gamma \quad (13)$$

where  $\rho_e$  is specular reflectivity of mirror.

**Incident Angle Modifier**

The effects of errors in the concentrating contours, tracking errors, and errors in displacement of receivers from the focus all lead to enlarged or shifted images and affect the intercept factor. These errors can be accounted for by using biaxial incidence angle modifiers. However, it is very specific for each collector, so in this study the LS3 specification is assumed to be used in Thailand. The simulation has included suggested Incident Angle Modifiers from reference<sup>2</sup> ( $K(\theta)_{dif}$ ) and that suggested for the LS3 design from reference<sup>3</sup> ( $K(\theta)_{LS3}$ ). The modifiers are given in equation (14) and (15):

$$K_{LS3}(\theta) = 1 + 0.078043 * \theta - 0.58047 * \theta^2 + 0.950559 * \theta^3 - 0.069092 * \theta^4 \quad (14)$$

where  $\theta$  is in radians.

$$K_{DUF}(\theta) = 1 - 6.74E^{-5} * \theta^2 + 1.64E^{-6} * \theta^3 - 2.51E^{-8} * \theta^4 \quad (15)$$

where  $\theta$  is in degrees.

However, there is a difference in the application of these two incidence angle modifiers. Equation (15) is to be applied to the Overall Optical Efficiency ( $\eta_o(\theta)$ ) where it is a function of the angle of incidence<sup>2</sup>. Equation (14) is to be applied to the Overall Optical Efficiency as a constant where it is evaluated at the normal<sup>3</sup>. Both of these calculations are made as shown in equations (16) and (17) and the smallest Modified Optical Coefficient ( $\eta_{o,mod}(\theta)$ ) is used in the simulation.

$$\eta_{o,mod}(\theta) = K(\theta)_{LS3} * \eta_o(0) \quad (16)$$

$$\eta_{o,mod}(\theta) = K(\theta)_{Duf} * \eta_o(\theta) \quad (17)$$

### End Effect Correction

The End Effect Correction for receivers of the same length as the reflectors is given by equation (18):

$$\xi(\theta) = 1 - (f/l) * \left( 1 + \frac{a^2}{48 * f^2} \right) * \tan(\theta) \quad (18)$$

### Absorbed Radiation

With all of the modifiers taken into account, the Absorbed Radiation (S) or the actual amount of radiation on the receiver is calculated by equation (19):

$$S = I_b * \eta_{o,mod}(\theta) * \zeta(\theta) * \cos \theta \quad (19)$$

### Thermal Performance and Losses

The thermal losses for a parabolic trough system are from convection and radiation from the absorber tube to ambient, and conduction from the absorber tube to the trough supporting structure itself. For the simulation, it was assumed that conduction from the absorber tube to the support structure is zero. It was also assumed that the cylindrical receiver is covered with a glass cover for the total distance of the trough. These assumptions were based on practical applications, but they may be areas for future improvement to the simulation.

### Heat Loss by Radiation

There are two radiation coefficients that are necessary to be calculated for a system with a glass cover. The first coefficient is that for radiation from the absorber pipe (receiver) to the cover. This coefficient is given in equation (20):

$$h_{r,r-c} = \sigma * \frac{(T_r^2 + T_c^2) * (T_r + T_c)}{\frac{(1 - \epsilon_r)}{\epsilon_r} + \frac{1}{F_{12}} + \frac{(1 - \epsilon_c) * D_o}{\epsilon_c * D_c}} \quad (20)$$

where  $T_r$  is temperature of the receiver (absorber pipe),  $T_c$  is temperature of the cover,  $\epsilon_r$  is emittance of the receiver (absorber pipe),  $\epsilon_c$  is emittance of the cover,  $D_o$  is receiver outer diameter,  $D_c$  is cover diameter (small thickness assumed),  $F_{12}$  is view factor between the receiver and the cover (assumed to be 1), and  $\sigma$  is Stefan Boltzman constant which  $5.67E^{-8}$ .

As reference<sup>2</sup> described, this equation has the assumptions that the surfaces are gray, are diffuse or specular-diffuse, are uniform in temperature and have uniform incident energy over the surface. As can be noted from equation (20), the temperature of the receiver and the temperature of the cover must be known to calculate this coefficient. The simulation attempts a logical estimate at the average receiver temperature for the length of the evaluation by equation (20a). This logical estimate is then refined by a second iteration of the evaluation length.

$$T_r = T_{in} + 0.25 * S * \left( \frac{(a - D_c) * l_{eval}}{m * C_p} \right) \quad (20a)$$

where  $l_{eval}$  is evaluation length of the trough.

The temperature of the cover is assumed in the simulation to be 20% of the difference between the receiver temperature and ambient greater than the ambient as described in equation (20a). The cover temperature is later checked in the simulation and a second iteration of the thermal losses is performed using the adjusted cover temperature.

$$T_c = 0.2 * (T_r - T_a) + T_a \quad (20b)$$

The second coefficient is radiation coefficient between the cover and the ambient air. This coefficient is described by equation (21):

$$h_{r,c-a} = 4 * \sigma * \epsilon_c * T_{loc}^3 \quad (21)$$

where  $T_{loc}$  is average (local) temperature between the cover and the ambient air =  $0.5 * (T_c + T_a)$ , and  $\epsilon_c$  is emittance of the cover.

### Convection to Ambient

The convection to ambient is a bit more complicated due to the fact that the fluid dynamics may call for a laminar or turbulent flow over the cover pipe depending on the calculated Reynolds Number (Re). The governing equation for the convection coefficient is described in equation (22), but the Nusselt Number (Nu) can be defined by two equations (22a) and (22b) as shown in reference<sup>2,4,5</sup>. The simulation incorporates equations

(22a) and (22b) in an *if-then* statement after evaluating the Reynolds Number (Re) in equation (22c).

$$h_w = \frac{Nu_a * k_a}{D_c} \quad (22)$$

$$Nu = 0.4 + 0.54 * Re_a^{0.53} \text{ for } 0.1 < Re_a < 1000 \quad (22a)$$

$$Nu_a = 0.3 * Re_a^{0.6} \text{ for } 1000 < Re_a < 50,000 \quad (22b)$$

$$Re_a = \frac{\rho_a * V_a * D_c}{\mu_a} \quad (22c)$$

where  $k_a$  is thermal conductivity of air,  $\rho_a$  is density of air,  $V_a$  is velocity of air,  $\mu_a$  is kinematic viscosity of air, and  $D_c$  is cover diameter.

The characteristics of air  $k_a$ ,  $\rho_a$ , and  $\mu_a$  are dependent on the air temperature. Therefore, the simulation incorporated these characteristics from a reference<sup>2</sup> appendix into a lookup table to be evaluated on a basis of a linear interpolation on the air temperature.

### Overall Loss Coefficient and Cover Temperature

The Overall Loss Coefficient combines the thermal losses into one coefficient. For the assumption that the area between the cover and the receiver (absorber pipe) is a stagnant quantity of air, the coefficient on a per length basis is calculated by equation (23):

$$U_L = \left( \frac{D_o}{(h_w + h_{r,c-a}) * D_c} + \frac{1}{h_{r,r-c}} \right)^{-1} \quad (23)$$

The Cover Temperature ( $T_c$ ) is evaluated by performing an energy balance on the cover thus generating equation (24):

$$T_c = \frac{D_o * h_{r,r-c} * T_r + D_c * (h_{r,c-a} + h_w) * T_a}{D_o * h_{r,r-c} + D_c * (h_{r,c-a} + h_w)} \quad (24)$$

A second iteration of the thermal losses and Overall Loss Coefficient is performed in the simulation with the adjusted cover temperature. This iteration was found to provide acceptable accuracy for the Overall Loss Coefficient; additional iterations were deemed unnecessary.

### Heat Transfer to Fluid

The heat transfer from the receiver pipe to the fluid (water or HTF) must be characterized by turbulent or laminar flow conditions. Accordingly, the simulation evaluates the Reynolds Number of the fluid ( $Re_f$ ) of

equation (25a) and uses an *if-then* statement between equations (25) (for turbulent flow) and (26) (for laminar flow) to calculate the Nusselt Number of the fluid ( $Nu_f$ )<sup>5</sup>. The Heat Transfer Coefficient to the fluid ( $h_f$ ) is then evaluated by equation (27):

$$Nu_f = \frac{(f/8) * Re_f * Pr_f}{1.07 + 12.7 * \sqrt{(f/8)} * \left( Pr_f^{2/3} - 1 \right)} * \left( \frac{u_f}{u_w} \right)^{n_1} \quad (25)$$

for  $Re_f > 2200$

$$Re_f = \frac{4 * m}{\pi * D_i * u_f} \quad (25a)$$

$$f = \left( 0.79 * \ln(Re_f - 1.64) \right)^{-2} \quad (25b)$$

$$Nu_f = 3.7 \text{ for } Re_f < 2200 \quad (26)$$

$$h_f = \frac{Nu_f * k_f}{D_i} \quad (27)$$

where  $Pr_f$  is Prandlt Number of the fluid,  $\mu_f$  is kinematic viscosity of the heat transfer fluid,  $\mu_w$  is kinematic viscosity of water at atmospheric conditions,  $m$  is mass flow of fluid,  $D_i$  is receiver inner diameter, and  $n_1$  is 0.11.

For the case laminar flow ( $Re_f < 2200$ ), a fully developed hydrodynamic and thermal profile was assumed. The Nusselt Number of the fluid was chosen to be 3.7 for the case of a constant wall temperature rather than the 4.4 for a constant heat flux. This choice was made because it results in a more conservative heat transfer coefficient although the theoretical performance of a solar collector lies somewhere between a constant temperature condition and a constant heat flux condition.

The characteristics of the fluid and water  $\mu_w$ ,  $\mu_f$ ,  $k_f$ , and  $Pr_f$  are dependent on the water and oil temperatures. Therefore, the simulation incorporated these characteristics from a reference<sup>4</sup> appendix and from reference<sup>6</sup> into lookup tables to be evaluated by linear interpolation based on the water and VP1 oil temperatures respectively. Unless later noted, for the first iteration for the evaluation of the trough performance, these characteristics are evaluated at the fluid inlet temperature.

### Overall Heat Transfer Coefficient and Factors

The Overall Heat Transfer Coefficient ( $U_o$ ) is the coefficient for heat transfer from the surroundings to the fluid, based on the outer diameter of the receiver

pipe. This is given in reference<sup>2</sup> as equation (28):

$$U_o = \left( \frac{1}{U_L} + \frac{D_o}{h_f * D_i} + \frac{D_o * \ln\left(\frac{D_o}{D_i}\right)}{2k} \right)^{-1} \quad (28)$$

where  $k$  is thermal conductivity of receiver pipe. It is convenient to define a Collector Efficiency Factor ( $F'$ ) at this point by equation (29):

$$F' = \frac{U_o}{U_L} \quad (29)$$

The Collector Flow Factor ( $F''$ ) is then described in equation (30):

$$F'' = \frac{F_R}{F'} = \frac{m * C_p}{A_r * U_L * F'} * \left( 1 - \exp\left(\frac{A_r * U_L * F'}{m * C_p}\right) \right) \quad (30)$$

where  $C_p$  is the specific heat of the fluid,  $A_r$  is the area of the receiver ( $D_o * l * n$ ),  $l$  is length of receiver per module,  $n$  is number of modules.

The specific heat of the HTF ( $C_p$ ) is dependent on the HTF temperature. Therefore, the simulation incorporated these characteristics from a reference<sup>4</sup> appendix and reference<sup>6</sup> into a lookup table to be evaluated by linear interpolation based on the fluid temperature.

The Useful Energy Gain ( $Q_u$ ) is then calculated as described in reference<sup>2</sup> with equation (31):

$$Q_u = F_R * A_a * \left( S - \frac{A_r}{A_a} * U_L * (T_i - T_a) \right) \quad (31)$$

where  $A_a$  is area of the aperture ( $(a-a_o)*l$ ),  $a$  is the width of the aperture,  $a_o$  is width of aperture shaded by the receiver, and  $T_i$  is fluid inlet temperature.

**First and Second Iterations**

The collector performance is dependant on the overall loss coefficient and the internal heat transfer coefficient to the heat transfer media. Both of these characteristics are a function of temperature, which requires that they are based on an estimate in the first iteration. The estimate for the heat transfer coefficient is calculated using the temperature into the trough for the first iteration. The estimate for the mean receiver temperature for the first iteration is calculated by equation (32).

$$T_{rm,1st} = T_i + 0.25 * S * \frac{A_a}{C_p * m} \quad (32)$$

A second iteration is then performed for each evaluation length of the trough using the performance factors calculated in the first iteration in order to refine these estimates. The mean fluid temperature ( $T_{fm}$ ) is calculated by equation (33), and the properties of the heat transfer fluid are then evaluated at this temperature.

$$T_{fm} = T_{in} + \frac{Q_u/A_c}{F_R * U_L} * (1 - F'') \quad (33)$$

The mean receiver temperature is calculated by equation (34).

$$T_{rm} = T_{in} + \frac{Q_u/A_c}{F_R * U_L} * (1 - F_R) \quad (34)$$

The iteration process is shown in Fig 2. The second iteration of the simulation refines the calculated performance and useful gain of the trough.

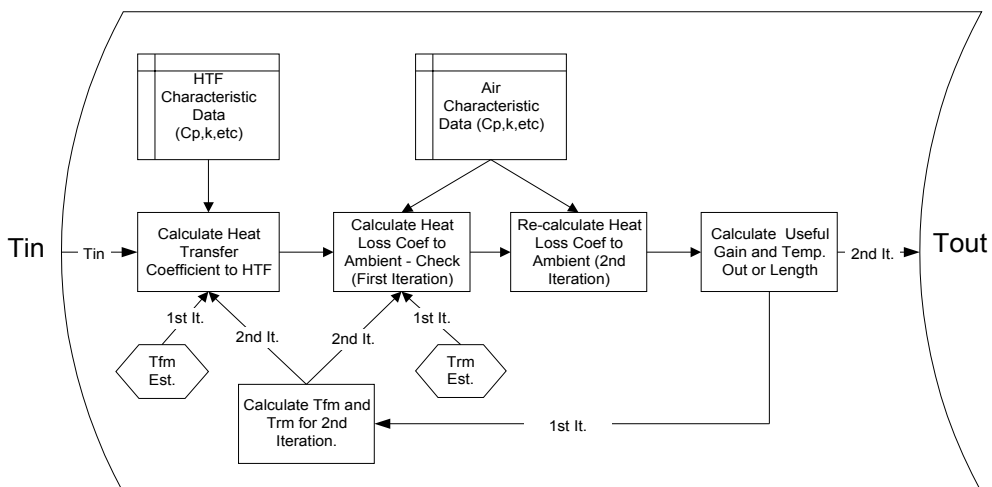
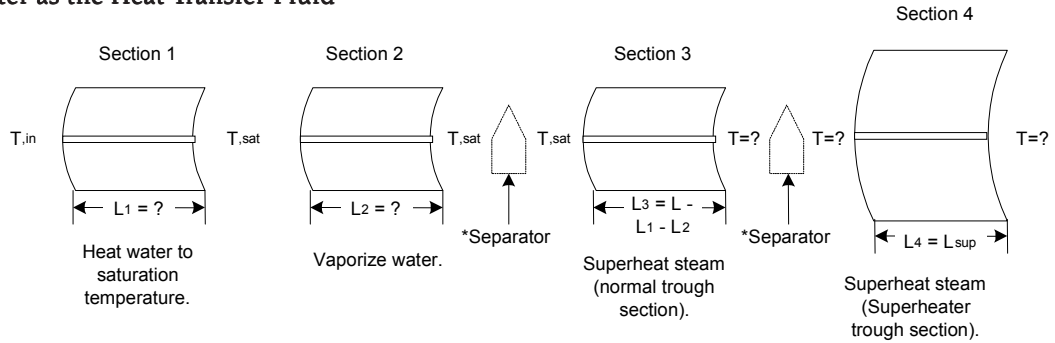


Fig 2. Iteration Process for Trough Performance Evaluation.

### Evaluation of the Parabolic Trough Water as the Heat Transfer Fluid



\*The separator may be in the first or second position depending on vaporizing conditions. There is only **one** separator.

**Fig 3.** Evaluation Sections of Parabolic Trough Using Water as the HTF.

Water as the heat transfer media makes the rather simple calculations up to this point much more complicated. This has to do with the phase change from liquid to gas that water goes through upon heating to temperatures in a parabolic trough application, and the property changes that occur during the phase changes (Fig 3).

#### First Section Evaluation: Heating the Water to Saturation Temperature

In this section, the output temperature is known rather than the length of heating. For this reason, equations (30) and (31), which depend on the length of heating for evaluation, could not be used. Since the length of the trough to heat the fluid temperature to the saturation temperature is not known, different means must be used to calculate the useful gain. The strategy adopted in the simulation is to use equation (35) from reference<sup>2</sup> which calculates the useful gain per meter based on the fluid temperature. The average fluid temperature ( $T_{fm}$ ) for this section of the trough can be assumed to be the average between the inlet fluid temperature and the saturation temperature. The average receiver temperature ( $T_{rm}$ ) is estimated as the average fluid temperature for the first iteration.

$$q'_{u1} = F' \cdot \frac{A_a}{L} \cdot \left( S - \frac{A_r}{A_a} \cdot U_L \cdot (T_f - T_a) \right) \tag{35}$$

where

$$T_{fm} = \left( \frac{T_{in} + T_{sat}}{2} \right) \tag{35a}$$

$T_{in}$  is temperature of the water into the trough,  $T_{sat}$  is saturation temperature of the water at the  $P_{in}$ ,  $P_{in}$  is pressure at inlet.

The characteristics of the fluid  $T_{sat}$  is dependent on

the water pressure. Therefore, the simulation incorporated these characteristics from a reference<sup>3</sup> appendix into a lookup table to be evaluated by linear interpolation based on the water pressure.

The section of the parabolic trough which serves to heat the water to its saturation temperature may have laminar and/or turbulent flow. The Nusselt Number and therefore the heat transfer coefficient of a turbulent fluid can be much higher than those for a laminar fluid. So it is not enough to take the Nusselt Number at the average temperature ( $T_p$ ) of the section because this may represent the characteristic flow for only slightly more than half the length. Instead, the simulation evaluates the Reynolds number at both the inlet temperature and saturation temperature to check for laminar and turbulent flow. If both are found to be turbulent or if both are found to be laminar, then equation (25) or (26) can be applied using the average Reynolds Number and Prandtl Number. If both laminar and turbulent flow exists, then equation (36) is applied to get a better estimate of the average Nusselt Number<sup>4</sup>. Fig 4 describes the flow characteristics of concern.

$$Nu_{ave} = 3.7 \cdot \frac{Re_{2200} - Re_{in}}{Re_{sat} - Re_{in}} + \frac{Re_{sat} - Re_{2200}}{Re_{sat} - Re_{in}} \cdot \frac{(f/8) \cdot Re_{turb,ave} \cdot Pr_{turb,ave}}{1.07 + 12.7 \cdot \sqrt{(f/8)} \cdot (Pr_{turb,ave}^{2/3} - 1)} \cdot \left( \frac{\mu}{\mu_w} \right)^{0.11} \tag{36}$$

$$f = \left( 0.79 \cdot \ln(Re_{turb,ave} - 1.64) \right)^{-2} \tag{36a}$$

where

$$Re_{turb,ave} = \left( \frac{Re_{2200} + Re_{sat}}{2} \right), \tag{36b}$$

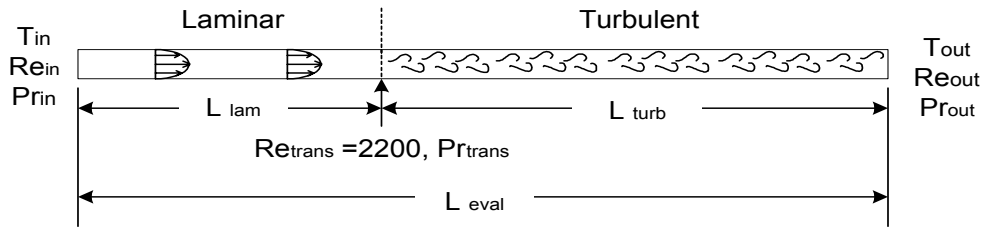


Fig 4. Transition of Flow as a Fluid Temperature Increases.

$$Pr_{turb,ave} = \left( \frac{Pr_{2200} + Pr_{sat}}{2} \right), \tag{36c}$$

$$x = \frac{L_{sep} - L_{sat}}{L_b} \tag{40}$$

where

$$L_{sep} = l * n_{sep}, \tag{40a}$$

$Re_{2200}$  is 2200 (turbulent flow beginning),  $Re_{sat}$  is Reynolds Number at saturation temperature,  $Re_{in}$  is Reynolds Number at inlet temperature,  $Pr_{2200}$  is Prandtl Number at  $Re$  equal to 2200,  $Pr_{sat}$  is Prandtl Number at saturation temperature,  $\mu$  is average of viscosity of water at  $T_{in}$  and  $T_{sat}$ , and  $\mu_w$  is viscosity of water at  $T_{rm}$

$n_{sep}$  is the number of the module that the separator follows. If the length to saturation ( $L_{sat}$ ) is greater than the length to the separator ( $L_{sep}$ ), then the quality of the steam is zero.

Using equation (37), the length to heat the water to saturation ( $L_{sat}$ ) is calculated.

The separator is assumed to be adiabatic in the simulation so that all the steam that enters also exits. This assumption is not conservative and may be an area for future improvement to the simulation. The mass flow out of the separator ( $m_s$ ) is calculated by equation (41):

$$L_{sat} = (T_{sat} - T_{in}) * \frac{C_p * m}{q_u} \tag{37}$$

$$m_s = x * m \tag{41}$$

For the second iteration of this section, the mean receiver temperature ( $T_{rm}$ ) is calculated using equation (38) and the characteristics from the first iteration:

The enthalpy of the steam as it enters the separator, or if there is no separator present, as it leaves the trough without being superheated ( $h_{w0,2}$ ), is calculated by equation (42):

$$T_{rm} = T_{fm} + Q_u' * \left( \frac{1}{h_f} + \frac{\ln(D_o/D_i)}{2\pi k} \right) \tag{38}$$

$$h_{w0,2} = h_f + x * h_{fg} \tag{42}$$

**Second Section Evaluation: Vaporizing the Water**

The second section for evaluating water as the heat transfer media in the parabolic trough is very similar to the first section in that the length of evaluation is not necessarily known, but the energy to vaporize the water ( $h_{fg}$ ) being based only on the pressure of the water is found using linear interpolation through a lookup table from reference<sup>2</sup>. The assumption made here is that the vaporization of the water in the receiver is a constant pressure process. The length to boil ( $L_b$ ) is then calculated by equation (39):

where  $h_g$  is the enthalpy of liquid water at the saturation temperature.

For the second iteration of this section, the mean receiver temperature ( $T_{rm}$ ) is calculated using equation (43) and the characteristics from the first iteration:

$$L_b = \frac{h_{fg} * m}{Q_u} \tag{39}$$

$$T_{rm} = T_{fm} + Q_u' * \left( \frac{1}{h_f} + \frac{\ln(D_o/D_i)}{2\pi k} \right) \tag{43}$$

**Third Evaluation Section: Superheating the Steam in the Normal Trough**

where  $Q_u'$  is useful gain per meter.

The quality of the steam ( $x$ ) is then calculated by equation (40):

The evaluation length of superheating in the normal trough ( $L_{e,w}$ ) can be defined by two means. If a superheating section of normal trough exists after the separator, then the length is defined by equation (44), and the mass flow for evaluation is the mass flow out of the separator ( $m_s$ ).



$$L_{e,w} = L - L_{sep} \tag{44}$$

where  $L = n * l$ . (44a)

If the water is fully vaporized before the separator, then the superheating evaluation length for the normal trough is defined by equation (45). This equation is applicable if there is a superheating section of the regular trough or not.

$$L_{e,w} = L - L_b \tag{45}$$

The average temperature of the receiver ( $T_{rm}$ ) for the first iteration is calculated using equation (32) as described in section 4.

The enthalpy of the water out of this section ( $h_{w03}$ ) is given by equation (46):

$$h_{w03} = h_g + \frac{Q_u}{m_s} \tag{46}$$

**Fourth Evaluation Section: Superheating the Steam in the Superheater Section**

The Superheater Section is a physically larger size section of the parabolic trough, which requires its own full analysis of the optical characteristics described in Section 3 using the geometry of the larger trough. The simulation incorporates this analysis and evaluates the Superheater Section similar to the previous sections. The input steam may be at one of two conditions: if there is no superheating of the steam in the regular section of the trough, then the input steam will be saturated vapor, if there is superheating in the regular section of the trough, then the steam will be at the output conditions of the third section. Since the likely gain for the steam in this section at a higher temperature is less than if the steam were at the saturation temperature, equation (47) is used to estimate the mean receiver temperature ( $T_{rm}$ ) for the first iteration if the input steam has been superheated. If the steam is saturated vapor, then equation (32) is used. This is accomplished through an *if-then* statement.

$$T_{rm,1st} = T_i + 0.05 * S * \frac{A_a}{C_p * m} \tag{47}$$

**Oil as the Heat Transfer Fluid**

The simulation divides the parabolic trough into three sections to evaluate oil as the HTF. The input for the second and third section is simply the output from the first and second section, respectively. It was deemed appropriate to use only three sections to evaluate the trough because of the iterative nature of the simulation (Fig 5).

With oil as the heat transfer fluid, the equations of Section 4 need very little modification. It is only worthy

to note that the evaluation length for this analysis is given by equation (48).

$$L_{e,H} = \frac{l * n}{3} \tag{48}$$

Once the useful gain for the trough is calculated, the temperature out of the oil ( $T_{oH}$ ) is calculated by equation (49).

$$T_{oH} = T_i + \frac{Q_u}{m * C_p} \tag{49}$$

The output enthalpy of the oil ( $h_{oH}$ ) is then evaluated by interpolating from the lookup table of the oil properties.

**Efficiency of the Parabolic Trough**

The efficiency of the parabolic trough is the amount of useful gain from the parabolic trough divided by the amount of energy input to the parabolic trough area. It is calculated by the simulation in equation (50).

$$\eta_{trough} = \frac{Q_{u,tot}}{I_{direct} * A_a} \tag{50}$$

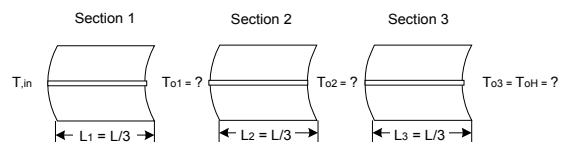
where  $Q_{u,tot}$  is the total useful gain for the trough for one hour,  $I_{direct}$  = the direct beam radiation for one hour, and  $A_a$  = the area of the aperture/reflector.

**Output to Thermal Storage and Efficiencies**

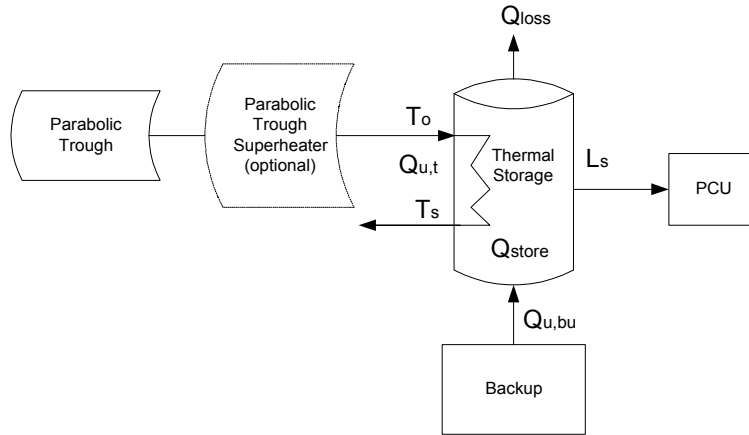
For the simulation, the trough is assumed to provide heat to a thermal storage system. The advantage of having a thermal storage system is that it can store the thermal energy generated during the peak radiation hours for later use when solar radiation is unavailable. The simulation assumes that there is a backup heating system to keep the thermal storage at a certain minimum temperature in the event that the solar energy input is not adequate to do so. The thermal storage system is used to power a steam engine and generator or Power Conversion Unit (PCU) via a thermal water loop. This load may be constant or may have a distinct hourly load profile.

**Thermal Storage**

The simulation models the thermal storage as an unstratified thermal storage tank as described in reference<sup>2</sup>. The storage tank temperature change is evaluated on an hourly basis corresponding to the



**Fig 5.** Evaluation Sections Using Oil as the HTF



**Fig 6.** Basic Thermal Analysis of Thermal Storage.

simulation of the parabolic trough. The governing equation is described in equation (51).

$$T_s^+ = T_s + \frac{\Delta t}{(mCp)_s} * \left[ \begin{matrix} Q_{u,t} + Q_{u,bu} - L_s \\ -(UA)_s * (T_s - T_a) \end{matrix} \right] \quad (51)$$

where  $T_s$  is the initial temperature of the storage media,  $T_s^+$  = the temperature of the storage media after one hour,  $(mCp)_s$  is mass times the specific heat of the storage media,  $\Delta t$  is one hour,  $Q_{u,t}$  is thermal input from the parabolic trough,  $Q_{u,bu}$  is thermal input from the backup heater,  $L_s$  is the thermal output to the PCU,  $(UA)_s$  is loss-coefficient-area product of the storage system, and  $T_a$  is ambient temperature.

**Parabolic Trough Input**

If the temperature of the heat transfer fluid from the parabolic trough is above the initial temperature of the storage media, the thermal input from the parabolic trough ( $Q_{u,t}$ ) is defined by equation (52). If the temperature of the HTF is below the initial temperature of the storage media, the HTF is incapable of providing thermal input.

$$Q_{u,t} = m_f * (h_{ti} - h_{to}) \quad (52)$$

where  $h_{ti}$  is the enthalpy of the HTF out of the trough,  $h_{to}$  is the enthalpy of the HTF at  $T_s$ , and  $m_f$  is mass flow of the HTF out of the trough.

**Output to PCU**

The thermal output to the PCU per hour is calculated by equation (53).

$$L_s = 3600 * \frac{gen_1}{E_{pcu}} \quad (53)$$

where  $gen_1$  is the load on the generator per hour, and

$E_{pcu}$  is efficiency of the PCU at load.

**Backup Heater Input**

The Backup System is used to keep the temperature of the storage media at the temperature set point. If the parabolic trough cannot provide enough input to the thermal storage, the input is defined by equation (54):

$$Q_{u,bu} = (T_{s,set} - T_s) * (mCp)_s + L_s + (UA)_s * (T_s - T_a) \quad (54)$$

where  $T_{s,set}$  is the temperature set-point of the thermal storage.

**Solar Fraction**

The solar fraction is the amount of input to the system that is from solar energy compared to the total input to the system. It is defined for one day by equation (55).

$$SF = \frac{Q_{u,t}}{Q_{u,t} + Q_{u,bu}} \quad (55)$$

**Use of the Simulation as a Sizing Tool**

The solar parabolic trough power plant specification for the simulation is shown in Table 2.

**Comparison of Water and VP1 Oil for Maximum Clear Sky Radiation**

In order to demonstrate the capability of the simulation, an evaluation was performed for a 96 meter parabolic trough with characteristics similar to the LS3 design from reference<sup>7</sup>. The length of 96 meters was chosen because it is twice the size of the existing trough at the School for Renewable Energy Technology (SERT), and is one proposed size for the BioSun project. The load on the PCU was considered to be a constant 25 kW and the efficiency of the PCU to be a constant 40%. The

**Table 2.** Input for the simulation.

Solar Trough Concentrator	
Width ( $w$ ) [m]	5.77
Module length ( $l$ ) [m]	6
Focal length ( $f$ ) [m]	1.71
Gap width ( $a_g$ ) [m]	0.1
Specific reflectance ( $rs$ )	0.9
Number of modules ( $n$ )	16
Incident angle modifier $K(\cdot)$	LS3
Dispersion angle ( $D$ ) [degree]	0.1
Solar Trough Receiver	
Cover diameter ( $D_c$ ) [m]	0.09
Receiver inner diameter ( $D_i$ ) [m]	0.05
Receiver outer diameter ( $D_o$ ) [m]	0.07
Absorber length ( $l$ ) [m]	12.5
Thermal conductivity ( $k_a$ ) [W/m.°C]	47.6
Receiver emittance ( $\mu_r$ )	0.91
Cover emittance ( $\mu_c$ )	0.88
Cover thickness ( $L$ ) [m]	0.0025
Cover extinction coefficient ( $K$ ) [ $m^{-1}$ ]	32
Specular absorbance ( $\pm_n$ )	0.93
Cover refractive index ( $n_c$ )	1.526
Heating fluid (Water = 1, HTF = 2)	1
Superheater Section Concentrator	
Superheater width ( $w_s$ ) [m]	5.77
Superheater module length ( $l$ ) [m]	20/40
Superheater focal length ( $f_s$ ) [m]	1.71
Number of superheater modules ( $n_s$ )	2
Superheater specific reflectance ( $rs_s$ )	0.9
Superheater incident angle modifier $K(\cdot)_s$	LS3
Dispersion angle ( $D$ ) [degree]	0.1
Superheater Section Receiver	
Superheater cover diameter ( $D_{c,s}$ ) [m]	0.15
Superheater receiver inner diameter ( $D_{i,s}$ ) [m]	0.08
Superheater receiver outer diameter ( $D_{o,s}$ ) [m]	0.1
Superheater absorber length ( $l_s$ ) [m]	6
Superheater thermal conductivity ( $k_{a,s}$ ) [W/m.°C]	47.6
Superheater receiver emittance ( $\mu_{r,s}$ )	0.91
Superheater cover emittance ( $\mu_{c,s}$ )	0.88
Superheater cover thickness ( $L_s$ ) [m]	0.0025
Superheater cover extinction coefficient ( $K_s$ ) [ $m^{-1}$ ]	32
Superheater specular absorbance ( $\pm_{n,s}$ )	0.93
Superheater cover refractive index ( $n_{c,s}$ )	1.526
Superheater after which module?	0
Other	
Latitude (Phitsanulok Province) [degree]	16.75
Day of the year ( $n$ )	116
Ambient Conditions	
Temperature ( $T_a$ ) [K]	298
Wind Velocity ( $V_a$ ) [m/s]	5
Fluid Conditions	
Inlet Temperature ( $T_i$ ) [K]	323
Mass flow ( $m$ ) [kg/s]	0.07
Inlet pressure ( $P_i$ ) [MPa]	1
Separator after which module? ( $n_{sep}$ )	16
Thermal Storage	
Mass of storage media ( $m_s$ ) [kg]	1,000
Set point temperature of storage ( $T_s$ ) [K]	498
Loss coefficient area product ( $UA_s$ ) [W/K]	10
Engine/Power Conversion Unit (PCU)	
Daily constant load on generator ( $gen_l$ ) [kW]	25
Efficiency of PCU ( $E_{pcu}$ )	0.5

trough was evaluated with both water and oil as the HTF, but with slightly different parameters including thermal storage mass and a Superheater trough section length.

**Water Output for a Maximum Clear Sky Radiation Input**

This evaluation was performed with two lengths of a Superheater section with optical characteristics similar to the LS3 design, 20 and 40 meters, in order to investigate if a greater length of trough is possible to produce superheated steam. The mass of the thermal storage ( $m_s$ ) was found to be inconsequential because the parabolic trough was not able to provide enough energy to charge the thermal storage. Fig 7 and Fig 8 show the temperatures and energy inputs of the system.

As would be expected, the trend shows that a larger Solar Fraction (SF) can be achieved with a longer Superheater section. However, the length of the regular trough section is the parameter that is more responsible for the energy output of the trough, because this dictates how much water is vaporized. If the amount of water that is vaporized and is able to be superheated is small, the temperature output will be high, but the energy output will be low. Therefore, the mass flow out of the separator and the placement of the separator are very important characteristics to study when using the simulation. These considerations must be taken into account when deciding the final size of the trough sections.

**VP1 Oil Output for a Maximum Clear Sky Radiation Input**

This evaluation was performed with two different arbitrary but realistic masses of thermal storage, 4,000 and 6,000 kg. The VP1 thermal oil was not evaluated with a Superheater section because the oil has a maximum temperature of 415° Celsius. Fig 9 and Fig 10 show the temperatures and energy inputs of the system.

The thermal storage mass plays a significant role for this evaluation because the parabolic trough is able to produce enough energy to charge the thermal storage. As can be seen from the results, a larger storage mass equates to a larger solar fraction. This is because the temperature of the storage increases slower with a larger mass, and the amount of energy input from the trough depends on the temperature difference between the oil and the thermal storage.

**CONCLUSION**

The simulation described in this paper was created to be a tool to determine the optimum parameters for a parabolic trough as the primary input to a hybrid power plant. This simulation tool is necessary due to

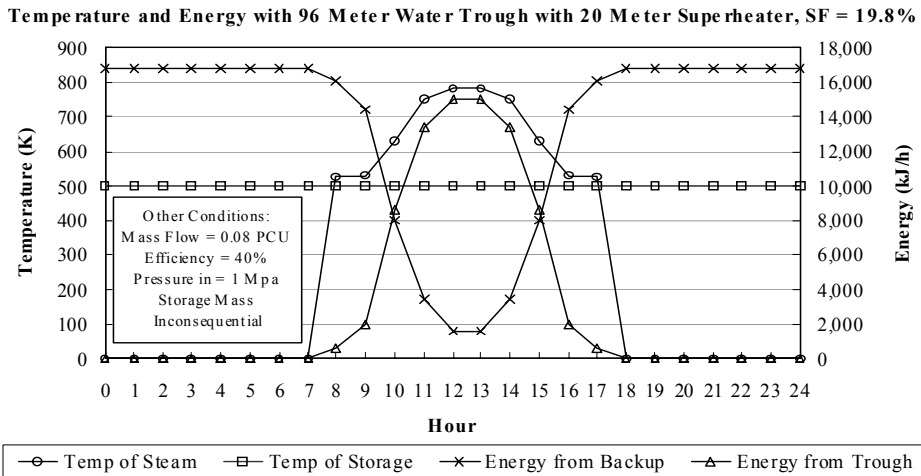


Fig 7. Water Output 1 for Maximum Clear Sky Radiation.

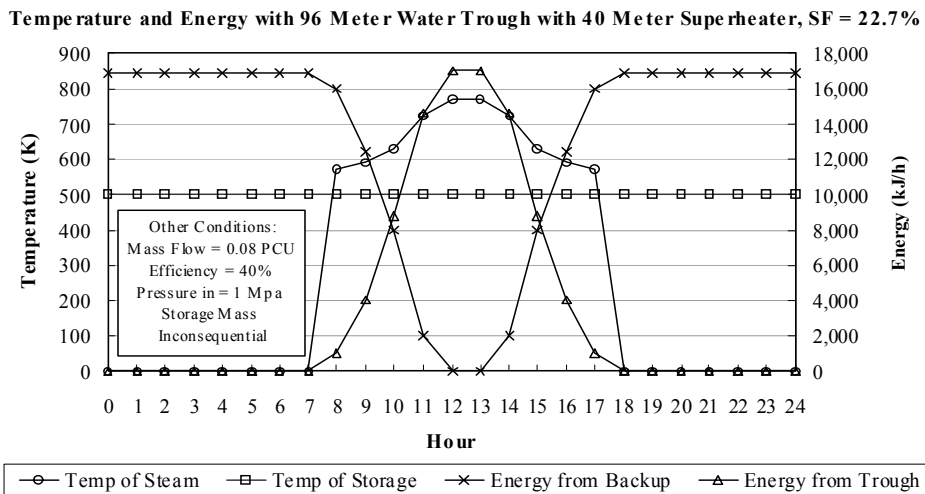


Fig 8. Water Output 2 for Maximum Clear Sky Radiation.

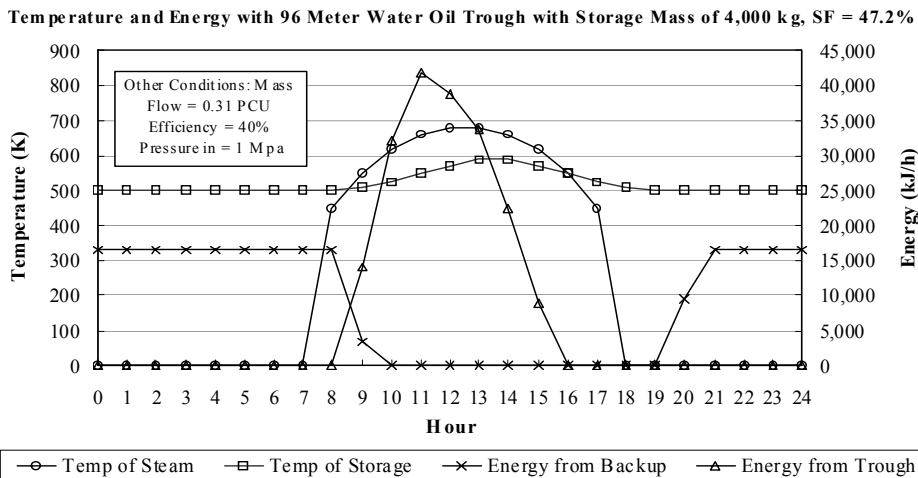


Fig 9. VP1 Oil Output 1 for Maximum Clear Sky Radiation.

Temperature and Energy with 96 Meter Water Oil Trough with Storage Mass of 6,000 kg, SF = 47.2%

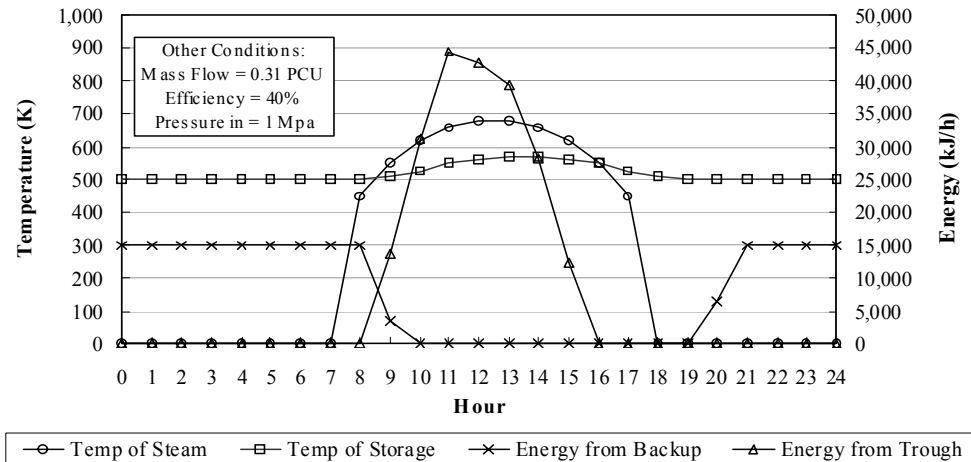


Fig 10. VPI Oil Output 2 for Maximum Clear Sky Radiation.

the many parameters that characterize the system that must be considered when determining the output of the system. It was constructed using methods from widely accepted reference materials and assumptions to the best ability. However, it should prove useful as a tool for approximately sizing a parabolic trough for a hybrid power plant or for building block to more complex simulations.

The result of this study gives guidance for the possible use of parabolic trough application for the BioSun project. However, the detail study of each component is needed for the final decision of the system. Further studies should include a dynamic simulation for the parabolic trough and a model for the power plant in order to predict the system behavior under climate of Thailand.

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