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Effect of the Condenser Pressure and Normalized Steam Mass Flow Rate on the Normalized Net Work Output of the Solar Power Plants

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Abstract**—***The growing problem of global warming and environmental pollution has become an urgent necessity in order to avoid these dangers and to solve these problems it is necessary to expand the use of solar power stations., in this study was conducted simulations for solar power plant with parabolic trough concentrates, and the study focused on the effect of pressure and mass flow rate of steam which is generated in the solar field, on the work, which is produced in the power plant in general. A comprehensive model for the simulation of a regenerative Rankine cycle was developed. The results obtained showed that the output power from the cycles is affected mainly by: the inlet heat transfer fluid (HTF) temperature, mass flow rate of the HTF and the condenser pressure. The cycle parameters were normalized and it was found that the performance was independent of the power block size. A linear regression was proposed by using the normalized variables, the results showed that the linear equation represents accurately the trend given by the results obtained in the simulation.*

Keywords—*Condenser pressure, steam mass flow rate, net work output,solar power plant.*

I. INTRODUCTION

The Rankin cycle is usually used in the power stations that use concentrated solar thermal energy [1,2,4]. The thermodynamic cycle is clearly recognizable by scheme 1. As shown in the diagram, the heat transfer fluid (HTF) is conducted through three heat exchangers: superheater, boiler and preheater. The condition of the saturated liquid in the preheater is achieved by using a closed type feed water heater (CF-1), which is supplied with pressurized water, the heater is usually a tube shell, and is one of the widely used heat exchanger models. When the saturated liquid is transferred through the boiler, it is subject to a phase shift from the liquid to the vapor. The saturated steam is transferred to the superheater from the boiler, which is considered as a steam generator, which is in the form of a shell and tube heat exchanger as the heat transfer fluid enters the tube and liquid of feedwater comes through the shell.

The addition of heat energy is caused by the increased heating of the steam in the superheater, and the steam is converted into superheated steam phase. The heat exchanger in the superheater is also a shell tube type, where the superheated steam (10) is pushed into the high-pressure turbine.

At this stage, two points are drawn, 11 and 12, which are extracted from the high-pressure turbine to the closed feedwater (CF-1, CF-2). For the purpose of increasing the temperature of the feedwater, closed feedwater heaters, which are usually shell tube type extraction device, are used by condensing the extracted steam [4,5].

When electrical stations include closed feedwaters, this means an increase in their thermal efficiency, and depending on the economic feasibility, the optimum number of heaters is determined, and through previous studies, five heaters are the optimal number for use [6,8]. To obtain a higher total thermal efficiency in the thermal cycle, the remaining steam is reheated at point (13) of the thermal cycle, and this process provides high pressure in the boiler and avoids the problem of low-quality steam in pressure of turbine exhaust. After the reheating process aforementioned, the reheated steam is expanded at point (14) in the low pressure turbine; four extraction operations are carried out after this stage (15-18), three of which are using closed feedwater (CF-3-5) and one using open feedwater (OF). A heat exchanger of direct contact type is used in open feedwater [4,7], where steam is obtained in the saturated state (6) at an average temperature by mixing of different streams in varying temperature. There is another purpose of the use of open feedwaters, namely the removal of air and other dissolved gases that may lead to reducing the overall thermal performance in addition erosion in the parts of the cycle. The feed water coming from the closed feedwater (CF-5) is mixed with the exhaust steam of the turbine (19). The point where the feed water is drawn by the closed feedwater is a valve that only allows water to pass through the low pressure area [4]. A change phase occurs in the coming mixture when entering the condenser, which is a type of tube and shell heat exchanger, as the mixture turns into a saturated liquid.

By the condenser pump, the feed water that leaves the condenser is pumped at point (1) which is in saturated liquid state to the open water feeder. The open water feeder (5) is equipped with feed water after being reheated with closed water feeders (CF3-5). At the end of the thermal cycle, the saturated liquid returns to the prheater (9) by pumping it at the boiler pressure and prheating it with closed water feeders (CF1-2) after it leaves the open water feeder (6).

The pressure, temperature and mass flow rate of the stream at each point of the thermal cycle are obtained by conducting an energy and mass balance under the steady state conditions of each part of the cycle. Thereafter the cycle is modeled under conditions of partial load thermodynamically.

Because the solar field intermittently absorbs energy from the sun during the day, this is lead to perform under partial load conditions.

II. MATH

The mass flow rate and temperature of the heat transfer fluid in the inlet of the power block is affected by the conditions described above. When the temperature of the heat transfer fluid is constant in this case the mass flow rate will be variable. Patnode approximation is used in heat exchangers under partial load conditions[1]:

$$\frac{UA}{UA_{ref}} = \left(\frac{\dot{m}_o}{\dot{m}_{o,ref}}\right)^{0,8} , \tag{1}$$

where: UA, UA_{ref} - the overall heat transfer coefficient for the current and reference conditions respectively,
A - the heat transfer area,
$\dot{m}_o$,$\dot{m}_{o,ref}$ - the mass flow rates of the outer fluid in the heat exchanger for the current and reference conditions respectively.

Under the Colborn equation that is used to describe fluid flow and to which this approach is subject, the mass flow rate of the incoming and outgoing fluids remains at a similar rate in both the partial and reference loads:

$$\frac{\dot{m}_i}{\dot{m}_o} = \frac{\dot{m}_{i,ref}}{\dot{m}_{o,ref}} , \tag{2}$$

$$\dot{W}_{HP,t} = \dot{m}_{10}h_{10} - \dot{m}_{11}h_{11} - \dot{m}_{12}h_{12} - \dot{m}_{13}h_{13} \tag{3}$$

$$\dot{W}_{LP,t} = \dot{m}_{14}h_{14} - \dot{m}_{15}h_{15} - \dot{m}_{16}h_{16} - \dot{m}_{17}h_{17} - \dot{m}_{18}h_{18} - \dot{m}_{19}h_{19} \tag{4}$$

$$\dot{W}_t = \dot{W}_{HP,t} + \dot{W}_{LP,t} \tag{5}$$

The work of pump is given by:

$$\dot{W}_{pump,cond} = \dot{m}_2 h_2 - \dot{m}_1 h_1 \tag{6}$$

$$\dot{W}_{pump,OF} = \dot{m}_7 h_7 - \dot{m}_6 h_6 \tag{7}$$

The net output of electric power, $\dot{W}_e$, is calculated by multiplying the net power product of the cycle by the generator efficiency:

$$\dot{W}_e = \dot{W}_t \eta_{generator} \tag{8}$$

The net work is given by:

$$\dot{W}_{net} = \dot{W}_e - \dot{W}_{pump,cond} - \dot{W}_{pump,OF} \tag{9}$$

Under partial load conditions, the relationship of generator efficiency to the load (Figure 2) is described as follows [1]:

$$\eta_{generator} = 0{,}90 + 0{,}258\,x - 0{,}3x^2 + 0{,}12x^3 , \tag{10}$$

$$x = \frac{\dot{W}_t}{\dot{W}_{t,nom}}$$

Under nominal conditions, by dividing the output cycle parameters by their respective values, the natural variables are obtained:

$$\frac{\dot{W}_{net}}{\dot{W}_{net,nom}}, \quad \frac{\dot{Q}_c}{\dot{Q}_{c,nom}}, \quad \frac{T_{HTF,rec}}{T_{HTF,rec,nom}} \tag{11}$$

III. RESULTS AND DISCUSSION

Using steam tables with the international standard, the implementation of the properties of water and steam thermodynamically [9,10]. Table 1. shows the parameters included in the simulations.

Table 2 presents the results obtained under nominal conditions, and Temperature - entropy diagram of the renovated Rankine cycle can be illustrated by in Figure 3.

The nonlinear algebraic equations for heat and mass balance were solved simultaneously by using computerized fluid dynamics and fortran power station 95. For the purpose of obtaining a comprehensive solution to the power block under partial load conditions, normalized variables were used.

Figure 4 shows the results obtained for 50 MWt and 80 MWt. The results proved that normalized attitude is not affected by the size of the power block. A comparison has been made between the power model proposed in this study (50 MWt) and the simulation that was applied by Patnode 35 MWt [1], in order to verify the last conclusion. As an input to the proposed model of power block, different boiling pressures, 90 and 100 bar, has been used, as shown in Figure 5. Whereas, 90 bar represents the nominal boiler pressure, while 100 bar represents the boiler pressure in the study carried out by Patnode [1]. The results are proven that the values obtained for the normalized gross electric output in both cases approximate the values obtained by Patnode, and that the boiler pressure has a slight effect on these values. The fact that any change in the input parameters will affect the accuracy of the results obtained in this study, so it is recommended to use the input parameters contained in Table 1, as well as we got the result that the proposed model fits the ability of different size.

The condenser pressure and the normalized steam mass flow rate affect adversely the normalized net work output. As shown in Figure 6, the normalized steam mass flow rate can be decreased down to a certain value, below which no feasible solutions are reached. The mass and energy balance of the open water feeder and the closed water feeder (CF-5) determine this imposed value.

Figure 6 shows the influence of normalized steam mass flow rate and condenser pressure on the normalized turbine extraction mass, $T_{HTF,a}$ = 310,350,390 °C.

The mass flow rates $\dot{m}_{15}$, $\dot{m}_{18}$ and the normalized steam mass flow rate $\dot{m}_{steam}/\dot{m}_{steam,ref}$ are reduced to achieve mass and energy balance.

The maximum values of the normalized net work output can be obtained by making low pressure for condenser and high inlet temperature of heat transfer fluid liquid heat transfer inside. Therefore, to optimize the net power output at a given condenser pressure, the HTF inlet temperature should be kept at the nominal conditions4 by adjusting the HTF mass flow rate.

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IV. CONCLUSIONS

The trend obtained is similar to the normalized net work output. Another important variable in the analysis is the return HTF temperature, the Results shows the normalized return HTF temperature for the condenser pressure at nominal conditions is independent of the condenser pressure and is affected by the HTF inlet temperature and the normalized steam mass flow rate.

TABLE 1. Cycle parameters assumed for the simulation

Parameter	Value	Reference
Heat Transfer Fluid (HTF)		
Inlet Temperature	390 ºC	[1]
Fluid	VP-1, Hitec	[15, 16]
Rankine Cycle		
Gross Electric Power, $\dot{W}_e$	50 MW	[3]
High Pressure	90 bar	[3]
High pressure turbine efficiency	85,50 %	[3]
Low pressure turbine efficiency	89,50 %	[3,17]
Reheat Pressure	0,19 P_{high}	[12, 13]
Condenser Pump Efficiency	75 %	[3]
Open Feedwater Pump Efficiency	78 %	[3]
Terminal Temperature difference Closed Feedwater	2,8 ºC	[14, 11]
Condenser Pressure	0,08 bar	[3]
ΔTpinch	10 ºC	[14]

TABLE 2. Cycle parameters obtained at nominal conditions

Variable		Value	Units
Cycle			
Power, work, thermal energy	$\dot{W}_{HP,t}$	15856,7	kW
	$\dot{W}_{LP,t}$	35453,6	kW
	$\dot{W}_{pump,cond}$	48,7	kW
	$\dot{W}_{pump,OF}$	654,95	kW
	$\dot{W}_{net}$	48641,8	kW
	$\dot{Q}_{boiler}+\dot{Q}_{superheater}$	134614,8	kW
Cycle efficiency	η_{cycle}	37,87	%
Energy losses in the condenser	$\dot{Q}_c$	79423,76	kW
Steam consumption	$\dot{m}_{steam}$	56,75	kg/s
Heat Transfer Fluid			
mass flow rate	$\dot{m}_{HTF,VP-1}$	507,87	kg/s
	$\dot{m}_{HTF,VP-1,superheater}$	53,34	kg/s
	$\dot{m}_{HTF,\ Hitec,superheater}$	76.2	kg/s
	$\dot{m}_{HTF,Hitec}$	783,83	kg/s
Temperature	$T_{HTF,VP-1,ret}$	293,7	ºC
	$T_{HTF,\ Hitec,ret}$	294,7	ºC

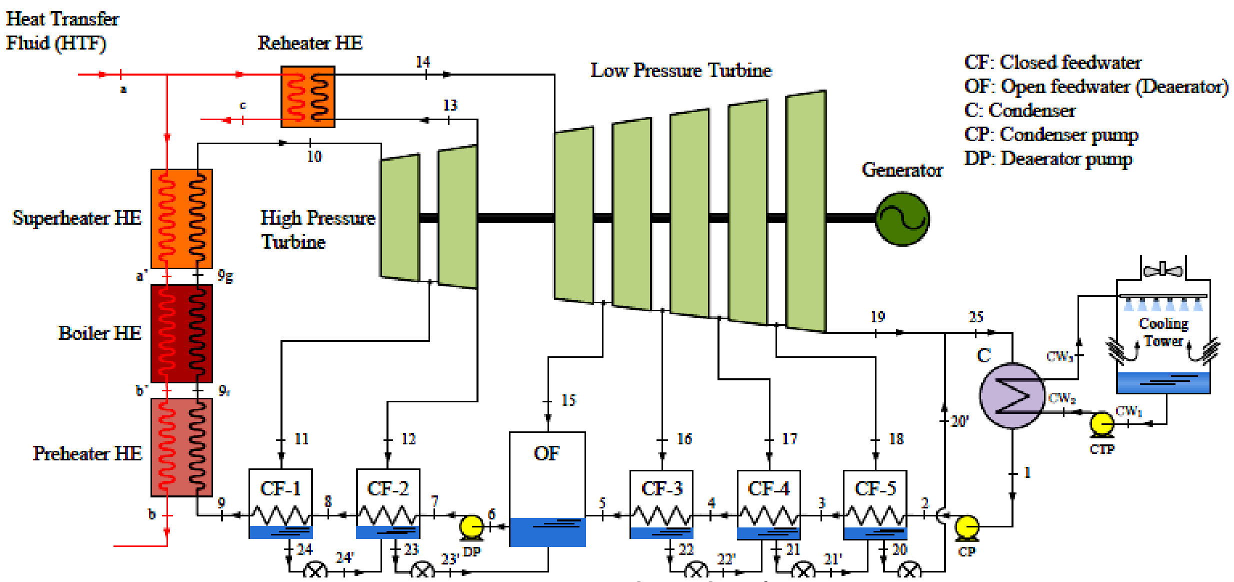

Figure 1. Regenerative Rankine cycle configuration

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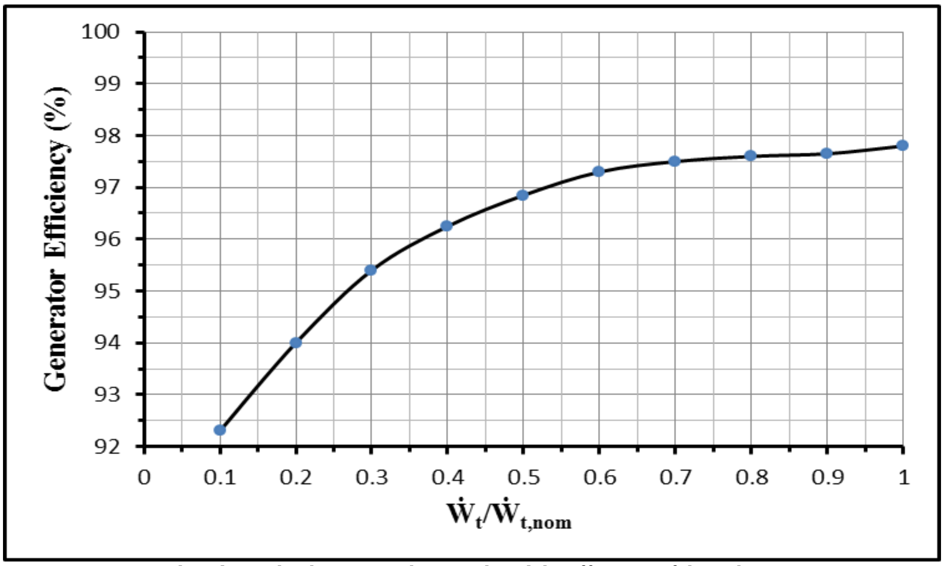


Figure 2. The relationship between turbine work and the efficiency of the turbine generator

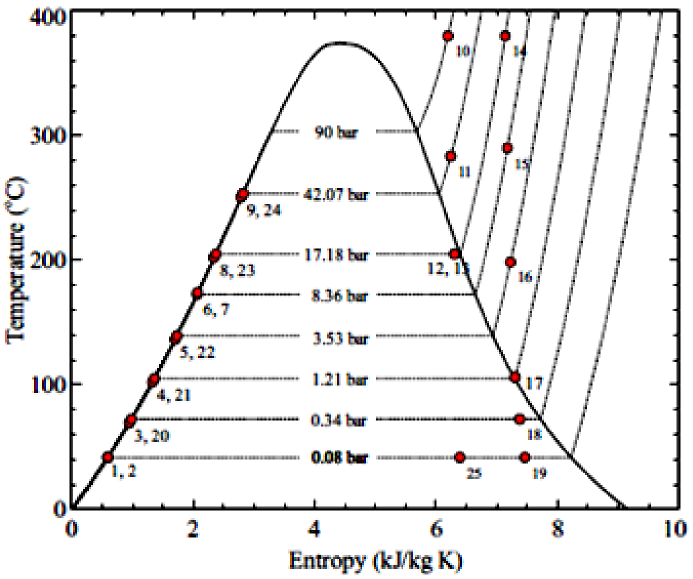


Figure 3. Temperature - entropy diagram of the renovated Rankine cycle

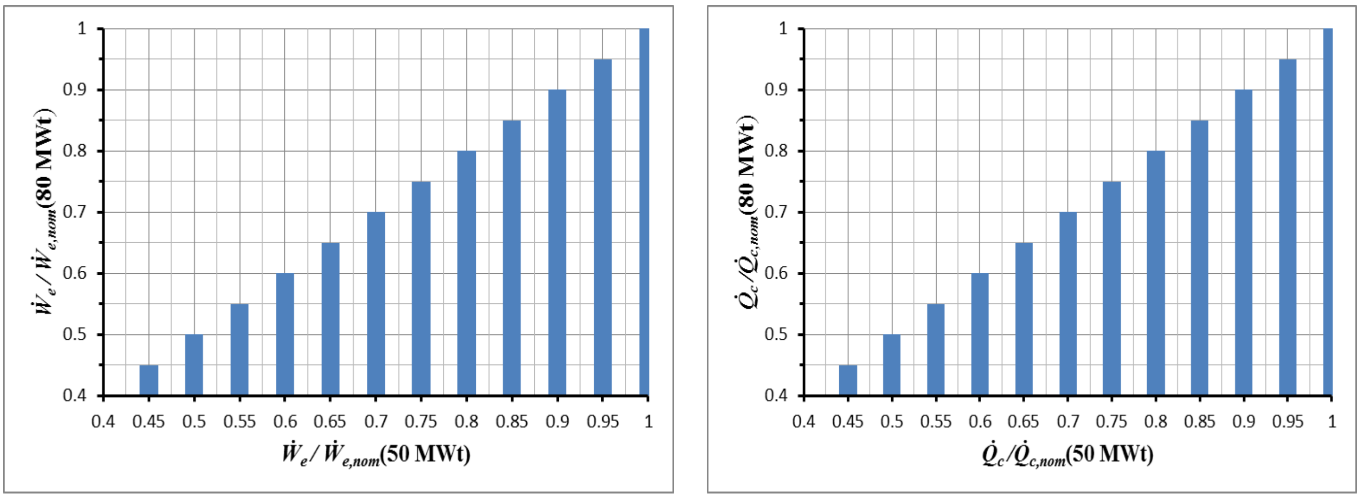


Figure 4. the influence of the size of the power plant on the normalized electrical work and the heat transfer rate of condenser

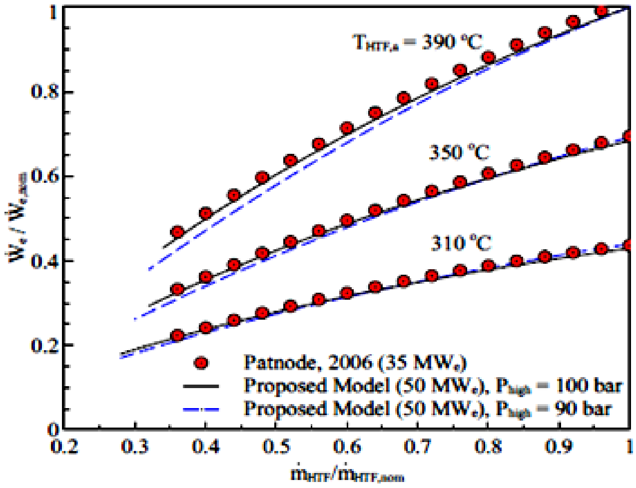
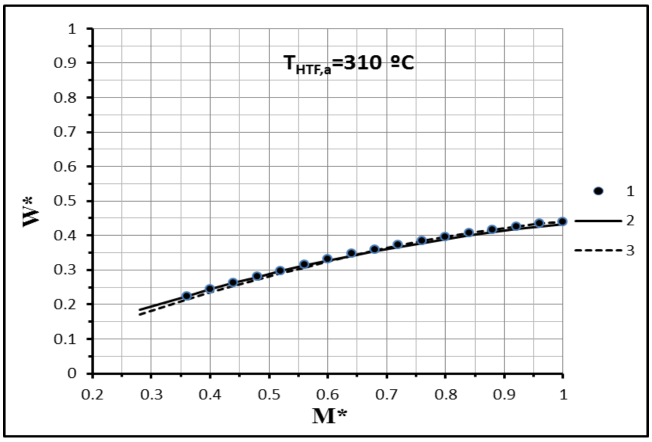
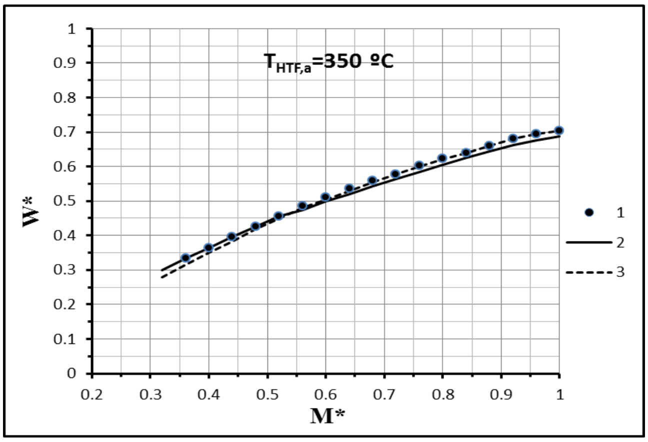
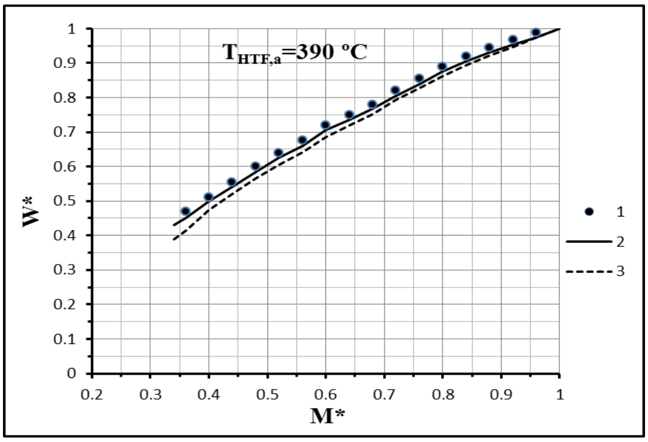


Figure 5. Comparison between the normalized electrical output obtained from the proposed model and the Patnode [1].







$W^{*}=\dot{W}_e/\dot{W}_{e,nom}$, $M^{*}=\dot{m}_{HTF}/\dot{m}_{HTF,nom}$

1- Model Patnode 2006 (35 MWt);
2- The present model (50 MWt), P_{high} = 10 MPa;
3- The present model (50 MWt), P_{high} = 9 MPa.

Figure 6. shows the results obtained for the normalized net work output with the mass flow rate of heat transfer fluid.

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