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Enhanced Model To Simulate The Performances Of A Steam Power Plant At Different Loads

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Abstract – *Thermal steam power plants are indispensable for global electricity generation. However, assessing their performances under different working loads is crucial for their enhancement. This study aims to develop a comprehensive model using Matlab to simulate the performance of an existing thermal steam power plant (Achouat plant), enhanced with regeneration and re-superheat features. Additionally, an energy analysis is conducted to evaluate its efficiency under both full and partial load conditions. This analysis aims to certify and compare the performance and parameters of different plant components across various operating loads. The obtained results demonstrate that the developed model performs well in modeling the studied plant under various loads. Moreover, the findings affirm that the plant operates most optimally at full load and higher operating loads, showcasing an overall energy efficiency ranging from 33.24% to 37.38% across operating loads from 20% to 100%.*

***Keywords**: Achouat power plant, Heat exchangers, Load, Off-design, Performance simulation.*

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I. Introduction

During the last century, global energy consumption has experienced rapid growth, driven by an expanding population and increasing prosperity. As a result of this growth, demand has increased for virtually all sources - coal, oil, natural gas, nuclear energy, and renewable energies. These additions to the global energy system manage to create a generally positive influence on higher living standards, the reduction of global poverty, and longer livelihoods.

In Algeria and in the recent years, the demand for electricity has increased significantly, especially during the summer when it reaches its high levels. This increase is due to population growth, improvements in life quality, as well as the importance given to both economic and industrial sectors. On the other side, steam thermal power plants play an important role in meeting Algeria's electricity needs. In 2017, the power generation from this type of plants reached 10,074 GWh, which represents 12% of the total national installed capacity [1].

Steam thermal power plants or plants based on steam turbines presents a fertile ground for scientific research.

The primary objectives in this field are to develop models to simulate the behavior of these systems and to enhance their performances. Over recent years, many studies have been performed to explore such topics. For instance, Ahmadi and Toghraie [2] conducted a detailed case study on the Shahid Montazeri steam power plant in Iran, which has a unit capacity of 200 MW. The study involved applying mass, energy, and exergy balance equations to analyze each component of the entire installation individually. They calculated metrics such as energy and exergy efficiencies and irreversibility using a model developed with Engineering Equation Solver software. Additionally, Akbari et al. [3] examined the influence of heat recovery steam generator (HRSG) pressure levels on the performance and efficiency of an existing combined power plant, using the Montazeri steam plant in Iran as a case study. Their research evaluated three different HRSG configurations and assessed their impacts on the boiler, steam turbines, and condenser design. Furthermore, Wang et al. [4] proposed and assessed a new coordinate control strategy to improve the operational flexibility of a double-reheat coal-fired power plant. The dynamic simulations validated the model, revealing that the reheat steam experienced significant overheating when the power ramp rate increased. To mitigate this issue, they introduced a new coordinated control strategy utilizing high-pressure extraction steam throttling, which effectively addressed the reheat steam overtemperature and enhanced the power ramp rate. Moreover, other works have been presented in the literature to showcase different models and configurations of steam power plants [5-9].

However, there are few studies that focus on simulating the performances of a steam power plant at different operating loads in the literature. Therefore, the objective of this work is to develop a new model to simulate the performances of a steam thermal power plant for electricity production at different operating loads, taking the existing "Achouat" plant as a case study. First, an energy analysis of the plant and its various components at full load will be conducted. Then, an energy analysis of the steam plant at different working loads is also presented. Furthermore, the results obtained from the different performances of the plant and its components during the variation of the operating load are considered to compare them and choose the optimal operating conditions of our studied system for improvement.

II. Data and methodology

II.1. Methodology

As stated in the introduction section, the primary aim of this study is to simulate the performance of an existing steam thermal power plant at various operating loads. To achieve this, a Matlab code has been developed utilizing the first law of thermodynamics, mathematical correlations, and design approaches for turbomachinery and heat exchangers, whether involving phase change or not, to simulate both overall plant performance and the performance of each component. Additionally, Refprop software was employed to extract the thermophysical properties of the working fluids (water/steam and air).

The methodology of the present study follows these main steps: Firstly, the main data and inputs of the considered existing steam plant are gathered and collected. Additionally, the main assumptions and boundary conditions assumed in the analysis are considered and presented. Then, the governing equations and correlation of each component of the whole installation are summarized. Subsequently, and in order to ensure the viability of the developed model, its results are validated with data of the considered plant. Finally, the effect of the working charge of the plant on the performances of the main components of the plant, as well as the global performances of the whole installation, is simulated and analyzed.

II.2. Studied plant

The studied thermal power plant of Jijel which named "Achouat" is located in the northeast of Algeria, on the edge of the Mediterranean Sea (Latitude: 36°49′13″ North, Longitude: 5°46′00″ East, Altitude relative to sea level: 9 m). It covers an area of 60 hectares; The thermal power plant is mainly composed of 03 turbo alternators (steam turbines) with a unit power of 210 MW. The schematic layout of the studied plant, including its components, is depicted in Figure 1, while the main inputs at nominal conditions (full working charge) were presented in Table 1.

Table 1. Summary of main data of the investigated plant at nominal conditions

Parameter		Value
Ambient conditions	Temperature [°C]	25.0
	Pressure [bar]	1.01325
Inlet conditions into the HPT	Temperature [°C]	540
	Pressure [bar]	127.5
Outlet conditions into the HPT	Temperature [°C]	540
	Pressure [bar]	26.7
Inlet conditions into the LPT	Temperature [°C]	540
	Pressure [bar]	23.48
Outlet conditions into the LPT	Temperature [°C]	33.81
	Pressure [bar]	0.0527
Isentropic/mechanical efficiency of HPT/LPT [%]		88/97.5
Isentropic efficiency of pumps [%]		87
Efficiency of the regenerator [%]		99
Pressure drop ΔP in the boiler [bar]		51
Pressure drop ΔP in the superheater [bar]		3.22
Pressure drop ΔP in the deaerator [bar]		0.6
Lower heating value of the fuel [kJ/kg]		28938
Condensing pressure [bar]		0.0527
Temperature at the outlet of the superheater [°C]		540
Generated power of the plant [MW]		210

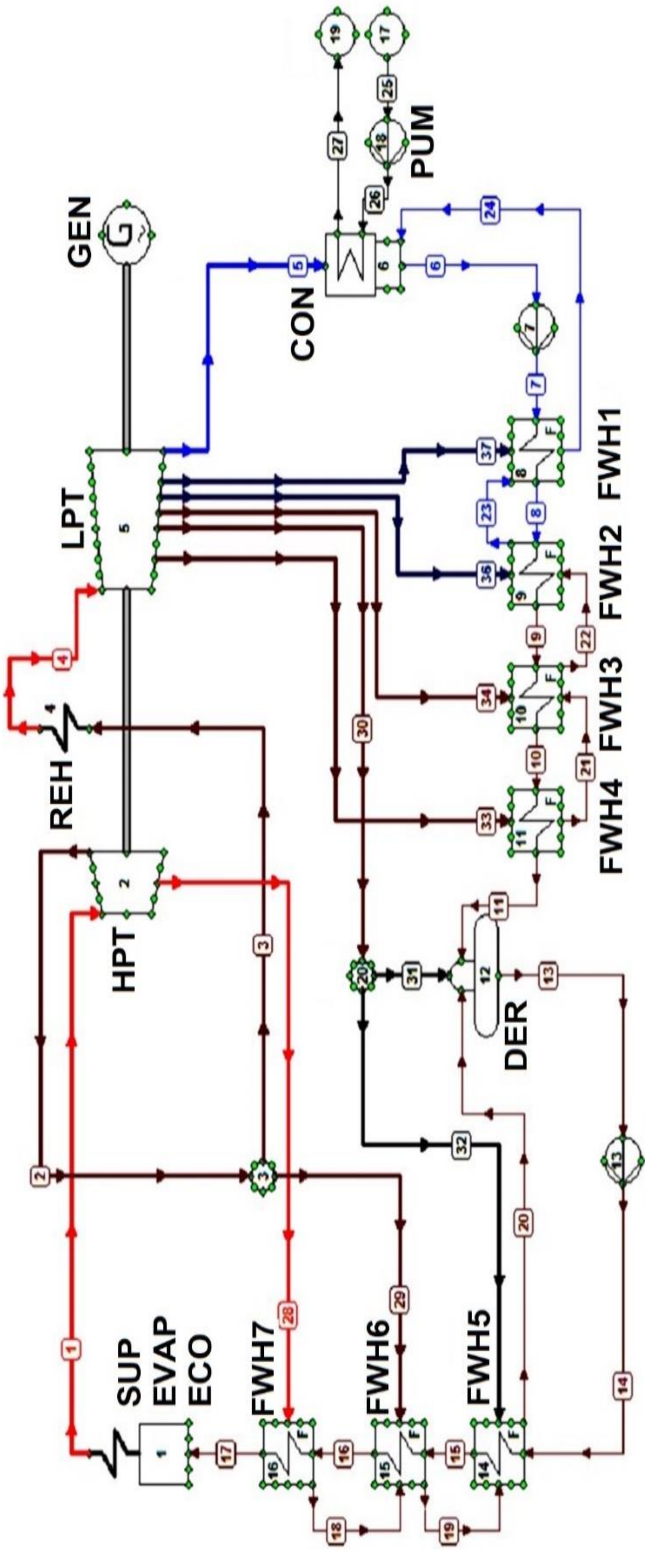

Figure 1. Schematic layout of Achouat steam thermal power plant

Furthermore, the dimensions of the heat exchangers including the preheater, the boiler, the superheater, the reheater, as well as the condenser, the seven feedwater heaters, and the deaerator, which are very important in the calculation of their global heat transfer coefficient at different charges, were summarized in Table 2.

Table 2. Summary of dimensions for integrated heat exchangers in the investigated plant

HE	D_i [m]	D_0 [m]	e [m]	N_{tube} [-]	L_{tube} [m]
FWH1	0.017	0.022	0.001	704	6.6
FWH2-4	0.016	0.022	0.001	1060	5.8
FWH5	0.022	0.039	0.005	228	26.2
FWH6	0.022	0.039	0.005	236	25.7
FWH7	0.022	0.039	0.005	248	26.1
Preheater	0.020	0.035	0.004	5948	3.18
Boiler	0.048	0.070	0.006	289	18
Superheater	0.024	0.043	0.006	1482	9.1
Reheater	0.034	0.049	0.004	1664	7.15
Condenser	0.024	0.048	0.0006	16680	9

II.3. Main assumptions

The energy modeling of a steam power cycle can be performed by combining the analysis of each subsystem. Therefore, the introduction of certain assumptions is of major importance for the implementation of mathematical models. The following assumptions are made in our study:

- All pumps and turbines within the system are assumed to operate adiabatically and under steady-state conditions.
- The properties of the working fluid of the cycle (water-vapor), such as density, specific heat, enthalpy, and entropy, vary depending on pressure and temperature.
- The feedwater at the exits of the condenser and the economizer is in a saturated liquid state, while it exits the evaporator as saturated vapor.
- Combustion in the boiler occurs at constant pressure, so the pressure of the burned gases equals atmospheric pressure throughout their circuit within the boiler.
- The gases burned in the boiler pass successively through the superheater, the evaporator, and the economizer, as well as the reheater, exchanging heat with each of these heat exchangers. Part of this heat is then recovered at the boiler combustion chamber, while the rest passes through an air preheater and exits the boiler through the chimney.
- Correction factors for the overall heat transfer coefficients (U) are not included in the calculations due to a lack of data.

II.4. Mathematical modelling

In this subsection of the paper, the mathematical model to simulate the performances of the installation is presented. Initially, an energy model based on the first law of thermodynamics is provided for the investigated plant's full-load operation. Subsequently, correlations utilized for estimating various parameters of the primary plant components, including heat exchangers, turbomachinery, and the generator, under varying loads (off-design), are summarized and categorized.

The energy analysis of a system uses the conservation of mass and energy (the first law of thermodynamics). To conduct this analysis, through which energy, mass, and work cross the boundaries of the system's components, such conservation laws must be applied. The equations of conservation (mass and energy balance) have been applied to the system's components:

$$\sum \dot{m}_{\text{in}} = \sum \dot{m}_{\text{out}} \quad (1)$$

$$\dot{Q} + \sum \dot{m}_{\text{in}}\, h_{\text{in}} = \dot{W} + \sum \dot{m}_{\text{out}}\, h_{\text{out}} \quad (2)$$

Moving to the mathematical equations for different heat exchangers in the installation, including those without phase change, such as superheater, reheater, and preheater, and those with phase change, like the boiler and condenser. First, superheaters, reheaters, and preheaters are modeled as counterflow heat exchangers using LMTD method. The following equations are used:

- Pressure drops across the exchangers:

$$P_{\text{out}} = P_{\text{in}} - \Delta P \quad (3)$$

- Log Mean Temperature Difference (LMTD):

$$\Delta T_{LM} = \frac{\Delta T_{\max} - \Delta T_{\min}}{ln\dfrac{\Delta T_{max}}{\Delta T_{\min}}} \quad (4)$$

$$\Delta T_{\max} = T_{h\,\text{in}} - T_{c\,\text{in}} \quad (5)$$

$$\Delta T_{\min} = T_{h\,\text{out}} - T_{c\,\text{out}} \quad (6)$$

- Heat transfer rate:

$$\dot{Q} = \dot{m}_h\, Cp_h\, \Delta T_h = -\dot{m}_c . Cp_c\, \Delta T_c = U.A.\Delta T_{LM} \quad (7)$$

- Where the global heat transfer coefficient is calculated by:

$$\frac{1}{U} = \frac{1}{h_{\text{conv},h}} + fe_o + \frac{e}{k} + fe_i + \frac{1}{h_{\text{conv},c}} \quad (8)$$

- Where h_{conv} is determined by:

$$h_{conv} = \frac{k\,.\,Nu}{Dh} \quad (9)$$

- And Nusselt number Nu is estimated for the two sides (hot and cold) by [10]:

$$Nu_h = 0{,}33.Re_h^{\ 0.6}.Pr_h^{\ 0.33} \quad (10)$$

$$Nu_c = 0{,}023.Re_c^{\ 0.8}.Pr_c^{\ 0.4} \quad (11)$$

- With Reynolds number Re:

$$Re = \frac{V.Dh}{\nu} \quad (12)$$

- Which is calculated from fluid velocity:

$$V = \frac{\dot{m}}{\rho.A} \quad (13)$$

- There are two surface area:

$$A_h = \frac{N_{\text{tube}}\,.\pi\,.D_i^{\ 2}}{4} \quad (14)$$

$$A_c = \frac{N_{\text{tube}}\,.\pi\,.D_o^{\ 2}}{4} - A_c \quad (15)$$

Secondly, for heat exchangers involving phase change, such as the boiler and condenser, the same procedures are followed to calculate pressure drops and global heat transfer coefficients. However, a key difference lies in the calculation of the convective heat transfer coefficients. In these cases, the side of the exchanger where the phase change occurs must be considered.

For the boiler, the convective heat transfer coefficient for steam with phase change can be determined using the Dittus-Boelter equation, as shown in the following equation [11]:

$$h_{conv} = 0.023\left(\frac{G \times D_i}{\mu}\right)^{0.8} \times \left(\frac{C_P \times \mu}{k_c}\right)^{0.4} \quad (16)$$

While for the condenser, it is determined by equation (9), with a difference in the calculation of Nu [12]:

$$Nu_c = \left(\frac{3{,}022 D_t}{k_c}\right) \times \frac{\left(k_c^{\ 3}.h.10^3.\rho_c.g\right)^{0.25}}{(e.\upsilon_c.\Delta T)} \quad (17)$$

Another key parameter in the modeling of this plant is the mass flow rate of natural gas used to fuel it. This mass flow rate is estimated by:

$$\dot{Q}_{\text{Boiler}} = \dot{m}_{\text{gas}} \times LHV \times \eta_{\text{comb}} \quad (18)$$

While for the feedwater heaters, NTU method was used to calculate the temperature of each fluid at the outlet of the heat exchangers. These exchangers are integrated to heat the feedwater using steam extracted from various points in the two turbines, to improve the efficiency of the whole installation. The primary goal is to raise the temperature of the feedwater to the saturation temperature corresponding to the pressure at the preheater inlet. This process involves the partial condensation of the steam, mixing with the feedwater, and subsequently recovering this mixture in the condenser. First, the efficiency of a heat exchanger ε is defined as the ratio of the actual heat flow exchanged $\dot{Q}_{\text{act}}$ to the maximum theoretical heat flow $\dot{Q}_{\max}$ that could be exchanged if the heat exchanger was perfect:

$$\varepsilon = \frac{\dot{Q}_{\text{act}}}{\dot{Q}_{\max}} \tag{19}$$

The $\dot{Q}_{\max}$ is calculated using the following equation:

$$\dot{Q}_{\max} = C_{\min} \times (T_{h\ \text{in}} - T_{c\ \text{in}}) \tag{20}$$

On the other hand, this efficiency is determined from the following equation:

$$\varepsilon = 1 - e^{-NTU} \tag{21}$$

Where NTU is given by:

$$NUT = \frac{U.A}{C_{\min}} \tag{22}$$

Now, the $\dot{Q}_{\text{act}}$ is calculated using eq (19), and the outlet temperature is presented by the following equation:

$$T_{out} = T_{in} - \frac{\dot{Q}_{\text{act}}}{\dot{m} \times Cp} \tag{23}$$

At the end of the heat exchanger modeling, the off-design cycle model uses specified conductance values to reflect the performance of the exchangers. These values can be defined using the Dittus-Boelter correlation, as presented by [13]:

$$UA = UA_{\text{design}} \left(\frac{\dot{m}}{\dot{m}_{\text{design}}}\right)^{0.8} \tag{24}$$

Regarding the pump, its consumed power is determined by:

$$\dot{W}_{\text{Pum}} = \frac{\dot{m}(h_{\text{out}} - h_{\text{in}})}{\eta_{Pum}} \tag{25}$$

While its efficiency at off-design conditions is determined based on its efficiency at the design point, which is already reported in Table 1 [14]:

$$\eta_{\text{Pum}} = (2 \times Load - Load^2) \times \eta_{\text{Pum design}} \tag{26}$$

Where *Load* define the load at which the plant is working:

$$Load = \frac{\dot{m}}{\dot{m}_{\text{design}}} \tag{27}$$

While for the two turbines, the high-pressure and low-pressure turbines, their generated works are calculated by:

$$\dot{W}_{\text{Turb}} = \dot{m}(h_{\text{in}} - h_{\text{out}}) \times \eta_{\text{Turb}} \tag{28}$$

Where their efficiencies at off-design conditions is determined based on their efficiencies at the design point presented in Table 1 [13]:

$$\eta_{\text{Turb}} = (1 - Reduction).\eta_{\text{Turb design}} \tag{29}$$

Where:

$$Reduction = 0.191 - 0.409 \times Load + 0.218 \times Load^2 \tag{30}$$

The last piece of equipment to be modeled at off-design conditions is the regenerator. Its efficiency is determined based on its efficiency at the design point using the following equation [13]:

$$\eta_{\text{gen}} = 0.9 + 0.258 \times Load - 0.3 \times Load^2 + 0.12 \times Load^3 \tag{31}$$

III. Results and discussion

III.1. Validation

The performances of our developed model, which simulates the installation operating at full capacity of 210 MW, are evaluated, and its results are compared with the actual data provided by the manufacturer of the Achouat power plant as summarized in Table 3.

After calculating the errors at several key points between the modeled power plant and the one presented by the manufacturer, varying degrees of discrepancies between the estimated values from the manufacturer and those obtained in our work were reported. The temperature error ranges from a minimum of 0% at the TMP inlet to a maximum of 29.69% at the deaerator outlet, with a notable 12.05% at the boiler inlet, which is considered high. The pressure error ranges from a minimum of 0% at most key points to a maximum of 0.39% at the boiler inlet. The steam mass flow rate error ranges from a minimum of 0.023% at the boiler inlet to a maximum of 13.67% at the condenser outlet. These errors may also occur due to the approach used to estimate the thermo-physical properties of water/steam and air in Refprop. Additionally, the lack of certain values from the manufacturer, such as the isentropic and mechanical efficiencies of some components and the correction factors for the heat exchangers, contributes to these discrepancies. Nevertheless, the model shows an acceptable correlation with the manufacturer's data, confirming the viability of our model

Table 3. Statistical comparison between our model results and the manufacturer's data at full operating load of the power plant

Point	Parameter	Actual data	Our model	Error e(%)
Boiler inlet	Temperature (°C)	244	217.76	12.05
	Pressure (bar)	178.5	179.2	0.39
	Mass flow rate (kg/s)	171.5	171.46	0.023
HPT outlet	Temperature (°C)	329	316.29	4.02
	Pressure (bar)	26.7	26.7	0
	Mass flow rate (kg/s)	160.27	165.7	3.28
LPT inlet	Temperature (°C)	540	540	0
	Pressure (bar)	23.48	23.48	0
	Mass flow rate (kg/s)	148.22	151.57	2.21
Condenser outlet	Temperature (°C)	33.5	33.81	0.91
	Pressure (bar)	0.0527	0.0527	0
	Mass flow rate (kg/s)	125.25	145.09	13.67
Deaerator outlet	Temperature (°C)	169.2	130.46	29.69
	Pressure (bar)	6.9	6.9	0
	Mass flow rate (kg/s)	171.5	171.46	0.023

Moving to the validation of our model to predict the performance of the installation operating at varying loads. This validation involves calculating the relative error between the model's parameter values and the actual data at different operating loads. Tables 4 and 5 provide a statistical comparison of the calculated values for overall efficiency and fuel consumption (natural gas) with the real data for the power plant at different loads ranging from 20-100%.

Table 4. Statistical comparison between the calculated values of overall efficiency and actual ones at different loads

Power (MW)	$\eta_{glob.actual}$ (%)	$\eta_{glob.model}$ (%)	*e (%)*
210	37.54	37.38	0.43
157.5	37.08	37.24	0.43
105	35.42	36.09	1.86
63	33.25	33.83	1.71

Table 5. Statistical comparison between the calculated values of consumed natural gas to feed the plant and actual ones at different loads

Power (MW)	$\dot{m}_{gaz,actual}$ *(kg/s)*	$\dot{m}_{gaz,model}$ *(kg/s)*	*e (%)*
210	19.5	19.42	0.41
157.5	14.67	14.61	0.41
105	10.25	10.06	1.89
63	6.94	6.44	7.76

After calculating the relative errors of several parameters at different operating loads between our modeled power plant and the actual data, we observed varying discrepancies. For overall efficiency, the relative error ranges from a minimum of 0.43% at full load to a maximum of 1.86% at 20% load. Similar trend was observed for the fuel mass flow rates, with relative errors ranging from 0.41% to 7.76%. These discrepancies may result from the approach used to estimate the thermo-physical properties of water/steam and air in Refprop, as well as the correlations used in our model for calculating the efficiencies of turbines, pumps, and generators at partial loads, and the overall heat transfer coefficients of the various heat exchangers in the power plant. Despite these errors, this developed model shows a good correlation with the actual operational data, confirming its viability.

Before analyzing the plant's behavior under varying loads, it's essential to size the heat exchangers at full-load operation. Table 6 summarizes this sizing, detailing the exchanged heat fluxes, surface areas, overall heat transfer coefficients, and efficiencies at full load.

The analysis of feed water heaters (FW1-FW7) and other heat exchangers in the steam thermal power plant highlights differences in performance based on U, A, $\dot{Q}$, and ε. FW6 has the highest U value (957.87 W/m².K) and a significant heat flux (35.288 MW), indicating superior design. FW1, despite a high U value (743.69 W/m².K), has the lowest effectiveness (27.95%) due to its small surface area (290 m²). FW5 has the highest effectiveness (89.98%) with a U value of 686.81 W/m².K and a large surface area (600 m²).

Comparing the other heat exchangers, The condenser, with a U value of 1483 W/m².K and a surface area of 11359 m², achieves the highest heat flux (269 MW) and effectiveness (86.33%), underscoring its role in condensing exhaust steam for reuse. The superheater, with U=677 W/m².K and a surface area of 2586 m², achieves a heat flux of 160.317 MW and effectiveness of 78.55%, crucial for raising steam temperature. The economizer, with U=382.71 W/m².K and a surface area of 2080 m², achieves a heat flux of 105 MW and effectiveness of 58%, heating feedwater using residual heat. The reheater, with U=323 W/m².K and a surface area of 2761 m², achieves a heat flux of 77.084 MW and effectiveness of 83.43%, essential for reheating steam.

Table 6. Sizing and characteristics of heat exchangers of the studied plant at full load operation.

Heat Exchanger	*U* (W /m².K)	*A* (m²)	$\dot{Q}$ (MW)	ε (%)
FW1	743.69	290	4.765	27.95
FW2	618.38	350	15.940	58.51
FW3	395.06	350	12.648	71.65
FW4	326.97	350	15.360	70.63
FW5	686.81	600	12.901	89.98
FW6	957.87	610	35.288	83.71
FW7	735.30	650	15.576	87.02
Economizer	382.71	2080	105.708	58.62
Evaporator	185.45	980	134.461	49.01
Superheater	677.34	2585.96	160.317	78.55
Reheater	323.57	2761	77.084	83.43
Condenser	1482.74	11359.04	268.919	86.33

To understand the plant's performance under varying conditions, the relationship between key metrics and operational load is examined. Figure 2 shows a direct relationship between the operational plant's load and the mass flow rates of both steam and burned natural gas. As the operational load increases, the steam mass flow rate is observed to increase linearly, reflecting the need for more steam to drive the turbines and generate higher power output. This linear relationship suggests a consistent steam-to-power ratio across different load levels. Similarly, the burned natural gas mass flow rate is found to increase linearly with the operational load, indicating a direct correlation between fuel consumption and power output. This linear trend signifies efficient and consistent combustion across varying loads, with the slope representing the plant's specific fuel consumption rate. For example, at 20% load, the steam mass flow rate is approximately 42 kg/s, while at 100% load, it increases to almost 172 kg/s (Figure 2). Likewise, the fuel consumption rate is observed to rise from around 4.75 kg/s at 20% load to almost 20 kg/s at 100% load.

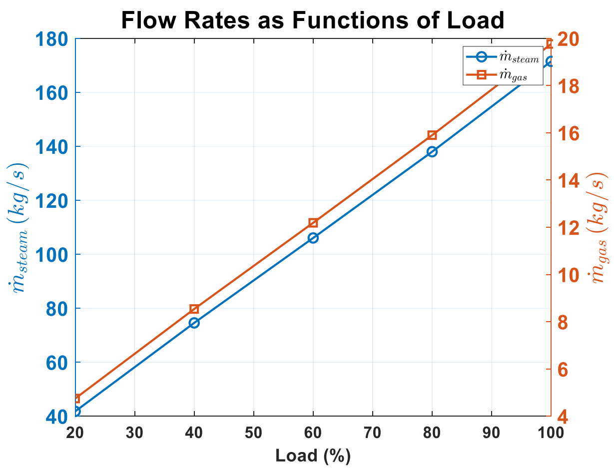

Figure 2. Variation of steam and burned gas mass flow rates with operational load

On the other hand, Figure 3 reveals the impact of operational load on different efficiencies within the steam thermal power plant. The plant's overall efficiency, along with turbine, pump, and generator efficiencies, show a decreasing trend as the operational load decreases. This phenomenon is attributed to the reduced efficiency of heat exchangers at partial loads, sub-optimal performance of turbines at lower loads, and the increased ratio of fixed losses (like pumping and friction) to the total energy output. For instance, at 100% load, the overall plant efficiency reaches approximately 37.4%, while it drops to around 33.2% at 20% load (Figure 3). Similarly, the efficiencies of the turbine, pump, and generator decrease from 88%, 87%, and 98%, respectively, at full load to about 77%, 32%, and 94%, respectively, at 20% load, highlighting the impact of partial load operation on these efficiencies.

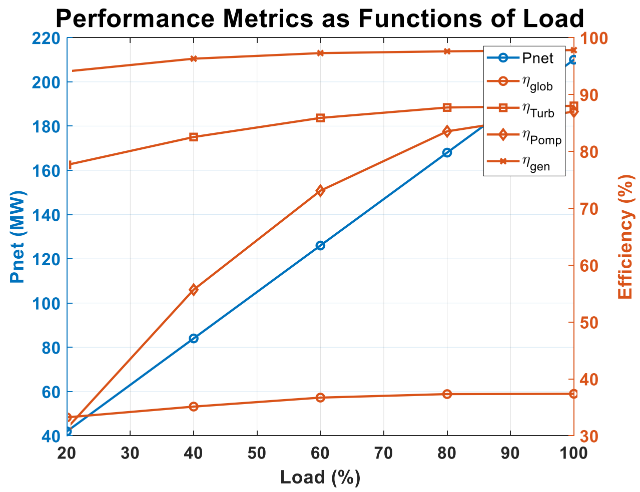

Figure 3. Variation of generated power and efficiencies with operational load

Figure 2 shows how steam mass flow rates and burned natural gas are affected by load. Figure 3 illustrates the impact of load fluctuations on the efficiencies of the turbine, pump, generator, and overall plant.

IV. Conclusion

This study investigates the performance of a steam thermal power plant operating at different load levels. The plant under consideration was modeled and simulated using a code developed under Matlab environment, with Refprop employed to calculate the thermophysical properties of the working fluids. These codes were developed in-house to size and calculate the performances of the plant's components, enabling an energy analysis of the plant at both full and partial loads. Initially, the model was validated against real data from the Achouat power plant at full and different partial loads to verify its accuracy. The results obtained from the developed model and the energy analysis of different parameters at full load and varying load levels are presented in tables and graphs. These findings can be summarized as follows:

- The plant operates most efficiently at full load, achieving a maximum overall efficiency of 37.4%. Efficiency significantly decreases with decreasing load, reaching 33.24% at 20% load. This underscores the importance of optimizing plant operation to maximize efficiency and minimize fuel consumption.
- The efficiencies of individual components, including the turbine, pump, and generator, also decrease with reduced load. At full load, the turbine efficiency is

88%, the pump efficiency is 87%, and the generator efficiency is 98%. However, at 20% load, these efficiencies drop to 77%, 32%, and 94%, respectively.

- Both steam and burned natural gas mass flow rates exhibit a linear relationship with operational load. For example, at 20% load, the steam flow rate is approximately 42 kg/s, while at 100% load, it increases to almost 172 kg/s. Similarly, the fuel consumption rate rises from around 4.75 kg/s at 20% load to almost 20 kg/s at 100% load.

Nomenclature

Latin Symbols
A: Surface area (m²)
Cp: Specific heat capacity (J/kg.K)
D_h: Hydraulic diameter (m)
D_i: Inner diameter (m)
D_0: Outer diameter (m)
e: Tube thickness (m)
h: Enthalpy (J/kg)
hconv: Convective heat transfer coefficient (W/m².K)
k: Thermal conductivity (W/m.K)
L_{tube}: Length of the tube (m)
LHV: Lower heating value of the fuel (kJ/kg)
$\dot{m}$: Mass flow rate (kg/s)
N_{tube}: Number of tubes (-)
Nu: Nusselt number (-)
Pr : Prandtl number (-)
$\dot{Q}$: Heat flux (MW)
P: Pressure (bar)
T: Temperature (K)
Re: Reynolds number
U: Overall heat transfer coefficient (W/m².K)
V: Velocity (m/s)
$\dot{W}$: Work (kW)

Greek Symbols
ε: Efficiency of the heat exchanger (-)
ν: Dynamic viscosity of the fluid (Pa.s)
η: Efficiency (-)
ρ: Fluid density (kg/m^3).
ΔP: Pressure drop (bar)
ΔT: Temperature difference (K)
ΔT_{LM}: Log Mean Temperature Difference (K)

Indices
c: Cold side
comb: Combustion
gas: Gas
gen: Generator
glob: Global
h: Hot side
in: Inlet
out: Outlet
pum: Pump
turb: Turbine

Abbreviations:
CON: Condenser
DER: Deaerator
ECO: Economizer
EVAP: Evaporator
FWH: Feedwater Heater
GEN: Generator
HPT: High-Pressure Turbine
LPT: Low-Pressure Turbine
PUM: Pump
REH: Reheater
SUP: Superheater

Declaration

- The authors declare that they have no known financial or non-financial competing interests in any material discussed in this paper.
- The authors declare that this article has not been published before and is not in the process of being published in any other journal.
- The authors confirmed that the paper was free of plagiarism

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