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Exergetic Analysis of Steam Turbine Power Plant Operated in Chemical Industry

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***Abstract*—**An Energetic and exergetic analysis is conducted on a Steam Turbine Power Plant of an existing Phosphoric Acid Factory. The heat recovery systems used in different parts of the plant are also considered in the analysis. Mass, thermal and exergy balances are established on the main compounds of the factory. A numerical code is established using EES software to perform the calculations required for the thermal and exergy plant analysis. The effects of the key operating parameters such as steam pressure and temperature, mass flow rate as well as seawater temperature, on the cycle performances are investigated.

A maximum Exergy Loss Rate of about 72% is obtained for the melters, followed by the condensers, heat exchangers and the pumps. The heat exchangers used in the phosphoric acid unit present exergetic efficiencies around 33% while 60% to 72% are obtained for steam turbines and blower. For the explored ranges of HP steam temperature and pressure, the exergy efficiencies of steam turbine generators STGI and STGII increase of about 2.5% and 5.4% respectively. In the same way optimum HP steam flow rate values, leading to the maximum exergy efficiencies are defined.

***Keywords*—**Steam turbine generator, energy efficiency, exergy efficiency, phosphoric acid plant.

I. Introduction

THE Tunisian production of Phosphoric Acid is among the five important ones in the world. Indeed the phosphate constitutes an important factor for the country economy balance. The annual production of Phosphoric Acid is about 500 000 tonnes. Phosphate and its derivatives such as phosphoric acid, DAP, TSP, DCP, etc. are exported to about fifty countries in the five continents. Although the economic importance of the phosphate industry the energy balances of the production cycle remand very substantial. To overcome this problem, the Tunisian Chemical Group (TCG) established programs in the purpose to improve the quality of production and increase the performance of the different plants. Among these programs, studies are conducted on the optimization of thermal power plants performances in order to provide the energy required for the phosphoric acid production.

Several investigations were conducted on energy and exergy optimization of chemical industrial factories.

Ray et al. [1] developed an exergy analysis for proper operating and maintenance decisions in a 500 MW steam power plant. The study is conducted for considering design and off-design conditions for various values of superheat and reheats sprays. Obtain results constitute guide procedures for exergy, economy and maintenance scheduling similar power plants.

An exergy analysis for thermal power plant is conducted by [2] using Aspen plus software. The effects of main operating parameters such as combustion exes air coefficient, steam temperature and pressure and combustion temperature on the exergy efficiency are analyzed. The obtained results reveal that the boiler engenders the maximum irreversibility rates followed by the turbine. Furthermore, the authors suggest that the increase of the combustion temperature as well as the steam pressure and temperature leads to improvement of exergy efficiency.

Regulagadda et al. [3] performed a thermodynamic analysis of a subcritical boiler-turbine generator for a 32 MW coal-fired power plant. Energy and exergy equation governing the cycle are established. A parametric study is conducted for a range of operating variables. That permits to define the optimum parameters leading to the best plant performances. The boiler and turbine engender the maximum exergy destruction rates in the power plant. The identification of the exergy losses in the different cycles has permitted to develop an environmental impact and sustainability analysis.

A comparison between nine coal-fired power plants in Turkey is conducted by [4]. For each plant a calculation model is proposed and the mass, energy and exergy balances are established. That permits to determine the energy and exergy efficiency as well the exergy destruction rate of each component. A comparison is then accomplished between the considered power plants. The obtained results may constitute helpful tools for further investigations in the field of energetic and exergetic industrial power plant analysis.

Ghannadzadeh et al. [5] developed a general methodology for exergy balance in chemical and thermal process integrated in the ProSimPlus code. In order to fully automate exergy analysis, the whole exergy balance of the system is presented under the form of single software. The essential elements for exergy analysis are provided that can be applied for every process or utility system.

Molés et al. [6] conducted a thermodynamic analysis of a combined organic Rankine cycle and vapor compression cycle system using two different fluids with low Global Warming Potentials GWP for each cycle. System performances are determined for ranges of operating conditions variations. Results show that the combined cycle COP varied between 0.30 and 1.10 while the computed electrical COP is varied between 15 and 110. Furthermore, for vapor compression

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system the selection of working fluid does not affect significantly the thermal and electrical efficiencies. Whereas, for the ORC working fluid has an important effect, especially on electrical efficiency.

An energy and exergy optimization of a drying plant is carried out by [7] in the purpose to define the optimum energy and exergy efficiencies. Real measurements are accomplished on the drying plants taking into consideration the different accuracies of the experimental devices. Exergy analysis is conducted in order to define the optimum mass and thermal values leading to the maximum system efficiencies. As result of the optimization study, a significant improvement in the energy and exergy efficiencies of about 41% and 43% respectively are obtained.

A Thermodynamic and exergoeconomic analysis of thermal power plant is performed by [8]. Using EES software the inlet and outlet thermodynamic properties of each component are determined. That permits to define energy and exergy efficiencies. Obtained results show that the main amounts of exergy losses are located in the boiler, in the turbine, in the condenser, in the heater and in the pump groups. While the highest amount of exergy loss costs are observed in the boiler, followed by the turbines and the condenser. Authors suggest that exergy and economic analysis of the thermal plants in project stage may be helpful to undertake future investigations and can minimize significantly the energy consumption of thermal systems.

J. Taillon et al. [9] illustrate a graphical representation of energy efficiencies related to Combined Heat and Power (CHP) and condensing plants. Basing exclusively on the energy efficiencies does not permit a suitable comparison between the different energy system performances. Therefore, authors conducted an exergy analysis on 24 existing industrial factories and established two news graphs: the first one illustrates the electrical, thermal, and total exergy efficiencies of condensing and CHP power plants. The second graph splits the thermal and exergy efficiency in two components: thermal losses and useful heat output quality.

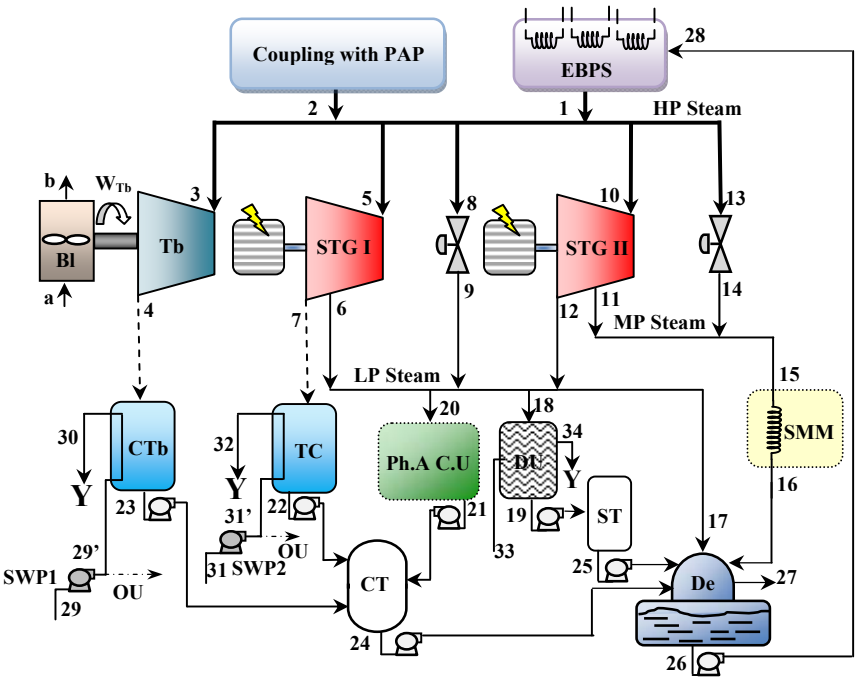


Fig. 1 Thermal Power Plant of Phosphoric Acid factory

O. K. Singh et al. [10] conducted a numerical study on Kalina cycle coupled with steam power plant stimulated by coal in order to valorize the exhaust gases at low temperature for electricity production. A model is developed in the purpose to optimize the cycle performances according to the main operating parameters. An optimum ammonia fraction value leading to the maximum cycle efficiency is obtained for a given turbine inlet pressure. Therefore, it has been demonstrated that the maximum cycle efficiency increases significantly with the turbine inlet pressure. For turbine inlet pressure of 4000 kPa and an ammonia fraction of 0.8, an improvement of 0.277% and 0.255% in the overall energy and exergy efficiency respectively.

In this work, an energetic and exergetic analysis is investigated on a Phosphoric Acid factory Power Plant. Mass, thermal and exergy balances are established for the different components of the power plant. A calculation code is performed using EES software. Real operating parameter ranges are considered to analyze the power plant performances.

II. System Description

The diagram of the Phosphoric Acid Thermal Power Plant is presented in Fig. 1. This plant is installed in the industrial area of the Tunisian Chemical Group (GCT) located in Gabes (South East - Tunisia). The main product of this factory is the

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Phosphoric Acid with about 1 500 t/day as daily production.

The thermal power plant of the indicated factory is mainly constituted by two steam turbine cycles STG I and STG II used to provide about 14 MW as total net electrical power required for the different units. The High Pressure steam (HP) mass flow rate, consumed by STG I and STG II is generated by an Evaporator Boiler Pre-superheater Superheater group (EBPS) at about 40 bars and 410°C.

TABLE I
OPERATING PARAMETER RANGES

Operating Parameters	Temperature Ranges (°C)	Pressure Ranges (bar)	Mass Flow Rate
HP Steam	386-395	39-41	179 t/h
MP Steam	190	12	18 t/h
LP Steam	165	5.7	135 t/h
Seawater	15-35	Input:1 Output:4	45-90 m^3/h for each pump
Seawater Salinity		0.039 kg/kg	
Humid Air RH		0.45-0.8	

The steam turbine cycle STGI is with extraction and condensation, while the second one STGII is with backpressure turbine. The Medium Pressure steam (MP) at 12 bars and 280 °C and Low Pressure steam (LP) at 6 bars and 230°C, used to supply the other different units, are obtained by the expansion of HP stream through appropriate devices (8-9) and (13-14). For the steam turbine STG I the input steam mass flow rate (point 5) is expanded to reach the extraction level at Low pressure (point 6). The remained steam flow rate is extended through the last stage of the turbine to reach the condensation pressure level at point 7. The condensation occurs in the seawater turbine condenser (TC). The tank (CT) is used for condensate storage. In the second steam turbine cycle STGII, HP steam expands from point 10 to reach the medium pressure MP at the extraction level (point 11). The remained stream is expanded to low pressure LP (point 12). The Turbo-blower Tb is used to provide compressed air for the sulfur combustion process in a sulfuric unit not presented on the diagram. The steam from the Turbo blower is condensed in the second seawater condenser CTb and then transferred to the storage tank CT. The two seawater pumps SWP1 and SWP2 are the most important among the other ones and require significant electrical energy supply. The streams of MP and LP steam are used in the units (SMM, Ph. A CU, DU, TC, CTb and De). The condensate issued from all the indicated units is submitted to the tank (CT) and then transferred to the deaerator (De). A water treatment is then performed in the purpose to supply the boilers. Appropriate sensors are used to measure the operating parameters such as: temperatures, pressures and mass flow rates. The variation ranges of these parameters are given on Table I.

III. Energy and Exergy Balances

In order to perform the energy and exergy analysis the following assumptions are considered [11]:

- All process are assumed as steady-state and steady flow
- The kinetic, potential and chemical exergy are neglected
- The dead state was considered as P_0=1.013 bar et T_0=293.15 K
- No chemical reaction is occurred in the different process
- For an open system and taking into account the indicated assumptions, the energetic and exergetic balances can be expressed as:

$$\dot{Q}-\dot{W}=\sum \dot{m}_{out}h_{out}-\sum \dot{m}_{in}h_{in} \quad (1)$$

$$\dot{X}_{heat}-\dot{W}=\sum \dot{m}_{out}\varepsilon_{out}-\sum \dot{m}_{in}\varepsilon_{in}+\dot{E}_D \quad (2)$$

where the exergy transferred by heat is given by:

$$\dot{X}_{heat}=\sum\left(1-\frac{T_0}{T}\right)\dot{Q} \quad (3)$$

and the specific exergy is showed as:

$$\varepsilon_i=\left(h_i-h_0\right)-T_0\left(s_i-s_0\right) \quad (4)$$

According to the stream identification indicated in Fig. 1, the energy and exergy balances for each component are given as:

- Steam Turbine Generator STG I

Energy balance:

$$\dot{W}_{STGI}=\dot{m}_5\left(h_5-h_6\right)+\left(\dot{m}_5-\dot{m}_6\right)\left(h_6-h_7\right) \quad (5)$$

Exergy balance:

$$\dot{E}_{D,STGI}=\dot{m}_5\left(\varepsilon_5-\varepsilon_6\right)+\left(\dot{m}_5-\dot{m}_6\right)\left(\varepsilon_6-\varepsilon_7\right)-\dot{W}_{STGI} \quad (6)$$

Energy Efficiency:

$$\eta_{e,STGI}=\frac{\dot{W}_{STGI}}{\dot{m}_5h_5-\dot{m}_6h_6-\dot{m}_7h_7} \quad (7)$$

Exergy Efficiency:

$$\eta_{ex,STGI}=\frac{\dot{W}_{STGI}}{\dot{m}_5\left(\varepsilon_5-\varepsilon_6\right)+\left(\dot{m}_5-\dot{m}_6\right)\left(\varepsilon_6-\varepsilon_7\right)} \quad (8)$$

- Steam Turbine Generator STG II

Energy balance:

$$\dot{W}_{STGII}=\dot{m}_{10}\left(h_{10}-h_{11}\right)+\left(\dot{m}_{10}-\dot{m}_{11}\right)\left(h_{11}-h_{12}\right) \quad (9)$$

Exergy balance:

$$\dot{E}_{D,STGII}=\dot{m}_{10}\left(\varepsilon_{10}-\varepsilon_{11}\right)+\left(\dot{m}_{10}-\dot{m}_{11}\right)\left(\varepsilon_{11}-\varepsilon_{12}\right)-\dot{W}_{STGII} \quad (10)$$

Energy Efficiency:

$$\eta_{e,STGII} = \frac{\dot{W}_{STGII}}{\dot{m}_{10}h_{10} - \dot{m}_{11}h_{11} - \dot{m}_{12}h_{12}} \quad (11)$$

Energy Efficiency:

$$\eta_{ex,STGII} = \frac{\dot{W}_{STGII}}{\dot{m}_{10}\left(\varepsilon_{10} - \varepsilon_{11}\right) + \left(\dot{m}_{10} - \dot{m}_{11}\right)\left(\varepsilon_{11} - \varepsilon_{12}\right)} \quad (12)$$

- Turbine Condenser (TC)

Energy balance:

$$0 = \dot{m}_7\left(h_7 - h_{22}\right) + \dot{m}_{31'}\left(h_{31'} - h_{32}\right) - Energy\ loss \quad (13)$$

Exergy balance:

$$\dot{E}_{D,TC} = \dot{m}_7\left(\varepsilon_7 - \varepsilon_{22}\right) + \dot{m}_{31'}\left(\varepsilon_{31'} - \varepsilon_{32}\right) \quad (14)$$

Energy Efficiency:

$$\eta_{e,TC} = \frac{\dot{m}_{31'}\left(h_{32} - h_{31'}\right)}{\dot{m}_7\left(h_7 - h_{22}\right)} \quad (15)$$

Exergy Efficiency:

$$\eta_{ex,TC} = \frac{\dot{m}_{31'}\left(\varepsilon_{32} - \varepsilon_{31'}\right)}{\dot{m}_7\left(\varepsilon_7 - \varepsilon_{22}\right)} \quad (16)$$

- Condenser of Turbo-Blower

Energy balance:

$$0 = \dot{m}_4\left(h_4 - h_{23}\right) + \dot{m}_{29'}\left(h_{29'} - h_{30}\right) - Energy\ loss \quad (17)$$

Exergy balance:

$$\dot{E}_{D,CTb} = \dot{m}_4\left(\varepsilon_4 - \varepsilon_{23}\right) + \dot{m}_{29'}\left(\varepsilon_{29'} - \varepsilon_{30}\right) \quad (18)$$

Energy Efficiency:

$$\eta_{e,CTb} = \frac{\dot{m}_{29'}\left(h_{30} - h_{29'}\right)}{\dot{m}_4\left(h_4 - h_{23}\right)} \quad (19)$$

Exergy Efficiency:

$$\eta_{ex,CTb} = \frac{\dot{m}_{29'}\left(\varepsilon_{30} - \varepsilon_{29'}\right)}{\dot{m}_4\left(\varepsilon_4 - \varepsilon_{23}\right)} \quad (20)$$

- Steam Turbine of Turbo-Blower

Energy balance

$$\dot{W}_{Tb} = \dot{m}_3\left(h_3 - h_4\right) \quad (21)$$

Exergy balance

$$\dot{E}_{D,Tb} = \dot{m}_3\left(\varepsilon_3 - \varepsilon_4\right) - \dot{W}_{Tb} \quad (22)$$

Energy Efficiency

$$\eta_{e,Tb} = \frac{\dot{W}_{Tb}}{\dot{m}_3\left(h_3 - h_4\right)} \quad (23)$$

Exergy Efficiency

$$\eta_{ex,Tb} = \frac{\dot{W}_{Tb}}{\dot{m}_3\left(\varepsilon_3 - \varepsilon_4\right)} \quad (24)$$

- Blower

Energy balance

$$\dot{W}_{Bl} = \dot{W}_{Tb}\eta_{Ge} = \dot{m}_{air}\left(h_b - h_a\right) \quad (25)$$

Exergy balance

$$\dot{E}_{D,Bl} = \dot{m}_{air}\left(\varepsilon_a - \varepsilon_b\right) - \dot{W}_{Bl} \quad (26)$$

Exergy Efficiency

$$\eta_{ex,Bl} = \frac{\dot{m}_{air}\left(\varepsilon_a - \varepsilon_b\right)}{\dot{W}_{Bl}} \quad (27)$$

- Deaerator

Energy balance

$$0 = \dot{m}_{16}h_{16} + \dot{m}_{17}h_{17} + \dot{m}_{24}h_{24} + \dot{m}_{25}h_{25} - \dot{m}_{26}h_{26} - \dot{m}_{27}h_{27} \quad (28)$$

Exergy balance

$$\dot{E}_{D,De} = \dot{m}_{16}\varepsilon_{16} + \dot{m}_{17}\varepsilon_{17} + \dot{m}_{24}\varepsilon_{24} + \dot{m}_{25}\varepsilon_{25} - \dot{m}_{26}\varepsilon_{26} - \dot{m}_{27}\varepsilon_{27} \quad (29)$$

Energy Efficiency

$$\eta_{e,De} = \frac{\dot{m}_{26}h_{26} + \dot{m}_{27}h_{27}}{\left(\dot{m}_{16}h_{16} + \dot{m}_{17}h_{17} + \dot{m}_{24}h_{24} + \dot{m}_{25}h_{25}\right)} \quad (30)$$

Exergy Efficiency

$$\eta_{ex,De} = \frac{\dot{m}_{26}h_{26} + \dot{m}_{27}h_{27}}{\left(\dot{m}_{16}h_{16} + \dot{m}_{17}h_{17} + \dot{m}_{24}h_{24} + \dot{m}_{25}h_{25}\right)} \quad (31)$$

IV. Results and Discussion

Analytic study is performed taking into consideration the real variations of main operating parameters such as mass flow rate, temperature, and pressure of HP steam. Furthermore seeing the sensibly variation of the seawater temperature for the local environmental conditions, the thermal power plant is analyzed for operating conditions during whole year. Calculations are carried out using a numerical code established according to EES software.

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Fig. 2 shows the Irreversibility rates of the different power plant components. The minimum irreversibility rates are obtained for the deaerators, condensers, and blower. The Steam turbine generators I and II present the maximum of irreversibility rates of about 4 MW and 2.5 MW respectively. This agrees with [8] where near 2.7 MW of irreversibility rate are obtained for High Pressure Turbine.

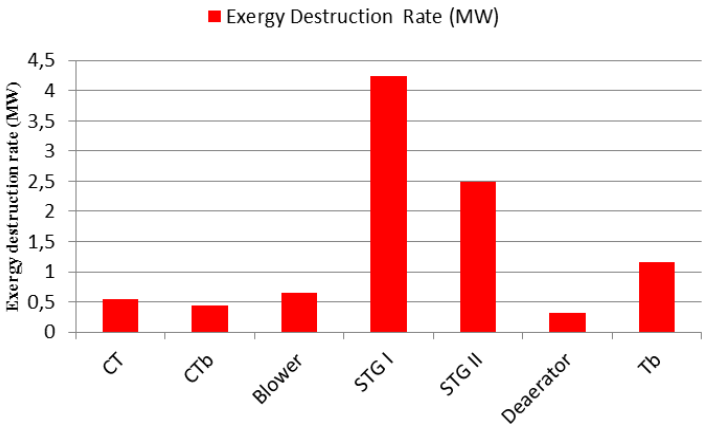


Fig. 2 Irreversibility rates of thermal power plant components

The energy and exergy efficiencies are showed in Fig. 3. The blower, turbines, and deaerators present the higher energy efficiencies. The minimum energy efficiencies are obtained for the condensers. The steam turbine generator STG II presents the maximum exergy efficiencies of about 71% followed by the Deaerator II 70%, Deaerator I 67%, the blower 64% and STG I 60%. The minimum values of energy efficiencies are obtained for the condensers 21% and 26%. Note that [12] obtained an exergetic efficiency of about 62% for the deaerator.

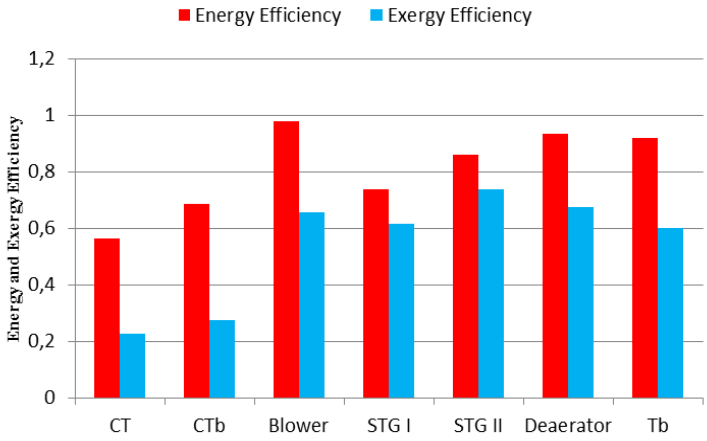


Fig. 3 Energetic and exergetic efficiency of the main components

The extraction mass flow rate in the steam turbine STGI (point 6) is defined according to operating conditions required for desired production rates. That affects the generated net power of the turbine as presented in Fig. 4. In fact, the generated power increases gradually with HP steam mass flow rate. For HP steam mass flow rate less than 35 t/h, the generated power is relatively low and is not significantly affected by the condensate rate. While for HP steam flow rate above 40 t/h the condensate rate affects sensibly the generated power. Indeed, in this range, increasing the condensate rate leads to the enhancement of the net generated power. A maximum net power of about 6 MW is obtained for 20 t/h of condensate mass flow rate.

For the back pressure steam turbine STGII, the variation of the generated power according to HP steam mass flow rate is presented in Fig. 5. The generated power increases linearly to achieve about 7 MW for a mass flow rate of about 82 t/h. That represents an increase of about 125 % for $\dot{m}_{HP}$ variation of about 95%. The total power generated by the two Steam turbines is widely sufficient for the power plant requirements. The generated power over than the plant supplies is transferred to the national electricity network.

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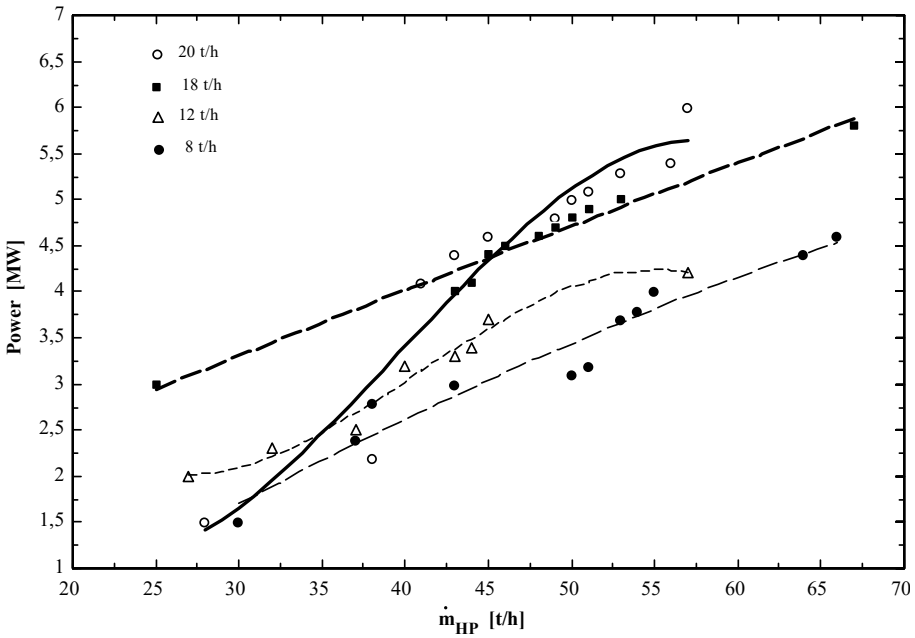


Fig. 4 Variation of STGI net power with HP steam flow rate

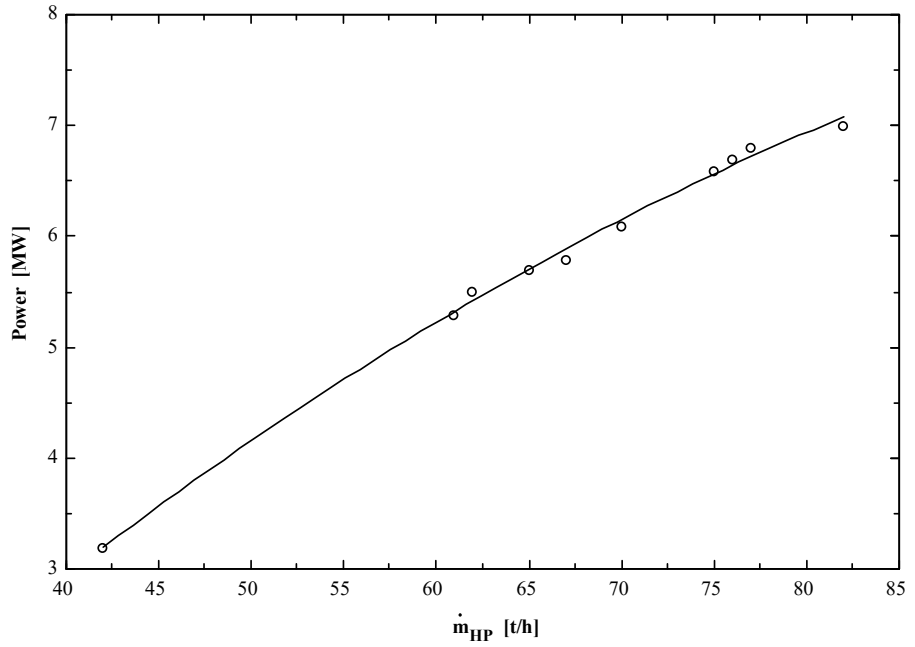


Fig. 5 Variation of STG II net power with HP steam flow rate

Fig. 6 depicts the variation of the exergetic efficiency of STGI according to HP steam flow rate for different values of condensate flow rate. The exergy efficiency increases with $\dot{m}_{HP}$ to reach maximum values of about 49%, 51%, 52% and 54% for condensation flow rates of 8, 12, 18 and 20 t/h respectively. The optimum $\dot{m}_{HP}$ values leading to the indicated maximum exergy efficiencies are respectively 60, 46, 52 and 54 t/h.

For the back pressure steam turbine STGII, the variation of the exergetic efficiency power according to HP steam mass flow rate is presented in Fig. 7. The exergetic efficiency increases sensibly with $\dot{m}_{HP}$ to reach a maximum value of about 75.5% for a mass flow rate of 73 t/h. That can be considered as an optimum value for the STGII supply.

Fig. 8 illustrates the variation of energy and exergy efficiencies of steam turbines STGI and STGII according to HP steam temperature. For the explored ranges of HP steam temperature, the energy efficiencies of steam turbine generators STGI and STGII increase of about 1.37 % and 8.8% respectively. While the exergy efficiencies increase of about 2.46 for STGI and 6.8% for STGII.

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Fig. 9 shows the variation of the condenser exergy efficiency according to seawater temperature. For an increase of 12°C leads the exergy efficiency increases of about 4 times for the turbo blower condenser CTb and 14 times for the turbine condenser CT. For T_{sw} above 25°C the exergy efficiency increases slightly with T_{sw} to reach maximum values of about 35 % and 45 % for the turbine condenser and the turbo-blower condenser respectively. The obtained exergy efficiency values are relatively low especially in cold seasons when the seawater temperature is less than 15°C. These results agree with [13] on energy and exergy analysis of a steam power plant. In fact, the authors obtained the same values of condenser exergy efficiency in similar operating conditions.

The variations of seawater pump exergetic efficiencies according to T_{sw} are presented in Fig. 10. For seawater temperature below 24°C the exergetic efficiencies of the two pumps are relatively low and very close to one another. For T_{sw} increase of about 16°C the pump exergy efficiencies increase of about 16 times to reach maximum values of 75% and 82%.

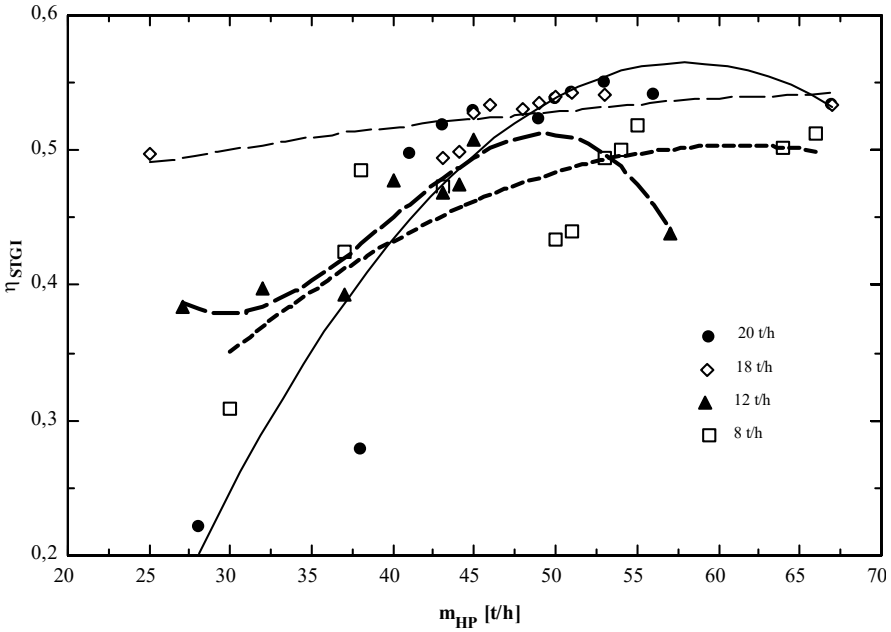


Fig. 6 Variation of exergetic efficiency of STG I with HP steam flow rate

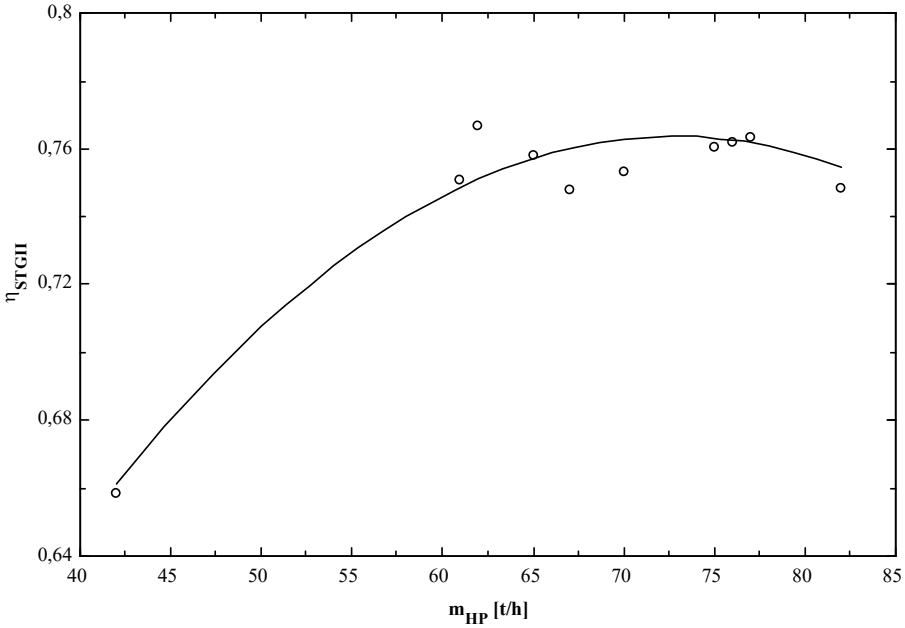


Fig. 7 Variation of exergetic efficiency of STG I with HP steam flow rate

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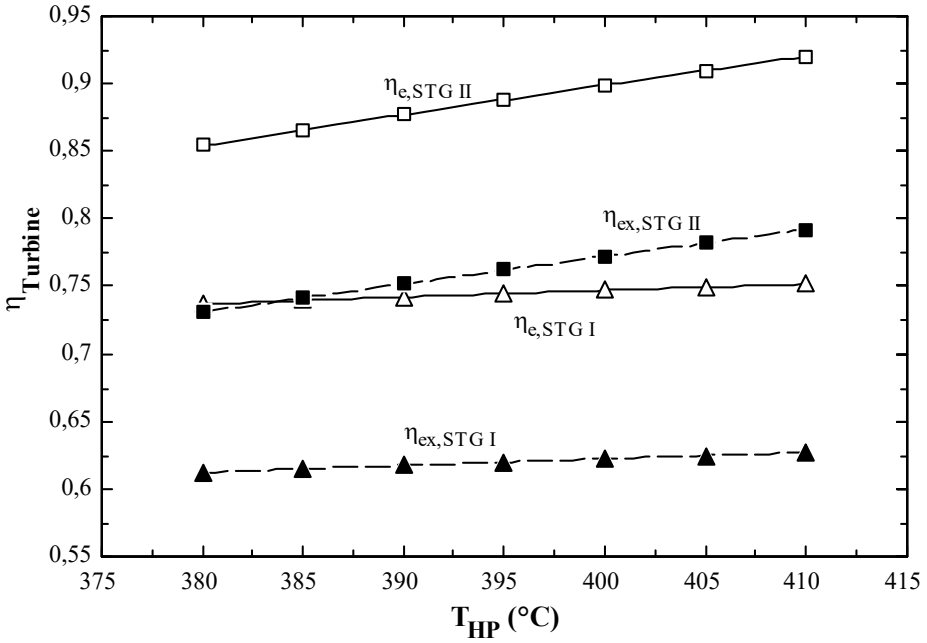


Fig. 8 Variation of energetic and exergetic efficiency of STG I and STG II with HP steam flow temperature

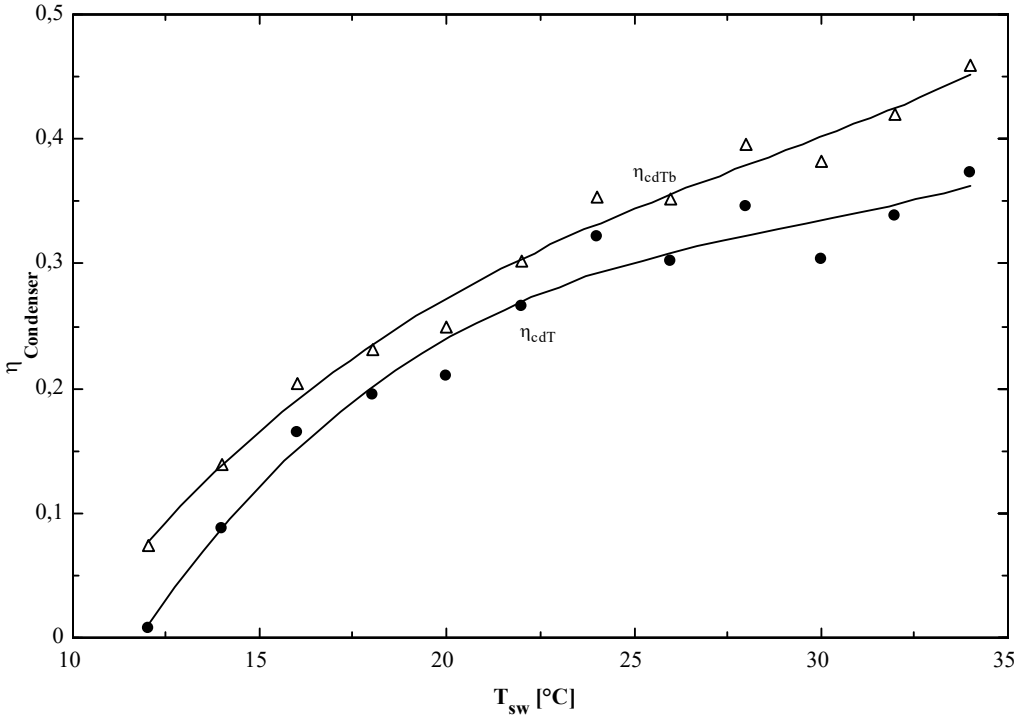


Fig. 9 Variation of exergy efficiency of condensers according to seawater temperature

V. Conclusion

A thermal and Exergetic Analysis on a Steam Turbine Power Plant used of a Phosphoric Acid Factory is performed. Heat exchangers and heat recovery systems are used in the purpose to enhance the power plant performances. Energy and exergy balances are established on the main compounds of the factory. Real variation ranges of the operating parameters are considered to analyze their effects on the cycle performances. The following concluding remarks are formulated in order to point out the main obtained results.

The Steam turbine generators STGI and SWTGII present the maximum irreversibility rates followed by the turbo-blower turbine, the condenser, and the deaerator. For variation ranges of HP steam temperature and pressure of about 30 °C and 4.2 bars, the exergy efficiencies of steam turbine generators STGI and STGII increase of about 2.5% and 6 % respectively.

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For the condensation mass flow rates of 8, 12, 18 and 20 t/h obtained after LP extraction in the, the optimum HP steam flow rate values leading to the maximum exergy efficiencies are respectively, 55, 46, 52 and 54 t/h. For STGII a maximum exergetic efficiency of about 75.5 % is obtained for HP steam flow rate of 73 t/h.

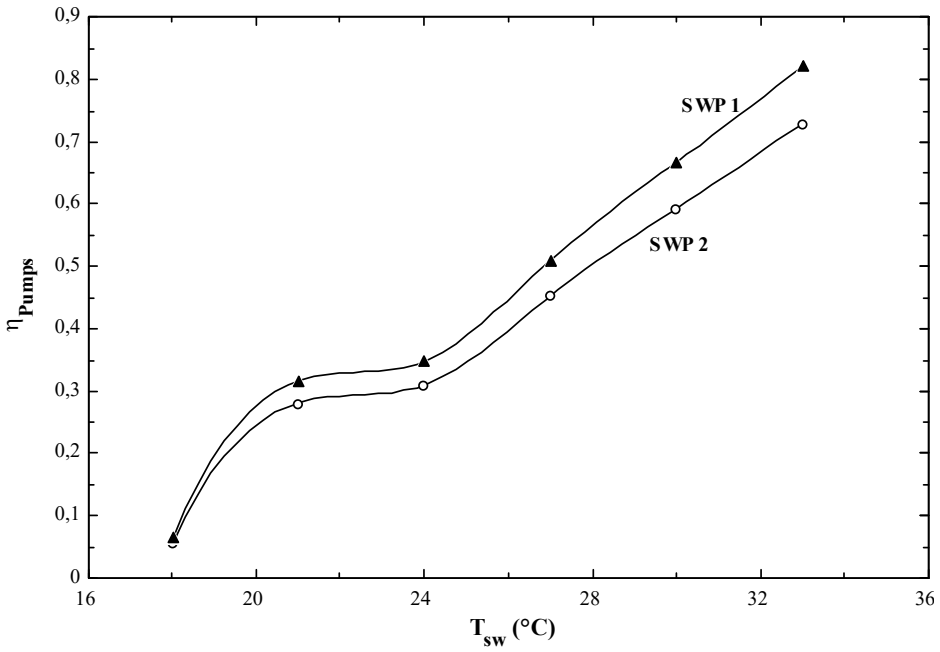


Fig. 10 Variations of seawater pump exergetic efficiencies according to T_{sw}

The seawater temperature affects significantly the exergy efficiency of the condensers. That should by taking into consideration for the operating condition in cold seasons.

The obtained results constitute helpful tools to analyze the real performances of industrial plants and permit to better undertake the future perfections that can be carried out on the different streams in order to improve the efficiency and reduce the energetic losses.

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