Univerzitet u Beogradu
Poljoprivredni fakultet
Institut za poljoprivrednu tehniku
Naučni časopis
POLJOPRIVREDNA TEHNIKA
Godina L
Broj 1., 2025.
Strane: 35 – 45

University of Belgrade
Faculty of Agriculture
Institute of Agricultural Engineering
Scientific Journal
AGRICULTURAL ENGINEERING
Year L
No.1, 2025.
pp. 35 – 45

Original Scientific paper
Originalni naučni rad
DOI: 10.5937/POLJTEH2501035I

METHODOLOGY FOR THE THERMODYNAMIC OPTIMIZATION OF STEAM TURBINE PUMP PERFORMANCE

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Abstract: Pumps are an integral part of steam-driven turbines since they are necessary to provide continuous kinetic energy to the working fluid and operate at considerably high pressures to meet boiler design conditions. Thermodynamic modeling and optimization from an energy viewpoint will establish the link between optimum operating conditions concerning boiler design. This was achieved through thermodynamic modeling and optimization of pump performance on a steam-driven turbine rated at 50 MW. Using developed energy models, a system performance simulation was made by systematically accounting for the variation in pump efficiency and overall turbine output. For the considered pumps, overall cycle thermal efficiency is slightly affected by the pump performance efficiency even at higher pressures. Increasing pump efficiencies has been shown to improve overall cycle efficiency comparatively.

The system recorded 80 % pump efficiency, 640 K temperature at the WHRB, 35 bar pressure at the inlet to the turbine, and a steam mass flow rate of 50.6 kg/s, respectively. The system's thermal efficiency was 38.39 % finally. Furthermore, in cases where a trade-off between cycle performance, efficiency, and cost of operation is required, these results will be helpful in the multi-objective optimization of similar systems in cost and thermal efficiency.

Keywords*:* *Steam turbines, modelling, energy equations, pump performance, efficiency, optimization.*

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INTRODUCTION

Energy availability is a critical indicator for industrialization, economic growth, and sustainable development. Energy demand is determined by the human aspiration to sustain and improve our families and communities. Also, energy demand is expected to widen annually due to rising population growth worldwide. Efforts to match this increasing population and economic development with adequate energy supply in many countries have severe implications for the environment because energy generation processes emit pollutants, many of which are harmful to the ecosystem [1,4].

As a result, energy conversion technologies have advanced significantly over the past few decades to keep up with the growing energy demands of both industrial and residential sectors. Burning fossil fuels, which comprise a significant portion of generation methods, has frequently been the most prevalent energy conversion technique [2]. One of these methods is the utilization of steam turbines as bottoming cycles in gas turbines. Steam turbines have many applications in various industrial sectors [3,4]. In areas where local waste heat is abundant, energy generation from steam turbines is a clear objective for powering the boilers, which drives steam to the turbine for expansion and conversion to mechanical work. Because boilers are the primary energy source for steam turbines and operate at specific high-pressure values, their link with the pumps must match to cater to the pressure variation needed for operation. In recent times waste heat from gas turbine plants has been utilised in waste heat recovery generators and waste heat recovery boilers, which substitutes traditional boilers for steam generation. In these applications, the required operating pressures for the gadgets are provided by pumps that handle condensed steam at the inlet to either the WHRB or HRSG. [5,6] presented an analysis of the cause of fracture in a steam turbine blade root using the finite-element analysis of the blade group concerning their natural frequencies. They revealed that an inappropriate mismatch between the contact areas of the blade root and rotor fastening tree was responsible for the blade root fracture. [7,8] analysed combined triple (Brayton/Rankine/Rankine) and (gas/steam/ammonia) power cycles. The triple process was investigated and optimized concerning essential system parameters, such as the gas topping cycle pressure ratio, gas turbine inlet temperature, HRSG pinch point, gas/steam approach temperature difference, rate of steam injection into the combustion chamber, and the effectiveness of the heat exchangers. They obtained the configuration that will achieve a thermal efficiency of 60 % when reasonably practical constraints for system parameters are used. [10,11] modeled and optimized the start-up of a combined cycle plant with steam as the bottoming cycle.

They decomposed the transients encountered in several phases that were optimized separately. A simple model was used for each phase to minimize its duration under constraints. Several state and control input constraints were given to imposing a trajectory that respects the stress limitation and ensure the damping of the responses. [12] performed an energy and exergy analysis of the Al-Hussein power plant in Jordan by analyzing the system components separately and to identify and quantify the sites having most significant energy and exergy losses. The boiler was the primary source of irreversibilities in the power plant.

RESEARCH METHODOLOGY

Plant description

The schematic diagram of the steam plant, where the performance of three pumps on its overall efficiency is considered, is shown in Fig. 1. The system comprises a waste heat recovery boiler (WHRB), a steam turbine for steam expansion, two feed water heaters positioned to obtain bled steam at the expansion line, a condenser, and three pumps for circulating the working fluid. The heat input to the system can reasonably be from a boiler. However, for this system, provision is made for waste heat from a possible topping cycle for powering the WHRB. At high pressure generated by pump 1, superheated steam is generated in the WHRB to the turbine, where expansion occurs. Due to the provision of two feedwater heaters, steam is bled at points 2 and 3 for preheating the condensate from the remaining two pumps. The last phase of the expansion process terminates at condenser pressure. A condenser is provided to exchange heat with a cooling fluid (water), condensing the exhausted steam to saturated liquid for handling by pump 3. The two feedwater heaters then add the heat to the incoming water, bringing it to near-saturation end route to the WHRB for the cycle.

Preliminary thermodynamic energy modeling of each component will reveal the interaction of the operating parameters at the state point level. Details of the first principles modeling of energy balances are found elsewhere [17,18] and are presented in the following section. From the schematics of Figure 3.1, the pumps, WHRB, steam turbine, steam condenser, and feedwater heater appear in different configurations for different plants with these components. However, for the case shown in Fig. 1, energy balances are presented in line with the stated objectives.

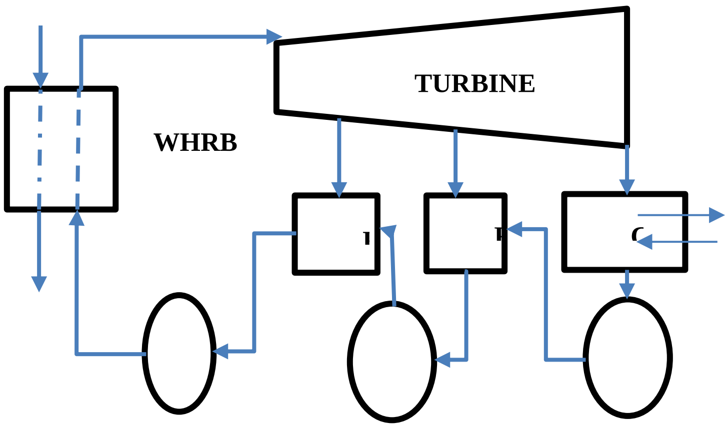

Fig.1. Schematic of a steam turbine plant with three pumps

Energy modeling of the steam turbine

The models presented for the turbine involve two feed water heaters, two pumps, and a condensing unit. For the steam turbine part, the energy balance is given in the schematic below to show values of bled steam at the feed heaters properly

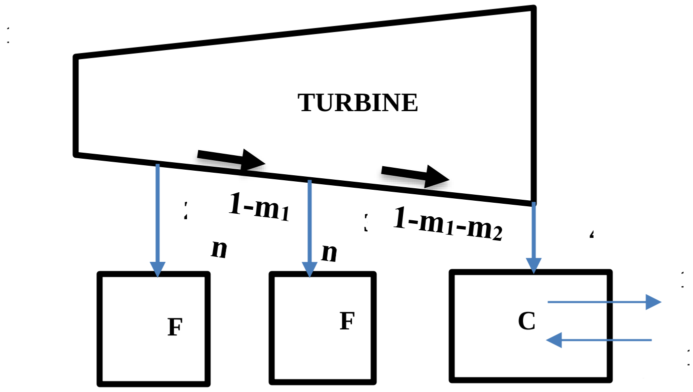

Fig. 2. The energy balance in the steam turbine.

The energy balance in the steam turbine [13] in consonance with Fig. 2, is as follows:

:$h_1 = m_1h_2 + m_2h_3 + (1 - m_1 - m_2)h_4 + W_{ST}$ ……. (1)

The pressures at points 2, 3, and 4 are known from the design conditions and are sufficient to obtain the enthalpies at these points in line with the entropy at the inlet to the turbine [13,14].

The work done by the steam turbine [15,16] is expressed as:

$$\mathbb{W}_{ST} = 1 * (h_1 - h_2) + (1 - m_1)(h_2 - h_3) + \left(1 - m_{1,} - m_2\right)(h_3 - h_4) \quad \text{…….(2)}$$

The energy balance for the condenser [8,2] takes the form:

$$h_4 + h_{w_{13}} = h_5 + h_{w_{14}} + Q_{condenser} \quad \text{…… (3)}$$

When full consideration for the cooling water is accounted for, the effectiveness of the intercooler, in terms of enthalpy values, can be obtained as [17,19]:

$$\varepsilon = \frac{h_4 - h_5}{h_4 - h_{13}} = \frac{h_{14} - h_{13}}{h_4 - h_{13}} \quad \text{……(4)}$$

Or neglecting the cooling water effect, Eqn 4 can be represented as:

$$h_4 = h_5 + Q_{condenser} \quad \text{……(5)}$$

The temperatures and enthalpies at the state points after them are represented in the normal form for real conditions. In the T-s diagram which caters for provision of the pump's efficiencies (assumed to be nearly isentropic), the ideal values are expressed with an apostrophe as used in the following derivations. The energy balance around the first pump is expressed for the ideal case as:

$$h_5 + W_{P_1} = h_6{}' \quad \text{……(6)}$$

Incorporating the efficiency of the pump, the mechanical work requirements of the pump can be obtained as:

$$W_{P_1} = \frac{h_6{}' - h_5}{\eta_{P_1}} \quad \text{……(7)}$$

Similarly, for the second and third pumps, respectively, energy balances and work rates [16, 18] are presented with the following expressions:

$h_7 + W_{P_1} = h_8{}'$ $\quad W_{P_2} = \frac{h_8{}' - h_7}{\eta_{P_2}}$ ...... (8)

$h_9 + W_{P_3} = h_{10}{}'$ $\quad W_{P_3} = \frac{h_{10}{}' - h_9}{\eta_{P_3}}$ ...... (10)

Values for the work rate are each expressed as functions of the pressure difference with the relationships:

$W_{P_1} = v_{f@P_5} * [P_6 - P_5]$: $W_{P_2} = v_{f@P_7} * [P_8 - P_7]$: $W_{P_3} = v_{f@P_5} * [P_{10} - P_9]$...(11)

For the first and second feedwater heaters, energy balance takes the form:

$h_3 + h_6 = h_7$: $\quad h_2 + h_8 = h_9$ ....(12)

Around the waste heat recovery boiler, under adiabatic boundaries, the energy balance is of the form:

$h_{11} + h_{10} = h_1 + h_{12}$ ......(13)

The effectiveness of the WHRB [15, 16] is obtained with the relationship:

$\varepsilon = \frac{h_1 - h_{10}}{h_{11} - h_{10}} = \frac{h_{11} - h_{12}}{h_{11} - h_{10}}$ ...... (14)

The energy efficiency of the plant is to be optimised and is obtained with the relationship expressed as:

$\eta_{therm.} = \frac{[1*(h_1 - h_2) + (1 - m_1)(h_2 - h_3) + (1 - m_{1,} - m_2)(h_3 - h_4)] - [W_{P_1} + W_{P_2} + W_{P_3}]}{h_1 - h_{10}}$ ......(15)

To account for the optimization routine, the values of pump performance are incorporated in Eqn (5) due to mechanical work requirements as follows:

$\eta_{therm.} = \frac{[1*(h_1 - h_2) + (1 - m_1)(h_2 - h_3) + (1 - m_{1,} - m_2)(h_3 - h_4)] - \left[\frac{h_6{}' - h_5}{\eta_{P_1}} + \frac{h_8{}' - h_7}{\eta_{P_2}} + \frac{h_{10}{}' - h_9}{\eta_{P_3}}\right]}{h_1 - h_{10}}$ ....(16)

The problem for optimization can be described as follows:

Find $X = \left[\eta_{P_1}, \eta_{P_2}, \eta_{P_3}\right]^T$ ......(17)

Minimise $\eta_{therm.} = f\left[\eta_{P_1}, \eta_{P_2}, \eta_{P_3}\right]$ ......(18)

Subject to: $\eta_{low} \le \eta_{P_1} \le \eta_{high}$: $\eta_{low} \le \eta_{P_2} \le \eta_{high}$: $\eta_{low} \le \eta_{P_3} \le \eta_{high}$ ..... (19)

RESULTS ANALYSIS AND DISCUSSION

Results analysis

The following thermodynamic assumptions are necessary:

- The whole system operates steadily, with the components having adiabatic boundaries.
- The air at entry to the air compressor is at a temperature and pressure of 25^0C and 1.013 bar, respectively.
- Heat addition is at constant pressure in the waste heat recovery boiler (WHRB).
- The temperature at the inlet to the WHRB from the topping cycle is approximately 95 per cent of the temperature of steam entering the steam turbine.
- The pressure at the inlet to the steam turbine is 35 bar and 0.08 bar at the condenser, while pressure for the bled steam is 5 bar and 1 bar, respectively, for the first and second feed heaters.
- The rated power of the steam turbine is 50 MW.

Furthermore, the exhaust air temperature, which drives the WHRB, forms the preliminary input to the system along with all the stated thermodynamic assumptions enlisted. Any operating parameters can be imposed on the system with variation parameters on certain decision variables. The operational parameters at the state points are therefore summarised in Tab. 2.

Tab. 2. Operating thermodynamic properties at the state points

State point	Temperature [K]	Pressure [Bar]	Enthalpy [kJ/kg]	Entropy [kJ/kg]
1	6.829	35	3215	670
2	6.829	5	2752	426.5
3	6.829	1	2478	372.8
4	6.829	0.08	2136	314.6
5	0.5921	0.08	173.7	314.6
6	0.5921	1	173.8	314.6
7	1.367	1	417.4	372.8
8	1.302	5	417.8	372.8
9	1.861	5	640.3	425
10	1.861	35	643.6	425.3

Discussion of results in Tab.2.

The enthalpy, entropy, temperature, and pressure at the state points are presented in Tab. 2. The pressure at the inlet to the turbine is kept at 35 and provided by pump 1. Pump 1 works between 5 and 35 bar due to its location between the first feedwater heater and the WHRB. For values shown in Table 1, the mass flow rate of steam is kept at 50.56 kg/s, while the power output from the steam turbine is at 50 MW. These values, especially the turbine-rated power, can be adjusted with a resultant increase/decrease in specific steam consumption. Mass fractions of bled steam at points 2 and 3 are 9.531 % and 9.565 %, respectively. The outlets at the feedwater heaters are at saturation temperatures corresponding to the pressures for which the bleeding was made. At these conditions, the corresponding temperatures are shown in Table 2. The enthalpy values follow a trend typical of temperature and pressure content at the state points. All computations are made with a written program developed by an engineering equation solver (EES). The efficiency of the cycle has been computed at 80 % efficiencies for the three pumps and stands at 38.39 %. This efficiency value gives credibility to the choice of operating temperatures, pressures, steam mass flow rate, etc. Accordingly, these operating properties can be used for practical design. Variation of heat input to the system at different state points is examined to determine the extent to which enthalpy changes occur in the turbine, as shown in Fig. 1.

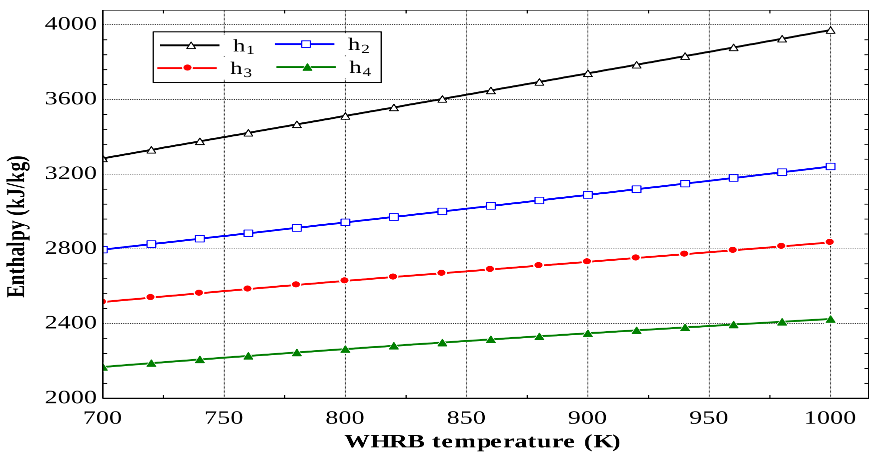

Fig. 1. The effect of enthalpies on WHRB temperatures

Enthalpy streams at the inlet to the steam turbine, first bled point, second, bled point, and after complete expansion in the turbine is shown in Fig. 1 at varying WHRB temperatures. When provision is made for a topping cycle, these temperatures will vary by the architecture of the process. In practical applications for topping gas cycles of up to 100 MW, temperatures after expansion hit up to 806 K. Here, the energy values at these points, as shown in Fig.1, represent higher steam propensity for doing work with increasing enthalpies at higher WHRB temperature values. Although this is expected, a particular point to note is the slope of these enthalpy lines against increasing temperature. The gradient of the enthalpy line at the inlet to the turbine is steeper and depicts sharp changes to energy quanta under comparatively low-temperature changes. Enthalpy-temperature changes for the bled points are less abrupt but remain constrained to the bleeding temperature. In contrast, the enthalpy at point 4 (exit from the turbine after expansion) is dictated by condenser pressure.

The choice of operating variables, especially the thermodynamic properties, resulted in very reasonable thermal efficiency values. At 80 % pump efficiency, 640 K temperature at the WHRB, 35 bar pressure at the inlet to the turbine, and a steam mass flow rate of 50.6 kg/s, the system's thermal efficiency was 38.39 %. The effect of some of these variables on the system's energy efficiency is shown in Fig. 2.

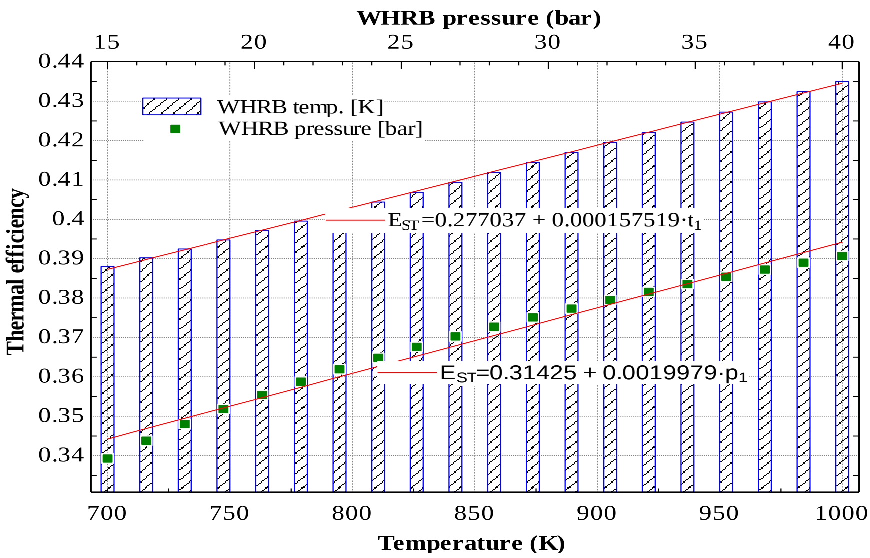

Fig. 2. Effect of WHRB temperature and pressure on thermal efficiency

Variation of temperature and pressure at the inlet to the steam turbine shows corresponding near-linear relationships (shown in Fig. 2). For variation in pressure, the temperature was kept constant at 670 K. Changes in system thermal efficiency are highly accentuated by pressure variations where a mean value of 31.425 % is readily available. Linear models of these relationships are shown in the fig. 2. The correlation coefficient for the two cases considered is nearly 1, indicating linear relationships for the variables.

The effect of pump performance on the system efficiency is investigated using the model, which relates thermal and pump efficiency.

Values for the enthalpies are shown in Tab. 1, obtained from the written program as shown in the appendix. The constraints for the pumps are kept within the following efficiency limits:

$0.50 \le \eta_{P_1} \le 0.85$: $0.50 \le \eta_{P_2} \le 0.85$: $0.50 \le \eta_{P_3} \le 0.85$.

With the objective function above, the optimization is implemented using efficiency variation in EES with the plot in Fig. 3.

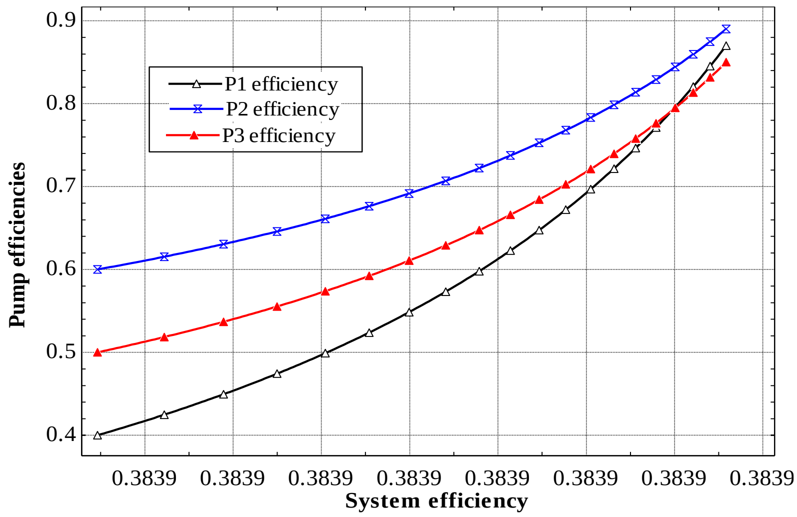

Fig. 3. Effect of pump performance on system efficiency

Variation in pump performance is shown in Fig. 3. For the considered pumps, overall cycle thermal efficiency is slightly affected by the pump performance efficiency even at higher pressures. Increasing pump efficiencies is shown to increase the overall cycle efficiency comparatively. In terms of optimum operating pump performance parameters, a limit may be placed on these values as an asymptote to the system's efficiency values as the pump performance curves approach 90 %, where the curves begin to intersect. Conclusively, the results provide leverage for turbine operation with a wide variety of pump performance efficiencies with minimal effect on system performance in terms of efficiency. In cases where a trade-off between cycle performance efficiency and cost of operation is required, these results will be helpful in multi-objective optimization of system cost and thermal efficiency without a compromise.

CONCLUSION

Pumps play a crucial role in steam-driven turbines because they are essential for supplying the working fluid with continuous kinetic energy and operate at extremely high pressures to meet boiler design requirements.

Thermodynamic modeling and optimization from an energy viewpoint will establish the link between optimum operating conditions concerning boiler design.

This was achieved through thermodynamic modeling and optimization of pump performance on a steam-driven turbine rated at 50 MW. With the selection criteria of the operating parameters, the results provide leverage for turbine operation with a wide variety of pump performance efficiencies with minimal effect on system performance in terms of efficiency. Furthermore, in cases where a trade-off between cycle performance efficiency and cost of operation is required, these results will be helpful in multi-objective optimization of system cost and thermal efficiency without trade-offs in other salient driving variables. In future work, other operational parameters of pumps on steam turbines should be considered concerning thermoeconomic and exergoeconomic cost analysis for optimum pump selection.

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METODOLOGIJA ZA TERMODINAMIČKU OPTIMIZACIJU PERFORMANSI PARNE TURBINSKE PUMPE

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Apstrakt: Pumpe su sastavni deo turbina na parni pogon jer su neophodne da obezbede kontinuiranu kinetičku energiju radnom fluidu i rade na znatno visokim pritiscima da bi se ispunili projektni uslovi kotla. Termodinamičko modeliranje i optimizacija sa energetske tačke gledišta uspostavlja vezu između optimalnih radnih uslova u pogledu dizajna kotla.

Ovo je postignuto termodinamičkim modeliranjem i optimizacijom performansi pumpe na turbini na parni pogon snage 50 MW. Koristeći razvijene energetske modele, napravljena je simulacija performansi sistema sistematskim uzimanjem u obzir varijacija u efikasnosti pumpe i ukupne snage turbine. Za razmatranu pumpu, ukupna termička efikasnost ciklusa je blago pod uticajem efikasnosti performansi pumpe čak i pri višim pritiscima. Pokazalo se da povećanje efikasnosti pumpe komparativno poboljšava ukupnu efikasnost ciklusa.

Sistem je zabeležio efikasnost pumpe od 80%, temperaturu od 640 K na VHRB, pritisak vrednosti 35 bar na ulazu u turbinu i maseni protok pare od 50,6 kg/s, respektivno. Konačno, toplotna efikasnost sistema bila je 38,39%.

U slučajevima kada je potreban kompromis između performansi ciklusa, efikasnosti i troškova rada, ovi rezultati će biti od pomoći u višestrukoj optimizaciji sličnih sistema u pogledu troškova i toplotne efikasnosti.

Ključne reči: *Parne turbine, modeliranje, energetske jednačine, performanse pumpe, efikasnost, optimizacija.*

Prijavljen/Registered:
***Submitted*: 04.11.2024.**
Ispravljen:
***Revised*: 02.12.2024.**
Prihvaćen:
***Accepted*: 01.01.2025.**