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RESEARCH ARTICLE

Energy and exergy analyses of a lab-scale open-cycle steam power plant at different boiler pressures

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Abstract

In contrast to conventional steam power plants, which employ high-pressure closed cycles, this study presents energy and exergy analyses of a non-conventional lab-scale steam power plant across different boiler pressures, with an emphasis on the effect of cycle openness. Energy loss from the condenser, turbine work, and thermal efficiency of the cycle have been investigated at different boiler pressures, using energy analysis. Exergy analysis is performed at varying boiler pressures to examine the exergy destruction in the boiler and the condenser, and to evaluate the cycle's exergetic efficiency. Experimental results show that minimizing exergy loss in one component of the cycle by varying the boiler pressure may increase it in another component; therefore, an optimal boiler pressure was investigated. As boiler pressure increases, turbine work and efficiencies initially increase because of higher exergy input but subsequently decrease as exergy destruction rises. The optimum boiler pressure is around 2.6 bar, as it minimizes exergy destruction in the cycle and energy loss in the condenser. Because the exergy input is less than the corresponding heat input, the energy efficiency (5-6.5 %) is found to be lower than the exergetic efficiency (15-21 %). In contrast to previous literature on conventional closed-cycle power plants, the current study finds condenser exergy destruction (542-564 kJ/kg) higher than that of the boiler (20-98 kJ/kg); this is due to the lack of condensate recirculation and consequent exergy loss at the condenser's open end.

Keywords: Steam power plant, boiler pressure, energy efficiency, exergetic efficiency, exergy destruction, entropy generation

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1. Introduction

Continuously increasing global energy demand is a critical issue, and steam power plants are leading to fulfil this need [1]. As they incorporate various irreversibilities, fuel waste, and environmental problems, researchers are working to minimize these issues and enhance efficiency by changing operational parameters. Conventional, first-law-based techniques are mainly concerned with energy inputs and outputs, but do not consider energy quality and thus obscure the actual inefficiencies of power plants. To overcome these shortcomings, the second law-based exergy analysis has become an effective performance assessment tool [2].

A large body of literature exists on energy and exergy analyses of conventional closed-cycle and high-pressure power plants. Researchers have investigated the impact of fuel type on the performance of power plants. Kaushik et al. reviewed energy and exergy studies of coal- and gas-powered thermal power plants and found that coal-fired power plants encounter significant exergy destruction in the boiler as compared to the gas-fired power plants [2]. Zueco et al. studied the performance of a steam power plant's components using a variety of fuels. Oxygen-rich fuels and regeneration cycles were found to improve performance. A major portion of exergy destruction occurs in the boiler, followed by the chimney stack and the steam turbine [3]. Aljundi et al. highlighted that the irreversibility of chemical reactions in the boiler is the primary cause of exergy

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destruction that can be reduced by the preheating of the combustion air and the optimization of the air-fuel ratio [4].

Some researchers have examined the effects of ambient conditions. Ameri et al. performed a combined energy, exergy, and exergo-economic analysis of a power plant. Exergetic efficiency was observed to be very sensitive to both load and ambient temperature changes [5]. Cetin et al. proved that when power plants used ultra-supercritical steam parameters with double reheat and extra feed water heaters, their thermal efficiency increased by 9.24% and exergetic efficiency by 8.06%, as compared to sub-critical power plants [6].

Some studies have reported that exergetic efficiency is lower than energy efficiency, while others have reported the opposite. Naik et al. found through exergy analysis of a 4.5 MW biomass-based plant that exergetic efficiency is lower than the energy efficiency, and the boiler is the major source of exergy destruction [7]. Arpit et al. conducted an analysis of a 120 MW sub-critical coal-fired power plant and found that its exergetic efficiency was lower than its energy efficiency. The boiler was identified as the main source of exergy destruction and the condenser as the primary site of energy loss [8]. Kanoglu et al. compared Carnot, Rankine, and flash-steam geothermal power cycles based on energy and exergy assessments and found that the energy efficiency was higher than the exergetic efficiency. The research explored that exergy analysis measures actual performance [9]. D. Mitrovic et al. found that the greatest energy loss occurs in the condenser, and major exergy destruction occurs in the boiler. In general, the energy and exergetic efficiencies achieved were 39% and 35.77%, respectively [10]. Shamet et al. indicated thermal efficiency (21.12%) to be less than exergetic efficiency (22.1%) [11]. Pilankar et al.'s analysis revealed that for individual components, energy efficiency was higher than exergetic efficiency. But for the overall power generation cycle, exergetic efficiency was found to be higher than energy efficiency [12]. Patel et al. studied the effect of powdered iron as a carbon-free sustainable alternative to fossil fuels, and the exergetic efficiency of the cycle was found to be higher than the energy efficiency [13]. Naik et al. found in another study of a 120 MW coal-based thermal power plant that exergetic efficiency is higher than energy efficiency [14]. Ahamdi et al. also reported exergetic efficiency to be higher than energy efficiency for the cycle as well as the individual components except the boiler [15].

Researchers have also explored advanced configurations such as combined heat and power (CHP) plants and combined-cycle power plants (CCPP). Noaman et al. examined the concept of integrating sCO_2 technology as a bottoming cycle in natural gas-fired CHP plants to improve energetic and economic performance. It introduced the sCO_2 cycles as an attractive but immature technology to the traditional steam or Organic Rankine Cycle bottoming systems, able to realize high thermal efficiencies up to 50%. [16]. The analysis of Massoud et al. proposed a new modification of an intercooled gas turbine-based CCPP and a steam injection system to alleviate efficiency losses due to intercooling, maximize heat recovery and minimize exergy destruction [17]. A comprehensive evaluation of CCPP done by Shireef et al. revealed that the combustion chamber is the major contributor to exergy destruction and the efficiency of the power plant is significantly improved by increasing the turbine inlet and decreasing the ambient temperature [18].

The effect of a waste heat recovery system has been studied by some researchers. Galal et al. modeled the waste heat recovery system using EES to analyze a steam power plant integrated with a sulfuric acid plant. It was revealed that higher steam inlet conditions lead to higher efficiency, whereas lower condenser vacuum pressure increases exergetic performance by 7% [19]. Mehrabi et al. found that the losses in the condenser were minimized by using exhaust steam in different refinery operations. Moreover, exergy performance was greatly enhanced by preheating the combustion air with waste heat [20].

Some researchers have investigated the effect of the pressure of different components. Rudiyanto et al. studied the effect of pressure variation in a 610 MW steam power plant in Indonesia and found that increasing boiler output pressure substantially decreases exergy losses and increases boiler exergetic efficiency. [21]. The study by Khaleel et al. showed that optimal feedwater and deaerator pressures vary as power plants age. Re-optimization can be performed using the Conjugate Directions Method to determine the optimal feed-water-heater pressure. Elimination of a feed water heater may increase or decrease the power output and efficiency, but it always deteriorates exergetic efficiency [22]. Satish et al. conducted an energy and exergy analysis of a 210 MW power plant in Vijayawada to identify inefficiencies. The results showed that low-pressure turbines had the greatest exergy destruction, while high-pressure heaters showed the maximum energy efficiency. The analysis emphasized the need to maximize the turbine performance and reduce the condenser losses to improve overall plant efficiency [23]. Tontu et al. carried out a comparison of ultra-supercritical, supercritical, and subcritical power plants, and found that the ultra-supercritical cycle had the highest efficiency because it consumed less coal. Energy and exergy efficiencies were highest in ultra-supercritical plants, followed by supercritical and subcritical plants. In all designs, the boiler irreversibility was decreased by preheating combustion air and tuning the air-fuel ratio [24]

The following table compares the present study with the previously published literature on conventional closed-cycle steam power plants. The quantitative comparison presented in Table 1 must be interpreted with caution because the two systems differ. This comparison is complemented by a detailed qualitative discussion that highlights the thermodynamic trends of the two systems, rather than establishing quantitative equivalence.

Table 1. Comparison between the present study and literature

Parameter	Present Study	Literature
Cycle	Open (finite mass / no recirculation)	Close (continuous recirculation)
Boiler Pressure	2-4 bar	30-220 bar (subcritical plants) > 220 bar (supercritical plants)
Temperature	< 200	400-540 (subcritical plants) 540-600 (supercritical plants)
Dominant exergy destruction	Condenser	Boiler
Operating nature	Quasi-steady	Steady
Energy Efficiency	5-6.5 %	30-48 %
Exergetic Efficiency	15-21 %	25-45 %

Existing studies focus on closed-cycle, high-pressure power plants in which a continuous supply of feedwater and steady-flow operation are available and heat addition in the boiler occurs at constant temperature and pressure. However, the current research analyzes a simple, non-conventional open-cycle configuration in which the boiler operates under non-isothermal and non-isobaric conditions and lacks a continuous supply of feed water. The current study deals with this distinct thermodynamic behavior, which is typically not addressed in the previous literature on conventional power cycles. These aspects clearly establish the novelty of the current study in comparison with the existing literature.

2. Methodology

Figure 1(a),1(b),2(a), and 2(b) show, respectively, the experimental setup, parts of the experimental setup, the schematic diagram, and the T-s plot for the steam power plant used in the experiment. The major components include a feed-water pump, a water-tube boiler, a turbine, and a condenser. The steam power plant, which provides all necessary conditions for energy and exergy analyses, is located on a stable surface. Temperatures at different locations in the cycle are measured by thermocouples having a measurement accuracy of ± 0.1°C, the boiler pressure is measured by the pressure gauge having a measurement accuracy of ± 0.1 bar, and flow rate is measured by the flow meter installed in the apparatus having a measurement accuracy of ± 0.005 L/min.

In this case, the cycle is not closed because the condensate from the condenser is not recirculated to the pump; instead, the condensate is wasted. Hence, before the start of the experiment, sufficient feed water is fed into the boiler from a container. As the lab-scale setup is small, this water is sufficient to produce the steam essential for the operation of the cycle for the time required to record output parameters at different boiler pressures. After closing the outlet valve of the boiler, a burner connected to an LPG cylinder is fired and inserted in the boiler for steam production. As shown in Figure 2(b), the water temperature increases continuously during processes FW–1 and 1–2. The heating process in the boiler occurs under non-isobaric and non-isothermal conditions because the boiler outlet remains closed during heat addition. Consequently, both pressure and saturation temperature increase, and the phase-change process is represented by the non-horizontal line 1 – 2. The outlet valve of the boiler is then opened, and the fuel-air ratio is correspondingly increased to achieve the desired constant pressure. Once the required pressure is achieved, it is maintained for a time sufficient to stabilize and record the required parameters.

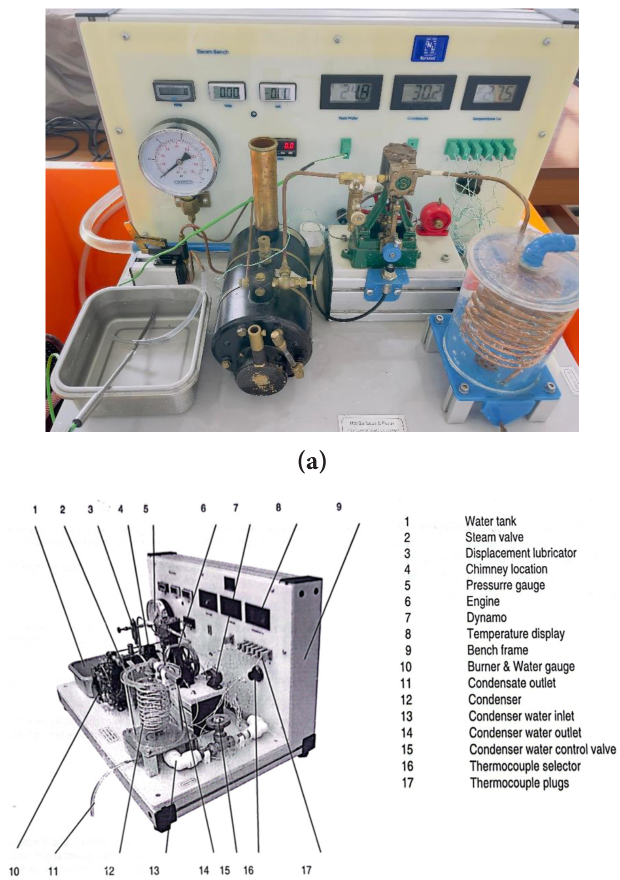
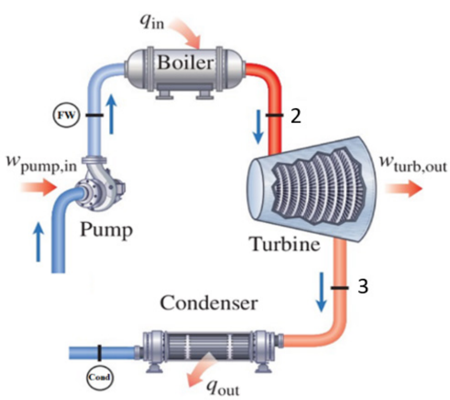
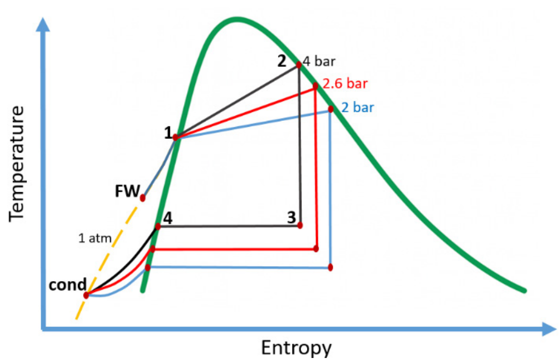


Figure 1. (a) Experimental Setup of steam power plant (photo taken by authors), **(b)** Parts of the Steam Power Plant (reproduced from lab manual of apparatus)

Experiments have been performed at boiler pressures of 2 bar, 2.2 bar, 3 bar, 3.5 bar, and 4 bar. The maximum stable pressure achievable in the experimental setup is 4 bars, due to safety considerations and the limited capacity of the heating system. Ranges below 2 bar have not been reported because output parameters, such as rpm and power, show only a slight response at boiler pressures below 2 bar owing to the low energy of the steam. The pressurized steam then expands to state 3 and rotates the turbine, converting the steam's thermal energy into mechanical energy. To provide an ideal reference for performance evaluation, the turbine expansion is assumed to be isentropic; however, the actual expansion is non-isentropic due to irreversibilities. After expansion in the turbine, the steam enters the condenser, where the cool water from the hydraulic bench condenses the steam to state "Cond". Since, it is an open-cycle and the outlet of condenser is open to atmosphere as shown in Figure 2(a), the exit state "Cond" in Figure 2(b) is always at 1 atm pressure. To perform energy and exergy analyses, experiments are performed at various boiler pressures, and the turbine rpm, ambient temperature (T_{sink}), feed water temperature (T_{FW}), effective source temperature (T_{source}), boiler outlet / turbine inlet temperature (T_2), turbine outlet / condenser inlet temperature (T_3), and condensate temperature (T_{cond}) are noted.



(a)



(b)

Figure 2. **(a)** Schematic of the experimental setup (adapted from [26]), **(b)** T-s plot of the steam cycle

The parameters are recorded only after they exhibit minimal fluctuation, thereby ensuring that reading is under stable conditions rather than as instantaneous measurements. To mitigate the effect of uncertainty, the experimental readings for each parameter are recorded three times, and the average value is used. Moreover, a detailed uncertainty analysis is presented in Table 2, which quantifies measurement uncertainties for all directly recorded parameters such as temperature and pressure. Furthermore, standard uncertainty propagation methods have been used to calculate the uncertainties in derived parameters such as turbine work, energy efficiency, exergetic efficiency, [25]. In the absence of literature on this novel open-cycle configuration, a comprehensive error analysis is conducted by considering instrumental uncertainties and propagation effects. The systematic error is dominant due to the pressure gauge uncertainty of ± 3.33 %, but the random errors are tried to be minimized by stabilizing and averaging the values. Despite higher uncertainty in boiler pressure, the calculated uncertainties for efficiency, exergy, and turbine work are quite low, indicating that the experiments are repeatable and the results are valid.

Table 2. Uncertainty analysis

Parameter	% Uncertainty
Temperature	± 0.11 %
Boiler Pressure	± 3.33 %
Condenser Heat Output	± 0.01 %
Exergy destruction in Condenser	± 0.86 %
Exergy destruction in Boiler	± 1.19 %
Turbine Work	± 0.38 %
Energy Efficiency	± 0.41 %
Exergetic Efficiency	± 0.46 %

3. Analysis

3.1. Energy analysis (First law analysis)

Energy analysis of a thermodynamic system indicates the interaction of mass, heat, and work with a thermodynamic component. It also provides information about the percentage of the thermodynamic energy that has been converted into work. Although the overall system does not remain steady because the working fluid continues to decrease, the data are recorded over short time intervals during which the key parameters such as rpm, temperature, and pressure show negligible variations; hence, the system can be approximated as quasi-steady. Moreover, the steady-flow energy and exergy balances are applied on a per-unit-mass basis to render them independent of mass. The heat input per unit mass of the working fluid in the boiler is given by:

$$q_{boiler,in} = h_2 - h_{FW} \tag{1}$$

As shown in Figure 2(b), the symbol h_2 indicates steam enthalpy at the boiler outlet or the turbine inlet. State 2 has been reasonably approximated as saturated vapor based on the presence of a two-phase mixture in the boiler during operation. As there is no recirculation of the condensed water, the boiler always contains liquid water during the experiment, and part of it continues to convert into steam, i.e, the working fluid inside the boiler exists as a two-phase liquid-vapor mixture. Under such conditions, the generated vapors are saturated. Although the boiler outlet temperature is measured experimentally, the thermodynamic condition is primarily determined by the two-phase system rather than by the temperature alone. Hence, in absence of the superheater and the presence of liquid water, the boiler outlet condition is reasonably approximated as saturated vapor, and the value of $h_2 = h_{g@T2}$ is taken from thermodynamic property tables [26] or Engineering Equation Solver (EES). State FW indicates the feed water, which is a subcooled liquid present at ambient condition, whose enthalpy $h_{FW} = h_{f@T_{FW}}$ is taken from property tables [26] or EES. State 3, as shown in Figure 2(b), represents the liquid-vapor mixture at the turbine outlet, whose enthalpy (h_3) is calculated as:

$$h_3 = h_{f3} + x_3 h_{fg3} \tag{2}$$

The numerical values of h_{f3} and h_{fg3} are obtained at T_3 from the property tables. To calculate h_3, steam quality (x_3) at state 3 is evaluated as:

$$x_3 = \frac{s_3 - s_{f3}}{s_{fg3}} \tag{3}$$

provide an ideal reference for performance evaluation, the turbine expansion is assumed to be isentropic; however, the actual expansion is non-isentropic due to irreversibilities. In the current study, turbine outlet temperature (T_3) is measured experimentally; however, other independent intensive properties, such as exit pressure or the turbine's isentropic efficiency, cannot be measured experimentally due to limitations of the experimental setup. Therefore, steam quality and enthalpy at the turbine exit are evaluated assuming isentropic expansion in the turbine i.e. $s_2 = s_3$, which is a widely used practice [27, 28]. This introduces a degree of modeling in the analysis; therefore, the turbine exit state is not purely experimental. Hence, the results are semi-empirical or partially model-driven, combining experimental measurements with thermodynamic modeling. Since this assumption is consistently applied to all calculations, the relative trend of performance parameters with respect to boiler pressure remains reliable. Measuring the turbine isentropic efficiency is recommended for future work, as it will improve the accuracy of the results. Therefore, equation (3) can be written as:

$$x_3 = \frac{s_2 - s_{f3}}{s_{fg3}} \tag{4}$$

where s_f and s_{fg} at T_3 are obtained from the thermodynamic property tables. Experimental readings show that the state "Cond" (as shown in Figure 2(b)) is a subcooled liquid, and its enthalpy (h_{cond}) is evaluated as $h_{cond} = h_{f@T_{cond}}$, where T_{cond} is the condensate temperature at the condenser outlet. To condense the steam, the heat rejected per unit mass ($\dot{q}_{cond,out}$) is evaluated as:

$$\dot{q}_{cond,out} = h_3 - h_{cond} \tag{5}$$

The pump work ($\dot{w}_{pump}$) is zero, as the pump is used only before the start of the cycle, and remains off during the experiment. Therefore,

$$\dot{w}_{pump} = 0 \tag{6}$$

The thermodynamic work produced by the turbine ($\dot{w}_{turb}$) is evaluated as:

$$\dot{w}_{turb} = h_2 - h_3 \tag{7}$$

The thermodynamic relations mentioned above quantify the energy interactions among the four basic components of the steam power plant: the boiler, condenser, pump, and turbine. This quantification has been used to analyze the various efficiencies within the cycle. For example, the thermal efficiency of the turbine (η_{turb}) is evaluated as follows:

$$\eta_{turb} = \frac{\dot{w}_{turb}}{\dot{q}_{boiler,in}} \tag{8}$$

The energy efficiency of the cycle (η_I) is evaluated as:

$$\eta_I = \frac{\dot{w}_{Net}}{\dot{q}_{boiler,in}} = \frac{\dot{w}_{turb} - \dot{w}_{pump}}{\dot{q}_{boiler,in}} \tag{9}$$

Since the work done by the pump through Eq. (6) is zero, therefore Eq. (9) is written as:

$$\eta_I = \eta_{turb} = \frac{\dot{w}_{turb}}{\dot{q}_{boiler,in}} = \frac{h_3 - h_2}{h_2 - h_{FW}} \tag{10}$$

3.2. Exergy analysis (second law analysis)

Developing an efficient thermodynamic system is a major challenge for energy engineers, and exergy analysis of such systems plays an important role in it [2, 9]. Exergy is the maximum useful work obtained from a thermodynamic system as it reaches the dead state [26]. Exergy analysis identifies the location and magnitude of thermodynamic losses, thereby indicating areas where improvements can be made. The specific exergy destroyed (χ_{dest}) for each component of the steam power plant is determined by the following general relation:

$$\chi_{dest} = T_{sink}\, s_{gen} \tag{11}$$

where T_{sink} is the temperature of the ambient where the waste heat is rejected, and s_{gen} is the specific entropy generation, which is evaluated as:

$$s_{gen} = \left(s_e - s_i + \frac{q_{out}}{T_{sink}} - \frac{q_{in}}{T_{source}} \right) \tag{12}$$

where s_e is the specific entropy at the exit, s_i is the specific entropy at the entrance of a component, and T_{source} is the effective source temperature, which is assumed to be equal to the measured boiler surface temperature. Although the heat addition in the boiler occurs over a range of temperatures involving combustion gases and metal walls, but their measurement is not possible in the current study due to experimental limitations. Therefore, the use of boiler surface temperature is an approximation which is adopted as an effective source temperature representing an average thermodynamic level at which heat is transferred to the working fluid of power plant. This approximation may affect the absolute values of exergy destruction, but it does not affect the mentioned trends of performance parameters with respect to the boiler pressure. Thus, exergy destruction per unit mass in the boiler ($\chi_{dest,\ boiler}$) can be evaluated as:

$$\chi_{dest,\ boiler} = T_{sink}\left(s_2 - s_{FW} - \frac{q_{boiler,in}}{T_{source}} \right) \tag{13}$$

The exergy destruction per unit mass in the condenser ($\chi_{dest,\ cond}$) is evaluated as:

$$\chi_{dest,\ cond\backslash} = T_{sink}\left(s_{cond} - s_3 + \frac{q_{cond,out}}{T_{sink}} \right) \tag{14}$$

Since, the process 2-3 is ideally isentropic, the enthalpy at state 3 is $s_3 = s_2 = s_{g@T_2}$ Similarly, specific entropy of the steam at state "cond" is evaluated as $s_{cond} = s_{f@T_{cond}}$. Since the heat loss and gain in the turbine can be assumed negligible as compared to that in the boiler and the condenser, the exergy destruction per unit mass in the turbine ($\chi_{dest,\ turb}$) is evaluated as:

$$\chi_{dest,\ turb} = T_{sink}\left(s_3 - s_2 \right) = 0 \tag{15}$$

because s_3=s_2 Exergy destruction in the pump is of no means since it does not work during the operation of cycle. Hence, the total exergy destruction per unit mass for the cycle ($\chi_{dest,\ total}$) is equal to the sum of all exergy destructions as:

$$\chi_{dest,\ total} = \chi_{dest,\ bolier\backslash} + \chi_{dest,\ cond} + \chi_{dest,\ turb} \tag{16}$$

After evaluating total exergy destruction per unit mass, the exergetic efficiency of the overall cycle is calculated as:

$$\eta_{II,cycle} = \frac{\text{exergyrecovered}}{\text{exergyexpended}} = 1 - \frac{X_{dest,total}}{X_{expended}}$$
$$= \frac{w_{net}}{X_{heat,in}} = 1 - \frac{X_{dest,total}}{X_{heat,in}} \tag{17}$$

As mentioned in Eq. (9), the network output per unit mass is evaluated as:

$$w_{net} = w_{turb} = h_3 - h_2 \tag{18}$$

The exergy expended ($\chi_{expended}$) is the exergy of heat energy provided to the boiler ($\chi_{heat,in}$), which is given as:

$$\chi_{expended} = \chi_{heat,in} = \left(1 - \frac{T_{sink}}{T_{source}} \right) q_{boiler,in} \tag{19}$$

Using these relations, Eq. (17) yields the final relation for exergetic efficiency of the cycle as:

$$\eta_{II,\ cycle} = \frac{h_3 - h_2}{\left(1 - \frac{T_{sink}}{T_{source}} \right)(h_2 - h_{FW})} = 1 - \frac{\chi_{dest,\ total}}{\left(1 - \frac{T_{sink}}{T_{source}} \right) q_{boiler,in}} \tag{20}$$

4. Results and discussion

The performance of the low-pressure open-cycle steam power plant has been investigated under various boiler pressures to determine the optimum operating condition. The analysis of experimental results discusses the effect of boiler pressure on various performance parameters such as condenser heat rejection and the turbine work, the effect of boiler pressure on exergy destructions in different components, and the effect of boiler pressure on energy and exergetic efficiencies.

4.1. Heat rejection from the condenser

The effect of the boiler pressure on the specific heat rejection by the condenser is shown in Figure 3. Heat rejection from the condenser reduces cycle efficiency. As shown in Figure 3, the initial decline in heat rejection between boiler pressures of 2.0 and 2.6 bar is due to an increase in the cycle's thermal efficiency. As boiler pressure rises, higher turbine-inlet enthalpy leads to a larger enthalpy drop during expansion, increasing turbine work. However, at pressures above 2.6 bar, irreversibilities (entropy generation) in the small-scale turbine increase, and steam exits the turbine at a higher enthalpy than in the lower-pressure experiments.

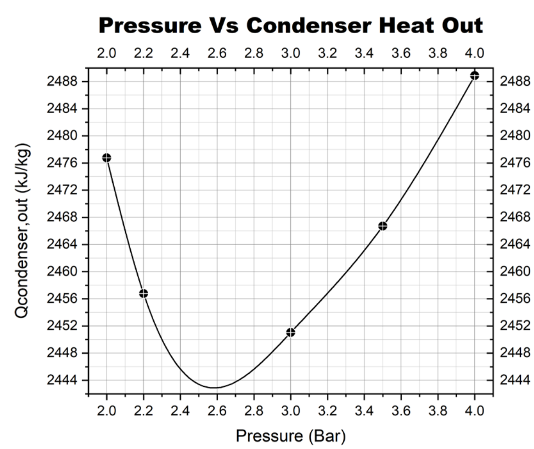


Figure 3. Effect of boiler pressure on heat rejection from the condenser

Since the condenser in this open-cycle configuration must reject this higher enthalpy energy to the environment to return the working fluid to ambient conditions, the amount of heat rejected increases. This behavior indicates that 2.6 bar is the optimal condition that balances energy supplied and entropy generation. In Figure 2(b), the state "cond" is fixed in all cases because the condensate from the condenser is always at ambient conditions.

4.2. Effect of boiler pressure on exergy destruction

The exergy profile of a component represents the lost work or degradation of energy quality. Exergy destruction primarily takes place in the boiler and the condenser. Ideal isentropic expansion in the turbine results in no exergy destruction, and the exergy destruction in the pump does not need to be evaluated since the pump is not involved during the operation of the cycle.

4.2.1. Exergy destruction in the condenser

The correlation between the boiler pressure and the associated exergy destruction in the condenser is shown in Figure 4. Since the principal cause of exergy destruction is entropy generation resulting from heat transfer across a finite temperature difference, Figure 4 follows a trend like that of Figure 3. The plot shows a maximum exergy destruction of 564 kJ/kg at 2 bars, decreasing to a minimum of 542 kJ/kg at around 2.6 bars. This reduction indicates that the condenser operates more efficiently at 2.6 bar due to minimum irreversibilities and a better thermodynamic match between the working fluid and the cooling water. A better thermodynamic match refers to an optimal balance between heat gain and irreversibility. However, above the boiler pressure of 2.6 bar, exergy destruction rises again because of thermodynamic mismatch and increased irreversibilities resulting from large heat transfer across finite temperature

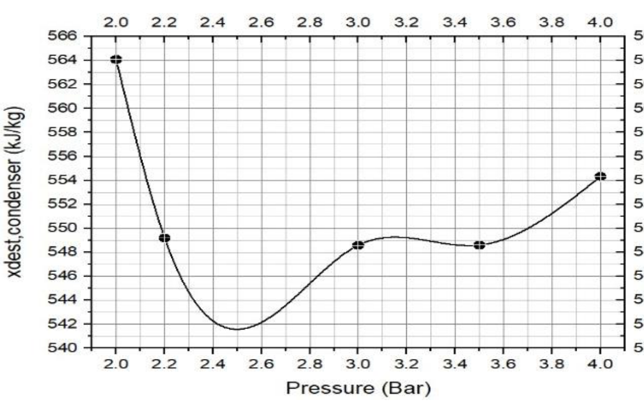


Figure 4. Effect of boiler pressure on exergy destruction in condenser

differences, as shown in Figure 3. The only difference from Figure 3 is that the highest exergy destruction occurs at 2 bars; this is due to the wider 3-cond gap for the case representing the condenser at 2.0 bar in Figure 2(b). This wider gap indicates a higher entropy difference and hence higher exergy destruction. This demonstrates how crucial condenser irreversibility is to the change in boiler pressure.

4.2.2. Exergy destruction in the boiler

Figure 5 shows how boiler exergy destruction varies with boiler pressure. Exergy destruction within the boiler is highly dependent on operating pressure, and the boiler exhibits a steep exergy destruction-pressure gradient, rising from a low value at 2 bar to approximately 98 kJ/kg at 4 bar. Entropy generation in the boiler is mainly governed by the temperature difference between the heat source and the working fluid [29]. Although the saturation temperature of water rises at higher boiler pressures, the flame temperature also increases, providing greater heat energy. The net effect is to increase the temperature difference between the hot working fluid and the heat source. The real boiler process involves non-uniform heat addition and higher local temperature gradients, which lead to increased exergy destruction at higher boiler pressures. Although higher pressure seems to be more energy efficient due to high-quality energy, it also increases exergy destruction in the boiler. This shows why exergy-based analysis is important in determining the system's actual performance limits. The irreversibilities in the boiler can be reduced by improving combustion control, implementing stage-wise or regenerative heat input, and improving insulation.

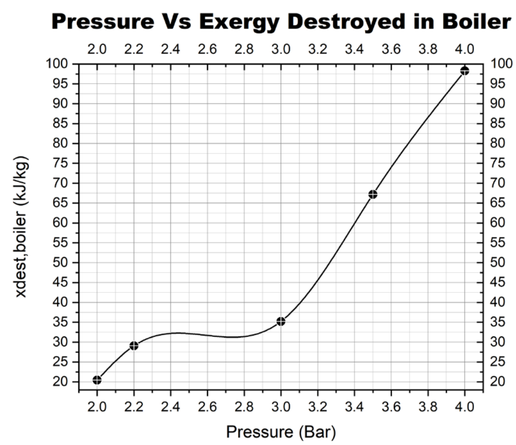


Figure 5. Effect of boiler pressure on exergy destruction in the boiler

4.2.3. Total exergy destruction

Due to the differing impacts of boiler pressure on the exergy losses in the boiler and condenser, optimizing boiler pressure for the overall cycle is therefore important. Figure 6 shows the dependence of total exergy destruction on boiler pressure. The experimental results demonstrate total exergy destruction of about 584 kJ/kg at 2.0 bar, which declines to a minimum of about 574 kJ/kg at about 2.6 bar and then rises to a maximum of 653 kJ/kg at 4.0 bar. The initial decline is due to improved exergy input, and the dramatic rise after 2.6 bar is attributed to rapidly increasing irreversibilities in the boiler and to increased energy loss through the condenser. As a result, the exergetic optimum is approximately 2.6 bars in the case under investigation, and total exergy destruction increases significantly when deviating from this condition. For design and operation, this finding indicates the procedure for determining the optimum boiler pressure to achieve optimal performance of the power cycle. Reducing exergy destruction in the boiler does not necessarily lead to optimal whole-plant performance unless the full-cycle performance is considered. Therefore, the optimal operating point at approximately 2.6 bar is not only a balance between component losses but also a state in which the entire system is deliberately compromised to give the best of low-temperature heat rejection and heat utilization.

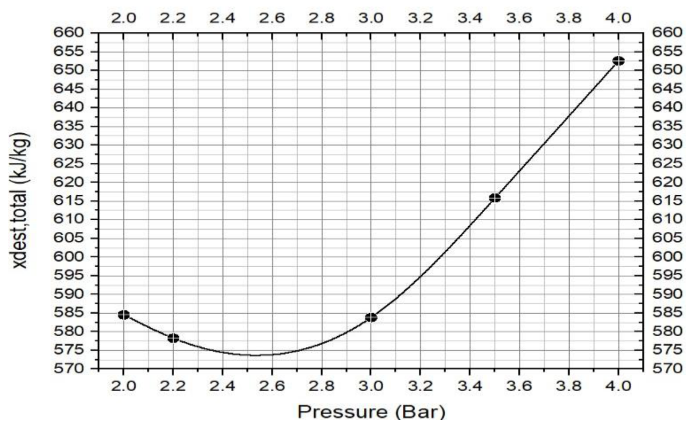


Figure 6. Effect of boiler pressure on total exergy destroyed

4.2.4. Comparison of exergy destructions

Figure 7 compares the exergy destruction in the boiler and the condenser at various boiler pressures. In contrast to the existing literature on conventional closed-cycle power plants, this study found exergy destruction in the condenser to be higher than that in the boiler. In conventional high-pressure power plants, the boiler dominates in the exergy destruction due to fuel combustion at higher pressure and temperature; and recirculation of the same working fluid in the cycle [2-4, 8, 10]. In the current study, the analysis has been expanded to demonstrate that condenser exergy destruction is dominant in the open-cycle power plant without condensate recirculation. The condenser condenses the steam to ambient conditions, which requires a large temperature gradient between the steam and the surroundings. Thus, a significant portion of the supplied exergy leaves the system, leading to substantial exergy destruction. This behavior is due to the lack of recirculation of the working fluid or to the openness of the cycle, which hinders effective

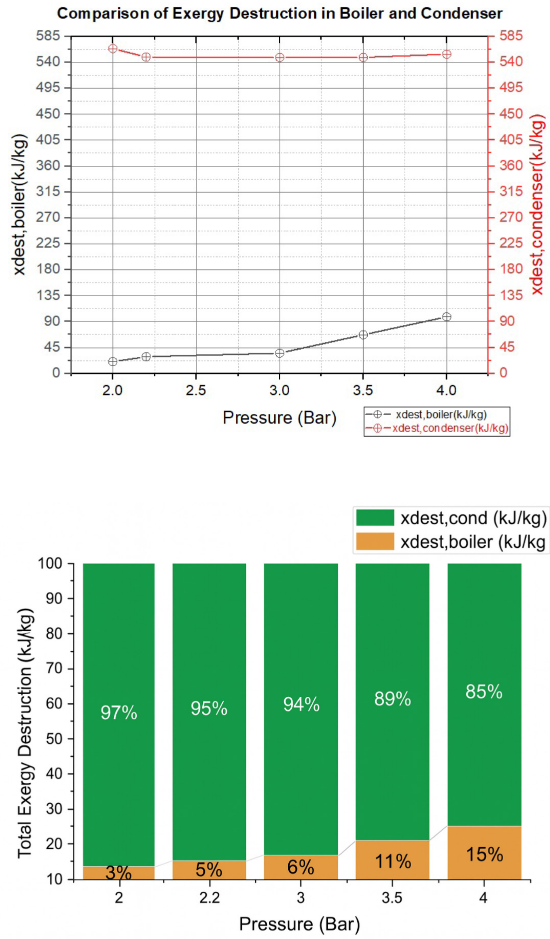


Figure 7. Comparison of exergy destruction in the boiler and the condenser

recovery of thermal energy and hence shifts the location of higher exergy destruction from the boiler to the condenser. Regarding the second law, it means that the open-cycle steam power plant is not boiler-dominated but is condenser-dominated. Figure 7(b) shows another important observation: the relative contribution of the boiler to total exergy destruction increases continuously. This is due to non-uniform heat addition and higher local temperature gradients at higher pressures, which raise the potential for irreversibility in the boiler.

4.3. Turbine work

Figure 8 shows how turbines work varies with boiler pressure. The experimental findings indicate that turbine work increases from 166 kJ/kg at 2.0 bar to 182 kJ/kg at 2.6 bar, then declines to 141 kJ/kg at 4.0 bar. This trend in the graph can be explained by Figures 3 and 6. In Figure 3, the heat rejected by the condenser represents the energy loss of the overall cycle. Hence, in the region of the graph where heat loss through the condenser decreases, turbine work output correspondingly increases, and vice versa. Thus, at the boiler pressure of

2.6 bar, minimal heat loss through the condenser results in maximal turbine work. Figure 6 shows that the minimum exergy destruction (irreversibilities) in the cycle occurs at the boiler pressure of 2.6 bar, resulting in a maximum turbine work output under this condition. The consistency of the results supports the accuracy of the experimental findings. The graph establishes that 2.6 bar is the optimum boiler pressure for maximum turbine work, and operating at pressures other than this would waste available energy. That is why the optimization of the steam cycle should focus on the choice of boiler pressure. The other form of work, i.e., electric work, is not measured due to certain limitations; the scope of the current study is limited to thermodynamic analysis.

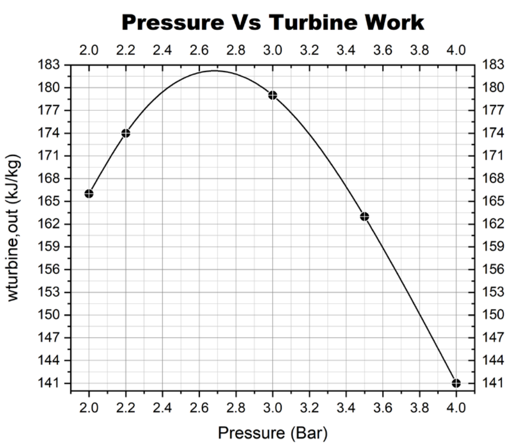


Figure 8. Effect of boiler pressure on turbine work

4.4. Effect of boiler pressure on efficiencies

4.4.1. Energy efficiency

Figure 9 shows the effect of boiler pressure on the cycle's energy efficiency. The energy efficiency, given by Eq. (10), is the ratio of the amount of work extracted from the turbine to the thermal energy input to the boiler. Since energy efficiency is proportional to turbine work output, it exhibits the same trend w.r.t boiler pressure as turbine work output. Figure 9 shows a noticeable improvement in efficiency as boiler pressure increases from 2.0 to 2.6 bar, with a maximum cycle efficiency of about 6.6% observed at 2.6 bar. The efficiency gradually decreases as boiler pressure increases up to 4.0 bar. This nonlinear behavior demonstrates the critical transition between increasing thermal potential and irreversibility. When boiler pressure rises from 2.0 to 2.6 bar, efficiency increases because the input thermal energy exceeds the irreversibilities. After the threshold of 2.6 bar, efficiency decreases because the irreversibilities increase more sharply than the input thermal energy. This decrease in energy efficiency can also be attributed to increased entropy generation or exergy destruction at elevated boiler pressures.

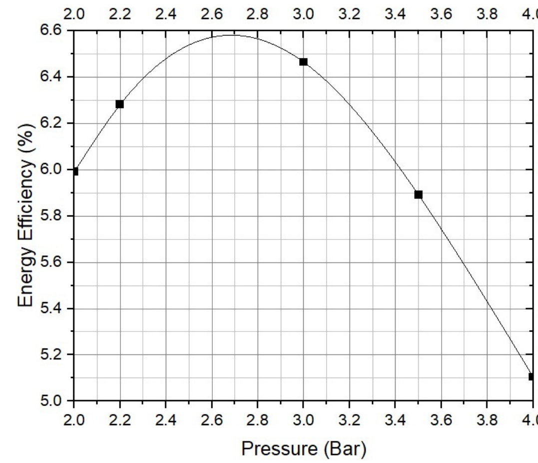


Figure 9. Effect of boiler pressure on energy efficiency

4.4.2. Exergetic efficiency

Exergetic efficiency is a measure of useful work extracted from the available energy. As given by Eq. (20), it is the ratio of the total exergy recovered (i.e., extracted turbine work) in the cycle to exergy of the heat energy provided for the working fluid in the boiler. Figure 10 shows that, at boiler pressures between 2 and 2.6 bars, the exergetic efficiency rises to a maximum of 21% because of an increase in recovered exergy, as shown in Figure 8. Although growth in heat exergy input to the boiler's working fluid is also observed, experimental data show that its rate of increase remains lower than the rate of exergy recovered. Hence, Figure 10 shows a trend like Figure 8. The findings indicate that, as the boiler pressure rises above 2.6 bar, exergetic efficiency gradually decreases, reaching 15.5% at 4 bar. This loss of efficiency recorded during the experiment is due to an increase in exergy destruction, which reduces the recovery, as depicted in Figure 6. Efficiency deterioration is a clear indication of irreversibilities at higher boiler pressures.

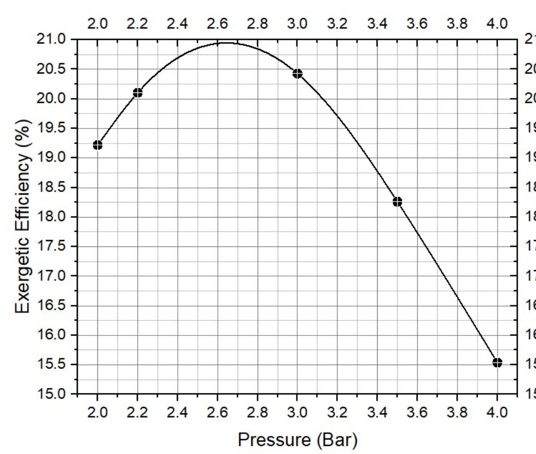


Figure 10. Effect of boiler pressure on exergetic efficiency

4.4.3. Comparison of energy and exergetic efficiencies

The comparative analysis of the two efficiencies in Figure 11 indicates that they follow similar trends, both rising as pressure increases, reaching their respective peaks at about 2.6 bar, and decreasing thereafter. This also indicates that the exergetic efficiency is higher than the energy efficiency at all boiler pressures; this difference can be explained by Eqs. (10) and (20), which define both efficiencies. The numerator of both equations is the same, but the denominator of Eq. (20) has an additional factor $\left(1 - \frac{T_{sink}}{T_{source}}\right)$. Since , $T_{source} > T_{sink}$ this explains that exergy input will always be lower than the corresponding heat input. This difference is particularly important for the current study because the lab-scale setup employs low-pressure, low-temperature heat addition. Therefore, in the current study, the exergy content of heat is lower compared with that of high-pressure power plants. It means that only a small portion of the supplied thermal energy can be converted into work. The impact of this factor on the performance comparison is as follows:

i The relatively low exergy input causes exergetic efficiency to be higher than energy efficiency.
ii Higher exergetic efficiency indicates better utilization of available energy.
iii Exergy input increases with boiler pressure, but irreversibilities increase more rapidly, reducing overall efficiency beyond a certain limit of 2.6 bar. Hence, an optimum pressure exists at which the balance between available energy and irreversibilities is most favorable.

The thermodynamic interpretation of this trend is that energy efficiency accounts for all the heat input energy provided to the boiler, while exergetic efficiency considers only a portion of the total input energy that can be converted into useful work, resulting in exergetic efficiency being higher than energy efficiency. Same result has been reported by Shamet et al. [11], Pilankar et al. [12], Patel et al. [13] and Ahamdi et al. [15]. These findings suggest that the most efficient operating pressure in the system is around 2.6 bar, at which both energy and exergy efficiencies are optimized.

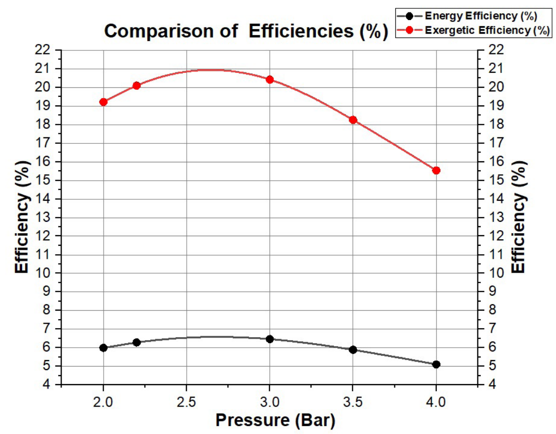


Figure 11. Comparison of energy and exergetic efficiencies

The following table summarizes the key numerical results at each boiler pressure. The table includes key parameters such as turbine work, energy efficiencies, exergy efficiencies, and exergy destruction.

Table 3. Summary of key numerical results

Boiler Pressure (bar)	Speed (rpm)	Turbine Work (kJ/kg)	Energy Eff. (%)	Exergetic Eff. (%)	Exergy Destruction in Boiler (kJ/kg)	Exergy Destruction in Condenser (kJ/kg)	Total Exergy Destruction (kJ/kg)	Condenser Heat Rejection (kJ/kg)
2.0	780	166	5.991	19.217	20.427	564.090	584.517	2476
2.2	830	174	6.282	20.101	29.095	549.193	578.288	2456
3.0	890	179	6.464	20.427	35.223	548.596	583.819	2452
3.5	1050	163	5.891	18.259	67.196	548.609	615.805	2468
4.0	1100	141	5.105	15.539	98.294	554.332	652.626	2488

This study has potential applications in the design and optimization of small-scale open-cycle steam power plants, such as educational test rigs, pilot plants, and laboratory-scale power units. It demonstrates the identification of the optimal boiler pressure for such small-scale power plants and the behavior of other performance parameters. This study is also applicable to open-cycle or once-through steam-based configurations, in which the working fluid is not recirculated. Since a large portion of exergy is destroyed in the condenser of such systems, integrating another heat-recovery system can significantly improve system efficiency.

5. Conclusion

Existing studies concern closed-cycle, high-pressure power plants in which a continuous supply of feed water and steady-flow operation are available, and heat addition in the boiler occurs at constant temperature and pressure. However, the current research analyzes a non-conventional low-pressure, open-cycle configuration in which the boiler operates under non-isothermal and non-isobaric conditions and has no continuous supply of feed water. The energy and exergy analysis at different boiler pressures have led to the insight that reducing exergy loss in one component of the cycle, by varying the boiler pressure may increase it in another. Therefore, the optimal state for a given power plant needs to be investigated; in the current study, this is at the boiler pressure of 2.6 bar. This is due to an optimal trade-off between heat exergy and irreversibilities, i.e., minimized exergy destruction in the cycle and energy loss in the condenser, resulting in peak turbine work and peak energy and exergetic efficiencies.

Another key finding is that open-cycle steam power plants are not boiler-dominated but condenser-dominated; i.e., the exergy destruction in the condenser is higher than in the boiler, which contrasts with existing literature on closed-cycle steam power plants. This is due to the openness of the cycle at the condenser exit, which prevents recirculation of the condensate and causes its exergy to be lost to the environment. However, the relative contribution of the boiler to total exergy destruction increases continuously with pressure owing to non-uniform heat addition and larger local temperature gradients at higher pressures, which increase the potential for irreversibility in the boiler. Moreover, the exergetic efficiency of the current cycle is higher than the energy efficiency because the energy analysis accounts for all the heat input to the boiler, whereas the exergy analysis considers only a portion of the total input energy that can be converted into useful work. The results of this study are valid only for a laboratory-scale, low-pressure, open-cycle steam power plant with no recirculation of the working fluid.

Nomenclature

Symbol

$\dot{Q}$	Heat transfer rate (kJ/s)
$\dot{W}$	Power (kW)
η	Efficiency (-)
η_I, η_{II}	Energy, Exergetic efficiency of cycle
h	Specific enthalpy (kJ/kg)
s	Specific entropy (kJ/kg.K)
s_{gen}	Specific entropy generation (kJ/kg.K)
x	Steam quality (-)
b_f, b_g, b_{fg}	Property (b) of saturated liquid, saturated vapor and difference between both, respectively
χ_{dest}	Specific exergy destruction (kJ/kg)
$\chi_{expended}$	Specific exergy expended (kJ/kg)
$\chi_{heat,in}$	Specific heat exergy provided to boiler (kJ/kg)
h_2	Specific enthalpy at turbine inlet / boiler outlet (kJ/kg)
h_3	Specific enthalpy at condenser inlet / turbine outlet (kJ/kg)
h_{FW}	Specific enthalpy of feed water (kJ/kg)
h_{cond}	Specific enthalpy of condensate (kJ/kg)
T_{source}	Effective source temperature (K)
T_{sink}	Ambient temperature (K)
T_{cond}	Condensate Temperature (K)

Authorship contributions

Abdul Samad Saleem: Literature review, design of experiments, editing of manuscript, analysis of results, supervision. Luqman Ali Khan: Original draft preparation, experimentation, plotting the graphs. Hasnain Ali Shah: Original draft preparation, literature review, experimentation. Malik Sarmad Zahid: Literature review, analysis of results.

Data availability statement

The experimental data is available from the corresponding author upon request.

Conflict of interest

The authors declare that they have no competing financial or non-financial interests.

Ethics

There are no ethical issues with the publication of this manuscript.

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Figure Captions

Figure 1. (a) Experimental Setup of steam power plant (photo taken by authors), (b) Parts of the Steam Power Plant (reproduced from lab manual of apparatus)

Figure 2. (a) Schematic of the experimental setup (adapted from [26]), (b) T-s plot of the steam cycle

Figure 3. Effect of boiler pressure on heat rejection from the condenser

Figure 4. Effect of boiler pressure on exergy destruction in condenser

Figure 5. Effect of boiler pressure on exergy destruction in the boiler

Figure 6. Effect of boiler pressure on total exergy destroyed

Figure 7. Comparison of exergy destruction in the boiler and the condenser

Figure 8. Effect of boiler pressure on turbine work

Figure 9. Effect of boiler pressure on energy efficiency

Figure 10. Effect of boiler pressure on exergetic efficiency

Figure 11. Comparison of energy and exergetic efficiencies