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RESEARCH ARTICLE ENERGY

Energy and Exergy Analysis of an Al-Hussein Steam Power Plant in Jordan Using Engineering Equation Solver (EES)

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ABSTRACT

This study independently reproduces the energy and exergy analysis of the Al-Hussein steam power plant in Jordan using Engineering Equation Solver (EES) under the same operating conditions and reference environment (298.15 K, 101.3 kPa) reported in the original work. The reproduced thermodynamic properties and component energy balance show close agreement with the published results, confirming the robustness of the first-law analysis. In contrast, the exergy results reveal notable deviations in specific components: the condenser exergy destruction is calculated as 18.03 MW (vs 13.74 MW reported), yielding a markedly lower condenser exergy efficiency (3.40%), while LPH5 exhibits a higher exergy efficiency (82.34% vs 67.3%). The distribution of irreversibilities indicates that the boiler remains the dominant source of exergy destruction (74.55% of the total), followed by the turbine (12.83%) and condenser (11.13%), highlighting the boiler as the primary target for performance improvement. These findings demonstrate that independent computational reproduction can confirm energy results while revealing sensitivity of component-level exergy metrics to modelling assumptions, improving the reliability of exergy-based performance assessments.

تحليل الطاقة والإكسرجي لمحطة الحسين لتوليد الطاقة البخارية في الأردن باستخدام برنامج حل المعادلات الهندسية (EES)

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الكلمات المفتاحية

تحليل الطاقة والإكسرجي
محطة قدرة بخارية
برنامج المعادلات الهندسية
تدمير الإكسرجي
اللاإنعكاسية
الحالة المرجعية

الملخص

تتناول هذه الدراسة إعادة إنتاج مستقلة لتحليل الطاقة والإكسرجي (Exergy) لمحطة الحسين البخارية في الأردن باستخدام برنامج Engineering Equation Solver (EES)، وذلك تحت نفس ظروف التشغيل والحالة المرجعية (298.15 كلفن، 101.3 كيلو باسكال) المعتمدة في الدراسة الأصلية. وقد أظهرت النتائج المتحصل عليها توافقًا وثيقًا في الخصائص الديناميكية الحرارية وتوازنات الطاقة على مستوى المكونات، مما يؤكد متانة تحليل القانون الأول ودقته في توصيف الأداء الطاقي للمحطة. في المقابل، كشفت نتائج تحليل الإكسرجي عن انحرافات ملحوظة في بعض المكونات. إذ تم حساب معدل تدمير الإكسرجي في المكثف بقيمة 18.03 ميغاواط مقارنةً بـ 13.74 ميغاواط في الدراسة الأصلية، ما أدى إلى انخفاض ملحوظ في كفاءته الإكسرجية لتصل إلى 3.40%. كما أظهرت وحدة التسخين ذات الضغط المنخفض (LPH5) ارتفاعًا في الكفاءة الإكسرجية (82.34% مقابل 67.3%). ويُبيّن توزيع اللاإنعكاسية أن الغلاية ما تزال المصدر الرئيس لتدمير الإكسرجي بنسبة 74.55% من الإجمالي، تليها التوربينة بنسبة 12.83% ثم المكثف بنسبة 11.13%، الأمر الذي يؤكد أن الغلاية تمثل الهدف الأساسي لتحسين الأداء وتقليل الفواقد اللاإنعكاسية. تؤكد هذه النتائج أن إعادة الإنتاج الحاسوبي المستقلة تمثل أداة فاعلة للتحقق من نتائج تحليل الطاقة، كما تكشف في الوقت ذاته عن حساسية مؤشرات الإكسرجي على مستوى المكونات تجاه افتراضات النمذجة والاختيارات الحسابية، مما يعزز موثوقية تقييمات الأداء المعتمدة على الإكسرجي ويوفر أساسًا أكثر دقة لاتخاذ قرارات التحسين والتطوير المستقبلي.

Introduction

The energy sector in Jordan is one of the most rapidly growing sectors in the country. Over the past years, the annual growth rate of electricity demand has exceeded 9%, with an installed capacity and annual production of around 9,000 GWh in 2006 [1,2]. The Central Electricity Generating Company (CEGCO) is operating as the primary electricity producer in Jordan, relying on heavy fuel oil, diesel oil, natural gas, and renewable

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energy sources. Spread over Jordanian cities, the electricity production stations of CEGCO transmit electricity to consumers via 132 kV and 400 kV overhead transmission lines [3,4]. Under these circumstances, the enhancement of the thermodynamic efficiency of existing thermal power stations is, on the one hand, an economic requirement to minimize fuel consumption, and on the other hand, an environmental requirement to restrict emissions [5,6].

Conventional energy analysis (First Law analysis) is very common for the calculation of heat and work interactions, but in many cases, it has been found to be less informative regarding the quality of energy and the actual sites of irreversibilities. For example, energy analysis may point out the significant amount of heat rejection in the condenser and, therefore, identify it as the major source of loss, although the rejected heat is mostly of low quality. Exergy analysis (Second Law analysis), on the other hand, makes use of the reference environment (dead state) and is able to directly calculate the irreversibilities and exergy destruction, which is more informative for the diagnosis of inefficiencies and the identification of improvement priorities [7-10].

A wide array of research has ascertained that exergy analysis usually pinpoints the boiler/steam generator as the most significant source of irreversibility, mainly due to combustion and large temperature differences, with considerable contributions from the turbine train and heat transfer equipment. For instance, Khaleel et al. [11] have particularly stressed that energy analysis can be misleading by giving excessive importance to condenser heat rejection, while exergy analysis points to the actual sources of irreversibilities and exergy destruction in the components, which are usually dominated by the boiler. Likewise, Galal et al. [12] have developed an EES model for a 5 MW steam power plant with waste heat recovery and found that the waste heat boiler is the dominant source of exergy destruction, and that exergy efficiency can be improved by operational optimization, such as reducing the condenser pressure. Similar results have also been obtained for large coal-fired power plants under different ambient conditions; Gungor Celik and Aydemir [13] have found that the boiler is still the dominant source of exergy destruction at all ambient temperatures, with considerable contributions from intermediate- and low-pressure turbines and the condenser.

More recent studies have then proceeded to extend conventional analysis by employing real plant data, incorporating real instrumentation assumptions, and proposing specific modifications. Mehrabi Gohari et al. [14] found boiler dominance in terms of energy loss and exergy destruction in a refinery steam cycle and proposed modifications such as combustion air preheating. Suryo et al. [15] and Hamayun et al. [16] similarly found the boiler to be the dominant source of irreversibility and proposed process and control/insulation modifications to mitigate avoidable destruction. Concurrently, more advanced methods such as advanced exergy analysis have been suggested to more accurately reflect inter-component interactions and realistic improvement potential; Azubuike et al. [17] demonstrated that a substantial portion of exergy destruction is endogenous, allowing for more informed prioritization than conventional analysis. Optimization-related studies also emphasize that practical improvement opportunities regularly follow a consistent thermodynamic principle: enhancing main/reheat steam quality and decreasing condenser pressure within limitations [18,19].

Notwithstanding this level of maturity in the area, there is an important remaining gap in that many existing plant analyses have published final exergy results without (i) an independent computational check, (ii) open verification of calculations of thermodynamic properties, and (iii) an explicit discussion of the implications of modeling decisions, particularly dead-state assumptions, on exergy destruction and efficiency at the component level. This becomes particularly important for sensitive components such as the condenser and low-pressure heaters, where relatively small differences in assumptions or property calculations can result in large differences in exergy results. This, in turn, can affect the reliability of decisions based on these results, such as the ranking of improvement priorities.

Consequently, the aim of this research is to carry out an independent recalculation and verification of the energy and exergy analysis of the Al-Hussein steam power plant in Jordan using the EES under the same operating conditions and reference environment as in the original research. The aim of this research is to: (i) carry out a recalculation of the energy and exergy balance for each component, (ii) calculate the exergy destruction and exergy efficiency for each of the major components, and (iii) determine the effect of changing the temperature of the reference environment on the major exergy parameters.

Materials and Methods

Plant description

The Al-Hussein power plant is located in Zarqa (altitude = 560 m), about 30 km northeast of Amman, with an installed capacity of 396 MW. The plant has been in operation since the mid-1970s and includes multiple steam-turbine units (3×33 MW and 4×66 MW) as well as two gas turbines (14 MW and 19 MW). The present case study focuses on a representative 66 MW steam unit. A schematic of the unit is shown in Figure 1 which includes a regenerative feedwater heating train comprising two high-pressure heaters (HPH1, HPH2), two low-pressure heaters (LPH4, LPH5), and a deaerator. Steam is generated and superheated to 793.15 K at 9.12 MPa before expansion in the turbine. The exhaust steam is condensed in an air-cooled condenser, and the condensate returns to the condensate return tank (CRT), completing the cycle and repeating continuously under steady operation.

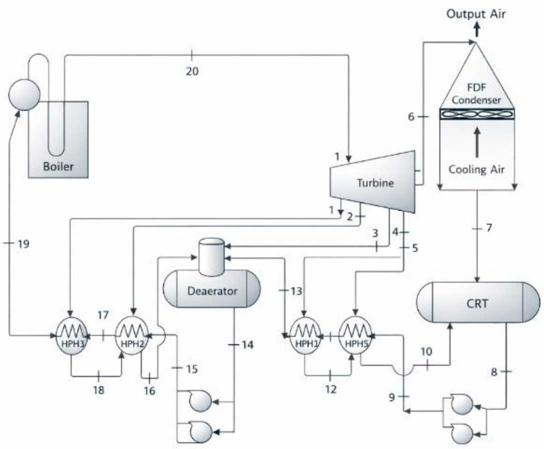

Figure 1: Schematic diagram of the power plant

Data collection and model inputs

Plant operating data used as direct inputs to the thermodynamic model are summarized in Table 1. These inputs include the fuel mass flow rate, steam conditions at turbine inlet, net power output, air-cooled condenser fan power and number of fans, cooling-air mass flow rate, and combined pump/motor efficiency. The calculations and thermodynamic property evaluations were implemented in EES using the same operating conditions adopted in the reference study.

Table 1: Provides the plant's operational conditions [20]

Operating condition	Value
Mass flow rate of fuel	5.0 kg/s
Inlet gas volumetric flow rate to burners	188,790 N m^3/h
Stack gas temperature	411.15 K
Feed water inlet temperature to boiler	494.15 K
Steam flow rate	275 ton/h
Steam temperature	793.15 K
Steam pressure	9.12 MPa
Power output	56 MW
Power input to FDC/fan	88 kW
Number of fans	18
Mass flow rate of cooling air	23,900 ton/h
Combined pump/motor efficiency	0.95

Heavy fuel oil is used as the primary energy source. In the present energy-exergy calculations, the fuel property required is the net heating value (NHV), which is 40504.58 kJ/kg. Other fuel characteristics (e.g., sulphur content, viscosity, flash point) are not used in the current energy-exergy balance framework and are therefore omitted from the model input set [21,22].

Mathematical model & assumptions

The thermodynamic analysis is conducted under steady-state operating conditions. Changes in kinetic and potential energies are neglected, and each component shown in Figure 1 is treated as a control volume. Mass, energy, and exergy balances are applied to all components to quantify (i) component-wise exergy destruction and (ii) component exergy efficiency. Thermodynamic properties of water/steam and air are evaluated using EES. The required inputs and state-point calculations are provided in Appendix B.

Fuel chemical exergy

The chemical exergy rate of the fuel is estimated using the exergy-grade-function method [2]:

$$\dot{X}_{fuel} = \dot{m}_{fuel} * NHV * \gamma_f \qquad (1)$$

where NHV is the net heating value and γ_f is the fuel exergy grade function. For petroleum-based liquid fuels, γ_f is commonly approximated as $\gamma_f \approx 1.06$. Accordingly, $\gamma_f = 1.06$ is adopted in this work, consistent with the reference methodology in [2].

Governing balances

For each component, the steady-state mass balance is:

$$\sum \dot{m}_{in} = \sum \dot{m}_{out} \qquad (2)$$

The steady-state energy balance is expressed as:

$$\dot{Q} - \dot{W} = \sum \dot{m}_{out} h_{out} - \sum \dot{m}_{in} h_{in} \qquad (3)$$

The corresponding exergy balance is written as:

$$\dot{X}_{heat} - \dot{W} = \sum \dot{m}_{out} \varphi_{out} - \sum \dot{m}_{in} \varphi_{in} + \dot{I} \qquad (4)$$

where $\dot{I}$ denotes the exergy destruction rate (irreversibility) of the component.

Exergy transfer with heat

The exergy transfer associated with heat interaction at boundary temperature T is:

$$\dot{X}_{heat} = \sum \left(1 - \frac{T_0}{T}\right) \dot{Q} \qquad (5)$$

where T_0 is the dead-state temperature.

Specific flow exergy

The specific flow exergy of steam/water streams (state points 1-20) is calculated as:

$$\varphi = (h - h_0) - T_0(s - s_0) \qquad (6)$$

where h_0 and s_0 are the specific enthalpy and entropy at the dead state (T_0, P_0).

For air streams (state points 21 and 22), assuming ideal-gas behaviour with constant c_p, the specific exergy is:

$$\varphi = c_p(T - T_0) - T_0[c_p \ln(\frac{T}{T_0}) - R\ln(\frac{P}{P_0})] \qquad (7)$$

Finally, the exergy rate associated with a fluid stream is:

$$\dot{X} = \dot{m}\varphi \qquad (8)$$

Component performance metrics

Using Eqs. (2-8), the exergy destruction rate $\dot{I}$ is computed for each component. Based on the obtained $\dot{I}$, component-wise exergy destruction percentages and component exergy efficiencies are evaluated and reported in Appendix A. For the overall power cycle, the exergy efficiency accounts for the irreversibility associated with heat transfer in the boiler as well as the exergy destruction due to fuel combustion and exhaust-gas interactions, consistent with the reference framework in [2].

Reference case

To validate the EES-based model and ensure reproducibility, a reference case is established using the operating conditions and reported results of the original paper [2]. The reference case corresponds to the Al-Hussein steam power plant under the same dead-state conditions adopted in [2], namely $T_0 = 298.15K$ and $P_0 = 101.3kPa$. All key input data required to run the model (e.g., mass flow rates, temperatures, pressures, and component performance parameters) are adopted directly from [2] and summarized in Table 1.

For comparison purposes, the component-wise heat-loss distribution and the exergy performance indicators (percentage exergy destruction and component exergy efficiency) reported in the original paper are used as baseline reference results. Figure 2 and 3 are replotted based on the data reported in [2] to provide a consistent visual comparison framework with the present EES results.

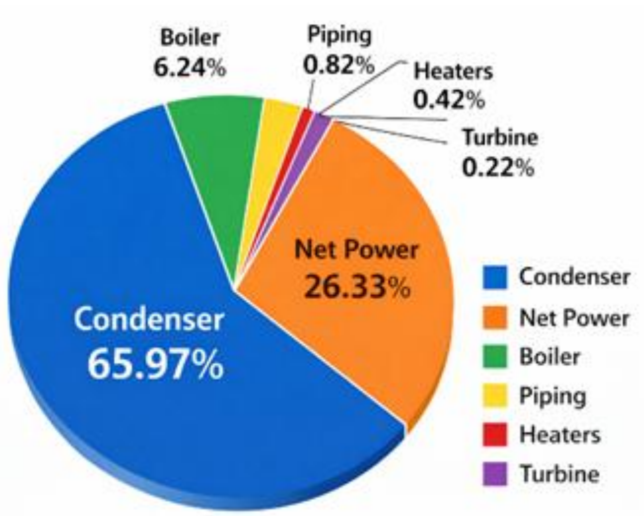

Figure 2: Component-wise heat-loss distribution of the Al-Hussein steam power plant. Replotted based on data reported in [2]

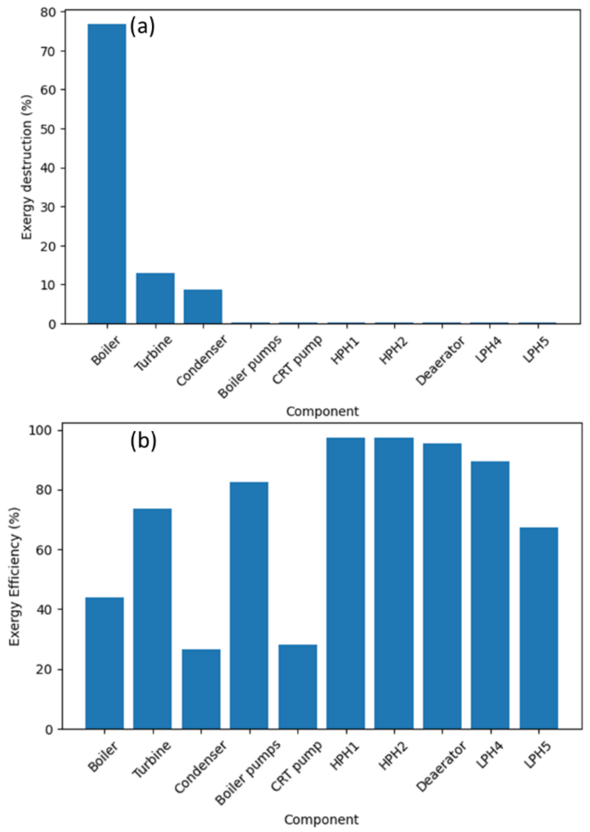

Figure 3: (a) Percentage exergy destruction and (b) exergy efficiency of the power plant components at dead state. Replotted based on data reported in [2]

Results & Discussion

The developed EES model was applied to the Al-Hussein steam power plant using the governing relations summarized in Appendix A. The reference (dead-state) conditions were set to $T_0 = 298.15\text{K}$ and $P_0 = 101.3\text{kPa}$, consistent with the original paper [2]. Thermodynamic properties for water/steam and air at the cycle state points shown in Figure 1were computed in EES and are reported in Appendix B. The resulting state-point properties and exergy rates are compiled in Table 3 and show close agreement with the values reported in the original paper [2], confirming that the present implementation reproduces the reference case with high consistency.

Energy balance and heat-loss distribution

The component-wise energy balance is summarized in Table 2 and visualized in Figure 4. The condenser represents the dominant heat-loss sink, followed by the net power output and boiler losses, while piping, heaters, and turbine losses contribute comparatively small fractions. Overall, the present energy-balance results remain in close agreement with those of the original paper [2], indicating that the first-law accounting of the cycle is correctly reproduced.

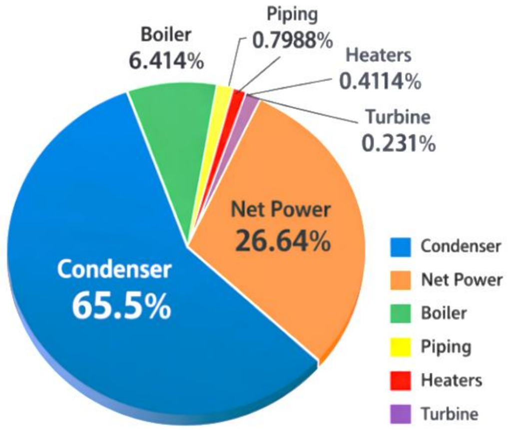

Figure 4: Component-wise heat-loss distribution of the Al-Hussein steam power plant

Table 2: Component-wise heat losses of the power plant based on the energy balance

Component	Heat loss (kW)
Condenser	133469
Net power	54287
Boiler	13069
Piping	1628
Heaters	838.2
Turbine	470.6
Total	**203761**

Exergy destruction and component exergy efficiency

Figure 5 presents (a) the percentage of the exergy destruction and (b) the exergy efficiency of the power plant components at the dead state. The results indicate that the boiler is the largest contributor to total exergy destruction, followed by the turbine and condenser, whereas the feedwater heaters and pumps have marginal contributions. In terms of component performance, the highest exergy efficiencies are obtained for HPH1 and HPH2, while the condenser exhibits the lowest exergy efficiency, reflecting the inherently irreversible nature of near-ambient heat rejection processes.

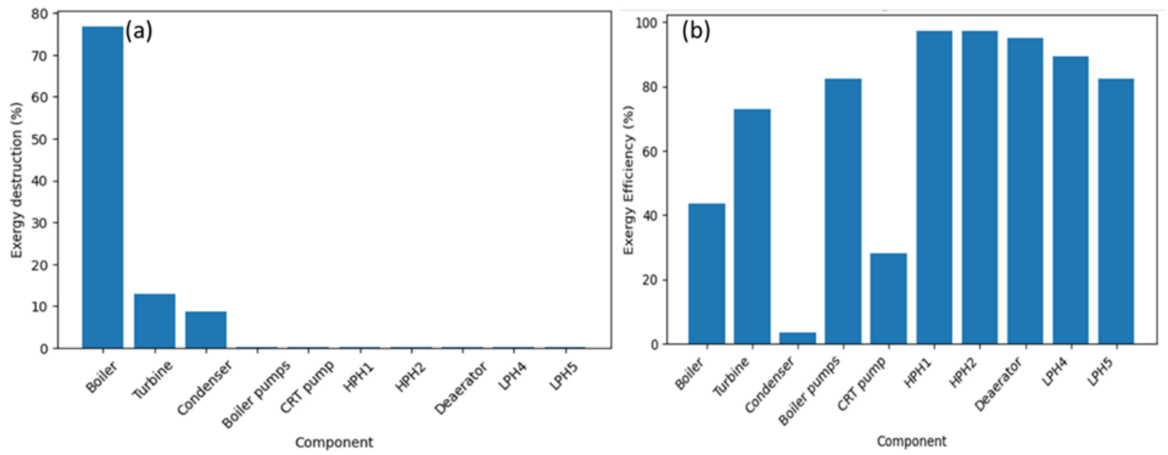

Figure 5: (a) Percentage of the exergy destruction and (b) exergy efficiency of the power plant components at dead state

Table 3: Exergy analysis of the power plant when $T_0 = 298.15$ K, $P_0 = 101.3$ kPa

State	T (K)	P (MPa)	$\dot{m}$ (kg/s)	h (kJ/kg)	s (kJ/kg.k)	φ (kJ/kg)	X (kW)
1	618.55	2.4231	4.94	3118.136	6.842	1083	5349
2	547.85	1.3244	4.14	2986.916	6.884	939.2	3888
3	463.65	0.5690	4.56	2831.432	6.951	763.5	3482
4	394.35	0.2060	3.88	2707.703	7.117	590.2	2290
5	360.45	0.0628	1.775	2655.193	7.517	418.6	743
6	343.15	0.0272	56.92	2626.909	7.819	300.2	17085
7	339.95	0.0272	56.92	279.656	0.9159	11.15	634.4
8	339.75	0.0270	62.78	278.818	0.9134	11.04	693.3
9	341.15	1.3734	62.78	285.789	0.9299	13.11	823.2
10	337.60	0.0245	5.86	269.813	0.8868	9.963	58.38
11	356.15	0.0536	62.78	347.611	1.111	20.9	1312
12	362.45	0.0687	3.87	374.095	1.185	25.42	98.38
13	390.15	0.1815	62.78	491.090	1.495	49.79	3126
14	428.15	0.6867	76.215	653.875	1.892	94.27	7185
15	430.15	12.2630	76.215	669.489	1.899	107.8	8218
16	436.15	0.6671	9.1	688.531	1.972	105	955.6
17	461.45	10.7910	76.215	804.434	2.206	151.4	11539
18	466.15	2.3544	4.94	821.281	2.263	151.3	747.2
19	494.15	10.3010	76.215	950.464	2.512	205.9	15696
20	793.15	9.1233	76.215	3436.251	6.717	1438	109612
Input air	298.15	0.1013	6638.9	298.586	5.696	0	0
Output air	318.15	0.1013	6638.9	318.690	5.761	0.6454	4285
Dead state	298.15	0.1013	-	104.920	0.3672	-	-

Overall, the obtained exergy results agree closely with those reported in the original paper [2], except for the condenser. The calculated condenser exergy destruction is 18,034 kW, compared to 13,738 kW in [2], corresponding to a deviation of 23.8%. This deviation also affects the condenser exergy efficiency. In addition, an approximately 18% deviation is observed in the exergy efficiency of LPH5, suggesting that low-pressure regenerative heater calculations are more sensitive to modeling assumptions and state-point mapping.

The larger discrepancy in LPH5 can be attributed to the sensitivity of low-pressure feedwater heaters to small variations in extraction-steam conditions, mass-flow splitting, and the treatment of drains/condensate mixing. Because LPH5 typically operates with small temperature driving forces, minor differences in assumed extraction fractions, terminal temperature difference (TTD), pressure drops, and saturation proximity may propagate into noticeable variations in enthalpy/entropy changes and, consequently, in stream exergy and exergy efficiency. These effects are amplified for exergy-based indicators compared with energy quantities, which explains why the overall energy balance remains consistent while LPH5 exhibits a larger deviation.

Sensitivity to dead-state temperature

To assess the sensitivity of the exergy metrics to the reference environment, T_0 was varied from 298.15 K to 318.15 K in 5 K increments while maintaining $P_0 = 101.3$kPa. The resulting parametric table is summarized in Table 4, reporting the total exergy rates at the state points defined in Figure 1**.** The trends remain consistent with those reported in the original paper [2], further supporting the robustness of the present calculations.

Table 4: Total exergy rate at different reference environment temperatures

X (kW)	T_0 (K)							
	283.15	**288.15**	**293.15**	**298.15**	**303.15**	**308.15**	**313.15**	**318.15**
1	5837	5672	5510	5349	5190	5033	4877	4723
2	4299	4161	4024	3888	3754	3621	3490	3360
3	3939	3785	3633	3482	3332	3185	3038	2894
4	2689	2555	2422	2290	2160	2031	1903	1777
5	936.2	871.2	806.8	743	679.9	617.4	555.4	494.1
6	23539	21367	19215	17085	14974	12882	10810	8757
7	1194	987.1	800.6	634.4	488.2	361.6	254.2	165.9
8	1308	1081	875.8	693.3	532.7	393.9	276.3	179.6
9	1454	1221	1011	823.2	657.5	513.5	390.7	288.9
10	113.5	92.99	74.64	58.38	44.18	31.99	21.8	13.56
11	2114	1824	1557	1312	1090	888.7	709.1	550.4
12	152.1	132.8	114.9	98.38	83.24	69.42	56.93	45.72
13	4289	3879	3491	3126	2783	2461	2161	1882
14	9050	8401	7779	7185	6617	6075	5559	5069
15	10092	9440	8815	8218	7647	7103	6585	6092
16	1189	1108	1030	955.6	884.1	815.8	750.6	688.4
17	13763	12994	12253	11539	10852	10191	9556	8946
18	895.6	844.3	794.9	747.2	701.2	657	614.4	573.5
19	18271	17385	16527	15696	14892	14114	13361	12635
20	116994	114506	112045	109612	107206	104826	102471	100143

As illustrated in Figure 6, the total exergy destruction rates of the major components (boiler, turbine, and condenser) follow trends that closely match the original paper [2].

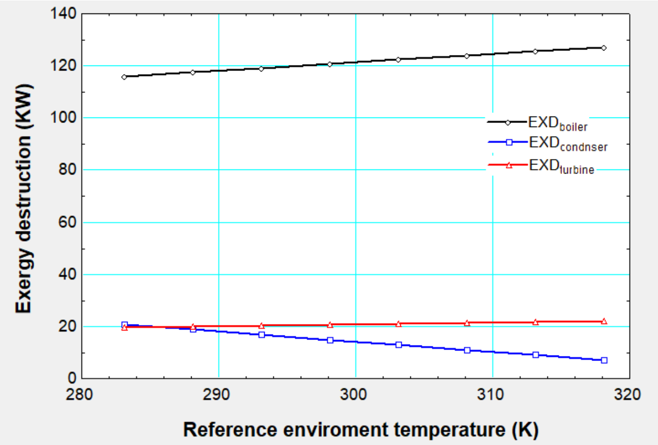

Figure 6: Effect of reference environment temperature on total exergy destruction rate in major plant components

Figure 7 shows the corresponding exergy efficiencies. While the boiler and turbine efficiencies remain nearly identical to [2], the condenser efficiency exhibits the same qualitative trend but lower absolute values. This behaviour is expected because condenser processes occur at temperatures close to the ambient reference condition; therefore, both the stream exergy (Eq. 6) and the heat-transfer exergy term (Eq. 5) depend explicitly on T_0. When $T \approx T_0$, the factor $(1 - T_0/T)$ becomes small, and small changes in T_0 can produce comparatively large relative variations in condenser exergy destruction and exergy efficiency, even though the condenser energy rejection remains large due to latent heat release during phase change. Hence, the observed condenser sensitivity is an inherent characteristic of second-law metrics for near-ambient heat-rejection processes.

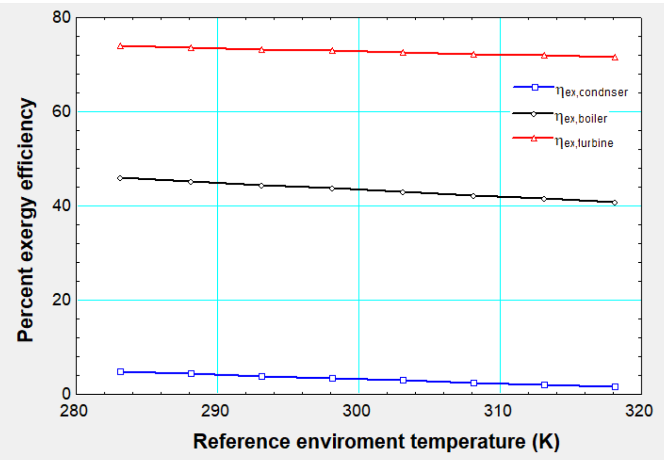

Figure 7: Effect of reference environment temperature on the exergy efficiency of major plant components

Conclusions

This study independently reproduced the energy and exergy assessment of the Al-Hussein steam power plant using EES under the same operating conditions and the same reference environment. The reproduced thermodynamic properties and the energy balance showed very close agreement with the original study, confirming the reliability of the energy-based results and validating the consistency of the implemented component energy balances.

The exergy results gave a clearer indication of where the irreversibilities are in the cycle and how the priorities for improvement should be established. The boiler continued to be the major contributor to exergy destruction, accounting for approximately 74.55% of the total exergy destruction, followed by the turbine with 12.83%, and then the condenser with 11.13%. This further confirms that the boiler, which relies on combustion and high-temperature heat transfer, is the major area of focus for improvement of the avoidable irreversibilities.

Although there was a general level of agreement, there were observable differences in certain component-level exergy values. The exergy destruction value of the condenser was determined to be 18,034 kW, which was a 23.8% difference from the original value of 13,738 kW, greatly influencing the condenser exergy efficiency. Furthermore, a difference of around 18% was also noticed in the calculated exergy efficiency of LPH5, which was sensitive to the small temperature driving force. The results of the parametric study also indicated that although the general exergy behaviour is not affected, the reference environment temperature has a significant influence on the condenser exergy values.

Recommendations For Future Work

Recommendations for future research include the development of the same steam cycle model using more sophisticated power plant simulators such as Thermoflow, Ebsilon Professional, IPSEpro, or Aspen HYSYS. Such simulators would offer better flowsheet diagrams and simulation capabilities.

Conducting energy, exergy, economic, and environmental analysis of steam power plants when replacing conventional fuel with solar fuel, as many local and regional studies have confirmed the feasibility of integrating solar concentrators as solar fuel instead of conventional fuel in power generation plants [23-26].

A focused sensitivity analysis needs to be performed for the condenser, as it had the maximum deviation in the exergy destruction and exergy efficiency. The sensitivity analysis needs to investigate the boundary conditions of the condenser and the consideration of exergy equations for the near-ambient heat transfer process.

The condenser analysis should also explicitly examine the impact of representation of auxiliary power (for example, fan power) and the definition of state points around saturation. These details can help to decrease uncertainty in condenser-related exergy ratio results.

For LPH5, the future research should focus on improving the modelling of assumptions in the extraction steam, such as mass splitting, drains/mixing treatment, and terminal temperature difference constraints. At the same time, uncertainty propagation can be considered to analyse the impact of small input changes on the exergy efficiency results.

Finally, the results of the study can be further developed with the help of advanced exergy analysis and optimization methods to estimate the avoidable improvement potential. AI-assisted surrogate modelling or sensitivity ranking can also be considered to improve the robustness of the calibration process with the help of real operational data.

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curation. **Mohammed and Yasser:** Formal analysis, Results interpretation, writing review and editing, Supervision. All authors have read and agreed to the published version of the manuscript.

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Appendix A

Energy and Exergy Calculations.
The specific exergy of the steam (φ) at (state 1) is:
$$\varphi_1 = (h_1 - h_0) - T_0(s_1 - s_0)$$
The total exergy rate at (state 1) is:
$$\dot{X}_1 = \dot{m}_1\varphi_1$$
The specific exergy (φ) at (state 2) is:
$$\varphi_2 = (h_2 - h_0) - T_0(s_2 - s_0)$$
The total exergy rate at (state 2) is:
$$\dot{X}_2 = \dot{m}_2\varphi_2$$
. .
. .
. .
. .
The specific exergy of the steam (φ) at (state 20) is:
$$\varphi_{20} = (h_{20} - h_0) - T_0(s_{20} - s_0)$$
The total exergy rate at (state 20) is:
$$\dot{X}_{20} = \dot{m}_{20}\varphi_{20}$$
The specific exergy of the input air (φ) at (state 21) is:
$$\varphi_{21} = c_p(T_{21} - T_0) - T_0\left(c_p Ln\,\frac{T_{21}}{T_0} - R * Ln\,\frac{P_{21}}{P_0}\right)$$
The total exergy rate at (state 21) is:
$$\dot{X}_{21} = \dot{m}_{21}\varphi_{21}$$
The specific exergy of the output air (φ) at (state 22) is:
$$\varphi_{22} = c_p(T_{22} - T_0) - T_0\left(c_p Ln\,\frac{T_{22}}{T_0} - R * Ln\,\frac{P_{22}}{P_0}\right)$$
The total exergy rate at (state 22) is:
$$\dot{X}_{22} = \dot{m}_{22}\varphi_{22}$$
$$\dot{X}_{fuel} = \dot{m}_{fuel} * NHV * \gamma_f$$

Heat Losses
$$\dot{Q}_{Condenser} = \dot{m}_{21}(h_{22} - h_{21})$$
$$\dot{Q}_{Boiler} = (\dot{m}_{fuel} * NHV * \gamma_f) - \dot{m}_{19}(h_{20} - h_{19})$$
$$\dot{Q}_{Net\ power} = (P_{output}) - \left(\dot{W}_{CRT\ pump} - \dot{W}_{Boiler\ pump}\right)$$
$$\dot{Q}_{Piping} = \dot{m}_8(h_9 - h_8) + \dot{m}_{14}(h_{15} - h_{14})$$
$$\dot{Q}_{Heaters} = (\dot{m}_1 * h_1 + \dot{m}_{17} * h_{17} - \dot{m}_{18} * h_{18} - \dot{m}_{19} * h_{19}) + (\dot{m}_2 * h_2 + \dot{m}_{18} * h_{18} + \dot{m}_{15} * h_{15} - \dot{m}_{17} * h_{17} - \dot{m}_{16} * h_{16}) + (\dot{m}_4 * h_4 + \dot{m}_{11} * h_{11} - \dot{m}_{12} * h_{12} - \dot{m}_{13} * h_{13}) + (\dot{m}_5 * h_5 + \dot{m}_9 * h_9 + \dot{m}_{12} * h_{12} - \dot{m}_1 * h_{11} - \dot{m}_{10} * h_{10})$$
$$\dot{Q}_{Turbine} = \dot{m}_{20}h_{20} - \left(\dot{m}_1h_1 + \dot{m}_2h_2 + \dot{m}_3h_3 + \dot{m}_4h_4 + \dot{m}_5h_5 + \dot{m}_6h_6 + P_{output}\right)$$
$$\dot{Q}_{Total} = \dot{Q}_{Condenser} + \dot{Q}_{Boiler} + \dot{Q}_{Net\ power} + \dot{Q}_{Piping} + \dot{Q}_{Heaters} + \dot{Q}_{Turbine}$$

Percent Ratio Heat Losses
$$R_{condencer} = \frac{\dot{Q}_{Condenser}}{\dot{Q}_{Total}} * 100$$
$$R_{Net\ power} = \frac{\dot{Q}_{Net\ power}}{\dot{Q}_{Total}} * 100$$
$$R_{Boiler} = \frac{\dot{Q}_{Boiler}}{\dot{Q}_{Total}} * 100$$
$$R_{Piping} = \frac{\dot{Q}_{Piping}}{\dot{Q}_{Total}} * 100$$
$$R_{Heaters} = \frac{\dot{Q}_{Heaters}}{\dot{Q}_{Total}} * 100$$
$$R_{Turbine} = \frac{\dot{Q}_{Turbine}}{\dot{Q}_{Total}} * 100$$

Exergy Destruction
$$\dot{I}_{Condenser} = \dot{X}_6 - \dot{X}_7 + (18 * \dot{W}_{fan})$$
$$\dot{I}_{Boiler} = \dot{X}_{fuel} + \dot{X}_{19} - \dot{X}_{20}$$
$$\dot{I}_{Boiler\ pump} = \dot{X}_{14} - \dot{X}_{15} + (\dot{m}_{14}(h_{15} - h_{14}))/\eta_{Pump}$$
$$\dot{I}_{CRT\ pump} = \dot{X}_8 - \dot{X}_9 + (\dot{m}_8(h_9 - h_8))/\eta_{Pump}$$
$$\dot{I}_{Deaerator} = \dot{X}_3 + \dot{X}_{16} + \dot{X}_{13} - \dot{X}_{14}$$
$$\dot{I}_{Turbine} = \dot{X}_{20} - \dot{X}_1 - \dot{X}_2 - \dot{X}_3 - \dot{X}_4 - \dot{X}_5 - \dot{X}_6 - P_{output}$$
$$\dot{I}_{HPH1} = \dot{X}_1 + \dot{X}_{17} - \dot{X}_{18} - \dot{X}_{19}$$
$$\dot{I}_{HPH2} = \dot{X}_2 + \dot{X}_{15} + \dot{X}_{18} - \dot{X}_{16} - \dot{X}_{17}$$
$$\dot{I}_{LPH4} = \dot{X}_4 + \dot{X}_{11} - \dot{X}_{12} - \dot{X}_{13}$$
$$\dot{I}_{LPH5} = \dot{X}_5 + \dot{X}_9 + \dot{X}_{12} - \dot{X}_{11} - \dot{X}_{10}$$
$$\dot{I}_{Cycle} = \dot{I}_{Condenser} + \dot{I}_{Boiler} + \dot{I}_{Boiler\ pump} + \dot{I}_{CRT\ pump} + \dot{I}_{Deaerator} + \dot{I}_{Turbine} + \dot{I}_{HPH1} + \dot{I}_{HPH2} + \dot{I}_{LPH4} + \dot{I}_{LPH5}$$

Percent Exergy Destruction
$$R_{Boiler} = \frac{\dot{I}_{Boiler}}{\dot{I}_{Cycle}} * 100$$
$$R_{Condenser} = \frac{\dot{I}_{Condenser}}{\dot{I}_{Cycle}} * 100$$
$$R_{Boiler\ pump} = \frac{\dot{I}_{Boiler\ pump}}{\dot{I}_{Cycle}} * 100$$
$$R_{CRT\ pump} = \frac{\dot{I}_{CRT\ pump}}{\dot{I}_{Cycle}} * 100$$
$$R_{Deaerator} = \frac{\dot{I}_{Deaerator}}{\dot{I}_{Cycle}} * 100$$
$$R_{Turbine} = \frac{\dot{I}_{Turbine}}{\dot{I}_{Cycle}} * 100$$
$$R_{HPH1} = \frac{\dot{I}_{HPH1}}{\dot{I}_{Cycle}} * 100$$
$$R_{HPH2} = \frac{\dot{I}_{HPH2}}{\dot{I}_{Cycle}} * 100$$
$$R_{LPH4} = \frac{\dot{I}_{LPH4}}{\dot{I}_{Cycle}} * 100$$
$$R_{LPH5} = \frac{\dot{I}_{LPH5}}{\dot{I}_{Cycle}} * 100$$

Exergy Efficiency
$$\eta_{ex,\ boiler} = \frac{\dot{X}_{20} - \dot{X}_{19}}{\dot{X}_{fuel}} * 100$$
$$\dot{W}_{CRT\ pump} = (\dot{m}_8(h_9 - h_8))/\ \eta_{Pump}$$
$$\eta_{ex,\ CRT\ pump} = 1 - \frac{\dot{I}_{CRT\ pump}}{\dot{W}_{CRT\ pump}}$$
$$\dot{W}_{Boiler\ pump} = (\dot{m}_{14}(h_{15} - h_{14}))/\ \eta_{Pump}$$
$$\eta_{ex,\ Boiler\ pump} = 1 - \frac{\dot{I}_{Boiler\ pump}}{\dot{W}_{Boiler\ pump}}$$
$$\eta_{ex,\ Deaerator} = 1 - \frac{\dot{I}_{Deaerator}}{\dot{X}_3 + \dot{X}_{16} + \dot{X}_{13}}$$
$$\eta_{ex,\ HPH1} = 1 - \frac{\dot{I}_{HPH1}}{\dot{X}_1 + \dot{X}_{17}}$$
$$\eta_{ex,\ HPH2} = 1 - \frac{\dot{I}_{HPH2}}{\dot{X}_2 + \dot{X}_{18} + \dot{X}_{15}}$$
$$\eta_{ex,\ LPH4} = 1 - \frac{\dot{I}_{LPH4}}{\dot{X}_4 + \dot{X}_{11}}$$
$$\eta_{ex,\ LPH5} = 1 - \frac{\dot{I}_{LPH5}}{\dot{X}_5 + \dot{X}_9 + \dot{X}_{12}}$$
$$\eta_{ex,\ Turbine} = 1 - \frac{\dot{I}_{Turbine}}{\dot{X}_{20} - \dot{X}_1 - \dot{X}_2 - \dot{X}_3 - \dot{X}_4 - \dot{X}_5 - \dot{X}_6}$$
$$\eta_{ex,\ Condenser} = 1 - \frac{\dot{X}_7}{\dot{X}_6 + (18 * \dot{W}_{fan})}$$
$$\eta_{ex,\ Cycle} = \frac{P_{output}}{\dot{X}_{fuel}}$$

Appendix B

EES Code for Detailed Tables of Thermodynamic Properties and Energy and Exergy Calculations

```
"Input Data Table"

m_fuel= 5"  kg/s"
NHV=40504.58" KJ/Kg"
P_output=56000" KW"
P_input= 88"  KW"
N_fans=18
m_cooling= 6638.9"  kg/s"
eta_pump=0.95

" States"

T[1] = 618.55"  K"
P[1] = 2.4231"  MPa"
m_dot[1] = 4.94"  kg/s"
h[1]= enthalpy(Steam,T=T[1],P=P[1])
s[1]=entropy(Steam,T=T[1],P=P[1])

T[2] = 547.85"  K"
P[2] = 1.3244"  MPa"
m_dot[2] = 4.14"   kg/s"
h[2]= enthalpy(Steam,T=T[2],P=P[2])
s[2]=entropy(Steam,T=T[2],P=P[2])

T[3] = 463.65"  K"
P[3] = 0.5690"  MPa"
m_dot[3] = 4.56"   kg/s"
h[3]= enthalpy(Steam,T=T[3],P=P[3])
s[3]=entropy(Steam,T=T[3],P=P[3])

T[4] = 394.35"  K"
P[4] = 0.2060"  MPa"
m_dot[4] = 3.88"   kg/s"
h[4]= enthalpy(Steam,T=T[4],P=P[4])
s[4]=entropy(Steam,T=T[4],P=P[4])

T[5] = 360.45"  K"
P[5] = 0.0628"  MPa"
m_dot[5] = 1.775"  kg/s"
h[5]= enthalpy(Steam,T=T[5],P=P[5])
s[5]=entropy(Steam,T=T[5],P=P[5])

T[6] = 343.15"  K"
P[6] = 0.0272"  MPa"
m_dot[6] = 56.92"   kg/s"
h[6]= enthalpy(Steam,T=T[6],P=P[6])
s[6]=entropy(Steam,T=T[6],P=P[6])

 T[7] = 339.95"  K"
P[7] = 0.0272"  MPa"
m_dot[7] = 56.92"   kg/s"
 h[7]= enthalpy(Steam,T=T[7],P=P[7])
s[7]=entropy(Steam,T=T[7],P=P[7])

 T[8] = 339.75"  K"
P[8] = 0.0270"  MPa"
m_dot[8] = 62.78"   kg/s"
 h[8]= enthalpy(Steam,T=T[8],P=P[8])
s[8]=entropy(Steam,T=T[8],P=P[8])

 T[9] = 341.15"  K"
P[9] = 1.3734"  MPa"
m_dot[9] = 62.78"  kg/s"
 h[9]= enthalpy(Steam,T=T[9],P=P[9])
s[9]=entropy(Steam,T=T[9],P=P[9])

 T[10] = 337.60"  K"
P[10] = 0.0245"  MPa"
m_dot[10] = 5.86"   kg/s"
 h[10]= enthalpy(Steam,T=T[10],P=P[10])
s[10]=entropy(Steam,T=T[10],P=P[10])

T[11] = 356.15"  K"
P[11] = 0.0536"  MPa"
m_dot[11] = 62.78"   kg/s"
 h[11]= enthalpy(Steam,T=T[11],P=P[11])
s[11]=entropy(Steam,T=T[11],P=P[11])

T[12] = 362.45"  K"
P[12] = 0.0687"  MPa"
m_dot[12] = 3.87"   kg/s"
 h[12]= enthalpy(Steam,T=T[12],P=P[12])
s[12]=entropy(Steam,T=T[12],P=P[12])

T[13] = 390.15"  K"
P[13] = 0.1815"  MPa"
m_dot[13] = 62.78"  kg/s"
 h[13]= enthalpy(Steam,T=T[13],P=P[13])
s[13]=entropy(Steam,T=T[13],P=P[13])

T[14] = 428.15"  K"
P[14] = 0.6867"  MPa"
m_dot[14] = 76.215"   kg/s"
h[14]= enthalpy(Steam,T=T[14],P=P[14])
s[14]=entropy(Steam,T=T[14],P=P[14])

T[15] = 430.15"  K"
P[15] = 12.2630"  MPa"
m_dot[15] = 76.215"   kg/s"
h[15]= enthalpy(Steam,T=T[15],P=P[15])
s[15]=entropy(Steam,T=T[15],P=P[15])

T[16] = 436.15"  K"
P[16] = 0.6671"  MPa"
m_dot[16] = 9.1"   kg/s"
h[16]= enthalpy(Steam,T=T[16],P=P[16])
s[16]=entropy(Steam,T=T[16],P=P[16])

T[17] = 461.45"  K"
P[17] = 10.7910"  MPa"
m_dot[17] = 76.215"   kg/s"
h[17]= enthalpy(Steam,T=T[17],P=P[17])
s[17]=entropy(Steam,T=T[17],P=P[17])

T[18] = 466.15"  K"
P[18] = 2.3544"  MPa"
m_dot[18] = 4.94"   kg/s"
h[18]= enthalpy(Steam,T=T[18],P=P[18])
s[18]=entropy(Steam,T=T[18],P=P[18])

T[19] = 494.15"  K"
P[19] = 10.3010"  MPa"
```

```
m_dot[19] = 76.215"   kg/s"
h[19]= enthalpy(Steam,T=T[19],P=P[19])
s[19]=entropy(Steam,T=T[19],P=P[19])

T[20] = 793.15"  K"
P[20] = 9.1233"  MPa"
m_dot[20] = 76.215"   kg/s"
h[20]= enthalpy(Steam,T=T[20],P=P[20])
s[20]=entropy(Steam,T=T[20],P=P[20])

T[21] = 298.15" K"
P[21] = 0.1013" MPa"
m_dot[21]= 6638.9"  kg/s"
s[21]=entropy(Air,T=T[21],P=P[21])
h[21]=enthalpy(Air,T=T[21])

T[22]= 318.15" K"
P[22] = 0.1013" MPa"
m_dot[22]= 6638.9"   kg/s"
s[22]=entropy(Air,T=T[22],P=P[22])
h[22]=enthalpy(Air,T=T[22])

T[0] = 298.15"   K"
P[0] = 0.1013" MPa"
h[0]=enthalpy(Steam,T=T[0], P=P[0])
s[0]=entropy(Steam,T=T[0],P=P[0])

Duplicate   i=1,20

    ex[i] = (h[i] - h[0]) - T[0] * (s[i] - s[0])
    E[i]=m_dot[i]*ex[i]

 End

ex[21] = 1.005*(T[21] - T[0]) - (T[21] *(1.005*(ln(T[21]/T[0])+0.287*ln(P[21]/P[0]))))
    E[21]=m_dot[21]*ex[21]


 ex[22] = 1.005*(T[22] - T[0]) - (T[0] *(1.005*(ln(T[22]/T[0])+0.287*ln(P[22]/P[0]))))
    E[22]=m_dot[22]*ex[22]

"Heat losses"

Q_condenser= m_cooling*(h[22]-h[21])

Q_boiler= m_fuel*NHV+m_dot[19]*h[19]- m_dot[20]*h[20]

Q_net_power= P_output-(W__CRT_pump+W_boiler_pump)

Q_piping= (m_dot[8]*(h[9]-h[8]))+(m_dot[14]*(h[15]-h[14]))

Q_heaters=(m_dot[1]*h[1]+m_dot[17]*h[17]-m_dot[18]*h[18]-m_dot[19]*h[19])+(m_dot[2]*h[2]+m_dot[18]*h[18]+m_dot[15]*h[15]-m_dot[17]*h[17]-m_dot[16]*h[16])+(m_dot[4]*h[4]+m_dot[11]*h[11]-m_dot[12]*h[12]-m_dot[13]*h[13])+(m_dot[5]*h[5]+m_dot[9]*h[9]+m_dot[12*[h[12]-m_dot[11]*h[11]-m_dot[10]*h[10([

Q_turbine=(m_dot[20]*h[20])-(m_dot[1]*h[1]+m_dot[2]*h[2]+m_dot[3]*h[3]+m_dot[4]*h[4]+m_dot[5]*h[5]+m_dot[6]*h[6]+P_output)

 Q_total = Q_condenser + Q_net_power + Q_boiler + Q_piping + Q_heaters + Q_turbine

"Percent ratio heat losses"

Ratio_condenser = (Q_condenser / Q_total) * 100
Ratio_net_power = (Q_net_power / Q_total) * 100
Ratio_boiler = (Q_boiler / Q_total) * 100
Ratio_piping = (Q_piping / Q_total) * 100
Ratio_heaters = (Q_heaters / Q_total) * 100
Ratio_turbine = (Q_turbine / Q_total) * 100

"Exergy Destruction"

EXD_deaerator=E[16]+E[3]+E[13]-E[14]

EXD_boiler= E[19]-E[20]+(m_fuel*NHV*1.06)

EXD_turbine= E[20]-E[1]-E[2]-E[3]-E[4]-E[5]-E[6]-P_output

EXD_condnser= E[6]-E[7]+(N_fans*P_input)

EXD_CRT_pump=E[8]-E[9]+(m_dot[8]*(h[9]-h[8]))/eta_pump

EXD_boiler_pump=E[14]-E[15]+(m_dot[14]*(h[15]-h[14]))/eta_pump

EXD_HPH1=E[1]+E[17]-E[18]-E[19]

EXD_HPH2=E[2]+E[18]+E[15]-E[17]-E[16]

EXD_LPH4=E[4]+E[11]-E[12]-E[13]

EXD_LPH5=E[5]+E[9]+E[12]-E[11]-E[10]

EXD_cyle=EXD_deaerator+EXD_boiler+EXD_turbine+EXD_condnser+EXD_CRT_pump+EXD_boiler_pump+EXD_HPH1+EXD_HPH2+EXD_LPH4+EXD_LPH5

"Percent Exergy Destruction"

Precent_destruction_deaerator= (EXD_deaerator/EXD_cyle)*100
Precent_destruction_boiler= (EXD_boiler/EXD_cyle)*100
Precent_destruction_turbine= (EXD_turbine/EXD_cyle)*100
Precent_destruction_condnser= (EXD_condnser/EXD_cyle)*100
Precent_destruction_CRTpump= (EXD_CRT_pump/EXD_cyle)*100
Precent_destruction_boilerpump= (EXD_boiler_pump/EXD_cyle)*100
Precent_destruction_HPH1= (EXD_HPH1/EXD_cyle)*100
Precent_destruction_HPH2= (EXD_HPH2/EXD_cyle)*100
Precent_destruction_LPH4= (EXD_LPH4/EXD_cyle)*100
Precent_destruction_LPH5= (EXD_LPH5/EXD_cyle)*100
Precent_destruction_cycle= (EXD_cyle/EXD_cyle)*100

"Exergy Efficiency"
```

```
eta_ex_boiler=(E[20]-E[19])/(m_fuel*NHV*1.06)*100

W__CRT_pump=m_dot[8]*(h[9]-h[8])/eta_pump

eta_ex_CRT_pump=(1-EXD_CRT_pump/W__CRT_pump)*100

W_boiler_pump=m_dot[14]*(h[15]-h[14])/eta_pump

eta_ex_boiler_pump=(1-EXD_boiler_pump/W_boiler_pump)*100

eta_ex_deaerator=(1-EXD_deaerator/(E[16]+E[3]+E[13]))*100

eta_ex_HPH1=(1-EXD_HPH1/(E[1]+E[17]))*100

eta_ex_HPH2=(1-EXD_HPH2/(E[2]+E[18]+E[15]))*100

eta_ex_LPH4=(1-EXD_LPH4/(E[4]+E[11]))*100

eta_ex_LPH5=(1-EXD_LPH5/(E[5]+E[9]+E[12]))*100

eta_ex_turbine=(1-EXD_turbine/(E[20]-E[1]-E[2]-E[3]-E[4]-E[5]-E[6]))*100

eta_ex_condnser=(E[7]/(E[6]+(N_fans*P_input)))*100

eta_ex_Cycle=(P_output/(m_fuel*NHV*1.06))*100
```