Energy and exergy analyses of a supercritical power plant

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Abstract: Energy and exergy efficiencies of a supercritical power plant have been studied in this paper. The effect of ambient weather condition was considered on the condenser pressure. It was shown that high ambient temperature has more adverse effect on the exergy efficiency than the energy efficiency. As ambient temperature increases, the exergy efficiency of the boiler, condenser, heaters and feed water pump decrease, while the exergy efficiency of the turbine improves slightly. The analysis showed that exergy efficiency of the supercritical boiler is considerably higher than the conventional boiler but it is still the main source of total irreversibility.

Keywords: supercritical power plant; irreversibility; exergy efficiency; condenser pressure.

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1 Introduction

Improving the performance of power plant is a never ending subject. In addition to energy analysis which elaborates on the quantity of efficiency, exergy analysis has been used to indicate the performance quality of the power plant. In exergy analysis, the second law of thermodynamics is used for furthering the goal of more efficient energy-resource use. Exergy analysis enables to accurately identify the locations, types and magnitudes of wastes and losses to determine the meaningful efficiency (Dincer and Rosen, 2007). Summaries of the evolution of exergy analysis are provided by Kotas.
In recent years, many researchers have used exergy analysis in the residential and industrial processes (Wall et al., 1994; Rosen et al., 2004; Pope, 2005; Utlu et al. 2006; Saidur et al., 2006; Saidur et al., 2007a, 2007b; Xi et al., 2009; Al-Ghandoor et al., 2010).

Dincer and Al-Muslim (2001) studied the way that energy and exergy efficiencies of a thermal power plant are affected by parameters such as the boiler pressure and temperature and the ratio of mass flow rate of the steam leaving the turbine for feed water heaters.

Kopac and Hilalci (2007) studied the effect of ambient temperature on the exergy efficiency of Catalagzi power plant in Turkey. They found that the highest exergy losses take place in the boiler, while the highest energy losses occur in the condenser. They also found that an increase in ambient temperature decrease the exergy efficiency of all power plant components except the condenser. In their analysis they assumed a constant condenser pressure at different ambient temperatures which is not consistent with actual situation.

Kanoglu et al. (2007) discussed various definitions of energy and exergy efficiencies in the most conventional power cycles and showed that these terms are frequently misused or misunderstood. They suggested using a careful definition of energy and exergy efficiencies before using them for designing, analysing and optimising any thermal systems. Rosen and Tang (2008) investigated the effect of changing excess air and stack gas temperature on the energy and exergy of a coal-fired power plant. They found that the performance increases by decreasing the fraction of excess air in the boiler and also reducing the stack gas temperature. Aljundi (2009) studied energy and exergy analysis of Al-Hossien power plant in Jordan and showed that maximum exergy destruction occurs in the boiler (77%) followed by the turbine (13%). He also discussed the effect of varying the reference environment state on the exergy analysis and found that for moderate change in the reference state, no drastic change in the performance of the major components are realised.

Saidur et al. (2010) applied the energy and exergy approaches to analyse an industrial boiler. They calculated the energy and exergy efficiencies of the boiler and found that the combustion phenomenon contributes the biggest amount of exergy destruction (65%) followed by the heat transfer (35%). They offered several energy saving measures such as using of variable speed drive in boilers fan and heat recovery from flue gas to increase the efficiencies. Regulagadda et al. (2010) studied thermodynamic analysis of a subcritical boiler–turbine generator in a 32 MW coal-fired power plant. They conducted a parametric study on the plant under various operating conditions, including different operating pressures, temperatures and flow rates, in order to determine the parameters that maximise plant performance. They found that boiler and turbine irreversibility yield the highest exergy losses in the power plant.

Reviewing studies on the exergy analysis of power plant shows that despite existence of many publications in this area, most of them used simplifying assumptions in the analysis which may not be consistent with actual conditions. For example, most researchers (Kopac and Hilalci, 2007; Aljundi, 2009) used a constant pressure assumption in the condenser even though ambient temperature varied considerably. The other deficiency in the literature is lack of investigation on the exergy analysis of a supercritical power plant, despite the fact that about one-third of fossil-fueled power plants in the world are working in supercritical cycle. In supercritical power plant,
once-through boiler is used instead of conventional steam-drum boiler. In this type of boiler the working fluid is heated directly from liquid state to supercritical state by bypassing the saturation region. This process allows a closer temperature match between vapour and gaseous product, resulting in less exergy destruction due to heat transfer irreversibility. Since the boiler is the main source of exergy destruction in the Rankine cycle; therefore, it is important to investigate the exergy efficiency of boiler in this situation. Supercritical power plant has higher efficiency than subcritical one but its exergy analysis has not been addressed yet. In this paper, an attempt was taken to analyse the effect of ambient temperature and relative humidity on the irreversibility rate and exergy efficiency of different components of a supercritical power plant while the change in the condenser pressure was taken into account. This helps to find a better understanding of supercritical power plant performance in the actual situation and its difference with subcritical one.

2 Exergy analysis

There is an increasing interest in using exergy analysis for different components of a thermal system since it can provide a better understanding of the process and quantifies the sources of inefficiency in each component. In the exergy analysis, complete equilibrium of the system with environment including the chemical and thermal equilibriums is considered. Exergy lost or irreversibility of every single component in the cycle is determined and the total irreversibility of the cycle can accordingly be calculated. The exergy balance in a system in contact with \( n \) heat sources which has a net generated work equal to \( W_{\text{gen}} \) and has multiple inlets and outlets is represented as follows (Bejan, 1995):

\[
\dot{E}_{x,w} = \sum_{j=1}^{n} \left( 1 - \frac{T_j}{T_i} \right) \dot{Q}_j + \sum_{\text{in}} \dot{m}e_{x,i} - \sum_{\text{out}} \dot{m}e_{x,o} - T_0 \dot{S}_{\text{gen}}. \tag{1}
\]

Flow exergy is generally divided into thermo-mechanical and chemical exergies which can be shown by the following equation:

\[
\dot{E}_x = \dot{E}^{\text{tm}}_x + \dot{E}^{\text{ch}}_x. \tag{2}
\]

Thermo-mechanical exergy includes kinetic, potential and physical exergies which can be represented as follows:

\[
\dot{E}^{\text{tm}}_x = \dot{E}^{\text{ke}}_x + \dot{E}^{\text{po}}_x + \dot{E}^{\text{ph}}_x. \tag{3}
\]

Physical exergy of the flow is calculated from the following relation (Cengel and Boles, 1994):

\[
e_{x, ph} = (h - h_0) - T_0(s - s_0). \tag{4}
\]

The standard chemical molar exergy of the fuel constituent parts \( (\tau_{x,i}^c) \) can be found in thermodynamic tables (Kotas, 1985). The molar chemical exergy of gas mixture is found from the following relation (Moran and Shapiro, 2000):
where \( y_i \) is the molar ratio of the fuel constituent part. The molar chemical exergy of the combustion gases is obtained from the following relation (Moran and Shapiro, 2000):

\[
\bar{e}_{\text{ch,combustion}} = R T_0 \sum_{i} X_i L n \left[ \frac{y_i}{y'_i} \right] 
\]

where \( y'_i \) is the molar ratio of the environment elements and \( X_i \) represents the unknown coefficients calculated in the combustion process.

### 2.1 Modelling and simulation of power plant components

In order to consider actual data for the analysis, an existing supercritical power plant (Ramin Power Plant in Ahvaz, Iran) working at very hot region in south of Iran was chosen for the analysis. It has a once-through supercritical boiler, three high, medium and low pressure turbines, seven close-type feed water heaters and one open-type feed water heater and generate net power of 315 MW.

Figure 1 shows a schematic diagram of the plant and Table 1 shows its operating values. Fuel of boiler is natural gas including \( CH_4, C_2H_6, C_3H_8, C_4H_{10}, ISO-C_4H_{10}, n-C_4H_{10}, ISO-C_5H_11, n-C_5H_{12}, CO_2, N_2 \).
Table 1  Operating values of the power plant

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Gas volume flow rate</td>
<td>69,000 m³/s</td>
</tr>
<tr>
<td>Gas lower heat value</td>
<td>48,500 kJ/kg</td>
</tr>
<tr>
<td>Air volume flow rate</td>
<td>690,000 m³/s</td>
</tr>
<tr>
<td>Maximum gas temperature in boiler</td>
<td>2000 ºC</td>
</tr>
<tr>
<td>Gas exit temperature from boiler</td>
<td>137 ºC</td>
</tr>
<tr>
<td>Feed water mass flow rate</td>
<td>1017.5 ton/hr</td>
</tr>
<tr>
<td>Feed water pressure</td>
<td>24 MPa</td>
</tr>
<tr>
<td>Feed water inlet temperature</td>
<td>282.5 kg/s</td>
</tr>
<tr>
<td>Steam temperature</td>
<td>540 ºC</td>
</tr>
<tr>
<td>Extraction steam pressure</td>
<td>4.48 MPa</td>
</tr>
<tr>
<td>Extraction steam temperature</td>
<td>303 ºC</td>
</tr>
<tr>
<td>Extraction steam mass flow</td>
<td>255.8 kg/s</td>
</tr>
<tr>
<td>Reheated steam temperature</td>
<td>540 ºC</td>
</tr>
<tr>
<td>Cooling water mass flow rate</td>
<td>10000 kg/s</td>
</tr>
</tbody>
</table>

The energy and exergy analysis of the cycle has been made using the 'EES' software. The combustion process is assumed to be complete (Jamil, 1994) as follows:

\[
\sum_{i=1}^{n} f_i C_i, H_{\text{in}} + a f_i \left( n_i + \frac{m_i}{4} \right) O_2 + 3.76 a f_i \left( n_i + \frac{m_i}{4} \right) N_2 \\
+ c CO_2 + d N_2 + n_{v,a} + H_2 O \rightarrow \sum \left( f_i n_i + c \right) CO_2 + \sum \left( \frac{m_i f_i}{2} + n_{v,a} \right) H_2 O \\
+ (a - 1) \left[ \sum f_i \left( n_i + \frac{m_i}{4} \right) \right] O_2 + \left[ \sum 3.76 a f_i \left( n_i + \frac{m_i}{4} \right) + d \right] N_2 \quad (7)
\]

where \( \alpha \) is the percentage of the excess air, \( f_i \) is the molar fraction of the fuel components parts and \( n_{v,a} \) is the number of moles of the humidity entering the combustion chamber with dry air. The unknown coefficients can be calculated by a molar balance and then the energy and exergy balance of the combustion gases can be performed. For different components of the cycle, the exergy destruction and the exergy efficiency can be obtained by applying exergy balance as follows:

**Boiler**

\[
\dot{E}_{\text{in,boiler}} = \dot{m}_f e_{s,f} + W_{\text{mech,boiler}} + \dot{m}_s e_{s,s} - \dot{m}_g e_{s,g} + \sum_{i \neq \text{dry air}} \dot{m}_i e_{s,i} - \sum_{\text{dry air}} m_{\text{out}} e_{s,w} \quad (8)
\]

\[
\eta_{\text{boiler}} = \frac{\sum_{i \neq \text{dry air}} \dot{m}_i e_{s,i} - \sum_{\text{dry air}} m_{\text{out}} e_{s,w}}{\dot{m}_f e_{s,f} + W_{\text{mech,boiler}} + \dot{m}_s e_{s,s} - \dot{m}_g e_{s,g}} \quad (9)
\]

The energy balance equation for calculating adiabatic flame temperature is:

\[
\sum N_i (\bar{h}_f^0 + \bar{h} - \bar{h}^0) = \sum N_i (\bar{h}_f^0 + \bar{h} - \bar{h}^0) \quad (10)
\]
The destroyed exergy due to combustion process and heat transfer can be expressed as:

$$\dot{E}_{x,d,\text{comb}} = \dot{m}_f e_{r,f}^h + \dot{m}_g e_{r,g} - \dot{m}_p e_{r,p,\text{adiabatic}}$$  \hspace{1cm} (11)

$$\dot{E}_{x,d,\text{heat}} = \dot{E}_{x,d,\text{comb}}.$$  \hspace{1cm} (12)

The second law efficiency of combustion process and heat transfer process in boiler can be expressed as (Saidur et al., 2010):

$$\eta_{II,\text{comb}} = \frac{\dot{m}_x e_{r,x,p,\text{adiabatic}}}{\dot{m}_x e_{r,x}^h + \dot{m}_x e_{r,x} - \dot{W}_{\text{in,b}}},$$  \hspace{1cm} (13)

$$\eta_{II,\text{heat}} = \frac{\sum_{\text{out}} \dot{m}_x e_{r,x} - \sum_{\text{in}} \dot{m}_x e_{r,x}}{\dot{m}_x e_{r,x,p,\text{adiabatic}} - \dot{m}_x e_{r,x}^h}.$$  \hspace{1cm} (14)

**Turbine**

$$\dot{E}_{x,t} = \sum_{\text{in},f} \dot{m}_e x - \sum_{\text{out},f} \dot{m}_e x - \dot{W}_t,$$  \hspace{1cm} (15)

$$\eta_{II,t} = 1 - \frac{\dot{E}_{x,t}}{\sum_{\text{in},f} \dot{m}_e x - \sum_{\text{out},f} \dot{m}_e x}.$$  \hspace{1cm} (16)

**Pump**

$$\dot{E}_{x,p} = (\dot{m}_e x)_{\text{in},p} - (\dot{m}_e x)_{\text{out},p} + \dot{W}_p,$$  \hspace{1cm} (17)

$$\eta_{II,p} = \frac{(\dot{m}_e x)_{\text{out},p} - (\dot{m}_e x)_{\text{in},p}}{(\dot{m} h)_{\text{out},p} - (\dot{m} h)_{\text{in},p}}.$$  \hspace{1cm} (18)

**Condenser**

$$\dot{E}_{x,c} = \sum_{\text{in},c} \dot{m}_e x - \sum_{\text{out},c} \dot{m}_e x,$$  \hspace{1cm} (19)

$$\dot{E}_{x,c} = \sum_{\text{in},c} \dot{m}_e x - \sum_{\text{out},c} \dot{m}_e x + \left(\dot{m}_e x_{\text{coolingwater}}\right)_{\text{in}} - \left(\dot{m}_e x_{\text{coolingwater}}\right)_{\text{out}} + Q_{\text{out,c}} \left(1 - \frac{T_c}{T_o}\right).$$  \hspace{1cm} (20)

**Closed feed water heaters**

$$\dot{E}_{x,he} = \sum_{\text{in},he} \dot{m}_e x - \sum_{\text{out},he} \dot{m}_e x,$$  \hspace{1cm} (21)

$$\eta_{II,he} = \left(\frac{\sum_{\text{out},he} \dot{m}_e x - \sum_{\text{out},he} \dot{m}_e x_{\text{cold}}}{\sum_{\text{in},he} \dot{m}_e x - \sum_{\text{out},he} \dot{m}_e x}\right).$$  \hspace{1cm} (22)
In the above equations, $H_{w,b}$ is the work consumed by the auxiliary equipments of the boiler, such as fans. Figure 2 shows temperature-entropy diagram of the plant.

Figure 2  The entropy-temperature diagram of Ramin power plant cycle (see online version for colours)

2.2 Modelling condenser pressure variation

As mentioned above, in the previous investigations a constant pressure assumption was used for condenser analysis regardless of any changes in the ambient temperature, which is not realistic. In practice, the condenser pressure of power plant varies considerably as ambient temperature and relative humidity changes due to the changes in the cooling water temperature. Therefore, it is required to model the variation of condenser pressure at different ambient conditions. Condenser pressure and inlet water temperature to the condenser were empirically measured at different ambient temperatures and relative humidities to model its behaviour. It was found that in most of the time the average local relative humidity was around 30%. Therefore, this relative humidity was used to calculate different wet bulb temperature of local area at ambient dry bulb temperature. Table 2 shows variations of the condenser pressure and the cooling tower temperature at different ambient temperatures assuming constant relative humidity equal to 30%. The condenser outlet water temperature is typically 10°C higher than the inlet water temperature.
Table 2
Variation of condenser pressure and inlet water temperature in terms of ambient temperature

<table>
<thead>
<tr>
<th>Ambient temperature (°C)</th>
<th>Ambient Wet-Bulb temperature (°C)</th>
<th>Condenser pressure (kPa)</th>
<th>Condenser inlet water temperature (°C)</th>
</tr>
</thead>
<tbody>
<tr>
<td>5.0</td>
<td>0.5</td>
<td>4.96</td>
<td>19.0</td>
</tr>
<tr>
<td>10.0</td>
<td>3.5</td>
<td>5.93</td>
<td>21.0</td>
</tr>
<tr>
<td>15.0</td>
<td>7.2</td>
<td>7.32</td>
<td>23.0</td>
</tr>
<tr>
<td>20.0</td>
<td>10.8</td>
<td>9.03</td>
<td>25.0</td>
</tr>
<tr>
<td>25.0</td>
<td>14.4</td>
<td>10.99</td>
<td>27.0</td>
</tr>
<tr>
<td>30.0</td>
<td>17.9</td>
<td>13.14</td>
<td>29.0</td>
</tr>
<tr>
<td>35.0</td>
<td>21.5</td>
<td>15.39</td>
<td>31.0</td>
</tr>
<tr>
<td>40.0</td>
<td>25.0</td>
<td>17.67</td>
<td>33.0</td>
</tr>
<tr>
<td>45.0</td>
<td>28.6</td>
<td>19.91</td>
<td>35.0</td>
</tr>
<tr>
<td>50.0</td>
<td>32.3</td>
<td>22.02</td>
<td>37.0</td>
</tr>
</tbody>
</table>

3 Results and discussions

Thermodynamic properties of the cycle at different points are listed in Table 3. The energy balance of power plant at normal condition is shown in Table 4.

Table 3
Thermodynamic properties of working fluid of the power plant at different points ($T_0 = 25$ °C)

<table>
<thead>
<tr>
<th>Node</th>
<th>$P$ (MPa)</th>
<th>$T$ (°C)</th>
<th>$h$ (kJ/kg)</th>
<th>$s$ (kJ/kg.K)</th>
<th>$\dot{m}$ (kg/s)</th>
<th>$c_v$ (kJ/kg)</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>0.066</td>
<td>49</td>
<td>205.2</td>
<td>0.691</td>
<td>186.0</td>
<td>3.93</td>
</tr>
<tr>
<td>2</td>
<td>2.290</td>
<td>51</td>
<td>215.5</td>
<td>0.716</td>
<td>186.0</td>
<td>7.39</td>
</tr>
<tr>
<td>3</td>
<td>2.080</td>
<td>67</td>
<td>244.5</td>
<td>0.805</td>
<td>186.0</td>
<td>9.17</td>
</tr>
<tr>
<td>4</td>
<td>1.870</td>
<td>91</td>
<td>382.5</td>
<td>1.210</td>
<td>218.1</td>
<td>27.18</td>
</tr>
<tr>
<td>5</td>
<td>1.660</td>
<td>120</td>
<td>504.8</td>
<td>1.526</td>
<td>218.1</td>
<td>54.31</td>
</tr>
<tr>
<td>6</td>
<td>1.450</td>
<td>149</td>
<td>628.6</td>
<td>1.831</td>
<td>218.1</td>
<td>87.54</td>
</tr>
<tr>
<td>7</td>
<td>0.700</td>
<td>164</td>
<td>693.1</td>
<td>1.983</td>
<td>282.6</td>
<td>106.70</td>
</tr>
<tr>
<td>8</td>
<td>33.130</td>
<td>172</td>
<td>730.1</td>
<td>2.062</td>
<td>282.6</td>
<td>120.89</td>
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<tr>
<td>9</td>
<td>32.800</td>
<td>201</td>
<td>857.9</td>
<td>2.340</td>
<td>282.6</td>
<td>165.80</td>
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<tr>
<td>10</td>
<td>32.470</td>
<td>252</td>
<td>1095.0</td>
<td>2.811</td>
<td>282.6</td>
<td>262.5</td>
</tr>
<tr>
<td>11</td>
<td>31.220</td>
<td>282</td>
<td>1243.0</td>
<td>3.085</td>
<td>282.6</td>
<td>328.98</td>
</tr>
<tr>
<td>12</td>
<td>24.000</td>
<td>540</td>
<td>3316.0</td>
<td>6.169</td>
<td>282.6</td>
<td>1482.00</td>
</tr>
<tr>
<td>13</td>
<td>6.918</td>
<td>360</td>
<td>3047.5</td>
<td>6.283</td>
<td>20.1</td>
<td>1180.00</td>
</tr>
<tr>
<td>14</td>
<td>6.640</td>
<td>261</td>
<td>1139.0</td>
<td>2.893</td>
<td>20.1</td>
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</tr>
<tr>
<td>15</td>
<td>4.480</td>
<td>303</td>
<td>2952.0</td>
<td>6.300</td>
<td>255.8</td>
<td>1080.00</td>
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<tr>
<td>16</td>
<td>4.280</td>
<td>208</td>
<td>889.4</td>
<td>2.402</td>
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<tr>
<td>17</td>
<td>4.080</td>
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<td>3536.0</td>
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<td>414</td>
<td>3282.0</td>
<td>7.249</td>
<td>11.8</td>
<td>1127.00</td>
</tr>
</tbody>
</table>
Table 3  Thermodynamic properties of the working fluid of the power plant at different points 
\((T_0 = 25 \, ^\circ\text{C})\) (continued)

<table>
<thead>
<tr>
<th>Node</th>
<th>(P) (MPa)</th>
<th>(T) (°C)</th>
<th>(h) (kJ/kg)</th>
<th>(s) (kJ/kg.K)</th>
<th>(m) (kg/s)</th>
<th>(e_x) (kJ/kg)</th>
</tr>
</thead>
<tbody>
<tr>
<td>19</td>
<td>1.620</td>
<td>180</td>
<td>763.6</td>
<td>2.139</td>
<td>62.4</td>
<td>130.70</td>
</tr>
<tr>
<td>20</td>
<td>1.110</td>
<td>355</td>
<td>3166.0</td>
<td>7.267</td>
<td>1.5</td>
<td>1005.00</td>
</tr>
<tr>
<td>21</td>
<td>0.532</td>
<td>264</td>
<td>2988.0</td>
<td>7.295</td>
<td>11.5</td>
<td>818.90</td>
</tr>
<tr>
<td>22</td>
<td>0.500</td>
<td>151</td>
<td>637.0</td>
<td>1.853</td>
<td>11.5</td>
<td>89.38</td>
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<tr>
<td>23</td>
<td>0.238</td>
<td>217</td>
<td>2901.0</td>
<td>7.494</td>
<td>33.0</td>
<td>673.05</td>
</tr>
<tr>
<td>24</td>
<td>0.228</td>
<td>236</td>
<td>2941.0</td>
<td>7.591</td>
<td>20.1</td>
<td>683.60</td>
</tr>
<tr>
<td>25</td>
<td>0.222</td>
<td>216</td>
<td>2901.0</td>
<td>7.523</td>
<td>9.2</td>
<td>663.90</td>
</tr>
<tr>
<td>26</td>
<td>0.222</td>
<td>102</td>
<td>427.6</td>
<td>1.329</td>
<td>20.7</td>
<td>35.95</td>
</tr>
<tr>
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<td>181</td>
<td>2830.0</td>
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Table 4  Energy balance of the power plant components and percent ratio to fuel energy input

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Table 5 shows total exergy of the working fluid at different points as ambient temperature change. The exergy analysis of different components can be obtained by using these data. Table 6 presents the first and second law efficiencies of major components of the cycle when ambient temperature is 25°C. Obviously, the boiler and turbine are the major sources of irreversibility in the cycle.

In the following, the effect of condenser pressure changes due to ambient condition changes, on the performance of the power plant are discussed. Figure 3 shows the effect
of ambient temperature on the energy and exergy efficiencies of the power plant when constant condenser pressure approach is used. Figure 4 shows the same results but when variable condenser pressure approach is taken into account. As shown, the energy efficiency is constant in the case of using constant pressure in the condenser but it decreases with ambient temperature when variable condenser pressure is taken into account. The exergy efficiencies decrease in both cases but the rate of reduction is higher when variable condenser pressure approach is taken into account. Actual data from the power plant agree with variable pressure approach in the condenser.

**Figure 3** Effect of ambient temperature on the energy and exergy efficiencies at constant pressure (see online version for colours)

**Figure 4** Effect of ambient temperature on the energy and exergy efficiencies at variable condenser pressure (see online version for colours)
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Table 6  The first and second law efficiency of different parts of the power plant (25 C)

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Figure 5 illustrates irreversibility of the different components of the plant at different ambient temperatures using constant condenser pressure approach. As shown, irreversibility of all components are increased as ambient temperature increases except the condenser. Irreversibility reduction in the condenser can be explained by noting the reduction of temperature difference between the steam and the cooling water, as the dead-state temperature is increased. This will decrease the external irreversibility and hence will increase the exergy efficiency. This result is in agreement with other findings (Kopac and Hilalci, 2007; Aljundi, 2009).

Figure 6 shows how the irreversibility of different components varies with ambient temperature, when variable condenser pressure approach is used in the analysis. As shown, irreversibility of all components except turbines is increased as ambient temperature increases. Turbine irreversibility reduction is the result of reduction of temperature difference between the turbine inlet and outlet temperature. Because by increasing the ambient temperature, the condenser pressure and temperature are increased.
The increase in the condenser irreversibility is the result of increase in the temperature difference between steam and cooling water as the dead state temperature is increased.

As seen, depending on the approach used in treating the condenser pressure, the final results may vary considerably. Because variable condenser pressure approach is close to the actual situation, it is suggested to use this approach in the exergy analysis of a power plant. The results reveal that for a 1°C increase in the ambient temperature, total irreversibility of the plant increases about 1.51 MW.

Moreover, Figures 5 and 6, indicate that boiler is still the main source of irreversibility in the power plant and total irreversibility largely follows by the boiler irreversibility. Comparison of exergy efficiency of supercritical boiler, Table 6, (45.48%) with conventional boiler which is around 25–30% (Saidur et al., 2010) shows that exergy efficiency of supercritical boiler has improved considerably. Two major sources of irreversibility in the boiler are combustion phenomena and heat transfer between hot
gas and working fluid. To make any improvement in the boiler performance, it is necessary to estimate the share of each source of irreversibility in the boiler. Figure 7 shows the share of each source of irreversibility at different ambient temperatures in terms of total irreversibility of the power plant. Clearly, irreversibility generation due to combustion is more than that of the heat transfer process. The figure also shows that increasing in the ambient temperature does not change irreversibility ratio of the heat transfer but slightly increases irreversibility ratio of the combustion process.

The high irreversibility in the boiler heat exchanger is due to finite temperature difference between the combustion gases and the working fluid (water and steam). Heat exchangers are generally inefficient from exergy standpoint because they have mostly been designed on the basis of low first cost. That dictates a minimum sized unit which results to cheaper heat exchanger; consequently, the temperature difference between the fluid streams is maximised. The larger is the temperature difference in heat exchanger, the greater will be the exergy loss during heat transfer. Optimising heat transfer area and its configuration, effective arrangement and better material selection are the primary ways to improve system performance.

**Figure 7** The share of irreversibility sources in the boiler at different ambient temperatures (see online version for colours)

![Figure 7](image_url)

A combustion process usually occurs simultaneously with heat transfer. The losses in the combustion chamber are due to the increase in the entropy of the combustion gases and heat loss within products of the combustion gases leaving the stack. Improvements in fuel combustion efficiency can greatly contribute to improvement in the boiler and system performance (Regulagadda et al., 2010).

Figure 8 shows how the energy efficiency of the power plant changes with ambient temperature at different relative humidities. As seen, when relative humidity increases, the performance of the power plant reduces. When ambient temperature is low, the rate of reduction is slightly higher than other condition. This decrease is due to the adverse effect of ambient humidity on the performance of the cooling towers, which causes an increase in the condenser pressure.
Figure 9 shows the variations of the power plant exergy efficiency with ambient temperature at different relative humidities. As seen, the exergy efficiency decreases as the relative humidity increases.

Figure 8 Variations of the energy efficiency with ambient temperature and relative humidity (see online version for colours)

4 Conclusion

Exergy analysis was used to investigate the irreversibility rate of different components of a supercritical power plant at different ambient temperatures and relative humidities. Two approaches, namely constant and variable condenser pressure, were used to model the condenser behaviour. It was found that the results of exergy analysis have considerable difference depending on the type of approached used in modelling the condenser. If constant pressure in condenser is assumed, the exergy lost in the condenser is decreased, as ambient temperature increases, but it will be increased if variable condenser pressure is taken into account.
Variable condenser pressure approach was recommended for exergy analysis since it is close to the actual condition. As ambient temperature increases, exergy efficiencies of the boiler, condenser, heaters and feed water pump decrease, while the exergy efficiency of the turbine improve slightly. For 1°C increase in the ambient temperature, total irreversibility increases 1.51 MW. Exergy efficiency of supercritical boiler is considerably higher than conventional one; however, it is still the main source of total irreversibility in the cycle followed by the turbine. The share of irreversibility due to the combustion process is more than the heat transfer process in the supercritical boiler. It was also found that not only increasing the ambient temperature, but also increasing the relative humidity deteriorates the energy and exergy efficiencies of the power plant.

References


**Nomenclature**

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<tr>
<th>Symbol</th>
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<tr>
<td>$e_s$</td>
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<td>$h$</td>
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