Performance characteristics of a Diesel engine power plant

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Abstract

Performance of an actual Diesel engine power plant with a rated output of 120 MW is analyzed based on the first and second laws of thermodynamics. The plant consists of seven identical Diesel engines and various subsystems including turbochargers, fuel heating units and heat exchangers performing various useful tasks. The engine runs on heavy fuel oil, and the pollutant emissions from the engine are greatly reduced by effective treatment systems. The characteristics and performance parameters of the internal combustion engines of the plant are evaluated. The mass, energy and exergy balances are verified for each flow stream in the power plant. The work and heat interactions, the exergy losses and the efficiencies of various components based on both energy and exergy concepts are evaluated. The thermal and the exergy efficiencies of the plant are determined to be 47% and 44%, respectively. The engine irreversibilities are due mostly to the irreversible combustion process and account for 32% of the total exergy input and 57% of the total irreversibilities in the plant. Most of the remaining irreversibilities in the plant occur in the desulphurization, intercooler, compressor and lubrication oil cooler units. The results should provide a realistic and meaningful ground for the performance evaluation of Diesel engine power units, and it may be used in the design and analysis of such systems.

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1. Introduction

Back in the 1970s, most power plants were large units based on steam turbines. At that time, both nuclear and hydropower were proving popular in the power range above 1000 MW. Public and private sector power plants also used gas turbines to some extent. Although smaller than the nuclear and hydropower plants, these were still quite large compared with power plants today [1,2].

Power generation using reciprocating engines was not as common three decades ago as it is today. The main application for engine derived power was in small backup plants for hospitals, airports, hotels and industry that needed to ensure a reliable power supply at all times. Engine based power production today represents some 10–15% of the total installed capacity all over the world. This is the result of its high efficiency, power concentration and reliability that have been improved considerably during the last decade. Reducing engine emissions to legally acceptable levels has been a challenge for large Diesel engine manufacturers. There is an increasing need to find more effective environmental technologies, and this has been a driver for more research activities [3,4].

Turkey’s energy market is one of the country’s fastest developing sectors. Annual demand for electricity has increased by more than 7% during recent years, and installed capacity and annual generation figures have today reached approximately 31.75 GW. Mobile Diesel power plants take a portion of 4.2% of this generation, whereas the other type of thermal plants such as steam and gas turbines having 47.7%, hydraulic power plants having 48% and geothermal power plants having 0.1% of the total capacity. At the present time, the installation of Diesel power plants with a total capacity of 262.4 MW is in progress. However, since consumption per capita per annum is still as low as 1800 kWh (compared to 8500 kWh in Germany and 17 500 kWh in Finland) and the economy continues to grow at high rates, the potential for further growth in this market is considerable [5,6].

Special attention has been given to the industrial autoproducer concept. An autoproducer is a company, or a group of companies, that produces its own electrical and thermal energy. The model offers interesting opportunities for private investors who are willing to form joint ventures with
local factories that consume power and energy. In their long term planning, TEAS (state owned public utility company) covers hydro, nuclear, coal and gas fired thermal power plants. However, these plants will take some years to be operational. To meet the intermediate needs for power capacity, TEAS introduced the “mobile power plant” concept in 1998, and a total of 75 MW of Diesel fired power plants were built in 1999. It seems that Diesel power plants may be a good solution as hundreds of megawatts can be operational in less than 12 months. These power plants run on cheap heavy fuel oil, which is the low grade product of all seven Turkish refineries. Such power plants have proven to be a reliable and economical means of power generation worldwide, and Turkey is no exception [7].

Exergy analysis is a powerful tool in the design, optimization and performance evaluation of energy systems and Diesel engine power systems are no exception. An exergy analysis is usually aimed to determine the maximum performance of the system and/or identify the sites of exergy destruction. Exergy analysis of a complex system can be performed by analyzing the components of the system separately. Identifying the main sites of exergy destruction shows the direction for potential improvements. Exergy analysis has been applied to both work producing and work consuming units [8,9].

The performance and emission characteristics of Diesel engines have been the subject of many studies. Some researchers studied the effects of certain operating parameters on the performance and emissions of the Diesel engine [10–12], while others analyzed Diesel engines using the second law aspects of their operation [13,14]. A case study on a Diesel engine powered cogeneration plant was also performed considering only the first law aspects [15]. In this paper, the performance of a Diesel engine power plant is analyzed. The paper is unique in that the analysis is not limited to the engine itself, but it covers all main components of the plant including the compressor, turbine, fuel and oil heating systems, boiler, emission treatment units and various coolers and that both the first and second laws of thermodynamics are utilized. The Diesel engine driven power plant is a relatively new concept in Turkey’s power market, and we hope that this study will provide valuable information regarding their operating and performance characteristics.

2. Power plant operation

The power plant has a total power capacity of 120 MW, while it was intended that an average of 100 MW were to be produced. It is located in the city of Batman located in southeastern Turkey, and it started to produce power in 2002. The power house consists of seven engine-generator sets, each having two turbochargers. The schematic of the plant is shown in Fig. 1 where only one turbocharger is demonstrated. The engine is a four stroke compression ignition engine with 18 cylinders in a V configuration. It uses heavy fuel oil, which is obtained from a nearby oil refinery. The permissible annual electrical energy production is 857 GWh, and the annual fuel consumption is nearly 180000 tons at designed operating conditions. The generated electrical energy is transferred at 154 kV and 50 Hz by transmission lines.

As shown in Fig. 1, when the engine starts, air is charged to the compressor of the turbocharger unit. The air is cooled by water in an intercooler before entering the engine cylinders. The exhaust gases leaving the engine flow through the turbine of the turbocharger to produce the necessary shaft work for the compressor. The exhaust gases leaving the turbine are sent to the DeNOx
(denitrification) unit in which the NO\textsubscript{x} emission is lowered to acceptable legal values by spraying a urea solution into the exhaust gases. The exhaust gases enter the boiler unit to transfer heat to the condensate return and make up water to produce saturated steam for preheating of streams in the auxiliary equipments such as the fuel forwarding module (FFM) and fuel and lubrication oil tanks. The exhaust gases flow through a DeSO\textsubscript{x} (desulphurization) unit before being exhausted to the atmosphere.

The water used in the plant is obtained from three wells. After the water is purified in the water treatment (WT) unit, it is distributed by the collectors to the boiler, the engine and the flue gas treatment unit. The fuel obtained in the refinery is first collected in the storage tanks. It is then preheated by steam before entering the FFM. Finally, it is injected to the engine cylinders through the nozzle. Oil is used for lubrication and cooling in the engine components. Table 1 lists the temperature, pressure and mass flow rate data in the power plant according to the nomenclature shown in Fig. 1.

3. Engine performance characteristics

Certain design, operating and performance data for the engine are given in Table 2. The characteristics and performance parameters of the engine are determined using the formulation given
The thermal efficiency of the plant is determined to be 47%. This appears to be high compared with a steam or gas turbine plant, which typically has efficiencies between 30 and 40%, and an automobile engine, which has typical average efficiencies in the range of 25–35%. This may be explained by the fact that stationary engines operate at their optimum values with a constant engine speed. The entire operation is optimized to minimize fuel consumption for a given power output. Inherent limitations such as space, weight, complexity and maintenance in the design of automobile engines are not crucial for stationary large engines. As a result, the brake
specific fuel consumption (BSFC = 195 g/kWh) is low compared with automobile engines whose BSFC values are typically around 400 g/kWh. A general trend is that the greater is the engine size,
the smaller is the BSFC. Two main reasons for this are less heat loss due to the higher volume to surface area ratio of the combustion chamber in a large engine and lower engine speeds, which reduce friction losses [16].

Although the volumetric efficiency is normally defined for naturally aspirated engines, we used the standard definition here to see how much the turbocharging process increased the air input to the engine cylinders compared with no pressure boosting system under ideal conditions. A value of 1.26 is obtained, which indicates that this increase is 26%. In a rough estimate, this corresponds to 26% increase in the power output of the engine due to the use of turbochargers compared to naturally aspirated engines. Some other engine performance data are indicative of large stationary Diesel engines. For example, the power output per displacement volume (OPD) value is 9.71 kW/L, whereas a typical value for an automobile engine is about 50 kW/L and that for a two stroke model airplane engine is 100 kW/L. The OPD value for a two stroke large stationary engine could be as small as 1.0 kW/L.

The emissions generated by the combustion process pollute the environment and harm human, animal and plant life. The major causes of these emissions are non-stoichiometric combustion, dissociation of nitrogen and the impurities in the fuel and air. The emissions of concern are hydrocarbons (HC), carbon monoxide (CO), oxides of nitrogen (NO\textsubscript{x}), oxides of sulphur (SO\textsubscript{x}) and solid carbon particulates. To minimize these emissions, flue gas treatment systems (DeNO\textsubscript{x} and DeSO\textsubscript{x} units) are installed. In Table 4, engine emission data measured before and after treatment are given. The values are in parts per million (PPM) by the mass basis, and they are dry values. The data clearly shows the effectiveness of these treatment units, and the final emission figures are all below the legal limits.

An air standard ideal Diesel cycle analysis of the engine based on engine inlet conditions and other engine data in Tables 1 and 2 under the consideration of variable specific heats gives an indicated net power of 21 707 kW and an indicated thermal efficiency of 55%. The corresponding actual brake values were obtained to be 18 900 kW and 47%, respectively. The differences between the actual brake values and ideal cycle indicated values are due to mechanical inefficiencies as the power is transferred from inside the cylinders to the crankshaft, heat losses, friction, ignition timing, finite time of combustion and blowdown and deviation from ideal gas behavior of the actual engine.

### 4. Thermodynamic analysis

Mass, energy and exergy balances for any control volume at steady state with negligible kinetic and potential energy changes can be expressed, respectively, by [18]
\[
\sum \dot{m}_i = \sum \dot{m}_e \tag{1}
\]
\[
\dot{Q} + \dot{W} = \sum \dot{m}_e h_e - \sum \dot{m}_i h_i \tag{2}
\]
\[
\dot{E}_{\text{heat}} + \dot{W} = \sum \dot{m}_e \psi_e - \sum \dot{m}_i \psi_i + \dot{I} \tag{3}
\]
where the subscripts \(i\) and \(e\) represent the inlet and exit states, \(\dot{Q}\) and \(\dot{W}\) are the net heat and work inputs, \(\dot{m}\) is the mass flow rate, \(h\) is the enthalpy, \(\dot{I}\) is the rate of irreversibility (exergy destruction) and \(\dot{E}_{\text{heat}}\) is the net exergy transfer by heat at temperature \(T\), which is given by
\[
\dot{E}_{\text{heat}} = \sum \left(1 - \frac{T_0}{T}\right) \dot{Q} \tag{4}
\]
The specific flow exergy is given by
\[
\psi = (h - h_0) - T_0(s - s_0) \tag{5}
\]
where the subscript 0 stands for the restricted dead state. The second law (exergy) efficiency of a turbine can be defined as a measure of how well the stream availability of the fluid is converted into shaft work output [19]
\[
\eta_{\text{T,II}} = \frac{\dot{W}_{\text{actual}}}{\dot{W}_{\text{rev}}} \tag{6}
\]
where \(\dot{W}_{\text{actual}}\) is the actual shaft-power and \(\dot{W}_{\text{rev}}\) is the reversible power, which is equal to \(\dot{W}_{\text{actual}} + \dot{I}\). The exergy efficiency of the compressor is defined similarly as
\[
\eta_{\text{C,II}} = \frac{\dot{W}_{\text{rev}}}{\dot{W}_{\text{actual}}} \tag{7}
\]
where \(\dot{W}_{\text{rev}}\) is equal to \(\dot{W}_{\text{actual}} - \dot{I}\). The exergy efficiency of the heat exchangers in the power plant is measured by the increase in the exergy of the cold stream divided by the decrease in the exergy of the hot stream
\[
\eta_{\text{HE,II}} = \frac{\dot{m}_c(\psi_e - \psi_i)c}{\dot{m}_h(\psi_i - \psi_e)h} \tag{8}
\]
where \(\dot{m}_c\) and \(\dot{m}_h\) are the mass flow rates of the cold and hot streams, respectively. The exergy efficiency may be defined for the DeSO\(_x\) and DeNO\(_x\) units as the ratio of the exergy output to the exergy input
\[
\eta_{\text{II}} = \frac{\dot{E}_{\text{out}}}{\dot{E}_{\text{in}}} \tag{9}
\]
This relation may also be viewed as the expression of the exergy efficiency for the entire power plant.

Energy and exergy analyses of the power plant are performed using actual operational data. Exergy rates for all flow streams in the plant are calculated and given in Table 1. The ideal gas model is applied for both air and the exhaust gases. Heat transfer or work, irreversibilities and exergy efficiencies are calculated for the main plant components and listed in Table 5. The
isentropic efficiencies for the compressor and turbine are also determined with values not close to the exergy efficiencies. This is due to the different ways they are defined. One of the reasons for the lower exergy efficiency for the compressor compared to the turbine is that there are desirable heat losses from the compressor, resulting in additional exergy destruction. The intercooler, boiler, radiator, lubrication oil cooler and condenser all have low exergy efficiencies. Exergy destructions in these heat exchange units are mostly due to the heat transfer across a high average temperature difference between the two fluids. The DeSO$\text{X}_x$ unit has high exergy destruction with a low efficiency, while the DeNO$\text{X}_x$ unit operates at high exergy efficiency. This is because there are much higher rates of heat removal from the exhaust gases in the DeSO$\text{X}_x$ unit compared with the DeNO$\text{X}_x$ unit.

The exergy input to the plant can be taken to be the chemical exergy of the fuel. The exergy of a high molecular weight fuel with an unknown composition may be taken as 1.065 times the lower heating value of the fuel [20]. Then, the exergy input to this power plant becomes 43110 kW. The only useful exergy output from the plant is the electrical power produced, which is 18900 kW. The difference from 24210 kW is the total irreversibility in the plant. The ratio of the exergy output to the exergy input for the plant is the exergy efficiency of the plant, which is 43.8%.

In order to determine the total exergy destruction in the engine unit, we write an exergy balance on the engine with the input exergies of the fuel, compressed air, lubrication oil and radiator water and the output exergies of the work produced, exhaust gases, lubrication oil and radiator water. The difference between the input exergies and output exergies for the engine is 13892 kW. This is the total exergy destruction in the engine, and it is mostly due to the highly irreversible combustion process. Heat losses from the engine and friction are the other causes of irreversibilities. Engine irreversibilities account for 32% of the total exergy input and 57% of the total irreversibilities in the plant. Note that in the exergy balance relation on the engine, the exergy of the fuel is calculated by adding the thermomechanical exergy of the fuel to its chemical exergy.

One may argue that the exergy of exhaust gases leaving the engine is not a total loss since it used for some useful purposes in the plant. For example, it is used in the boiler to produce steam, which is used to heat fuel and lubrication oil before they enter the engine. On the other hand, one can

<table>
<thead>
<tr>
<th>Component</th>
<th>Rates of heat transfer or work (kW)</th>
<th>Exergy destruction (kW)</th>
<th>Exergy efficiency (%)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Compressor</td>
<td>6170</td>
<td>2221</td>
<td>64.0</td>
</tr>
<tr>
<td>Intercooler</td>
<td>4642</td>
<td>3533</td>
<td>13.0</td>
</tr>
<tr>
<td>Turbine</td>
<td>6873</td>
<td>569</td>
<td>91.7</td>
</tr>
<tr>
<td>DeNO$\text{X}_x$</td>
<td>904</td>
<td>400</td>
<td>92.0</td>
</tr>
<tr>
<td>Boiler</td>
<td>670</td>
<td>327</td>
<td>18.5</td>
</tr>
<tr>
<td>DeSO$\text{X}_x$</td>
<td>10760</td>
<td>6840</td>
<td>16.7</td>
</tr>
<tr>
<td>Engine jacket radiator</td>
<td>800</td>
<td>625</td>
<td>38.0</td>
</tr>
<tr>
<td>Lubrication oil cooler</td>
<td>2070</td>
<td>1896</td>
<td>34.5</td>
</tr>
<tr>
<td>Condenser</td>
<td>20</td>
<td>17</td>
<td>18.7</td>
</tr>
</tbody>
</table>

$^a$ Isentropic efficiency.
rightfully counter argue that the only useful output from the plant is the power produced and the use of exhaust gases in the boiler is an internal process aimed toward more effective power production. Our treatment would be different if the exhaust gases were used for process heating in a manufacturing plant as in a cogeneration scheme.

5. Conclusions

Energy and exergy analyses of an actual Diesel engine power plant are performed. Both the performance characteristics of the internal combustion engine unit and the supporting components in the plant are evaluated. The thermal and exergy efficiencies of the plant are calculated to be 47% and 44%, respectively. The sites of exergy destructions in the plant and the exergy efficiencies of various components are determined. Such an analysis is useful in that higher exergy destruction sites represent the most potential for possible improvements in the performance. The study is useful and provides some comprehensive information regarding the characteristics of Diesel engine power plants as the use of these systems has been increasing considerably in recent years. The results should provide a realistic and meaningful ground in the performance evaluation of Diesel engine power units, and it may be used in the design and analysis of such systems.

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