OPERATION OF THE MULTI-FUEL CYCLES UNDER OFF-DESIGN REGIME

The paper presents an analysis of a multi-fuel power cycle. The cycle includes a gas and steam turbine. The methods and tools are described, which allow to model the operation of the cycle in the off-design regime.

Keywords: multi-fuel cycles, off-design.

1. Introduction

The multi-fuel power cycle become an essential alternative for conventional and combined heat and power plants [1,3,4,7]. The main advantage of such cycles is the possibility of their better adjustment to the user demands and in consequence the decrease of the operation costs [2,4]. The selection of the basic parameters such as the power output or the amount of the generated heat is strictly connected to the conditions under which a specified cycle is to operate.

Such strictly limited selection results in a very good efficiency of the operation in the design regime. However the range of the off-design regime in which the operation is profitable may be very small. Therefore even the initial choice of the cycle components requires also an analysis of the off-design operation.

This paper presents the tools and results for such an analysis. The cycle under investigation is a multi-fuel power plant with a gas and steam turbines. The article describes a model applied to the simulation of the operation under various regimes. The following issues are analyzed:

• the adjustment of the gas turbine in order to satisfy the power demands,
• the efficiency of the combined gas and steam cycles for a changing ambient temperature,
• the division of the load between the gas and steam sub-cycles.

The research regards especially the effectiveness criterion as it has a direct influence on the costs of operation.

2. The analyzed cycle

The cycle under the investigation consists of a gas turbine, a heat recovery steam generator (HRSG) and a condensing turbine. The configuration of the cycle is shown in fig. 1 together with notation of the cycle nodes. The flue gas from the gas turbine feeds the HRSG and allows to generate a part of the livesteam for the steam turbine. A parallel coal boiler produces the rest of the livesteam. Such arrangement allows better flexibility in terms of the fuel.

The gas turbine has an open cycle with one compressor, combustor and expander. The expander has an internal cooling. The cooling air is extracted from the last stage of the compressor and delivered to the first stage of the expander. Since the exhaust gas has quite high temperature at the expander outlet it is further to feed the HRSG. This temperature may even be increased in the afterburner. Such necessity arises in the off-design operation when the stream of the exhaust gas is smaller.

The steam cycle consists of two expanders: a single-flow HP part and double-flow LP part. The cycle includes also exchangers for heat recovery. They are fed by the steam extracted from expander bleedings in both parts of the steam turbine.

3. The modeling of the off-design operation

The analysis of the operation under off-design regime requires appropriate tools for the simulation of the operation of the cycle. A simulation module was designed for this purpose [6].

The module consists of several components, which simulate the operation of the chosen machines of a thermal cycle, such as the compressor, combustor, gas expander, steam expander, heat exchangers and so on. This allows to analyze various arrangements of combined multi-fuel cycles.

The simulation module for off-design operation must take into account the influence of the user as well as the atmospheric conditions. The design power output for the gas turbine is 66.6 MW and for the steam turbine 48 MW. Several design values are gathered in table 1.

Tab. 1. Design values for the chosen parameters of the cycle

<table>
<thead>
<tr>
<th>Quantity</th>
<th>Point</th>
<th>Unit</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>ambient air pressure</td>
<td>201</td>
<td>kPa</td>
<td>101</td>
</tr>
<tr>
<td>ambient air temperature</td>
<td>201</td>
<td>oC</td>
<td>217</td>
</tr>
<tr>
<td>ambient air humidity</td>
<td>201</td>
<td>%</td>
<td>60</td>
</tr>
<tr>
<td>inlet guide vane angle</td>
<td>201</td>
<td></td>
<td>1</td>
</tr>
<tr>
<td>compressor outlet pressure</td>
<td>202</td>
<td>kPa</td>
<td>1800</td>
</tr>
<tr>
<td>fuel mass flow</td>
<td>203</td>
<td>kg/s</td>
<td>20.3</td>
</tr>
<tr>
<td>combustor outlet temperature</td>
<td>204</td>
<td>oC</td>
<td>1100</td>
</tr>
<tr>
<td>turbine inlet temperature</td>
<td>204</td>
<td>oC</td>
<td>200</td>
</tr>
<tr>
<td>power output</td>
<td></td>
<td>MW</td>
<td>66.63</td>
</tr>
<tr>
<td>livesteam pressure</td>
<td>1</td>
<td>MPa</td>
<td>5.7</td>
</tr>
<tr>
<td>livesteam temperature</td>
<td>1</td>
<td>oC</td>
<td>480</td>
</tr>
<tr>
<td>livesteam mass flow rate</td>
<td>1</td>
<td>t/h</td>
<td>200</td>
</tr>
<tr>
<td>condenser pressure</td>
<td>1.24</td>
<td>kPa</td>
<td>6</td>
</tr>
<tr>
<td>feed water flow rate at the HRSG inlet</td>
<td>124</td>
<td>t/h</td>
<td>100</td>
</tr>
<tr>
<td>power output</td>
<td></td>
<td>MW</td>
<td>47.86</td>
</tr>
</tbody>
</table>

Both turbines are selected in such a manner that for design regime the afterburner does not operate and most of the heat from the exhaust gas is regenerated in the HRSG. The exhaust gas temperature at the HRSG outlet (point 208 in fig. 1) is 200 °C.
(ambient) conditions on the operation of the machines. Therefore the input data for the analysis consist of a set of parameters, which are either set directly by a user or derive from the atmospheric conditions. For the analyzed cycle the set of the input data includes:

- ambient air pressure, temperature and humidity,
- compressor inlet guide vane angle (IGV A),
- gas fuel flow rate to the combustor,
- gas fuel flow rate to the afterburner,
- livesteam pressure, temperature and flow rate,
- cooling water temperature at the condenser inlet,
- feed water flow rate at the HRSG inlet.

The simulation allows to determine the parameters of the flowing media (air, exhaust gas, steam and water) in chosen nodes of the cycle, that is at the inlets and outlets of each of the machines. In addition the simulation module calculates the total power output, amount of fuel supplied to the steam boiler and the efficiency of electric power generation.

The efficiency of the gas turbine in terms of the lower heating value of the gas fuel.

\[ \eta_{GT} = \frac{N_{GT}}{m_{i,GT} \cdot LHV_{gas}} \]  

(1)

and for the steam cycle:

\[ \eta_{ST} = \frac{N_{ST}}{m_{i,ST} \cdot LHV_{gas}} \]  

(2)

The total efficiency of the electric power generation for the whole cycle is calculated for the total actual amount of the fuel supplied to the system:

\[ \eta = \frac{N_{GT} + N_{ST}}{(m_{i,GT} + m_{i,ST}) \cdot LHV_{gas} + m_{i,GT} \cdot LHV_{gas}} \]  

(3)

The symbols applied in the above equations are: \( m, i \) - mass flow rate and specific enthalpy, \( N_{GT}, N_{ST} \) - electric power output of the gas and steam turbine, \( \eta_{ST} \) - steam boiler efficiency, \( LHV_{gas} \) - lower heating value of the gas fuel.

Numbers refer to the cycle nodes according to the fig. 1.

In the design conditions the electric efficiency is 34.50% for the gas turbine and 33.48% for the steam cycle. The total electric efficiency is 42.46%.

4. Matching of the cycle sections in the off-design operation

The machines in the thermal cycle cooperate with each other. This cooperation is especially important when modeling the off-design operation since the alteration of one of the input parameters usually influences and changes other parameters in several nodes of the cycle [5,6]. These changes derive from the altered equilibrium state of the machines. The simulation module determines the equilibrium states for the machines, which match the equilibrium states of other machines in the cycle. This approach allows to calculate the values of thermal parameters in all nodes of the cycle.

The matching of three main sections is especially important in the analysis of the off-design operation for a gas turbine. These sections are: the compressor, combustor and expander. There are several parameters, which describe the operation of these sections but may also be treated as constrains, which force the sections to cooperate. The parameters are:

- Rotational speed. The expander drives the compressor through either one shaft or a transmission gear.
- Air flow. The amount of the exhaust gas depends on the amount of the inlet air to the compressor. The mass balance includes also the amount of the fuel gas, cooling air and flows through the seals.
- Pressures and temperatures. The expander inlet pressure depends on the compressor delivery pressure. The temperature of the air delivered to the combustor depends on the compressor isentropic efficiency and the turbine inlet temperature derives from the combustion process.

As the equilibrium states for the main sections are compared a single matching point may be determined, which uniquely identifies the off-design equilibrium. This point is usually plotted in the compressor characteristic. The equilibrium states for the main gas turbine sections derive from:

- The line of the compressor inlet guide vane angle. The compressor characteristic shows a dependency between the amount of the inlet air and the outlet pressure for various values of the IGV A.
- Mass and energy balance for the combustor. The compressor delivery pressure and its isentropic effectiveness determine the compressor outlet temperature. The mass and energy balance for the combustor determines the amount of air of this temperature, which is required to obtain the desired turbine inlet temperature for a given amount of the fuel.
- Absorption capacity for the expander. There is a strict relation between the pressures at the expander inlet and outlet and the amount of gas which is expanded.

As for the steam cycle, the off-design simulation matches the turbine sections with the regeneration system. The amount of steam extracted from turbine bleedings to feed the heat exchangers depends on the heat exchange conditions in the exchangers. These in turn depends on the pressure of the extracted steam.

5. Gas turbine adjustment to satisfy actual power demands

There are two basic method for an adjustment of a gas turbine, that is for fitting the current power output to the desired demand. The first one is to change the amount of fuel delivered to the combustor. It derives from the mass and energy balance that decreased amount of fuel results in lower turbine inlet temperature. Then the enthalpy drop in the expander is smaller and the output power decreases. Figure 2 presents such a power drop for the analyzed cycle.

The efficiency of the gas turbine drops as the amount of the fuel is decreased. This derives from the thermodynamics of the open gas cycles. This adjustment does not affect much the amount of compressor inlet air and delivery pressure.

The second method to adjust a gas turbine is the change of the compressor inlet guide vane angle (IGV A). The design applied to modern gas turbines allows a flexible change of the IGV A.

Figure 3 shows the matching points of the whole gas turbine plotted in the compressor characteristic. This characteristic relates the compressor pressure ratio and its flow factor for the inlet air. In addition this two parameters are shown as relative to the design values.

The characteristic shows the matching points for four distinct values of the IGV A including the design matching point. The decrease of the IGV A results in smaller amount of the inlet air and lower pressure ratio (see also fig. 5). On the other hand the
increase of the IGVA results in a larger amount of the inlet air but since the amount of the fuel remains unchanged the turbine inlet temperature significantly drops. As an effect the power output becomes smaller.

The alteration of the power output for the IGVA adjustment is shown in fig. 4. This figure presents also the change of the gas turbine electric efficiency.

If the amount of the inlet air is reduced due to small value of the IGVA but the fuel flow remains unchanged than the temperature of the exhaust gas from the combustion process significantly increases as shown in fig. 5. This may lead to a situation when the combustor outlet temperature exceeds its limiting value, which is extremely dangerous for the expander components in the first stage (blades and rotor). The limiting value is usually slightly higher than the design exhaust temperature. Therefore the IGVA adjustment should always be accompanied by a fuel adjustment. Only such simultaneous adjustment protects the expander against high inlet temperature.

The power output of the gas turbine changes also with the fuel lower heating value (LHV) when the amount of the fuel remains the same. However this method for the alteration of the power output cannot be treated as an adjustment for a standard operation.

6. The efficiency of the combined cycle for a varying ambient temperature

The ambient conditions have an essential influence on the regime of the operation. The gas turbine power output and efficiency depend on the ambient air temperature. Lower temperature results in higher power and better efficiency, which is shown for the analyzed cycle in fig. 6. The change of the temperature affects also the compressor pressure ratio and combustor outlet temperature (fig. 7) - even though the IGVA and the amount of the fuel remain both unchanged.

In case of the steam cycle the ambient conditions affect mainly the condenser pressure. Lower temperature of the cooling water allows to achieve lower pressure and therefore better efficiency and higher power output of the steam turbine as seen in fig. 6.

When the ambient air temperature drops down the total efficiency of the whole cycle rises at first but then starts to fall down. This happens because lower ambient temperature results in lower exhaust temperature at the gas turbine outlet. At some point it is necessary to turn on the afterburner in order to maintain the required exhaust temperature. However afterburner utilizes additional fuel, which counts against the efficiency.
It should also be emphasized that the ambient air temperature may change with quite high frequency but the temperature of the water cooling the condenser does not change as often.

7. The division of the load between the gas and steam sub-cycles

The analyzed system allows to divide the demanded power output between the gas and steam sub-cycle. The division may be completed according to various criteria. In addition the demanded level of the generated power may be achieved through various methods of turbine adjustment.

The following are the results of the sample analysis for a demanded power of 80 MW, which is 69.5 % of the design load. The first method of load division is performed for a constant combustor outlet temperature. First the inlet guide vane angle is set and then the amount of the fuel is adjusted in such a manner that the combustor outlet temperature does not exceed its design value (1150°C, table 1). This is repeated for various IGV As. The obtained consecutive matching points are plotted in fig. 8.

Additional assumptions are that the steam boiler may operate with the minimum 50% design load and that the HRSG generates as much steam as possible but with respect to the first assumption.

The load division is shown in fig. 9a. The sum of the power generated in the gas and steam turbines equals 80 MW as demanded. Figure 9b presents the efficiencies of both turbines and the whole cycle. The analysis of the efficiency proves that there is an optimal IGVA, which guarantees the highest total efficiency for this method of load division.

The total efficiency drops below the optimal value for smaller IGVAs because the afterburner has to be turned on. This is due to the small amount of the exhaust gas, too small to generate the requested amount of steam in the HRSG while keeping the assumptions of this method correct. Figure 10 shows the dependency between the IGVA and the amount of the fuel supplied to the combustor and afterburner.

The second method to divide the load between both turbines assumes that the afterburner is always turned off. Once again the IGV A is set firstly for the gas turbine. Then the amount of the fuel supplied to the combustor chamber is adjusted in such a manner that the afterburner is not required to operate and the total power output equals 80 MW. Also the assumption is still valid as for the minimum load in the steam boiler and that the HRSG carries as much load as possible.

Figure 11 demonstrates the gas turbine matching points obtained for this method of load division and various IGVAs. The power outputs and efficiencies are shown in fig. 12 for both turbines and the whole cycle.
The efficiency of the whole cycle increases for smaller inlet guide vane angles. However there is a constrain - the limiting value of the combustor outlet temperature. Figure 13 presents how the combustor outlet temperature changes for various IGVAs.

The plot in fig 13 clearly suggests that the angles lower than 0.7 of the design IGV A value cause the combustor outlet temperature to rise over the its limiting value. The increased temperature increases the metal temperature in the components of the expander, which are heated by the hot exhaust gas. Therefore the optimization of the load division must consider also the constrain described above.

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**Fig. 9.** Power (a) and efficiency (b) for the load division with constant combustor outlet temperature

**Fig. 10.** Amount of the gas fuel for constant combustor outlet temperature $(t_{204})$

**Fig. 11.** Gas turbine matching points for the cycle with the afterburner turned off

**Fig. 12.** Power (a) and efficiency (b) for the load division with the afterburner turned off
8. Conclusions

The simulation module for the modeling of the off-design operation presented in this paper allows to assess the influence of various operating regimes on the efficiency of the operation. This assessment determines the optimal method to perform the operation, that is most of all to divide the load between the turbines.

The analysis described here proved that high total efficiency may be achieved despite the decrease of the sub-cycles efficiencies resulting from the operation in the off-design regime. This derives from the fuel savings when gas and steam sub-systems are adjusted and better combined.

Fig. 13. Combustor outlet temperature for various IGVAs with the after-burner turned off

9. References


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