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OFF-DESIGN OPERATION OF THE MULTI-FUEL CHP CYCLES

Wojciech Kosman
Institute of Power Engineering and Turbomachinery
Silesian University of Technology
Gliwice, Poland

ABSTRACT

The paper describes the problems when the operation regime of the CHP cycle is adjusted according to the user demands. The analysis concerns a multi-fuel system, which utilizes biomass, natural gas and coal. The analyzed system consists of a combined gas and steam cycles.

The analysis presented here concerns off-design operation caused by part loading, varying ambient conditions and power demands. An appropriate division of the load between the sub-cycles increases the overall efficiency. The analysis concerns also various options of biomass supply to the system. The research regards mostly the effectiveness criterion as it has a direct influence on the costs of operation.

INTRODUCTION

Multi-fuel power cycles have become an significant alternative for conventional and combined heat and power plants. One main advantage is a fuel flexibility. Their design is usually fitted for specific user demands and in consequence the operation performs with optimized operation costs. The selection of the basic parameters (power output, amount of the generated heat) is strictly connected to the conditions under which a specified cycle is to operate.

This 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.

The issue of the off-design operation is well-known and studied for most of the machines, which are found in a typical combined cycles [1,2]. On the other hand the problem is not so obvious in case of the whole heat and power generating cycle [3-5]. The change in the operation of the various machines affects the performance of the whole cycle in different scale. This is due to the thermal arrangements and connections between machines in a CHP plant, which are location- and user- specific.

Another reason for the analysis of the off-design regime even during the design stage is that almost every CHP plant is expected to operate under varying heat and power demand throughout a year. Optimizing the performance for the maximal load is a common practice since then potential power losses would be the highest. Nevertheless lower load levels should also be optimized to avoid unnecessary losses, which over time could result in a considerable increase of the operation costs.

As mentioned earlier the optimization process must take into account the cooperation of the machines in the CHP plant [6]. It means that there is a need to establish part load characteristics not only for the single units (supplied by the manufacturer) but for the whole cycle.

The issue becomes especially important for the part loading at about 60 - 80 % of the design maximal load. For the lower levels some of the machines (turbines, boilers) may be switched off as the power demand is totally satisfied by the rest of the cycle. The tradeoff is lower efficiency when compared to the combined cycle. At the level of about 70 % all sub-cycles in a CHP plant must operate and so an appropriate load division becomes an essential task.

CHP plant users may complete load division through several approaches and also the assessment of the performance under off-design regime may be based on various criteria.

In the scope of the assessment of the part load operation the analysis described below bases on the efficiency criterion. Any losses appearing in the power cycle result in decreased efficiency. This in turn affects the operation costs, which derive mainly from the fuel consumption. Operations costs are usually monitored by the CHP plant users and therefore create an appropriate basis for the assessment of the operation.

The analysis regards several approaches when completing the load division. This approaches involve various control procedures for the gas and steam turbines as well as the limitations, which restrict the range of the possible off-design operation.

NOMENCLATURE

Symbols:

IGVA - relative compressor inlet guide vane angle [-],
H - total enthalpy [MW],
h - specific enthalpy [kJ/kg],
M - Mach number,
m - mass flow rate [kg/s],
N - electric power output [MW],
p - pressure [MPa]
T - temperature [ºC],
TIT - turbine inlet temperature [ºC],
η - efficiency [%].

Subscripts and superscripts:

aft - afterburner,
bio - biomass,
dim - dimensionless,
gas - gas fuel,
GT - gas turbine,
i - isentropic,
in - inlet,
out - outlet,
ref - reference (design),
ST - steam turbine,
tot - total,
* - critical.

ANALYZED CYCLE

The cycle under the investigation is a 50 MW steam turbine system expanded through the application of the 92.5 MW gas turbine. A heat recovery steam generator (HRSG) connects two sub-cycles. The basic configuration of the CHP plant is shown in Fig. 1. The flue gas from the gas turbine feeds the HRSG and allows to generate a part of the live steam for the steam turbine. A parallel coal boiler produces the rest of the live steam. Such arrangement allows better flexibility in terms of the fuel consumption.

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

The steam cycle consists of two turbine parts: a single-flow HP part and double-flow LP part. The cycle includes also exchangers for heat recovery. Steam bleedings in both parts of the steam turbine supply extracted steam to the recovery heat exchangers.

Two configurations of the cycle are further analyzed. They differ in the location of the main heat exchanger, which covers the heat demand. In the first option the exchanger is located in the steam sub-cycle. Steam taken from the outlet of the high pressure part of the turbine feeds the exchanger. This arrangement derives from the primary solution for the steam-only power system. The second configuration has the main heat exchangers located in the HRSG section, that is in the gas sub-cycle. Figure 1 presents both options for comparison purposes.

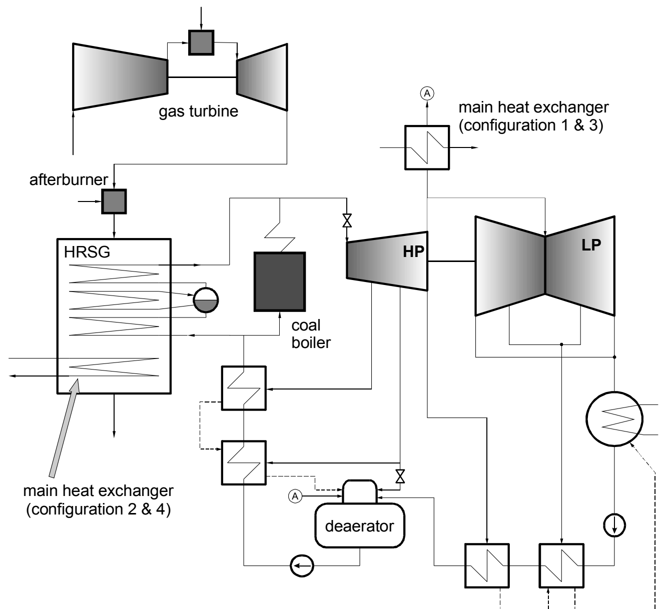

Fig. 1 Thermal configuration of the analyzed cycle

In addition there is a possibility to utilize a biomass fuel in the considered CHP plant. The biomass may be supplied either to the gas or steam sub-cycle. Figures 2 and 3 show the biomass utilization system.

The system shown in figure 2 is a hard coal boiler with biomass co-firing. The biomass is supplied to the same pulverizing mill as the coal.

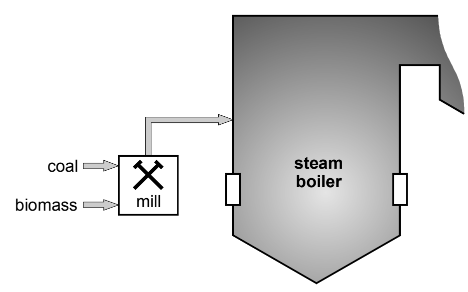

Fig. 2 Biomass supply to the coal boiler

Due to the form of the biomass the pulverized coal and biomass mixture is more coarse then a pulverized coal and therefore the fraction of the biomass admissible for co-firing is limited. This method allows to increase the biomass fraction up to 10 % in a hard coal boiler or even up to 40 % in case of a brown coal without any significant change in the boiler performance (meaning also, apart from the boiler efficiency, the NO_x, SO_2 and CO_2 emissions).

Another option for a biomass supply is the external combustor. This combustor may operate in atmospheric pressure but its application requires a modification of the gas turbine. The exhaust gas from the biomass combustor chamber feeds an additional heat exchanger, which raises the temperature of the air delivered by the compressor to the gas turbine combustor chamber. A gas sub-cycle modified according to the described solution is shown in Fig. 3.

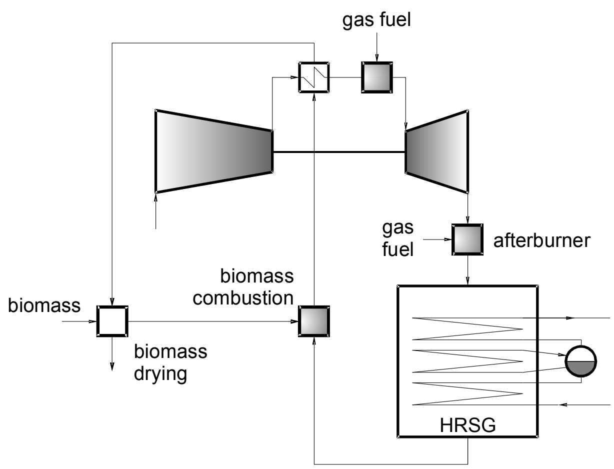

Fig. 3 Biomass supply to the gas sub-cycle

The additional exchanger may be viewed as a part of a heat recovery system.

The above description may be summarized in four distinct options of the cycle configuration:

1. Biomass supplied to the steam boiler, heat exchanger located in the steam sub-cycle.
2. Biomass supplied to the steam boiler, heat exchanger located in the HRSG.
3. Biomass supplied to the gas sub-cycle, heat exchanger located in the steam sub-cycle.
4. Biomass supplied to the gas sub-cycle, heat exchanger located in the HRSG.

OFF-DESIGN REGIME MODELING

The analysis of the operation under off-design regime requires appropriate approach towards the simulation of the operation of the cycle. A simulation module was designed for this purpose [6,7].

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

The basic assumption is that the off-design simulation requires only such input data, which in a real power plant derive from three categories:

1) user dependent parameters,
2) ambient dependent parameters,
3) health state.

For the analyzed cycle the first two sets (1 and 2) of the input data include:

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

It is worth to notice that this list does not include for example compressor delivery pressure, which is usually measured in gas turbines. This is because the user cannot set the pressure ratio directly but by setting the IGVA value. The simulation tool is built to model the

Apart from the above list the input includes health state data such as efficiency characteristics for the turbines, efficiency and pressure ratio maps for the compressor, temperature characteristics for the heat recovery exchangers and so on.

The next significant assumption regards the cooperation of the machines in the thermal cycle. This cooperation is especially important when modeling the off-design regime since the alteration of one of the input parameters usually influences and changes other parameters in several nodes of the cycle [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.

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

- **Rotational speed**. The turbine 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 turbine 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 simulation module compares the equilibrium states for the main sections it determines a single ***matching point***, 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 lines in the compressor characteristic show a dependency between the amount of the inlet air and the outlet pressure for various values of the IGVA.
- **Mass and energy balance for the combustor**. The compressor delivery pressure and isentropic efficiency 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 turbine**. There is a strict relation between the pressures at the turbine inlet and outlet and the amount of gas which is expanded.

The compressor characteristic is usually described as the dependency between the pressure ratio and the amount of air for various IGVA. The most appropriate for the numerical calculations is the dimensionless form defined for:

- dimensionless pressure ratio

$$\beta_{dim} = \frac{p_{out}/p_{in}}{p_{out,ref}/p_{in,ref}} \tag{1}$$

- dimensionless flow factor

$$m_{dim} = \frac{m_{in}\sqrt{T_{in}}}{p_{in}} \bigg/ \frac{m_{in,ref}\sqrt{T_{in,ref}}}{p_{in,ref}} \tag{2}$$

- dimensionless isentropic efficiency

$$\eta_{dim} = \frac{\eta}{\eta_{ref}} \tag{3}$$

The range of possible operation defined for a compressor has some restrictions. These are:

Surge limit - exceeding this restriction causes compressor to choke. The corresponding line that limits the compressor characteristic is usually located with some margin to the real surge limit to assure save operation.

IGVA limits - the lines corresponding to the maximal and minimal IGVA possible to set in the compressor.

Minimal efficiency - the line corresponding to the minimal efficiency calculated from the measurements of the analyzed compressor.

The area limited by the restrictions described above contains several points, which define the lines of the compressor characteristic. These points create a basic data used to approximate an arbitrary operation point.

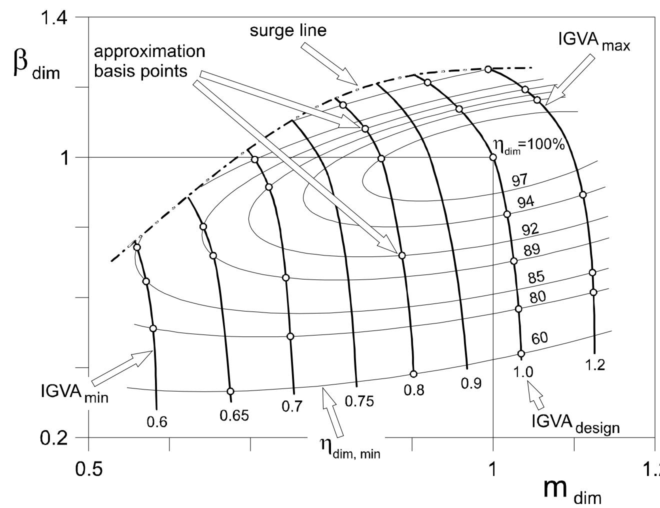

Fig. 4 Compressor characteristic

Figure 4 presents an example of the dimensionless compressor characteristic used in the calculations for the analyzed power plant. The splines were applied to approximate the lines from the set of points. This method incorporates polynomials of the third degree and approximates a line segment basing on control points obtained from the test measurements as the compressor is manufactured.

Since the gas turbine described here is a single shaft unit, its rotational speed cannot be adjusted. However, it is straightforward to enhance the compressor characteristic with the lines corresponding to the rotational speed.

An appropriate approach is also required for the turbine section with internal cooling. The amount of the gas expanding in the turbine changes in the consecutive stages as the coolant mixes with the main stream. The presence of the coolant (air from the compressor extractions) does not only lower the temperature in the turbine components but also affects the expansion line. The latter influence is essential for the performance analysis [6,7]. The relations for the absorption capacity in a cooled turbine is presented in the appendix.

As for the steam cycle, the simulation module distinguishes three main sections: the steam boiler, turbine and heat regenerators. The turbine is further divided into groups of turbine stages in such a manner that one group is located between steam extractions. The regeneration system is divided into separate heat exchangers.

The main task during off-design simulation is to match 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.

Since the simulation module divides the turbine into groups of stages, the description of the health state - taken as input data - also refers to these groups and not to the whole turbine. Therefore the efficiency characteristics and absorption capacity relations are given for consecutive stages not separated by any extraction.

The most difficult is the description of the first group of stages in a turbine. Located upstream of this group there is the steam distribution system and the control stage. In the turbines with the nozzle governing the steam distribution system is more complex than in turbines with the throttle governing. The expansion process is different.

The comparison of the expansion processes in a turbine with the nozzle and throttle governing is presented in Fig. 5.

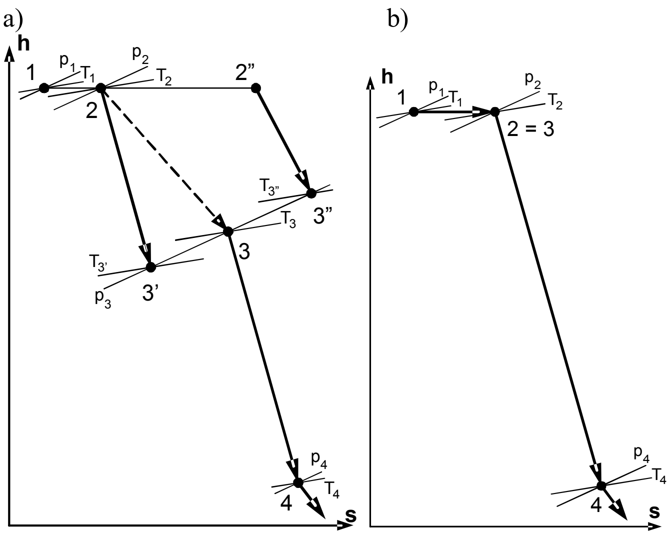

Fig. 5 Steam expansion in the HP part of a turbine with nozzle (a) and throttle (b) governing

Point 1 refers to the steam at the turbine inlet, while point 4 to the outlet of the first group of turbine stages (first extraction). Pressure p_3 is the pressure in the chamber of the control stage for the nozzle governing (Fig. 5a) or the pressure after the control valve for the throttle governing (Fig. 5b). Pressure p_2 refers to the steam after the control valve, so for throttle governing points 2 and 3 coincide.

The first group includes all the stages between points 3 and 4. A simplified yet very comfortable solution regards this group together with the steam distribution system and the control stage. This is more comfortable for the turbines with nozzle governing. The simplification leads then to the following definition of the isentropic efficiency for the first group of stages:

$$\eta_{i,1-4} = \frac{h_1 - h_4}{h_1 - h_{4ss}} \tag{4}$$

The efficiency characteristics are defined separately for each group of turbine stages. They are applied to the modeling process as a relations between the isentropic efficiency and amount of the steam expanding in the group of stages.

The off-design flow through a steam turbine may be described by the absorption capacity equation. This equation relates the pressures at the inlet and outlet of a group of stages and the amount of steam which is expanded in that group (constant amount since no extractions are included). In general form it may be expressed as a Stodola-Flügel relation:

$$\frac{m}{m_{ref}} = \sqrt{\frac{T_{in,ref}}{T_{in}}} \sqrt{\frac{\varepsilon_{in}^2 - \varepsilon_{out}^2}{\varepsilon_{in,ref}^2 - \varepsilon_{out.ref}^2}} \tag{5}$$

where $\varepsilon_i = p_i / p_{0,ref}$ and $p_{0,ref}$ is a chosen characteristic pressure value, for example the maximal pressure level at the turbine inlet.

If the transonic flow occurs in the off-design regime then the absorption capacity equation becomes:

$$\frac{m}{m_{ref}} = \sqrt{\frac{T_{in,ref}}{T_{in}}} \sqrt{\frac{\varepsilon_{in}^2 - \varepsilon_{out}^2 - a\left(\varepsilon_{in} - \varepsilon_{out}\right)^2}{\varepsilon_{in,ref}^2 - \varepsilon_{out,ref}^2 - a\left(\varepsilon_{in,ref} - \varepsilon_{out,ref}\right)}} \tag{6}$$

where $a = \varepsilon_* / (1 - \varepsilon_*)$ and ε_* is such a pressure ratio, under which the critical velocity occurs in a stage (usually in the last stage of the group).

Further assessment of the off-design regime bases on three efficiencies defined as follows:

- gas turbine electric efficiency

$$\eta_{GT} = \frac{N_{GT}}{H_{expander,in} - H_{compressor,out}} \tag{7}$$

- steam cycle efficiency

$$\eta_{ST} = \frac{N_{ST}}{H_{live\,steam} - H_{feed\,water}} \tag{8}$$

- total cycle efficiency

$$\eta_{tot} = \frac{N_{GT} + N_{ST} + Q}{m_{gas}\, LHV_{gas} + m_{coal}\, LHV_{coal} + m_{bio}\, LHV_{bio}} \tag{9}$$

It is worth to emphasize here that the steam cycle efficiency is defined in reference to the amount of energy supplied to the steam cycle and not to the energy in fuels as the livesteam is produced in both, the HRSG and coal boiler. As a result steam cycle efficiency does not include boiler and HRSG efficiencies and is higher than the gas turbine electric efficiency.

LIMITATIONS FOR THE CHP PLANT ADJUSTMENT

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. Decreased amount of fuel results in lower turbine inlet temperature (TIT). In consequence the enthalpy drop in the turbine is smaller and the output power decreases. Figure 4 presents such a power drop for the turbine in 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. Fuel amount cannot be increased too much above the design value since then the temperature at the turbine inlet would exceed its limiting value. Such situation would be extremely dangerous for the turbine components in the first stage (blades and rotor). The limiting temperature value is usually slightly higher than the design TIT value.

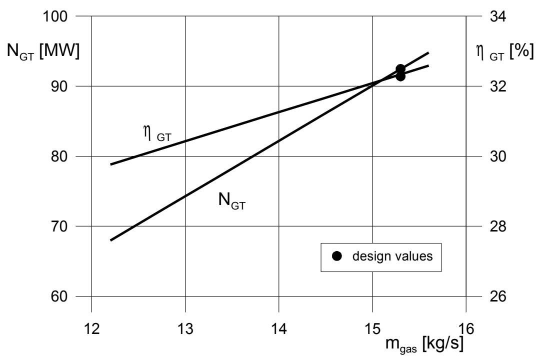

Fig. 4 Gas turbine efficiency and power output for the fuel adjustment

This adjustment does not affect much the amount of the compressor inlet air.

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

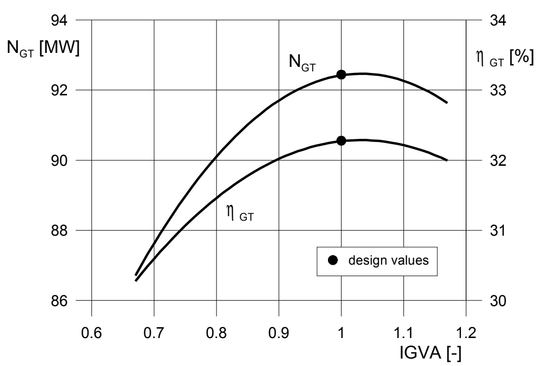

Fig. 5 Gas turbine efficiency and power output for the IGVA adjustment

The decrease of the IGVA results in smaller amount of the inlet air and lower pressure ratio, which leads to lower power output. 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 combustor outlet temperature significantly drops. As an effect the power output also becomes smaller.

The alteration of the power output for the IGVA adjustment is shown in Fig. 5. 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. 6. Therefore again the combustor outlet temperature becomes the limiting value for this kind of adjustment. The IGVA adjustment is always proceeded together with the simultaneous fuel adjustment.

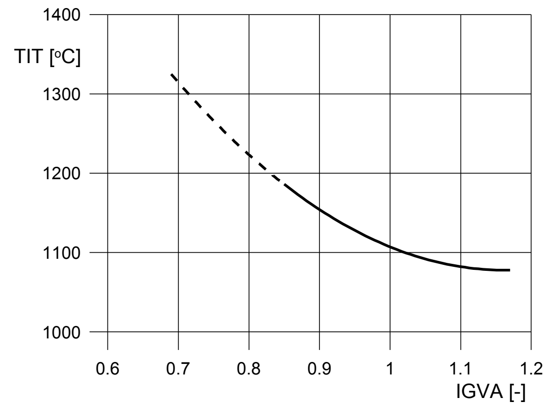

Fig. 6 Combustor outlet temperature for the IGVA adjustment

Further limitations derive from the absorption capacity of the turbine (as mentioned earlier) and the choke line for the compressor. However addition limitations arise when a gas turbine cooperates in a CHP cycle. In the analyzed cycle the gas turbine must generate an appropriate amount of exhaust gas to feed the HRSG. The conditions of the heat exchange in the HRSG may be assessed through the ***pinch point*** analysis [8]. Also the exhaust temperature at the HRSG outlet must not drop below the temperature of sulfur acid saturation in order to avoid the sulfur corrosion.

PART LOAD DIVISION - METHOD 1

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 electrical power of 105 MW, which is 70 % of the design power output. The simulations are presented for two locations of the system heat exchanger (in the HRSG or connected to the steam turbine) and for two options of the biomass supply. Together this gives four analyzed cases.

The first method of load division is performed under a constant combustor outlet temperature. This is due to the gas turbine control system, which tends to maintain the highest possible TIT for every IGVA. The reason for this is that higher TIT assures better gas turbine efficiency.

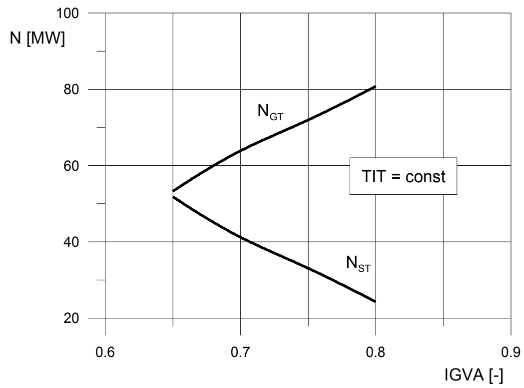

Fig. 7 Configuration 1: part load division with the TIT = const

Figure 7 shows the part load division for this method of the CHP adjustment. This graph corresponds to the configuration option number 1. The load at 105 MW may be achieved for the values of the IGVA within the range 0.65 - 0.8. Larger IGVA would require a level of power generation too low for the steam turbine. Lower IGVA would cause the gas turbine to choke.

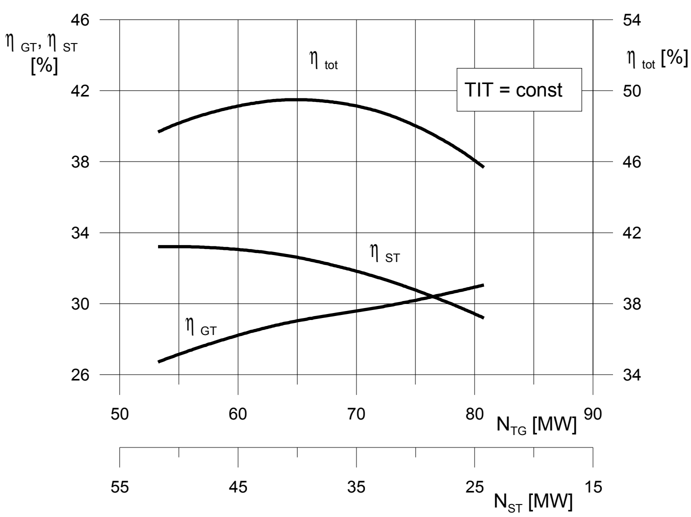

Fig. 8 Configuration 1: efficiencies for part load division with the TIT = const

The efficiencies of the gas and steam sub-cycles as well as the total cycle efficiency are plotted in Fig. 8. All the values refer to the configuration option number 1.

The simulations prove that there exists an optimal load division for this method of the CHP adjustment.

The calculations for the configuration option number 2 provide similar results. The efficiencies for the cycle with the main heat exchangers located in the HRSG section are presented in Fig. 9.

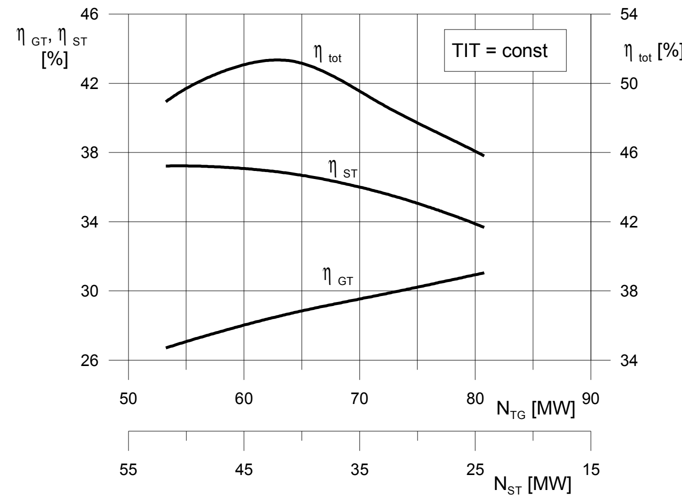

Fig. 9 Configuration 2: efficiencies for part load division with the TIT = const

While the optimal load division is very similar for both configurations, the second one assures higher total efficiency. Since the main heat exchanger is relocated from the steam sub-cycle into the HRSG in the latter configuration, the usage of the energy supplied to the system in fuels is better.

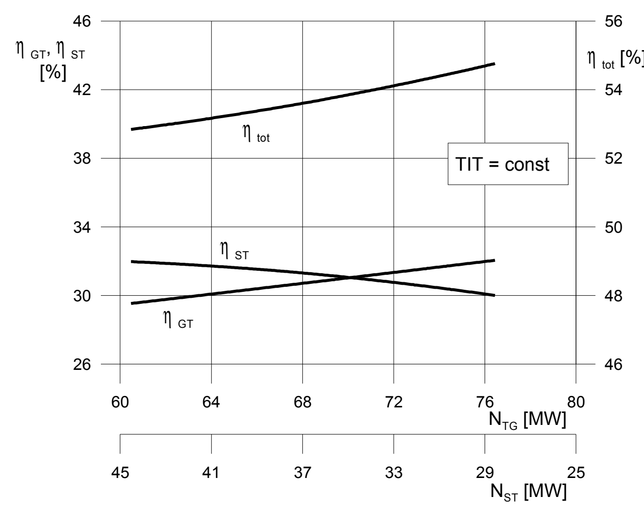

Fig. 10 Configuration 3: efficiencies for part load division with the TIT = const

The lines of the efficiencies are totally different for the cycle with the biomass supplied to the gas sub-system. The efficiencies for the configuration options 3 and 4 are presented in Fig. 10 and 11 respectively.

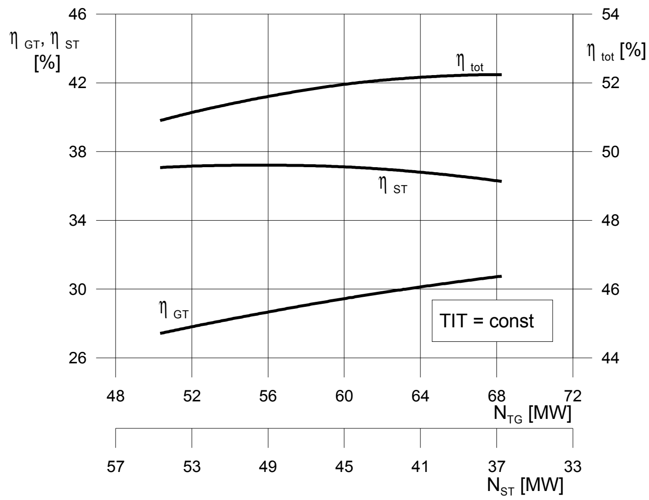

Fig. 11 Configuration 4: efficiencies for part load division with the TIT = const

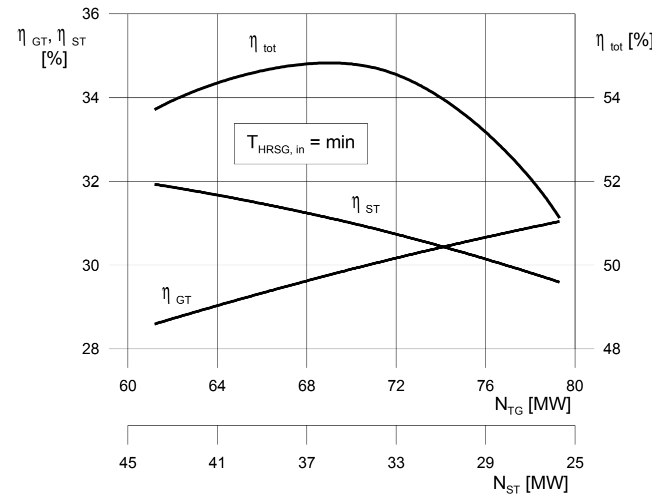

Fig. 12 Configuration 1: efficiencies for part load division with the $T_{HRSG,in}$ = min

There is no clear optimal load division within the analyzed range. However the load division is restricted here by the steam turbine. The cycle operates with the maximal total efficiency when it reaches the minimum value of the steam turbine load.

PART LOAD DIVISION: METHOD 2

The second method of load division allows to decrease the combustor outlet temperature to better fit the heat recovery steam generator. The pinch point analysis conducted for the HRSG estimates the minimal temperature of the exhaust gas required at the HRSG inlet to produce the necessary amount of live steam and hot water. Then the amount of fuel directed into the gas turbine combustor may be set according to the above restriction. This method ensures that there is very little margin of the unused energy.

Table 1: Range of the IGVA for load division according to method 2

Cycle configuration	IGVA
1	0.75 - 0.9
2	0.6 - 0.9
3	0.75 - 1
4	0.7 - 0.8

In this case the cycle configuration has an essential influence on the range of possible part load adjustment. Table 1 lists the values of the inlet guide vane angle, which may be set when applying the above method. There are configurations which allow a wide range of possible IGVA values but also such that narrow the CHP adjustment.

Figure 12 shows the efficiencies for the configuration number 1 adjusted with the temperature of the exhaust gas at the HRSG inlet set to minimal value. When compared with the previous method for the part load adjustment (fig. 8) it is clear that the second method provides higher total efficiency. This is because the gas and steam sub-cycle become more strictly combined through the HRSG.

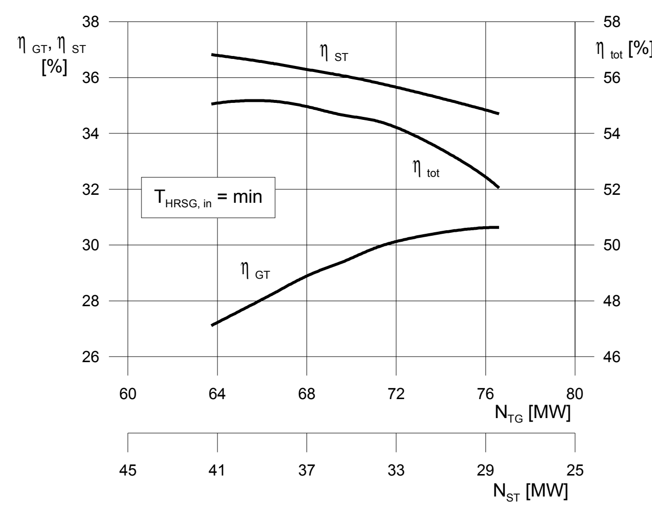

Fig. 13 Configuration 2: efficiencies for part load division with the $T_{HRSG,in}$ = min

The simulations show the same trend for the configuration option number 2. The total efficiency increases yet the optimal load division still exists (fig. 13). It should be emphasized here that - as the picture shows - the efficiencies of the sub-cycles do not change in a significant manner.

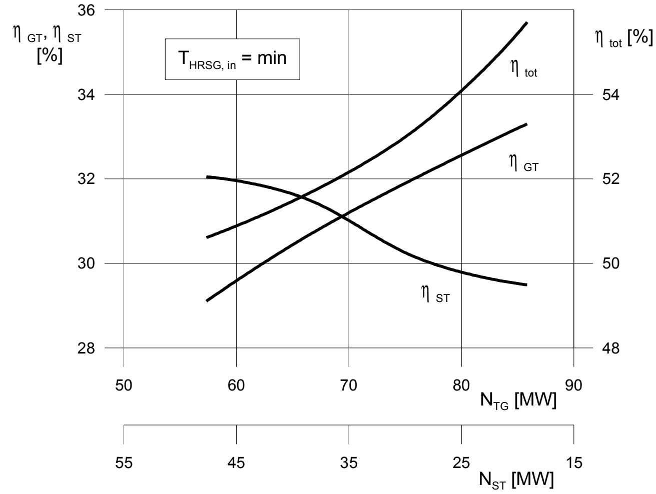

Fig. 14 Configuration 3: efficiencies for part load division with the $T_{HRSG,in}$ = min

Figures 14 and 15 present the efficiencies for configurations 3 and 4 respectively. As for method 1 the lines of the total efficiency have no maximal point. Again the optimal load division corresponds to the minimal load of the steam turbine.

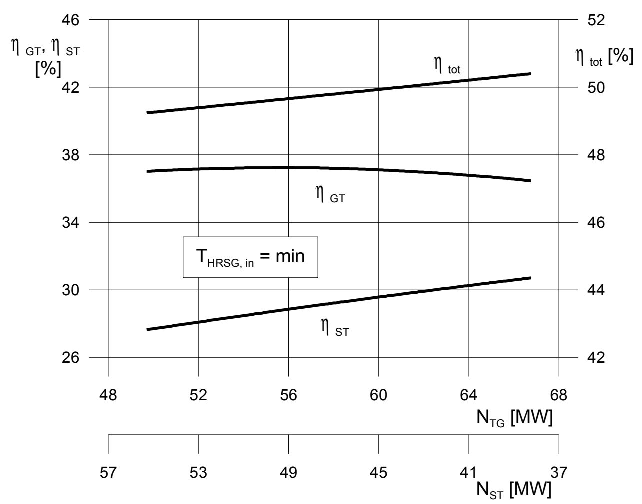

Fig. 15 Configuration 4: efficiencies for part load division with the $T_{HRSG,in}$ = min

None of the above simulations required the afterburner to operate. The demanded load was small enough to be satisfied by the gas and steam turbine without a supplementary firing.

INFLUENCE OF THE AMBIENT CONDITIONS

The ambient conditions have an essential influence on the regime of the operation. The gas turbine power output and efficiency increase for lower ambient air temperature even though the IGVA and the amount of the fuel remain both unchanged.

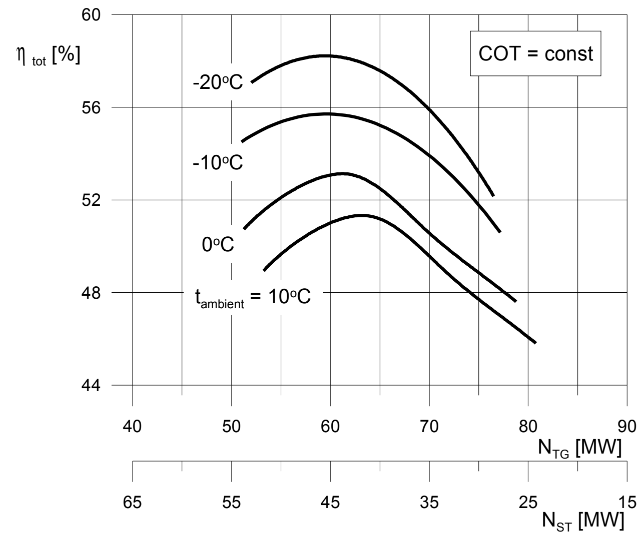

Fig. 16 Optimal load division for various ambient temperature

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 in the condenser and in the result better efficiency and higher power output of the steam turbine.

The results of the simulation (fig. 16) show that not only the efficiency increases for lower ambient temperature (as expected) but also the optimal part load division is obtained for different turbines adjustment.

CONCLUSIONS

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. Optimal load division does not always correspond to the optimal loading for a single units. This derives from the fuel savings when gas and steam sub-systems are adjusted and better combined.

The optimization of the load division strongly depends on the thermal configuration of the CHP plant. Therefore a separate analysis for the off-design regime is required for each specific multi-fuel cycle, yet it may be performed using the modeling tool presented in this paper.

APPENDIX: COOLED EXPANSION IN GAS TURBINE

The internal cooling affects the main stream mainly by mixing. Heat transfer caused by internal cooling may be neglected when analyzing expansion of the main stream. Therefore, for the purpose of determining the off design performance, it is sufficient to model the internal cooling as mixing of air and main stream.

The composition of exhaust gases in gas turbines is usually similar to the composition of air. Perfect gas analysis may assume equal values of specific heat and isentropic exponent for the main stream and coolant. Thus the energy balance of mixing at the outlet of a preceding row of blades (i-1) may be written as:

$$T'_{i-1,out} + \alpha_{i-1} T_a = (1 + \alpha_{i-1}) T_{i-1,out} \tag{A.1}$$

where $T'_{i-1,out}$ is the temperature of the main stream before it mixes with the coolant. The flow factors (A.1) in two consecutive rows of blades are related by the equation similar to (A.2):

$$m_{i,dim} = (1 + \alpha_{i-1}) m_{i-1,dim} \left(\frac{\overline{T}_{i,in}}{T_{i,in}} \frac{T_{i-1,out}}{T'_{i-1,out}} \frac{T'_{i-1,out}}{T_{i-1,in}} \frac{T_{i-1,in}}{\overline{T}_{i-1,in}} \right)^{\frac{1}{2}} \ldots$$

$$\ldots \frac{\overline{p}_{i-1,in}}{p_{i-1,in}} \frac{p_{i-1,in}}{p_{i-1,out}} \frac{p_{i,in}}{\overline{p}_{i,in}} \tag{A.2}$$

where total and static parameters are linked by the relations:

$$\frac{\overline{T}}{T} = 1 + \frac{k-1}{2} M^2 \tag{A.3}$$

$$\frac{\overline{p}}{p} = \left(1 + \frac{k-1}{2} M^2 \right)^{\frac{k}{k-1}} \tag{A.4}$$

The above equations are then inserted into (A.2) together with the equation of polytropic expansion between the inlet and outlet of the preceding row. As a result (A.2) becomes:

$$m_{i,Z} = (1 + \alpha_{i-1}) m_{i-1,Z} \left(\frac{1 + \frac{k-1}{2} M^2_{i-1,in}}{1 + \frac{k-1}{2} M^2_{i,in}} \right)^{\frac{k+1}{2(k-1)}} \left(\frac{p_{i-1,in}}{p_{i,in}} \right)^{\frac{n+1}{2n}} \tag{A.5}$$

Using the above equation in a recursive manner leads to a similar relation between any two flow factors for arbitrary rows of blades. It is the most convenient to relate an arbitrary flow factor with the one for the first row of blades:

$$m_{i,dim} = m_{1,dim} \left(\frac{1 + \frac{k-1}{2} M^2_{1,in}}{1 + \frac{k-1}{2} M^2_{i,in}} \right)^{\frac{k+1}{2(k-1)}} \left(\frac{p_{1,in}}{p_{i,in}} \right)^{\frac{n+1}{2n}} \ldots$$

$$\ldots \prod_{j=1}^{i-1} \left\{ (1 + \alpha_j) \left[1 + \alpha_j \left(1 - \frac{T_a}{T_{j,out}} \right) \right]^{\frac{1}{2}} \right\} \tag{A.3}$$

This equation relates the main stream parameters at the inlet of a group of stages and the parameters at any chosen cross-section along the turbine.

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