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ENERGETIC ANALYSIS OF A BIOGAS COMBINED CYCLE / CHP SYSTEM BASED ON ORENDA OGT2500 GAS TURBINE

BY

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Abstract. The paper presents the results of a study concerning Power and heat generation with biogas in CHP / Combined Cycle systems based on ORENDA OGT2500 Gas Turbine. Actually, the energetic analysis of three thermodynamic schemes (Gas Turbine Engine + Hot Water Boiler, Gas Turbine Engine + Steam Turbine Engine, Gas Turbine Engine + Steam Turbine Engine + Hot Water Boiler) had been made.

Key words: Biogas, Combined Cycle, CHP, home made code.

1. Introduction

Biogas is an important source of energy, which can be exploit for heat or / and power generation. The capacity of CHP biogas plants ranges typically from less than 250 kWe to 2.5 MWe, with conversion efficiencies to electricity of 32…45% (JRC-IET, 2011). There are also biogas plants over 2.5 MWe, but they are uncommon. In this category can be mentioned: three units of 4.5…4.6 MWe

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(Racine / Wisconsin / US, East Bay / Oakland / US and Istanbul / Turkey), based on Mercury or Centaur 50 Solar Gas Turbine; one unit of 12 MWe in Toledo / Ohio / US, based on a Taurus 60 Solar Gas Turbine; one Combined Cycle (CC) Plant of 15 MWe installed outside Paris / France, that uses one Mars 100 Solar Gas Turbine; one 35 MWe CC Plant, in operation from 1984, serving Los Angeles Country / US, which uses three Mars Solar Gas Turbines (COSPP, 2012).

The capital cost of a biogas plant with a gas engine or turbine is estimated to be in the range of €2500–5000/kWe (Van Tilburg, 2008).

In the present paper are energetically analyzed three different configurations (thermodynamic schemes) of CHP/CC Biogas systems based on Orenda OGT2500 Gas Turbine. Operation with biogas of this Gas Turbine in simple scheme was discussed in (Bălănescu *et al*., 2012).

2. Analyzed Configurations. Methodology of Analysis

There are defined three basic components (Fig. 1): Gas Turbine Engine (GTE), Steam Turbine Engine (STE) and Hot Water Boiler (HWB). Based on these components, the three analyzed configurations are:

− Configuration 1: GTE + HWB
− Configuration 2: GTE + STE
− Configuration 3: GTE + STE + HWB

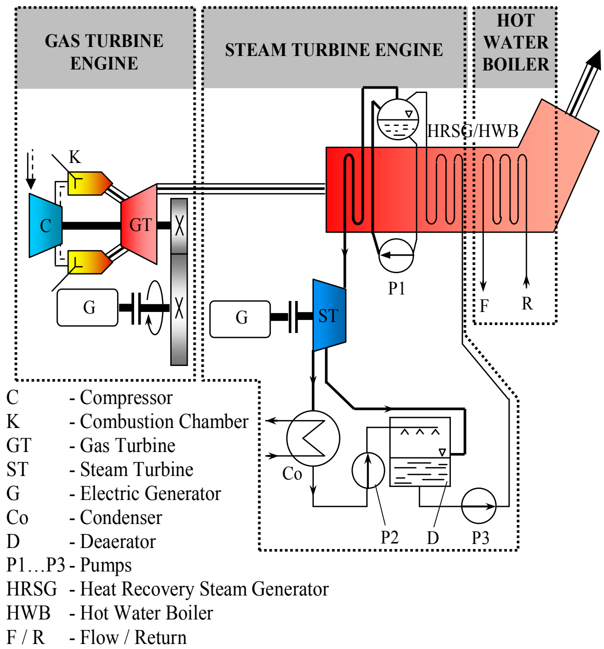

Fig. 1 – Analized CHP/CC system – general scheme.

The energetic analysis of the three configurations was made by using the home made code BIOTURBO.

In order to proceed with energetic analysis, the assumptions from (Bălănescu *et al.*, 2012) were made. Besides, the following values are assumed:

− heat transfer efficiency (for HRSG/HWB heat exchangers): $\eta_x = 0.95$
− HRSG feed water temperature: $t_a = 100$°C
− bled steam pressure: $p_{BS} = 1.2$ bar
− condensing pressure: $p_C = 0.05$ bar
− pinch point (minimum admitted value): $\Delta t_p = 25$ deg
− steam turbine isentropic efficiency: $\eta_S = 0.86$
− steam turbine mechanical efficiency: $\eta_m = 0.99$

As long there is no supplemental firing, HRSG inlet gas temperature and GT exhaust gas temperature (t_E) are equal.

The nominal values of steam absolute pressure (p_s) and temperature (t_s) as well the nominal temperatures of water – flow (t_{we}) and return (t_{wi}) –, were defined in accordance with STAS 2764-86.

Characteristic parameters of GTE, namely gas mass flow (G_g), power output (P_{GTE}), fuel consumption (FC), and efficiency (η_{GTE}) are calculated using the procedure presented in (Bălănescu *et al.*, 2012). Besides, by using procedures from (Ishigai, 1999; Drbal, 1996) there are calculated the following parameters:

• STE output (configurations 2 and 3)

$$P_{STE} = \left[G_s \cdot \left(i_S - i_{BS}\right) + G_{co} \cdot \left(i_{BS} - i_{co}\right)\right] \cdot \eta_m \quad [\text{kW}], \tag{1}$$

where: G_{co} − steam mass flow on the steam turbine exhaust, [kg/s]; i_S − steam turbine inlet enthalpy, [kJ/kg]; i_{BS} − bled steam enthalpy, [kJ/kg]; i_{co} − steam turbine exhaust enthalpy, [kJ/kg]; G_s − steam mass flow delivered by HRSG, [kg/s]; is calculated with following formula:

$$G_s = \frac{Q_u}{i_S - i_w} = \frac{G_g \cdot \left(i_i - i_e\right) \cdot \eta_x}{i_S - i_w} \quad [\text{kg/s}]. \tag{2}$$

In eq. (2) we denoted: Q_u − heat absorbed in HRSG, [kW]; i_w − HRSG feed water enthalpy, [kJ/kg]; i_i − HRSG inlet enthalpy, [kJ/kg]; i_e − HRSG exhaust enthalpy, [kJ/kg].

• STE efficiency (configurations 2 and 3)

$$\eta_{STE} = \frac{P_{STE}}{G_g \cdot \left(i_i - i_e\right)} \cdot 100 \quad [\%] \tag{3}$$

• Heat absorbed in HWB (configurations 1 and 3)

$$Q_{HWB} = G_g \cdot \left(i_e - i_f\right) \cdot \eta_x \quad [\mathrm{kW}], \tag{4}$$

where i_f is HWB exhaust (flue gas) enthalpy, [kJ/kg].

• Hot water production (configurations 1 and 3)

$$G_w = \frac{Q_{HWB}}{i_{we} - i_{wi}} \quad [\mathrm{kg/s}], \tag{5}$$

where i_{we}, i_{wi} are HWB water outlet / inlet enthalpies, [kJ/kg].

• Overall output (electrical + thermal) of the plant

$$P = P_{GTE} + P_{STE} + Q_{HWB} \quad [\mathrm{kW}] \tag{6}$$

• Electrical efficiency of the plant

$$\eta_{el} = \frac{3600 \cdot \left(P_{GTE} + P_{STE}\right)}{FC \cdot LHV} \cdot 100 \quad [\%] \tag{7}$$

• Specific fuel consumption of the plant

$$SFC = FC / P \quad [\mathrm{kg/kWh}] \tag{8}$$

• Overall efficiency (electrical + thermal) of the plant

$$\eta = \frac{3600 \cdot P}{FC \cdot LHV} \cdot 100 \quad [\%] \tag{9}$$

Obviously, in the case of configuration 1, $P_{STE} = 0$ in eqs. (6) and (7), while $\eta_{el} = \eta_{GTE}$ and $i_e = i_i$. In the case of configuration 2, $Q_{HWB} = 0$ in eq. (6) while $\eta = \eta_{el}$ and $i_f = i_e$.

3. Analysis of the Results

The results of the study are presented in Tables 1÷3.

Table 1
Characteristic Parameters for Configuration 1 (*GTE* + *HWB*)

Parameter	Values					
P_{GTE}, [kW]	3043.6					
$\eta_{GTE} = \eta_{el}$, [%]	28.8					
G_g, [kg/s]	15.15					
t_E, [°C]	451					
FC, [m^3_N/h]	1921.2					
t_f, [°C]	110					130
t_{we} / t_{wi}, [°C]	90/70	120/70	130/70	150/70	160/70	130/90
Q_{HWB}, [kW]	5286.4					4987.6
Q_{HWB}, [Gcal/h]	4.54					4.29
G_w, [t/h]	226.4	90.2	75.1	56.1	49.8	106
η, [%]	**74.1**					71.4
SFC, [m^3_N/kWh]	0.231					0.239

Table 2
Characteristic Parameters for Configuration 2 (*GTE* + *STE*)

Parameter	Values				
P_{GTE}, [kW]	3010				
η_{GTE}, [%]	28.5				
G_g, [kg/s]	15.15				
t_E, [°C]	453				
FC, [m^3_N/h]	1921.2				
$t_e = t_f$, [°C]	178	179	180	183	183
p_s, [bar]	15.7	16.7	15.7	16.7	15.7
t_s, [°C]	250	250	300	320	350
p_{sat}, [bar]	16.5	17.6	16.5	17.6	16.5
t_{sat}, [°C]	203	206	203	206	203
G_s, [t/h]	6.19	6.18	5.87	5.72	5.58
P_{STE}, [kW]	1245.5	1252.6	1257	1264	1267.2
η_{STE}, [%]	27.5	27.8	28	28.4	28.5
P, [kW]	4255.5	4262.6	4267	4273.9	4277.2
$\eta = \eta_{el}$, [%]	**37.8**	**37.9**	**37.9**	**38**	**38**
SFC, [m^3_N/kWh]	0.451	0.451	0.45	0.45	0.449

Table 3

Characteristic parameters for Configuration 3 (*GTE* + *STE* + *HWB*)

Parameter	Values				
P_{GTE}, [kW]	2984.7				
η_{GTE}, [%]	28.2				
G_g, [kg/s]	15.15				
t_E, [°C]	455				
FC, [m^3_N/h]	1921.2				
t_e, [°C]	177	179	180	183	182
p_s, [bar]	15.7	16.7	15.7	16.7	15.7
t_s, [°C]	250	250	300	320	350
p_{sat}, [bar]	16.5	17.6	16.5	17.6	16.5
t_{sat}, [°C]	203	206	203	206	203
G_s, [t/h]	6.24	6.21	5.91	5.75	5.63
P_{STE}, [kW]	1256.9	1259.7	1264.2	1271.2	1279.1
η_{STE}, [%]	27.5	27.8	28	**28.4**	**28.5**
P, [kW]	4241.7	4244.5	4248.9	4255.9	4263.8
η_{el}, [%]	**37.7**	**37.7**	**37.8**	**37.8**	**37.9**
t_f, [°C]	110				
t_{we} / t_{wi}, [°C]	90 / 70				
Q_{HWB}, [kW]	1003.6	1033.9	1049	1094.4	1079.3
Q_{HWB}, [Gcal/h]	0.863	0.889	0.902	0.941	0.928
G_w, [t/h]	43	44.3	44.9	46.9	46.2
η, [%]	46.6	46.9	47.1	**47.6**	**47.5**
SFC, [m^3_N/kWh]	0.366	0.364	0.363	0.36	0.36

Excepting p_{sat} and t_{sat}, which are the pressure (absolute) and temperature of steam at saturation state (in the drum), all parameters from Tables 1÷3 had been described above.

It can be observed that P_{GTE}, η_{GTE} and t_E change their values when configuration is changed. Compared with GTE in single operation (Bălănescu *et al*., 2012), P_{GTE} and η_{GTE} have lower values in all three cases while t_E is higher. This indicates an output loss of GTE, induced by a shorter expansion. The loss is caused by the gasodynamic resistance of HRSG or / and HWB, which must be passed by GTE exhaust gases.

Analyzing the results presented in the tables above it can't be concluded that one configuration is the best choice in any conditions. The highest overall efficiency and, consequently, the lowest specific fuel consumption are obtained in the case of Configuration 1. But, as long the internal energy consumption (electrical and thermal) of an Anaerobic Digestion plant is about 22% for dry process (dry mass: < 15%) and 37% for wet process (dry mass: 20…40%), Configuration 1 can be the most attractive only if there are heat consumers for the rest of thermal energy.

When major demand is for power, Configurations 2 and 3 are the most interesting; by adding a STE to ORENDA OGT2500 the electrical efficiency of the system increases with 9.3 ÷ 9.7%.

The values from Table 2 and Table 3 indicate that variation of the steam parameters in the assumed range has no significant influence over the parameters of the CC systems: variation of η_{STE} is maximum 1% while variation of η_{el} is maximum 0.2%.

5. Conclusions

The balance of both heat and power demand is the only criteria that decides optimum CHP / CC configuration.

The highest overall efficiency (74.1%) is offered by Configuration 1 when the proper heat consumption ensures heat generation at the full capacity of 4.54 Gcal/h. Obviously, SFC has minimum value in this case, namely 0.231 m^3_N /kWh. GTE output loss caused by the gasodynamic resistance of HWB is 6.4 kW, which means 0.1%.

If only power is required, Configuration 2 is the best option. In this case, electrical efficiency is about 38% and SFC = 0.45 m^3_N/kWh, regardless of the steam parameters as long they are kept in the assumed range. The use of STE rises the electrical efficiency with 9.3 ÷ 9.7%. GTE output loss caused by the gasodynamic resistance of HRSG is 40 kW, which means 0.4%.

If major demand is for power but heat is also required, Configuration 3 is the most attractive. In this case, electrical efficiency reaches 37.9%, overall efficiency is 47.5%, SFC = 0.36 m^3_N/kWh and HWB delivers 0.93 Gcal/h. GTE output loss caused by the gasodynamic resistance of HRSG + HWB is 65.3 kW, which means 0.7%.

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ESTIMAREA PERFORMANȚELOR UNUI SISTEM ENERGETIC COGENERATIV / CU CICLU MIXT GAZE-ABUR AVÂND LA BAZĂ TURBOMOTORUL CU GAZE ORENDA OGT2500 FUNCȚIONÂND CU BIOGAZ

(Rezumat)

În lucrare este analizată posibilitatea producerii energiei electrice şi termice prin arderea biogazului în instalații energetice cogenerative / cu ciclu mixt gaze-abur având la bază turbomotorul cu gaze ORENDA OGT2500. Practic, sunt analizate, din punct de vedere energetic, trei posibile variante de instalație: turbomotor cu gaze + cazan recuperator de apă caldă, turbomotor cu gaze + turbomotor cu abur, turbomotor cu gaze + turbomotor cu abur + cazan recuperator de apă caldă. Sunt prezentate rezultatele acestei analize energetice.