EXERGY ANALYSIS OF A BIOMASS COGENERATION SYSTEM

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This paper is dedicated to energy and exergy analysis of a usual biomass cogeneration system with a thermal power in the range from 100 to 300 kW. The analyzed system consists of a downdraft biomass gasifier, a producer gas cleaning system and a gas engine. The heat exchangers in the system use the sensible heat of producer and flue gases and waste heat from the gas engine to heat water in a district heating system 90/70 0C and for a domestic hot water supply 25/60 0C. The system has the overall electrical efficiency of 22.04%, whereas the total efficiency is 96.72% due to the omission of the system heat loss. The exergy efficiency of the system is barely 35.83%. The reason for this are the irreversibilities created in the gas engine, gasifier and in the heat exchanger network. The system efficiency can be improved by the use of waste heat to preheat gasifying air, to produce gasifying steam, or to perform biomass pyrolysis prior to its gasification. These measures enable production of a more valuable producer gas with a higher calorific value and a smaller amount of nitrogen. The use of a better gas produces higher temperature in the gas engine, which result in a higher gas engine efficiency.

Key words: biomass gasification, exergy, cogeneration, efficiency

0. INTRODUCTİON

High cost of energy and limited energy resources for the present knowledge require the improvement of energy efficiency of the existing energy transformation processes. This together with the need for the use of electrical energy as a final energy led to the development of cogeneration plants, which produce electricity and useful heat from the same plant. The introduction of a feed-in tariff system for the electricity produced from renewable sources spark many activities in developing biomass cogeneration plants in Serbia.

Karl [1] cited by Karellas et al. [2] gave typical power ranges and efficiencies of currently available power systems. Figure 1 shows that gas engines, microturbines and fuel cells are well-suited systems for small scale power production because of their high efficiency at low-power ranges. These technologies are only compatible with gaseous fuels. This is the reason for developing better biomass gasification systems. Gas engines is the most suitable technology for electricity production in the middle-power range, from a few kW up to 10 MW. For electricity production in large-power ranges of more than 10 MW combine process are the most desirable option. Ahrenfeldt et al. [3] give the state-of-the-art of biomass gasification and cogeneration technology with the most important pilot and commercial products for the future.

The availability of biomass in Serbia, the size of lumber industry and agriculture facilities make the biomass cogeneration system in the range from 100 to 300 kW very interesting. The most common are plans of the biomass cogeneration system shown in Fig. 2. The goal of this paper is to analyze the biomass cogeneration system shown in Fig. 2 and to recommend the measures for its improvement.
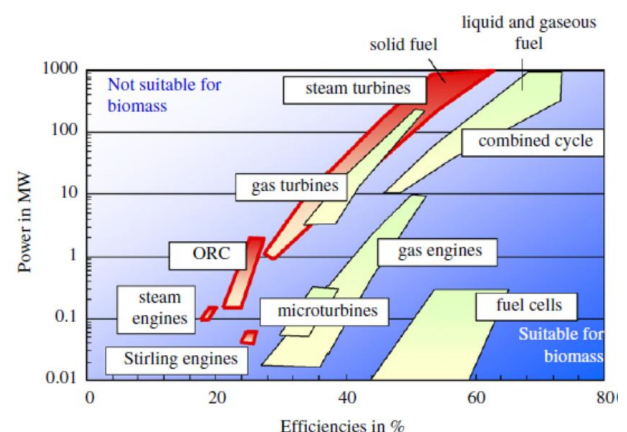

Fig. 1. *Efficiency and power range of known plant types.ORC – Organic Rankine Cycle [1] taken from [2].*

1. SYSTEM DESCRIPTION

Figure 2 shows the analyzed cogeneration system, whose main components are a gasifier and a gas engine. Downdraft biomass gasifiers are the most frequently used reactors due to their capability to prduce tar-free gas. Tar is a mixture of heavy hydrocarbons that condenses below 500 °C and presents the main obstacle for the wider use of producer gas. Low retention time of gases in the rector, low temperatures, and the lack of oxygen are the main reasons for tar creation. Opposite, presents of steam, high gasification temperatures, and passing of gases through a char layer leads to tar destruction. The shortcomings of downdraft gasifiers are a low calorific value and high temperature of produced gas. The latter is the reason for the use of two heat exchangers between the gasifier and the gas engine in the system shown in Fig. 2.

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The first heat exchanger is placed between the gasifier and a filter that removes particulate matter. This exchanger heats water from 70 to 90 °C in a district heating system by decreasing the producer gas temperature from 900 to 800 °C.

It is supposed that all the exchangers work in the same 90/70 °C system except the second one, which is placed downstream of the filter and is used to heat water in a domestic hot water system 25/60 °C.

Except the gasifier the other important part of the examined cogeneration system is the gas engine. Diesel engines have certain advantages such as their higher efficiency due to a greater compression ratio which usually varies between 12 and 24 [4], their better durability and, in some cases, the lower maintenance compared to spark-ignition engines. Generally, design modifications required in diesel engines in order to make these machines run on 100% producer gas include the installation of additional equipment incorporating spark ignition and air-gas mixing systems. In spark ignition and diesel engines producer gas and air are usually mixed in an intake collector and then the air-fuel mixture now ready for combustion enters the cylinders of the engine [4].

In the analyzed system the waste heat of the engine and the sensible heat of the flue gas that leaves the engine are used for heating water in a district heating system.

All the analyses are conducted per 1kg of biomass. To obtain the analysis that can be used for different biomass fuels, a general ash-free biomass given by the formula CH1.4O0.59N0.0017 with 15 wt% of moisture is used. This formula is frequently used [5, 6, 7, 8] and is suitable because the organic masses of different biomass fuels differ slightly. The composition of mineral matters has an important influence on choosing the energy transformation process of biomass. The lower heating value and the exergy of the used biomass are presented in Fig. 2.
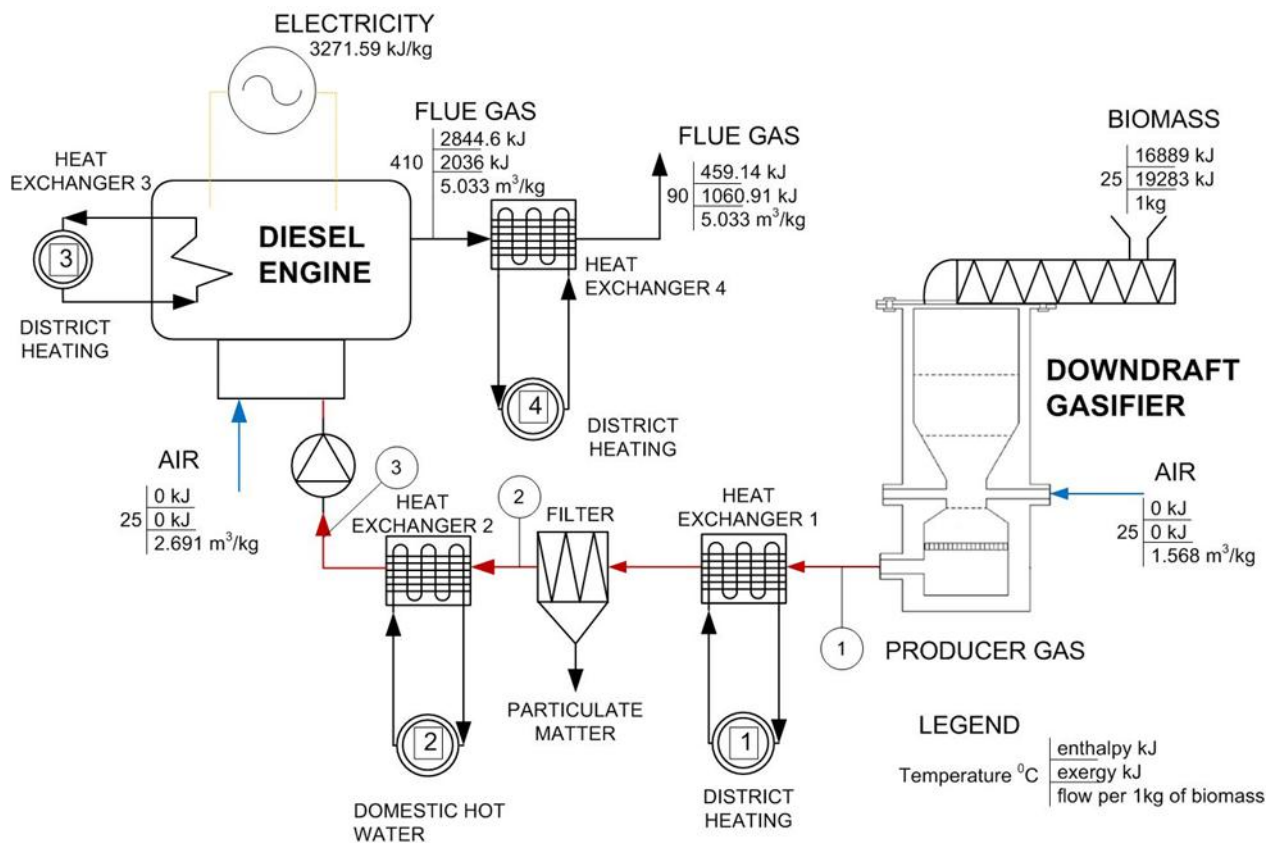

Fig. 2. *Schematic of the analyzed biomass cogeneration system with enthalpies, exergies, temperatures, and mass flows per kg biomass of some matter flows. The date for other flows are presented in Tables 2 and 3.*

2. METODOLOGY

2.1 Gasifier

The examined gasifier is modeld by the already used and verified model [7].

Lower heating value [kJ/m^3_N] of the producer gas, whose composition and amount are shown in Table 1 is calculated by the following equation

$$LHV_{gas} = 282993 x_{CO} + 802303\, x_{CH_4} + 241827\, x_{H_2} \qquad (1)$$

In this equation x_i are the molar or volume fractions of gas species. The sum of the gas species $i = CO, CO_2, CH_4, H_2, N_2, H_2O$ in the producer gas is $\sum x_i = 1$.

2.2 Enthalpies and entropies

All the calculations in the analysis are performed at the standard reference state (25 ^{0}C and 0.1 MPa). Also it is assumed that all the gas species are ideal with specific molar heat capacities given by the third level polynomial equations taken from [9].

Molar specific enthalpies of producer and flue gas are calculated by the following equation

$$h_{gas} = \sum_i x_i \int_{T_0}^{T} c_{p,i} dT \,.[kJ/kmol], \tag{2}$$

whereas the enthalpy is

$$H = n_{gas} h_{gas} [kJ]. \tag{3}$$

In this equation n_{gas}[kmol] is the amount of gas produced in the gasifier by the gasification of 1 kg of biomass or the amsount of the flue gas that leaves the gas engine.

Having defined water temperature regimes in all heat exchangers, water mass flow rates are calculated by:

$$\Delta H_{gas} = \dot{m} c_p \Delta t, \tag{4}$$

where $\dot{m}$ [kg/s] is the mass flow of water, c_p its specific heat capacity, and.Δt its temperature difference at the inlet and outlet of a heat exchanger.

The entropy of a gas is

$$(s_T)_i = \int_{T_0}^{T} c_{p,i} (dT/T) - \mathcal{R} \ln(x_i), \tag{5}$$

where $\mathcal{R}$ = 8.3145 kJ/kgK is the universal gas constant. The last term in the above equation is due to the assumption that all the gas species before and after the engine are at pressure of 1 bar.

Entropies of producer and flue gas are

$$S = \sum x_i (s_T)_i, \tag{6}$$

whereas the change of the entropy of water is

$$\Delta S = \dot{m} \int_{T_0}^{T} c_p (dT/T). \tag{7}$$

2.3 Gas engine model

The simple thermodynamic model of a Diesel cycle [9, 10] is used with the following assumptions:

- pressure of air-producer gas mixture in front of the engine is 1 bar,
- the compression ratio ε is 17.5,
- adiabatic flame temperatue is calculated by assuming that 2% of the producer gas heating value is transferred to the surrounding surfaces,
- complete combustion with the excess ratio of 1.01 takes place in the engine, and

- the temperature of the flue gas at the exit of the gas engine is 410 °C. This is an empirical datum taken from the relevant literature [4]..

By the use of these assumptions, the theoretical efficiency of the Diesel cycle was calculated. The obtained value is multiplied by the mechanical efficiency to obtain actual thermal efficiency of the engine. The empirical value for the mechanical efficiency of the Diesel cycle is 50% [9].

2.4 Exergy analysis

In an irreversible process, the energy is conserved, but the quality of energy decreases because energy is converted into a different form of energy, from which less work can be obtained due to an irreversible increase in entropy. Exergy is the thermodynamic property that considers this irreversible increase in entropy, i.e. the second law of thermodynamics, and is suitable for analyzing energy conversion processes. Exergy is defined as a measure of the actual potential of a system to do work. All real thermodynamic processes are accompanied by an irreversible increase in entropy, which leads to a decrease in exergy (available energy).

The exergy balance of the biomass conversion process is

$$\sum_{in} E_j = \sum_{out} E_k + I. \tag{8}$$

In this equation, $\sum_{in} E_j$ and $\sum_{out} E_k$ represent exergy flows of all entering and leaving streams, respectively. The term I[kJ] is called irreversibility and is the measure of dissipation of exergy in an energy conversion process.

The exergy of stream of matter E depends on its composition (chemical exergy E_{ch}) and its temperature and pressure (physical exergy (physical exergy E_{ph}) $E = E_{ch} + E_{ph}$.

The standard chemical exergy of a pure chemical compound ech is equal to the maximum amount of work obtainable when a compound is brought from the environmental state, characterized by the environmental temperature To and environmental pressure po, to the dead state, characterized by the same environmental conditions of temperature and pressure, but also by the concentration of reference substances in standard environment. The standard chemical exergies of the gases present in the product gas are obtained from Szargut et al. [11]. The chemical exergy of the mixture $e_{ch,m}$ is determined by the composition and concentration of components in the mixture:

$$e_{ch,m} = \sum_i x_i e_{ch,i} + RT_0 \sum_i x_i \ln x_i. \tag{9}$$

The physical exergy of a pure compound   and of a mixture can be calculated using the following formulae, respectively:

$$e_{ph,i} = (h_T - h_{T_0})_i - T_0(s_T - s_{T_0}^0)_i, \tag{10}$$

$$e_{ph} = \sum_i x_i e_{ph,i}. \tag{11}$$

Enthalpies and entropies in the above equations are calculated by the use of Eqs. (2-7).

The equation given by Szargut and Styrilska [12] is used to calculate the exergy of solid biomass. It is interesting to note in Fig. 2 that the chemical exergy of biomass is larger than its lower heating value. This is because the biomass chemical exergy considers its chemical structure.

Exergy efficiency of a system or a component is defined as the ratio of all the streams leaving and entering the system or the component [10].

3. RESULTS

The compositions, lower heating values and amounts of the producer gas, air-producer gas mixture, and flue gas at the exit of the engine are given in Table 1.

Table 1. *Compositions, heating values and amounts of the producer gas, air-producer gas mixture and flue gas.*

Producer gas		Mixture air-producer gas		Flue gas	
CO	19,97%	CO	10,36%	CO_2	16.47%
CO_2	8,60%	CO_2	4,46%	H_2O	16.06%
CH_4	12ppm	CH_4	11 ppm	N_2	67.36%
H_2	18,59%	H_2	9,64%	O_2	0.11%
H_2O	43,57%	H_2O	60,62%	V	5.033 m^3
N_2	9,27%	N_2	4,81%		
LHV	4529 kJ/m^3	O_2	10,10%		
V	2.903 m^3/kg$_{biomass}$	V	5.593 m^3/kg$_{biomass}$		

Enthalpies, exergies and mass flow rates per 1 kg of biomass of all mass streams are given in Fig. 2 and Table 2 and 3.

Table 2. *Mass flow rates of water and their exergies per 1kg of biomass in the district heating system (heat exchangers 1,3, and 4) and in the system for domestic hot water preparation (heat exchanger 2).*

Heat exchanger		1	2	3	4
t_{ul}	^0C	70	25	70	70
t_{izl}	^0C	90	60	90	90
ΔH	kJ/kg$_{biomass}$	3535	118	6574	2386
$\dot{m}$	kg/kg$_{biomass}$	42,14	0,80	78,36	28,43
E_{in}	kJ/kg$_{biomass}$	546,2	0,0	1015,8	368,6
E_{out}	kJ/kg$_{biomass}$	1096,2	6,4	2038,6	739,7

Table 3. *Temperatures, enthalpies, and exergies of the producer gas in the designated in Fig. 2.*

		1	2	3
t	^0C	900	80	50
H	kJ/kg$_{biomass}$	3751,1	215,9	97,9
Ech	kJ/kg$_{biomass}$	12714	12714	12714
Eph	kJ/kg$_{biomass}$	2104	121,132	97,884

Fig. 3 shows the exergy efficiencies of the system components. All these components, except the gas engine have the energy efficiencies of 100% because heat losses in these components are neglected. The gas engine has the energy efficiency of 78.35% because 21.75% of the input energy is retained in the flue gas. Fig. 3 shows that the least efficient component in the system is the gas engine due to the following irreversibilities: producer gas-air mixing, friction, heat transfer with finite differences, and combustion reactions. Fig. 4 gives the working parameters of the engine is characteristic points in the T-s diagram.
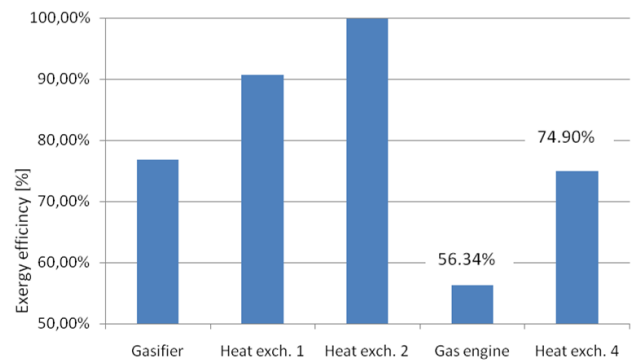

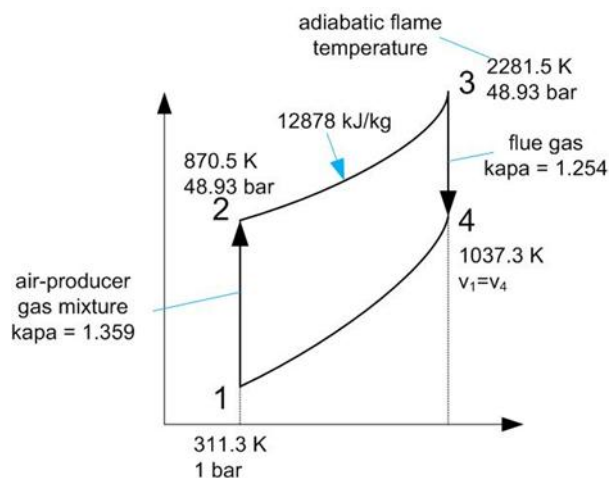Fig. 3. *Exergy efficiencies of the system components*

Fig.4. *T-s diagram of the ideal Diesel cycle*

Ideal Diesel cycle shown in Fig. 4 has the energy efficiency of 57.8%, which means that the actual efficiency of the engine is 28.9%. The electric efficiency of the whole cycle is 22.04%. The obtained values agree with [4]. The thermal energy efficiency of the system is 74.68%. In comparison to real systems, which have the thermal efficiency between 50 i 60% [13] this is a higher efficiency due to the omission of heat loss to surroundings.

The real exergy efficiencies of the heat exhcangers 1, 2, and 4 are lower than those shown in Fig. 3, because the values shown in Fig. 3 take chemical exergies of the gases. These chemical exergies are larger than corresponding physical exergies and do not change during the heat exchange process. If we do not take into account chemical exergies in heat exchange processes the exergy

efficiencies of the heat exchangers 1, 2, and 4 are 45.93%, 93.85%, and 64.45%, respectively. These data shows that in comparison with the heat exchanger 1, the heat exchanger 4 is more efficient due to a smaller mean temperature difference. The obtained data show the expected values: exergetically more efficient are the heat exchangers with smaller temperature differecnces.

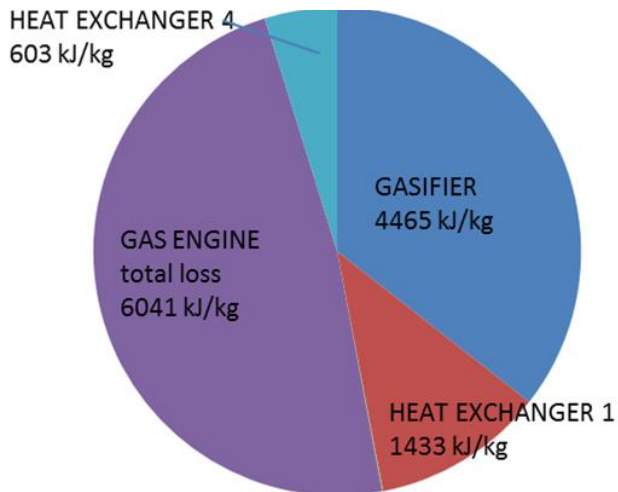Fig. 5 shows the exergy losses in the components of the examined cogeneration system.

Fig. 5. *Exergy losses in the system components*

The total exergy efficiency of the system is 35.83% when the exergy of the flue gas at the exhaust is neglected. This datum shows that 2/3 of the valuable biomass exergy is wasted in the process. The electrical exergy efficiency of the process is reliable 27.15%, whereas the thermal exergy efficiency is bearly 8.69% although almost all the wasted thermal exergy is used by heat exchanging.

4. CONCLUSIONS

The exergy efficiency of the analyzed system is small due to large exergy losses in the gasifier and the gas engine. What we can do to improve it? We should design the system so as the high efficient components serve to those that are less efficient. How this can be achieved? By:

- designing heat loops that will decrease exergy losses of the gasifier by preheating gasifying medium and biomass (pyrolysis) as close as possible to the gasifying temperature. The exergy loss due to gasifying reaction is very difficult to decrease at present. Technically, the decreases are achieved by preheating gasifying air [7] or by use of the producer gas physical heat to preheat steam for gasification, or by use of the process waste heat for pyrolysis of biomass prior to its gasification. In comparison to a simple downdraft gasifier, there are reactors that are more preferable for the use in cogeneration systems. These are the Viking and FICFB reactors [13,14].

These reactors use both medium preheating and biomass pyrolysis.

- the use of solar energy intermittently for medium preheating and biomass pyrolysis.
- designing heat exchanger networks in an optimal manner within the system by the use of pinch analysis or an similar methodology [14,15].
- improving the efficiency of the gas engine by the use of more quality producer gas that has less amount of nitrogen in its composition and larger heating value and/or preheating the combustion mixture. This enable achieving a larger combustion temperatures, but on the other side risk larger emission of nitrogen oxides NOx.

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