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Gas Turbine Cycle with External Combustion Chamber for Prosumer and Distributed Energy Systems

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Abstract: The use of various biofuels, usually of relatively small Lower Heating Value (LHV), affects the gas turbine efficiency. The present paper shows that applying the proposed air by-pass system of the combustor at the turbine exit causes tan increase of efficiency of the turbine cycle increased by a few points. This solution appears very promising also in combined gas/steam turbine power plants. The comparison of a turbine set operating according to an open cycle with partial bypassing of external combustion chamber at the turbine exit (a new solution) and, for comparison, a turbine set operating according to an open cycle with a regenerator. The calculations were carried out for different fuels: gas from biomass gasification (LHV = 4.4 MJ/kg), biogas (LHV = 17.5 MJ/kg) and methane (LHV = 50 MJ/kg). It is demonstrated that analyzed solution enables construction of several kW power microturbines that might be used on a local scale. Such turbines, operated by prosumer's type of organizations may change the efficiency of electricity generation on a country-wide scale evidently contributing to the sustainability of power generation, as well as the economy as a whole.

Keywords: biofuels; sustainable power generation; microturbines

1. Introduction

An increase in the demand for energy and its carriers, as well as the growing legal requirements in the aspect of harmful emissions cause [1–3] the necessity to continuously increase the share of energy from renewable sources [4–6]. In addition to energy from water, wind or solar radiation, biomass is the most commonly used source of renewable fuel [7–9]. Its importance as an inexpensive and renewable source of energy grows particularly when it is produced from agricultural and forestry residues [10,11]. The use of by-products or residues of the agri-food industry for biogas or biofuel production is frequently observed. The growing level of public awareness together with the tightening of legal requirements in environmental protection issues add dynamics to the increase in the level of attractiveness of activities and technologies leading to the reduction of adverse human impact on the ecosystem [12–14]. The sustainability of biomass derived energetics based on distributed electricity generation can be assessed by means of life cycle assessment (LCA) methodology, as well as the direct approach offered by [15–17]. During managing of the production process, the amount of energy

consumed and the production of post-production waste emitted to the atmosphere are analyzed at each stage of the production process. The analysis of energy expenditure is carried out in a comprehensive manner, concerning the whole life cycle of the product. In this context, even electromobility requires a huge amount of energy used to extract and process lithium, cobalt, and manganese; the key raw materials needed to produce batteries for electric cars. Thermodynamic cycles and the use of modern calculation tools are also important in this process [18,19].

The development of prosumer energetics, is expected to assure strengthening of energy security, to implement climate policy, but also create the opportunity to solve grid problems and also to mitigate an increase of energy prices [20–22]. The observed dynamic changes in the structure of the electricity mix are caused, among others, by changes in the development strategy of individual countries, which are motivated by principles of sustainable development and, as a consequence, climate policy being an indispensable element of planning reforms in the power sector [23]. This leads to a visible increase in the number of prosumer generation centers being built (also biogas production plants/biorefineries with polygeneration devices) [24,25]. In these types of units, most often turbines, attention is paid to durability, reliability, low price or efficiency of components and the entire installation [26,27].

Biogas plants and biorefineries are, apart from technology, connected to either dedicated energy crops (biogas-biomethane) or to lignocellulosic wastes of plants used for edible crops (second generation biofuels-biomethanol) [28–30]. Biogas obtained in the process of anaerobic methane fermentation can be considered as a substitute for natural gas and a universal source of cheap energy produced and used locally [31,32]. Production of biomethane is, in turn, the clean for the environment, and the fuel is characterized by a higher EROEI index than vegetable oils [33]. The division of biofuels into individual generations is based mainly on the type of raw material processed [34]. Currently, R&D activities in biorefineries are aimed at the bioconversion of lignocellulose to simple sugars and fermentation to ethanol. In addition, the work focuses on breeding new varieties of energy plants with high biomass efficiency (e.g., 2nd generation bioethanol from hemp mass). Third generation biofuels (products obtained as a result of the conversion of new raw materials intended for this purpose) seem to be still a distant future [35–37]. The only enterprise on the wider scale operating in Poland that attempts to produce third generation biofuels is PKN Orlen [38,39]. The experimental station launched in Płock, in which the technology for the production of biocomponents derived from oil algae is being developed [40]. For their production, post-production water and CO_2 obtained from refining processes will be used [41,42].

The innovativeness of modern biogas plants is primarily connected with new technologies for the production of energy crops and models of "tapes" of plant production and preservation assuring the continuous supply of energetic biomass along with the necessary logistic security [43,44]. In addition, innovative processes for the fermentation are used due to the microbiological composition of the fermentation flora depending on the biomass feedstock parameters. R&D works are conducted towards the optimization of the biochemical processes of the bioreactor depending on the type of plant biomass and the use of biogas in the fuel cell [45,46]. The disadvantage of the domestic power generation sector is the relatively low efficiency of energy production from coal, and in the case of dispersed power engineering, the efficiencies of small power plants are even lower, and thermal power plants based on circuits with organic agents achieve efficiencies of only a dozen or so percent [47–50]. In addition, there is the issue of high carbon dioxide emissions [51].

As part of the electric micro-plants' development, one can indicate turbine micro-turbines [52,53], bladeless-free adhesive turbines [54] or solar collectors [55,56]. Turbines of small output can be arranged in simple or complex cycles, including regenerators, interstage coolers or successive combustion chambers. Turbines burning biofuels in external combustion chambers arouse particular interest. In this solution, clean air flows in compressor and turbine during the whole period of operation. In the case of an external combustion chamber, it is possible to bleed some turbine exhaust air and omit a combustion chamber. This can improve turbine unit efficiency.

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A key role in the modern cogeneration units belongs to the high efficiency and (in parallel) compact heat exchangers with passive techniques of heat transfer augmentation. Preferential are the heat exchangers with minichannels of cylindrical construction [57] and plate ones [58]. Very promising are also the heat exchangers with the micro-jets' technology—very intensive experimental and numerical investigations of their development are conducted at the moment.

Particular biofuels can differ depending on their chemical composition and the heating values which influence the relations between the temperatures and the mass (and volume) flow rates of the working media. This, in turn, shows some impact on the power plant overall efficiency and the design of turbomachinery flow parts. In the case of various biofuels (especially pellets) so called "external combustion systems" may be used, which allows to burn different sorts of fuel (liquid, gas or solid), even of poor quality, because in these units' clean air flows through the compressor and the turbine.

Currently, it is possible to obtain a stable flame during the combustion of low calorific fuels in a wide range of operating parameters, such as the molar composition of the fuel and the excess air coefficient. However, the biogas must be properly cleaned and dried so that it does not damage the turbine. Depending on the origin, the biogas composition is variable. The calorific value depends primarily on the methane content. Currently, biogas, that is combusted in gas turbines, has the methane content from 35% to 100%. As a result of continuous combustion with excess air and low pressures in the combustion chamber, turbines as well as microturbines have a significantly lower value of exhaust emissions as compared to the reciprocating engines. The combustion of low calorific gases has a significant impact on the natural environment by reducing the emission of nitrogen oxides [59,60].

If the fuel of low value of Lower Heating Value (LHV) is used in gas turbine units, the mass flow rate of a turbine exhaust air can be remarkably higher than mass flow rate of compressor air. Thus, in the case of an external combustion chamber, it is possible to bleed some turbine exhaust air and omit a combustion chamber. This original arrangement was compared with the open gas turbine cycle with regenerator. Thus, two variants were considered during the calculations:

- Variant 1: turbine set operating according to the open cycle with regenerator (Figure 1),
- Variant 2: turbine set operating according to the open cycle with partial bypassing of external combustion chamber at turbine exit and with high-temperature heat exchanger (Figure 2).

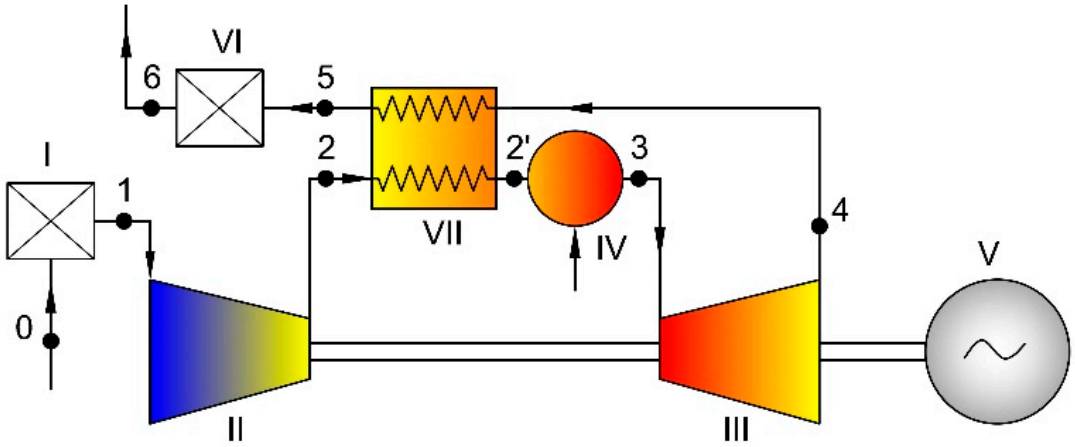

Figure 1. Turbine set operating according to the open cycle with regenerator (Variant 1-V1). I–filter; II–compressor; III–turbine; IV–combustion chamber; V–electric generator; VI–silencer; VII–regenerator.

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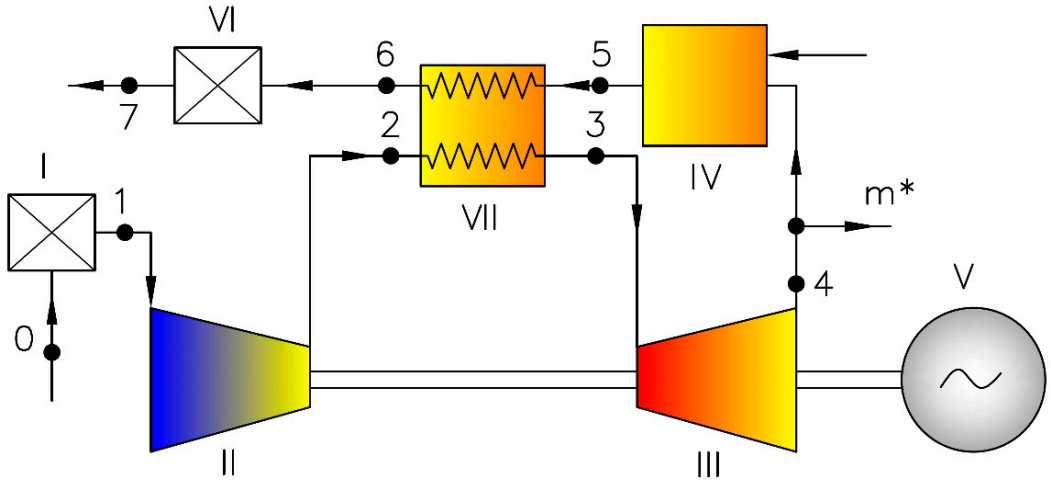

Figure 2. Turbine set operating according to the open cycle with partial bypassing of the external combustion chamber at the turbine exit and with a high-temperature heat exchanger (Variant 2 -V2). I–filter; II–compressor; III–turbine; IV–external combustion chamber; V–electric generator; VI–silencer; VII–regenerator.

2. Materials and Methods

2.1. Nomentclature and Units

The following list contains the most important symbols and units (Tables 1 and 2) used in the formulae and in the figures.

Table 1. Symbols and units used in calculations.

Symbol	Description	Unit
c_p	specific heat at constant pressure	[kJ/kg·K]
h	enthalpy of unit mass	[kJ/kg]
p	pressure	[Pa]
η	efficiency	[–]
$\dot{m}$	mass flow	[kg/s]
l	work of unit mass	[kJ/kg]
c_p	specific heat at constant pressure	[kJ/kg·K]
κ	exponent of isentropic process	[–]
W	power	[kW]
$\dot{Q}$	heat flux	[kW]
T	temperature	[·K]
LHV	lower heating value	$\left[MJ/m^3\right]$ or [MJ/kg]
$\prod$	compression ratio	[–]

Table 2. List of used subscripts.

Symbol	Description
f	fuel
m	mechanical
opt	optimal
meth	methane
C	compressor
CC	combustion chamber
T	turbine
GT	gas turbine set
1, 2, ...	points on diagrams

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2.2. The Computation Algorithm

In the case of small power plants (from several kW to several hundred kW), the maximum temperature 900 °C was assumed before the turbine, and the low efficiency of the components was assumed. Assumptions adopted for the analysis are presented in Table 3.

Table 3. Assumptions adopted for the design analysis of turbine generator variants [27,51].

Description	Unit	Value
compressor efficiency	[-]	0.800
turbine efficiency	[-]	0.820
mechanical efficiency	[-]	0.980
leakage losses	[-]	0.02
generator efficiency	[-]	0.900
efficiency of the combustion chamber	[-]	0.950
pressure loss in the filter	[-]	0.995
pressure loss in the silencer	[-]	0.995
pressure losses in the combustion chamber	[-]	0.98
pressure loss in the regenerator	[-]	0.98

The results of thermodynamical calculations of the cycles with gas turbines using fuels of different caloric values were presented in the paper. The results are presented for a gas from biomass gasification and for a biogas. For comparison, the analysis was also carried out for methane (the main component of LNG or natural gas). The characteristics of the considered gases are shown in Table 4.

Table 4. The list of analyzed gases with different heating values [61,62].

Fuel Type	Volumetric Composition	Density	Calorific Value, LHV
	[-]	[kg/m^3]	[MJ/kg]
gas from biomass gasification	CH_4 0.09; CO_2 0.133; CO 0.147; H_2 0.073; N_2 0.42; H_2O 0.137	1.2107	4.4
biogas	CH_4 0.4; CO_2 0.23; H_2 0.16; CO 0.1; N_2 0.11	0.9438	17.5
methane	CH_4 1.0	0.6660	54.1

The thermodynamical calculations were performed following classical approach known from the bibliography [27,63–65]. The calculations were performed in the following order: the calculations of compression process in the compressor, calculations of expansion line in the turbine, calculations of regenerator and combustion chamber energy balance equations. The following main relationships were used in the analysis, including energy balances and definitions for the efficiencies.

The power and specific work of the gas turbine set are determined accordingly to the dependencies:

$$W_{GT} = \eta_m \cdot \dot{m}_T \cdot l_T - \dot{m}_C \cdot l_C$$

$$l_{GT} = \frac{W_{GT}}{\dot{m}_C} = \eta_m \cdot \left(\frac{\dot{m}_T}{\dot{m}_C}\right) \cdot l_T - \left(\frac{\dot{m}_C}{\dot{m}_C}\right) \cdot l_C = \eta_m \cdot \left(\frac{\dot{m}_T}{\dot{m}_C}\right) \cdot l_T - l_C$$

The specific work of the compressor and turbine are defined by relations:

$$l_C = \frac{1}{\eta_C} \cdot c_{pC} \cdot T_1 \cdot \left(\left(\frac{p_2}{p_1}\right)^{\frac{\kappa_C - 1}{\kappa_C}} - 1 \right)$$

$$l_T = \eta_T \cdot c_{pT} \cdot T_3 \left(1 - \left(\frac{p_4}{p_3}\right)^{\frac{\kappa_T - 1}{\kappa_T}} \right)$$

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The efficiency of the gas turbine cycle is defined as:

$$\eta_{GT} = \frac{W_{GT}}{\dot{Q}_1}$$

where the heat flux brought to combustion chamber:

$$\dot{Q}_1 = \left(\dot{m}_T \cdot h_3 - \dot{m}_C \cdot h_{2'}\right)\frac{1}{\eta_{CC}} = \dot{m}_f \cdot LHV$$

After the transformations of the above equations, it was obtained:

$$\eta_{GT} = \frac{\eta_m \cdot \dot{m}_T \cdot l_T - \dot{m}_C \cdot l_C}{\left(\dot{m}_T \cdot i_3 - \dot{m}_C \cdot i_{2'}\right) \cdot \frac{1}{\eta_{CC}}}$$

and:

$$\eta_{TG} = \eta_{KS} \cdot \frac{\eta_m \cdot \eta_C \cdot c_{pT} \cdot \frac{T_3}{T_1}\left[1 - \left(\frac{1}{\prod_T}\right)^{\frac{\kappa_T - 1}{\kappa_T}}\right] \cdot \frac{\dot{m}_T}{\dot{m}_C} - \frac{1}{\eta_C} \cdot c_{pC} \cdot \left[\left(\prod_C\right)^{\frac{\kappa_C - 1}{\kappa_C}} - 1\right]}{\frac{\dot{m}_T}{\dot{m}_C} \cdot c_{pCC} \cdot \frac{T_3}{T_1} - \frac{1}{\eta_C} \cdot c_{pC} \cdot \left[\left(\prod_C\right)^{\frac{\kappa_C - 1}{\kappa_C}} - 1\right]}$$

The heat flux transferred from the exhaust fumes in the regenerator is that:

$$\dot{Q}_{VII,\,T} = \dot{m}_T \cdot (h_4 - h_5)$$

The heat flux received by the air in the regenerator is that:

$$\dot{Q}_{VII,\,C} = \dot{m}_C \cdot (h_{2'} - h_2)$$

Average values of specific heat for particular states of working media were determined on the basis of their chemical composition and thermodynamical parameters using REFPROP software.

3. Results

Calculations of two variants of cycles were performed (Figures 1 and 2) for three fuels with different lower heating values (Table 4). The reference was made to the results for methane as a comparative fuel. For each of the analyzed variants, the compression ratio was optimized to obtain maximum efficiency. The effect of the compression on the value of the relative efficiency (referred to the maximum value) for three fuels used in variant 1 is shown in Figure 3, and for variant 2-in Figure 4. As can be seen, the calorific value shows an effect on the optimal compression (maximum efficiency) only for variant 1, whereas in variant 2 the type of fuel has a little effect on the optimal compression value. This is confirmed by the results shown in Figure 5 (optimal compression values for different types of fuel and both variants).

The Figure 6, in turn, shows the influence of the structure of the cycle and calorific value of the fuel on the relative efficiency (referred to methane) for both variants of turbine sets. In each case considered, the decrease of calorific value leads to a reduction in the efficiency of the turbine set (from a few to a dozen or so percent). The subsequent figures show the influence of the cycle structure and of calorific value of the fuel on the relative mass flow of fuel combusted in the combustion chamber (Figure 7), on the relative mass flow of exhaust (Figure 8) and on the relative power of the turbine set all related to the corresponding variants of the cycles with methane used as a fuel (Figure 9).

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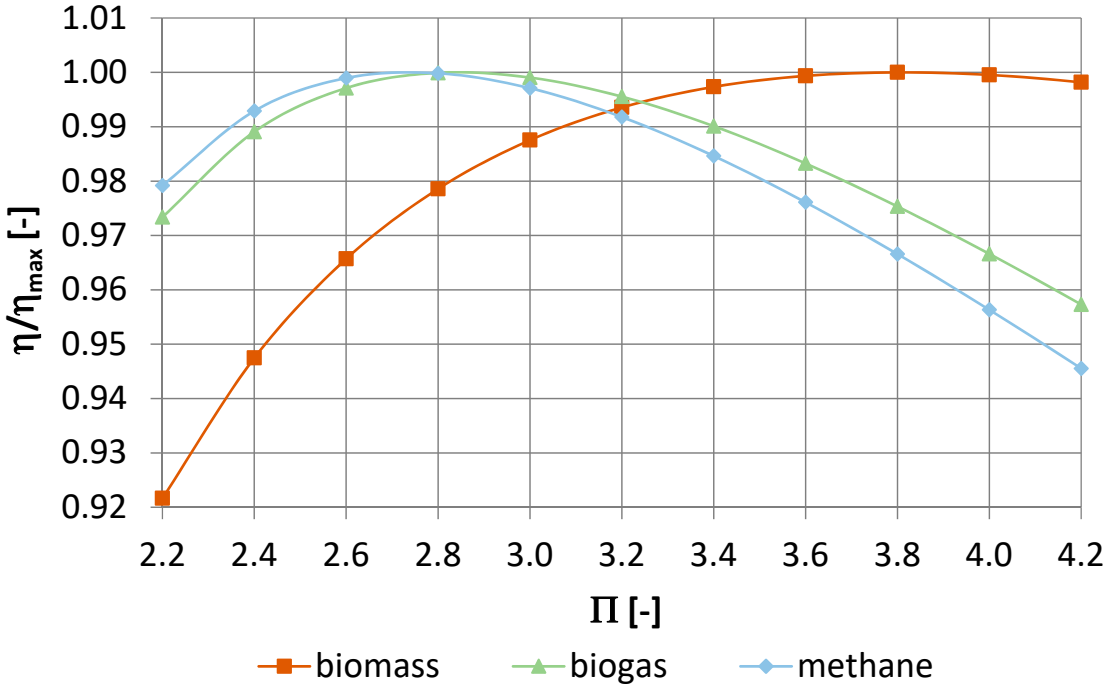

Figure 3. Effect of compression ratio ∏ on the value of relative efficiency (referred to the maximum value) for three example fuels in variant 1.

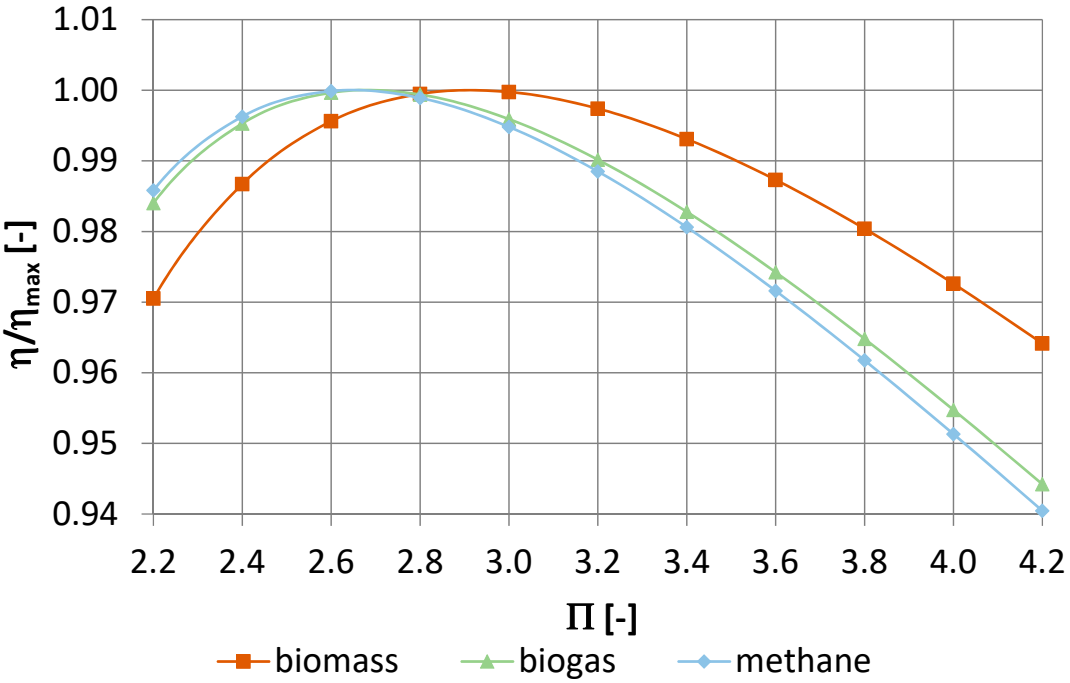

Figure 4. Effect of compression ratio ∏ on the value of relative efficiency (referred to the maximum value) for three exemplary fuels in variant 2.

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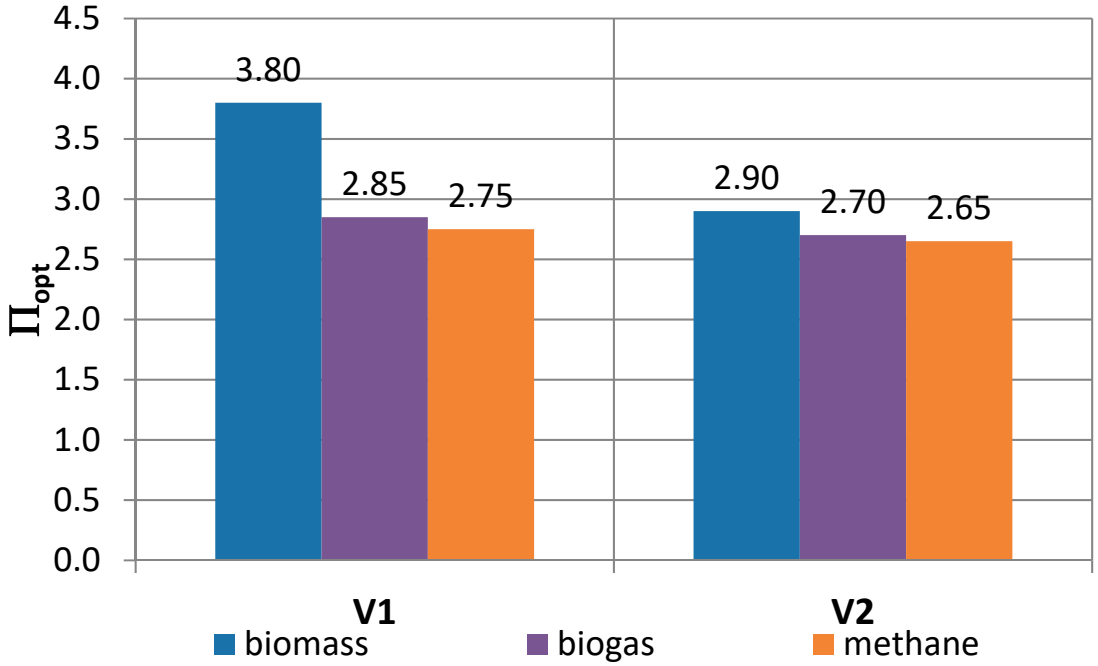

Figure 5. Influence of the structure of the cycle and calorific value of fuel on the optimal compression.

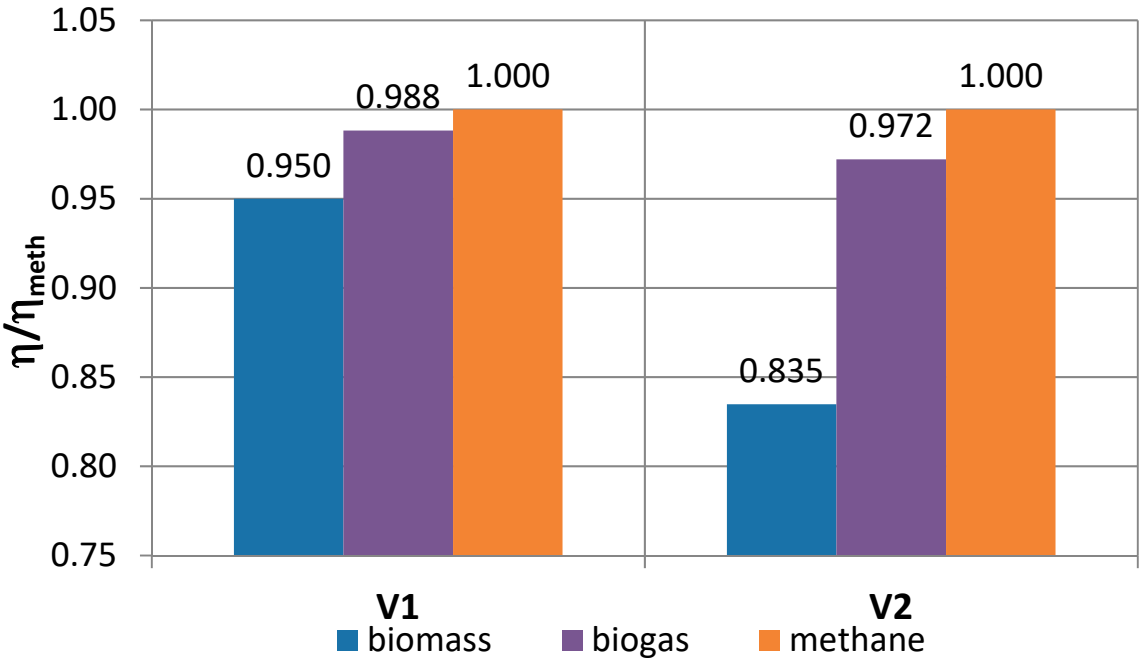

Figure 6. The influence of the structure of the cycle and calorific value of the fuel on the relative efficiency (referred to methane) for both variants of turbine sets.

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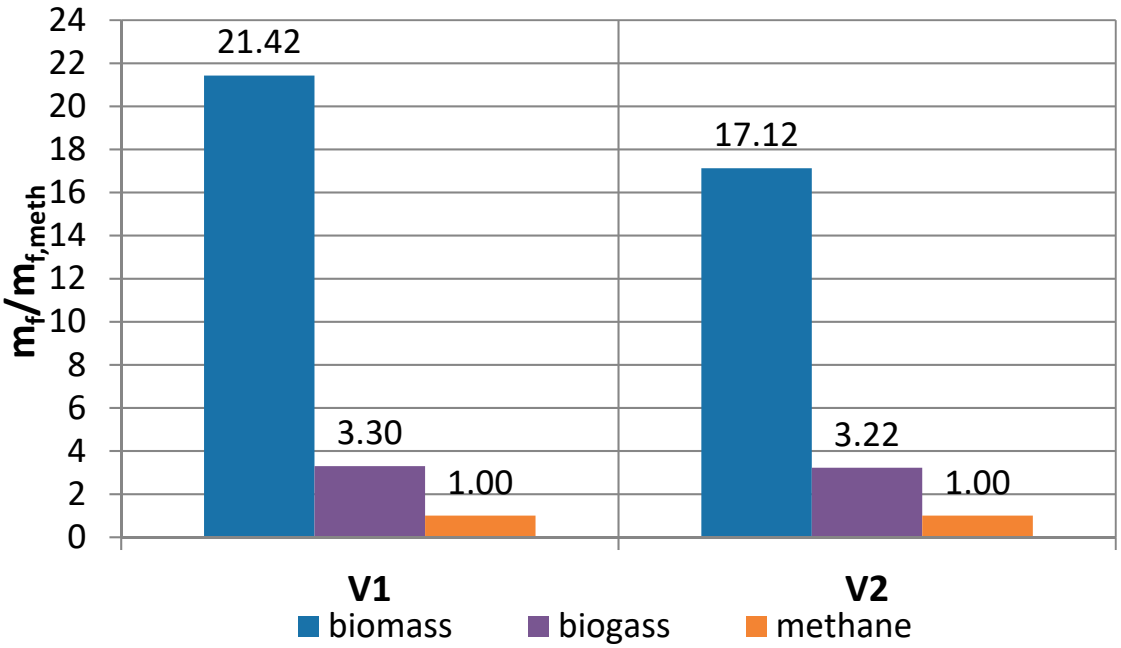

Figure 7. The influence of the structure of the cycle and calorific value of fuel on the relative mass stream of the fuel in the combustion chamber for both variants (referred to methane as a fuel).

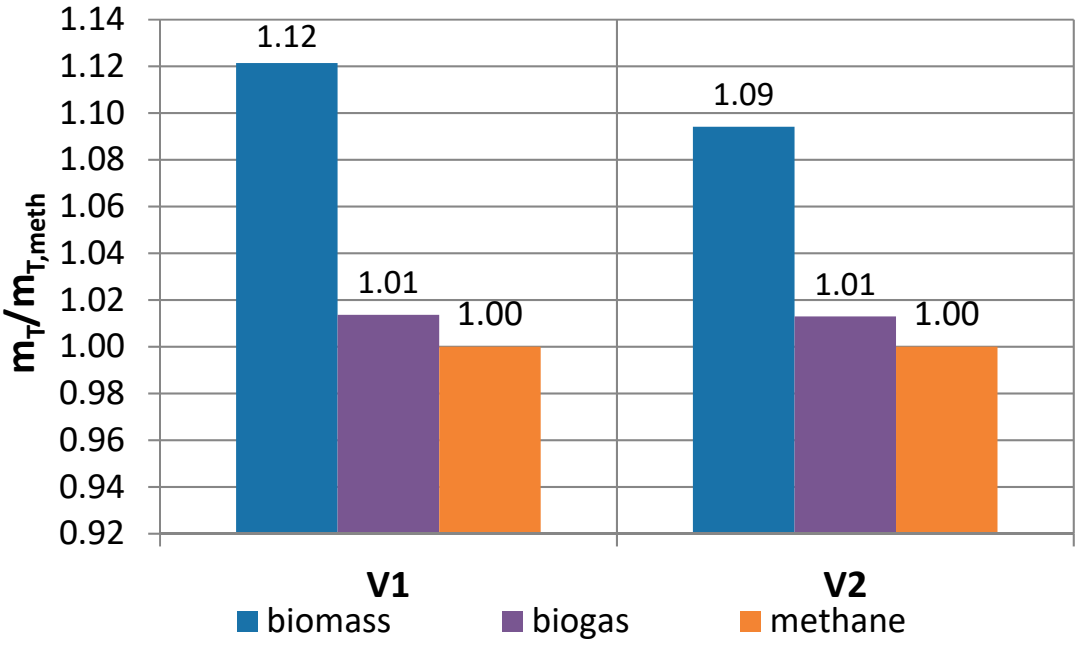

Figure 8. Effect of the structure of the cycle and calorific value of fuel on the exhaust gas mass flow for both variants (related to methane used as a fuel).

A decrease of the lower heating value leads to a clear, multiple increase in mass fuel consumption, especially when using low calorific biomass gasification fuel (Figure 7). However, due to the small mass fraction of fuels in relation to air, this does not significantly affect the increase of exhaust gas flow (Figure 8). Only in the case of fuel from biomass, the increase of the flue gas stream may amount to approx. 10% and only in variant 1 this results in a very large increase in the unit power of the turbine set.

It is interesting to compare directly the efficiency of both variants of the turbine set cycles (V1 and V2). Only for biomass fuel, the variant with the external combustion chamber (V2) appears worse than

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the classic turbine set with the regenerator (V1). For higher caloricity of the fuel, variant 2 achieves higher efficiency than variant 1, Figure 10.

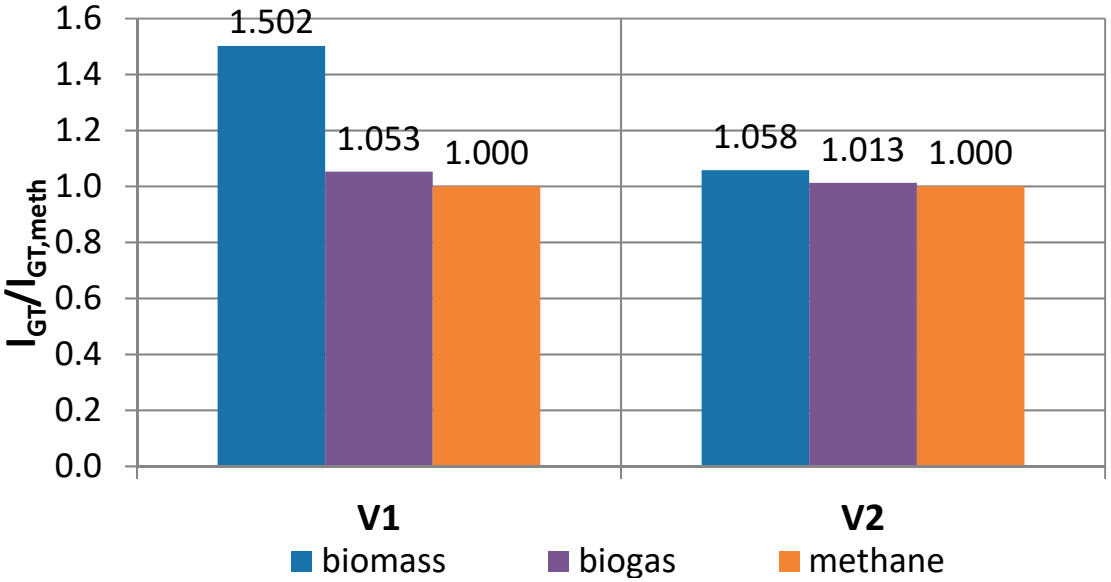

Figure 9. The influence of the structure of the cycle and calorific value of fuel for both variants on the effective power related to that with methane as a fuel.

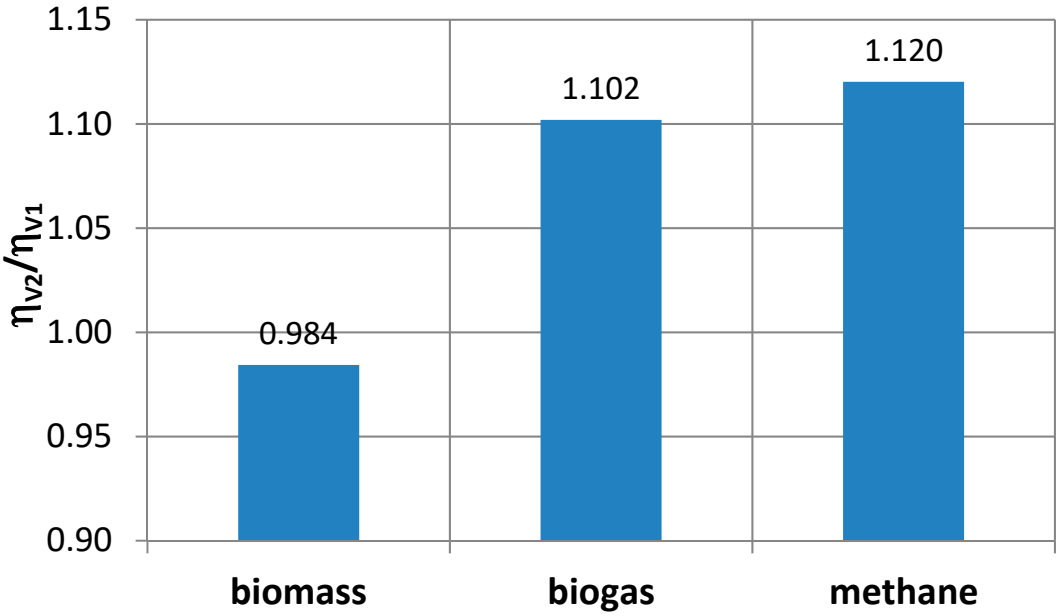

Figure 10. The influence of calorific value of the fuel on the efficiency ratio of the cycles with the external combustion chamber and bypass (V2), and for with the regenerator (V1).

In the analysis of the work of cycles, one should also consider the possibilities of using energy of exhaust gas in cogeneration systems for generating electricity and heat as well as in combined systems (cooperation with other types of thermal cycles). Therefore, in this analysis the values of the stream intensity and the temperature of the medium leaving the turbine set are important. Figure 11 shows the outlet air temperature (after the regenerator) for both variants V1 and V2, and for all the fuels considered. The highest outlet temperatures are for biomass fuel (with the lowest caloricity LHV) for both the regenerator cycle (278 °C) and the cycle with the external combustion chamber (160 °C).

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Cycles with an external combustion chamber are usually characterized by a lower outlet temperature of the working medium.

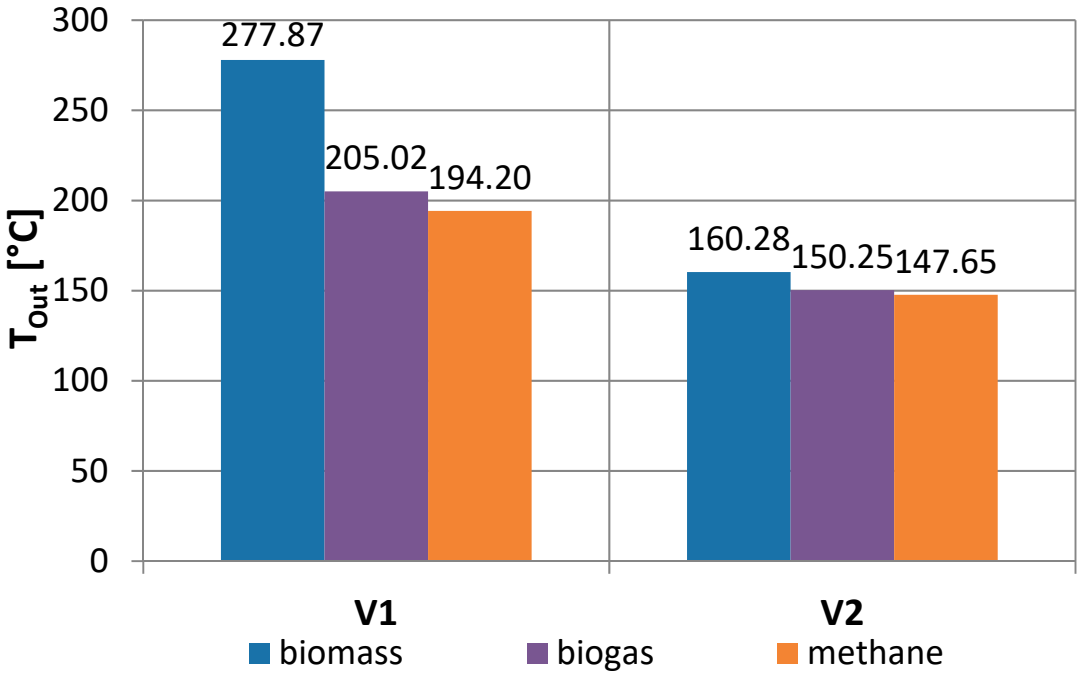

Figure 11. The influence of the calorific value of the fuel and the structure of the cycle on temperature of exhaust gas leaving the turbine set.

Figure 12 shows the relative air flow drawn from behind the turbine in relation to the air flow through the compressor. It ranges from a few to a dozen or so percent (for the fuel from biomass), but it should be taken into account that it is a high temperature air, which creates greater opportunities for its use, also in combined systems. The possibility of heat production in cogeneration with electricity generation for variants with a regenerator and an external combustion chamber is shown in Figure 13 (variant 3) and Figure 14 (variant 4), respectively.

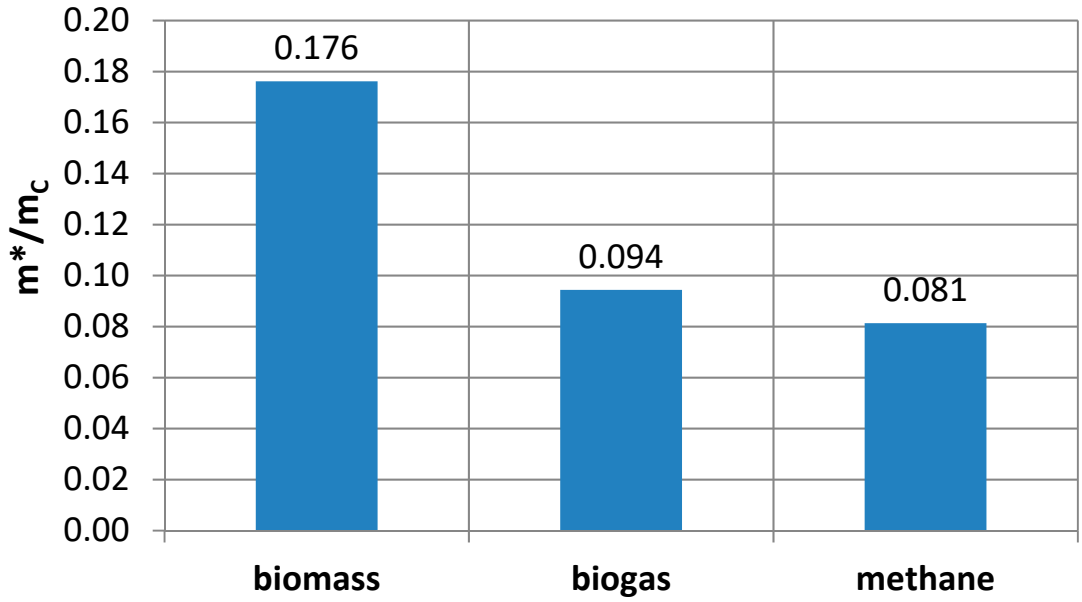

Figure 12. The influence of the calorific value of the fuel on the relative mass stream of the bypass to the mass flow of air flowing through the compressor for the optimal compression ratio.

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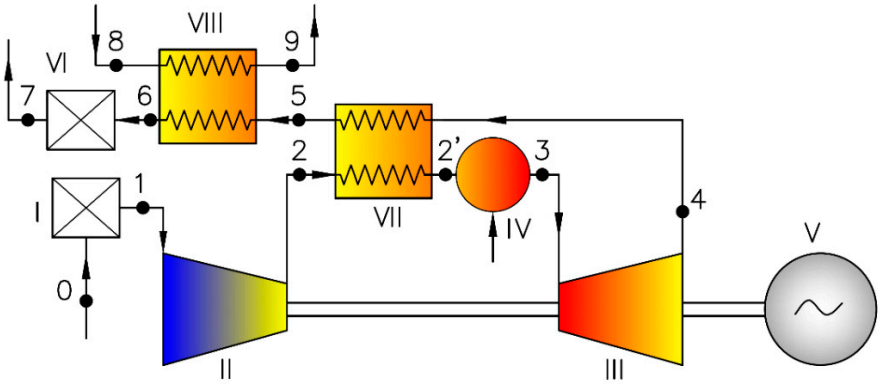

Figure 13. Circulation diagram with a gas turbine with a regenerator and cogeneration (Variant 3-V3). I–filter; II–compressor; III–turbine; IV–combustion chamber; V–electric generator; VI–silencer; VII–regenerator; VIII- heat exchanger (for cogeneration heat).

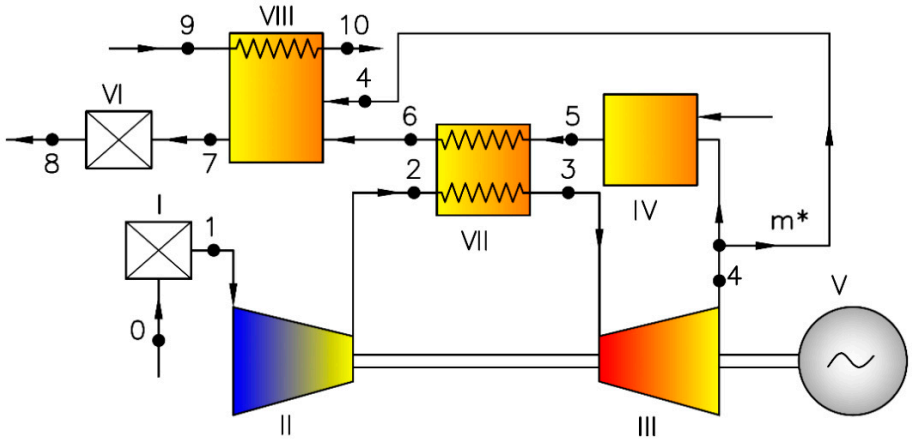

Figure 14. Circulation diagram with air turbine, external combustion chamber and bypass and cogeneration (Variant 4-V4). I–filter; II–compressor; III–turbine; IV–external combustion chamber; V–electric generator; VI–silencer; VII–regenerator; VIII- heat exchanger (for cogeneration heat).

Figure 15 shows a gas turbine with an external combustion chamber and air intake from behind the turbine for combined cycle with a steam turbine (variant 5). This solution increases the electric power generated, and the efficiency of the entire power plant. The comparison of the efficiency of the combined machine (variant 5) with the efficiency of the system with regenerator (variant 1) is shown in Figure 16. Regardless of the fuel used, the efficiency of the combined system (variant 5) exceeded about 30% efficiency of turbine set with regenerator (variant 1).

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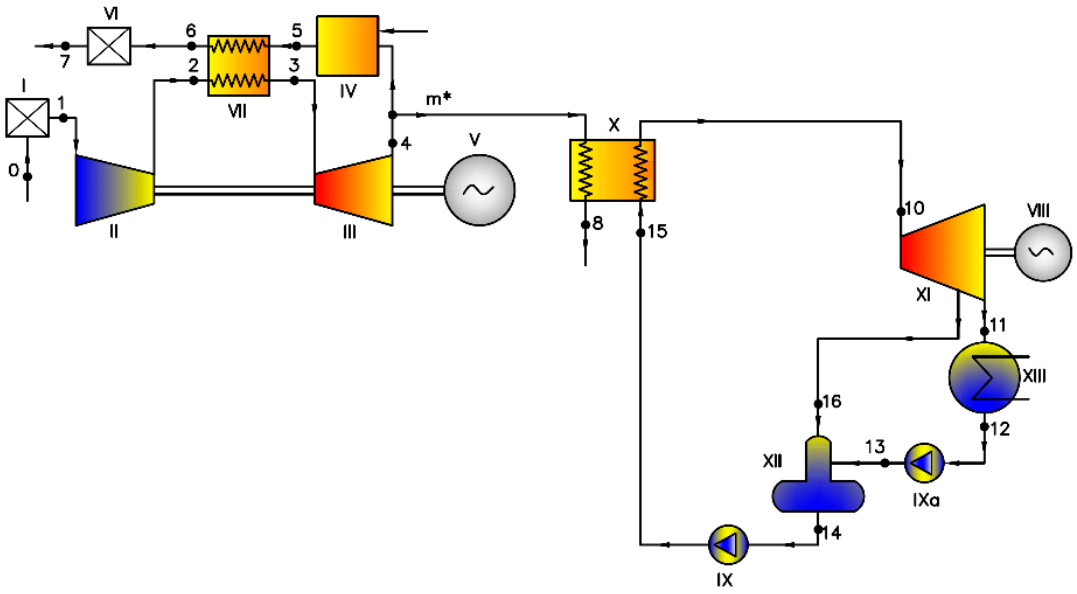

Figure 15. Circulation diagram with an air turbine, an external combustion chamber and a bypass with an additional steam circuit (Variant 5-V5). I–filter; II–compressor; III–turbine; IV–external combustion chamber; V–electric generator of gas turbine; VI–silencer; VII–regenerator; VIII- electric generator of steam turbine; IX–main pump; IXa–condensate pump; X–heat recovery steam generator; X1–steam turbine; XII–direct contact heat exchanger; XIII–condenser.

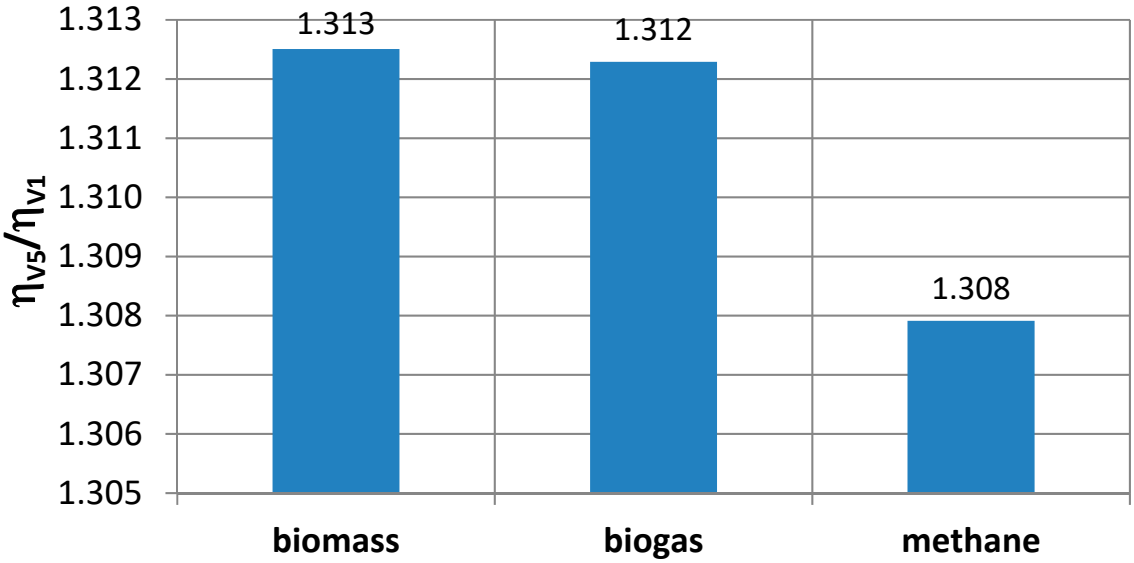

Figure 16. The effect of calorific value of fuel on the relative efficiency of the cycle with the external combustion chamber and bypass and additional steam circuit (V5) referred to the cycle with regenerator (V1).

The results of calculations for the analyzed fuels are presented in Tables 5 and 6.

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Table 5. Summary of the results of calculations carried out for the different calorific values.

Parameter	Variant	Biomass	Biogas	Methane
Π_{opt}	V1	3.80	2.85	2.75
	V2	2.90	2.70	2.65
η/η_{meth}	V1	0.950	0.988	1.000
	V2	0.835	0.972	1.000
$m_f/m_{f,meth}$	V1	21.42	3.30	1.00
	V2	17.12	3.22	1.00
$m_T/m_{T,meth}$	V1	1.12	1.01	1.00
	V2	1.09	1.01	1.00
$l_{GT}/l_{GT,meth}$	V1	1.502	1.053	1.000
	V2	1.058	1.013	1.000
T_{out}	V1	277.87	205.02	194.20
	V2	160.28	150.25	147.65

Table 6. Summary of the results of calculations carried out for the different calorific values of fuel for selected cycles.

Fuel	η_{V2}/η_{V1}	m^*/m_c
Biomass	0.984	0.176
Biogas	1.102	0.094
Methane	1.120	0.081

All the calculations were performed assuming efficiencies of the main turbine power plants elements. Efficiencies of compressors, turbines, combustion chambers and regenerators (Table 3) can be treated as quite good values for gas turbine sets of small output. If we consider the other values of these parameters, we may expect a change of overall power plant efficiency, for example:

- applying lower compressor efficiency equal to 0.75 (instead 0.80) we observe the drop of overall efficiency by about 7%–8% for all considered variants while assuming higher compressor efficiency 0.85 the increase in overall efficiency can reach 6.5%–7.5%,
- decreasing turbine efficiency from 0.82 to 0.77 we lower the overall efficiency by about 9%–10% and increasing it to 0.87 we have the rise in overall efficiency by about 8%–9%,
- changing the combustion chamber efficiency by 5% the overall efficiency also changes by about 5%,
- assuming high values of all elements (compressor efficiency −0.85, turbine efficiency −0.87, combustion chamber efficiency −0.985, regenerator efficiency 0.98) we may increase the overall power plant efficiency by 18%–21%,
- using low values of all elements (compressor efficiency −0.75, turbine efficiency −0.77, combustion chamber efficiency −0.9, regenerator efficiency 0.8) we may reduce the overall power plant efficiency by 28%–32%.

All the considered variants, in spite of the fuel used, are more sensitive to reduction than to growth of the efficiency of particular power plant elements.

For many years, installations with electric power of several hundred kilowatts and more have been used in biomass-burning power plants. Cogeneration systems working with organic media are already available in a wide range of electrical and thermal power, e.g., a power plant with an electrical power of 300–600 kW and a heat power of 1500–2800 kW or a heat and power plant with an electric power of 200–1000 kW and a thermal power of 1000–6000 kW [66]. However, only a few examples of Organic Rankin Cycle (ORC) installations with an output power less than 100 kW [67] or less than 30 kW [68] of electrical power can be found. Only a few examples of ORC cogeneration systems

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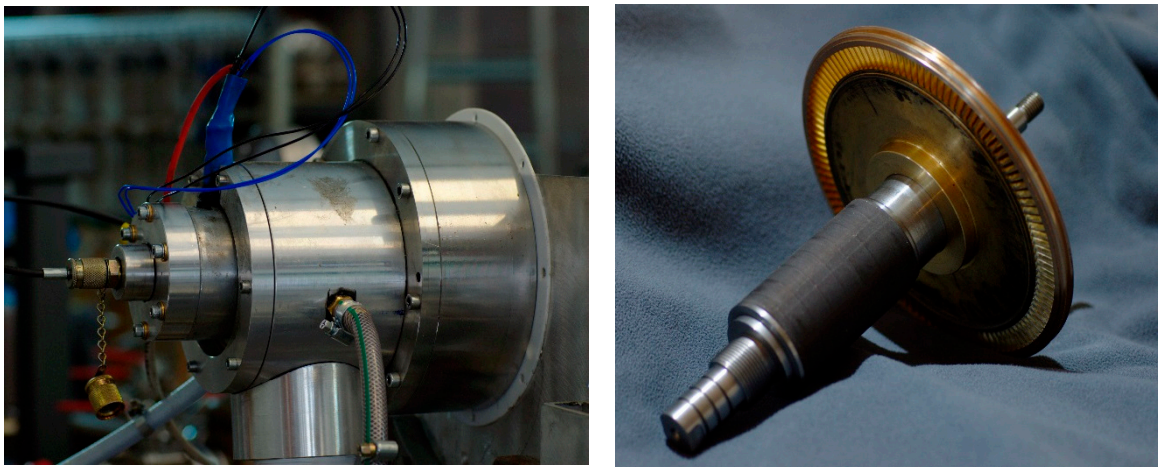
(Organic Rankin Cycle) with an electric power below 5 kW can be found on the market. The published results of experimental studies on the operation of microturbine sets can be found, for example, in [69].

Previous research has shown that it is possible to build a set of microturbines with a capacity of about 2 kW with a higher efficiency than in existing machines [70]. It is worth noting that the relatively high efficiency of microturbines can be achieved thanks to a very careful and advanced design process (Figure 17). The safe and reliable behavior of the microturbine set has been confirmed during operational tests. The results indicate the need to consider the interaction between components of the ORC installation and microturbine.



(**a**) (**b**)

Figure 17. Example of experimental single stage micro turbine. (**a**) view of experimental micro turbine; (**b**) bladed rotor of micro turbine with shaft and electrical generator rotor.

4. Final Conclusions

The performed analysis have shown that:

1. Systems with an external combustion chamber and air intake from behind the turbine can be competitive in terms of efficiency with classic turbine sets, also those with regeneration.
2. The use of fuels with lower calorific value usually leads to a lower efficiency of the power plant.
3. The structure of the gas turbine power plant and the type of fuel should be taken into account at the stage of determining the main design parameters of the turbine set (e.g., optimal compression ratio, unit power).
4. Turbosets with an external combustion chamber and air intake from behind the turbine are characterized by wide possibilities of applications in cogeneration systems and high efficiency combined systems.

Conducted analyses provide knowledge to help to mitigate potential environmental hazards through introduction of biofuels into distributed energy generation and optimization of turbines to such, locally available fuels.

Energy efficiency solutions provide opportunities for diversification and reduce energy consumption as well as primary consumption. It seems that the efficiency of turbine sets for micro-energetics will clearly increase after the introduction of new high-efficiency thermodynamic cycles. Such effective technological changes and improvements to the system may be one of the stages in the fight against the energy crisis, as well as an element of strategic development in the shaping of the national power generation sector.

The problem still requires further research, but implementation of the final might contribute to the reduction of environmental burdens.

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At present two experimental micro power plants with external combustion chambers are being built: a domestic cogeneration power plant of 20 kW heat power and 2 kW electric power as well as a high efficiency turbine power plant with an innovative isothermal turbine of 5 kW (Figure 18).

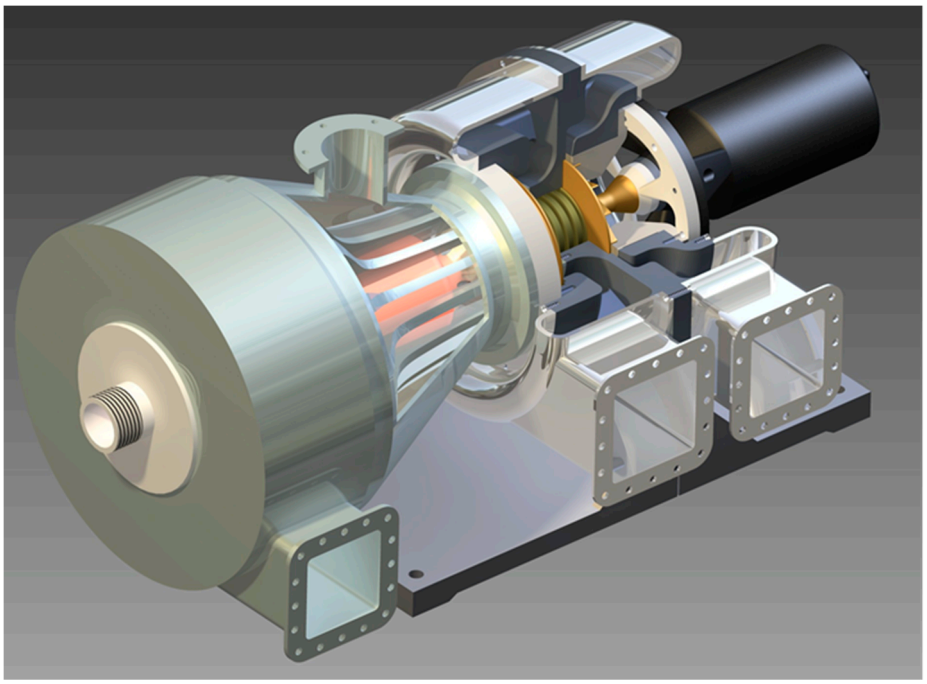

Figure 18. Example of an experimental variant of the innovative isothermal turbine.

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