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TECHNICAL AND ECONOMIC OPTIMIZATION OF COGENERATION TECHNOLOGY USING COMBUSTION AND GASIFICATION

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Abstract. This paper presents the technical and economic optimization of new microcogeneration technology with biomass combustion or biomass gasification used for cogeneration of electrical energy and heat for a 200 kW unit. During the development phase, six possible connection solutions were investigated, elaborated and optimized. This paper presents a basic description of the technology, a description of the technological solutions, and especially the results of balance and financial calculations, ending with a comparison and evaluation of the results.

Keywords: microcogeneration, biomass, combustion, heat balance, economic indicators.

1. Introduction — microcogeneration

The Energy Institute at the Faculty of Mechanical Engineering, Brno University of Technology, has been conducting research and development in the area of gasification since 2000. The Institute's laboratory is equipped with a Biofluid 100 atmospheric fluid generator with 150 kWt capacity [1]. The development activities focus on catalytic technologies for cleaning gases. A cogeneration unit based on a combustion engine with 22 kWe capacity is attached to Biofluid 100, and this enables tests to be run on operations of the unit fired by gas generated from the gasification process. The main problem when applying gasification technologies integrated with combustion engines is unreliability of the units [1–3]. Gas cleaning is also an energy-intensive and expensive process. The Energy Institute at the Faculty of Mechanical Engineering has therefore participated in several projects working on the design and manufacture of brand-new cogeneration units. This involves the use of biomass combustion or a gasification unit. Flue gas released from the boiler heats the compressed air from the compressor in a heat exchanger. The heated air is supplied to a turbo-set, where it expands.

Cogeneration of energy from biomass and from wastes produces heat that is often difficult to utilize. It is therefore beneficial to design small-capacity units, so-called microcogeneration stations, where the problem with heat utilization is not so striking. These units are constructed directly on the site where the fuel comes from, which is most commonly logging, the wood-processing industry or agricultural industry. Transport costs may have a negative impact on the overall cost-effectiveness of the operation of these energy stations, especially when fuel is to be transported over long distances. Fuel transport may also contribute to heavy traffic on the roads. All of these drawbacks are eliminated when microcogeneration units are introduced. Other benefits include reduction of electrical energy losses in the power transmission system, since the electrical energy is consumed in the close vicinity of the microgeneration unit. Research carried out at the Energy Institute at Brno University of Technology (EI BUT) has recently focused on microcogeneration units of this kind. In cooperation with commercial businesses, we have been participating in developing the Stirling engine, an efficient steam engine designed for generating electrical energy from steam under low capacities. We are also developing a cogeneration station incorporating gasification technology and a hot-air microturbine.

The basic principle is shown in Fig. 1. It involves the use of a biomass combustion or gasification unit. Flue gas released from the boiler heats up the compressed air from the compressor in a heat exchanger. The heated air is supplied to a turbo-set, where the air expands. The air leaves the turbine at a high temperature, and can be used for combustion. The rest is mixed with the flue gas to dry biomass, or serves for heating purposes.

2. Design and balance calculations for a microcogeneration unit

We will now compare six basic options, and modifications to them, for a unit with 200 kW designed electric power which utilizes waste heat for drying biomass. Due to limited space in this paper, we will not provide full specifications of each solution, but only basic parameters and the main differences between the designs.

The basic design parameters were: mass flow of the air, temperature and pressure of the air prior to and beyond the turbine. The desired flue gas temperature at the end point of the technology, i.e. at

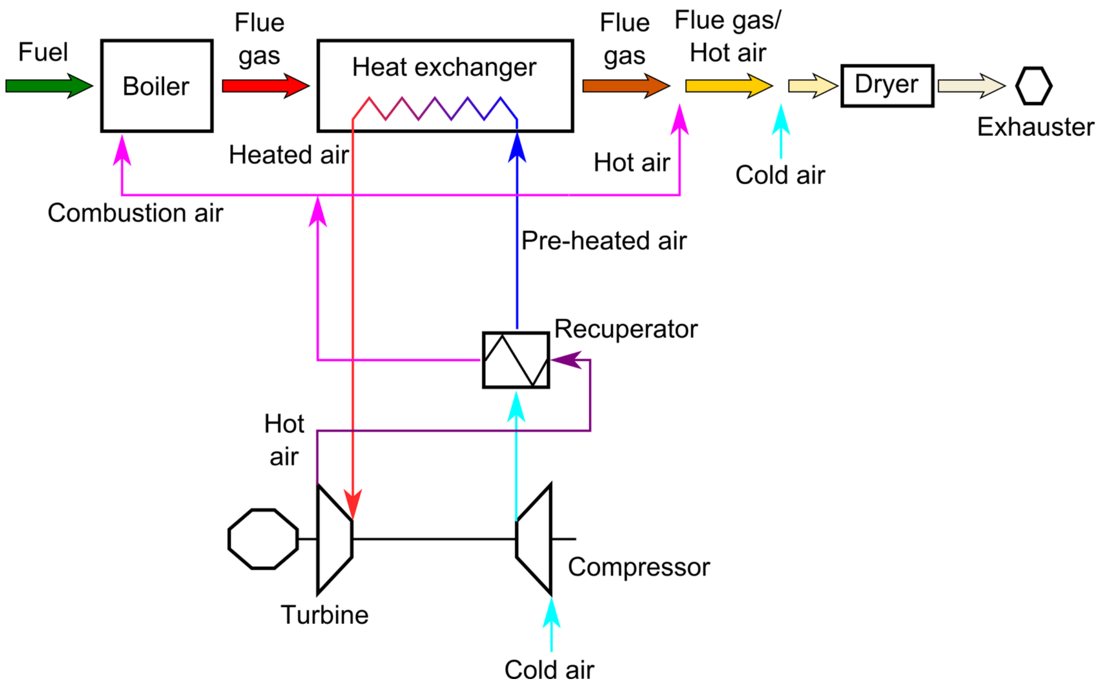

Figure 1. Scheme of the designed technology — Solution 1.

Solution	Amount of air	Temperature prior to turbine	Temperature beyond turbine	Pressure beyond turbine	Amount of fuel	Temperature of combustion air	Flue gas flow rate	Electric power	Thermal power	Electrical efficiency	Thermal efficiency	Overall efficiency
	[kg/s]	[°C]	[°C]	[kPa]	[kg/h]	[°C]	[kg/h]	[kW]	[kW]	[%]	[%]	[%]
1	3.21	750	460	101.3	608.4	240	27195	200	790.7	9.86	39.0	48.9
2	2.59	750	503	101.3	416.0	503	20688	200	601.4	14.42	43.4	57.8
3	3.13	750	518	111.3	490.7	518	25343	200	737.1	12.23	45.0	57.3
4	2.33	850	597	111.3	379.5	597	18112	200	527.3	15.81	41.7	57.5

Table 1. Basic parameters and calculation results for Solutions 1–4.

the inlet to the laboratory oven, was 175 °C. The flue gas temperature at the inlet to the heat exchanger was restricted to 1000 °C because of the service life of the exchanger and the material intensity. A decrease in flue gas temperature was achieved via higher amounts of excess air during the combustion process; however, this requires bigger dimensions of the exchanger. The air pressure prior to the turbine was 400 kPa. The inlet data for the solutions are given in tables, see below.

The solutions differ in the parameters of the working media, and also in several key factors in their circuits. The basic construction characteristics and differences can be briefly summed up as follows:

Solution 1 — air is heated in an exchanger, which is located beyond the boiler for sawdust. Flue gas at a temperature of roughly 1000 °C from the boiler is the heat source. Air is compressed in the compressor, passes through the exchanger and enters the turbine. A certain amount of the air leaving the turbine is used as combustion air; the rest is mixed with the flue gas beyond the heat exchanger. The recuperator for preheating the air beyond the compressor is located in the path of the air stream beyond the turbine.
The degree of recuperation was selected as $\varepsilon = 0.73$. The mixture of flue gas and hot air is blended with cold air, so that the flue gas reaches the desired temperature prior to entering the dryer. The air pressure beyond the turbine is atmospheric, so there has to be an exhauster integrated into the system. For a scheme, see Fig. 1.

Solution 2 — differs from Solution 1 through the absence of a recuperator integrated into the system. The parameters of the air flow rate and the air temperature beyond the turbine are also different.

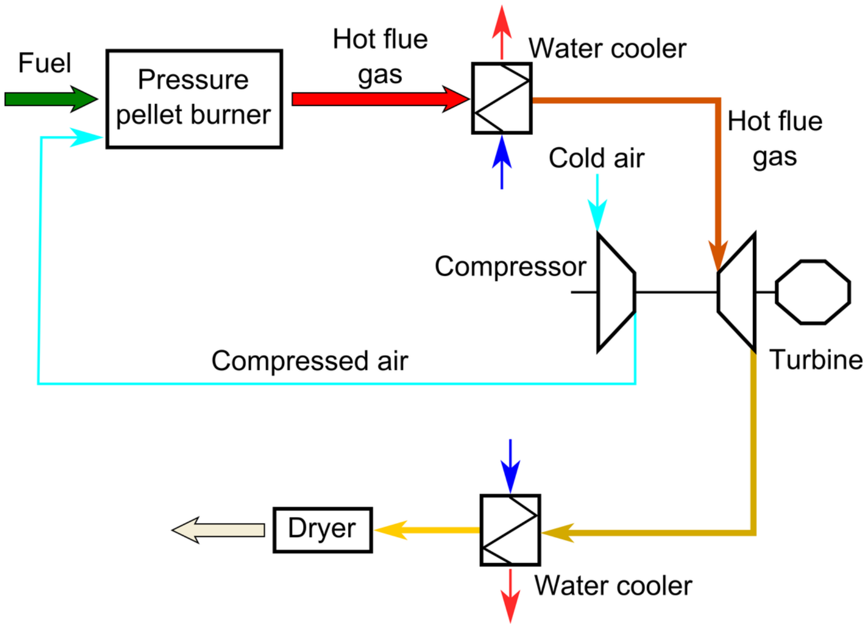

FIGURE 2. Scheme of the technology — Solution 6.

Coefficient of excess air	Amount of fuel	Temperature of flue gas from the burner	Thermal power of cooling beyond the burner	Thermal power of cooler beyond the turbine	Thermal input of the dryer	Total thermal power	Electric power	Electrical efficiency	Thermal efficiency	Total efficiency
	[kg/h]	[°C]	[kW]	[kW]	[kW]	[kW]	[kW]	[%]	[%]	[%]
3.6	427.6	807	21	851	270	1142	193.2	9.4	55.6	65.0
3	508.5	913	346	855	271	1772	193.2	7.9	60.2	68.1
2.4	627.2	1060	815	862	272	1950	193.2	6.4	64.7	71.1
1.8	818.0	1281	1559	872	275	2706	193.2	4.9	68.8	73.7

TABLE 2. Comparison of solutions with a pellet burner — Solution 5.

Solutions 3 and 4 — also do not integrate the recuperator. The overpressure of the air leaving the turbine equals 10 kPa. The combustion chamber will therefore be designed as an overpressure chamber, and again no exhauster will be installed. A major difference in Solution 4 is that the temperature of the air prior to the turbine is 850 °C, much higher than for the other solutions.

2.1. PARTIAL EVALUATION OF BASIC SOLUTIONS

These four basic solutions were compared on the basis of balance calculations, which accounted for the heat losses and pressure drops. In addition, the internal consumption of the equipment, especially the input of the exhauster, was not accounted for. The basic design parameters for particular solutions and key results are presented in Table 1.

A comparison of the solutions showed that Solution 4 seems to be the most useful, as it has the highest rate for electrical efficiency. The low rate of overall efficiency must be caused mainly by significant dilution of the flue gas, with cold air carried out so that the temperature decreases prior to entering the dryer. If recuperation is used, preheated air enters the exchanger with the temperature of the outlet flue gas higher by approximately 200 °C. The flue gas flow rate after dilution then basically doubles. The loss due to the heat in the flue gas increases, and the input of the exhauster becomes disproportionate. Solution 1 has therefore not been included for further specifications. On the other hand, Solution 4 has an integrated overpressure system, so there is no need to install an exhauster. However, this may complicate the design of the overpressure furnace, and may cause difficulties in regulating the boiler and in maintaining optimum efficiency of the operations [4].

Solutions	Air mass flowair	Temperature prior to turbine	Temperature beyond turbine	Pressure beyond turbine	Amount of fuel	Flue gas flow rate	Input of exhauster	Electrical power	Thermal input of dryer	Thermal output for heating	Electrical efficiency	Thermal efficiency	Overall efficiency
	[kg/s]	[°C]	[°C]	[kPa]	[kg/h]	[kg/s]	[kW]	[kW]	[kW]	[kW]	[%]	[%]	[%]
2	2.59	750	503	101.3	416.0	5.75	0	200	791	0	14.4	43.4	57.8
2A	2.59	750	503	101.3	559	5.30	35	165	557	0	8.85	29.9	38.8
2B	2.59	750	503	101.3	559	2.77	18	182	297	378	9.75	36.2	45.9
3	3.13	750	518	111.3	490.7	7.04	0	200	601	0	12.2	45.0	57.3
3A	3.13	750	518	111.3	652.7	6.48	0	200	681	0	9.2	32.3	40.5
3B	3.13	750	518	111.3	652.7	3.34	0	200	364	461	9.2	37.9	47.1
4	2.33	850	597	111.3	379.5	5.03	0	200	737	0	15.8	41.7	57.5
4A	2.33	850	597	111.3	515	4.76	0	200	501	0	11.6	29.2	40.8
4B	2.33	850	597	111.3	515	2.50	0	200	268	336	11.6	35.2	46.8
5	3.6	750	503	101.3	428	2.59	6.8	193.2	270	872	9.4	55.6	65.0
6	3.6	750	518	111.3	517	3.13	0	200	326	1106	8.0	57.7	65.7

Table 3. Overall comparison of all solutions.

2.2. Specification and extension of the designed solutions

On the basis of the acquired data and values, we decided to specify our calculations for Solutions 2–4 in two more solutions.

Solutions A — heat loss in the circuit will be taken into account. The heat loss was determined partially by calculation, and partially from operational experience with a similar 80 kWe unit [5]. Further, the assumed input of the exhauster was specified on the basis of the known flue gas flow rate, [6]. These solutions always take subsequent cooling of the flue gas into account by mixing with cold air and utilization of the mixture in a biomass dryer.

Solutions B — these solutions, in contrast to Solutions A, substitute mixing of flue gas and cold air by cooling in a flue gas/water exchanger and subsequent utilization of the flue gas in a dryer. This design reduces the flow of the flue gas into the stack, losses due to heat in the flue gas, and also the input of the exhauster.

We further evaluated modifications to previous solutions which integrate an overpressure burner for pellets and direct supply of flue gas into the turbo-set. Compressed air will be supplied into the overpressure burner for the pellets. Hot flue gas will be cooled in a flue gas/water exchanger to the desired temperature at the turbine inlet, and the required flow rate also has to be specified. The flue gas leaving the turbine will be cooled again in a flue gas/water exchanger so that the temperature required for the flue gas entering the dryer is achieved. Solution 5 integrates a turbo-set identical to Solution 2 (750 °C, 101.3 kPa beyond the turbine); Solution 6 integrates a turbo-set identical to Solution 3 (750 °C, 101.3 kPa beyond the turbine), i.e. a dryer with overpressure without an exhauster — see Fig. 2.

Calculations of Solutions 5 and 6 were always carried out with different values of the excess-air coefficients. The calculations used the α coefficient in the interval 1.8–3.6. An increase in excess air leads to a decrease in the temperature of the flue gas in the chamber, and also to a decrease in the need to cool the flue gas prior to entering the turbine (input of the hot-water exchanger). The boundary value is roughly $\alpha = 3.6$, where the final flue gas temperature, including the heat loss, reflects the required temperature of the flue gas prior to entering the turbine — see Table 2. In general, greater excess air leads to greater electrical efficiency and lower thermal power, and also lowers the overall efficiency. Considering the analysed solutions, we recommend adopting the solution with maximum excess air, which does not require the installation of an exchanger between the chamber and the turbine, and the flue gas temperature will be controlled directly in the chamber. This solution requires less investment and also eliminates the need to utilize the acquired heat; it also offers maximum electrical efficiency.

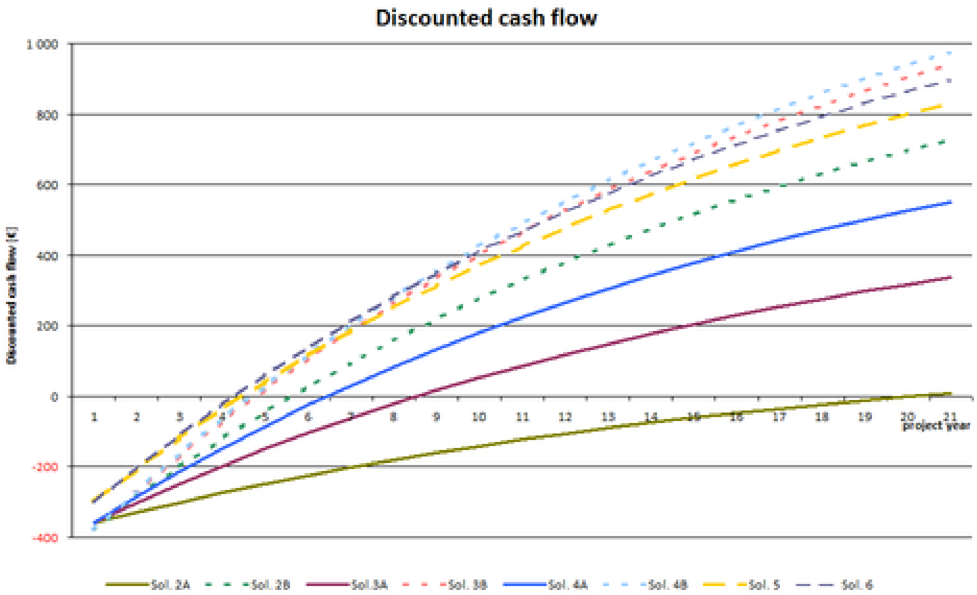

FIGURE 3. Discounted Cash Flow.

Investment	
Gasifier	€ 80000
Turbo-generator	€ 200000
Flue gas/air exchanger	€ 80000
Flue gas/water exchanger	€ 20000
Fixed costs	
Services	€ 15000/year
Service costs	€ 3005/year

TABLE 4. Overview of investments costs and fixed costs.

Variable expenses and selling prices	
Fuel price – sawdust	€ 84/t
Fuel price – pellets	€ 168/t
Feed-in price of electricity	€ 155.2/MWh
Selling price of heat (including green premium)	€ 14/GJ

TABLE 5. Overview of fuel prices and feed-in energy prices.

2.3. OVERALL EVALUATION AND A COMPARISON OF THE SOLUTIONS

Table 3 presents an evaluation of Solutions 2A, 2B, 3A, 3B, 4A, 4B, 5 and 6, and a comparison with the original Solutions 2–4, without considering the heat losses, the pressure drops and the internal consumption of the exhauster. The results correspond with data obtained while participating in the development and commissioning of a similar 80 kWe unit [5].

3. ECONOMIC COMPARISON OF THE PROPOSED SOLUTIONS

Several solutions for microcogeneration units combusting biomass were proposed in previous sections. The optimum solution has to be selected on the basis of the cost-effectiveness of the whole project. An assessment may be carried out using generally known and beneficial indicators. This includes the Net Present Value (NPV), which expresses the appreciation of the investment, including the cost of capital in a given period of time, usually the service life of a particular piece of equipment. Other indicators include the simple return on the investment (which does not take into account the change in the value of money in time) and the discounted cash flow (DCF) or the Internal Rate of Return (IRR). Less significant indicators are the Payback Period (PP), index rentability (IR) and Return on Investment (ROI).

A basic economic assessment requires precise identification of the inlet and outlet commodities. Above all, it is crucial to have available the real value of the investments and the fixed costs. For the purposes of our analysis, the prices of the components of basic units were determined. Small payments and extra work are either included in the components or are neglected. The fixed costs include only costs directly linked with operations of the unit. Expenses linked with other services (e.g. book-keeping, company's overhead, etc.) are neglected.

In addition, we have to identify the prices of the input materials (fuel) and the purchase price of output commodities (electrical energy and heat). Energy prices may be identified using trends in market prices or, if possible, using state-guaranteed feed-in

	Simple payback period	Discounted payback period	PP	NPV	IRR	IR	ROI
	[years]	[years]		[€]	[%]		[%]
2A	11.19	19.5	−63.03	9145	0.30 %	1.03	5.10 %
2B	3.94	6.86	11.14	727423	18.00 %	2.91	14.60 %
3A	5.93	10.34	−1.17	336364	9.40 %	1.93	9.70 %
3B	3.3	5.75	15.56	940695	22.80 %	3.48	17.40 %
4A	4.53	7.9	7.3	550978	14.70 %	2.53	12.70 %
4B	3.21	5.6	16.22	977548	23.60 %	3.57	17.90 %
5	3.04	5.3	17.52	832697	25.30 %	3.78	18.90 %
6	2.87	5.01	18.8	897129	27.10 %	3.99	20.00 %

Table 6. Discounted Cash Flow.

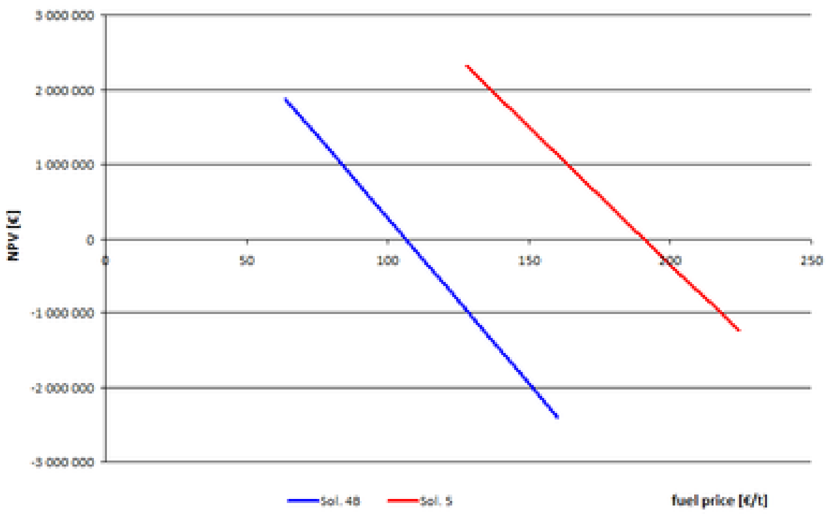

Figure 4. Chart showing the dependence of NPV on fuel price.

prices and green premiums. The calculations provide results for more quantities — operational time 7500 hours/year, discount 6 %. We may neglect pressure drops and heat losses, as mentioned above, as well as potential changes in fuel prices and the prices of both generated energies in time. Predicting these prices is rather complex, and lies beyond the scope of this paper.

We analysed the solutions on the basis of these values for expenses and revenues. The results of the analysis are given in the form of a chart in Figure 3 and DCF Table 6. It is clear that Solutions 5 and 6 have the shortest payback period, thanks to the low initial investment costs. However, the highest revenue over the whole service life may be expected with the application of Solutions 4B and 3B. The solutions that do not integrate use of the residual heat in an exchanger (Solution A) seem to be the worst options. Investment in Solution 2A pays off at the very end of its service life. Solutions 3B, 4B, 5 and 6 all have a reasonable payback period of 5–6 years.

Assessing the economic analyses, we also have to carry out a sensitivity analysis of the quantities, so that we know in advance how a change in one quantity may influence the expected outcome. The following basic quantities were selected for the purposes of our sensitivity analysis: investments, fuel prices, feed-in price of electrical energy and selling price of heat. The sensitivity of each solution to changes in these quantities was basically the same; therefore a thorough calculation was therefore performed for Solutions 4B and 5. We examined the influence of selected quantities on NPV and on discounted cash flow.

The results of the sensitivity analysis are shown in the charts in Figures 4–9, which show the dependence of NPV and discounted cash flow on the price of heat and on investment. On the basis of these calculations, we may say that NPV and discounted cash flow are strongly dependent on fuel prices, the feed-in price of electrical energy, and the selling price of heat. The dependence of discounted cash flow is exponential. In contrast, the dependence on investment is minimal.

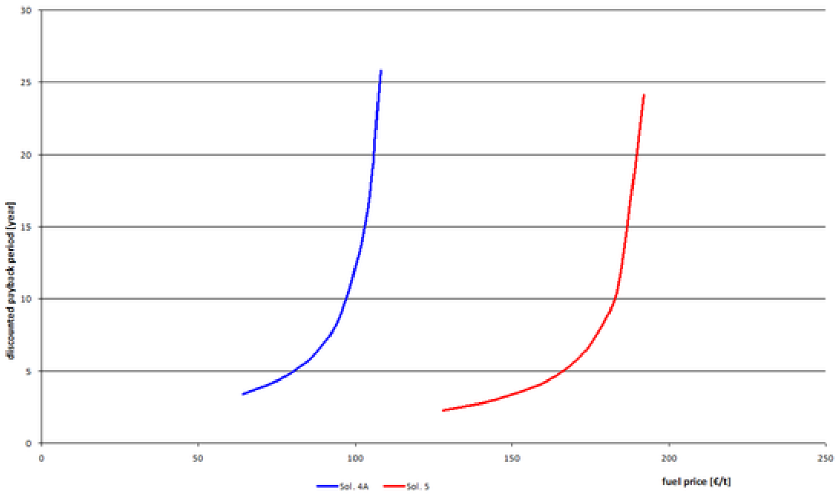

FIGURE 5. Chart showing the dependence of the discounted payback period on fuel price.

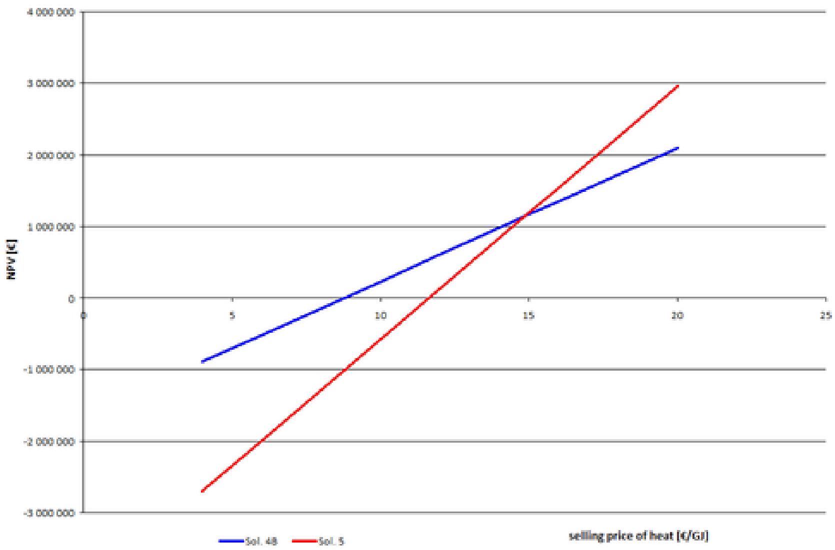

FIGURE 6. Chart showing the dependence of NPV on the selling price of heat.

4. Overall evaluation and a comparison between the options

In Solution 2, the electrical efficiency decreased by 5 % and the total efficiency decreased by more than 10 %, due to the impact of heat loss and the decrease in net electrical power. Although Solution 3 does not have to account for the input of the exhauster, the electrical efficiency and the total efficiency are not significantly higher than for Solutions 2A and 2B. The highest electrical efficiency among all originally designed solutions was achieved with Solution 4, thanks to the higher temperature of the air prior to entering turbine. The internal consumption of the exhauster is also eliminated; however, the whole technology would need to have higher internal pressure, which may make it problematic to seal the combustion equipment. However, a technical solution for this problem may be possible. A possible technical solution might be to cool the boiler and the grate during combustion using very hot combustion air, which enables even inferior and very wet types of fuel to be combusted.

All values prove that Solution B is the most suitable option. Instead of using a supply of cold air, there is an exchanger for heating the water that is designed to reduce the flue gas temperature to the temperature required by the exchanger (175 °C). The greatest electrical efficiency was achieved for the original Solution 4, with a conventional boiler and air heating in the exchanger. The greatest overall electrical efficiency was identified for solutions with a direct supply of flue gas into the turbo-set using a pellet burner.

Although the electrical efficiency did not achieve the expected values of 15–20 %, the results are adequate

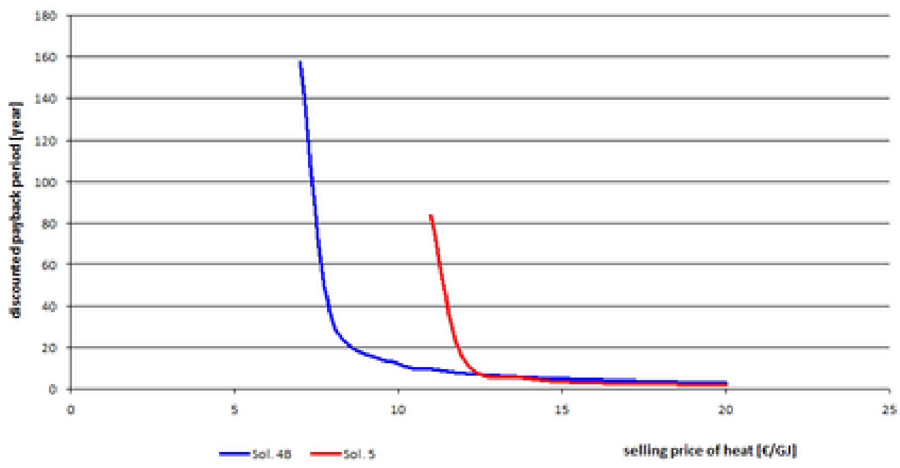

FIGURE 7. Chart showing the dependence of the discounted payback period on the selling price of heat.

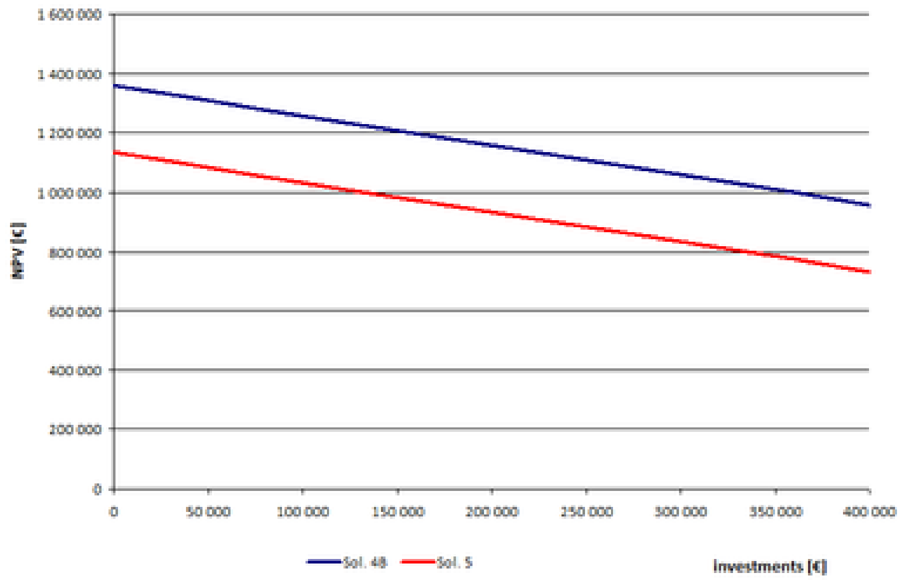

FIGURE 8. Chart showing the dependence of NPV on investments.

in comparison with ORC technology. We also have to bear in mind that the published data describing the electrical efficiency of the ORC technology in the interval 17–18 % [7] refers to the ORC unit only. Heat transferred by the oil heat medium is considered as an input energy flow. Boiler efficiency, internal consumption of the boiler, and consumption of the oil heat medium circuit (oil pump) are not included in the provided data. It is therefore somewhat misleading. After we have included all losses and drops, and also the internal consumption and efficiency of the boiler, we may ascertain that the net electrical efficiency of the ORC technology in relation to the energy in the fuel is equal to approximately 11 %. If the power output is reduced to 50 %, the total electrical efficiency decreases below 9 % for a 1 MWe unit.

Solutions 3B, 4B, 5 and 6 offer the best results in the economic analyses, with little difference between their performances. Their payback period is 5–6 years. We may therefore choose the desired solution from among these options on the basis of the technical criteria. The choice is then presented to the client. The benefits and drawbacks of particular solutions may be briefly summed up as follows:

- Solutions 3B and 4B produce less heat with the same amount of electric power, which will make easier and complete use of the heat.
- Compared with Solution 3B, Solution 4B requires a higher air heating temperature of 850 °C, which is the limit value of the turbo-set manufacturer and may complicate the size and heat load and/or the cooling of the exchanger. This has been substantiated by practical experience from other installations.
- However, Solution 4B achieves higher electrical effi-

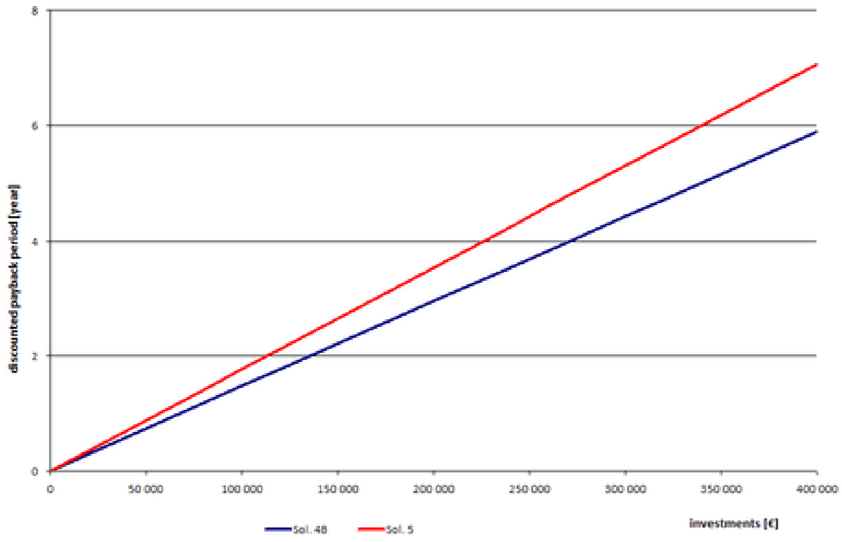

Figure 9. Chart showing the dependence of the discounted payback period on investments.

ciency than Solution 3B, and the overall efficiency is comparable.

- Both solutions account for use of hot combustion air with 10 kPa pressure, so the combustion unit will be operated with higher internal pressure. This leads to problems such as the threat of fires during fuel transportation in the combustion chamber.
- Solutions 5 and 6 do not require the installation of a flue gas/air exchanger, which significantly reduces the investment. In addition, the total efficiency is much higher.
- However, the drawbacks of Solutions 5 and 6 include greater heat production, the need to use standardized high-quality pellets, burner design for combustion at 40 bar pressure and zero content of solid particulate matter in the flue gas, which flows directly into the turbo-set. It is important to have in mind that modification and compression of the fuel is a expenzive and energy-intensive process. This will have a great impact on energy savings and the environmental benefits of the whole technology [8–10].

5. Conclusions

This paper has summarized the situation in energy production in the Czech Republic, with special focus on targets and the direction for the energy sector in the Czech Republic. We have also discussed the use of renewable energy sources and wastes. The second part of the paper presents the main research areas being studied at the Energy Institute at the Faculty of Mechanical Engineering, Brno University of Technology, focusing on microcogeneration using biomass combustion. Technical and economic optimization of the design of a new 200 kW unit is outlined. The paper ends with a comparison of results. The range of electrical efficiency is 9–12 %; the overall efficiency of the unit is 65 %. Our economic analysis shows a payback period of 5–8 years for the unit. These units may be operated in logging, in the wood-processing industry, and in agricultural production, where fuel is provided in the form of waste products. At the same time, all-year-round consumption of residual heat must be secured. The head of the research project anticipates that the technology can be exported and used in developing countries with a poor distribution network. There are typically abundant supplies of very cheap biomass in these regions, and a lack of electrical energy. We conclude that microcogeneration technology seems to be a viable and plausible option.

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