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A STUDY OF THE APPLICABILITY OF A STRAW-FIRED BATCH BOILER AS A HEAT SOURCE FOR A SMALL-SCALE COGENERATION UNIT

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Straw-fired batch boilers, due to their relatively simple structure and low operating costs, are an excellent source of heat for a wide range of applications. A concept prototype of a cogeneration system with a straw-fired batch boiler was developed. The basic assumptions were based on the principles of the Rankine Cycle and the Organic Rankine Cycle systems with certain design modifications. Using the prototype design of a system that collects high-temperature heat from the boiler, studies were performed. The studies involved an analysis of the flue gas temperature distribution in the area of the oil exchanger, a comparison of the instantaneous power of the boiler's water and oil circuits for different modes of operation, as well as an analysis of the flue gas. In the proposed system configuration where the electricity production supplements heat generation, the power in the oil circuit may be maintained at a constant level of approx. 20-30 kW. This is possible provided that an automatic fuel supply system is applied. Assuming that the efficiency of the electricity generation system is not less than 10%, it will be possible to generate 2-3 kW of electricity. This value will be sufficient, for an on-site operation of the boiler.

Keywords: cogeneration, biomass systems, micro and small scale systems

1. INTRODUCTION

The appliances available in the Polish market and intended for using biomass in energy production in small or micro-scale applications are primarily based on thermochemical processes of fuel conversion into heat, i.e. combustion, gasification, and pyrolysis. Depending on the local availability of biomass, its price and quality, as well as the investor's expectations concerning the operational parameters, various types of heating boilers are used. In case of residential buildings, particularly single-family houses, the most common solutions include multi-fuel boilers, pellet-burning boilers (with a retort feeder or a gutter feeder) and wood gasification boilers. When it comes to farms, housing estates, industrial facilities as well as schools and other public buildings, the application of straw-fuelled boilers is becoming increasingly popular since straw is inexpensive and widely available fuel. The solutions available in the market include batch boilers for periodic and cyclic combustion of baled straw, devices based on ground straw combustion technology using burning cigar method (continuous operation) and automatic devices for combustion of straw cut into pieces of 5-10 cm in length (continuous operation as well). Due to the low price and the possibility to use a wider assortment of biomass (e.g. wood, woodchips, energy willow, textile waste, sawdust, etc.), the first of the above-mentioned solutions has gained more popularity in Poland.

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The power output of typical straw-fired batch boilers ranges between 40 and 700 kW and their application is primarily associated with the generation of low-temperature heat used in heating systems and domestic hot water systems. The heat also finds special applications in places such as greenhouses, drying rooms, distilleries, etc. In case of such applications, besides the typical heating boilers, it is also common to use oil-fired air heaters with thermal oil as the operating medium, heated to the temperature of 150 – 200 °C. The possibility to generate heat of such high parameters justifies the attempts to use straw-fired devices (including batch boilers) as the source of heat supplied to small- and micro-scale cogeneration systems. These endeavours are in line with the current research work that is being conducted all over the world – the possibility to use biomass boilers in small- or micro-scale distributed cogeneration systems (encompassing systems of up to 10 kW_e) has been the subject of studies for several years (Bernotat and Sandberg, 2004; Brain et al., 1998; Dong et al., 2009). At this moment, there are generally two combined heat and power generation technologies used that are considered in view of biomass utilization: the internal combustion piston engine (IC Engine) and the turbine based CHP plant with either a steam turbine or a vapour turbine working with organic fluids (ORC). Other CHP technologies, such as those based on gas turbines, Stirling engines and fuel cells, are still in the development phase (Borsukiewicz-Gozdur et al., 2014). Special attention should be paid to the ORC system characterized by an electricity output of 2 kW_e. The system was developed by the scientists from the University of Nottingham. The electrical efficiency of the developed CHP system with the selected ORC fluids (HFE700, HFE7100, n-Pentane) is predicted to be within the range of 7.5% – 3.5%, depending mainly on the hot water temperature of the biomass boiler and the ORC condenser cooling water temperature. It has also been experimentally demonstrated that it is possible to generate electricity in such a system (Liu et al., 2011). The same scientific institution developed an experimental ORC system based on a 50 kW pellet-fired boiler, where the electrical power reached approx. 860 W. In this case, the CHP system consists mainly of a biomass boiler, an evaporator, an ORC expander, an alternator, a heat recuperator and a condenser. The heat obtained from biomass combustion in the boiler is used to produce hot water, which is then used to heat up and vaporize the organic working fluid by means of the evaporator. The organic fluid vapour drives the expander rotating the alternator, thus producing power (Qiu et al., 2012). The energy and economic analyses conducted for various configurations of the biomass-fuelled ORC circuits confirm the potential behind the application of this type of installations in small- and micro-scale systems (e.g. in the residential sector). This is illustrated by the installation cost of 5 thousand €/kW with a 3-year pay-back period (Algieri and Morrone, 2013). When it comes to each of the developed solutions, the crucial factor is the optimal choice of the operating medium. For this purpose, special computational models were created, including advanced models with sufficient accuracy. The models were based on the database of the Design Institute for Physical Properties (DIPPR) which includes nearly 1800 substances and the Peng-Robinson-EOS (Drescher and Bruggemann, 2006; Wilding et al., 1998). Another example of an ORC thermodynamic model was built by the scientists of the Chinese Academy of Science using Matlab together with REFPROP. Outcomes indicate that R11, R141b, R113 and R123 manifest slightly higher thermodynamic performance than other substances. On the other hand, R245fa and R245ca are the most environment-friendly working fluids for engine waste heat-recovery applications (Wang et al., 2011). Another issue considered in relation to biomass combustion systems is their environmental impact. It was found that under certain circumstances emissions resulting from biomass combustion do not exhibit a significant impact on the environment. Such calculations were conducted for a known amount of wooden biomass, in case of which an analysis of their thermogravimetric and physicochemical properties was carried out. This data was introduced in numerical calculations performed in CHEMKIN (the calculations were conducted for 5 kg of wood biomass combusted in a grade furnace; the combustion process was conducted under the air excess ratio of 2). Numerical modelling enables the identification of hazardous gaseous pollutants forming during the biomass combustion process such as NO, CO and other substances. It was discovered that the decrease of the CO molar fraction was significant up to 1000 K and was barely observable above 1000 K. The character and the rate of the change of the NO concentration within the considered temperature range, on the other hand, was initially very low, that is, between 600 to 1000 K. Subsequently it increased significantly, which was connected with the increase of the temperature in the combustion chamber. It has also been found that increasing the pressure in the combustion chamber resulted in a higher NO concentration during biomass combustion (below 1200 K), while the concentration of CO was reduced when the pressure was reduced (Magdziarz et al., 2011). To obtain the optimal, lowest possible emission for a given boiler heat output, it is necessary to ensure proper operational parameters of the devices including the stand-by/operation time ratio, the fuel mass streams and the air stream. Rough measurements of dust concentration have exhibited values between 30 and 50 mg/m^3 (Juszczak, 2014).

Promising results of research conducted currently worldwide confirm the rationale of the research that was undertaken. The study encompasses both the assessment of the practical possibilities of the application of high-temperature heat obtained from a straw-fired batch boiler and the construction of a complete small-scale cogeneration system (including an oil circuit, a steam circuit and a cooling circuit), as well as the development of a control system. An additional aspect of the ongoing research covers the analysis of the environmental impact, which has so far been conducted with a flue gas analyser. The subsequent part of the paper presents the results of the research conducted to date, aimed at determining the possibilities of using a batch boiler as a source of high-temperature heat for a small-scale CHP system.

2. METHODS

2.1. The structure of the experimental rig

The basic component of the system for examining the potential use of straw to generate energy was a 180 kW straw-fired batch boiler. The structure of the boiler is characterised by simplicity and functionality. The combustion of straw is conducted with a counter-flow system consisting in the separation of the air supplied to the boiler into primary and secondary streams. The boiler has two combustion chambers. In the first chamber, fuel combustion takes place in controlled oxygen deficiency conditions and the primary air stream is used. The produced gases are then recycled and, while mixing with the secondary air stream, flow into the second chamber where post-combustion occurs. Before entering the stack, the hot flue gas gives up its heat in the fire-tube exchanger. After it passes through, its temperature is approx. 200 – 250 °C. Such an organisation of the combustion process ensures the proper gasification of straw and combustion of the arising gas, thus enhancing the energy-related and ecological efficiency of the boiler (see Fig. 1).

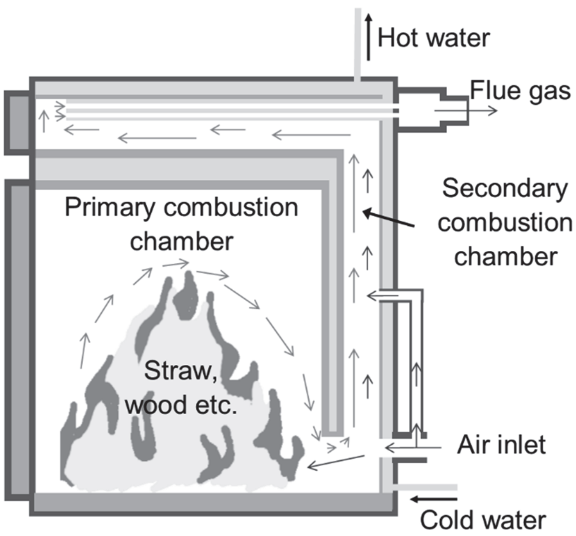

Fig. 1. The structure and operational principles of the biomass-fired batch boiler

The fuel (cubic or round bales) may be fed into the combustion chamber manually, by means of a dedicated fuel feeder, or mechanically. Due to a short combustion time of a single fuel load, a 4 m^3 buffer tank has been used. The tank accumulates excess heat and gives it up to the heating installation at a rate adjusted to the current needs. The tank is connected with air heaters, a heat exchanger and various heating systems, such as wall-mounted heaters, suspended infrared electric radiators, underfloor heating, etc. (Sornek et al., 2014a).

2.2. The use of the biomass-fired batch boiler in a CHP system

The fundamental assumption that was made at the beginning of the investigation was that the basic functionality of the boiler as a heat source had to be maintained. Electricity generation was intended to be an additional feature which would not have a significant effect on the thermal efficiency of the device. The developed solution should be applicable both in brand new and existing units. An additional limitation was constituted by the low investment costs which should characterise the new system. As a result, it became necessary to construct a dedicated cogeneration system from scratch, using our own prototype solutions. The new structure was created based on a typical Rankine Cycle (RC) and Organic Rankine Cycle (ORC). However, because it was very difficult to re-scale the solutions applied in the average and large-scale systems, the steam turbine was replaced with a specially-designed steam engine, the simplest version of which can be obtained after modifying an air compressor (Sornek et al., 2014b). The distribution of temperature in the tested batch boiler exhibits a quite high variability over time in different points of the primary (main) combustion chamber. This is largely dependent on the batch size and the arrangement of straw bales. The air is blown from the back of the chamber, which means that the flame (and hence the temperature field) moves forward as subsequent batches of straw burn out. The flame temperature exceeds 1000 °C, thus being in the required range for optimum combustion of straw (850 – 1100 °C). A far more even distribution of temperature occurs in the area between the post-combustion chamber and the fire-tube exchanger, where temperature reaches 500 – 700°C (Sornek et al., 2014c). Taking the previous assumptions into account (electricity production as an additional functionality of the device), the use of heat from this part of the boiler seems to be the most favourable option. The reduction in the temperature of the flue gas, which leads to the minimisation of the stack loss of the boiler and the increase of the overall efficiency of the device, should be considered an advantage in that case.

The concept of the cogeneration system, designed to work with the straw-fired batch boiler, has been presented in Fig. 2.

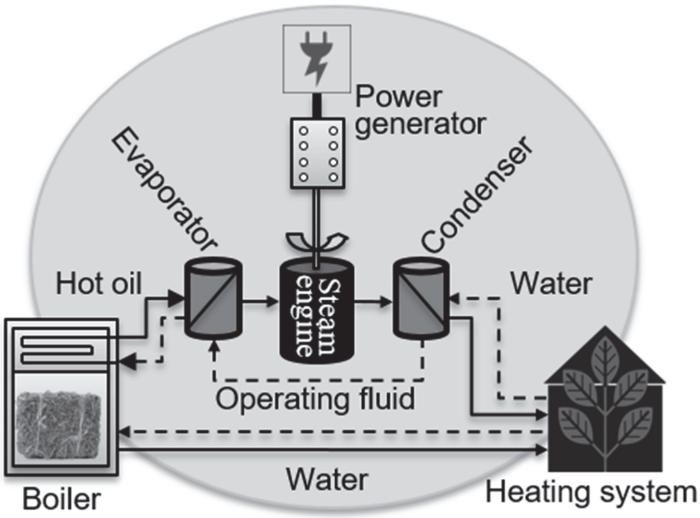

Fig. 2. The general idea behind the use of the biomass-fired batch boiler in a CHP system (Sornek et al., 2014d)

2.3. The structure of the hot oil circuit

The design and structure of the individual components of the heat-to-electricity conversion system required adapting the structural parameters of the batch boiler. For this reason, a special design of the oil exchanger was prepared and developed. The exchanger was placed in the space between the post-combustion chamber and the fire-tube exchanger. In order to maximise the use of available space, the assumed shape was a double meander, which allowed to reach the total length of nearly 20 m. The hot oil heated in the oil exchanger was next transported through a special circuit to the evaporator, which consisted of two commercially available shell-and-tube heat exchangers characterized by a heat exchange area of 1.2 m^2 and by 2.3 and 2.6 dm^3 tube side and shell side capacity, respectively. The maximum operating parameters were 203 °C in case of temperature and 16 bars in case of pressure (for both the tubes and the shell sides). The exchangers serve to combine the oil circuit with the steam circuit, and the way they are connected makes it possible to choose one of three operating modes: a single heat exchanger, two heat exchangers working in a serial connection and two heat exchangers working in a parallel connection. It thus makes it possible to increase the amount of heat given up to the operating medium on the cold side of the exchangers, e.g. in order to reheat it (in a serial connection) or supply an additional device, e.g. an absorption or adsorption chiller (in a parallel connection). The configuration of the tested hot oil system has been shown in Fig. 3.

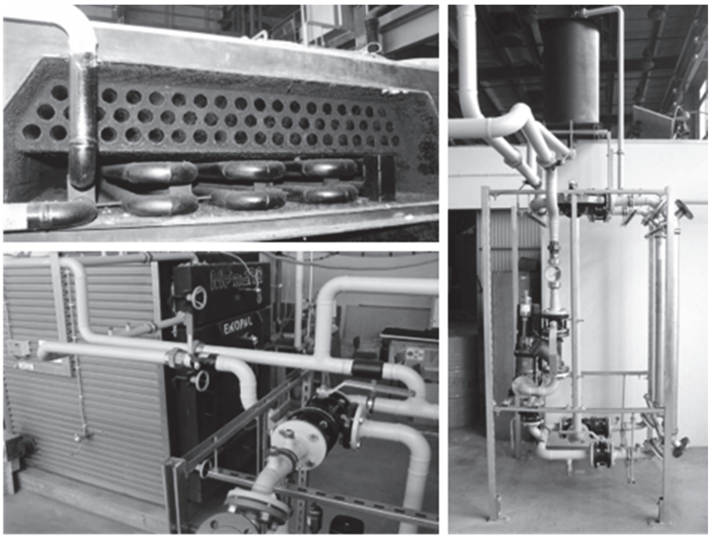

Fig. 3. The configuration of the tested hot oil system

The oil used as a working medium was made of a refined, mineral base oil dedicated for heating devices. The oil contained dispersing and washing additives as well as additives improving the resistance to foaming. Due to that, it exhibited good resistance to thermal degradation and oxidation as well as sufficient viscosity, which results in a smooth system start-up and a highly efficient oil circulation, a long operating time without decomposition products, an increase in the product viscosity and a high heat exchange index. The oil used is dedicated for use as heat carrier in closed-circuit heating systems (for a temperature range from -10 to 285 °C), industrial heating systems, heaters and oil systems for heating including, among others, furnaces fired with solid fuel, provided with additional heat absorption systems.

2.4. Assumptions behind the steam circuit

The steam generated in the shell of the heat exchangers is assumed to be used to drive a specially-designed steam engine, which was constructed as a modification of a piston compressor with cylinders in V-arrangement. The possibility to use the compressor's structure (or the design of other piston-based devices, such as a two- or four-stroke engine) arises from a high degree of structural similarity compared with a steam engine. In the discussed solution, the cylinders and pistons were retained while the inlet and outlet valves were removed and additional pipes were added for the inflow and outflow of steam. Moreover, a flywheel and a timing gear system were incorporated. The resulting structure, although less efficient than the steam microturbine, is much cheaper and less complicated to build. Details related to its operation will be determined in the course of further tests. The first version of the steam engine has been shown in Fig. 4.

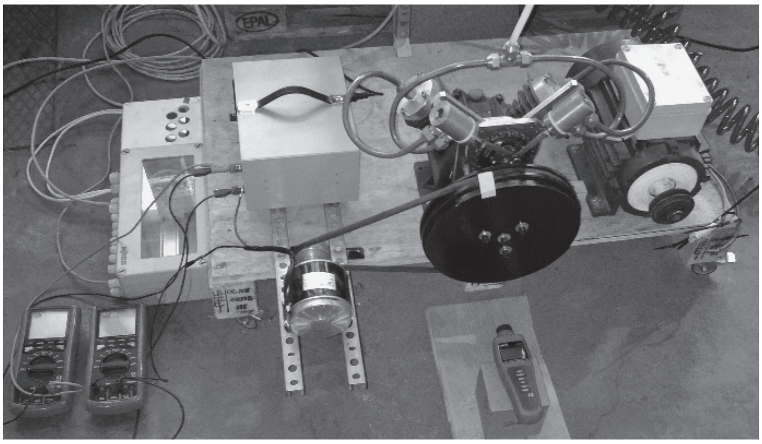

Fig. 4. The first (test) version of a steam engine during tests

At the current stage of investigations, the operation of a prototypical steam engine has been tested with compressed air only (preliminary tests, before connecting the engine to the system). In further stages, it is planned to first use water as a working medium and then test the operation with a low-boiling fluid (the selection of the exact type is currently under consideration). In case of using water and wet low-boiling fluids as working mediums, it is required to equip the steam installation with steam treatment elements (such as a dehydrator, a separator and a tank with a degasser column).

2.5. The measurement and control system

The boiler was originally equipped with a high-pressure fan with an automatically adjustable degree of damper opening as well as with a microprocessor-based controller that managed the straw combustion process according to factory-set parameters. The control signal was the flue gas temperature, the maximum value of which was defined by the user before the start of the combustion process (the default flue gas temperature was set to 200 – 250°C). Additional protection was offered by the water jacket temperature control, which helped to avoid the overheating (boiling) of the water.

The solution applied in the boiler does not allow for a more advanced control of the combustion process, such as the use of an instantaneous power value or the emission of selected contaminants into the atmosphere as the control signal. Therefore, an alternative control and measurement system with a PLC was developed for the purposes of the research. The system is equipped with over 40 temperature sensors (located in different parts of the facility), an electromagnetic flow-meter, a flue gas analyser and inverters used for controlling the operation of the boiler's blowing fan. It also includes a circulation pump in the water jacket circuit of the boiler – an accumulation tank as well as the circulation pump in the accumulation tank circuit – heat recovery units. The analyser uses the electrochemical method of O_2 concentration measurement (0-21% range) and the NDIR method for measuring the concentrations of CO (0-100,000 ppm range), CO_2 (range 0-20%), NO (range 0-2,500 ppm), NO_2 (0-500 ppm range) and SO_2 (0-1,000 ppm range). Moreover, the NO_x concentration is calculated. The dedicated visualisation created using CoDeSys software makes it possible to regularly control the combustion process as well as acquire measurement data in any defined time range – typically from one second up to a few minutes (Filipowicz et al., 2011). The position of the control and measurement elements used in the described tests has been shown in Fig. 5.

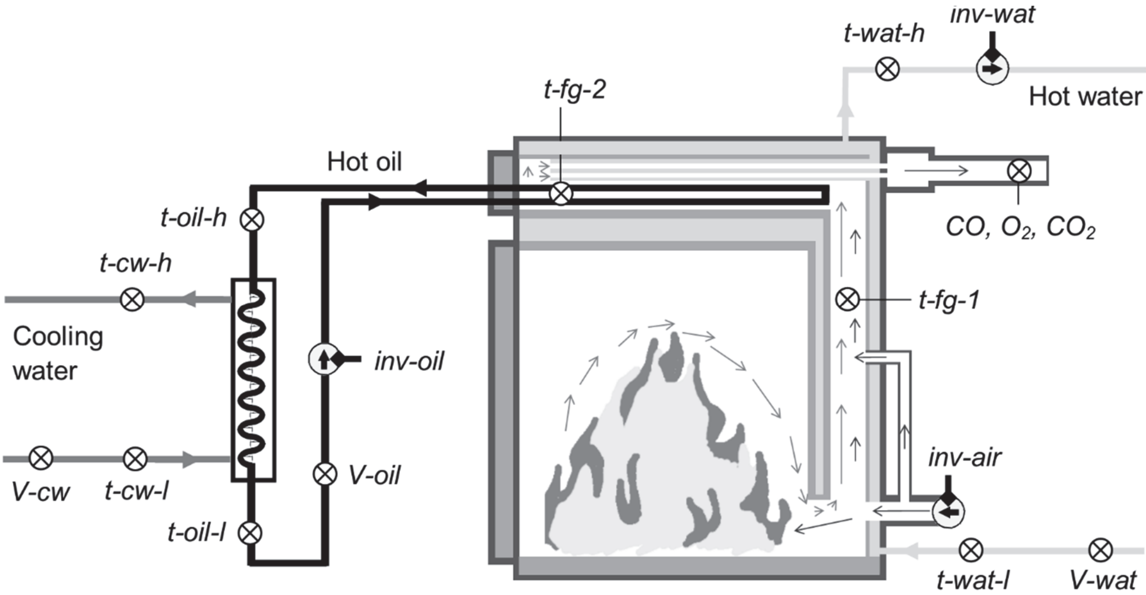

Fig. 5. The position of the control and measurement elements

From the perspective of the research aimed at determining the possibilities to use a straw-fired batch boiler as a heat source for a small or micro-scale cogeneration system, it is crucial to identify the volume of the high-temperature heat that can be obtained from the unit. In this case, the following measurements were conducted: the temperature of flue gas in the post-combustion chamber of the boiler (*t-fg-1*) and at the outlet from the fire-tube exchanger (*t-fg-2*), the temperature in the water circuit (*t-wat-h*, *t-wat-l*), in the oil circuit (*t-oil-h*, *t-oil-l*) and in the cooling water circuit (*t-cw-h*, *t-cw-l*). In order to determine the instantaneous power obtained in the water circuit, the oil circuit and the cooling water circuit, the flow of the operating medium was measured in each of the circuits (*V-wat*, *V-oil* and *V-cw*, for the water, the oil and the cooling water circuits, respectively). Moreover, the concentrations of selected pollutants in the flue gas (CO, CO_2 and O_2) were measured at the section of the stack behind the boiler's flue.

The control of the facility's operation was performed by means of inverters adjusting the flow of the air blown into the combustion chamber (*inv-air*), the flow of water in the water circuit (*inv-wat*), the flow of oil in the oil circuit (*inv-oil*) and controlling one of the shut-off valves (for the adjustment of the cooling water stream).

3. RESULTS OF THE EXPERIMENT

3.1. The course of the research

The tests were divided into a few series. In the A case, measurements were commenced with a cold boiler (the temperature of the boiler's structure, water and oil was equal to the ambient temperature), and in the B case, the measurements were conducted with a pre-heated boiler (the temperature of the elements resulted from the first series of tests). 80 kg of straw with an average humidity of 11 – 12% was burned in each of the series.

Table 1. Operational parameters of the biomass boiler's system elements

Parameter	Symbol	Case A	Case B
Air blowing into the boiler	*air-inv*	variable during the combustion process	variable during the combustion process
Flow of the water in the water circuit	*V-wat*	70-72 l/min	100-103 l/min
Flow of the oil in the oil circuit	*V-oil*	variable during the combustion process	variable during the combustion process
Flow of the water in the cooling water circuit	*V-cw*	20-21 l/min	10-11 l/min
Pre-heating	-	No	Yes

The straw was formed in rectangular bales with the dimensions of 80×40×40 cm. The hot oil temperature (*t-oil-h*, Fig. 5) was selected as the control signal of the system's operation. In this part of the study, it was assumed that the temperature of the hot oil should not exceed 150 °C (in practice, the temperature was maintained at 130 – 140 °C) and the temperature variations were stabilized by modifying the inlet air stream (*inv-air*) and the oil flow (*inv-oil*). The operational parameters of the biomass boiler's system elements have been shown in Table 1.

In practice, the air fan (*inv-air*) operated with maximum power except for the starting phase and for the times when the hot oil temperature (*t-oil-h*) was too high (in this case the power of the air fan was reduced by 5% at a time step of 10 s and then increased again). The graph of the air fan's relative power (*air-inv*) controlled based on the temperature of the hot oil (*t-oil-h*) has been presented in Fig. 6.

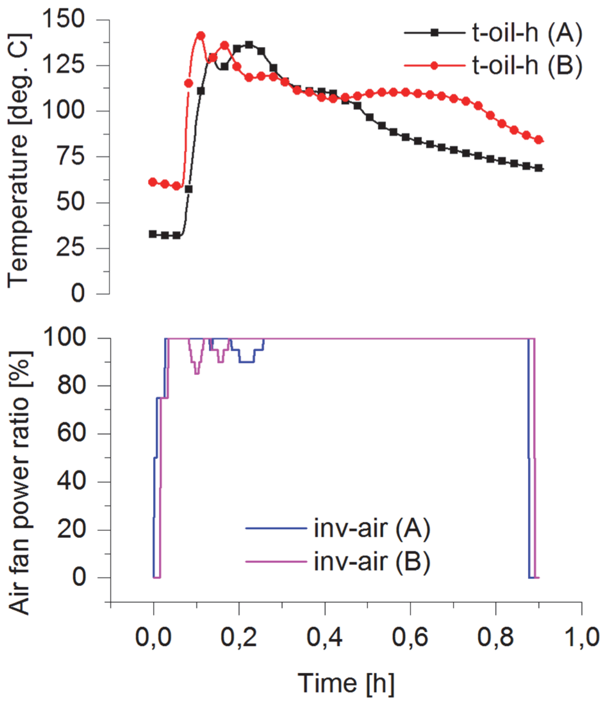

Fig. 6. Air fan's relative power controlled based on the temperature of the hot oil

3.2. Temperature distribution in the oil heat collection system

During normal operation of the boiler, the temperature of the flue gas flowing through the oil exchanger area decreased from approx. 800 °C (value measured at the *t-fg-1* point) to approx. 400 °C (value measured at the *t-fg-2* point) – see Fig. 7. Due to the variability of the fan power depending on the hot oil temperature (*t-oil-h*), as well as the different working parameters of the entire system, in the A case the flue gas temperature in the *t-fg-1* point remained at the level of approx. 800 °C significantly longer than in the B case. In the B case, however, the increase of the oil temperature at the beginning of the combustion process was notably faster – this resulted from the fact that both the boiler and the oil were pre-heated. The changes of the temperature in the oil exchanger area as well as the temperature of the hot oil (*t-oil-h* point) have been presented in Fig. 7.

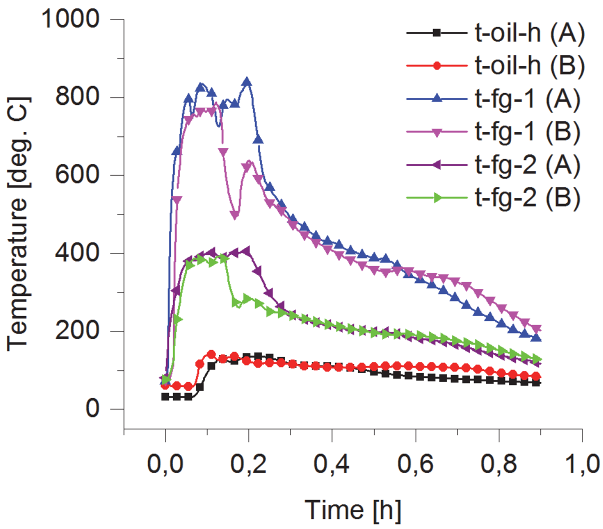

Fig. 7. Temperature variations in the oil heat collection system

3.3. Variation of the power of the boiler and the oil circuit

The power was measured according to the well-known formula:

$$P = \dot{m}\, c\, \Delta T \tag{1}$$

It may be estimated that the accuracy of the mass flow was at a level of 0.5 – 1.0% of the measured value for water and 2.0 – 3.0% for the oil flow meter. The temperature difference accuracy was ca. 0.5 °C both for water and the oil circuit. The final value was ≈ 1.0% for power of the water circuit and ≈ 3.0% for the oil circuit.

Due to the fact that the flue gas temperature at the inlet to the oil exchanger area is higher in the A case and the temperature drop in that area is practically identical in both cases, the amount of heat received by the oil is higher in the A case. The maximum values of the instantaneous power of the water and oil circuits, however, are at a similar level. As indicated by the graphs presented in Fig. 8, the highest power of the oil circuit (nearly 40 kW) was reached in the initial phase of the combustion process, when the power of the water part of the boiler increases. Next, the power of the oil circuit decreased to approx. 14-18 kW and maintained this level until the fuel feed was burned and the power of the water part of the boiler decreased.

The effect of the time shift between the maximum power of the oil and the water circuits is a result of different dynamics of heat transfer in case of water and oil. The water system contains much more water as a working medium compared to the oil circuit (5 m^3 compared to 0.1 m^3). Also the volume of the water jacket of the boiler is larger than the volume of the oil heat exchanger (1.5 m^3 vs. 0.04 m^3). Of course, the surface of heat transfer is significantly higher in case of the water part than in the case of the oil part, but this does not constitute a significant impact. It is also necessary to take into account the fact that the specific heat of water is approximately two times higher as compared to oil. In the subsequent part of the process, the power of the oil circuit is too low to be used efficiently. Therefore, it is necessary to provide an automatic fuel supply system in order to avoid long downtimes in the generation of high-temperature heat.

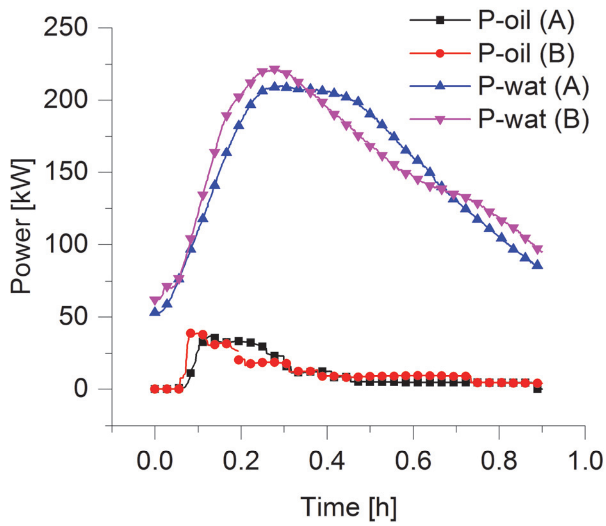

Fig. 8. The histogram of the power of water and oil circuits

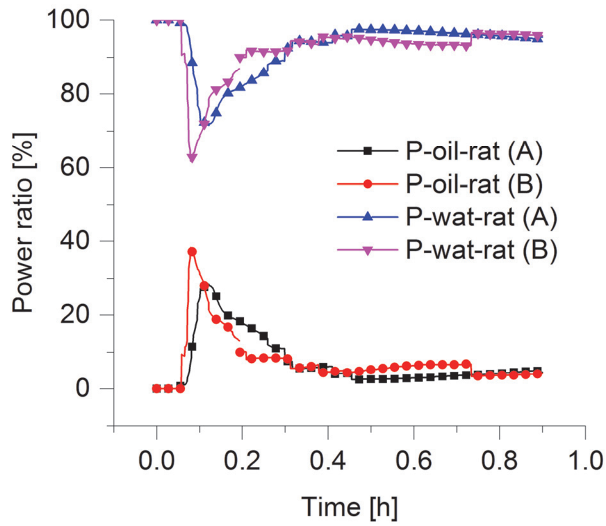

Fig. 9. Share of the oil circuit power and the water circuit power in the overall power of the boiler

The average ratio of the oil circuit power to the boiler's total power (defined as the sum of the water part power and the oil circuit power) is equal to 6.3% for Case A and 6.6% for Case B (details are presented in Tab. 2). The maximum share of the oil circuit power in the boiler's total power output is reached at the time when the power of the oil circuit is at the peak value and amounts to 28.4% for Case A and 38.2% for Case B, respectively. However, in the time between the 4^{th} and 23^{rd} minute, the average share is only 14.2% in the A case and 13.3% in the B case – this is when the oil share is higher taking into account the entire combustion process. The shares of the oil circuit power (*P-oil-rat*) and of the water part power (*P-wat-rat*) in the boiler's total power have been shown in Fig. 9.

Assuming that the heat from the straw combustion is approximately 12.5 MJ/kg (in conditions of 12% humidity), one may conclude that the theoretical heat coming from the combustion of 80 kg of the fuel load is approximately 1.0 GJ. Integrating the boiler power curves *P-wat (A)* and *P-wat (B)* as well as the *P-oil (A)* and *P-oil (B)* from Fig. 8 in the whole time allows for the estimation of the energy transferred from the boiler to the water and oil circuits (see Tab. 2).

In Cases A and B, on average 61.0% of the heat is transferred to the water tank, 4.2% to the oil circuit and 34.8% of the heat is lost. This poor efficiency of the boiler may have resulted from the operation in non-nominal conditions (partial fuel load and additional flow resistance caused by the oil exchanger; the nominal efficiency is ≈ 82%).

Table 2. Energy flow in the system

Case	Theoretical fuel heat	Energy transferred to water	Energy transferred to oil	Total energy transferred to mediums	Energy transferred to oil/Total energy
	MJ	MJ	MJ	MJ	%
A	1000	597	40	637	6.3
B		620	44	664	6.6
Average	1000	608.5	42	650.5	6.45

3.4. Emissions from biomass combustion

The comparison of two fuel combustion processes shows differences in the carbon oxide emission related to the control mode of the operation of the boiler and the water circuit (different powers of the water circulation pump). A crucial role is also played by the fact that the boiler was cold at the beginning of the first combustion process, while it was already pre-heated at the beginning of the second process.

When analysing the curves presented in Fig. 10, one may note that in the first phase of combustion there is a relatively high carbon oxide emission. The emission subsequently decreases reaching a value below 10,000 ppm. This is related to a very low content of oxygen in the flue gas at the beginning of the process. It might be the result of an inadequate air stream being supplied to the combustion chamber and an insufficient mixing of air and flue gas in this area (which consequently leads to improper post-combustion of the flue gas).

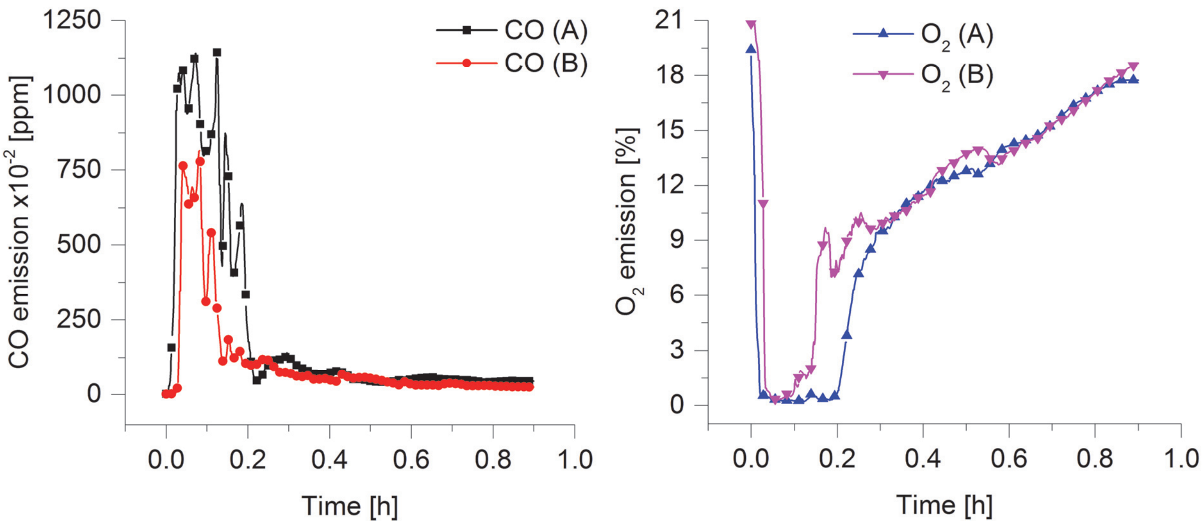

Fig. 10. The amount of CO (normalized to 13% O_2) and O_2 emissions to the atmosphere

The analysis shows that the CO emission rapidly increases for temperatures above 600 °C and for high power of the oil loop. Considering the relation between the CO emissions and the generated power, it is notable that operation at a high power is the most favourable. However, this is not a strong correlation (CO concentration is dependent on other factors). It is, however, possible to control the combustion process by trying to minimize CO content. Special software and improvements of the air mixing system are necessary. This factor also has to be considered in the control software.

4. CONCLUSIONS

The research conducted so far confirms the possibility to use straw-fired batch boilers as sources of high-temperature heat for combined heat and power production (co-generation) systems. In the proposed system configuration, the power obtained in the oil circuit with a single fuel load reached the maximum value of approx. 40 kW only for a short time (~10 min.). On average, this value was significantly lower. As a result, the ratio of the oil circuit power to the boiler's total power (defined as the sum of the water part power and the oil circuit power) has not exceeded 7% (while the maximum share of the oil circuit power reached a level of 38.2%). It is expected that using an automatic fuel supply system, the power of the oil circuit could be maintained at a constant level (e.g. approx. 20 – 30 kW). At the same time it is necessary to introduce automatic operation (using continuous fuel load) of the boiler, which would guarantee continuous operation of the device and proper distribution of the boiler's thermal power between the water circuit and the oil circuit.

The use of the proposed configuration of hot oil collecting system, on the other hand, has a negative impact on the boiler's efficiency. On average, in Cases A and B 61.0% of the heat is transferred to the water tank, 4.2% to the oil circuit and 34.8% of the heat is lost. This poor efficiency of the boiler may result from its operation in non-nominal conditions (partial fuel load and additional flow resistance caused by the oil exchanger; the nominal efficiency is ≈ 82%). To avoid this problem in commercial applications, it is proposed to fill the water jacket with oil and directly heat the oil used to evaporate the water or low boiling fluid in the evaporator.

The selection of a proper control signal for the operation of both the boiler and the oil circuit is very important (the operation of the low-boiling medium circuit and the electricity generator is significant as well). The hot oil temperature should be considered the optimal control signal, because – as demonstrated in the study – it does not directly translate into the power output (and hence, into the amount of heat). The measurement of the instantaneous power could be a good solution to this problem. Such a solution, however is relatively expensive due to the necessity to measure the mass stream of oil as well as the supply and return temperatures.

The obtained results allowed to develop another design of the combined heat and power system with a biomass boiler. This system will be based on the RC/ORC solution and will not be equipped with an additional hot oil heat exchanger (the oil will be filled in the boiler's jacket).

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SYMBOLS

c	specific heat of working medium, $kJ \cdot kg^{-1} \cdot K^{-1}$
$\dot{m}$	mass flow rate of working medium, $kg \cdot s^{-1}$
ΔT	temperature difference between hot (h) and cold (l) point, K

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