

GENERATOR GAS AS A FUEL TO POWER A DIESEL ENGINE

by

Wojciech TUTAK* and Arkadiusz JAMROZIK

Institute of Thermal Machinery, Czestochowa University of Technology, Czestochowa, Poland

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The results of gasification process of dried sewage sludge and use of generator gas as a fuel for dual fuel turbocharged compression ignition engine are presented. The results of gasifying showed that during gasification of sewage sludge is possible to obtain generator gas of a calorific value in the range of 2.15-2.59 MJ/m³. It turned out that the generator gas can be effectively used as a fuel to the compression ignition engine. Because of gas composition, it was possible to run engine with part-load conditions. In dual fuel operation the high value of indicated efficiency was achieved equal to 35%, so better than the efficiency of 30% attainable when being fed with 100% liquid fuel. The dual fuel engine version developed within the project can be recommended to be used in practice in a dried sewage sludge gasification plant as a dual fuel engine driving the electric generator loaded with the active electric power limited to 40 kW (which accounts for approx. 50% of its rated power), because it is at this power that the optimal conditions of operation of an engine dual fuel powered by liquid fuel and generator gas are achieved. An additional advantage is the utilization of waste generated in the wastewater treatment plant.

Key words: engine, generator gas, gasification, sewage sludge, dual fuel

Introduction

Sewage sludge is generated in the sewage treatment plant as a by-product of the biochemical processes of sewage treatment [1, 2]. This sludge is separated from the sewage in settling tanks and then, after the addition of coagulants, these are thickened on belt presses or in centrifuges to form dense pulp. So obtained sludge contains approx 12% of organic substances that are subject to biodegradation and approx. 79% of water, approx 9% of ash, and some small amounts of detrimental substances, e. g. heavy metals – which usually prevent the sludge to be utilized for agricultural purposes. The calorific value of that sludge usually does not exceed 1 MJ/kg, which makes it unable to sustain the combustion process by itself. Within the meaning of the waste management regulations in force, sewage sludge is regarded as hazardous waste, and therefore its storage and processing is subject to regulatory restrictions [3]. To reduce the mass and density of the sludge and to sanitize it, some municipal sewage treatment plants use drying of the sludge in driers at a temperature of approx. 220-240 °C. The calorific value of dried sludge is approx. 11 MJ/kg and is comparable with the calorific value of crude wood (8 MJ/kg), brown coal (9-11 MJ/kg), and dry peat (14 MJ/kg), and this means that the dried sludge can be utilized as an energy raw material. Sewage sludge of a moisture content not exceeding 10% and a calorific value not lower than 10 MJ/kg are not biodegradable and may only be disposed of by thermal transformation [4].

One of the technologies for the thermal neutralization of waste, combined with the use of energy for an energetic purposes, it is gasification process. Obtained in this way generator gas

* Corresponding author; e-mail: tutak@imc.pcz.czest.pl

can be used for decentralized electricity production in combined heat and power generating sets (CHP) of stationary gas turbines or stationary internal combustion engines [5-7]. Sewage sludge gasification is a variation of thermal waste neutralization processes [8-15].

In accordance with the current rules [3, 4, 16] for restrictions on the storage of biodegradable waste, at the Institute of Thermal Machinery (ITM) of the Czestochowa University of Technology, the technology processing of fermented, dried, organic sludge, which are by-product of the process of wastewater treatment in sewage, in electricity and heat was developed.

The currently available systems for gasification of organic substance, mostly wood derived producer gas supply to power generating sets, the installations with co-gas generators with fixed bed and fresh air as the oxidizing medium are dominated. These installations achieve relatively high efficiency exceeding 80% and produce hydrogen-rich gas of the calorific value from 4.0 to 5.5 MJ/m³ [14, 15, 17]. As part of the developed in ITM technology, the prototype installation of gasification of dried sewage sludge was made and tested. The installation also included the gas cleaning installation and electric generator of nominal value of active electrical power of 80 kW powered by turbocharged IC engine adapted to operate as the dual fuel engine: gas and liquid fuel.

The installation dried sewage sludge gasifying and generator gas cleaning

The process of gasification of a solid fuel or biomass (most often wood) is a variation of incomplete combustion conducted with a considerable oxygen deficiency. This process is run in a gas generator called also a gasifier or gas producer. The concept of the proposed gasification installation based on co-gas generator with fixed bed, which as the gasifying medium is used air [18, 19].

Figure 1 presents diagram of the installation dried sewage sludge gasifying and generator gas cleaning cooperating with generating set.

In the case of the co-current gasification the solid fuel is supplied to the top of the chamber, the air is delivered by nozzles situated in the central part of the chamber on its conical constriction. Directly next to the nozzles the combustion zone is placed in which the gasification process occurs. There mainly carbon dioxide and water are produced and in this layer of the red-hot and incompletely burned fuel, which is placed below the combustion zone, the reduction reactions occurred. As a result of these reactions the carbon dioxide and hydrogen are produced. Above the combustion zone the pyrolysis zone is placed where as a results of complex chemical reaction the heavy hydrocarbons, tar-forming gas liquor are produced [11, 12]. Above the pyrolysis zone is placed the drying and degassing zone, in which the release of volatile substances contained in the fuel. The acquired gas during the gasification process is removed through the grate below the lower narrowed part of the chamber of the generator. The vapour gas pitch and other volatile products of gasification process must pass through the combustion and reduction zones where much of it is burned and reduced. Ash remaining after the gasification process is discharged in a natural way by the grate into the ash directly below the grate.

The main problem occurring in gasification systems is a tar, its constituents are polycyclic aromatic hydrocarbons, belonging to the most common, persistent organic pollutants [14, 15]. The amount and composition of tar depends on the type of the reactor, the process parameters (pressure, temperature, and time), the charge properties (type, moisture content, fragmentation of the feed). In the case of use of the generator gas to power an internal combustion (IC) engine is necessary to keep in mind the relatively stringent requirements for gas purity [20,

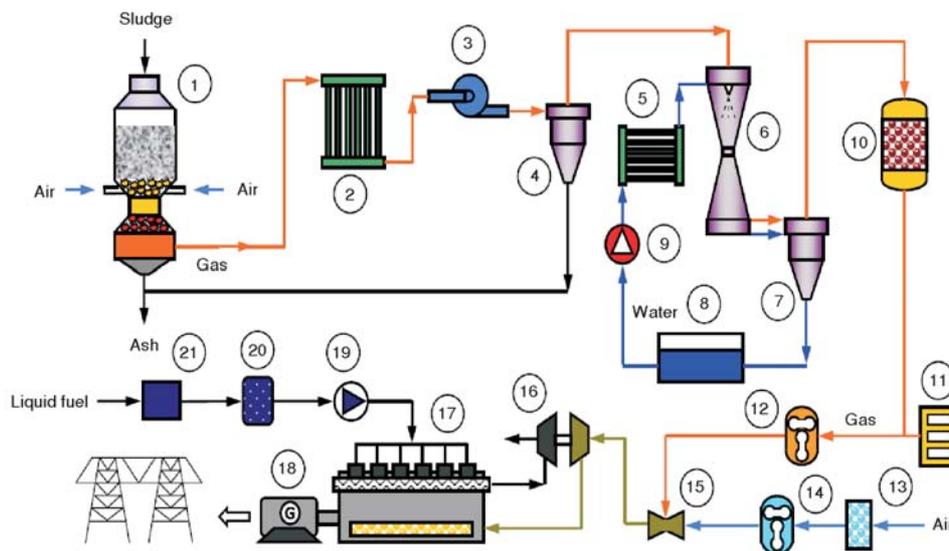


Figure 1. Diagram of the installation dried sewage sludge gasifying and generator gas cleaning cooperating with generating set (combustion engine and generator)

1 – gasifier, 2 – gas cooler, 3 – blower, 4 – preliminary cyclone, 5 – water cooler, 6 – Venturi scrubber, 7 – cyclone, 8 – water tank, 9 – water pump, 10 – gas filter, 11 – gas analyzer (H_2 , CO , CO_2 , CH_4 , O_2), 12 – gas flowmeter, 13 – air filter, 14 – air flowmeter, 15 – gas mixer, 16 – turbocharger, 17 – combustion engine, 18 – generator, 19 – injection pump, 20 – liquid fuel filter, 21 – liquid fuel tank

21]. According to the project of sewage sludge gasification assumptions for gas purification system used cyclone dust collector with a venturi wet scrubbers called [22, 23].

The cleansing installation has enabled the separation of solids parts, dust, and tars included in the received gas which are unacceptable by the engine, especially by the turbocharger and other parts of intake system of engine. The main element of the generator gas cleansing system is a cyclone which uses the centrifugal forces acting on the particles and aerosols. In the classical cyclone dusty gas is introduced tangentially to the cylindrical housing. The shape of the cyclone causes the swirl of gas stream which moves through spiral down the device. As a result of rotational motion the centrifugal force is generated which consequently reject the dust grains of the gas on the outer wall of the device, after which they slide down into the dust collector. The gas stream at the bottom of the cyclone changes direction by 180 deg. and with spiral motion is removed through the outlet pipe of the device. The main advantage of this element is simple and compact design without moving parts and the ability to work at high temperatures and high pressure. The main disadvantage is the significant drop in gas pressure required for effective cleaning. The classical solution as used in the generator gas cleaning system, the gas stream is introduced tangentially to the cylindrical part of the cyclone. Another element of the cleansing installation is a wet dust collector called scrubber. The suction gas stream is cleaning by contact with the liquid spray. The wet cleaning process most commonly used liquid is water. The scrubber also takes part in the process of cooling the gas. The gas stream is supplied to the scrubber, where pollutants are moved from the gas to the liquid. In the gas cleaning system a venturi scrubber with adjustable throat was used. In this type of scrubber, high turbulence and high relative velocities of gas and water drops determine the efficiency of the gas cleaning. A

very important parameter is the length of the Venturi scrubber throat. The liquid is supplied through nozzles which disperse the water into droplets which are entrained with the gas into the throat. An important parameter affecting the efficiency of the scrubber is the ratio of liquid volume to gas volume. With increasing gas velocity the droplet size decreases but the number of droplet increases, which increases the efficiency of the cleaning process. As a result of the turbulent gas-liquid contact to form droplets which are entrained with the gas cleaned. In order to remove liquid droplets from the gas cyclone demister is used. In this process the liquid absorbs heat from the hot gas, therefore there is the additional cooling system required to lower the temperature of the fluid which supplied the nozzle of the scrubber. Radiator cooling system is closed. After the cleaning process the gas is provided to filter chamber containing a porous and hence to the generator.

The modernization efforts brought about reasonably stable operation of the gasifier fed with dried sewage sludge. The tests of the operation and functionality of the gasification installation and the generator gas purification installation included, among other things, the following: composition of the obtained generator gas, the leakproofness of the gasifier (the measure of the leakproofness is the oxygen content of the obtained gas), the quality of gas purification from pitchy substances, and ash content. The stability of the gasification process was achieved after 3 hours from the gasifier start-up. As a result of gasification of 1 ton of sewage sludge, at least 1450 m³ of generator gas of a calorific value in the range of 2.15-2.59 MJ/m³ can be obtained.



Figure 2. View of grains of dried sewage sludge before and after gasification

Figure 2 presents view of sewage sludge before and after gasification. The residue from gasification mainly consists of minerals. Table 1 presents the chemical composition of dried sewage sludge used during researches.

In order to determine the composition of the generator gas, it was used a set of two gas analyzers Signal and Siemens:

- Siemens COLOMAT 5 – H₂,
- Siemens Ultramat 23 and methane-only FID gas analyzer Signal 3000HM – CH₄,
- Signal 7000FM and Siemens Ultramat 2 – CO,
- Signal 7000FM and Siemens Ultramat 23 – CO₂,
- Signal Paramagnetic Oxygen Analyzer Model 8000M – O₂, and
- Hydrocarbon analyzer heated FID – Signal 3000HM – THC.

Some measurement systems have been duplicated in order to increase the reliability of the measurements. In both cases, similar values were obtained. The chemical composition of the

Table 1. The average chemical composition of the dried sewage sludge

Parameter	Value
Carbon	32.2%
Hydrogen	4%
Nitrogen	3.5
Oxygen	16.3%
Sulfur	1.5%
Humidity	5%
Ash	37.3%
Calorific value	12.3 MJ/kg
Sludge gasification productivity	1.45 m ³ /kg

sludge was determined in a specialized laboratory: Institute of Chemical Processing of Coal with the Accreditation Certificate of the Research Laboratory awarded by the Polish Accreditation Centre.

Table 2 provides the average generator gas composition values (H_2 , CO , CO_2 , CH_4 , O_2) along with calorific value (W_g) and theoretical air demand value (L_t), as obtained from a series of measurements carried out after the final upgrading of the sewage sludge gasification installation. After attaining the stable operation of the gasifier and satisfactory composition of the generator gas, tests of the generator gas and diesel oil-fuelled internal combustion engine were carried out.

Table 2. Averaged values of the composition of the generator gas obtained by gasification of dried sludge and calorific value and the theoretical air requirement

	H_2 [%]	CO [%]	CO_2 [%]	CH_4 [%]	O_2 [%]	W_g [MJm ⁻³]	L_t [m ³ m ⁻³]
Average value	3.81	13.40	7.69	0.97	3.84	2.44	0.50
Maximum value	4.4	14.18	8.10	0.98	3.99	2.59	0.52
Standard deviation	0.13	0.79	0.41	0.01	3.69	0.09	0.02

Figure 3 presents the example of gas composition which was obtained during the gasification process. Over the 2-2.5 hours after the start of the installation the stable parameters were obtained.

The generating set

The object of investigation was a 6CT107 turbocharged auto-ignition internal combustion engine powered by diesel oil fig. 4(a), installed on an ANDORIA-MOT 100 kVA/80 kW power generating set in a portable version. The engine was equipped with pressure sensors in each cylinder [24]. Figure 4 presents the test engine equipped with measuring system and section of the engine cylinder with visible valve, injector and pressure sensor. The shape of combustion chamber is visible as well.

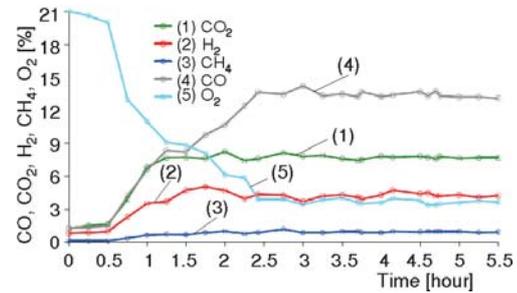


Figure 3. The gas composition in the gasification process

Figure 4. The 6CT107 test internal combustion engine

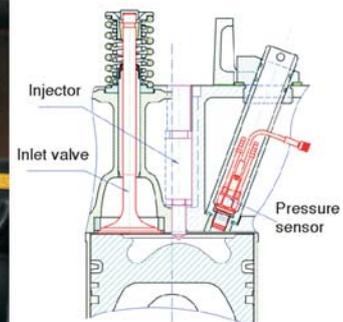


Table 3. Engine specification

Parameters	Value
Power	80 kW
Number of cylinders	6
Rotational speed	1500 rpm
Displacement	6.54 dm ³
Stroke	60.325 mm
Cylinder bore	107.19 mm
Compression ratio	16.5
Injection angle	9° ± 1.5°

The generating set driver by the 6CT107 turbo-charged engine was adapted to dual fuel operation by equipped it with an additional gas supply system and liquid and gaseous fuel dosage control systems based on WOODWARD parts. The generator gas – air mixture was delivered to the engine with the assumed ratio. Roots-type flowmeters were used. Mixing occurred before the turbine in a classic mixer. The engine of this set was adapted to simultaneous indication of the all six cylinders and equipped with measuring instrumentation necessary for taking measurements of basic load characteristics and with a set of analyzers to measure chemical

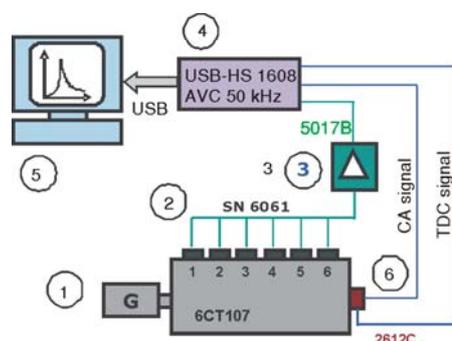
composition of generator gas supplied to the engine. The engine and electric generator control systems allow either the powering of a group of loads isolated from the power grid or parallel operation with this power grid. Table 3 presents the engine specification used during researches.

The tests included the simultaneous, synchronized measurement, and recording of pressure variations in the individual cylinders of a compression ignition (CI) turbocharged six-cylinder engine of a displacement of 6.54 dm³. During the measurements, the engine was driving, at a constant rotational speed of 1500 rpm, an asynchronous electric generator being loaded with resistances switched on sequentially. In order to be able to make the analysis of the recorded pressure variations as a function of both time (especially in the combustion phase, when large high-frequency pulsations are likely to occur) and the crank angle (CA), simultaneous recording of the variations of pressure and CA and TDC tracer signals as a function of time at a frequency of 50 kHz was carried out.

The study was conducted on a test stand consisting of the following measurement apparatus:

- pressure sensor – Kistler 6061 SN 298131,
- measurement computing USB-based 8 channel 16 Bit 50 kHz data acquisition device – USB-HS 1608,
- crank angle encoder – Kistler CAM 2612c – resolution 1024 pulses/rev, and
- charge amplifier Kistler type 5017B multichannel.

Recorded in a digital form, the signals were subjected to resampling. At the moments of successive CA tracer pulses, which made it possible to obtain the variations of pressure as a function of CA with a step of 0.5 deg. At the same time, the basic parameters of gasification unit operation and composition of produced generator gas were also recorded. For the acquisition of all fast-varying signals, a USB-HS 1608 eight-channel module operated by its own program was used. In each recorded measurement series, 500 cycles were recorded for each of the six cylinders. Diagram of the engines indication system is shown in fig. 5.

**Figure 5. Diagram of the engine indication system**

1 – generator set, 2 – pressure sensor, 3 – eight-channel charge amplifier, 4 – eight-channel module of signals acquisition type, 5 – PC computer, 6 – crank angle encoder

The excess air ratio was determined on the basis of simultaneous measurement of fuel consumption of liquid and gaseous and an analysis of the generator gas composition.

Research results

Engine test was carried out after thermal stabilization, the two operating points, at partial and full load, respectively equal to 40 kW (50% of nominal power an electric generator) and 80 kW (100% of nominal power an electric generator).

The different ratios of the energy delivered to the engine in the liquid fuel (diesel) – Q_d , to the sum of the energy supplied in diesel and generator gas – Q_{tot} was analyzed. In the part-load conditions for $Q_d/Q_{tot} = 1.0$, for $Q_d/Q_{tot} = 0.41$, and for $Q_d/Q_{tot} = 0.24$ the indication of the test engine was conducted (tab. 4). Excess air ratio λ of the mixture was: 2.3 for $Q_d/Q_{tot} = 1.0$, 2.7 for $Q_d/Q_{tot} = 0.41$, and 2.85 for $Q_d/Q_{tot} = 0.24$. In the full-load conditions for $Q_d/Q_{tot} = 1.0$, for $Q_d/Q_{tot} = 0.71$ and for $Q_d/Q_{tot} = 0.63$ the indication of the test engine was conducted as well. Excess air ratio λ of the mixture was: 1.6 for $Q_d/Q_{tot} = 1.0$, 1.75 for $Q_d/Q_{tot} = 0.71$, and 1.5 for $Q_d/Q_{tot} = 0.63$ (tab. 4).

Table 4. The analysis results of the test engine indication

Load	Q_d/Q_{tot} [-]	λ [-]	O ₂ [%]	IMEP [MPa]	η_i [%]
50%	1.00	2.3	21	0.65	37.8
	0.41	2.7	15		44.3
	0.24	2.85	14		46.1
100%	1.00	1.6	21	1.2	38.2
	0.71	1.75	17		39.9
	0.63	1.5	15		36.5

Figure 6 shows courses of change of pressure of the test engine recorded at partial load (50%), powered by the diesel, and 41% and 24% share of diesel, as well as at full load (100%), engine powered by diesel fuel, and 71% and 63% share of diesel.

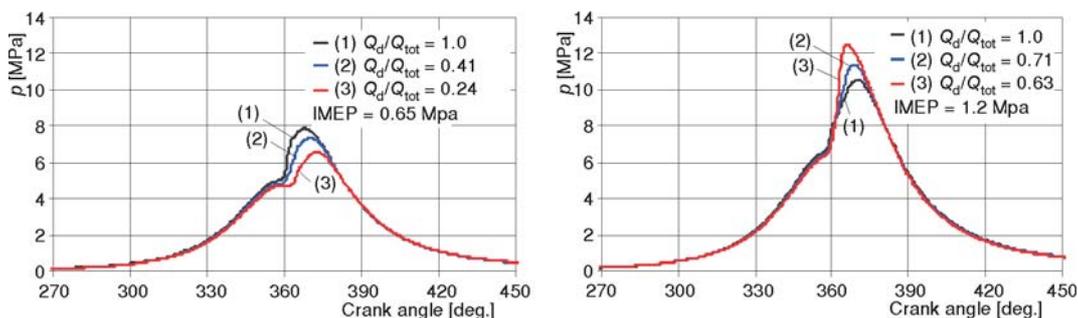


Figure 6. Indication diagrams of dual fuel engine with different liquid fuel energy share for the partial load

The IMEP (indicated mean effective pressure) and indicated efficiency η_i were calculated on the basis of the recorded pressure courses and fuel consumption recorded during researches [25, 26]. At 50% of engine load the IMEP was equal 0.65 MPa and at 100% load the

IMEP was equal 1.2 MPa. The value of the indicated efficiency depended on the energetic share of diesel in the total energy supplied to the engine. The indicated efficiency at 50% of engine load was: 37.8% at $Q_d/Q_{tot} = 1.0$, 44.3% at $Q_d/Q_{tot} = 0.41$, and 46.1% at $Q_d/Q_{tot} = 0.24$. The indicated efficiency at 100% of engine load was: 38.2% at $Q_d/Q_{tot} = 1.0$, 39.9% at $Q_d/Q_{tot} = 0.71$, and 36.5% at $Q_d/Q_{tot} = 0.63$ (tab. 4).

Figure 7 presents the rate of pressure rise as a function of crank angle – $dp/d\phi$ at the various energetic share of liquid fuel (diesel), at part load (IMEP = 0.65 MPa) and full load (IMEP = 1.2 MPa). The rate of pressure rise $dp/d\phi$ is the increase of pressure at 1° CA [20]. In part-load conditions, the largest rate of pressure rise was observed in the engine powered by the diesel and was equal 1.03 MPa/deg. and this value was achieved 1° aTDC. On the other hand the smallest value of this parameter was achieved for 24% share of diesel and was equal 0.39 MPa/deg. and this value was achieved 5° aTDC. In the full-load conditions the largest value of rate of pressure rise was achieved at 63% share of diesel in the fuels (diesel + generator gas) mixture and was equal 1.73 MPa/deg. and this value was achieved 3° aTDC. Engine powered by only diesel fuel the rate of pressure rise was the smallest and was equal 0.72 MPa/deg. and this value was achieved at aTDC.

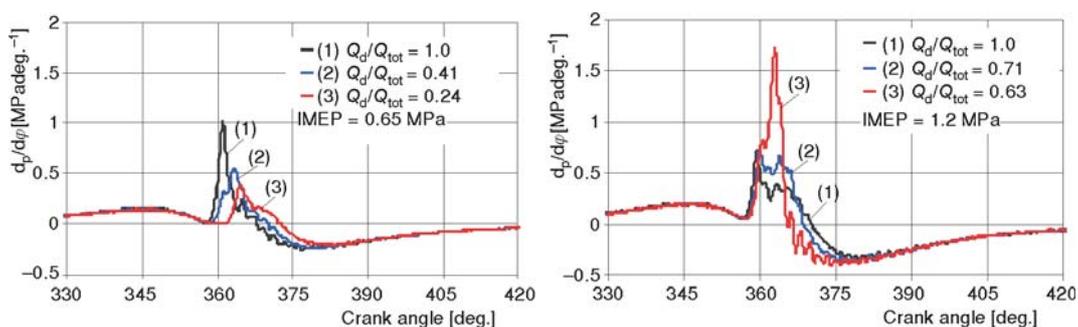


Figure 7. The rate of pressure rise of dual fuel engine combustion at different liquid fuel energy share for the partial load

The rate of pressure rise has a significant impact on the character of engine working, so-called hard work. The mean rate of pressure rise should not exceed 0.3-0.6 MPa/deg. At this rate of pressure rate it can be speak of a relatively soft operation of engine. If, however, $dp/d\phi$ reaches value of 0.7-1.0 MPa/deg., the engine work becomes hard, noisy, with a clearly audible characteristic knocks [27]. On the degree of the engine work hardness the ignition delay has a direct influence. Extending the period of ignition delay causes the rapid combustion process, which subsequently causes a high rate of pressure rise and hard work of the engine. The rate of pressure rise during the combustion process has the decisive influence on: engine noise (hard work) and dynamic load of the crankshaft, and above all, on the piston pin bearing and crankpin.

Figure 8 shows the heat release rate – HRR, as a function of crank angle at various energetic share of liquid fuel, for part – (IMEP = 0.65 MPa) and full-load (IMEP = 1.2 MPa). These values were calculated using the first thermodynamics law on the basis of pressure obtained from measurements [28]. In part-load conditions, the largest heat release rate was achieved in engine powered by diesel and was equal 279.5 J/deg. and was reached 1 deg. aTDC. In full load conditions, the highest rate of heat release in the engine was achieved in the case of 63% share of diesel and was equal 557.5 J/deg. and was achieved 3 deg. aTDC.

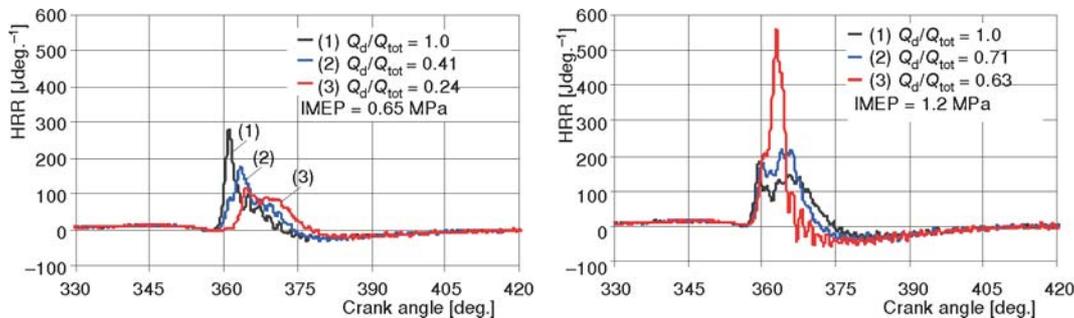


Figure 8. The rate of heat release (HRR) of the dual fuel engine with different liquid fuel energy share for the partial load

It is supercharged engine, consequently, at partial load less charge is delivered to the cylinder during the intake stroke, the lower temperature is reached at the time of ignition. In addition, the engine has fixed constant angle of injection of diesel fuel (tab. 3). These conditions cause the ignition delay is much higher here than at full load. This is clearly shown in fig. 8 which showing the rate of heat release. I case of full load, to the cylinder more air or a mixture of generator gas and air is supplied, and at the beginning of the diesel fuel injection the higher pressure and thus also a higher temperature is reached. Although in both cases at the time of fuel injection, the temperature reached is higher than the auto-ignition temperature of diesel fuel, on the basis of the combustion kinetics theory the oxidation reaction rate strongly depends on the temperature [25]. In fig. 8 for a full load can be seen that for all three cases, the ignition occurs almost at the same time. The courses of HRR which are a very good source of information about the combustion process in the engine, shows that with the increase in the share of generator gas combustion process occurs faster and faster. It is related with the properties of gases contained in the generator gas composition, such as H_2 , CH_4 , and CO . Especially hydrogen and methane has a much higher laminar flame speed than diesel fuel. In addition, the calorific value of methane and hydrogen in particular is very high. It is these properties, is likely to result in a different combustion characteristics at partial and full load.

Ignition delay is one of the most important parameters of diesel engines which will directly affect the performance, emissions and combustion. A number of investigations have been conducted to study the ignition delay of diesel fuel. The results showed that the ignition delay depends on fuel parameters and pressure, temperature and excess air fuel ratio. The ignition delay is the time between the start of injection and the start of combustion. It is widely accepted that the ignition delay has a physical and a chemical delay. The physical delay is the time required for fuel atomization, vaporization and mixing with the air, whereas the chemical delay is the pre-combustion reaction of fuel with air [25]. In addition to these effects the recent trend of changing fuel quality and types has a great effect on ignition delay.

The ignition delay in a diesel engine is defined as the time interval between the start of fuel injection and the start of combustion [25]. Many defining methods were suggested to obtain the start of combustion for diesel engines [25, 29]. Assanis *et al.* [29] used the first derivative of the net HRR curve, that is, the second derivative of the cylinder pressure to define the combustion start of the turbocharged diesel engine. Combustion duration and intensity are estimated from the HRR, which is the most valuable source of information for the combustion mechanism

in Diesel engines [25]. The heat release rate diagram also provides valuable information for the initial stage of combustion where most of the NO is formed.

Conclusions

In this study, the engine was fed with a mixture of gases from the gasification process as well as diesel, so it was a dual fuel engine. The gas composition and gas generator parameters are shown in tab. 1. For both loads with different energetic ratios of generator gas and diesel were determined for both the ignition delay and the combustion duration. The ignition delay and combustion duration determined on the basis of the course of heat release rate [25].

In both cases, with the increase in the share of generator gas there was a slight increase in ignition delay, this can be seen in the figure showing the HRR. At partial load with an increase in the share of generator gas followed an increase of combustion duration. The opposite phenomenon was observed at full load. At partial load at $Q_d/Q_{tot} = 1.0$ the combustion duration was equal 14° CA, at $Q_d/Q_{tot} = 0.24$ combustion duration was equal to 18° CA. At full load at $Q_d/Q_{tot} = 1.0$ the combustion duration was equal 18° CA, at $Q_d/Q_{tot} = 0.63$ this parameter decreased to 10° CA.

The result showed that as a result of gasification of 1 ton of sewage sludge, at least 1450 m^3 of generator gas of a calorific value in the range of (2.15- 2.59) MJ/m³ can be obtained.

The tests performed on the supercharged piston engine dual fuel supplied with generator gas and liquid fuel batched to engine in varying energy share proportions have demonstrated that the dual fuel feeding of an engine is an effective method to reduce the engine susceptibility to variations in chemical composition of generator gas and its calorific value. The most favorable engine operation conditions occur in the situation of a partial electric generator load of approx. 40 kW corresponding to approx. 50% of the generator rated power attained with the liquid fuel alone. The process of combustion of fuel in the dual fuel engine powered by generator gas (with an energy share of 76%) and liquid fuel (with an energy share of 24%) is run with a excess air ratio of approx. 2.85, corresponding to 14% of oxygen content in the generator gas-air mixture and proceeds correctly. The engine attains then a high overall efficiency of 44.3%, so better than the efficiency of 37.8% attainable when being fed with 100% liquid fuel (tab. 4). A maximum combustion pressure of 7.8 MPa occurs during feeding the engine with the sole liquid fuel, and then it monotonically decreases down to 6.5 MPa as the liquid fuel energy share is reduced to 24% (fig. 6).

In the conditions of the full load corresponding to approx. 100% of the generator's rated electric power, which is 80 kW, the combustion process in the dual fuel fed engine has characteristics differing unfavorably from the respective characteristics occurring within partial loads corresponding to engine loading with a power of 40 kW. With the full engine load, the energy share of liquid fuel could not be lower. Even with the 71% share of diesel fuel when the combustion process was carried out with excess air ratio reduced to about 1.75 (corresponding to 17% oxygen in the gas mixture), obtained the correct engine work. Below this value (*e. g.* for $Q_d/Q_{tot} = 0.63$) the process of fuel combustion is an invalid. There will be too high-speed combustion pressure rise, which can cause mechanical damage to the engine (fig. 7).

At partial load with an increase in the share of generator gas followed an increase of combustion duration. The opposite phenomenon was observed at full load. At partial load at $Q_d/Q_{tot} = 1.0$ the combustion duration was equal 14° CA, at $Q_d/Q_{tot} = 0.24$ combustion duration was equal to 18° CA. At full load at $Q_d/Q_{tot} = 1.0$ the combustion duration was equal 18° CA, at $Q_d/Q_{tot} = 0.63$ this parameter decreased to 10° CA.

The dual fuel engine version developed within the project can be recommended to be used in practice in a dried sewage sludge gasification plant as a dual fuel engine driving the elec-

tric generator loaded with the active electric power limited to 40 kW (which accounts for approx. 50% of its rated power), because it is at this power that the optimal conditions of operation of an engine dual fuel fed with liquid fuel and generator gas are achieved. Under the conditions of a continuous round-the-clock operation regime, the gasification installation operating with the engine in question is able to gasify about 1.8 tons sewage sludge per day and generate 0.96 MWh electric energy during this time, while consuming approx. 50 kg of diesel oil for this purpose.

Nomenclature

CA	– crank angle, [deg.]	Q_{tot}	– sum of the energy supplied in diesel and generator gas, [J]
$dp/d\varphi$	– rate of pressure rise, [MPadeg. ⁻¹]	W_g	– calorific value of the gas, [MJkg ⁻¹]
HRR	– heat release rate, [Jdeg. ⁻¹]	<i>Greek symbols</i>	
IMEP	– indicated mean effective pressure, [MPa]	η_i	– indicated efficiency, [%]
L_t	– theoretical air demand value, [m ³ m ⁻³]	λ	– excess air factor
p	– pressure, [MPa]		
Q_d	– energy delivered to the engine in the liquid fuel (diesel), [J]		

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