Article citation info:
STELMASIAK Z. Limitations of enrichment of gaseous mixture in dual fuel engines. Eksploatacja i Niezawodnosc – Maintenance and Reliability 2014; 16 (4): 537–544.

Zdzisław STELMASIAK

LIMITATIONS OF ENRICHMENT OF GASEOUS MIXTURE IN DUAL FUEL ENGINES

OGRANICZENIA WZBOGACANIA MIESZANINY GAZOWEJ W SILNIKACH DWUPALIWOWYCH*

Exhaust gases from dual fuel engines are nearly smokeless, while maximal engine load is limited by allowable thermal load, not by smoke limit. This encourages users to enrich gas-air mixture, leading to increased output power of the engine, what partially compensates reduction of maximal output power connected with gaseous fuelling. Excessive enrichment of the charge, resulting from concentration of gas in the gas-air mixture and size of Diesel oil's dose, can lead to a phenomenon of knocking combustion and thermal overloading, and in result – to serious damage of the engine. In the paper is presented a calculation methodology of maximal enrichment of the gaseous mixture, above which a thermal overloading of the engine can occur. On example of a breakdown of three dual fuel engines of Caterpillar 3516A and SW 680 type, effects of thermal overloading, and a hazards connected with excessive enrichment of the gaseous mixture are shown. Conclusions of a general nature can serve as a guide to selection of control systems to dual fuel engines.

Keywords: *dual fuel engine, gaseous mixture, thermal load, ignition of gas, knocking combustion, combustion.*

Spaliny silników dwupaliwowych są prawie bezdymne, a maksymalne obciążenie silnika jest ograniczone dopuszczalnym obciążeniem cieplnym, a nie granicą dymienia. Skłania to użytkowników do wzbogacania mieszaniny gaz-powietrze prowadzącego do zwiększania mocy silnika, co częściowo kompensuje zmniejszenie mocy maksymalnej związane z gazowym zasilaniem. Nadmierne wzbogacanie ładunku, wynikające ze stężenia gazu w mieszaninie gaz-powietrze i wielkości dawki oleju napędowego, może prowadzić do zjawiska spalania stukowego i przeciążenia cieplnego, a w efekcie do poważnego uszkodzenia silnika. W pracy przedstawiono metodykę obliczania maksymalnego wzbogacania mieszaniny gazowej, powyżej którego mogą występować przeciążenia cieplne silnika. Na przykładzie trzech awarii silników dwupaliwowych Caterpillar 3516A i SW 680 pokazano skutki przeciążenia cieplnego silnika i niebezpieczeństwa związane z nadmiernym wzbogaceniem mieszaniny gazowej. Wnioski ogólne mogą być wskazówką do doboru sterowania silników dwupaliwowych.

Słowa kluczowe: *silnik dwupaliwowy, mieszanina gazowa, obciążenie cieplne, zapłon gazu, spalanie stukowe, spalanie.*

1. Introduction

In the next decades compressed natural gas (CNG) shall belong, except petroleum fuels, to important engine fuels. According to forecasts, in the year 2020 it should constitute about 8% of all engine fuels consumed in European Union countries [11]. Such market share shall increase in the next decades. It requires development of a microstructure connected with gaseous fuelling and operation of gaseous engines, but first of all, with construction of large number of gas filling stations.

Gaseous fuel supply can be accomplished both in spark ignition and compression ignition engines. Due to low cetane number and high self-ignition temperature of the gas, stable ignition of the gas-air mixture is accomplished by injection of small dose of liquid fuel, i.e. Diesel oil mainly. Due to this reason, such system was assumed to be called as a dual fuel system or jet ignition (multi-spot, spatial ignition of the gas and Diesel oil) [2, 4, 5, 7, 11].

In a spark ignition engine, one should take into account reduction of maximal engine output power in range of 8÷10%, what results from reduced filling ratio of the cylinder and smaller calorific value of the gas-air mixture [10, 11]. In an engine adopted to operation on the gas only, decrease of the output power can be partially reduced by compression ratio increased with 1,5÷2,5 units, what results from high knock resistance of gaseous fuels. In an engine, which should be alternately fuelled with the gasoline or the gas, so called *flexi-fuel* technology is impossible to be used, and nearly 10% reduction of the output power can be unwillingly received by users.

The dual fuel system enables us to keep the most of positive features of compression ignition engine, and because of it, recently we can see its renaissance in many applications. It results from electronic control systems of injection of liquid fuel and the gas, allowing maintaining optimal composition of the charge in altering conditions of engine operation. Gradual development of fuel supply systems to dual fuel compression ignition engine results from many benefits offered by such type of fuelling. To the most important should be included:

- possibility of maintaining of the output power at unchanged level [3, 4, 5, 10],
- high engine efficiency [1, 2, 4–6],
- possibility of alternating operation in dual fuel system and on Diesel oil only,
- possibility of combustion of the gas-air mixtures in wide range of change of the excess air ratio [2, 4, 6, 10],
- lower emission of nitrogen oxides NO_x and particticulate matter PM,
- lower costs of engine operation.

The last from above mentioned factors is the most decisive when decision about gaseous fuelling is to be taken. It results from a big difference in price of gaseous and liquid fuel. Generation of unit work of the engine in case of gaseous fuelling is nearly twice cheaper than in case of fuelling with Diesel oil.

(*) Tekst artykułu w polskiej wersji językowej dostępny w elektronicznym wydaniu kwartalnika na stronie ***www.ein.org.pl***

Exhaust gases from compression ignition engines operated on the gas are nearly free of smoke, while maximal portion of energy supplied to the engine is limited by allowable thermal load, and is not limited by the smoke limit. This leads many users to reduce average excess air ratio and to increase maximal engine output power, what compensates its reduction due to gaseous fuelling.

The work presented in this paper points at limitations present when the maximal engine output power is increased, and a phenomena connected with it.

2. Limitations of enrichment of the charge in dual fuel engine

Investigations of dual fuel engines operated on the CNG, performed by the author showed that both in case of piston pumps and mixer-type supply with the gas, as well as in case of high pressure common rail injection systems of the initial dose, and injection of the gas to manifold, it is possible to develop the output power similar to the one developed by the engine operated on the Diesel oil only, Fig. 1. Simultaneously, development of similar output power is possible for different size of the initial dose, what can be implemented in traction engines. In the Fig. 1a, together with change of the initial dose one adjusted quantity of the gas to generate engine output power identical like in case of operation on Diesel oil only. Due to this reason, the lines of the output power for various dosages are overlapping. In case of common rail system (Fig. 1b), for the doses 20,3 mm³/cycle and 31,2 mm³/cycle in area of a higher rotational speeds occurred reduction of the output power reaching 6÷8%. This was caused by emerging phenomenon of knocking combustion. Only further increase of the dose to 39,3 mm³/cycle (and the same, leaning of the gaseous mixture) allowed development of the output power comparable to traditional fuelling in complete range of change of rotational speed. It should be also noted that in a system of injection of the dose through *common rail* system, further increase of the dose favors increasing of the output power and simultaneously can lead to improvement of

a) b)

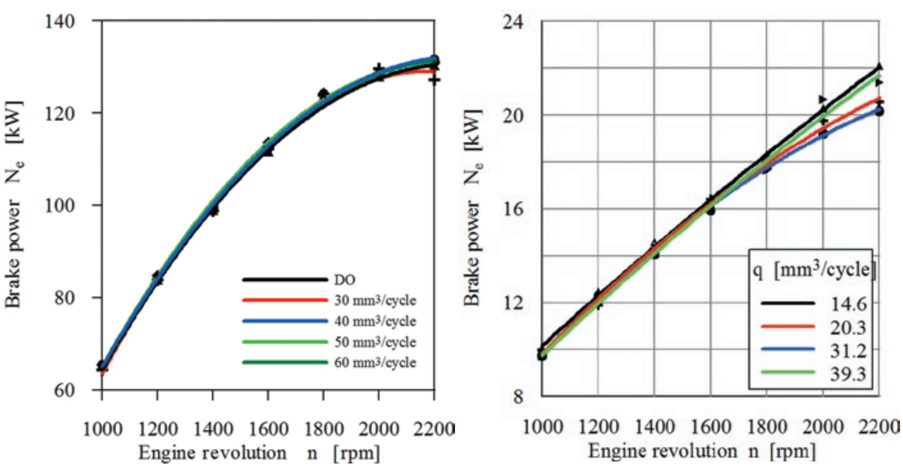


Fig. 1. Comparison of the output power of dual fuel engine powered traditionally, and operated in dual fuel system with CNG: a) SW 680 engine with in-line P56-01 pump and mixer-type feeding with the gas, b) SB3.1 engine with injection of the initial dose through common rail system and injection of the gas to suction manifold

thermal efficiency of the engine [14].

Investigations performed earlier showed that in dual fuel engines, even slight addition of the gas can substantially reduce smokiness of the exhaust gases [1, 3, 4, 10, 14]. It may encourage using of the charge with reduced average excess air ratio, what partially compensates loss of the output power resulting from gaseous fuelling. Reduction of the average excess air ratio λ can be attained by increase of the gas flowing to the engine, and the same by decrease of the excess air ratio λ_o of the gaseous mixture. However, enrichment of the gaseous mixture can lead to thermal overload of the engine, and in consequence to its damage [12, 13].

To avoid damage of the engine, it is safer to adopt such principle, that at nominal load streams of energy at traditional and dual fuel supply should be the same. It will enable to evaluate such minimal number $\lambda_{o\,min}$ of the gaseous mixture, which shouldn't be exceeded. Hereinafter in the paper shall be presented a methodology of evaluation of this number.

Stream of energy supplied to the engine, when the engine is run on Diesel oil, results from quantity of intake air and applied excess air ratio λ. This can be calculated from the following equations:

– maximal quantity of Diesel oil possible to be combusted at the excess air ratio λ:

$$\dot{m}_{on} = \frac{\dot{V}_p}{\lambda \cdot L_{on}} \tag{1}$$

where: $\dot{m}_{on}$ – stream of liquid fuel supplied to the engine [kg/s],
$\dot{V}_p$ – stream of the air sucked by the engine [nm³/s],
L_{on} – theoretical air demand for liquid fuel [nm³/kg],

– stream of energy supplied together with Diesel oil ($\dot{Q}_{on}$):

$$\dot{Q}_{on} = \frac{\dot{V}_p \cdot H_{on}}{\lambda \cdot L_{on}} \tag{2}$$

where: H_{on} – calorific value of liquid fuel [MJ/kg].

In case of dual fuel supply with initial dose of Diesel oil with fraction of energy U_{on}, stream of energy supplied together with gaseous fuel ($\dot{Q}_{g\max}$) is expressed by the formula:

$$\dot{Q}_{g\max} = (1 - U_{on}) \cdot \dot{Q}_{on} \tag{3}$$

or

$$\dot{V}_g \cdot H_g = (1 - U_{on}) \cdot \frac{\dot{V}_p}{\lambda \cdot L_{on}} \cdot H_{on} \tag{4}$$

where: $\dot{V}_g$ – stream of the gas flowing to the engine [nm³/s] ,
H_g – calorific value of gaseous fuel [MJ/nm³].

Maximal stream of the gas sucked to the engine ($\dot{V}_{g\max}$) is determined by the formula:

$$\dot{V}_{g\max} = (1 - U_{on}) \cdot \frac{\dot{V}_p}{\lambda \cdot L_{on}} \cdot \frac{H_{on}}{H_g} \tag{6}$$

Minimal excess air ratio for the gas-air mixture, assuming that volumetric efficiency of the engine at traditional and dual fuel supply does not change:

$$\lambda_{o\min} = \frac{\dot{V}_p - \dot{V}_{g\max}}{\dot{V}_{g\max} \cdot L_g} = \frac{\dot{V}_p}{\dot{V}_{g\max} \cdot L_g} - \frac{1}{L_g} \tag{7}$$

Substituting into the formula (6) for $\dot{V}_{g\max}$ the formula (5), we finally receive:

$$\lambda_{o\min} = \frac{\lambda}{1-U_{on}} \cdot \frac{H_g}{H_{on}} \cdot \frac{L_{on}}{L_g} - \frac{1}{L_g} \qquad (6)$$

From the formula (7) follows, that maximal enrichment of the gas-air mixture (the lowest number λ_o) is directly proportional to the average excess air ratio λ, equal at traditional and dual fuel supply, and inversely proportional to energetic portion of the gas $U_g=1-U_{on}$. It depends also on calorific values and theoretical demand of air for the both fuels.

In course of tuning of dual fuel engine, one should pay attention on the following cases:

- λo < λo min – stream of energy supplied to the engine is bigger than the stream supplied in case of fuel supply with Diesel oil only,
- λo > λo min – stream of energy at dual fuel supply is smaller than in case of run on Diesel oil only.

Feed of the engine with gaseous mixture having the excess air number $\lambda_o < \lambda_{o\,min}$ is connected with thermal overload of the engine, comparing to traditional fuelling. This should be applied, therefore, very carefully, because in a compression ignition engine are present a small clearances in piston-cylinder bore system, and growth of temperature of these components can lead to seizure of the engine.

It should be clearly underlined, that in case of fuelling with the gas one should expect a drop of maximal output power, connected with volume taken by the gas in the charge, and smaller calorific value of gaseous mixture. Therefore, both spark ignition engine and dual fuel engine should have, as a standard, diminished nominal output power, depending on these values. Although efficient combustion process in dual fuel engine points at a possibility of reduction of average number λ, such operation should be used very carefully and should be based on tests of the engine.

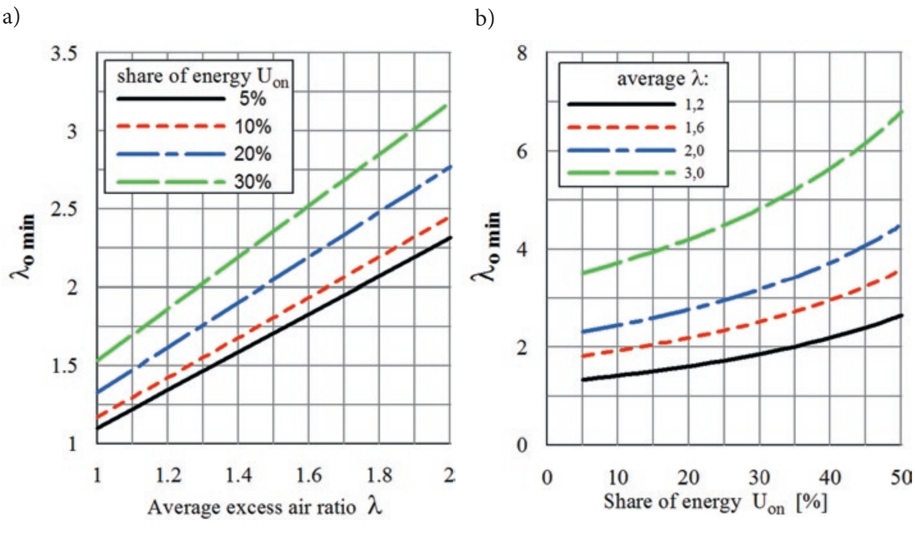


Fig. 2. Changes of maximal excess air number of gaseous mixture $\lambda_{o\,min}$ for the Diesel oil and natural gas: a) in function of the excess air ratio λ for constant energetic fractions of Diesel oil U_{on}, b) in function of change of the fraction U_{on} for constant values of the excess air ratio λ

From analysis of the runs presented in the Fig. 2 is seen that fuel supply in naturally aspirated engine at nominal load with charge having the average excess air ratio λ=1.3 enables usage of the initial dose with energetic fraction up to 30% of the nominal dose. It allows to keep composition of the gaseous mixture in range of combustibility in ambient conditions $\lambda_{o\,min}$<2.0. In case of supercharged engines, where bigger excess air ratio are used λ=2.0, the initial dose should be reduced to about 20% of energetic portion. Then, minimal excess air ratio $\lambda_{o\,min}$ of the gaseous mixture amounts to $\lambda_{o\,min}$<2.8. Such composition in conditions of increased temperatures, present in supercharged engines, should assure correct combustion of the gaseous mixture.

Dependence of the number $\lambda_{o\,min}$ on portion of energy of the initial dose is hyperbolic, Fig. 2b. It means, that, initially change of size of the dose slightly affects value of the $\lambda_{o\,min}$ regardless of the number λ. However, considerable growth of the dose leads to excessive leaning of the gaseous mixture, what could result in presence of negative phenomena accompanying combustion of lean mixtures. When compression ignition engine is adapted to dual fuel supply, selection of size of the initial dose should take into account analyses of heat release from the gaseous mixture, as a carrier of main energy supplied to the engine.

Due to easiness of mixing of the gas with the air and complete homogeneity of gaseous mixture before self-ignition of the liquid fuel, even in case of direct injection of the gas into cylinder, in a dual fuel engine at maximal load it is possible to slightly reduce the average excess air ratio λ for total charge, in relation to the one used at fuelling with Diesel oil only. This enables to maintain the nominal output power of the engine fuelled traditionally, and in some cases to increase the output power. However, this should be applied very carefully with use of electronic control system and engine control in connection with controlling of knocking and temperature of exhaust gases. In stationary engines with number of cylinder bigger than ten, level of engine vibrations should be also controlled, which increases together with appearance of the knocking at enriched gaseous mixture or non-uniform combustion in the cylinders, when the mixture is excessively leaned. Electronic system controlling quality of the charge, used in dual fuel Navistar DT 466 engine with swept capacity of 7,6 dm^3 enabled development of the same engine output power in complete range of engine rotations, or slightly higher like in case of traditional fuelling [3].

3. Analysis of damages in dual fuel engines

Thermal overloading belongs to frequent reasons of damage of dual fuel engines operated both in spark ignition system and dual fuel system. To the most frequent reasons of the damage belong:

- excessive thermal load in piston-cylinder bore system [8, 9, 13],
- excessive thermal load of exhaust valves and cylinder head,
- engine operation at knocking combustion limit or with distinct knocking [2, 8, 9, 10].

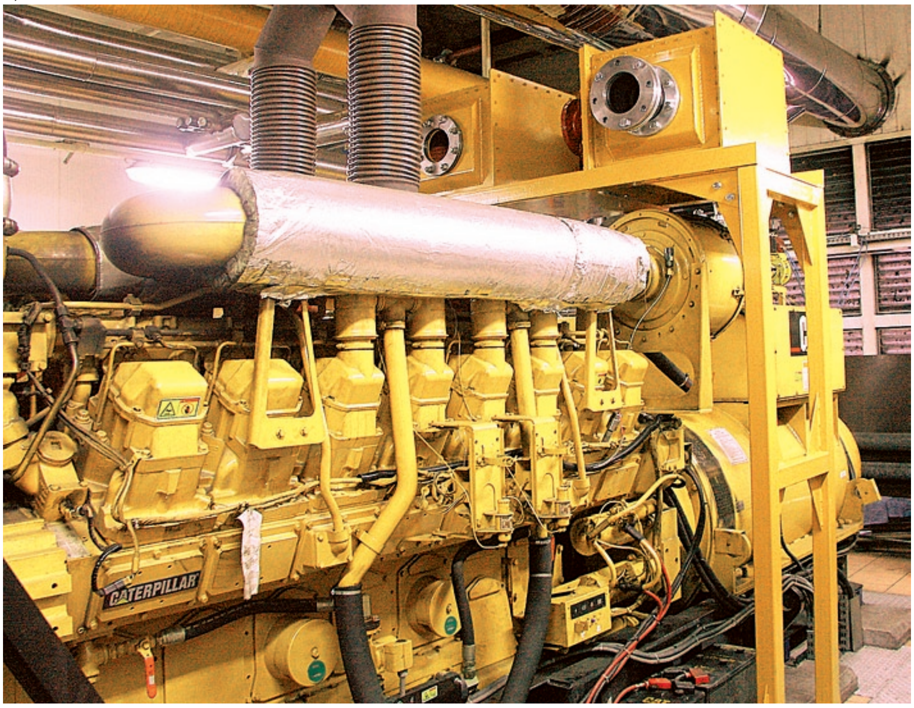
In the last reason are consciously distinguished operations at knocking combustion limit and with distinct knocking, although the both phenomena are classified as a knocking combustion. The most often, operation at knocking combustion limit occurs without any distinct symptoms of the knocking, manifested by pressure waving in high pressure range. This phenomenon is not always diagnosed by sensors as a knocking, and virtually unnoticeable by service personnel, especially in traction engines during motion. While, operation in area of the knocking combustion gives a distinct vibro-acoustic symptoms, which can be detected by knocking sensors, which should be installed in gaseous dual fuel engines.

Below are presented some cases of engine damage arisen in result of thermal overloading and irregular combustion of gaseous mixture.

In the Fig. 3 is presented a 16 cylinder Caterpillar 3516A engine with output power of 1,5 MW, with diagnosed breakdown of the first cylinder. The engine was fuelled in dual fuel system of mine gas with mixer-type supply of the gas. The mixer was installed in front of the engine, upstream the system of turbochargers. Gaseous mixture was supplied through a gas distribution system to the turbocompressors supplying individually the both rows of the cylinders. The breakdown occurred after about 6.500 h of engine operation. The engine was stopped by service personnel due to increased knocking in crankshaft assembly, without any symptoms of seizure. Diagnostic system controlling increased vibrations of the engine was not activated.

Seizure of the piston in its carrying part, in the first cylinder from the RH row of the cylinders, was identified as the reason of increased knocking of the engine (Fig. 3b, 4a, 4b). After disassembly of the cylinder head and the piston, clear traces of seizure were seen on the carrying surface (Fig. 4a-b). Piston crown, piston ring area and the pistons themselves didn't show any distinct traces of damage. All piston rings were loose and freely movable in the grooves. Also condition of the cylinder head and the valves didn't show any traces of irregular combustion (Fig. 4d).

a)



b)

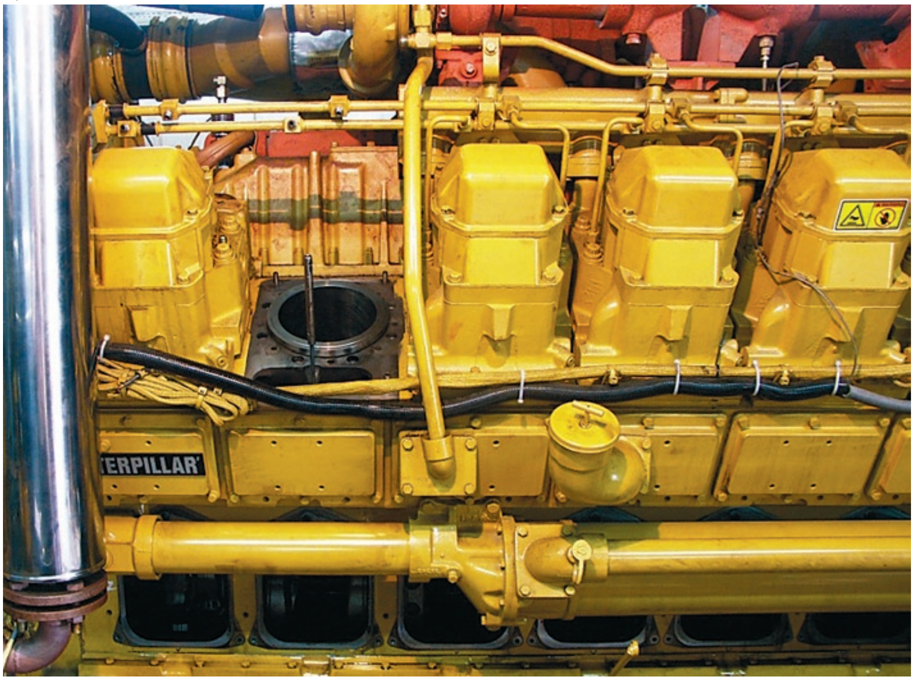


Fig. 3. Dual fuel Caterpillar 3516A engine in state of breakdown of the first cylinder (photo of the author):a) the engine during operation, b) the engine after disassembly of cylinder head from the first cylinder

Presented facts allow assuming that abnormal composition of the gaseous mixture sucked to the first cylinder and connected with it thermal overloading were the reasons of engine breakdown. It could result from enrichment of the gaseous mixture aimed at a higher output power, or sudden growth of methane contents in the mine gas, not discovered by the systems monitoring operation of the engine. As this phenomenon didn't occur in the cylinder the closest to the mixer, non-uniform composition of the gas-air mixture could be also a reason of the breakdown. With average contents of methane in the mine gas of 45÷50%$_V$ volume of the gas supplied to the mixer is relatively high. In short pipes with small diameters, not complete blending of the gas with the air can not occur during feed to the mixer, and there exists a probability that the first cylinders were supplied with more rich mixture than other cylinders. In case of the discussed engine, size of the initial dose was fixed by the Woodward regulator, which selects size of the dose to keep the output power and engine rotations at desired level. Size of the dose of Diesel oil was the same for all cylinders.

The next damage of the discussed engine, arisen due to unsufficient cooling of piston crown, was the next confirmation of the breakdown resulted from thermal overloading [12, 13]. Fatigue failure of the piston ring, retaining the oil sprinkled by nozzeles to channel in the piston crown when the piston is near the BDC, was the reason of this damage (Fig. 5). In correct condition of the system, the oil supplied to the channel fills it completely, and the oil is retained for a longer time, effectively carrying away the heat from the piston crown.

After damage of the piston ring, cross-section of the holes carrying off the oil increased significantly, what resulted in rapid loss of the oil and not complete contact of the oil with piston crown. It effected in worsened cooling of the piston crown, gradual heating of complete piston, and in result, seizure of it carrying part. Also in this case, both the piston crown, piston rings and cylinder head with valves didn't show any symptoms of irregular combustion.

It is worth to underline, that the engine discussed here was not equipped with individual system to monitoring of the cylinders, what should be considered as a standard for such engine size. In the Fig. 6 is presented a simple control method of thermal load of the cylinder, implemented in the engines produced by

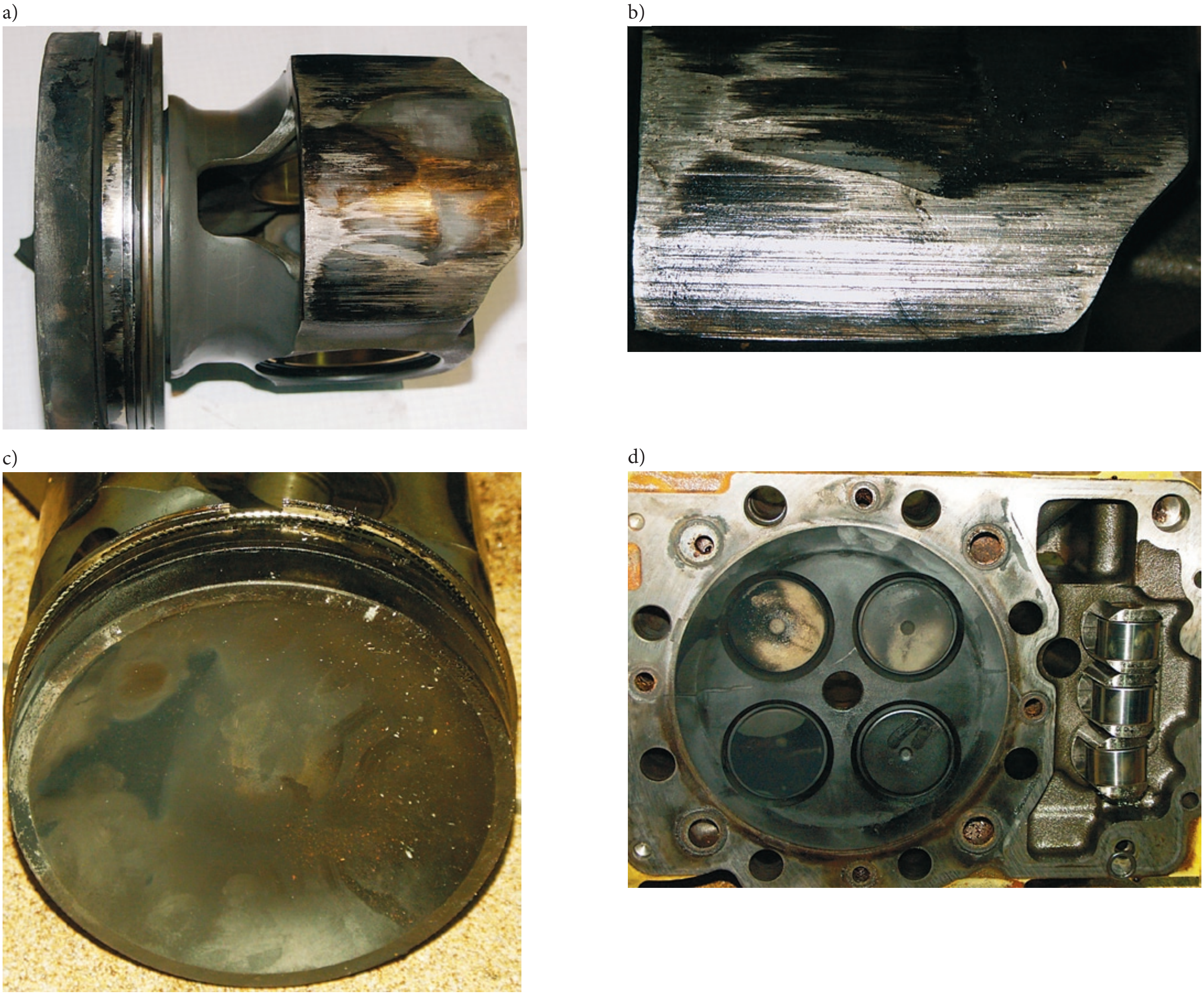


Fig. 4. View of components of the first cylinder of the Caterpillar 3516A engine with traces of seizure (photo of the author): a) piston of the first cylinder, b) view of carrying sector of the piston with traces of seizure, c) view of piston crown of the first cylinder, d) cylinder head of the first cylinder

Deutz Co. In a special channel in cylinder head of each cylinder is assembled a thermocouple. Tip of the thermocouple is in contact with the working medium during complete cycle through an additional hole, and through housing with temperature of the cylinder head. In result, the thermocouple measures temperature which is lower than temperature of the working medium during combustion, and is higher than temperature of the cylinder head. Basing on experience of the manufacturer, in memory of diagnostic system is recorded range of temperature corresponding to correct engine operation. Growth of temperature detected by the thermocouple is pointing at irregular operation of a cylinder.

Cost of the system is low, while durability of thermocouple is high. Continuous recording of temperature in all cylinders allows signaling of engine failure well in advance.

Dual fuel engine operation should occur in areas far from knocking combustion limit and limit of misfiring, what is schematically presented in the Fig. 7.

The mixture in range of stoichiometric composition λ_o=1.0-1.2 is the most prone to the knocking combustion. Tendency to knocking combustion decreases together with leaning of the gas-air mixture, however, simultaneously decreases calorific value of the mixture. Obtainment of a higher average effective pressure requires, therefore, increased degree of supercharging or size of initial dose of Diesel oil. From the Fig. 7 is seen that in naturally aspirated engines it is possible to obtain average effective pressure of about 1.0 MPa with knocking-free engine operation, what is not a small value for a naturally aspirated gaseous engine.

Tendency to knocking combustion belongs to individual features of dual fuel engine and depends on the following factors: type of combustion chamber, size of initial dose, composition of combustible mixture, cylinder bore diameter, rotational speed, type of cooling. Therefore, in selection of control system to dual fuel engine, tendency to occurrence of such phenomenon should be always taken into considerations.

Knocking combustion is very hazardous for each combustion engine, especially compression ignition engine, where small clearances in piston-cylinder system are present [8, 9, 11, 12, 13].

In the Fig. 8 are presented damaged pistons of the 4th and 5th cylinder from the SW 680 engine fuelled in dual fuel system with CNG gas. The damage took place during the testing, when course of the

Fig. 5. Piston of dual fuel Caterpillar 3516A engine seizured due to failure of oil cooling system of piston crown (photo of the author)

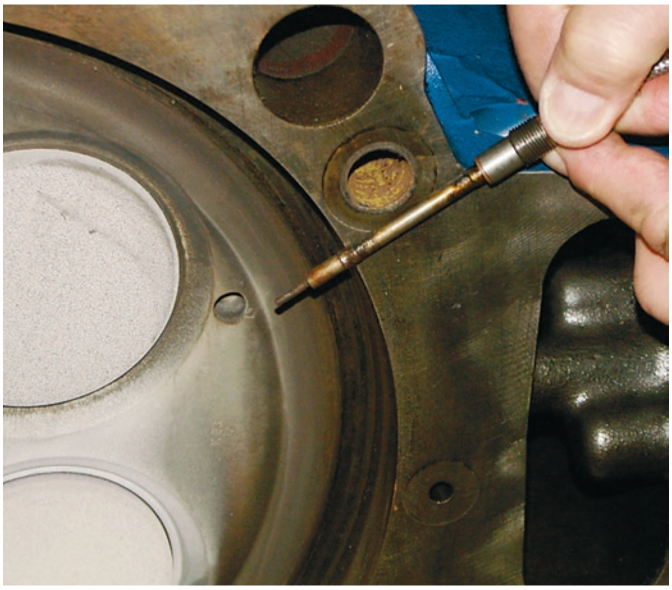


Fig. 6. Monitoring of thermal load of a cylinder of gaseous engine from Deutz Co. (photo of the author)

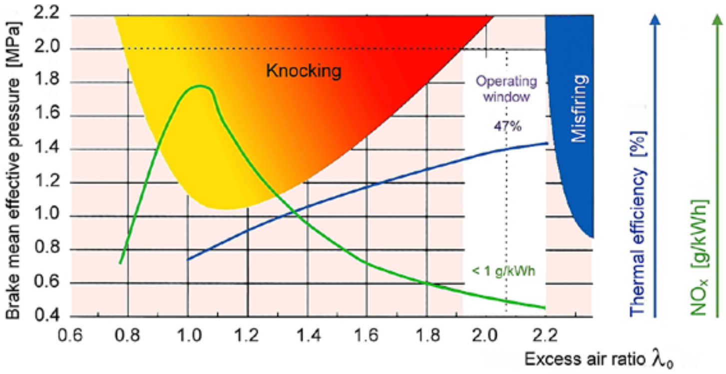


Fig. 7. Operational ranges of dual fuel Wärtsila 50 DF engine [15]

pressure in the 2nd cylinder was traced by the author on computer's screen. Pressure in the engine was increased by increase of quantity of the gas which powered the engine. After several minutes of engine operation at a load bigger with about 5% from the nominal load, the engine was stopped, while repeated start-up in cold and hot conditions was not possible. It should be underlined that on observed indication diagram, the author didn't notice any distinct symptoms of the knocking combustion.

Monitoring of flow rate of the gas enabled calculation of the excess air ratio λ_o for the gas-air mixture. It was found, that the excess air ratio was much below the $\lambda_{o\ min}$ calculated for the investigated engine. Excessive enrichment of gaseous mixture did not result in growth of engine torque, and operators still increased supply of the gas to the engine, leading to engine operation at knocking combustion limit.

After disassembly of the engine it was found that the piston of the 4th cylinder underwent seizure. Crown and groove's part of the piston exhibited numerous partial melting, Fig . 8a-b. Sealing rings no. 1 and no. 2 underwent baking on about 70% of the circumference. Also the piston from the 5th cylinder underwent seizure and partial melting, Fig 8c-d. However, extent of the damage in this cylinder was smaller and took place from side of the 4th cylinder. It suggest that the damage could arise in result of overheating due to excessively thermally overloaded 4th cylinder, because knocking combustion should result in the damage on full circumference of the piston.

As fuel supply system assured fuelling of each cylinder with homogenous mixture having the same composition, it can be assumed that the damage occurred due to phenomenon of knocking combustion, which appeared first in the 4th cylinder due to excessive enrichment of the gaseous mixture. Short time of engine operation caused that such phenomenon did not occur in the other cylinders. Presence of knocking combustion in the 4th cylinder suggests only, that the engine was operated at knocking combustion limit, while cooling conditions of the 4th cylinder were worse than in the others.

It is also worth to underline, that after occurrence of the damage described here, in course of further investigations one controlled the excess air ratio λ_o when engine load was increased. It allowed increasing maximal output power with about 10% comparing to nominal output power of the engine operated on Diesel oil only. In spite of long lasting engine operation under increased load, none anomalies in combustion process or changes in components of crank system, in valves and in cylinder heads, were confirmed.

It means, that in dual fuel engines it is necessary to restrict enrichment of the gaseous mixture; while range of these limitations belongs to individual features of the engine. In the cases where the enrichment cannot be preceded by investigations, it is better to limit maximal output power of the dual fuel engine to 92÷95% of the nominal power, while in the cases where maximal output power is required; engine load should be increased by addition of Diesel oil instead of the gas.

4. Summary

On the base of performed analyses and investigations it is possible to formulate the following observations concerning controlling of engine load in gaseous engines with spark ignition and dual fuel engines.

- Gaseous fuel supply results in decreased maximal output power of the engines in scope of 8÷10% due to worsening of filling and lower calorific value of the gas-air mixture.
- In a spark ignition engines adapted to gas fuel only, decrease of the output power can be reduced by increase of the compression ratio with 1.5÷2.5 units, what is allowed due to high knocking resistance of majority of gases.

a) b) c) d)

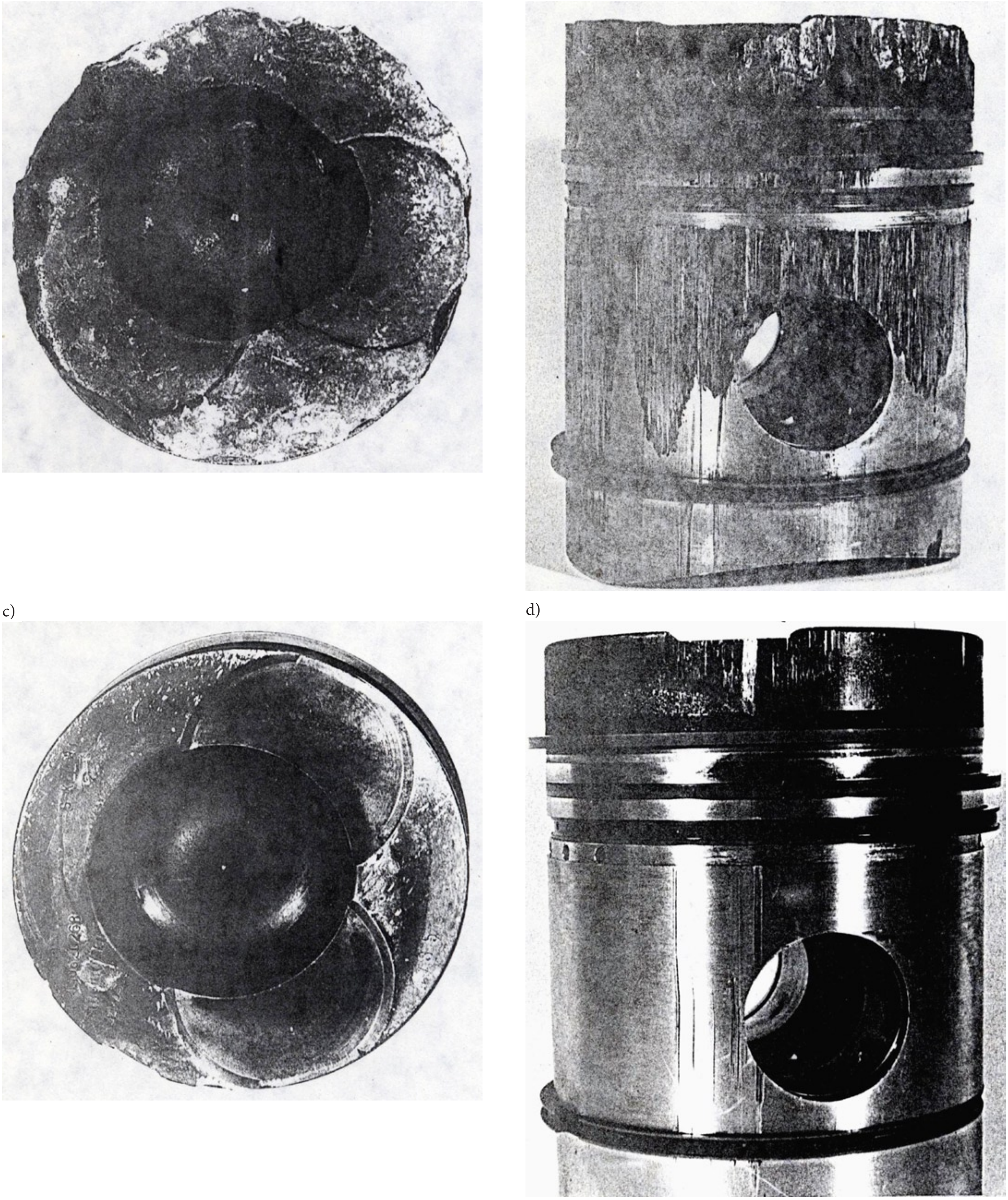


Fig. 8. Pistons of the SW 680 engine run in dual fuel system after several minutes of operation with knocking combustion: a), b) – piston from the 4th cylinder, c), d) piston from the 5th cylinder, (photos of the author)

- In dual fuel engines, lower calorific value can be compensated by decrease of the average excess air ratio. It results from significantly lower smokiness of exhaust gases. In case of some engines, it allows development of maximal output power, the same like in case of operation on Diesel oil only.

- Lower smokiness of exhaust gases in dual fuel engines may lead the users to increase the maximal output power by significant enrichment of the gas-air mixture. Such actions may lead, however, to uncontrolled thermal overloading of the engine, or engine operation at knocking combustion limit. The both phenomena in smaller engines are difficult to be discovered by service personnel and engine control systems. Such phenomena may lead to serious failure of the engine.
- In bigger stationary engines with mixer-type fuelling, thermal overloading of some cylinders can occur during enrichment of the gas-air mixture, what results from asymmetry of fuel supply. Therefore, in such engines one should control temperature of each cylinder, while threshold level of the temperature should be set individually for a given type of the engine, and memorized in ECU of the engine.
- Dual fuel engine operation at the knocking combustion limit is especially hazardous. In traction engines with small number of cylinders, such operation can not be detected by sensors of knock, and a failure in piston-cylinder system can occur in a short time.
- Enrichment of the gas-air mixture leading to reduction of the average excess air ratio below value used at fuelling with the Diesel oil only should be preceded by investigation of thermal loads of engine components, and should be used very carefully.

References

1. Alla G H A, Soliman H A, Badr O A, Rabbo M F A. Effect of pilot fuel quantity on the performance of dual fuel engine. Energy Conversion & Management 2000; 41: 559-572.
2. Badr O, Karim G A. An examination of the flame spread limits in a dual fuel engine. Applied Thermal Engineering 1999; 19: 1071÷1080.
3. Barroso P, Ribas x, Domingues j, De Sella E, Garcia J M. Study of dual-fuel (diesel+natural gas) particle mater and CO_2 emissions of a heavy-duty diesel engine during transient operation. Combustion Engines; 2/2013 (153) ISSN 0138-0346: 28-39.
4. Clark N N, Atkinson Chr M, Atkinson R J, McDaniel T, Park T. Optimized Emission Reduction Strategies for Dual Fuel Compression Ignition Engines Running on natural Gas and Diesel. http://www.cemr.wvu.edu 2002: 1-6.
5. Friedeman Z. Gasmotoren. Würzburg: Vogel Buchverlag, 2001.
6. Gebert K, Beck J, Barkhimer R L, Wong H Ch. Strategies to Improve Combustion and Emission Characteristics of Dual-Fuel Pilot Ignited Natural Gas Engines. SAE Paper 1997; 971712: 79-87.
7. Jarg Chr, Sutkowski M. The Wärtsila 32GD engine for heavy gases. Combustion Engines 2009; 2/2009(137): 3-11.
8. Kowalewicz A. Adaptacja silnika wysokoprężnego do zasilania gazem naturalnym. Czasopismo Techniczne Wydawnictwo Politechniki Krakowskiej 2008; 7-M/2008: 67-78.
9. Kozioł S. Ocena trwałości układu tłokowo-korbowego dwupaliwowego silnika o zapłonie samoczynnym zasilanego głównie mieszaniną propan-butan. Radom: Rozprawa doktorska Politechnika Radomska, 2004.
10. Luft S, Kozioł St. Ocena trwałości elementów układu tłokowo-korbowego silnika o ZS zasilanego głównie LPG. Journal of KONES Internal Combustion Engines 2002, 9(3-4):153-160.
11. Sawłowicz W, Stelmasiak Z, Wojciechowska A. Wstępne badania funkcjonalne mieszalników gazu z uwzględnieniem prawidłowej współpracy organów sterujących silnika. Bielsko-Biała: Sprawozdanie z pracy naukowo-badawczej nr ITS/089/066/S, 1986.
12. Stelmasiak Z. Studium procesu spalania gazu w dwupaliwowym silniku o zapłonie samoczynnym zasilanym gazem ziemnym i olejem napędowym. Bielsko-Biała: Wydawnictwo ATH w Bielsku-Białej Rozprawy naukowe Nr 5, 2003.
13. Stelmasiak Z, Wencelis J, Larisch J. Analiza przyczyn pękania pierścieni osadczych w układzie olejowego chłodzenia denka tłoka dwupaliwowego silnika zasilanego gazem kopalnianym. Silniki Spalinowe 2007; SC3: 361-368.
14. Stelmasiak Z, Larisch J, Wencelis J. Reasons of a thrust ring damage in oil-cooled piston head in a dual fuel CI engine fuelled with gas and Diesel oil. Journal of POLISH CIMAC 2008, 3(2): 163-172.
15. Stelmasiak Z, Larisch J, Gilowski T, Fic A, Knefel T. Sterowanie procesem spalania w silniku dwupaliwowym z wtryskiem sprężonego gazu ziemnego (CNG) i wysokociśnieniowym wtryskiem dawki pilotującej Common Rail w aspekcie: optymalnej sprawności ogólnej silnika; minimalnej emisji HC I NOx. Bielsko-Biała, Sprawozdanie z pracy naukowo-badawczej nr 5 T12D 011 24, 2006.
16. Wärtsila 50DF Technologie Reviev. Wärtsila: Materiały reklamowe, 2010.

Zdzisław STELMASIAK
Internal Combustion Engines and Automobiles Branch
Technical University of Bielsko-Biala
ul. Willowa 2, 43-309 Bielsko-Biala, Poland
E-mail: zstelmasiak@ath.bielsko.pl