Marcin K. Wojs¹, Piotr Orliński², Stanisław W. Kruczyński³

COMBUSTION PROCESS IN DUAL FUEL ENGINE POWERED BY METHANE AND DOSE OF DIESEL FUEL

1. Introduction

Compression ignition engines are the most common source of power, which use oil fuels and can be applied in a wide variety of uses that support our lives because of its originally high efficiency, easy operability, and high levels of safety [1]. Leading into use the compression ignition engines powered by gas fuels is problematic on account of their design assumptions. Majority of suggested solutions is providing changes in the structure of the engine through the application of additional ignition systems. Another solution is implementing the preliminary dose of diesel fuel to gas fuel ignition [2, 3].

At present the market is dominated by fuels of the oil origin what results in heavy costs of using them. Using gas for powering engines of machines will allow from one side for limiting consuming fossil fuels still becoming more expensive [4]. However, on the other side it can contribute to increase the energy independence, by using another type of fuel such as methane.

2. Model and conditions of simulation in AVL Boost program

The simulation model of the engine CDC 6T-590 was prepared using AVL software. It was in particular AVL BOOST ver. 2013 program. Conducted simulations of working cycle of dual fuel engine powered by diesel fuel and gas were aimed at the analysis of selected phenomena connected with possible course of the combustion process of dual fuel blend delivered to the cylinder. As a gas fuel methane CH₄ was used together with the preliminary dose of diesel fuel.

In Fig. 1 the schema of CDC 6T-590 engine simulation model is shown.

Fig. 1. The schema of CDC 6T-590 engine simulation model prepared in AVL BOOST software

---

¹ Mgr inż Marcin K. Wojs, Institute of Vehicles, Warsaw University of Technology
² Dr hab. inż. Piotr Orliński, Institute of Vehicles, Warsaw University of Technology
³ Prof. dr hab. inż. Stanisław W. Kruczyński, Institute of Vehicles, Warsaw University of Technology
In Fig. 2 it is presented the window of the program allowing to put the engine CDC 6T-590 basic geometrical data. However, Fig. 3 shows the window of the program on which the fuels applied for the simulation are defined. Presented case assumes 20% of diesel fuel and 80% of methane.

![Fig. 2. Basic dimensions of CDC 6T-590 engine](image)

![Fig. 3. Fuels selection and their ratios](image)
Fig. 4. Thermodynamics properties of diesel fuel

Nevertheless, in Fig. 6 there is presented the way how to put the data for simulation of combustion process through Vibe function with division into two combustion phases.
Simulations were performed for rotational velocity referring to the maximum torque for the following engine working conditions:

- fuel: 20% diesel fuel, 80% methane
- tense of the compressor: 1.4
- engine rotational velocity $n=1600$ obr/min

Division of the quantity of the ensuing heat during combustion process:

- combustion phase I - 5% of the heat release in the time of 15 deg of crankshaft rotation,
- combustion phase II - 95% of the heat release in the time of 55-145 deg of crankshaft rotation (with calculation step 20 deg);
- combustion phase I - 20% of the heat release in the time of 15 deg of crankshaft rotation.
- combustion phase II - 80% of the heat release in the time of 95 deg of crankshaft rotation (with calculation step 20 deg)

The beginning of combustion: 6 deg before TDC.

3. Results of simulation for dual fuel powered CDC 6T-590 engine

Through simulations a series of results was obtained, from which were presented only selected calculations, referring to combustion process. Effects of simulations of ensuing heat, pressure in combustion chamber, temperature in combustion chamber and indicated mean pressure are presented suitably in Fig. 7 – 9.
Fig. 7. The course of pressure inside combustion chamber as a function of crankshaft rotation angle (a fragment) for differentiated number of ensuing heat during first and second combustion phase:

- **p1** – combustion phase I: 5% of ensuing heat in the time of 15 deg of crankshaft rotation, combustion phase II: 95% of ensuing heat in the time of 55 deg of crankshaft rotation;
- **p2** – combustion phase I: 5% of ensuing heat in the time of 15 deg of crankshaft rotation, combustion phase II: 95% of ensuing heat in the time of 75 deg of crankshaft rotation;
- **p3** – combustion phase I: 5% of ensuing heat in the time of 15 deg of crankshaft rotation., combustion phase II: 95% of ensuing heat in the time of 95 deg of crankshaft rotation;
- **p4** – combustion phase I: 5% of ensuing heat in the time of 15 deg of crankshaft rotation., combustion phase II: 95% of ensuing heat in the time of 105 deg of crankshaft rotation;
- **p5** – combustion phase I: 5% of ensuing heat in the time of 15 deg of crankshaft rotation., combustion phase II: 95% of ensuing heat in the time of 125 deg of crankshaft rotation;
- **p6** – combustion phase I: 5% of ensuing heat in the time of 15 deg of crankshaft rotation, combustion phase II: 95% of ensuing heat in the time of 145 deg of crankshaft rotation.
Fig. 8. The course of temperature inside combustion chamber as a function of crankshaft rotation angle (a fragment) for differentiated number of ensuing heat during first and second combustion phase:

T1 – combustion phase I: 5% of ensuing heat in the time of 15 deg of crankshaft rotation, combustion phase II: 95% of ensuing heat in the time of 55 deg of crankshaft rotation;

T2 – combustion phase I: 5% of ensuing heat in the time of 15 deg of crankshaft rotation, combustion phase II: 95% of ensuing heat in the time of 75 deg of crankshaft rotation;

T3 – combustion phase I: 5% of ensuing heat in the time of 15 deg of crankshaft rotation, combustion phase II: 95% of ensuing heat in the time of 95 deg of crankshaft rotation;

T4 – combustion phase I: 5% of ensuing heat in the time of 15 deg of crankshaft rotation, combustion phase II: 95% of ensuing heat in the time of 105 deg of crankshaft rotation;

T5 – combustion phase I: 5% of ensuing heat in the time of 15 deg of crankshaft rotation, combustion phase II: 95% of ensuing heat in the time of 125 deg of crankshaft rotation;

T6 – combustion phase I: 5% of ensuing heat in the time of 15 deg of crankshaft rotation, combustion phase II: 95% of ensuing heat in the time of 145 deg of crankshaft rotation.
Fig. 9. Velocity of heat ensuing as a function of crankshaft rotational angle for differentiated number of ensuing heat during first and second combustion phase:

Q1 – combustion phase I: 5% of ensuing heat in the time of 15 deg of crankshaft rotation, combustion phase II: 95% of ensuing heat in the time of 55 deg of crankshaft rotation.

Q2 – combustion phase I: 5% of ensuing heat in the time of 15 deg of crankshaft rotation, combustion phase II: 95% of ensuing heat in the time of 75 deg of crankshaft rotation;

Q3 – combustion phase I: 5% of ensuing heat in the time of 15 deg of crankshaft rotation, combustion phase II: 95% of ensuing heat in the time of 95 deg of crankshaft rotation;

Q4 – combustion phase I: 5% of ensuing heat in the time of 15 deg of crankshaft rotation, combustion phase II: 95% of ensuing heat in the time of 105 deg of crankshaft rotation;

Q5 – combustion phase I: 5% of ensuing heat in the time of 15 deg of crankshaft rotation, combustion phase II: 95% of ensuing heat in the time of 125 deg of crankshaft rotation;

Q6 – combustion phase I: 5% of ensuing heat in the time of 15 deg of crankshaft rotation, combustion phase II: 95% of ensuing heat in the time of 145 deg of crankshaft rotation.

4. Conclusion

Performed simulations of working cycle of the dual fuel engine powered by diesel fuel and methane were aimed at the analysis of selected phenomena connected with possible course of combustion process of dual fuel blend delivered to the cylinder. The most important of a point of view of the efficiency and durability of the engine phenomena are:

1. increase of the maximum cycle pressure,
2. extending the combustion process.
The results of computer simulation indicate that very high level of maximum pressure inside combustion chamber can be reached with relatively high quantity of heat, (20%) that will ensue in the first phase of combustion. Such phenomenon is possible in case, when blend ratio methane – air in the whole combustion chamber volume is on the border of combustion (i.e. the ratio of methane in air - fuel blend in the range 2.2 - 9%), which means relatively low value of air – fuel equivalence ratio for compression ignition engine powered by two fuels. Such blend after being ignited in many points inside combustion chamber through pilot injection of diesel fuel, is combusting rapidly, which results in very high increase of maximum cycle pressure and in the end with possible knock. In case of powering the engine by the blends with such composition it has to be taken into account the decrease of the tense of the compressor. It is aimed at limiting the pressure in the beginning of compression stroke in the combustion chamber. Certainly, the limited tense will effect in delivering to the cylinder lower quantity of air. So that, it will contribute to generate the conditions, in which the methane – air blend will be on the border of combustion in the whole volume of combustion chamber.

References:

Abstract
The paper shows selected results of simulations performed in AVL Boost software, covering several aspects of methane combustion in compression ignition engine with the participation of pilot injection of diesel fuel.

Keywords: dual fuel engine, methane, AVL Boost

PROCES SPALANIA W SILNIKU DWUPALIWOWYM ZASILANYM METANEM I DAWKĄ PILOTUJĄCĄ OLEJU NAPĘDOWEGO

Streszczenie
Artykuł przedstawia wybrane wyniki symulacji przeprowadzonych w programie AVL Boost obejmujące wybrane aspekty procesu spalania metanu w silniku o zapłonie samoczynnym przy udziale dawki wstępnej oleju napędowego.

Słowa kluczowe: silnik dwupaliwowy, metan, AVL Boost