EXPERIMENTAL AND COMPUTATIONAL STUDY OF A LEAN BURN NATURAL GAS ENGINE

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Abstract. Turbocharged diesel engine used for electric generator was converted to run on natural gas. Modifications of the engine included engine head, piston, spark ignition set, fuel delivery system and intake system.

During laboratory tests the performance maps of the gas engine were taken at 1500rpm. The following measurements were performed:
- Composition of exhaust gases (NOx, CO, CO2 etc.);
- Temperature of exhaust gases;
- Pressure in combustion chamber;
- Fuel consumption;
- Electric power produced by generator.

Computations of combustion process in gas engine were made with the use of KIVA3V computer code. Several ignition angles and equivalence ratios were considered in the study. Results of computations include:
- Temperature distribution in combustion chamber;
- Pressure distribution in combustion chamber;
- NOx emission concentration;
- CO2 and CO emissions;
- OH concentration distribution in combustion chamber;
- Velocity and turbulence kinetic energy field in combustion chamber;
- Overall indicated pressure and temperature of the cycle.

All calculation were performed for original engine combustion chamber geometry.

1. Introduction

Natural gas engines are being used increasingly in stationary applications, most notably in Heat and Power systems (CHP). The engines are usually adapted from automotive diesel engines. The conversions are not usually made by the original manufacturers and in consequence the combustion systems have not been optimized. The conversions are made by replacing the fuel injection equipment with a gas carburettor and spark ignition and by modifying the induction and combustion systems for more appropriate air motion, chamber shape and compression ratio. Until recently such engines operated close to the stoichiometric air-fuel ratio (16.5-17.0 by mass) where performance is stable with acceptable efficiency but the exhaust emission of NOx is unacceptably high by today’s standards. The use of lean air-fuel mixtures is one well known way of increasing engine efficiency and reducing the exhaust emissions of SI engines. A disadvantage of lean operation, however, is that the burning rate of mixtures is reduced as a result of increase in the overall combustion duration, which in turn leads to increased heat-transfer losse.
Successful implementation of lean-burn strategy to minimize emissions and maximize thermal efficiency requires proper combustion chamber design which permits the fastest possible combustion rate.

The work presented here describes an experimental study of 13.3 litre turbocharged lean operated gas engine and the advanced modelling of gas engine combustion characteristics with the use of KIVA3V code.

2. Experimental

The engine used in the studies was a prototype, turbocharged, in-line 6 cylinder engine, adapted from diesel engine. The specification of the engine is given in Table 1. The engine was supplied with the natural gas of the following composition: \( \text{CH}_4 = 96\% \), \( \text{C}_2\text{H}_6 = 1.2\% \), \( \text{CO}_2 = 0.2\% \), \( \text{N}_2 = 2.4\% \), \( \text{O}_2 = 0.2\% \).

Table 1: Specification of the gas engine

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Configuration</td>
<td>In-line 6 cylinder</td>
</tr>
<tr>
<td>Bore [mm]</td>
<td>135</td>
</tr>
<tr>
<td>Stroke [mm]</td>
<td>155</td>
</tr>
<tr>
<td>Compression ratio</td>
<td>11.5</td>
</tr>
<tr>
<td>Capacity [dm³]</td>
<td>13.32</td>
</tr>
<tr>
<td>Power [kW]</td>
<td>200</td>
</tr>
</tbody>
</table>

The tests were conducted with constant rotational speed \( n = 1500 \text{ rev/min} \) and with constant ignition timing \( \alpha_2 = -12° \). The parameters varied during the tests were: engine load and excess air ratio. In the tests following measurements were made:

1. Engine rotational speed;
2. Generated electric power output;
3. Concentration of CO, HC, NO\textsubscript{x}, CO\textsubscript{2}, O\textsubscript{2} in exhaust gases;
4. Mean temperature of exhaust gases in the manifold and in individual outlet channels from cylinders;
5. Pressure in the combustion chamber of 6\textsuperscript{th} cylinder;
6. Temperature in the combustion chamber of 6\textsuperscript{th} cylinder.

The instrumentation of the test facility is shown in Fig.1. The exhaust emissions were measured by multicomponent analyser Oliver. The pressure in cylinder 6 was measured by AVL 8QP500c pressure transducer and charge amplifier. A crankshaft encoder and interface was used, that provide TDC and BDC flags, and pulses every 1° of crank angle.
Fig. 1. Schematic of gas engine and associated instrumentation
1-6 – mean gas temperature transducers at the outlet ports of individual cylinders; 7 – pressure transducer; 8 – mean gas temperature in outlet manifold of cylinders 1-3; 9 – mean gas temperature in outlet manifold of cylinders 4-6; 10-11 – air temperature transducer before and behind cooler; 12 – fluid temperature in cooling system; 13 – mean gas temperature in exhaust manifold; 14-15 – mixture temperature transducer before and behind turbocharger; 16 – air temperature transducer at engine inlet; 17 – oil temperature transducer; 18 – pressure drop transducer at lemniscate; 19-20 – water temperature transducer before and behind mixture cooler; 21 – mixture composition regulating valve; 22 – mixture pressure transducer behind turbocharger; 23 – ignition timing regulator; 24 – mixture pressure transducer behind turbocharger; 25 – air filter

Figures 2 and 3 show the effect of engine load on emissions, maximum pressure, indicating pressure and temperature. Within loads tested the largest influence was detected on maximum pressure and NO\textsubscript{x} concentration.

Fig. 2. Concentration of NO\textsubscript{x}, CO\textsubscript{2} and HC in exhaust gases as a function of engine load
Fig. 3. Maximum pressure, indicating pressure and temperature in 6th cylinder as a function of engine load

Figures 4 and 5 show the effect of mixture stoichiometry (AFR – air-fuel ratio by mass) on emissions, maximum pressure, indicating pressure and temperature.

Fig. 4. Concentration of NOx, CO2 and HC in exhaust gases as a function of air-fuel ratio

Fig. 5. Maximum pressure, indicating pressure and temperature in 6th cylinder as a function of air-fuel ratio
The following conclusions can be drawn from engine tests:

1. The variation of air-fuel ratio can be used for control of mechanical and thermal loads of the engine as well as NO$_x$, CO$_2$ and HC emissions in exhaust gases.
2. The increase of engine load results in the increase of engine mechanical and thermal load and NO$_x$ emission.
3. NO$_x$ emission is most sensitive to the changes of engine loads and air-fuel ratio.

3. Computer modelling

Calculations were performed with KIVA3V second release code [1]. Postprocessor GMV was used to prepare simulation. Calculation cycle took only two strokes (compression and expansion) of four-stroke engine and was based on simple chemistry model given below:

\[
\begin{align*}
\text{CH}_4 + \text{O}_2 &= \text{CO}_2 + \text{H}_2\text{O} \\
\text{N} + \text{NO} &= \text{N}_2 + \text{O} \\
\text{N} + \text{O}_2 &= \text{NO} + \text{O} \\
\text{N} + \text{OH} &= \text{NO} + \text{H} \\
\text{N}_2 &= \text{N} + \text{N} \\
\text{O}_2 &= \text{O} + \text{O} \\
\text{CO} + \text{CO} + \text{O}_2 &= \text{CO}_2 + \text{CO}_2 \\
\text{H}_2 &= \text{H} + \text{H} \\
\text{H}_2 + \text{O}_2 &= \text{OH} + \text{OH} \\
\text{H}_2\text{O} + \text{H}_2\text{O} &= \text{H}_2 + \text{OH} + \text{OH}
\end{align*}
\]

Computational mesh, had 180 degrees and about 26000 cells. Shape of the mesh cells allows for analysis of such parameters as pressure, temperature, turbulence kinetic energy, velocity and concentrations of species (NO$_x$, CO$_2$, CO, OH) in piston vertical intersection.

Composition of flammable mixture used in computations was based on composition of mixture used in laboratory tests with some some modifications forced by chemistry model. Gas used in calculation was a mixture of methane (97.62%), carbon dioxide (0.4%) and nitrogen (1.98%). Computations were made in two following steps:
1. Experimental verification of results from calculations;
2. Part-load performance and equivalence ratio maps.

3.1. Verification of results

Computational model includes many simplifications and assumptions, so experimental verification was necessary to check the correlation of results with real engine. Verification was made for full load, equivalence ratio 0.625 and spark ignition at 12 degrees before TDC. Comparison of cylinder pressure profiles from simulation and laboratory test is shown in Fig. 6.
The difference between calculational and experimental pressure profile results from simplyfied chemistry model used in the KIVA code. Also computational mesh density and pressure measurement error (in experiment pressure was measured locally but in calculation it was a mean value in the cylinder) can have an influence on it. Differences are not big and this simple model is close to real condition in piston engine.

### 3.2. Performance maps

Engine calculations were made for 100%, 80% and 60% of full load. In all cases equivalence ratio was kept at 0.625 and spark ignition at 12 degrees before TDC. Results are shown in Table 2 and in Figs. 7 and 8.

#### Tab.2. Computed results for different loads

<table>
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<tr>
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<td>100</td>
<td>2.21</td>
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<td>10.44</td>
<td>317</td>
<td>&lt;1</td>
<td>6.04</td>
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<tr>
<td>80</td>
<td>1.84</td>
<td>23.09</td>
<td>8.71</td>
<td>125</td>
<td>&lt;1</td>
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<tr>
<td>60</td>
<td>1.37</td>
<td>17.11</td>
<td>6.45</td>
<td>46</td>
<td>&lt;1</td>
<td>5.92</td>
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Fig. 7. Formation of CO₂, CO and NOx for different loads
Larger load causes higher pressure in combustion chamber, which results in higher temperatures and then in larger NO\textsubscript{x} formation. Indicated mean effective pressure (IMEP) and NO\textsubscript{x} emission grow with load but emission grows faster than IMEP.

Next series of calculations were made for different equivalence ratios. Calculations involved following cases: equivalence ratio: 1.0, 0.833, 0.714, 0.625 and 0.556; spark ignition at 12 degrees before TDC; 100% load. Results are shown in Table 3 and in Figures 9 and 10.

**Tab. 3. Results for different equivalence ratio.**

<table>
<thead>
<tr>
<th>Equivalence Ratio</th>
<th>Ind. Work [kJ]</th>
<th>Ind. Power [kW]</th>
<th>IMEP [bar]</th>
<th>NO\textsubscript{x} [ppm]</th>
<th>CO [ppm]</th>
<th>CO\textsubscript{2} [%]</th>
</tr>
</thead>
<tbody>
<tr>
<td>1.000</td>
<td>3.25</td>
<td>40.62</td>
<td>15.32</td>
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<td>48</td>
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<tr>
<td>0.833</td>
<td>2.83</td>
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<td>13.35</td>
<td>6534</td>
<td>1</td>
<td>7.89</td>
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<tr>
<td>0.714</td>
<td>2.50</td>
<td>31.26</td>
<td>11.79</td>
<td>2129</td>
<td>&lt;1</td>
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<td>0.625</td>
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<td>27.68</td>
<td>10.44</td>
<td>317</td>
<td>&lt;1</td>
<td>6.04</td>
</tr>
<tr>
<td>0.556</td>
<td>1.95</td>
<td>24.35</td>
<td>9.18</td>
<td>32</td>
<td>&lt;1</td>
<td>5.40</td>
</tr>
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</table>

Fig. 9. CO\textsubscript{2}, CO and NO\textsubscript{x} formation for different equivalence ratio.
Lower equivalence ratio means that there is less fuel in the combustion chamber with the same amount of air. Because of this smaller amount of heat is generated during combustion. This smaller quantity of energy has to heat the same part of gas, so the temperature is decreasing with equivalence ratio increasing. Smaller temperature has strong influence on chemical reactions speed and in spite of this emission of NOX is lower.

Conclusions

An experimental study of 13.3 litre turbocharged lean operated gas engine and the advanced modelling of gas engine combustion characteristics with the use of KIVA3V code were performed. Laboratory tests of the engine have shown that variation of air-fuel ratio can be used for control of mechanical and thermal loads of the engine as well as NOx, CO₂ and HC emissions in exhaust gases. The increase of engine load results in the increase of engine mechanical and thermal load and NOx emission.

Engine simulations have proved to be a valuable tool in qualitative analysis of engine performance and emissions with the change of operating parameters.

References