FREQUENCY ANALYSIS OF THE VARIATIONS OF PRESSURE IN THE BRANCHED INLET SYSTEM OF A 8A20G GAS ENGINE

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Abstract
The results of the indication of the individual inlet channels of an 8A20G 8-cylinder gas engine are presented in the article. On this basis, the filling factor has been evaluated for particular cylinders.

1. Introduction

The simultaneous indication of the variations of pressure occurring in the individual channels of the branched inlet system of an 8A20G gas engine (Fig. 1) was carried out by using an 8-channel digital measuring system composed of: an ADC488/8SA 16-bit A/C converter with its own software, supplied by IOtech; piezo-quartz pressure sensors, type SN6061B, supplied by KISTLER; an 8-channel charge amplifier, type 5017, supplied by KISTLER; a crankshaft rotation angle transmitter, type CAM 2611, supplied by KISTLER; a PORTABLE PCII computer; and a system for the measurement of time and the number of recorded engine operation cycles.

The simultaneous indication of the 8 inlet channels of the 8A20G engine was carried out for different degrees of loading (0÷110%), with 110% of loading corresponding to the engine effective power of Ne=680kW (Le=1.32MJ/m³).

Figure 2 shows examples of expanded indicator diagrams of the variations of pressure pulsation in the inlet channels for Le=1.32MJ/m³. In order to illustrate particular variations simultaneously, the variations were synchronized in relation to the inner dead centre (IDC) position of the cylinder being measured, and were shifted vertically by a value of 10kPa for each cylinder.

2. Analysis of the uniformity of fresh charge distribution

The analysis of the effect of pressure pulsation in the engine inlet manifold on the uniformity of distribution of a fresh charge was carried out by computational-experimental methods based on the numerical analysis of engine cycle performed by using the SILNIK computer program [2], where preset values were input, including real pressure variations measured in all inlet channel of individual cylinders (immediately before the heads), and the entire cycle of engine operation was computed for those variations, with particular consideration being given to the charge exchange process.
The cylinder filling process was calculated under the assumption that the flow characteristics of all pairs of suction valves and exhaust valves, as well as respective valve lift curves and valve clearances are identical. The probability of thermal processes occurring in cylinders after the closure of valves was also assumed. With thus adopted assumptions, the differences in the calculated values of the filling factor of particular cylinders will be the result of non-identical variations of pressure in particular channels caused by transient flows in a relatively long (2.4 m) inlet manifold of the engine under consideration. The non-identicalness of pressure variations,
occurring during opening of the suction valves will be of significant importance here. Figure 3 shows the load characteristics of filling factor variations for particular cylinders of the 8A20G engine. The results of numerical computations of the filling factor and indicated work in the respective cylinders of the engine operating at 110% load are given in Fig. 4 and in Table 1.

**Fig. 3.** Load characteristics of the filling factor of particular cylinders of the 8A20G engine (C1÷C8 – filling factors of cylinders 1÷8, respectively, obtained by means of numerical modeling; SU – filling factor determined from the measurement of the consumption of air and gas by the engine)

**Fig. 4.** Results of the numerical computations of the filling factor and indicated work of the engine operating under 110% load (nv - flow – the line denotes the filling factor determined from the measurement of the consumption of air and gas by the engine)

\[ L_\text{e} = 1.32 \text{ MJ/m}^3, \ h_\text{e} = 1.7574 \]
Table 1

Results of the computations of the filling factor for particular cylinders and the average value and the limits of dispersion obtained under the engine load of 1.32 MJ/m³

<table>
<thead>
<tr>
<th>Cylinder No.</th>
<th>1</th>
<th>2</th>
<th>3</th>
<th>4</th>
<th>5</th>
<th>6</th>
<th>7</th>
<th>8</th>
</tr>
</thead>
<tbody>
<tr>
<td>Filling factor</td>
<td>1.832</td>
<td>1.692</td>
<td>1.654</td>
<td>1.622</td>
<td>1.679</td>
<td>1.619</td>
<td>1.620</td>
<td>1.575</td>
</tr>
<tr>
<td>Computed value, averaged for eight cylinders</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td>1.662±0.078(±4.7%)</td>
</tr>
<tr>
<td>Measured value, averaged for eight cylinders</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td>1.757±2.5%</td>
</tr>
</tbody>
</table>

The filling factor computation results summarized in Table 1 enable the quantitative evaluation of the effect of pressure variation non-identicalness on the process of filling particular cylinders, and indicate that the maximum difference in the filling factor for particular cylinders does not exceed 5% and is little significant.

3. Frequency analysis of recorded pressure variations

From the variation of pressure in the inlet channel, averaged for 180 successive cycles, a part was selected, which was comprised in a “window” contained within a range of approx. 40°÷226.5°CR (crankshaft rotation) angle corresponding to the opening of the inlet valve from the moment of exhaust valve closure to the moment of inlet valve closure. Of 720 values for the entire cycle, from 210 to 443 values contained in this window were subject to frequency analysis. Figure 5 shows the analyzed variations of pressures in the inlet channels for the engine load of $L_e = 1.32\text{MJ/m}^3$.

![Figure 5](image-url)

Fig. 5. Fragments of the pressure variation in the inlet channels contained in the “window”, subjected to frequency analysis (the horizontal lines are the lines of the ±10% tolerance of pressure for the end of the inlet phase in a given channel)
Considering the average values of the parameters of medium in the channels in this angular interval (p=123.8kPa, T=320K), the frequencies of the basic form of medium free vibration in the channel were calculated. The channel is regarded as a one-sidedly opened straight constant cross-section conduit filled with gas; the length of this conduit corresponds to ¼ of pressure wavelength for the basic form of the free vibration of the gas column, while the frequencies of the successive forms of free vibration are defined by the relationship below:

\[ f_{\lambda/4} = \frac{(2n+1)c}{4(l + \Delta l)} \quad n = 0, 1, 2, k , \]

where:
- \( c \) – velocity of acoustic wave propagation within the medium filling the conduit \( c = \sqrt{\kappa \cdot R \cdot T} \),
- \( \kappa \) – adiabatic exponent,
- \( R \) – individual gas constant,
- \( T \) – temperature,
- \( l \) – conduit length,
- \( \Delta l = 8\cdot r/3/\pi \) – so called outlet correction [6],
- \( r \) – conduit radius,
- \( n \) – number of free vibration form).

Thus calculated frequency of free vibration in the crankshaft rotation angle range 40\(^\circ\)–226.5\(^\circ\) for inlet channels is given in Table 2.

### Table 2

<table>
<thead>
<tr>
<th>Vibration form number</th>
<th>0</th>
<th>1</th>
<th>2</th>
</tr>
</thead>
<tbody>
<tr>
<td>( f_{\lambda/4} ) Hz</td>
<td>157.7</td>
<td>473</td>
<td>788.4</td>
</tr>
</tbody>
</table>

Figures 6-13 show the frequency spectra of the amplitude of pressure analyzed, with marked average values of 180 cycles analyzed. In Fig. 14, comparison of the analyzed averaged pressure variations is made for all inlet channels of the 8A20G engine.
Fig. 6. Frequency spectrum of the variation of pressure in the inlet channel of cylinder no. 1 (averaged values of 180 cycles of engine operation) for an effective work of $L_e=1.32\text{MJ/m}^3$.

Fig. 7. Frequency spectrum of the variation of pressure in the inlet channel of cylinder no. 2 (averaged values of 180 cycles of engine operation) for an effective work of $L_e=1.32\text{MJ/m}^3$. 
Fig. 8. Frequency spectrum of the variation of pressure in the inlet channel of cylinder no. 3 (averaged values of 180 cycles of engine operation) for an effective work of $L_e=1.32\,\text{MJ/m}^3$.

Fig. 9. Frequency spectrum of the variation of pressure in the inlet channel of cylinder no. 4 (averaged values of 180 cycles of engine operation) for an effective work of $L_e=1.32\,\text{MJ/m}^3$. 
Fig. 10. Frequency spectrum of the variation of pressure in the inlet channel of cylinder no. 5 (averaged values of 180 cycles of engine operation) for an effective work of $L_e=1.32\text{MJ/m}^3$.

Fig. 11. Frequency spectrum of the variation of pressure in the inlet channel of cylinder no. 6 (averaged values of 180 cycles of engine operation) for an effective work of $L_e=1.32\text{MJ/m}^3$. 
Fig. 12. Frequency spectrum of the variation of pressure in the inlet channel of cylinder no. 7 (averaged values of 180 cycles of engine operation) for an effective work of $L_e = 1.32 \text{MJ/m}^3$.

Fig. 13. Frequency spectrum of the variation of pressure in the inlet channel of cylinder no. 8 (averaged values of 180 cycles of engine operation) for an effective work of $L_e = 1.32 \text{MJ/m}^3$. 
The performed frequency analysis of the variations of pressure recorded in the inlet channels show that for the maximum load of the engine ($L_e=1.32\text{MJ/m}^3$) the participation of the basic harmonic component of the variation of pressure in the inlet channel is predominant (Table 3). The amplitude of this component is greater than the amplitude of higher components by a factor of 14.5 to 35.1.

$$\frac{A_1}{A_{\text{max}(2\div19)}}$$

<table>
<thead>
<tr>
<th>Channel No.</th>
<th>1</th>
<th>2</th>
<th>3</th>
<th>4</th>
<th>5</th>
<th>6</th>
<th>7</th>
<th>8</th>
</tr>
</thead>
<tbody>
<tr>
<td>Harmonic No.</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>1</td>
<td>0.2260</td>
<td>0.2256</td>
<td>0.2229</td>
<td>0.2241</td>
<td>0.2238</td>
<td>0.2251</td>
<td>0.2241</td>
<td>0.2247</td>
</tr>
<tr>
<td>2</td>
<td>0.0065</td>
<td>0.0141</td>
<td>0.0077</td>
<td>0.0074</td>
<td>0.0050</td>
<td>0.0131</td>
<td>0.0064</td>
<td>0.0089</td>
</tr>
<tr>
<td>3</td>
<td>0.0066</td>
<td>0.0059</td>
<td>0.0153</td>
<td>0.0051</td>
<td>0.0110</td>
<td>0.0045</td>
<td>0.0055</td>
<td>0.0097</td>
</tr>
<tr>
<td>4</td>
<td>0.0040</td>
<td>0.0034</td>
<td>0.0051</td>
<td>0.0021</td>
<td>0.0043</td>
<td>0.0031</td>
<td>0.0029</td>
<td>0.0034</td>
</tr>
</tbody>
</table>

$$\frac{A_1}{A_{\text{max}(2\div19)}}$$

|          | 34.5 | 16  | 14.5 | 30.2 | 20.4 | 17.1 | 35.1 | 23.1 |

Fig. 14. Comparison of frequency spectra for all inlet channels ($k1$–$k8$ – number of the inlet channel of a particular cylinder) – average values of 180 cycles of engine operation for an effective work of $L_e=1.32\text{MJ/m}^3$.
4. Conclusions

- During the process of filling the cylinder with a fresh charge, i.e. from 33°CR crankshaft rotation) before IDC (inner dead centre) to 46°CR after ODC (outer dead centre), the variations of pressures measured in particular inlet channels are characterized by a good agreement, which indicates that those phenomena occur in the inlet system of an 8-cylinder in-line engine and do not cause any significant differences in the variations of pressures in the inlet channels of individual cylinders and do not create any risk of an uniform distribution of the fresh charge.

- The above finding confirms the previous conclusions [4, 5] that the inlet manifold of the 8A20 engine has been designed correctly and, despite its quite large length, does not introduce any significant disturbances in the uniformity of fresh charge distribution among individual cylinders.

5. References


3. Gruca M.: Program komputerowy „Ind488”.
