Analysis of changes in the angular velocity of the crankshaft of the marine engine for diagnosing the wear and location the failure of the fuel injection system

1 Introduction

The mode of operation of the internal combustion engines involves the conversion of chemical energy of fuel into mechanical work. The combustion process can be simply described as periodic explosions in a cylinder with frequency dependent on the number of cylinders and the rotational speed of the shaft. In practice, uniformity of combustion parameters in every cylinder is almost impossible. Due to this fact, instantaneous angular acceleration does not remain the same at the ends of the crankshaft. These observations formed the basis for the investigation of the instantaneous angular speed of the crankshaft ends. To investigate the influence of the failure behavior of the fuel systems during shaft's rotational movement, a series of experiments were planned.

For the simulations, a medium-speed marine engine driving electro generator was selected. The failure simulation was based on the installation of clogged spray holes, draining of part of fuel dose from high-pressure pump and decreasing of injection pressure by a lower tension of the injector spring. The results of measurement were processed and analyzed through a comparison of the fast Fourier transform spectra. As a general conclusion, a difference between general harmonics order of magnitudes was detected.

Keywords: medium speed reciprocating marine internal combustion engine, instantaneous angular speed, fast Fourier transform, failure simulation of fuel delivery system
sional vibration is a result of the periodical character of gaseous forces and the reaction of load torque. Therefore, all deviations or disturbances of excitation are reflected in the angular speed and torsion magnitudes [4, 9]. A medium-speed marine engine driving electro-generator was a selection for simulation. The engine, mounted at the laboratory testbed, allows the installation of the defective fuel valves and for the simulation of high-pressure fuel pump failures. In order to simulate engine work under the conditions of malfunction, three types of failure were selected. The simulation of fuel leak from the high-pressure pump was simulated by unscrewing the bypass ball valve. This operation results in a drain of the part of the fuel dose from the high-pressure pump. This operation affects the combustion pressure, especially by lowering the maximum value. The level of changes can be observed and adjusted by observing the pressure run displayed at the electronic indicator screen. Also, failures of the injection process were the object of investigation. The simulation of failure was done by the installation of a partly clogged sprayer nozzle, draining of part of fuel dose from the high-pressure pump and decreasing the injection pressure by lower tension of injector spring. The results of the measurement were processed and analyzed by a comparison of the fast Fourier transform spectra.

2 Test rig description

The simulation of the diesel engine fault was carried out at the test rig of the Maritime University in Gdynia. The test engine was a three-cylinder, four-stroke medium-speed marine diesel, driving a three-phase electro-generator. The nominal speed of the engine was 750 rev/min, and the generator frequency was 50 Hz. In Figure 1, the optical encoders used for angular speed registration, mounted at opposite sides of the engine, are presented. Recording starts simultaneously, but the master disc is disc number 1, installed at the free end of the shaft. That disc has an additional starting slot, which also allows the determination of the position of the first piston TDC.

The signal from encoders was received by the processing module and finally converted into several laser impulses received in the relevant time of the passage of tooth and slit through the laser ray. This formed the basis of speed calculation when the angular width of slit and tooth were known. To avoid errors due to manufacturing accuracy, the pair of tooth–slit was considered for the calculation of speed.

3 Analysis of failure impact at instantaneous angular speed run

In order to determine or detect possible deviations of IAS, a reference is needed for comparison. The reference measurements must be carried out with an engine in a healthy condition. It means that all regulations and adjustments of fuel system elements shall be done following the manufacturer’s manual. The schedule of experiments was composed of measurements at a wide range of loads, but the results of only the 70% MCR (250 kW) were taken for further processing and presented in this paper.

3.1 Healthy engine analysis

The measurements of the IAS of a healthy engine were repeated 20 times, during independent runs of the engine, and at different outer temperatures. It was done in order to evaluate the uniformity of results and to determine the impact of outer conditions. The obtained results showed the occurrence of differences between the records from free
and loaded ends of the shaft. In Figure 2, the runs of IAS after processing (smoothing) are presented. The differences in either magnitudes or peak phases are clearly visible. Because of the small width of the band of speed variations, degrees per second was treated as the unit of measurement.

The subsequent step of the analysis was the comparison of the spectra of harmonic components of both ends. The expected differences between the magnitudes and frequencies of leading harmonics were observed (see Figure 3). A difference can be observed in the magnitudes of major frequencies. Free end disc (disc1) IAS had a leading frequency of approximately 56 and 62Hz, which is referred to as the 8th and 9th order (Table 1).

| Order of a harmonic (k) | Frequency [Hz]               |
|-------------------------|------------------------------|
| ½                       | 6.25 (one cylinder combustion) |
| 1                       | 12.5 (basic harmonic component) |
| 1½                      | 18.75 (combustion harmonic component) |
| 2                       | 25                           |
| 2½                      | 31.25                        |
| 3                       | 37.5                         |
| 3½                      | 43.5                         |
| 4                       | 50 (polar pulsation for four pairs of poles of a single voltage phase) |
| 4½                      | 56.25                        |
| 5                       | 62.25                        |
| 5½                      | 68.25                        |
| 6                       | 75                           |
| 6½                      | 81.25                        |
| 7                       | 87.25                        |
| 7½                      | 93.75                        |
| 8                       | 100                          |

The major magnitude at the generator’s end occurred at 18.6 Hz, which is a reflection of the contribution from the pistons. It must be stated that “healthy engine” means that the fuel system adjustments were corrected according to the manual. As a very old engine is under consideration, this procedure does not ensure the ideal regularity of the engine run.

For further analysis of fluctuations at both ends, a differential value of harmonic magnitudes ($\Delta m$) was calculated, which represents the difference of subsequent order of magnitude of disc 1 and disc 2.

$$\Delta m_i = \Delta 1_i - \Delta 2_i;$$

where:

$\Delta 1_i$ – magnitude of $i$th-order of disc 1

$\Delta 2_i$ – magnitude of $i$th-order of disc 2

Figure 4 shows that the major excitation frequencies responsible for disc 1 fluctuations were 55.6 and 62.2 Hz. For disc 2, the major difference in magnitude was observed at 18.5 Hz. The engine’s rotational frequency was 12.5 Hz, and the engine cycle occurred at a frequency of 6.25 Hz. It means that the frequency of disc 2 is strictly attributed to the excitation coming from three cylinders. Figure 4 shows this excitation because the contribution from cylinders has the strongest impact on the IAS fluctuation of disc 2 (negative value for $\Delta m$). The fluctuation of disc 1 is represented by the highest orders of frequency – 55.9 and 62.2 Hz (positive value of $\Delta m$). These observations lead us to conclude that
the malfunctions of the fuel system are reflected in the relations between the ends records, and parameter $\Delta m$ can be used to study the equality of contribution of cylinders.

### 3.2 Simulation of HP pump leak

The malfunction of the pump was simulated by the release of the drain ball valve spring. The strength of impact at maximum combustion pressure was controlled by reading the electronic indicator. The results of IAS measurements are presented in Figure 5.

The obtained results differ vastly from that for a healthy engine. The strongest excitation of disc 2 was observed for the frequency of approximately 80.7 Hz. This magnitude is four times higher than for the healthy condition. This could be clearly inferred from the value of the $\Delta m$ parameter (value of $-20$). The positive value appears for 43.6 Hz, which means that disc 1 excitation is stronger. Nevertheless, the most significant finding is the huge rise of negative value in the range of the 13th order (80.66 Hz) (see Figure 6).

**Figure 7**: Spectra of IAS in case of low tension of fuel valve spring.

**Figure 8**: Comparison of IAS harmonic components for simulation of HP pump leak and difference between magnitudes.

In general, the consequences of that failure are similar to those observed when a pump leak occurred but a twice higher negative $\Delta m$ value ($-35$) in the range of 88.6 Hz is seen. It means that the fuel injection failure caused by a lower valve of the opening pressure had a stronger impact on shaft torsion and subsequently on the difference between acceleration at the shaft ends.

### 3.3 Simulation of weak spring of fuel valve

The simulation of the failure of fuel valve spring was done through the diminishing of spring tension, and as a consequence, lower injection pressure. The results of the measurements are presented in Figures 7 and 8.

### 3.4 Simulation of the clogged sprayer nozzle

This problem was simulated by installing a special sprayer nozzle with only seven holes instead of the original nine. The measurements were carried out in order to find out the possibility of detecting the malfunction and to compare the results with stronger combustion malfunctions that had occurred previously. The obtained spectrum of harmonic components and values of parameter $\Delta m$ are presented in Figures 9 and 10. It is a very interesting observation, because in general, the differential value was very low, which means that the fluctuation of IAS of the free end and the loaded end of the shaft was almost identical. It leads to the conclusion that engine run is more regular when the injection process in cylinder number 2 is disturbed. This change can be caused by the general condition of the engine due to wear and tear (lower compression). Thus, it is clear that even optimum parameters of the injector do not guarantee a smooth run of an engine.
4 Conclusions

The experimental results demonstrate that the fluctuation of IAS of the free and loaded ends of the shaft is different. The difference can be detected by observing the spectrum of harmonic components, which are characterised by different frequencies and magnitudes. Serious disturbances of fuel delivery, such as pump leak or diminishing of injection pressure, cause stronger deviations of the spectrum. When the engine run is highly irregular, the difference between IAS spectra grows, and as a consequence, there is an increase in comparison parameters.

Figure 11 shows the influence of strength of disturbance on the $\Delta m$ value. A serious malfunction of the injector creates bigger peaks of differential parameters (solid line) for some frequencies. So the diagnostic utility of $\Delta m$ for detecting malfunction is thus established.

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