Mechanics and Mechanical Engineering 2021; 25 (1):72–76

Research Article

Sebastian Drewing, Norbert Abramczyk, and Daria Żuk*

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

https://doi.org/10.2478/mme-2021-0010
Received Jul 08, 2021; accepted Jul 23, 2021

Abstract: The combustion process can be simply described as periodic explosions in a cylinder with its 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 a 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

***Corresponding Author: Daria Żuk:** Gdynia Maritime University, Faculty of Marine Engineering, Morska str. 81-87, 81-225 Gdynia, Poland. ORCID: 0000-0002-0810-0626
Sebastian Drewing: Gdynia Maritime University, Faculty of Marine Engineering, Morska str. 81-87, 81-225 Gdynia, Poland. ORCID: 0000-0002-2388-4342
Norbert Abramczyk: Gdynia Maritime University, Faculty of Marine Engineering, Morska str. 81-87, 81-225 Gdynia, Poland. ORCID: 0000-0003-4556-7994

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 rotational speed of the shaft. In theory, one can assume ideal equality between the cylinders. The older constructions, with camshaft valve opening control, due to wear and tear of elements, show deviations of the contribution between individual cylinders [1].

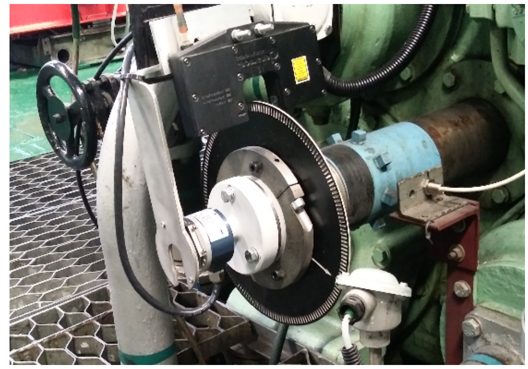
These observations formed the basis for the investigation of the instantaneous angular speed (IAS) of crankshaft ends [2, 3]. Despite fluctuations in the instantaneous angular speed, at the end of every cycle (two revolutions for four-stroke engine), the mean value of angular speed is equal for both ends of the shaft. Therefore, a comparison of speeds should give information on how the combustion process occurs. The point of highest interest is the evaluation of the contribution of a single piston and, eventually, the detecting reasons for inequality. Considered as a non-intrusive method, the measurement of IAS, due to significant development of data acquisition and processing, has found a broad spectrum of applications for condition monitoring. The IAS technique was used for cylinder pressure reconstruction, monitoring of gear transmission, fault detection of high-speed diesel engines [4] and multicylinder diesel engines driving electro-generator sets [1, 5]. Instantaneous angular speed analysis can be used for the monitoring of engine performance because oscillations in the angular speeds of crankshaft directly reflect the gas pressure torque produced by piston-conrod assembly during the combustion process.

The second phenomenon of interest from the diagnostic point of view is torsional vibration [6]. In many works, torsional vibration is pointed out as a reason for the heavy damage of reciprocating and rotating machines [7, 8]. Torsional 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.

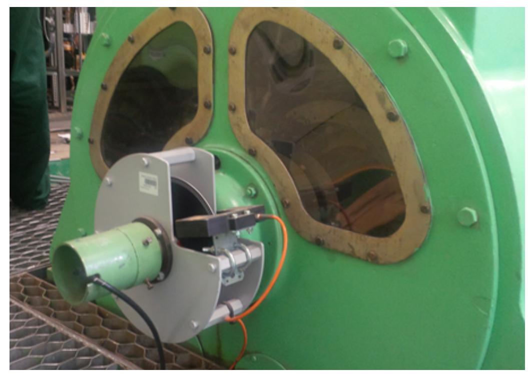
2 Test rig description

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.



(a)



(b)

Figure 1: Encoders mounted at free end (a) and generator’s end side (b).

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.

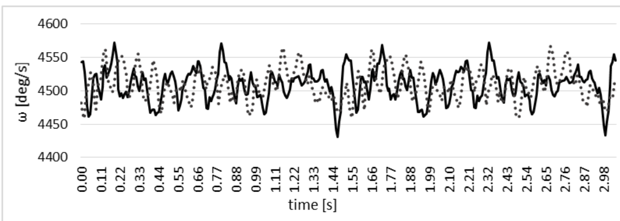


Figure 2: Run of IAS of disc 1 and 2 recorded during healthy engine test (70% MCR).

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).

Table 1: Orders of selected harmonic components and corresponding frequencies.

Order of a harmonic (k)	Frequency [Hz]
$\frac{1}{2}$	6.25 (one cylinder combustion)
1	12.5 (basic harmonic component)
$1\frac{1}{2}$	18.75 (combustion harmonic component)
2	25
$2\frac{1}{2}$	31.25
3	37.5
$3\frac{1}{2}$	43.5
4	50 (polar pulsation for four pairs of poles of a single voltage phase)
$4\frac{1}{2}$	56,25
5	62,25
$5\frac{1}{2}$	68,25
6	75
$6\frac{1}{2}$	81,25
7	87,25
$7\frac{1}{2}$	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.

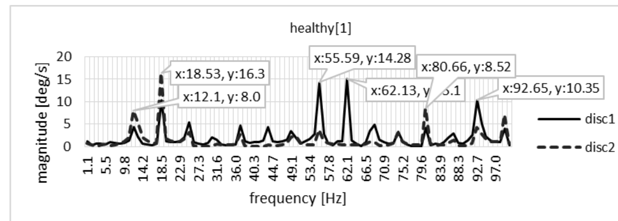


Figure 3: Comparison of harmonic components of disc 1 and disc 2 of healthy engine.

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

$$\Delta m_i = \Delta_{1i} - \Delta_{2i}; \tag{1}$$

where:
Δ_{1i} – magnitude of ith-order of disc 1
Δ_{2i} – magnitude of ith-order of disc 2

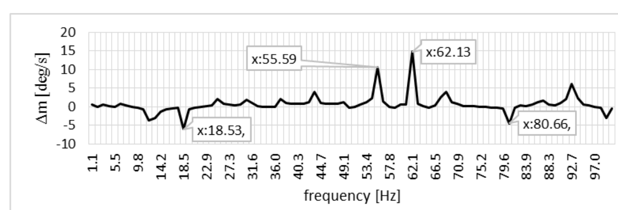


Figure 4: Differential value Δ_m for healthy engine.

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 Δm). The fluctuation of disc 1 is represented by the highest orders of frequency – 55.9 and 62.2 Hz (positive value of Δm). These observations lead us to conclude that the malfunctions of the fuel system are reflected in the relations between the ends records, and parameter Δm 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 Δm 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).

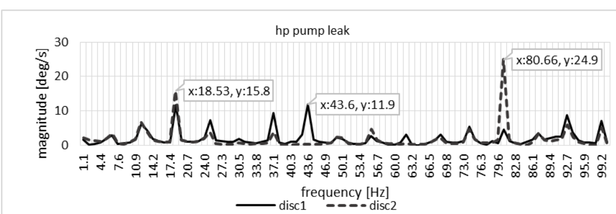


Figure 5: Comparison of IAS harmonic components for simulation of HP leak pump.

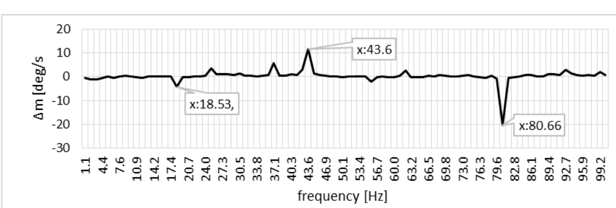


Figure 6: Comparison of IAS harmonic components for simulation of HP pump leak and difference between magnitudes.

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.

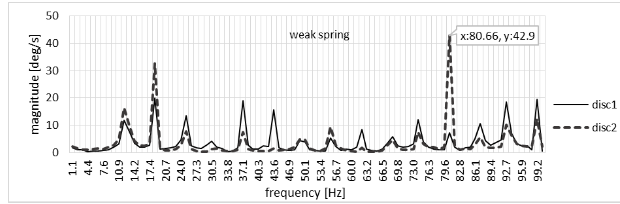


Figure 7: Spectra of IAS in case of low tension of fuel valve spring.

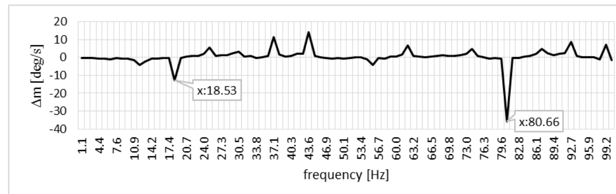


Figure 8: Comparison of IAS harmonic components for simulation of weak spring and difference between magnitudes.

In general, the consequences of that failure are similar to those observed when a pump leak occurred but a twice higher negative Δm 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.

3.4 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 Δm 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.

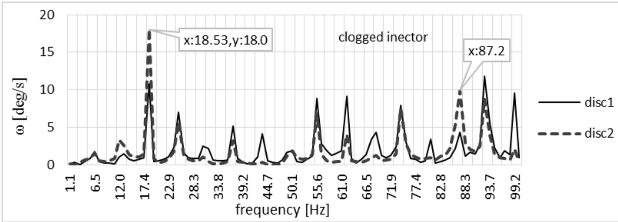


Figure 9: Spectra of IAS in case of partly clogged sprayer nozzle.

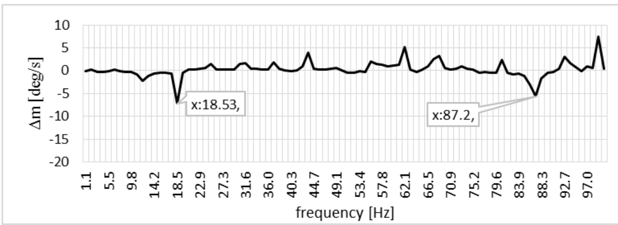


Figure 10: Differential value of harmonics in simulation of clogged injection nozzle.

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.

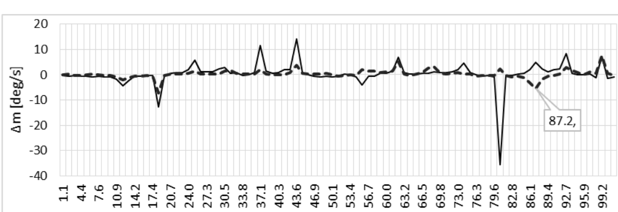


Figure 11: Comparison of *Δm* parameter of weak spring (solid line) and clogged nozzle (dotted line) of fuel valve.

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

References

[1] **Lin T. R., Tan A.C., Ma L., Mathew J.**: Estimating the loading condition of a diesel engine using instantaneous angular speed analysis, *Proceedings of the 6th World Congress on Engineering Asset Management*, Cincinatti, Ohio, USA, **2011**.

[2] **Desbaseille M., Randall R.B., Guillet F., El Badaoui M., Hoisnard C.**: Model–based diagnosis of large diesel engines based on angular speed variations of the crankshaft, Mechanical Systems and Signal Processing, 24, pp 1529 – 1541, **2010**.

[3] **Williams J., Witter, M.**: Individual Cylinder IMEP Estimation Using Crankshaft Angular Velocity Measurements, SAE Technical Paper 2001-01-0990, ,https://doi.org/10.4271/2001-01-0990, **2001.**

[4] **Geveci M., Osburn A.W., Franchek M.A.**: An investigation of crankshaft oscillations for cylinder health diagnostics, Mechanical Systems and Signal Processing. 19 ,pp. 1107 – 1134, **2005.**

[5] **Gawande S.H., Navale L.G.,. Nandgaonkar M.R, Butala D.S., Kunamalla S.**: Detecting power imbalance in multi-cylinder inline diesel engine genset, Journal of Electronic Science and Technology, Vol.8, No3, **2010**.

[6] **Adamson S.**: Improved Approaches to the Measurement and Analysis of Torsional Vibration, SAE Technical Paper 2004-01-1723, https://doi.org/10.4271/2004-01-1723, **2004.**

[7] **Feese T.**, *Coupling failures in VFD motor/fan system due to torsional vibration*, Torsional Vibration Symposium, Salzburg, paper No.7B-2, **2017.**

[8] **Ravindran R., Das D., Kamani K., Sivaraman P. et al.**: Methodology Development for Torsional Vibration Measurement and Processing in Powertrains, SAE Technical Paper 2015-01-2278, https://doi.org/10.4271/2015-01-2278, **2015.**

[9] **Malaczynski G., Van der Poel R.**: Phase Diagrams of Different Modes of Misfire Calculated from the Digital Fourier Transformation of Angular Crankshaft Velocity, SAE Technical Paper 2010-01-0167, https://doi.org/10.4271/2010-01-0167, **2010.**