DESIGN OF DETECTOR OF INTERNAL COMBUSTION ENGINE'S CRANKSHAFT TORSIONAL VIBRATIONS BASED ON ACCELEROMETER METHOD

Oleksander Veretennik
Doctor of Engineering Sciences, Professor
Department of Ship's power plants[1]
vamod@ukr.net

Dmitry Kardashev
PhD
Department of Theoretical Mechanics[1]
dlkardashev@ukr.net

Sergey Aboleshkin
PhD
Department of Ship's power plants[1]

[1]*National University "Odessa Maritime Academy"*
8 Didrikhsona str., Odessa, Ukraine, 65029

Abstract: Detector of internal combustion engine's crankshaft torsional vibrations based on accelerometer method is designed. It consists of a measuring and transmitting part, which is mounted on the crankshaft of a diesel engine or the propeller shaft of a vessel and a receiving and recording part placed remotely. The measuring and transmitting parts of detector is easily mounted on area of the shaft of the vessel important for the measurement, which are difficult to access for other methods of vibration measuring. It allows directly in real time to measure the tangential accelerations of the shaft without interfering in the propulsion unit work.
Carrying out such measurements allows to control the arising torsional stresses and signalling the excess of their safety limit. The measurement results of engine's pulsations of rotation of the crankshaft, arising during operation, allow to watchkeeping engineer to analyze the quality of the working process of each engine's cylinder in real time and at different operating modes of the vessel. This analysis makes it possible to assess the technical condition of the engine's parts and assemblies (cylinder liner, piston, o-rings), the quality of the fuel equipment of the diesel engine and, in general, allows to control the technical condition of the entire engine.
Keywords: sensor-accelerometer, torsional vibrations, torsional stress, crankshaft, damper, propeller shaft, amplitude-frequency characteristic.

1. Introduction

The appearance during the operation of a ship's propulsion unit of torsional vibrations of ship shaft lines causes dangerous dynamic loads in this unit, which can lead to damage to elements of power plants that provide safe operation of the ship, such as crankshafts [1], gearboxes, couplings, torsional vibration dampers [2] and other elements of the vessel. Requirements for compulsory theoretical calculation and experimental control of parameters of torsional vibrations on vessels in operation are determined by the rules of international classification societies [3–6].

2. Methods

As is known theoretical calculation of parameters of torsional vibrations of ship's propeller shaft includes the following stapes [6–10]:

1) bringing the mechanical system to discrete mass system;

2) harmonic analysis of the change in torque for one revolution of the crankshaft;

3) determination of natural frequencies, relative amplitudes of angular oscillations of masses, positions of nodes;

4) calculation of the dependence of shear stress on the engine speed of the crankshaft, taking into account the damping in the system and the harmonic components of the torque, the definition of resonance zones;

5) calculation of vibrations with the cylinder disconnected, with the damper locked and disconnected.

The reliability criterion is the conformity of calculations with subsequent real torsiographic measurements.

3. Results

The authors develop the method for measuring dynamic loads on a crankshaft described below, using the method for determining the dynamic shear stress of a crankshaft section based on the mathematical terms (1) [9, 10]

$$\tau = I\,\varepsilon / W, \quad (1)$$

where τ is the twisting stress of the shaft section; W is the polar moment of resistance of the cross section of the shaft section; I is the moment of inertia of the concentrated mass; ε is the angular acceleration of the corresponding concentrated mass.

Fig. 1 shows a block diagram of measuring the instantaneous value of the angular acceleration of the shaft.

The moment of inertia of the mass and the moment of resistance are determined by calculation in accordance with the technical characteristics of the engine.

The method of measurement with the help of the following equipment is implemented.

The measuring and transmitting unit is attached to the free end of the engine crankshaft, flywheel or damper. It includes a pair of three-axis accelerometers with the same sensitivity parameters, which are mounted opposite each other at the same distance from the axis of rotation. This condition is fulfilled to exclude the influence of the field of gravity. The measuring unit also includes an electronic unit consisting of an adder, an amplifier, a modulator and antenna.

The receiver unit includes a receiver, a demodulator and a four-channel recording oscilloscope. The receiver unit includes a receiver, a demodulator and a four-channel recording oscilloscope (**Fig. 2**).

A specialized software product is using for mathematical calculation of the obtained measurements.

Under the influence of tangential acceleration, accelerometers generate analog electrical signals for each channel separately, which are summed in the adder, amplified in the amplifier and modulate the high-frequency signal in the modulator. The carrier frequency of the frequency-modulated signal is selected in the interference free region with a bandwidth of 1 Hz to 2 kHz. The resulting modulated signal is emitted by a transmitter with an antenna that has a circular pattern.

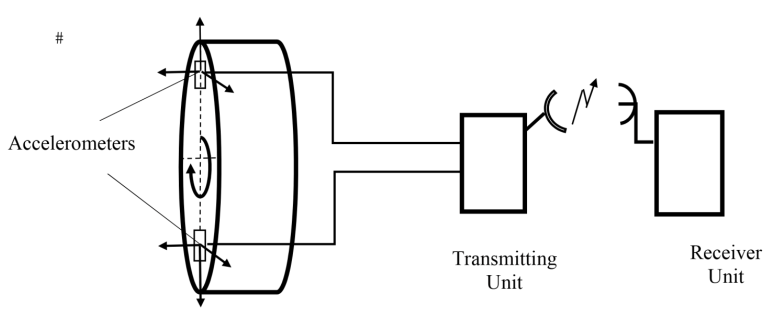


Fig. 1. Block diagram of the measuring unit

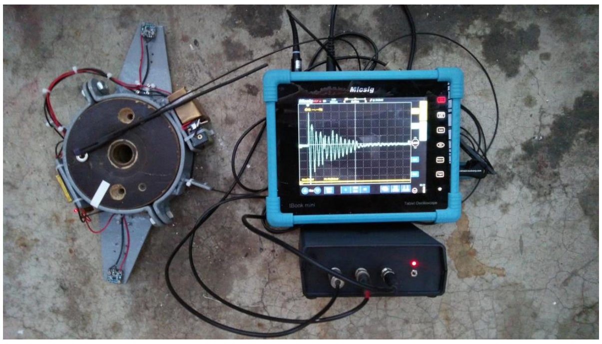


Fig. 2. View of the measuring unit

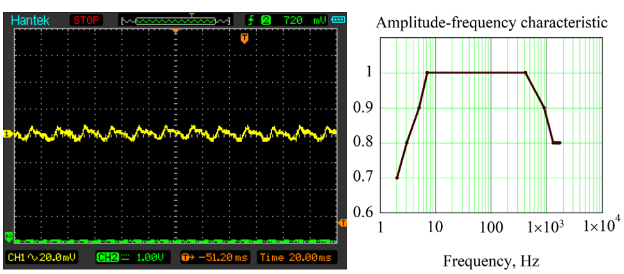
In the receiving path the demodulator decodes the signal and displays on the oscilloscope screen, which allows to get information in digital form. For convenience of the subsequent processing of the information and a binding to parameters of working process of the engine the counter of revolutions is used. The counter signals are output to a separate oscilloscope channel. The sensitivity of the acceleration sensors is 6.5 mV/g at a radius of 0.17 m. The self frequency of the accelerometer sensor is 2.5 kHz.

To determine the level of the shear stress dependence of time let's use the working formula

$$\tau(t)=Ig[U(t)-U_n(t)]/\lambda RKW, \qquad (2)$$

where I=6,7 kgm^2 (calculated) – moment of mass inertia on which accelerometers are installed; g=9.81 m/s^2 – acceleration of gravity; $U(t)$ – the output time signal of the oscilloscope, mV; $U_n(t)$ – signal of the noise of the transmit-receive path mV (**Fig. 3, *a***); R=0.17 m – the distance of the sensor from the axis of rotation; K – coefficient of nonlinearity of the amplitude-frequency characteristic of the receiving-transmitting path (**Fig. 2, *b***); λ=6.5 mV/g – sensitivity of the accelerometer sensor; W=138.4 m^4 – the moment of resistance of shaft section.

The receiving and transmitting tract's noise is recorded on a separate channel.

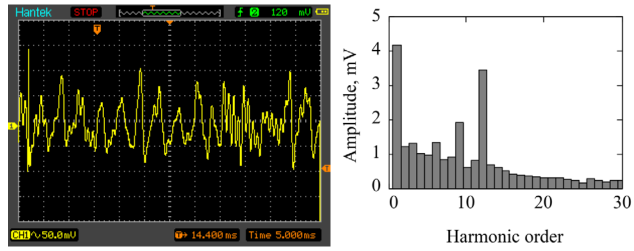


a *b*

Fig. 3. Characteristics of the receiving and transmitting: *a* – Noise; *b* – amplitude-frequency characteristic of the receiving and transmitting

The measurements were carried out on a laboratory diesel engine in the range of rotational speeds from minimally stable to 1400 rpm with 50 rpm increments.

Fig. 4 shows the oscillogram of the signal in the near-resonance region (1000 rpm), as well as the Fourier spectrum of tangential stresses, obtained from (2) after expansion in mathematical series.



a *b*

Fig. 4. Signal in the near-resonance region: *a* – Analog signal; *b* – Fourier spectrum of the obtained crankshaft tangential acceleration

4. Discussions and conclusions

Fig. 5 shows the frequency response of tangential stresses on the measured section of the crankshaft obtained after mathematical processing of the spectra. The boundaries of permissible and maximum stresses are shown.

Resonances were observed at frequencies of 857 rpm (the amplitude of the tangential stress is 18 % of the allowable by the Russia Maritime Register od Shipping (RMRS) requirements); 1270 rpm (tangential stress amplitude is 60 % of permissible by the RMRS requirements).

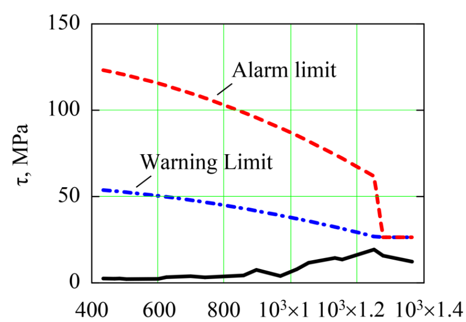


Fig. 5. Tangential stress τ depending on the engine speed (rpm)

The proposed methodology has a number of advantages:

– ability to continuously record in real time the signal in the entire range of measured rotation speeds of the crankshaft;

– direct determination of torque and tangential stresses on the shaft;

– allows to record the signal remotely.

The reliability of the obtained results is confirmed by the satisfactory confirmation of the last one with theoretical calculations of the parameters of torsional vibrations of the engine.

The disadvantages include the impossibility of fixing the measuring unit in the shafting areas, the most interesting for research, which is typical for most techniques, as well as the need to provide filtering of the noise of the transmit-receive path.

References

1. Homik, W. (2011). Damping of torsional vibrations of ship engine crankshafts - general selection methods of viscous vibration damper. Polish Maritime Research, 18 (3), 43–47. doi: https://doi.org/10.2478/v10012-011-0016-9

2. Efremov, L. V. (2007). Teoriya i praktika issledovaniy krutil'nyh kolebaniy silovyh ustanovok s primeneniem komp'yuternyh tehnologiy. Sankt-Peterburg: Nauka, 276.
3. ISO 3046-5: Reciprocating internal combustion engines-Performance – Part 5: Torsional vibrations (2001). International Organization for Standardization, 10.
4. Lloyd's Register of Shipping (2000). Main & Auxiliary Machinery. Rules & Regulations for the Classification of Ships, Part 5. London.
5. Rossiyskiy rechnoy registr. Rukovodstvo R.009-2004. Raschet i izmerenie krutil'nyh kolebaniy valoprovodov i agregatov (2016). Moscow, 68.
6. Rossiyskiy morskoy registr sudohodstva. Prilozheniya k rukovodstvu po tehnicheskomu nablyudeniyu za sudami v ekspluatatsii ND No. 2-030101-009 (2013). Sankt-Peterburg, 227.
7. Willson, W. K. (1964). Amplitude calculation. Practical solution of torsional vibration problem, Volume two; Chapman and Hall. LTD. London, 381–388.
8. Den Hartog, J. P. (1985). Mechanical Vibrations. New York: Dover Publications, Inc., 464.
9. Terskih, V. P. (1953). Raschety krutil'nyh kolebaniy silovyh ustanovok. Vol. 1. Moscow: Mashgiz, 160–161.
10. Istomin, P. A. (1968). Krutil'nye kolebaniya v sudovyh DVS. Leningrad: Sudostroenie, 306.

Received date 07.10.2019
Accepted date 08.11.2019
Published date 23.11.2019