Turbine Engine Resonance Parameter Monitoring as a Condition Prediction Tool

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The operation of turbine engines, which are machines with a high degree of construction complexity and characterized by rotational speeds of high values, forces users to pay special attention to their condition. There are many standardized diagnostic methods relating directly to turbine engines, but they are still being expanded and improved. The guidelines contained in the standards (i.e. ISO 10816-4) usually do not oblige the user to predict the future condition. Most often, only limit values are given, the exceeding of which gives information about the need to repair the engine or its component. The paper proposes a method which, when properly used, can serve as the basis for the condition assessment system built for the operation of marine turbine engines. Collecting an appropriate reference database from engines in different conditions will allow to adequately predict the repair time with an indication of critical elements that require special attention.

The possibilities of using resonance parameters of turbine engines to assess the condition of the rotor system and foundation are presented. Measurements were made during the run-down and with the use of a modal hammer. The subjects of the research were the GTD 350 turbine engine and a model of the rotor system on which damage in the form of imbalance and misalignment were simulated. From the point of view of operational diagnostics, measurements during engine start and stop are of key importance, because they allow the comparison of current vibration parameters with historical parameters. Tracking changes in the values of these parameters enables inference about the condition of the rotor system and can be a good supplement to the diagnostic systems of both aviation and marine turbine engines. The paper presents two alternative methods of obtaining information about the instantaneous value of rotational speed. An alternative to classic measurements with the use of a tachometric probe, measurements are made using the vibration signal as a source of information about the instantaneous rotational speed.

Starting point for the proposed method of predicting the condition of the device is experimental modal analysis (EMA). EMA is a process based on the experimental determination of the resonance frequencies of the tested system. Using appropriate techniques, it is also possible to determine the mode of natural vibrations. EMA has many advantages as it is a fast and relatively cheap method of determining resonance frequencies. In the proposed method, the authors assume that changes in the stiffness of the rotor system caused by changes in its condition will affect on the values of resonance frequencies. The output equation for modal analysis is the general equation of the oscillating motion:

$$m\ddot{x}(t) + c\dot{x}(t) + kx(t) = f(t) \tag{1}$$

where: *m* – mass, *c* – damping ratio, *k* – elasticity ratio, *x* - displacement

Equation (1) might be transformed to:

$$[-m\omega^2 + jc\omega + k]X(\omega) = F(\omega) \tag{2}$$

where: ω – is system natural frequency. Assuming that:

$$H(\omega) = \frac{1}{-m\omega^2 + jc\omega + k} \tag{3}$$

The quantity $H(\omega)$ is known as the frequency response function (FRF). Finally it might be written:

$$H(\omega) = \frac{X(\omega)}{F(\omega)} \tag{4}$$

The FRF describes the ratio of the Fourier transform at the output of the $X(\omega)$ system to the Fourier transform of the forcing applied to the $F(\omega)$ system. Three types of estimators *H1*, *H2* and *H3* are used when determining the FRF value. The correct use depends on the expected input and output signal components. The use of the H3 estimator allows to minimize the influence of input and output signal noise on the results of the frequency response of the system. The H3 estimator is given by the equation:

$$H3 = \sqrt{H1 \cdot H2} \tag{5}$$

H1 is the estimator used when the output signal has a high signal to noise ratio, *H2* is the estimator used to minimize the impact of input signal noise. *H1* and *H2* estimators are given by equations (6) and (7).

$$H1 = \frac{G_{XF}}{G_{XX}} \tag{6}$$

The H1 estimator should be interpreted as the ratio of the cross spectrum of the response and excitation signal G_{XF} to the mutual excitation spectrum G_{XX}.

$$H2 = \frac{G_{FF}}{G_{XF}} \tag{7}$$

The H2 estimator is interpreted as the ratio of the cross-excitation signal spectrum G_{FF} to the excitation response cross-spectrum G_{XF}.

During the preliminary tests, changes of foundation stiffness of rotor system model (Fig. 1) on its resonance frequencies was verified. For this purpose, the device was placed on 4 identical shock absorbers. The tests were carried out in three series for three different shock absorber stiffnesses (55, 65 and 75 ShA). In each series, three extortions were performed using a modal hammer.

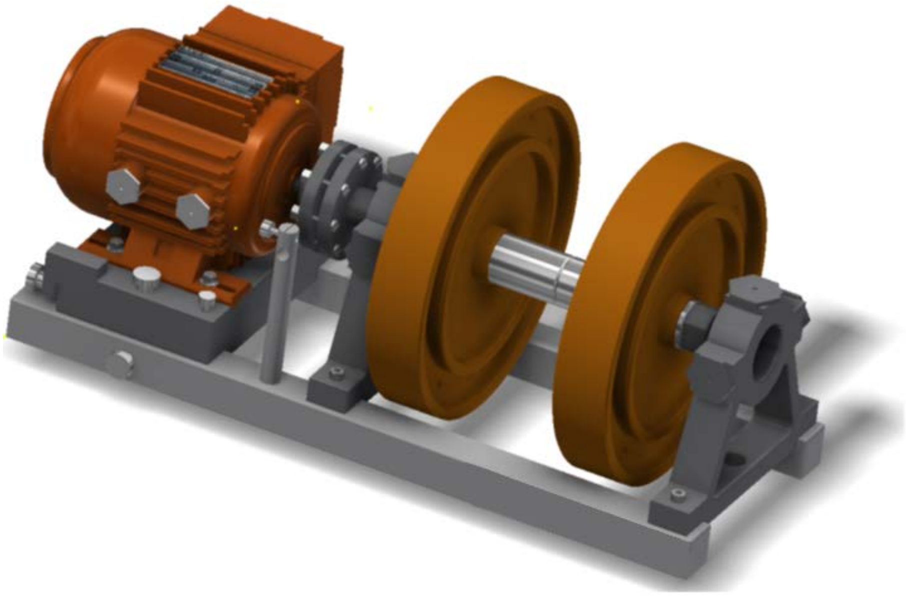

Figure 1: Model of the tested rotor system

Changes in the stiffness of metal-rubber shock absorbers on which the propulsion systems of vessels with turbine engines are installed are a frequent phenomenon during their operation. These changes cause changes in the parameters of the shaft line alignment, which in a short time may damage its elements. The spectra of the frequency response function obtained during the preliminary tests are shown in Figure 2. The colors represents different stiffness of the same material, all absorbers were made of chloroprene rubber. Black color - shock absorbers with a stiffness of 65 ShA, green color - shock absorbers with a stiffness of 65 ShA, red color - shock absorbers with a stiffness of 65 ShA.

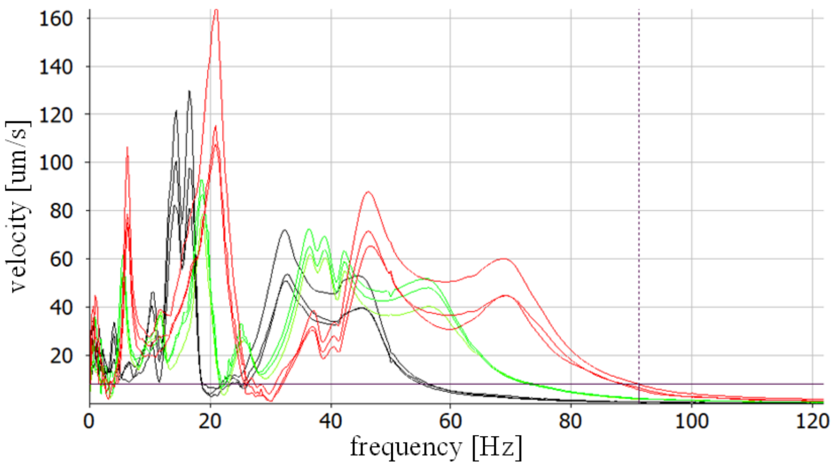

Fig. 2 Velocity vibration spectrum obtained during modal hammer tests of rotor model mounted on metal-rubber shock absorbers with different stiffness (single integrated signal).

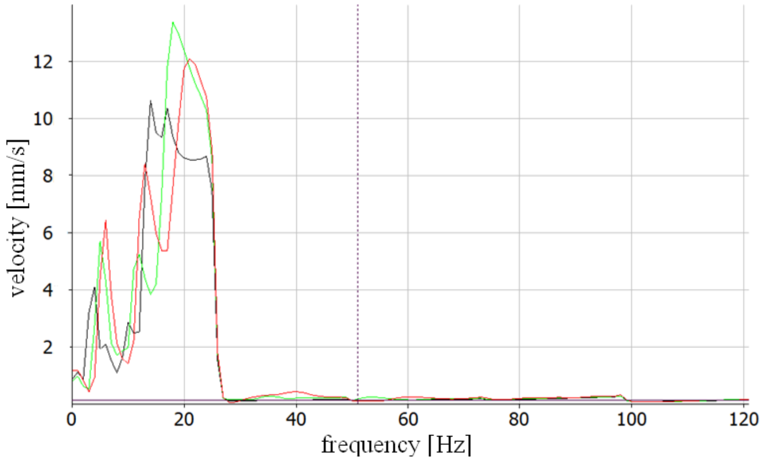

Fig. 3 Spectrum of the maximum amplitude velocity obtained during run down tests of rotor model mounted on metal-rubber shock absorbers with different stiffness (single integrated signal).

The analysis of the waveforms presented in Figure 2 and 3 allows the observation of significant changes in the frequency of the system's response to the excitation depending on the stiffness of the shock absorbers. It is important that for each stiffness, three measurements were made and the responses from these give the same characteristic frequencies. They differ in the values of the maximum amplitudes, but this is mainly due to the different values of the force input signal. The results of the preliminary tests give grounds for the conclusion that it will also be possible to detect reasons of other changes in the system stiffness. Relating the results of future measurements to the reference results and determining the trend of changes will allow for determining and predicting the condition of the tested device.