Vibration Diagnostics Methods of Marine Diesel Engines with Turbocharger

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ABSTRACT

This paper discusses the marine diesel engine vibration diagnostics methods, according to the international standards. The analysis of vibration diagnostics methods applied to piston engines is being shown. The analysis of root-mean-square value of vibration velocity (RMS) for various classes of mechanisms is being shown. The main defects of various diesel units and the frequencies of the harmonics that indicates their occurrence have been presented. A method for eliminating the "leakage effect" in the discrete spectrum DFT is offered. The proposed method is based on solving a system of complex equations. The article considers the method for diagnosing a turbocharger, based on the spectral analysis of vibroacoustic signals of a compressor.

1.0 INTRODUCTION

The structure of marine diesel engines combines various mechanisms and units associated with different functions and operations, such as:

- reciprocating motion mechanisms (the crank mechanism, pistons);
- rotary mechanisms (gear, belt and chain drives, oil and water pumps, crankshafts and camshafts);
- turbocharger;
- high pressure fuel injection elements (high pressure fuel pumps, valves and injectors);
- gas distribution mechanism (inlet and exhaust valve drives);
- roller bearings and connecting couplings;
- other components (for instance, generator as a part of a diesel-generator set).

Normal functioning of a marine diesel engine is ensured by a complex of different operational mechanisms. Each one of these mechanisms generates its own specific vibrations, which greatly complicate the task of vibration diagnosis of marine diesel malfunctions, despite relatively easy malfunction diagnosis based on analyzing vibrations of rotary type machinery. The fact that the vibration signal at some location results from a sum of very heterogeneous signals from the different nearby mechanisms complicates the task even more. In this case, it is logical to simplify the task of diagnosis by analysing the individual components and mechanisms [1].

2.0 VIBRATION UNITS ANALYSIS OF MARINE DIESEL ENGINES ACCORDING TO ISO 10816 STANDARD [2]

Individual vibration harmonics are being calculated for each engine unit. Basic vibration frequencies for all types of diesel engines in all cases are the following:

- crankshaft rotation frequency harmonics $f_n = n_{RPM} / 60;$
- cylinder harmonics $f_{cyl} = f_n \times i_{cyl} \times Coef.stroke,$

where n_{RPM} *is number of rotations per minute, icyl is* number of cylinders and *Coef.stroke* = 0,5

for four-stroke and *Coef.stroke* =1,0 for two-stroke engines.
For example, the four-stroke diesel engine MAN 6D20 [www.imes.de] with the engine speed 1500 RPM has basic harmonic equations the following:

$$f_n = 1500 / 60 = 25\ Hz$$
$$f_{cyl} = 25 \times 6 \times 0{,}5 = 75\ Hz$$

There is an evaluation method of engine technical condition based on analyzing vibration frequencies of non-rotating parts (ISO 10816[2]). According to the requirements of [2], technical condition diagnosis is dependent on general vibration class for each mechanism. In order to analyze this, certain mechanisms RMS vibrospeed values between 10-1000 Hz are being used. For a continuous signal $V(t)$, RMS is defined as $\tilde{V}_{RMS} = \sqrt{\frac{1}{T}\int_0^T V^2(t)dt}$, where T is the sampling period, which should be longer (at least 10 times) than any basic period of any analyzed signal fundamental frequencies contained in $V(t)$.

In case of a discretely recorded vibration signal V_i ($i = 1,2,\ldots N$) and availability of N values of vibrospeed V_i, the RMS value is being defined as

$$V_{RMS} = \sqrt{\frac{1}{N}\sum_{i=1}^{N} V_i^2}\ .$$

Certain filters have to be applied for choosing between necessary frequency range of 10 - 1000 Hz when calculating RMS for a time signal. V_{RMS} calculations are done easier using spectrum amplitude values s_j

$$V_{RMS} = \sqrt{\frac{1}{2}\sum_{j=k_1}^{k_2} s_j^2}\ ,$$

where k_1 and k_2 are harmonics indexes between 10 Hz and 1000 Hz, respectively.

It is possible to estimate overall technical condition of the mechanism by calculating the RMS value using the Table 1, where vibrospeed RMS limits for normal (A, B) and abnormal (C, D) conditions for mechanisms of distinct classes are specified. MAN D20 diesel corresponds to a class III in case of its installation on the rigid basis and to a class IV in case of an installation on flexible support.

Table 2-1: ISO 10816: zones* vibrospeed RMS for mechanisms of various classes [2].

<table>
<tr><th>R.m.s. vibration velocity
mm/sec</th><th>up to 15 kW
Class I</th><th>15 to 75 kW
Class II</th><th>> 75 kW
(rigid)
Class III</th><th>> 75 kW
(soft)
Class IV</th></tr>
<tr><td>0,28</td><td rowspan="3">A</td><td rowspan="4">A</td><td rowspan="5">A</td><td rowspan="6">A</td></tr>
<tr><td>0,45</td></tr>
<tr><td>0,71</td></tr>
<tr><td>1,12</td><td rowspan="2">B</td></tr>
<tr><td>1,8</td><td rowspan="2">B</td></tr>
<tr><td>2,8</td><td rowspan="2">C</td><td rowspan="2">B</td></tr>
<tr><td>4,5</td><td rowspan="2">C</td><td rowspan="2">B</td></tr>
<tr><td>7,1</td><td rowspan="5">D</td><td rowspan="2">C</td></tr>
<tr><td>11,2</td><td rowspan="4">D</td><td rowspan="2">C</td></tr>
<tr><td>18</td><td rowspan="3">D</td></tr>
<tr><td>28</td><td rowspan="2">D</td></tr>
<tr><td>45</td></tr>
</table>

*** A - good condition; B - satisfactory; C - unsatisfactory; D - an emergency condition, the operation is dangerous**

Vibration level higher than the normal indicates existing problems in the mechanism, like cylinder unbalance, breaking or misalignment of couplings, fasteners weakening, mounting problems, etc. Since each unit of a diesel engine is unique in its vibration, the most effective method of diagnosis is an over-time monitoring of measurement results. Such monitoring can be accomplished by trending (scatter plan of RMS values of vibration velocity in time), see Fig. 2-1.

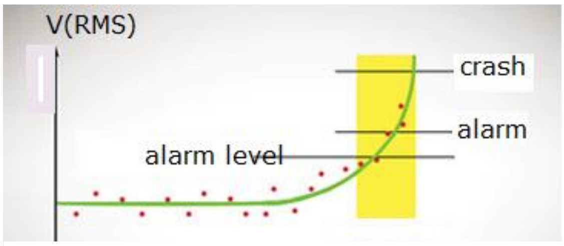


Figure 2 - 1: RMS time trend

In the absence of problems, overall vibration mechanism will remain stable for some time. Subsequent periodic measurements are compared with the baseline level. Frequency measurement depends on the type of engine and on its condition. Warning and alarm levels can be set on the trend graph [1,3]. Warning levels are setting borders within which the increase in signal may occur randomly. Steady increase in the amplitude above the warning level shows the need to increase the frequency of the measurements in order to prevent a malfunction. When amplitude approaches to the alarm level, it is necessary to undertake a more detailed spectral analysis of vibration signals in order to detect the defect and take measures to eliminate it and prevent mechanism crash (Figure 2-1).

Table 2-1 shows that the vibration level is determined by the vibration RMS and describes the general state of the mechanism. Specific future defects in individual units can be predicted when analyzing the vibration spectrum, as shown in Table 3-1.

3.0 SPECTRAL ANALYSIS OF ENGINE COMPONENTS VIBRATION

Spectral analysis of individual harmonics can give more detailed information about the nature of diesel defects. Table 3 lists and summarizes the main defects of various diesel units and the frequencies of the harmonics for which these defects occur. The data (Table 3) resulted from the analysis of references [1,4,5]. According to [3], as a result of irregularities in the cylinders fuel supply (power imbalances), "... steady increase the depth of modulation (15%) of the vibration at the frequency of the working cycle (fcyl), and also increase the depth of modulation is present (7%) at the fundamental frequency of a diesel engine (fn)" occurs. In any case, information about power imbalances between cylinders, when detected, is not reliable diagnostic information, which determines the specific defect and points the way to its correction.

Table 3 - 1: Diesel defects and frequency characteristic harmonics [1, 4, 5]

Malfunction	Vibration frequency	Description
Fixing defects	*fn,* *2× fn*	Disproportionate growth of the first two harmonics of the fundamental frequency
Defects in alignment with the diesel generator	*fn,* *2× fn,* *3× fn*	Arise in the misalignment of the crankshaft and the generator rotor. In a spectrum there are first three multiple harmonicas of the main frequency
Defects of fastening of the generator	*fn,* *2× fn*	Disproportionate growth of the first two harmonics of the fundamental frequency. It is most pronounced in one direction
Increased vibration of the oil and water pumps' drives as well as the camshaft drive	$Fz = fz \times z$ f_Z *- frequency of the rotor* *z - number of teeth*	Predominant frequency of engaging the gear Fz. Signs: growth of the harmonic amplitude Fz and the appearance of harmonics shaft gears fz

Table 3 - 1. Continued		
Vibration couplings	*fn* *in the axial direction*	Parallel shear or fracture of coupling cause mass unbalance of rotating rotor due to the axis displacement of the mass centers. Axial vibration with frequency *fn* appears
Increased vibration of crankshaft bearings	F_{CR} =(0,42÷0,48) × *fn* *in the radial direction*	Vibration depends on a condition of working surfaces of sliding bearings, gaps between shaft and bearings as well as on properties of lubricating oil. Under certain conditions oscillations of the shaft in the radial direction may arise. Appearance in the spectrum of harmonics F_{CR} may characterize defect inserts or shaft
Vibration of rolling bearings (generators etc.)	- defect in an outer ring BPFO: $f_0 = \frac{z}{2} f_n \left(1 - \frac{d}{D}\cos\beta\right)$ - defect of the inner ring BPFI: $f_i = \frac{z}{2} f_n \left(1 + \frac{d}{D}\cos\beta\right)$ - separator defect FTF: $f_c = \frac{f_n}{2} \left(1 - \frac{d}{D}\cos\beta\right)$ - defect of the rolling elements BSF: $f_r = \frac{D}{d} f_n \left(1 - \left(\frac{d}{D}\cos\beta\right)^2\right)$	Defects or damage to the rolling contact surfaces (outer / inner ring, separator or rolling elements). More often vibration acceleration is analyzed (or vibration velocity in some cases). *z* - number of rolling elements; *D* - diameter of a circle passing through the center of the rolling elements (average diameter of the separator); *d* - the diameter of the rolling element; β - contact angle of the rolling element.
Vibration of rolling bearings (simplified formula, through RPM)	- BPFO: $f_0 \approx 0{,}45 \cdot z \cdot RPM$ - BPFI: $f_i \approx 0{,}55 \div 0{,}6 \cdot z \cdot RPM$ - BSF: $f_r \approx 3{,}5 \cdot RPM$	. *z* - number of rolling elements; *RPM* - revolutions per minute of crankshaft
Turbocharger rotor unbalance	f_{TUR}	Deviation from axial symmetry of the rotor and the heterogeneity of the material. Causes forced oscillation with a speed of the turbocharger due to rotation of the unbalanced centrifugal forces.
Vibration of rotor blades of the air compressor (gas turbine) turbocharger	$f_{TUR} \times n_{b.comp}$ $n_{b.comp}$ - *number of the compressor wheel blades of turbocharger*	Compressor blade vibration is always present in the vibration spectrum and does not indicate a defect *. Defect diagnosed by increased levels of blade harmonics over time.
Defect turbocharger mounting	*fn* *in the vertical and transverse directions*	Defect formed during the assembling or during operation. The general level of vibration and harmonic spectrum rises at engine speed in the vertical and transverse directions.

* Presence of a bladed compressor harmonica in a spectrum of vibration allows carrying out the turbocharger express analysis [7].

3.1 Diagnostics of piston engines is based on amplitudes of multiple harmonics

References [6, 7] propose an interesting method to diagnose reciprocating engines failures based on analysis of multiple harmonics of the main frequency (from f_n up to the 9th harmonica). Vibration is measured at points located nearby to diagnosed units of the engine; then, amplitudes of harmonics f_n , $2f_n$,...,$9f_n$ are examined. Estimation of technical condition of units and engines components is based on values of coefficients K, which are equal to the quotient of the squares harmonics sums:

$$K_{(j\div l)}^{(p\div s)} = \sum_{m=p}^{s} A_m^2 / \sum_{i=j}^{s} A_i^2 ,$$

where A_m, A_i *with* $i \in [1...9]$ are the amplitudes of *the m-th and i-th* harmonics in the spectrum.

The references [4, 6, 7] – from which Table 3-2 was extracted – propose a number of criteria for the condition diagnosis of reciprocating engines. We think that those criteria and table, directly or in a modified form, can also be used for marine diesel engines.

Table 3 - 2. Coefficients of multiple harmonics for diagnostics of piston engines [4, 6, 7]

Measurement place	Mechanism or unit	Type of malfunction	$K_{(j\div l)}^{(p\div s)}$
Crosshead	Crank mechanism	Gaps, condition of sliding surfaces, surfaces of the pin and bearing of the small end of the rod, rigidity of fastening of a piston rod	$K_{(1\div 3)}^{(1)}$
Crosshead	Crank mechanism	Gaps, condition of connecting rod big end and main bearings, rigidity of fastening of the rod big end	$K_{(1\div 5)}^{(3\div 5)}$
Valves; the cylinder in a zone of valves	Valves	Breakdown of springs, change of "time-section" parameter, hydraulic hammer, deviation of valve timing	$K_{(1\div 3)}^{(1)}$
Cylinder cover	Details of the cylinder unit	Clearances, wear of piston rings, cylinder liner surface, loosening of the piston rod, increased "piston-cylinder liner" clearance	$K_{(1\div 3)}^{(1)}$
Bearing of reciprocating engine from the drive side or flywheel	Rotating details of a shaft, the coupling	Imbalance	$K_{(1\div 9)}^{(1)}$
	Shaft of the engine and generator	Misalignment	$K_{(1\div 9)}^{(2\div 3)}$
	Coupling	Increased clearance, slackening, decreasing of rigidity	$K_{(1\div 9)}^{(3\div 9)}$

Two facts emerge from the analysis of the fore mentioned proposed reciprocating engines diagnostic method:

- The use of the squares of multiple harmonics quotient normalizes levels of diagnostic coefficients in the range [0 to 1];
- Identical coefficients calculated from sensor data installed at various locations contain information about the corresponding units and mechanisms condition.

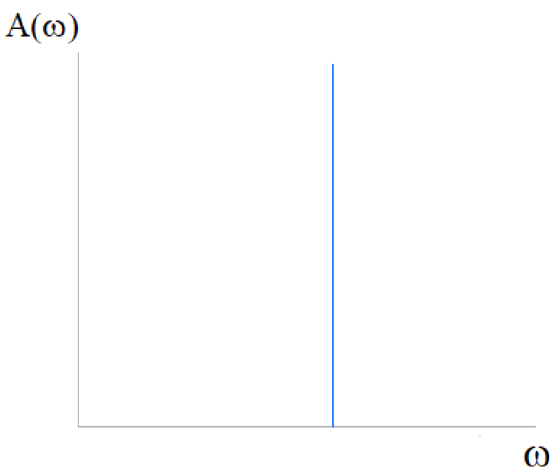
Unfortunately, this technique [6, 7] could not be extended yet to the most important units of modern diesels - the high pressure fuel equipment (nozzles and high pressure fuel pumps). It is known that the more frequent operational failures of ship diesels - from 50% to 80% - correspond to the high pressure fuel equipment [8, 9].

The proposed methodology in [4, 6, 7] has to be checked for high-speed diesels. The equations for calculation of diagnostic coefficients have to be specified and modified if necessary. Thus it is possible to recognize that in a spectrum of diesel vibration the most informative harmonics are $0{,}5f_n$, f_n, f_{cyl} and multiples of them up to 20-th harmonic.

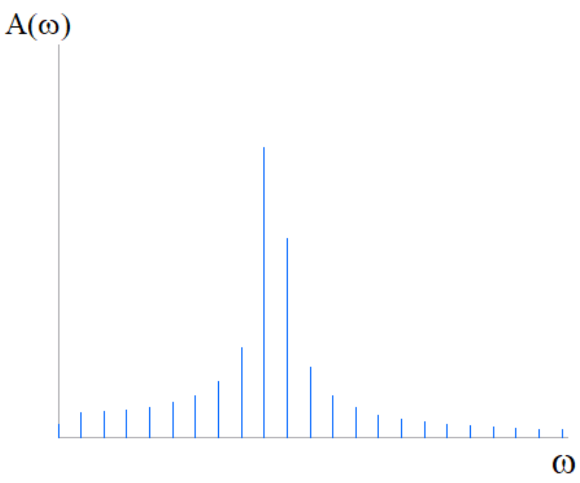
4.0 ELIMINATING THE "LEAKAGE EFFECT" OF DISCRETE SPECTRUM

In the process of analyzing the discrete spectrum of vibroacoustic signals in order to estimate their frequency and amplitude characteristics, it is necessary to solve the problem of eliminating the effect of "leakage". This effect is a consequence of the finiteness of the analyzed temporal realization and its discrete representation. The effect of "leakage" or outflow of power from the spectral peaks into the adjacent spectral lines is considered to be one of the main DFT errors [10 - 12].

As an example, Fig. 4-1 shows the amplitude spectra of the same sinusoidal signal with an integer (a) and a non-integer (b) number of samples per one signal period. Let the frequency of a signal be represented by $\gamma = M/T$, where T is the period of the signal; $M = n + \sigma$, where n is an integer and $0< \sigma <1$, then the maximum distortions of the amplitude, frequency and phase of the central harmonica and leakage of power into the neighbouring ones will be observed at $\sigma = 0{,}5$ See [12].



a) σ = 0



b) σ ≠ 0

Figure 4 - 1: DFT leakage effect

Thus, when analyzing the parameters of the original spectrum signal, i.e. the central harmonic, the resulting amplitude, frequency and phase will be distorted in the case of a non-integer number of signal samples per its period. In practice, for discrete recording of signals, an ADC with a selected and fixed sampling rate is used. It is understandable that the number of samples per period will never be an integer and the value of σ will change from 0 to 1 depending on the natural frequency of the measured signal, and the accuracy of estimating the signal parameters along the central harmonic will change.

The most common solution for leakage effect reduction is based on window transform methods. The essence of the method is simple: to reduce the number of discontinuities at the edges in order to reduce leakage it is necessary to reduce the amplitude of the signal near the edges. This scaling is carried out during the implementation of the multiplication by the window with the special form

$$s_j^w = s_j \cdot W(j),$$

where *W(j)* is stands for Window functions (see Table. 4-1)

As a result of applying window functions, the spectrum of the original signal is changed and its RMS decreases by RMS Coeff times, as shown in Table 2. The decrease in *RMS* when using the Hanning window is 0,707 / 0,433 = 1,633. Thus, the dependence of the amplitude, frequency and phase of the fundamental harmonic in the spectrum from the value of σ decreases. This means that the fundamental harmonic s^w_j can be used to approximate the signal parameters with a certain constant error, which can be taken into account.

Table 4-1. Window functions and RMS coefficient used to reduce the effect of "leakage" [12]

Hemming window	*RMS Coeff = 1,414*	$\varpi(n) = 0{,}53836 - 0{,}46164\cos\left(\frac{2\pi n}{N-1}\right)$
Hanning window	*RMS Coeff = 1,633*	$\varpi(n) = 0{,}5\left(1-\cos\left(\frac{2\pi n}{N-1}\right)\right)$
Kaiser window	*RMS Coeff = 1,61*	$\varpi(n) = \frac{\left\lvert I_0\left(\beta\sqrt{1-\left(\frac{2n-N+1}{N-1}\right)^2}\right)\right\rvert}{\lvert I_0(\beta)\rvert}$
Blackman-Harris window	*RMS Coeff = 1,585*	$\varpi(n) = 0{,}42 - 0{,}5\cos(2\pi n/(N-1)) + 0{,}8\cos(4\pi n/(N-1))$

More precisely, we can eliminate the “leakage” effect by a numerical method based on the processing of the complex DFT results. In [12], a suggestion was made that the frequency m, the phase φ, and the amplitude A of the original signal from the values of two maximum harmonics in the spectrum X_k, X_{k+1} should be specified. For this it is proposed to solve numerically the system of complex equations. To do so, the system of complex equations is proposed to be solved numerically:

$$\left\{\begin{array}{l} \lvert E(m,\phi)_k / E(m,\phi)_{k+1}\rvert = \lvert X_k / X_{k+1}\rvert \\ \operatorname{Arg}(E(m,\phi)_k) = \operatorname{Arg}(X_k) \end{array}\right\} \tag{1}$$

where the parameters of the k-th harmonic are specified as:

$$X_k = \mathrm{Re}_k + j\,\mathrm{Im}_k\,;$$

$$X_k = NA_k e^{j\varphi_k},$$

$$A_k = \frac{1}{N}\sqrt{\mathrm{Re}^2{}_k + \mathrm{Im}^2{}_k}\,,$$

$$\phi_k = \mathrm{arctg}\left(\frac{\mathrm{Im}_k}{\mathrm{Re}_k}\right) = \operatorname{Arg}(X_k)\,.$$

The harmonic coefficients can be represented in the form $X_k = (A_k / 2)E(m,\phi)_k$, where $E(m,\phi)_k$ is a complex function independent of the amplitude, but dependent on the frequency and phase:

$$E(m,\phi)_k = e^{j\phi} \frac{e^{2\pi j(m-k)} - 1}{e^{\frac{2\pi j(m-k)}{N}} - 1} + e^{-j\phi} \frac{e^{-2\pi j(m+k)} - 1}{e^{\frac{-2\pi j(m+k)}{N}} - 1}$$

The system of equations (1) must be solved in the case where the harmonics to the left and right of the central one are not equal to zero (in practice it is more than a given small value δ):

$$X_{k-1} > \delta, X_{k+1} > \delta .$$

If $X_{k-1} = 0, X_{k+1} = 0$, then the leakage effect is absent and the frequency, amplitude and phase of the central harmonica correspond to parameters of the measured initial signal See Fig. 4-2d.

When solving the system (1) for the situation of strong leakage effects (σ ~ 0,5 See Fig. 4-2c), only five full iterations were required to provide a specified error of less than 0.5% in frequency and phase. For a sinusoidal signal, the amplitude and frequency are recovered to the value specified in the original signal with accuracy to 5 decimal places. In this case, the amplitude of the central harmonic in the spectrum after the DFT before the recovery procedure was with an error of 35% (!) See Fig. 4-2c [10, 11]

An error in estimating the frequency of the original signal with respect to the frequency of the central harmonic can also be significant. It depends on the frequency of the ADC and the frequency of the original signal. As the frequency of the ADC increases, the error in estimating the frequency will decrease.

The solution of the system (1) is not associated with additional memory as is the case for the fast Fourier transform (FFT). Despite the iterative numerical solution for system (1), such procedure only very slightly increases the overall computation time, and make it possible to obtain not only the spectrum of the signal, but also the restored value of the fundamental frequency, amplitude and phase of the measured signal, when it is close to sinusoidal.

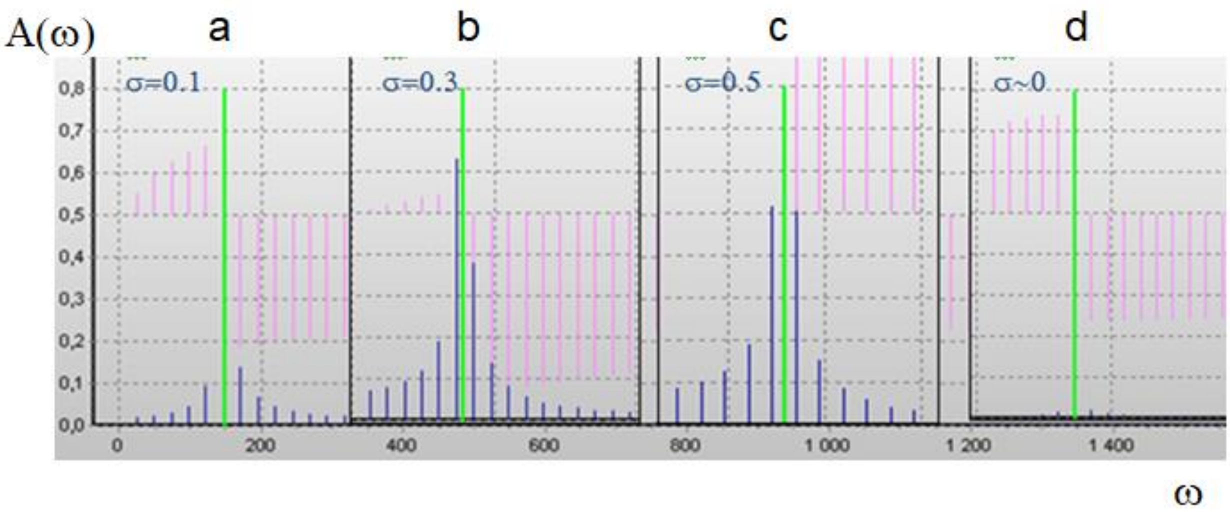


Figure 4 - 2: DFT leakage effect eliminating [10, 11]

The solution of system (1) is not connected with allocating additional memory for storing volumetric data sets and computed coefficients, as in case of fast Fourier transform (FFT). In this regard, the algorithm can be programmed on a modern DSP controller which implements the FFT.

Despite the iterative numerical solution (1), such a recovery procedure increases the total calculation time very slightly and it allows recovery of not only the signal spectrum, but also the restored value of the fundamental frequency and amplitude and phase of the measured signal, if it is close to sinusoidal.

This method was investigated in case of noise in the original signal (with a white noise of 5% and 10% of the amplitude of the sinusoid). Fig. 4-2 shows the solution of the system (1) for a sinusoid with an amplitude of 0,8 and for the cases a) $\sigma = 0{,}1$, b) $\sigma = 0{,}3$ c) $\sigma = 0{,}5$ and c) $\sigma = 0$. The central green line in each figure 4-2 a, b, c, d is the main harmonic of a sinusoid with amplitude of 0,8 with the restored amplitude, frequency and phase, being a result of solving the system of equations (1).

For all the cases, not more than 5 complete iterations were required to ensure a given accuracy. As a result of the solution of the system (1), the phase and frequency of the signal with the addition of white noise to 10%, are restored to the initial value with an error of not more than 0.5%.

5.0 VIBRO-ACOUSTIC DIAGNOSTICS OF TURBOCHARGER

Turbochargers are an integral part of most modern marine diesel engines. Modern turbochargers provide a high value of charge air pressure (π_k up to 5) and highly efficient operation of marine diesel engines with low emissions of carbon oxides and soot [11].

High efficiency of MAN ME and MAN MC diesel engines (with a real specific effective flow rate of 160-170 g/kWh) is provided by the high charge air pressure, in particular. When the efficiency of the turbocharger decreases, the efficiency of the diesel engine drops abruptly, the level of emission of carbon oxides and soot increases.

During operation of marine diesel engines, the exhaust manifolds become clogged with products of incomplete combustion. As a result, the throughput of the exhaust manifolds and the nature of the internal flow of gases before the blades of the turbocharger impeller may vary. In this case, the appearance of pulsations is possible which leads to rotor oscillation. The increased level of rotor oscillation creates additional loads on the turbocharger bearings and reduces their life. In the event of microdefects in the bearings of the turbocharger, the vibration level of the rotor increases even further that may lead to a severe accident.

Constant operational monitoring of the vibration level of the turbocharger rotor can prevent an emergency situation.

The experiments on diesel engines in laboratory and in sea conditions have revealed that the turbocharger compressor blades generate oscillations which are always present in the overall vibration spectrum, regardless of the technical condition of the turbocharger. The spectral analysis of the turbocharger vibration has shown that the compressor blades generate a vibroacoustic signal with a frequency equal to the speed of the turbocharger rotor multiplied by the number of air blades [11, 12],

$$\upsilon_b = n_b \times RPMtur / 60,$$

where υ_b - blade frequency of the turbocharger compressor, Hz; n_b - the number of compressor air blades, *RPMtur* - the speed of the turbocharger rotor min^{-1}.

To determine the blade frequency of the turbocharger compressor and the subsequent calculation of the turbocharger speed, the amplitude spectrum of vibroacoustic signals was used. The recording was made opposite the compressor air filter (see Figure 5-1) through the use of a broadband industrial microphone with a frequency bandwidth of 10 Hz - 20 kHz.

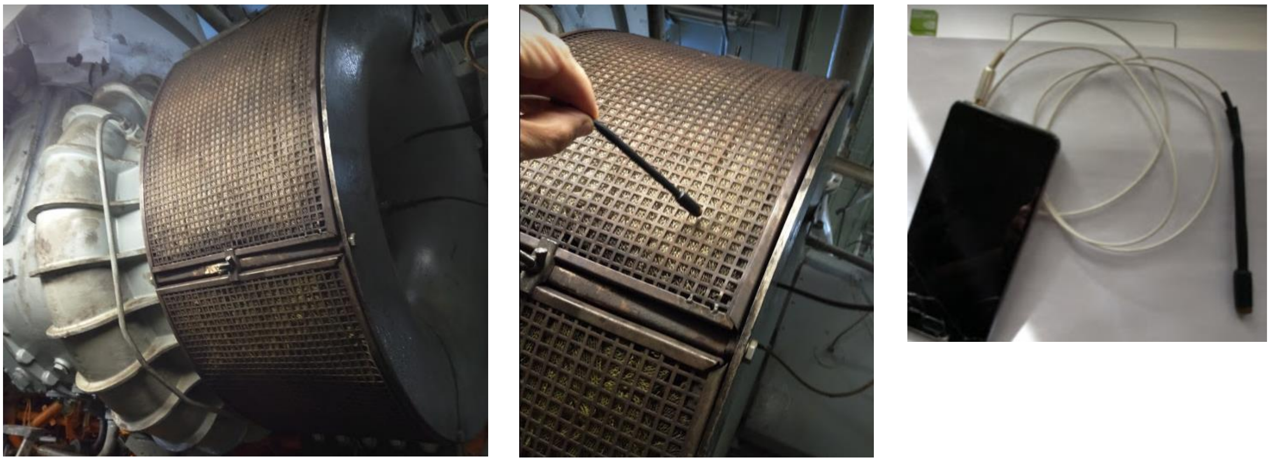


Figure 5 - 1: Recording the vibration of turbocharger through the use of a broadband microphone

In the case of recording vibration of the turbocharger of diesel engine 6L80MCE with the TURBOCHARGER VTR 564-31 (20 compressor blades), the expected blade frequency of the compressor was calculated on the basis of the turbocharger rotor speed rating at the nominal conditions:

$$\upsilon_b = 20 blades \times 9000\ rpm / 60 = 3\ kHz.$$

Due to the fact that the operating mode of the diesel engine was at a lower load, the expected speed of the turbocharger rotor shall be less than the nominal one. Thus, the value of the blade frequency calculated for the nominal conditions can be used as the upper limit for determining the actual operational value.

Figure 5-2 shows the vibration spectrum of the TURBOCHARGER VTR 564-31 recorded at a load close to the nominal one. It can be seen from Figure 5-2 that the harmonic closest to 3 kHz has a frequency of 2948 Hz. The nearest harmonic on the left has a frequency of 1474 Hz and is a subharmonic with a frequency equal to half of the blade frequency $\upsilon_b / 2$. This leftmost subharmonic in the spectrum can be considered as the left boundary when determining of the harmonic corresponding to the blade frequency of the turbocharger compressor.

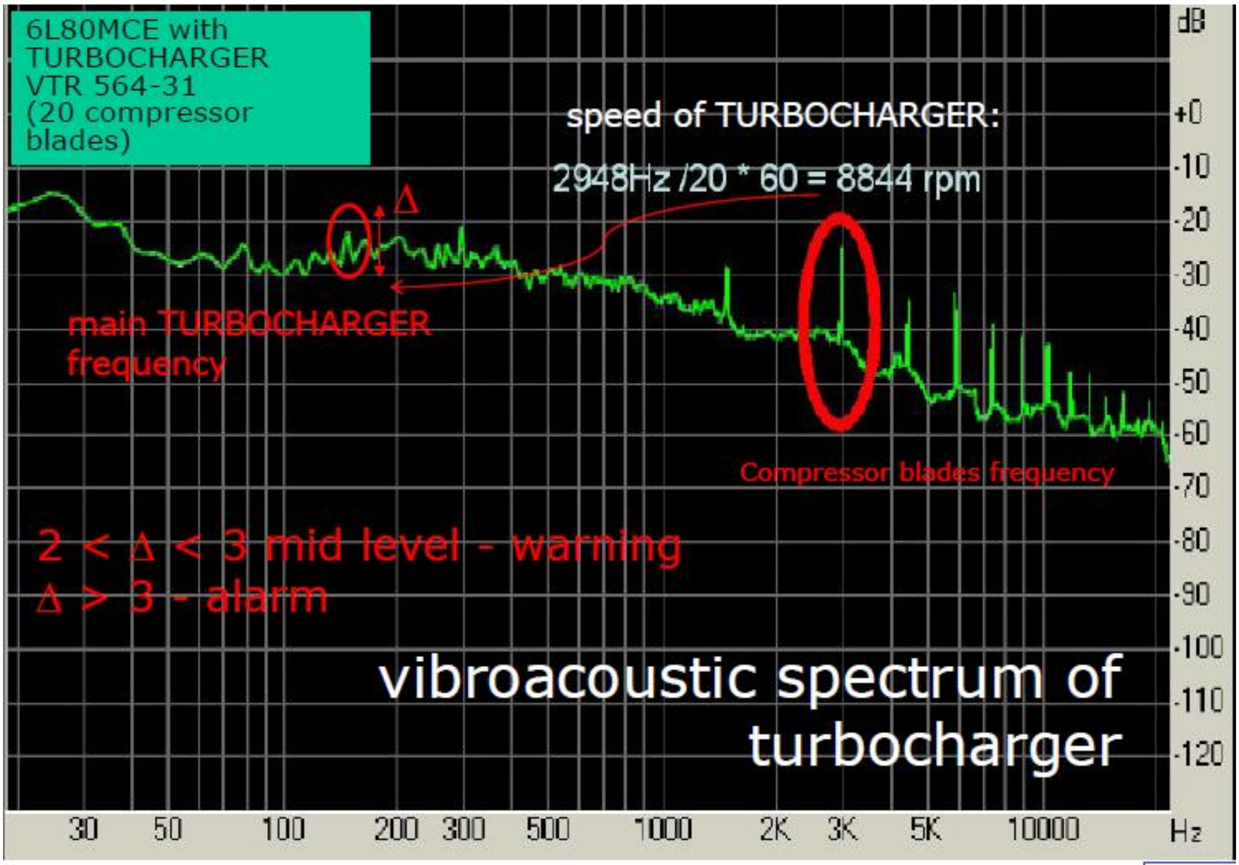


Figure 5 - 2: Vibroacoustic spectrum of turbocharger VTR 564-31

Thus, according to the blade frequency of the VTR 564-31 turbocharger compressor determined in the spectrum in the operational conditions, we calculate the speed of the turbocharger rotor:

$$RPMtur = 60 \times \upsilon_b / n_b,$$
$$RPMtur = 60 \times 2948 \text{ Hz } / 20 = 8844 \; RPM,$$

The regular tachometer of the turbocharger showed a rotation speed of 8800 RPM, which in comparison with the value determined by the spectrum gives a relative error of 0,5%.

It is necessary to take into account the industrial accuracy class of the standard tachometer (division scale of 200 RPM).

Spectral analysis of a vibroacoustic signal recorded at a frequency of 44,1 kHz makes it possible to analyze harmonics in steps of less than 1 Hz at a recorded signal frequency up to 20 kHz [12]. The blade frequency of the turbocharger compressor is significantly lower. Thus, as a result of the spectral analysis of the vibroacoustic signal of the turbocharger compressor, an error in determining the frequency less than 1 RPM can be reasonably obtained. Such accuracy is much higher than the accuracy of the standard tachometers, which makes it possible to use the blade frequency of the turbocharger compressor in accurate calculations of the main rotational speed of the turbocharger and the subsequent estimation of the diesel engine power.

After determining the compressor blade frequency and the main speed of the turbocharger (RPMtur), we can analyze the harmonic amplitude at the main speed of the rotor.

$$\upsilon_{turbocharger} = \upsilon_b / n_b$$

In the case shown in Fig. 5 - 2

$$\upsilon_{turbocharger} = 2948 \text{ Hz } / 20 = 147{,}4 \text{ Hz}$$

We eliminate the "leakage effect" for the harmonic at the fundamental frequency υ of the turbocharger, using the algorithm described in paragraph 4, solving the system of equations (1). After recovering the amplitude of the υ turbocharger , we analyze it.

Obviously, if there is a significant increase in the amplitude Δ of the harmonic at the main speed of the turbocharger rotor, this demonstrates an increased vibration of the rotor. Fig. 5 - 2 shows a slight increase in the amplitude of the fundamental harmonic Δ, which characterizes the permissible vibration level of the turbocharger rotor.

Preliminary experiments on MAN MC diesel engines have shown that an increase in the amplitude of the harmonic at the main frequency υ turbocharger in 2-3 times regarding the average level of the amplitude spectrum characterizes the dangerous vibration level of the turbocharger rotor. The average level of the harmonic amplitudes was estimated in the frequency range

$$[\upsilon_{turbocharger} - 50 \text{ Hz} .. \upsilon_{turbocharger} + 50 \text{ Hz}]$$

To better quantify the limits of vibration level of the turbocharger rotor, further research is required. It may be noted that the spectrum analysis of vibroacoustic signals of the turbocharger compressor can be made quickly under operating conditions.

6.0 CONCLUSIONS

The methods of vibrodiagnostics of marine diesel engines with turbocharging considered in the article can be helpful for practical use. The "leakage effect" method improves the reliability of diagnostic findings.

A vibroacoustic method for determining the speed of a turbocharger rotor and estimating the level of the oscillation amplitude at the main rotational speed can be used as a basis for the express diagnostics of the turbocharger under operating conditions.

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