Trending of Vibration Spectrum in Marine Diesel Engine, Marine Gear Box and Variations with Rotational Speeds

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Abstract - *Noise and vibration on board ships may cause discomfort to passengers and crew. It may also impair the efficient execution of the crew's duties, be the cause of damage to sensitive equipment, structural parts of the ship and cargo, and even compromise the safety of the vessel. In military scenario, this may influence the combat readiness of the vessel too. Nowadays, people on board are less willing to accept discomfort due to noise and vibration, leading to increasingly strict requirements. As these are usually not easily met, noise and vibration have become important factors in ship design. Noise and vibration levels are determined by the characteristics of source, transmission and receptor. Low frequency noise and vibration are difficult to damp and addition of mass and stiffness in the 'remedial' design stage is costly, and an effort multiplier. Therefore, noise and vibration problems must be avoided through identification and treatment of the major sources as and when found.*

It is appropriate that the principal vibration exciting sources be addressed first, since with high excitation levels excessive vibration can occur almost independently of system structural characteristics. In general, the major sources are the high-speed diesel main engine and the propeller (impeller). Hull excitations are generally considered to give less excitation than diesel engines. Thus, in this section, attention is paid to the excitation of the high speed main diesel engine.

The main source of vibration generator is the propulsion system for a ship. Correct identification of its signature and spectrum with the trending behaviour is a better method to predict whole ship's vibration levels. In order to identify the same, a fast attack craft fitted with 02 Nos 2000 hp each engines were earmarked to conduct trials. Sea trials at following conditions had been carried out.
Sea State : 1-2, 2-3;
Loading Condition : Full Load, Half Load;
Wind Condition : Moderate< 15 knots;
Sea Direction : Ahead, Astern

Key words: **vibration spectrum, vibration analyser, marine engine, thrust bearings**

I. INTRODUCTION

Prediction and diagnostics of rotating machinery using the envelope spectrum of high frequency vibration exited by internal forces of the engines is becoming widely used by many experts. In recent years, automatic diagnostic systems based on this method have been developed mainly for rolling element bearings. These systems can provide detailed diagnostics and condition prediction of a bearing by a single vibration measurement. Automatic diagnostic systems for simultaneous diagnostics of gears and rolling element bearings in gear transmissions have also been developed. But until recently, there were no diagnostic systems that used envelope spectra of random vibration for detection and identification of defects in fluid film/ thrust bearings. The typical use of standard enveloping methods developed for the diagnostics of rolling element bearings in the diagnostics of thrust bearings are discussed in this paper. Methods for comparison of rolling element results with thrust bearings of a marine diesel engine and rolling and gear elements of a marine gear box are analyzed together with the practical results achieved in this field. The results that are discussed in this paper were practical outcomes of a vessel whilst patrolling.

The aim of this study was to identify and differentiate the spectrum patterns generated from rolling element and thrust bearings of marine engines and gear boxes to identify an optimum maintenance schedule in order to minimise the risks of failures originated from vibration defects.

II. METHODOLOGY

Vibrations of the machineries of the craft are analyzed using Fast Fourier Transform (FFT) algorithm in two frequency bands, the first from 0 to 100 Hz, and the second from 100 Hz to 10 kHz. Taking into account the overall machinery readings, low and medium frequency ranges of the vibrations are of particular interest, and therefore the limits of the above frequency bands are defined. Particular interest is in the low frequency band, which is defined from zero to 100 Hz. Vibrations – displacement, velocity and accelerations were measured during the movement of the vessel in the measuring points defined.

A. Vibration Analyser and Software

The trial team used 02 Nos vibration data analysers (Frequency Range: 0 – 40,000 Hz) integrated with supported software system. The averaging used is 5 times and FFT uses Hanning window method for data filtering. One analyser is having two channels and the sampling frequency for each channel is 102.4 kHz.

B. Measurement Procedure(As per ISO 6954: 2000)

a. Measurements were recorded in all three directions at a minimum of two locations on each deck. At other locations, measurements are only required in the vertical direction.
b. The combined frequency weighting curve according to ISO 2631-2 was applied to all measurements irrespective of their direction.
c. The frequency range evaluated was 1 Hz to 10000 Hz. (Analysed separately in low, medium and high frequency ranges)
d. The measurement duration was above 1 min. for all machinery locations. For hull locations measurement duration of at least 2 min is required.
e. The result of each measurement shall be the overall frequency-weighted r.m.s. value.

Measurements were recorded in radial direction at three locations near the crank shaft.

C. Locations for Data Recording

Table 1.Locations onboard Ship

Sr No	Location	Direction
Machinery Locations		
1	Gear Box Free End (G/B F/E)	Vertical Horizontal Axial
2	Gear Box Drive End (G/B D/E)	Vertical Horizontal Axial
3	Main Engine Free End (M/E F/E)	Vertical Horizontal Axial
4	Main Engine Drive End (M/E D/E)	Vertical Horizontal Axial

D. Engine Specifications

Cylinder arrangement	: 60 deg V
No. Of Cylinders	: 16
Stroke	: 160 mm
Bore	: 132 mm
Cylinder capacity	: 2.19 dm3
Displacement (Total)	: 35.04 dm3
Turbo charged	
Compression Ratio	: 15:1
HP	: 1200 at 1500 RPM
KW	: 1050 KW
Max RPM	: 1800

E. Vessel Particulars

Length overall	-	24.05 m
Breadth, moulded (max)	-	5.7 m
Draught Max	-	1.45 m
Half load	-	1.3 m
Minimum	-	1.15 m
Displacement at full load	-	59.65 Ton
Max. Speed at 100% engine	-	50 kn (knots)
Class	-	Fast Attack Craft

III. CONDUCT OF SEA TRIALS AND SALIENT RESULTS

The trial team focused on recording data at two fixed engine RPMs and analysing the spectrums resulted from earmarked machinery locations.

A. Main Machinery Vibration Spectra at 800 rpm

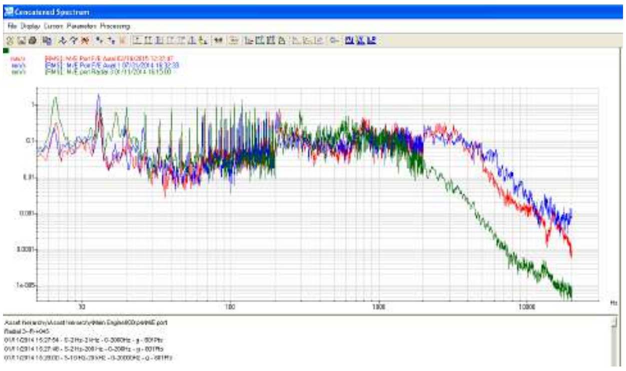

Figure 1 : Port Main Engine Free End Readings

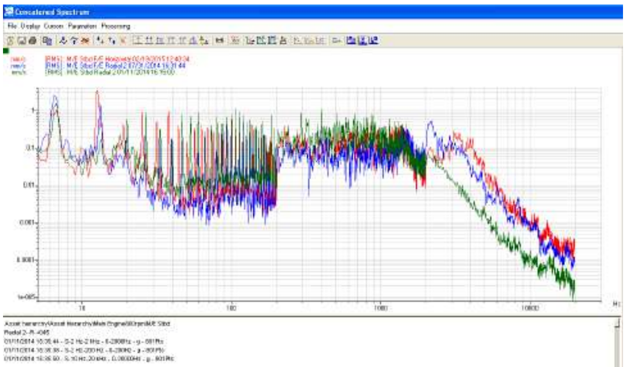

Figure 2 : Stbd Main Engine Free End Readings

B. Marine Gear Box Spectra at respective 800 rpm

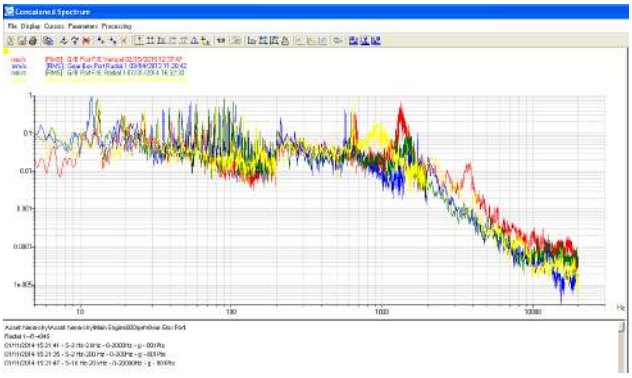

Figure 3: Port Gear Box Free End Readings

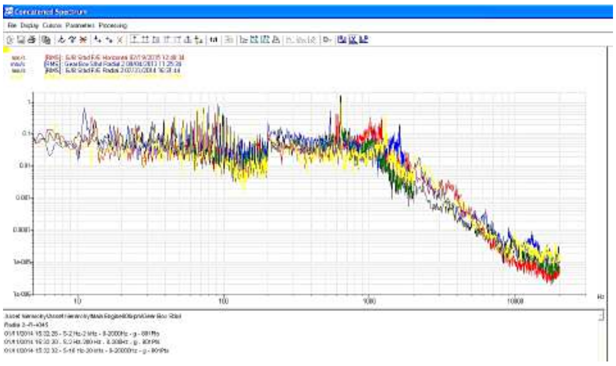

Figure 4: Stbd Gear Box Free End Readings

The method of rolling element bearings predictions by the spectrum of high frequency vibration envelope is based on the analysis of characteristics in the formation of friction forces in good and defective bearings as well as in the features of shock pulses that appear in the interaction of rolling surfaces with cavities, spalls, or cracks in the bearing elements.

The vibration spectra on marine engine and connected gear box were separately recorded in order to compare the differences between rolling elements and thrust bearings in the aspects of both numerical and graphical comparison. It was displayed from the spectrum patterns that 10 -1000 Hz the spectrum covers dominant crank shaft vibration signatures in case of engine and drive gear signature in case of gear box. The friction forces depend on the rolling friction coefficient and the load on rolling elements. In good bearings, the friction is uniform in time, i.e. it does not depend on the rotation angle of the rotating race or of the cage. As a result, the friction forces, together with the random vibration excited by them, become amplitude modulated. In bearings with non-uniform wear of inner and outer races and rolling elements, the friction coefficient in turn depends on the rotation angle of rotating race and cage which results in similar amplitude modulation of the friction forces and the resultant high frequency vibration. Finally, shock pulses in bearings with cavities and cracks on rolling surfaces and races produce vibration as well. On the resonant frequencies of rolling elements and races, this vibration is actually attenuated self-oscillations that should not be considered as random vibration. At other frequencies, shock pulses excite random, fast attenuated vibration that is also modulated in amplitude.

The resulted spectrum patterns of gear boxes (at 800 rpm) on four days i.e. trending pattern (04 Sep 2013, 11 Jan 2014, 31 Jul 2014, 19 Feb 2015) had seen not much of a deviation from the standard pattern except for few isolated valleys and peaks. This had been a healthier sign of a marine gear box with a demonstration of trouble free operation more than two years of continuous operation.

IV. COMPARISION OF RESULTS

High frequency random vibration is excited by friction forces in both rolling element and thrust bearings. When defects of friction surfaces develop in these types of bearings, the friction forces and high frequency vibration acquire amplitude modulation and thus bearing defects can be detected by the analysis of envelope spectrum of this vibration. In the same situation, in the case of thrust bearings, there are much more problems in the detection and identification of defects than in rolling element bearings. The first problem is connected with the limited diagnostic information that can be derived from the modulation frequencies. Rolling element bearings have at least 4 types of friction surfaces with different rotation speeds. These are outer and inner races, rolling elements, and the cage.

In case of thrust bearings, there are only two friction surfaces and one fundamental frequency. The second problem is due to the characteristics of pressure pulsation formation in the lubrication layer and the vibration excited by this pulsation. The pulsation power is determined by the velocity gradient in the lubrication layer, but it increases not only with rotation speed, but also with the decrease of lubrication layer thickness. The thickness of the lubrication layer, in turn, depends on the bearing design and on the relative position of the shaft axis and bearing shell axis.

In a number of bearings, the shaft axis moves and, as a result, the high frequency vibration may be modulated even in good bearings. The third problem occurs in the detection of the defect severity. In the case of thrust bearings, not only the modulation index in the envelope spectrum, to be considered, but the thickness of the lubrication layer (and bearing thickness) as well. Reliable information about the thickness of the lubrication layer, as a rule, is not available for a user, not generally provided by manufacturer, thus the levels for defects have to be adapted for every machine type that has some special design characteristics of the bearings or of the machine. Following many years of investigations and practical diagnostics of bearings in rotating machines using the envelope spectrum of high frequency vibration, it was concluded that the above problems are difficult to be solved in practice.

A number of diagnostic symptoms were found which allow successful condition diagnostics of fluid film bearings. These are based on the characteristics of shaft journal oscillations in the lubrication layer of the fluid film bearing. Consider three of these characteristics. The first characteristic is the possible appearance of short pulses with increased oil pulsation during the movement of oil wedge on the bearing shell surface. The rotation of the oil wedge is an indication of, first of all, of shaft wobbling. In a good bearing, such a pulse can appear when the oil wedge passes joints of the shell sectors. In the case of a worn bearing, when the oil wedge passes non-uniform wear zones on the bearing shell, cracks, and so forth, such short pulses can be considered as shock pulses in fluid film bearings analogous to the shock pulses in rolling element bearings. The second characteristic is possible appearance of shaft vibration at frequencies different from the harmonics of rotation speed. Most often, this is a self-sustained oscillation of the shaft in bearings with loose clearances or defects in the oil supply system. In most practical cases, self-sustained oscillations of the shaft synchronize with one of the sub-harmonics of the rotation speed.

Sometimes it is observed that the pendulum shaft oscillation in very loose bearings which also, as a rule, synchronize with one of rotation speed sub-harmonics. And the third characteristic is the appearance of low frequency random oscillations of the shaft relative to the bearing shell surface. This situation can be found in the bearings with non-uniformly worn shells. These oscillations are defined by the unstable shape of the oil wedge with small and random changes. This can be detected by changes in the shape of the envelope spectrum and background level increases on frequencies below the shaft rotation speed. Such increases of background level in the envelope spectrum should be considered as an effective symptom of the bearing shell wear. The use of the above diagnostic symptoms allow the thrust bearings diagnostics without needing a relative displacement transducer installation in the bearing unit.

The experimental concern of the study is to comply with the periodicity of measurements. The intervals between measurements can be rather long, about a few months when the service life of the bearing is 5 to 8 years and when the machine is operated in its standard modes If the above conditions are met, it is enough to make 10 to 20 measurements of the high frequency vibration envelope spectra during the whole service life of a bearing to eliminate its un-predicted failure. The levels for defects that are used for the estimation of the detected defect severity in rolling element bearings can be similar for all types of machines and bearings. They monotonously increase with the increase of rotation speed and bearing dimensions.

A number of different problems occur in the diagnostics of thrust bearings using an envelope spectrum of high frequency vibration. These problems are solved by the added analysis of diagnostic information from the auto spectra of bearing housing vibration. The first problem is connected to the design characteristics of some machines where the shaft motion oscillates in the stationary bearing shell, even with no defects present. In this case, an envelope spectrum at this bearing will contain harmonic components proportional to the rotation frequency in spite of the absence of defect.

The vibration measurements, in turn, should be done more often, than rolling elements. The second problem is closely connected with the variety of fluid film bearings designs, each of which has its own lubrication layer thickness. For this reason, similar defects in the bearings with different lubrication layer thickness lead to different changes in the normalized thickness and thus, different modulation of random vibration. The best solution is to correct levels for defects according to the data derived from the analysis of autospectra, i.e. the increase of vibration components after the detection of severe defects in similar bearings of other engines. For the reasons discussed above which are different from the rolling element bearing case, fluid film bearings are best diagnosed by the parallel analysis of autospectra and envelope spectra of the bearing shield vibration.

V. CONCLUSION

Comparative analysis of possibilities for rolling element and fluid film bearings (in marine gear boxes and engines respectively) diagnostics by the analysis of high frequency vibration envelope spectra presents the following conclusions:

(I) Vibration diagnostics is the most efficient method for detection, identification and differentiation of incipient defects in rolling element and thrust bearings. Analysis and predictions on rolling element bearings are comparatively easier than thrust bearings. However, it is recommended to use same analysis technique for both the instances.

(II) The typical interval between measurements for rolling element bearings in absence of detected defects is several months (6 - 12 months) and for thrust bearings is less than that. The levels for the detection of defects in rolling element bearings and estimation of their severity are nearly independent on the machine/ engine and bearing design. Only some small dependence of levels for defects on the rotation speed and bearing dimensions exists. For machines with thrust bearings, the levels for defects also depend on the bearing design.

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