A Diagnostic Approach to the Vibration Measurements and Theoretical Analysis of a Dredger Propulsor System

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Fault finding through vibration and noise study is more potential non-intrusive tool in the contemporary days. For any machinery as the time progresses and various components are either replaced or modified, the stiffness and dampers vary, there is a need for theoretically analyzing the entire system time to time for a comprehensive analysis of vibrations. A diagnostic analysis has been carried out on a dredger which is propelled by twin propeller system powered by two diesel engines for different cruising speeds of the dredger to critically analyze the possible defects which may surface at any of the operating speeds with dredger loaded and unloaded conditions. The theoretical analysis of the diagnostic problem was done by considering the dredger propulsive system consisting of four discrete masses with springs and dampers. The relevant frequencies concerning to every mode of vibration involved are calculated and different parameters of the vibration calculated by theoretical evaluation are compared with the practically measured values by plotting the graphs and the velocity spectrums.

Keywords: Dredger; Vibration; Modelling; Propulsion; Diagnosis

NOTATION

g : acceleration due to gravity

k_e : radius of gyration of connecting rod

P : gas pressure force

P : R/I ratio

R : crank radius

r : distance of centre of piston pin bearing on rod to its centre of gravity

W_E : weight of the piston

W_2 : weight of the connecting rod

W_3 : weight of crank effective at its radius

x : displacement of piston

θ : crank angle

φ : connecting rod angle

ω : angular velocity of crank

INTRODUCTION

Vibration has traditionally been associated with trouble in machines-wear malfunction, noise and structural damage. In more recent years, however vibration has been used to save the industry millions of rupees in machine downtime. Sophisticated tools, such as, computer software, measuring equipment and analytical device, have been developed and are applied to provide more detailed and accurate analysis of complex vibration problems in the name of predictive maintenance. The development of new engine generations for minimum fuel consumption involving greater stroke/bore ratios, lower running speeds and higher combustion pressure, for lower installation and operating costs as well as for higher reliability and a very large spectrum of different shaft line arrangements largely influences the vibration analysis of a modern ship installation[1].

Vibration is the subject of continuous research in terms of developing new computer software, analysis methods and measuring techniques for dealing with complex vibration problem.

Based on long experience, the vibration of ship machinery is well understood and can be predicted with sufficient accuracy. This confidence in the calculations is needed to be sure that the vibration levels will remain below the corresponding admissible limits. It is important that vibration aspects are considered at the earliest stage possible in the ship design process to find a cost-effective installation with the best combination of countermeasures against vibration. It also calls for close collaboration between the shipyard, the engine and propeller suppliers, and other suppliers.

Although vibration behaviour is now well understood, there will thus always be potential for further advances in the state-of-the-art. Further developments of analytical techniques are in progress and experience is continually being accumulated.

MATHEMATICAL EVALUATION

The first important way to avoid vibrations is to prevent resonance conditions. The procedure is successful as long as the natural frequencies and excitation frequencies can be regarded as being independent of environmental conditions. In questions of ship technology, this pre-requisite frequently

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remains unfulfilled. If the stiffness and mass matrices are known then natural vibration calculations can be performed. For this, numerically effective approximation methods are used. Once the criticality conditions are set, then the system responses due to various excitation sources will be considered. Present work is related to the vibration analysis of a dredger's propulsor system, which comprises multi-cylinder engine, gearbox, thrust block and propeller. This system is considered as four degree-of-freedom system in calculating the natural frequencies at four modes. The vibration parameters at various lumped masses are calculated at various orders of the rotating frequency.

Different Excitation Sources

In ship technology, with the exception of special problems, (*eg*, impact excitation), periodically varying excitation forces are of interest[2].

- Main engine and auxiliary machinery: excitation frequencies are half and/or whole multiples of the frequencies of revolution.
- Shaft machinery: excitation frequencies are equal to the frequencies of revolution.
- Compressors: excitation frequencies are equal to the frequencies of revolution and to twice the frequency.
- Gear boxes: excitation frequencies are equal to the frequencies of revolution and meshing.
- Propellers: excitation frequencies are equal to the frequencies of blade and its multiples.

But the main sources of excitations of concern are the excitation effects stemming form the main engine and propeller. But in the present studies, only the main sources of excitations are dealt with, *ie*, the excitation effects stemming form the main engine and propeller (Figure 1).

Main Engine

Basically, the main engine of a ship introduces excitation forces into the foundation at all frequencies that are half and/ or whole multiples of the frequency of revolutions.

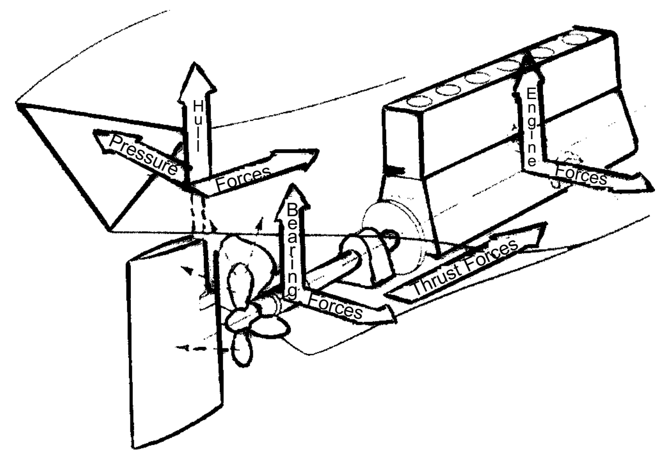

Figure 1 Main excitation sources

For the calculation of excitation forces generated by slow running main ships, forces occurring within one cylinder are taken as the starting point and the forcing functions like the inertia force of the reciprocating parts, inertia forces of crank and other parts, the inertia couple due to the angular acceleration of the connecting rod and the reaction to the torques acting on the crankshaft. These forcing functions are transformed into the frequency domain by means of a Fourier analysis. For considering the vibrations of the crankshaft or shaft line, the harmonic components of the tangential and radial forces are applied as the sources of excitation. For engine housing vibrations vertical and transverse vibrations are to be considered.

If the forces acting in a single cylinder unit are known for the individual orders, their phase relationship with other cylinder units can be calculated, considering the ignition sequence and the total excitation forces are calculated. Another advantage of this procedure lies in the simple method of taking account or irregular ignition sequences and in the possibility of simulating ignition failures, for instance.

Propeller

From the propeller, excitation forces are transmitted into the ship via the shaft line in the form of pressure pulses acting on the ship shell. These thrust forces are the significant factors for the vibration of shaft lines, the predominant factor for vibration of ship structures are pressure fluctuations. For computation of axial and torsional vibrations of the shaft line, fluctuations in the thrust are taken into account.

The forcing functions of engine are:

Reciprocating inertia force	: $F = -W_E\,\ddot{x}/g$
Rotating inertia force	: $F_R = W_3\,R\omega^2$
Residual couple	: $M_R = -W_2\,r\ddot{\varphi}(k_e^2 - 1)$
Gas pressure torque	: $M_P = P(\dot{x}/\omega)$
Reciprocating inertia torque	: $M_E = -W_E\,\ddot{x}(\dot{x}/\omega)/g$
Correcting couple	: $M_M = M_R\left[p\sec\varphi\cos\theta\right]$

To express the forces and couples acting on the engine frame in a series of harmonic terms, it is necessary to express velocity, acceleration and connecting rod angle in terms of crank angle.

The inertia forcing functions expressed in terms of θ are

$$F = -(W_E/g)R\omega^2\,\Sigma A_n\cos n\theta$$

$$M_R = -(W_2/g)R\omega^2\left(r - k_e^2/l\right)\Sigma E_n\sin n\theta\,(n\,\text{odd})$$

$$M_E = -(W_E/g)R^2\,\omega^2\,\Sigma\sin n\theta$$

$$M_M = (W_2/g)R\omega^2\left(r - k_e^2/l\right)\Sigma\sin n\theta$$

$$F_1 = -(W_E/g\,x\,p)R\omega^2\,\Sigma A_n\cos n\theta$$

In-line Engines

For engines with cylinders arranged in-line, the following design features are normally adopted in order to balance the engine as well as possible:

(1) uniform firing intervals,

(2) identical moving parts in each cylinder,

(3) an arrangement of cranks along the shaft such that the rear half of the crankshaft as a mirror image of the front half.

This arrangement of cranks provides the same degree of balance of moments and also the forces.

For the four-stroke cycle engines, the vector summation of harmonics for N identical cylinders, firing at equal intervals results in an expression of the form

$$A_n = \sum_{i=0}^{i=N-1} \cos\left(\frac{n\theta + 4\pi ni}{N}\right)$$

where, A_n is the amplitude of the nth harmonic, $4\pi / N$ is the firing interval, θ is the crankshaft angle measured from the line of stroke, and i is an integer enumerating the cylinders from the reference cylinder ($i = 0$) to the ($N - 1$) cylinder. The sum of such series is always zero unless $2n$ is a multiple of N. The same result can be obtained from a series of sine terms. A series proportional to gas pressure torque is of the form

$$M_{Pn} \, \Sigma \sin\left(\frac{n\theta / 2 + 4\pi ni}{\cos N}\right)$$

The multi cylinder, in-line, four stroke cycle engines, firing at equal intervals and with lines of stroke parallel to are unbalanced with those harmonics whose order is $n/2$ whenever n is an integral multiple of the number of cylinders N.

Theoretically, there are always some harmonics which are unbalanced in any in line engines.

VIBRATION RESPONSE

The dynamic response of the shafting may conveniently be divided into longitudinal vibrations, torsional vibrations and lateral vibrations.

Longitudinal Vibrations

Shafting longitudinal/axial vibrations are characterized by shafting segments oscillation in afore — and — aft direction around some neutral position. They are mainly excited by the propeller thrust variations and forces generated by the engine's crank mechanism[3]. The longitudinal vibrations in the line shafting may

- excite structural vibration of the engine room, double bottom and other local structures in the engine room as well as the local and global vibration of the superstructure through the thrust bearing,
- excite the propulsion machinery itself, *ie,* engine, reduction gear, shafting components.

Torsional Vibrations

Shafting torsional vibrations are characterized by shafting variable speed of rotation. In contrast to other, easily visible or perceptible kind of vibrations like axial or lateral vibrations, shafting is invisible. However, they are significant and a cause of serious vibration damages, even fractures of shafting. These vibrations are caused by variable gas pressure in the cylinder of the engine, inertia forces of a crank mechanism and fluctuation of seawater flow around the propeller.

For counteracting these vibrations the most usual measures are by selecting appropriate dimensions and materials, selection of proper turning and tuning wheel and selection of appropriate engine location. The can also be resolved by appropriate operations, *ie,* rapid pass through units or mounting of a torsional vibration damper.

Lateral Vibrations

Shafting lateral vibrations are characterized by shafting segments oscillations in a plane passing through the shaft neutral position. They are a special case of more general whirling vibrations.

Lateral vibrations are mainly excited by the propeller weight, shafting segments' weights and unbalance. The amplitudes of lateral vibrations are enlarged by the increased span between the shaft line bearings.

They are significant because they may cause

- additional dynamic stresses in the propeller shaft,
- disturbance of the stern bearing as overheating or wear down,
- dynamic magnification of bearing reactions, being the cause of structural vibration in the after body.

The basic design countermeasure is to ensure that lateral natural frequencies are positioned sufficiently away from the rotational speed.

The axial or lateral vibrations coupled with torsional vibrations makes a greater contribution to severe damage.

Natural Frequencies and Mode Shapes (eigen values and eigen vectors)

When a system requires more than one coordinate to describe its motion, it is called a multi-degrees-of-freedom system or an N-DOF system where N is the number of coordinates required. The N-degrees-of-freedom has N natural frequencies, and for each of the natural frequencies, there corresponds a natural state of vibration with a displacement configuration known as normal mode. Mathematical terms, that correspond to these values are known as eigen values and eigen vectors, respectively.

By considening a multi-degree-of-freedom system and in the matrix form, the differential equation of motion of the system, in fact for any system can be written in the form

$$[M]\{\ddot{x}\} + [K]\{x\} = \{0\}$$

considering $[M][M]^{-1} = [I]$, a unit matrix and $[M]^{-1}[K] = [D]$, a dynamic matrix.

For free body vibrations, having harmonic motion of frequency ω, *ie,*

$$\{x\} = \{X\} \sin \omega t$$

thus

$$\{\ddot{x}\} = -\omega^2 \{x\} = -\lambda \{x\} = -\lambda \{X\} \sin \omega t$$

where $\lambda = \omega^2$ is the eigen value and $\{X\}$ is the column giving the amplitudes of the respective masses, *ie,* eigen vectors.

Thus

$$[[D] - \lambda [I]] \{X\} = \{0\}$$

The determinant formed from the above equation

$$[[D] - \lambda [I]] = 0$$

is the frequency equation and gives n values of $\lambda_i \left(= \omega_1^2\right)$ for n degrees-of-freedom.

By substituting the λ_i in the matrix equation the mode shape $\{X\}_I$ called the eigen vector, for the ith mode of vibration can be obtained. Thus, an n-degrees of freedom system will give n eigen values and corresponding n eigen vectors.

Numerical Integration Methods for Theoretical Analysis of Vibrations

When the differential equations of a vibrating system cannot be integrated in a closed form, a numerical approach can be used.

$$[M]\{\ddot{x}\} + c[\dot{x}] + [K]\{x\} = [F]$$

The following finite difference expansions are employed:

$$\dot{\vec{x}}_{i+1} = \frac{1}{6\Delta t}\left(11\vec{x}_{i+1} - 18\vec{x}_i + 9\vec{x}_{i-1} - 2\vec{x}_{i-2}\right) \quad (1)$$

$$\ddot{\vec{x}}_{i+1} = \frac{1}{(\Delta t)^2}\left(2\vec{x}_{i+1} - 5\vec{x}_i + 4\vec{x}_{i-1} - \vec{x}_{i-2}\right) \quad (2)$$

To derive equations (1) and (2), consider the function $x(t)$. Let the values of x at the equally spaced grid points $t_{i-2} = t_i - 2\Delta t$, $t_{i-1} = t_i - \Delta t$, t_i and $t_{i+1} = t_i + \Delta t$ be given by x_{i-2}, x_{i-1}, x_i and x_{i+1}, respectively, as shown in Figure 2. the Taylor's series expansion, with backward step, gives several possibilities.

- With step size $= \Delta t$:

$$x(t) = x(t+\Delta t) - \Delta t\, \dot{x}(t+\Delta t) + \frac{(\Delta t)^2}{2!}\ddot{x}(t+\Delta t) - \frac{(\Delta t)^3}{3!}\dddot{x}(t+\Delta t)$$

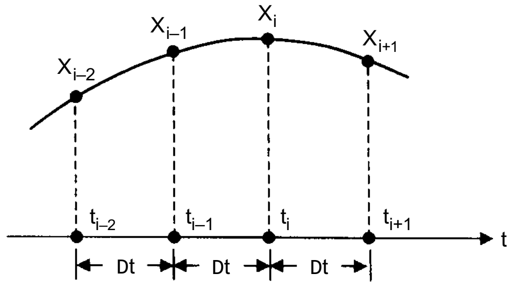

Figure 2 Equally spaced grid points

or

$$x_i = x_{i+1} - \Delta t\, \dot{x}_{i+1} + \frac{(\Delta t)^2}{2}\ddot{x}_{i+1} - \frac{(\Delta t)^3}{6}\dddot{x}_{i+1} + \ldots \quad (3)$$

- With step size $= 2\Delta t$:

$$x(t-\Delta t) = x(t+\Delta t) - (2\Delta t)\dot{x}(t+\Delta t) + \frac{(2\Delta t)^2}{2!}\ddot{x}(t+\Delta t) - \frac{(2\Delta t)^3}{3!}\dddot{x}(t+\Delta t) + \ldots$$

or

$$x_{i-1} = x_{i+1} - 2\Delta t\, \dot{x}_{i+1} + 2(\Delta t)^2 \ddot{x}_{i+1} - \frac{4}{3}(\Delta t)^3 \dddot{x}_{i+1} + \ldots \quad (4)$$

- With step size $= 3\Delta t$:

$$x(t-2\Delta t) = x(t+\Delta t) - (3\Delta t)\dot{x}(t+\Delta t) + \frac{(3\Delta t)^2}{2!}\ddot{x}(t+\Delta t) - \frac{(3\Delta t)^3}{3!}\dddot{x}(t+\Delta t) + \ldots$$

or

$$x_{i-2} = x_{i+1} - 3\Delta t\, \dot{x}_{i+1} + \frac{9}{2}(\Delta t)^2 \ddot{x}_{i+1} - \frac{9}{2}(\Delta t)^3 \dddot{x}_{i+1} + \ldots \quad (5)$$

By considering terms up to $(\Delta t)^3$ only, equations (3) to (5) can be solved to express $\dot{x}_{i+1}$, $\ddot{x}_{i+1}$ and $\dddot{x}_{i+1}$ in terms of x_{i-2}, x_{i-1}, x_i and x_{i+1}.

$$\dot{x}_{i+1} = \frac{1}{6(\Delta t)}\left(11x_{i+1} - 18x_i + 9x_{i-1} - 2x_{i-2}\right) \quad (6)$$

$$\ddot{x}_{i+1} = \frac{1}{(\Delta t)^2}\left(2x_{i+1} - 5x_i + 4x_{i-1} - x_{i-2}\right) \quad (7)$$

Equations (6) and (7) represent the vector form of these equations.

To find the solution at step $i+1\left(\vec{x}_{i+1}\right)$, in the forcing equation,

$$[m]\ddot{\vec{x}}_{i+1}+[c]\dot{\vec{x}}_{i+1}+[k]\vec{x}_{i+1}=\vec{F}_{i+1}\equiv\vec{F}(t=t_{i+1}) \tag{8}$$

By substituting equations (1) and (2) into equation (8), one obtains

$$\left(\frac{2}{(\Delta t)^2}[m]+\frac{11}{6\Delta t}[c]+[k]\right)\vec{x}_{i+1}$$

$$=\vec{F}_{i+1}+\left(\frac{5}{(\Delta t)^2}[m]+\frac{3}{\Delta t}[c]\right)\vec{x}_i$$

$$-\left(\frac{4}{(\Delta t)^2}[m]+\frac{3[c]}{2\Delta t}\right)\vec{x}_{i-1}+\left(\frac{1}{(\Delta t)^2}[m]+\frac{[c]}{3\Delta t}\right)\vec{x}_{i-2} \tag{9}$$

SPECTRUM ANALYSIS

Routine vibration amplitude against frequency analysis techniques applied to reciprocating pumps, compressors and gasoline engines are generally quite effective for diagnosing mechanical problems, such as, rotating unbalance, misalignment, looseness etc. In reciprocating machines also the vibrations resulting from these forces often have frequency characteristics similar to those associated with mechanical problems as above but characterized mainly by the inertia of the reciprocating components plus varying pressure on the pistons.

The vibration frequencies normally encountered can be those at 1 and 2 × rpm due to mechanical problems; however, higher order frequencies are also common with some designs, depending on the number of cylinders and their relationships with others. Apart form these vibration can present at half order harmonics $1\frac{1}{2}$, $2\frac{1}{2}$, $3\frac{1}{2}$, $4\frac{1}{2}$ etc. These related vibration frequencies are common on reciprocating compressors and engines that work on a four-stroke cycle. These units will typically have a camshaft rotating at half rpm, which contribute further to the vibration at one-half rpm. Although these frequencies may not indicate the problems but it has to be checked for ignition, compression and carburetion (fuel injection) problems[5].

Typical operational problems which can cause excessive vibration at higher order frequencies include excessive wear of rod and main bearings, piston slap, valve clash, blow by, compression leaks, faulty ignition., leaking valves, misfiring and fuel injection.

The frequency characteristics can be studied by frequency against displacement or velocity or acceleration curves, additional details about the vibration can be obtained by techniques like phase analysis, time wave form analysis and mode shape analysis. Mode shape analysis techniques can be the most useful for identifying the sources of mechanical looseness.

To express the forces and couples acting on the engine frame in a series of harmonic terms, it is necessary to express velocity, acceleration, and connecting rod angle in terms of crank angle. Since the period of the gas pressure cycle requires two revolutions of crank, only one-half cycle of the gas pressure variations occurs during one revolution of the crankshaft. Hence, the harmonics of the gas pressure torque are expressed as half integer multiple of the crankshaft rotation.

Instrumentation and Data Collection

Portable vibration measuring instrument and accessories are used on board of the dredger to monitor vibration. DC-11 vibration data logger is used to monitor the vibrations of the propulsor system as well as the hull vibration. Piezoelectric accelerometer and optical stroboscope are used to record acceleration amplitude of vibration and phase of the vibration. The data logger saves the data in the form of FFT spectrums. Since the rotating speeds involved with the propulsive system are of medium speed range, it is expedient to choose velocity spectrums only in lieu of other spectrums. Troubleshooting can readily be measured with flat velocity spectrums, giving valuable information in the required frequency range. The displacement amplitudes of the vibration can also be considered in case the velocity spectrums reveal little about the fault. The measurements have been taken in rms value in 1/3 octave mode to suppress the stray noise due to the adjacent machinery.

The dredger is propelled at 5 practically possible speeds and at different load conditions. The concept of speed variation is mooted to evaluate the vibration trend of the engine, gearbox and thrust block. The measurements are taken in tri-axial directions, *ie,* longitudinal, vertical and horizontal directions to critically analyze the vibration conditions.

Scheme for the Analysis of the Propulsion System

In the scheme for the analysis of the propulsive system the engine, gearbox, thrust block and propeller are considered as lumped masses connected through the shaft. Dampers are provided in between engine-gearbox, thrust block-propeller and are for consideration in the longitudinal vibration. The equivalent spring system has been considered for the shaft for all four components as shown in Figure 3. The components of the radial forces and propeller thrust force are considered as the exciting forces acting at engine and propeller, respectively.

For transverse vibrations the entire propulsive system was considered as a cantilever beam and the four components are considered as four lumped masses and each supported on a damper and spring.

The numerical methods for calculating various parameters like displacement, velocity and acceleration were calculated by using C programming.

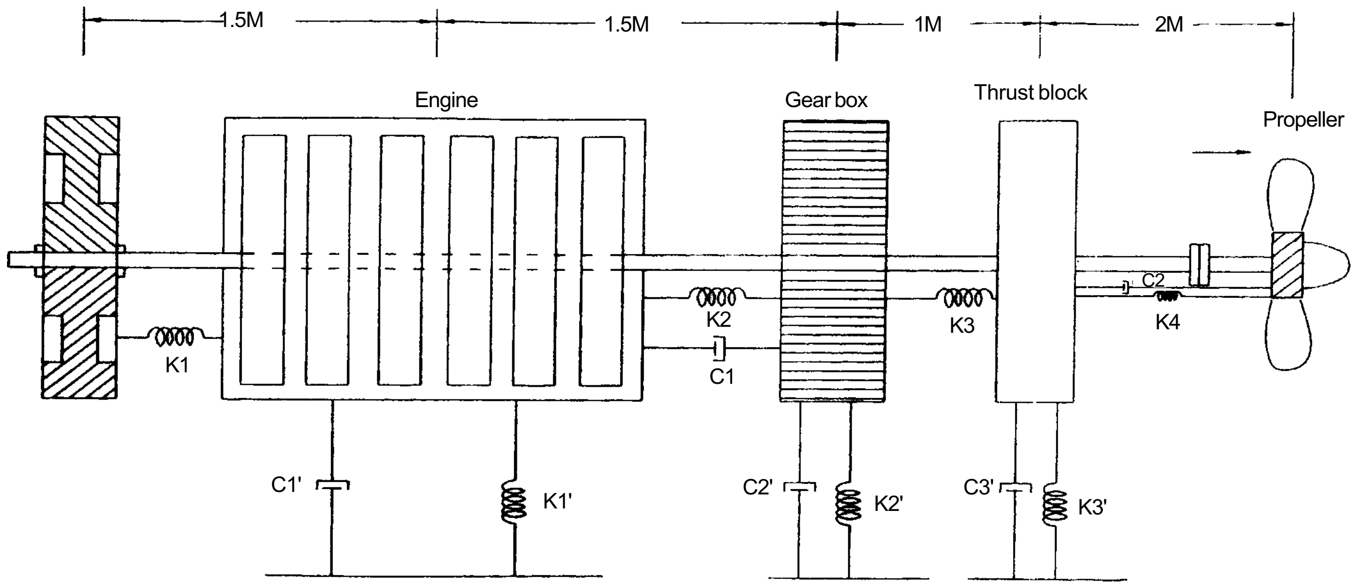

Figure 3 Scheme for propulsion plant

SYSTEM RESPONSE ANALYSIS

When a system requires more than one coordinate to describe its motion, it is called multi-degree-of-freedom system and the present system consists of four degrees-of-freedom. The generalized forcing equation is a second order differential equation formulated by considering the inertia forces, damping forces, spring forces and the exciting forces.

The eigen values and vectors are calculated by using matrix iteration methods[6] both for longitudinal and transverse mode and the fundamental natural frequencies are calculated as 326.124 rad/sec and 61.78 rad/sec, respectively. The response of the system to the forcing functions is calculated by using a numerical approach method[7]. The rms values of various parameters like displacement, velocity and acceleration are calculated taking into consideration the rotating frequency of the system.

These theoretically calculated values are compared with the practically measured values taken on the dredger.

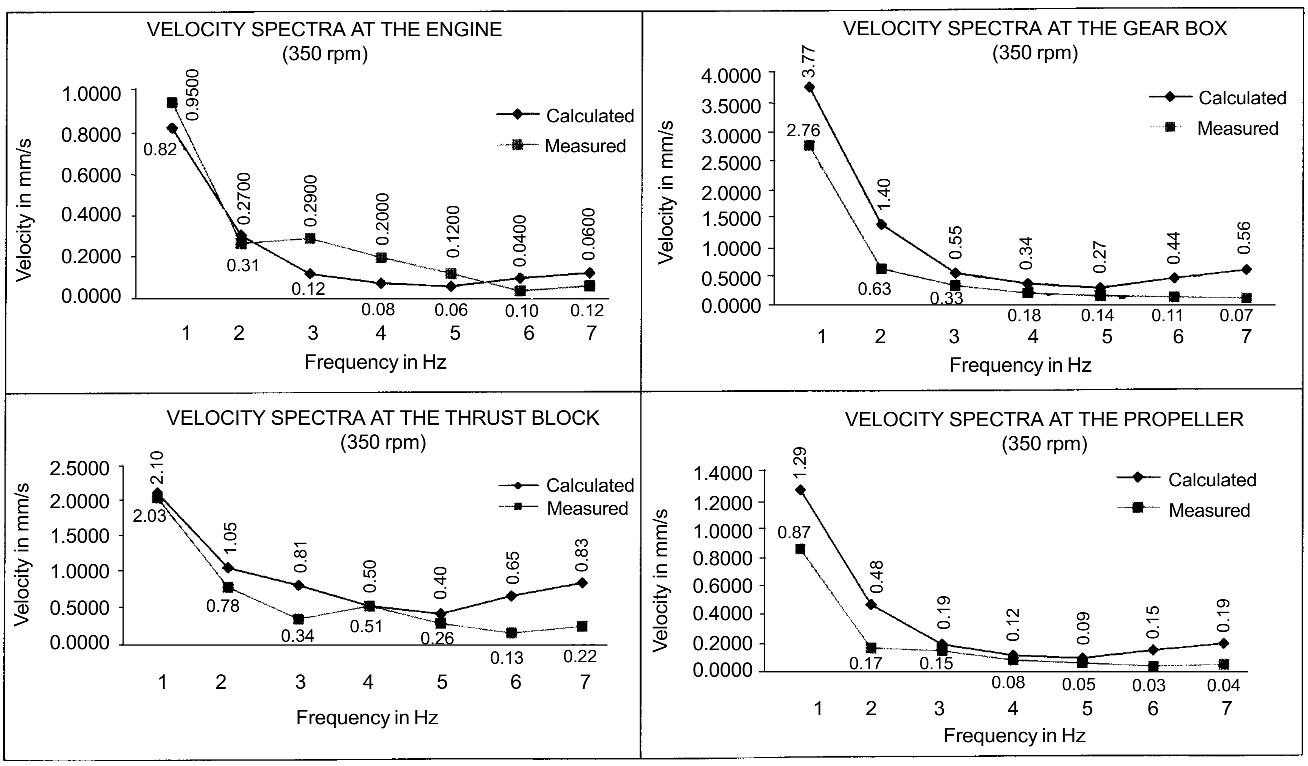

Figure 4 Comparison of calculated and measured values of transverse vibrations at 350 rpm

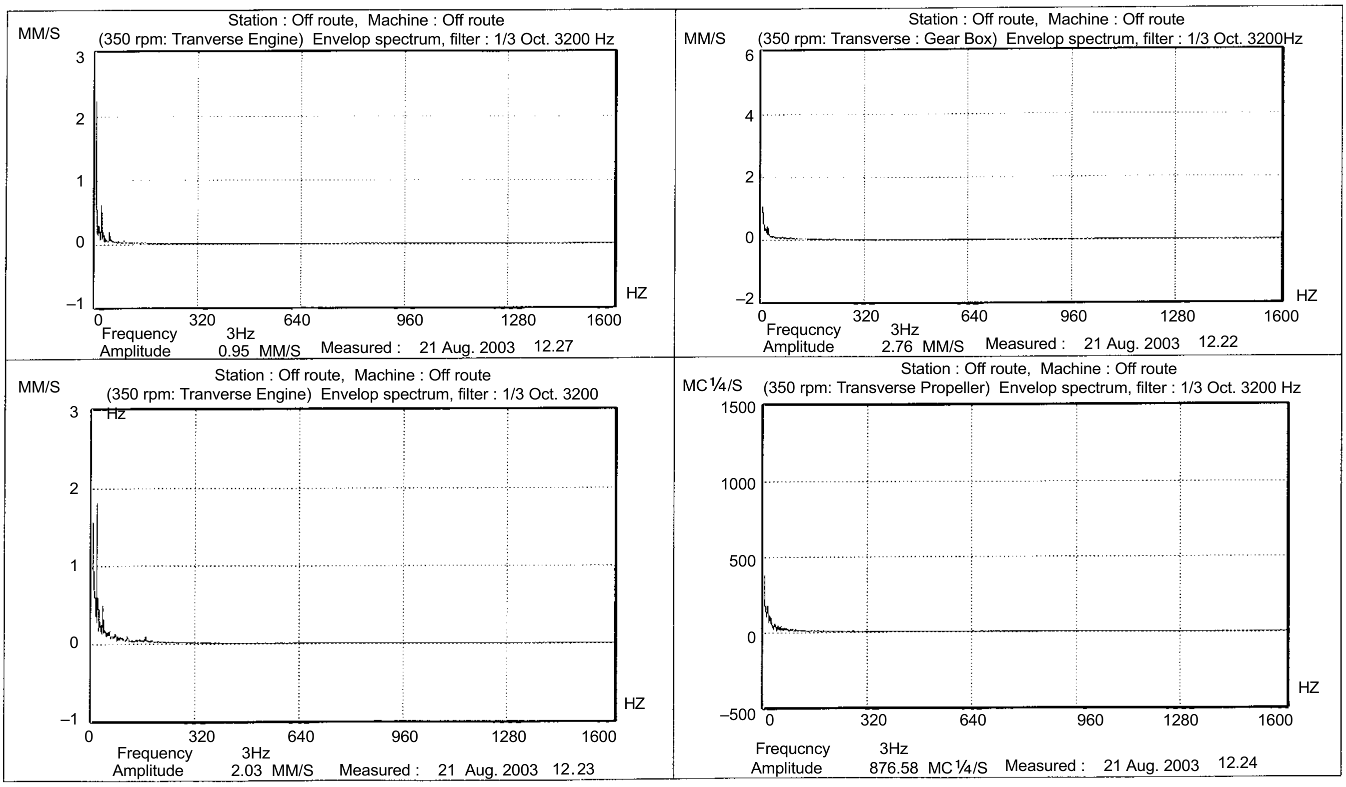

Figure 5 Velocity spectrums at 350 rpm of transverse vibrations

A typical graph for the measured and calculated values of velocity parameter at an engine speed of 350 rpm of the engine is shown in Figure 4. The vibration velocity spectrum at the particular speed given in the Figure 5.

CONCLUSION

It needs a non-intrusive tool to assess the incipient faults in the marine propulsion systems employed for ship operations. This is because of the reason the systems cannot be dismantled during cruise of the ship. The vision of fault finding through vibration monitoring became powerful tool in the contemporary days. For the dredger mentioned in this thesis, since it belongs to old fleet, the base line spectrums are unavailable. With this reason, it created necessity in calculating the modal frequencies of the equivalent multi-degree of freedom system. It is ascertained that incidentally the modal frequencies are far away from the operating speeds of the engine. That is the reason for the resonant effects of the critical frequencies have been ruled out and scrutiny by evaluating the vibration amplitudes at different orders of the operating frequencies is taken up.

The survey of vibration velocity amplitudes reveals that vibration excess is there at gearbox of starboard side engine. The fault with the above gearbox is confirmed with the acceleration diagram concerned at 350 rpm. Except this the system has never gone critical vibrationally at any other component of the system when the dredger is being run at different speeds. The masses of various components remain same in the system starting from the commissioning of the dredger, the stiffness may vary as the time passes. That is the reason the theoretical calculations may detract to some extent depending on the variation in the stiffness implicit in the system. The actual vibration measurements taken on the dredger give the true picture about the condition. However, it is observed that these measurements are well within the levels indicated by ISO vibration standards.

REFERENCES

1. J Jenzer. 'Some Vibration Aspects of Modern Ship Installations.' *System Dynamics and Vibration Analysis, Wartsila NSD Switzerland Ltd,* Winterthur, July 1996.
2. 'Ship Vibration.' Iwer Asummen, Wolfgang Menzel Holger Mumm, *et al,* Hamburg, 2001.
3. 'Prevention of Harmful Vibrations in Ships.' *Guideline Det Norske,* Veritas, Norway, July 1983.
4. J Jenzer. 'Vibration Analysis for Modern Ship Machinery.' *Wartsila NSD Switzerland Ltd,* 1991.
5. Ronald L Eshleman. 'Basic Machinery Vibrations, an Introduction to Machine Testing.' Analysis and Monitoring, *Vibration Institute VI Press,* Illinois.
6. V Ramamurthi. 'Computer Aided Mechanical Design III.' *Tata McGraw Hill Publication.*
7. S S Rao. 'Mechanical Vibrations Fourth Edition.' *Pearson Education Publication.*
8. C M Harris and C E Crede. 'Shock and Vibration Handbook.' *McGraw Hill Publication.*