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SOURCES OF VERTICAL VIBRATION GENERATING DYNAMIC LOADS IN THE NAVAL BEAMS

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Abstract: The main reason causing general stabilized hall vibration is the main mechanisms functioning, and periodic variation of hydrodynamic pressure force that strains the plating of the ship. This hydrodynamic pressure reaches maximum strain around the propellers zone. General vibration of the hall cause local vibrations of different parts of the ship since general vibrations causes movement of support structures of the hull. In some cases considerable local vibrations are caused by the auxiliary mechanisms, especially of those having pistons. Auxiliary mechanisms functioning in unfavorable conditions may cause general vibrations, too.

1. INTRODUCTION

The main reason causing general stabilized body vibration, also called running vibrations (vibrations when engine is running), is the main mechanisms functioning, and periodic variation of hydrodynamic pressure force that strains the plating of the ship.

This hydrodynamic pressure reaches maximum strain around the propellers area.

General vibration of the body cause local vibrations of different parts of the hull - floors, beams, frameworks, plates, etc., since general vibrations cause movement of support structures of the hull.

In some cases considerable local vibrations are caused by the auxiliary mechanisms, especially of those having pistons, for example, piston auxiliary engine or piston pumps.

Auxiliary mechanisms functioning in unfavorable conditions may cause general vibrations, too.

2. VIBRATIONS EFFECTS ON NAVAL BEAMS

Ship‘s motions at a stabilized agitation cannot cause stabilized vibrations on a maritime ship, because the period of the perturbation forces is 10 $\div$15 times bigger than the oscillations of the first frequency tone of the ship and a static deformation occurs during the balance of the hull, too.

General vibrations during agitation may occur only on barges – large inland tankers (length of 150 ÷ 200 m), when they tread water.

During the ship’s balance there may appear quite large damped general oscillations of the hull, caused by the wave hitting the stem.

In this case a detailed examination of the nature of forces that cause stabilized vibrations of the hull should be done.

In piston engines there appear periodic forces of inertia, when the piston is working; these forces are also observed in the crank gear, their frequency being equal to that of the crankshaft’s rotation.

The main vector and main moment of these forces decompose into Fourier series, resulting in a lot of harmonic perturbation forces, whose frequency is multiple to the crankshaft speed.

Without insisting too much on the issue of inertia forces in piston machines, we have to note that multi-cylinder machines are able to fully balance the inertial forces and moments of inertia forces of internal order, whose amplitudes register maximum values.

For the reason set out in four-cylinder engines there is a possibility to get a pretty good balance of inertia forces.

However, at a total balance of inertia forces, which basically takes place in turbine installations and installations with multi-cylinder engines, there is a first-order vibration whose frequency is equal to the speed of the propeller.

The appearance of such vibrations can be caused by one of the following: dynamic imbalance of the shaft, propeller, gearbox (reducer), lack of precision in assembling shaft, as well as by inaccuracies in propeller building – difference in the pitch (this can happen if there is a dynamic equilibrium).

Today's requirements for the dynamic equilibrium are so strict that the shafts, blades and gearboxes are considered perfectly balanced dynamically.

Therefore we must concentrate only upon examining the influence of deviations from the execution precision of the shaft and to difference in the pitch of the propeller blades.

First, we should examine the influence of reciprocal movement of the bearings, that appears when the crankshaft is assembled, and which is inevitable at general bending of the hull.

If the initial axis of the shaft was straight, when the supports of the bearings move aside there will be bearing reactions, whose magnitude and direction to the rotation of a circular shaft remains unchanged and, therefore, can not cause vibrations.

Variable reactions may occur only in a rotating non-circular shaft.

However, if the shaft axis had a deviation from the straight line caused either by the nonlinearity of the straight sections of the shaft, or by the lack of precision in installation - i.e. reciprocal movements of the centers of shaft, or the rigidity of the shaft axis on the joining of shaft sectors flange then, at the rotation even of a circular shaft, by reaction, the bearings will turn anticlockwise, i.e. vertical and horizontal forces, harmonically variated will be transmitted to the hull by means of bearings.

3. CASE STUDY

The simplest case will be examined, when the distorted axis of the shaft is a flat curve, all of the bearings are in the same vertical plane, but are not located on the same line (are not aligned).

If the plane of the initial bending of the shaft is horizontal, then horizontal strains should be applied to the shaft, so as the support sections of the shaft coincide with the verticals passing through the bearings, and further on, by applying vertical strains the support sections of the shaft should coincide with the shaft's bearings.

In this case the reactions of the bearings will be inclined, i.e. when the shaft rotates anticlockwise the magnitude and direction of the reactions will probably vary, according to the harmonic law, with the speed of the shaft, causing vibrations to the hull.

However, as shown by calculations, for the existing hypothesis, adopted at the assembly shaft line, the magnitude of these variable reactions is insignificant, and cannot cause general perceptible vibration to the hull if, within the system propeller-shaft do not appear resonance phenomena caused by the propeller's reaching the so called critical rotative speed (frequency) of the shaft.

The critical frequency of shaft's rotation is determined taking into account from the following considerations:

- take a shaft with metrical mass "γ", which rotates with angular velocity "ω", and is deviated from the linear's form with value "v";
- then, all along the shaft length "a" of the centrifugal forces of inertia the metrical load acts, and is equal to:

$$\frac{\gamma}{g}\omega^2 v\,, \tag{3.1}$$

so that the differential equation of the bending must be written as:

$$EIv^{IV}(x) = \frac{\gamma}{g}\omega^2 v + q \tag{3.2}$$

where "q" is the - intensity of the transverse load.

Note.
In this case the so called "gyro effect" will not be taken into consideration.

Differential equation:

$$EIv^{IV}(x) = \frac{\gamma}{g}\omega^2 v + q \tag{3.3}$$

coincides with the differential equation:

$$EIv_0^{IV}(x) - m\omega^2 v_0(x) = q(x), \tag{3.4}$$

which determines the shape of the beam oscillations on which a harmonic disruptive load of the:

$$q\cos\omega t \tag{3.5}$$

type acts. When the value of frequency ω comes closer to the frequency of free transverse vibrations of propeller-shaft system, (i.e. the critical frequency of shaft rotation), the bearing reactions, and also the disruptive forces that cause ship vibrations may increase substantially.

4. CONDITIONS FOR DESIGNING SHAFTS DEPENDENT ON PROPELLER

Shaft lines are always designed in such a way that the critical frequency is substantially higher than the maximum speed of the shaft in operational mode.

For this reason virtually the first order disruptive forces arising from the lack of precision in shaft assembly are not taken into account.

The disruptive forces arising from the unequal pitch of the propeller blades are to be examined.

In this case the resistance developed by each blade will be different, the resultant of the maximum resistance of the propeller will not pass through its center, at the end of the propeller not only the force of resistance will be applied but also the momentum created by it. The action plan of this moment will rotate with angular velocity of the shaft.

In addition, the profile resistance of the blades will be different, the range of profile forces resistance of the blade profile will not be closed, and they will report to the resultant that rotates with angular rotation speed of the shaft.

In this way, the difference in blades pitch gives rise to the momentum and first order disruptive force which cause vertical bending oscillations, first order combined vertical bending and torsional oscillations of first order.

For this reason, at present, there have been set stricter rules for the tolerances of blades pitch.

In real conditions the momentum and the first order force will have a varying amplitude, because the resistance will vary for each blade if the propeller happens to be in a non-uniform current, caused by the coming closer of the hull.

In this case the disruptive momentum will be the periodic function of time, but at the Fourier series decomposition the main term will be the first one.

It should be noted that the second order vibrations with a frequency two times bigger than the shaft's rotation is practically not observed.

Besides first order vibrations, another type of vibration is observed, the so called propeller (or blade) vibration which is of "z" order, in which "z" is the number of propeller blades.

These vibrations are caused mainly by pulsatory pressures that are transmitted to the hull through water.

The frequency of pressure pulsation is most probably equal to the frequency at which the blades rotate near the hull, which is equal with "$z\omega$".

A certain part of the disruptive effort, resulted from the resistance pulsation with frequency "$z\omega$" is transmitted to the hull by axial bearings.

The pulsatory pressure that acts on the hull is a periodic function of time. When it is decomposed into Fourier series there occur variable harmonic pressures with "$z\omega$", "$2z\omega$" value, etc..

Series coefficients decrease rapidly, especially when going away from the location of the propellers, and, for example, to calculate the generalized force for general vibrations we can leave apart the second term of frequency decomposition with frequency "$2z\omega$".

The amplitude of harmonic variable pressure can reach considerable magnitude only within the range of propeller location.

This pressure must be taken into account when calculating shell plating vibration.

In single screw propeller ship the pulsatory pressures will not be symmetrical in relation to the diametrical plane (due to rotation of propeller), and will be linked to both vertical and horizontal components of these pressures resultant.

Besides vertical vibrations, horizontal and torsion vibrations also occur. In multi screw ships, propellers, being located on different boards turn in opposite directions.

In this case the horizontal component of the pulsatory pressure should be taken into account because propellers are in different phases in relation to the hull, i.e. the position of two propellers which are symmetrical to each other is not exactly the inverse image (in the mirror).

In multi screw ships a clattering of the propellers can appear, and is due to the differences between the angular speeds of propellers rotations.

By adding up two harmonic loads:

$$P \cos \omega t \tag{3.6}$$

and

$$P \cos \omega_1 t \tag{3.7}$$

there results:

$$2P\cos\frac{\omega-\omega_1}{2}t\cos\frac{\omega+\omega_1}{2}t \tag{3.8}$$

with amplitude:

$$2P\cos\frac{\omega-\omega_1}{2}t \tag{3.9}$$

slowly varying (for the small difference: $\omega-\omega_1$).

Determination of the value of disruptive forces mentioned above is a rather complex hydrodynamic problem.

For this reason only the possible ways to decrease disruptive forces will be mentioned.

4. DISCUSSIONS, INTERPRETATIONS AND CONCLUSIONS

The value of pulsatory hydrodynamic pressures depends, to a great extent, on the existing space between propeller and hull, and it decreases as the space increases.

Besides this, the decrease in pulsatory pressures can be achieved by increasing the number of propeller blades, and fitting up special parts along the propeller, to equalize the water flow driven to the propeller. We can conclude that a precise calculation of maintained vibrations of the hull has not been studied to the end.

Along with the complexity of the problem referring to determination of disruptive forces value another issue rises, i.e. the value and the principle of internal energy diffusion to the hull's oscillations. A correct calculation of resistance determines the calculation result of vibrations while the propeller is in motion.

However, avoidance of resonance by appropriate spreading of frequencies (that means that the minimum frequency of the free oscillations should be at least 15 ÷ 20 % higher than the upper limit of the disruptive force frequency) can only be possible in low speed engines.

Nowadays engines speeds are so great that in common case even tone $4\div5$ frequency free oscillations of the hull are below third order disruptive forces, therefore, it is necessary to calculate the maintained oscillations, whose frequency comes closer to the frequency of free oscillations (quite high), of a quite high tone.

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