Practical confirmation of mechanical balancers effectiveness to reduce vibration of marine main diesel engines

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Abstract. The article discusses compensators of unbalanced moments from inertial forces of the second and variable components of the overturning moment of the main order and the elastic moment at the resonance of torsional vibrations of the shaft line of modern marine low-speed diesel engines. The efficiency of the diesel engine's operation is analyzed based on the vibration measurements on ships and some problems caused by these devices.

1. Introduction
Unbalanced moments from the forces of inertia of the second order of modern marine low-speed diesel engines depend on the number of cylinders and the order of their operation [1, 2]. These moments because specific problems associated with the vibration of low-speed diesel engines themselves, as well as the vibration of the ship's hull and superstructure [3, 4, 9].

The moment from the inertia forces of the 2nd order is determined relative to the center of mass of low-speed diesel engines by the formula

$$M_2 = \lambda m_s \omega^2 \sum_{j=1}^{i} L_j \cos 2 \alpha_j$$

where
- $\lambda$ – constant of low-speed diesel engines, equal to the ratio $R$ - radius of the crank to the length of the connecting rod $L_{sh}$;
- $i$ – the number of cylinders of low-speed diesel engines;
- $m_s$ – the mass of translationally moving parts;
- $\omega$ – nominal rotational speed of the crankshaft;
- $L_j$ – the distance of the j-th cylinder from the center of mass of low-speed diesel engines;
- $\alpha_j = (\alpha_j + \xi_j)$ is the phase angle of the j-th crank, here $\xi_j$ is the wedge angle of this cylinder.

The values of the unbalanced moments $M_2$ of modern long-stroke low-speed diesel engines of the LMC, SMC, SME, and GME types increased significantly due to the increase in $\lambda$, i.e., the use of shortened connecting rods for these machines [5, 9]. For example, for ultra-long-stroke diesel engines of the GME type, $\lambda = 0.465$. This data must be considered at the design stage of ships to assess the vibration activity adopted for the installation of low-speed diesel engines. That is, the ability to cause an increased vibration level of the ship's hull and the main elastic system in the engine room "diesel
bottom” during resonance vibrations. The previously proposed to unbalance criteria for low-speed diesel engines make it possible to evaluate all unbalanced moments, including $M_2$. At the same time, evaluating the $M_2$ imbalance criteria, its value with the number of cylinders more than six may not be considered. This data is confirmed by the experience of diesel engine manufacturers who use different designs of $M_2$ expansion joints (Figure 1), only for 4-6 cylinder low-speed diesel engines [3].

![Figure 1. Kinematic diagram of the compensator](image)

To assess the imbalance of low-speed diesel, Pentatech Co., Ltd. suggests using the ratio of the $M_2$ value to the diesel power (Power Related Unbalance, PRU) [8]. This ratio can be used for determining the acceptable value of the unbalanced moment $M_2$ from the 2nd order inertia forces.

$$PRU = \frac{M_2 [Nm]}{N [kW]}$$

The obtained PRU value will give the designer an idea of the need to use $M_2$ expansion joints in low-speed diesel engines.

The diesel engine manufacturer manufactures the unbalanced moment compensator. The unbalanced moment compensator is an additional rotary device, which is selected in such a way that the moment created by two centrifugal forces is in antiphase $M_2$ and compensates for its effect on the skeleton of low-speed diesel engines. The need to install a compensator is determined based on the PRU values (Table 1).

| PRU | Necessity of installation of a compensator |
|-----|----------------------------------------|
| Less than 120 | Not necessary |
| 120-220 | Desirable |
| More than 220 | Needed |

These conditions are in good agreement with the criteria for the imbalance of low-speed diesel engines, developed based on vibration standards for a small fishing vessel and given in [3, 4].
2. Problems of increased vibration on ships from an unbalanced moment of the 2nd order

On the tanker "Asia," there was an increased vibration of the main diesel engine of MAN B&W type 5L60MC-C from the resonance of the H-form of vibrations of the flexible system "diesel bottom" at the nominal speed. Based on the results of vibration research, a new design of links for the upper attachment of the skeleton of low-speed diesel engines was developed and patented [3, 6, 7]. Figure 3 shows the design of the conical damper pivot of this connection.

Let us turn to the problem of vibration on this tanker. At first, the operation of the tanker in the fleet of the Primorsky Shipping Company was observed:

• damage to parts of the 2nd order unbalanced moment compensator (remote nasal block);
• increased vibration of the main diesel engine;
• Damage to the structures of fastening the diesel engine to the ship's hull (upper links of the B&W design, Fig. 2).

Diesel engines of this type have a significant unbalanced moment of the 2nd order, which is associated with a peculiarity of the layout of long-stroke diesel engines, i.e., a consequence of a significant increase in the ratio of the radius of the bloodworm to the length of the connecting rod (1 - 0.4). For this reason, the company has taken constructive measures to reduce moment of the 2nd order using a specially installed compensator.

![Figure 2. The structure of the connection of the upper frame of the low-speed diesel engines from B&W](image)

This device has a nasal block, consisting of 2 oppositely rotating masses and creating a vertical force that changes harmonically with a frequency of the 2nd order (Fig. 1). The nose block is driven from the camshaft. The feed unit is located in the chain compartment, where similar masses are attached to the intermediate gears of the chain drive. The pair of forces created by these devices act in antiphase to the moment of the 2nd order. The nose block is taken out separately from the cylinder block and mounted on two brackets connected to the frame with flanges. However, some damages appeared in the design of the bow unit during operation in the form of:

• fatigue cracks in the body walls;
• damage in the details of its attachment to the diesel engine frame (cracks in brackets, flanges, destruction of bolts).

Cracks in the case, made of sheet steel, and in the details of its fastening were repeatedly welded, but this did not give a positive effect. For eliminating them, it was necessary to reduce the excessive vibration of the nasal block by increasing its rigidity, i.e., eliminating resonance vibrations. Oscillations of the wall were caused by an alternating load transmitted from an intermediate gear, cantilevered on this structure. When developing measures to modernize the expansion joint body (increasing the block rigidity), it was generally achieved by installing knits from the bottom and upper longitudinal ties with the cylinder block.

On passenger ships of the "Mikhail Kalinin" type, the flexible system "2 diesel-bottom" had a resonance of the 1st vibration mode with excitation from 2 unbalanced moments of the 2nd order of diesel engines operating in phase. The shafting frequencies are set manually and slightly different,
leading to a phase shift between the unbalanced moments acting on the flexible system "2 diesel-bottom" (disturbance and vibration have the form of a beat, as shown in Fig. 4).

Similarly, the oscillations had the form of a beat with a period depending on the difference in the rotational speeds of the left and right diesel engine, which is determined by the formula [3]

$$T_b = \frac{\pi}{(\omega_{port} - \omega_{std})} = \frac{\pi}{\Delta\omega}$$

where $\omega_{port}$ is the rotational speed of the left diesel engine, $\omega_{std}$ is also the right diesel engine.

![Figure 3](image_url) Conical hinge-damper connection of the upper attachment (patented by the author design [10]).

![Figure 4](image_url) Summation of two unbalanced moments of the 2nd order

The analysis of this dependence follows that with the frequency increment $\Delta\omega$ tending to zero, the period increases to $\infty$. In this case, if the phases coincide (or in antiphase) of the two harmonic moments acting on the system "2 diesel-bottom", the oscillations will have a resonance (the beat will disappear). Otherwise, if the frequency difference increases by more than 3, the value of the period will decrease to a few seconds.

Figure 5 shows a graph of the vibration amplitudes of the nose of the cylinder block of the right main low-speed diesel engines when the rotational speeds of the two diesel engines are equal and the unbalanced moments in phase coincide.

The beat period tends to infinity when the rotation frequency is equalized and decreases with an increase in the difference in frequencies. With a beat period of more than 5 seconds (the difference in rotational speeds is 6 min$^{-1}$), the vibration amplitudes reached the maximum permissible value for low-speed diesel engines, which was confirmed by experimental studies carried out on the turbo electric vehicle "Abkhazia." With a further increase in the period, the amplitudes exceed the permissible value
according to the norms of a small fishing vessel. However, with a beat period of more than 6.5 seconds, the vibration process is stabilized, the amplitudes remain constant (Figure 6).

In this case, eliminate the resonance by increasing the frequency of free oscillations of the system, i.e., in a constructive way, it was impossible due to the required large material costs. Therefore, it was recommended to set the operating mode of the main diesel engines with the parameters given in Table 2.

![Graph of the dependence of the maximum amplitudes of vertical vibration on the rotational speed of the main diesel engines](image)

**Figure 5.** Graph of the dependence of the maximum amplitudes of vertical vibration on the rotational speed of the main diesel engines

**Table 2.** Permissible beating periods according to vibration standards

| Parameter                        | Vibration standard for engines rules of small fishing engines | Sanitary Standards for residential premises |
|----------------------------------|-------------------------------------------------------------|---------------------------------------------|
| Beating period, s (less)         | 5                                                           | 3.5                                         |
| Speed difference, min¹*          | 6                                                           | 8                                           |

*Note: the difference in rotation speeds leads to overload of one of the diesel engines. Therefore, it was recommended to alternate the operation of the left and right diesel engine at an increased speed every 4 hours.

With the development of designs of low-speed diesel engines (models such as LMC, SMC, SME, and GME by MAN), compensators’ unbalanced torque of the second-order were improved. Let us consider the main ones:

- a compensator with an external nose unit (given in the 2nd section);
- a compensator with a nasal block built into the frame;
- compensator with a remote aft electromechanical unit in the tiller compartment and a bow unit in the skeleton of low-speed diesel engines [9].
Figure 6. Dependence of the relative vibration amplitudes on the beat period or the difference between the rotational speeds of the crankshafts of diesel engines (at the resonance of the 1st vibration mode of the "2 diesel-bottom" system with a frequency of the 2nd order)

The first expansion joint design had design flaws, discussed in Section 2, and was used only for low-speed diesel engines of the LMC type.

The second is the most advanced and widely used in all models of super-long-stroke low-speed diesel engines (Figure 7). It has a mechanical drive from the crankshaft of the diesel engine of the bow and stern blocks.

Figure 7. Bow and stern blocks of the compensator of 6-cylinder low-speed diesel engines of the SMC type manufactured by MAN B&W

The third one with a remote aft electromechanical unit in the tiller compartment and a bow unit in the skeleton of low-speed diesel engines (Figure 8) is the most complex design. In it, one block is mechanical and is located in the frame of the diesel engine, and the second is placed in the tiller compartment. At the same time, the in-phase rotation of the shafts of the remote block and the
The crankshaft of low-speed diesel engines is maintained to create a pair of forces that create a moment in antiphase to the unbalanced moment of the second-order of the main diesel engine. The advantage of this design is that the arm of the pair of forces is much larger than the length of the skeleton of low-speed diesel engines, which is the arm of the pair of forces for type 1 and 2 expansion joints. The disadvantages of this design include the local effect of harmonic forces with a frequency of the 2nd order on the skeleton of low-speed diesel engines (common to all three designs) and the hull's structure in the tiller compartment since they are not absolutely rigid. This design of the compensator can be used for reducing the vibration level at resonances of the vessel's hull vibrations from the impact of the second-order unbalanced moment of low-speed diesel engines [9].

**Figure 8.** Layout of various types of balancers [8]: 1 – with a screw of adjustable pitch, horizontal balancer, compensating for the effect on the thrust bearing of the variable stop; 2 – for 8, 9, 10, 11, and 12-cylinder low-speed diesel engines to compensate for the action of the horizontal torque MX, horizontal balancer; 3 – on 4, 5 and 6-cylinder engines to compensate for the action of the overturning moment and under the influence of the reaction from the elastic moment at the resonance of torsional vibrations of the shafting; 4 – for 4, 5 and 6-cylinder low-speed diesel engines to compensate for the action of the second-order M2 torque; 5 – for ships with 3, 4, 5 and 6-blade propellers; 6 – with superstructure vibration, horizontal balancer

The same company offers compensators of different types (vertical and horizontal) to compensate for alternating loads from main diesel engines (second-order unbalanced moments, twisting moments and torsional moments) and propellers. Figure 8 shows a diagram of the possible use of these compensators in the aft position of the engine room and superstructure to reduce vibration levels on ships.

The effectiveness of the use of compensators is assessed by the frequency of reducing the vibration level, especially during resonances, and, finally, by reducing this level below the permissible values (for the ship transportation system according to the norms of small fishing vessels, and workplaces and living quarters – sanitary standards).

Before and after installation of expansion joints, Pentatech Co., Ltd. reported Vibration test results in work [8] (reduction of vibration level by several times).
There are several methods for reducing ship vibration, namely [9]:

1. Reduction of the exciting force (harmonic forces or moment) during forced vibrations in the "H-form" when passing through the resonance zone of torsional vibrations of the shafting (i.e., from the effect of the elastic moment). In the crossheads, it is transformed into harmonic forces with a frequency of the 6-th order and causes transverse core vibrations with increased amplitudes. In this case, vibration is transmitted along with the hull structures to the superstructure. The balancer creates a horizontal harmonic force with a frequency of the 6th order. This force acts in antiphase to the moment from the normal forces in the crossheads and thereby reduces the forced vibrations of the engine cylinder block. This result is noticeable from the results of vibration measurements with the compensator turned on and off (a decrease in the amplitudes of vibration velocities by more than two times). However, the level of torsional vibration at resonance (Fig. 9) in this case remains unchanged since the presence of a compensator does not affect the torsional shafting scheme and the torque disturbing the torsional vibration (a variable component of the motor torque). In this case, it should be considered more rational to install a torsional vibration damper (silicone on the nose end of the crankshaft with the transfer of the tacho system encoders to it) similar to the system of their attachment to the flywheel. In this case, it will be possible to reduce the elastic moment at the resonance of the torsional vibration and at the same time eliminate its reaction in the crossheads, thereby lowering the vibration level of the diesel cylinder block. In this case, the installation of the compensator is not required.

2. An increase in the rigidity of the elastic system and, at the same time, natural resonant frequency, i.e., an increase in $\Delta \omega$ – the difference in the frequencies of the harmonic component of the disturbing force and free vibrations with an exit from the near-resonance zone ($\pm 30\%$ for mechanical vibrations of real elastic systems with dynamic coefficients equal to $\beta = 5 \div twenty$);

3. Increasing the damping properties of the system in order to reduce the vibration amplitudes at resonance;
4. Appointment of "forbidden zones" in frequency in the area of resonant oscillations, the width of which depends on the permissible level of vibration (or cyclic stress) on the curve of the peak of the resonance. If the peak of the resonance is in the range of the specification mode, it is necessary to take the constructive measures provided in clause 2;

5. Constructive changes in the elastic system (reconstruction of the structural diagram of the system with the replacement of materials);

6. Use of vibration isolation.

The first of them is the most radical and is produced at the expense of compensators. Low-speed diesel engines with cylinders from 4 to 6 with second-order unbalanced torque compensators are carried out based on the requirements to reduce their vibration activity when installed on any vessel. However, if increased vibration (resonance) is detected during the delivery tests at full speed, it is necessary to analyze the vibration of this flexible system and develop constructive measures (2nd method). In this case, if the resonance occurs in full stroke mode, then the task is simplified by the fact that it is more rational to introduce additional constraints into the elastic system to increase its rigidity and, consequently, natural resonant frequency [4]. The links should be installed in the direction in the direction of the main vector of displacements (according to the results of vibration measurements in 3 directions) so that the links work in tension-compression but not bending [3, 8]. However, this method requires certain costs for vibration research and additional costs for installing compensators. Suppose the installation of additional connections is a problem. In that case, forbidden zones can be assigned according to the crankshaft speed of low-speed diesel engines, or a mode with a separation of the speeds of 2 low-speed diesel engines can be selected, as was the case on ships of the "M. Kalinin" (section 2). This process makes it possible to reduce the period of the beating of vibration, and the amplitude of vibration does not reach maximum values [3].

3. Conclusion

1. Installation of the second-order moment unbalanced moments compensators is a radical method of reducing vibration of ship hulls.

2. This method of reducing ship vibration is applicable for elastic subsystems of the "diesel-bottom" type (at resonances of the main vibration modes: 1st, H, and X).

3. The use of other types of compensators should also be considered an effective method of reducing vibration levels of elastic subsystems in the engine room and superstructure.

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