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A Review of Marine Engine Vibration Performance Analysis and Counteract Measures

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Abstract: *This paper gives a review of the vibration characteristics and its analysis associated with two stroke low speed marine diesel engines and its counteract measures.Marine diesel engines are variable speed engines and face resonance vibration in a narrow band speed within its speed range and this band is called as barred speed range. The historical strategy for vibration control has been sub-critical design criteria. This strategy adopts vibration analysis of local primary forces of machinery components and thereby neglects the impact of secondary forces. Recently, Global vibration analysis has been adopted to reduce the influence of barred speed rangeand secondary excitations. In this paper, we study the performance comparison of both sub-critical design criteria and global vibration analysis.*

Keywords: *Vibration, secondary excitations, barred speed range, sub critical design, Global vibration analysis.*

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Nomenclature:
SMCR: Specified Maximum Continuous Rating
NCR: Normal Continuous Rating
SDARI: Shanghai Merchant Ship Design And Research Institute
FEM: Finite Element Method
AVMS: Axial Vibration Monitoring System
IACS: International Association of Classification Societies

I. Introduction

Although ship designers make every effort to prevent objectionable and detrimental vibrations before a vessel is constructed, there are times when vibrations are at an unacceptable level after construction or major modification. Vibration aboard ship can result in fatigue failure of structural members or major machinery components, can adversely affect the performance of vital shipboard equipment and increase maintenance costs, and result in discomfort or annoyance to passengers and crew [6].The concern about vibrations on board ships most often stems from a wish to provide comfortable conditions. However, if not adequately dealt with, vibrations can reach a level which threatens the safe operation of mechanical and electronic components and even the stability of major parts of the ship's steel structure [4].A ship is an extremely complex assembly of structural and mechanical components which are, in turn, stimulated by a large number of dynamic forces both transient and periodic in nature which may be significantly Increasedin severity bysea and operating conditions [1] [6].In this review, an effort is made to present sufficient information to understand the basis for the generally observed vibration phenomenon.And relates to the control and attenuation of those design factors (exciting forces or dynamic response characteristics) which are built into the ship and which contribute to alternating stresses or a vibratoryenvironment which may prove to be unsatisfactory to the ship's structure, machinery, equipment, or personnel. It will also apply to an increase in vibration resulting from damage, such as bent propeller, or maintenance problems,such as excessive bearing wear down [6-7].The principal engine unbalance encountered with slow-speed diesel driven ships is the primary and secondary free engine forces and moments. Of particular concern is the magnitude of the forces and moments, the location ofthe engine, and the possible correlation of these inputs with the lower vertical and athwart ship natural frequencies of the hull girder. The primary forces and moments occur at shaft frequency and the secondary forces and moments occur at twice shaft frequency [1-3].The combustion forces occurring during ignition plays a dominant role as excitation source coming from the engines. Due to these forces, transferred through the crank gear, transversal and vertical forces act on the engine structure and excite the typical engine vibration modes: the transverse vibration about the longitudinal axis (H-type mode), the torsional vibration about the vertical axis (X-type mode) as well as the longitudinal vibration about the transversal axis (L-type mode). While the longitudinal vibration may be excited by vertical forces, the H-type and X-type vibrations are predominantly caused by the transverse guide forces. As it is extremely costly to change the vibration behavior of the engine/foundation once the vessel is in service, a procedure has been developed for assessing the risk of resonance at the design stage using the finite element (FEM) technique.Complete FEM models are made to predict hull/ engine interaction, where FEM vibration and stress analyses are performed for hull structure, shafting system and propeller blades based on the calculated loads.

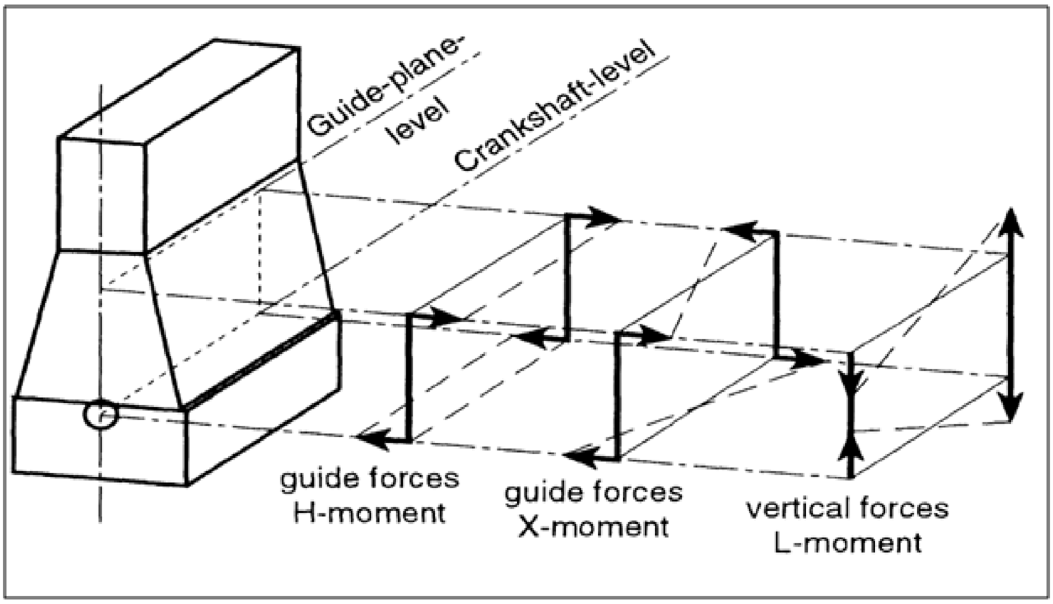

Figure 1. Excitationforces and moments of engines.

II. Marine engine vibration theory

[Vibration level] = [Magnification Factor] x [Static Displacement]

$$[\text{Vibration level}] = \frac{1}{\sqrt{(1-R^2)^2 + (2\cdot\delta\cdot R)^2}} \cdot \frac{M_G}{K_{\theta\_DB} + K_{\theta\_TB}}$$

Where,

Magnification factor (M_{Dyn}) = $\frac{1}{\sqrt{(1-R^2)^2 + (2\cdot\delta\cdot R)^2}}$

Static displacement (θ_{Static})= $\frac{M_G}{K_{\theta\_DB} + K_{\theta\_TB}}$

M_G = guide force moment

δ =Vibration damping factor

K_{θ_DB} and K_{θ_TB} = Double Bottom Torsional Stiffness and Stiffness contribution from Top Bracings respectively.The amplitude ratio or magnificationfactor is a measure of the severity of the vibration. For R=1 maximum vibration occurs (Resonance).

$R=F_E/F_{Nat}$ =1, Resonance -> High vibration.

In order to avoid a vibration problem, the magnification factor must be kept small.

$R=F_E/F_{Nat}$< 0.5, Flank vibration -> Low vibration.

In practical means, this is done by applying sufficient stiffness so the natural frequency (F_{Nat}) is much higher than the forcing frequency (F_E). This method is called “Sub critical Design Criteria”. The natural frequency is only dependent on engine/vessel mass/inertia and installed stiffness [5].

III. Sub-critical design criteria

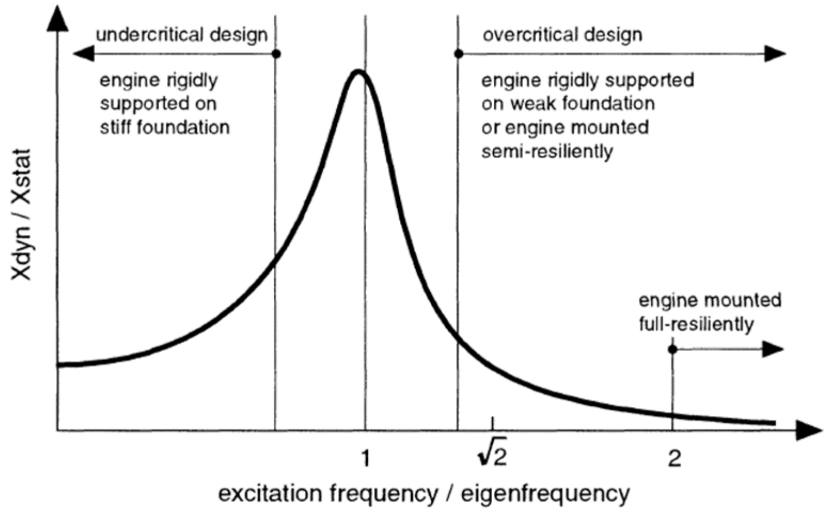

Figure 2: Possible counter measures to detune engine/foundation vibrations.

In the presence of resonance, the forces introduced into the foundation and thus into the ship may be magnified dynamically 6 to 9 times. Therefore, its recommended to initiate countermeasures in order to detune the engine/foundation system. Possible measures with respect to the recommended frequency ranges of a low-vibration design are shown schematically in Fig. 2. Here the transfer function of a single degree of freedom (H-type motion of engine) system is plotted. The counter measures can be varying the foundation stiffness; frequency variation incase of a rigidlymounted engine can possibly be achieved by modifying thefoundation design. An effective sub-critical or under-critical design of a foundation having a safety margin of 30 % requires highly stiff foundations. It appears appropriate only for engines with relatively low firing frequencies. For engines with higher firing frequencies an overcritical design may be more appropriate. The foundations will be of a more elastic construction, so that after passing the resonance a condition with low amplitudes will occur. However, this method also has its limitations. Apart from possible strength problems and classification requirements to be observed, it has to be ensured that the next higher vibration modes will not enter into the excitation frequency range.The natural frequency of the one-node vibration is so adjusted that resonance with the main critical order occurs about 35-45% above the engine speed at SMCR. Such under-critical conditions can be realized by choosing a rigid shaft system, leading to a relatively high natural frequency. Thus, characteristics of under-critical system are: to have a short shafting arrangement, probably no tuning wheel, with large diameters of shafting as it enables the use of shafting materials with a moderate ultimate tensile strength, though requiring careful shaft alignment due to relatively high bending stiffness and to be without barred speed range.

3.1. Critical running:

When running in critical, significant varying torque at MCR conditions of about 100-150% of the mean torque is to be expected which under adverse conditions, might excite annoying longitudinal vibrations on engine/double bottom and/or deck house.The shipbuilders are aware of this and they ensure that the complete aft body structure of the ship, including the double bottom in the engine room, is designed in such a way to be able to cope with these vibrations.

3.1. Overcritical running:

The natural frequency of the one node vibration is to be adjusted such that the resonance of the main critical order occurs at about 30-60% of the engine speed at SMCR. Such overcritical conditions can be achieved by choosing an elastic shaft system, leading to low natural frequency values.The characteristics of overcritical conditions are: to have a tuning wheel on crankshaft fore end, a turning wheel with high inertia, small diameter shafts requiring shafting material with high ultimate tensile strength and with barred speed range about ±10% with respect to the critical engine speed.Overcritical layout is normally applied for engines with more than four cylinders.

IV. Influential factors in order to obtain low vibration

F_{Nat} and F_E are the influential factors affecting the vibration levels in marine diesel engines. The strategy is to increase F_{Nat} and decrease F_E to get lower vibration levels. Higher $F_{Nat}$is achieved by higher stiffness of double bottomK_{θ_DB}, top bracing systemK_{θ_TB} and number of top bracings. Lower F_E canbe achieved by lower guide force moment $M_G$which again depends on lower engine rated power. G type engines are designed to give same ship speeds at lower engine rated power. Lower F_E can also be achieved by reduced hull induced secondary excitation. In G type engines both higher F_{NAT} and lower $F_E$has been achieved effectively [5-6].

V. Barred Speed Range

Position of barred speed rangeor critical speed range is mainly influenced by the number of cylinders, shafting length and shafting material strength and causes resonance within this range, leading to vibrations. Due to excessive torsional vibrations at certain shaft speeds, many shaft lines have a barred speed range.Normally, the barred speed range is 42 – 51 % of SMCR RPM and within this range fatigue failure may occur due to increased guide forces in the system [1].In case of resonance operational speed range vibrations can be amplified by a factor of 5-10 times as compared to the non-resonance situation. As a general rule, the barred speed range should be passed in seconds, not minutes. Hence a quick passage of the barred speed range (~5 seconds) is required to avoid these fatigue failures.As the actual level of the guide moment has less importance to a successful installation aslong as a non-resonant situation is achieved for an engine –hull combination. The increase in guide force moments between the G and S-type engines can be a factor of only 1.25 - 1.5.Therefore,the strategy for G-type engine is to adopt non-resonance situation within the operating range. The fig.3 below shows the influence of number of cylinders on barred speed range[5].

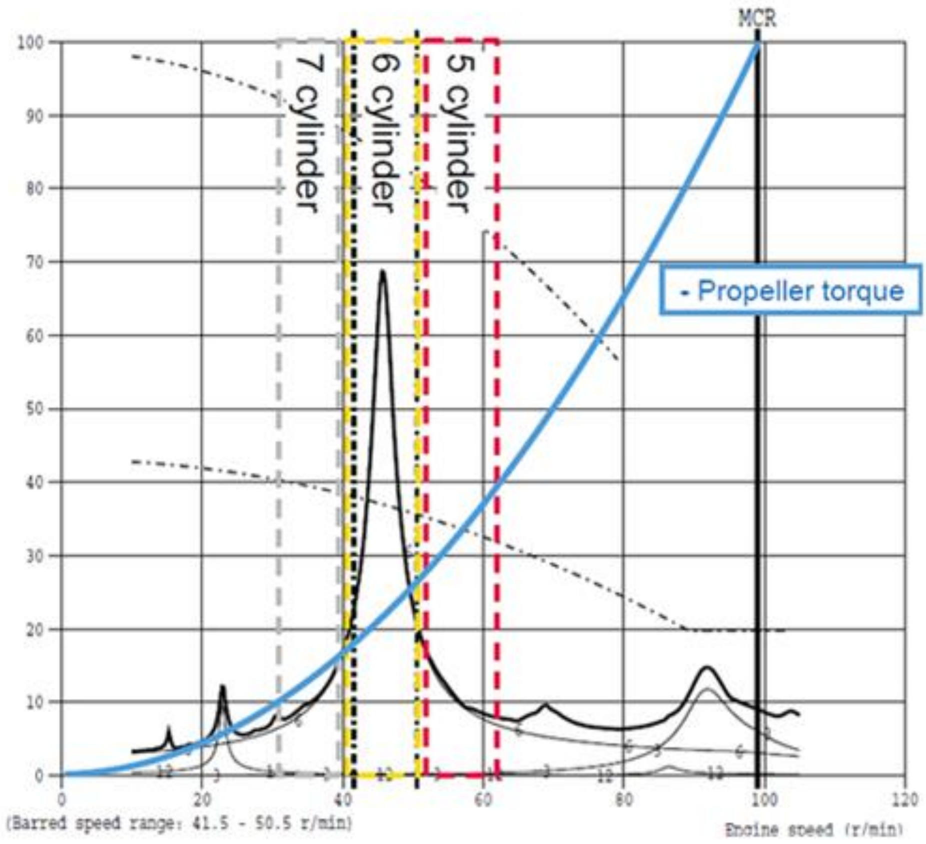

Fig.3: Barred Speed Range. [5].

VI. Global vibration analysis

Previously the ship builder and engine manufacturers have been adopting a procedure called local vibration analysis. But, to achieve low vibrations, the ship builders and engine manufacturers collaboration has shifted to a procedure called Global vibration analysis. This is used to investigate the overall vibration characteristics of whole ship. Includes excitation force, secondary excitation, resonance check and response check to measure vibration level with increase of RPM by using fixed monitoring system [1].

VII. G-Type engine counter measures forsecondary excitations

SDARI- Shanghai Merchant Ship Design And ResearchInstitute in collaboration with MAN Diesel & Turbo Diesel and gave corrective measures for secondary vibrations [1] [5] [7], which resulted in lower F_E. Some of the modifications done in collaboration with hulldesigner is, to make vertical bulkheads (longitudinal and transverse) as continuous as possible from deck to deck, heavy pieces of equipment are installed over beams, bulkheads, frames, or webs in both directions, avoid cantilevers to supportequipment,sturdy bearing foundations and double bottoms should be deep, and should change depth gradually if a lesser or greater height is needed to accommodate machinery or equipment.[3] [6].

VIII. Performance of global vibration analysis

The study of hull secondary vibration influence on main engine is termed as global vibration analysis in ship building. MAN Diesel & Turbo and hull designer (SDARI) jointly worked on the secondary excitations from hull and its influence on main engine vibration characteristics and performance [1]. MAN Diesel & Turbo has supplied the engine calculation model (5G60ME-C9) used by the hull designer when performing the global vibration analysis on a 64,000-dwt bulk carrier. The G-type delivers higher, but fully controllable, guide force moments compared with the S-type engine. Special attention has therefore been directed towards the structural vibrations related to this excitation source. The combination of global hull and local main engine vibration performance was carefully measured and analyzed, covering the full operational speed range of the vessel. The G-type engines had much lower resonance excitations than S-type. For G-type engines vibration limits were 8 mm/s at MCR but for S-type its much higher at 20 mm/s at MCR [1-2].

IX. Vibration levels and their Acceptability

There are two basic criteria for determining acceptability level of vibrations,

a) The vibration level must not result in stress levels that may cause fatigue damage to the engine, or the connected hull structure.

b) Vibration must not result in annoyance and/or discomfort for the operating personnel.

The limits applying to marine low speed diesel engines are shown in table.1. They aregiven as single order peak amplitudes.

ZONES	VALUE	STATUS
Zone 1	0-25 mm/s	Recommended
Zone 2	25-50 mm/s	Acceptable for Main Engine. Under adverse conditions, annoying/harmful vibrations may occur in the connected structure/vessel.
Zone 3	> 50 mm/s	Not Acceptable for Main Engine.

Table.1.: Vibrational limits.

X. Conclusion

Mechanical vibrations of steel structures are of a complex nature. When the steel structure comprises a ship and a two-stroke low speed diesel engine, a cogency of the excitation sources and the natural frequencies of the structures may lead to situations of annoying vibration unless due consideration is paid to this point [4]. Vibration measurements of the new G-type engine, show low and fully acceptable global vibration conditions of both M/E frame and vessel super structure. Comparing the G-type installation (Tankers / Bulkers), with other S-type installations, it is clear that the vibration response is reduced compared to most S-type installations. Main critical excitation frequency (Cylinders x speed / 60.) at MCR speed is well below H-mode nat. frequencies for G-types installed in Tankers / Bulkers. This results in low “flank” vibration at MCR speed. Avoiding problematic resonance conditions is a key factor in achieving a successful vibration performance.

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