Vibration Characteristics of Two-stroke Low Speed Diesel Engines

Abstract

This paper gives a general introduction to the vibration characteristics associated with two-stroke, low speed marine propulsion diesel engines, and ou#ines measures that can be taken to counteract any adverse influences arising in the ship.

A few years ago, vibrations were encountered in some ships propelled by engines with a low number of cylnders. These cases led to intensified investigations of the vibration conditions on board some of these ships, and prompted a further careful theoretical investigation into the vibratory excitation sources.

The vibratory conditions relating to the coupling between torsional vibrations of the propeller and axial vibrations of the shaft system, engine and huff wi// be thoroughly dealt with. These appear especialy where shaft system diameters have been increased considerably in order to avoid a "barred speed range':

Introduction

Developments in world economy during the last two decades have led to drastic changes in the traditions of the shipping and shipbuilding industries.

On the technical side, two-stroke, low speed diesel engines with a low number of cylinders have become very popular for the propulsion of ocean-going ships, mainly on account of their low instal!ation and operating costs.

Fig. 1 shows how the number of 4 and Scylinder engines has increased over the years at the expense of 7 and 8-cylinder engines. The same illustration also indicates how the stroke/bore ratio and the ratio between mean indicated pressure and maximum pressure have developed with the aim of reducing the specific fuel consumption and reducing the engine speed, with consequently increased propeller efficiency.

In the same period, it has bean verified that constant pressure turbocharging and uniflow air scavenging are the working principles which ~provide the lowest specific fuel consumption of two-stroke low speed diesel engines.

From a vibration point of view, the above changes have resulted in certain vibration characteristics playing a more domi- nant role than others. However, the fundamental excitation principles in the engine remain the same. The enclosed reference list refers to some recently published papers dealing with these subjects.

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.

As a major licensor, MAN B&W Diesel are obviously interested in having as many MAN B&W engines as possible installed with the optimum overall cost efficiency. With rward to vibrations, this means that the optimum combination of vibration countermeasures are to be implemented on every propulsion unit.

According to the authors' experience, actual contract conditions have in some cases prevented the necessary countermeasures from being taken or have even caused unnecessary countermeasures to be implemented.

Terminology

Before undertaking a detailed examination of the vibration characteristics of the diesel propulsion plant, it may be useful to study a simple mass-spring system in order to recapitulate the terminology used in a discussion of vibrations.

Fig. 2 illustrates the following:

1) Mass-elastic system:
model used to calculate the physical system comprising masses, spring and damping elements

2) Excitation:
Forces or moments acting on the mass-elastic system

3) Mode shape, or vibration modes, and natural frequency:
A characteristic deflection form of the mass-elastic system and a corresponding characteristic frequency at which the system can perform sinusoidal vibrations once excited, after which jt is left to vibrate freely

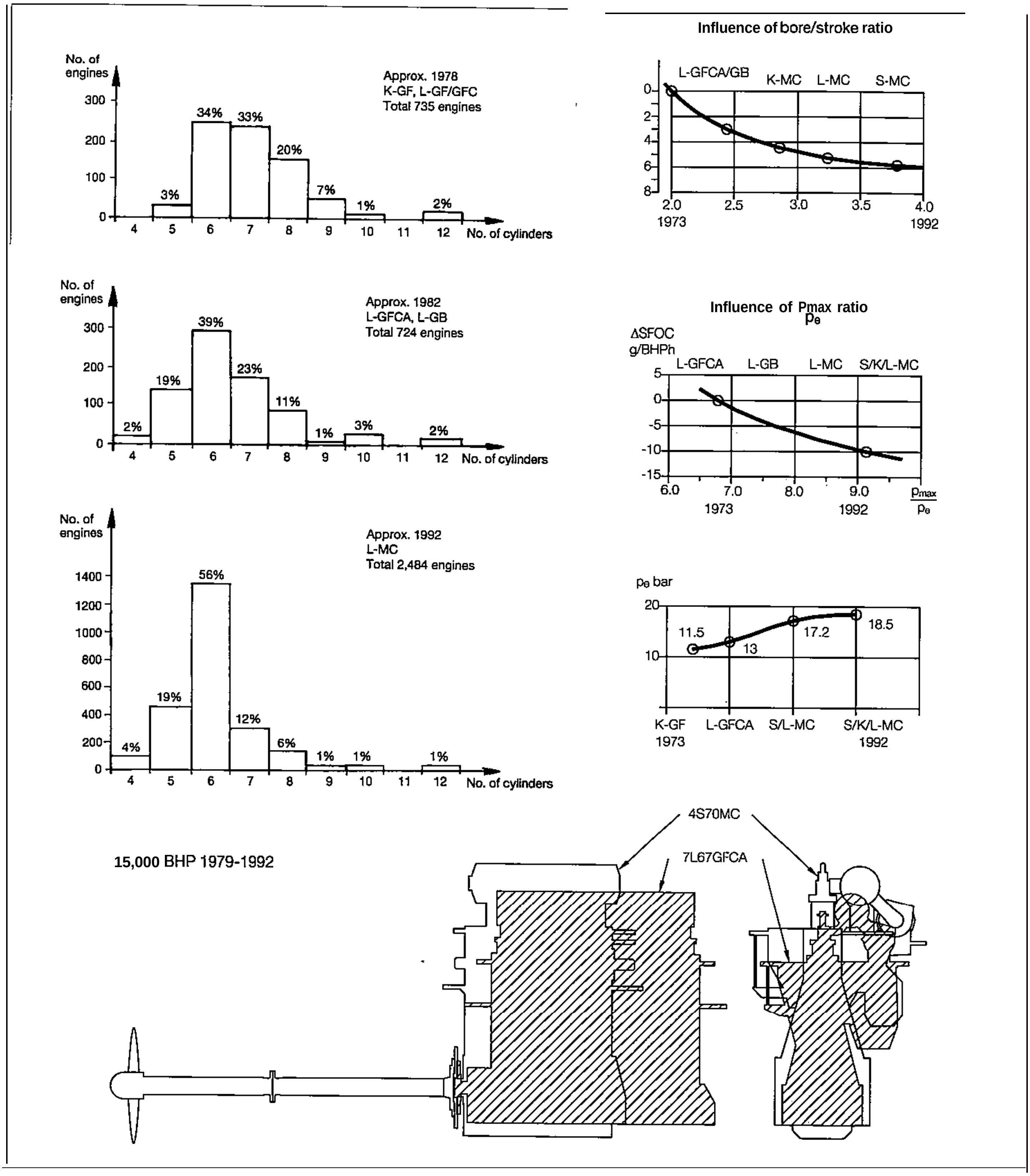
4) Harmonic excitation:
In the case of a periodic excitation, it is possible to describe the excitation as a sum of sine functions with different amplitudes, phase angles and periods (Fourier analysis). The periods of the sine functions will be 1, 1/2, l/3, i/4 ... of the period for the basic excitation. These sine excitation components are also called the lst, 2nd. 3rd, 4th order harmonic excitations

5) Resonance:
The frequency of a harmonic excitation coincides with the natural frequency of the mass-elastic system. Depending on the damping of the system, a considerable magnitication of the response will take place at resonance. Magnifications of 5 to 50 times will not be unusual

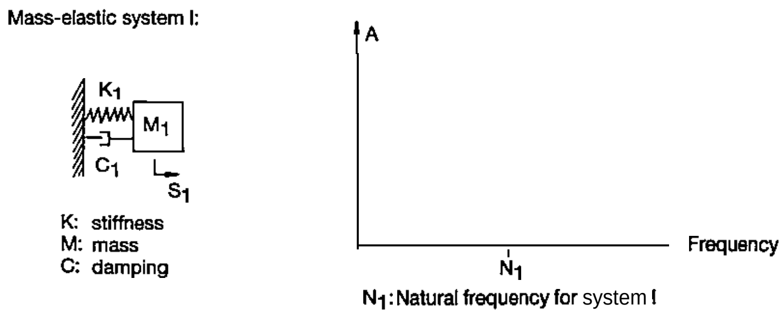
6) Main critical resonance is the condition at which the main harmonic excitation has resonance

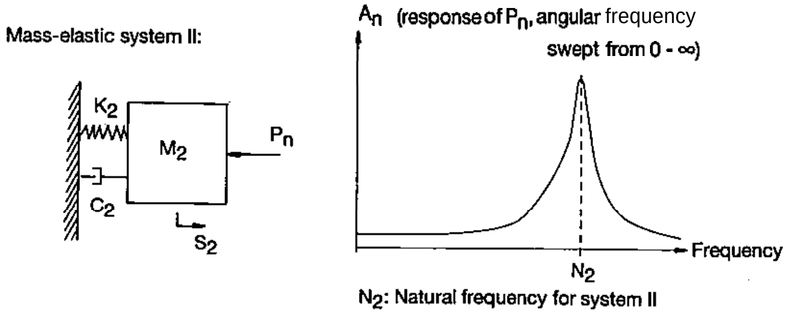
Overcritical condition refers to the condition at which the frequency of the main harmonic excitation is higher than the natural frequency.

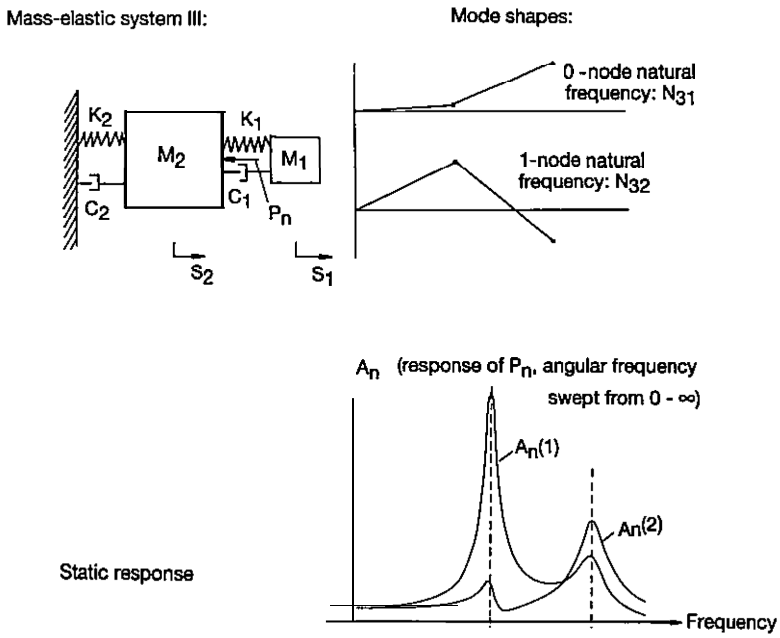
Conversely, undercritical condition refers to the condition at which the frequency of the main harmonic excitation is lower than the natural frequency.



Fig, 1: Developments in the parameters of two stroke low-speed diesel engines







P_n excitation of n'th order:

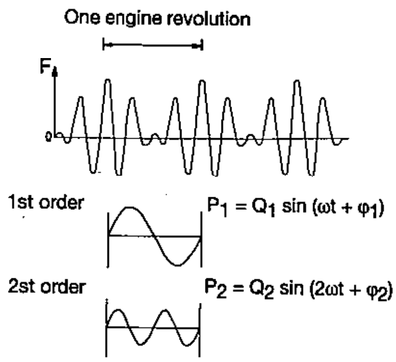
$P_n = \quad Q_n \sin(n\varpi t + \varphi_n)$

Q_n: n'th order excitation amplitude

ϖ: engine angular velocity

φ_n: n'th order excitation phase angle

A periodic excitation F can be split into a number of harmonic orders P_n:



$$F = \sum_{n=1}^{N} Q_n \sin(n\omega t \times \varphi_n)$$

(excitation synthesis)

X_n = response of n'th order

$X_n = A_n \sin(n\varpi t + \psi_n)$

A_n = n'th order response amplitude

ψ_n = n'th order response phase angle

For linear system n'th order excitation P_n will give only n'th order response X_n

$$S = \sum A_n \sin(n\omega t \times \psi_n)$$

(responses synthesis)

Sub- and super-harmonic response

Non-linear response of order different than the excitation order

These effects are considered in standard torsional vibration calculations

atural
equency: Characteristic frequency for the mass-elastic system

'ode shape: Characteristic "deflection form" in which mass-elastic system responds when excited at a frequency equal to the natural frequency

esonance: Where natural frequency coincides with excitation frequency $n\varpi$

Fig. 2: Explanation of vibration terms

Excitation - General

Excitations generated by the engine can be divided into two categories:

1) Primary excitations, which are forces and moments originating from the combustion pressure and the inertia forces of the rotating and reciprocating masses. These are characteristics of the engine as such, and they can be calculated in advance and be stated as part of the engine specification, with reference to a certain speed and power

2) Secondary excitations, stemming from a forced vibratory response in a sub-structure. The vibration characteristics of sub-structures are almost independent of the remaining ship structure

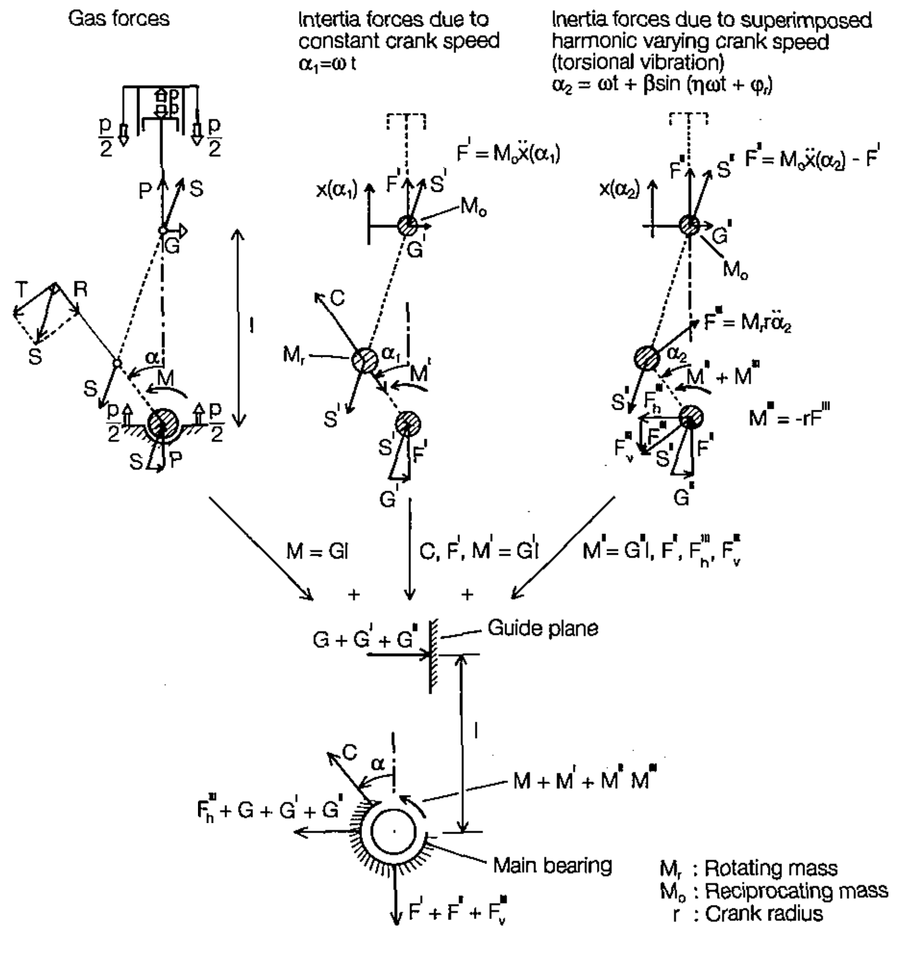
Examples of secondary excitation sources from sub-structures could be anything from transverse vibration of the engine structure to longitudinal vibration of a radar or light mast on top of the deckhouse. Such sub-structures of the complete ship might have resonance or be close to resonance conditions, resulting in considerable dynamically magnified reaction forces at their interface with the rest of the ship.

Secondary excitation sources cannot be directly quantified for a certain engine type, but must be calculated at the design stage of the specific propulsion plant.

Primary excitation sources

The primary excitation sources are very closely connected to the crankshaft/connecting rod mechanism and the engine process pressure acting through it. Even though the function of this mechanism is simple, it can be difficult to axplain the origin and distribution of its associated internal/external forces and moments.

Fig. 3 shows the forces and moments of a 1-cylinder engine. As an approximation for calculation purposes, the



S, S' and S'':
Connecting rod force acting on crosshead, equal to connecting rod force acting on crankpin, equal to force on main beating journals

M, M' M'' and M''':
Torque on main bearing journals from combustion pressure forces and inertia forces

T and R:
S, S' and S'' can at the crankpin be given as a sum of a radial component Rand a tangential component T

Resulting forces on engine frame in vertical direction: C, F', F'' and F_v'''

Resulting forces on engine frame in horizontal direction: C and F_h'''

Resulting moment on engine frame: $M + M' + M'' = l \times (G + G' + G'')$

Fig. 3: Resulting forces and moments on the engine frame from one cylinder

mass of fhe connecting rod has been divided into two and concentrated at the centre of the crankpin and the centre of the crosshead, respectively. This means that only inertia forces acting on two masses, i.e. the reciprocating mass at the centre of the crosshead and an equivalent rotating mass at the centre of the crankpin, need to be considered.

The gas force P will, through the connecting rod, act *on* the crankshaft with a torque M, causing an equivalent reaction torque on the engine frame G x I = M. M and G will contain harmonic excitations of all orders. In MAN B&W's experience, only excitations of the 1 st to 16th order need to be considered.

At a certain uniform speed of the crankshaft, an inertia force F arises from the accelerations of the reciprocating mass M_o, and a centrifugal force C acts on the rotating mass M_r. The force F will contain harmonic excitations of the 1st 2nd, 4th, 6th and higher even orders, however, normally only the 1st and 2nd order are taken into account. Force C will only give 1st order excitation.

For a multi-cylinder engine, Fig. 4. the firing order will determine the vectorial sum of the forces and moments from the individual cylinders.
Distinction should be made between:

External forces and moments, and
Internal forces and moments

The external forces and moments will act as resultants on the engine and thereby also on the ship through the foundation and top bracing of the engine. The internal forces and moments will tend to deflect the engine as such.

External forces and moments:

. *1st order moments* in vertical and horizontal direction. These are of equal size in MAN B&W engines with standard balancing.

. *2nd order moments* in vertical direction. 4th and higher even order external forces and moments will exist *on* engines with certain numbers of cylinders, however, they will be small and can be ignored.

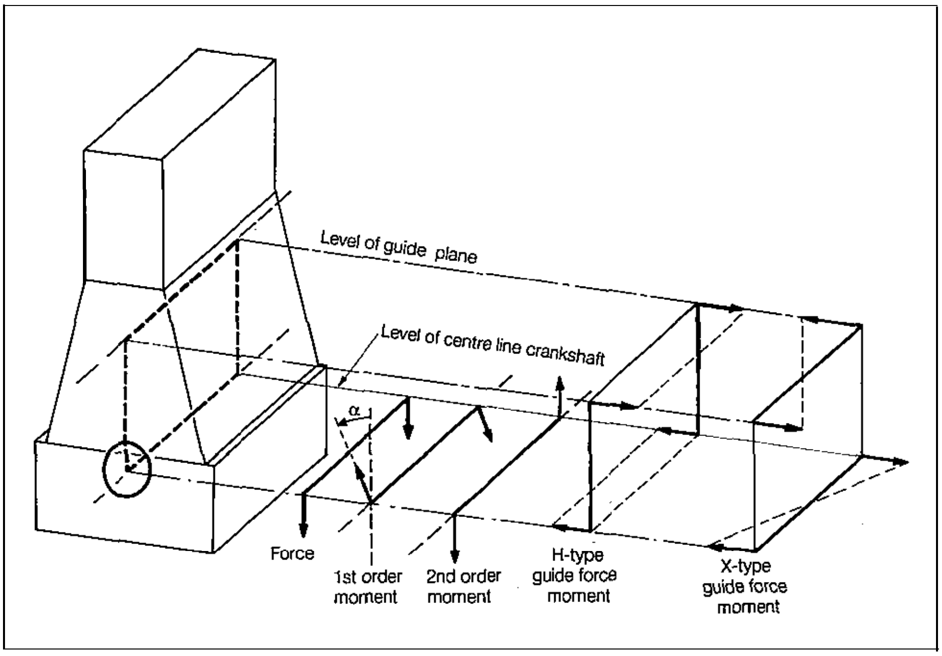


Fig. 4: Forces and moments of a multi-cylinder engine

- *The H-type guide force moment* is a moment between the stationary engine frame and the rotating/oscillating parts of the engine. From a practical engineering point of view, it should be applied to the engine frame as an external moment.

Internal forces and moments:

It is the responsibility of the engine designer to provide the engine frame with sufficient stiffness to cope with the internal forces and moments so that deflections and corresponding stresses can be kept within acceptable limits.

If the engine frame could be assumed to be infinitely stiff, internal moments and forces would not be able to give excitations to the ship's structure. However, it is obvious that an infinitely stiff engine frame cannot be obtained and, therefore, it is the relative stiffness between the engine frame and the connected hull structure which has to be considered.

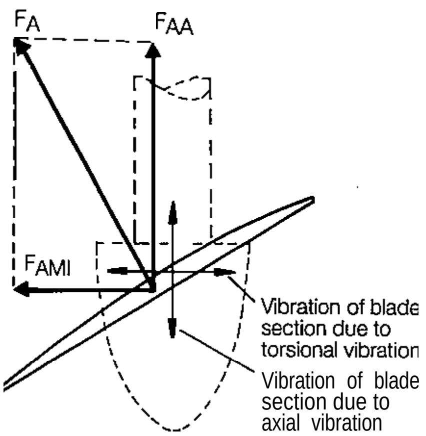
In MAN B&W's experience, the internal forces and moments of 1st and 2nd order, caused by the inertia forces on rotating and reciprocating masses, will not be able to excite vibrations in the ship.

The X-type guide force moment should, however, be taken into account because of its higher excitation frequencies and because it acts on the engine in one of its less rigid directions, particularly in the case of engines with a high number of cylinders.

Secondary excitation sources

Torsional vibrations

Torsional vibrations of the entire shaft system are mainly excited by the tangential force T, Fig. 3. Torsional vibration can, as will be demonstrated in the *next* paragraphs, excite vibration in the hull through the coupling phenomena present in the connecting rod mechanism and in the propeller.



Coupling between **vibratory torsional** torque and vibratory thrust due to added mass.

F_A: Total force on propeller blade from added mass will be perpendicular to the blade, independent of vibration direction

F_{AMI}: Force component contributing to added moment of inertia (entrained water) used in torsional vibration calculation

F_{AA}: Force component contributing to added mass used in axial vibrationcaluculation

Fig. 5: Torsional vibration induced propeller thrust

Torsional vibration induced moments and forces due to connecting rod mechanism
If a harmonic angular velocity is superimposed upon the normal uniform rotation of the crankthrow, as in the case of torsional vibrations, this will cause harmonic forces and moments to occur. However, due to the connecting rod mechanism, the reaction forces will not solely be of the same order as the superimposed torsional vibration, but significant orders of n-2, n-l, n+1 and n+2 will also appear.

One of the best known effects of this is the (n-2)th and (n+2)th orders in the tangential force T, which are responsible for the so-called sub- and super-harmonic torsional vibrations. But also external **forces** F'' of (n+1)th and (n-1)th order will appear for the 1 -cylinder engine, see Fig. 3.

Appendices A, B and C give, as examples, the values of these secondary forces and moments for relevant torsional vibration condition of multi-cylinder engines.

Torsional vibration induced propeller thrust
The propeller can be considered as a "screw", optimized to transform power from a uniform rotating torque into a uniform translatory moving force, pushing the ship (the propeller thrust).

With this concept in mind, it is not difficult to imagine that if a varying component is superimposed on the mean input rotational speed (or input torque) due to vibration of propeller and shafting, this variation will also appear in the propeller thrust. An investigation of such an effect is given in Ref. (2).

Fig. 5 shows that this coupling effect can be explained partly as an added mass effect, which is also in accordance with the theory in Ref. (2).

Hydrodynamic forces on the propeller due to vibration of propeller and shafting will also be able to set up pressure fluctuations on the hull surface above the propeller, which can give rise to annoying vibrations.

These phenomena have nothing to do with the non-uniform wake field.

Axial vibrations

Axial vibrations are excited in the crankshaft from the radial force R as well as the tangential force T, Fig. 3. The beforementioned torsional vibration induced propeller thrust will also excite axial vibration in the shaft system. Axial vibrations will create a reaction force in the thrust bearing which can be considered as an excitation source for the rest of the ship.

Propeller excitations due to non-uniform wake field

Excitations due to the propeller working in the non-uniform wake field will be transmitted to the hull either through the shaft system as forces and moments or through the water as pressure fluctuations acting on the hull surface,

The forces and moments should also be considered when calculating the torsional, axial, and lateral vibrations of the shaft system.

The excitation can be reduced by modifying wake field and propeller design, however, this subject is beyond the scope of this paper.

Vibration Modes, Their Excitation and Control

The general excitations listed in the previous section and the vibration modes on which they act will be discussed theoretically and illustrated by relevant examples in this section. Furthermore, available countermeasures will be discussed.

Torsional vibrations

The control of torsional vibrations is of vital importance for the propulsion plant because excessive vibration of this kind can lead to damage or even fracture of the crankshaft or the propulsion elements, such as intermediate shafts, propeller shaft, gears and flexible couplings.

This is also the reason why the classification societies, since the early days, have required calculation and verification by measurements for this kind of vibrations.

The classification societies prescribe two limits, τ_1 and τ_2, for the torsional stress in the speed range up to 80 per cent of MCR, see Fig. 6.

At engine speeds where the lower limit τ_1 is exceeded, it will be necessary to

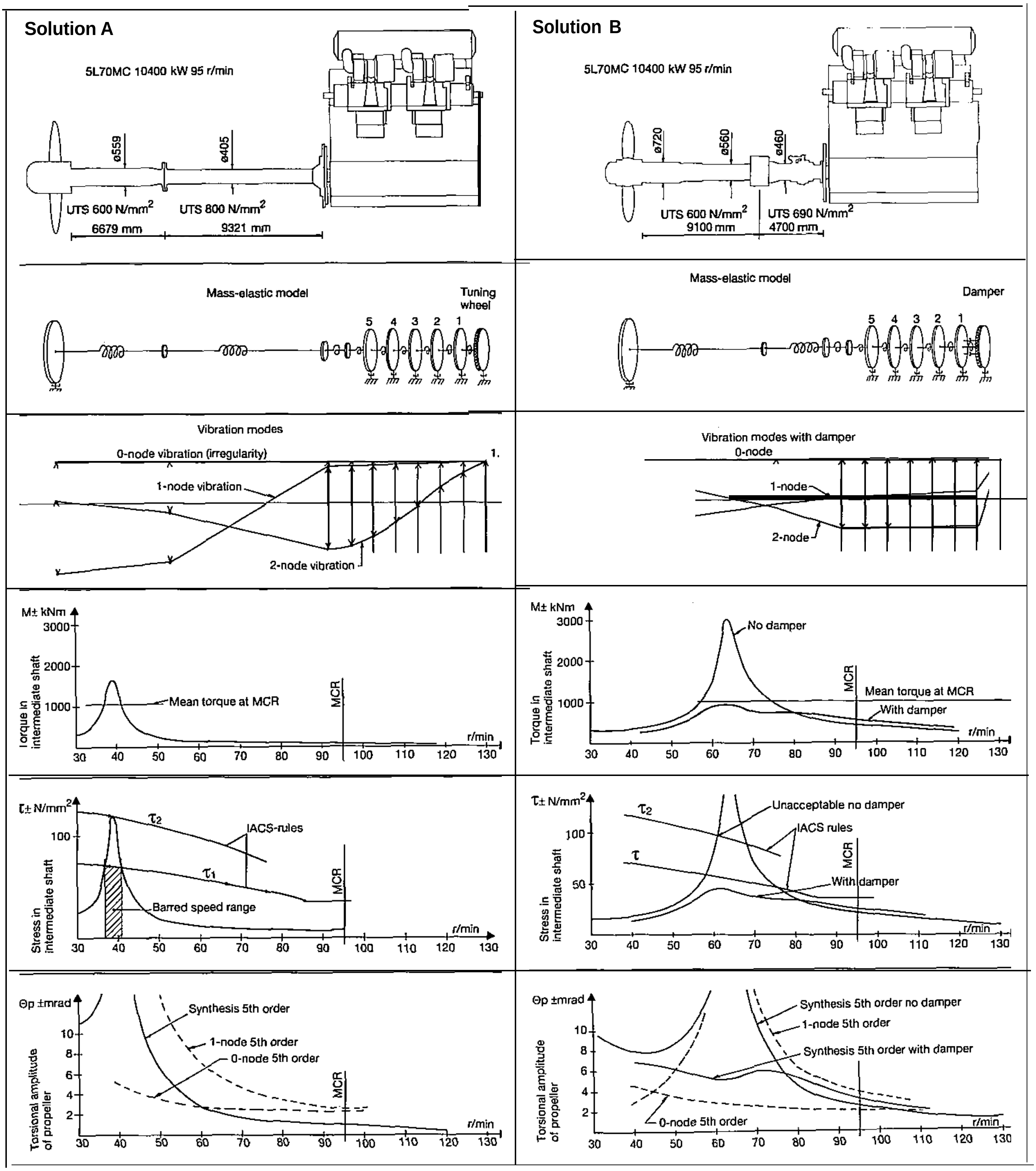


Fig. 6a and 6b: Different shaft systems for a 5L7OMC engine

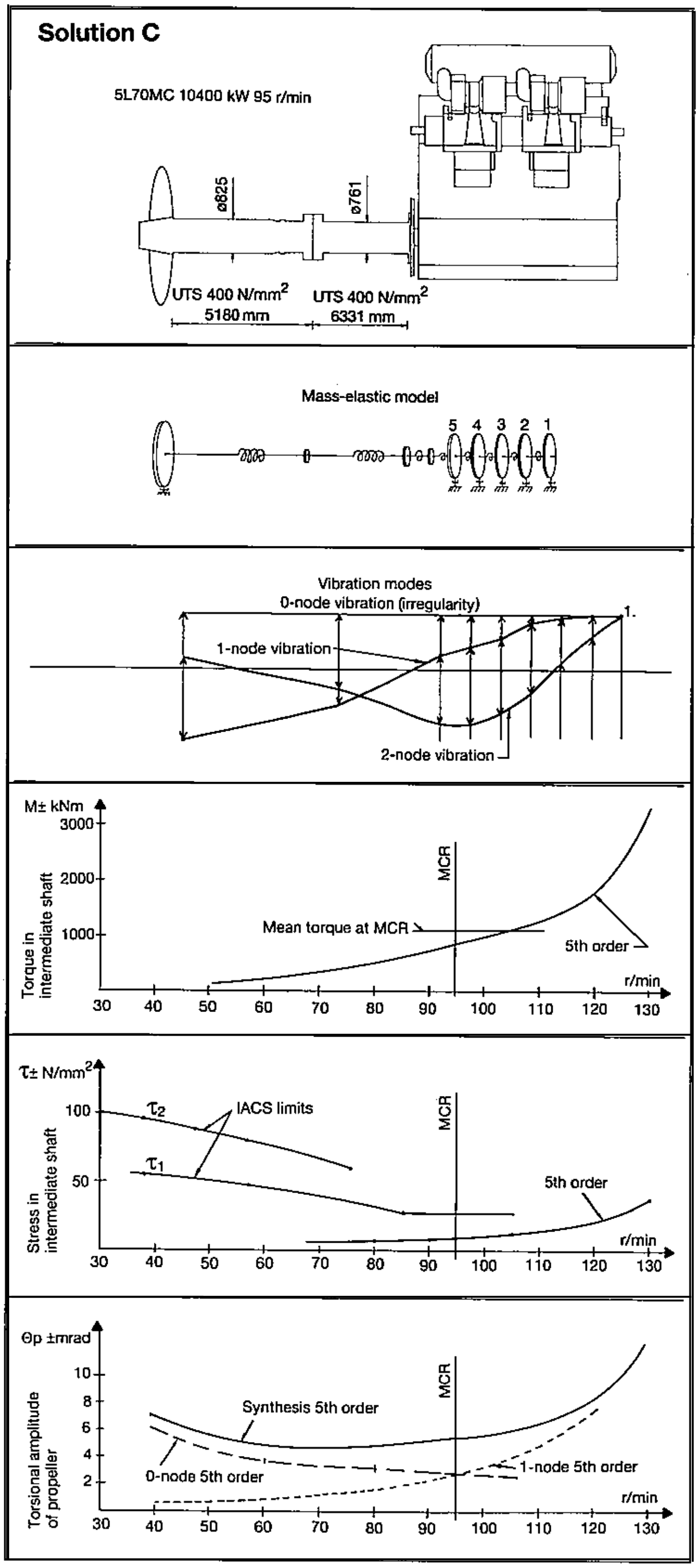


fig. 6c: Different shaft systems for a 5L7OMC engine

introduce a "barred speed range" in which continuous operation is prohibited. The upper limit τ_2 must not be exceeded. Above 80 per cent speed only limit τ_1 is applicable.

Ihe following propulsion systems and their torsional vibration characteristics will be treated in the following:

1) Engines with 4, 5 and 6 cylinders, directly coupled to the propeller

2) Engines with more than 6 cylinders, directly coupled to the propeller

3) Engines directly coupled to the propeller and with a small power take-off

4) Engines with a large power take-off and the possibility of disconnecting the propeller

Engines with 4,5 and 6 cylinders

With the conventional aft end engine installation, the torsional characteristics of these engines are dominated by a resonance of the 1 -node torsional vibration mode excited by the harmonic order equal to the cylinder number (i.e. 5th order 1 -node resonance in case of a 5-cylinder engine, referred to as main critical resonance).

This resonance will normally occur somewhere in the middle between minimum and maximum speed of the engine, mainly depending on the lengths and diameters of the shaft system (i.e. total torsional flexibility of shaft system between propeller and engine).

The response at resonance will lead to torsional stresses in the shaft system which have to be compared to the limits stipulated by the classification society in question. The magnitude of the resonance stresses will depend on the excitations and the damping of the system. Generally, it can be said that the excitation increases with increasing engine speed. The system damping will depend on the ratio between moment of inertia for propeller and engine, the damping of the propeller, and the presence of possible torsional vibration dampers.

Figs. 6 and 7 illustrate three possibilities A, B and C, which are relevant for the "layout" of the shafting system for a 5-cylinder engine directly coupled to the propeller, but which have widely different torsional vibration conditions.

Solution *A* is characterised by a relatively flexible shaft system. The material strength has been increased in order to reduce the diameter of the shafts, thereby making them even more flexible. A tuning wheel has been mounted on the front end of the crankshaft to increase the ratio between engine and propeller mass moment of inertia, resulting in higher damping in the system. The resonance will occur below engine MCR speed (over-critical).

The torsional stresses will be below the 2-limit, but above the 1-limit, with a 'barred' speed range as a consequence.

Solution *B* is characterised by a relatively stiff shaft system, e.g. due to the short distance between propeller and engine or due to implemented stiff shaft elements such as shrink-fit couplings, oil distribution box to CP-propeller and ice class requirements on shaft diameters. This has brought the main critical resonance relatively close to MCR, and the resonance stresses will exceed the upper limit 2 prescribed by the classification society. In this case there will be four possibilities:

1) Mounting a torsional vibration damper of appropriate size which will reduce stresses to below the 2 limit, and the plant will have a barred speed range

2) Mounting a torsional vibration damper of appropriate size which will reduce stresses to below the 1 limit, and the plant will have no barred speed range

3) Increasing shaft diameters in order to move the main critical resonance to above MCR (solution C)

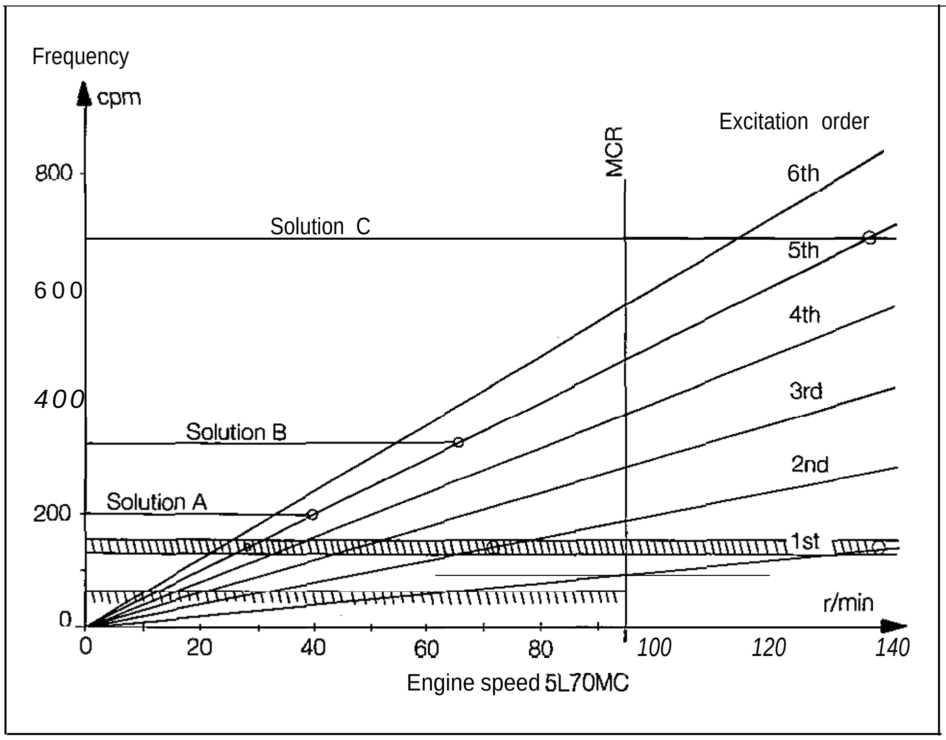


Fig. 7: Engine speed versus excitation frequency. Natural frequencies for 1 -node torsional vibration modes art? indicated for solutions A, B and C. Optimum positions of natural frequency of PTO systems indicated by shaded areas

4) QPT (Quick Passage through a barred speed range Technique) for CPP-installation, Ref. 12

The procedure can be described as follows:

a. The carrying out of ordinary torsional vibration calculations in maximum pitch condition

b. The carrying out of ordinary torsional vibration calculations in minimum pitch condition using rather pessimistic propeller damping values

c. The establishment of a simulation model of ship/propeller/shafting/engine/governor which, in the barred speed range during steady state operation, Fig. 8 (upper part), gives results coinciding with the results of the ordinary torsional vibration calculations

d. The simulation of engine starting and stopping with rapid passage through the barred speed range, Fig. 9 (upper part)

e. The evaluation of stresses in the barred speed range based on results of the start and stop test simulations for rapid passage through the barred speed range

In order to ensure the rapid passage through the barred speed range, a so-called "critical speed unit" should be installed. This unit operates on the speed setting signal in such a way that automatic rapid passage through the barred speed range is obtained when the engine is operated via the bridge manoeuvring system, as well as when operating from the engine control room

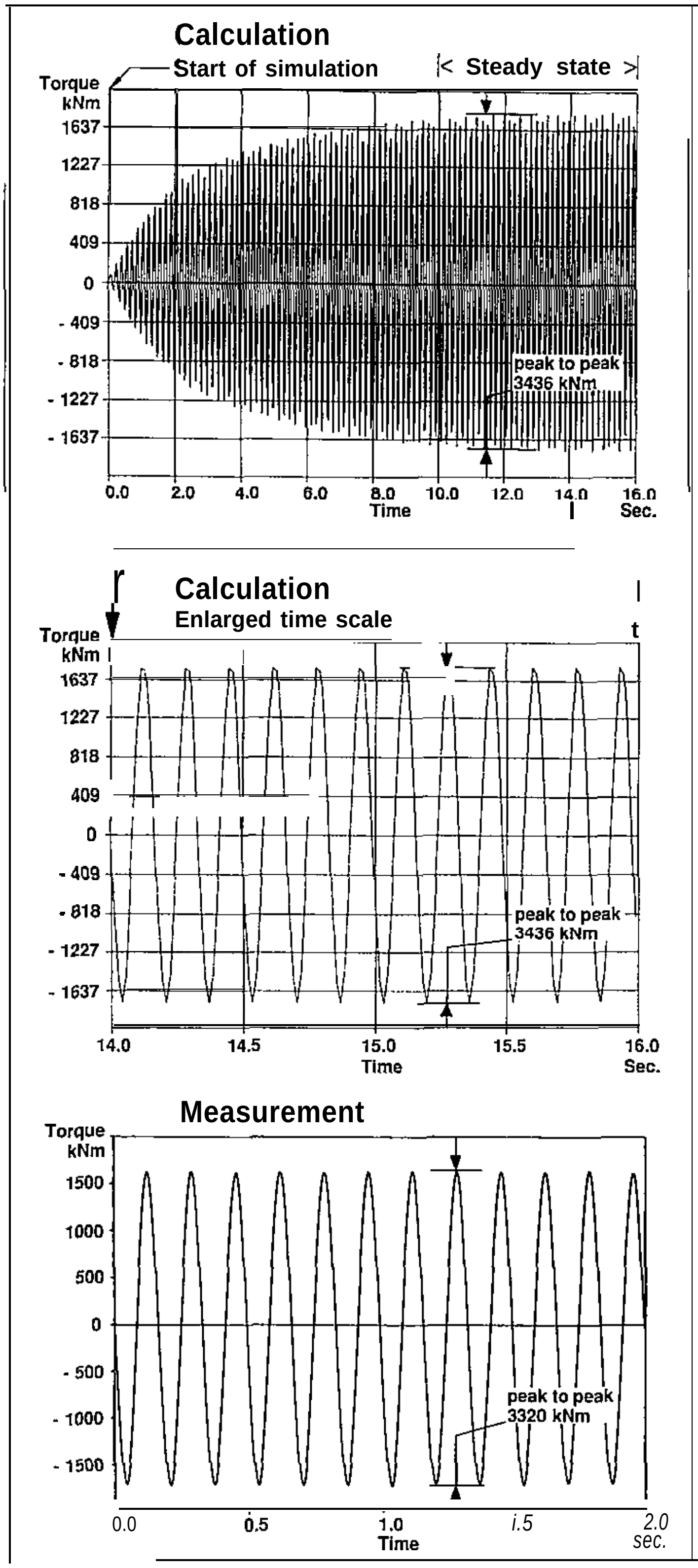


Fig. 8: Simulated steady state torque and measured torque at minimum pitch. (As an example, a 5L50MC engine is used)

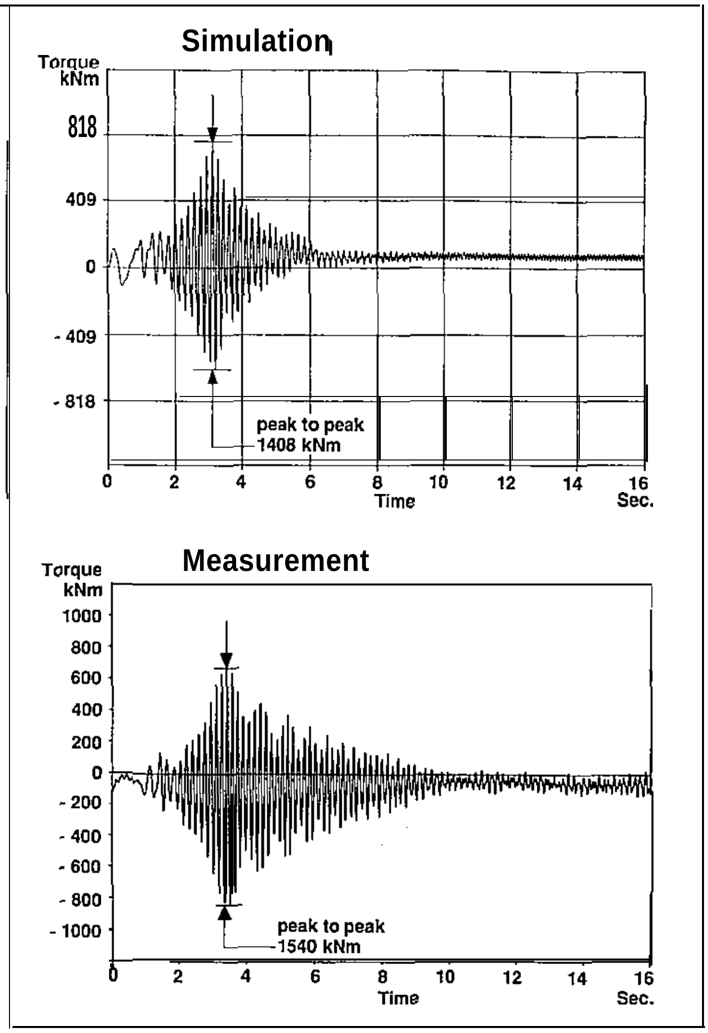


Fig. 9: Comparison between simulated and measured torque in the intermediate shaft during starting of main engine and passage through the barred speed range (minimum propeller pitch). (As an example, a 5L50MC engine is used)

Solution C is obtained by increasing the diameter of shafts until the main critical resonance is positioned approximately 40-45 per cent above the nominal speed (undercritical). Due to the large shaft diameter (large moment of resistance), only moderate torsional stresses appear even though the varying torsional torque in the shaft is high.

Solution C is chosen either as an unavoidable consequence for a very short shaft system or because the shipowner has specified that the ship must not have any barred speed range.

Besides avoiding a barred speed range, solution C is characterised by a rather high varying torque in the shaft which,- as already explained, will induce a rather high varying thrust, called torsional vibration induced propeller thrust.

It should be mentioned that under adverse conditions the varying thrust can reach levels of up to 50% per cent of the mean thrust, which is far above what a propeller designer would accept as an excitation from the non-uniform wake field.

Of the three alternatives, A will normally involve the lowest cost. Solutions A, B or C cannot be directly related to a specific engine type, only detailed torsional vibration calculation at the design stage will reveal the optimum solution.

Trends in the choice and feasibility of the solutions can be summarised as follows:

4-cylinder engines:
- Solution A:
  less feasible -not very common
- Solution B:
  feasible -not very common
- Solution C:
  feasible -very common

5-cylinder engines:
- Solution A:
  feasible -very common
- Solution B:
  feasible -not very common
- Solution C:
  feasible -common

6-cylinder engines:
- Solution A:
  feasible also without tuning wheel – very common
- Solution B:
  feasible -not very common
- Solution C:
  not feasible

Engines with 7 or more cylinders

For such engines, the 1 -node main critical resonance is not normally important, because it will occur close to or below the minimum speed of the engine. Furthermore, the 7th order or higher order excitations of the torsional vibration are considerably smaller than the 4th, 5th and 6th order, which means that barred speed ranges are not normally required.

However, the Z-node torsional vibration mode (one node in the crankshaft) begins to be important and needs attention.

Major resonances for this vibration mode should be avoided close to MCR. Small torsional vibration dampers might be relevant.

Engines with small power take-off

It has become very popular to connect a power take-off (PTO) to the crankshaft or propulsion shaft system. Some years ago, also exhaust gas driven power turbines were introduced, delivering their power to the propulsion shafting power take-in (PTI). A common feature of both the PTO and the PTI is that they are connected to a gear system which needs protection from the relatively high torsional excitation from the crankshaft.

In the MAN B&W standard designs, the PTO and PTI are mounted on the fore end of the crankshaft, as a compact unit, and the protection is obtained by installing an elastic coupling between the propulsion shafting and the abovementioned gear.

Normally, when the PTO/PTI represents a power of less than 10 per cent of the main engine power, the vibration modes of the PTO/PTI system will not influence the vibration modes of the propulsion shaft system. This means that the main propulsion shaft system can be designed and determined regardless of whether a PTO/PTI is to be installed later on.

As a “rule of thumb”, the lowest natural frequency of the PTO/PTI mass-elastic system should not be higher than 75 per cent of the frequency corresponding to the main engine speed. This will give low torsional loads in the PTO system due to the fact that overcritical vibration condition is obtained for all harmonic excitations from the main engine at MCR, see Fig. 7.

Engines with large power take-off

Certain types of ships, such as ferries, cement carriers and shuttle tankers, have operating conditions that require high auxiliary power, simultaneously with propulsive power. This has been met by controllable pitch propellers combined with relatively large power take-offs comprising clutches, elastic couplings, gears, generators and/or hydraulic power packs.

The torsional vibrations of such installations are very complex, and need careful investigation during the design stage.

The design philosophy with respect to torsional vibrations in a power take-off is:

- The elastic couplings are necessary to facilitate alignment and to protect the gears from the high frequency torsional excitation of the main engine. Such excitation may, in combination with backlash, produce harmful vibration in the gear
- The elastic coupling, or couplings, should be sufficiently flexible to ensure a natural frequency in the PTO-system of either approx. 1.5 times the mainengine speed, or below 0.75 times the main engine speed, see Fig. 7. This will give main critical resonances in the PTO-system (4th, 5th and 6th order) at very low speed or even below the minimum speed. Furthermore, the 1 st and 2nd order excitation, which becomes dominant in case of misfiring, will have resonance away from the nominal speed. Such tuning of the natural frequencies will normally require very elastic couplings

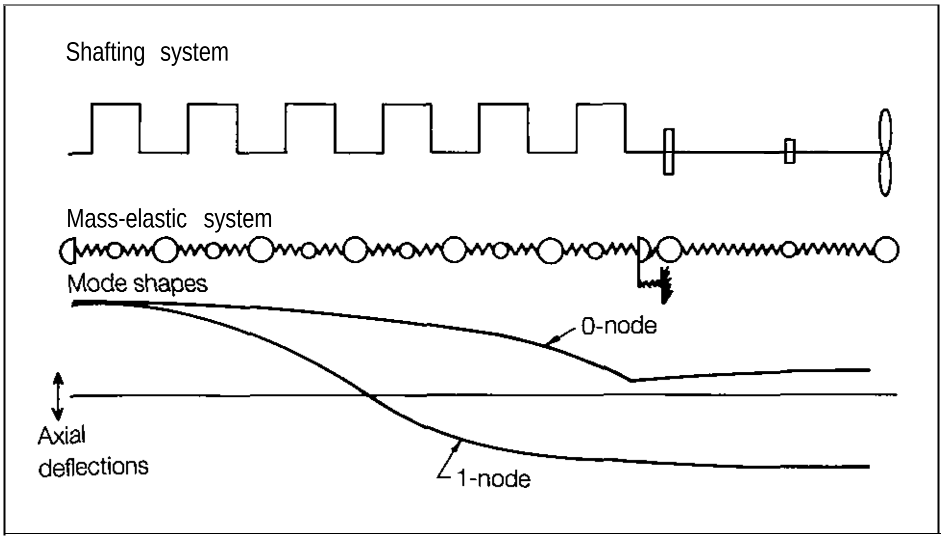


Fig. 10: Mass-elastic system for axial vibration calculations. Axial deflections of O-node and I-node mode shape

The combination of low natural frequency and high moment of inertia in the PTO-system will require special facilities in the engine governor if instabilities in the system are to be avoided.

Axial Vibrations

Axial vibrations are longitudinal shafting vibrations. Fig. 10 shows the mass-elastic system used for axial vibration calculations and the mode shapes of the two lowest modes which are of relevance.

MC engines with more than six cylinders will have main critical resonance with O-node vibration mode below MCR speed. For 4,5, and 6-cylinder engines, the main critical resonance will occur outside the normal speed range. However, for 5cylinder S-MC and 6-cylinder K-MC, L-MC, and S-MC engines, the main critical 5th and 6th order resonance, respectively, will be situated very close to MCR speed.

The 1 -node vibration mode is normally of less importance. Its natural frequency is determined by the mass and stiffness of the entire shafting system. Especially the stiffness of the thrust bearing and its support is very decisive.

Normally, the natural frequency is so high that no dynamic amplification of this mode will occur.

Axial vibrations are excited by:

. Radial and tangential components of the combustion pressure and mass forces in the individual cylinders. The fact that the p_{max}/p_e ratio of modern engines has increased considerably (see Fig. 1) means that especially the radial components of orders higher than the 4th order have increased

. Propeller excitation of the blade frequency and multiples hereof from the non-uniform wake field

Excitations caused by *responses* from other vibration modes, such es:

1) Torsional vibration induced propeller thrust, the magnitude of which depends on how the torsional vibrations (see Fig. 6) are situated. This excitation may initiate heavy varying forces in the thrust bearing

2) Coupling of torsional vibrations of the crankshaft to responses in the axial direction (mechanism: twist of crankshaft will cause axial deflection). This coupling depends on the geometry of the crankshaft and is found where pronounced torsional responses exist. For engines with a relatively low number of cylinders, it will almost exclusively be found in connection with barred speed ranges

For the following reasons, the axial vibration damper is standard for *all* cylinder numbers of MC engines:

- First and foremost, the axial vibration amplitudes are to be kept below a certain level to protect the crankshaft against too heavy extra stresses caused by axial vibrations. For this reason, MC engines with six or more cylinders are provided with an axial vibration damper

- The second reason for installing an axial vibration damper is to be able to control varying forces in the thrust bearing, which may excite the hull structure. In order to control these varying forces, 4 *and 5-cylinder engines* are also provided with an axial vibration damper

The axial vibration damper effectively reduces the varying forces generated in the crankshaft and acting on the thrust bearing. The varying forces originating from torsional vibration induced propeller thrust are, on the other hand, left practically unaffected by the axial vibration damper.

Fig. 11 shows measurements of the varying thrust in the intermediate shaft of a 4L60MCE engine with an active axial vibration damper. The figure shows the thrust originating from the 8th order 1 -node torsional resonance and the 4th order flank. With an inactive damper, the magnitude of the varying thrust is the same.

For plants on which the torsional vibration induced propeller thrust is negligible, it will still, despite the use of an axial vibration damper, be necessary to

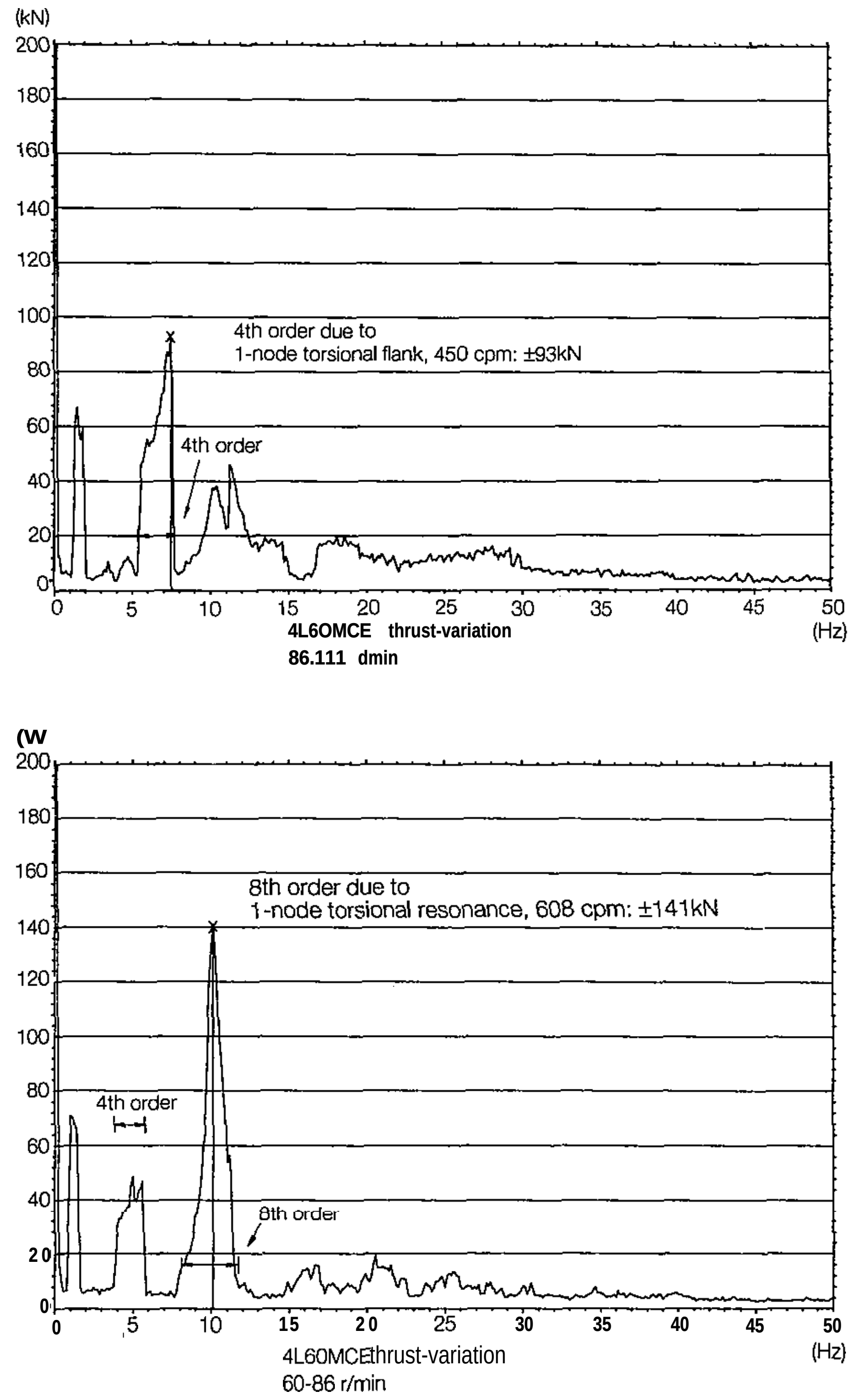


Fig. 11: Torsional vibration induced propeller thrust measured in the intermediate shaft of a 4L60MCE engine. Response of 8th order 1 -node torsional resonance and 4th order flank is seen

predetermine the varying force in the thrust bearing. The damper will leave a varying force in the thrust bearing of a magnitude corresponding to the static deflection of the crankshaft caused by the mass and gas forces. On account of the evolution in the p_{max}/p_e ratio of modern longstroke engines, especially the radial components of these forces have increased, resulting in a considerable static deflection of the crankshaft.

The above shows that the axial vibration and related torsional vibration problems should be solved at the design stage of the ship through cooperation between the engine builder and the shipyard. Especially the force in the thrust bearing should be determined in each individual case by combining the forces from the torsional vibration induced propeller thrust, the forces from axial vibrations of the crankshaft, and the forces from coupled axial and torsional vibrations in the crankshaft.

In order to be able to present solutions to such cases, MAN B&W use a computer program including FEM (Finite Element Method) based crankshaft models (Fig. 12) which allows the firing order, crank throw geometry, and bearing stiffnesses to be represented. It also allows the model to be excited directly by the tangential and radial forces actingon the crankpins, Fig. 13.

An example:
When designing the shafting for a 5-cylinder engine, a choice can be made between an overcritical layout, i.e. using small diameter shafts and, if necessary, a barred speed range, or an undercritical layout, i.e. the use of large diameter shafts and no barred range.

The reason for the interest in 5-cylinder engines is that many of these installations are apparently designed for undercritical operation without sufficient allowance being made in the hull structure for excitations from the thrust bearing originating from axial vibrations and the vibrations coupled to them.

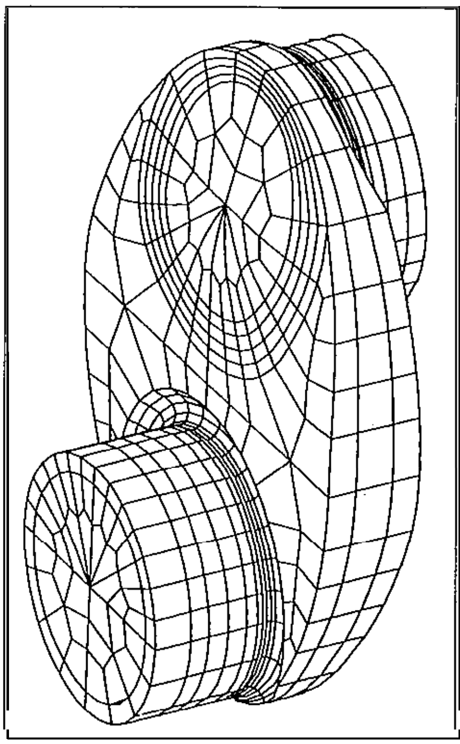


Fig. 12: Half crankthrow FEM-mode/led in order to carry out vibration and stress analysis. Complete crankshaft is mode/led from this half crankthrow by the dedicated system HIFINEL-CRANK

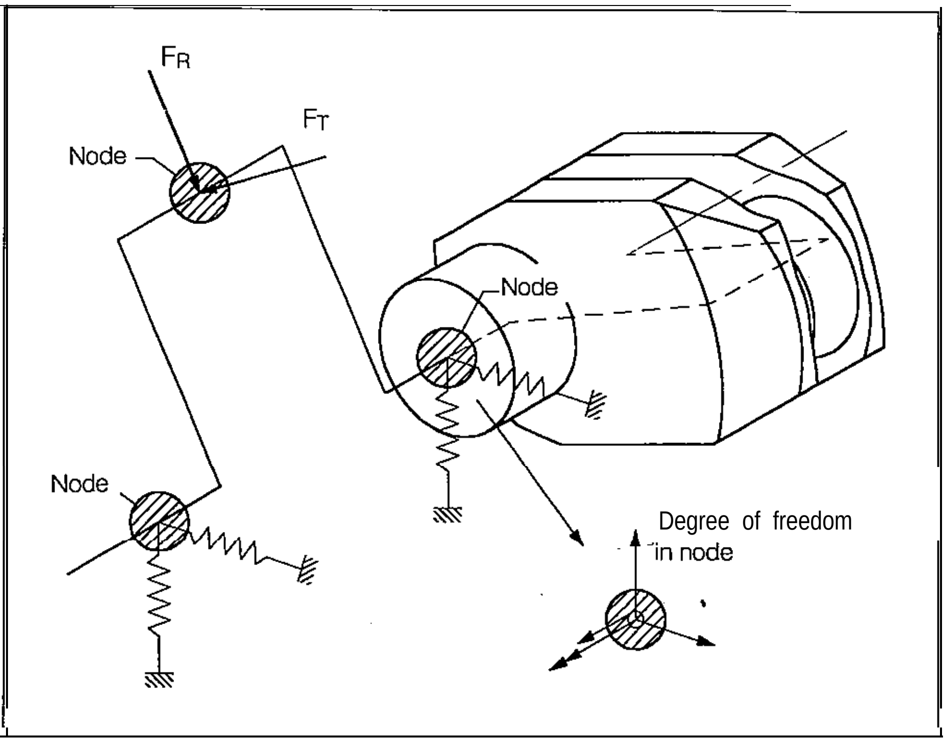


fig. 13: Crankshaft model

To illustrate the issues with, in particular, undercritical operation, it will be necessary to recapitulate some conclusions relating to torsional vibrations in the two layouts.

The aspects of the undercritical layout are illustrated in Fig. 14, upper. At MCR, the 5th order l-node torsional vibration resonance is situated above MCR. The 5th order O-node torsional vibration mode, normally referred to as irregularity, can be considered as over-critical. Thus it can be assumed that the torsional amplitude on the propeller is a sum of the two contributing torsional amplitudes.

When passing through a resonance, Fig. 14, lower, the phase of the pertaining amplitude is changed 180 degrees. Accordingly, when the shaft system layout is overcritical at MCR, relative to both the 5th order 1 -node torsional mode and the O-node torsional mode, the conditions in Fig. 14, centre, are obtained. The torsional amplitude on the propeller is the difference between the two contributions. As these are typically of the same magnitude, the resultant propeller amplitude at MCR is small compared with that of an undercritical layout.

In the case of undercritical layout, the resonance at the main critical order (in this case 5th order) should be situated 40-45% above MCR. Measuring results on the thrust bearing of a 5L70MC confirm the calculations of the 5th order varying thrust. Depending on propeller and shafting, values of +200 to +400 kN have bean found at MCR.

Corresponding values for the 5th order varying thrust on the thrust bearing, when having overcritical layout, are typically 60 to 100 kN.

For a shafting design where the main critical is situated below but rather close to MCR, and a torsional vibration damper is used to control the stress-levels (Fig. 6, solution B), the result will be a relatively high torsional amplitude at the propeller. This will give a significant contribution to torsional vibration induced propeller thrust.

The FLS-M vibration compensator, Fig. 15, has been successfully applied as a so-called thrust pulse compensator in order to reduce the torsional vibration induced propeller thrust on 5-cylinder engines that are coupled to large-diameter shafting.

The thrust pulse compensator counteracts the varying thrust from a position on a foundation on the tanktop close to the thrust bearing; Fig. 16, of a 5L60MC engine.

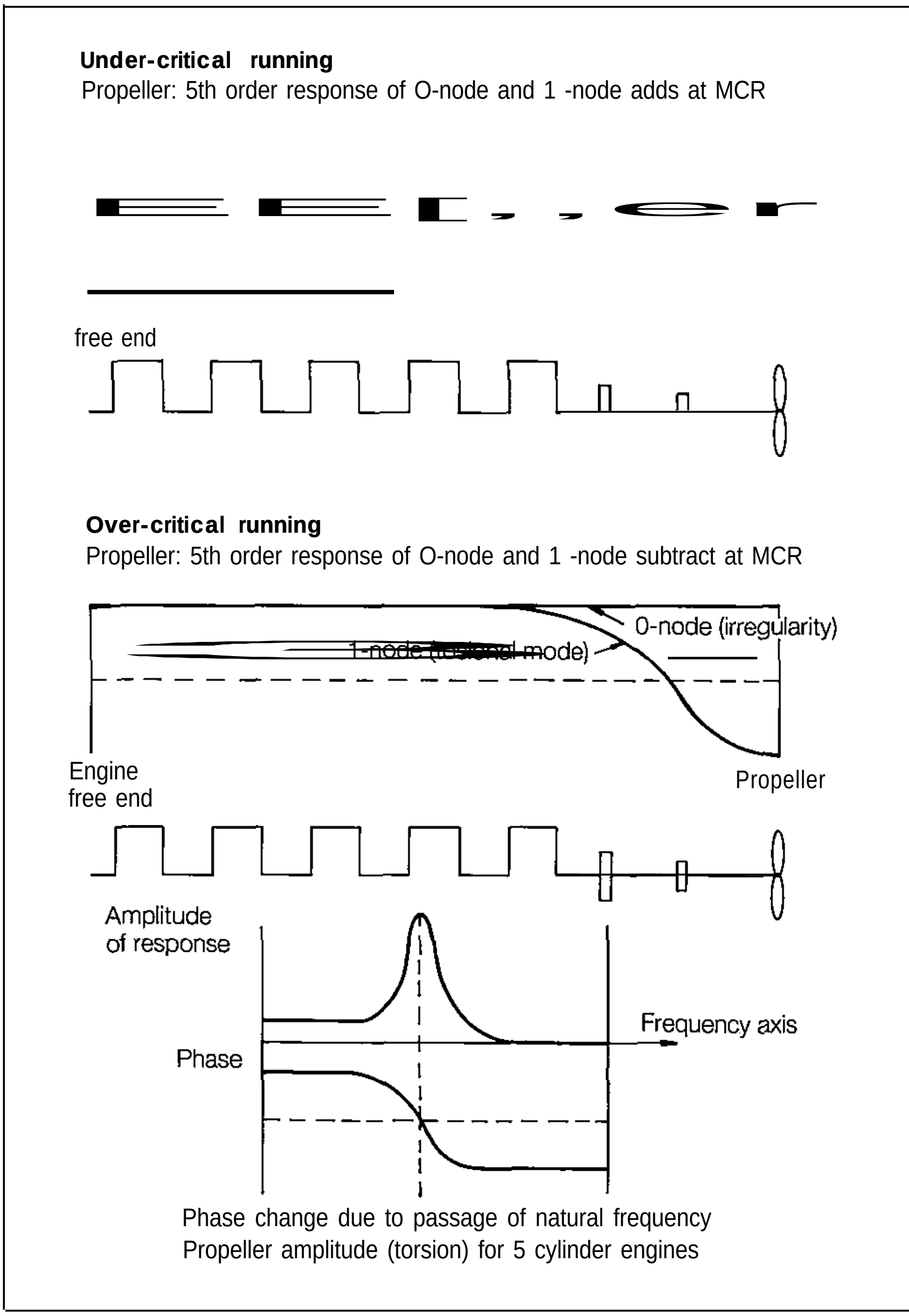


Fig. 14: Torsional response for 5-cylinder engines shown in order to illustrate torsional vibration induced propeller thrust
Upper : situation when running under-critical
Centre: situation when running over-critical
Lower : phase change when passing through a resonance

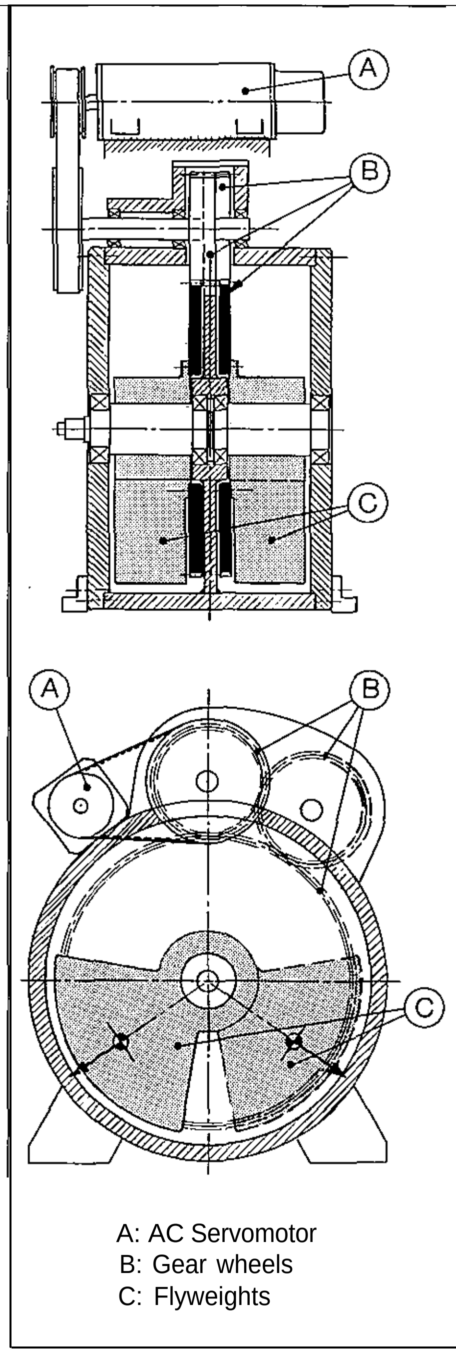


Fig. 15: FLS-M Vibration compensator

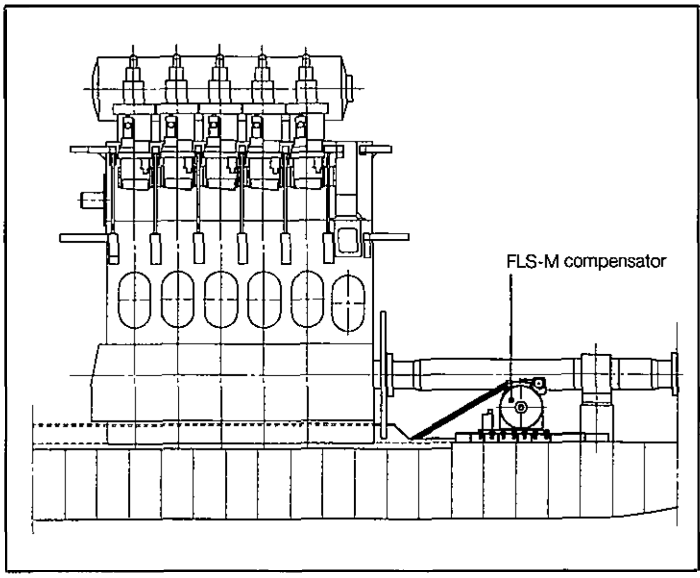


Fig. 16: FLS-M Vibration compensator used as a thrust pulse compensator

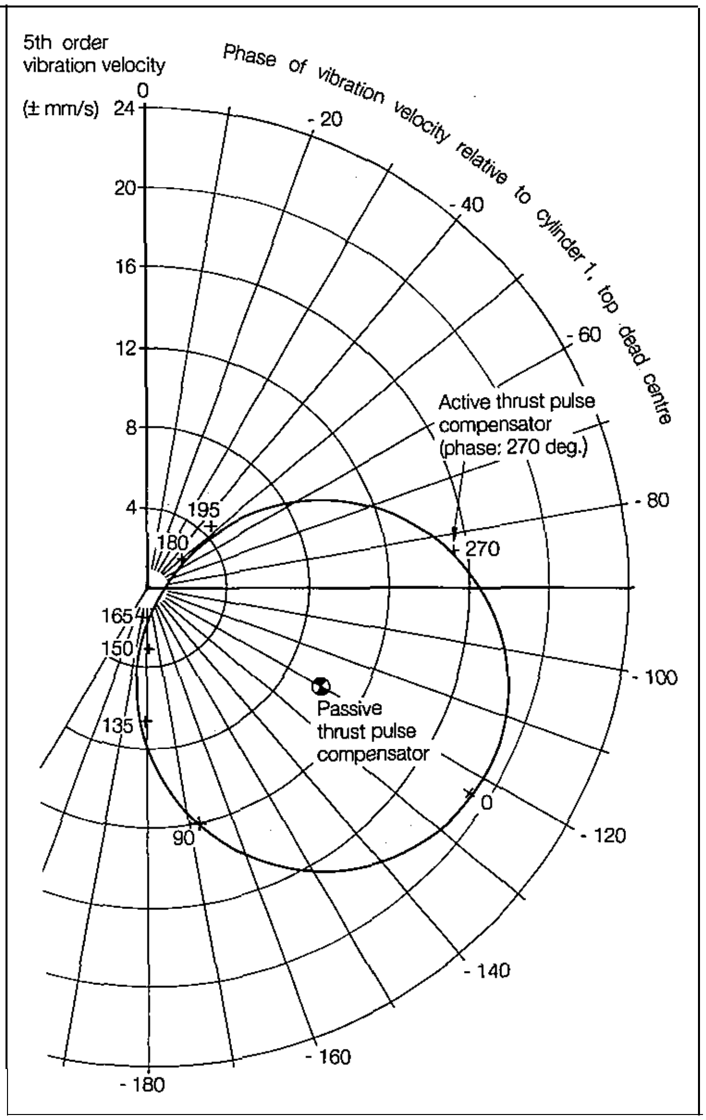


Fig. 17: 5L60MC, Measurements at 92 r/min at the wheelhouse in longitudinal direction

Fig. 17 outlines the results of vibration measurements at the wheelhouse level of the superstructure in the longitudinal direction. It is seen that with the optimum phase of the counterweights of the thrust pulse compensator, a reduction of the vibration level with a factor of 8-10 is obtained.

External Forces and Moments

The entire ship forms a mass-elastic system, with natural frequencies and vibration modes. The horizontal and vertical bending modes of the hull girder and the corresponding natural frequenties can be calculated. Determination of vibration modes with 4, 5 and more nodes requires comprehensive calculating procedures, whereas modes with 2 and 3 nodes can be calculated by more simple procedures.

In order to obtain the correct decision basis at the engine contract stage, information about these vibration modes should be available, as this will make it possible to decide the measures to be taken to control the responses from these vibration modes.

Hull girder vibration modes are excited by forces and moments acting on the hull girder, Fig. 18. Excitations of hull girder vibration modes originating from the engine are external forces and moments generated by the inertia forces of unbalanced rotating and reciprocating masses.

For MAN B&W engines, the external forces can -for all practical purposes – be considered to be zero, due to their small size. Normally, only the external moments of 1st and 2nd order need to be considered. However, modest moments of other orders exist; an example is shown in Appendix A.

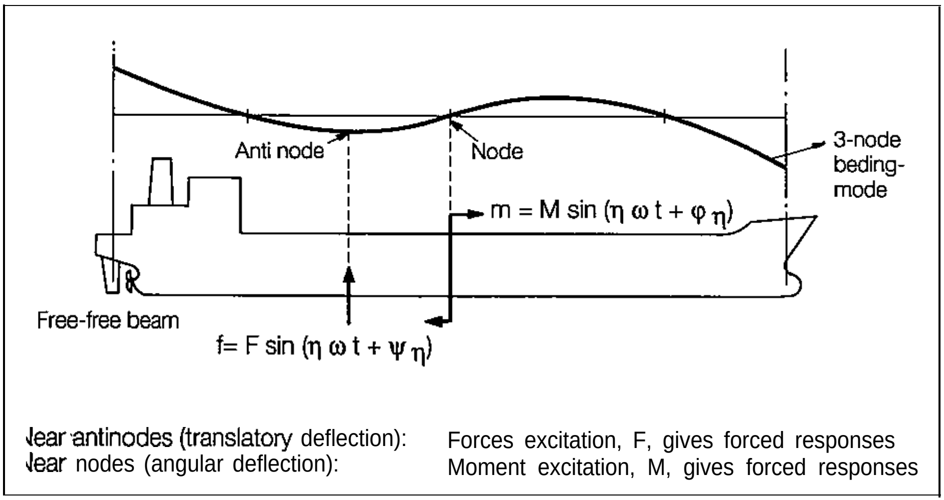


Fig. 18: Excitation of the hull girder modes
Characteristics of mode shapes and their excitations

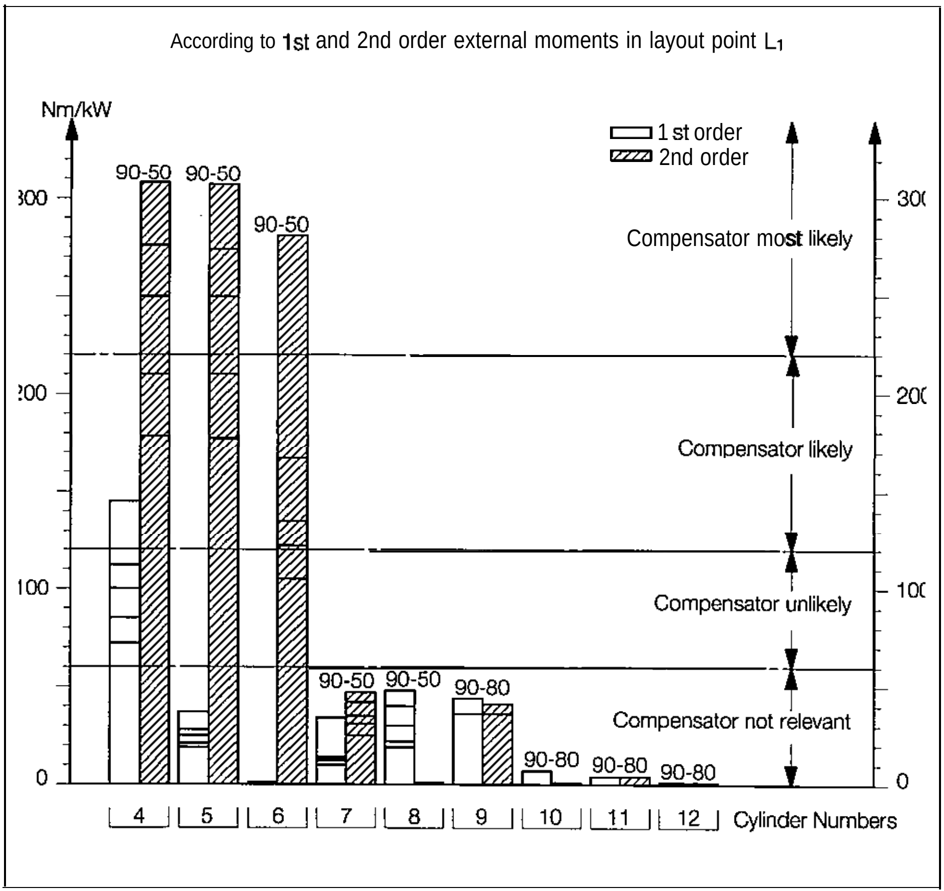


Fig. 19: Power related unbalance for the MC engines
Defined to judge the size of the external moments

1 st order moments act in both the vertical and horizontal directions. For MAN B&W engines with standard balancing, these moments are of identical magnitudes.

For engines with five cylinders or more, the 1 st order moment is very rarely harmful to the ship. However, with 4-cylinder engines, precautions need to be considered.

The 2nd order moment acts only in a vertical direction Precautions need only be considered for 4, 5, and 6-cylinder engines.

To judge the size of the external moments, the so-called Power Related Unbalance (PRU) has been defined, Fig. 19.

On 4-cylinder engines, the 1 st order moment is controlled in the following way:

- standard:
  adjustable counterweights

- option:
  1 st order moment compensator

Resonance between the vertical moment and the 2-node vertical hull girder mode may often be critical, whereas the resonance between the horizontal moment and the 2-node horizontal hull girder mode normally occurs at engine speeds higher than nominal. As standard, 4-cylinder engines are fitted with adjustable counterweights. as illustrated in Fig. 20. These counterweights reduce the vertical moment to an insignificant value (although simultaneously increasing the horizontal moment); thus, this resonance of the 2-node vertical hull girder mode is easily dealt with.

In rare cases, where the 1 st order moments will cause resonance with both the vertical and the horizontal 2-node hull girder mode in the normal speed range of the engine, a 1 st order moment compensator, as shown in Fig. 21, can be introduced in the chain tightener wheel, reducing the horizontal 1 st order moment to a harmless value. The com-

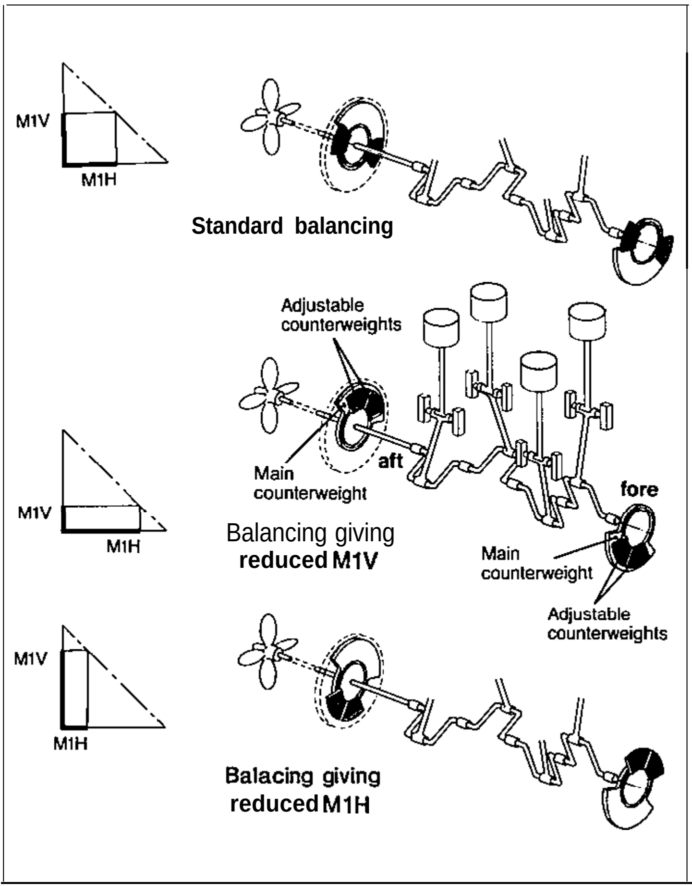


Fig. 20: Adjustable counterweights for 1st order external moment control

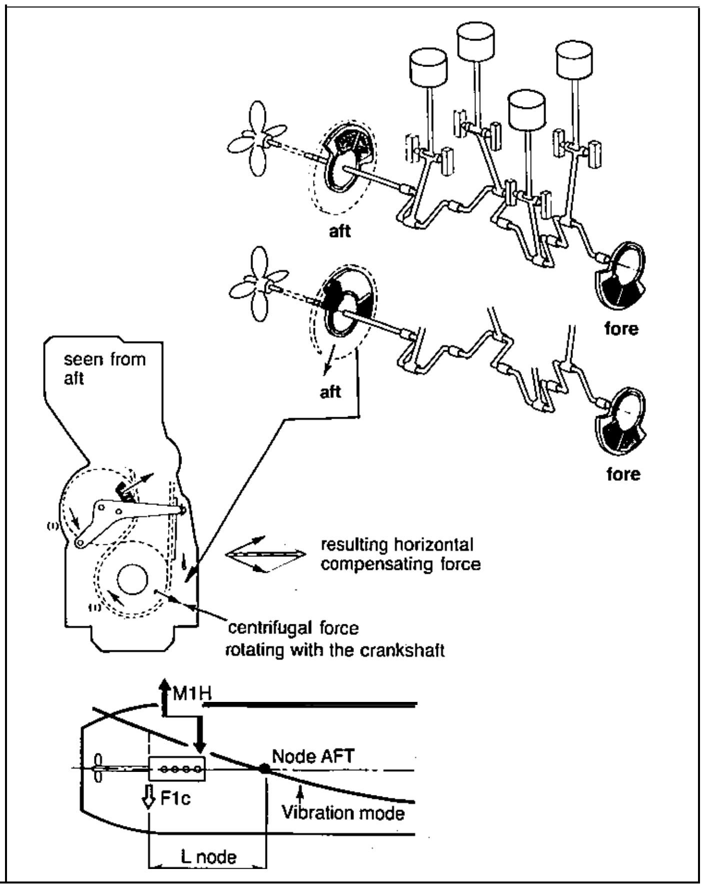


Fig. 21: Compensation of 1st order external moment

pensator comprises two counter-rotating masses, rotating at the same speed as the engine is running.

Since resonance with both the vertical and the horizontal hull vibration modes is rare, the standard engine is not prepared for the fitting of such compensators

Resonance between the 2nd order vertical moment and the 3, 4, and 5-node hull girder vibration modes are possible in the normal running range of the engine, Fig. 22, upper. In order to control the resulting vibratory responses, a second order compensator can be installed on 4, 5, and 6-cylinder engines.

Several solutions, from which the most cost-efficient one can be chosen, are available to cope with the 2nd order vertical moment:

a) No compensators, if considered unnecessary on the basis of the natural frequency, nodal point, and size of the 2nd order moment

b) A compensator mounted on the aft end of the engine, driven by the main chain drive, Fig. 22

c) A compensator mounted on the fore end, driven from the crankshaft through a separate chain drive

d) An electrically driven compensator, synchronized to the correct phase relative to the free moment. This type of compensator requires an extra seating to be prepared, preferably in the steering gear room where deflections are largest and the compensator therefore has the greatest effect

e) Compensators on both the aft and fore ends of the engine, completely eliminating the external 2nd order moments, Fig. 22

Solutions (b), (c) and (d) are force generating compensators, which are ineffective if they are placed in a node of the actual hull girder mode, but effective if they are placed away from the node, i.e. close to an antinode.

If the node of the critical hull girder mode is situated close to the engine, solutions (d) and (e) should be considered.

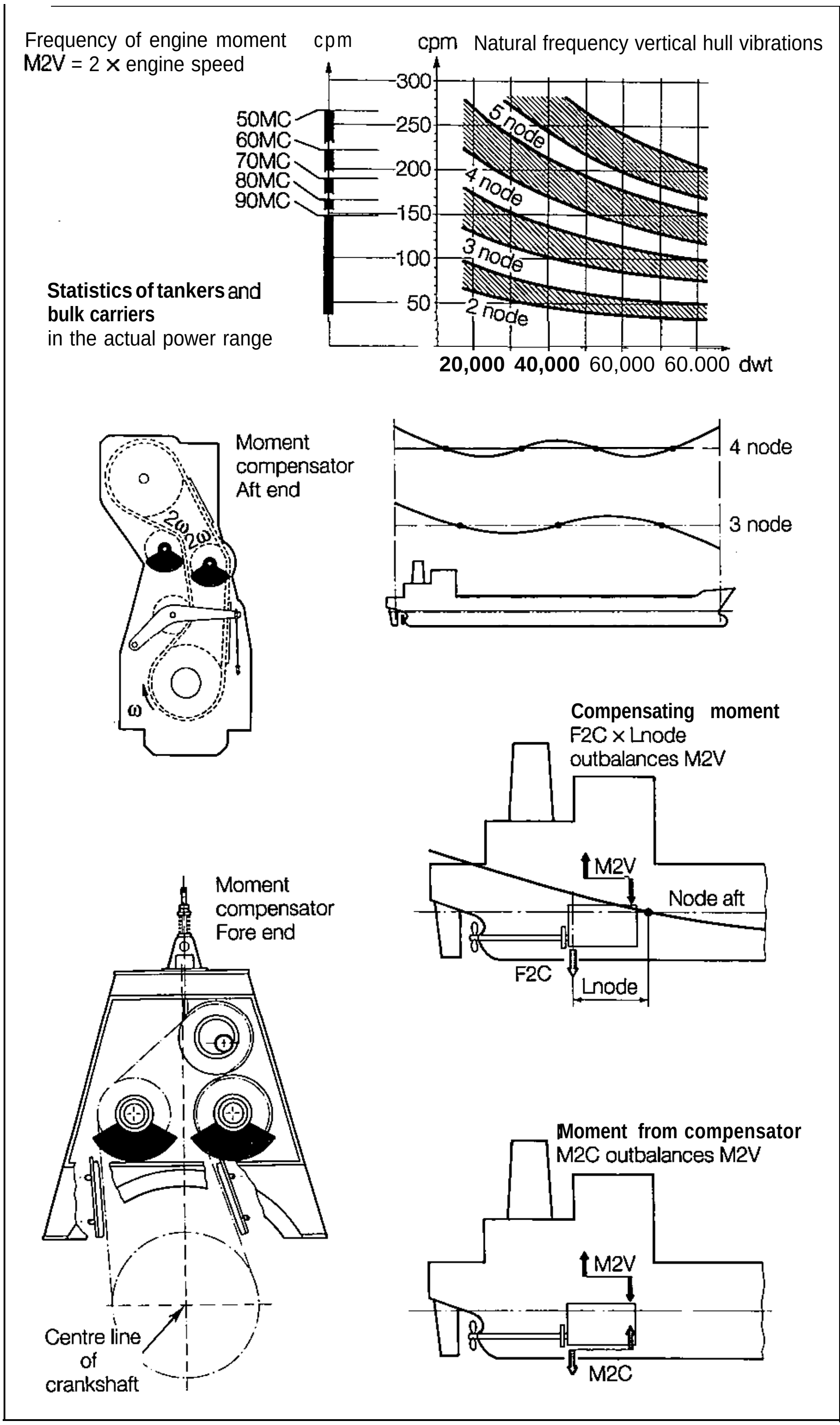
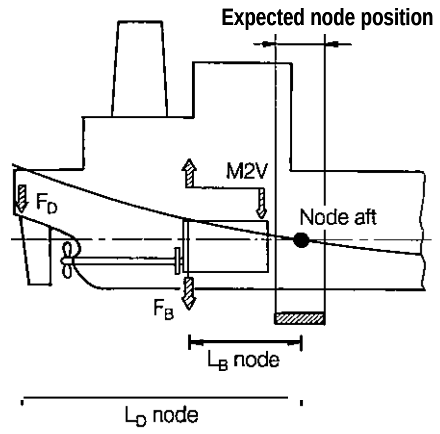


Fig. 22: Compensation of 2nd order vertical external moment. 3, 4 and 5-node vertical hull girder mode should be considered



F_D: **Compensating force of** elcectrically **driven compensator**

F_B: Compensating force of compensator in the main chain drive

M2V: **2nd order external moment to be outbalanced**

$$F_D = \frac{M2V}{L_D \text{ node}} \qquad F_B = \frac{M2V}{L_B \text{ node}}$$

As L_B node is given with greater relative **uncertainty than L_D node, F_D can be -determined** more **precisely than F_B**

Fig. 23: 2nd order moment balancing. Sensitivity of force generating compensators due to the node position

If placed in the steering gear room, the electrically driven compensator d) has the advantage -compared to the other compensators (b) **and (c) -that it is not as sensitive to the positioning of the node, Fig. 23.**

If compensator(s) are omitted, the engine can be delivered with preparation for the later fitting of compensators. This preparation must be decided at the contract stage of the engine. Measurements taken during the sea trial, or later during service with special loadings of the ship, will show whether compensator(s) have to be fitted or not.

In addition to these above discussed external forces and moments, there are also secondary external forces and moments originating from torsional vibrations. An example concerning an 8S60MCE is given in Appendix A. Secondary external forces and moments originating from torsional vibrations are mainly of higher order and will be capable of exciting local vibration modes. These secondary forces and moments, which have not so far been reported as a source of vibration, should be considered at the design stage. They will appear as a result of thorough torsional vibration calculations.

Engine Structure and Double Bottom Vibrations

The vibration modes of the engine frame are part of more comprehensive vibration modes in the aft end of the ship. There are three major modes:

1. H-mode:
   Transverse vibration mode with anti-node at the engine top level.
   In-phase amplitudes from the first cylinder to the last cylinder, Fig. 24

2. X-mode:
   Transverse vibration mode of engine top where the foremost part and the aftmost part of the engine are 180 degrees out of phase, having node at the centre part of the engine, Fig. 24

3. L-mode:
   Longitudinal vibration mode with anti-node at the engine top level, Fig. 24

The natural frequencies of these vibration modes are to a large extent determined by the stiffness of the seating and the double bottom on which the engine is installed. Fig. 25 shows the measured mode shape of a longitudinal double bottom engine column vibration mode for a 5L80MCE engine. It appears that the majority of the elastic deformations occur in the double bottom. The natural frequency is therefore mainly determined by this structure, as the en-

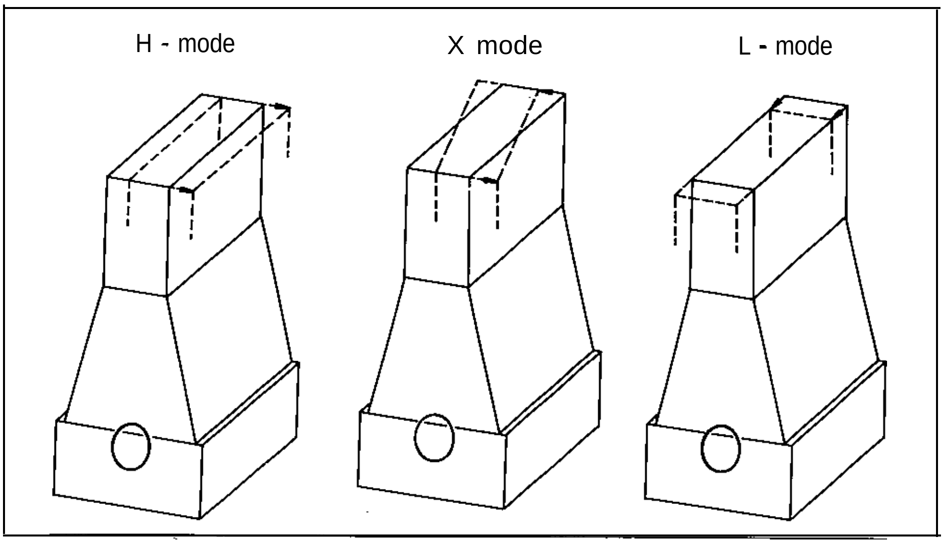


fig. 24: The three major modes of the engine column structure, H, X and L-mode

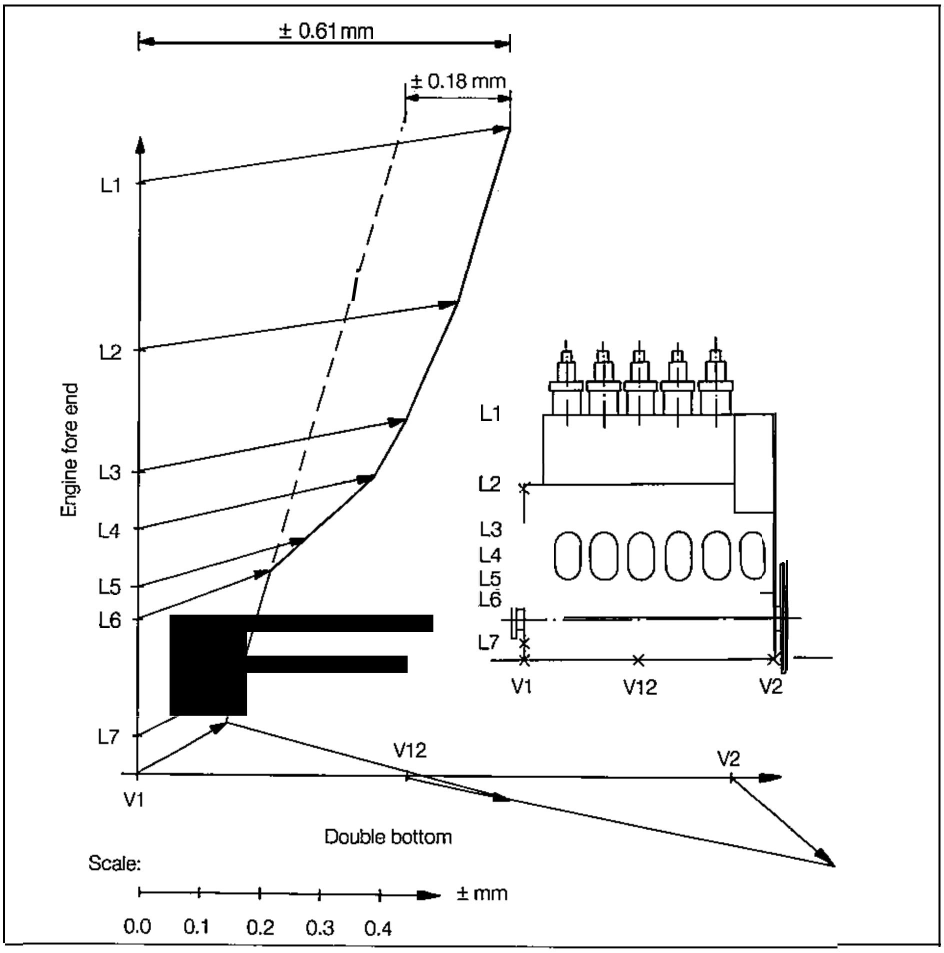


Fig. 25: Longitudinal double bottom engine column vibration mode measured on a 5LBOMCE engine. Main critical 5th order has resonance in the running range

gine as such, compared to the double bottom, has a much greater stiffness. A similar example could be given for the H-vibration mode.

H and X-modes are excited by guide force moments of the H and X-types, Fig. 4. The primary values of these guide force moments can be calculated for each engine type on the basis of its gas and mass forces. This kind of excitation is inherent in all engines.

Secondary values of the H and X-type guideforce moments originate from torsional vibrations and will, therefore, be different for each installation, even for the same type of engine. Appendix B shows primary and secondary values of guide force moments for a 5L70MC engine with two different shafting layouts. It is seen that the secondary values are moderate.

The guide force moments of an 6S6OMC engine for two alternative firing orders are given in Appendix C. In this case, values for the guide force moments originating from the Z-node torsional mode are noticed.

L-modes are excited by secondary phenomena only, i.e. installation dependent forces. The example in Fig. 25 shows a case where the L-mode shape is excited mainly by varying forces in the thrust bearing initiated by torsional vibration induced propeller thrust.

Another source of excitation is varying forces in the thrust bearing caused by axial vibrations of the crankshaft.

H and X-vibration modes are traditionally controlled by bracing the engine top to the hull structure so as to obtain resonances with the critical orders situated above the relevant speed range, thus detuning the system. In order to obtain a sufficient detuning effect, i.e. to bring certain resonances with critical orders above the relevant speed range, stiffness requirements are specified for the attachment of the bracing to the engine room structure.

We have experienced some cases where a proper detuning effect was not obtained, even though the stiffness requirements had been fulfilled: In this connection, it is relevant to bear in mind the development during the past fifteen years, see Fig. 1. Today an engine with a specified output typically has fewer cylinders and is considerably higher than previous engine types, i.e. the height/length ratio is different. To cope with this development, classification societies and shipyards should consider revising their requirements to the double bottom design.

For two reasons, L-vibration modes have attracted increasing attention:

- The excitation of the thrust bearing has increased
- The L-mode has in many cases become resonant with the main critical order in the relevant speed range (example: see Fig. 25). This is the most important reason and, again, it is suggested that the requirements to the double bottom design should be reconsidered

In certain cases, longitudinal top bracing has been introduced in order to detune critical orders and the natural frequency of the L-mode. By means of this arrangement, vibrations in the longitudinal direction have been reduced to a satisfactory level.

Where axial vibrations of the crankshaft are the main source of excitation, the longitudinal vibration levels can also be reduced by means of an axial vibration damper.

As mentioned earlier, L-mode excitations are of a secondary type. This means that the excitation level is determined by vibration characteristics of other vibration modes. L-mode excitations are determined by means of axial and torsional vibration characteristics, and these are to be calculated and synthesized at the design stage of the ship in order that appropriate precautions can be taken.

As an alternative to the traditional friction type of top bracing, Fig. 26, the hydraulically adjustable top bracing has been designed for use on vessels having large deflections due to heavy sea, loading/unloading, etc.

This system, shown in Fig. 27, consists basically of a hydraulic cylinder and two spherical bearings. Oil is supplied from the camshaft lubricating oil system, and a relief valve prevents the build-up of excessive forces.

This hydraulically adjustable top bracing is intended for one-side mounting, and will provide a constant force between engine and hull, irrespective of deflection and, as such, will still act as a detuner of the double bottom/main engrne system.

The system has been commercially applied in a number of newbuildings with good results.

Obviously, this system increases the overall costs and, therefore, it will replace the friction type only when necessary.

It should be noted that the hydraulically adjustable top bracing does not increase the building width of an engine, compared to the friction type bracing.

Vibration Levels and Their Acceptability

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

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

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

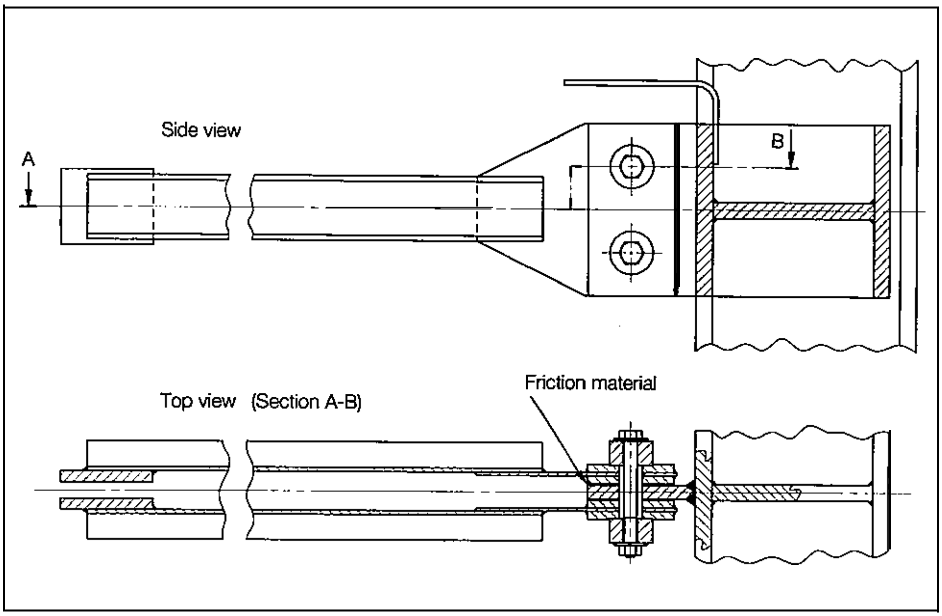


Fig. 26: Friction type top bracing

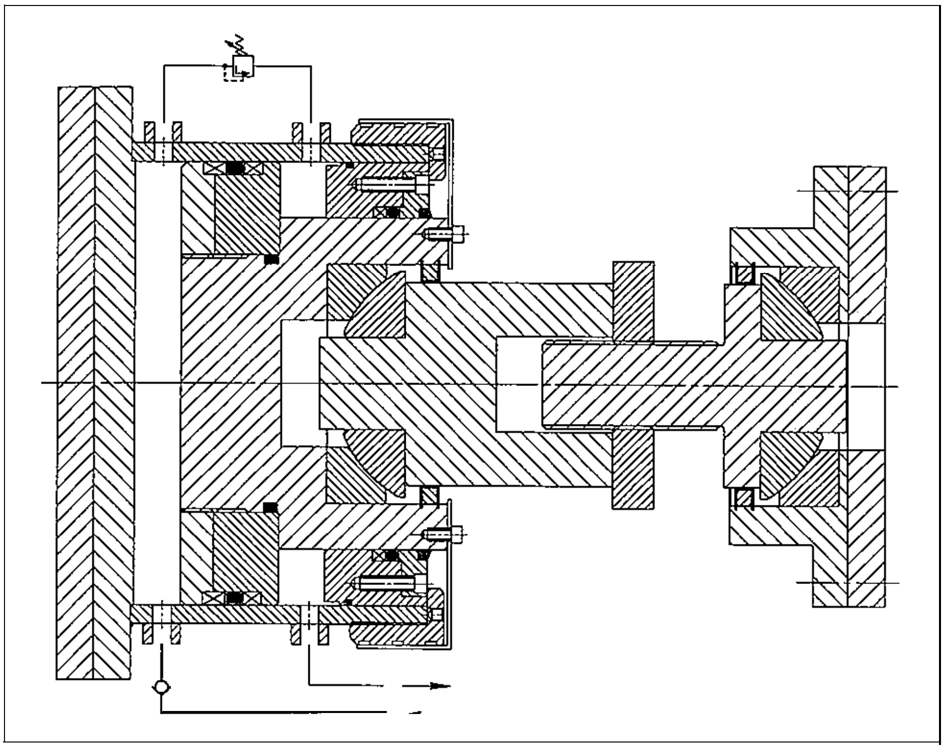


Fig. 27: Hydraulically adjustable top bracing

With a view to fulfilling these criteria, certain limits to vibration levels can be prescribed, and it is common practice to specify different limits in different frequency ranges:

a) Lower frequency range: displacement limit

b) Intermediate frequency range: velocity limit

c) Upper frequency range: acceleration limit

The displacement limit in the lower frequency range is determined by static stress level considerations, In the intermediate range, the velocity limit will keep the kinetic energy constant throughout the range, resulting in decreasing permissible displacements, The acceleration limit in the upper frequency range decreases the permissible displacements further so as to control noise radiation.

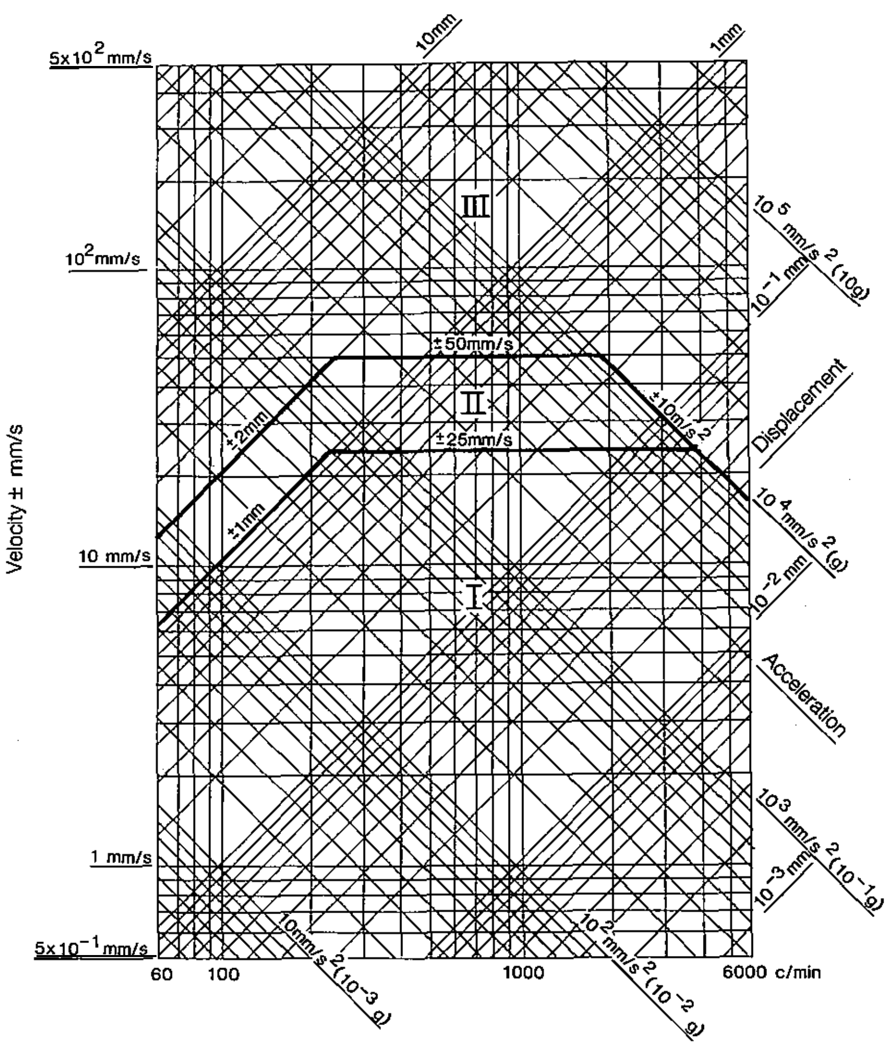
The limits applying to MAN B&W two-stroke engines are given as single order peak amplitudes, X:

$$S = \pm x \sin(\eta \varpi t + \phi \eta)$$

The two-stroke low speed diesel engine is designed to cope with rather high internal varying forces and, consequently, rather high limits are allowed for vibration levels in its main structure. Fig. 28 shows the limits which are acceptable for MAN B&W two-stroke engines.

If vibration levels in zone II (see Fig. 28) have been measured under a certain condition of the ship, it should be borne in mind that zone III readings, i.e. not acceptable, might occur under other conditions, such as:

1: different draught of the vessel
2: different trim of the vessel
3: different distribution of ballast load
4: different engine loading
5: changing sea condition



Zone I: Acceptable
Zone II: **Vibration will not damage the main engine, however, under adverse conditions, annoying/harmful vibration responses may appear in the connected structures**
Zone III: Not acceptable

Fig. 28: Vibration limits

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, a coagency 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.

Based on all the experience gathered up till now, we are confident that the necessary means for predicting and counteracting vibration on board ships with two-stroke diesel engines are available today.

The fact that, even today, ships are, from time to time, delivered with unsatisfactory vibration conditions reflects that the whole procedure from project to actual ship in service is subject to compromises which consider other aspects than the vibrational and that predictions of vibrational behaviour, even when based on advanced computer programs, are still subject to uncertainties.

In order to utilise the available possibilities, its is recommended that yards, owners, and engine builders discuss the vibration aspects at an early stage, at least before signing the contract, so as to ensure that the best possible solutions are selected and incorporated in the project right from the start.

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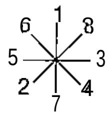
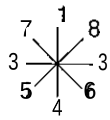
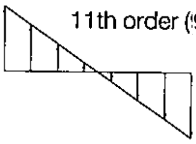
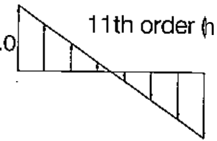
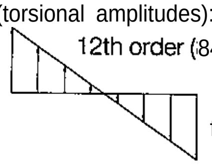
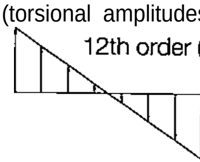
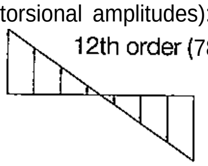
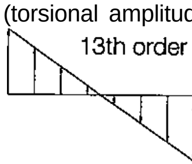
Appendix A

<table>
<tr><td colspan="4">Example: 8S60MCE, 102 r/min, 12,000 kW
External Forces and Moments (vertical)</td></tr>
<tr><td colspan="2">Firing order:
</td><td colspan="2">Firing order:
</td></tr>
<tr><td>Primary values given at 102 r/min</td><td>Secondary values due to Torsional Vibration Responses</td><td>Primary values given at 102 r/min</td><td>Secondary values due to Torsional Vibration Responses</td></tr>
<tr><td rowspan="3">1 st order moment:
±348 kNm

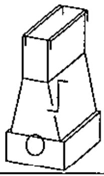
4th order moment:
±75 kNm</td><td>Excitation (torsional amplitudes):
±2.5 mrad 11th order (92 r/min)

±2.5 mrad
Cyl. No. 1 2 3 4 5 6 7 6
Secondary values at 92 r/min:
Free forces: 12th order: ±198 kN
13th order: ±104 kN
Free moments: 10th order: ±966 kNm</td><td rowspan="3">1st order moment:
f174 kNm

4th order moment:
±300 kNm</td><td>Excitation (torsional amplitudes):
±1.0 11th order (h order (92 r/min)

±1.0 mrad
Cyl. No. 1 2 3 4 5 6 7 8
Secondary values at 92 r/min:
Free forces: 12th order: ±299 kN
Free moments: 10th order: ± 92 kNm</td></tr>
<tr><td>Excitation (torsional amplitudes):
±0.5 mrad 12th order (84 rlmin)

fo.5 mrad
Cyl. No. 12 3 4 56 78
Secondary values at 84 r/min:
Free forces: 1 lth order: ±101 kN
13th order: ±122 kN
Free moments: 14th order: ± 38 kNm</td><td>Excitation (torsional amplitudes):
±2.0 mrad 12th order (84 r/min)

f2.0 mrad
Cyl. No. 1 2 3 4 5 8 7 6
Secondary values at 64 r/min:
Free forces: 11 th order: ±201 kN
13th order: ±245 kN
Free moments: 11 th order: ± 61 kNm
13th order: ± 75 kNm
14th order: ± 38 kNm</td></tr>
<tr><td>Excitation (torsional amplitudes):
±0.85 mrad 12th order (78 r/min)

±0.85 mrad
Cyl. No. 1 2 3 4 5 6 7 8
Secondary values at 78 r/min:
Free forces: 12th order: ± 52 kN
Free moments: 14th order: ±376 kNm</td><td>Excitation (torsional amplitudes):
±0.8 mrad 13th order (78 rlmin)

f0.8 mrad
Cyl. No. 12 3 4 56 78
Secondary values at 78 r/min:
Free forces: 12th order: ±208 kN
Free moments: 14th order: ± 94 kNm
15th order: ± 67 kNm</td></tr>
</table>

Appendix B

Example: 5L70MC, 95 rlmin. 10,400 kW

Guide Force Moments -H-type



Primary values given at 95 r/min

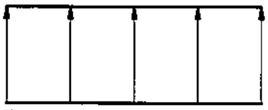
Order	Moment
5th	±1200 kNm
10th	± 97 kNm

Secondary values due to Torsional Vibration Responses

Shafting layout: Overcritical

Excitation (torsional amplitude): 5th order at 95 r/min

±3.0 mrad

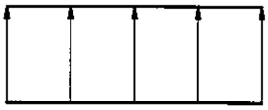


Cyl. No. 1 2 3 4 5

Secondary values of guide force moments (95 r/min):
5th order: ±213 kNm

Excitation (torsional amplitudes): 5th order at 55 r/min

±24 mrad



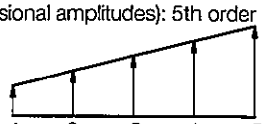
Cyl. No. 1 2 3 4 5

Secondary values of guide force moments (55 r/min):
5th order: ±571 kNm

Shafting layout: Undercritical

Excitation (torsional amplitudes): 5th order at 95 r/min

±0.5 mrad ±1.5 mrad

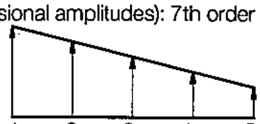


Cyl. No. 1 2 3 4 5

Secondary values of guide force moments (95 r/min):
5th order: ±71 kNm 3rd order: ±6 kNm
7th order: ±13 kNm 2nd order: ±4 kNm
8th order: ± 7 kNm

Excitation (torsional amplitudes): 7th order at 96 r/min

±0.7 mrad ±0.35 mrad

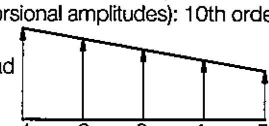


Cyl. No. 1 2 3 4 5

Secondary values of guide force moments (96 r/min):
7th order: ±15 kNm
5th order: ±25 kNm
10th order: ±23 kNm

Excitation (torsional amplitudes): 10th order at 68 r/min

±1.7 mrad ±0.85 mrad

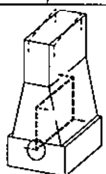


Cyl. No. 1 2 3 4 5

Secondary values of guide force moments (68 r/min):
10th order: ±174 kNm

Example: 5L70MC, 95 rlmin, 10,400 kW

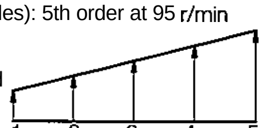
Guide Force Moments **–X-type**



Order	Moment
1st	±151 kNm
2nd	±250 kNm
3rd	±378 kNm
4th	± 63 kNm
5th	0
6th	± 31 kNm
7th	±222 kNm
8th	±137 kNm
9th	± 6 kNm
10th	0
11th	± 3 kNm
12th	± 20 kNm

Excitation (torsional amplitudes): 5th order at 95 r/min

±3.0 mrad ±0.5 mrad ±1.5 mrad



Cyl. No. 1 2 3 4 5

Cyl. No. 1 2 3 4 5

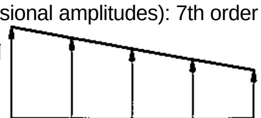
Secondary values of guide force moments (95 rlmin):
7th order: ±75 kNm 3rd order: ±36 kNm
8th order: ±39 kNm 2nd order: ±27kNm

Excitation (torsional amplitudes): 5th order at 95 r/min

Secondary values of guide force moments (95 r/min):
5th order: ±18 kNm 3rd order: ±12 kNm
7th order: ±25 kNm 2nd order: ± 9 kNm
8th order: ±13 kNm

Excitation (torsional amplitudes): 7th order at 96 r/min

±0.7 mrad ±0.35 mrad



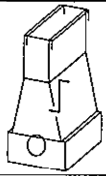
Cyl. No. 1 2 3 4 5

Secondary values of guide force moments (96 rlmin):
7th order: ±35 kNm

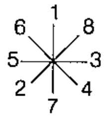
Appendix C

Example: 8S60MCE, 102 r/min, 12.000 kW

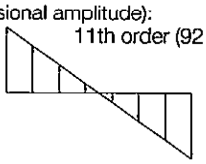
Guide Force Moments **– H-type**

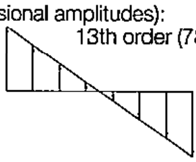


Firing order:

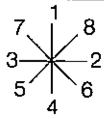


Primary values given at 102 r/min	Secondary values due to Torsional Vibration Responses
8th order: ±392 kNm	Excitation (torsional amplitude): 11th order (92 r/min) ±2.5 mrad … ±2.5 mrad; Cyl. No. 1 2 3 4 5 6 7 8; Secondary values of guide force moments (92 r/min): 11th order: ±388 kNm; 13th order: ±222 kNm
	Excitation (torsional amplitudes): 13th order (78 r/min) ±0.85 mrad … ±0.85 mrad; Cyl. No. 1 2 3 4 5 6 7 8; Secondary values of guide force moments (78 r/min): 13th order: ±133 kNm; 11th order: ± 54 kNm; 18th order: ± 83 kNm

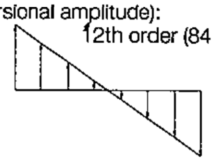


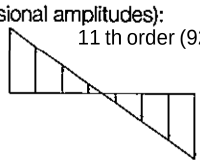


Firing order:



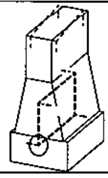
Primary values given at 102 r/min	Secondary values due to Torsional Vibration Responses
8th order: ±392 kNm	Excitation (torsional amplitude): 12th order (84 r/min) ±2.0 mrad … ±2.0 mrad; Cyl. No. 1 2 3 4 5 6 7 8; Secondary values of guide force moments (84 r/min): 12th order: ±392 kNm
	Excitation (torsional amplitudes): 11 th order (92 r/min) ±1.0 mrad … ±1.0 mrad; Cyl. No. 1 2 3 4 5 6 7 8; Secondary values of guide force moments (92 r/min): 11th order: ± 76 kNm





Example: 8S60MCE, 102 r/min, 12.000 kW

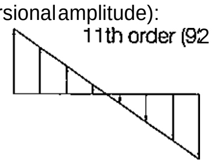
Guide Force Moments -X-type



Order	Moment
1st	±181 kNm
2nd	0
4th	±350 kNm
5th	±902 kNm
8th	±417 kNm
7th	± 29 kNm
6th	0
9th	± 11 kNm
10th	0
11th	± 56 kNm
12th	± 11 kNm

Excitation (torsional amplitude): 11th order (92 r/min)

±2.5 mrad



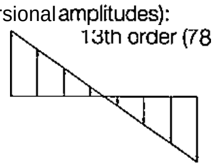
±2.5 mrad

Cyl. No. 1 2 3 4 5 6 7 6

Secondary values of guide force moments (92 r/min):
8th order: ± 83 kNm
9th order: ± 45 kNm
10th order: ± 40 kNm

Excitation (torsional amplitudes): 13th order (78 r/min)

±0.85 mrad



±0.85 mrad

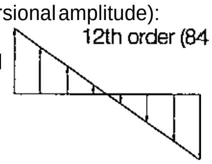
Cyl. No. 1 2 3 4 5 6 7 6

Secondary values of guide force moments (78 r/min):
16th order: ± 35 kNm
15th order: ± 21 kNm

Order	Moment
1st	± 91 kNm
2nd	0
3rd	± 209 kNm
4th	±1401 kNm
5th	± 451 kNm
6th	0
7th	± 14 kNm
6th	0
9th	± 5 kNm
10th	0
11th	± 28 kNm
12th	± 43 kNm

Excitation (torsional amplitude): 12th order (84 r/min)

±2.0 mrad



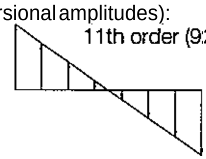
±2.0 mrad

Cyl. No. 1 2 3 4 5 6 7 6

Secondary values of guide force moments (64 r/m
9th order: ± 39 kNm
10th order: ± 16 kNm

Excitation (torsional amplitudes): 11th order (92 r/min)

±1.0 mrad



±1.0 mrad

Cyl. No. 1 2 3 4 5 6 7 8

Secondary values of guide force moments (92 r/min):
9th order: ± 34 kNm
10th order: ± 35 kNm