Case Studies (Dynamic responses) of Aluminium SURV Vessels at Sea

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SURVEILLANCE, PILOT AND RESCUE CRAFT

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CASE STUDIES (DYNAMIC RESPONSES) OF ALUMINIUM SURV VESSELS AT SEA

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SUMMARY

It is a known fact that fast aluminium vessels are difficult to design properly. In many designs, problems have occurred related to cracking of aluminium construction. This is generally the result of either poor detail construction design, high power propulsion installations and heavy external loading due to motions in a seaway or a combination of all or some of these issues.

The paper deals with an introduction into the problems at hand, a number of practical examples of on board trouble shooting (i.e. measurements), problem solutions and recommendations for future design approach to be taken to avoid these problems for the next generation of fast aluminium vessels.

AUTHORS' BIOGRAPHIES

Jakob Pinkster holds the current position of Assistant Professor of Ship Hydromechanics, Technical University of Delft, The Netherlands. In the past he has been involved with fast marine vehicles with regard to design, construction, testing, trouble shooting etc.

Ron Uitermarkt is now retired but has for many years been involved in measuring vibrations on board marine vehicles. His experience in this matter is therefore very wide. He has also been called upon many times in trouble shooting exercises and asked to seek solutions for vibration problems.

1. INTRODUCTION

Fast aluminium vessels are difficult to design properly, everybody who has been involved in this field knows that. Even if the mission profile is well defined there are still many difficulties that must be overcome before the vessel has been properly designed. After a first rough design estimate is made, an attempt is made for a first estimation of the required propulsion and powering installations. Of course the difficulty to create the proper body plans for such fast vessels that gives an acceptable trade off between vessel power requirements and ship motions in a seaway should not be forgotten.

Along the way the ship designer seems to be making some mistakes which are going to cause many headache after the vessel has been built and in full service. In this respect many designs, problems have occurred related to cracking of aluminium construction. This is generally the result of either poor detail construction design and fatique performance of welded aluminium, high power propulsion installations and heavy external loading due to motions in a seaway or a combination of all or some of these issues.

The paper deals with an introduction into the problems at hand, a number of practical examples of on board trouble shooting (i.e. measurements), problem solutions and recommendations for future design approach to be taken to avoid these problems for the next generation of fast aluminium vessels. But, first of all, what are the mission profile requirements of such SURV vessels to start with and which problems occur with these marine vehicles?

2. FAST ALUMINIUM VESSELS, MISSION PROFILE AND PROBLEMS

In general, the mission profile of a fast SURV vessel can be summed up as follows - fast and safe deployment for SAR purposes in rigorous weather conditions. This simple sentence can be unraffled into a very long list of design specifications depending on the local nature of the SAR service to be rendered. Besides being a long list of design specifications, it is obvious that this is also a very difficult list to satisfy simultaneously when the vessel is finally built and in service.

The joint experience of the authors is that problems concerning fast aluminium vessels is largely related to poor construction details, fatique, and unexpectedly high external loading or a combination thereof. This leads to cracking of the aluminium construction, in general at points of low material properties, i.e. in the vacinity of heat affected welding zones and at points of high stress concentrations i.e. spray rails, sharp corners, poor welding techniques etc. (see Fig. 1, see [1]). Fatique also plays an important role regarding resonance of aluminium construction parts with engine frequencies, propeller blade frequencies, gearbox frequencies, universal joint frequencies etc. This is a direct result of insufficient vibration estimation calculations in the design stage as well as the wrong choice of propulsion system concepts.

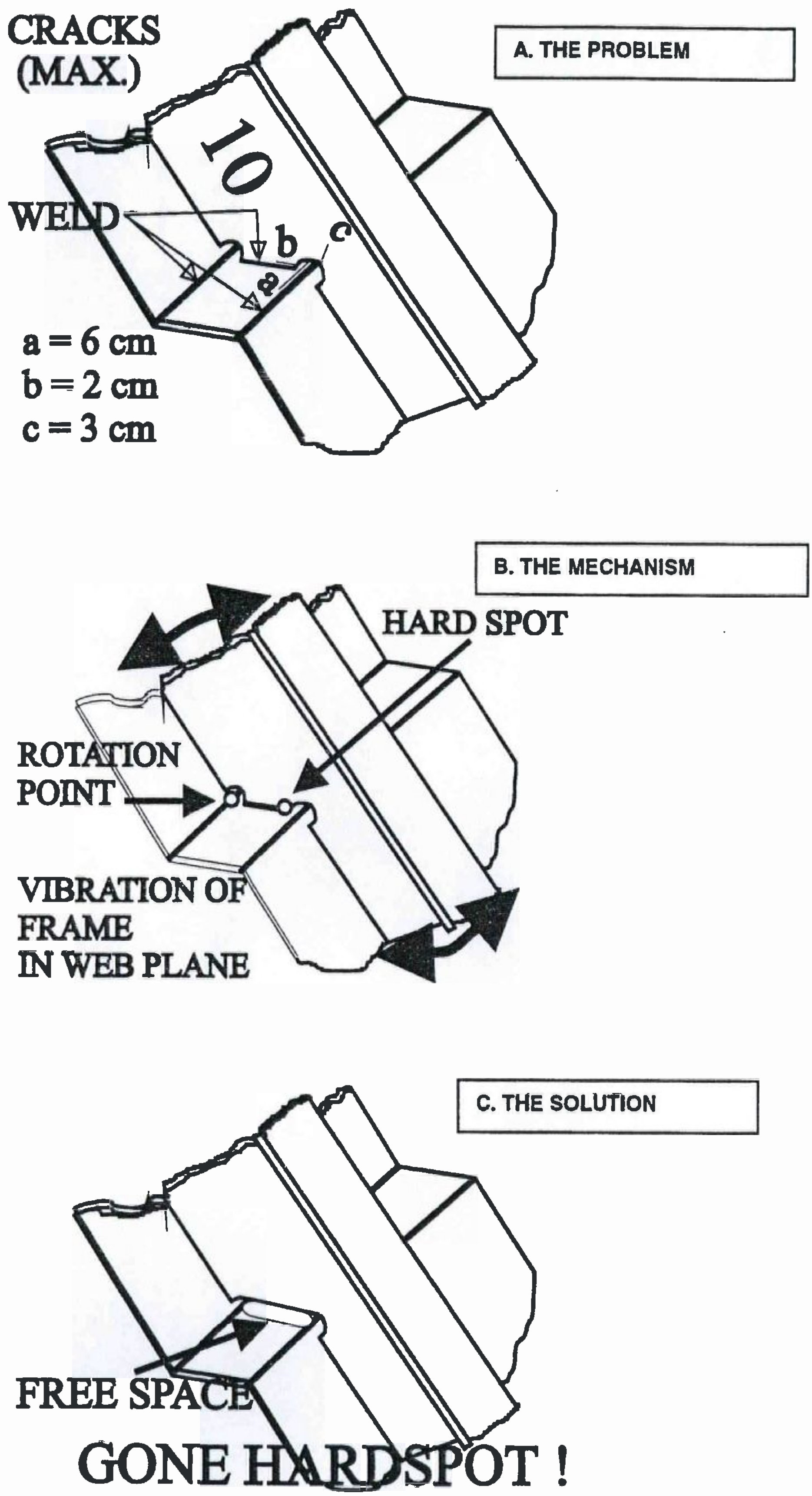


Fig. 1 Damage to a spray rail due to in plane vibration of a web frame of a fast pilot (and SAR) vessel, the mechanism and the problem solution

3. IMPACT LOAD FORCES AS A RESULT OF VESSEL MOTIONS AND THE RESULTING CONSTRUCTION DAMAGE

Due to high speeds of the these vessels in a seaway, large ship motion accelerations may frequently occur, i.e. verticle accelerations up to 6 g in sea conditions with wave heights of 2.5m according to [1]. Also slamming is not uncommon and is an accepted phenomenon for these types of vessels due to the very nature of their SAR tasks. However, this slamming phenomenon which is characterised by the application of very high impact loads on the forward bottom structure of the vessel, can cause very serious structural damages. This damage is not only due to structural failure from overloading but can also become apparent in the form of fatique cracking of aluminum plates and frames. Not only is the frequency of slamming and the force of the slamming important for the fatique life of the aluminium structure but also the actual transient loading of the structure itself. As a result of the transient loading which the structure experiences from the slamming, initially the structure is displaced from the equilibrium position and then oscillates about this position for a number of cycles until the vibration motion is damped out completely. Thus within the slamming frequency there is also an extra redution in fatique life from load cycles due to transient loading.

Another aspect to be taken into account with these high speed aluminium vessels is that of hydroelastic effects, which is due to the rather high flexibilty of the construction of these aluminium vessels. Normally when considering hydrodynamic loads the assumption is made that the body penetrating the water is absolutely rigid. Recent research, [2], has shown that when hydroelasticity is taken into account the hydrodynamic forces due to bottom slamming are reduced when compared to a rigid contruction but seem to have a certain cyclic characteristic (see Fig. 2). When one takes the short time span of this actual bottom slamming into account and the inertia of the bottom construction itself, the resulting cyclic loading of the construction would appear to be of no significance.

How can one reduce bottom slamming in the bottom of the bow section of fast SURV vessels? A larger deadrise angle in the bow section would be a step in the right direction of course but another option would be to utilise a larger vessel with the same payload capacity and speed, i.e. make use of the enlarged ship concept ([3], and [4]). Examples of successful application of ESC are shown in the following figures for a fast patrol craft (see Fig. 3) and a SAR rescue boat (see Fig. 4). The larger vessels have better seakeeping performance (less motions and lower accelerations), better resistance characteristics (less exploitation costs) while the first costs are not substantially increased (since the main extra costs are due to an extra (cheap) midship piece of the hull).

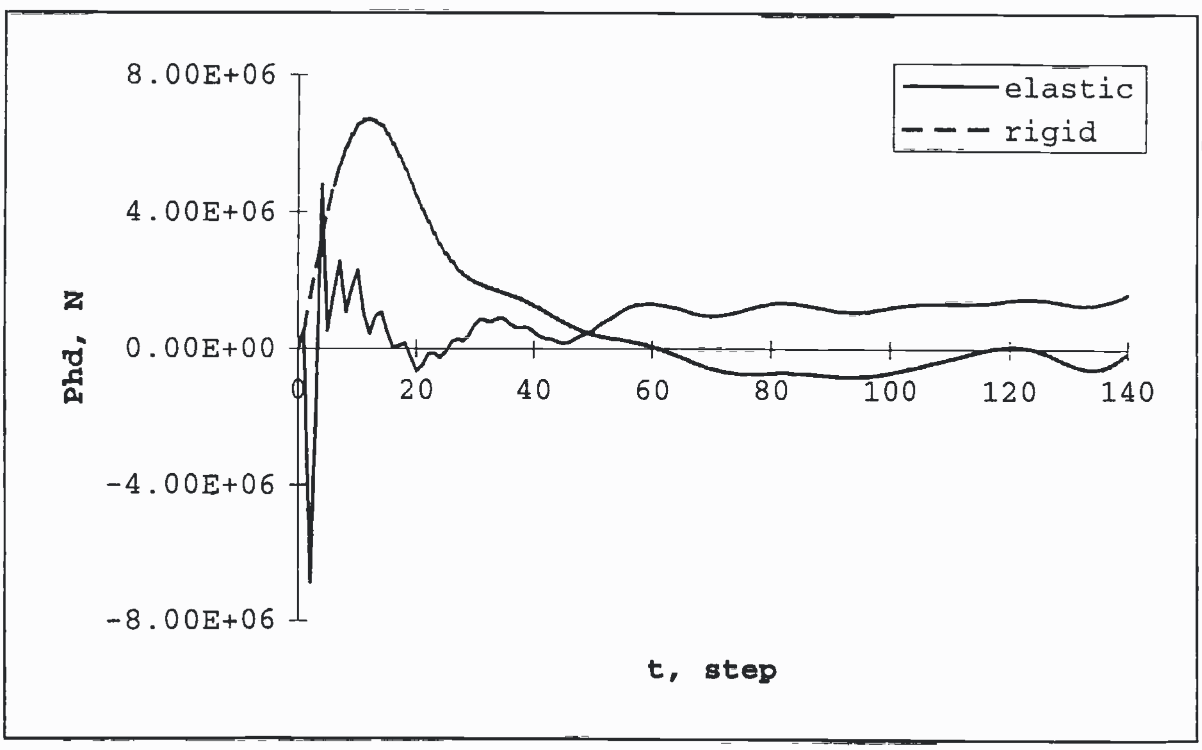
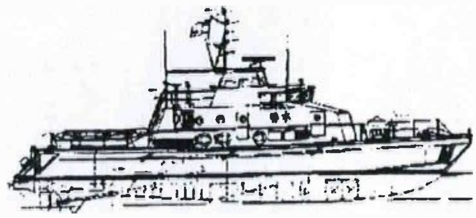
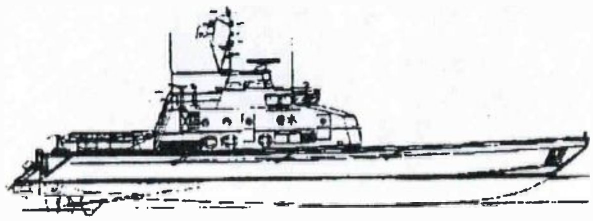


Fig. 2 Hydrodynamic forces on the bottom as a function of time ($\Delta t_i = 0.1 \cdot 10^{-4}$ sec), from [2]



1.0 L



1.25 L

1.50 L

Fig. 3 ESC applied to a 26 knot patrol vessel ([3])

4. CRACKING IN ENGINE FOUNDATIONS

In general engine foundations of aluminium vessels are welded open box constructions between web frames and ships hull. Engines can be installed in a fixed or elastic support. By an elastic support the dynamic forces coming from the engine are smaller than in the case of a fixed mounting but the support itself is much smaller so the local strain increases. In case of cracks engineers try to increase the rigidity by welding more plates in the foundation. The success of this activity is poor. Vibration measurements show that the local resonance plays a main part here. Increasing the rigidity of aluminium constructions with the purpose to realise higher resonance frequencies ends up in the neighbourhood of 80 to 100 Hz. The problem is that it just happens to be these frequencies that are present (and therefore met) in the generated spectrum of high speeds diesel engines.

The elasticity of aluminium is 1/3 of that of steel and causes a much lower structure resonance frequencies. Fig. 5 shows experiences from in practice measured character of stiffening constructions. The maximum reachable value lies between 90 to 100 Hz.

By fast vessels it is common to try to keep the weight of the propulsion installations rather as low as possible and therefore shipbuilders mainly choose high speed engines. The spectra of dynamic forces which are generated at these rpm's have important components in certain frequency range (75 – 135 Hz) as shown in Table 1.

TABLE 1 Fast diesel engines and important vibration frequency components

Engine [rpm]	No. of cylinders [-]	Important order [-]	Frequency [Hz]
1800	6 in line	2.5	75
		3	90
		4.5	135
1800	8 in line	3.5	105
		4	120
2400	6 in line	1.5	90
2400	8 in line	2.5	90

V engines (12 & 16 cylinder) have the same dominating orders produced by V angle related combinations.

The transmitted forces of elastic supports of mountings are dependent on the choice of the type of rubber used in the resilient mounting. Important is to mention that the use of super elastic systems results in the risk of too high a vibration level of the engines themselves and/or in an unacceptably large displacement of the centre of the shaft line.

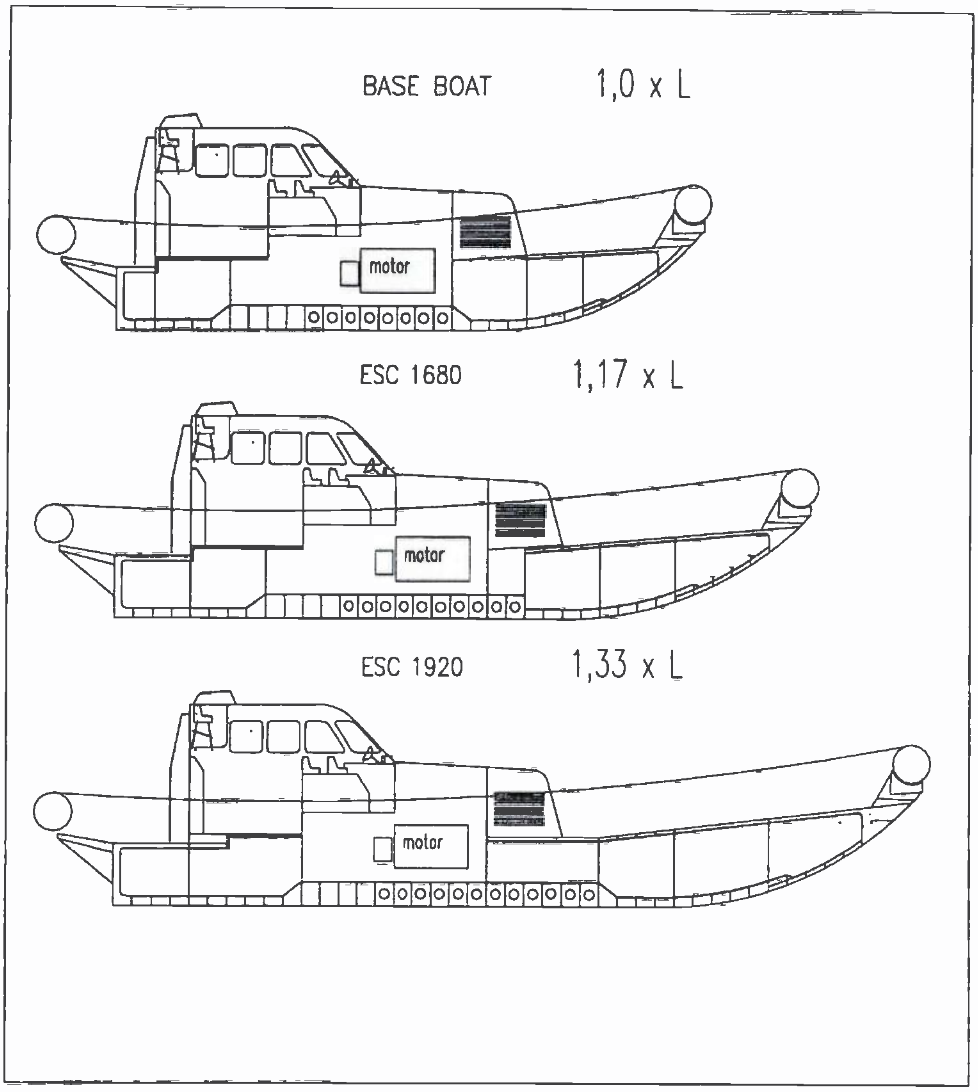


Fig. 4 ESC applied to a fast SAR vessel ([4])

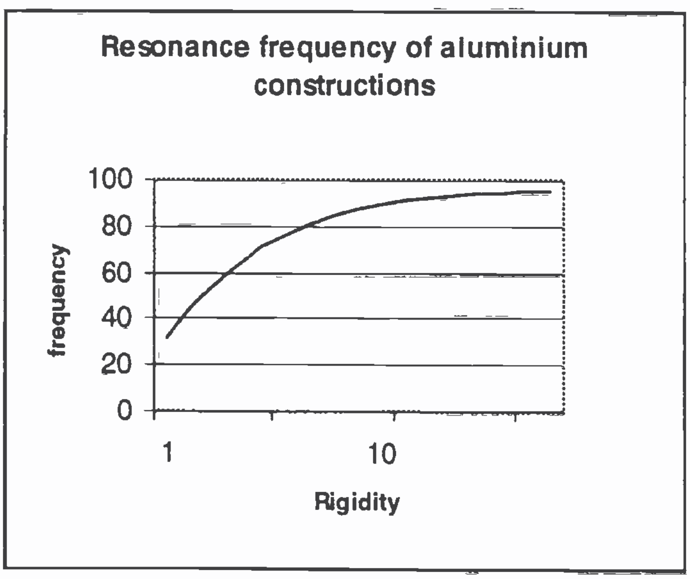


Fig. 5 Maximum stiffness in relation to resonance frequency for aluminium constructions

5. BREAKING OF PROPULSION SHAFTS BY V-DRIVE APPLICATIONS

Installation of a V–drive design of the propulsion installation is very attractive due to space savings. Fig. 6 shows a V-drive with an intermediate shaft between engine and gearbox. The V–angle is accommodated by a universal joint.

The universal joint generates a rotating force which is supported by the nearby located bearing. The bearing is mounted on the ship construction. The radial rotating force has a speed of two times the shaft rpm. (second order). The size of the force depends on shaft torque and V angle. By engines with a speed of 2400 rpm at nominal speed a rotating force with a frequency of around 80 Hz can be expected. A rigidly constructed support for the bearing will promptly start to vibrate. An answer to this problem would be to reduce the stiffness of the support. The effect of a reduction of the stiffness of the support on the resulting forced vibration is shown in Fig. 7.

The generated second order radial force and moments is indicated in the catalogue of the manufacturers of the universal joint. Surprising is that the dynamic torsion moments generate radial forces in the same way. So therefore, it is not only the second "universal" orders giving problems with the bearing support but also the torsion vibration spectrum. Experiences learn that there are some parts of the universal joint that suffer destruction due to fatigue as a result of these radial forces and resulting movements within the system.

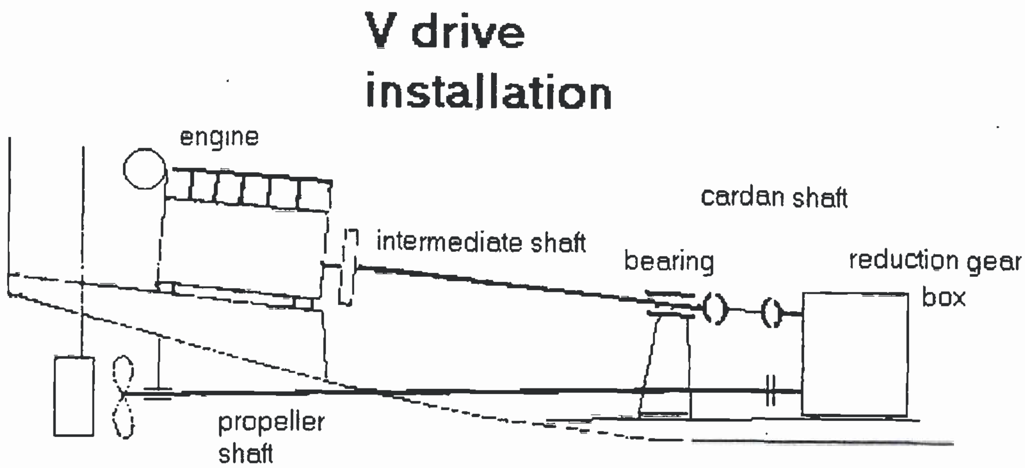


Fig. 6 A typical V-drive installation for a fast vessel

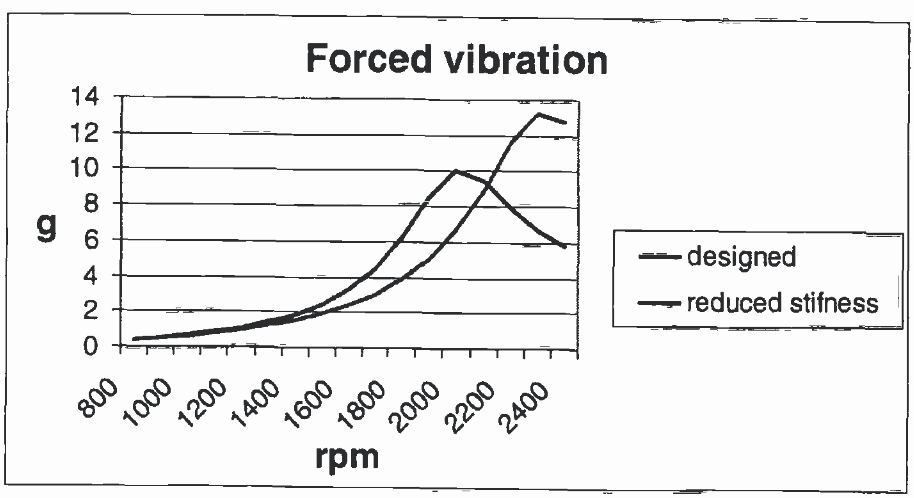


Fig. 7 Effect of reducing stiffness of bearing support on vibration level

6. CRACKING AT BEARING SUPPORTS OF PROPULSION SHAFTS

In some cases the irregular flow of water to the propeller can, more or less, result in larger dynamic forces than caused by cavitation. The dynamic forces will be transmitted to the propeller shafts resulting in a static and dynamic deformation of the shaft line. There will be reaction forces in the system. These reaction forces and the amplification due to resonance of the shaft line load the aft support bearings. The bearings, constructed in brackets, are mounted at one of the aft web frames. The generated dynamic forces have frequencies related to the propeller rpm and the number of propeller blades or multiples thereof (i.e. harmonics). Only the harmonics reach a frequency in the neighbourhood of 80-100 Hz.

Although, in general, resonance of the bracket construction is never met, the resulting dynamic load transmitted to the aft ship is underrated. The dimensioning of the shaft line between propeller and the first bearing inside the vessel is critical. The number of bearings, shaft diameter and propeller overhang (i.e. distance centre propeller - centre support point aft bearing) are very important factors with regard to the resonance frequencies and whirling of the shaft line.

Measuring cavitation of propellers show a wide frequency spectrum from 30 to 130 Hz without dominating components. Although the resulting energy is small, it however creates a lot of noise easily transported through the aluminium structure. Dynamic forces coming from induced pressure forces from the propeller tips and the rotation of the propeller shaft are stable in frequency and can result in a strong dynamic load of the system.

7. CRACKING OF THE BEARING SUPPORTS JET PUMP PROPULSION SHAFT

The situation with jet pumps is analog to propellers. The impeller also rotates in an irregular wakefield due to obstacles (i.e. supports etc.) in the jet pump housing. The difference with a propeller propulsion system is a double bearing near the impeller. So the shaft line in actual fact appears to be better supported.

The generated frequencies are higher and may in many cases reach 80-100 Hz. The displacements of movements are smaller but the generated forces could however be large (due to the high loading of the impeller). Fig. 9 shows a shaft line whereby the overhang is supported by a bearing. The bearing has of course a clearance and the impeller is axial static and dynamically loaded. The load distribution is irregular. The impeller shaft moves up in the bearing and vibrates within the available clearance. The small bending displacements are transmitted to the coupled intermediate shaft.

Since the intermediate shaft turns in air, system damping will not be large. The dimensioning of the intermediate shaft (diameter, bearing distance etc.) make it easily possible that bending resonance will occur. The resulting frequencies of such movements for aluminium constructions are not particularly welcome as mentioned before.

In some cases also resonance can occur locally. Cracking starts, is discovered and results in welding work being carried out on the damaged part of the construction. The strength of the material decreases and the problem will remain.

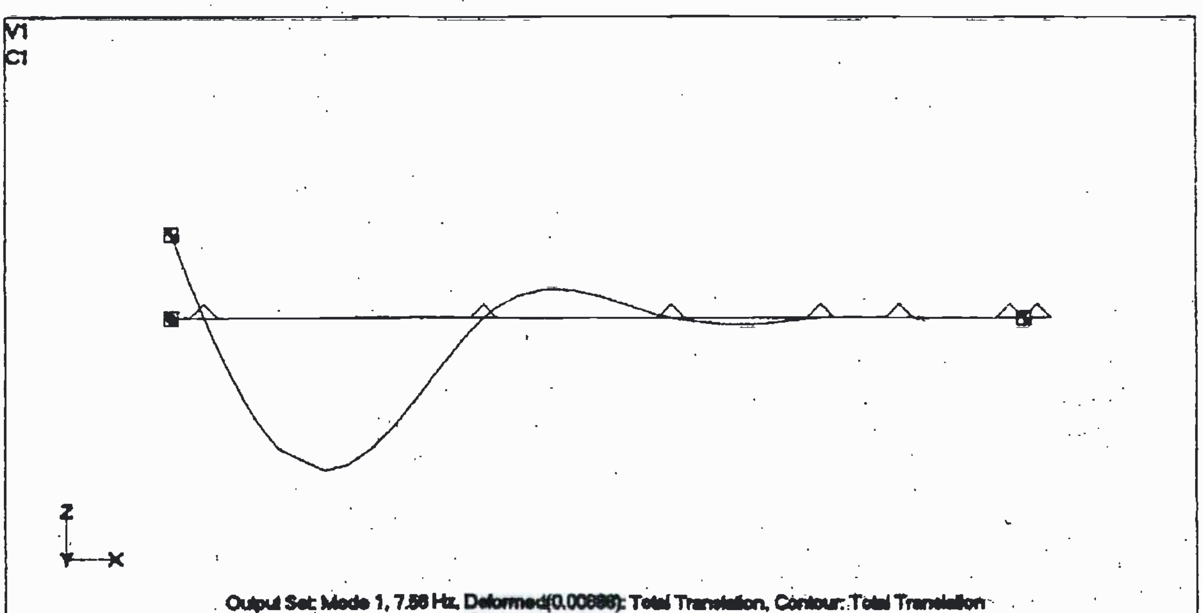


Fig. 8 Rotating propeller shaft line showing the 1st whirling mode

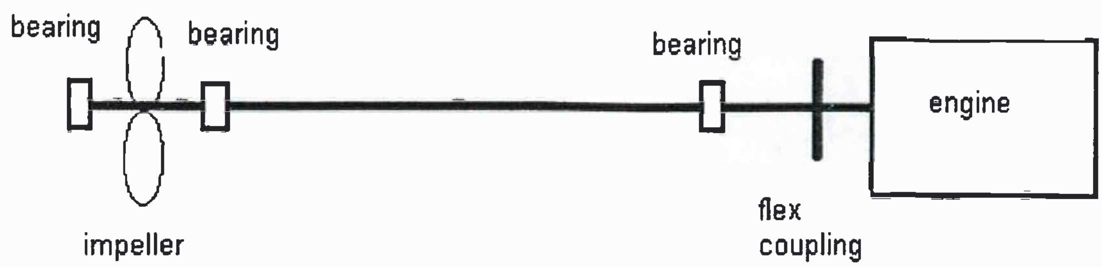


Fig. 9 Waterjet system shaft support for a fast vessel

8. CRACKING OF PLATING IN THE AFT PART OF THE VESSEL

Ships with a heavy vibration in the aft part have the chance of cracks due to local resonance of plate fields. Although it is often tank walls in the vacinity that are damaged, sometimes cracks appear in the hull plates near the web frame which support the necessary system brackets. Relation of cracks with cavitation phenomena is exceptional since, as mentioned before, the dynamic energy is too small. The hull plates which are in contact with water are (well) damped (due to added mass) so a lot more energy is necessary to create cracking thereof.

9. SOME DESIGN SOLUTIONS TO SOME OF THE ABOVE PROBLEMS

9.1 VIBRATION BEHAVIOUR OF ENGINES

Engines generate free moments and forces due to flying mass via the crankshaft construction. The timing, phase and amplitude of these free moment and forces differ depending on the number and configuration of cylinders and injection sequence. Also manufacturers fit in some cases a balanced shaft to solve (i.e. minimise) the vibration problems.

This means that the choice of engine is important. A large number of cylinders means a better distribution of forces and, in relation to the generated power, lower values. Disadvantages will include an increase in maintenance and engine prime cost. It is clear that in the relationship weight versus power the high speed engines are most attractive. A benefit is also profit of volume and the vessel prime cost.

It is clear that torsion vibration have to be as small as possible. A large number of cylinders make the generated moments in the engine smaller. On the other hand the resonance characteristics of the spring-mass system play an important role. Propeller, gearbox and shafting are elements with less freedom of parameters to choose from. The choice of the flexible coupling in the shaft line behind the engine offers more possibilities. On the torsion side, torque-stiffness and damping play a dominating role. Manufacturers produce a wide range of stiffness values by the structure and dimensioning of rubber elements. With the aid of torsion calculations the resulting effect of the designers choice of system parameters can be studied.

The lower resonance frequencies is dependent on the chosen parameter values. It is not only the vibration level but also the expected life cycle that is dependent on the torsion behaviour. A calculation in the design stage is a necessity and in relation to ship vibration attention had to be paid to:

- no critical rpm's in the operating range;
- introduction of the second order universal load in the calculation of V-drive installations;
- attention to effects of propeller or impeller blade frequency.

The choice of flexible supporting elements for the engine (sometimes coupled gearbox) had to be based on static and dynamic load scenarios. Also the radial stiffness of flexible coupling in the shaft line had to be incorporated and accounted for within these calculations. To this end, two means of supporting engines may be recognised:

- flat bed construction. The elements are loaded in a vertical sense;

  V-bed construction. The elements are loaded under an angle between 20 to 45°.

In a flexible support construction, a continuous shaft misalignment takes place. The flexible coupling has to solve the displacement problem. It will be clear that V-bed construction minimises the displacements however the static load on the rubber elements increase. Due to the load distribution over the engine the choice of the elements differ resulting in a different local stiffness.

For a problem-free design a proper finite element design calculation is necessary. After dimensioning the element on static load and acceptable deformation the six degrees of freedom in the case of dynamic movement has to be calculated. That means 6 resonance frequencies. These resonance frequencies are not to coincide (match) with the operating range with regard to torsion and the lateral free moments and forces of the engine.

Often, the main purpose for the use of flexible support constructions is the desire to minimise on board noise levels. The flexible supports have therefore to be designed in such a manner that the 6 resonance frequencies are below 50 Hz. Ideally there are no problems in engine support constructions as they are loaded in the supercritical zone. Only one thing is forgotten in this approach, what happens to the engine? With a super elastic support the transmitted forces are small, but the existing forces and moments cause heavy vibration of the engine. This has a negative effect on the life cycle of the engine. So again back to the calculations. With the use of finite element calculations a balance may then be chosen between transmitted and remaining forces and thereby save the life cycle of the engine.

To this effect, it is in any case advisable to distribute (with the use of steel plates) the support zone over a bigger area of the aluminium foundation.

9.2. SHAFT LINE VIBRATIONS

When designing fast vessels attention is paid to the hydrodynamic parameters and the wake field around the propellers. Obstacles which effect the wake field (i.e. shaft brackets etc.) are a necessary evil from a construction point of view. Solutions have to be found in the shaft line dynamics. First of all, modelling of the system in a finite element calculation is also necessary. Again the calculations are to be divided in a static and dynamic mode.

The static calculation gives information regarding the slope of the shaft line bending as a result of the shaft weight and that of the propeller. It also informs the designer about the slope in the bearings and the resulting supporting forces. When these are near zero first life cycle problems arise. It is considered advantageous to include in the calculation the thrust and eccentric thrust moment.

The dynamic calculation results in the vibration modes, frequencies and bending forms. It will be clear that the non matching of blade frequency and the different vibration modes is high on the designers agenda at this stage of the calculations. When these frequencies match problem(s) arise. Necessary input data for these calculations are shaft diameter, bearing distance, material, mass of the propeller and bearing support centre-point. When there is no static load on the bearing the support point had to be obliged. Many questions arise around the stern gland. Is that a support point or not.

When there is a bending resonance the aft bracket bearing receive the main dynamic load. A very critical factor in this is the distance between centre propeller and aft bearing centre bearing point. The slope of the shaft in the bearing brings the support point aftwards. This results in an increase in the subsequent resonance frequency. In order to prevent this from occurring the bearing can be bored in a slope or conic.

Waterjet pump installations also require the above mentioned shaftline calculations. The impeller is subjected to a lifting force which is the result of the thrust moment due to the effect that the wake field has on the impeller. The aft bearing load is zero and the shaft moves in a circular fashion within the available bearing clearance. Whirling of the intermediate shaft must be eliminated therefore by the use of correct design parameters. These can be deduced by making a number of vibration calculations using different design parameters. Finally apply flexible couplings to isolate bending moments but be aware of torsion vibrations.

Application of V-drive installations needs an intensive study of the torsion vibrations. Important elements are:

- no resonance in the speed range of the second order;
- no resonance of the main engine orders in the speed range;
- application of a (max) multi cylinder engines;
- no resonance of blade frequency or the first harmonic in the speed range;
- calculate the second order torque amplitude;
- dimension the elastic coupling in relation to above remarks;
- apply if necessary rotating mass.

It's clear that the V–angle must be kept as small as possible. Pay attention to mounting instructions of the manufacturer (advice fork positions).

Distribute bearing support as wide as possible using steel bed plates. Also the gearbox requires attention. The gearbox has not only to deal with the thrust (static and dynamic load) but also to content with (and therefore withstand) the forces from the universal joint at the other end.

Calculate the whirling resonance frequency of the intermediate shaft. Resonance of the second order within the speed range + 1.5 max rpm is not acceptable. This is so since there is no damping.

10. CONCLUSIONS

Slamming loads on fast aluminium vessels can cause excessive damage. Not only in the short term but also over a longer period of time (i.e. due to fatigue). This should be taken into account when design the vessel with regard to hydrodynamic loading and construction details.

The relationship between installed power and the mass of aluminium fast vessels assure a large chance for life cycle problems of the aluminium construction parts.

The main problem are the vibration levels caused by a lack of mass.

The maximum realisable resonance frequency of aluminium welding construction match, in a worst sense, those frequencies generated by the propulsion installation.

Torsion vibrations and finite elements calculations are necessary to design and create reliable installation.

Based on practical experience some important remarks have been made in which it is advised to compare design and the calculation outputs as much as possible.

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