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Interaction and Compatibility Between Machinery and Hull from a Static and Vibratory Point of View

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1. INTRODUCTION

The rapid growth of the world economy over the past thirty years, together with such political events as the closing of the Suez canal led to drastic modifications of the pattern of maritime transport.

As a result, new types of vessels have emerged such as fast container and Ro-Ro ships, huge and powerful LNG carriers as well as giant tankers commonly called VLCC and ULCC, their tonnage exceeding 500,000 DWT.

On Figure 1 is visualised the increase of the dwt of tankers within the past years.

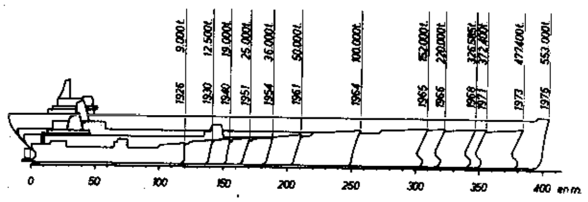

Fig. 1 - Increase of the DWT of tankers in the last years

The introduction of welding accompanied by recently developed calculation methods (e.g. finite element technique and creation of speedy powerful electronic computers) have allowed the considerable lowering of steel-work scantlings.

But on these bigger and bigger, and more flexible structures, have been installed more and more powerful propulsive plants characterized by shorter and much stiffer line shaftings.

During the operation of these newly delivered ships, troublesome problems adversly affecting the behaviour of machinery and ships'steel work started to occur. These problems being investigated by Bureau Veritas in order to discover the causes and propose remedies quickly led to the conclusions that the influence of hull deformations adversly affecting the correct behaviour of propulsive plants are at the origin of these troubles.

In addition to the interaction between static phenomena affecting both hull and machinery, vibratory problems have been also more and more frequently encountered leading to the emerging philosophy for a simultaneous treatment of :
- hull and machinery,
- and static and dynamic phenomena occurring on ships.

In this paper are briefly analyzed, in view of the gained experience, particularities of behaviour from point of view of compatibility and interaction with the hull of :
- tail shafts and line shaftings,
- Diesel engine driven propulsive plants,
- steam turbine and gearing driven propulsive plants.

Also practical and theoretical means and ways to assure this compatibility are presented, which are leading to the philosophy of integrated and simultaneous treatment of static and vibratory phenomena of hull and machinery.

2. INVESTIGATION OF DAMAGES OCCURRED TO :

2.1. Stern-gears and tail-shafts

Among the most frequently encountered damages affecting the liners of tail-shafts, supported by lignum vitae bushing of stern tubes are those due to erosion-corrosion (or pitting).

On Figure 2 are presented photos showing such damage patterns, which correspond to the number of blades of the concerned propeller.

The lignum vitae supporting such destroyed tail-shaft liners have been rapidly worn, showing the most important wear down at the aft extremity of aft bush, the rate of wear often decreasing in time.

Experimental investigations of such damages have revealed the presence of severe vibrations of tail shaft and even of aft part of the hull accompanied with leakages of stern glands, and being due to non rational, or old fashioned straight line, alignment of line shafting (1), (4), (12).

As to the most current damages of white metal bushings, two examples are presented on Figures 3 and 4.

The damage of aft bush shown on Figure 3 was detected during the guarantee inspection of the tail shaft after only six months of service.

Figure 4 shows the photo of the forward bushing of the stern tube of a tanker, which has been seized at the beginning of sea trials, only after six hours of main engine operation.

The analysis of the causes of these both damages have shown that the non-rational distribution of static reactions in way of tail-shaft bearings was at the origin of the incidents (see § 3.2.1. and Figure 28).

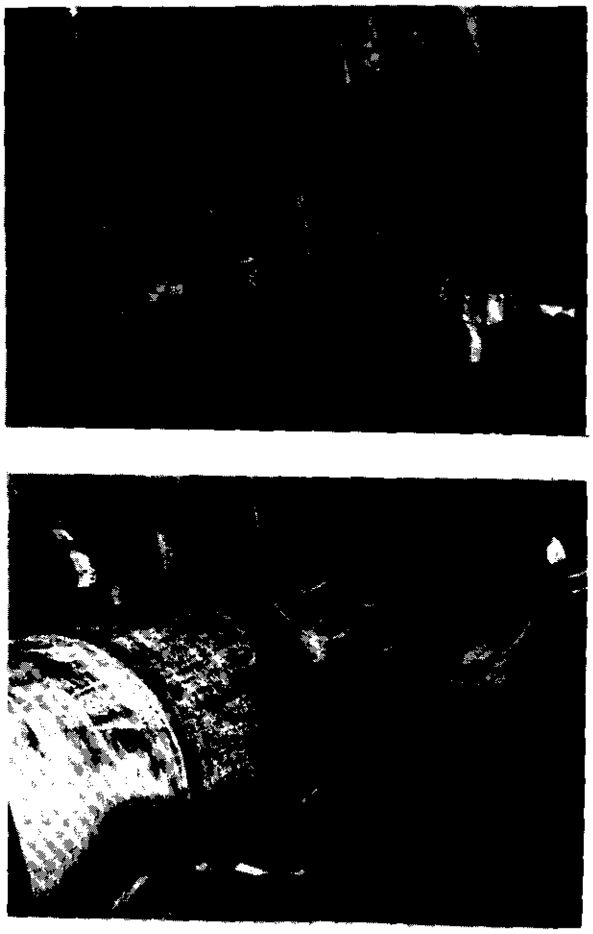

Fig. 2 - Damage by erosion-corrosion (pitting) of tail-shaft liners

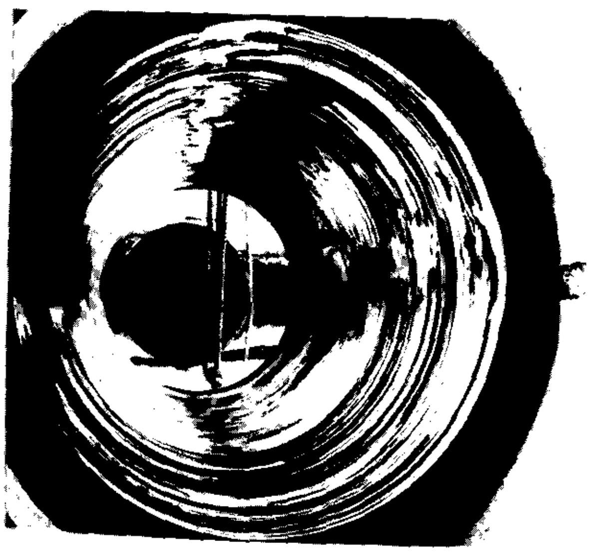

Fig. 3 - Seized white metal stern-bushing

Fig. 4 - Upper part of babbitted forward bushing seized after sea-trials

In fact, if the alignment of line shafting is executed according to straight and not precalculated curved line :

- local overload occurs in way of aft extremity of aft bushing,
- loss of contact between forward tail shaft journal and its bushing occurs leading often to appearance of negative reaction affecting even more unfavourably the local overload of the aft extremity of aft bushing.

Both above mentioned phenomena lead to the appearance of lateral vibrations of tail shaft, breaking of oil film and destruction of white metal.

The shocks of tail shaft journal against its support (bearing reaction forces) are at the origin of the forced vibrations of the aft part of the hull as it is illustrated on Figure 5.

On this Figure are shown the recordings of vertical and transverse vibrations of tail-shafts being at the origin of stern gland leakages but also of forced vibrations of the aft part of the hull (as shown on the upper part of the Figure). This case illustrates well the interdependence of static and vibratory phenomena of hull and machinery.

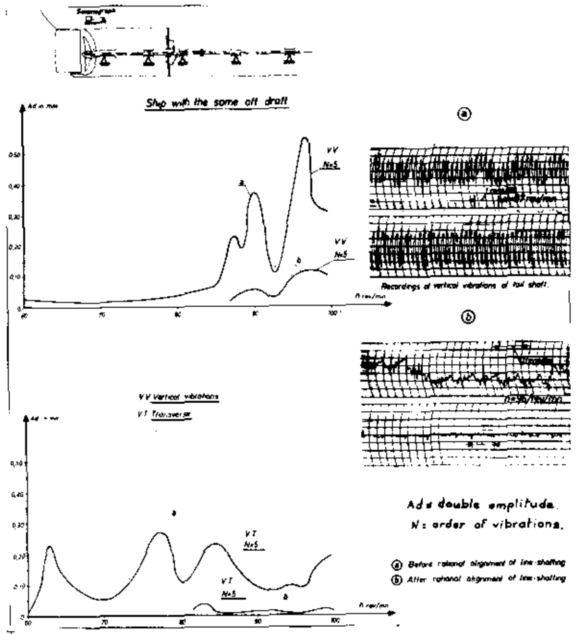

Fig. 5 - Forced vibrations of hull excited by tail-shaft vibrations

On the lower part of this Figure are shown the recordings of vertical and transverse vibrations of tail shaft and corresponding response of the aft part of the hull after execution of ratinnal alignment (4).

This operation aims at suppressing not only leakages of stern gland and severe hull vibrations but also at assuring the correct behaviour of tail shafts and their bushings of lignum vitae or white metal type if this is the case.

On Figure 6 is presented the aft bushing of the stern tube of a bulk carrier of 18,000 dwt, the boring of lignum vitae being executed according to calculations of rational alignment and with sloped axis,

The increase of clearance after one year has been of 0.25 mm and the darker zones are well indicating the evenly distributed, over the whole length of the bushing, contact pattern.

Also for this ship, previous very heavy hull vibrations have been completly eliminated.

But as the size of ships and the power of machinery have increased, it has been stated that slope boring of stern tube was not sufficient to assure the correct behaviour of white metal, when anew severe damages to white metal in way of aft part of aft bushings have been encountered. On Figure 7 is presented a photo of such a damage.

In order to cope with such difficulties it was necessary to calculate the alignment and contact conditions by taking into account the flexibility not only of shaft but also of white metal, these calculations leading to the adoption of double or even multi-sloping of white metal (see § 4.1. and Figure 35).

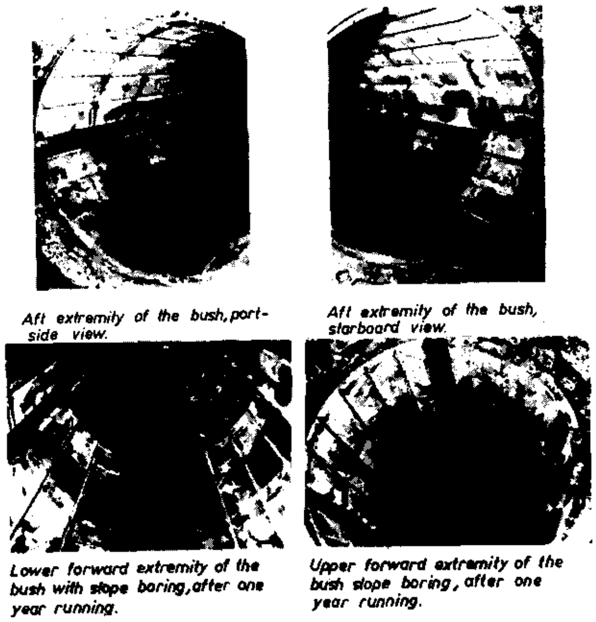

Fig. 6 - Aft bushing of stern tube after one year's operation, the boring-out of lignum vitae being carried out according to a sloped axis.

Fig. 7 - Deteriorated aft part of aft bushing of a 260,000 DWT. VLCC.

2.2. Diesel engine crankshafts and their bearings

Damages to Diesel engine driven plants originating in the interaction phenomena between hull and shaftings, have mainly affected the behaviour of crankshafts and their bearings (3), (7), (12).

They have been accompanied also by vibratory phenomena unavoidably leading to accelerated comsumption of fatigue potential of the involved material.

On Figure 8 is presented a fracture of a web due to bending strains at the origin of which was the non-rational alignment of the crankshaft.

The executed investigations of the causes of crankshafts damages have proved that before damages to crankshafts, their main bearings are seriously affected as it is shown on Figure 9.

Fig. 8 - Fracture of a web due to bending.

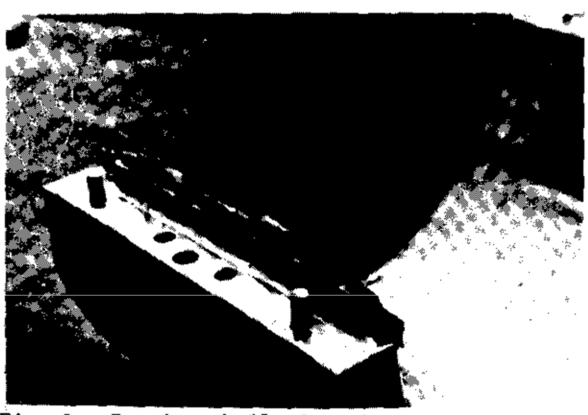

Fig. 9 - Bearing shells deteriorated by seizure

As regards these damages it concerned a series of 14 dry cargo ships where the repeated deterioration of white metal, always affected the last-but-one main bearing (7). In these cases the adverse external influences coming from line- and thrust-shaft alignment have been stated.

On Figure 10 are presented recordings related to :

- engine bed-plate deformations due to waves,
- displacements and vibrations of thrust-bearing and crankshaft,
- stresses in bed-plate and bending moments in thrust-shaft.

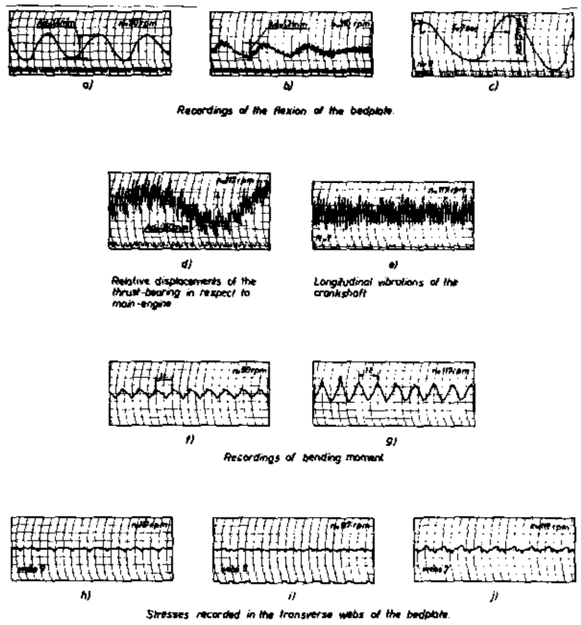

Fig. 10 - Recordings of vibrations of Ship A before realignment

From these recordings the importance of deformation of double bottom steel-work and thrust bearing foundations due to sea-swell, affecting thrust- and crank-shaft alignment conditions, could be stated.

Figure 11 shows the recordings of thrust-shaft and fly-wheel vibrations, excited by combustion effects, favourable conditions for these vibrations being created by the non-rational alignment of line shafting and the deformability of double bottom steel-work.

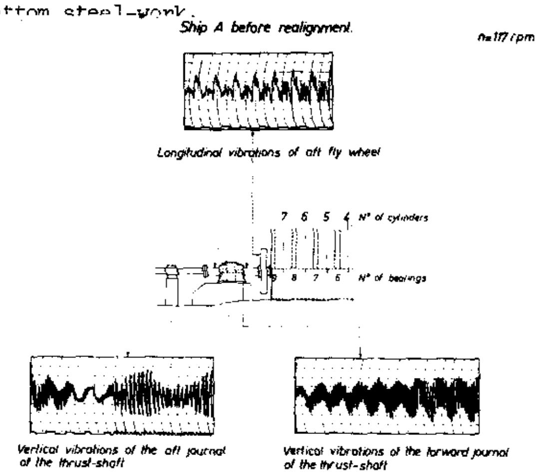

Fig. 11 - Vertical vibrations of thrust shaft excited by combustion effects

On Figure 12 are presented the alignment conditions of line and crank-shafts of two of the concerned ships, A and H, on which the above described damages have been also encountered.

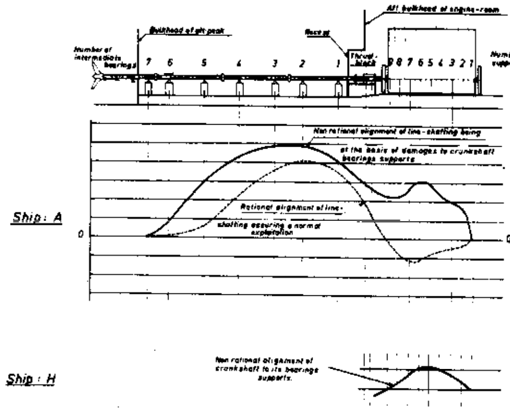

Fig. 12 - Résumé of alignment conditions of ships A and H

The remedy consisted in the execution of :
- double bottom reinforcement,
- realization of rational alignment and distribution of reactions of line shafting,
- sagging predeformed realignment of main engine crankshaft.

On Figure 12 are also presented the new realignment conditions of ship A (see also § 4.2. and Figure 38) and on Figure 13 are shown the vibrations recorded after these operations.

The comparison with Figure 12 is self explicit. After these realignments the correct behaviour of bearings and crankshafts is to be reported for all ships (7).

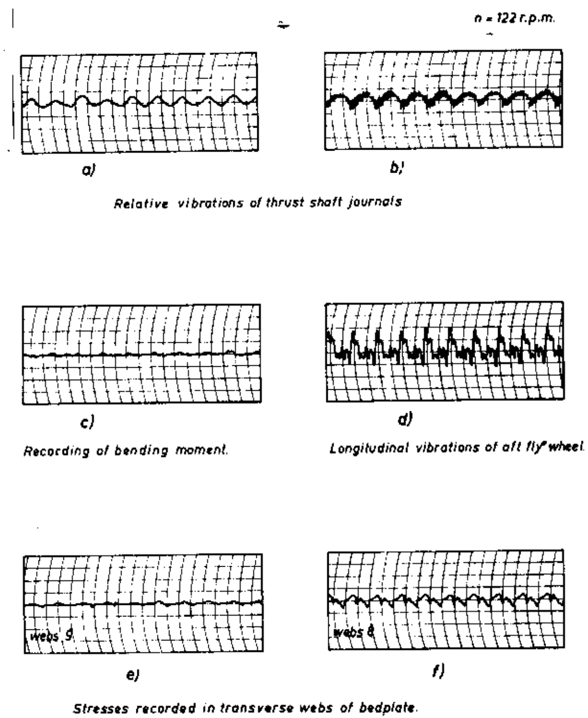

Fig. 13 - Recordings of vibrations of ship A after realignment

Another type of damages to crankshaft bearings in the form of mirorlike crazing of white metal affecting this time several bearings situated in the aft part of 12 cylinder engines, and which could be attributed to external influences coming from steel-work deformations and non-rational alignment of line- and crank-shafts, will be shortly presented hereafter (3). Figure 14 shows this type of damages.

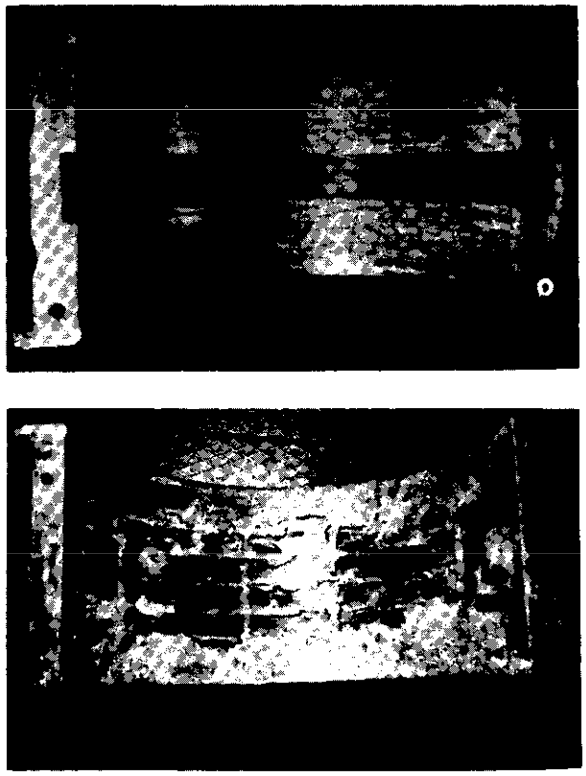

Fig. 14 - Crazing of the bearing metal (after one month of operation) as a result of hammering

This time they concerned a series of 6 ships who were at the time of their building the biggest in the world Diesel engine driven tankers (3).

On Figure 15 are presented the results of deformations of engine bedplates of three tankers of this series due to loading conditions, where important hogging deformatinns could be seen.

On the lower part of Figure 16 are also shown the recordings of deformations of bed-plate due to loading conditions of one of the concerned tankers.

If it can be stipulated that 4.2 mm deformation of bed-plate occurred on waves of effective height of 2.8 mm, the influence of steel-work deformations on the behaviour of diesel engine and its crankshaft is self-evident.

After several damages to bearings, as shown on Figure 14, it has been also stated a fracture of the pin in way of one of the cranks of the main bearings having suffered repetitive damages as described.

Before replacement of this crankshaft, supplementary investigations and measurements have been also executed, especially in order to check the contact conditions between journal and bearing in way of the cracked crankpin. On the upper part of Figure 17 are shown the recorded vertical vibrations of the journal.

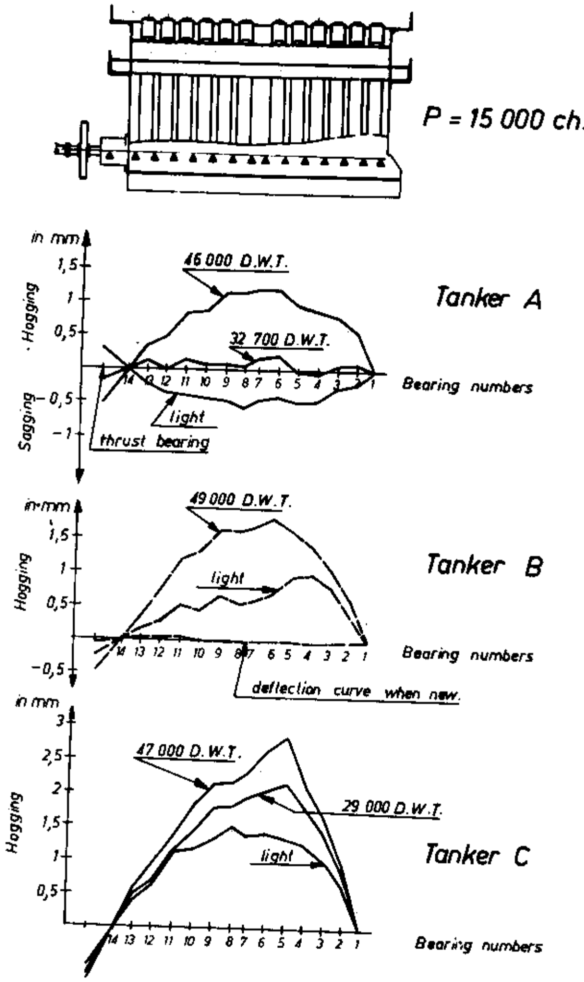

Fig. 15 - Deformations of engine bed-plates of tankers due to loading conditions

The double amplitudes, being of 0.42 mm and equal to the clearance, indicate that the crankshaft journal has not been in permanent contact with its lower shell, hence breaking of oil film and rapid deterioration of white metal due to fatigue phenomenon have occurred.

The remedies applied to these engines consisted in :

- rational realignment of line shafting,
- supporting of vertically elastic thrust bearing,
- sagging predeformed realignment of crankshaft (see tanker A of Figure 15).

On the lower part of Figure 17 are shown the recordings of vertical vibrations of the journal after realignment operations. The comparison with recordings shown on the upper part of this Figure is self-evident (3). Since these operations the main engines are behaving correctly, proving the necessity of taking into account the interdependence between hull and machinery from static and dynamic point of view.

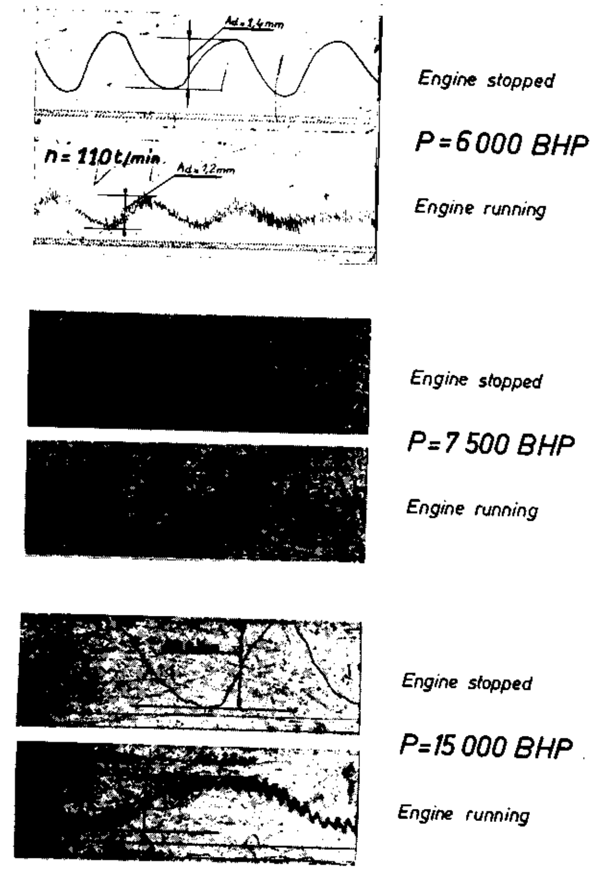

Fig. 16 - Recordings of bending bed-plates of Diesel engines due to sea-swell.

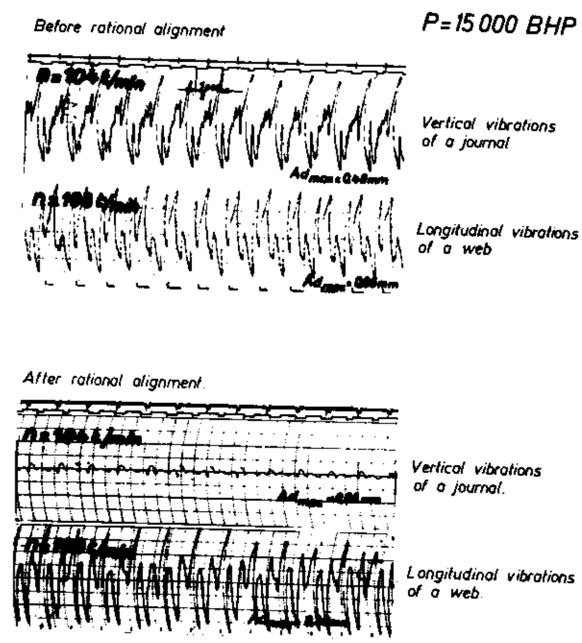

Fig. 17 - Recordings of vibrations of crankshaft journal and web for two alignment conditions

2.3. Gears of turbine driven ships

The interface between hull and machinery of turbine driven ships have been felt in the past mainly on last reduction of main gearings (bull gear). Figure 18 shows one of the extreme damages encountered by the Author.

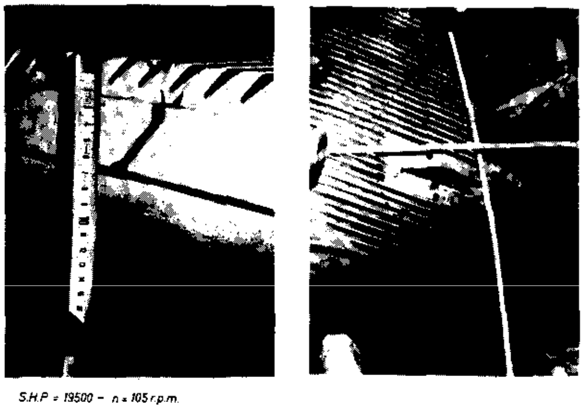

Fig. 18 - Cracked teeth and rim of bull gear wheel

Fortunately, before the appearance of such cases, some warning signals can be observed on the meshing teeth.

Figure 19 indicates such appearances which are to be ascribed to external influences due to lack of particular attention paid to interdependence problems between hull flexibility and line shafting stiffness (5), (9), (10), (12), (15).

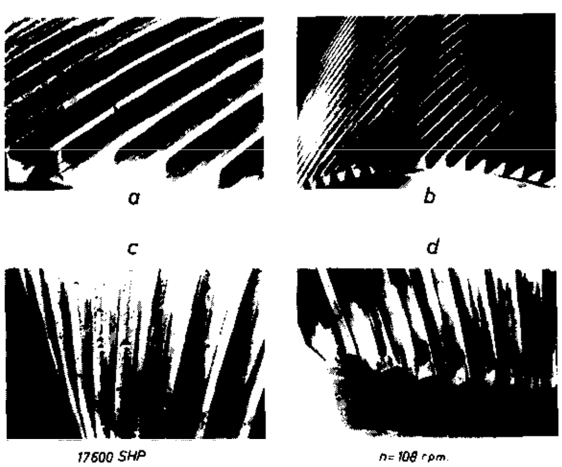

Fig. 19 - Pitting, spalling and heavy wear of after helix due to tilting of thrust block and non rational alignment of line shafting.

This Figure shows the teeth of aft helix of bull gear which have to be changed after two years of operation. The executed investigations have shown the very detrimental influence of non-rational alignment of shafting, and the adverse influence of thrust-block tilting, on bull gear behaviour.

On Figure 20 are shown vertical relative vibrations of thrust shaft indicating corresponding contact conditions with thrust bearing journals and journal bearings (see also § 3.1.4. and Figure 27, § 3.2.3.2. Figure 32 and § 4.3. Figure 43)

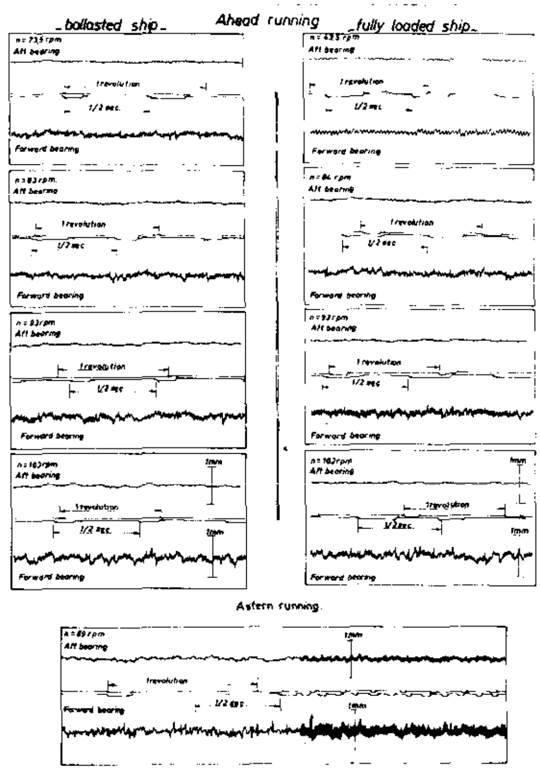

Fig. 20 - Vertical relative vibrations of thrust shaft

Another interesting example of the influence of steel-work deformation (without interaction of thrust block) concerns two tankers, gear damages of which are shown on Figure 21.

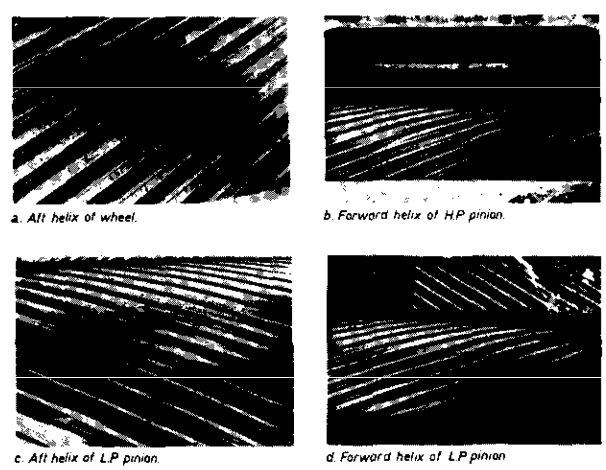

Fig. 21 - Pitting, spalling and undercutting of teeth of three mobils of bull gear

Figure 22 presents the time history of deterioration and then its stabilisation after execution of rational alignment (10) analogue to the above mentioned case.

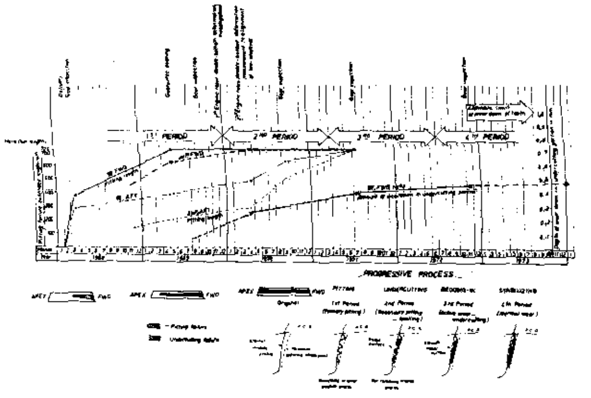

Fig. 22 - Time history of pitting and undercutting and stabilizing of wear down.

3. INTERACTION PROBLEMS BETWEEN MACHINERY AND HULL

In view of the experiences collected and lessons drawn during investigations and trouble shooting work, some examples being presented above, it becomes obvious to the Author that in the past not enough attention has been paid to the problems of interaction between the flexibility of steel-work in way of engine room and the stiffness of line shafting and this from both static and dynamic point of view. In what follows it will be briefly presented the results of the experience gained from theoretical and experimental investigations executed by the Author.

3.1. Flexibility of steel-work in way of engine room

As the compatibility between flexibility of steel-work and stiffness of line shaftings of propulsive plants is to be assured in the region of the engine room, the attention will be particularly centered on some aspects of the problems involved in this region of the hull.

3.1.1. Hull girder

Deformations due to loading conditions will affect the contact conditions between the line shafting and its bearings connected rigidly to the steel-work, in different ways and this at first according to engine room location along side the hull girder (2), (9), (12).

Figure 23 shows the deformations of the ships analyzed in § 2.2. with engine room situated in way of 3/4 L aft.

As it can be seen, in way of engine room there is an inflexion point, influencing in an important way the predeformation of the crankshaft (see Figures 12, 38). In order to determine these values it is necessary to proceed with calculations of deformations where the loading are determined by strips, the final loading being :

$$\pm\, q(x) = d(x) - p(x) \qquad (1)$$

and the final deformation is determined as follows :

$$f(x) = \frac{1}{EI}\int_0^L\int_0^L\int_0^L\int_0^L q(x)\, dx\; dx\; dx\; dx \qquad (2)$$

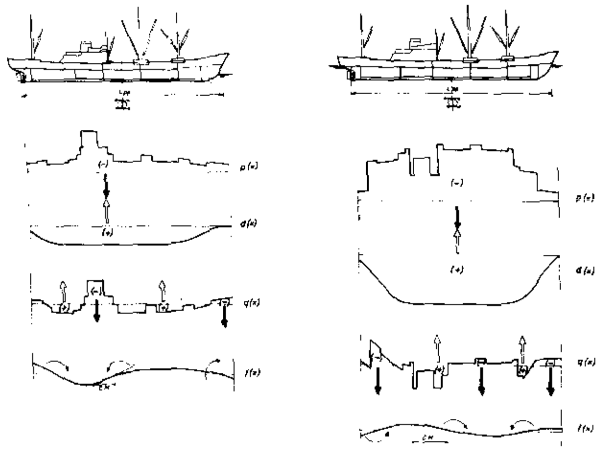

Fig. 23 - Deformations of hull girder of a ship with engine room situated 3/4 aft, for different loading conditions.

The particularly interested hull-girder deformations have been discovered by the Author for ships with engine room situated aft. Figure 24 presents the results of such experimental and theoretical studies.

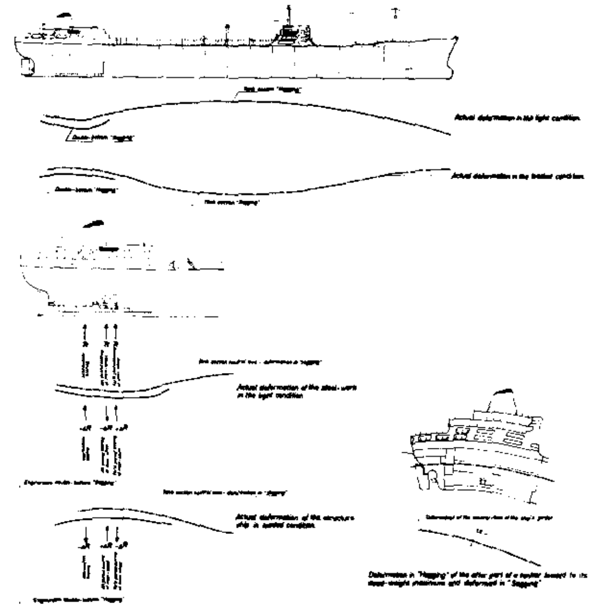

Fig. 24 - Influence of hull structure deformations of a tanker due to loading conditions on distribution of bearing reactions.

In fact, according to the classic theory of hull girder deformations, for loaded ship the deformation appears in sagging and for ballasted ship in hogging along the tanks. But this is not the case for the aft part of such ships where hogging deformation appears for loaded ship and sagging deformation for empty ship. Such deformations affect the alignment conditions of tail shaft, line shafting as well as crankshaft or bull gear, as shown on Figure 24.

3.1.2. Double bottom - Thermal expansion

Supplementary deformations affecting propulsive plant are due to local deformations of double bottom of engine room, behaving as a membrane embedded in outside plating and both engine room bulkheads. As the deformability of the floors is function of their length l^4 the increase of breath B of ships has exerted a very great influence on the flexibility of the double bottom membrane.

On Figure 25 are presented such results of calculations of a 250,000 DWT tanker, where hogging deformation for loaded ship is to be noted (9), (10) (12).

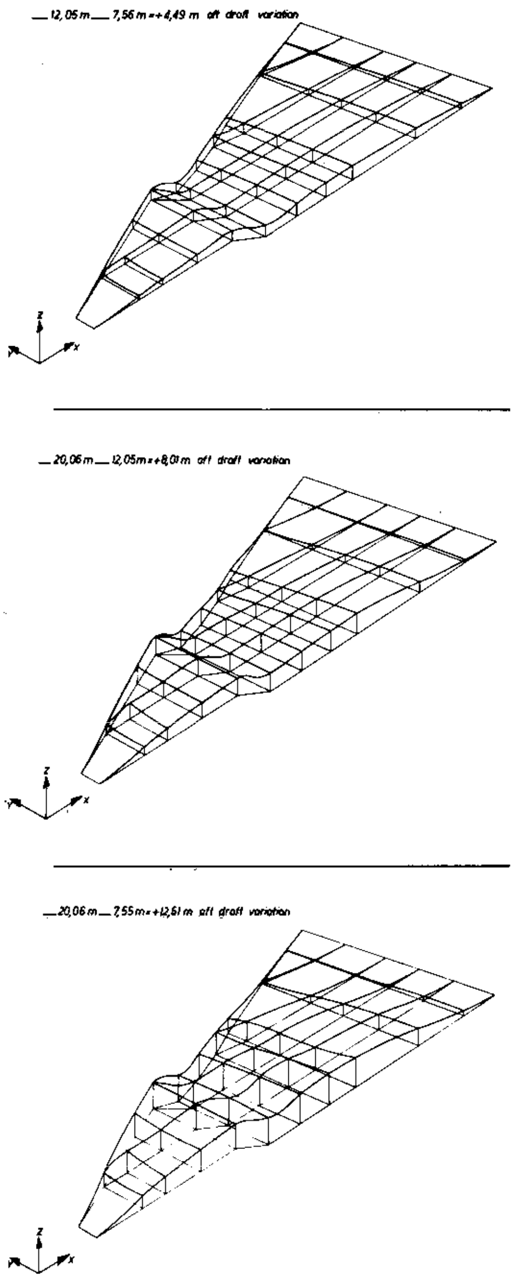

Fig. 25 - Deformations of double bottom steel-work in engine room due to loading conditions

It is also interesting to note alongside the outside shelling the hogging deformation due to before mentioned non-conventional hull girder deformation calculations.

For calculations of shafting alignment due to deformation of double bottom steel-work it is also necessary to take account of thermal expansion of steel-work foundation incorporated into double bottom as well as of bed-plate (for Diesel engine) or casing (for gear).

3.1.3. Outside shelling - Pillars - Superstructures

For tankers of about 300,000 DWT and above it has been stated that for some configurations of double bottom and webs of outside shelling and their respective stiffnesses in transverse direction, for loaded ships the double bottom is deforming in sagging (9), (10), as it is shown on the lower part of Figure 26.

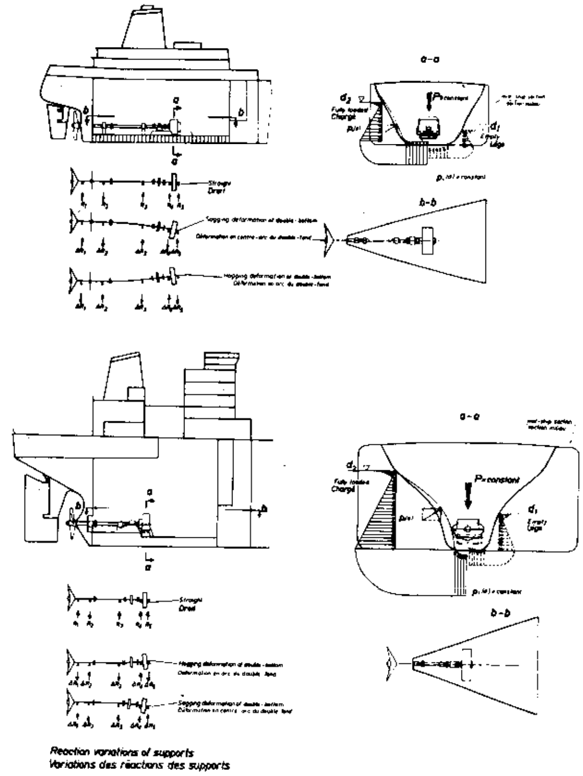

Fig. 26 - Influence of deformations of outside shelling on vertical displacement of bull gear as function of draught variations and aft body shape.

This Figure is also giving explanations of this phenomenon, due to bigger efforts acting on outside shelling webs than on double bottom floors and provoking their sagging deformations.

The implications of such deformations on line shafting alignment are also indicated on the same Figure (9), (12).

It should not be either forgotten when calculating double-bottom deformations the presence of pillars supporting the outside bulkheads of superstructures or engine room casing, and having an important influence on the deformability of the double bottom itself.

3.1.4. Tilting and rocking of thrust block

To flexibility problems of steel-work in engine room belong also the question of tilting, and rocking, of the thrust block of propulsive apparatus, being either main gear or Diesel engine.

In fact, under the influence of propeller thrust and its variations, the thrust bearing foundations are deflecting provoking the S shape deformations of the double bottom on which they are installed (11), (12).

The consequence of these phenomena is :

- modifications of executed alignment conditions of line shafting and propulsive apparatus shafts,
- appearance of vibratory phenomena of thrust shaft (see Figure 11 and Figure 20) and even crankshaft (as it has been described in §2.2. see Figure 9).

On Figure 27 is presented graphically this phenomenon and its repercussions on the behaviour of bull-gear (5), (10).

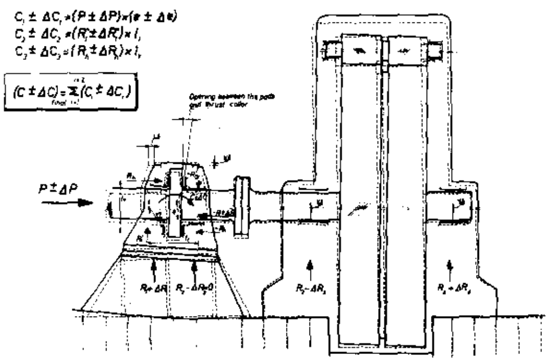

Fig. 27 - Tilting of thrust-block and its influence on behaviour of bull-gear.

This Figure is self-explicit and is to be analyzed in connection with § 2.3. see Figure 19. As to countermeasures against this phenomenon please refer to § 4.2. Figure 38 and § 4.3. Figures 43, 44 and 47.

3.2. Stiffness of line shaftings - Influences of support foundations

It has been shown in the previous chapters that the steel-work of big ships becomes flexible (2), (3) (12). But on these ships it has been needed to install more powerful propulsive plants with rpm tending to decrease. Hence for these increased torques for constant shearing stresses it was necessary to increase shafting diameters D which, over last thirty years, have more than doubled. Consequently, their moment of inertia to bending $I = f(D^4)$ has abnormally increased leading to the before mentioned incompatibility between steel-work flexibility and line shafting stiffness. Hereafter are briefly analyzed some aspects of this phenomenon for different parts of line shaftings of propulsive plants.

3.2.1. Tail-shafts (in conjunction with propeller and stern tube)

In parallel with the increase of power it was obvious that the weight of the propellers has also increased being at the origin of the increase of loads and local specific pressures in way of aft extremities of aft stern tube bushings (4).

On another hand, due to the conservative approach of shipbuilders to line shafting problems, the number of bearings of line shaftings has been maintained being at the origin of their discharging. Consequently with the overload of aft bushings it occurred in way of some shafting supports (especially forward support of tail shaft) negative reactions leading to a loss of contact between corresponding journals and lower shells, as shown on Figures 28 and 29.

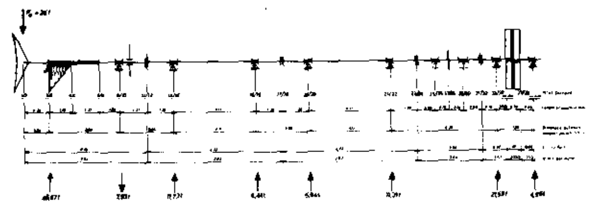

Fig. 28 - Distribution of static reactions of line shafting supports of a 48,000 DWT tanker (see Figures 3 and 4).

As the natural frequency of a beam supported on two bearings is the inverse function of the square root of the cube of span between them

$$f = f(l^{-\frac{3}{2}}) \tag{3}$$

this frequency has rapidly decreased often falling into resonance with propeller blade excitations being at the origin of problems described in § 2.1. (see also Figures 3, 4 and 5 as well as § 3.3. Figure 33).

In order to cope with these phenomena it was necessary to increase the natural frequency by decreasing the span between supports, or make them effective without negative reaction.

This has been achieved by introducing the rational misalignment of line shafting (see § 2.1. and 4.1.).

3.2.2. Intermediate shafts

On Figure 29 are presented the deformations and reaction distribution of supports of tail- and intermediate shafts, as function of their stiffnesses (I/l^3)

The tail shaft shown on the lower part of this Figure is analogue to the one shown on Figure 28, and for which it occurred harmful lowering of natural frequency (see also Fig. 33).

The realization of the increase of natural frequency of aft part of line shafting by suppression of negative reaction in way of forward support of tail-shaft and assuring correct behaviour of stern-gear of this shafting could be only achieved by curved rational alignment of line shafting. It was realized by lowering the last intermediate bearing, and even disloading it, in order to counterbalance the effect of heavy propeller on the reaction of forward support of tail-shaft.

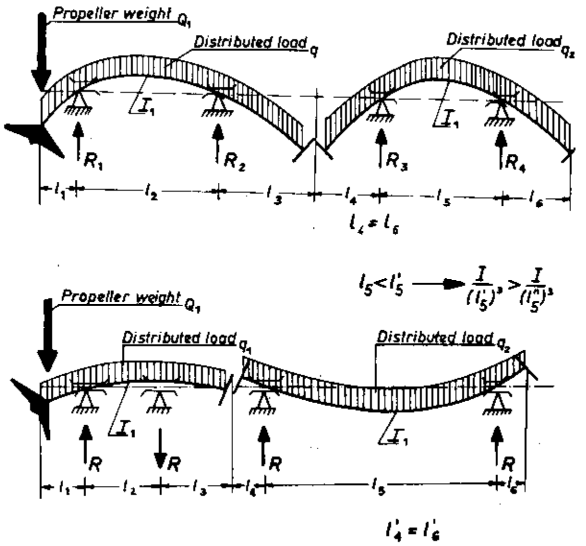

Fig. 29 - Deformation of tail and intermediate shafts as function of their stiffnesses

From what has been told above it follows that in order to create favourable conditions for the correct behaviour of line shafting it is necessary to realize :

- some parts of shafting rigid (to assure their elevated frequency in order to avoid the resonant response and its harmful dynamic amplification) and this concerns tail-shaft directly exposed to hydrodynamic excitations,
- other parts flexible (in order to facilitate the correct loadings of tail shaft supports).

This concept, the logic of which is obvious, has a supplementary advantage regarding the flexibility of the steel-work, namely that it allows to adapt the stiffness of line shafting, by its decrease, to the flexibility of the foundation basis on which the line shafting is installed, hence facilitating correct behaviour of propulsive apparatus, what will be discussed hereafter.

3.2.3. Thrust-shafts and shaftings of propulsive apparatus

The analyzed cases of investigations carried out for two main types of propulsion plants being Diesel engines and steam turbines with gears, are bringing indications on different requirements regarding the requested behaviour of thrust shafts. So hereafter are stipulated necessary requirements.

3.2.3.1. Crankshafts

As it has been exposed in § 2.2. damages to crankshaft bearings as shown on Figure 9 have been due to vibrations of thrust- and intermediate shafts (and fly-wheel) excited by the combustion effects in last cylinder. The fact that, after realignment and once the thrust shaft has been supported by the aft journal bearing, in which correct reactions have been realized by convenient rational alignment of crankshaft, bearing destruction has been stopped, indicates the necessity of having in way of the last journal of the thrust shaft a journal bearing.

This journal bearing is opposing the pivoting of thrust shaft around the last crankshaft bearing under the influence of combustion effects and consecutive loss of contact with last-but-one crankshaft journal leading to its hammering against the lower shell of the bearing.

In fact the Author's researches have proved that stiffness of thrust-shaft is much higher than the stiffness of crankshaft, the last one being much more flexible than it could be thought when analyzing Diesel engine Manufacturer and Classification Society formulas determining the scantlings of crankshaft webs and journals (or pins) (3), (8).

On Figure 30 are presented the values of real moment of inertia in bending and equivalent diameter of cranks.

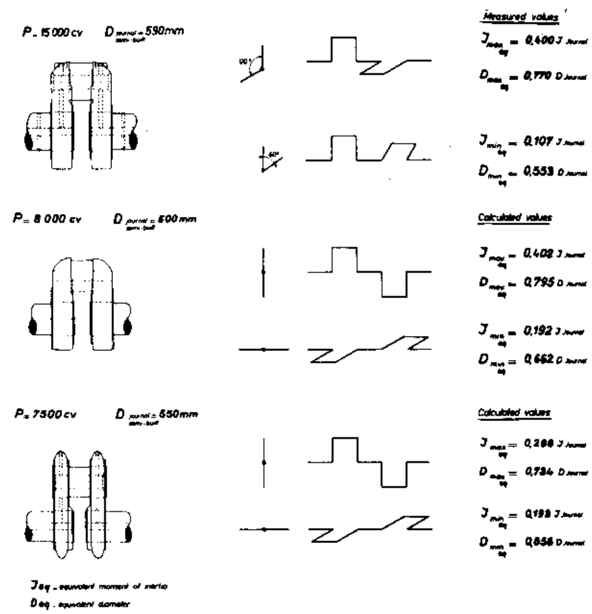

Fig. 30 - Values of real moment of inertia in bending and equivalent diameters of cranks.

As it can be seen they are function of angle between consecutive cranks and may vary from 0.1 to 0.4. of the moment of inertia of the crankshaft journal. This conclusion is well indicating the importance of thrust-shaft behaviour and its correct alignment conditions.

On Figure 31 are presented the values of stresses which may occur in cranks in case of the loss of contact between journal and lower shell of bearing as it has been analyzed in §2.2 and shown on Figures 14 and 17. These values indicate also clearly the importance of a correct internal alignment of crankshaft, taking into account external influences due to steel-work deformations as analyzed in § 2.2. and illustrated on Figures 12, 15 and 16.

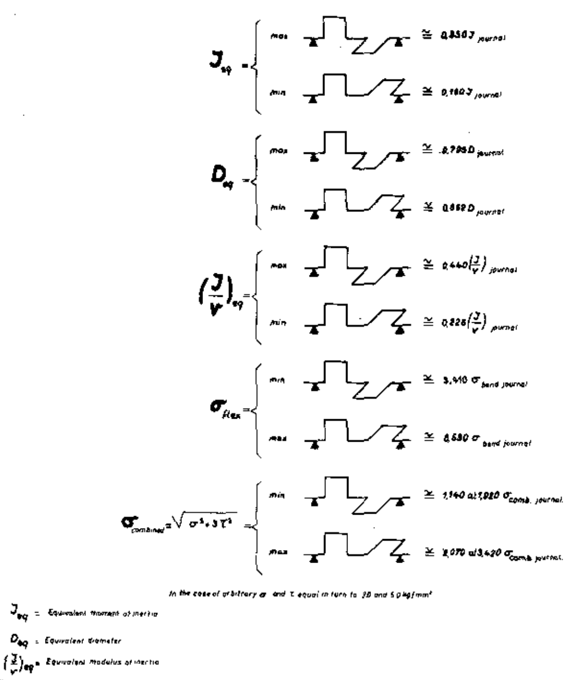

Fig. 31 - Stresses in cranks as result of loss of contact between a journal and its bearing.

3.2.3.2. Main-wheel shaft

The results of investigations presented in § 2.3. (see Figures 19, 20) together with considerations exposed in § 3.1.4. (see Figure 27) indicate that contrary to internal architecture of thrust bearings of Diesel engine, the thrust bearings of gear driven shaftings should not have any journal bearing in thrust bearing (5), (10), (15).

On Figure 32 are presented influences acting on bull-gear shaft bearing reactions. From the Author's experience it can be stated that the action of an eventual journal bearing in thrust-bearing, besides the influence of hull deformations and non-rational alignment of shafting, is the most harmful for correct behaviour of main gear and should never be admitted.

3.3. Mutual interdependence between static and vibratory phenomena

At the occasion of different investigations as well as some theoretical considerations the attention has been drawn on this phenomenon. However, it has been thought useful to explain this question by a typical example which has been also at the basis of the Author conceiving a philosophy of treatment of vibratory phenomena by research and consecutive detuning of forced vibration resonators (4), (6), (12).

Figure 33 visualizes this case.

It is question of a steam turbine driven 58,000 DWT tanker for which the Shipyard has executed careful studies related to hull girder vibrations and has kept the clearances in way of stern aperture in accordance with universally recommended values. Also calculations of natural frequencies of lateral and whirling vibrations have been executed. In view of all these studies no hull vibration should occur. But already during trials very heavy vibrations of the aft part of the hull have been stated.

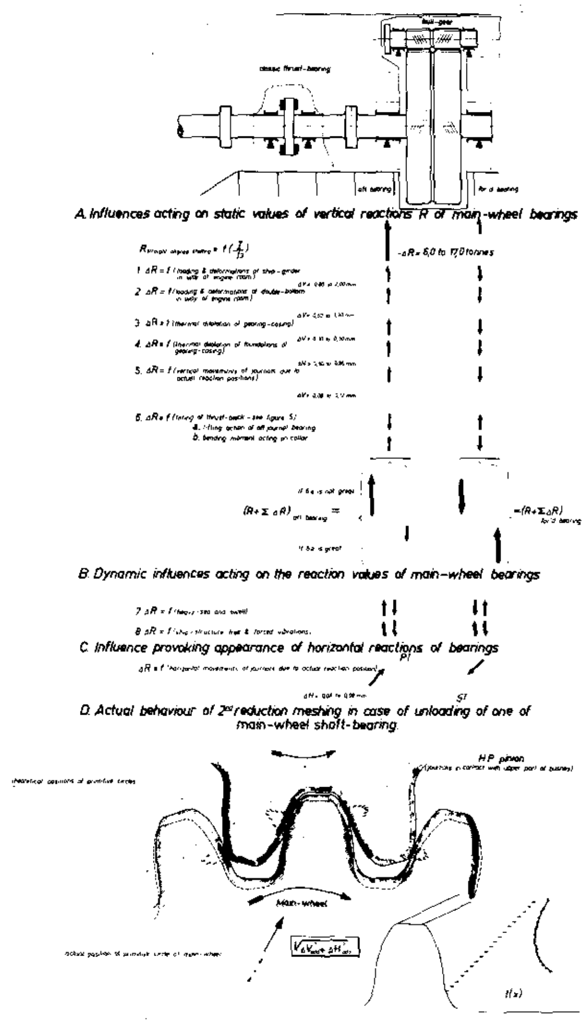

Fig. 32 - Influences acting on bull-gear wheel shaft bearing reactimns and their

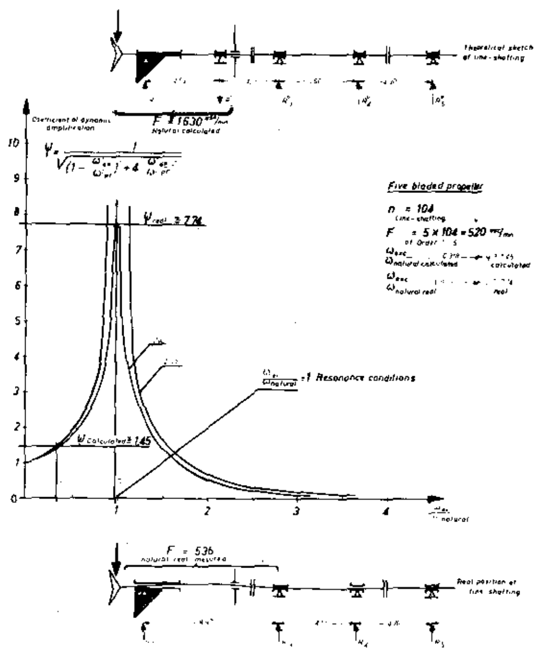

Fig. 33 - Effect of static contact conditions between tail-shaft journal and its forward support on the dynamic behaviour of this shaft (Fig.5)

The consecutive measurements executed by the Author have shown that they have been excited by the five order lateral vibrations of line shafting (see Figure 5 upper part). Nevertheless calculation results shown on the upper part of Figure 33 (and as it can be seen from recording of Figure 5) it has been stated the resonant response in lateral vibrations of tail shaft to fifth order excitation coming from the propeller At the origin of this unfortune has been the fact that when calculating the natural frequency of lateral vibrations, it has not been taken account of static conditions of alignment where there is no contact between journal and bushing in way of forward stern bushing due to the presence of negative reaction. Hence the real span of aft part of shaftings has been such as shown on the lower part of Figure 33 leading to natural frequency quite equal to this of fundamental excitations of propeller.

Hence the appearance of forced vibration resonators has led to the resonant response of the tail shaft, its hammering against forward stern tube leading to hull vibrations.

The problem has been solved by rational alignment of line shafting leading to detuning the resonator and cancelling the dynamic amplification of the response of the shafting. The decrease of vibrations of tail shaft and hull is also indicated on the lower part of Figure 5.

4. PRACTICAL AND THEORETICAL MEANS AND WAYS TO ASSURE COMPATIBILITY BETWEEN MACHINERY STIFFNESS AND HULL FLEXIBILITY

Some firm conclusions and criteria are emerging from the experience gained and lessons collected.

First of all it can be thought that it is better to foresee the troubles at the beginning of the construction by adopting the solutions which have given proofs in the past than to cure them.

Afterwards it is always necessary to follow carefully and survey the behaviour of ships and their propulsive plants in order to dispose of feed-back data allowing improvements of theoretical treatments at the project stage.

It is also necessary to carry on the theoretical research of newly occurring phenomena, forging necessary theoretical tools to tackle with them at project and building stages.

In view of such principles and following adopted order, it will be analyzed the problem of compatibility by dealing separatly with :
- line shaftings,
- Diesel engines,
- main-gears.

4.1. Rational treatment and concept of line shaftings and their foundations

One of the fundamental question, related to line shafting and aiming at assuring the correct behaviour of this part of the propulsive plant from point of view of compatibility with the hull, is the execution of rational fair curved line shafting alignment (1), (4). This problem is today belonging to the classic routine and such studies are mentioned here only for order sake.

But the lessons of the past, even with shafting rationally aligned, are drawing the attention on some particular points which may be useful to treat more in detail.

In fact, in many cases, several problems with stern gear, having bushings being equipped with lignum vitae, white metal,... have been due to some errors of boring-out, assembling or machining of conventional old-fashioned stern-tube with two bushings.

The Author's experience leads him to conclude that one never can be really sure what has been done with stern-tube, how it has been bored out, etc, hence a conclusion :

In order to be able to cope with some human errors, if happened, why not renouncing on this old solution (dating yet from big BRUNELL time) and foreseeing a one bushing stern-tube, and even with split shells, Such solutions exist (GLACIER, THURNBULL).

Introduction of one bushing stern-tube with roller bearing (SKF) or tilting pads (WAUKESHA) could be recommended also.

On the lower part of Figure 34 is presented a rational architecture of aft part of line shafting with one bushing stern-tube. The forward tail shaft support, being amovible and allowing corrections of alignment, is situated in longitudinal direction to meet the desiderata exposed in § 3.2.1. and aiming at stiffening this part of the line shafting which natural frequency of lateral and whirling vibrations is doing the line shafting overcritical.

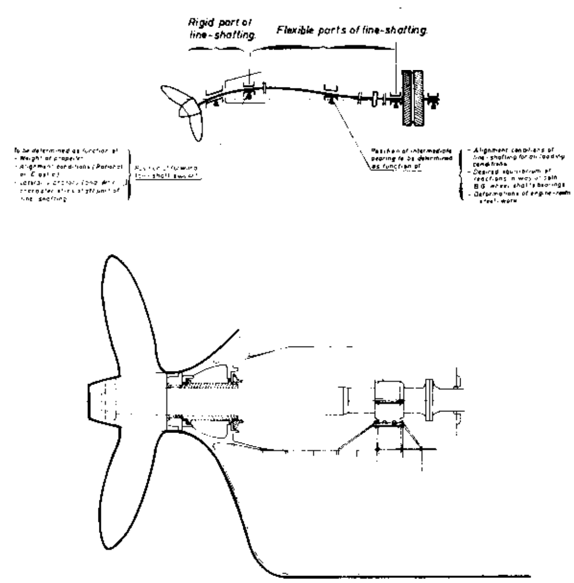

Fig. 34 - Rational architecture of aft part of line shafting with one bushing in stern-tube.

As it can be seen also from the upper part of this figure, forward of tail shaft, the long span of the intermediate shafts (of course previously calculated in order to avoid any lateral resonant response) assures the compatibility of the line shafting stiffness with the flexible steel-work of the hull.

But if white metal bush is adopted it is also necessary, especially for high power installation, to assure correct contact conditions between bended journal of tail shaft and white metal in order to avoid the destruction of the last one due to local overloads of aft extremity and especially for low rpm, when no sufficient oil film is present. Figure 35 presents the results of such calculations for cylindrical boring-out of the bushing and another one with double slope. The indicated comparative values are self-evident, such technique and calculations being adopted in BUREAU VERITAS from more than ten years.

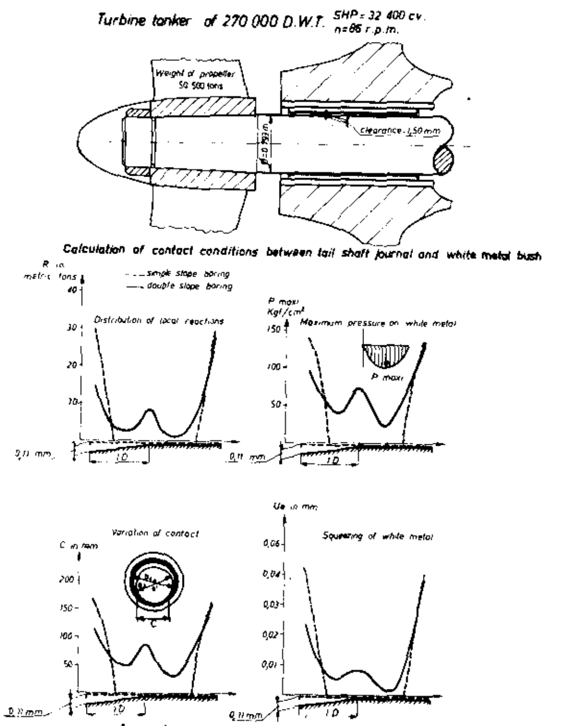

Fig. 35 - Contact distribution between tail-shaft journal and white metal of aft bushing of stern tube of a 270.000 DWT tanker

A further step in studies is also needed and this in respect of oil film building-up in stern-tube for normal and low rpm. Such calculations are executed in BUREAU VERITAS taking account of bended journal, double or multi-sloping of white metal and skewed (in two surfaces occurring) misalignment of tail-shaft.

Figure 36 shows an example of the results of such calculations (see also § 4.3. and Figure 47).

It is also necessary to note the increasing importance of 6 components of propeller forces and moments occurring on the propeller and being at the origin of important displacements and vibrations of tail-shaft journal. Several measurements already executed in way of aft stern-bushing have shown that tail-shaft at its normal rpm is often laying at 9 or 10 o'clock.

On Figure 37 are shown the effects of propeller generated forces and moments on tail-shaft.

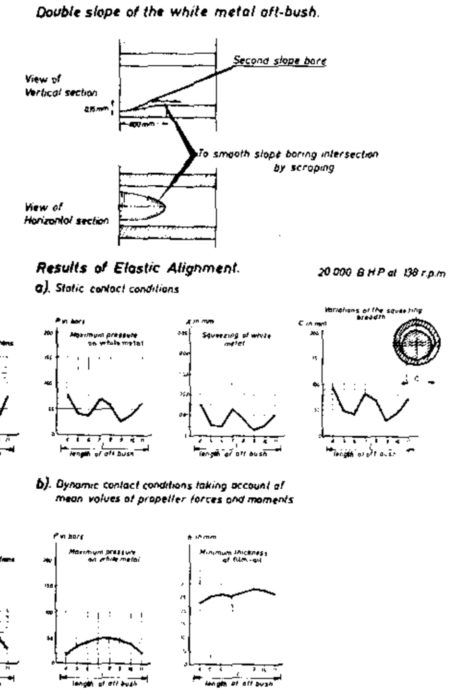

Fig. 36 - Contact conditions and building-up of oil film in stern tube bushing.

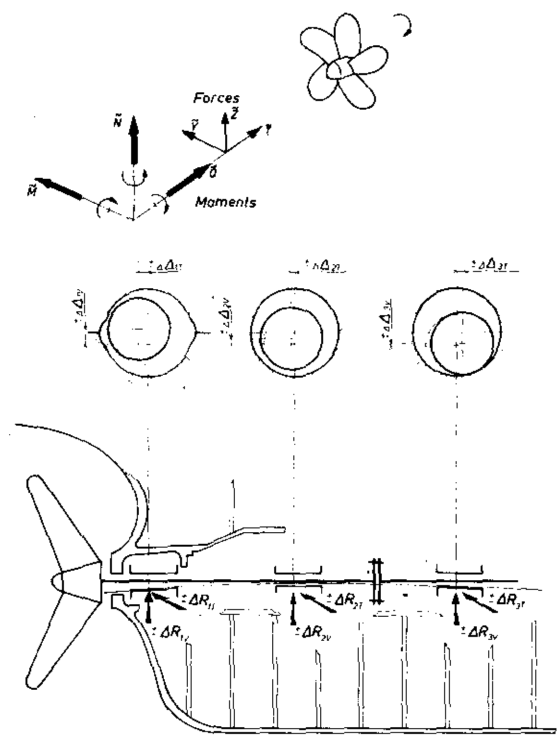

Fig. 37 - Effects of propeller generated forces and moments on tail and intermediate shafts

4.2. Rational treatment and concept of Diesel engines and their foundations

This type of propulsive plants requests a treatment from three points of view :

- first, concerning external influences of line shafting,

- second, concerning development of engine beam concept in view of steel-work deformations,
- third, concerning the proper alignment of crankshaft to counteract detrimental steel-work deformations.

The different examples and analysis of damages exposed in § 2.2. as well as considerations presented in § 3.2.3.1. furnish relevant proofs and indications on the importance of rational alignment of line shafting (the stiffness of which is much more important than this of the crankshaft) for the correct behaviour of the crankshaft and its bearings (3), (7). The question of particularities of thrust bearing concept and behaviour on correct behaviour of crankshaft and its supports are already well-exposed. An example of such rational alignment of line shafting together with crankshaft pre-deformed in sagging is presented on Figure 38. It concerned the realignment of 14 ships investigated and analyzed previously (see § 2.2.).

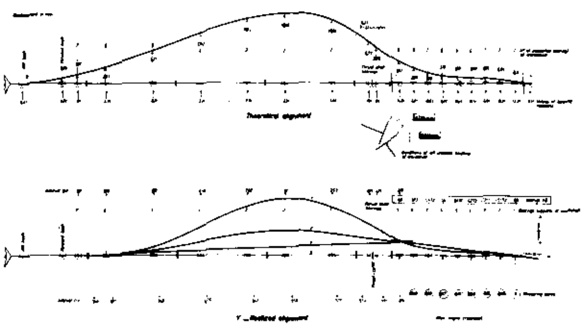

Fig. 38 - Rational realignment of line shafting assembly, the crankshaft being predeformed in sagging.

But besides such countermeasures aiming at coping with detrimental effects of steel-work deformations by acting on external and internal predeformed alignment of assembly of shaftings, the need has appeared to reanalyze the rationality of the engine beam in view of problems encountered (2).

A closer analysis of conventional two stroke engine beam concept from point of view of flexural solicitations is showing its weak points and even inadequacy.

On Figure 39 are presented the Author's reflexions on this subject (3), (12).

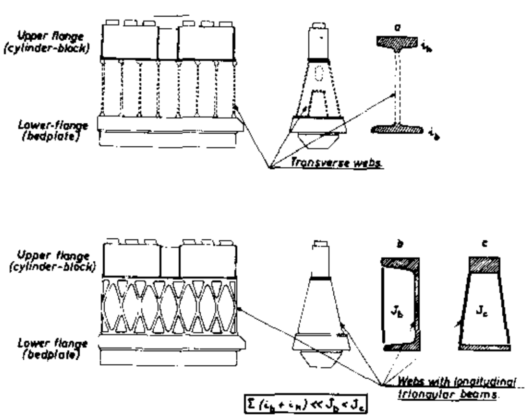

Fig. 39 - Development of concept of the engine as a beam.

In fact an engine is constituted of three main parts : bed-plate, transverse webs supporting upper part being the cylinder block. But from point of view of vertical flexural solicitations such a structure does not present a rigid beam as two lower and upper flanges (being bed-plate and cylinder block) are unable to work together not being connected by elements being able to transmit the shearing forces, hence no beam structure.

This is shown on the upper part of Figure 39 sketch a. This fact may also explain the flexibility of bed-plates (see Figures 15 and 16) and consecutive troubles to main-bearings and crankshafts (see Figures 8, 9 and 14).

Due to some particularities of different types of engines they could be considered as quasi-beams as shown on sketch b) of the same figure. But in order to have the engine working as a beam, it is necessary to conceive the triangular structure between cylinder block and bed-plate, as shown on Figure 39 sketch c).

Such a concept has been adopted by realization of "Sea Horse" type British built Diesel engine. Another also very rational solution of a new concept of engine beam has been realized by M.A.N. in the form of box like structures of assembly of main engine type B in the development of which BUREAU VERITAS has also assured its assistance by executing needed F.E.M. calculation.

On Figure 40 and for comparison sake are presented the new and old concepts of M.A.N. engine beams.

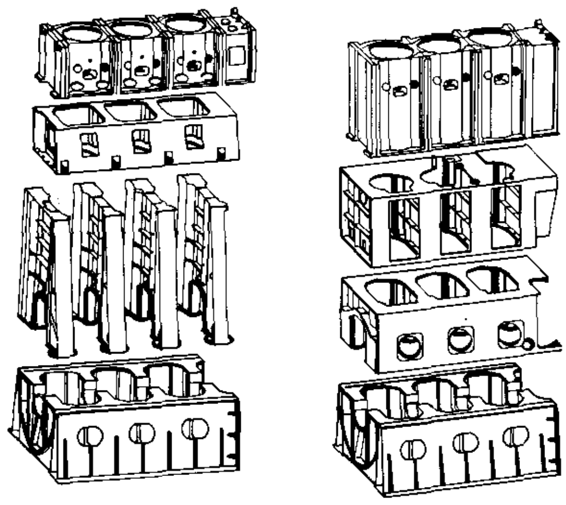

Fig. 40 - Old and new concepts of M.A.N. engine beam.

Such solutions, in conjunction with rational architecture of foundations incorporated into the double bottom, will surely create more favourable conditions for the behaviour of the crankshaft and its bearings. As to the third aspect of assuring compatibility for this type of propulsive plant, it is related to the realisation of a curved internal alignment of the crankshaft. Such studies have been also executed in the past in BUREAU VERITAS when the criteria of practical realization of curved alignment have been based on measurements and correct interpretation of values of crankshaft deflections (8), (12).

The theoretical determination of angular rotations and deformations have been executed by calling for two different methods :

At first, the use of general equations of elasticity, starting from expressions of elastic potential

$$U = \int \left[\frac{N^2}{2ES} + \frac{M_z^2}{2EI_z} + \frac{M_y^2}{2EI_y} + \frac{M_k^2}{2G\tau} + \frac{kT^2}{2GS} \right] ds \qquad (4)$$

where :

- N = normal force
- T = shearing force
- M_z, M_y = bending moments
- M_k = torque
- S = section area
- E = Young modulus of elasticity
- G = shear modulus of elasticity
- I_z, I_y = flexural moments of inertia
- τ = torsional constant
- k = sectional coefficient

Secondly, use of grapho-analytic method of Mohr.

For each surface the theory of plain flexion has been used by writing that :

$$\frac{d^2y}{dx^2} = - \frac{M}{EI} \qquad (5)$$

On Figure 41 are presented the summary of angular deformations and moments in way of journal bearing of two adjacent crank-throws.

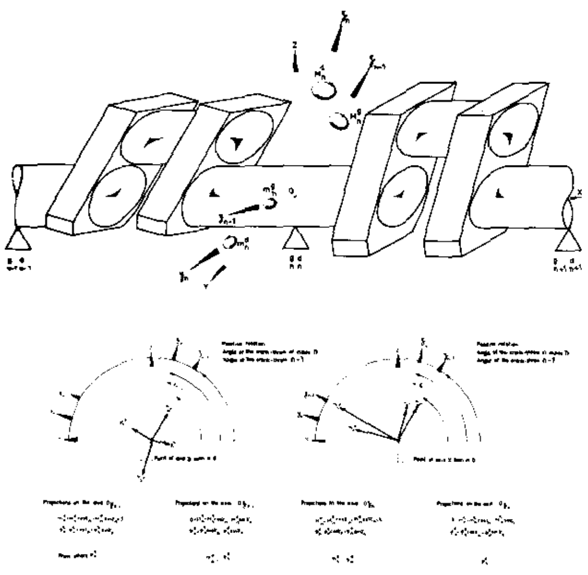

Fig. 41 - Summary of angular deformations and moments in way of a journal bearing of two adjacent crank-throws.

These theoretical calculations, backed by experimental measurements on the turning lath of a new crankshaft, have been also applied to real bulk carrier main engine.

In fact, it has been recalculated, on the basis of measured deflections of a crankshaft predeformed in sagging at 0.50 m, the positions of crankshaft journals in vertical and horizontal directions and on Figure 42 are presented the calculated deformations of the concerned crankshaft. A rather good correlation between calculated and measured values is to be noted.

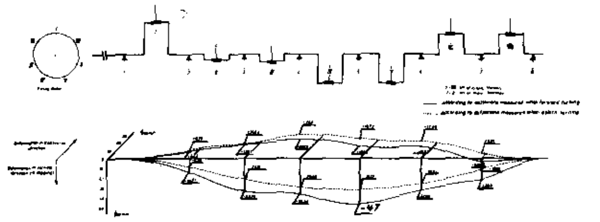

Fig. 42 - Real deformations of curved crankshaft determined on the basis of measured deflection values.

4.3. Rational treatment and concept of marine gears and their foundations

As to propulsive plants in which the torque is transmitted to line shafting through the main gear, for assuring the correct behaviour of the last one, the attention should be drawn on the following :

- correct assessment of all external influences which may be introduced into the main gear by the line shafting,
- creation of favourable conditions in the environment of main gear aiming at diminushing the harmfulness of external influences.

From different examples of investigations analyzed in § 2.3.accompanied by considerations developed in § 3.1. and § 3.2.3.2. some indications in this respect are clearly emerging (5, 9, 10, 12, 15). They will be shortly summarised or completed.

In order to assure the correct behaviour of the last reduction it is indispensable to realize the correct meshing conditions between mobiles. This can be achieved if the desequilibrium of reactions in way of main wheel shaft bearings lies within tolerances indicated by gear Manufacturer.

For this reason a correct rational alignment of line shafting should be executed taking account of all external influences (10) such as hull and double bottom steel-work deformations due to loading and sea-swell conditions. The effects of thermal dilatation of foundations and casing together with foundation deformations due to tilting and rocking of thrust-block should be well assessed.

Besides the harmfulness of steel-work deformations it is absolutely important to adopt the thrust-bearings in which there is no thrust shaft journal bearing (contrary to thrust-bearings of Diesel engines), as it was the case with the first type of investigations into damages of main gear described in § 2.3. On Figure 43 are shown the old and corrected architectures of thrust-bearing where also thrust-pad support have been realized in the form of a spherical tilting crown in order to eliminate the appearance of bending moment on the thrust collar

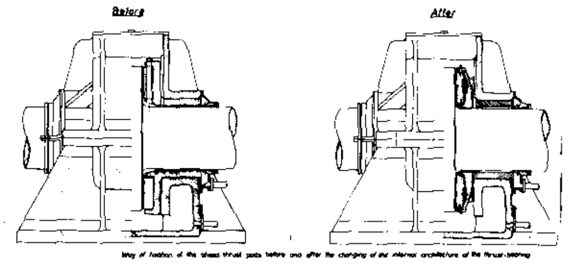

Fig. 43 - Modifications of the internal architecture of the thrust bearing

It must be mentioned that the external shape of this old fashioned thrust-bearing should be considered as non rational, the support points being situated far below the shaft axis ; this concept is favourable to tilting of thrust block and S shape deformations of foundations and double bottom.

Also according to the Author's experience (10), the concept of thrust bearing incorporated into main engine casing, especially for higher out-put, could be judged as non rational In fact it is better to divide the functions allowing the main gear and its casing to serve for transmission of torque only and the thrust bearing for transmission of thrust only.

The modern rational concept of thrust bearing is such that its support points are situated at the level of shaft axis, as shown on Section GH of Figure 44, where also rational concept of its foundations working on shearing and traction as well as the rational concept of deep double bottom creating favourable conditions working for main gearing are shown (5), (10), (12).

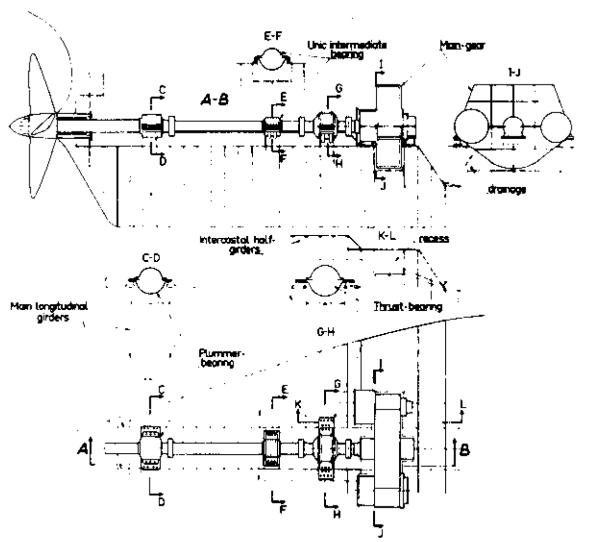

Fig. 44 - Rational foundations of line shafting thrust-bearing and main gear of big propulsive plant.

An analogue solution has been also adopted, to full satisfaction of Shipyard and Shipowner, for the series of the biggest in the world 553,000 DWT tankers type "BATILLUS" (16),(17),(18)

In order to correctly assess the values of all deformations of steel-work needed for rational line shafting alignment it is necessary to proceed with building-up of a correct F.E.M. model of the aft part and the engine room of the ship as shown on Figure 45 being a 250,000 DWT tanker.

With this elasto-dynamic model can be executed needed static and vibratory calculations (9), (10), (12), (14), (16).

On Figure 46 are presented the good degree of correlation which has been stated by the Shipyard in respect to calculations executed by BUREAU VERITAS.

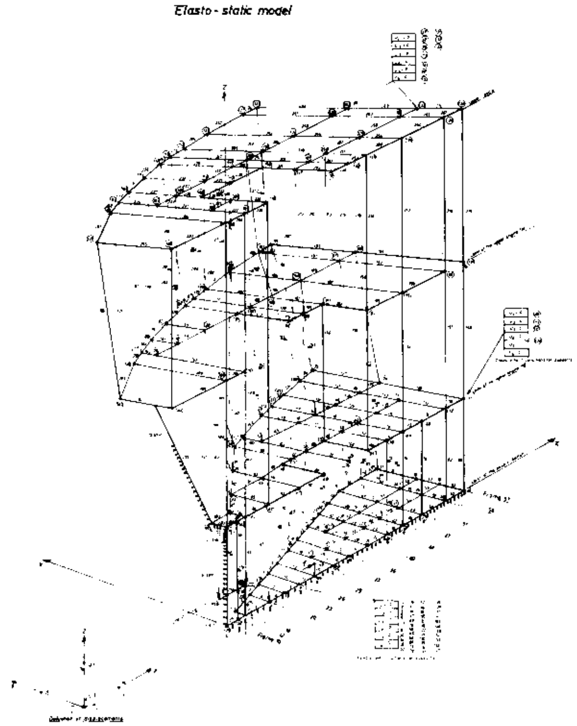

Fig. 45 - Equivalent elastic system of aft part of 250,000 DWT tanker used for deformation calculations

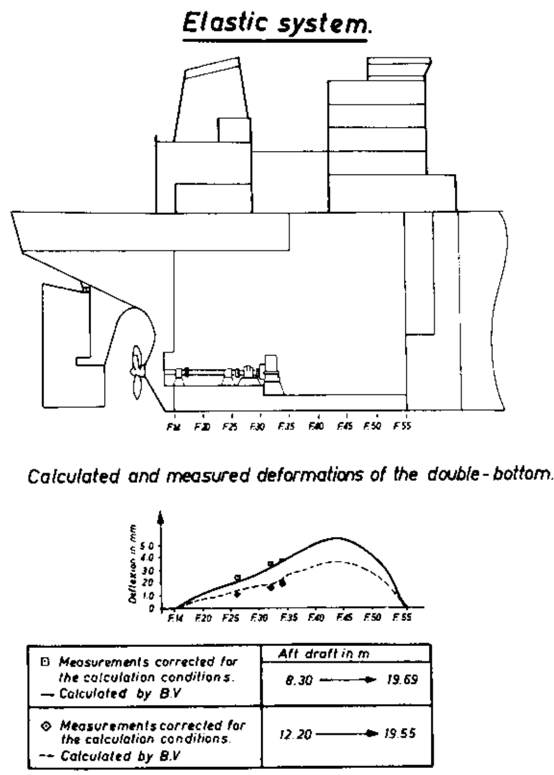

Fig. 46 - Correlation between calculated and measured values of deformations in way of line shafting.

An example of rational alignment calculations (14), (16) related to line shafting of the biggest in the world built Ro-Ro ships equipped with the biggest in the world c.p. propellers of 45,000 HP (and driven by three medium speed engines though one main gear) is shown on Figures 47.

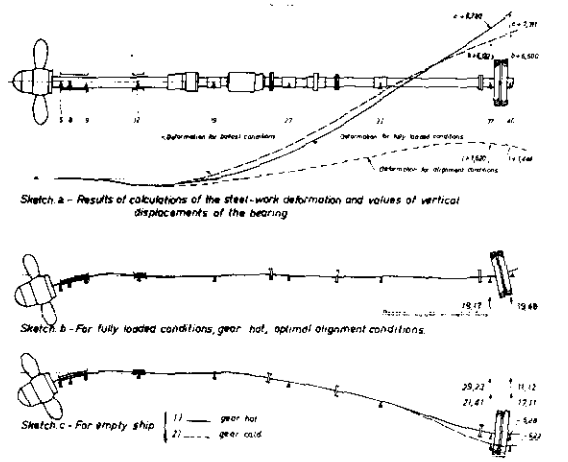

Fig. 47 - Rational alignment calculations taking into account hull deformations of 46,000 BHP c.p. propeller Ro-Ro ship.

There it can be also seen the calculated deflections of the steel-work to loading conditions. At this occasion it is interesting to note that the weight of c.p. propeller, together with tail-shaft and control box, is 170 tons to be supported by two stern tube bushings which have been double sloped in order to diminush the local specific pressures at aft extremity which nevertheless has been at rpm = 0 equal to 120 Kg/cm^2 with corresponding white metal squeezing of 0.035 mm. At normal rpm this pressure has fallen to 30 Kg/cm^2 due to the presence of oil film of calculated thickness of 0.13 mm (see § 3.2.1. and § 4.1.) and the stern gears are behaving well since already more than two years.

5. CONCLUSIONS AND RECOMMENDATIONS

It could be difficult to summarize even shortly all conclusions and recommendations exposed in detail in different chapters and which are the results of the collected experience. Hence only the most salient items will be remembered hereafter.

A. The rapid growth, during the past thirty years, of dimensions of ships and outputs of their propulsion plants, together with the introduction of powerful theoretical means (FEM calculations, speedy computers) allowing the decrease of scantlings, have led to :
- increase of flexibility of steel-work of ships,
- increase of stiffnesses of their line shaftings.

B. These facts together with often severe difficulties appearing in operation of propulsive plants have created an incompatibility between respective flexibilities and stiffnesses.

C. The above mentioned difficulties, described and analyzed in § 2., have been accompanied by the appearance of annoying vibratory phenomena affecting machinery and hull.

D. The powerful modern experimental means (strain gauges and pick-ups, electronic amplifiers, recorders, analysers, telemetry, etc...) have allowed to obtain more insight into the causes of occurred difficulties and have led to the conclusions that in the past not enough attention has been paid to the interaction between machinery and hull.

E. The lessons drawn from the investigations and the experiences gained during trouble shooting activity are leading to the philosophy of necessity of :
- simultaneous treatment of hull and machinery,
- from static and vibratory point of view.

On Figure 48 is presented the chain of integrated calculations needed for realizing this aim.

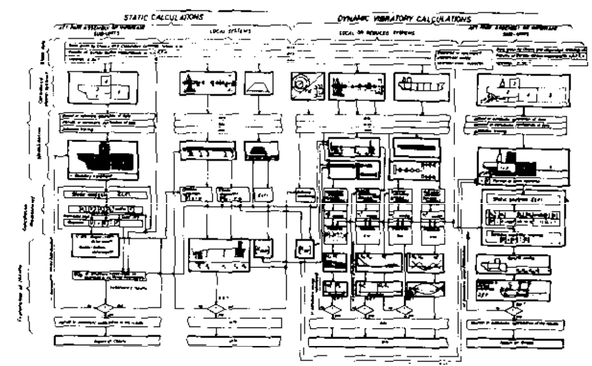

Fig. 48 - Chain of integrated calcultations of static and vibratory phenomena simultaneously of hull steel-work and machinery.

F. The vibrations of forced character, of big ships having mainly affected the aft part and not the hull girder, it has been stated that their appearance was mainly due to the presence of forced vibration resonators, increasing the vibratory level in their neighbourhood. Hence the effective treatment of vibrations occurring on modern ships consisted in their research and detuning.

G. Along the text of this paper it has been presented the detailed analysis (in § 3.) of problems related to flexibiltiy of steel-work (hull girder, double bottom of engine rooms and the influences exerted on them by outside shelling steel-work, pillars, superstructures as well as foundations of thrust blocks) and stiffness of line shaftings as well as consecutive repercussions on Diesel engine crankshafts and bull gear assemblies with respective repercussions on them of thrust block. Particular emphasis has been put on mutual interdependence between static and dynamic phenomena affecting the ship.

H. Practical and theoretical means and ways to assure the compatibility between machinery stiffness and hull flexibility havebeen also presented in § 4. by proposing rational treatments and concepts of line shaftings driven by Diesel engines or marine gears together with analysis of particularities related to these two types of propulsion.

6. ACKNOWLEDGEMENTS

The Author wishes to express his thankfulness for the Management of all Shipyards, Shipowners and Machinery Manufacturers for whom he has had the pleasure and honour to work during the past 20 years, for the confidence and help granted to him during his work on their behalf.

Without this attitude, he could not achieve the results reported in this Paper.

He wishes to thank also the Board of Directors of BUREAU VERITAS for permission granted to him to publish the results of his investigations carried out during his activity in Applied Study and Research Department.

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