LECH MURAWSKI, D.Sc.,M.E.
Ship Design and Research Centre, Gdańsk
Ship Structures Division

Analysis of measurement methods and influence of propulsion plant working parameters on ship shafting alignment

Part II

SUMMARY

In the paper three different experimental investigation methods of propulsion shaft alignment are discussed. A new original method of direct measurement of the bearing reactions on the foundation pads is presented. Possible application of the method was analyzed by means of computations and experimental investigations carried out on a 2000 TEU containership at quay and during sea trials. Discussion of results of the analysis and tentative recommendations are included.

Part I of the paper was published in the Polish Maritime Research, No 2 (12), June 1997.

MEASUREMENTS DURING SEA TRIALS

In the shafting alignment technology presently applied by the shipyard, no control or other works are provided to be executed during sea trials. The only warning is excessive bearing temperature signal; there is no warning against the unsufficent loading on the intermediate bearing . It was the reason to carry out extensometric investigations of the changes of loading on the intermediate bearing foundation pads. Two main measurement variants were assumed: static measurements when turning the shaftline by a turning gear, and dynamic measurements carried out in function of the engine rotational speed. Denotation of the dynamometers was maintained the same as that assumed in Fig.4 (see part I of the paper).

Measuring channels were set to zero just before the ship's departure for sea trials. The ship's ballast draught was as follows: fore 6.00 m, aft 7.47 m. The main engine was in the cold condition, no hydrodynamic loads were applied. The state was close to that during the ship trials at quay.

The strain-gauge bridges were set to zero at the shaftline rotation angle equal to 90° (the location of the first crank in respect to the top dead centre). Zeroing the measuring system led to the investigation of changes (increments) of the intermediate bearing reactions in respect to the results of their measurements at quay. It was necessary to account for the „screw - orifice" cooperation, because of the loading exerted on the pads by the foundation bolts in tension. The total load increment of the bolt - pad system can be calculated from the following expression:

$$\Delta R = \Delta R_K \left(1 + \frac{k_s}{k_k} \right)$$

where:

- ΔR - total load change
- ΔR_K - pad load increment
- k_s - foundation bolt stiffness
- k_k - pad stiffness

In the case of the system in question the multiplier value was 1.1905. Measured load values were to be multiplied by the multiplier value in order to obtain the total load value.

When the measuring system was set to zero the first group of measurements were carried out during which the load changes on the dynamometers versus the shaft angle of rotation were investigated. Results of the measurements, converted into real total load changes, are presented in Tab.1.

Tab.1. Loads on the dynamometers before the ship's sea trials

Shaft rotation angle [°]	D1	D2	D3	D4	Σ
	[kN]				
180	0.23	-3.01	0.15	-7.39	-10.02
270	2.96	-1.29	-0.46	-11.33	-10.12
0	2.74	0.86	0	-3.12	0.48
90	0	0	0	0	0
Load change per rotation	2.96	3.87	0.61	11.33	

The next group of measurements was performed after changing the ship's ballast condition. The actual fore draught was 5.50 m, and stern draught 8.80 m. During the measurements the engine was also in the cold condition and no hydrodynamic load was present. The measurements were carried out during overhaul works of the engine, therefore rotating the shaftline was not possible. Results of the measurements are presented in Tab.2. They are related to the state just before sea trials (setting to zero was not performed again). The numbers given in Tab.2 are values of the absolute change of the load increment which resulted from the difference between two ship ballast conditions. The mark „-" means unloading the pads, i.e. decreasing the total reaction.

Tab.2. Loads on the dynamometers after changing the ship ballast condition, the main engine in cold condition

Shaft rotation angle [°]	D1	D2	D3	D4	Σ
	[kN]				
270	-16.43	-7.31	-0.46	-83.09	-107.29
Abs. load change	-19.39	-6.02	0	-94.42	-119.83

Last static measurements of the bearing reactions on the pads were carried out in the same ballast condition as that during the preceding measurements. An important difference of the system's state was the engine in hot condition. The investigations were performed just after running the engine for many hours. Results of the measurements are presented in Tab.3. In the table the change of load increment in relation to the cold engine state of the system is included (Tab.2), as well as the change of load resulting from the angular position of the crankshaft.

Tab.3. Loads on the dynamometers after changing the ship ballast condition, the main engine in hot condition

Shaft rotation angle [°]	D1	D2	D3	D4	Σ
	[kN]				
180	-11.40	-4.30	-4.36	-130.22	-150.28
270	-7.99	-3.87	-4.82	-128.74	-145.42
0	-7.54	-2.58	-2.80	-120.36	-133.28
90	-13.69	-3.01	-4.20	-130.38	-151.28
Load change per rotation	6.15	1.72	2.02	10.02	18.00
Abs. load change at 270°	8.44	3.44	-4.36	-23.00	-15.48

The static investigations performed during sea trials confirmed that the dynamometer D3 was not entirely loaded - the intermediate bearing was supported on three pads only. The port side of it was more loaded therefore the pad 4 transferred the prevailing part of the load. An important asymmetry of loading on the port-side and starboard pads of the bearing was observed. Therefore any further experimental - computational analyses should be carried out in respect of two planes: vertical and horizontal.

The influence of the ship ballast condition on shafting alignment, not taken into account so far, should not be neglected. A substantial increase of the ship loading resulted in the drop of the intermediate bearing reaction by 70.5 % of its initial value (170 kN). It can cause a dangerous unloading of the intermediate bearing in result of which, for instance, the stern bearing can be overloaded. Therefore in any further computational analyses not only the overall ship hull deformations but also the local stiffness of the bearing foundation on the ship double bottom should be accounted for.

The influence of the engine state (cold or hot condition) on the magnitude of bearing reactions was confirmed. A drop of the intermediate bearing reaction after heating the engine should reach 13 % according to the shipyard's computation analysis, but it was only about 9 % as the measurements revealed. The change is rather low especially when 6 % change of the bearing load due to changing the rotation angle of the crankshaft is taken into account.

The measurements of the dynamic changes of the loading on the intermediate bearing pads (bearing reaction) were carried out in function of the engine speed. They were performed in parallel with the torsional vibration measurements. Both measurement results were stored in the same computer memory therefore the mutual phase shifting of the torsional and transverse vibrations of the intermediate shaft and of the variable bearing reactions could be analyzed.

The measurement results were processed by means of FFT harmonic analysis procedure and transformed to the real changes of the load on the foundation pads. Measurement results for the D4 dynamometer were excluded because of an incorrect course of that part of measurements and it was not possible to repeat them. Fig.7 to 9 comprise: the mean changes of the load on the particular dynamometers, peak-to-peak load amplitudes and their fifth harmonic components in function of the engine speed.

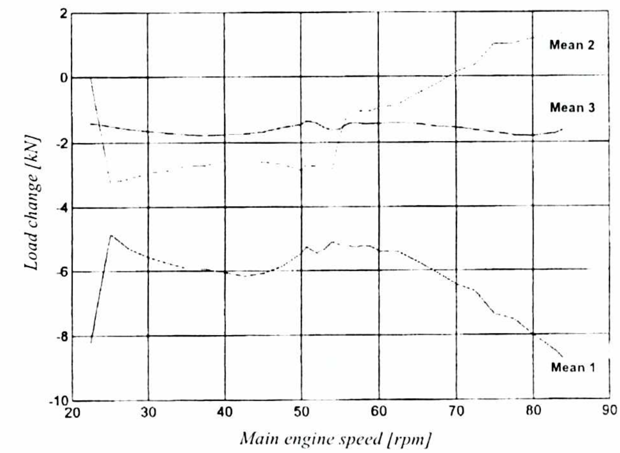


Fig.7. The mean changes of the variable load on the particular dynamometers (D1, D2, D3 dynamometer mean values respectively)

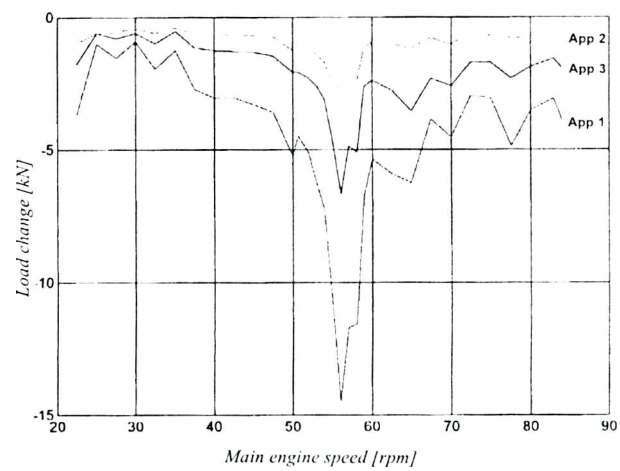


Fig.8. The peak-to-peak amplitudes of the variable load on the dynamometers (D1, D2, D3 dynamometer amplitude values respectively)

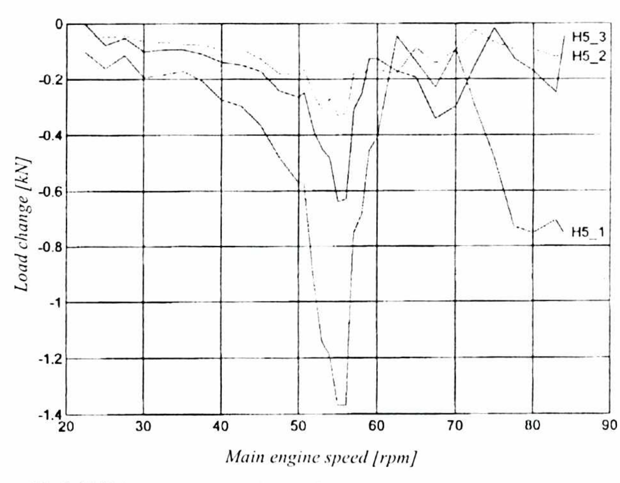


Fig.9. Fifth harmonic components of the variable load on the dynamometers (D1, D2, D3 dynamometer component values respectively)

Changes of the load on the foundation pads during crankshaft rotation are only slightly different from those during „static" turning. Propeller harmonic components of the variable load on the dynamometers are negligible (Fig.9). It can be stated on the basis of the experimental investigations that the influence of the propeller-induced hydrodynamic forces on the alignment of the power transmission system in question is small. However, verifying the calculation assumptions with the use of the presented measurement results was not possible because total values of the reaction changes were lacking (due to the failed measurement at D4 dynamometer). Higher harmonic components revealed zero response level.

INTERMEDIATE SHAFT BENDING STRESS MEASUREMENT

The measurements were performed by means of the complete strain-gauge bridges glued on the intermediate shaft of the ship power transmission system. The measuring systems were adjusted for measuring the shaft bending stresses only. Telemetric systems were utilized for transmitting signals from the extensometric bridges to a digital recorder. Therefore carrying out the measurements while running the main engine was possible. Two measuring systems were installed: the first one 2160 mm distant (marked: ZG_SG) and the second one (marked: ZG_L) 5580 mm distant from the intermediate shaft - engine crankshaft coupling. The first measurement point was located close to the main engine and the second point close to the intermediate bearing (from the side of the engine). The strain gauges were glued in such a way that they maintained vertical position when the first crank was at the top dead centre. The experimental investigations were carried out in two variants: static and dynamic measurements in function of the engine speed.

The measured shaft bending stresses were related to the calculated static stresses. They were determined for an assumed shaft deflection line and loading (especially that propeller induced), the same as those used in the shipyard's report. In Tab.4 the calculated bending stress levels at the selected measurement points are presented.

Tab.4. The calculated bending stresses at the selected measurement points

Loading variant	ZG_SG	ZG_L
	[MPa]	
Straight shaftline, no hydrodynamic forces	-0.16	0.35
"Cold" engine, no hydrodynamic forces	-5.41	-1.48
"Hot" engine, no hydrodynamic forces	-4.44	-2.53
"Hot" engine, hydrodynamic forces: accounted for	-4.43	-2.39

The extensometric measurements of bending stresses were carried out in parallel with the direct measurements of the bearing reactions (the extensometric measurements of the loads on the bearing pads) which were performed before the ship's departure for sea trials. The measuring channels were set to zero at 90° angle of rotation of the first crank in respect of the top dead centre (the strain gauges were placed at the neutral bending axis). Measurements at various turning positions of the shaftline were then carried out. Measurement results (converted to bending stresses) are presented in Tab.5. Just after the measurements the channels were set to zero again.

Tab.5. The measured bending stresses during tests at quay

Shaft angle of rotation [°]	ZG_SG	ZG_L
	[MPa]	
180	6.25	0.88
270	0.62	0.21
0	-5.36	-0.67
90	0.41	-0.05
Abs. stress change	11.61	1.52

The next group of measurements were performed after changing the ship ballast condition (as in the case of the measurements of the loads on the bearing pads). During the measurements the main engine was in the cold condition and no hydrodynamic forces were present. Measurement results are given in Tab.6. The table contains the results related to the state before the ship's sea trials (Tab.5)

Tab.6. The measured bending stresses after changing the ship's ballast condition, cold engine

Shaft angle of rotation [°]	ZG_SG	ZG_L
	[MPa]	
270	1.38	0.06

The last, static measurements of the intermediate shaft bending stresses were carried out at the same ship's ballast condition as applied during the preceding measurements (Tab.6). An important change of the system's state was the hot condition of the main engine. Measurement results are given in Tab.7.

Tab.7. The measured bending stresses after changing the ship's ballast condition, hot engine

Shaft angle of rotation [°]	ZG_SG	ZG_L
	[MPa]	
180	6.45	3.09
270	1.06	0.36
0	-6.66	-3.61
90	-1.99	-0.26
Abs. stress change	13.11	6.70

The recorded changes of the intermediate shaft bending stresses less distinctly show the influence of different working conditions of the propulsion system than the changes of loading on the foundation pads. Nevertheless a greater part of the conclusions was confirmed. The stress increase measured by ZG_SG system, given in Tab.6, means that the shaftline deflection increases due to the influence of substantial ship hull deformations. The decaying bending stresses in the vicinity of the intermediate bearing (given in col. ZG_L, Tab.6) firmly confirms the decreasing level of the intermediate bearing reaction. The conclusion about the necessity of accounting for the actual ship's ballast condition in analyzing the shafting alignment was thus confirmed. The relatively small stress level difference for two conditions of the main engine (compare Tab.6 and 7) confirms the previous conclusion, based on computational and experimental analyses, that the engine condition slightly influences correctness of the shafting alignment. Among the measurement results given in Tab.7 no zero stresses can be found at the theoretical neutral bending axis (vertical plane). It evidences a substantial deformation of the intermediate shaft in the horizontal plane, it means that any computational analyses should be carried out for the two planes. These conclusions make qualitative assessment of the performance of the power transmission system possible. A quantitative assessment is possible only in a indirect way by applying numerical simulation of the system's behaviour. The calculated bending stress levels (Tab.4) satisfactorily comply with those measured (given in Tab.5).

Measurements of the dynamic bending stresses of the intermediate shaft were carried out in function of the engine speed. They were performed in parallel with the torsional vibration measurements. Results of both were stored in the same computer memory. The recorded measurement courses were then analyzed in frequency domain. In Fig.10 ÷ 12 measurement results of the torsional and transverse vibrations at all measurement points are presented in function of frequency (engine speed).

In Tab.8. the most important measurement results (stresses) of the transverse vibrations are collected. Changes in stress values due to the applied minimum and maximum engine speed are presented (a distinct stress increase resulting from torsional vibration resonance was omitted).

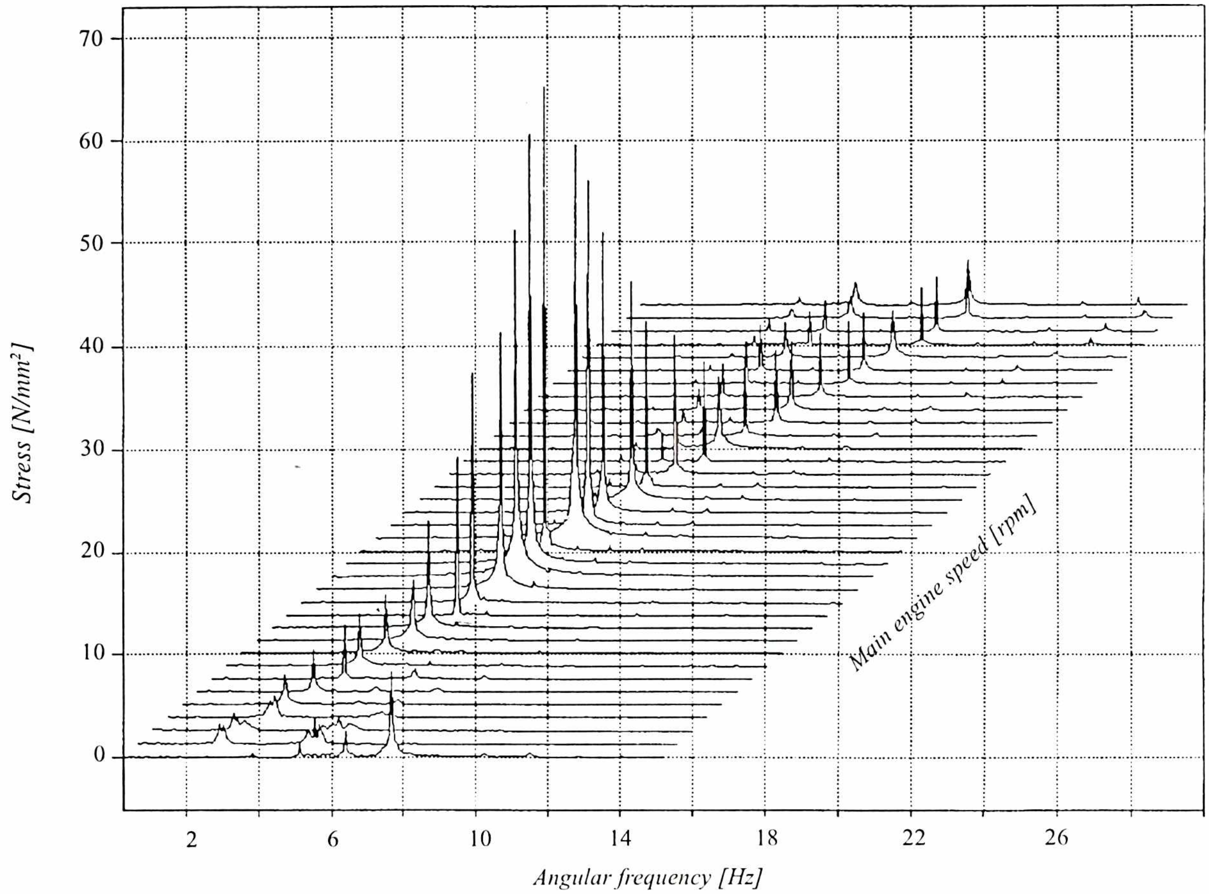


Fig.10. Changes in torsional vibration stresses of the shaftline versus engine speed

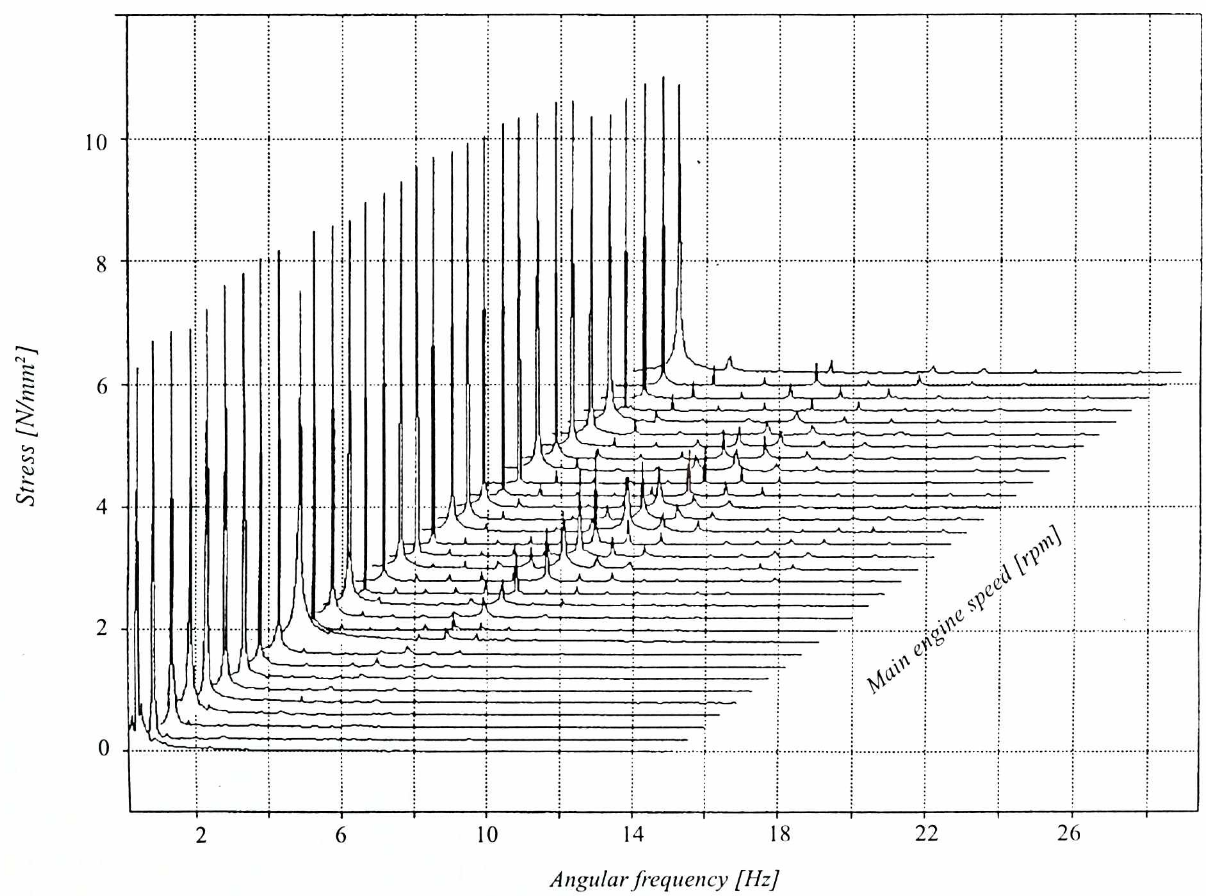


Fig.11. Changes in transverse vibration stresses of the shaftline close to the main engine (ZG_SG) versus engine speed

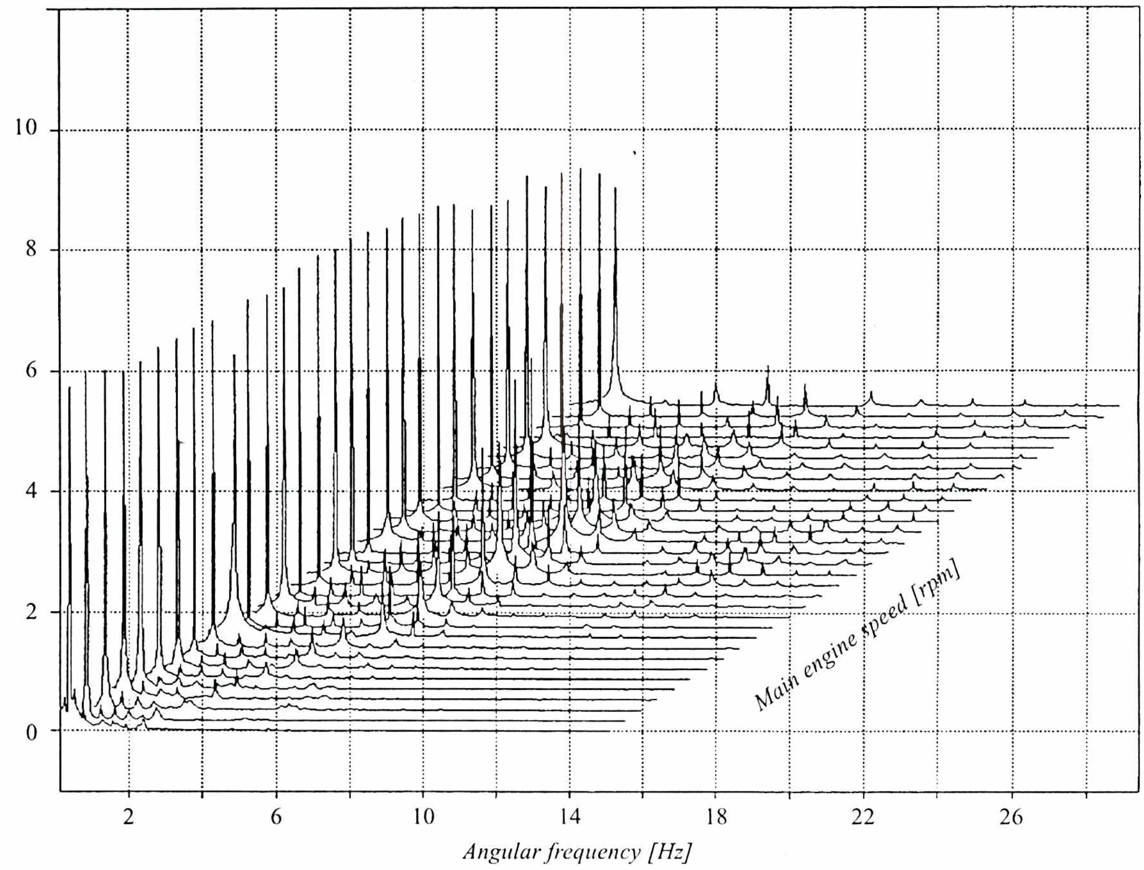


Fig.12. Changes in transverse vibration stresses of the shaftline close to the intermediate bearing (ZG_L) versus engine speed

Tab.8. Changes in bending stresses due to the applied minimum and maximum engine speed

	ZG_SG	ZG_L
	25 => 83 rpm [MPa]	
Total Σ	6.33 => 4.93	6.58 => 5.37
1 H	5.90 => 4.03	6.55 => 5.11
5 H	0.12 => 0.65	0.02 => 0.09
10 H	0.01 => 0.15	0.02 => 0.05
15 H	0.02 => 0.05	0.01 => 0.01

The bending stresses vary to a small extent only in function of the engine speed. Therefore constant components of the propeller-induced hydrodynamic forces should be accounted for when calculating the shafting alignment. However the influence of the blade harmonic components is very low.

CONCLUSIONS

1. Three different experimental investigation methods of ship shafting alignment were discussed. A new original measurement method of the bearing reactions on the foundation pads was presented. Possible applications of the method was analyzed on the basis of computations and experimental investigations carried out on a 2000 TEU containership at quay and during sea trials.

2. As far as the new method is concerned, highly nonlinear stiffness characteristics of the measuring dynamometers were observed. It was caused by a various extent of spherical contact surfaces of the particular dynamometers under loading. On the basis of the performed analysis it can be stated that the accuracy of workmanship of the spherical contact surfaces was not sufficient.

The histeresis discrepancy of the tested dynamometers was rather small. However, dynamometer characteristics should in any case be determined by using a testing machine, otherwise investigation results would be unreliable.

3. The experimental investigations revealed that the shafting alignment procedure applied to the tested ship was in compliance with the shipyard's usual technological method. However it can be stated on the basis of the extensometric measurements that the reaction distribution over particular foundation pads was very non-uniform, e.g. the intermediate bearing was supported by three pads only.

4. The extensometric measurements revealed also an asymetry of loading on the intermediate bearing foundation in the horizontal plane. It means that any computation - measurement analyses should be performed for two planes: horizontal and vertical.

5. The influence of ship ballast condition on the shafting alignment, not taken into account so far, seems not to be negligible. A substantially increased ship loading resulted in lowering the intermediate bearing reaction to an important extent. It can cause a dangerous situation of unloading the intermediate bearing and overloading the stern tube bearing. It means that any computation analyses should be carried out with accounting for not only the overall ship hull deformations, but also local stiffness of bearing seating on the double bottom.

6. The influence of an engine thermal condition (hot or cold) on the bearing reaction level was confirmed. Such changes are not substantial: of the similar order as those due to changing the crankshaft angle of rotation.

7. The changes of dynamic loading on the foundation pads (when operating the propulsion system) are only slightly different from those appearing at the „static" change of the shaftline angle of rotation. Propeller harmonic components of the variable loading on the dynamometers are also negligibile. Therefore it can be stated that the influence of the blade components of the propeller-induced hydrodynamic forces on the alignment of the power transmission system in question is small. However accounting for the constant components of hydrodynamic forces is reasonable.

8. The observed changes of the intermediate shaft transverse stresses revealed the influences of the different propulsion system conditions less distinctly than the changes of loading on the foundation pads. Nevertheless majority of the above given conclusions (p.4,5,6,7) was confirmed by the results.

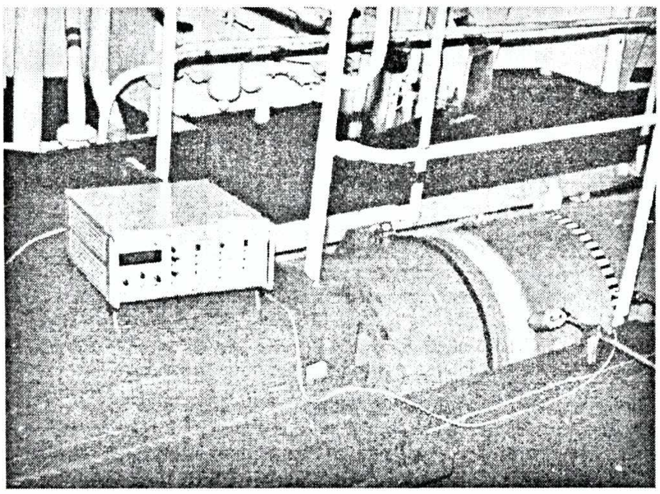
9. The shafting alignment procedure, still used by the shipyard, consisting in generating the set bearing reactions is very good because of its simplicity and reliability. However it does not make possible to control: tensioning the foundation bolts, reaction distribution on the particular pads, as well as the final shafting alignment state after completing the whole procedure. Therefore it should be supplemented by the recommendation on tightening the bolts with the use of a torque spanner. It would be also useful to control distribution uniformity of bearing reactions by means of the extensometric measurements of loading on the foundation pads also after completing the alignment procedure, i.e. during ship sea trials. The transverse vibration measurements of the intermediate shaft could supplement the above specified means of controlling the shafting alignment.

The above mentioned conclusions from the measurement investigations should be taken very cautiosly because of their preparatory character. In particular they should not be considered as a basis for changing the shafting alignment procedure intended for application on other ships.

BIBLIOGRAPHY

1. Gawlik B., Kralewski S.: „Calculation of shafting alignment". Gdynia, 1996
2. Pressicaud J.P.: „Correlation between theory and reality in alignment of line shafting". Bureau Veritas, III. Paris, 1986
3. Ville R.: „Actual dynamic behaviour and calculated approach to stern tube white metal bush". Bureau Veritas, III. Paris, 1986
4. Voley G.C.: „Wzajemne oddziaływanie układu napędowego i kadłuba oraz ich swobodne i wymuszone drgania". Zeszyty Problemowe CTO. Gdańsk, 1982
5. Voley G.C.: „Forced vibrations of the hull and rational alignment of the propeller shaft". Bureau Veritas, I. Paris, 1976
6. Voley G.C., Ville R.: „Actual tail shaft behaviour on oil film taking account of propeller forces and moments". Bureau Veritas, II. Paris, 1981
7. Wieczorek J., Dunia Z.: „Instrukcja montażu linii wałów na statkach". Gdynia, 1986
8. Wojdała H.: „Analiza obciążeń statycznych łożysk i momentów gnących oraz osiowania linii wałów". Szczecin, 1985
9. „Shaft alignment for direct coupled low-speed diesel propulsion plants". MAN B&W Diesel A/S. Copenhagen, 1995

Appraised by Marek Krawczuk, Assoc.Prof.,D.Sc.



The equipment for measuring the bending stresses in the intermediate propulsion shaft

Current reports

GDYNIA MARITIME ACADEMY
FACULTY OF MARINE ENGINEERING

Loading operations simulator for the liquefied petroleum gas tankers

Dramatic development of the liquefied gas cargo sea trade and serious hazards connected with it cause that the crews of gas tankers are faced with special requirements dealing with careful handling of such cargo and understanding the operation processes. For that reason the idea of building a stand for the simulation of the loading and transporting processes of the liquefied petroleum gas (LPG) with the use of real gases emerged at the Department of Marine Power Plants, Gdynia Maritime Academy, which educates the officers - operators of the ship power plants and relevant mechanical equipment.

All operations at the stand designed and manufactured by the Academy's laboratory, are realized between two tanks :one which simulates a land-based pressurized or semi-pressurized tank and the other which simulates a similar ship tank.

Controlling such equipment as : transport pumps, gas liquefying aggregates, electromagnetic valves, is activated with the use of the central console which is additionally equipped with control-measurement panels and alarm signalling devices. Collecting the characteristic values, setting the alarm tresholds, recording graphically the measured values is realized by means of the UNITEST, special computer software.

The stand makes it possible to simulate the following cargo handling operations:

- Preliminary preparation of the land-based tank (pressurized or semi-pressurized one) by refrigerating or not refrigerating the gas contained in it, with the use of the gas liquefying aggregate.
- Preliminary refrigeration of the ship tank from the ambient temperature to the temperature in the land-based tank with the use of the gas liquefying aggregate to reduce pressure and temperature of the vapour and liquid in the tank.
- Loading the cargo to the ship tank by means of the transport pump with draining simultaneously the cargo vapour back to the land-based tank. About 800 kg of the liquefied gas takes part in the operation.
- Loading the cargo to the ship tank without draining the cargo vapour back to the land-based tank, which requires applying additionally the liquefying aggregate to keep the pressure inside the ship tank constant.
- Liquefying again (by means of the liquefying aggregate) the cargo vapour produced due to the heat transmission through insulation into the tank.
- Unloading the liquefied gas from the ship tank to the land-based tank:
  - unloading (by means of the transport pump)with draining simultaneously the cargo vapour from the land tank back to the ship tank
  - unloading (by means of the transport pump)without draining back the cargo vapour from the land tank, which requires transferring a part of the cargo through an evaporator to the tank to avoid dropping the pressure in the ship tank
  - unloading with heating simultaneously the pumped liquefied gas (if the ship tank temperature is lower), in a heater to avoid generating the thermal stresses.
- Emptying totally the ship tank out the liquefied gas (after pumping it out by the transport pump) by means of the liquefying aggregate.

Ryszard Dziubek, M.Sc., M.E., is the designer and builder of the described stand.