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THE INFLUENCE OF SHAFT LINE ALIGNMENT ACCURACY ON THE OPERATIONAL RELIABILITY OF MARINE PROPULSION SYSTEMS

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Abstract

The paper presents a method of identification parameters of shaft line alignment and its influence on operational reliability of marine propulsion system. The discussion about shaft line parameters (bearings' reactions, bending moment and shear forces acting on crankshaft or gear box, and stresses distribution in the shaft line) which has an influence on marine propulsion system reliability was presented. Proper shaft line alignment is often a problem for repair shipyards, for aged ships without sufficient documentation. Some data can be draw by measurements; e.g. shaft line dimensions or intermediate bearings' reactions. Other data (stern tube bearing load, real axis of shaft line) have to be determined on the base of calculations. Author proposed combined experimental-analytical method for identified and optimization (correction) some existing parameters and checking power transmission system's foundation. Specialised software (based on Finite Element Method) has been developed for shaft line alignment calculations. Main novelties of the software are elastic supports (model of bearings – boundary conditions) of the shaft line, continuous support as a model of stern tube bearing and influence coefficients calculations. An example analysis with discussion has been performed for cargo ships with medium-speed main engine. Multi-variant computations supported by measurements of the ships' shaft line have been carried out. Changes in shaft line alignment have been proposed in order to increase reliability of propulsion system.

Keywords: *shaft line alignment, marine propulsion systems, propulsion system's bearings damages, operating parameters of the power transmission system*

1. Introduction

Slow-speed main engine connected directly by shaft line (intermediate shafts and propeller shaft) with propeller is typical for merchant ships [5]. In that propulsion system, there are no gears or flexible couplings. Scheme of typical marine propulsion system is presented in Fig. 1. Efficiency is a main reason for so simple propulsion system. Usually main engine is powerful – above 20000 kW.

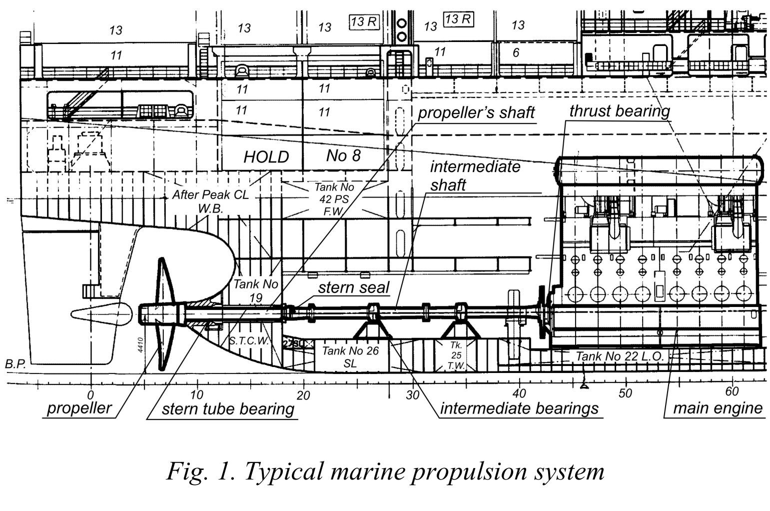

Fig. 1. Typical marine propulsion system

Power transmission system (crankshaft plus shaft line) is loaded by strongly unsymmetrical perpendicular forces. Especially stern tube bearing is loaded by very heavy propeller from one side. Shaft line's rotational speed is very low (60-180 rpm), also owing to efficiency. Therefore, stern tube bearing has to be relatively long. Its ratio length to pin diameter is about equal to 2. Unsymmetrical loads and high L/D ratio is a source of strongly variable eccentricity along bearing length [4]. It is one of the main reasons for the necessity of shafting alignment. During shaft line, alignment the propulsion system's bearings (engine main bearings, intermediate bearings, stern tube bearings) are relatively vertically displaced [5] – shaft line axis cannot be straight line. An example of shaft line alignment of medium size container ship (2000 TEU) is presented in Fig. 2. Shaft line alignment analysis should be performed by multi-variant calculations [3], for different operating parameters of the propulsion system and ship exploitation conditions (e.g. ballast load).

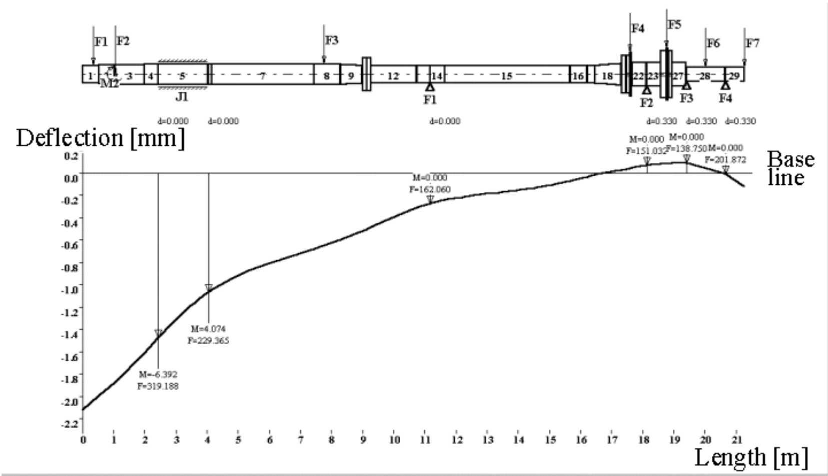

Fig. 2. Shaft line alignment for the container ship

Shaft line alignment is performed and checked (by measurements) usually only during shipbuilding process. It is not monitored during ship exploitation. Shaft lines' improper operational parameters can be checked only indirectly, e.g. by bearings oil film temperature. Shaft line alignment can be dangerously changed under the influence of excessive operational loads, random events (ship grounding), repairing process of propulsion system or ship hull in the engine room area. Some of the researchers [1, 2, 8] proposed nonstandard methods for shaft line alignment diagnostic, but those methods they have not been introduced to seagoing. Incorrect shaft line alignment can be a cause of the following unpropitious events [5, 7]:

- overloading of some bearings of propulsion system,
- stern tube bearing can have local null backlash (especially in the aft part),
- crankshaft can be overload by bending moment and shear force coming from shaft line,
- shaft line bending stresses can be excessive.

2. Shaft line alignment parameters affecting on operational reliability of propulsion system

Several targets should be fulfilled during the shaft line alignment [5, 10]. All targets have to be achieved in all exploitation conditions of the marine propulsion system. The fulfilment of any parameter can cause serious malfunction of propulsion system.

Firstly, appropriate loadings of the shaft line and crankshaft bearings are crucial target of the shaft line alignment. Reactions in the bearings do not have to be too high or too small in all the propulsion's service conditions. In the case of low static bearings' reactions, the possible influence on lateral vibrations (dynamic reactions) should be considered. If the static reaction is similar to the dynamic one, the loading direction might be changeable. Impact loading (hammering) might be the cause of a quick deterioration of the bearing. The Sommerfeld number (see equation (1)) is the best parameter (according to the author experience) to characterize the bearings load. After assuming that Sommerfeld numbers should be equal in all the power transmission systems bearings (as well as all the revolutions and viscosity being the same in all the bearings), equation (2) describes the desirable ratio of the different bearings reactions. A low bearings loading is dangerous especially for the intermediate bearings (ship hull deformation may be the cause of decreasing relatively small reactions). Therefore, it is reasonable that the Sommerfeld number of the intermediate bearings is sometimes 30-50% greater than other bearings.

$$S = \frac{R}{\eta \cdot U} \cdot \left(\frac{d}{c} \right)^2 , \tag{1}$$

where:

S – Sommerfeld number,
R – bearing loading unitary force,
η – lubricating oil absolute viscosity,
U – peripheral speed,
d – diameter of shaft journal,
c – bearing slackness.

$$\frac{R_1}{R_2} = \frac{L_1}{L_2} \cdot \frac{d_2}{d_1} \cdot \left(\frac{c_1}{c_2} \right)^2 \cdot \frac{S_1}{S_2} , \tag{2}$$

where L – bearing length.

Secondly, the loading distribution of the stern tube bearing should be taken into consideration. The stern tube bearing is the heaviest loaded bearing as it is relatively long (L/D ≅ 2). What is more, loading is strongly asymmetrical (by the propeller's forces) so part seizure of the bearings is a danger. Therefore, the reactions distribution should be checked, for instance by comparing reactions on both the bearing edges (both reactions should be positive even with the dynamic component). Checking the relative deflection between the line of the journals and tube axis is an alternative method. In this method, deformation of the shaft line, as well as the ship hull has to be determined.

Thirdly, the crankshaft's loadings by shear force and the bending moment, coming from the shaft line, must be appropriate. All the world producers of marine engines define the allowable values of the crankshaft's loading. An example of allowable forces and moments is presented in Fig. 3. The crankshaft's shear force and bending moment have to be included in the allowable area in all the propulsion's service conditions. Recently this requirement has changed to another: reactions, coming from shaft line loading, in the engine main bearings are given. In this case, the model of the crankshaft must be more detailed. Additionally, propeller thrust includes the moment inducted by the eccentricity of the thrust bearing. The thermal rise of the engine body (main bearing) should be also taken into account together with thermal deformation of ship hull and hull deformation under the influence of sea wave and ship loading conditions.

Finally, the bending stresses of the shaft line in the given final alignment should be checked. Usually (for most typical marine power transmissions systems) these quasi-static stresses are not very high for all service conditions. Relatively large shaft line diameters are determined by the torsional vibration requirements. On the other hand, limits of the bending stresses should be relatively low due to the expected high level of torsional vibration stresses.

The most popular method used for shaft line alignment calculations is the Finite Element Method. The shaft line is modelled by linear beam elements. According to world producers of marine engines, the model of a crankshaft is simplified there are few inline beams (in the model there is no geometry of the cranks). Thermal expansion of the main engine body is modelled as a vertical, parallel movement of the main bearings. The value of this movement is specified by the engine's manufacturers. Calculations have to be done for all the typical ships propulsion systems service conditions. A power transmission system working in nominal conditions (with hydrodynamic forces in a nominal propulsion speed and a "hot" main engine) is the most important variant. Calculations must also be prepared for a disconnected power transmission system (during instalment). A shift of the shafts necks (SAG and GAP) has to be defined.

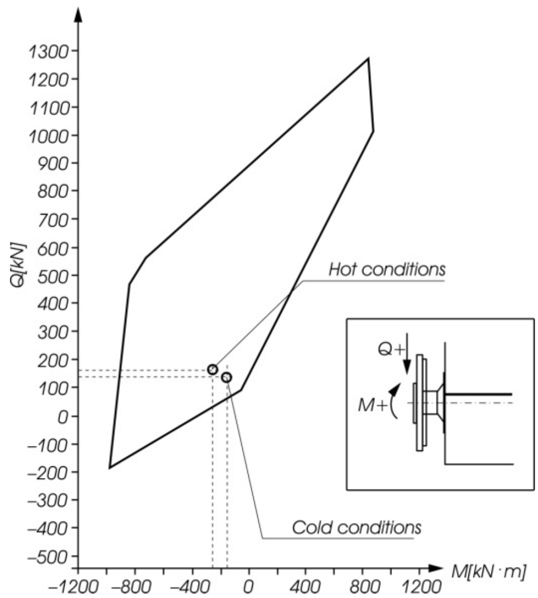

Fig. 3. An example of allowable area of crankshaft loading

As was already mentioned, beam model of the power transmission system is isolated from the ship hull. Therefore, determining the correctness of the boundary conditions is one of the most important, difficult and debated, during the marine power transmission systems static and dynamic calculations. Shaft line alignment and lateral vibration analysis are especially sensitive to proper boundary conditions' determination. Ship hull deformations, under different load conditions and regular sea waves, should also be analysed.

The author developed specialized software dedicated for optimization analyses of shaft line alignment [5, 6]. The software is based on finite element method and was written in Borland Builder C++ (with Open GL procedures) language. An example of model of container's shaft line with crankshaft is presented in Fig. 4.

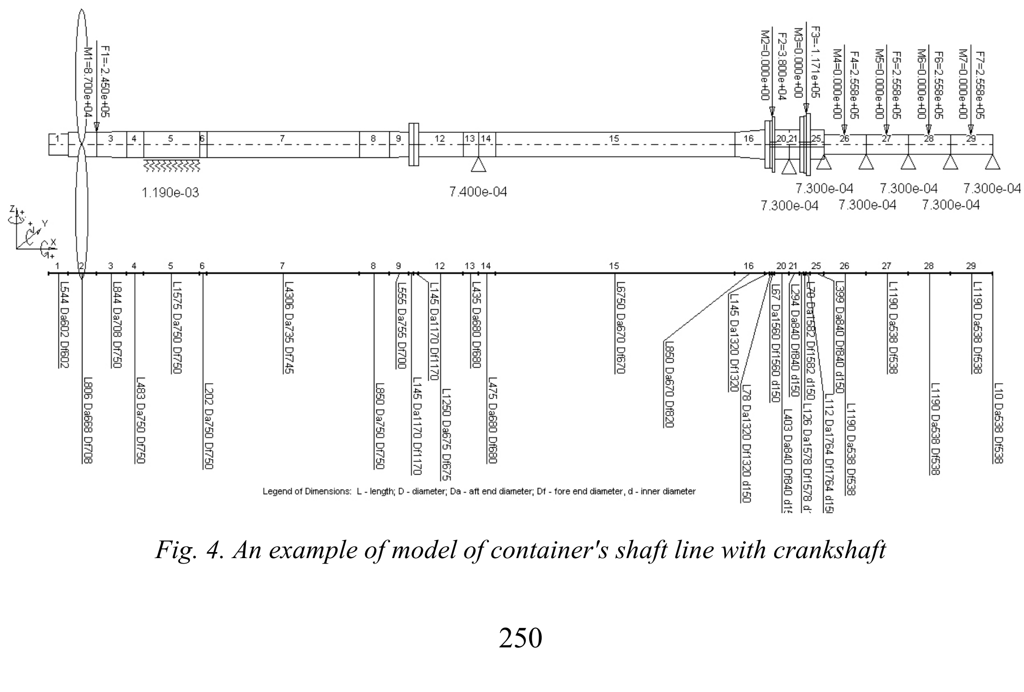

Fig. 4. An example of model of container's shaft line with crankshaft

3. An example of shaft line alignment optimisation

An example of shaft line alignment optimization was performed for universal supply ship with double-shaft and middle-speed propulsion system. The target of those analyses is improving reliability of the propulsion system. The shaft line is more flexible then typical (with slow-speed main engine) – ϕ282 is a diameter of intermediate shaft and ϕ400 is a diameter of propeller shaft. In the ship's documentation, only design bearings' reactions are available. Unknown are calculation assumptions and producers' norm of the propulsion system elements [6]. Real reactions are known after measurements.

Right shaft line was analysed because of worse bearings' reaction distribution. Two calculation variants were performed. First, one (according to the most popular analysis method) for ideal stiff boundary conditions (bearings' foundation). Second, one was performed for elastic supporting points. The elasticity was assumed according to the author's experience. Calculation results of the bearings' reactions, reconstructed designed state of the shaft line, for the elastic supports, are presented in Fig. 5. Distributions of the bearings' reactions are similar for both analyses methods. Only stern tube bearing reaction distribution shows significant differences. Design shaft line alignment is better copied by model with elastic boundary conditions. This model shows that aft part of stern tube bearing is underloaded.

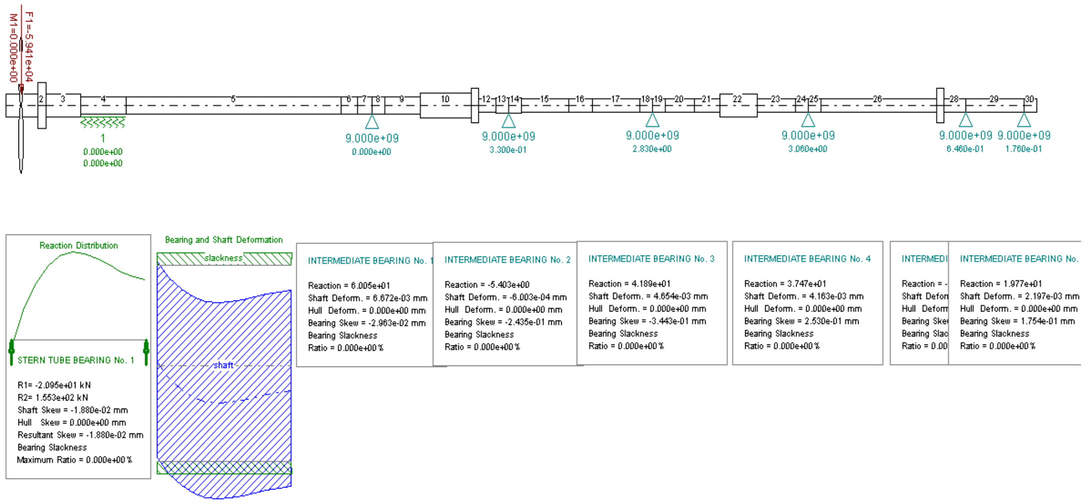

Fig. 5. Design shaft line alignment for elastic support assumption

According to the author, already during the design process the error was committed. Stern tube bearing should be raised vertically with simultaneously first intermediate bearing's levering. Some part of stern tube bearing loads would be taken by the intermediate bearing.

Software made by the author, repeats the design assumption in the first stage. Secondly, searches the solution area to find the shaft line alignment with the measured bearings' reactions (identification). An analysis of identified shaft line alignment shows that bearings' loadings should be improved (mainly stern tube bearing). Gearbox is also too high loaded by bending moments. Improvement in the shaft line alignment can be achieved by the following, technological operations:

– all intermediate bearings should be reclined in order to fore stern tube bearing higher load. It will be a cause of lower loading of aft stern tube bearing and better loading distribution,
– intermediate bearing No. 3 should be relatively (in comparison to No. 2) more reclined. It will be a cause of lower loading of intermediate bearing No. 2 and smaller values of bending stresses and loads reduction (bending moment) of gear box.

Shaft line alignment optimization is difficult without presented software. In the current practice all changes was performed by trial and error method. High experience of the chief engineer was essential. In some technological instructions, it can be found that the deviation of the measured reactions of shaft line bearings ±50% is permissible!

In the third step, sensitivity coefficients are determined by the author's software. The influences of bearings moving on shaft line alignment parameters are calculated. Therefore, optimal shaft line alignment can be determined relatively easily. For the analysed shaft line, fore stern tube bearing was raised by 4.8 mm, and intermediate bearings No. 1, 2 and 3 was raised by 6.1 mm, 4.5 mm and 2.0 mm respectively. Shaft line alignment parameters, after optimization, are presented in Fig. 6-8. Aft stern tube bearing is uniformly loaded. Especially loading of its fore edge is decreased two times. In addition, fore stern tube bearing has better loading because of reducing the risk of “hammering”.

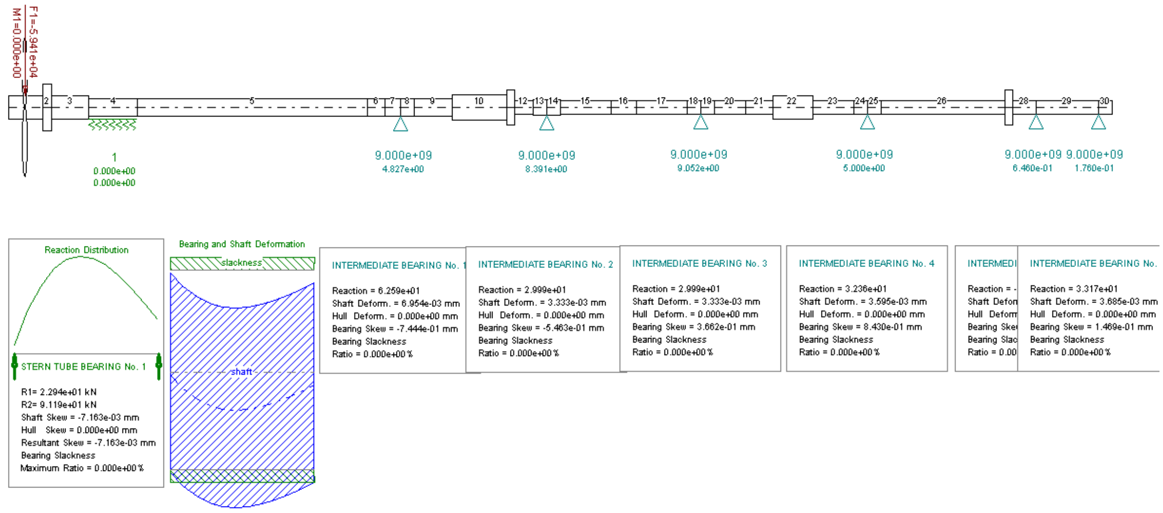

Fig. 6. Bearings reactions after optimisation

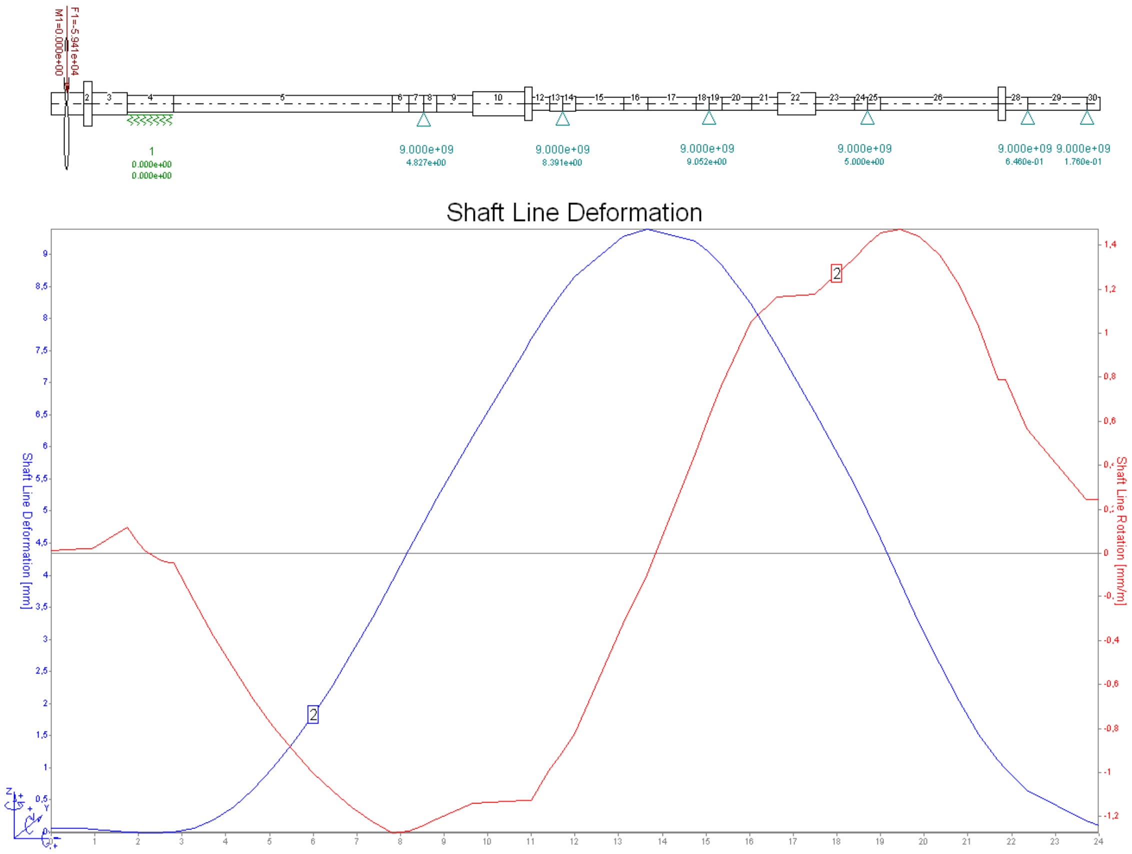

Fig. 7. Shaft line deformation after optimisation

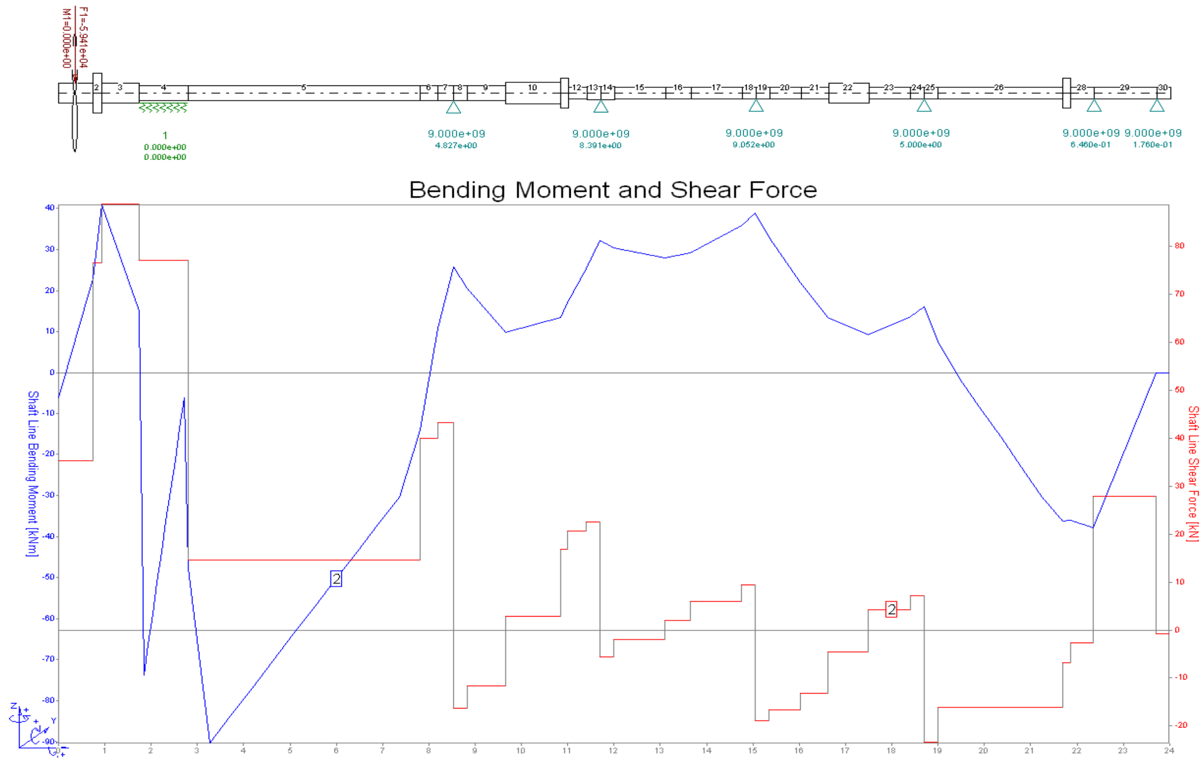

Fig. 8. Bending moments and shear forces after optimisation

4. Conclusions

Changes of shaft line alignment of aged ship with insufficient data availability have been difficult for proper realisation with standard methodology. Usually, it is very costly and the cost of the process is depended on engineer’s experience. In the paper author proposed a method for identification and optimisation of shaft line alignment parameters. Specialised software has been developed and verified for shaft line alignment.

Proper shaft line alignment has a big influence on operational reliability of propulsion system's bearings and so on reliability of navigation safety. Commonly used calculations methods might be not enough accurate. Especially main stern tube bearing should be modelled as a continuous support. In addition, elasticity of ship hull (mainly double bottom) should be taken into account. Shaft line optimisation is also difficult especially with insufficient data availability. Presented software was verified on several ships and different types of propulsion system.

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