FEM Supported Alignment of Power Train Components

Jan Henrik Weychardt, Berend Bohlmann, Flensburger Schiffbaugesellschaft GmbH &Co. KG, Flensburg/Germany, Weychardt@FSG-ship.de , Bohlmann@FSG-ship.de

Abstract

The paper presents lessons learnt from FEM computations on the interaction of the aft body and the gear boxes of a RoPax ferry. The calculations were conducted due to insufficient tooth grip pattern which were observed at trial trip conditions. Other examples on the deflection behavior of modern RoRo vessels, powered conventionally by 4-stroke plants on gear boxes as well as gearless 2-stroke plants address some aspects on assembly modes and operational conditions.

1. Introduction

Ship yards, engine builders and gear box makers put great efforts to increase the efficiency of their products. The steel weight of modern ship hulls is significantly reduced compared to former generations. Hull form and compartmentation are much more efficient and this process is still ongoing, thus allowing the yards to improve their product performance in terms of payload capacity, speed performance, fuel consumption etc. Analogously, this applies for the suppliers of propulsion trains: Engines, gearboxes, clutches and bearings generate and carry higher power densities than ever.

1.1. General problems

Unfortunately, the progress in overall power/mass efficiency has in certain cases resulted in insufficient power/stiffness relations of components which in turn affects the whole system. Beside the conventional assessment of the single components, the interaction of the hull and the power train components is to be evaluated. How ever, local weaknesses are not generally of disturbing influence, but can be advantageous for a working alignment. Another point is that mechanical engineering components are quite sensitive against misalignment and because of an insufficient knowledge about interactions to hull close tolerances are predicted to avoid failures.

1.2. Definition of sufficient alignment

"The alignment of a propulsion train is carried out while a berth and/or building state (irrelevant for service) in a way, that no combination of loading, ballasted and swell conditions, propulsion parameters, local temperatures etc. (relevant for service) leads to irregular states for bearings and (crank) shaft."

This alignment definition is "permanent under construction" and has to be fitted to each special application by weighting the single items. It does not demand an optimal alignment for the design load case because the changes need not to be distributed symmetrically around it. The definition can include the regarding of states corresponding to the likelihood of appearance (service strength). It is intended to allow additional bendings to enlarge alignment tolerances. Another point of "sufficient" is the realization of calculation and modelling within a tenable time!

1.3. Goals

It has to be find a far-reaching knowledge of the interactions between shipbuilding and mechanical engineering components to enlarge alignment tolerances and reliability. This knowledge can also be used to create revised or even new building methods e.g. alignment on building place before launching or before locating major block weights and/or pre-outfitting of blocks with propulsion train components in preassembly stages.

2. Correction of a RoPax ferry's gear box alignment

The example is suitable for the interactions between shipbuilding and any mechanical engineering components, like gear boxes, main and auxiliary engines, PTOs etc. It deals with a twin-screw RoPax ferry, powered by 2x2 4-stroke engines with a total power of about 22000kW, Fig.1.

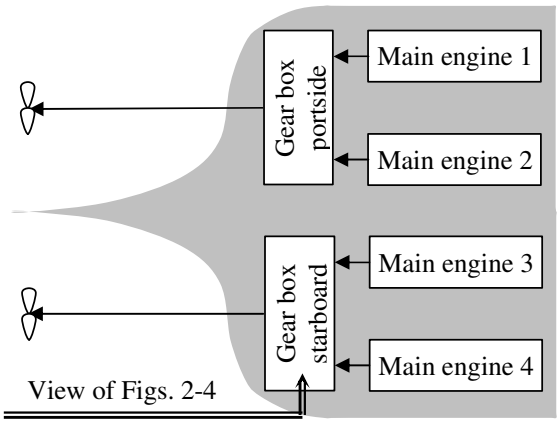

Fig.1: Plant arrangement in aft ship

2.1. Problem

The alignment of the gear boxes had to be corrected due to insufficient tooth grip pattern on all four engagements of driving pinions and following gears, which were observed at trial trip conditions, Fig.2. It has to be taken into account that after some hours of service conditions a tooth grip pattern is in state of development. Nevertheless, at the first sight the foundation seemed to be too weak. This point of view was emphasised by the fact that this vessel was weight critical while construction and as a result there were some rumours that the foundations of the gear boxes appeared too weak. However, a point of weakening gear foundations is the fact, that the following wheel's big diameter demands a cut-out in the upper deck of double bottom, directly on the fore side of the thrust bearing. FSG was asked to evaluate the influence of foundation to the tooth grip patterns and for suggestions to improve them to everyone's satisfaction.

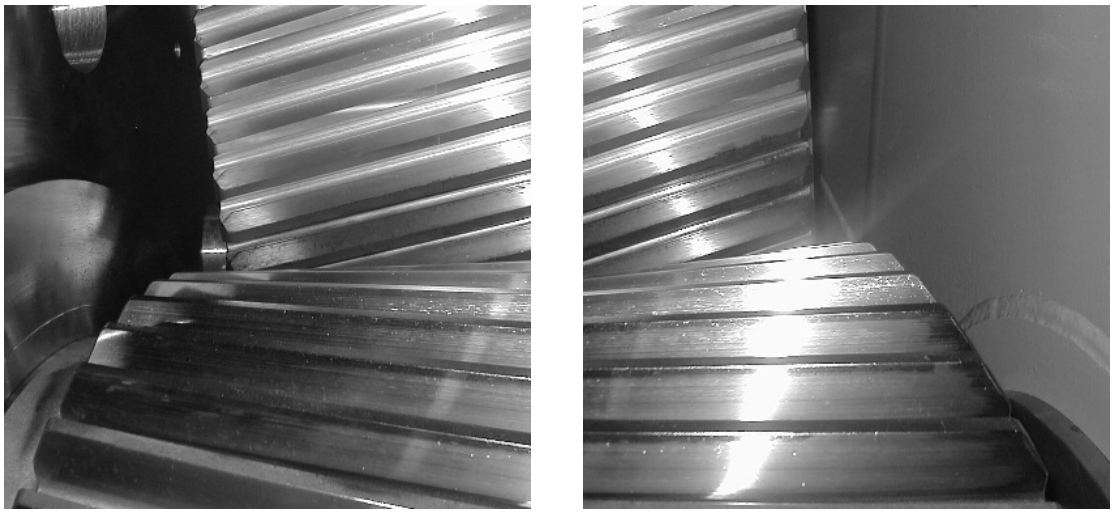

Fig.2: Insufficient tooth grip pattern on starboard gear box, view as shown in Fig.1.
In the foreground the starboard driving pinion, in the background the following gear

2.2. Causes

First researches resulted in a more complex problem: In the early design stage the gear supplier thought FEM modelling and calculations to the gear itself are unusual. On the other hand he demanded from the yard to guarantee a maximum gear shafts crossing of 0.1mm/m without recognizing the gear house stiffness. So the yard took a “dummy gear” FEM calculation to design a light weight foundation with sufficient stiffness. Obviously there was no more relevant communication. Each partner designed his own part as shown in Fig.3 on the final FEM model, realized after trial trip.

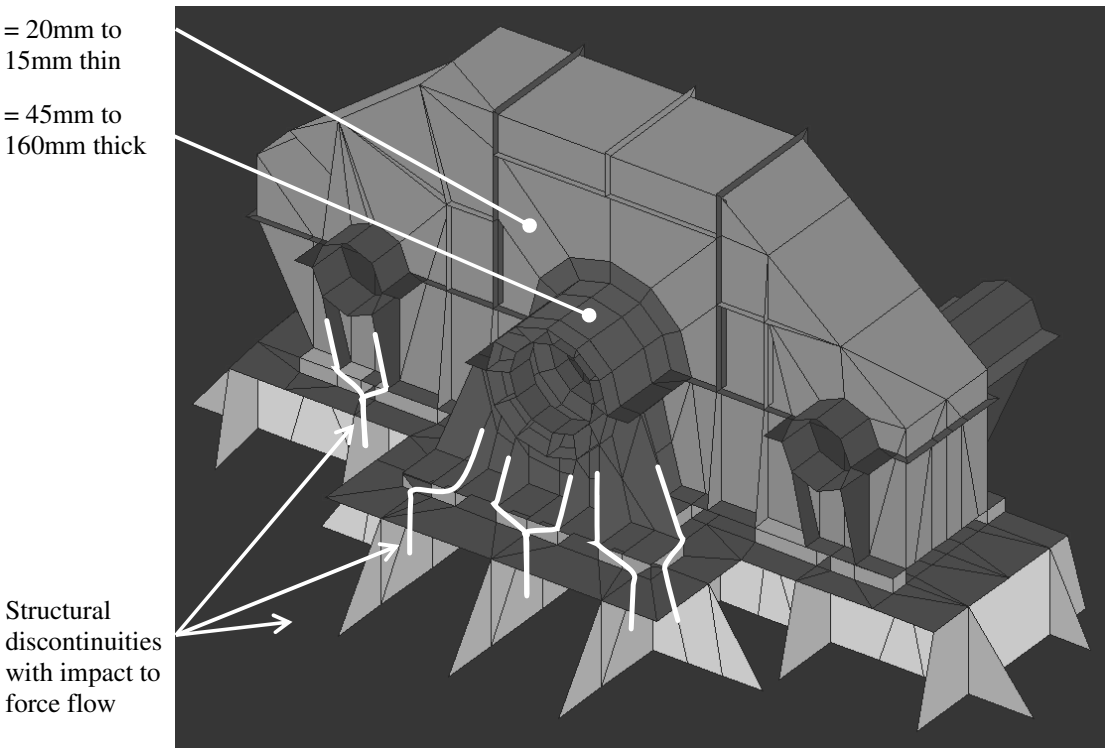

Fig.3: Disregarded basic design rules at the cut of gear box and foundation:

The gear box is coupled to two engines on the back side of view and the shaft line on the fore side. There is also placed the dark grey coloured thrust bearing. The following basic design rules were disregarded:

- No needless jump in material thickness (the thinner, the lighter drawn). No material of 20mm to 45mm is used. The thrust bearing is stiff against environment.
- No needless structural discontinuities with impact to force flow at the transition of gear and foundation.

Conclusion ➔ more stiffness was possible with less material.

2.3. FEM-calculations

The FEM model shown above is necessary to calculate the actual deformations in the system of interacting gear box and foundation. Bearing stiffnesses, shafts and engaged gear wheels are not necessary to evaluate the quality of foundation. According to the service condition “both engines full power” the model was loaded with the maximum thrust and bearing loads resulting from maximum torque. The deformation behaviour can be explained by the vertical displacements shown in Fig.4.

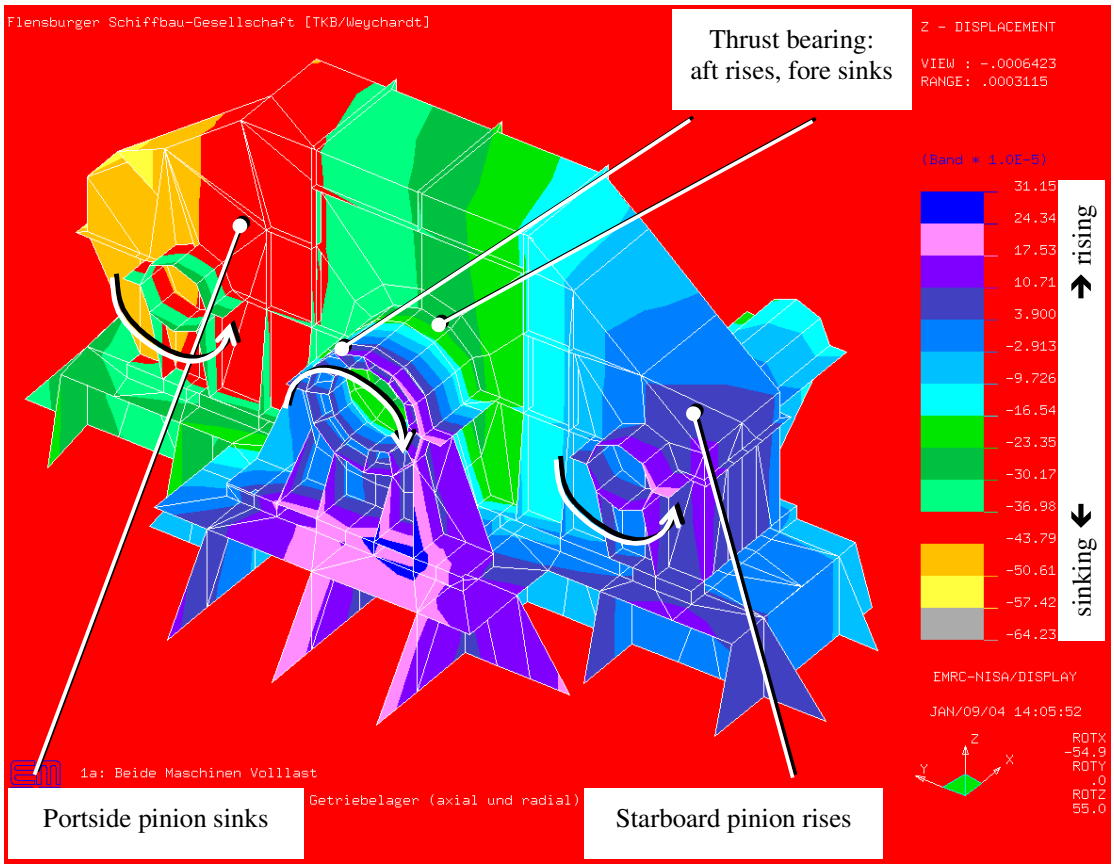

Fig.4: Vertical displacements at portside gear, propeller rotates clockwise, engines counter-clockwise, sinking and rising of pinions due to tooth force directions

The sinking of the portside pinion can easily be detected by the darkest grey colour in grey scale. It sinks because the downwards directed reaction force on the following gear due the counterclockwise rotating engines. Persecuting the grey scale on the housing to the rising starboard pinion indicates a smooth tilting of the whole gear box. A much stronger tilting can be detected at the thrust bearing with more slender grey colour stripes.
The relevant results were:

- Crossings of the shaft's projection lines on the top plate of foundation between shaft's bearings: 0.04mm/m<0.1mm/m. Result: The structural stiffness of the vessel is sufficient.
- Crossings of the shaft's lines between shaft's bearings in gear house: 0.14mm/m>0.1mm/m, the gear house's part is 0.1mm/m and does not fulfil the demand of its own manufacturer. The calculated crossing will even increase by taking into account bearing play and weakness of shafts, gear wheels and tooth engagement.
- Tilting of thrust bearing to cross axis (not demanded): 0.4mm/m, as consequence the shaft tilts itself and moves up in the thrust bearing.

Other asymmetrical load cases caused smaller crossings.

2.4. Problem solution

A new shaft alignment towards the gear box was calculated. Some journal bearings had to be raised. After the second trial trip the tooth grip patterns were almost satisfying, Fig.5. Another optimisation was carried out by the gear manufacturer by smooth tensings of the housing.

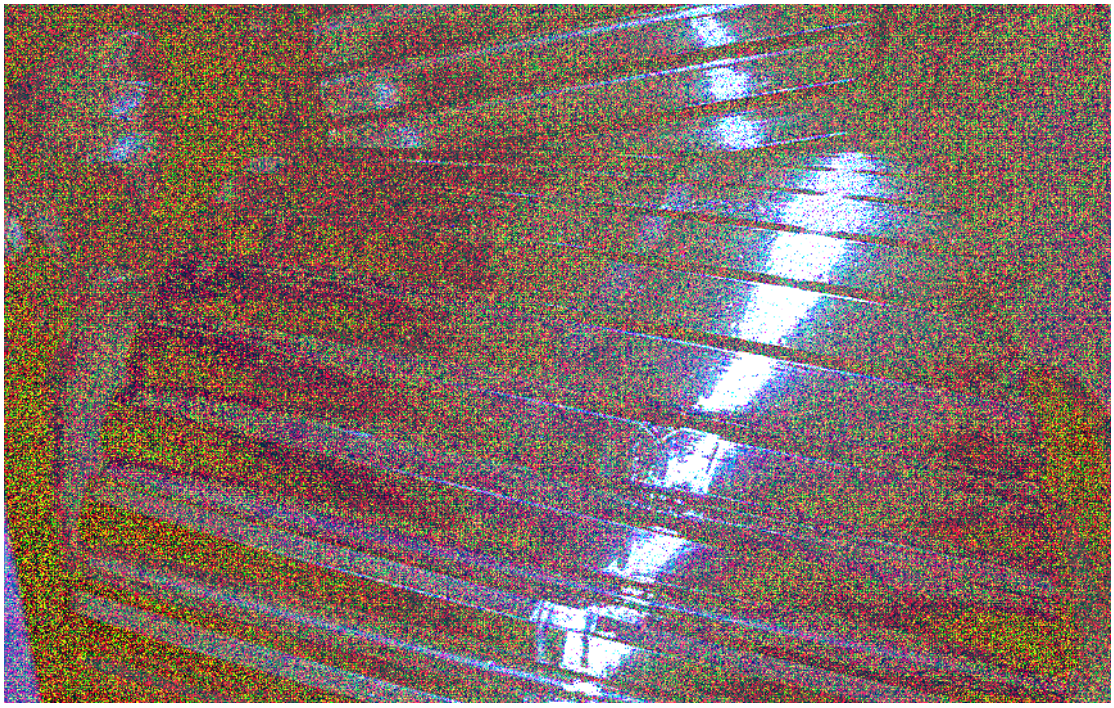

Fig.5: Sufficient tooth grip pattern, same pinion/wheel engagement as in Fig.2

2.5. Lessons to be learnt

Costly amendments can be avoided by using available technologies in early design stage. In this case the FEM-modelling of the gear box, its foundation and the close environment would have been sufficient. To recognize basic design rules an approach of mechanical and ship building engineers is necessary regarding the fact that supply components only can be reasonable in series production. That is only possible with a frictionless communication.

FEM calculations in early design stage can be a stable base for supplier guarantees and for the yard a help to find the best fitting component. A close attendance by classification society is desirable. Such calculations also can avoid illogical demands like a maximum shaft crossing without modelling the shafts and their environment. In this case the quality of tooth grip should have been demanded because it was the reason of query.

3. Crank web clearance calculations by FEM

FSG carried out some studies referring the crank web clearance of 4-stroke 9-cyl. engines, because it is one of the criteria for a sufficient alignment of main engine. The crank web clearance is the distance change of two shaft segments neighbouring to the same crank at one revolution. The amplitude can change at different service conditions, e.g. caused by thermic deformations. So the interaction of crank shaft, engine and double bottom is interesting to be evaluated.

A sufficient modelling of double bottom's and engine's steel structures and lubricating films is state of the art, see Ch.4.1. However, it is not a standard computation at yards, classification societies and engine makers and even in special cases rarely used until today. The emphasis was laid to the crank shaft with its unsteady geometry. Two starting points were discussed: The modelling with beam elements or brick elements.

Beam elements are easy to define and quick to calculate. But a model reflecting all the mechanical behaviour at tensile, bend and torque loads was not found with a tenable expenditure. Further computation rates increased so that big models can be calculated in a sufficient time. So a model of one crank segment was made out of brick elements, Fig.6. A crankshaft can be modelled by connecting any number of these segments at any angles by rigid links at the marked centre point. So it is not necessary to adapt the meshing on the end wall to different angles resulting from the number of cylinders and/or the ignition sequence. Most certainly other inaccuracies in modelling cause bigger deviations. However, a linear deflection behaviour is assumed for the transition which might give reasons for further discussions.

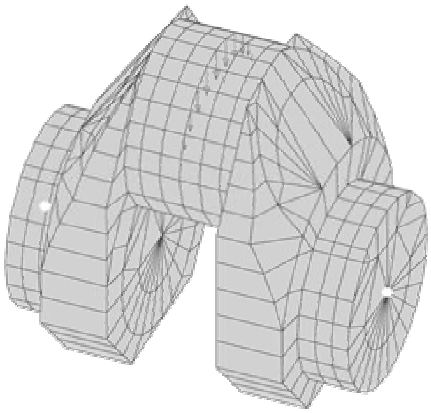

Fig.6: Crank segment for one cylinder made of brick elements

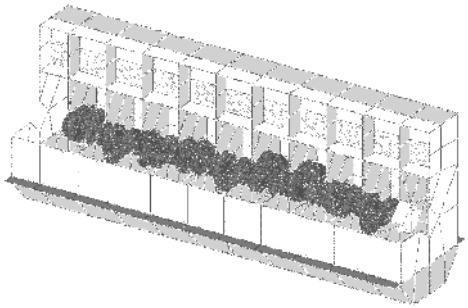

Fig.7: Crank shaft made of 9 segments in engine housing

A 9-cyl. crank shaft embedded in the engine housing is shown in Fig.7. This ensemble was connected to a double bottom structure and bended by a reference moment. The FEM-model of crankshaft was revolved in 40° steps for each of 9 calculations and the crank web clearance was determined for each cylinder. The results are sufficient, Fig.8. All curves are nearly sinusoidal as expected. The amplitudes depend on the relative angle between two cranks. Small amplitudes are calculated for the cylinders no 3, 4, 6 and 7. The reason is that the relative angles from no 3 to 4 and 6 to 7 is 80° instead of 40° for all other neighbouring cylinders.

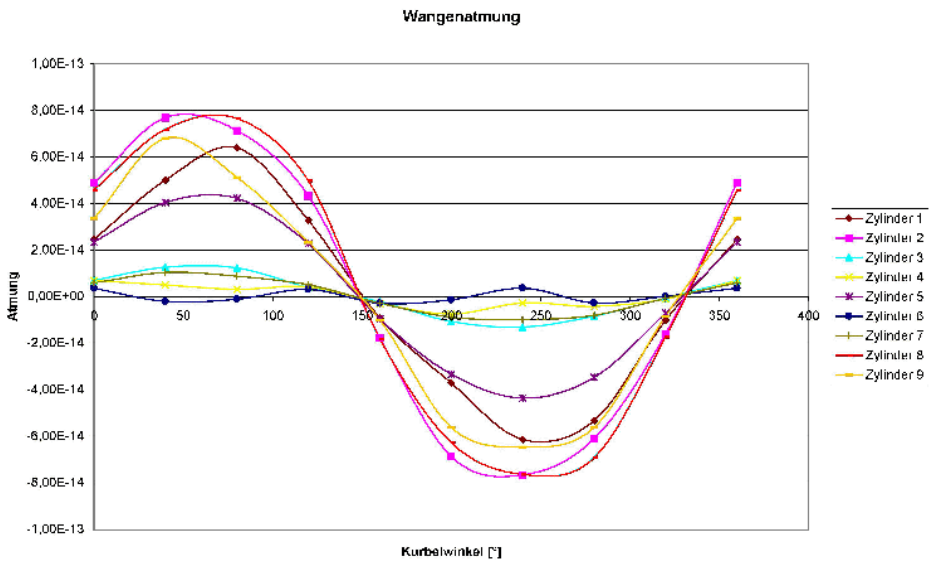

Fig.8: FEM calculated crank web clearances at 9 cylinders

4. Alignment of a RoRo vessel's propulsion train

At present FSG builds 5 RoRo vessels with a FEM supported alignment. Bearing displacements resulting from hull deformations of all relevant building, outfitting and service cases were regarded in bending line calculations.

4.1. FEM models

All calculations for different cases were carried out with variants of a global FEM model taken from the vibration analysis. The meshing in the lower aft ship area had to be refined, Fig.9.

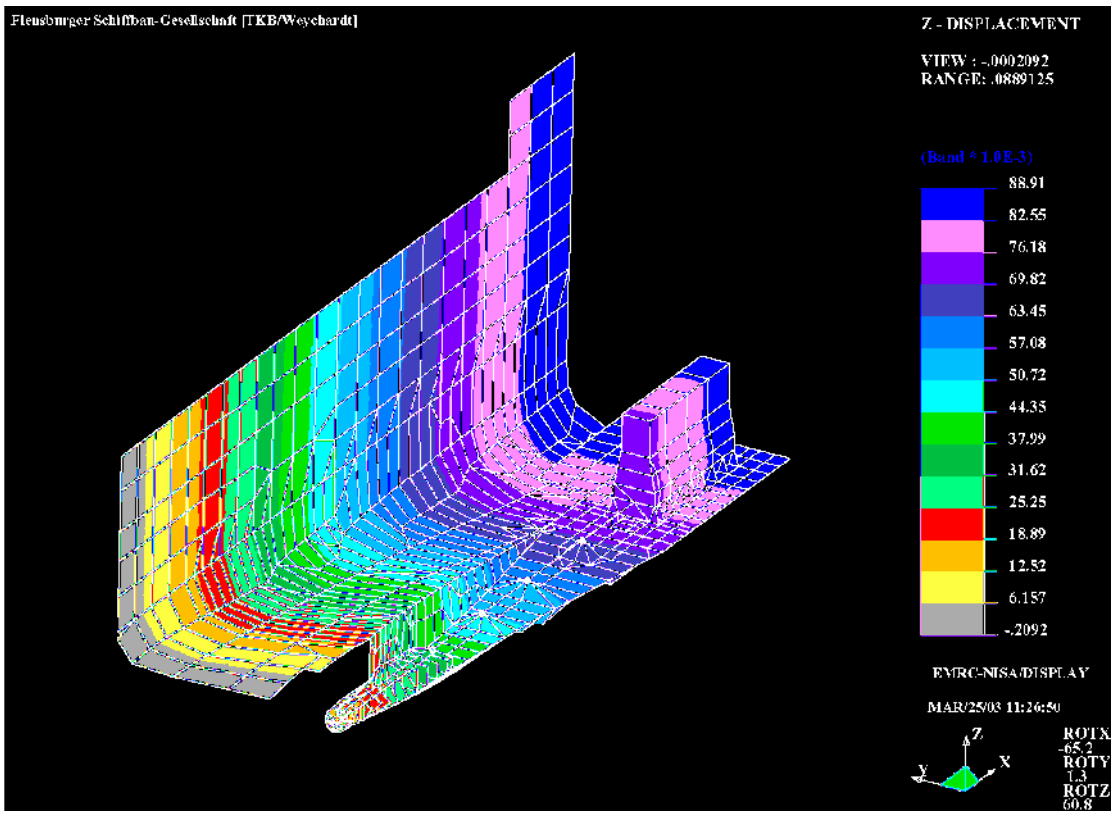

Fig.9: Cut out of global FEM model
showing aft ship (shell) with refined meshes, main engine and shaft line

The steel structure was modelled by shells and beams with linear deflection approaches and varies with the building state. Individual mass distributions are realized by single masses on each node. There are also inducted case specific service forces. The water pressure distribution regarding draught, trim and heel is inducted normal to dipped shell elements. Shaft and hull are connected by springs representing the stiffness of lubricating films in each bearing at specific service conditions.

4.2. FEM calculations

As all results, the calculated displacements in Fig.9 refer to an imaginary state without any deformations. So they are not suited for conclusions and have to be referenced to other deformation cases. In this application a reference deformation case "according to specification" (typical loading conditions simulating cargo load, bunker etc. according to owners requirements) for best alignment could be defined, because the main changes resulting from maximum wave sagging and hogging are nearly symmetrical and the different load cases cause only negligible changes. Some building and

outfitting states had to be regarded to calculate the changes from alignment states to service state. About 20 deformation cases were defined, e.g. to evaluate changing conditions while building or outfitting. The most important were:

1. According to specification (still-water service conditions)
2. Maximum hogging
3. Maximum sagging
4. Outfitting alongside quay (alignment finish)
5. Assembling aft ship block (first alignment under slipway conditions, w/o main engine and deck house)

Deformation changes inside propulsion train can be evaluated, if all calculated displacements are referenced to any two points, e.g. aft stern tube bearing and fore end of engine, Fig.10. The first three graphs in agenda show the vessel's deflections caused by the maximum vertical design wave bending moment according to class, e.g. the maximum change between maximum hogging and sagging ("3-1") is about 5mm. The other three give information, which displacements have to be provided while alignment to reach the best result in service conditions, e.g. the influence of launching is about 3mm ("5-4").

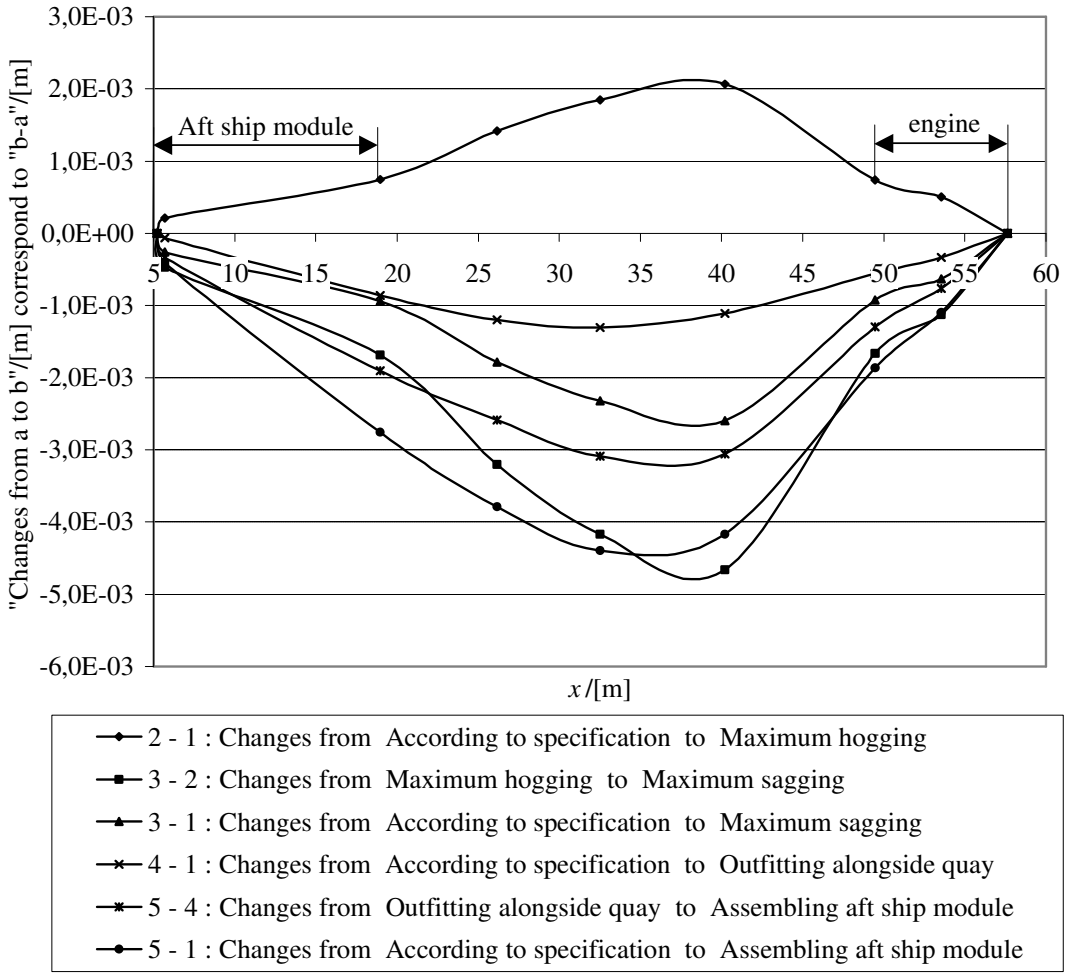

Fig.10: Deformation behavior (vertical) of a RoRo vessel's aft ship structure changing between different deformation cases

4.3. Calibration/Validation of the FEM model

The most sensitive location against misalignment in the propulsion train is the shaft bossing on the aft ships block, Fig.11. It is designed for optimal propeller inflow, so the lever arm from aft stern to the 35t propeller is approx. 6.5m. The decision to calibrate the FEM model here was founded in three reasons:

- All FEM calculations of vessel show the most significant bending in this region of lowest stiffness, Fig.10,
- the calculated alignment value for the aft stern tube bearing of about 15mm above center line is unusual high and
- a damage of this bearing is most costly because of docking.

The bending behavior was tested with a beam, welded in two points of 8m distance under the block and a weight force on the aft end, Fig.11. The weight force bends only the shaft bossing but not the quasi elastic linked test beam.

So the relative deformations were measured without boundary effects and could be compared by a FEM calculation. The requirements to the weight were:

- At least 20t to cause measurable deformations (pre-calculated),
- easy to transport and
- height incl. lifting device less then 3.5m to fit under test beam.

The choice was the elevating platform truck that normally carries ship blocks. To be carried by a block as shown in Fig.12 is a special load case that had to be approved by the manufacturer.

Fig.13 shows the good agreement between calculation and measurement. A last validation of the unusual alignment values are successful trail trips and service performance.

Fig.11: Aft ships module with shaft bossing, test beam and weight force ***F***

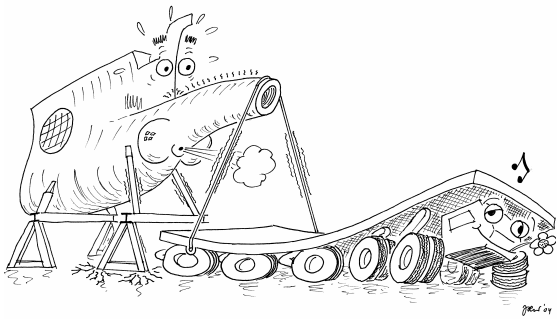

Fig.12: Topsy-turvy world:
Module carries elevating platform truck

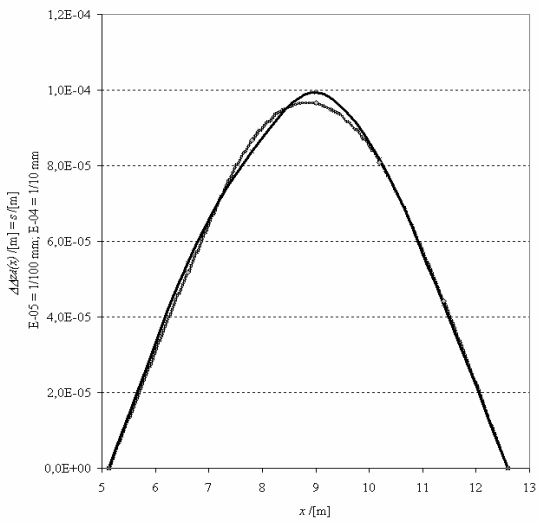

Fig.13: Comparison calculation-measurement converted to a weight of 10t,
FEM (grey), measurement (black)

References

BOURCEAU, C.; WOJCIK, Z.C. (1966), *Die Beanspruchung von Kurbelwellen in Dieselmotoren,* Jahrbuch der Schiffbautechnischen Gesellschaft

CASTLE, D., (1969-1979), *Alignment Investigation following a Medium-Speed Marine Engine Crank Shaft Failure,* Proceedings of The Institution of Mechanical Engineers

DeGEORGE, V.A. (1982), *Combined Effects of vertical and horizontal Shaft Alignment on Main Reduction Gear,* Marine Technology, pp.178-184

ILLIES, K. (1969), *Wechselwirkungen zwischen Maschine und Schiff,* Jahrbuch der Schiffbautechnischen Gesellschaft

N.N. (2000), *Rigid shafting and flexible hulls*, The Motor Ship, pp.87-92