Some Concerns on Shaft Alignment Onboard Ship

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Abstract. As aft stern tube bearing damages were reported quite often over the past years, which may pose threats to the safe operation of the ships in the worst cases, shaft alignment has become a more heated topic than ever before in the industry. Root cause behind the damages is of interest to all parties involved. In this regard, concerns have been paid to the "uncertainties" in the shaft alignment process i.e. from the early shaft alignment design to final execution of shaft alignment onboard and their contributions to the damage are discussed as well in the paper. The uncertainties emphasized in the paper as sensitive and crucial to the shaft alignment behavior are, to the best practice and knowledge of CCS, identified and elaborated in a practical and effective way. The uncertainties are mainly focused on:

-the propeller hydrodynamic loads

-alignment procedures for one stern tube bearing arrangement shafting

-instability of viscosity of EALs (Environmentally acceptable lubricants) against pressure

It is noted it is rather difficult to get some uncertainties quantified precisely even with advanced computer programs for the time being. One of the best solutions to cope with the uncertainties is to improve the safety margin in the early design stage by reducing the mismatch angle between bearing surface and shaft. With the above in mind, a robust design of aft stern tube bearing, e.g. optimized double slope bearing, as a countermeasure is presented in the late part of the paper.

Keywords: shaft alignment, mismatch angle, double slope bearing, uncertainty

1 Introduction

As aft stern tube(S/T) bearing damages were reported quite often over the past years, shaft alignment has been paid more attentions by the industry. Root cause behind the damages is of interest to all parties involved. A thorough investigation has been carried out based on the failure cases. Concerns have been paid to the "uncertainties" in the shaft alignment process i.e. from the early shaft alignment design to final execution of shaft alignment onboard, which could affect the shaft alignment quality in one way or another.

2 Uncertainties

2.1 A Propeller Hydrodynamic Loads

Over the past years during sea trials, aft stern tube bearing overheating damages have been reported where aft stern tube bearing suffered high-temperature alarm during a manoeuvre to starboard and the bearing temperature increased to an ever higher than 110 degrees Celsius in a very short time.

Upon removal of the propeller shaft, the stern tube bearing was found to have suffered extensive overheating damage with circumferential scoring and melting of the white metal of bearing as shown in Fig.1. However at the same time it was observed that nothing happened during a manoeuvre to portside. Obviously the propeller hydrodynamic loads counted somehow in such cases. Unfortunately classical shaft alignment calculation usually focuses more on static conditions, less on such dynamic running conditions.

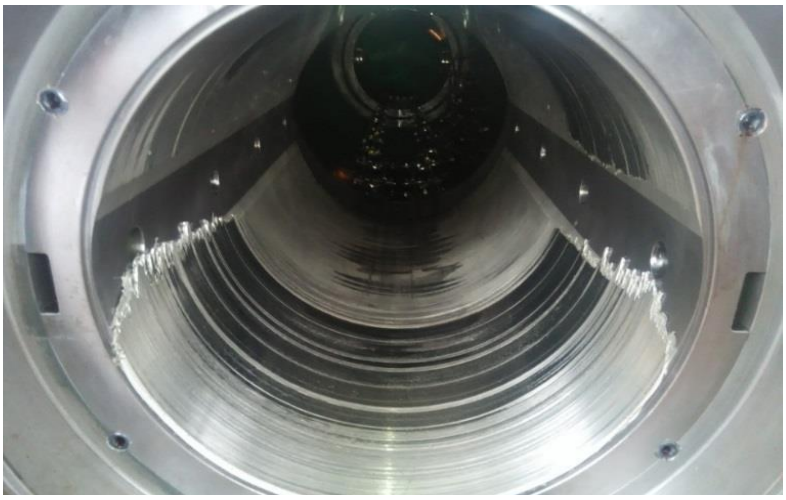

Fig. 1. Overheating damage of aft stern tube bearing

Investigations on the hydrodynamic propeller loads have since been carried out either by means of CFD or measurements. Both methods have shown that additional hydrodynamic bending moments imparted from the propeller during manoeuvres are evident. Particularly a bending moment My, as shown in Fig.2, brings the propeller downward during a manoeuvre to starboard, which may enlarge the mismatch angle between the aft stern tube bearing surface and propeller shaft journal significantly to exceed the normal acceptance limit of 0.3 mm/m. As a consequence, an undesired hard edge loading occurs to damage the bearing. Therefore in theory it is necessary to get the hydrodynamic propeller loads considered case by case in the shaft alignment design to quantify its effect on the bearing performance. However to get the hydrodynamic propeller loads on all conceivable operation conditions is not only a time-consuming work with an advanced CFD method, but also difficult to get it with a satisfactory degree of accuracy, which is thus thought an uncertainty in a sense.

Based on the experience on numerous successful solutions to the bearing failure cases and from a practical engineering point of view, the propeller hydrodynamic loads can be simply simulated by an application of the moment My, which brings down or lift up the propeller. Normally an assumption of the moment My as ±30% of the nominal propeller torque is practical and on the safe side for the conventional tankers, bulk carriers and container ships.

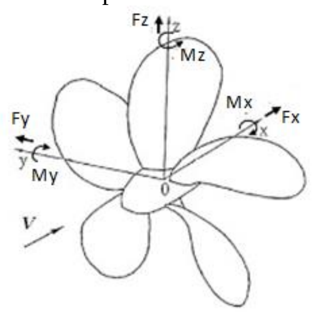

Fig. 2. Hydrodynamic propeller loads

2.2 Alignment Procedures for One Stern Tube Bearing Arrangement Shafting

It is observed a tendency in the design of shaft line over the years. More shaft lines are designed with only one stern tube bearing, i.e. no forward stern tube bearing out of various reasons. The critical reason is that the shaft line without forward stern tube bearing will become less sensitive to the shaft alignment thanks to the increased bearing spans, leading less difficult for shaft alignment design. However an interesting phenomenon is also noted with such design for a bulk carrier as shown in Fig. 3.

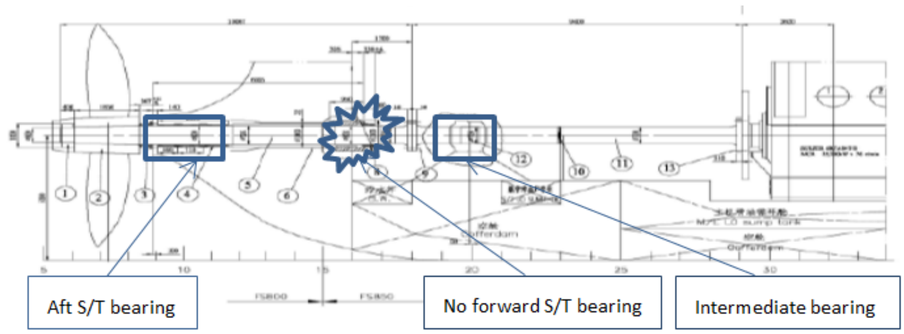

Fig. 3. Shaft arrangement

It is a typical direct coupled plant where the propeller is directly driven by a two-stroke engine through a propeller- and an intermediate shaft. An equivalent crankshaft model(beam model) for shaft alignment purpose supplied by the engine maker was used in the calculation. The complete model for shaft alignment calculation is shown in Fig.4.

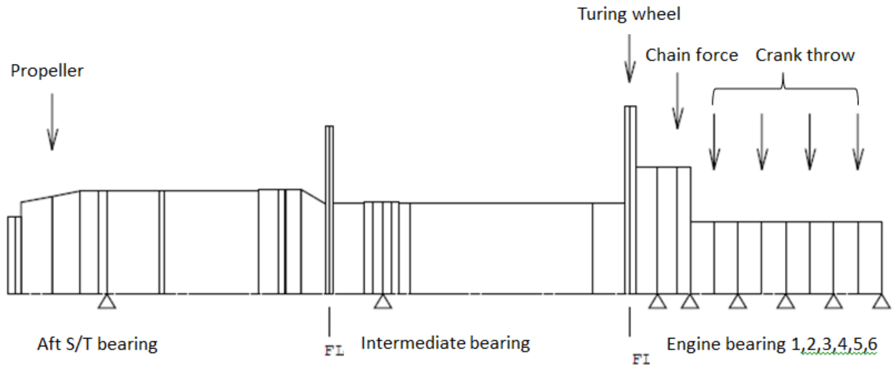

Fig. 4. Shaft model of alignment calculation

Some results of the shaft alignment with two different sets of bearing offsets are shown in Table 1, 2 and Fig.5, 6.

Table 1. Bearing loads and mismatch angle-1 (cold static)

	Aft S/T bearing	Inter. bearing	No.1 M/E bearing	No.2 M/E bearing	No.3 M/E bearing	No.4 M/E bearing	No.5 M/E bearing
Bearing Offset [mm]	0.00	-1.75	-4.79	-4.79	-4.79	-4.79	-4.79
Bearing reaction[kN]	1004	203	72	418	384	358	453
Mismatch Angle [mm/m]	0.20						

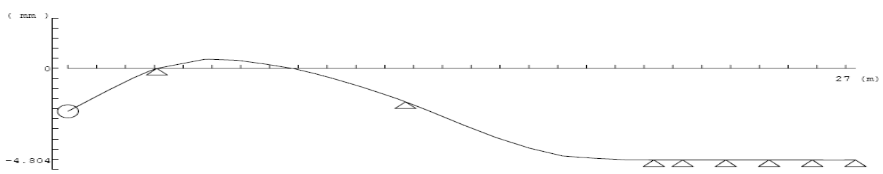

Fig. 5. Shaft deflection-1 (cold static)

Table 2. Bearing loads and mismatch angle- 2 (cold static)

	Aft S/T bearing	Inter. bearing	No.1 M/E bearing	No.2 M/E bearing	No.3 M/E bearing	No.4 M/E bearing	No.5 M/E bearing
Bearing Offset [mm]	0.00	0.00	-2.62	-2.62	-2.62	-2.62	-2.62
Bearing reaction[kN]	996	219	72	409	384	358	453
Mismatch Angle [mm/m]	0.43						

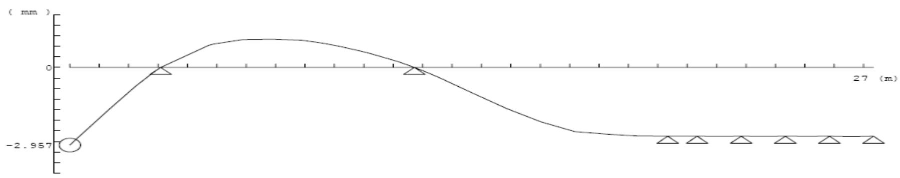

Fig. 6. Shaft deflection-2 (cold static)

As can be seen, different bearing offsets could lead to more or less the same bearing load, but with different mismatch angle between the propeller shaft and the aft stern tube bearing, e.g., 0.20mm/m and 0.43mm/m respectively in this case. The mismatch angle is crucial to the bearing lubrication condition and that is why a limited mismatch angle of 0.3mm/m is specified by the classification societies as a more practical guide to safeguard a satisfactory hydrodynamic lubrication. The one with mismatch angle of 0.43mm/m is far beyond the general limit 0.3mm/m, not acceptable, while the other with mismatch angle of 0.2mm/m is below the limit, acceptable. Although the bearing loads are very close, the mismatch angle is not only different in magnitude but also totally different in nature in this case. It comes to the conclusion that unlike shaft line with forward stern tube bearing, for shaft line without forward stern tube bearing the verification of bearing loads, is necessary condition but not sufficient and necessary condition, simply because it can't ensure the uniqueness of the mismatch angle. But the normal practice to check the shaft alignment is just a verification of the bearing loads by comparison between the measured bearing loads and the theoretical ones. And in practice the intermediate bearing offset is normally adjusted during alignment process simply to achieve the acceptable bearing loads. However in fact, the mismatch angle varies with the adjustment of the intermediate bearing as an uncertainty, which unfortunately escapes our attention somehow.

Obviously the check of the bearing loads only is not enough for the one stern tube bearing arrangement shaft to ensure a satisfactory shaft alignment to be in compliance with the designed one. An additional check is necessary. One additional economic and effective way is to check the shaft deflection in relation to the forward sealing in way of the forward sealing, ensuring the actual measured deflection to be same as the designed one that corresponds to the designed mismatch angle.

2.3 Instability of Viscosity of EALs(Environmentally acceptable lubricants) against Pressure

Since the VGP regulation entered into force by the US EPA on December 19, 2013, cases of overheating of aft stern tube bearing have been reported at mooring trial or at sea trial due to application of the EALs in stern tube. It is very interesting to experience at some vessels where such bearing overheating incident disappeared simply after changing the lubricant to mineral oil from EALs. The difference between the normal mineral oil and EALs is evident in terms of lubrication performance. Therefore tribological study & measurement of EAL's basic properties have been executed by some EALs suppliers. One of the unfavored EALs properties found by the oil suppliers is the instability of viscosity characteristics against pressure, namely, the viscosity gets lower than mineral oil with the increase of local pressure. The pressure-viscosity coefficients of EALs are normally around 20~30% lower than mineral oil, but it could further up to 40% under severe load conditions and oil film thickness becomes thinner accordingly. This kind of instability of viscosity should be well addressed as an uncertainty in the application of the EALs.

In view of the nature of the EALs, in order to minimize the impact of the instability it is very important to avoid severe load condition by reducing the above-mentioned mismatch angle. Of course use of EALs with higher viscosity is another option.

3 Robust design of aft stern tube bearing

3.1 Principle

With the above-mentioned uncertainties in mind, a practical way to deal with the uncertainty, based on best CCS practice and experience, is

- to increase the safety margin in the shaft alignment design stage by reducing the mismatch angle to be less than 0.1mm/m in static condition from the normal acceptance level 0.3mm/m. As such, it leaves enough tolerance to the unfavorable effect of the uncertainties
- to design the aft stern tube bearing that is less sensitive to the effect of the hydrodynamic propeller load

Theoretically, a satisfactory build-up of oil film ensuring hydrodynamic lubrication is paramount to a satisfactory shaft alignment for the safe and reliable operation of a ship during its lifetime. So bearing hydrodynamic performance analysis based on well recognized Reynolds equation is used for the robust design of the aft stern tube bearing. The optimized targets are as follows:

- Be of a double slope bearing

- Maximum mismatch angle(alpha A~C as defined in Fig.7) on the static conditions: 0.1mm/m
- Minimum oil film thickness on dynamic conditions: 0.02mm

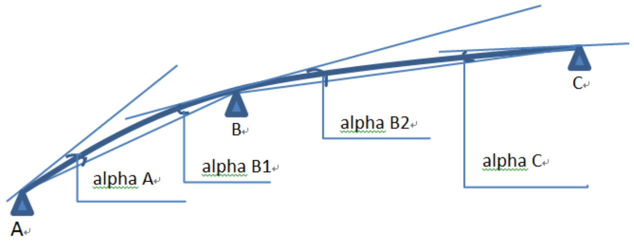

Fig. 7. Mismatch angle between bearing surface and shaft

3.2 Case Study

In order to illustrate the robust design of the bearing, an example of a bulk carrier is presented here.

The shaft arrangement is similar to the fig.3, without forward stern tube bearing. The plant is powered by a 2-stroke engine with nominal power of 8500kw at 77rpm. EAL is used in this case. Therefore an additional calculation in which the viscosity uncertainty is simulated by using 70% of its nominal viscosity was also made on the safe side.

The optimized bearing is a double slope bearing. The primary and secondly slopes are approx. 0.17mm/m and 0.4mm/m respectively, as shown in Fig.8.

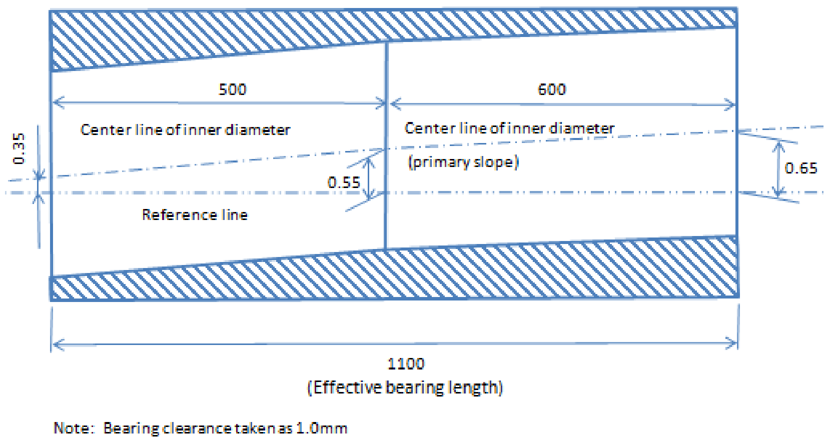

Fig. 8. Optimized double slope bearing geometry

The analysis is done in two steps. First make a traditional shaft alignment calculation to get bearing loads and bearing mismatch angles. With the calculated bearing loads and mismatch angles as input, then make a bearing hydrodynamic performance analysis using Reynolds equation to get oil film thickness. The results of the analysis are summarized in Table 3 and 4.

Table 3. Bearing loads

	Aft S/T bearing load [kN]			Inter. bearing load [kN]	No.1 M/E bearing load [kN]	No.2 M/E bearing [kN]	No.3 M/E bearing [kN]
	Part A	Part B	Part C				
Static warm	53	122	114	65	64	125	198
Warm running with -30%T downward	294	23	0	35	74	122	195
Warm running with 30%T upwards	0	0	245	115	51	130	202

Table 4. Mismatch angle and oil film thickness

	Aft S/T bearing mismatch angle [mm/m]				Oil film thickness at 77rpm [mm]	
	alpha A	alpha B1	alpha B2	alpha C	viscosity 100cSt	viscosity 70cSt
Static warm	0.06	0.07	0.09	0.09	0.17	0.12
Warm running with -30%T downward	0.14	0.14	0.35	0.06	0.11	0.06
Warm running with 30%T upwards	-0.68	-0.63	-0.19	-0.21	0.11	0.06

As can be seen, the oil film thickness is well above the minimum limit of 0.02mm and thus the design is on the safe side with an expected higher safety margin.

For better understanding of the optimized bearing geometry, the same analysis was carried out for the original bearing design. The original bearing is a single slope design with a slope of 0.2mm/m, as shown in Fig.9.

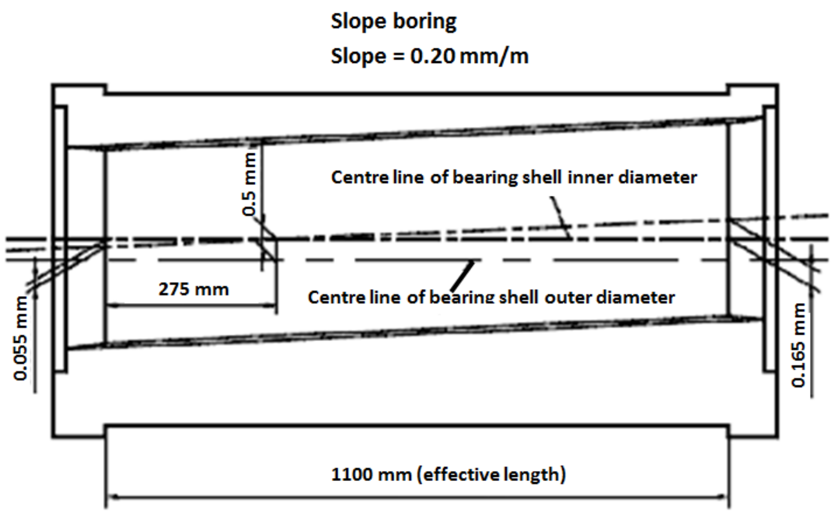

Fig. 9. Original single slope bearing geometry

The calculated results of oil film thickness, on warm running condition with -30% torque downwards, are presented as compared to the optimized double slope bearing in fig.10.

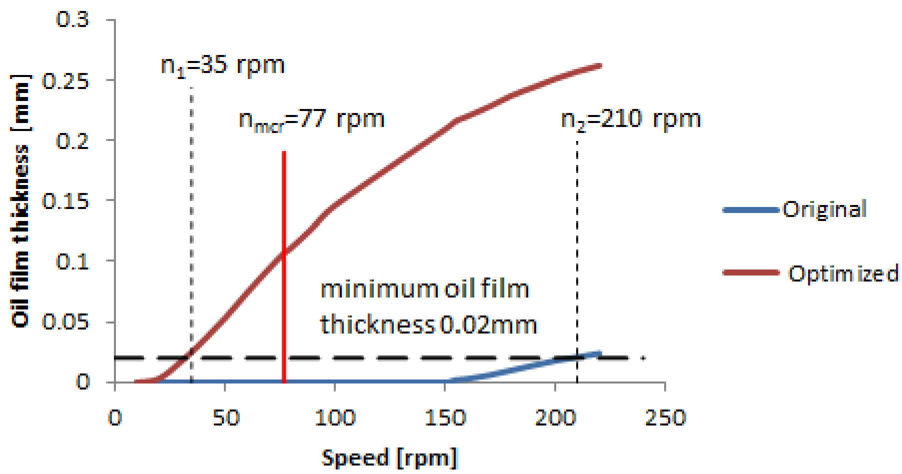

Fig. 10. Oil film thickness vs. speed

The difference is significant. As the required minimum oil film thickness is 0.02mm, for the original design the minimum speed n_2 ensuring hydrodynamic lubrication is 210rpm, far beyond the nominal speed $n_{mcr,\ 77rpm}$. However for the optimized design the required minimum speed n_1 is 35rpm, well below the nominal n_{mcr}. One thing here we have to point out is that this comparison does not justify the original bearing design is definitely not suitable for the plant, but more sensitive to the hydrodynamic propeller loads anyway. On the contrary, in other word, the optimized bearing is much less sensitive to the hydrodynamic propeller loads as a robust design.

4 Conclusion

Since the uncertainties are there, what we can do is to make a robust design of the aft stern tube bearing on the safe side, leaving more tolerance to the uncertainties. With an optimized double slope bearing, a robust design with higher safety margin is possible.

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