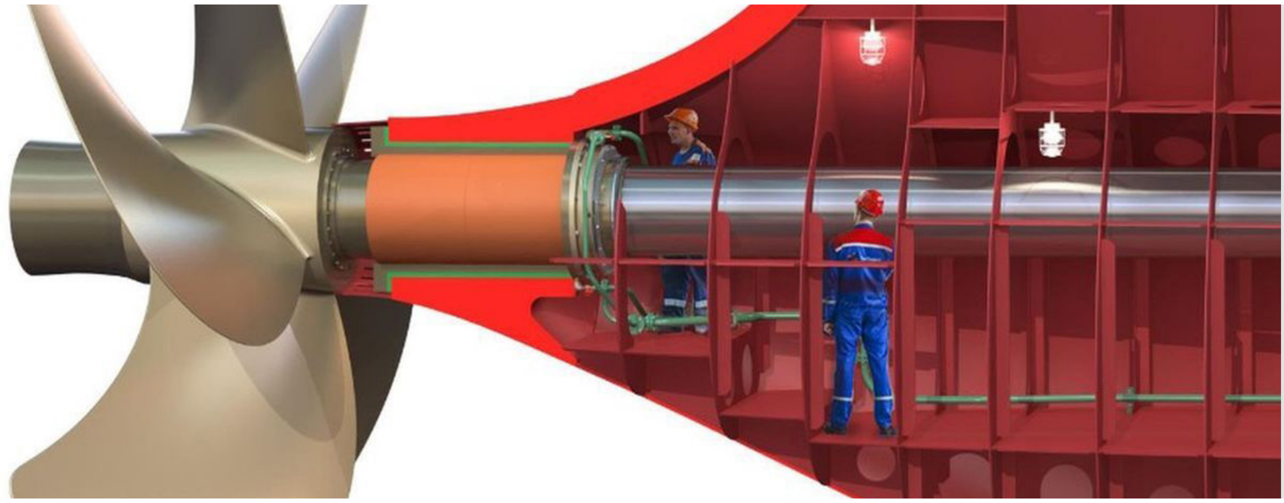
TECHNICAL NOTES

Water Lubricated Stern Bearing



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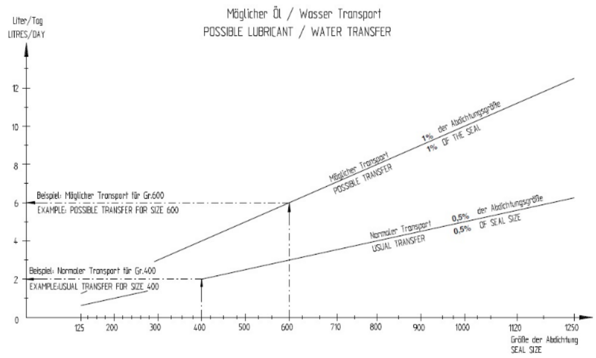
Abstract: Water lubricated stern bearing with Elastomeric/composite material and modified stern boss design having a single stern bearing eliminating the stern tube cylinder as seen in a conventional Oil lubricated bearing design. The new design gives advantage to lower/ same capital investment as compared to an oil lubricated conventional stern tube design during the new building stage. No risk of oil pollution from the stern tube/ seals, lower operating costs in terms of spares and fuel saving due to reduced frictional drag at the stern bearing, reduced time during the docking of the vessels and last but not the least the overhaul of the stern bearing can be done in floating condition to the extent of replacing the bearing.

Introduction

The majority of the vessels in the current age are installed with Oil lubricated stern tube having stern sealing system with lip seals at the aft when using mineral oil, The lip seals will always migrate some amount of fluid across the seals and there is some oil leakage that is happening even with a fully functional seal system, some makers like SKF have defined the rate of leakage across the seals which is shown in the graph below which is considered as acceptable depending on the shaft diameter.

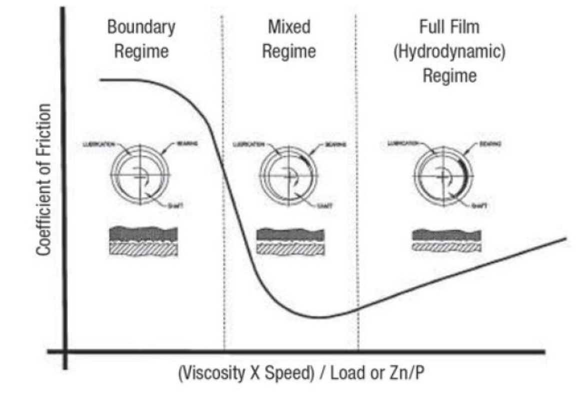
> *No risk of oil pollution from the stern tube/ seals, lower operating costs and fuel saving due to reduced frictional drag at the stern bearing*



In the current times where we have regulation and governing bodies like EPA (Environmental protection agency) of USCG who have defined the design of the seals to prevent oil /water interface with a neutral chamber of air (Air guard design) that are to be applied when using mineral oil as lubricant for the stern bearing or use of Environmentally friendly Lubricating oil (EAL) which is made from Plant based products.

Both the above designs of lubricating the stern bearing and seals suffer with complexities, it is worthwhile to mention deliberate pollution of EAL in the sea is still considered as pollution by the EPA (USCG) and have stated on means to adopt water lubricated stern tube in the future to eliminate Oil pollution at sea.

Water lubricated stern bearings have been in use since 1874 on SS Britanicia, using Lignum Vitae (type of wood) as bearing material having shaft protected with a bronze liner in the past. However, operation. This is due to the low viscosity of water. An interesting observation is that in the high-speed range, the water lubricated frictional force is lower than with oil. Once hydrodynamic operation is achieved, friction increases. However, the higher viscosity of oil results in greater shearing forces and higher friction than with water combined with the greater cooling effect of water results in a bearing running at a lower temperature as compared to that with oil lubrication. This is illustrated by a typical "Stribeck curve" which plots the coefficient of friction against the hydrodynamic parameter Zn/P. The curve is divided into three main lubrication regimes.



The Stribeck curve, illustrating the buildup of hydrodynamic film

In the first (Boundary) regime, direct contact exists between the shaft and the bearing resulting in high friction values. In this region of the curve, high bearing self-lubricity is of significant benefit. As the shaft speed increases, we move into the second (Mixed) regime of the curve where the hydrodynamic film starts to build and effectively "lift" the shaft from the bearing surface. The result is less shaft to bearing contact and friction drops rapidly. Further increases in speed take us into the third (Hydrodynamic) regime where the hydrodynamic film builds sufficiently to eliminate all direct contact. As speed continues to increase, friction begins to increase because of the increasing shear resistance imparted by the viscosity of the lubricant. The transition between lubrication regimes during operation of a bearing depends primarily on lubricant properties, velocity, and load. The curve profile and definition of transition points will depend on the bearing geometry, clearance ratio, self-lubricity of the bearing material and surface finish. A higher viscosity lubricant results in the generation of a hydrodynamic film at a lower shaft speed and effectively moves the transition points to the left. Increasing the viscosity, however, also increases the minimum operating coefficient of friction. Lowering the coefficient of friction of the bearing material results in decreased friction at shaft speeds below the point where full hydrodynamic operation occurs. The geometry of the bearing, and, whether the bearing is grooved also affects the curve. A continuous bearing surface without grooves allows the hydrodynamic film to build quicker than one with grooves. Hydrodynamic calculations show that the necessary speed to achieve

> *Water has its clear advantages as a lubricant with better heat transfer and thermal conductivity*

a hydrodynamic film is double that for an ungrooved bearing. Wet lubrication also has the added benefit of being able to carry away frictionally generated heat, the enemy of all bearings. This is especially significant with all the bearing makers because the low thermal conductivity of the material does not allow much heat to be dissipated through the bearing wall (metallic bearings have much higher thermal conductivity and can dissipate more heat through the bearing wall). Wet lubrication can be supplied by several methods varying in complexity and performance. There are drip feed systems (normally oil), which are appropriate for slow to intermediate speeds where heat buildup is not a concern. Bath systems are also used - where the bearing is fully or partially submerged in a limited quantity of lubricant. The limiting factor with bath systems is that the whole bath can become overheated if the assembly generates significant heat. A third method is a continuous flow of fresh cool lubricant from an external source, usually force-fed. This method is essential for applications such as marine propeller shafts, vertical pumps and turbines where high RPM and/or significant loads lead to levels of heat generation which cannot be dissipated by a bath of lubricant.

Wear: Wear is the destructive removal of material from contacting surfaces moving relative to one another. Wear can take several forms and, as a highly complex process, is difficult to predict.

i) Adhesive Wear: Adhesive wear occurs when minute peaks of two rough surfaces contact each other and weld or stick together, removing a wear particle. Adhesive wear of composite/elastomeric bearing materials is very minimal at normal temperatures and pressures but becomes the dominant wear mode as the operating temperature reaches maximum operating limits. The maximum operating temperatures of bearing materials are defined to try and avoid this mode of wear. The amount of adhesive wear is related to the friction between the two surfaces, the pressure on the working surface, and the type and amount of lubrication provided.

ii) Abrasive Wear: Abrasive wear involves the wearing of a softer surface by a hard particle. Examples are sandpaper or a grinding wheel (two body abrasion) or sand particles between a bearing and a shaft (three-body abrasion). Actual abrasive wear will vary with the quantity of abrasives present and with the size, shape, and composition of the abrasive particles. The best approach to minimizing abrasive wear is to reduce or

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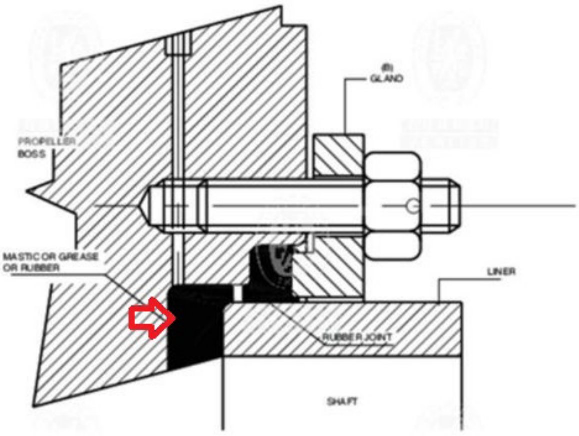
eliminate the quantity of abrasives by using filters or clean water injection. If this is not possible, then a satisfactory alternative for minimizing abrasive

wear is to have one surface very hard and the second relatively soft and compliant. Abrasive particles are allowed to be pushed into the softer surface and roll or slide through the contact area with very little damage to the shaft or bearing. The elastomeric nature of bearings facilitates abrasive wear resistance because the material flexes when it encounters abrasive particles. With shaft rotation the particles are moved along the bearing surface until flushed out through a lubrication groove. With more rigid materials the abrasive particles tend to become embedded in the material and may cause shaft wear. A continuous flow of fresh lubricant (as in a propeller shaft application), and grooves in the bearing, will help to flush out abrasive particles and reduce the amount of wear.

SYSTEM DESIGN: Companies Like Wartsila and Thordon have come out with a design concept where the stern tube is eliminated from the design and the stern boss casting is modified to accommodate the stern bearing. The design concept is the same however this is named the EVOTUBE by Wartsila and T-BOSS by Thordon.

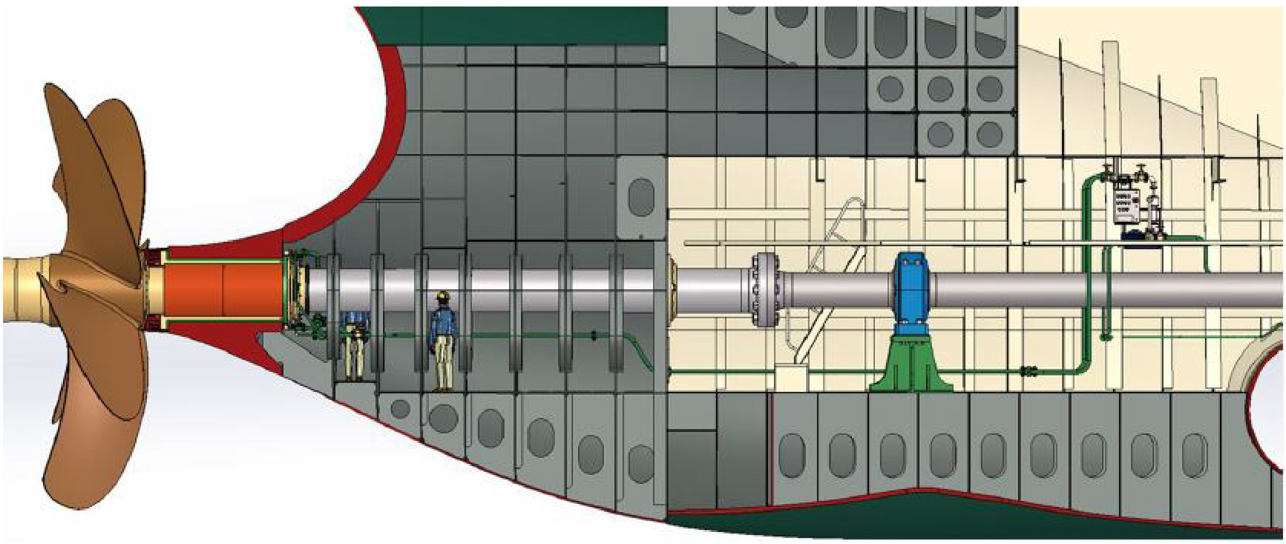
The design is such that it can be applied to the new building of the vessel. however, would face challenges to retro fit on existing vessel as the stern boss casting will need to be modified which is easily done at the new building. There is no limitation on the application of this design on the size of the vessel and can be applied to large as well as small vessels. The design of the water lubricated stern bearing would require only one forward seal in the engine room and sea water for lubrication of the bearing is fed from the engine room using a sea water treatment plant with filters to ensure uniform water quality. The shaft in way of the bearing is protected with a bronze/Epoxy liner which is sealed at the propellor boss as per drawing which is shown below.



The Liner can be inspected as required in service using an endoscope without the need to withdraw the tail shaft.

The aft of the bearing is installed with an inflatable seal which is retracted when in service and can be inflated on a stationary shaft when the vessel is afloat to allow persons to work from the engine room side on the forward seal overhaul or bearing overhaul.

The bearings installed are of the elastomer / composite type having a length to diameter ratio of 2:1 and bearing pressure of 0.6 MN/m2 whereas for class calculation it is considered at 0.55 MN/m2. The forward stern tube bearing is in the engine room like the other intermediate bearings of the shafting system. The aft stern tube bearing is installed with tapered key design in segments where the lower half is without grooves for better hydrodynamic



lubrication. The tapered key design enables the bearing segments to be withdrawn into the engine room during the overhaul of the bearing and thus enabling the renewal of the bearing without the need to withdraw the tail shaft.

The propellor weight will need to be supported using external hooks to enable withdrawal of the bearing.

The aft section of the ship will be designed with framing to enable access the stern bearing from the engine room, this design enables a shorter engine room where the main engine is positioned closer to the propellor thus giving the advantage of larger cargo space as compared to a vessel with conventional stern tube design thus improving the Energy efficiency design Index (EEDI) of the vessel.

The Design is patented by Wartsila additionally Thordon bearing Inc, ABS, the design house the Shanghai merchant Ship Design and Research Institute (SDARI) and the National Technical University of Athens have collaborated to bring forward the design, ABS has in principle granted approval to the design.

In addition, a shorted shaft line with one less bearing implies a different dynamic behaviour both in terms of torsional vibration as well as lateral vibration, known as whirling. To compensate for whirling, the intermediate bearing must also move afterwards so that the bearing span is acceptable as far as whirling vibration is concerned. While keeping the engine intact, the shorter shaft line would increase the torsional stiffness of this system and therefore the barred speed range would increase. The torsional stiffness of the system and therefore the barred speed range would increase towards higher rpm, together with higher torsional vibration amplitudes and stresses on the shaft line. Depending on the power train technical characteristics, such a reduction in length could cause the torsional stresses to exceed the IACS transient shear stress limit. Therefore, the installation of a torsional damper may be considered. The additional mass , inertia and damping coming from the torsional damper installation decreases the torsional vibration amplitudes, potentially to an extent that the barred speed range may be eliminated.

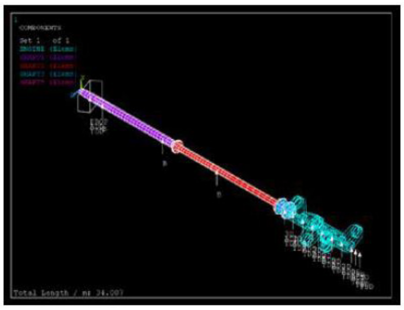
The elimination of the stern tube cylinder casting together with its piping and oil storage tank, opens up a new space which together with the trimming of the stiffeners around it, can create new chamber with sufficient size to allow for human inspector entry to a temporary opening in the bulkhead of the engine room. Such an inspection chamber would serve the purpose of shaft line inspection, seal inspection, aft bearing inspection and bearing replacement from inside the vessel, while the vessel is afloat. Such an installation would eliminate shaft line withdrawal, while the vessel is afloat.

SHAFT ALIGNMENT: ABS has studied the shaft alignment for a 3800 TEU container vessel which is described below. According to class rules the shaft line is subjected to satisfactory shaft alignment. An independent model is created for the original design using in house ANSYS customized finite element software. Based on the standard engineering procedure and practices for plan approval. The mathematical model is solved and post processed using static steady state analysis to determine the bearing reaction loads, the shaft inclination and relative misalignment angles between the shaft and the bearings. The modeling involves 6 degree of freedom pipe type of elements with shear effect to model intermediate and tail shaft, while the propellor is modeled as a thin disc with mass and inertia of propellor as per pertinent original plan. The engine crankshaft is geometrically modelled by meshing the crankshaft as a solid model and converting it into super elements using the sub structuring technique. The bearings are modelled as linear spring -damper elements with radial stiffness and damping and gap as clearance. All additional weights (crank throw weights, chain forces, etc.) are modelled as point loads. The bearing reactions are assessed based on the ABS Rule limits as well as the bearing maker limits. The same applies for shaft inclination as well as misalignment angles. More detailed bearing modeling on the Fluid Structure Interaction (FSI)between the oil or water film between the shaft and the bearing is performed to ensure that metal to metal contact of film breakage, which would cause increase bearing wear or even damage and failure, is avoided.

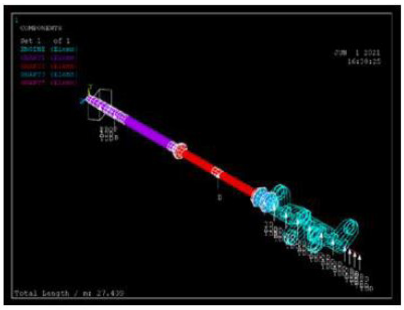
The aft of the bearing is installed with an inflatable seal which can be inflated on a stationary shaft when the vessel is afloat

To validate the proposed model the initial simulation is carried out for the original design which contains 2 stern tube oil lubricated bearings. The results are compared to those of the shipyard during the original plan approval process. Once the comparison is satisfactory the proposed modification is implemented on the model, namely, the shortening of the shaft and the move of the Main Engine afterward by 2m, as well as the elimination of the forward stern tube bearing and the replacement of the aftmost oil-lubricated bearing with a water lubricated bearing.

See figure shown below.



Original Design

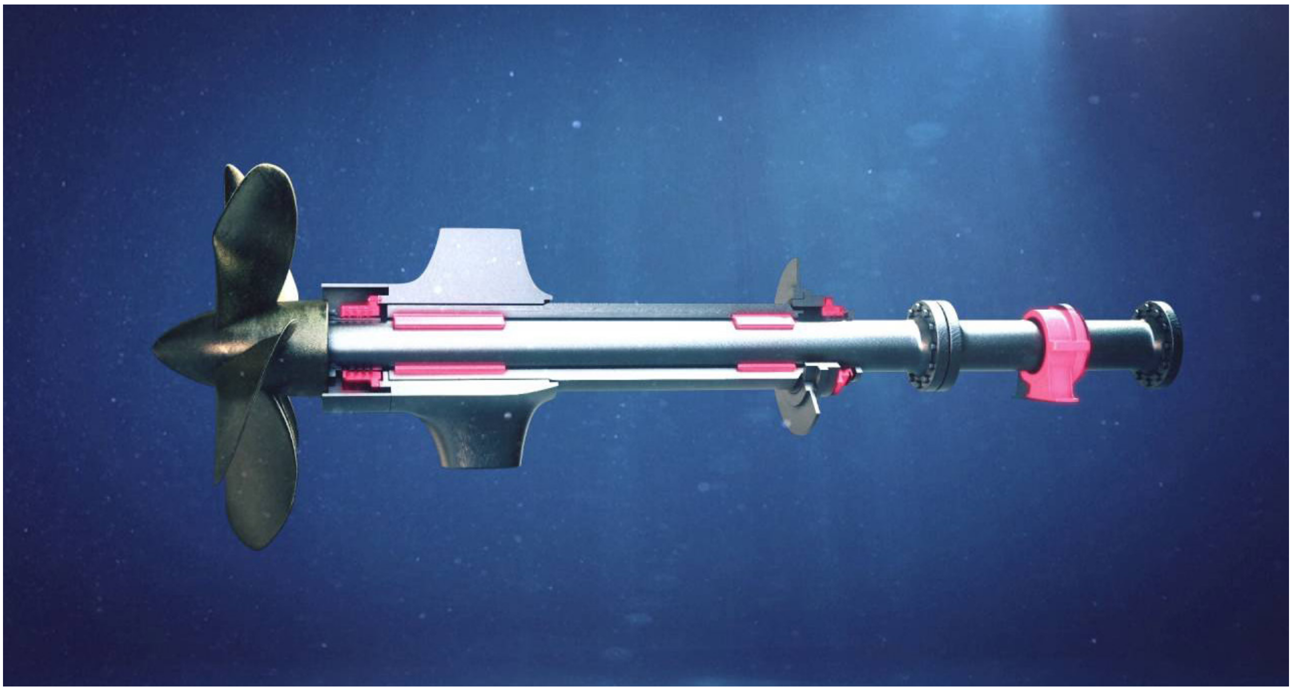


Shortened Design

The shaft alignment of the modified design was carried out including hot static and cold static as well as dynamic conditions including propeller loads for port and starboard vessels turning, for various propeller depths. Hull deflections were also considered in the analysis, as per the ABS Rule requirements for drydock or lightship, ballast and laden conditions. The bearing offsets both those of the shaft line and of the engine were optimized for best possible bearing load distribution using an ABS inhouse proprietary genetic algorithm for mathematical optimization. The result showed that for the modified design, all the bearing loads were well within acceptable limits including misalignment angles below IACS limits of 0.3 mrad. An FSI also confirmed that the water film pressure distribution was acceptable as well as the thickness under all loading conditions.

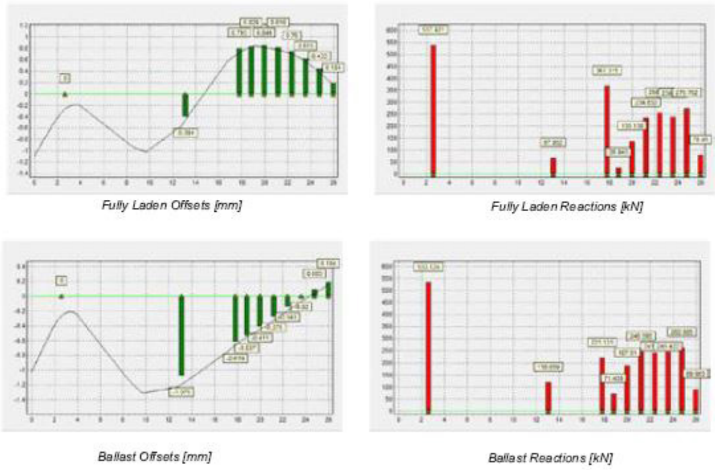
All in all, a total win-win design which should be our industry standard going forward

The criterion of the "Engine shear force -bending moment envelope" was also examined and was found to be within the engine maker's limits. To examine the possibility of accelerate bearing wear during the operational life of the vessel , a sensitivity analysis was conducted considering the bearing loads as well as misalignment angles for zero and maximum wear down of the aft most bearing (10-15 mm) , throughout the analysis the bearing load and misalignment angle values were found to be satisfactory both in terms of makers limit and ABS Rule limits.



- Shaft Alignment Optimization

	Optimised Drydock offsets [mm]	Fully Laden Hot Static Offsets [mm]	Ballast Hot Static Offsets [mm]
ASTB	0	0	0
I/M Bearing	0	-0.394	-1.075
M/E 1	-0.146	0.792	-0.619
M/E 2	-0.146	0.828	-0.536
M/E 3	-0.146	0.846	-0.41
M/E 4	-0.146	0.816	-0.276
M/E 5	-0.146	0.75	-0.143
M/E 6	-0.146	0.615	-0.02
M/E 7	-0.146	0.433	0.092
M/E 8	-0.146	0.184	0.184



All bearings are positively loaded and within their maker's limits

VIBRATION: The vibration assessment involved torsional, lateral, and axial vibration of the modified shaft line. The original design features a critical speed at about 47 rpm, where the tail shaft and intermediate shaft stresses exceeded the IACS continuous torsional stress limits and thus a barred speed range for torsional stress was imposed.

CONCLUSION: The design offers several advantages as compared to the conventional stern tube design with aft and forward seals. The most obvious and important ones being on the zero-oil pollution risk and ability to overhaul the seals, bearings without the need to the vessel to go on blocks and replacement of the bearing can be done in a time frame of 24 hours. The initial capital cost of the system is the same as that for a conventional oil lubricated stern tube system however there is a reduction in operating and overhaul cost as lesser number of seal parts are required for the system and saving in cost of the LO. There is also a fuel saving potential which is calculated basis the losses seen due to the frictional drag of the LO as compared to that of water for a water lubricated stern bearing. For a typical 950 mm shaft the annual saving in the fuel cost is 50000 US Dollars. Also, the composite/Elastomeric bearing that are produced have zero carbon emission during production which is not the case with white metal bearings. All in all, a total win-win design which should be our industry standard going forward.

ACKNOWLEDGEMENT

My colleague Mr Klaus Jorgensen (Senior marine Engineer) in our new building machinery team who is a Pioneer in this field and we have been working together for last 5-6 years on the concept of water lubricated bearing design.

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