METAL FATIGUE IN VESSEL STRUCTURES DUE TO DYNAMIC MOVEMENT

Ty A. Smith[1]
Kiril I. Tenekedjiev[2,3]

[1] Australian Maritime College, National Centre for Maritime Engineering and Hydrodynamics, e-mail: tasmith9@utas.edu.au
[2]Australian Maritime College, National Centre for Maritime Engineering and Hydrodynamics,
[3]Nikola Vaptsarov Naval Academy, Varna, Bulgaria
e-mail: Kiril.Ttenekedjiev@utas.edu.au; Kiril.Tenekedjiev@fulbrightmail.org

Abstract: Metal fatigue that occurs due to dynamic movement in a vessel. Two case studies using reliability engineering are diagnosed with modern technology, using vibration analysis and laser alignment on their propulsion systems. These types of ships have characteristics where both have different hull structure and material type; dynamic movement will occur no matter the features of the vessel. In both studies, it was found that the propulsion systems were not installed correctly and dynamic movement was detected which results in the re-occurring faults that the vessels were experiencing. The understanding of reliability engineering needs to be used when diagnosing metal fatigue.

Keywords: Reliability engineering, metal fatigue, Modern Technology, Dynamic Movement, Laser Alignment.

1. Introduction

The fatigue of metal occurs when a structure starts to fail from considerably low stress levels, or the metal is subjected to cyclic fluctuating stresses. Normally it occurs over long periods. Approximately 90% of all field failures of metal is due to fatigue [1]. One common source of metal fatigue is due to repeated variations of stress and excessive vibration. This is a significant re-occurrence in all types of drive train structures throughout the marine industry.

The major fault of any rotating machinery in the industrial world is the incorrect installation of the rotating equipment. An error can be captured using Vibration Analysis (VA) also known as condition monitoring. Using a transducer to obtain the complex frequency that a piece of machinery produces, then using the Fast Fourier Transfer(FFT) the faults can be read from a spectrum; these can be executed when following an ISO standard for example, ISO 10816. VA is used to detect many errors, a well-known error that occurs in a spectrum is misalignment from incorrect installation [2]. The basic understanding of alignment is needed, this is described when two or more shafts centre lines must be within a tolerance of each other in both offset and angular parameters at the coupling. These tolerances are checked in both the horizontal and vertical position when taking measurements [3]. Aligning of two shafts could be done by eye or a straight edge, if you want the life span of any machinery to last longer more precise tooling should be used. To gain these tolerances a dial indicator or laser is needed to achieve the tolerance within an acceptable range. Conducting an alignment on a vessels propulsion system can be carried out with a well-known, yet older technology such as one or two dial indicators which would carry out the alignment between couplings to ensure these types of machinery are positioned correctly. Examples of such machinery are; steady bearings, gearboxes and motor/engines.

Traditionally a piano wire is used to locate the centre line of the machinery and the final coupling of the drive train; for example, a stern tube or Z-drive thruster. Piano wire and dial indicators have been out of date since the introduction of laser technology which is today's modern technology. The laser has many other features that help perfect any alignment, such as: Drive train, Straightness, Offset, Off Line 2 Run (OL2R) and Geometric alignments. Another laser that is hardly used in the marine industry in Australia is a 3D laser tracker. The tracker involves a laser mounted on a tripod and a prism. The laser tracks, the reflected lens when placed or dragged over a machined surface. Two case studies are examined in this paper, one being a ductile vessel (Vessel x) and the other being a brittle vessel (Vessel y). Shaft alignment of both vessels has been undertaken using a combination of the above methods. These measurements are taken to determine whether dynamic movement is a primary cause of metal fatigue in ship structures.

2. Case Studies

2.1. Case Study 1 (Vessel x)

Vessel *x* is a 55m planning monohull, that has two high revving 3500hp diesel engines with a down angle propeller shaft (see Figure 1). Vessel *x* had three major faults that were occurring, identified by the vessels operator:

- Bearings in the stern tube were failing which would in turn cause the shaft to bind up.
- The vessel experiences excessive vibration over 1450rpm.
- Cracking was observed occurring between the engine beds, frame 33 and at the king post.

2.1.1 First Test

Vibration analysis was executed at three different rpm ranges on the drive train at the following locations.

Horizontal Drive End = HDE
Horizontal None Drive End = HNDE
Axial = A.

The test points were attained at the following positions:

- Engine: HDE and A
- Gearbox Input Shaft: HDE, A and HNDE
- Gearbox Output shaft: HDE, HNDE and A
- Stern Tube HDE and A

The three different rpm ranges tested were at idle at 1000rpm, 1450rpm and full speed at 1900rpm. The readings showed that there was an increase in misalignment, this was a good indication of dynamic movement occurring, as shown in figure 2.

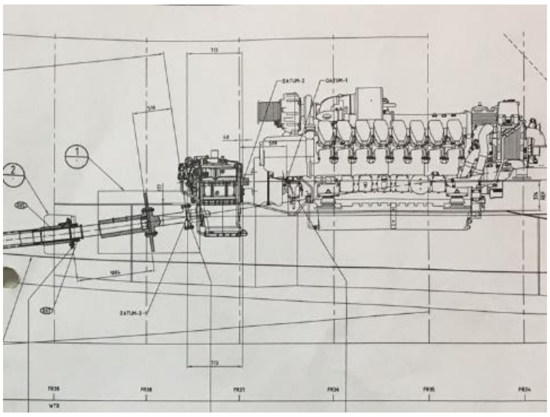
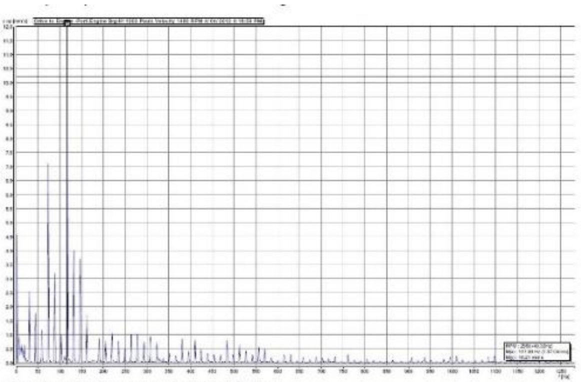


Figure 1 - Drive train configuration



Above is a vibration spectrum indicating findings.

Figure 2 – VA misalignment spectrum

2.1.2 Second Test

Following the first test there was recorded data of progressive movement present, therefore the next step was to capture the dynamic movement. The lasers were set up between the engine and gearbox (see figure 3). A misalignment of greater than 20mm, which can be recorded by the detectors was present. Due to the excessive movement, a third test was required. The lasers were set up from the engine bell housing of the engine to the stern tube bulkhead in the engine room. At the highest rpm of the engines (1900RPM), a movement of roughly 35mm was measured as it was outside the detectors, a rule was used.

2.1.3 Third Test

Due to time restraints, only the starboard drive train could be tested. Adjustable rollers had to be constructed so that the centre of the stern tube bearing could be found. When the centreline of the bearing was found, the rollers were then locked off to simulate a steady bearing for the propeller shaft (Both the propeller shaft and the gearbox output shaft should rotate together for precision alignment when lasers are being used). The alignment between the shaft and gearbox were greater than the allowable tolerances. The engine to gearbox check was just within tolerance. On the inspection of the soft footings on the engine, the front left mount was loosened, a 14mm gap was present. Due to the high ductility of the vessels hull, a soft footing of this magnitude would put high stress on both the engine bed and hull of the vessel (see figure 4).

Figure 3 – Dynamic movement set up on Vessel *x*

Figure 4 – Shaft alignment on Vessel *x*

2.1.4. Discussion

The vibration readings that were recorded were over 12mm/s rms when the vessel exceeded 1450rpm. At the full speed of the vessel (1900RPM), the vibration readings were much higher, as the dynamic movement readings attempted were not achievable a shaft alignment was carried out to reduce the misalignment around 1450rpm +/- 5%. When undertaking the alignment, it must be ensured that the couplings are in tolerance and that all soft footings are removed. As the results of the sea trial following the realignment, the outcome was more than satisfied as the vessel could now achieve higher rpm range without vibrations throughout the vessel. The vessel is now achieving 1600rpm cruise speed this an acceptable change from 1300rpm. The vibration was still present from 1600-1900rpm due to the dynamic movement on the engines.

Recommendations for Vessel *x*:

- Put in place a 3-monthly VA routine to monitor the misalignment that is occurring.
- Rewrite the alignment procedure.
- Implement dynamic movement alignment procedures.

Continuous work has been done on this vessel since this job; Vessel *x* went through a remediation process. The ship was then strengthened by inserting four longitudinal beams through the engine room. Due to these four beams in place the cracks at frame 33 and the king posts have shown no signs of growing. Although forward of the vessel at frame 25 cracking has commenced. This will only keep occurring as the major fault of this ship is due to the incorrect installation and dynamic movement that is taking place in the drive train of the vessel.

2.2. Case Study 2 (Vessel y)

Vessel *y* is a 58m displacement hull that has two medium revving diesel engines at 3500hp with a *Z*-drive thruster (see figure 5).

Vessel *y* had four major faults that were occurring identified by the vessels operator:

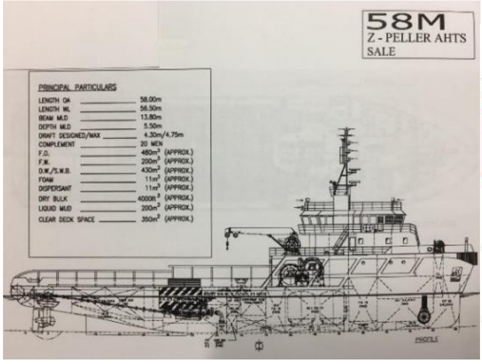
- High vibration that was felt throughout the ship at most rpm ranges.
- Bolts were loosening on bearing 1 on both Starboard and Port first intermediate shaft.
- Welds were cracking on the base plates of bearing housing 1 on both Starboard and Port intermediate shaft 1.
- Overheating on bearing 6 both Port and STBD intermediate shafts.

2.2.1 First test

Vibration Analysis was taken at three different rpm ranges. The same keys from 2.1.1 apply below. The test points were taken at the following positions:

- Engine: HDE and A
- Clutch box input shaft: A, HDE and HNDE
- Clutch box output shaft: A, HDE and HNDE
- Intermediate shaft 1: HDE, A, HNDE and A
- Intermediate shaft 2: HDE, A, HNDE and A
- Z-drive thruster: A, NDE and HNDE.

The three different readings were at idle 450rpm, 600rpm and full speed of 750rpm. Readings showed that there was an increase in misalignment, this was a good indication of dynamic movement occurring. Offset and angular misalignment was observed at each coupling and Cardan shafts (see figure 6, highlighted in red).



PRINCIPAL PARTICULARS

LENGTH OA	58.00m
LENGTH WL	56.50m
BEAM MLD	13.80m
DEPTH MLD	5.50m
DRAFT DESIGNED/MAX	4.30m/4.75m
COMPLEMENT	20 MEN
F.O.	$480m^3$ (APPROX.)
F.W.	$200m^3$ (APPROX.)
D.W./S.W.B.	$430m^3$ (APPROX.)
FOAM	$11m^3$ (APPROX.)
DISPERSANT	$11m^3$ (APPROX.)
DRY BULK	$4000ft^3$ (APPROX.)
LIQUID MUD	$200m^3$ (APPROX.)
CLEAR DECK SPACE	$350m^2$ (APPROX.)

Figure 5 -Vessel *y*

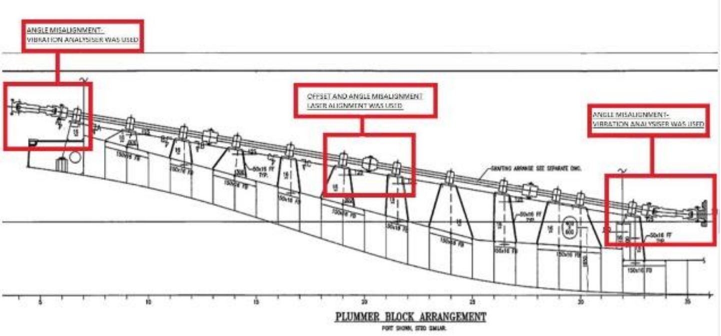


Figure 6 – VA misalignment position

2.2.2. Second test

Executed alignment measurements throughout the drive train both Port and Starboard. Figure 7 is the reading from the coupling between the two intermediate shafts. This was the indication that bearing six would be over heating due to the misalignment. It was revealed that bearing 6 was a thrust bearing which was not designed to hold an 8m shaft on a 7 degree down angle. Offset brackets are used for aligning Cardan shafts, although this couldn't be done due to the intermediate shafts being at an angle of 7 degrees, therefore a third test was required.

2.2.3 Third test

A 3D laser Tracker was used to undertake this test due to complexities of the drive train system. To begin the alignment the engine block was used as the reference, then the tracker zeroed in the coupling on the clutch box, both intermediate shafts and *Z*-drive thruster. The distance between the clutch box and *Z*-drive thruster was 16m apart. The offset misalignment was 13.7mm and the angle angular misalignment was 6.5mm. The two intermediate shafts had high stress applying to the bearing housing as these shafts carried the misalignment (see figure 8 and 9).

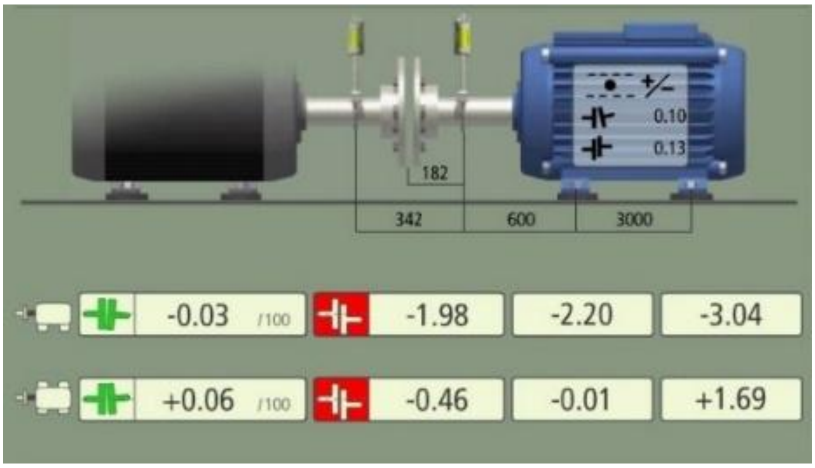


Figure 7 – Shaft alignment result

Figure 8 – 3D laser

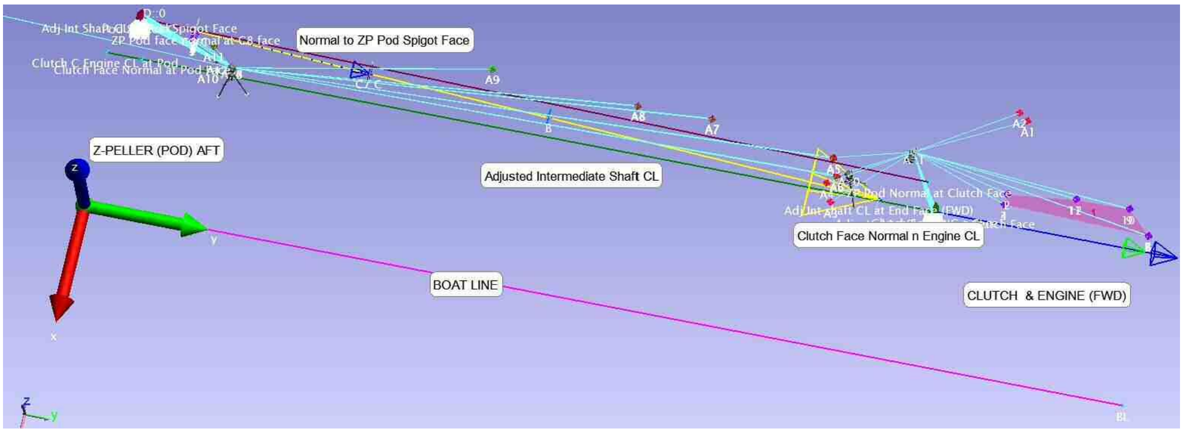


Figure 9 – 3D laser measurement results

2.2.4 Discussion

The vibration readings where over 12mm/s rms through the drive shaft. The starboard bearing 1 on the intermediate shaft was 23mm/s rms due to one loose bolt and cracked welds in the base plate. The intermediate shafts needed to be redesigned between the clutch box and the Z-drive thruster. The Z-drive also required repositioning to be parallel to the clutch box. Neither of these modifications were an option at the time. With the 3D tracker, an accurate calculation could be made on the Cardan shafts. After looking at the specification of the Cardan shaft, the shaft works highly within the spline tolerance between the two universal joints. This spline allows for the absorption of axial loads when the couplings are not parallel with each other. The Cardan shaft was on the limit of its design for axial movement. Due to the maximum limit on the Cardan shafts when the dynamic movement occurs, the Cardan shaft then became overloaded. Causing the faults in bearing 1 such as the bolts coming loose and cracking welds in the base plate. This explains the high vibration throughout the whole vessel.

Recommendations on Vessel *y*:

- Redesign the drive train between the clutch box and z-drive thruster as this will fix bearing faults, specially bearing 1 and 6.
- Introduce new alignment procedure with new drive train to allow for dynamic movement. The failure of the high vibration throughout the vessel will be minimised

3. Conclusion

Dynamic movement caused the same faults that occurred in both cases due to incorrect installation of drive train. The studies show that these faults are independent of the ductility of the hull. Further studies will have to be done to compare the present procedures of fault finding metal fatigue to the procedures of introducing reliability engineering tooling, for example the use of vibration analysis and dynamic movement into detecting the cause of metal fatigue.

Acknowledgments

This work has been supported by Northern Maritime and Industrial Services P/L. The author would also like to thank all the experts for providing their opinion and knowledge in this study.

References

[1] Lisson, D.L.(2017) Materials Technology (Chapter 2 Mechanical Properties (10th edition), Australian Maritime College, National Centre for Maritime Engineering

[2] Mobius Institute (2006) Vibration Analysis – Training manual (Category II)

[3] Piotrowski, J.(2006) Shaft Alignment Handbook (3rd Edition), Parkway NW, CRC Press.