ICMES 2003
INTERNATIONAL COOPERATION ON MARINE ENGINEERING SYSTEMS
Paper presented at the 9TH International Conference on Marine Engineering Systems
at the Helsinki University of Technology (HUT) Ship Laboratory and on board MS SILJA SERENADE
19-21 May 2003

Solutions to Failures of Shaft Bearings on Vessels Fitted with Z-Drive Thrusters No. 27

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ABSTRACT

Z-Drive thrusters with lineshafts are installed on a number of ships, including Offshore Supply Vessels, Passenger-Car Ferries, and Tug Boats. A typical arrangement has Cardan shafts, with universal joints, that couple the Z-Drive and diesel engine to a lineshaft. Roller element bearings are used to support the lineshaft. On some systems a stub shaft, supported by two roller element bearings, is used to couple the main engine to the forward Cardan shaft. A number of bearing failures have occurred on these systems in recent years. This paper discusses the results of investigations to determine the source(s) of failures, and the corresponding modifications conducted to prevent further failures. Alignment and vibration measurements, theoretical analyses and criteria are discussed. An example alignment procedure is also provided.

INTRODUCTION

Propulsion systems using Z-Drive Thrusters that are directly driven by diesel engines are a popular choice for ship owners and operators. These systems do not require rudders while offering good maneuvering characteristics at a relatively low capital cost. In recent years a number of failures have occurred to lineshaft bearings on these systems, which have resulted in significant costs to shipyards, owners, and propulsion system suppliers. In several cases the initial failure(s) were attributed to improper installation, bearing selection, or insufficient structural support stiffness. However, after further investigation it was revealed that the lineshaft alignment condition was the root-cause of failure(s). This paper presents the results of failure investigations, and provides design and alignment guidelines to prevent such failures. The investigations included the development of comprehensive alignment and vibration criteria, a review of the design arrangements, theoretical analysis and on-site measurements.

PROPULSION SYSTEM ARRANGEMENT

A typical shaftline arrangement has Cardan shafts with universal joints that are used to couple the Z-Drive thruster and diesel engine to a lineshaft, as shown in Figure 1. Roller element bearings are used to support the lineshaft. A roller bearing with axial load capability is fitted to the lineshaft and stub shaft. Systems with three to seven lineshaft bearings were investigated, some fitted with stub shafts, some without. Those with stub shafts have two bearings that are used to resist the bending moments applied by the Cardan Shaft [1]. Systems without the stub shaft have bearings that are integral to the torsional coupling. Photos 1 and 2 illustrate a damaged lineshaft bearing.

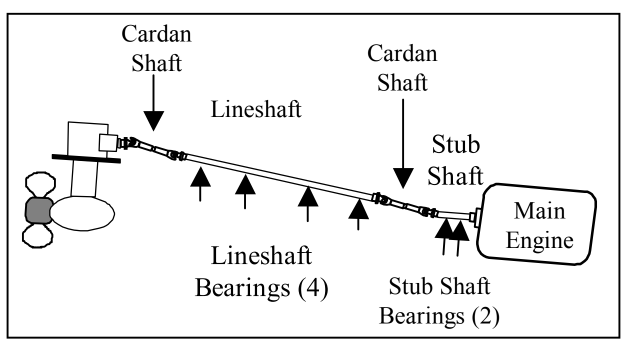


Figure 1 Schematic of Propulsion Shaftline

Photo 1 Lineshaft Bearing – Bearing Cap Removed
(Note: Left side shows over-heating has occurred)

Photo 2 Inside of Bearing Cap

ALIGNMENT CRITERIA

Propulsion suppliers typically provide procedures that call for alignment using "pointers" mounted on the aft and forward ends of the lineshaft and the Z-Drive thruster and engine output shafts. The primary purpose of the pointers is to ensure the Cardan Shaft Yoke angles are equal. They also align the main engine with the Z-Drive, and the aft and forward most lineshaft bearings, as shown in Figure 2. A typical tolerance for alignment of the pointers is ±1 to 3 mm (runout). The stub shaft is aligned within the tolerances specified by the main engine and torsional coupling manufacturers, accounting for differential thermal growth. A typical allowance for the thermal growth is 0.25 mm, with an allowable radial offset of about ±0.5 to 1.0 mm.

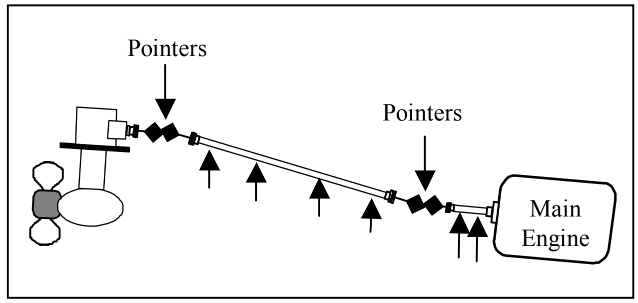


Figure 2 Alignment of Main Engine and Z-Drive using Pointers

Alignment procedures for the inner lineshaft bearings, and the stub shaft bearings, provide limited direction, such as a reference to a jack-up load test. Therefore, more comprehensive alignment criteria were defined to help establish the root cause of failure(s) and to provide guidance for other similar installations. These included allowable bearing loads, shaft bending stress, and bearing offsets. Maximum allowable bearing loads specified by the manufacturers, are in the range of 9 to 27 kN (2000 to 6000 lbs), which far exceed the applied loads on the bearings, and are therefore not a concern. Minimum bearing loads are typically in the range of 2 to 6 kN (450 to 1350 lbs). Low loads on grease filled roller bearings can result in damage to the bearing because the roller elements may not fully rotate with the shaft, resulting in a condition commonly called “skidding”. In a number of designs it is impractical to achieve the minimum load requirement for every lineshaft bearing. Therefore, a minimum load of 1 kN (225 lbs) is recommended if possible, and zero or negative (upward) loads should be avoided. A maximum allowable static shaft bending stress of 13.8 MPa (2000 psi) is recommended [2]. A maximum bending stress of less than 7 MPa (1000 psi) is typically achieved if the criteria for bearing offsets and loads are achieved. Allowable bearing offsets were established from estimates of the centrifugal forces and from on-board measurements. Experience has shown that the lineshaft support bearings are typically required to be aligned within less than 0.13 mm (0.005") of their prescribed position, to satisfy all the alignment criteria. To achieve this, the strain gauge technique is recommended to be included in the alignment procedure [3,4,5,6,7].

Table 1 summarizes guidelines for alignment criteria. The manufacturer should be consulted for each case. In all of the failure investigations conducted one or more of these alignment criteria were not satisfied.

Table 1 Alignment Criteria Guideline

Description	Value
"Pointer" Offset (Alignment of Main Engine to Z-Drive)	±1 to 3 mm (0.040" to 0.120")
Maximum Stub Shaft Offset From Main Engine (Warm)	±0.5 to 1.0 mm (±0.020" to 0.040")
Max. Load on Bearings	9 to 27 kN (2000 to 6000 lbs)
Min. Load on Bearings	1 to 3 kN (225 to 675 lbs)
Max. Shaft Bending Stress	13.8 MPa (2,000 psi)
Max. Bearing Offset from Prescribed Position	±0.13 mm (±0.005")

VIBRATION CRITERIA

Shaft Torsional Vibration

Allowable torsional alternating stress is in the range of 25 MPa (3,600 psi) [8,9,10]. Alternating shaft torque corresponding to these stresses can be as high as 80% of the maximum continuous rated torque. Since the Cardan shaft produces lateral forces on the shaftline from the applied torque [1], a more restrictive criterion is recommended. Figure 3 illustrates shaft torsional vibrations measured in the middle of a lineshaft, which were found to be excessive. Torsional natural frequencies that are equal to 1.0, 1.5, and 2.0 times the maximum rated shaft speed are recommended to be avoided. The allowable alternating torque is recommended to not exceed 25% of the full power mean torque [8].

Shaft Lateral Vibration

To avoid excessive lateral vibrations the first natural frequency of vibration is required to be at least 230% of the maximum rated shaft speed, because the Cardan shaft induces a dynamic bending moment on the lineshaft [8]. This moment is proportional to the shaft torque, is at a frequency equal to twice the shaft speed, and results in dynamic lateral loads (vertical and horizontal) on the lineshaft bearings [1]. These loads, and the relatively high shaft speeds, impose significant constraints on the shafting design. More bearings are typically required than conventional shafting systems to increase the lateral natural frequency to an acceptable value. This results in low loads on the bearings and a relatively stiff shafting system which require the lineshaft bearings to be positioned very accurately. Figure 4 shows the shaft orbit measured prior to and after a re-alignment of a lineshaft, which illustrates how a mis-alignment can result in excessive lateral vibrations. The top plot in Figure 4 illustrates the shaft movement exceeding the specified bearing clearance. For the shaft systems examined, a critical shaft speed for 2nd order lateral vibrations occurs within the operating shaft speed range if one of the lineshaft bearings is removed. Therefore, resonant lateral vibrations may occur if a lineshaft bearing is unloaded.

Allowable vibrations on the lineshaft bearing housings vary from as low as 10 mm/sec to as high as 32 mm/sec (RMS 1 to 1 kHz) depending upon the reference cited and the application [11,12,13,14,15].

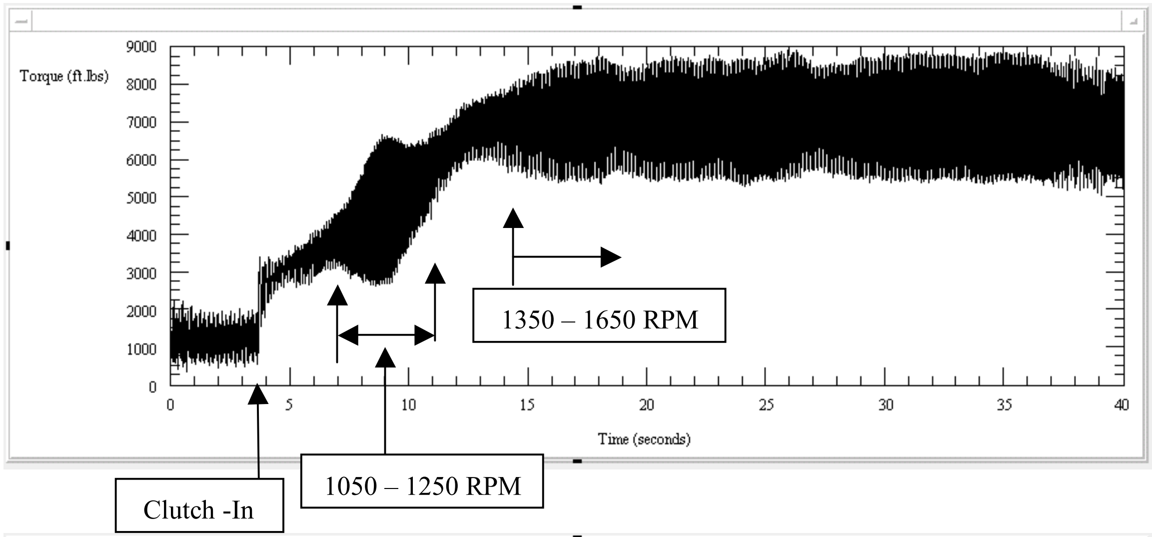
Table 2 summarizes guidelines for the torsional and lateral vibration criteria. The manufacturer should be consulted for each case.

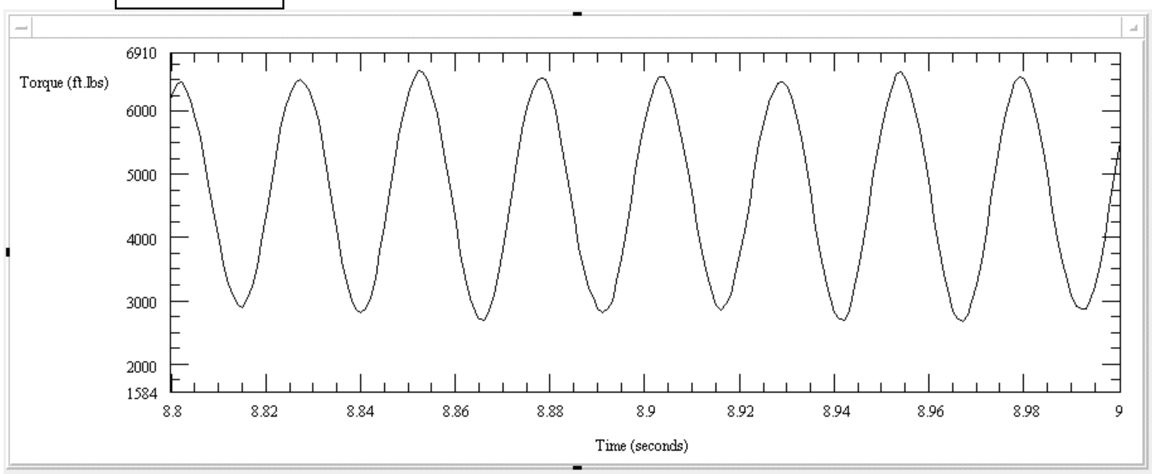
Table 2 Vibration Criteria Guideline

Torsional Vibration
Alternating torque to be less than 25% of full power mean torque.
If feasible, avoid a shaft critical at maximum rated shaft speed for 1.0, 1.5, and 2.0 order vibrations.
Lateral Vibration
1st Natural Frequency > 230% of Max. Shaft Speed
Bearing Housing Vibrations < 10 to 32 mm/sec

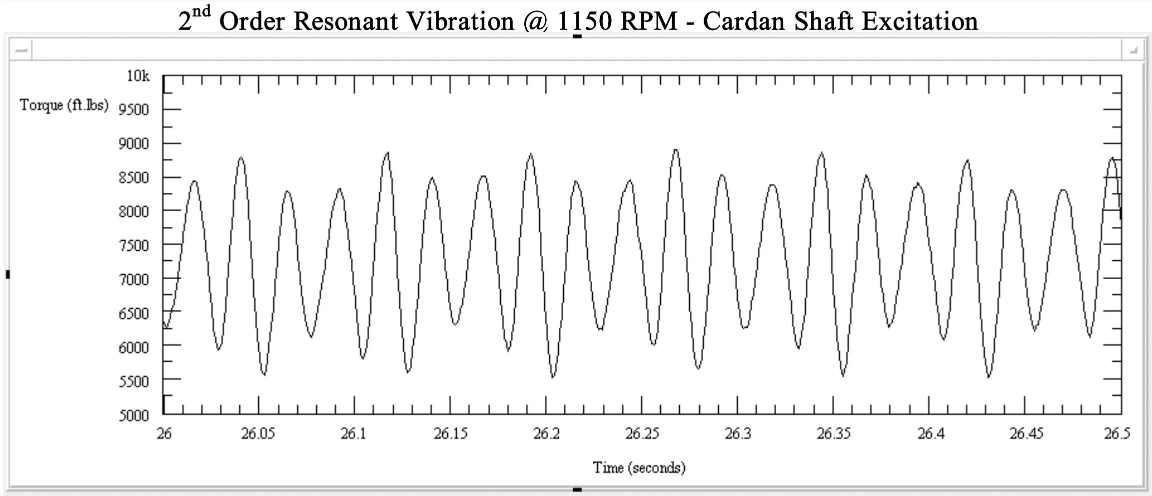
THEORETICAL ANALYSIS

Finite Element Analyses (FEA) of the shafting systems were used to calculate the theoretical alignment condition, as well as the torsional and lateral (whirling) vibration characteristics. Models were constructed of shaft elements of uniform section, and concentrated springs (bearings). Alignment models were used to calculate the bearing reaction influence numbers, bearing loads, and the shaft stresses and deflection, for a given set of bearing positions. The bearing reaction influence numbers represent the change in the load of each bearing as a result of raising (or lowering) any one bearing. The lateral and torsional models were used to calculate the natural frequencies of vibration, and if necessary a forced-damped response. The theoretical calculations were compared to the measured results to assist in the development of a least-cost practical solution to the bearing failures, and to assess the shaftline arrangement.



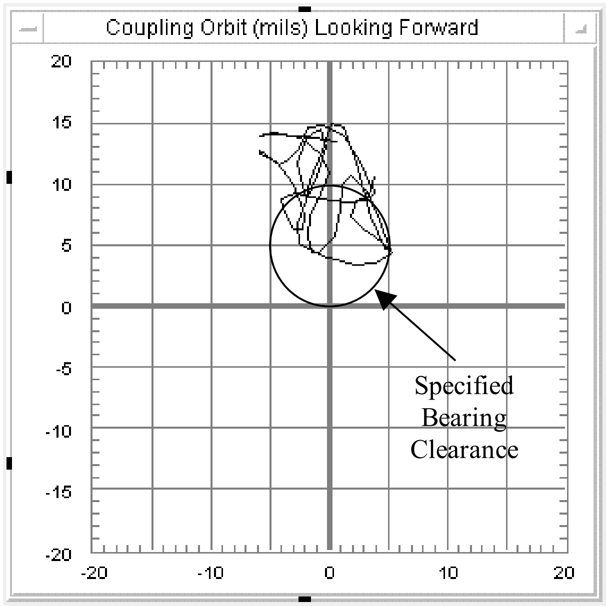


2nd Order Resonant Vibration @ 1150 RPM - Cardan Shaft Excitation

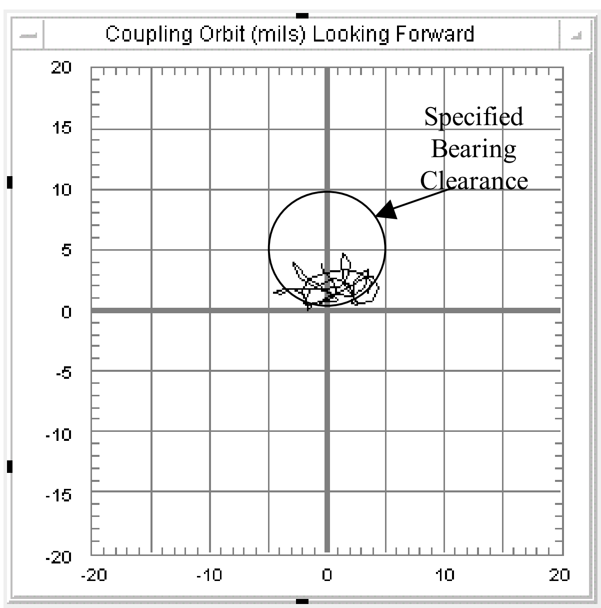


1.5 Order Resonant Vibration @ 1650 RPM – Engine Excitation

Figure 3 Measured Resonant Torsional Shaft Vibrations during Fast Acceleration (Idle to 1650 RPM)
Note: Loud Noise Heard from 1350 to 1650 RPM



1250 RPM Prior to Re-Alignment



1250 RPM after Re-Alignment

Figure 4 Measured Shaft Orbit at of Stub Shaft
(Conversion Note: 1 mil = 0.001" = 0.0254 mm)

ALIGNMENT MEASUREMENT TECHNIQUE

The strain gauge technique was used to measure the alignment condition of the lineshafts [3,4,5,6,7]. The primary advantages of using this technique for these shafting systems are:

- Accurate measurement of bearing loads (typically better than ±50 lbs).
- Measurement of horizontal and vertical bearing loads.
- Lineshaft and bearings remain connected.
- A line of sight is not required.
- Shaft bending stress is directly measured.
- Shaft hog and sag are easily determined.
- After the gauges are installed, the alignment condition can be re-measured within minutes.
- Bearings can typically be positioned to within ±0.13 mm (±0.005") of the prescribed positions.

Jack-up load tests were also conducted on selected bearings to provide an independent check on the strain gauge results [6].

CASE STUDIES

Three case studies are presented below, which present failures, solutions, and recommendations.

Case 1: Offshore Supply Vessel; Three Lineshaft Bearings, No Stub Shaft

The propulsion system has two thrusters (port and starboard), each driven by a Cat 3512B diesel engine, with a rating of 1120 kW (1500 HP) at 1600 RPM. Each propulsion shaftline is about 4700 mm (15 ft.) long and is supported by 3 lineshaft bearings. The forward lineshaft bearing also acts as a "fixed" bearing to carry axial loads from the shaftline. The vessel was delivered in early 2001, after which a number of damages occurred to the propulsion lineshaft bearings. The following performance items related to the propulsion shaft were noted:

- **July 2001**: High vibration at 1650 RPM.
- **August 2001**: Middle bearing on starboard shaft was seized. The port shaft forward bearing had a crack in roller cage. Both bearings were replaced.
- **September 2001**: Middle bearing on port shaft running at 220°F.
- **October 2001**: Middle bearing on port shaft running hot and damaged. Bearing was replaced.
- **January 2002**: Aft bearing failed on starboard shaft. Shaft journal damaged. Shaft and bearing replaced.

An investigation was undertaken to determine the cause of the bearing failures after the last reported failure in January 2002. It involved a combination of theoretical modeling, design review, assessment and on-site measurements. Alignment measurements were taken using the strain gauge technique and jack-up load method. The alignment condition was found to be unacceptable on both the port and starboard shafts, with the middle bearing top loaded and the shaft bending stress excessive. This condition can result in excessive lateral vibration, and high dynamic loads on the bearings, and was concluded to be the root cause of the bearing failures experienced.

A realignment of the propulsion shafts was conducted from January 14 to 15, 2002. Table 3 presents the measured bearing loads. Figure 5 shows the results of the jack-up load tests on the No. 2 and 3 port shaft bearings prior to realignment, which are similar to the strain gauge results. To obtain an acceptable alignment condition, the middle bearings on the port and starboard shaft were raised 2.2 mm and 2.8 mm, respectively. The maximum measured shaft static bending stress was reduced from 17.2 MPa to 1.2 MPa as a result of the realignment. Based upon the bearing load measurements and the influence numbers, it was estimated that the shafts were straight aligned to less than ±0.10 mm.

Table 3 Lineshaft Bearing Loads: Case 1

	Port Shaft Bearing Load (kN)			
	Initial		Realigned	
Bearing	Vertical	Horiz.	Vertical	Horiz.
No. 3	6.5	0.2	3.5	-0.1
No. 2	-3.3	-0.1	2.5	0.0
No. 1	6.4	-0.1	3.6	0.1
	Starboard Shaft Bearing Load (kN)			
No. 3	6.0	-0.6	3.6	-0.5
No. 2	-2.1	1.3	2.6	0.8
No. 1	5.6	-0.7	3.3	-0.3

Conversion Note: 1 kN = 225 lbs

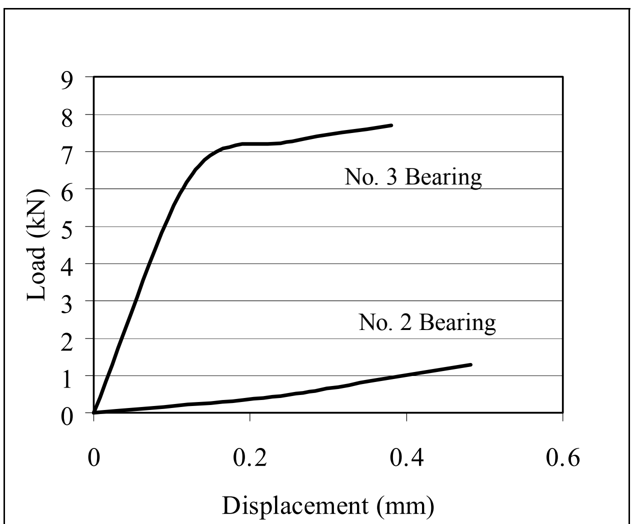


Figure 5 Jack-Up Load Tests Port Shaft: Case 1
(Prior to Realignment)

The bearing reaction influence numbers indicated that the shafting system flexibility was acceptable, such that the tolerance for shaft alignment during operations and construction was reasonable, as shown in Table 4. For example, a movement of 0.50 mm of No. 2 bearing results in a 1.2 kN change in load, which is about 50% of the static load on the bearing. Another indication of the flexibility of a shaftline is the ratio of the distance between shaftline bearings to the shaft diameter, which is 15 for this system. The minimum ratio between two successive bearings is recommended to be 12 and the maximum is 22 [6].

Table 4 Bearing Influence Numbers: Case 1
(Change in Load by Lowering Bearing)

	Reaction Influence (kN/mm)		
Bearing	No. 1 (Fwd)	No. 2	No. 3 (Aft)
No. 3 (Aft)	-0.60	1.20	-0.60
No. 2	1.20	-2.39	1.20
No. 1 (Fwd)	-0.60	1.20	-0.60
	Reaction Influence (lbs/mil)		
No. 3 (Aft)	-3	7	-3
No. 2	7	-14	7
No. 1 (Fwd)	-3	7	-3

No lineshaft bearing failures have occurred since the realignment. Two other subsequent vessels of the same class included the strain gauge alignment technique as part of the alignment procedures during construction. No lineshaft bearing failures have occurred on either of these vessels.

Case 2: Thruster Tug; Seven Bearings, No Stub Shaft

The propulsion system has two thrusters (port and starboard), each driven by a MTU / Detroit 16V-4000 diesel engine, with a rating of 2254 kW (3020 HP) at 2000 RPM. Each propulsion shaftline is about 9754 mm (32 ft.) long and is supported by 7 lineshaft bearings (numbered forward to aft). The aft lineshaft bearing also acts as a "fixed" bearing to carry axial loads from the shaftline. The vessel was delivered in the August of 2001, shortly after which a number of damages occurred to the propulsion shaftline bearings. The following performance items related to the propulsion shaft were noted:

- **September 2001**: The No.1 bearing (forward bearing) on the port shaft was damaged.
- **October 2001**: The No.1 and No. 4 bearings on the port shaft failed. After replacement they were reported to be running hot, and the engine vibration was high.
- **November 2001**: The port Cardan shaft was replaced, and a number of bearing positions were adjusted by shims.

A number of bearings continued to run hot, and further failures were expected. An investigation was undertaken to determine the cause of the bearing failures, and to implement the solutions in December 2001. The alignment condition was measured using the strain gauge technique and jack-up load method. The alignment condition was found to be unacceptable on both the port and starboard shafts. Two bearings were top loaded on port shaft (No. 4 and No. 6), and three bearings were found to be top/unloaded on the starboards shaft (No. 3, 5, and 6). The No. 7 bearing (fixed) on the port shaft was not parallel to the shaft and required shims on one end to align it the shaft axis. To achieve an acceptable alignment condition, the No. 6 bearing was raised 0.75 mm, and the No. 5 bearing was moved 0.5 mm outboard, on the port shaft. The No. 6 and No.5 bearings on the starboard shaft were required to be raised 0.75 mm and 0.50 mm, respectively. Table 5 presents the measured bearing loads on the starboard shaft. Based upon the bearing load measurements and the influence numbers, it was estimated that the shafts were straight aligned to less than ±0.13 mm.

The bearing reaction influence numbers indicate that the shafting system is stiff, as shown in Table 6. For example, a movement of 0.254 mm on No. 5 bearing results in a 4.5 kN change in the load, which is about 3 times the static load on the bearing. The ratio of the distance between shaftline bearings to the shaft diameter is 11 for this shafting system.

Table 5 Lineshaft Bearing Loads: Case 2

Bearing	Load (kN)			
	Initial		Realigned	
	Vertical	Horiz.	Vertical	Horiz.
No. 7	5.7	-0.3	5.0	0.7
No. 6	-1.8	1.1	1.6	-0.4
No. 5	5.1	-2.5	1.1	-0.5
No. 4	-0.8	0.8	0.5	0.1
No. 3	3.0	1.5	3.2	0.5
No. 2	2.4	-1.0	2.0	0.3
No. 1	1.7	0.8	2.1	0.4

Conversion Note: 1 kN = 225 lbs

Table 6 Bearing Influence Numbers: Case 2

	Reaction Influence (kN/mm)						
Bearing	No.7	No.6	No. 5	No.4	No.3	No.2	No.1
No.7	-3.5	5.4	-3.3	1.0	-0.3	0.1	0.0
No.6	5.4	-11.9	11.1	-5.2	1.4	-0.3	0.1
No.5	-3.3	11.1	-17.6	14.4	-5.9	1.4	-0.3
No.4	1.0	-5.2	14.4	-19.2	12.9	-4.7	0.9
No.3	-0.3	1.4	-5.9	12.9	-15.6	10.4	-3.2
No.2	0.1	-0.3	1.4	-4.7	10.4	-11.8	5.4
No.1	0.0	0.1	-0.3	0.9	-3.2	5.4	-3.6

Conversion Note: 1 kN/mm = 5.7 lbs/mil

The No. 7 bearing continued to run at high temperatures after the realignment. Further investigation indicated that shaft speed of 2000 RPM exceeded the rated speed for the bearing. Modifications were incorporated to provide oil lubrication for this bearing. Prior to delivery of the next vessel of this class in late December 2002, strain gauge alignment was included in the alignment procedures during construction. No more lineshaft bearing failures have occurred on either of these vessels.

Case 3: Thruster Tug; Four Lineshaft Bearings, Two Bearings on Stub Shaft

The propulsion system has two thrusters (port and starboard), each driven by a GM-EMD 16-645 E5 diesel engine, with a rating of 1604 kW (2150 HP) at 900 rpm. Each lineshaft is about 6100 mm (240") long and is supported by four roller bearings. The third bearing from the aft end is a fixed bearing, to carry axial loads from the shaftline. There is an 890 mm (35") stub shaft that connects the main engine to the lineshaft, which is supported by 2 spherical roller bearings. The aft bearing is a fixed bearing, to prevent axial movement of the stub shaft. Photo 3 shows the aft two bearings on the lineshaft.

Photo 3 Aft Two Bearings on Lineshaft

The vessel was delivered in 2000, after which excessive vibrations were observed around both shaftlines, and in particular the port shaftline from the No. 6 (aft) to No. 4 bearing. The starboard shaftline had considerable less vibration. The vibration levels on both shafts on the sister vessel were reported to be satisfactory. In 2000 structural vibration measurements and analysis were conducted. In accordance with recommendations from previous vibration measurements and analyses (not LamaLo Technology Inc.), the bearing supports were stiffened. Additional angle bar was added to tie the bearing seatings together and steel pipes were welded from the seating to the shell plating stiffeners. Photos 4 and 5 show the lineshaft and bearing seating supports prior to and after modifications, respectively. About 20 months after the modifications, the vibration levels increased again, and cracking of the welds where the added angle bar was fitted to the bearing seating occurred on a number of occasions, and the vibration levels increased.

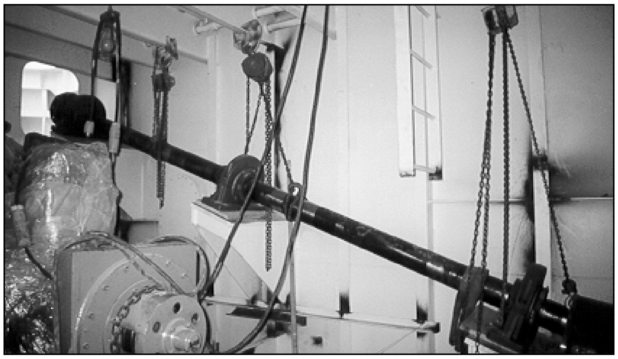


Photo 4 Original Bearing Supports: Case 3


Photo 5 Modified Bearing Supports: Case 3

To determine an optimum solution to the excessive vibrations, an investigation was conducted that involved a combination of theoretical modeling, design review, and on-site measurements. The installed alignment condition was measured August 5, 2002. It was found that both the port and starboard shafts were misaligned, such that the No. 5 bearing on the port shaft, and No. 4 bearing on the starboard shaft were top-loaded (numbered forward to aft). In addition, the port shaft was hogging by over 2.54 mm (0.100"), but the starboard shaft was near the straight-aligned condition. Both shafts were realigned to provide for satisfactory bearing loads on August 6, 2002. The hog of 2.54 mm was still present in the port shaft, while the starboard shaft was straight aligned. The vessel was tested and the vibration levels were found to be satisfactory on the starboard shaft. However, the port shaft vibrations were excessive, in particular the No. 4 lineshaft bearing, which demonstrates that although the bearing loads were satisfactory the alignment condition was not. The port shaft was re-aligned to be near the straight-aligned condition on August 7, 2002 and tested that evening. All bearing temperatures and vibrations were then found to be acceptable. Table 7 lists the measured bearing loads prior to and after the realignment. The maximum measured shaft static bending stress was reduced from 18.6 MPa (2,700 psi) to 4.1 MPa (600 psi) on the starboard shaft, and from 24.4 MPa (3,450 psi) to 3.3MPa (480 psi) on the port shaft, as a result of the realignment work. The strain gauge measurements indicated that the shafts were straight-aligned to less than ±0.13 mm (±0.005"), after the realignment.

Table 7 Lineshaft Bearing Loads: Case 3

Port Lineshaft Bearing Loads (kN)

	Initial August 5, 2002		First Re-Alignment August 6, 2002 No.5 Up 0.76 mm		Second Re-Alignment August 7, 2002 No.5 Down 1.3 mm; No.4 Down 3.1 mm	
	Vertica l	Horizontal	Vertica l	Horizontal	Vertica l	Horizontal
No. 6 (Aft)	4.0	-0.3	2.6	-0.4	4.0	-0.6
No. 5	-0.9	0.0	2.7	0.2	1.5	1.6
No. 4	6.8	0.8	3.8	0.8	1.7	-1.3
No. 3 (Fwd)	1.3	-0.4	2.1	-0.6	4.0	0.4

Jack-Up Load Test Initial Condition: No. 5 = Top Loaded; No. 4 = 6.7 kN

Starboard Lineshaft Bearing Loads (kN)

	Initial - August 5, 2002		After Re-Alignment - August 6, 2002 No. 5 Up 0.76 mm; Inboard 0.76 mm; No. 4 up 1.14 mm	
	Vertical	Horizontal	Vertical	Horizontal
No. 6 (Aft)	3.9	1.6	3.9	0.3
No. 5	3.0	-4.5	1.7	-1.1
No. 4	-1.1	3.9	1.5	1.2
No. 3 (Fwd)	5.4	-1.0	4.1	-0.4

Conversion Note: 1 kN = 225 lbs; 1 mm = 0.039"

Jack-up tests were also conducted on the No. 4 and No. 5 bearings on port shaft. No correction factor was applied to the jack-up load test results since the jack was placed relatively close to the bearing. The results from the jack-up loads tests were within 10% of those measured using the strain gauge technique.

Linear vibration measurements were taken using a Bruel and Kjaer Type 2513 Vibration Meter. Vibration levels below 20 mm/sec RMS can be considered acceptable for this system [11,15]. The results are shown in Table 8, which indicate that the vibration levels were excessive prior to the realignment, and were acceptable after the realignment.

The bearing reaction influence numbers indicate that the shafting system is stiff, as shown in Table 9. For example, a movement of 0.254 mm on No. 5 bearing results in a 1.4 kN change in the load, which is about 80% of the static load on the bearing. Therefore, the tolerance for shaft alignment is about 0.13 mm (0.005"), which corresponds to about 45% of the static load on the bearing. To align the bearings within this tolerance, it is recommended that strain gauge alignment method be used to position the inner lineshaft bearings (No. 5 and 4) for these systems.

Table 8 Linear Vibration Measurements: Case 3

	Vibration Level (mm/sec RMS)			
Bearing	Prior to Realignment		After Realignment	
	Port	Stbd.	Port	Stbd.
No. 6	30	15	10	7
No. 5	15	15	7	7
No. 4	45	20	15	15
No. 3	5	5	5	5

Table 9 Bearing Influence Numbers: Case 3

	Reaction Influence (kN/mm)			
Bearing	No. 6	No. 5	No. 4	No. 3
No. 6	-1.0	2.1	-1.4	0.2
No. 5	2.1	-5.4	4.7	-1.4
No. 4	-1.4	4.7	-5.4	2.1
No. 3	0.2	-1.4	2.1	-1.0

Conversion Note: 1 kN/mm = 5.7 lbs/mil

The vibration levels on the lineshaft continue to be acceptable. It is planned to incorporate strain gauge alignment on the next vessel of this class.

EXAMPLE ALIGNMENT PROCEDURE

The following example alignment procedure applies to a shafting system similar to that in shown in Figure 1 and in Case 3. This procedure is designed to produce the prescribed bearing offsets and loads listed in Table 10.

Table 10 Prescribed Bearing Loads and Offsets

Bearing	Bearing Load (kN) ± 0.3	Bearing Offset (mm) ± 0.13
Lineshaft		
No. 6 (Aft)	4.0	0.0
No. 5	1.6	0.0
No. 4	1.6	0.0
No. 3	4.0	0.0
Stub Shaft		
No. 2	1.3 to 2.0	0.25 to 0.60
No. 1 (Fwd.)	1.3 to 2.0	0.25 to 0.60

Alignment of the shafting system should be conducted using the following four techniques:

(i) "Pointer Method": This is the method specified by manufacturer of the Z-Drive. It is primarily used to align No. 3 and 6 Bearings, and the Main Engine and Z-Drive. The alignment of No. 1 & 2 bearings (stub shaft) is also specified in this procedure.

(ii) Optical (Wire/Laser/Feeler Gauge): This method is used to rough align No. 5 and 4 lineshaft bearings.

(iii) Strain Gauge Technique: This method is used to conduct the final alignment (horizontal and vertical) of the complete assembled propulsion system prior to chocking the bearings. The Main Engine and Z-Drive should already be positioned.

(iv) Jack-Up Load Method: This method is used to final-align the stub shaft bearings, and provide a check on the strain gauge method for the lineshaft. The Main Engine and Z-Drive should already be positioned.

With the vessel in the water, all relevant main machinery installed and major structural work completed, conduct the following tasks.

1. Follow the procedure specified by the manufacturer of the Z-Drive (Pointer Method).
2. Support the bearings by jacking screws.
3. Position No. 5 and 4 bearings within ±1.0 mm along a straight line drawn through No. 6 and No. 3 bearing centers. Any suitable optical method can be used. This step is used to roughly position these bearings.
4. Align No. 5 and 4 bearings to reduce the horizontal load on the bearing to a minimum. This can be done by any available method, such as rotating the shaft and allowing the bearing to move sideways. Again, this step is used to roughly position these bearings.
5. Install the Cardan Shafts, and connect all shafting, including the torsional coupling to the stub shaft.
6. Conduct a strain gauge alignment to finalize the position of No. 4 and 5 bearings.
7. Align No. 1 and 2 bearings (stub shaft) using the jack-up load method. Ensure that these bearings will be down-loaded when the machinery is at its operating temperature.
8. Chock all bearings and the main engine in place.
9. Measure the alignment condition using the strain gauge technique after all bearing hold down bolts are torqued.
10. This step is optional. After sea trials, and with the vessel fully loaded and the machinery at operating temperature, measure the alignment condition using the strain gauge technique.

CONCLUSIONS

A number of shaft bearing failures due to misalignment have recently occurred on vessels with Z-Drive thrusters fitted with Cardan shafts and lineshafts. Excessive vibration and failure of bearings can occur even if the bearing loads are satisfactory. Bearing failures can be prevented by ensuring that all alignment criteria (listed herein) are satisfied. To satisfy the alignment criteria the bearings are typically required to be positioned with a tolerance of ±0.13 mm (±0.005"). Incorporating the strain gauge alignment technique into the alignment procedure has shown to be an effective means of obtaining a satisfactory alignment condition, and preventing bearing failures.

AKNOWLEDGEMENTS

The authors acknowledge, with thanks, the efforts and cooperation provided by the manufacturers, ship owners, and ship builders. Special thanks are given to the ship's crew and shipyard workers whose hard work and perseverance made the successful repairs and realignment possible.

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