SHAFT ALIGNMENT USING STRAIN GAGES: CASE STUDIES

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Two case studies are presented in which measurement and modeling programs were undertaken to assess the installed shaft alignment condition on marine vessels. One on the CCGS Earl Grey, and the other on the HMCS Huron. In both cases catastrophic damages had previously occurred to shaftline components, which were suspected to be a result of shaft mis-alignment. The damaged components were repaired, and the shaftlines were re-aligned using traditional methods (optical/laser). To assess the alignment condition of the installed shaftlines, measurements were taken by using the strain gage technique. In one case the shaft was realigned according to the strain gage measurements. Mathematical modeling was also conducted to assess the implications of the alignment measurements, and to provide the means to estimate the offsets of the installed bearings from their prescribed positions.

INTRODUCTION

Shaft alignment and vibration measurements, and modeling, are increasingly becoming an integral part of pro-active cost effective maintenance programs. With an analytical model, and measurements of the alignment and vibration conditions, intelligent decisions with respect to repair/maintenance of marine shaftline components can be made. Traditionally decisions on repairs and maintenance have been conducted based upon limited information or understanding of the shaftline vibration and alignment characteristics. Typically no measured data is available prior to the damage occurring, making it difficult to assess the cause of the damage, and therefore, the most cost-effective method of repair. Costs associated with shaft alignment problems have been reported to be in the order of $100,000 per vessel and have been over $1,000,000 for some extreme cases. Therefore, only a small percentage reduction in these types of costs would result in significant financial benefits, in addition to improved vessel service capabilities.

In this paper two case studies are described in which measurement and modeling programs were undertaken to assess installed shaft alignment conditions. One on the CCGS Earl Grey, and the other on the HMCS Huron.

MODELING TECHNIQUE

The software program called "SHAFTKIT Version 2.0" was used for the theoretical modeling analysis. The program can be used to calculate the theoretical static alignment condition, as well as the forced-damped response for axial, torsional, and lateral vibrations. The solution algorithm was adapted from the finite element program "SAP IV", and uses the Wilson-Theta Method of direct integration to solve the equations of motion of the shafting system. The eigen value problem is solved to compute the natural frequencies directly. Propeller damping, concentrated damping, and Raleigh damping can also be included. Shaft models are composed of a linear system of shaft elements of uniform section, with concentrated springs (bearings) and masses (e.g., propeller).

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The static alignment algorithms provide calculations of the bearing reactions, influence coefficients, and the shaft slope, displacement, bending moment, bending stress, and shear force.

SHAFTKIT is has been sold to over 35 marine organizations throughout the world, and has been verified against other finite element programs and full-scale measurements.

MEASUREMENT TECHNIQUE

The strain gage method was used to measure the bearing reactions. The alignment of an installed and complete shafting system can be determined by measuring the strain in the shaft at appropriate locations. The advantages of using the strain-gage method of determining shaft alignment are as follows:

- Both horizontal and vertical alignment can be checked simultaneously.
- Strain gage readings can be taken on an installed shaft while the vessel is in the water.
- Once the gages are installed the strain readings can generally be taken within an hour.
- Reactions for bearings that are inaccessible for jacking can be estimated with a degree of accuracy
- Once strain gages are installed on the shafts, repeat measurements can easily be taken at various times of day to account for varying ambient conditions, and in the case of cargo vessels under different loading conditions.

The output of the strain gages provides a measurement of the bending strain in the shaft, from which the bending moment can be determined. Force and moment equilibrium equations are used to develop equations that relate the bending strain measurements to the bearing reactions. The bearing offsets from the straight-line condition (all bearing centers concentric) are estimated by computing which offsets, when applied to the influence coefficients, would result in the same reactions as the difference between the measured and the calculated straight-line reactions.

CASE STUDY 1 - CCGS EARL GREY PROPULSION SHAFLTINE ALIGNMENT

Shaftline Description

The CCGS Earl Grey is a Type 1050 Navaids Vessel, and has a twin screw propeller configuration. The two propulsion shaftlines are identical, each consisting of two KHD Deutz (SBV 9M628) medium speed diesel engines rated at 1,625 kW (2,178 HP) at a service speed of 900 RPM, driving a single screw controllable-pitch propeller, through a twin input, single output, single reduction gearbox (4.5:1). Power Take Offs (PTOs) are connected to both gearboxes, and are used for generating power for the bow and stern thrusters and an external fire pump. The propulsion shaft is approximately 16.9 m long, and is supported by two oil lubricated stern tube bearings and one intermediate shaftline bearing.

Background

The vessel was scheduled for a routine refit and dry-docking. There were no reports of any abnormal condition prior to the docking (i.e., vibrations, high temperatures etc.). With the vessel up on the slips inspection of the after end indicated welding cracks in the struts, skeg, kort nozzle and the stern tube. It is suspected that these cracks developed during operations in ice. Both tailshafts were withdrawn and it was decided to renew all the sterntube bearings and all the cracked welding was repaired. Unfortunately the alignment was not checked at this time. During subsequent trials, vibrations were noted in the shaftlines and high sterntube bearing temperatures were recorded. Inspection of the oil filters indicated that there was damage to the bearings. The vessel was docked again and the tailshafts withdrawn. The alignment on both drive trains was checked, and both gearboxes were found to be out of line. Both sterntube liners were line-bored, and both gearboxes were realigned using the optical/laser method. It was also necessary to realign all four engines, and PTOs. New sterntube bearings were installed and tailshafts were reinstalled. Subsequent sea trials proved that the repairs were satisfactory. In order to establish a database for future reference, and to verify the installed alignment condition, bearing reactions were measured using the strain gage technique.

Measurements

Each shaftline of the CCGS Earl Grey has 5 bearings. Therefore, an additional 3 data points must be known for the shaft alignment to be determined. Strain gages were installed at the break points indicated in the free body diagrams of the shaftline shown in Figure 1.

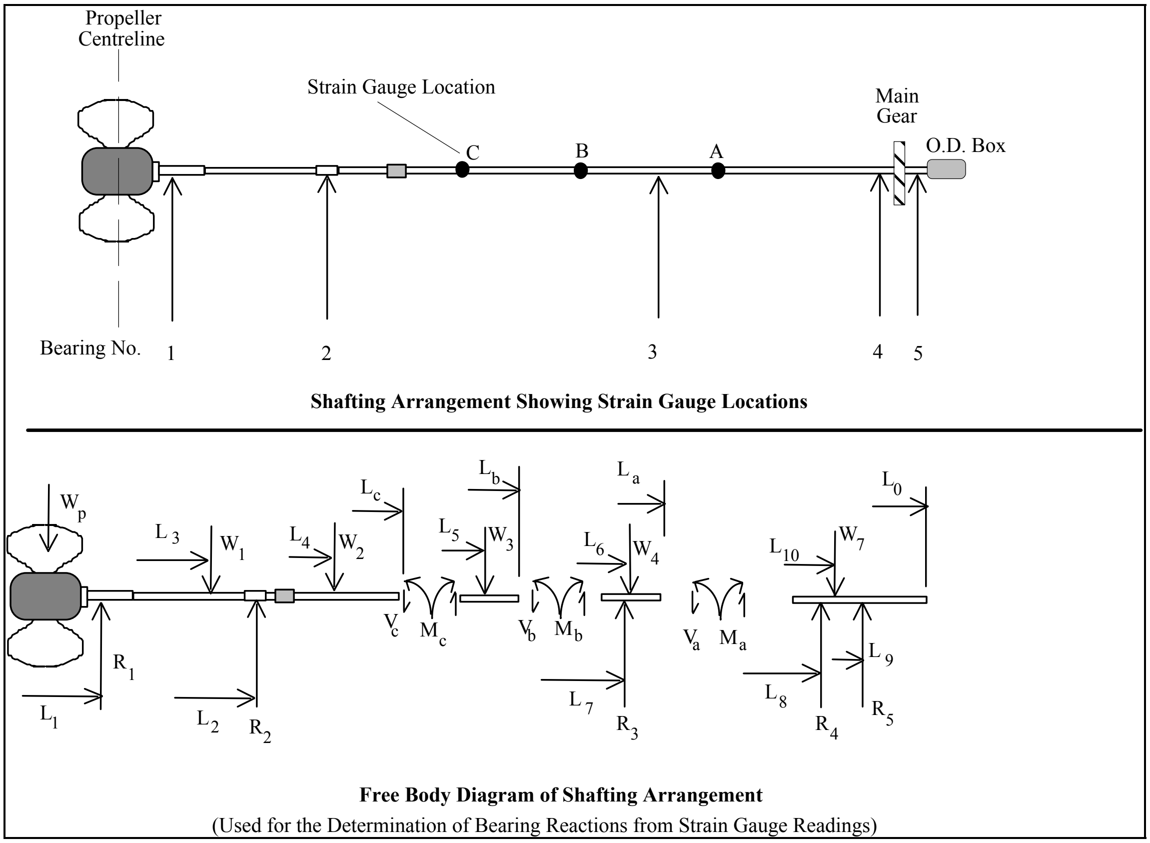

Figure 1 Shaftline Arrangement Showing Strain Gages and Free Body Diagrams:
CCGS Earl Grey

Data and Results

Measurements were taken at two gearbox lube oil temperatures, ambient and warm, to obtain data on the effect of the thermal rise associated with the gear bearings during the hot running condition. An Excel spreadsheet was developed to compute the vertical and horizontal bearing reactions from these measurements. In addition to the strain gage measurements, the intermediate bearing reaction was measured by means of the jack-up method. The measurements were taken November 4-5, 1996, while the vessel was alongside the CCG dock at Charlottetown, PEI.

Table 1 presents the vertical bearing reactions for the measured, the estimated hot running, and the prescribed alignment condition. The results of the jack-up tests on the intermediate bearing are also listed in Table 1, and the data is plotted in Figure 2. The jack-up test results agreed well with those from the strain gage based measurements. Both measurement techniques indicated that the starboard shaft intermediate bearing reaction was significantly less than that for the prescribed condition. The reactions for the hot running condition (Gear Lube Oil T=50°C) were estimated by assuming a linear change in bearing reaction with rise in temperature. As a check on this assumption, an independent estimate was made by using the influence coefficients and the projected thermal rise of the gear bearings. This analysis produced approximately the same bearing reactions for the hot running condition.

Figures 3 and 4 illustrate the vertical bearing reactions for the hot running condition. The bearing reactions on the port shaft were in close agreement to the prescribed condition. However, the starboard intermediate bearing reaction was 60% (11.2 kN) less than the prescribed value, and the differential gear bearing reaction was 9.5 kN, which was very close to the maximum allowable of 11 kN. An analysis was conducted to determine a simple solution to resolve these discrepancies. It was determined that raising the intermediate bearing by 0.20 mm from its present position would result in relatively equal gear bearing reactions, and a 2.2 kN increase in the intermediate bearing reaction. Table 2 lists the reactions for this solution, and Figure 5 provides an illustration. It was recommended that the intermediate bearing be raised 0.20 mm.

Table 1 Static Vertical Bearing Reactions: November 1996

	Vertical Bearing Reactions (kN)						
	Starboard			Port			Prescribed
Bearing	Measured		*Est.*	Measured		*Est.*	
	T=22°	T=43°	*T=50°*	T=20°	T=34°	*T=50°*	T=50°
Aft Stern Tube	91.3	90.9	*90.8*	90.8	93.8	*97.2*	95.6
Fwd. Sterntube	57.7	58.6	*58.9*	54.6	51.4	*47.8*	48.3
Intermediate	18.2	17.0	*16.6*	25.0	24.8	*24.6*	27.8
Aft Gear	40.6	42.6	*43.3*	33.3	34.9	*36.8*	35.9
Fwd. Gear	35.7	34.2	*33.8*	39.8	38.5	*37.1*	36.2
	Jack-up Tests						
Intermediate	18.5				27.5		

Notes:
1. Temperatures are for the gearbox lube oil.
2. Estimates at 50°C were based upon projecting a linear change with temperature from the measured data. Estimates also made using the influence coefficients produced approximately the same results.

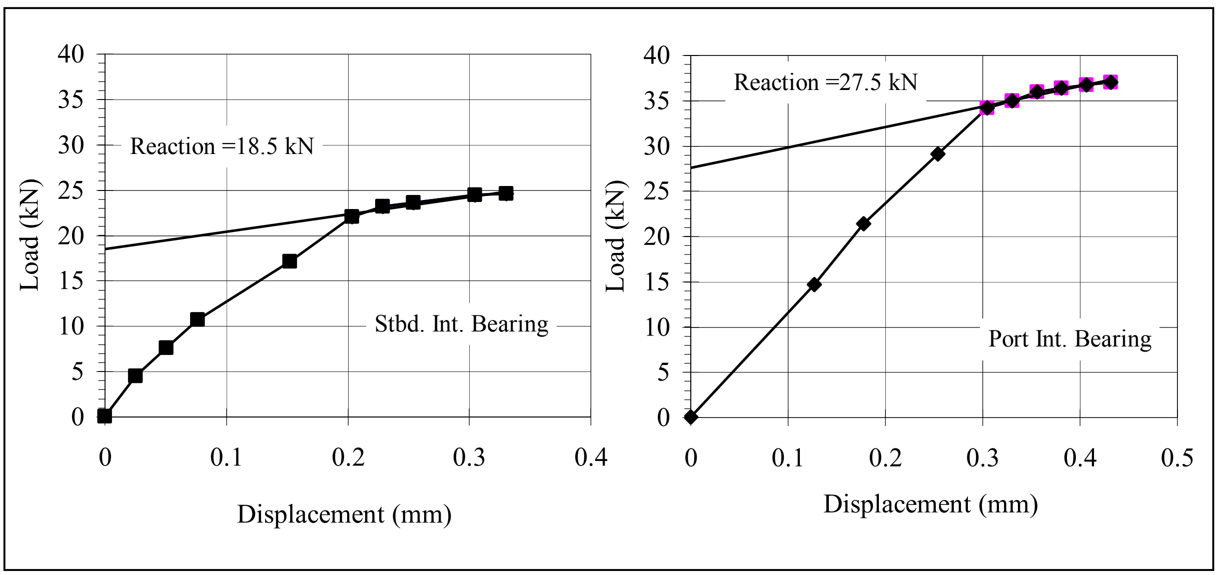

Figure 2 Jack-up Test Results

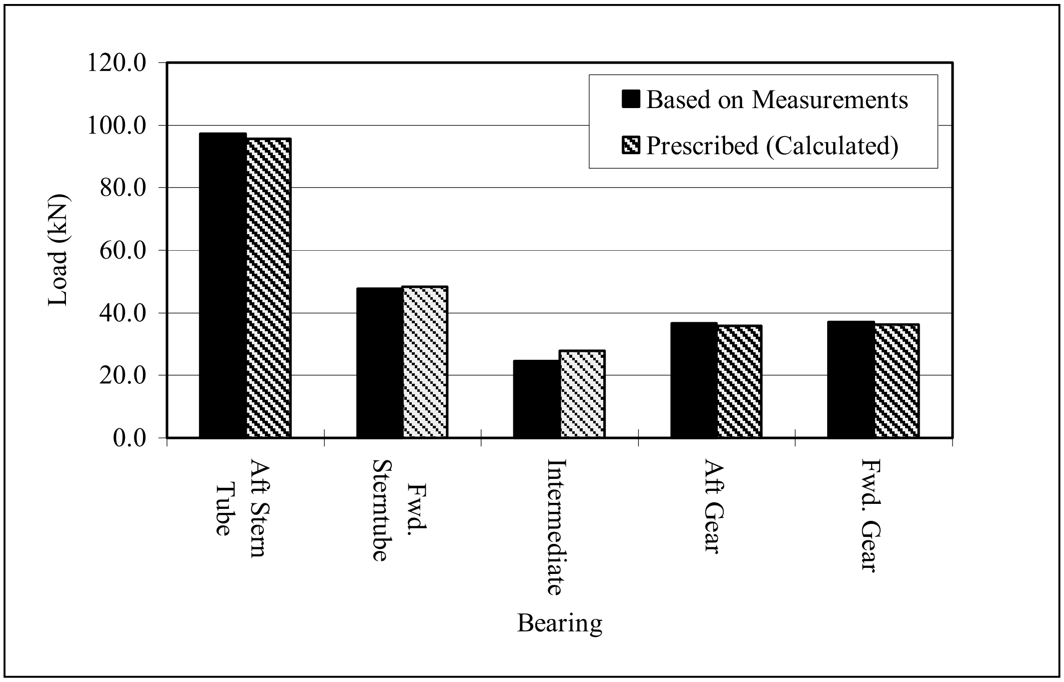

Figure 3 Port Shaft Bearing Reactions
(both inboard and outboard engines developing full power with zero power take-off)

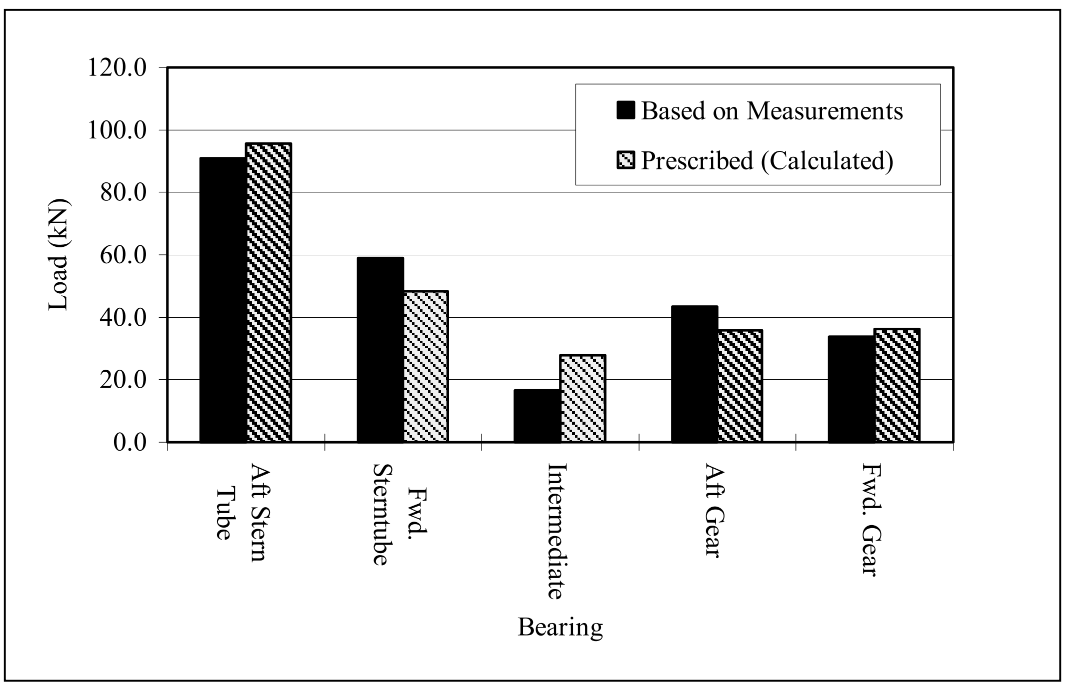

Figure 4 Starboard Shaft Bearing Reactions
(both inboard and outboard engines developing full power with zero power take-off)

Table 2 Proposed Starboard Shaft Intermediate Bearing Positions

Starboard Shaft						
Alignment Status		***Current***	***Proposed***	***Prescribed***	***Current***	***Proposed***
Conditions						
Gear Lube Oil Temp. (°C)		50°	50°	50°	22°	22°
Int. Bearing Position (mm) *(Change from Present Position - Upward Positive)*		0.00	0.20	0.25	0.00	0.20
			Reactions (kN)			
Aft Stern Tube		90.8	91.4	95.6	91.3	91.9
Fwd. Sterntube		58.9	57.5	48.3	57.7	56.3
Intermediate		16.6	18.8	27.8	18.2	20.4
Aft Gear		43.3	37.9	35.9	40.6	35.2
Fwd. Gear		33.8	37.8	36.2	35.7	39.7

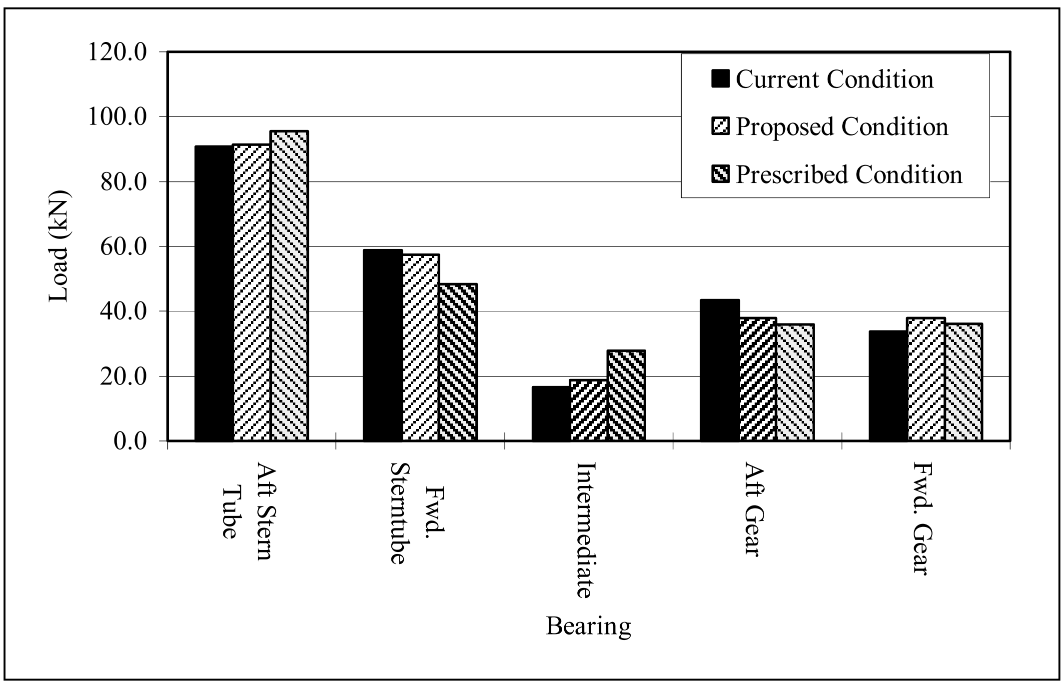

Figure 5 Proposed Starboard Shaft Re-Alignment Condition: Hot Running

Unequal forces and moments are applied to the main gear bearings while using only one of the two engines for each shaft. These forces are generated by the interaction of the pinions with the main gear. Table 3 lists the resulting starboard gear bearing reactions for each operational mode. The results indicate that the differential gear reactions (15.8 kN) exceeded the design limit (11 kN), when using only the inboard engines. This design limit was not exceeded for the proposed condition.

Table 3 Starboard Shaft Gear Reactions for Different Operational Modes (Current and Proposed Condition)

	Gear Bearing Reactions (kN)			
Operational Mode	**Current Condition**		**Proposed Condition**	
(Engines at Full Power)	**Forward**	**Aft**	**Forward**	**Aft**
Both Engines	33.8	43.3	37.8	37.9
Inboard Engine Driving Shaft	-7.6	8.2	-3.6	2.8
Outboard Engine Driving Shaft	75.1	78.4	79.1	73.0

The horizontal bearing reactions were also computed from the strain gage data. Table 4 lists the results. The horizontal reactions were all negligible, and were less than 5% of the corresponding vertical reaction. Therefore, the horizontal alignment condition was considered to be satisfactory.

Table 4 Static Horizontal Bearing Reactions: November 1996

	Horizontal Bearing Reactions (kN) (Positive Towards Stbd.)					
Bearing	**Starboard**			**Port**		
	T=22°	T=43°	*T=50°*	T=20°	T=34°	*T=50°*
Aft Stern Tube	-3.7	-3.2	*-3.0*	0.5	1.4	*2.3*
Fwd. Sterntube	4.2	3.4	*3.1*	-0.6	-0.7	*-0.9*
Intermediate	-0.9	0.3	*0.7*	-0.2	-1.5	*-2.9*
Aft Gear	4.4	-0.1	*-1.6*	0.7	1.1	*1.6*
Fwd. Gear	-4.1	-0.4	*0.8*	-0.5	-0.3	*-0.1*

CCG Earl Grey Conclusions

- The measured bearing reactions on the port shaft were in close agreement to the prescribed condition.
- The jack-up test results on the intermediate bearing agreed well with those based upon the strain gage based measurements.
- Both measurement techniques indicated that the starboard shaft intermediate bearing reaction was significantly less than that prescribed.
- In the hot running condition, with both engines on, the starboard intermediate bearing reaction was computed to be 60% less than the prescribed value. The differential gear bearing reaction was computed to be 9.5 kN, which is very close to the maximum allowable of 11 kN.
- With only the inboard engine on-line, the measurements indicated that the starboard shaft differential gear reaction would be 15.8 kN, which considerably exceeds the 11 kN design limit.
- It was determined that a satisfactory alignment condition would result by raising the starboard intermediate bearing by 0.20 mm from its present position.

CASE STUDY 2 - AUXILIARY GEARBOX OUTPUT SHAFT ALIGNMENT ON THE HMCS HURON

Shaftline Description

The HMCS Huron is one of four Iroquois Class Destroyers built in the early 70's, and most recently modernized in the early 90's by the Tribal Class Update and Modernization Program (TRUMP). The propulsion system installed in the updated Class consists of two propeller shaftlines, each of which is operated by a main or cruise gas turbine engine. The main engines are Pratt & Whitney FT4A-2 marine gas turbines rated to deliver 18,500 kW. The cruise engines are Allison 570-KF marine gas turbines rated to deliver 4806 kW. The engines drive their respective CRP propellers though two independent double input, single output, MAAG main reduction gearboxes. Both the Port and Starboard cruise engines input their respective main reduction gearing via a smaller auxiliary reduction gearbox.

The auxiliary gearbox output shaft is supported by four bearings, two in the auxiliary gearbox (AGB) and two in the main gearbox (MGB). The bull gear of the AGB is located between the two AGB bearings, and a primary pinion of the MGB is between the two MGB bearings. The shaft is approximately 2.9 m long. The shaft section between the AGB and MGB has an outside diameter of 0.130 m and an inside diameter of 0.116 m, and is $42NiCrMo_6$ steel. There are three bolted flanged connections. Figure 6 provides a schematic of the shaftline.

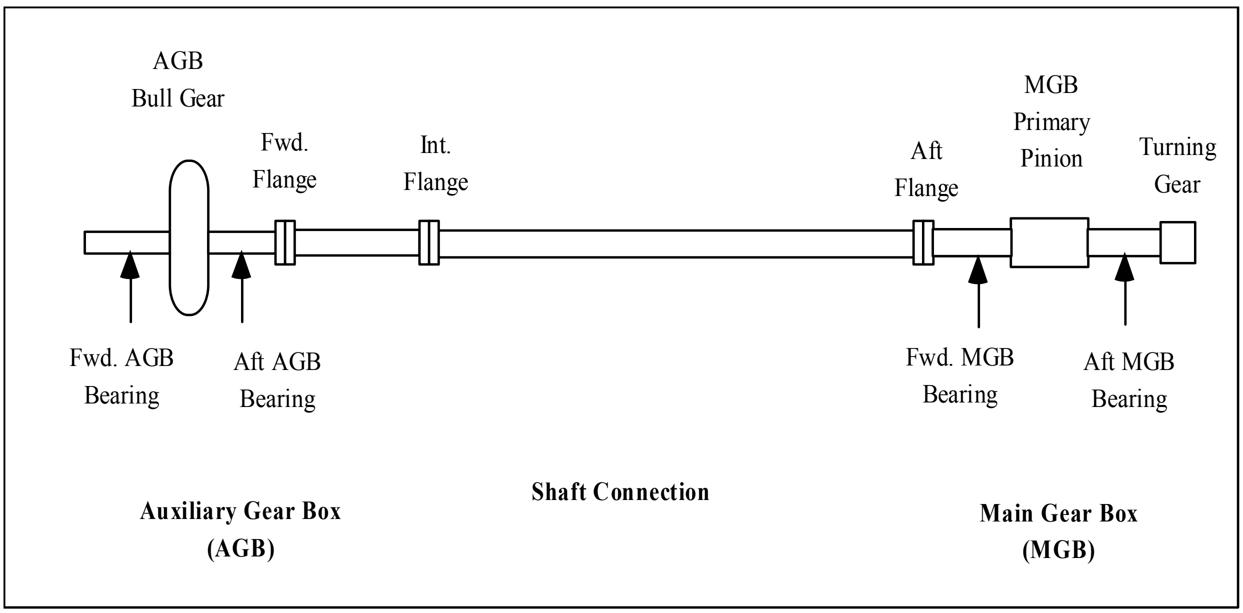

Figure 6 Schematic of Shaftline

Background

HMCS Huron was the last of four IROQUOIS Class Destroyers modernized. Each vessel underwent the same work package, which included replacement of the cruise engines and installation of the new auxiliary gearbox on the original cruise input of the main gearbox. The new auxiliary gearbox was required to reduce the higher input speed of the new Allison cruise engines.

The installation and alignment procedure of the auxiliary gearbox used an empty bearing shell in the main gearbox as a datum to align the auxiliary gearbox to the main gearbox. All shaft flange alignment readings were taken with dial indicators. No laser or other methods were used for alignment. With two of these installations per ship, a total of 8 cruise drivetrains were modified using this procedure. All ships have operated successfully with this procedure, with the exception of HMCS Huron.

On 24 April 1996, HMCS Huron experienced a catastrophic failure on her Port auxiliary cruise drivetrain. After approximately 1700 hours operation on the Port cruise engine, the bolts between the Port Auxiliary output and the Port main gearbox input flange failed, resulting in significant damage to both the Port auxiliary and main gearboxes. As a result of the primary bolt failure, the auxiliary gearbox was removed and sent to the OEM for a new output flange, and the main gearbox underwent 10 weeks of in-situ work to repair secondary damage to gearing meshes caused by ingress of failed bolt pieces.

The initial failure investigation revealed use of flange bolts not conforming to the required dimensional specifications, and this was thought to be the cause of the catastrophic failure. The repaired auxiliary gearbox was eventually reinstalled in the ship using the same alignment procedure, and the proper flange bolts. All alignment readings indicated a satisfactory installation. As a final precaution, two of the flange bolts were removed and inspected after 200 hours operation.

This subsequent bolt inspection revealed significant fretting on the fitted bolt shanks after only 200 hours, and was immediate cause for concern. At this stage it was decided by the Department of National Defence (DND) to verify the installed alignment using other means than the original dial indicator method. The limited space within the gearbox precluded the use of laser equipment to verify in-situ alignment unless the main gearbox cover was lifted, and so the strain gage method was used requiring only the removal of two inspection ports. This method resulted in savings of approximately 28 man-days of dockyard work that would have otherwise been expended to verify alignment using a laser.

The strain gage readings indicated a significant misalignment between the main and auxiliary gearboxes. It is suspected that this misalignment was induced by using the empty bearing shell as a datum for the dial indicator readings. While this procedure had been successful on the other ships, the empty bearing shell on this gearbox was not in the correct position to begin with (for reasons unknown), and this resulted in an original installed but undetected misalignment. This was believed to be the contributing cause, along with the non-conforming bolts, to the original catastrophic failure.

The port auxiliary gearbox was repositioned using strain gage readings, and the port cruise engine realigned to the new position of the auxiliary gearbox. New conforming bolts were installed in the flange that previously failed, to replace the fretted bolts. The total cost of the entire repair, including the second realignment, approaches $500k for this ship, and does not include approximately 3 months of downtime.

HMCS Huron was deployed in the South Pacific, and reported satisfactory performance of her Port cruise drivetrain. Upon her return in late June 97, a complete alignment verification was conducted, with both strain gages and lasers, to ensure a satisfactory alignment was achieved and to provide comparisons between various measurement methods. Due to the minimal cost and impact of the strain gage readings, it is intended to verify in-situ the similar installations on the three other vessels of the Class as well.

Measurements

The Auxiliary Gearbox (AGB) shaftline of the HMCS Huron has 4 bearings. Therefore, an additional 2 data points must be known for the shaft alignment to be determined. Strain gages were installed at the break points indicated in the free body diagrams of the shaftline shown in Figure 7. Measurements were taken of both the starboard and port side shaftlines.

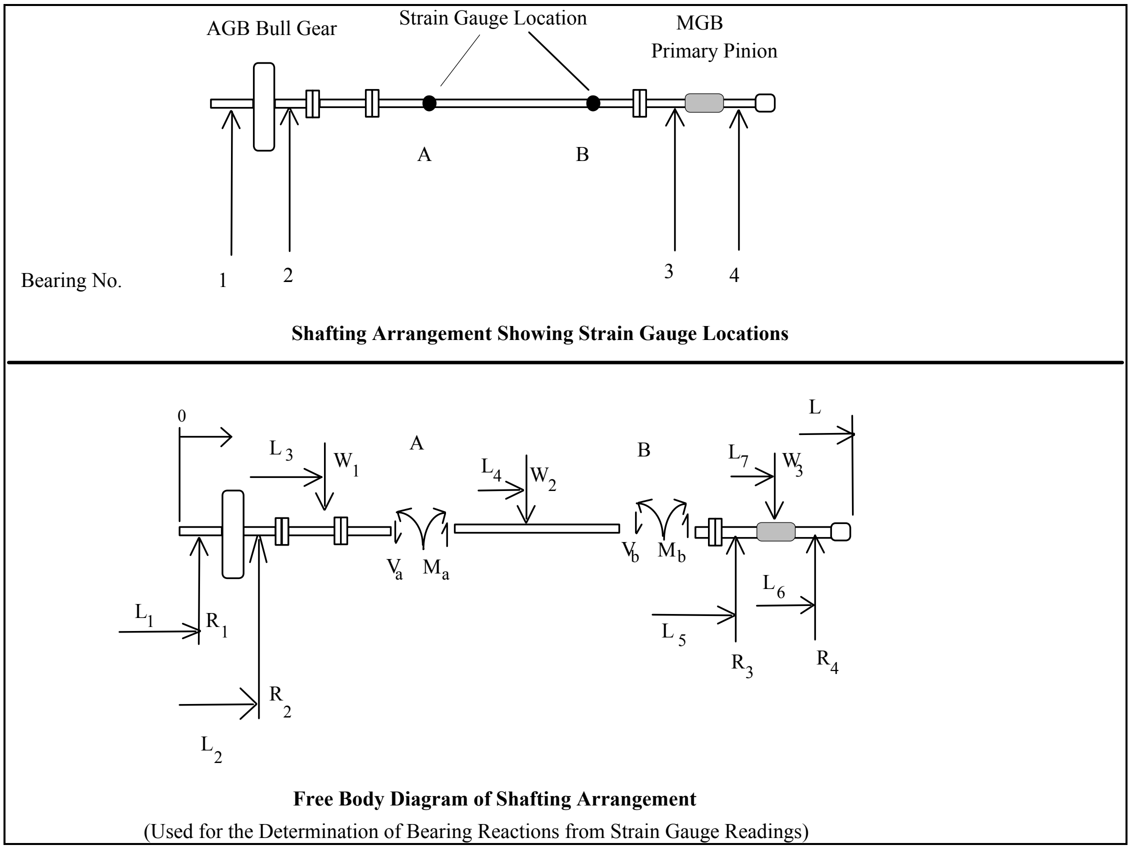

Figure 7 Shaftline Arrangement Showing Strain Gages and Free Body Diagrams

Measurements were taken at two gearbox lube oil temperatures, ambient and warm, to obtain data on the effect of the thermal rise associated with the gear bearings. An Excel spreadsheet was developed to compute the vertical and horizontal bearing reactions from these measurements. The measurements were taken March 17 & 18, 1997, while the vessel was alongside the dock at CFB Esquimalt.

Table 5 and 6 present the measured bearing reactions. The computed straight line alignment vertical reactions are also presented for comparison. The theoretical horizontal bearing reactions for a straight aligned shaft are zero. The results indicated that the port shaft was significantly mis-aligned both vertically and horizontally. In particular, the AGB vertical bearing reactions were in the opposite direction and up to 5 times higher in magnitude than the straight line reaction. The horizontal reactions were also in the opposite direction and up to 5 times higher in magnitude than the corresponding vertical straight line bearing reaction. The maximum static bending stress measured in the port shaft was 15.3 Mpa, and the corresponding stress in the starboard shaft was 1.8 Mpa, a difference of over 700%.

The measured bearing reactions on the starboard shaft were in close agreement with the straight-line condition. There was no significant change in the alignment condition between the cold and warm conditions.

An analysis was conducted to estimate the bearing offset from the straight line condition (all bearing centers concentric) for the port shaft. The influence coefficients and the difference between the measured and calculated straight line reactions were used to estimate the bearing offsets. The results indicated that the AGB aft and fwd bearing centers were approximately 0.6 mm to 0.7 mm above and 1.2 to 1.4 mm outboard of the MGB bearing centers.

Table 5 Static Vertical Bearing Reactions: March 17& 18, 1997

	Vertical Bearing Reactions (kN)					
	Starboard Shaft			Port Shaft		Calculated
Bearing	Measured			Measured		Straight Line
	Cold	Warm		Cold	Warm	
R1 - Fwd. AGB	0.42	0.32		-3.10	-3.09	0.67
R2 - Aft AGB	1.35	1.44		5.77	5.74	1.13
R3 - Fwd. MGB	0.98	1.10		-0.21	-0.17	0.66
R4 - Aft MGB	0.17	0.07		0.48	0.44	0.47

Table 6   Static Horizontal Bearing Reactions: March 17 & 18, 1997

	Horizontal Bearing Reactions (kN) (Positive Towards Port)				
Bearing	Starboard Shaft			Port Shaft	
	Cold	Warm		Cold	Warm
R1- Fwd. AGB	-0.16	-0.18		4.81	4.81
R2 - Aft AGB	0.13	0.14		-5.67	-5.67
R3 - Fwd. MGB	0.26	0.35		-0.06	-0.02
R4 - Aft MGB	-0.23	-0.31		0.92	0.89

It was recommended that the port side AGB be re-aligned to the MGB. However, due to the time constraints imposed by the HMCS Huron’s schedule, the more traditional optical alignment method was not feasible. Therefore, the re-alignment was conducted with the aid of the strain gage readings, because measurements could be taken without removal of the AGB shaft. This was done by modifying the horizontal position of the AGB until the strain gage output indicated negligible horizontal bearing reactions (i.e., low output voltage). Then the vertical position of the AGB was adjusted until the strain gage readings provided similar bearing reactions to the straight-line condition. The measurements were taken during March 19, 1997, while the vessel was alongside the dock at CFB Esquimalt.

Table 7 and 8 present the measured horizontal and vertical bearing reactions respectively. Figure 8 presents the port shaft vertical bearing reactions for each position of the AGB. The computed straight-line alignment reactions are also presented for comparison. The results indicated that under the re-aligned condition the port shaft horizontal bearing reactions are approximately zero, indicating a straight line condition. Although the forward AGB bearing is slightly unloaded, the corresponding vertical bearing reactions agree reasonably well with the straight-line condition. While in operation the pinion forces on the Bull Gear of the AGB produce a significant downward force on the AGB bearings, which ensures that these bearing are down-loaded.

Table 7 Static Horizontal Bearing Reactions (kN)

Bearing	18-Mar	19-Mar
	(Original)	(Re-Aligned)
R1 - Fwd. AGB	4.81	0.00
R2 - Aft AGB	-5.67	0.00
R3 - Fwd. MGB	-0.06	0.01
R4 - Aft MGB	0.92	-0.01

Table 8 Static Vertical Bearing Reactions (kN)

Bearing	18-Mar	19-Mar	Calculated
	(Original)	(Re-Aligned)	(Straight Line)
R1 - Fwd. AGB	-3.10	-0.04	0.67
R2 - Aft AGB	5.77	1.91	1.13
R3 - Fwd. MGB	-0.21	0.92	0.66
R4 - Aft MGB	0.48	0.14	0.47

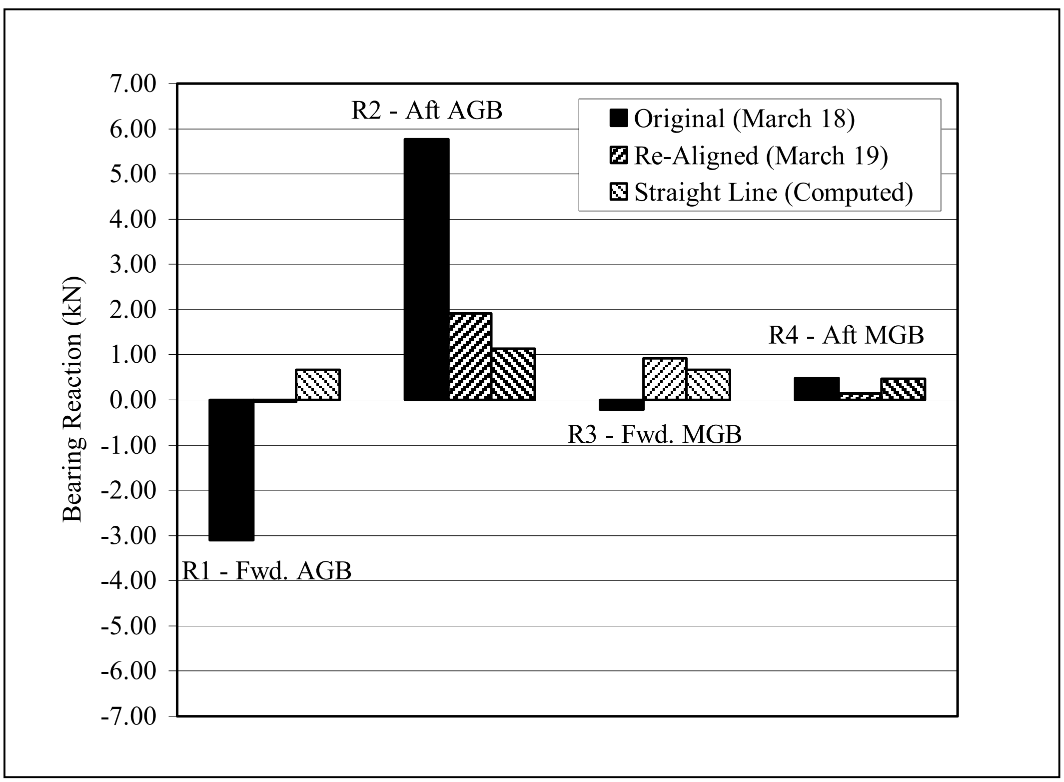

Figure 8 Port Shaft Vertical Bearing Reactions

The influence coefficients and bearing reactions computed from the mathematical model, and the measured bearing reactions, were used to estimate the bearing offsets from the straight-line condition (all bearing centers concentric). Figures 9 and 10 illustrate the estimated original mis-alignment condition, and the results of the re-alignment done on March 19, in terms of relative bearing positions. After re-alignment, the aft AGB bearing center was approximately 0.08 mm (.0032") above a line through the MGB bearing centers, and the fwd AGB bearing was on-line. All bearing centers were aligned horizontally after the re-alignment.

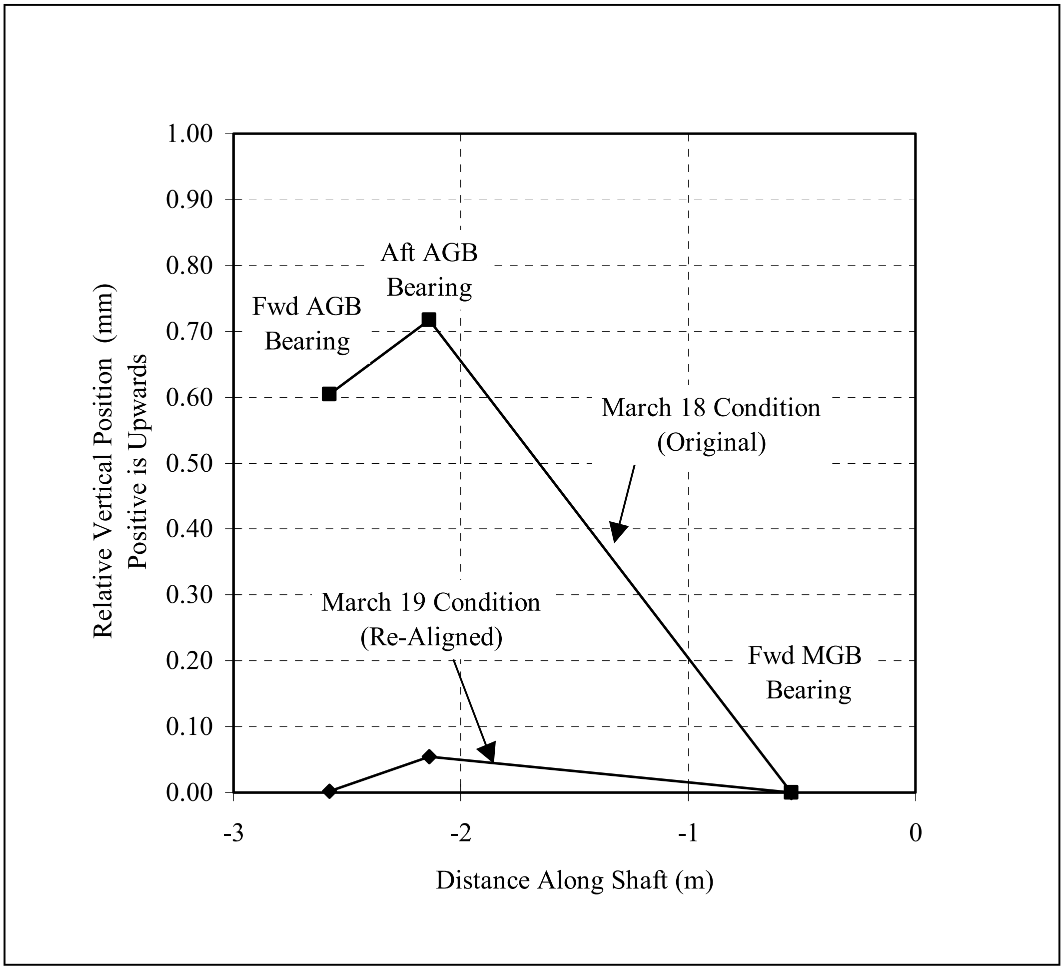

Figure 9 - Relative Vertical Position of Bearings: HMCS Huron Port AGB

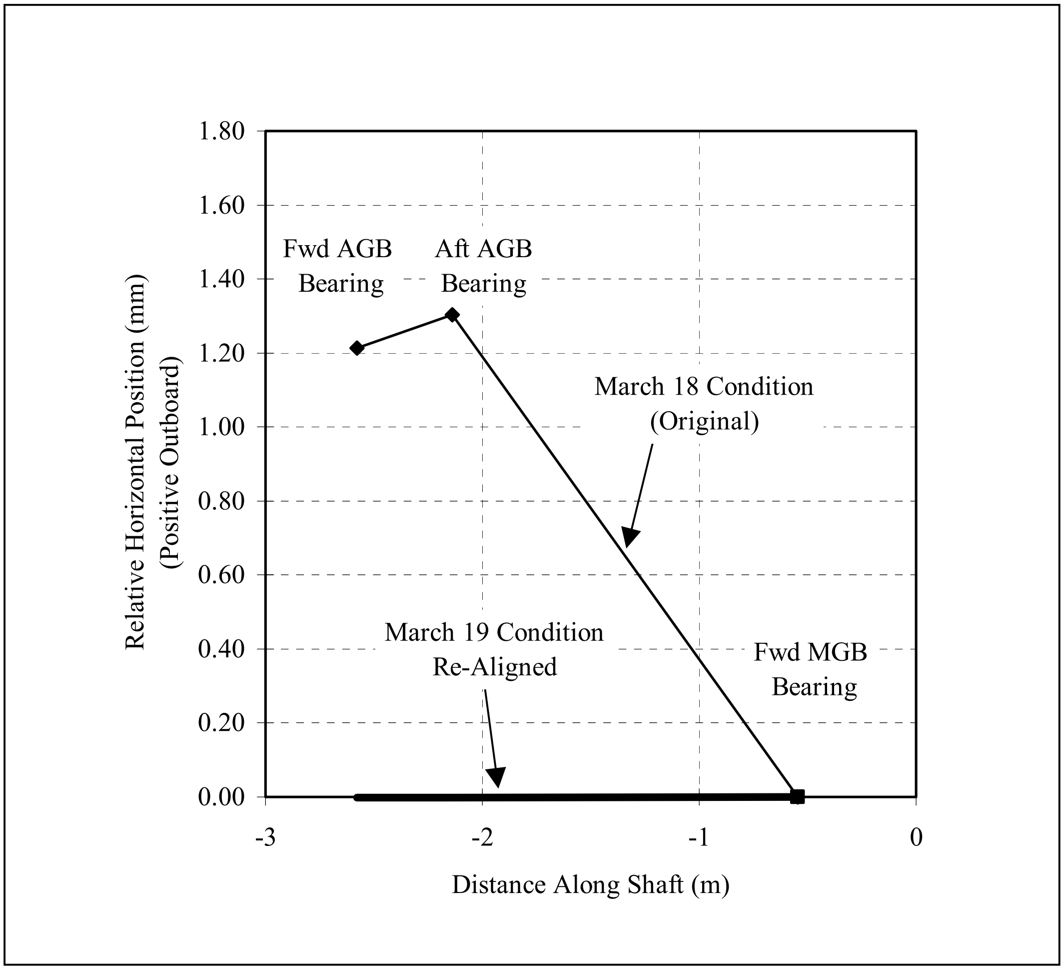

Figure 10 - Relative Horizontal Position of Bearings: HMCS Huron Port AGB

HMCS Huron Conclusions

- The strain gage technique used for the re-alignment of the auxiliary gearbox (AGB) was very successful, and cost effective.
- After re-alignment, the measurements indicated that the port side auxiliary gearbox (AGB) was in reasonable alignment with the main gearbox, both in the vertical and horizontal directions.
- It was recommended that similar shaft alignment measurements be taken on other vessels of the same class to assess their respective alignment condition.

OVERALL CONCLUSIONS

The shaft alignment measurement and modeling program conducted on the HMCS Huron and CCGS Earl Grey proved to be extremely successful. If such a program was conducted prior the damages experienced on these vessels significant repair costs may have been saved.

Shaft alignment measurement and modeling should be considered to be an integral part of a pro-active cost effective maintenance program, considering the potential significant cost benefits. Extending the program to a fleet of vessels, and having a centralized data set, has a number of advantages. For example, comparisons of the shaftline alignment and vibration characteristics can be made between vessels. As such, the effect of operational profiles on bearing wear-down, vibrations, and other items can be analysed to help determine cost-effective changes to the existing repair/maintenance/operations.

DISCLAIMER

The opinions expressed in this paper are those of the authors, and not necessarily the Canadian Coast Guard, or the Canadian Department of National Defence.

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