Propulsion Shaftline Bearing and Gear Teeth Failure Investigation on the "C" Class British Columbia Ferries

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ABSTRACT

British Columbia Ferry Services Inc. operates five "C" class passenger-car ferries that were delivered in 1976 (3 ships) and 1981 (2 ships). The vessels have a service speed of 19.0 knots, are 457 ft. long, and have a gross tonnage of 6,550 tons. They have a capacity of about 360 cars and 1,500 passengers. Their route varies from Vancouver-Vancouver Island–Sunshine Coast, with up to 9 round trips per day. The vessels have a double-ended single screw propeller configuration. Each of the controllable pitch propellers are driven by one or two diesel engines rated at 5,930 HP, through single reduction gearboxes. The two gearboxes are connected via quill shafts that are concentric with the pinion shafts. The pinion shafts are engaged via clutches at the end of each of the quill shafts. This arrangement enables one or both of the diesel engines to drive either one or both of the propellers. In 1998 a catastrophic failure occurred to the bull wheel on one vessel. The bull gear teeth broke off taking out both pinions. In 1999 the bull gear on another vessel was replaced as a result of fractured gear teeth, found during an inspection. In addition, a number of failures have occurred to both the lineshaft and pinion shaft bearings. An investigation was conducted to determine the source of the failures and the most prudent and economical course of action. To accomplish these objectives a review of the system design and maintenance/failure history was conducted, as well as theoretical modeling and alignment and vibration measurements. It was found that the failures occurred due to misalignment. A realignment program has been undertaken, with the first realignment completed in February 2002. This paper describes the failure investigation and the results of the realignment work conducted.

INTRODUCTION

The five "C" class ferries operated by British Columbia Ferry Services Inc. form a major part of their fleet. Three vessels were built in 1976 (MV Queen of Alberni, Coquitlam and Cowichan) and two more were built in 1981 (MV Queen of Oak Bay and Surrey), each with a length of 457 ft and a service speed of 19 knots. The vessels operate scheduled ferry service from Vancouver-Vancouver Island-Sunshine Coast, British Columbia. Up to 9 round trips are made per day, with over 5000 hours of running time per year. A number of damages to the intermediate shaft and gearbox pinion shaft bearings have occurred in the last 10 years, and a catastrophic failure occurred to the gearbox on one vessel in 1998 (Alberni). A fractured bull gear tooth was found on another vessel in 1999 (Surrey). Excessive wear and damages to the aft sterntube bearings have also occurred. An investigation of the failures was conducted to determine the source(s) of the failures and the most prudent and economical course of action. To accomplish this objective the component and system design of the shafting system were reviewed and assessed, a theoretical alignment and vibration analysis was conducted, and the alignment condition and torsional vibration characteristics were measured. This paper presents the results of the work conducted on the failure investigation and realignment.

PROPULSION SYSTEM ARRANGEMENT

The "C" Class Ferries have a specialized double-ended single screw propeller configuration. Figure 1 is a schematic of the propulsion arrangement. Each of the 4 bladed controllable pitch propellers are driven by diesel engines rated at 5,930 HP at 425 RPM, through single reduction gearboxes (ratio of 2.16:1). The gearboxes are connected via quill shafts that are concentric with the pinion shafts. The pinion shafts are engaged via clutches at the end of each of the quill shafts. This arrangement enables one or both of the diesel engines to drive either one or both of the propellers. There are two normal operating modes: Mode 1: Both engines are driving one propeller, used during transit; and Mode 2: Both engines are driving both propellers, used in the harbor and during docking.

The main gear shaft is supported by two journal bearings. Each quill shaft is supported by two roller bearings located at the ends of the shafts and mounted in the clutch housings. The roller bearing nearest to the engine is a "fixed type", and the other roller bearing outer ring is not fixed. The hollow pinion shafts are each supported by two journal bearings located on either side of the pinion. The pinions are maintained in their axial position by "thrust stones" mounted on either side of the pinion. The thrust collar is mounted on the main gear shaft on the propeller side of the gearbox. Each propulsion shaftline is approximately 188 ft. long, and is supported by two sterntube bearings and seven lineshaft bearings. The original clutches were replaced by heavier clutches in 1991, on four of the five vessels. A picture of a gearbox is shown in Photo 1. An installation of a pinion shaft, with clutch attached, is shown in Photo 2. Figures 2 and 3 are schematics of the propulsion shafting and quill-pinion shafting arrangement, respectively. A section of the intermediate shaft is shown in Photo 3. The component parameters of the propeller, clutch, flexible coupling, and gearbox are listed in Table 1. The bearing parameters are listed in Table 2 and Table 3.

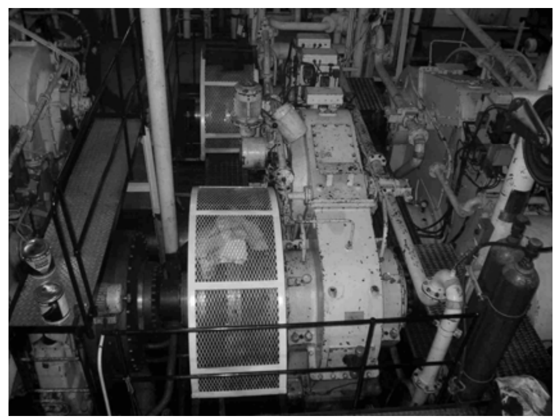

Photo 1 Gearboxes and Coupling to No. 1 Engine

Photo 2 Installation of Pinion Shaft with Clutch

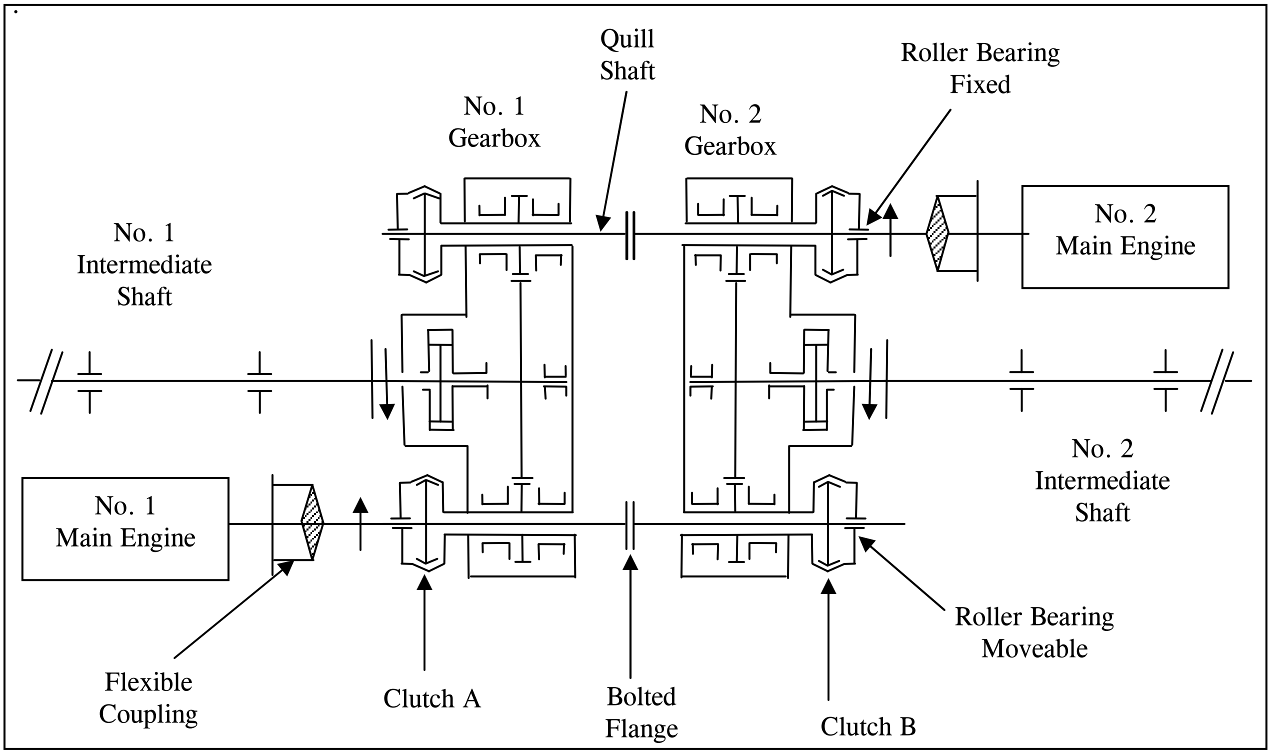

Figure 1 Propulsion System Arrangement Schematic

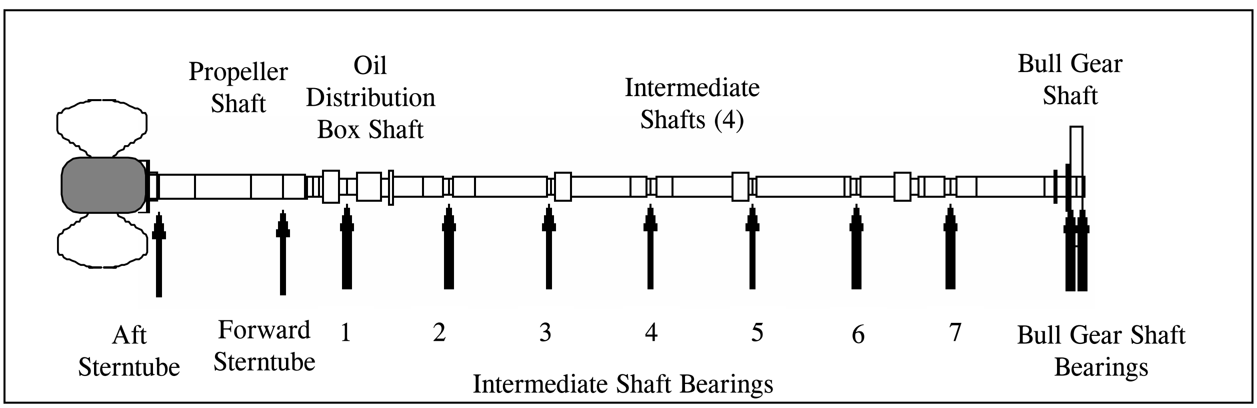

Figure 2 Propulsion Shaftline Model Schematic

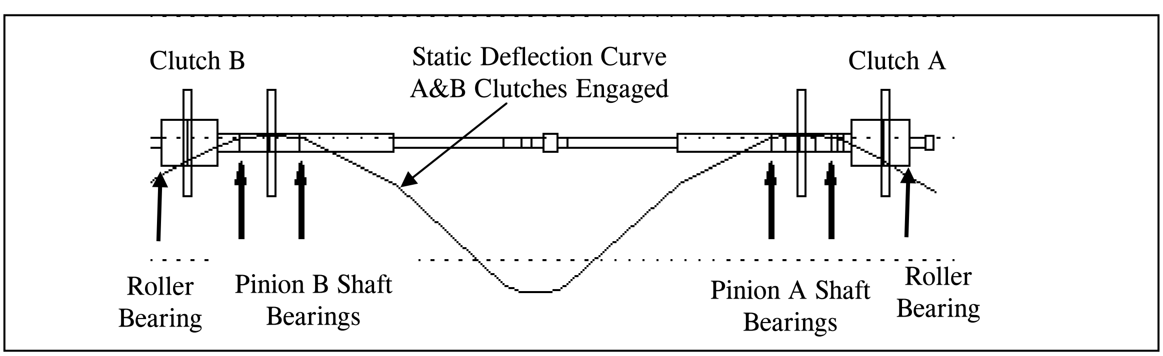

Figure 3 Quill-Pinion Shaft Model Schematic

Photo 3 Intermediate Shafting

Table 1 Shaftline Component Parameters

Propeller	
Total Mass in Air	35,500 lbs
Diameter	150 inches
Number of Blades	4
Design RPM	196
Clutch	
Post-1991 Mass	8,000 lbs
Pre-1991 Mass	5,300 lbs
Main Gear	
Single Reduction	2.164:1
Mass	10,400 lbs
Diameter	95 inches
Pinion	
Mass	3,200 lbs
Diameter	43 inches

Table 2 Propeller Shafting Bearing Parameters

	Aft Sterntube	Fwd. Sterntube	Lineshaft
Type	Journal	Journal	Roller
Lubrication	Water	Water	Grease
Length (inches)	79	50	NA
Diameter (inches)	22.0	21.7	15
Length/Dia.	3.6	2.7	NA
Clearance (mils)	40	35	*6*
Rated Pressure (psi)	40	40	NA
Rated Load (lbs)	69,400	51,000	38,000

Table 3 Gear Shafting Bearing Parameters

	Gear Shaft	Pinion Shaft	Quill Shaft
Type	Journal	Journal	Roller
Lubrication	Oil	Oil	Grease
Length (inches)	8.6	8.6	NA
Diameter (inches)	16.6	16.9	7.9
Length/Dia.	0.5	0.5	NA
Clearance (mils)	*25*	*25*	*8*
Rated Pressure (psi)	*120*	*120*	*NA*
Rated Load (lbs)	*30,000*	*17,500*	*11,300*

Note: Numbers in *italics* are estimates.

ALIGNMENT CRITERIA

As part of the failure investigation a comprehensive set of alignment criteria were developed, and are shown in Table 4. Criteria were developed based upon industry standards, classification requirements, manufacturer's specifications, and the author's experience [1,2,3,4]. These criteria were compared to the measured alignment condition to determine the possible source(s) of failure.

Table 4 Alignment Criteria

Description	Design Limit
Relative Shaft Deflections Across Sterntube Bearings Across Pinion and Gear Shaft Bearings	 0.020 inches 0.012 inches
Shaft Bending Stress	6,000 psi
Bearing Athwartships Offsets Sterntube Bearings Pinion and Gear Shaft Bearings	 0.020 inches 0.012 inches
Minimum Bearing Loads Fwd. Stern Tube Bearing Lineshaft Bearings Pinion Shaft Bearings	 9,000 lbs. 4,700 lbs. 2,000 lbs.
Difference in Load Between Aft and Forward Gearshaft Bearings Vertical Athwartships	 4,700 lbs. 3,000 lbs.
Allowable Vertical Loads	
Aft Sterntube	69,400 lbs.
Forward Sterntube	51,000 lbs.
Lineshaft Bearings	38,000 lbs.
Main Gearshaft Bearings	17,000 lbs.
Pinion Shaft Bearings	17,500 lbs.
Quill Shaft Bearings	11,300 lbs.
Allowable Athwartships Loads	
Aft Sterntube	20% of Vertical Load
Forward Sterntube	30% of Vertical Load

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 matrix, bearing loads, and the shaft stresses and deflection. 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 failures, and to assess the shaftline arrangement. Since the failures were found to be a result of misalignment, the results of the vibration analysis and measurements are not presented in this paper.

Propulsion Shaftline Flexibility

The ratio of the distance between shaftline bearings to shaft diameter provides an indication of the flexibility of a shaftline, and are shown in Table 5.

Table 5 Shaft Span Ratio of Bearings

Span	Ratio of Span to Shaft Diameter
Aft Sterntube to Fwd. Sterntube	13
Fwd. Sterntube to No. 1 Lineshaft	9
Between Lineshaft Bearings	15
No. 7 Lineshaft to Aft Gearshaft	17

The span to diameter ratio between two successive bearings is recommended to be between 12 and 22 [5]. The distance from the aft lineshaft bearing to the forward sterntube bearing is lower than recommended, which results in a relatively stiff shaft in this region. For example, a relative displacement of 0.020 inches of the forward sterntube bearing, results in a load change of 5,080 lbs on the forward sterntube bearing and 4,700 lbs on the No. 1 lineshaft bearing. It is important to ensure that the static load on the forward sterntube bearing be sufficient to ensure it remains loaded under all operating conditions. Therefore, the specified minimum load for the sterntube bearing is 9000 lbs (see Table 4).

Table 6 presents the bearing reaction influence numbers for the propulsion shaftline. They can be applied to both vertical and athwartships bearing loads. The intermediate shafting is relatively flexible, with influence numbers in the range of 100 lbs/mil for lineshaft bearings 2 to 7.

Table 6 Bearing Reaction Influence Numbers for Propulsion Shaftline
Change in Bearing Load by Dropping Bearing 1 mil (0.001") [lbs/mil]

Bearing	Aft Sterntube	Fwd. Sterntube	Line. No.1	Line. No.2	Line. No.3	Line. No.4	Line. No.5	Line. No.6	Line. No.7	Aft Gear	Fwd. Gear
Aft Sterntube	-21	67	50	6	-1	0	0	0	0	0	0
Fwd. Sterntube	67	-254	235	-59	13	-3	1	0	0	0	0
Line. No. 1	-50	235	-267	112	-38	10	-2	1	0	0	0
Line. No. 2	6	-59	112	-112	80	-33	8	-2	1	0	0
Line. No. 3	-1	14	-38	80	-107	77	-30	8	-2	1	-1
Line. No. 4	0	-4	10	-33	77	-102	77	-33	8	-6	5
Line. No. 5	0	1	-2	8	-30	77	-107	80	-32	23	-17
Line. No. 6	0	0	1	-2	8	-33	80	-108	79	-101	77
Line. No. 7	0	0	0	1	-2	8	-32	79	-106	311	-259
Aft Gear	0	0	0	0	1	-6	23	-101	311	-1,962	1,734
Fwd. Gear	0	0	0	0	-1	5	-17	77	-259	1,734	-1,538

The flexibility factor provides an indication of the flexibility of the shaftline from the forward lineshaft bearing to the gearbox. The flexibility factor (allowable setting error) is defined as the allowable difference between the gear bearing loads divided by the difference between the influence number of the forward slow-speed gear bearing on itself and the after slow-speed gear bearing on itself. An absolute minimum acceptable value for the flexibility factor has been recognized to be 0.01 inches [6]. The flexibility factor is calculated as follows:

$$FF = \Delta R / (I_{11} - I_{22}) = 0.011 \text{ inches.}$$

Where,

FF = Flexibility Factor (minimum allowable of 0.01 inches)

ΔR = allowable difference between the two slow-speed bearing static reactions (4,700 lbs)

I_{11} = bearing reaction influence number of the forward slow-speed gear bearing on itself

I_{22} = bearing reaction influence number of the aft slow-speed gear bearing on itself

Therefore, the flexibility factor is acceptable for the shafting system.

Quill-Pinion Shaft Flexibility

Tables 7 and 8 present the bearing reaction influence numbers for the quill-pinion shaft with clutch A engaged, Clutch B engaged, and both Clutch A & B engaged. These influence numbers indicate that the quill-pinion shafts are relatively flexible if only one clutch is engaged. When both clutches are engaged the system is about 3 times stiffer. For example, if only one clutch is engaged and both the pinion bearings are raised 20 mils, the load on each pinion bearing increases about 2,000 lbs; however if both clutches are engaged the corresponding increase is 6,000 lbs.

Table 7 Quill-Pinion Shaft Bearing Reaction Influence Numbers [lbs/mil]
Change in Bearing Load by Dropping Bearing 1 mil

Quill-pinion Shaft – Clutch A Engaged

Bearing	Roller	Pinion 2A Gear Side	Pinion1A Engine Side
Movable Roller	-14	106	-92
Pinion 2A Gear Side	106	-808	702
Pinion1A Engine Side	-92	702	-610

Quill-pinion Shaft – Clutch B Engaged

Bearing	Pinion 2B Gear Side	Pinion 2A Engine Side	Fixed Roller
Pinion 2B Gear Side	-604	695	-91
Pinion 2A Engine Side	695	-800	105
Fixed Roller	-91	105	-14

Table 8 Quill-Pinion Shaft Bearing Reaction Influence Numbers [lbs/mil]
Quill-pinion Shaft – Clutch A& B Engaged

Pinion Bearing	2B Gear Side	1B Engine Side	2A Gear Side	1A Engine Side
Gear Side (2B)	-981	1,210	-886	657
Engine Side (1B)	1,210	-1,503	1,181	-888
Gear Side (2A)	-886	1,181	-1,515	1,220
Engine Side (1A)	657	-888	1,220	-990

ALIGNMENT MEASUREMENT TECHNIQUE

The strain gauge technique was used to measure the alignment condition of the lineshafts [7,8,9,10]. The primary advantages of using this technique for these shafting systems are:

- Accurate measurement of bearing loads
- Measurement of athwartships and vertical bearing loads.
- Measurement of inaccessible bearing loads, such as sterntube and bull gear shaft bearings.
- 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 of the entire shafting system can be re-measured within minutes.

A sensitivity analysis was conducted on the measurement of the bearing reactions to determine the range of error (measurement resolution) using the strain gauge technique. The following were the estimated errors:

- $\pm$2,000 lbs for the sterntube bearings and No. 1 lineshaft bearing
- $\pm$1,000 lbs for the lineshaft bearings
- $\pm$1,500 lbs for the gear shaft bearings

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

MV QUEEN OF ALBERNI PROPULSION SHAFTLINE ALIGNMENT

Alignment Measurements

Table 9 lists the measured bearing loads on the MV Queen of Alberni, which had a catastrophic failure of the No. 2 bull gear teeth in 1998. Table 10 lists the measured loads on the No. 1 shaft when the vessel was loaded (15 semi-trailers, 8 trucks, and 63 cars). Figure 4 shows the results of jack-up loads tests on the No. 7 and 6 bearings on the No. 1 shaft. The following conclusions are drawn from these measurements:

- The difference between the aft and forward gear bearing vertical loads was excessive on the No. 1 and 2 shafts. The difference between the aft and forward gear bearing athwartships loads was excessive on the No. 2 shaft. This condition is considered to be the primary cause of the bull gear tooth failure in 1998.
- Lineshaft bearing No. 6 on the No. 1 shaft was top-loaded/unloaded. This condition can result in excessive lateral shaft vibrations, and is considered to be the source of failure of the No. 5 bearing on this shaftline.
- The vertical load on the forward sterntube bearing on the No. 2 shaft is too low. This can result in damages to the sterntube bearings and aft intermediate shaft bearings, due to excessive shaft lateral vibrations.
- The change in the bearing loads between the loaded and unloaded condition was not significant, such that an unacceptable alignment condition would not result if the bearing loads were satisfactory.
- The No. 7 lineshaft bearing is overloaded on the No. 1 shaftline.
- The jack-up loads test results agree with the strain gauge alignment measurements.

Table 9 MV Queen of Alberni Propulsion Shaft Bearing Loads: Vessel Unloaded (Nov. 2001)

Bearing	**No. 1 Shaftline Load (lbs)**	
	Vertical	Athwartships
Aft Sterntube	53,634	-127
Fwd. Sterntube	12,677	-278
Line. No. 1	13,947	1,171
Line. No. 2	17,472	-1,157
Line. No. 3	19,110	-68
Line. No. 4	15,164	681
Line. No. 5	24,724	-387
Line. No. 6	**-2,919**	3,048
Line. No. 7	**38,146**	-3,510
Aft Gear	4,156	1,786
Fwd. Gear	10,601	-1,159
Aft-Fwd Gear	***-6,446***	*2,945*
Bearing	**No. 2 Shaftline Load (lbs)**	
	Vertical	Athwartships
Aft Sterntube	54,836	254
Fwd. Sterntube	**5,936**	-1,672
Line. No. 1	18,568	2,094
Line. No. 2	19,202	-558
Line. No. 3	21,761	-260
Line. No. 4	7,531	66
Line. No. 5	26,971	275
Line. No. 6	12,058	-1,036
Line. No. 7	21,357	1,788
Aft Gear	5,333	**-6,114**
Fwd. Gear	13,162	**5,163**
Aft-Fwd Gear	***-7,829***	***-11,277***

Note: Bold Numbers indicate unacceptable loads.

Table 10 MV Queen of Alberni Propulsion Shaft Bearing Loads: Vessel Loaded (Nov. 2001)

Bearing	**No. 1 Shaftline Load (lbs)**	
	Vertical	Athwartships
Aft Sterntube	52,948	122
Fwd. Sterntube	16,335	-1,338
Line. No. 1	9,670	2,196
Line. No. 2	19,233	-1,341
Line. No. 3	18,174	-196
Line. No. 4	17,184	1,335
Line. No. 5	21,110	-2,066
Line. No. 6	**244**	3,002
Line. No. 7	**36,883**	-3,009
Aft Gear	4,958	3,954
Fwd. Gear	9,972	-2,659
Aft-Fwd Gear	***-5,014***	***6,613***

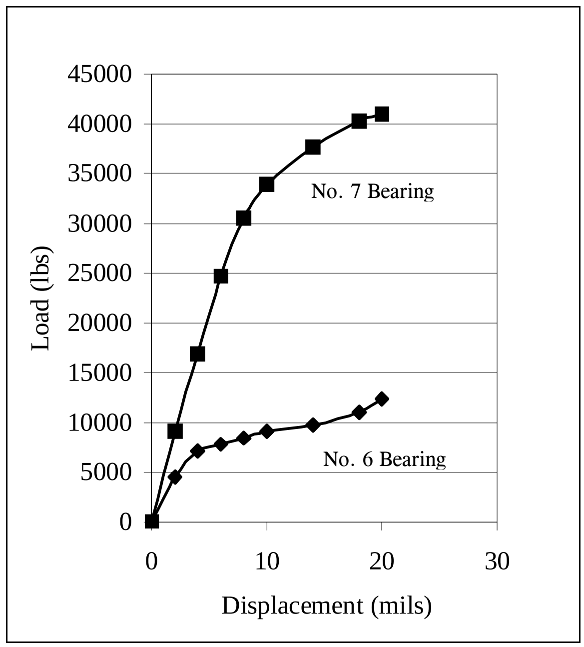

Figure 4 MV Queen of Alberni No. 1 Shaft Jack-Up Load Tests

Realignment

The required adjustments to the bearing positions to provide for an acceptable alignment condition were estimated using the bearing reaction influence numbers and the measured bearing loads. A combination of bearing adjustments was chosen that resulted in both a satisfactory alignment condition and cost-effect procedure. The shafts were realigned in January 2002. Tables 11 and 12 list the measured bearing loads after realignment, which indicates an acceptable alignment condition for both shaftlines. Subsequent to the realignment of the No. 1 shaftline, repair/refurbishing of the lineshaft bearings, shaft journals and clutches was conducted. This work resulted in excessive athwartships bearings loads on the bull gear bearings. Subsequent checks of the bull gear – pinion contact using blue dye indicated near full face contact on the No. 2 gear teeth, but as little as 50% contact on the No. 1 gear teeth. There is a possibility that the two gearboxes may be misaligned with each other. If this is the case, the realignment procedure would require re-positioning one gearbox and likely a main engine. Further investigation is being conducted to determine the source of the change in the alignment condition on the No. 1 shaft and the most prudent and economical course of action.

Table 11 MV Queen of Alberni Propulsion Shaft Bearing Loads: Realigned

No. 1 Shaftline January 2002 No. 5 Down 0.120"; No. 6 Athwartships -0.040"		
Bearing	**Load (lbs)**	
	Vertical	Athwartships
Aft Sterntube	53,445	127
Fwd. Sterntube	13,406	278
Line. No. 1	13,209	-1,288
Line. No. 2	18,252	1,438
Line. No. 3	16,008	-407
Line. No. 4	23,579	-115
Line. No. 5	12,050	1,190
Line. No. 6	8,512	-3,131
Line. No. 7	32,543	2,944
Aft Gear	7,476	-201
Fwd. Gear	8,231	-836
Aft-Fwd Gear	-755	635

Table 12 MV Queen of Alberni No. 2 Propulsion Shaft Bearing Loads: Realigned

No. 2 Shaftline January 2002 No. 2 Up 0.040", No. 4 Up 0.080" No. 6 Atwhartships+0.110" No. 5 Atwhartships+0.130"		
Bearing	**Load (lbs)**	
	Vertical	Athwartships
Aft Sterntube	53,172	306
Fwd. Sterntube	14,489	-1,522
Line. No. 1	7,448	1,565
Line. No. 2	27,712	-33
Line. No. 3	12,402	723
Line. No. 4	17,469	-3,682
Line. No. 5	18,548	2,112
Line. No. 6	17,118	3,670
Line. No. 7	19,147	-4,133
Aft Gear	9,827	-1,153
Fwd. Gear	9,380	2,147
Aft-Fwd Gear	447	-3,300

Visual Inspection

The bull gear and pinion teeth were visually inspected through the ports fitted to the top of the gear boxes. Photo 4 shows the wear pattern observed on the No. 2 bull gear. As low as 20% contact was observed on the engine side of the gear teeth, with heavier pitting in this area. At the time of this observation the bull gear had been in service for approximately 2 years, subsequent to replacement because of a catastrophic failure in 1998. Photo 5 shows the corresponding wear pattern on the No.1 bull gear. About 80% contact was observed on the loaded side of bull gear with mild polishing and light pitting. Of the 8 shaftlines (4 vessels) investigated to date, 4 bull gears were found to have about 20% to 30% teeth contact wear, in each case measurements indicated a significant misalignment between the bull gear shaft and intermediate shaft. Similarly the other 4 bull gears showed near full contact wear, and the corresponding measurements indicated acceptable alignment.

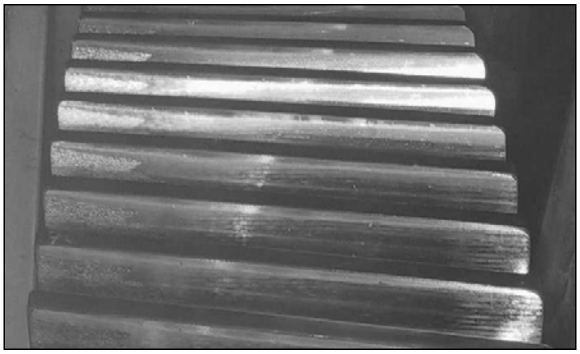

Photo 4 No. 2 Bull Gear Teeth (Excessive Athwartships Misalignment)

Photo 5 No. 1 Bull Gear Teeth (Acceptable Athwartships Alignment)

Photo 6 shows the bottom half of an aft bull gear shaft bearing. The shaftline was realigned one year previous to this photo being taken. The bearing wear pattern indicates that an athwartships misalignment was present, as well as an acceptable alignment. It is believed that the realignment resulted in the acceptable alignment wear pattern, and operations prior to the realignment produced the athwartships misaligned wear pattern.

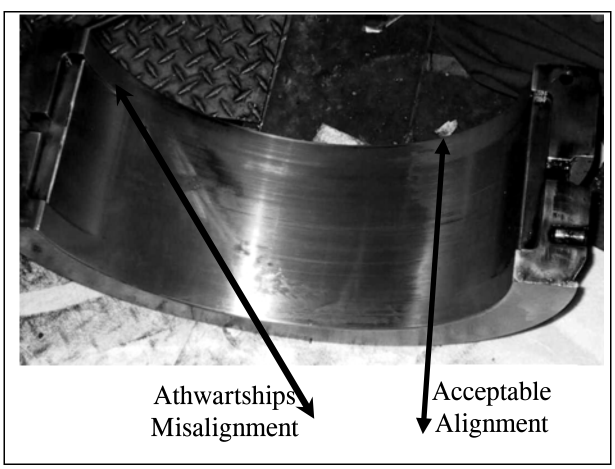

Photo 6 Bottom of Aft Gear Shaft Bearing

Photo 7 shows a damaged race of the No. 5 intermediate shaft roller bearing (3rd aft of gear box). This damage is consistent with the measured alignment condition. Alignment measurements showed that the No.6 bearing was unloaded/top-loaded, which can result in excessive vibration of the shaftline, particularly from the No. 7 to No. 5 bearings. The realigned condition resulted in more than 8,000 lbs load on the No. 6 bearing, which should prevent such damages in the future.

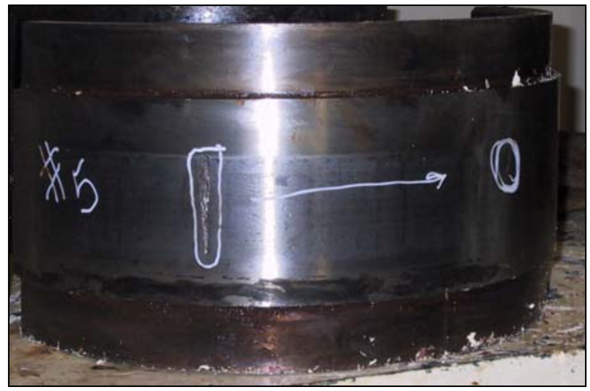

Photo 7 Damaged Race of Roller Bearing

MV QUEEN OF ALBERNI QUILL-PINION SHAFTLINE ALIGNMENT

Quill-Pinion Shaft Alignment Measurements

Table 13 lists the measured bearing loads with only one clutch engaged on each quill shaft. Measurements could not be taken with both clutches engaged because the turning gear was not capable of turning the shafts when both clutches were engaged. The theoretical loads when both clutches are engaged are listed in Table 14. All bearing loads are acceptable with either Clutch A or Clutch B engaged. Pinions 1B and 2A are unloaded or top-loaded when both clutches are engaged. This is considered unsatisfactory and can result in wiping of the pinion shaft bearings and improper bull gear – pinion teeth contact.

Table 13 Measured Pinion-Roller Bearing Loads

	Clutch B Engaged			
Bearing	**Load (lbs)**			
	No. 1 End		**No. 2 End**	
	Vert.	Horiz.	Vert.	Horiz.
Pinion Fwd	7,712	-1,045	9,221	-1,170
Pinion Aft	8,187	1,202	6,451	1,328
Fixed Roller	5,158	-158	5,386	-175
	Clutch A Engaged			
	Vert.	Horiz.	Vert.	Horiz.
Pinion Fwd	5,005	310	5,093	-78
Pinion Aft	6,972	-2,369	6,305	592
Fixed Roller	9,081	2,059	9,660	-515

Table 14 Theoretical Pinion Shaft Bearing Loads with Both A& B Clutches Engaged

Bearing	Vertical Load (lbs)	
	1991-Present Clutches	Original Clutches
Pinion 2B	15,008	10,058
Pinion 1B	-833	1,530
Pinion 2A	-1,035	1,328
Pinion 1A	15,458	10,440

The unsatisfactory pinion bearing loads when both clutches are engaged is a result of transferring the load from the roller bearing of the disengaged clutch, to the pinion bearings when the second clutch is engaged. This results in the over-hanging weight of the clutch acting on both pinions. When torque is applied to the pinion from the engine, the vertical load on the pinions driven by the No. 2 engine increases and thereby provides for a more favorable loading condition. However, the corresponding loads on the pinions driven by the No. 1 engine are upwards, and thereby produce a worse loading condition.

To provide for acceptable pinion bearing loads with both clutches engaged, pinion bearings 1B and 2A would be required to be shimmed up 20 mils. Unfortunately, this would result in an unacceptable pinion bearing loads when only one clutch is engaged. Installing a bearing on the quill shaft between the two gearboxes would require the bearing to be top-loaded, and would double the stiffness of the quill-pinion shafting system, both of which are considered to be detrimental. An option that shows promise would be to add a weight to the flange on the quill shaft. The added mass would help counter-balance the over-hanging weight of the clutches. It is estimated that a mass of 2,200 lbs. added to the flange on the quill shaft would produce acceptable bearing loads. Consideration should also be given to installing lighter clutches.

Visual Inspection

Pinion shaft bearings are inspected upon removal from the gearbox. A number of pinion shaft journal bearings have been found to be wiped in varying degrees over the last 10 years. Typically, bearings furthest from the clutch are wiped at the top, and the bearings closest to the clutch are wiped at the bottom. Photo 8 shows the top half of a pinion shaft bearing. Wiping of the white metal has occurred at of the top of a bearing. Photo 9 shows the bottom half of the bearing, where the wear pattern indicates that full contact is not achieved and over-loading has resulted in wiping at one end of the bearing. This is indicative of clutches that are too heavy, such that over-hanging weight of the clutch results in top-loading of the pinion shaft bearing furthest from the clutch, and over-loading of the bearing closest.

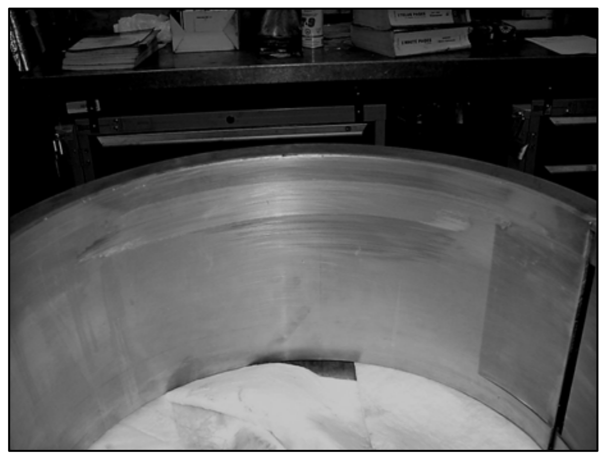

Photo 8 Top Half of Pinion Shaft Bearing

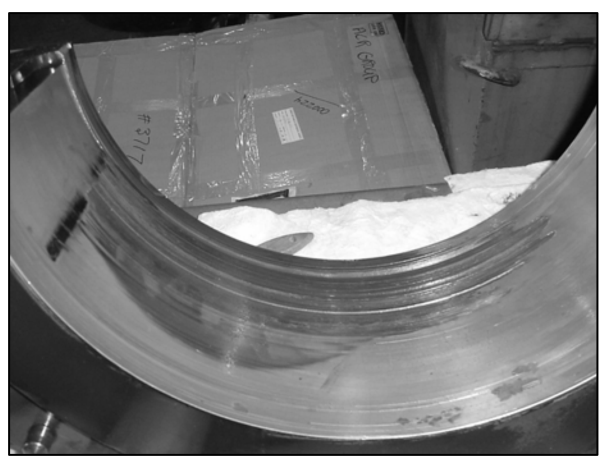

Photo 9 Bottom Half of Pinion Shaft Bearing

CONCLUSIONS

The failures of the bull gear teeth and poor pinion-bull gear tooth contact can be attributed to misalignment of the main gear shaft with the intermediate shafting. Heavy clutches result in wiping of the pinion shaft bearings, and may be a contributing factor to the poor pinion-bull gear teeth contact. Failures of the lineshaft bearings can be attributed to misalignment of the intermediate shafting resulting in top/under-loaded bearings. Excessive wear and damages to the aft sterntube bearings can be caused by unloading of the forward sterntube bearing. This condition may also result in failure of the aft lineshaft bearing.

The alignment measurements were consistent with the observed failures and bull gear teeth wear patterns. Repositioning the intermediate shaft bearings can result in an acceptable alignment condition.

A program is currently under-way to realign the propulsion shafting on the "C" class ferries. It is anticipated that this program will result in a significant reduction in maintenance costs and an increase in the reliability of the propulsion system.

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DISCLAIMER

The views expresses in this paper are those of the authors and not necessarily the British Columbia Ferry Services Inc.

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