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CRACKED SHAFT DETECTION ON LARGE VERTICAL NUCLEAR REACTOR COOLANT PUMP

L. Stanley Jenkins
Westinghouse Electric Corporation
Cheswick, Pennsylvania 15024

Due to the difficulty and radiation exposure associated with examination of the internals of large commercial nuclear reactor coolant pumps (RCP's), it is necessary to be able to diagnose the cause of an excessive vibration problem quickly without resorting to extensive trial and error efforts. Consequently, it is necessary to make maximum use of all available data to develop a consistent theory which locates the problem area in the machine.

This type of approach was taken at Three Mile Island (TMI), Unit #1, in February 1984 by the author to identify and locate the cause of a continuously climbing vibration level of the pump shaft. The data gathered necessitated some in-depth knowledge of the pump internals to provide proper interpretation and avoid misleading conclusions. Therefore, the raw data included more than just the vibration characteristics.

Pertinent details of the data gathered is shown herein and is necessary and sufficient to show that the cause of the observed vibration problem could logically only be a cracked pump shaft in the shaft overhang below the pump bearing.

MACHINE DESCRIPTION

TMI #1 is a Babcock and Wilcox Nuclear Steam Supply System (NSSS) with four 93A spool piece design reactor coolant pumps designed and manufactured by Westinghouse, Electro-Mechanical Division. Pump head and flow are about 107 m/7.20 m^3/sec. at 288°C in the loop. The pumps are vertical and driven by Allis Chalmers 6700 kw (rated) motors (totally enclosed, water/air cooled). The spool piece design is a common feature to this type of pump whereby an 457 mm long section of shaft can be removed to permit maintenance of the seals without removal of the motor and breaking of the pump/motor alignment. The 93A pumps in this plant are a special adaption of the very common series of 93A pumps found in the majority of Westinghouse NSSS's. A cutaway view of the typical 93A RCP is shown in Figure 1. Detail differences between the TMI #1 pumps and the typical 93A pump design are insignificant relative to the discussion details in this paper.

The TMI #1 Reactor Coolant System (main primary piping only) is shown diagramatically in Figure 2. The range of flows possible with this system design (as a percentage of total four pump operation "best estimate" flow) is depicted in Table 1 for typical combinations of pump operation. The negative values indicates reverse flow in the loop. Increasing flow rates generate radial loads (on the impeller, fixed relative to the casing) which increase approximately as the square of the flow rate. This load is estimated to be in excess of 2300 kgf for single pump operation at TMI #1.

Vibration probe locations are shown in Figure 1. This is the standard probe location for all RCP's of this design. The shaft probes "see" the 347 stainless steel shaft surface. Nominal pump shaft diameter is 200 to 230 mm.

INITIAL DETECTION OF THE PROBLEM AND VIBRATION CHARACTERISTICS OBSERVED

In late January 1984, EMD Engineering was alerted by plant personnel of a worsening vibration problem on "1B" RCP (referred to as B). The shaft vibration had been in the 225 - 280 micron (9 - 11 mil) range for several months prior to this, during which time the pump was run primarily alone. Approximately 1400 hours had been logged on the pump in this mode. A tape of the vibration signals was made prior to shutdown of the pump as the levels neared 750 microns (30 mils) and the rate of increase climbed to 25 microns/hr (1 mil/hr) (instruction book limits, single pump operation).

The investigation into the cause of the problem took the following direction:

1. Compare frequency content, vibration levels, etc., on all RCP's in the plant.

2. Determine the effect of balance weight changes on B RCP.

3. Run the motor on B RCP "no-load" and compare data to the loaded operation.

The following was noted during these tests:

1. Comparison of vibration levels between pumps indicated nothing unusual. However, B RCP contained a large amount of 2X vibration component not present to any significant degree in the other RCP's. Table 2 shows the vibration level and frequency content comparison.

2. A reduction in 1X (running speed = 20 Hz) motion due to the addition of weight to a pump coupling bolt had no impact on the 2X component. Table 3 shows the results of this test. Response to the weight was normal (approximately 3.5 grams/micron, 40°lag).

3. During no-load motor operation, all 2X components disappeared from frame and shaft (motor) data (a probe had been mounted to read motor shaft motion specifically for this one test).

Figures 2 through 6 are samples of the waveform and spectral data taken during the above tests on B RCP.

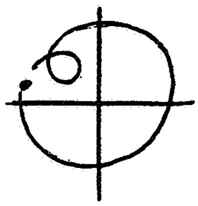
ASSESSMENT OF THE FACTS

In addition to the facts noted above, the following was known:

1. The 2X vibration level would "ratchet" up on each start of the coupled pump assembly (see Figure 7).

2. The pump and motor were found to be 812 microns (32 mils) out of alignment, laterally (centerline to centerline shift). The effect of this is depicted in Figure 8.

3. A shaft orbit with an inner loop was observed during coupled pump operation, like that shown here:



Per standard references on machinery malfunction diagnostics, it was apparent that two possibilities for the cause of the problem were pump/motor misalignment or a cracking shaft. Other possibilities such as a loose impeller, bearing deterioration, or full or partial rubs were eliminated since they were inconsistent with some or all of the data taken. The primary problem was then determining which of these two possibilities was actually correct.

RESOLUTION OF THE ALIGNMENT/CRACK CONTROVERSY

Assuming that a cracked shaft was the most logical choice of malfunctions, to rationally explain the observations and data it was necessary to: 1) choose a location for a crack in the shaft and predict the shaft motion to determine if it yields the observed shaft motion, and 2) show that the misalignment observed could not produce sufficient shaft bending to cause the observed 2X motion. The obvious method of realignment and rerunning the pump was discarded due to the time consumption for this effort, unnecessary radiation exposure, and the desire to minimize damage to the stationary parts due to additional operation.

The rationale for elimination of misalignment induced 2X is based on bearing clearances in the pump and motor plus the beneficial effect of long distances between bearings. Again, referring to Figure 8 it is easy to see that a minimum of 457 microns (18 mils) centerline shift, without any tilting of the rotor on the Kingsbury thrust bearing, is possible without incurring any bending of the rotor. Furthermore, the long distance between bearings will permit additional tilting of the motor rotor (the Kingsbury thrust bearing can accommodate this easily as will the spherically seated pump bearing) to accommodate the measured 812 micron shaft lateral misalignment with introduction of some edge loading of the motor radial tilted pad bearings. Hence, it is highly unlikely that 2X would be generated by bearing misalignment. Note that labyrinth rubbing (which was predicted on the basis of a separate evaluation of the direction of misalignment coupled with anticipated radial motion of the impeller under a fixed load in the casing) does not produce shaft bending since rubs in water wear out in a relatively short time.

The rationale for showing that the observed motion is predictable is more difficult. First of all, it is logical to place the crack below the pump bearing. (The presence of a fixed radial load in the casing, and the observation that balance weight response was normal suggest a crack below the pump bearing.) Then, by utilizing a simplified model of the pump shaft and taking the motor shaft between bearings as a "built-in beam," it is possible to relate a dynamic motion observed <u>at the pump coupling</u> to a stationary -- fixed in the casing -- hydraulic load <u>at the impeller</u>. The end result of this analytical exercise (using the principal of superposition of loadings and deflections to establish the proper phasing of the motions at the impeller and coupling) is shown in Figures 9 and 10. This effect was successful in not only predicting the observed 2X motion qualitatively but also, when the 1X motion is added in (generated by the "hinging" action of the impeller mass below the crack) the observed composite waveform was predicted. Compare Figure 10 with Figure 4 where 1X and 2X amplitudes are nearly equal.

ADDITIONAL (NON-ESSENTIAL) VERIFICATION OF CRACK EXISTENCE

For obvious reasons, there was an extreme reluctance to accept that the difficult to access pump shaft was cracked. Hence, another independent method of crack verification was desired. An Ultrasonic Test (UT) of the 3.3 m long pump shaft was attempted from the accessible coupling end. Data obtained was compared to another survey on a new pump shaft on site. The best available comparison of data (actually obtained after removal of the pump shaft) is in Figure 11. During the investigative effort and afterwards as well, it was found to be nearly impossible to predict by UT, from the coupling end of the shaft, if a crack existed in the overhang and where in the overhang it was located.

It was noted, however, that once the impeller end of the shaft was exposed that a UT of this end of the shaft produced very strong evidence of a sizeable discontinuity. The axial location predicted and its orientation relative to the coupling keyway was very close to the predictions made on the basis of vibration analysis described in this paper. From the UT check of the impeller end, approximately 60% of the cross-section was estimated to be cracked through.

CONCLUSIONS

It is evident from by this effort that it is possible to positively identify the presence of a crack in a radially loaded vertical machine by means of vibration analysis techniques and use of very simple analytical methods. A determination of response to balance weight and separation of the machine train into separate operable components to the greatest extent possible is very important to positively identifying the location of the suspected crack.

Non-destructive examination by UT of the shaft of a 93A RCP from the coupling end has not yet proven to be a reliable indicator of the presence of a crack. However, a prediction of anticipated shaft motion based on the expected location of the crack (in a bearing span or overhang) produced the observed motion with simple rotor assembly models. Knowledge of some of the characteristics of the machine is a prerequisite for this approach.

SYMBOLS AND UNITS

1 mil = 25.4 microns = .001 inch.

m	meter	kgf	kilogram (force)
°C	degrees Celsius	MPa	megapascal
kw	kilowatt	1X	running speed component of vibration
sec	second	2X	twice running speed component of vibration
mm	millimeter		

TABLE 1

TYPICAL RCP FLOWS AS A FUNCTION OF PUMP COMBINATIONS

(REFERENCE: TMI #1 OPERATING INSTRUCTIONS, VOL. 1)

PUMPS RUNNING	% FLOW PER PUMP A	B	C	D
A B C D	25	25	25	25
A B C	26	26	32	-10
A B	27	27	-4	-4
A C	33	-9	33	-9
A	33	-8	-2	-2

Since all RCP's have nominally identical impellers, any permutations of the above (e.g., CD instead of AB) are omitted as they are equivalent to that shown.

TABLE 2

SHAFT AND FRAME VIBRATION

RCP

A, B, C, D

RCP	TIME/ DATE,ETC	SHAFT VIBRATION					FRAME VIBRATION
		CPLG. WEIGHTS	COMPOSITE (ALL PASS)	1/2 X	1X	2X	
A	1719 2/3/84 54°C 2.17MP$_a$	CPLG. WEIGHTS: Unchanged since March 1979	H 30 V 24	H .4 V .4	H 25.2 V 21.6	H .9 V 1.3	H .8 V .9
B	1228 2/1/84 54°C 2.17MP$_a$	CPLG. WEIGHTS: Unchanged since March 1979	H 25 V 28	H .3 V .5	H 16.8 V 20.8	H 8.0 V 7.2	H — V —
C	1103 2/1/84 54°C 2.17MP$_a$	CLPG. WEIGHTS: Unchanged since March 1979	H 15 V 14	H .3 V .3	H 11.2 V 11.2	H .4 V .3	H — V —
D	1550 2/3/84 54°C 2.17MP$_a$	CPLG. WEIGHTS: Unchanged since March 1979	H 21 V 23	H .3 V .3	H 18.8 V 20.4	H .3 V .5	H .6 V .9

All vibration levels in mils, peak-to-peak

TABLE 3

SHAFT AND FRAME VIBRATION

CHANGES DUE TO BALANCING

RCP B

TIME/ DATE,ETC	SHAFT VIBRATION					FRAME VIBRATION
	CPLG. WEIGHTS	COMPOSITE (ALL PASS)	1/2 X	1X	2X	
1228 2/1/85 54°C 2.17MP$_a$	CPLG. WEIGHTS: Unchanged since 1979	H 25 V 28	H .3 V .5	H 16.8 V 20.8	H 8.0 V 7.2	H — V —
1938 2/3/84 54°C 2.17MP$_a$	CPLG. WEIGHTS: Add 881g to Bolt #5	H V	H .2 V .4	H 7.8 V 11.2	H 8.8 V 8.2	H .6 V 1.5

All vibration levels in mils, peak-to-peak

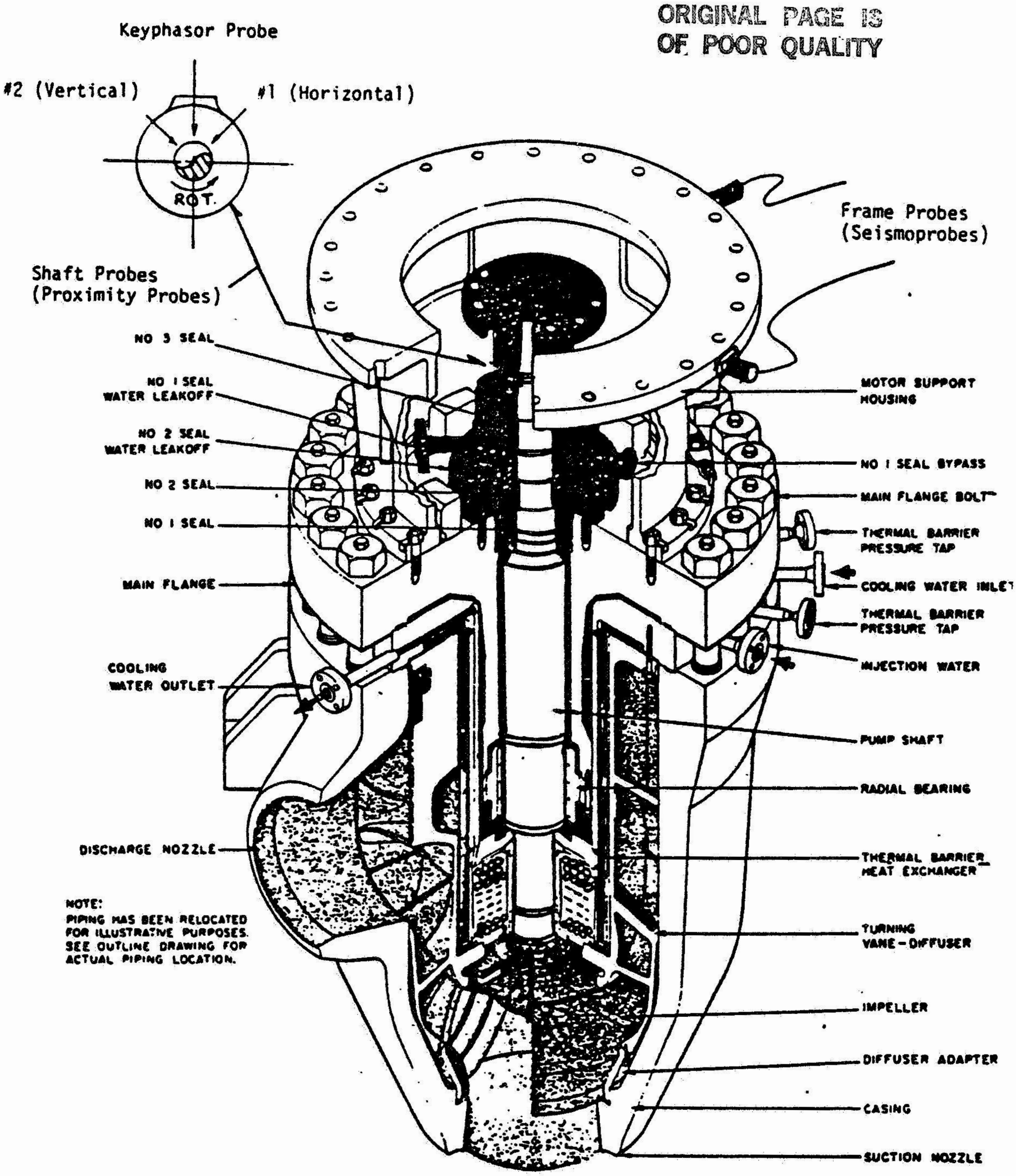
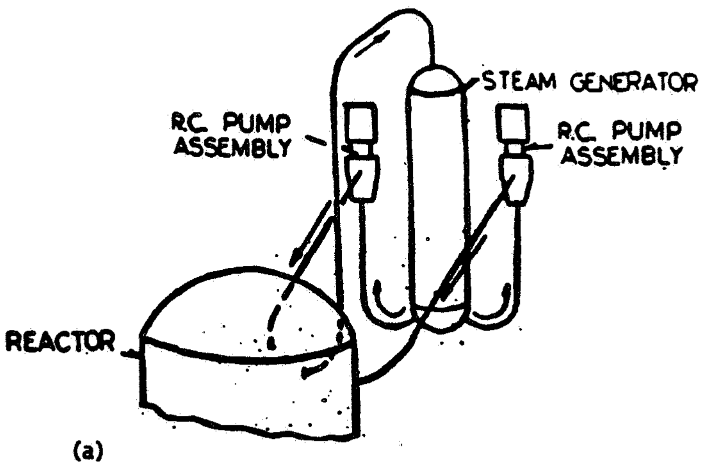
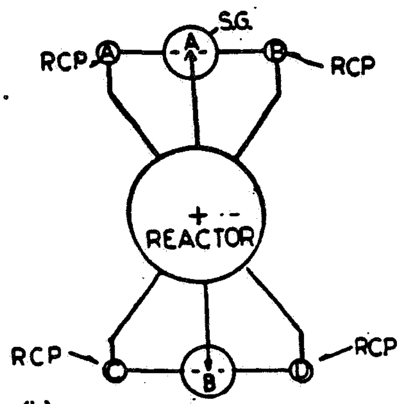


Figure 1. - Cutaway view of typical reactor coolant pump.



(a)

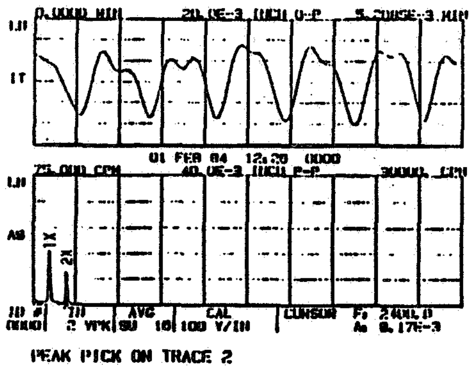


(b)

(a) Isometric view of typical primary piping at TMI #1 (one S.G. loop shown).
(b) Plan view of primary piping at TMI #1.

Figure 2. - Piping schematic.

RC PIB HORIZONTAL



INCH P-P	CPM
16.8E-3	1200.0
8.00E-3	2400.0

01-FEB-84

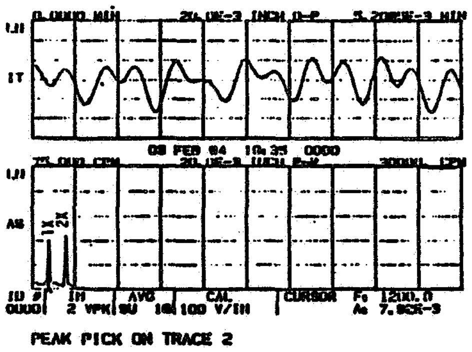
12:28:40

TIME/DATE,ETC	SHAFT VIBRATION			FRAME VIBRATION
1228	CPLG. WEIGHTS: Unchanged since March 1979			H V
2/1/84	COMPOSITE (ALL PASS): H 25, V 28			
54°C				
2.17MPa	1/2X: H .3, V .5	1X: H 16.8, V 20.8	2X: H 8, V 7.2	—

All vibration levels in mils, peak-to-peak

Figure 3. - Sample of waveform and spectral data taken on B RCP: February 1, 1984 (original data).

RC PIB HORIZONTAL



INCH P-P	CPM
7.80E-3	1200.0
8.80E-3	2400.0

03-FEB-84

19:38:05

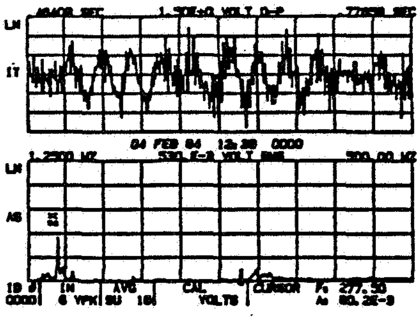
TIME/DATE,ETC	SHAFT VIBRATION			FRAME VIBRATION
1938	CPLG. WEIGHTS: Add 880g to Bolt #5			H V
2/3/84	COMPOSITE (ALL PASS): H 18, V 22			
54°C				
2.17MPa	1/2X: H .2, V .4	1X: H 7.8, V 11.0	2X: H 8.8, V 8.5	.6 1.5

All vibration levels in mils, peak-to-peak

Figure 4. - Sample of waveform and spectral data taken on B RCP: February 3, 1984 (after balance).

R C PIB

FRAME HORIZ



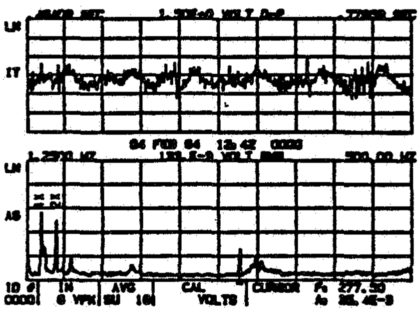
VOLT RMS	HERTZ
196. E-3	40. 000
63. 6E-3	46. 252
58. 3E-3	277. 52
58. 3E-3	297. 52

04-FEB-84

12: 39: 35

RC PIB

FRAME VERT



VOLT RMS	HERTZ
75. 8E-3	20. 001
37. 2E-3	23. 751
66. 5E-3	40. 000
34. 6E-3	277. 52

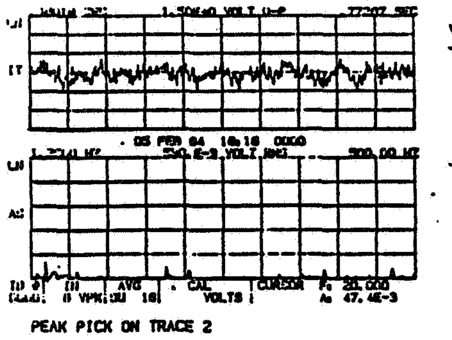
04-FEB-84

12: 43: 46

AFTER BALANCE

Figure 5. - Frame vibration spectral data: February 4, 1984 (after "balancing").

RC PIB



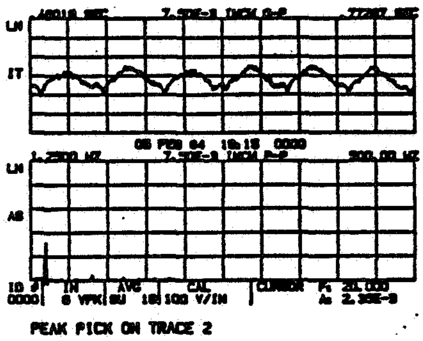
VOLT RMS	HERTZ
42.4E-3	20.001
47.7E-3	176.26
42.4E-3	470.02

Uncoupled Frame Motion

05-FEB-84

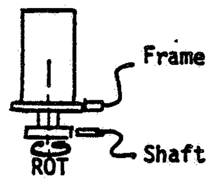
18:18:06

RC PIB



INCH P-P	HERTZ
2.33E-3	20.001

Motor
Stand
Not
Shown



Uncoupled Motor Shaft Extension Motion

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18:16:07

Figure 6. - Spectral and waveform plots from uncoupled motor run of B RCP, showing frame and shaft motion.

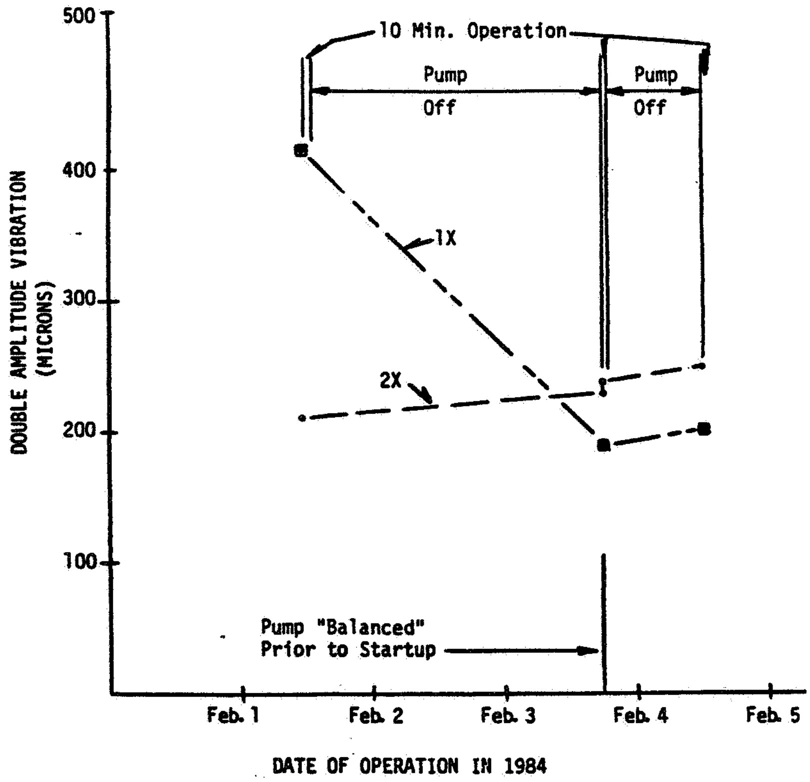

Figure 7. - Trend plot for 1X and 2X vibration levels for B RCP shaft.

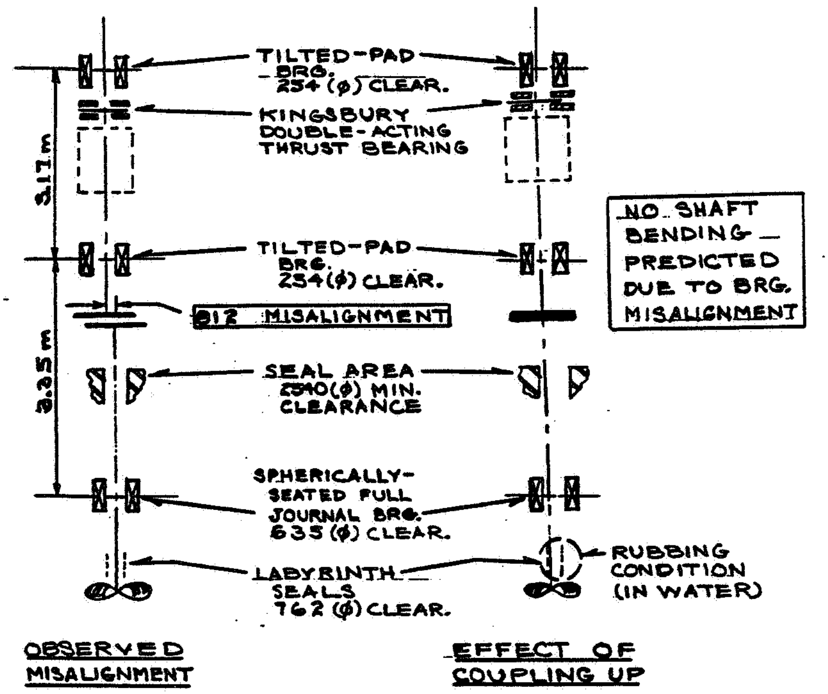

Figure 8. - Effect of 0.032-inch-bearing centerline offsets on shaft bending.

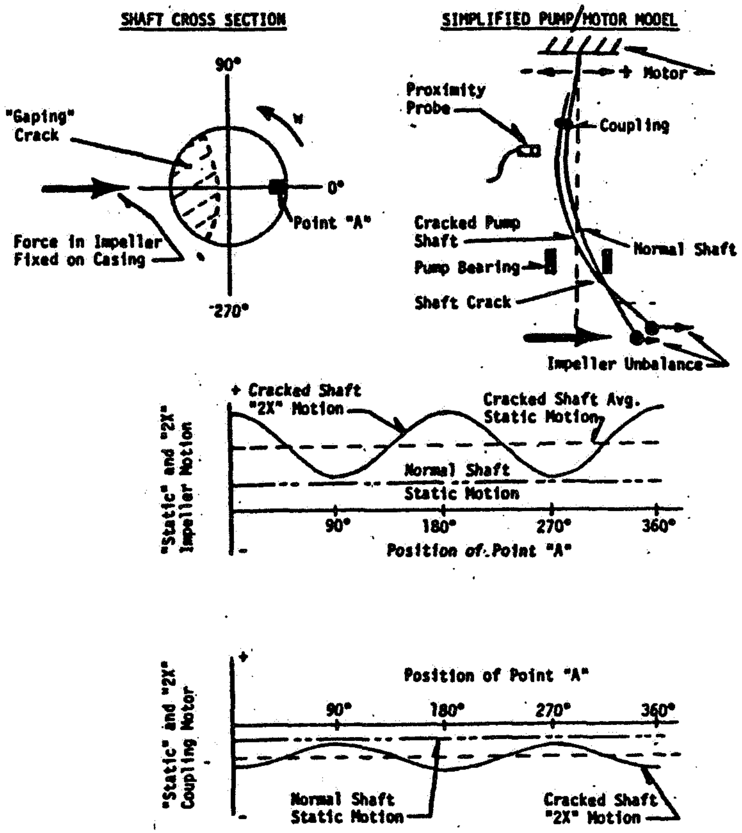

Figure 9. - Dynamic (2X) and static impeller and coupling motion.

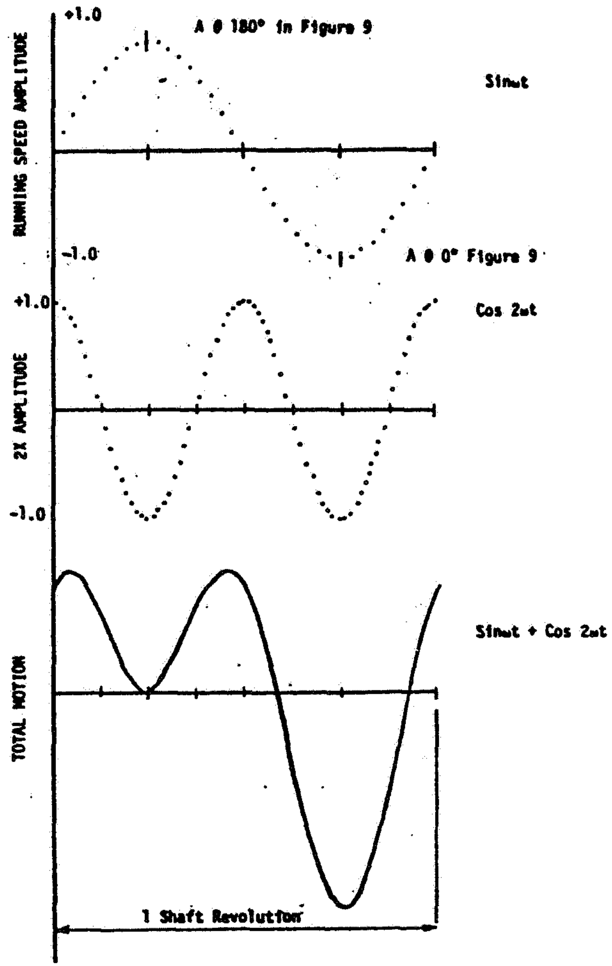

Figure 10. - Graphical summation of predicted 1X and 2X motions at pump coupling.

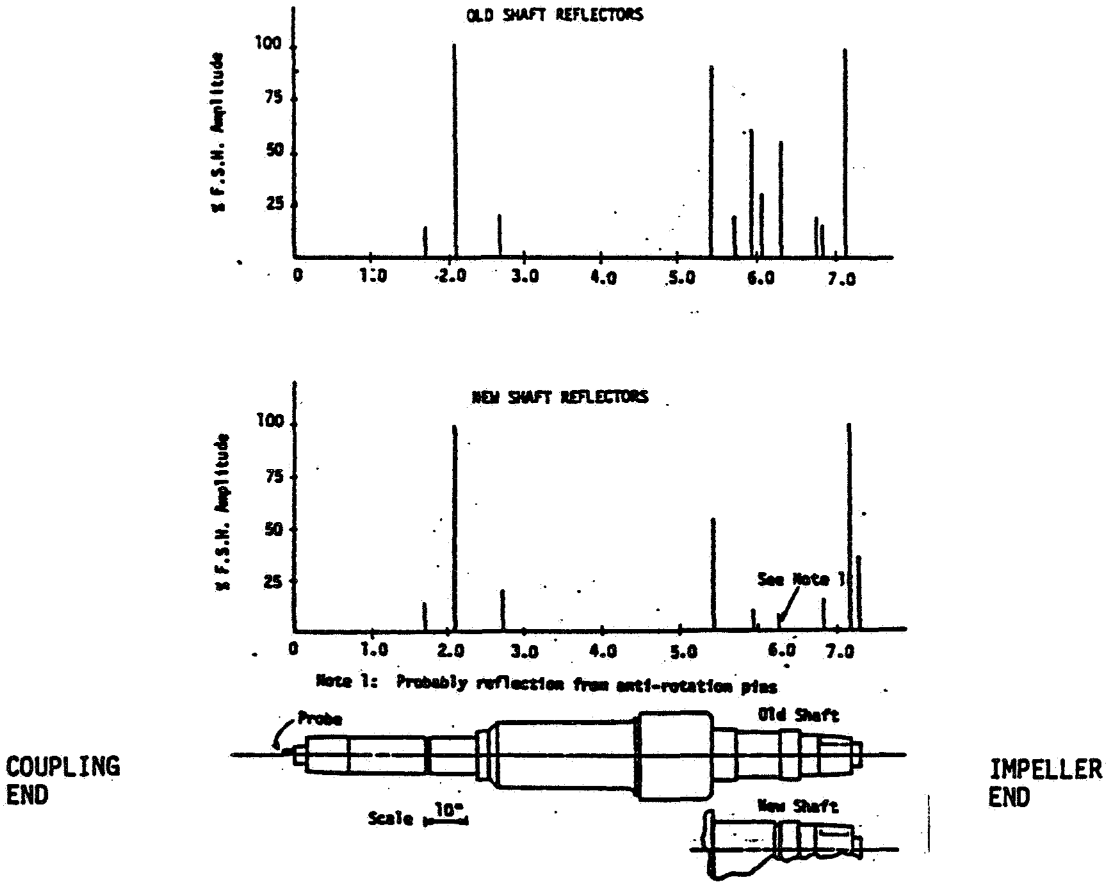

Figure 11. - Comparison of old shaft and new shaft UT reflectors.