UNITED STATES
NUCLEAR REGULATORY COMMISSION
WASHINGTON, D. C. 20555

Docket File
DCS MS-016

NOV 12 1981

Docket No. 50-306

LICENSEE: Northern States Power Company (NSP)

FACILITY: Prairie Island Unit 2 (PI-2)

SUBJECT: SUMMARY OF MEETING ON OCTOBER 29, 1981 - REACTOR COOLANT PUMP SHAFT CRACK

On October 29, 1981, the NRC staff met with NSP and Reactor Coolant Pump Manufacturer (Westinghouse Corp.) to discuss the failure of the Reactor Coolant Pump No. 21 (RCP No. 21) that was observed at PI-2 on June 11, 1981. By letter dated August 24, 1981 we requested that the licensee and pump manufacturer inform the staff of the nature of the failure in detail. This letter also transmitted a proposed agenda for a meeting to be held in Bethesda, Maryland covering areas of interest to the staff. The purpose of this meeting was to inform the NRC of the details of the failed RCP at PI-2. A list of attendees and a copy of the viewgraphs presented during the meeting are enclosed (Enclosure 1&2).

The meeting opened with a presentation of a modified agenda similar to the one proposed by the staff on August 24, 1981. The modified agenda consisted of the following.

1. Operating experience leading to the discovery of the pump failure at PI-2.

2. Operating experience of other pumps of similar design including operating history as of June 1, 1981.

3. Detailed description of the design, manufacturing history, pump shaft material selection, and mechanical loading.

4. Failure Mode of PI-2:
   - Metallurgical Examination
   - Fatigue and Fracture Mechanics Analysis
   - Dynamic Analysis

5. Overview of the safety significance of the pump shaft failure at PI.

6. Discussion of the pump failure at PI as compared to Surry.

7. Monitoring Methods and Criteria for Measuring Pump Vibrations:
   - At PI
   - Other Plants

Description of Operating Experience Leading to the Discovery of RCP No. 21 Failure

On June 11, 1981 at 1530 hours the high vibration alarm was activated on RCP No. 21 indicating a shaft vibration of .010 inches. RCP No. 21 is a Westinghouse pump Model No. 93A for which, under normal operation, the vibration is

.003 to .005 inches. After the alarm sounded, the shaft vibration was monitored every 30 minutes and between 2000 and 2030 hours the vibration increased to .015 inches. Between 2100 and 2200 hours the seal leakoff flow rate was observed to be 4.2 GPM (the normal flow rate is 3.0 to 3.5 GPM) and the measured shaft vibration was .020 inches. Between June 12 and 14, 1981 the lower motor bearing was disassembled and the support ring was repaired. On June 15, 1981 when the pump was restarted and an unacceptable level of vibration was observed, the pump was shut down and disassembled. When the pump shaft was removed from the pump, a crack was found under the sleeve which measured 270-300° around the circumference of the shaft. The pump shaft was sent to Westinghouse to have the failure analyzed.

Description of the Failure

Westinghouse's detailed examination revealed the crack was initiated from a pin hole in the pump shaft. The metallographic examination of the fractured surface indicates that the crack propagated due to high cycle fatigue. A review of the manufacturing history of the failed pump showed the pin that was inserted into the pin hole of the shaft was twice welded to the sleeve that could possibly impose a high residual stress in the vicinity of the pin hole of the shaft. This twice welding operation and the closeness of the welding operation to the shaft pin hole is unique to the failed PI pump. Although this condition could reasonably explain the high residual stress that could result in the crack initiation localized in the vicinity of the pin hole, the propagation of the crack (i.e., 270°-300° around the shaft) cannot be explained by any identified dynamic loads used in the fatigue stress analysis. Using the most conservative load values, the fatigue stress analysis shows the Factor of Safety to be 1.61 (61% margin) indicating that the observed failure should not have occurred. The licensee has engaged a third party consultant to review Westinghouse's effort regarding the cause of crack propagation.

Safety Significance of Reactor Coolant Pump (RCP) Shaft Failure

The RCPs are designed and fabricated to the safety criteria of ANSI N18.2. Although the RCPs are considered a non-nuclear safety component, the pressure retaining parts of the pump are designed to the ASME Safety Class 1 and the rotating parts are Safety Class 2. The RCPs with their protection systems and associated supports are designed to withstand a sudden stopping of one RCP due to seizure or other similar causes without causing failure to the RCP pressure boundary. The shaft break transient resembles a locked rotor transient in which a high pressure spike (peak pressure 2737 psia) occurs in a few seconds and then decays away. On this basis the licensee and Westinghouse do not consider the event as being a safety significant issue.

Generic Considerations

Seventeen utilities are currently using the Westinghouse Model 93A pump. Four of the 17 utilities have pumps that have approximately the same or slightly greater number of operating hours as the failed pump at PI. The shaft failure at Surry was different than the failure that occurred at PI. In the case of Surry, the crack location was at the groove in the shaft. This groove is no longer in use. The PI pump failure occurred at the pin hole. In addition, the Surry pump has a much shorter operating history than the PI pump (i.e., 9,697 vs 51,643 hour). Westinghouse has concluded that the PI pump failure should not be classified as a generic issue based on a single failure from a total of 106 pumps operating throughout the country.

OFFICE							
SURNAME							
DATE							

Pump Monitoring

Westinghouse recommends to its clients that vibration monitoring devices be installed on the primary coolant pumps. According to Westinghouse's records six plants operating with Model 93A RCPs and nine plants operating with non 93A RCPs do not monitor pump vibration. Based on operating expe ce Westinghouse recommends the following:

Shaft Vibration	Action
.003-.005 inches	Normal
.015 inches	Alarm
.020 inches	Recommend Shutdown

The NRC did not take a position on this matter during the meeting since the information presented at the meeting still had to be reviewed.

Original signed by:

Dominic C. DiIanni, Project Manager
Operating Reactors Branch #3
Division of Licensing

Enclosures:
1. List of Attendees
2. Viewgraphs Used During Mtg.

cc w/enclosures:
See next page

OFFICE	ORB#3:DL	C-ORB#3:DL	ORB#3:DL				
SURNAME	DDiIanni/cb	RClark	PMKreutzer				
DATE	11/ /81	11/ /81	11/ /81				

MEETING SUMMARY DISTRIBUTION

Licensee: Northern States Power Company

*Copies also sent to those people on service (cc) list for subject plant(s).

Docket File
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ACRS (10)
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Federak Activities Branch
Region V Office
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Enclosure 1

ATTENDEES AT THE OCTOBER 29, 1981 MEETING

REACTOR COOLANT PUMP CRACK PROBLEM

NRC

Wm. J. Collins

Earl J. Brown

H. L. Brammer

Keith Wichman

Warren S. Hazelton

C. D. Sellers

D. C. DiIanni

NSP

Gerald Nells

Dean Hannam

Ben Stephens

NUTECH

Pete Riccardella

WESTINGHOUSE

Andrew Madeyski

Gary Elder

Allan Hribar

Ed Burns

Alan Dietrick

Agenda Item 1.

Enclosure 2

PRAIRIE ISLAND NO. 2

RCP 21 OPERATING HISTORY

YEAR	HOURS COLD (<350°F)	HOURS HOT	STARTS
'74	48	650	0(?)
'75	73	8147	21
'76	210	7085	9
'77	6	8055	2
'78	14	8242	24
'79	9	8694	20
'80	113	7617	47
'81 → 6/11	109	3083	30
	581	51,573	153
	(4.2×10^7)	(3.7×10^9)	

21 HEAT-UPS

20 COOL-DOWNS

MAINTENANCE C -1 RCP

Date	Description
11/27/74	Checked seals, fo..d "O" Ring missing in #2 installed new #3 seal.
12/1/76	Replaced #1 seal, #2 seal, and #3 seal.
12/1/77	Inspected seals, replaced upper seal housing and #3 seal ring.
1/28/78	Replaced #1 seal insert, #1 seal runner, #1 seal ring, #2 seal ring. Seal had eratic leakoff.
12/10/78	Removed 21 RCP Motor for cleaning and inspection.
1/19/80	Inspected seals changed out #2 & 3 seals.
3/5/81	Inspected seals.

EVENTS LEADING TO 21 RCP SHAFT FAILURE

JUNE 11

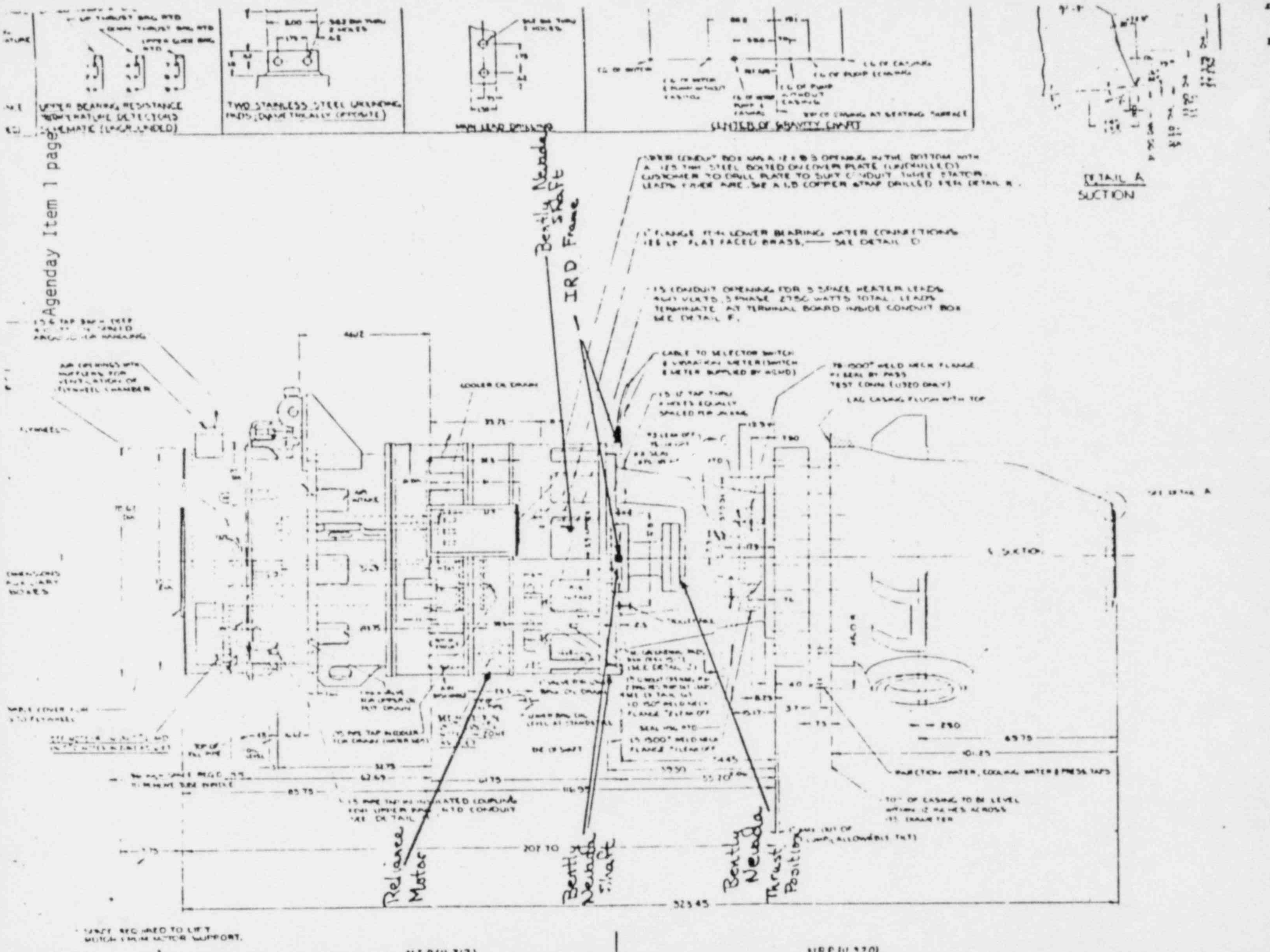
Time	Event
1530	High Shaft Vibration Alarm received. Alarm Set Point $\geq$15 mils Local Reading 10 mils Starting checking local reading every half-hour.
2000-2030	RCS was diluted; number one seal leakoff increased; vibration 15 mils.
2100-2200	Seal leakoff increased to about 4.2 gpm; Shaft vibration 20 mils.
2200	Shutdown begun.
2230	Unit off line, 21 RCP stopped; vibration 27 mils. Seal leakoff returns to normal. Cooldown commenced.

JUNE 12	Inspection found 21 RCP motor lower radial bearing loose.
JUNE 13 & 14	21 RCP motor lower radial bearing repaired.
JUNE 15	21 RCP restarted; initial vibration 7 mils. After 20 minutes - 12 mils. An attempt to balance was made, but vibration kept increasing. Pump stop cooldown commenced.
JUNE 16	Determination made to disassemble the pump - Shaft would not rotate Shaft would not drop

REACTOR COOLANT PUMP
VIBRATION MONITORING

	Type	Location	Output
1.	Frame Vibration Reliance	RCP Motor	Cabinet Analog Computer
2.	Frame Vibration IRD	RCP Motor to Motor Stand	Meter in Containment
3.	Frame Vibration Bentley Nevada	RCP Motor to Motor Stand	Cabinet Control Board Alarm
4.	RCP Shaft Vib Bentley Nevada	RCP Shaft Just Below Motor Lower Radial Bearing	Cabinet Control Board Alarm
5.	RCP Shaft Thrust Position Indication	RCP Shaft at the Pump Coupling	Cabinet

Agenday Item 1 page



93A REACTOR COOLANT PUMPS

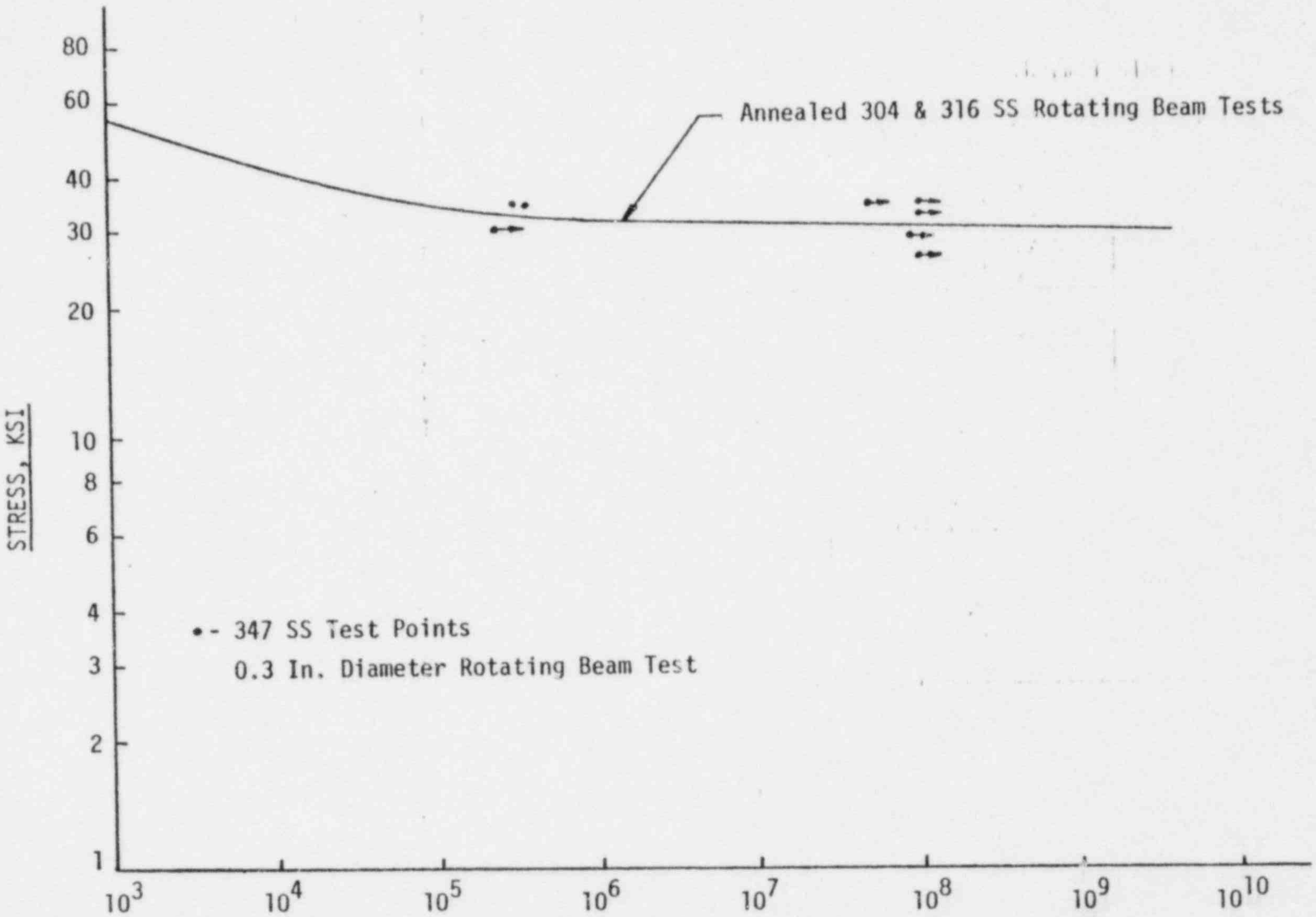
Group	Description	Number
1	Same as Prairie Island #2	106
2	Lowered Turning Vanes (Lower Hydraulic Loads)	26
3	Tack Welded Pins (Lower Residual Stress)	10
4	Lowered Turning Vanes and Tack Welded Pins (Lower Hydraulic Loads and Residual Stress)	30
	TOTAL	172

93A RCP OPERATING HISTORY

(As of June 1, 1981)

Plant	No. of RCP's	Operating Hours	Shaft Revolutions
Oconee	4	53,000	3.8×10^9
Prairie Island #1	2	53,000	3.8×10^9
Kewaunee	2	53,000	3.8×10^9
Zion #1	4	47,200	3.4×10^9
Surry #1	3	45,300	3.3×10^9
Zion #2	4	45,000	3.2×10^9
D.C. Cook #1	4	45,000	3.2×10^9
Surry #2	3	40,000	2.9×10^9
Three Mile Island #1	4	33,000	2.4×10^9
Trojan #1	4	27,000	1.9×10^9
D.C. Cook #2	4	22,000	1.6×10^9
Salem #1	4	20,500	1.5×10^9
North Anna #1	3	20,000	1.4×10^9
Farley #1	3	19,300	1.4×10^9
Beaver Valley #1	3	18,000	1.3×10^9
Diablo Canyon #1	4	9,000	6.5×10^8
North Anna #2	3	5,000	3.6×10^8
Foreign Plants	18	------	$> 10^9$
Foreign Plants	6	------	$> 10^8$
Total	82		

10^8 Revolutions = 1,389 Hours in approximately two months.



S-N CURVE FOR 300 SERIES SS

agenda item 3

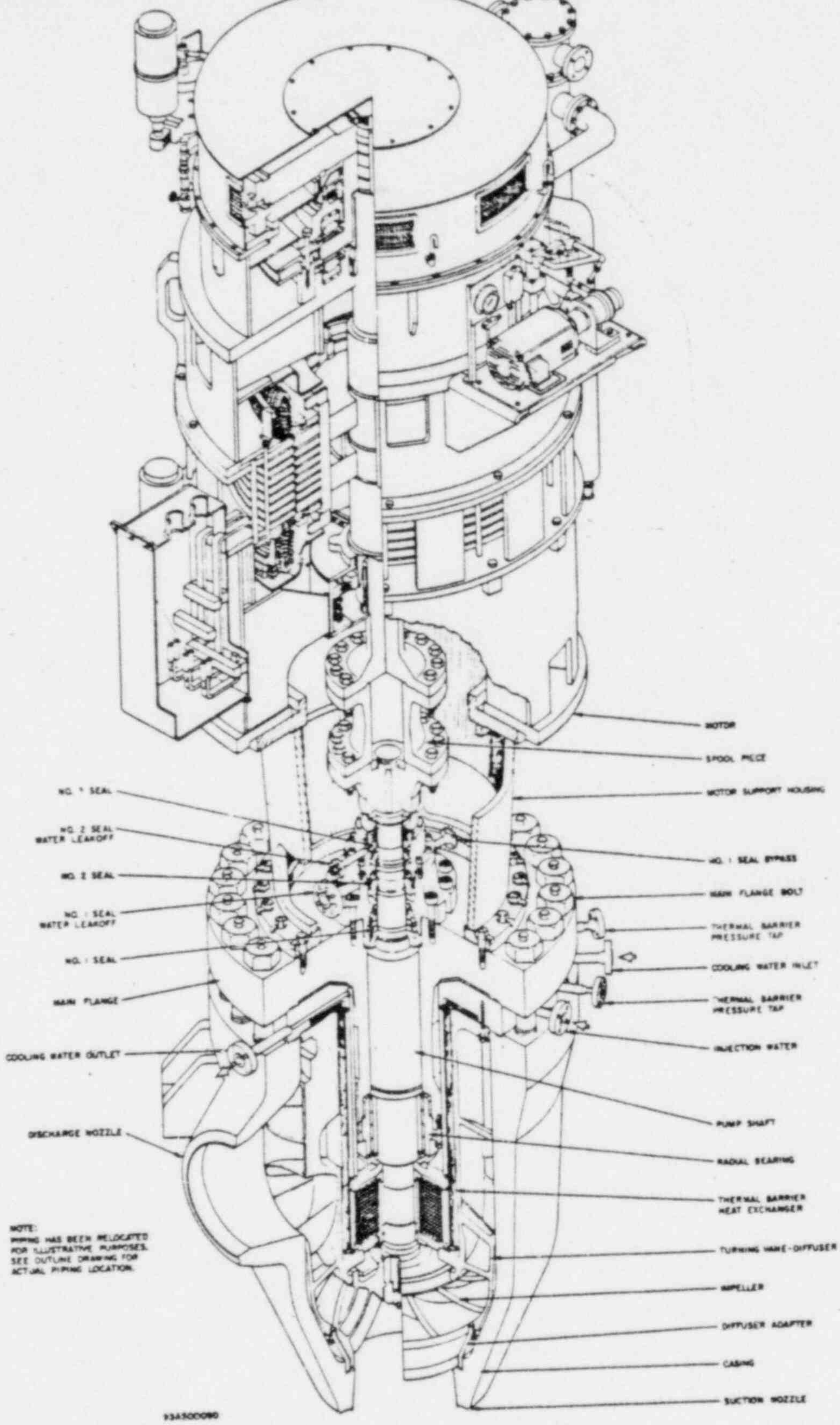
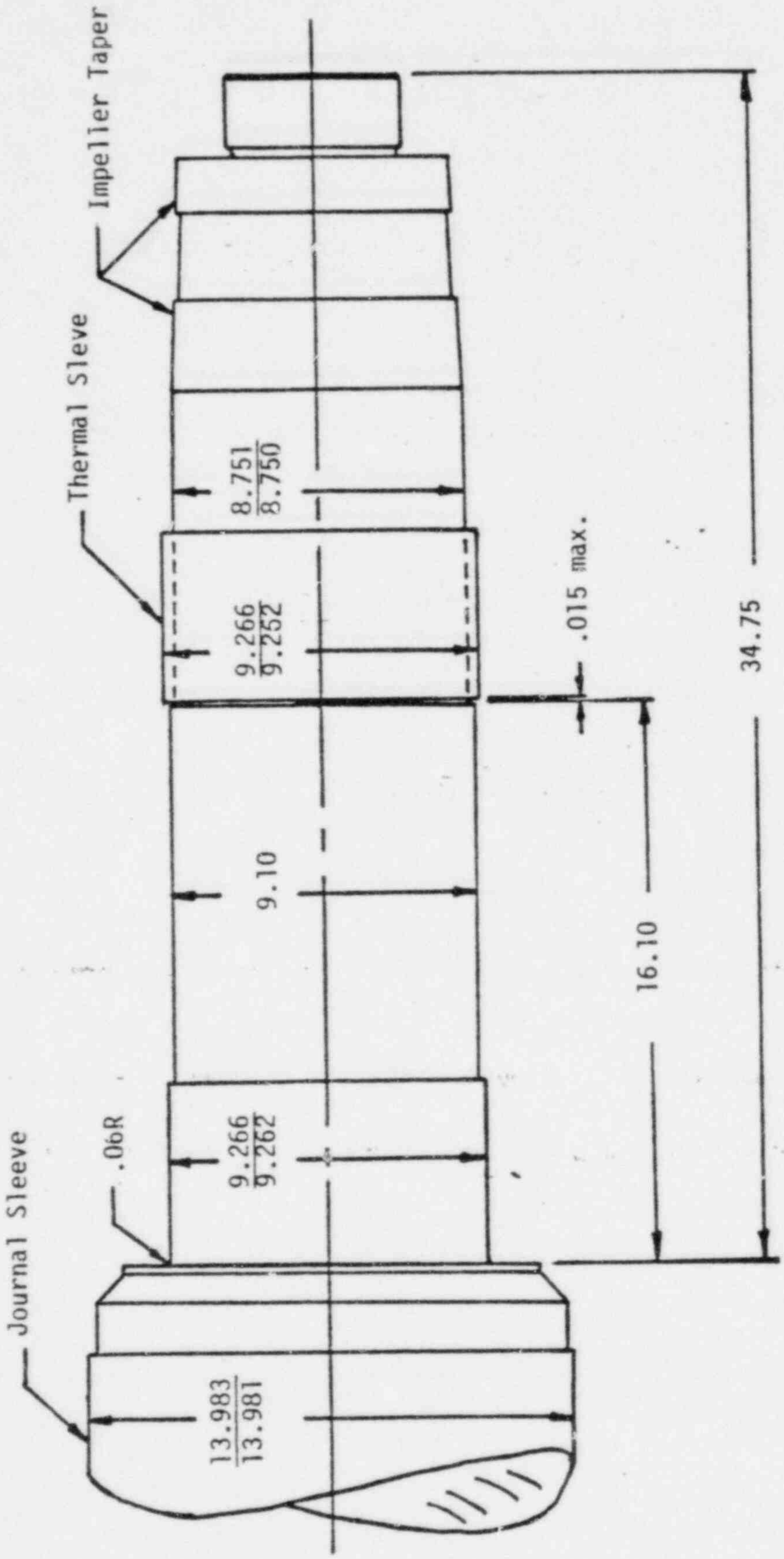
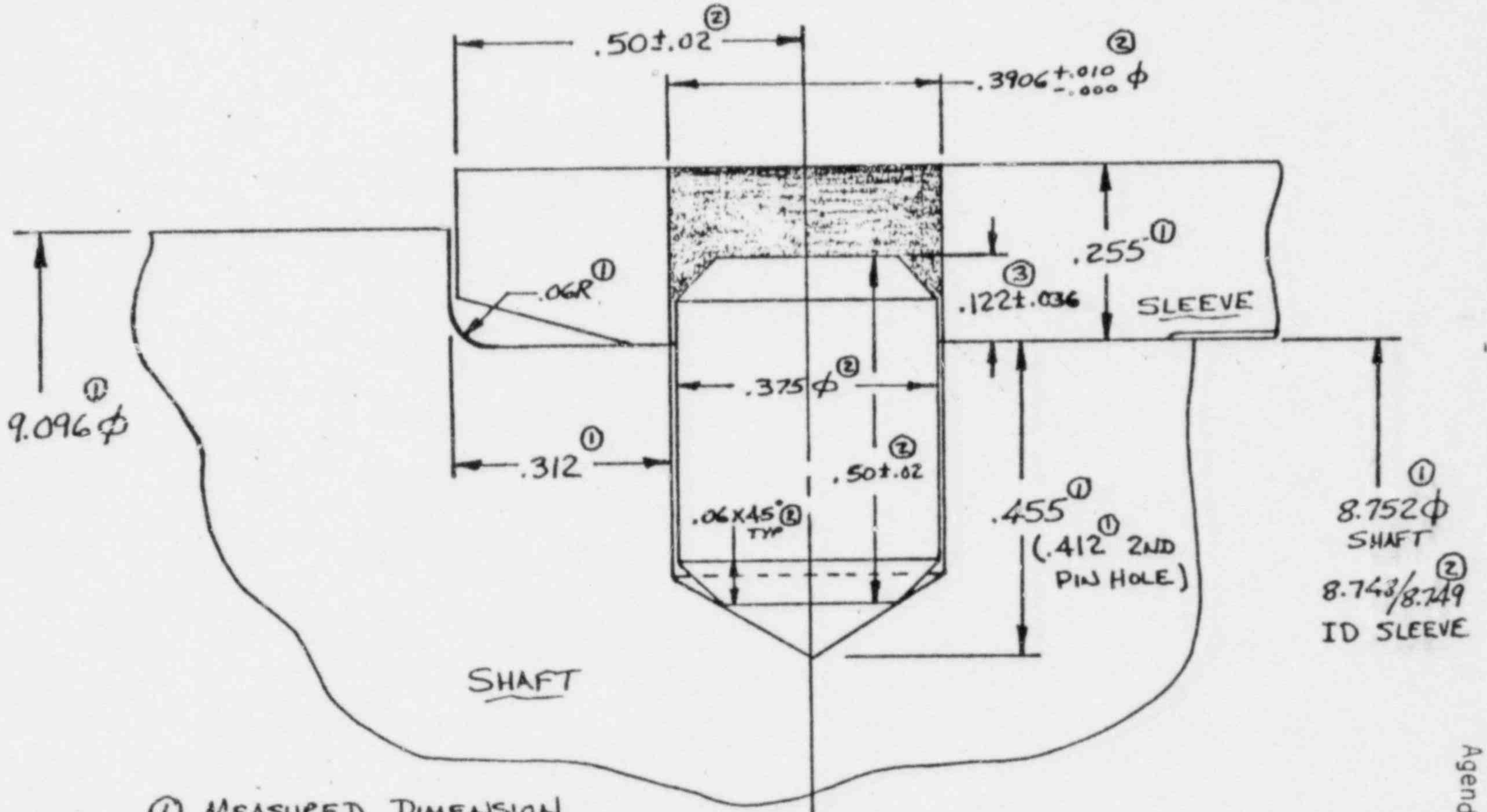


FIGURE I-1 TYPICAL PUMP, CUTAWAY VIEW



LOWER SHAFT DIMENSIONS

NRP PIN DIMENSIONS
(CRACKED PIN HOLE)



① MEASURED DIMENSION
② DRAWING DIMENSION
③ CALCULATED

SHAFT S/N 516 FABRICATION

Rough Turned forging Mfg. by Standard Steel

↓

Shaft Preliminary Machining

↓

Assemble Bearing Journal Sleeve and Thermal Sleeve

↓

Shaft Final Machnining

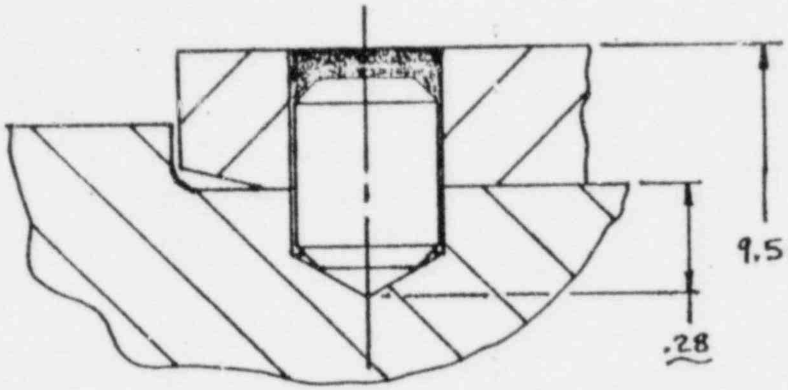
↓

Assembly Coupling and Balance

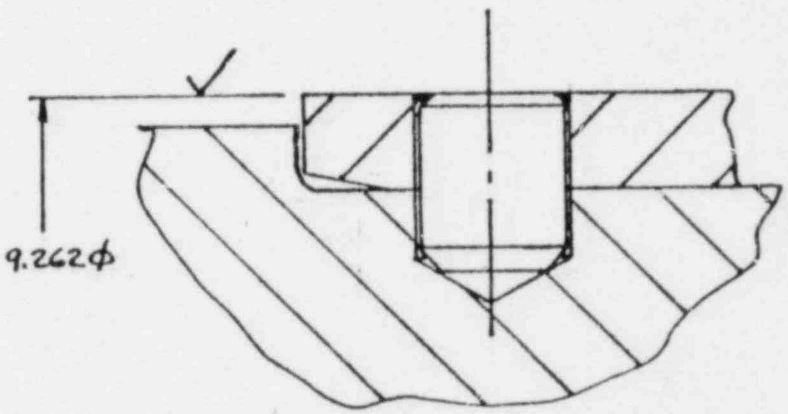
↓

Assembly Spool and Balance

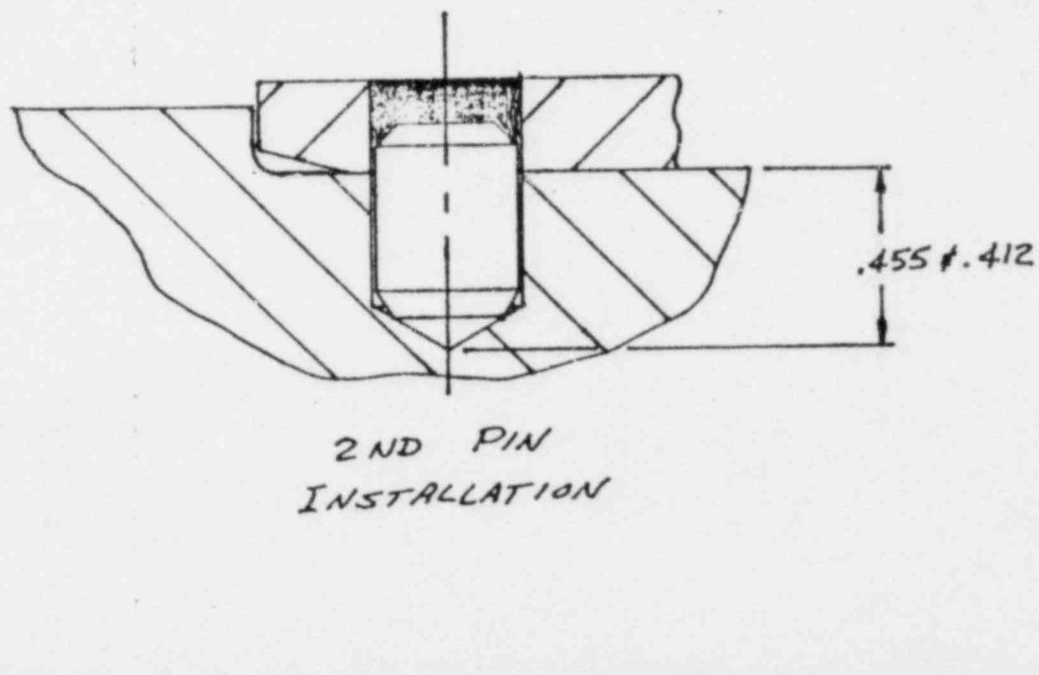
SHAFT REWORK ~ MRR 9052



1ST PIN
INSTALLATION



SLEEVE MACHINING
REMOVED WELD



2ND PIN
INSTALLATION

SHAFT MATERIAL

- ASTM-A-182 GRADE F-347

- HEAT TREATED FOR DIMENSIONAL STABILITY

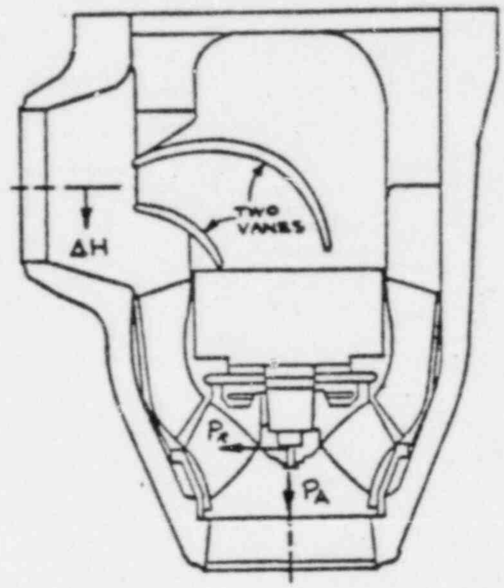
  1500 - 1600°F FOR TWO HOURS PER INCH OF THICKNESS
  COOLED TO 300°F AT 200°F/HR. MAX.

- EXAMINED BY U/T AND LP

- MATERIAL PROPERTIES:

YIELD STRENGTH	-	37,000 PSI
TENSILE STRENGTH	-	85,000 PSI
ENDURANCE LIMIT (10^9)	-	31,000 PSI

PRIMARY SHAFT LOADS

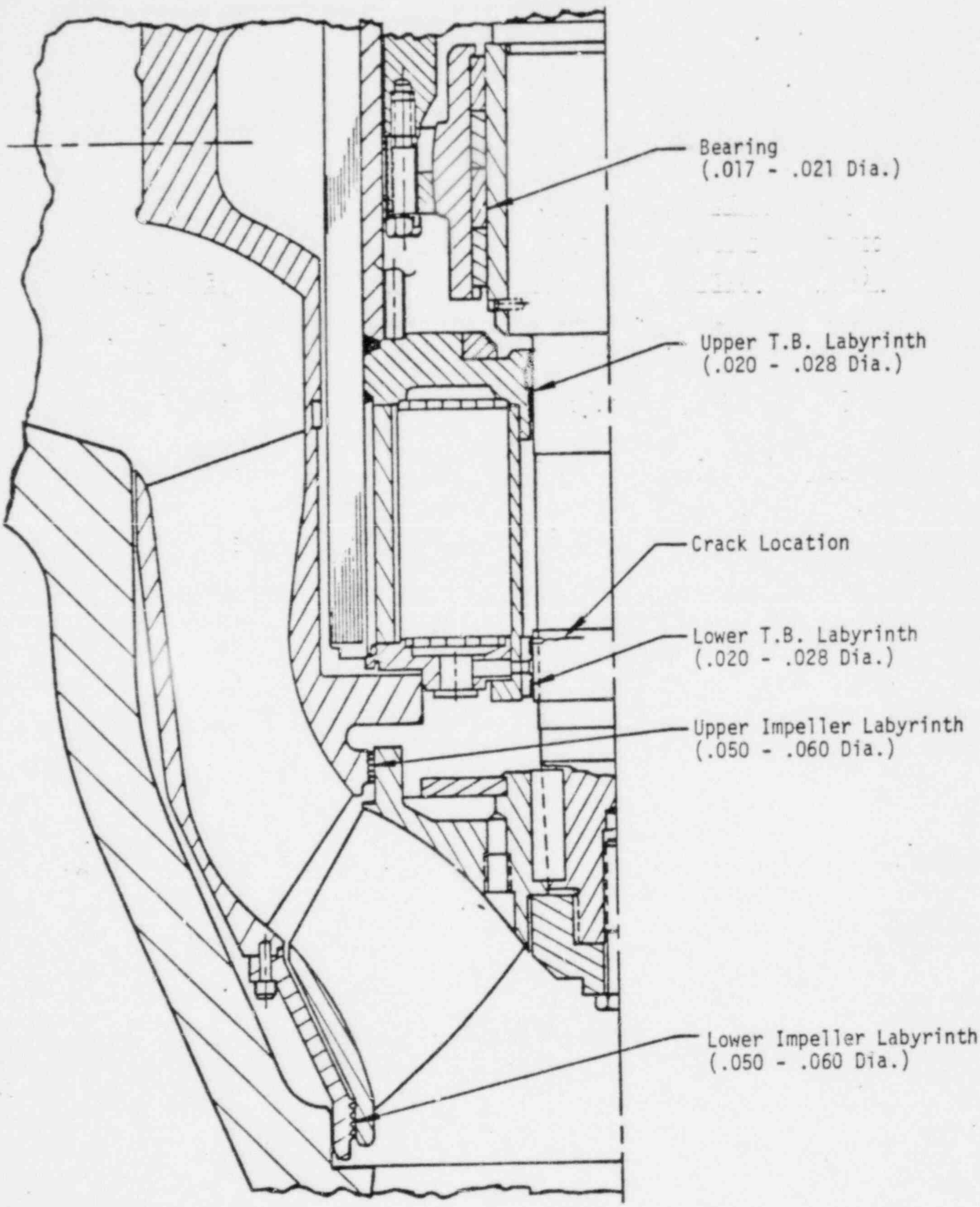


P_R = Radial Thrust Load

P_A = Axial Thrust Load + Deadweight - Hydrostatic Load

HYDRAULIC LOADS-PRAIRIE ISLAND RCP'S

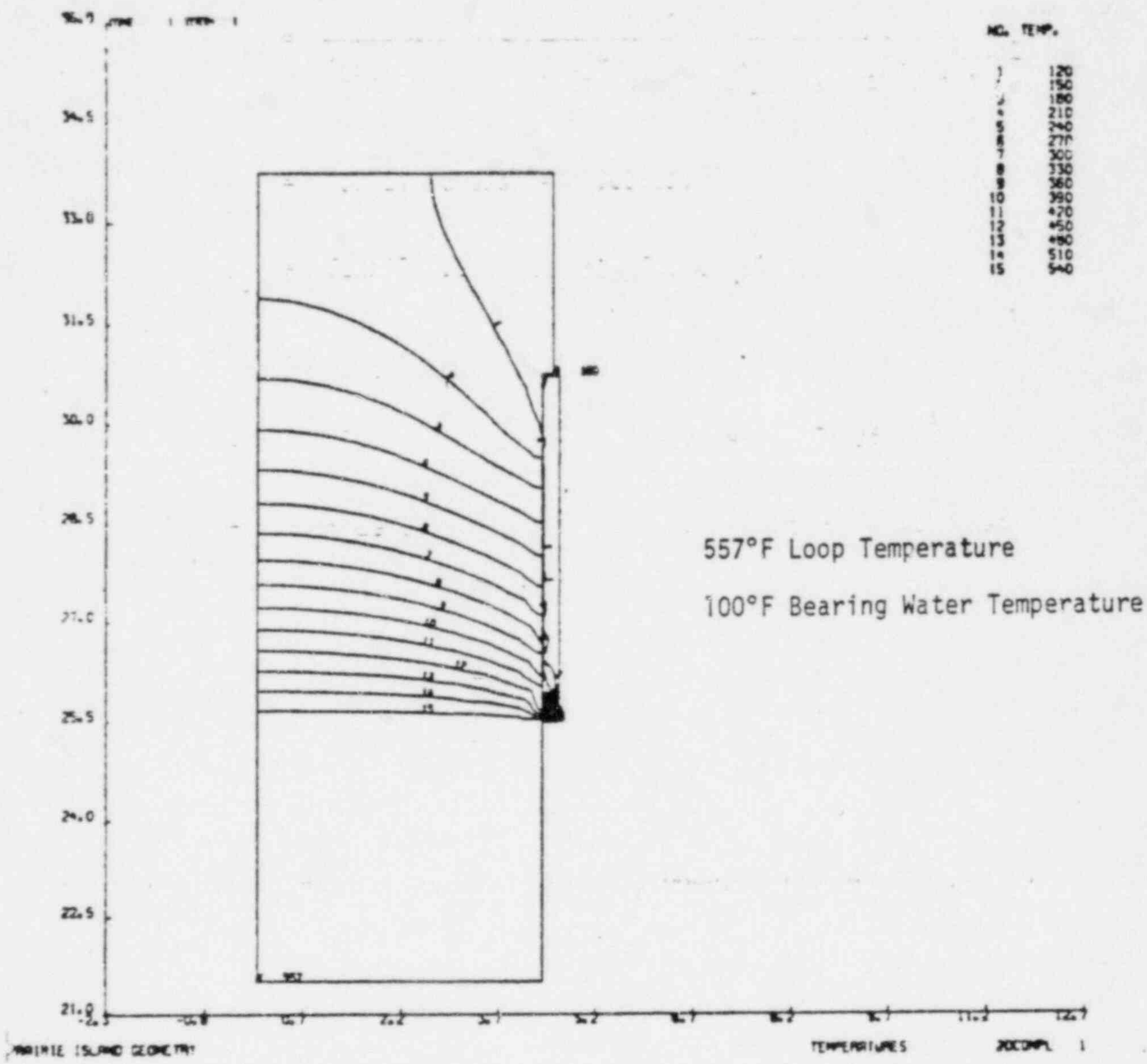
	COLD RUNOUT 1 PUMP	COLD OPERATION 2 PUMPS	HOT OPERATION 2 PUMPS
FLOW (GPM)	102,870	94,390	98,200
TORQUE (FT.-LBS.)	31,560	33,147	25,546
AXIAL THRUST (LBS.)			
NOMINAL	64,670	68,789	50,726
MAXIMUM	71,140	75,670	55,800
RADIAL THRUST (LBS.)			
NOMINAL	4,083	3,589	2,884
MAXIMUM	5,104	4,486	3,605
MAX. CYCLIC TORQUE	180	118	89
MAX. CYCLIC AXIAL LOAD	297	194	147
ROTATING RADIAL LOAD	635	635	480



SHAFT CLEARANCES

ISOTHERMAL PLOT

STEADY STATE - NORMAL OPERATIONS



SHAFT STRESSES

	Cold Runout		Hot Operation	
Load Source	Max. Load (lbs.)	Stress (psi)	Max. Load (lbs.)	Stress (psi)
Motor Torque	31,560	2880	25,546	2330
Axial Thrust	71,140	1183	55,800	928
Radial thrust	5140	±1425	3605	±999
Cyclic Torque	180	±16	89	±8
Cyclic Axial Load	297	±5	147	±2
Rotating Radial Load	635	176	480	133

FATIGUE ANALYSIS

Basic Equation

$$F.S. = \frac{1}{\{[\sigma_o/\sigma_u)+(k_f\sigma_a/\sigma_f)]^2+3[(\tau_o/L_s\sigma_y)+(k_{sf}\sigma_a/\sigma_f)]^2\}^{1/2}}$$

σ_o = Normal Stress = 1360 psi

σ_u = Tensile Strength = 85,000 psi

k_f = $Q(k_t - 1) + 1$ = 2.5 (Q = .8)

σ_a = Alternating Bending Stress = 1425 psi

σ_F = Size Corrected Endurance Strength = 21,000 psi

τ_o = Steady Torsional Stress = 2880 psi

L_s = Limit Factor for Shear = 1.33

k_{sf} = 3.2

F.S. = $\frac{1}{.31}$ = 3.20 (220% Margin)

For A $\sigma_o = \sigma_y$ = 37,000 psi

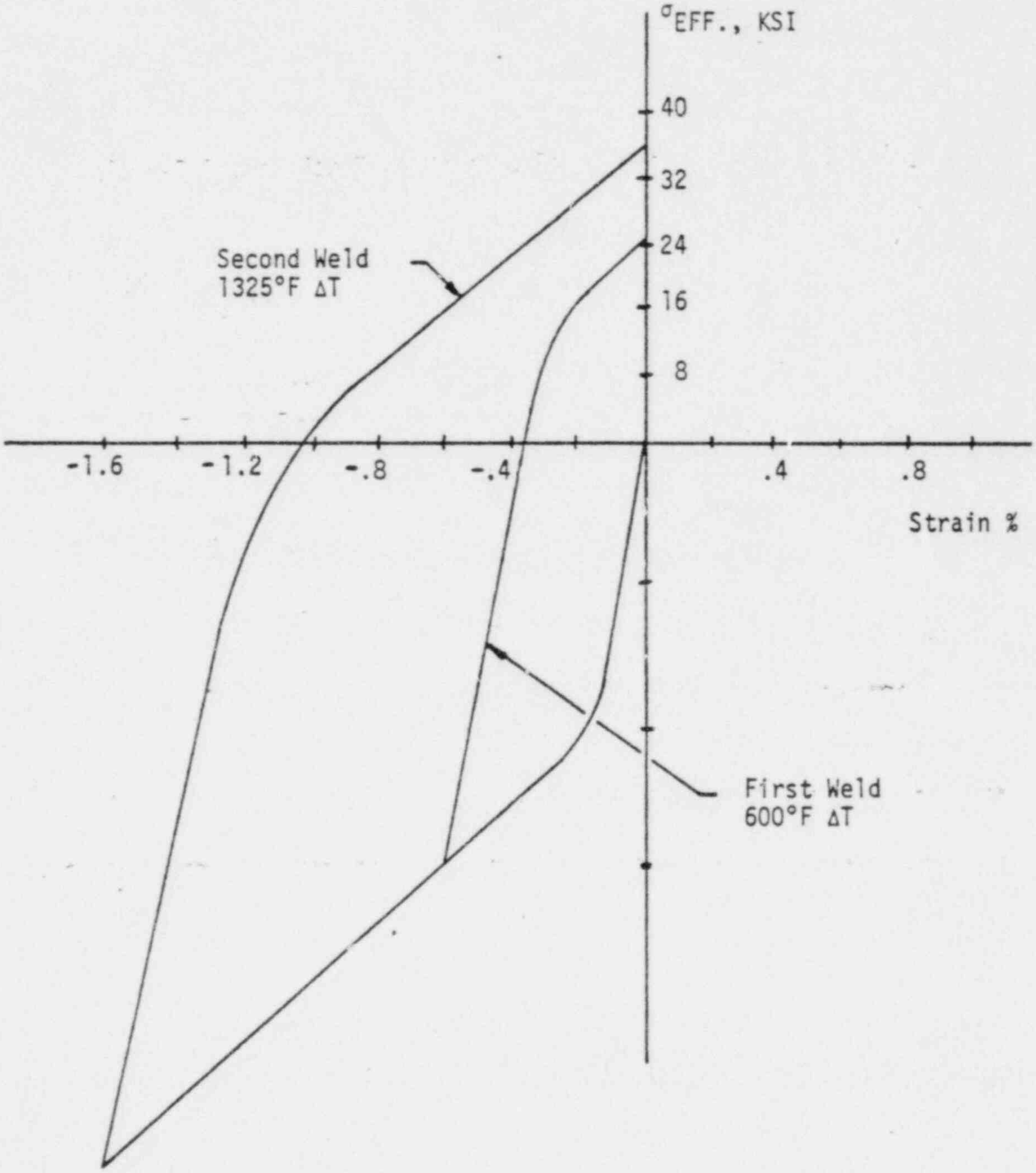
F.S. = $\frac{1}{.62}$ = 1.61 (61% Margin)

RESIDUAL STRESS DUE TO WELDING

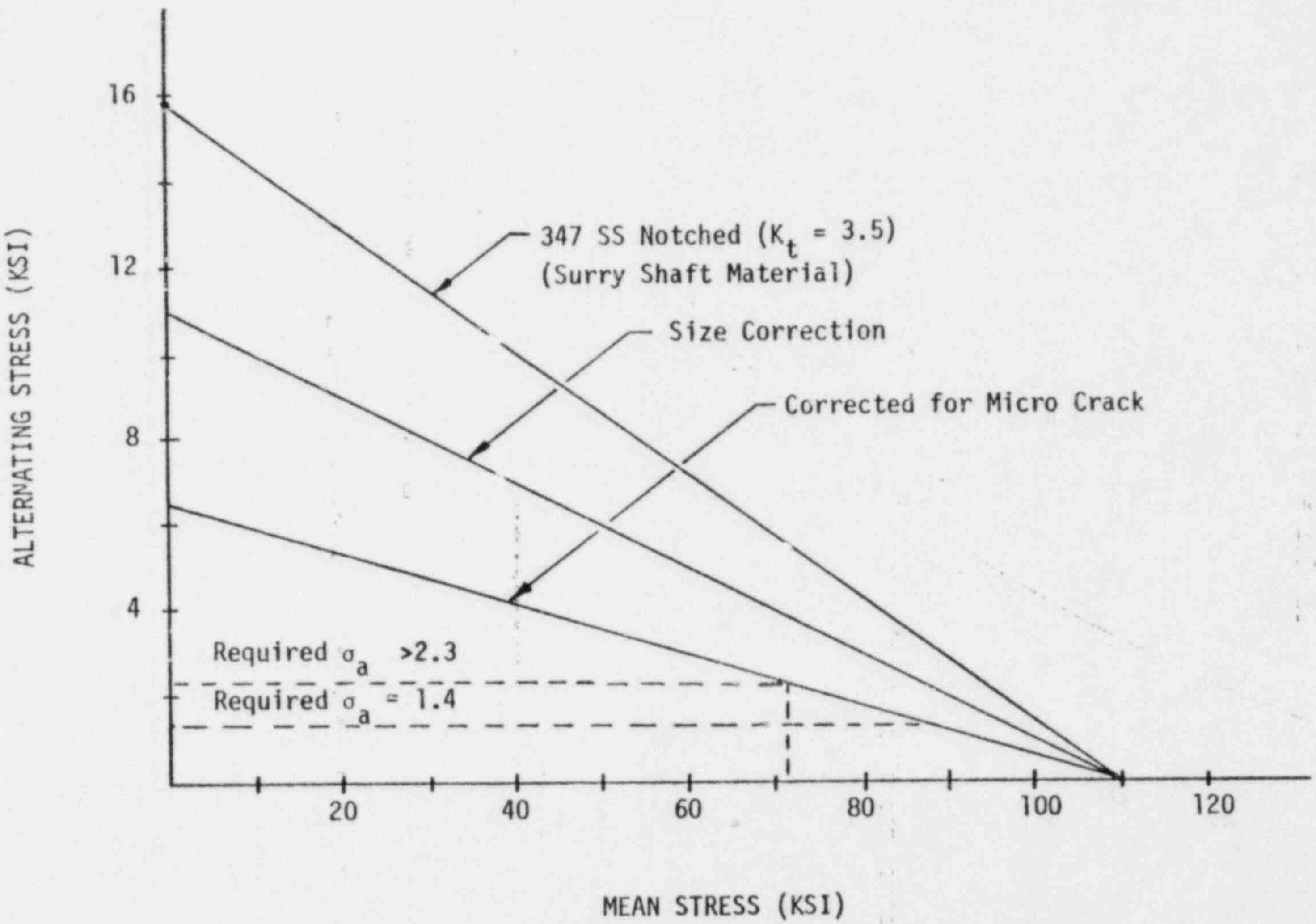
$$\sigma_{EFF.} = \{\tfrac{1}{2}[\sigma_1-\sigma_2)^2 + (\sigma_2-\sigma_3)^2 + (\sigma_3-\sigma_1)]\}^{\frac{1}{2}}$$

σ_1 = 72 KSI (just below surface)

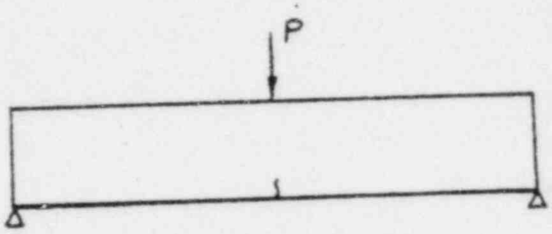
$\sigma_2 = \sigma_3$ = 36 KSI

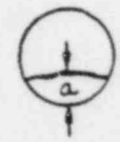


ALTERNATING STRESS REQUIRED TO PROPAGATE A SMALL CRACK



FRACTURE MECHANICS MODEL





Bush's K-Solution for Edge Cracked Round Bar in 3-Point Bending

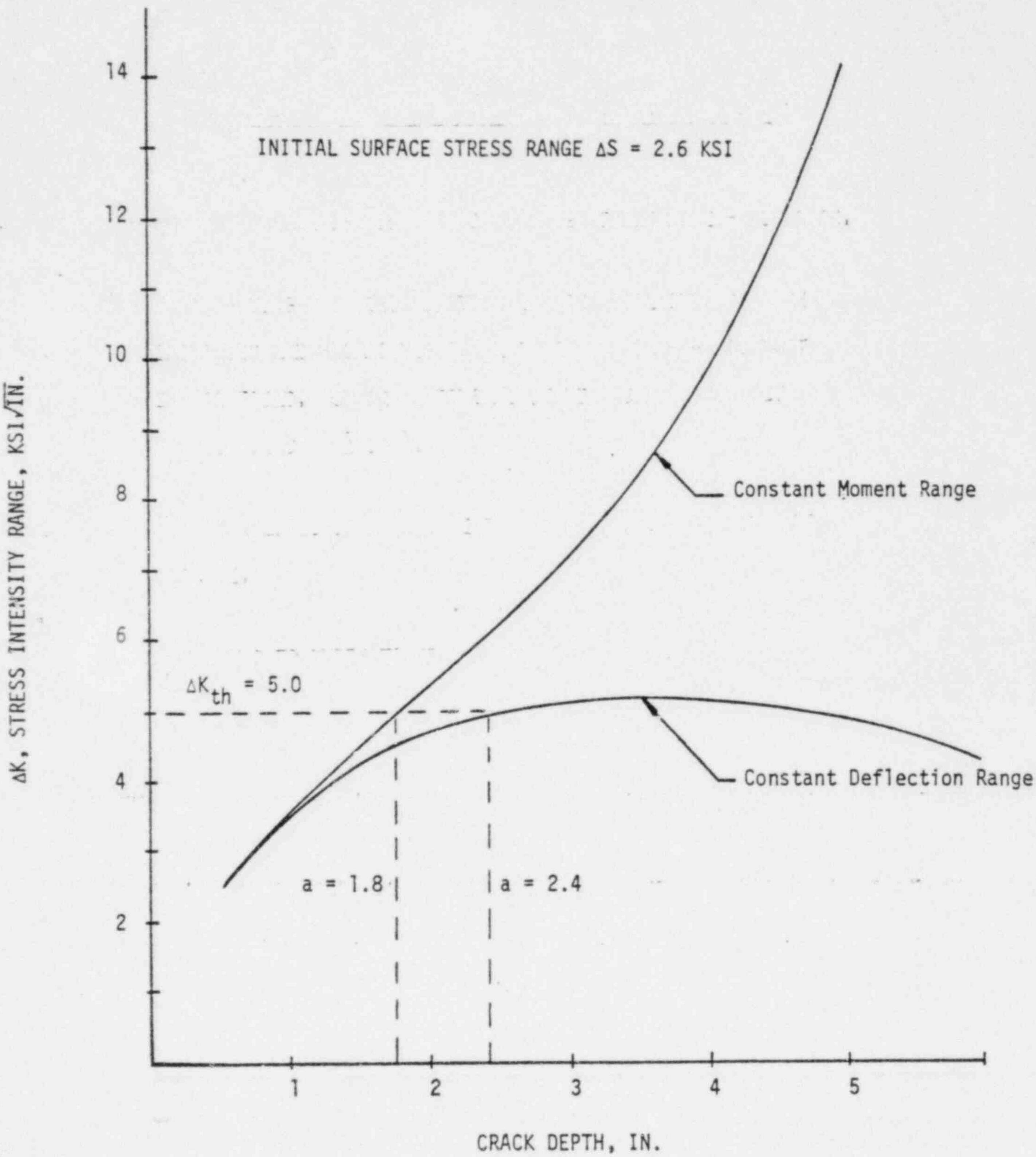
$$\Delta K = \frac{Y'\Delta M}{D^{2.5}}$$

$$\Delta M = \frac{\Delta S \pi \left(\frac{D}{2}\right)^3}{4}$$

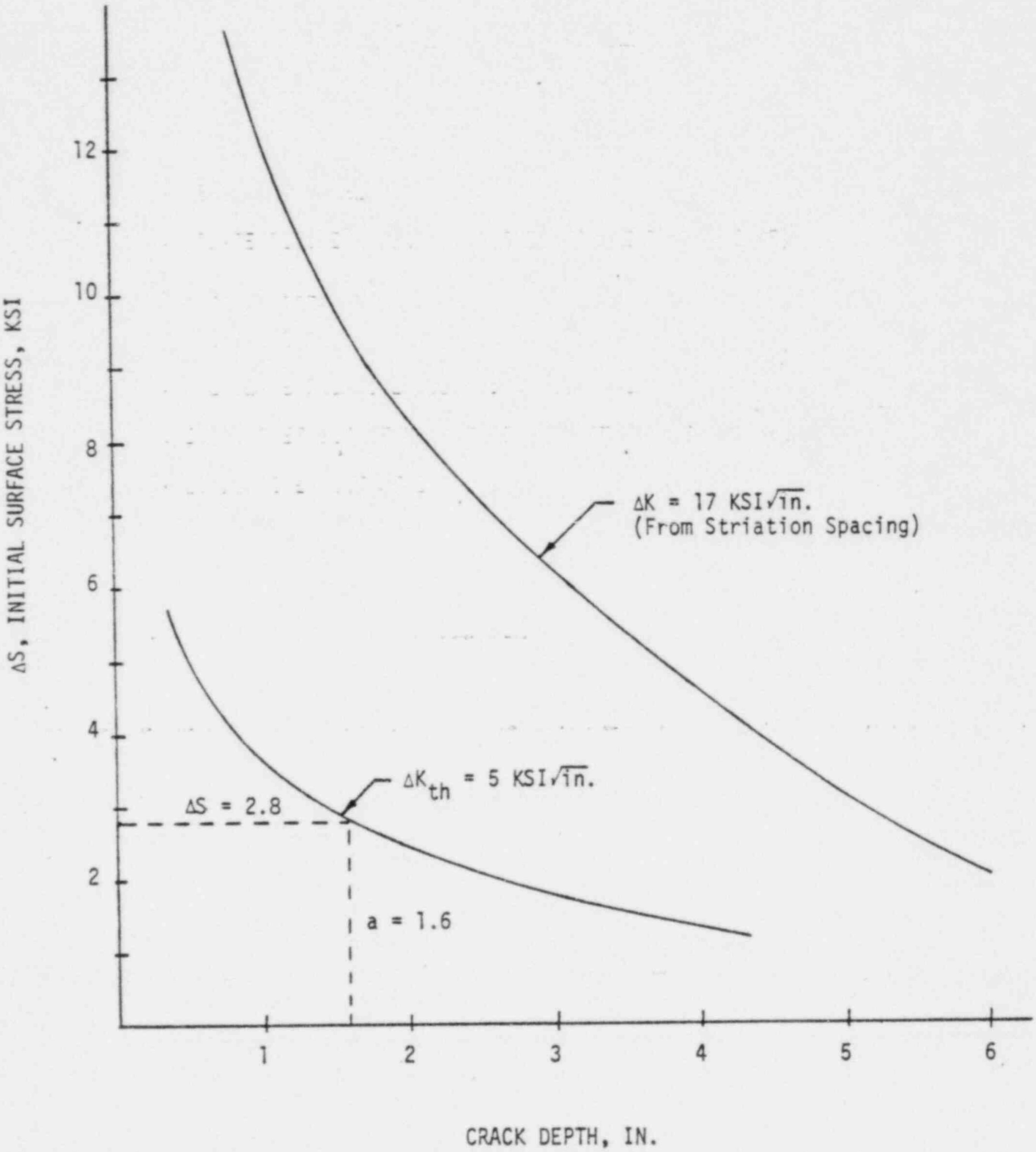
For $\Delta S = 2.6$ KSI ($\pm$ 1.3)

$$\Delta M = 171 \text{ Kip-In.}$$

STRESS INTENSITY VERSUS CRACK DEPTH



STRESS VERSUS CRACK DEPTH



CRACK FORMATION SCENARIO

- LOCALIZED HIGH STRESS DUE TO WELDING AT PIN HOLE - NO CRACK.

- PIN FRETTING PRODUCES ADDITIONAL HIGH LOCAL STRESS WHICH INITIATES AND PROPAGATES SMALL CRACK WITHIN HIGH CONTACT STRESS FIELD. CRACK ARRESTS AS IT LEAVES HIGH CONTACT STRESS FIELD (0.1 IN.).

- NO KNOWN LOADS OF MAGNITUDE REQUIRED TO PROPAGATE CRACK TO FAILURE.

  (NEED TWO TO THREE TIMES KNOWN LOADS)

Agenda Item 4

DYNAMIC ANALYSIS REVIEW OF
NSP PUMP CHARACTERISTICS

- RESURRECT MODEL
- UPDATE/REVISE MODEL (TO REFLECT CURRENT KNOWLEDGE)
- CHECK MODEL VALIDITY
- CALCULATE AXIAL, TORSIONAL AND LATERAL RESONANCES
- CALCULATE RELATIVE VIBRATION RESPONSE
- EXAMINE EFFECT OF THRUST BEARING STIFFNESS CHANGES
- EVALUATE EFFECT OF LOOSE LOWER MOTOR BEARING
- EVALUATE POTENTIAL EXCITATION SOURCES
- EVALUATE EFFECT OF LOCALIZED STIFFNESS CHANGES

RCP DYNAMIC MODEL PARAMETERS

- WECAN, WESTINGHOUSE ELECTRIC COMPUTER ANALYSIS FINITE ELEMENT PROGRAM
- 108 UNIQUE NODES
- 165 ACTIVE DYNAMIC DEGREES OF FREEDOM
- 193 ELEMENTS, 7 DIFFERENT TYPES AS FOLLOWS:

. STIF 7	STRAIGHT PIPE	AXISYMMETRIC STRUCTURAL MEMBERS SUCH AS THE ROTOR ASSEMBLY
. STIF 4	STRAIGHT BEAM	NON-SYMMETRIC STRUCTURAL MEMBERS SUCH AS THE MOTOR FRAME
. STIF 44	LUMPED MASS	TO ADD MASS TO PRESERVE RIGID BODY MASS AND INERTIA DISTRIBUTION
. STIF 14	UNIAXIAL SPRING	LINEAR REPRESENTATION OF BEARING FLUID FILM STIFFNESS
. STIF 38	FLUID ELEMENT	HYDRODYNAMIC MASS INTERACTION OF THE FLUID AND STRUCTURE
. STIF 29	CURVED PIPE	PIPING ELBOWS IN THE MOTOR OIL COOLER PIPING SYSTEM
. STIF 27	GENERALIZED STIFFNESS MATRIX	FOUNDATION SUPPORT AND PRIMARY PIPING INTERFACES WITH RCP

CALCULATED AXIAL MODE FREQUENCIES

MODE	FREQUENCY, HZ	MODE DESCRIPTION
1	18.3	RIGID BODY DISPLACEMENT OF ENTIRE ROTOR
2	44.4	RIGID BODY DISPLACEMENT OF STATOR ON FOUNDATION
3	169.3	FLYWHEEL AND MOTOR ROTOR CORE MOVING OPPOSITE IMPELLER
4	286.2	FLYWHEEL AND IMPELLER MOVING OPPOSITE DIRECTION OF MOTOR ROTOR CORE

CALCULATED TORSIONAL MODE FREQUENCIES

MODE	FREQUENCY, HZ	MODE DESCRIPTION
1	30.4	FIRST TORSIONAL MODE OF ROTOR
2	43.2	SECOND " " " "
3	104.2	THIRD " " " "
4	309.5	FOURTH " " " "

CALCULATED LATERAL MODE FREQUENCIES

MODE	FREQUENCY, HZ	MODE DESCRIPTION
1	7.91 7.93	BENDING OF LOWER PORTION OF ROTOR
2	8.78	ROCKING OF ENTIRE RCP ON FOUNDATION - SLIGHT LOWER ROTOR BENDING
3	23.1 24.4	BENDING OF MOTOR ROTOR NEAR FLYWHEEL - SLIGHT BENDING NEAR IMPELLER
4	32.0	MID-ROTOR BENDING NEAR SHAFT SEALS AND COUPLING
5	32.5	"S" BENDING OF ENTIRE ROTOR
6	32.6	PUMP INTERNALS BENDING AND "S" BENDING OF ENTIRE ROTOR

RELATIVE VIBRATION RESPONSE

METHOD - · REPRESENTATIVE LOADINGS SELECTED

· APPLIED LOAD SWEPT THROUGH FREQUENCY RANGE OF 200 OR 500 RPM TO 2500 RPM

· RELATIVE VIBRATION AMPLITUDE RESPONSE NEAR CRACK LOCATION OBSERVED

· ALTERNATING STRESSES ASSOCIATED WITH SPECIFIC POINTS OF THE RESPONSE CURVES CALCULATED

SELECTED LOADS:

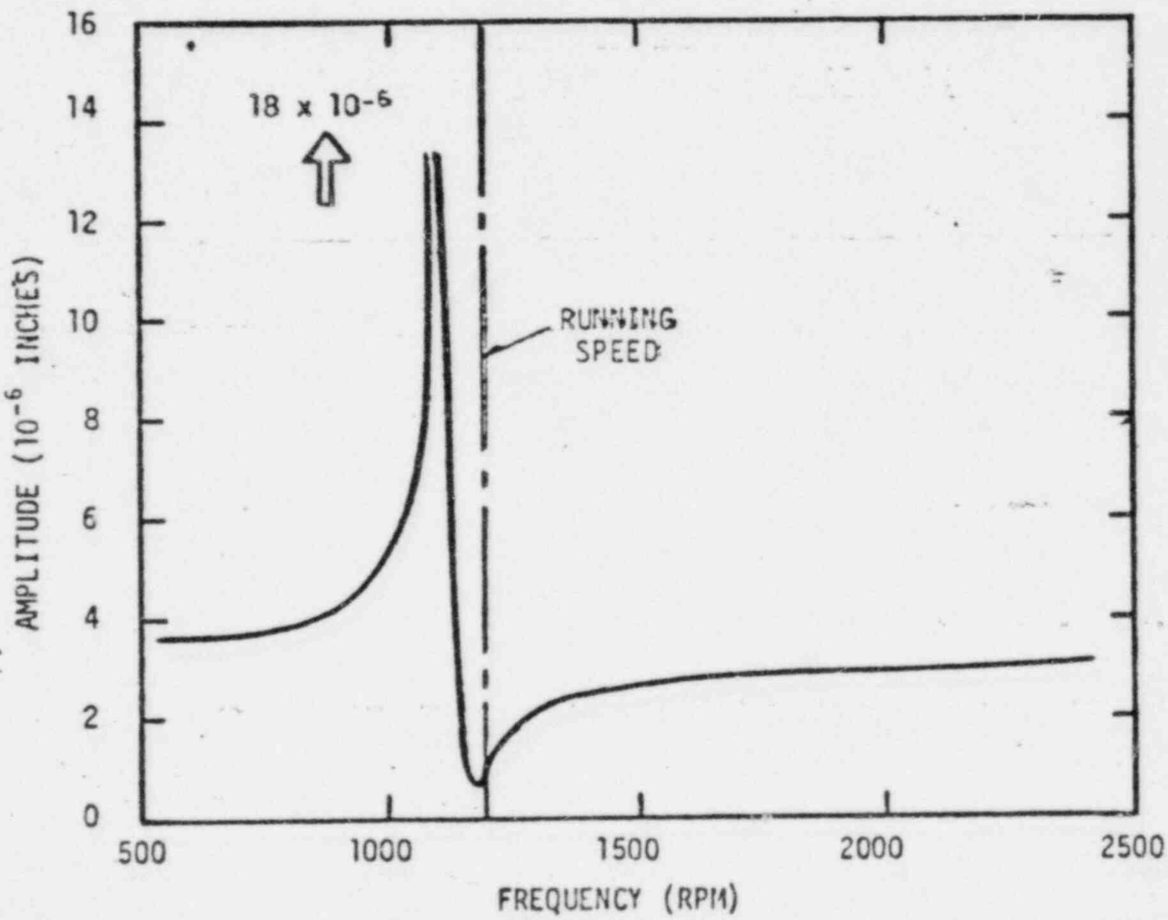
AXIAL - 1000 LB OSCILLATING VERTICAL FORCE AT IMPELLER

TORSIONAL - 1000 IN-LB OSCILLATING TORQUE AT IMPELLER

LATERAL - 1 IN-LB UNBALANCE AT IMPELLER

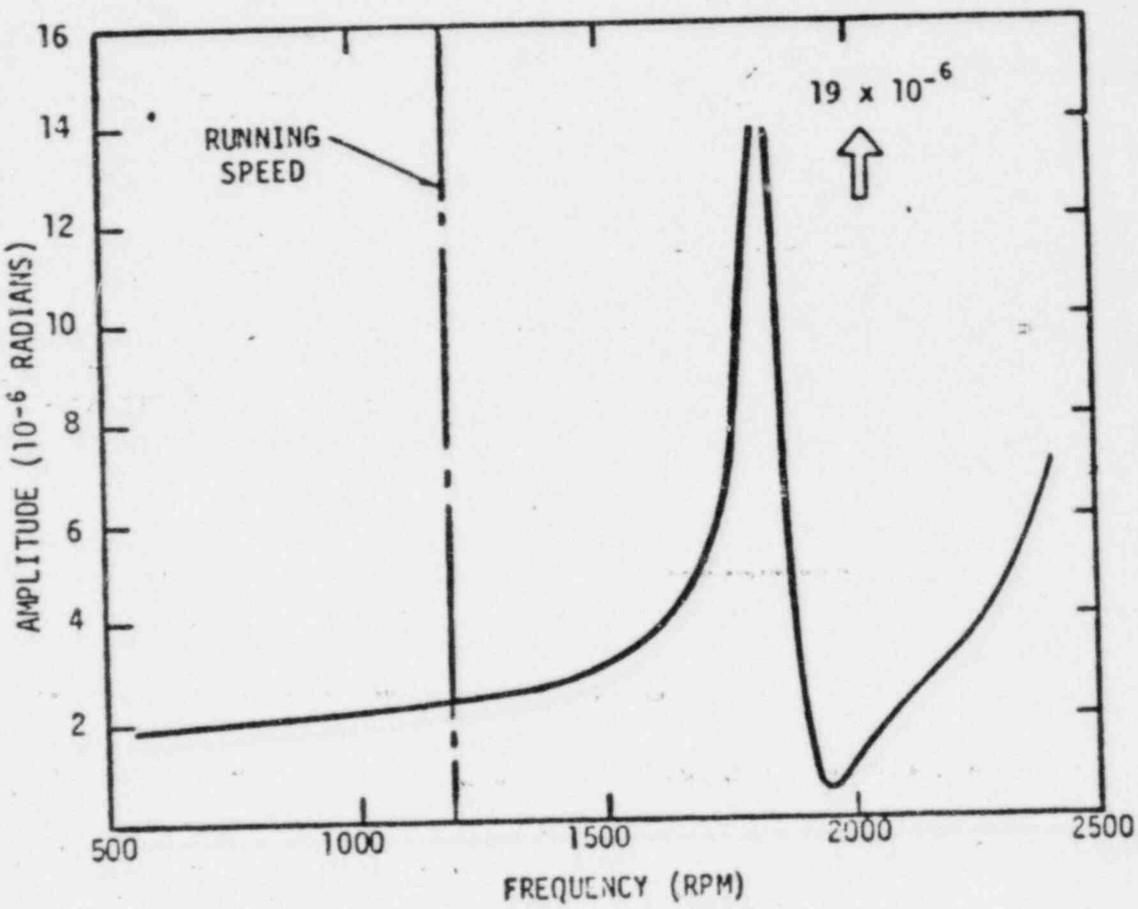
AXIAL RESPONSE

RELATIVE DISPLACEMENTS IN VICINITY OF CRACK



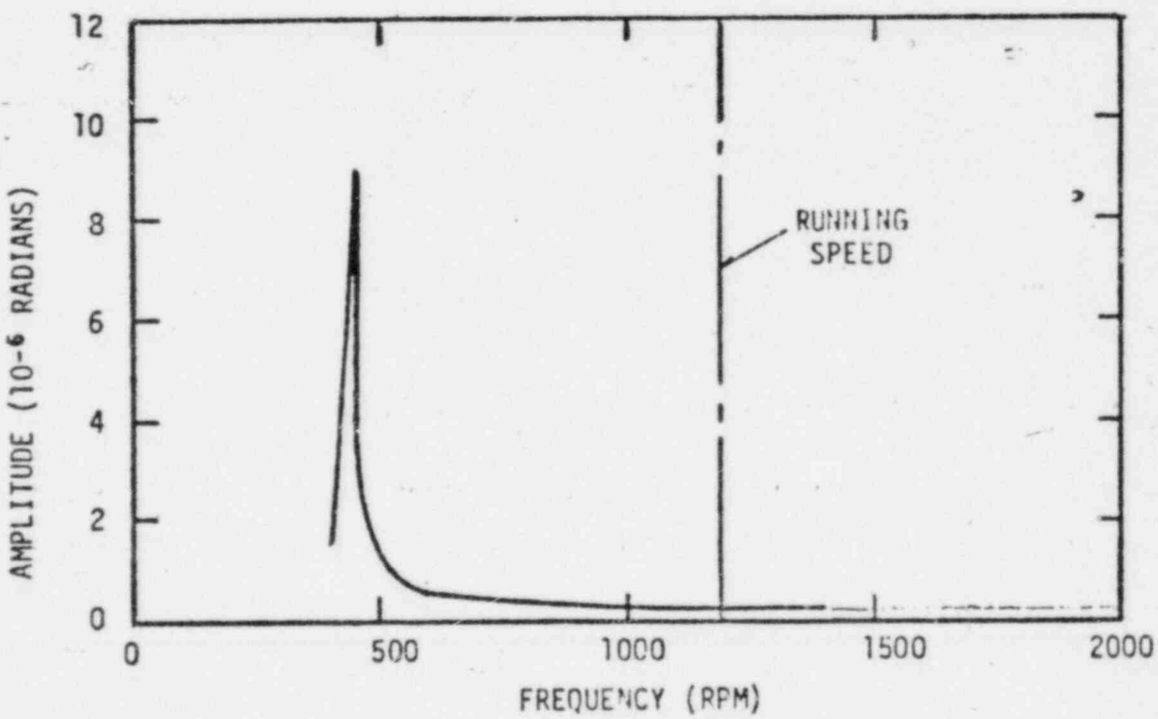
TORSIONAL RESPONSE

RELATIVE ROTATION IN VICINITY OF CRACK



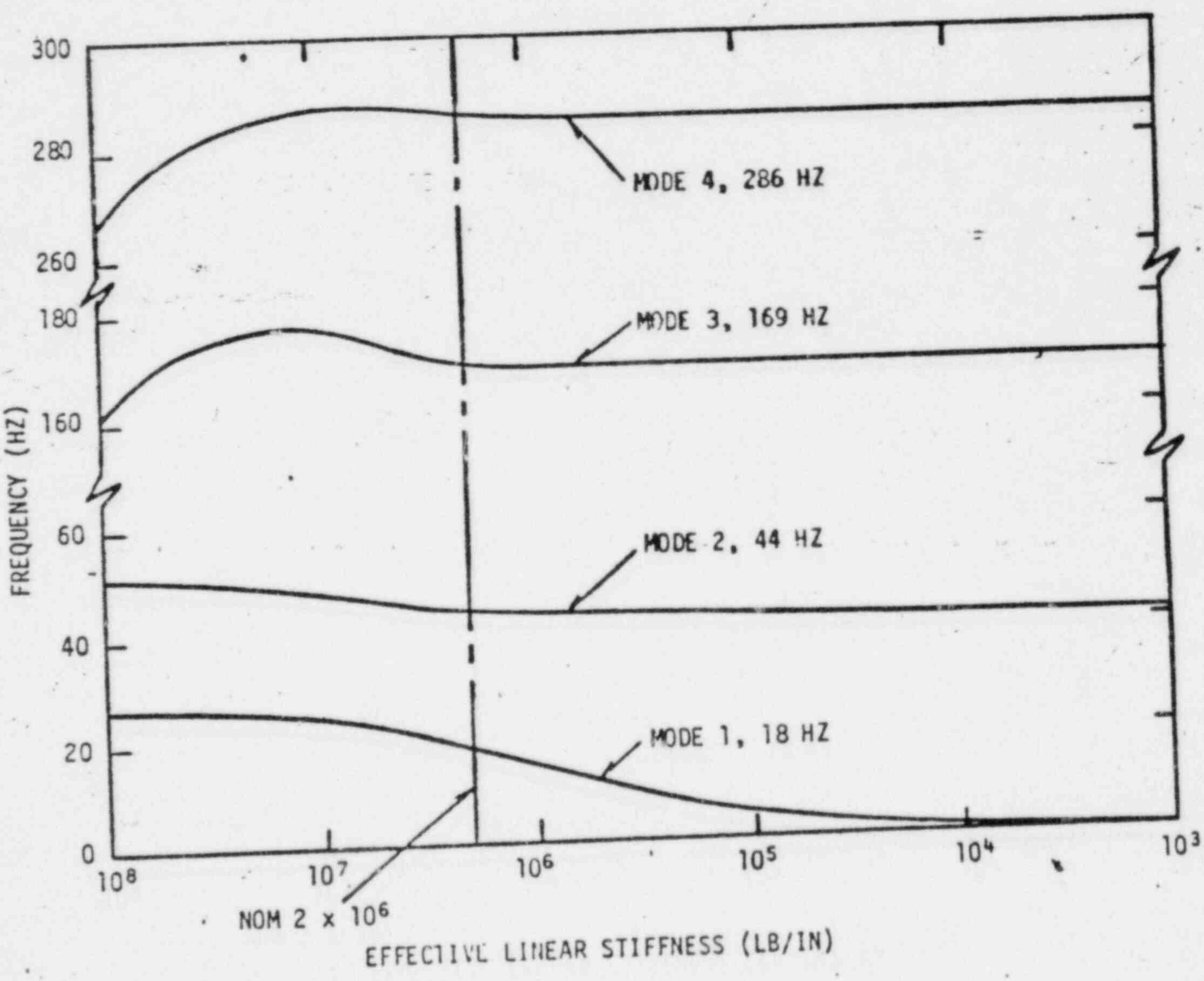
LATERAL RESPONSE

RELATIVE BENDING ROTATION IN VICINITY OF CRACK

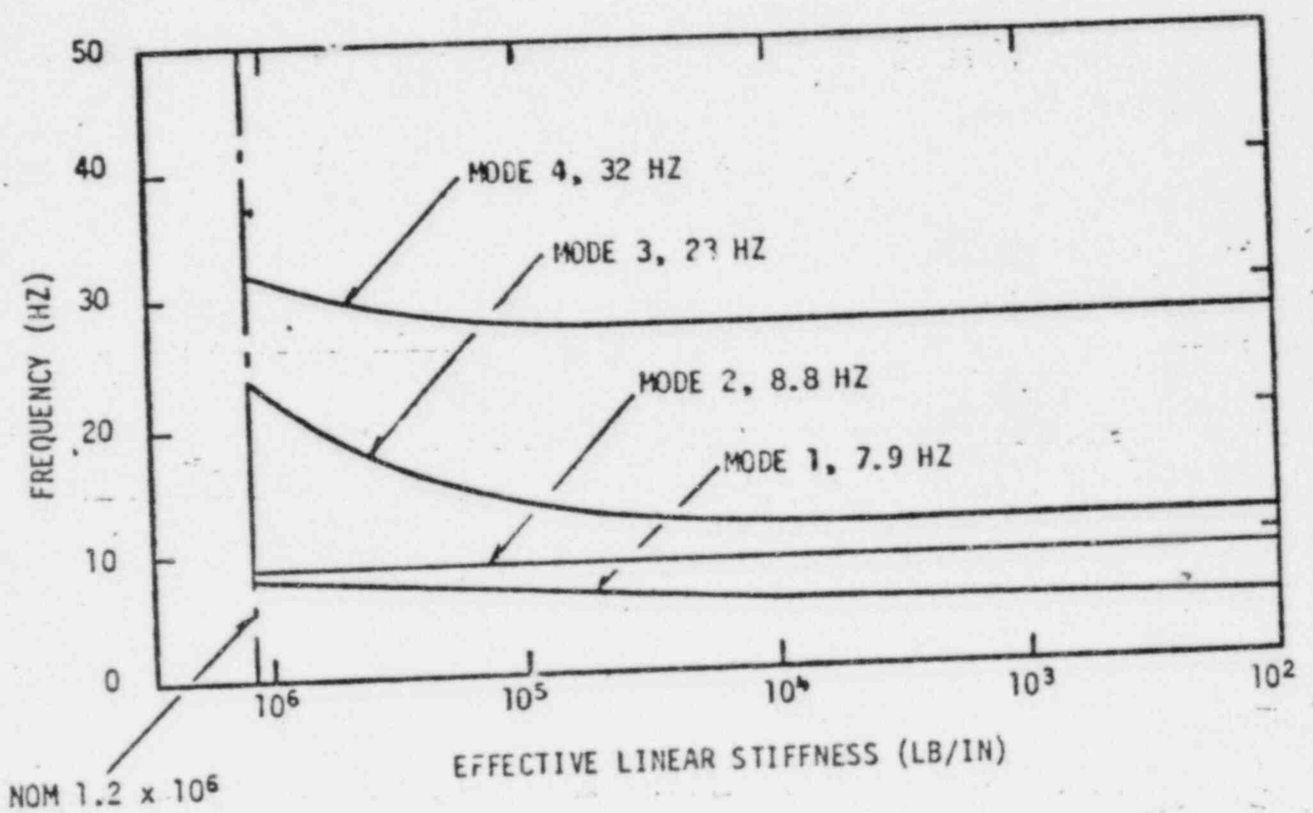


EFFECT OF THRUST BEARING STIFFNESS CHANGES

ON AXIAL MODE FREQUENCIES



EFFECT OF LOOSE LOWER MOTOR BEARING ON LATERAL MODES



SUMMARY OF RESULTS

- AXIAL AND TORSIONAL MODES DO NOT APPEAR TO BE POTENTIAL SOURCES OF THE PROBLEM
- THE LATERAL MODE WHICH PRODUCES THE HIGHEST RELATIVE VIBRATION RESPONSE IS THE 7.9 HZ BENDING MODE
- THE ALTERNATING STRESSES ASSOCIATED WITH THIS MODE ARE SMALL
- THERE IS ALSO NO EVIDENCE OF LARGE DEFLECTIONS ASSOCIATED WITH THIS MODE IN PUMPS WHICH ARE IN NORMAL CONDITION

OTHER ACTION

WORK IS CONTINUING TO ACHIEVE A BETTER UNDERSTANDING OF THE FAILURE EVENT.

SAFETY SIGNIFICANCE OF RCP SHAFT FAILURE

- SAFETY CRITERIA (ANSI N18.2)

  THE RCP IS NON-NUCLEAR SAFETY.

  THE PRESSURE RETAINING PARTS (RCPB) ARE SAFETY CLASS 1.

  THE ROTATING PARTS ARE SAFETY CLASS 2.

- THE ANSI STANDARD STATES:

  "THE REACTOR COOLANT PRESSURE BOUNDARY TOGETHER WITH ITS PROTECTION SYSTEMS SHALL BE DESIGNED SO THAT SUDDEN STOPPING OF ONE REACTOR COOLANT PUMP (CONDITION IV) DUE TO SEIZURE OR OTHER SIMILAR CAUSE WILL NOT RESULT IN FAILURE OF THE REACTOR COOLANT PRESSURE BOUNDARY."

- ANALYSES HAVE BEEN PERFORMED FOR:

  FLOW COASTDOWN.

  LOCKED ROTOR.

  SHAFT BREAK.

REACTOR COOLANT FLOW STOPPAGE ANALYSES

- FLOW COASTDOWN VS. INSTANTANEOUS STOPPAGE (LOCKED ROTOR)

  FLOW COASTDOWN ANALYSES FOR ONE AND TWO PUMPS HAVE BEEN PERFORMED (FSAR CHAPTER 14).

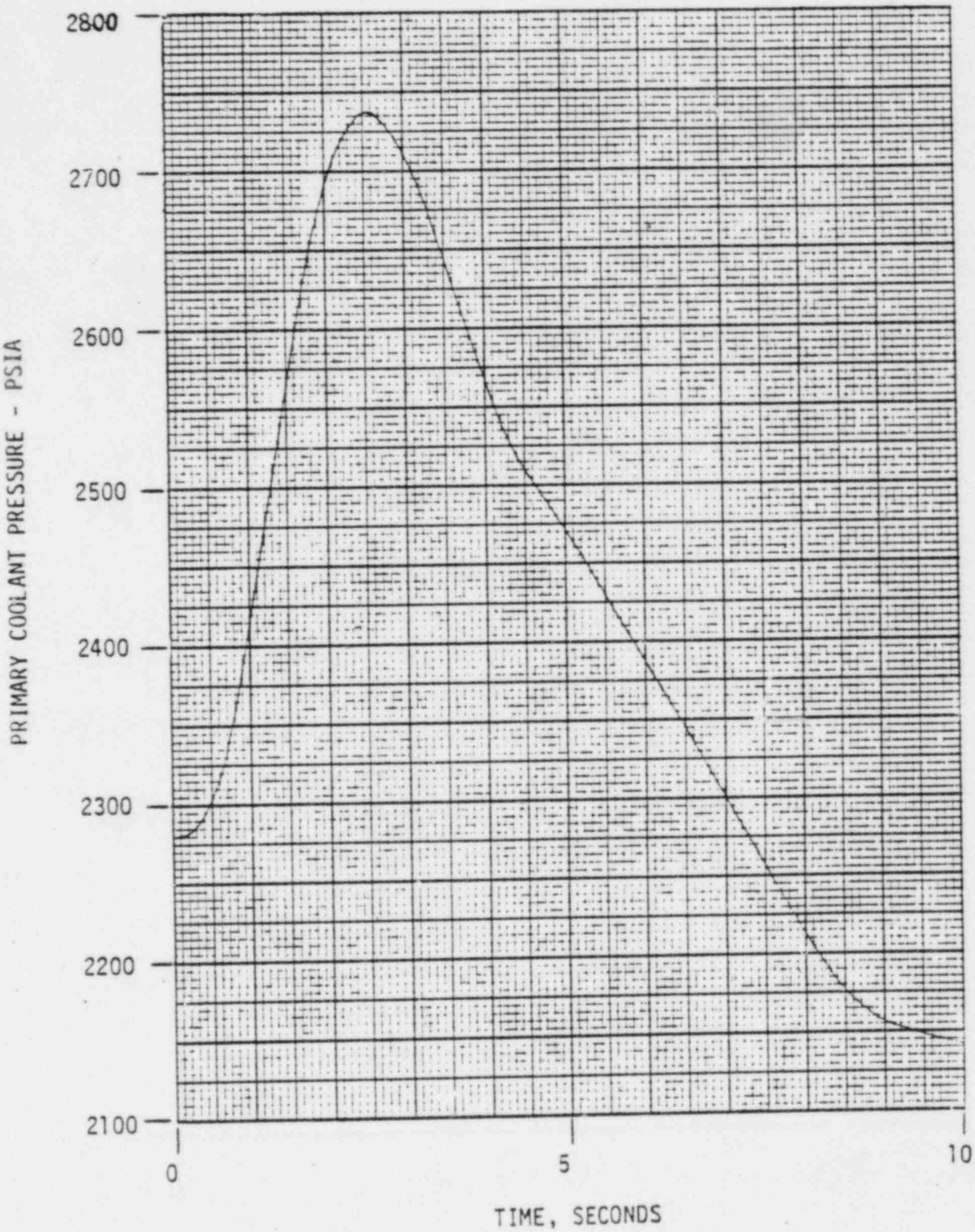
  LOCKED ROTOR ANALYSES FOR ONE PUMP HAVE BEEN PERFORMED (FSAR CHAPTER 14 AND, MORE RECENTLY, IN SUPPORT OF EPRI S&RV TEST PROGRAM).

  SHAFT BREAK ANALYSES HAVE BEEN PERFORMED.

- SHAFT BREAK TRANSIENTS RESEMBLE LOCKED ROTOR TRANSIENTS.

  A HIGH-PRESSURE "SPIKE" OCCURS IN A FEW SECONDS, THEN DECAYS AWAY.

  FOR NORTHERN STATES POWER, THE PEAK RCS PRESSURE CALCULATED FOR A LOCKED ROTOR TRANSIENT WAS 2737 PSIA (FSAR). IN SUPPORT OF THE EPRI VALVE PROGRAM, THE PEAK PRESSURE CALCULATED (2-LOOP REFERENCE PLANT) WAS 2745 PSIA.



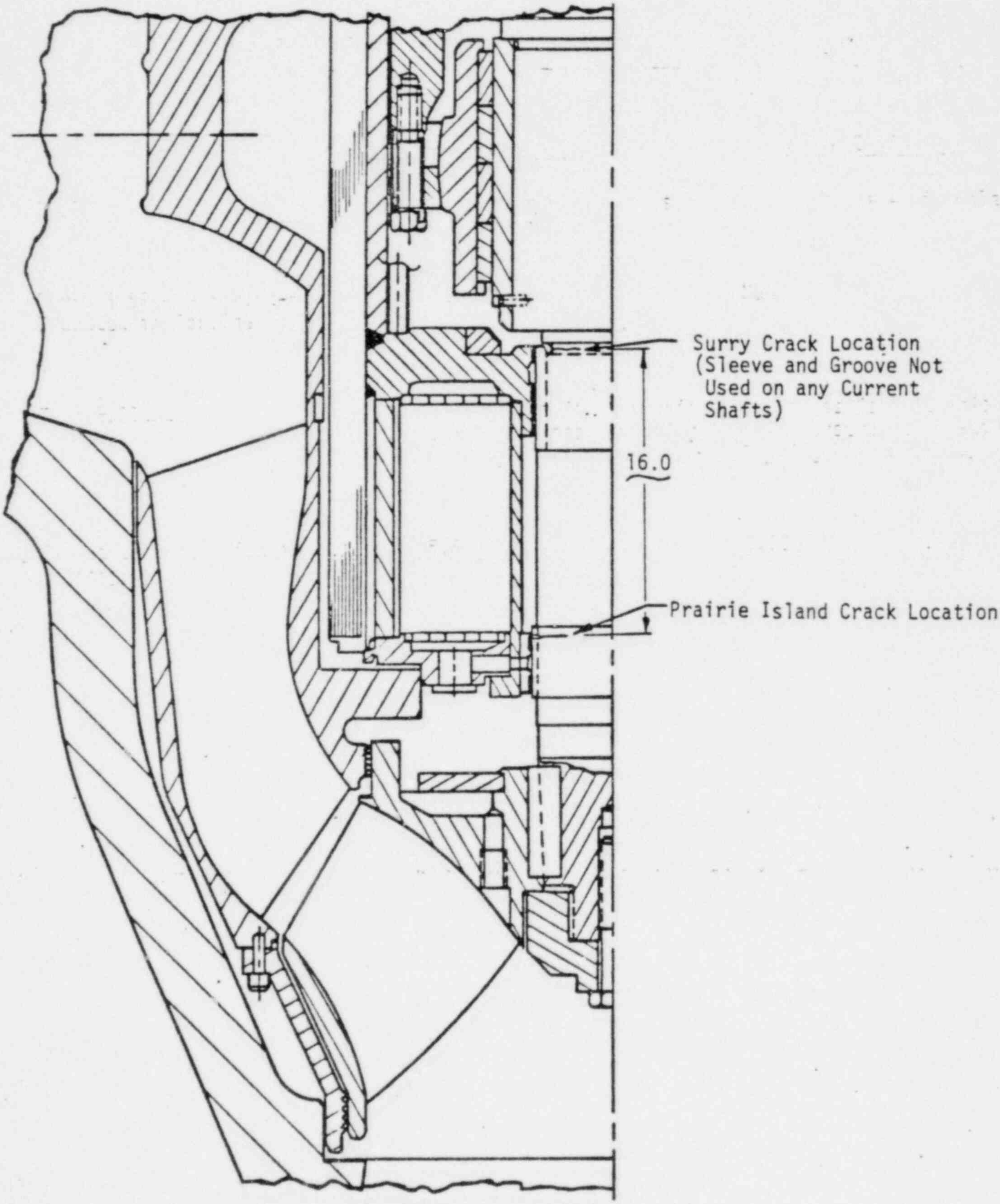
LOCKED ROTOR

~~Agenda~~ Agenda Item 6

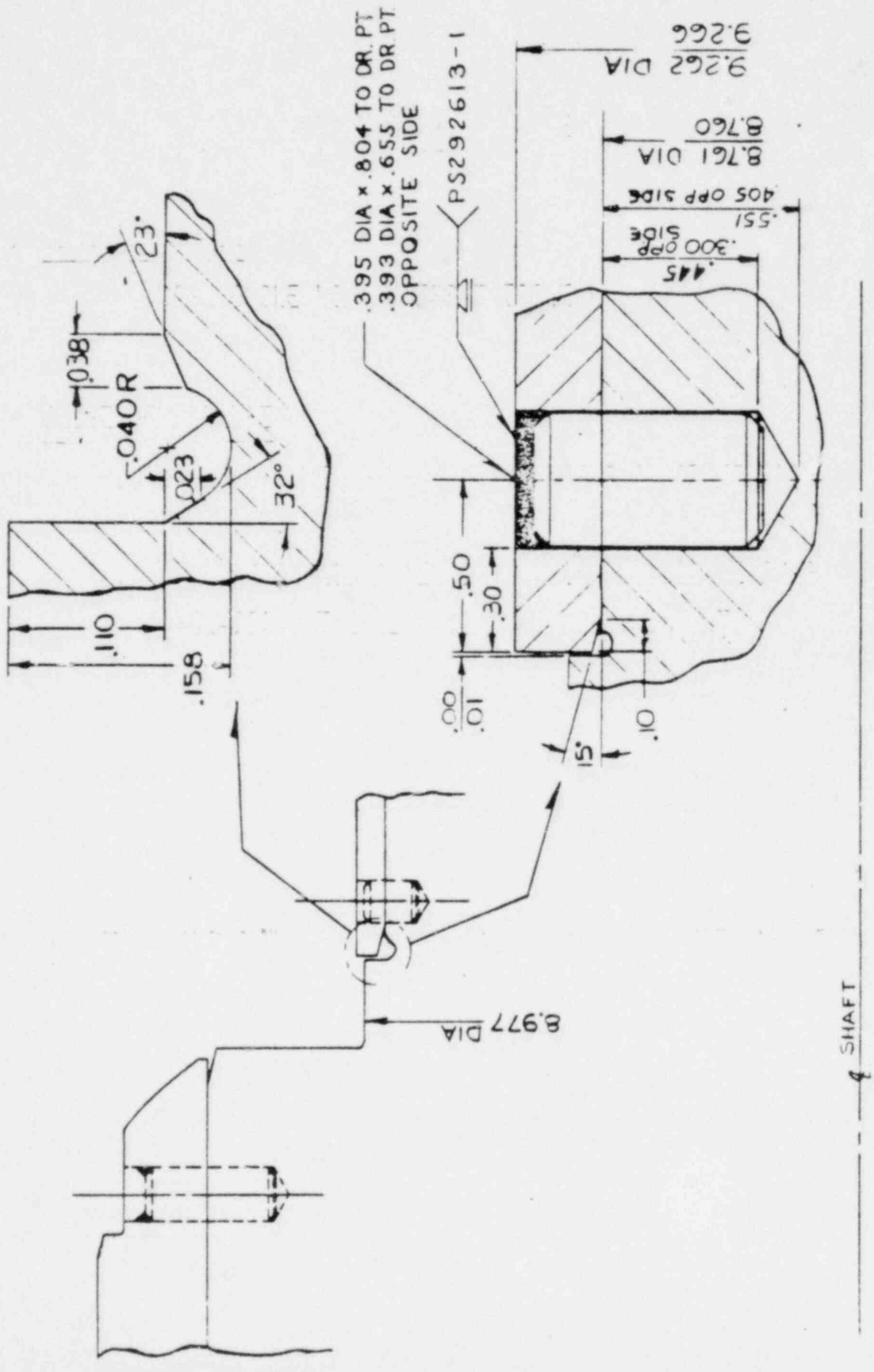
OPERATING HISTORY

PRAIRIE ISLAND VERSUS SURRY

	Hours Cold	Hours Total
Plant Operation		
Surry	1288	8994
Prairie Island	581	51,573
EMD Test		
Surry	354	703
Prairie Island	≈35	70
Total		
Surry	1642	9697
Prairie Island	616	51,643



SURRY VERSUS PRAIRIE ISLAND CRACK LOCATION



FAILURE LOCATION DETAILS
SURRY

RCP SHAFT FAILURE COMPARISONS

	SURRY	PRAIRIE ISLAND
Failure Location	Upper Sleeve - Grooves	Lower Sleeve - Pin Hole
Thermal Sleeve Shrink-Fit Expected in Service	Yes	No
Type Failure	High Stress Concentration, Low Overstress, Rotating Bending	No Stress Concentration, Low Overstress, Rotating Bending
Stress Concentration in Bending, K_t	5.5	2.88
Hydraulic Radial Thrust (lbs.)	5300	5104
Maximum Bending Stress (psi)	2850	1425
Pin Weld Residual Stress	Yes	Yes
Number of Times Weld Applied	1	2
Shaft Material	347 SST	347 SST

SURRY SHAFT FAILURE

CLASSIFICATION

HIGH CYCLE LOW STRESS FATIGUE

CAUSE OF FAILURE

- SHARP GROOVE -- VERY HIGH STRESS CONCENTRATION
- MAXIMUM ROTATING BENDING STRESS AT SINGLE PUMP OPERATION AT COLD SYSTEM CONDITIONS JUST SUFFICIENT TO PROPAGATE CRACK ONCE INITIATED
- HIGH LOCALIZED TENSILE RESIDUAL STRESS AT PIN WELDS PROVIDED SITES OF PRIMARY CRACK INITIATION

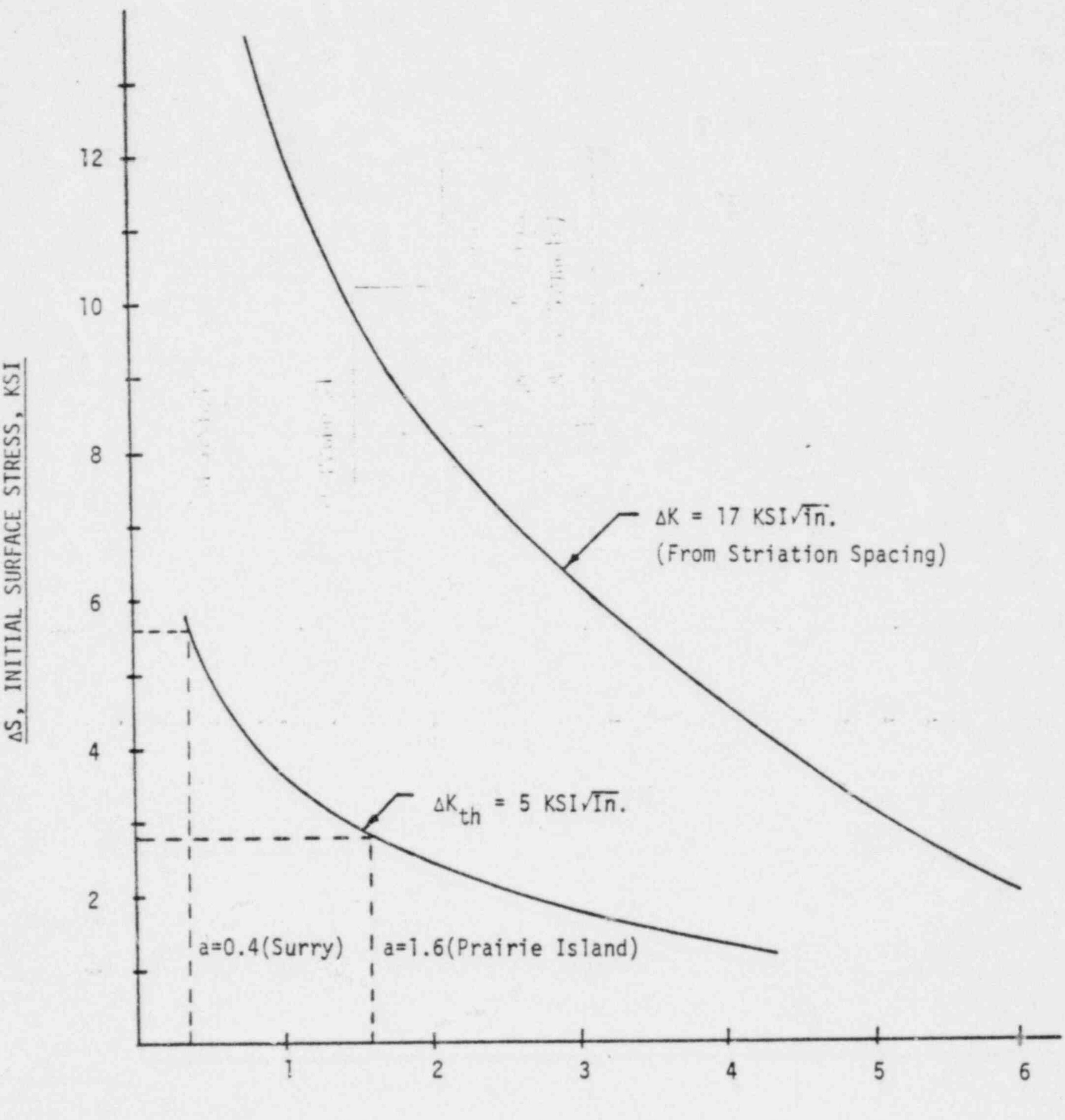
SURRY FATIGUE ANALYSIS

σ_a = 2850 psi

$k_f = Q(K_t - 1) + 1 = 4.6$

F.S. $= \frac{1}{1.07} = .93$ No Margin

STRESS VERSUS CRACK DEPTH



FAILURE NON-GENERIC

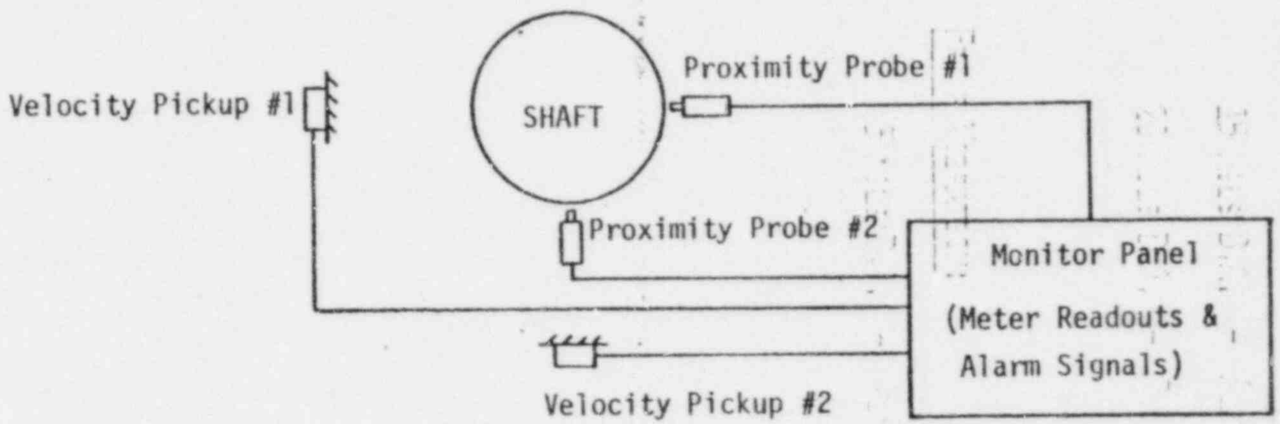
- SURRY FAILED SHAFT DESIGN DIFFERENT THAN ALL OTHER MODEL 93A REACTOR COOLANT PUMPS.

- PRAIRIE ISLAND SHAFT IS 1 OF 84 WITH SIGNIFICANT OPERATING HOURS.

- UNIQUE FEATURES OF PRAIRIE ISLAND UNIT #2 PUMP:
  - DOUBLE PIN WELD
  - LOOSE LOWER MOTOR BEARING

- NEED ABNORMALLY HIGH LOADING TO PROPAGATE CRACK. SOURCE OF LOADING UNKNOWN.

SHAFT VIBRATION SYSTEM



Agenda Item 7

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SHAFT VIBRATION MONITORING SYSTEM

OPERATING 93A RCP'S:

WITH WESTINGHOUSE SYSTEM	- 10 PLANTS
WITH OTHER SYSTEMS	- 10 PLANTS
NONE (PER WESTINGHOUSE RECORDS)	- 7 PLANTS (6 DOMESTIC)

NON-OPERATING 93A RCP'S:

WITH WESTINGHOUSE SYSTEM	- 8 PLANTS
WITH OTHER SYSTEMS	- 5 PLANTS
NONE (PER WESTINGHOUSE RECORDS)	- 10 PLANTS (9 DOMESTIC)

VIBRATION LEVEL CRITERIA FOR WESTINGHOUSE SYSTEM

SHAFT VIBRATION:

15 MILS D.A. - ALARM

20 MILS D.A. - RECOMMEND SHUTDOWN

FRAME VIBRATION:

3 MILS D.A. - ALARM

5 MILS D.A. - RECOMMEND SHUTDOWN