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Hydraulic Laboratory Technical Memorandum PAP-1167

Senator Wash Penstock Stress and Vibration Testing

Prepared for Lower Colorado Region and Yuma Area Office

U.S. Department of the Interior
Bureau of Reclamation
Technical Service Center
Hydraulic Investigations and Laboratory Services Group
Denver, Colorado

March 2018

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Introduction

The Senator Wash Pump-Generating Plant, located near Yuma, AZ was constructed in the mid-1960s as an off-stream pumping facility to help improve water scheduling of the Colorado River. Pumping allows part of the river flow to be stored in Senator Wash Reservoir upstream of Imperial Dam and then released (using generating mode) to the river for downstream use when needed [1]. The facility is operated and maintained by the Imperial Irrigation District (IID). The plant contains six vertical shaft Francis-type pump turbines, each with pumping capacities of about 165 ft^3/s at the rated head and generating release capacities of approximately 200 ft^3/s [2]. A header pipe connects the six units to a common steel penstock. The 10-ft-diameter penstock has ½-inch wall thickness, and is supported by ring stiffeners approximately every 40-ft.

During an inspection in early 2017 Reclamation personnel noticed movement in the penstock while operating which appeared to produce lateral deflections of about ¼-inch. Previously the penstock expansion joint had seized and was replaced in January 2017 before the inspection was made. IID personnel suspected that the expansion joint replacement may have caused increased penstock vibrations but this could not be confirmed. In March 2017, Josh Mortensen from TSC's Hydraulic Investigations and Laboratory Services Group visited the site to observe the penstock vibrations. Based on those observations, the decision was made to perform an operational field test to quantify penstock vibrations and associated stresses and determine whether they are within allowable limits [3]. The field test was performed November 13-16, 2017.

The main objective of this field study was to determine whether penstock vibrations and operating stresses at Senator Wash Pump-Generating Plant are acceptable or warrant modifications or operating restrictions.

Field Test Approach

The test plan included measuring both vibration and stress for a range of operating conditions. A vibration survey was initially completed using 8 unidirectional accelerometers arranged both axially and circumferentially along and around selected 40-ft and 32-ft spans (location of new expansion joint) of the penstock. These measurements helped determine modes of vibration, dominant frequencies, and deflections of the pipe wall for operation in both generating and pumping modes. Strain gages were then installed at locations of maximum deflections and other strategic locations to quantify penstock stresses during operation. Stress measurements were compared to the endurance limit of steel for fatigue life assessment.

Endurance Limit

For this analysis it was assumed that the endurance limit is 0.504 times the minimum ultimate tensile strength of the steel (S_{ut}) [4]. If the fluctuating stresses due to vibration during operation exceed the endurance limit the penstock will experience fatigue and has a limited operational life before cracking eventually forms and propagates to failure. Since the actual grade of the penstock steel could not be identified, ASTM A283/A283M steel (S_{ut} = 55,000 psi) was assumed [5], which results in an endurance limit of 27,720 psi. According to the AWWA M11 standard, another metric that may be used is to limit the penstock hoop stress to 75% of the yield strength [5], resulting in a maximum hoop stress of 22,500 psi which is more conservative in this case.

Instrumentation

Vibration Measurements

Eight accelerometers were used to measure both axial and circumferential vibration modes. Measurements were made with Wilcox Research Model 797L accelerometers, with a range of 0-10g, mounted on magnetic bases which allowed them to be securely affixed to the penstock and easily removed to various locations (Figure 1). For measurements of axial vibration, the accelerometers were equally spaced on the top of the penstock, longitudinally along half of the span between ring supports on both the 32-ft and 40-ft sections (Figure 16, Appendix A). For measurements of circumferential vibration accelerometers were equally spaced azimuthally over both 90° and 180° spans as shown in Figure 12 of Appendix A. Accelerometer data were collected using a Measurement Computing 1616HS data acquisition module connected to a laptop computer. Accelerometer signals were simultaneously recorded with a sampling frequency of 1,000 sample/s for 60 seconds during each test condition.

The root-mean-square (RMS) of each accelerometer was used as a quick indicator in the field to determine locations of maximum pipe movement. Locations with high RMS values were chosen for strain gage installation in an attempt to measure maximum stress. An in-depth modal analysis to define operating deflection shapes was not performed for this study.

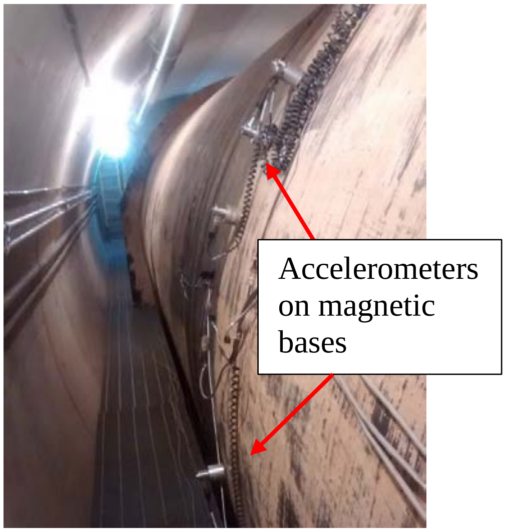

Figure 1. Accelerometers mounted circumferentially on the penstock.

Stress Measurements

Stresses were obtained using ten strain gages installed at selected locations on the outside of the penstock (Table 2, Appendix A). Strain values were multiplied by the modulus of elasticity of steel (assumed 30×10^6 psi) to obtain the stress. The strain gages were glued to the bare metal of the penstock by following an established procedure for surface preparation and installation. Measurement locations were determined from vibration testing that was performed previously.

The 350-Ohm strain gages (Micro-Measurements) were wired in quarter-bridge configurations to obtain independent, simultaneous strain measurements from each gage. Gages were installed in two orientations to obtain both axial and hoop stresses during penstock operation (Figures 2 and 3). Strain data were collected using a 24-bit Micro-Measurements D4 signal conditioning system and laptop computer with the sampling frequency set at the maximum rate of 8 sample/s during the entire test period.

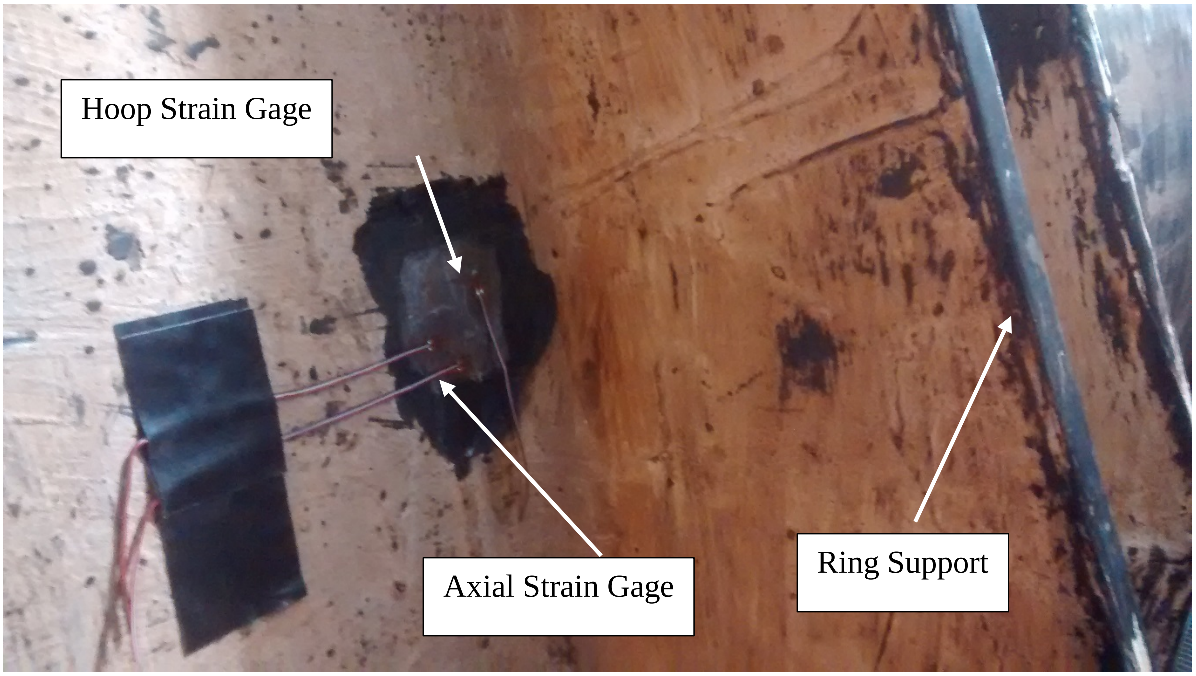

Figure 2. Strain gages installed on the 40-ft span near a penstock ring support.

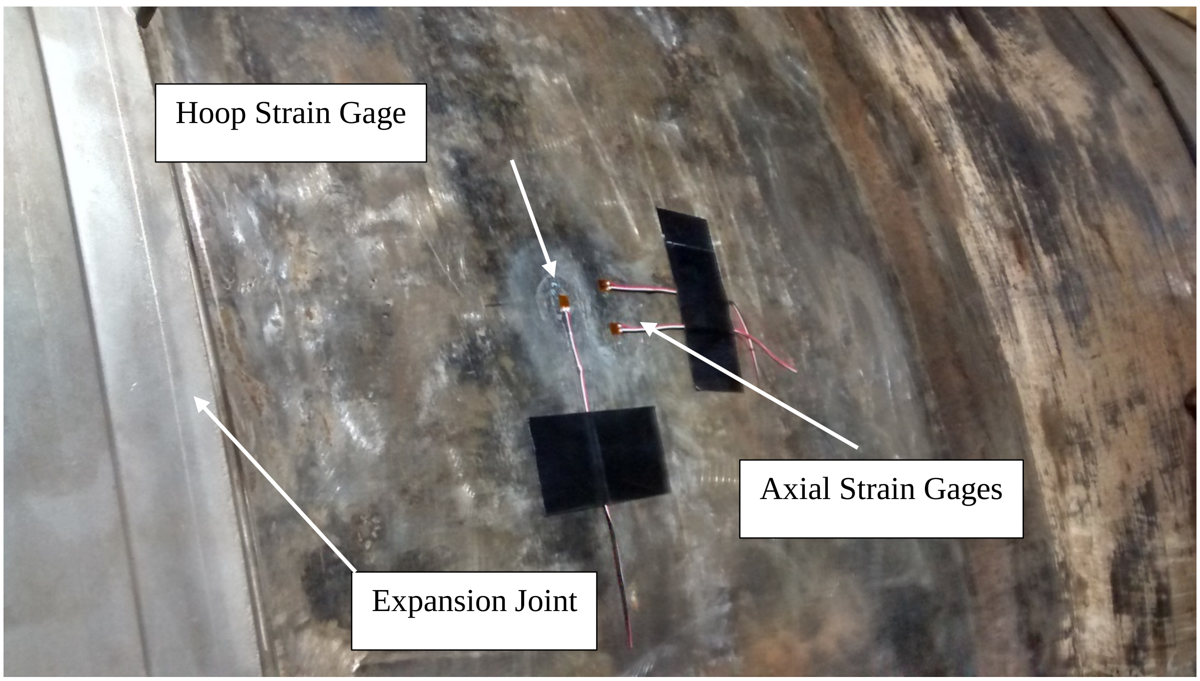

Figure 3. Strain gages installed on the 32-ft span near the new expansion joint.

Auxiliary Measurements

Other data collected during both vibration and stress testing included penstock discharge and water surface elevations of Senator Wash Reservoir and Squaw Lake. Direct penstock pressure measurements were not made due to difficulty in accessing existing pressure ports on the penstock. Runner speed was not measured but assumed constant at 360 rpm for every operating condition [2]. This speed results in a running frequency of 6 Hz and blade passing frequency of 42 Hz as the turbine runner has seven blades.

An externally mounted acoustic flow meter was used to measure penstock discharge during testing. Discharge measurements were successful for test conditions with up to three units operating simultaneously. However the signal was lost each time a fourth unit was brought online. The reason for this could not be identified nor corrected during testing. Water surface elevations were read from the Senator Wash SCADA system.

Field Test Results and Discussion

Operational vibration testing was performed on Tuesday 11/14/2017 and stress testing on Thursday 11/16/2017. Attempts were made for both tests to be performed under the same operating conditions, but due to different river operations two days apart, there were differences in water surface elevations and hence discharges through the penstock. Both vibration and stress testing began with a no-flow condition and then the units were brought online one at a time first in generating mode. The maximum number of simultaneous units online was four since Units 4 and 6 were undergoing maintenance at the time. This process was then repeated in pumping mode. Test logs documenting operating conditions and measurements during testing are shown in Table 3 of Appendix B and Table 8 of Appendix C, respectively.

Vibration Testing

Modes of vibration

A full modal analysis to resolve operating deflection shapes was not performed for this study. However, vibration results helped determine locations for strain gage measurements. The RMS of each accelerometer signal was used as an indicator of locations of maximum deflection which could quickly be determined in the field. While the actual operating modes cannot be determined from RMS data only, Figure 4 suggests that a k=3 mode for axial vibrations is possible for both the 32-ft and 40-ft sections. Circumferential vibration results in Figures 5 and 6 from the 90° orientation suggest a possible n=4 mode for both penstock sections. These can be compared to the nodal patterns shown in Figure 7.

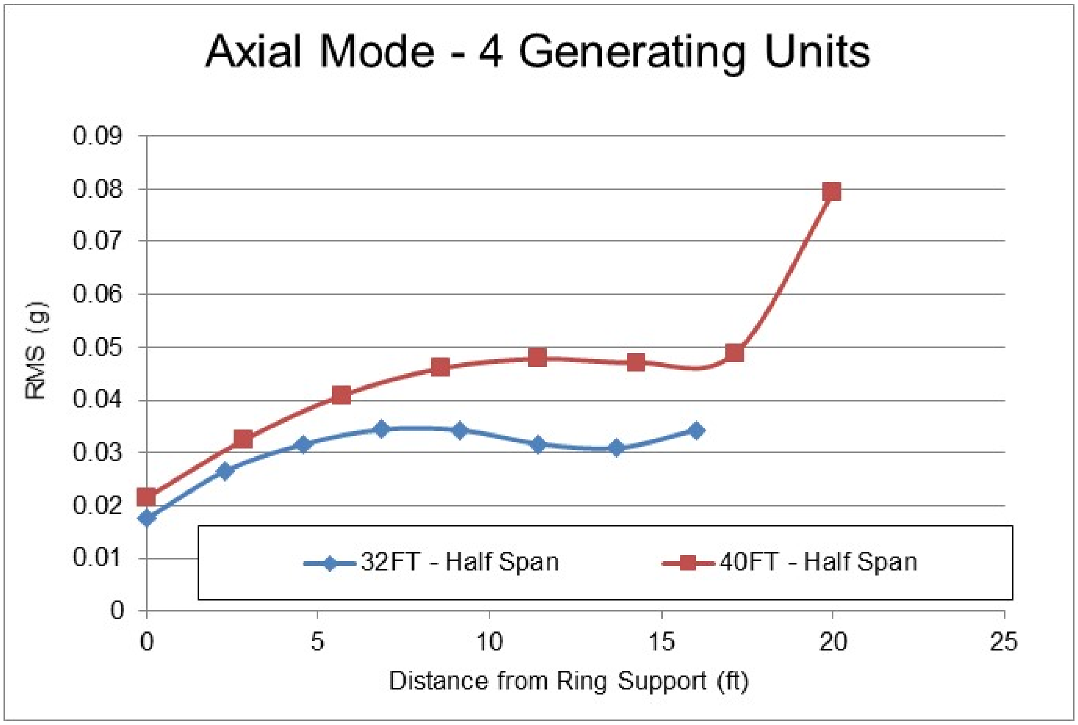

Figure 4. RMS accelerations from axially mounted accelerometers for both 32-ft and 40-ft penstock sections.

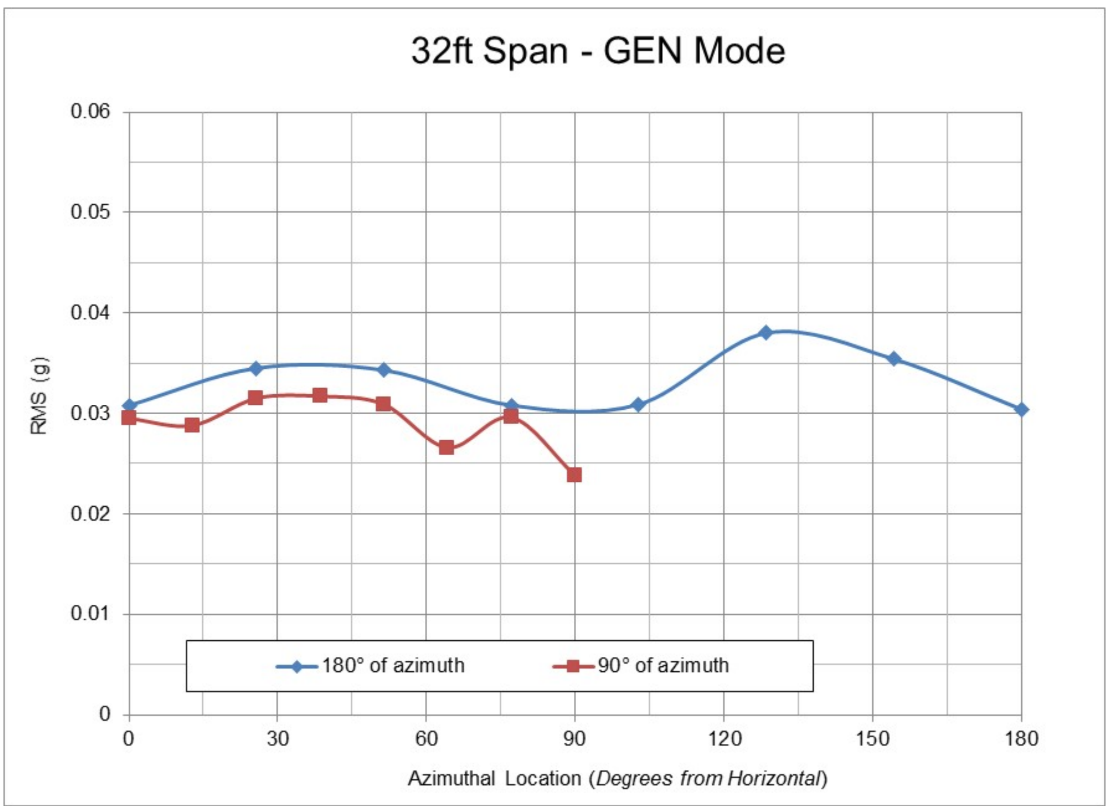

Figure 5. RMS accelerations from circumferentially mounted accelerometers for the 32-ft span.

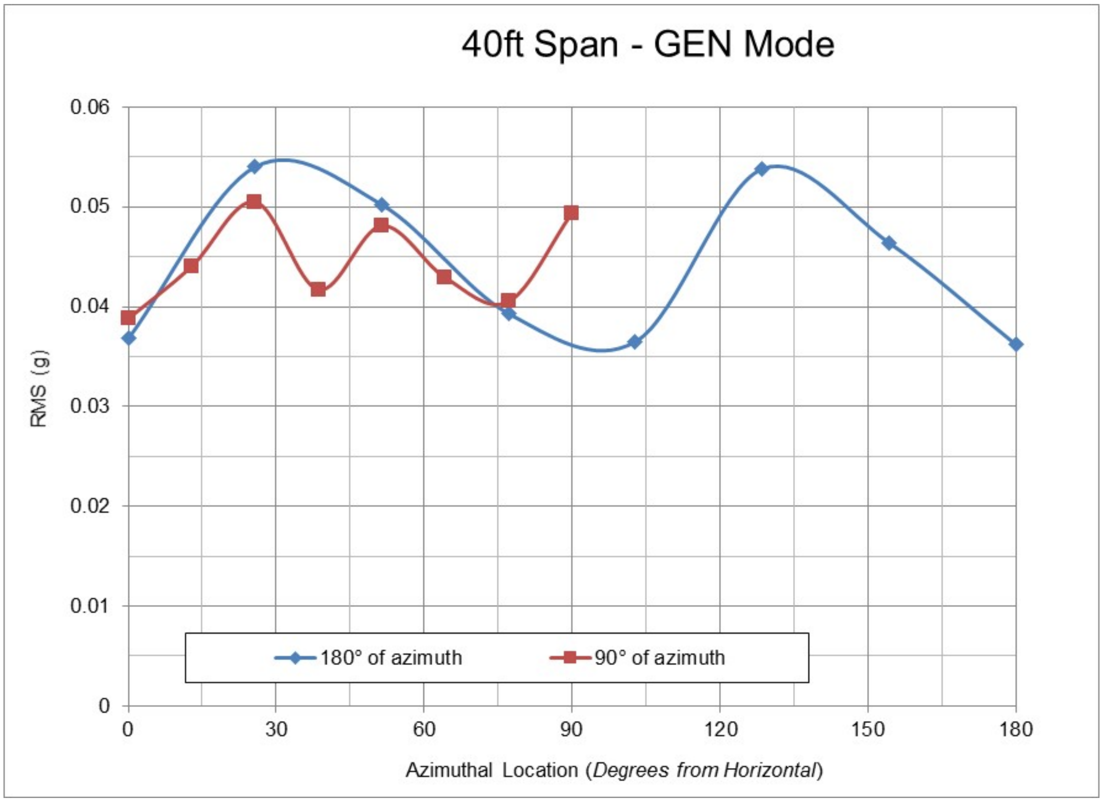

Figure 6. RMS accelerations from circumferentially mounted accelerometers for the 40-ft span.

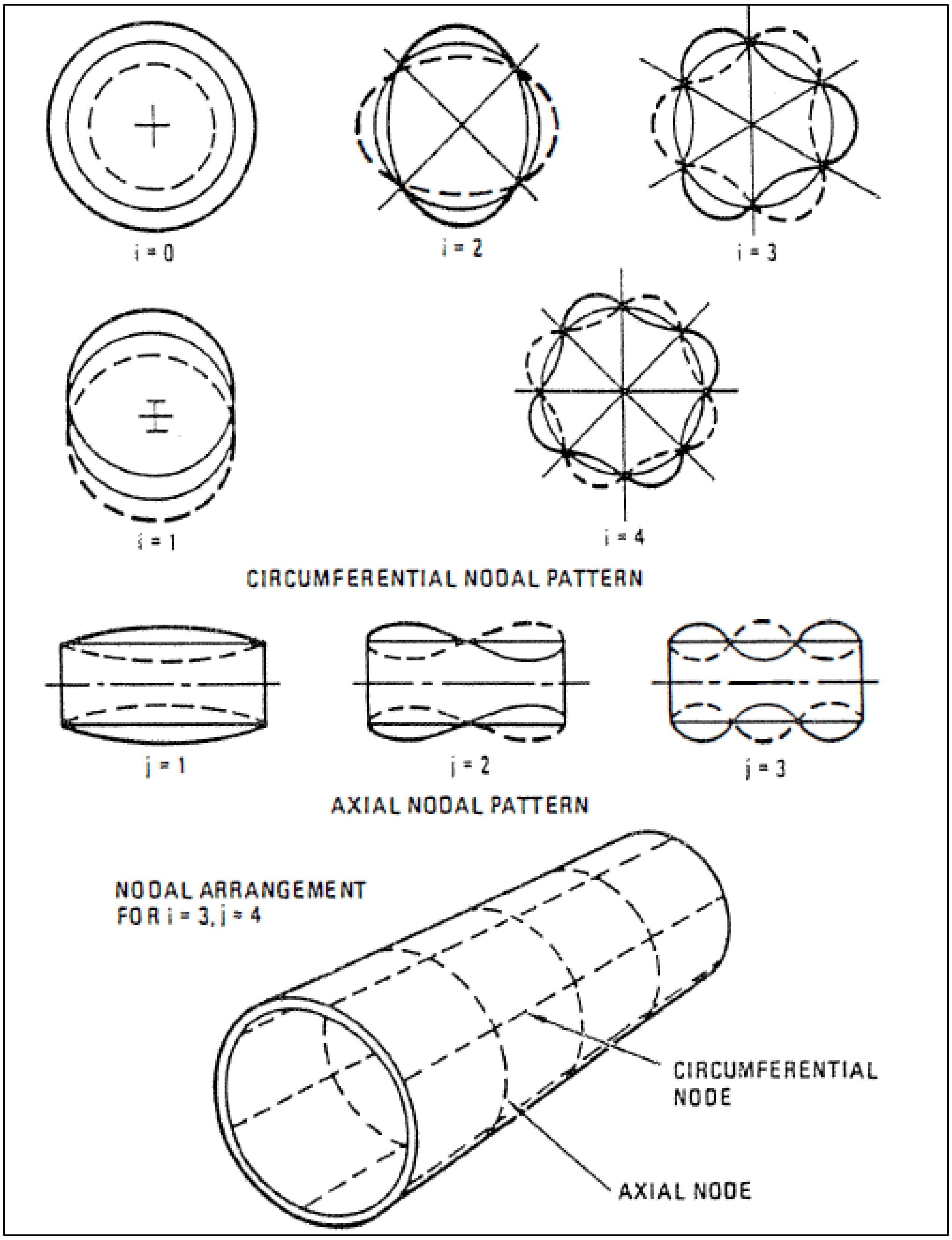

Figure 7. Schematic of circumferential and axial vibration modes according to Wachel, et al. [6].

Penstock Natural Frequency

To help understand the nature and implications of operational vibrations, efforts were made to identify the resonant frequencies of the penstock. For circumferential vibrations Equations 1 and 2 [6] were used to estimate the first natural frequency of the penstock, resulting in 19.6 Hz. It is unknown how much the added mass of the water in the penstock affects the accuracy of this result, but it is expected that it will act to reduce natural frequencies.

$$f_n = \frac{\lambda_i}{2\pi R}\left[\frac{E}{\gamma(1-\upsilon^2)}\right]^{1/2} \qquad \text{Eq. (1)}$$

$$\lambda_i = \frac{1}{12^{1/2}}\,\frac{h}{R}\,\frac{i(i^2-1)}{(1+i^2)^{1/2}}; i = 2,3,4 \ldots \qquad \text{Eq. (2)}$$

Where:

f_n = shell wall natural frequency (Hz)
λ_i = frequency factor (-)
R = radius of pipe wall (inch)
E = modulus of elasticity (lb/in^2)
υ = Poisson's ratio
γ = mass density of pipe wall (lb-s^2/inch^4)
h = pipe wall thickness (inch)
i = mode number

Prior to testing, the penstock was struck with an impulse hammer in an attempt to excite the resonant frequencies. This was done at a no flow condition with the penstock pressurized, which would account for the added mass of the water. The excitation was produced by striking the penstock with a specially designed impulse hammer with a rubber tip and the response was measured using the eight accelerometers. Figures 8 and 9 show the response of the 32-ft and 40-ft sections respectively which represent the average of three strikes.

Results indicate that the dominant frequencies are much higher than those expected during operation i.e., greater than the blade passing frequency of 42 Hz. While not as strong, the first natural frequency of approximately 10 Hz, approximately half of the estimated natural frequency without the added mass of the water, does appear in the frequency spectrum. This is of interest as it is near the running frequency of the plant (6 Hz) and within the range of operational frequencies.

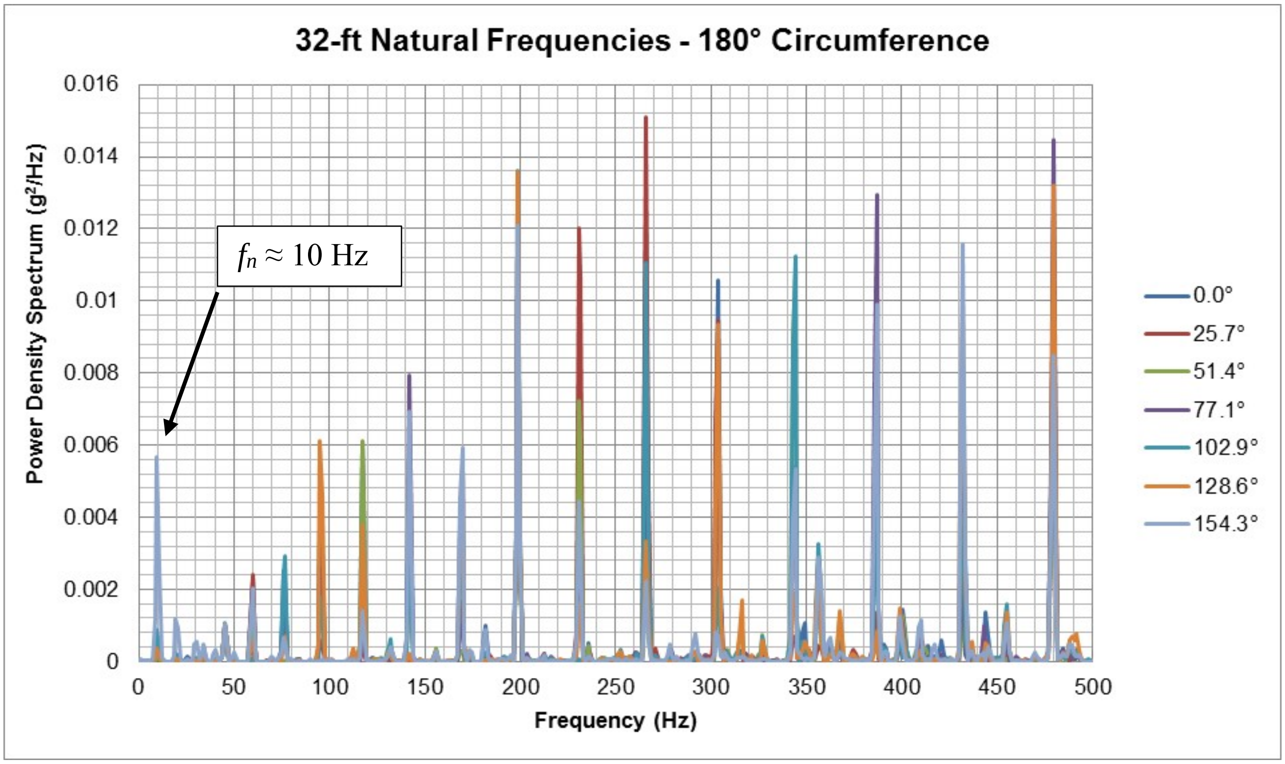

Figure 8. Power Density Spectrum vs. frequency from impulse hammer on the 32-ft section.

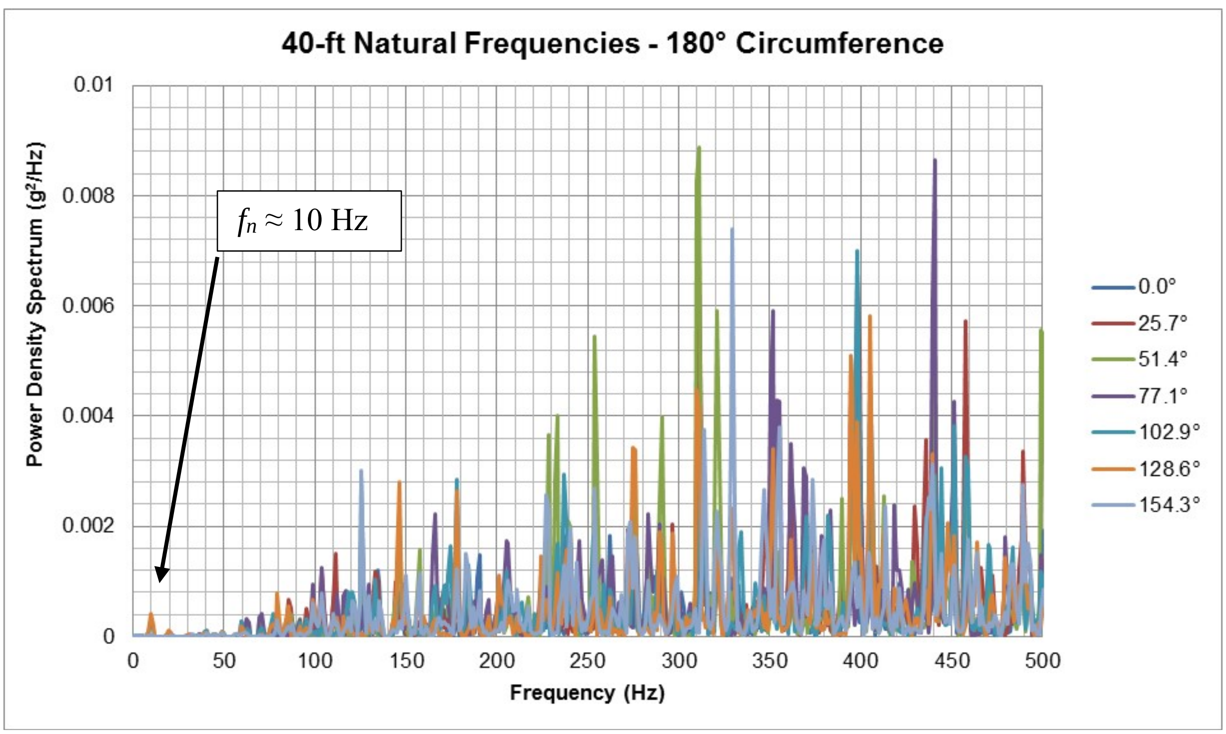

Figure 9. Power Density Spectrum vs. frequency from impulse hammer tests on the 40-ft section.

Dominant Operating Frequencies

From operational vibration tests dominant frequencies were identified near or within the range of running and blade passing frequencies (6 – 42 Hz). Circumferential frequencies are shown in Figures 10 through 12 and all operational frequency data are summarized in Tables 4 and 5 of Appendix B. Vibration at the running frequency dominated in most test conditions and the frequency increased as more units were brought online as shown in Figure 11. Operation in pumping mode produced vibrations dominant at the running and blade passing frequency but amplitudes were significantly less compared to generating mode.

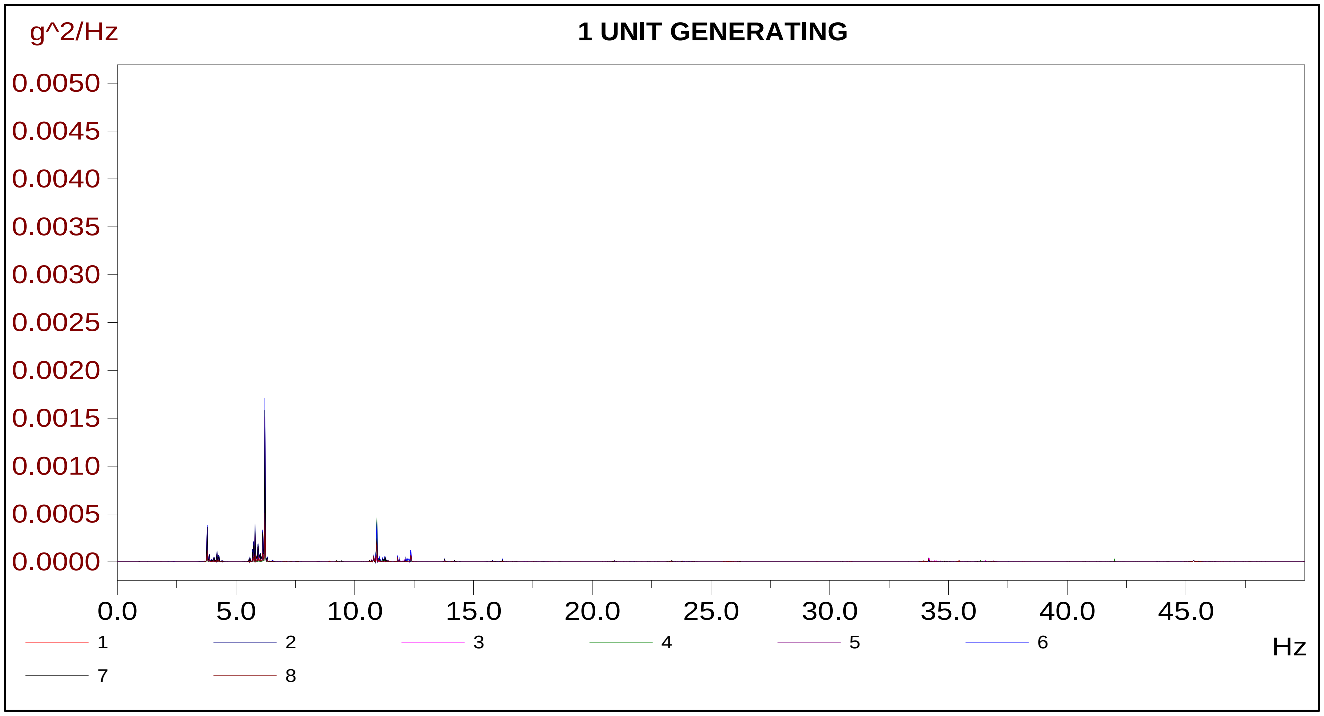

Figure 10. Power Density Spectrum for 1 unit generating (G1).

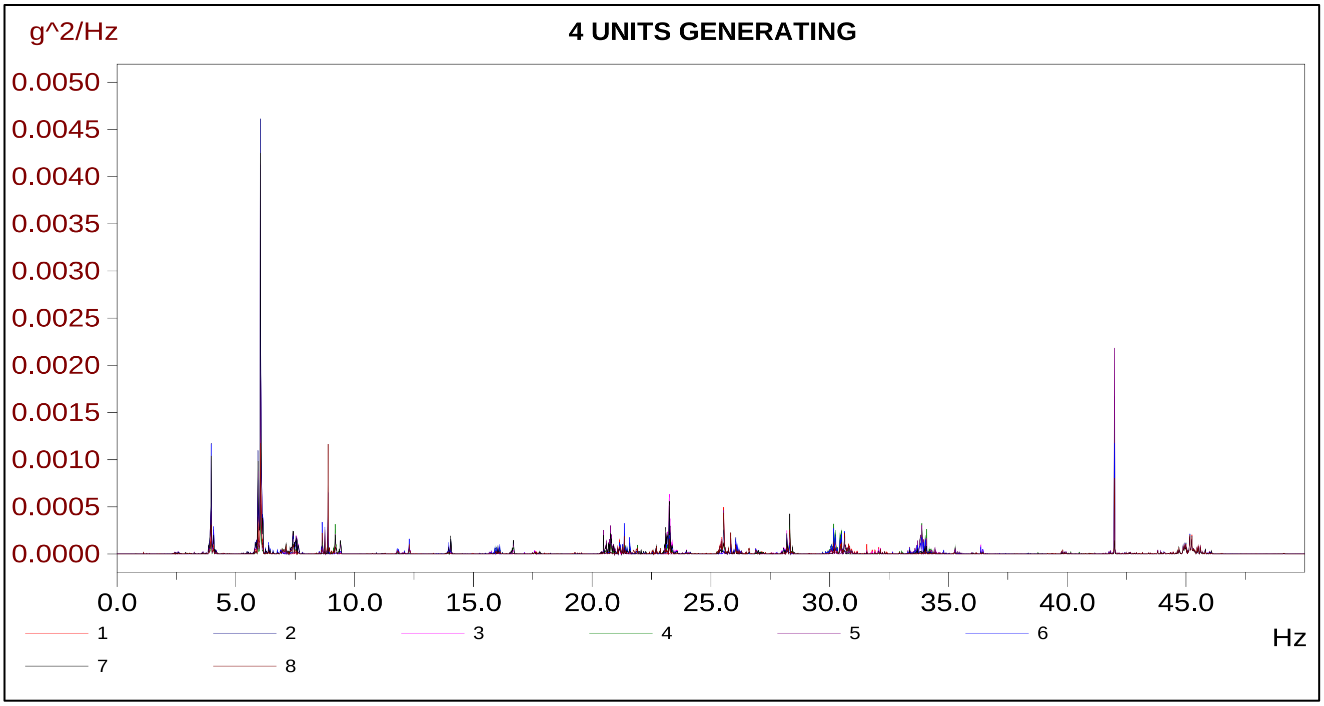

Figure 11. Power Density Spectrum for 4 units generating (G1, G2, G3, G5).

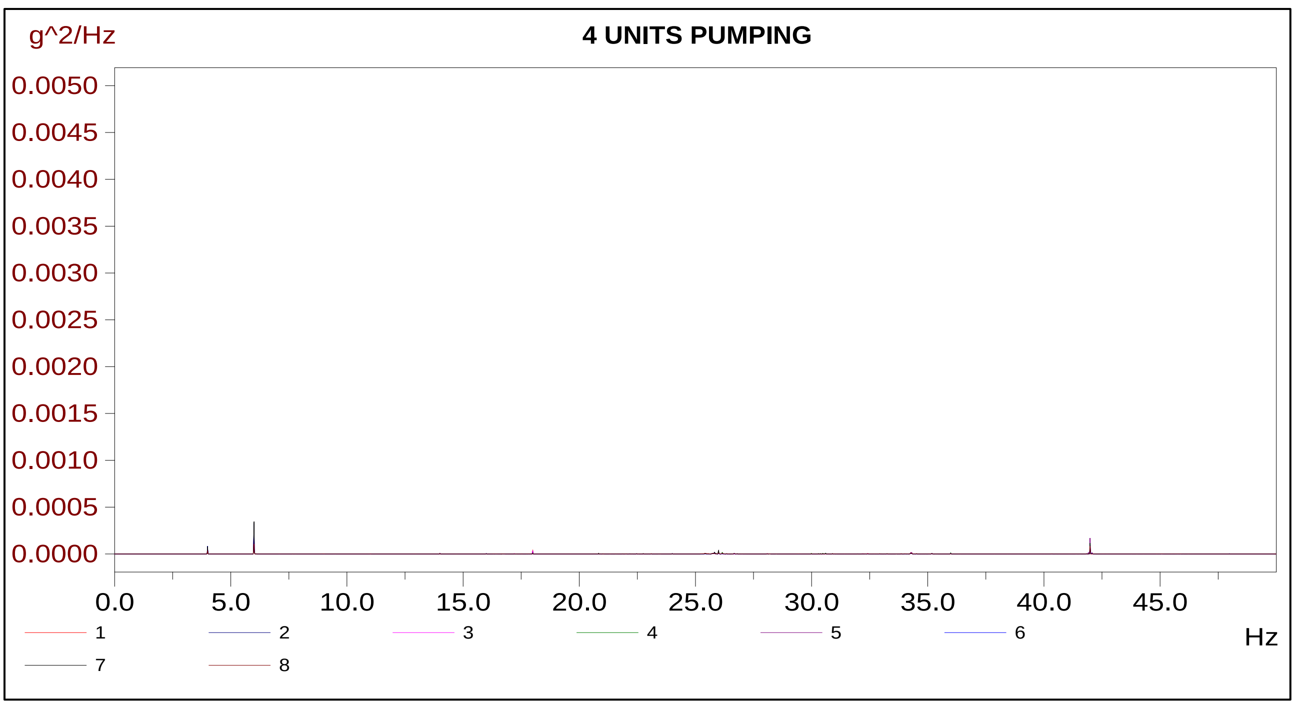

Figure 12. Power Density Spectrum for 4 units generating (P1, P2, P3, P5).

Penstock Displacement

Deflections in the penstock wall may be determined by double integration of the acceleration-time signals to obtain displacements (Table 1). The results indicate that deflections increased with additional units in generating mode, producing a maximum peak-to-peak displacement of approximately ¼-inch. This result agrees well with visual observations made from the initial inspection when vibration was first identified. While these deflections appear significant visually when observing the penstock in operation, they are not likely to have adverse impacts to the penstock and its components as long as they do not produce excessive stresses. In pumping mode the maximum displacement was on the order of 1/16-inch peak-to-peak which was hardly noticeable.

Table 1. Summary of peak-to-peak penstock displacements derived from acceleration data.

Operation	Penstock Span [ft]	Avg P-P [in]	Max P-P [in]
G1	32	0.051	0.087
G1	40	0.088	0.151
G1, G2	32	0.062	0.083
G1, G2	40	0.099	0.163
G1, G2, G5	32	0.122	0.172
G1, G2, G5	40	0.192	0.278
G1, G2, G3, G5	32	0.136	0.229
G1, G2, G3, G5	40	0.163	0.255
P1	32	0.030	0.042
P1	40	0.057	0.083
P1, P2, P3, P5	32	0.037	0.068
P1, P2, P3, P5	40	0.047	0.070

Potential Sources of Vibration

While test data from the accelerometers were helpful in identifying dominant frequencies, the exact cause of the vibration could not be identified. However, some potential causes have been excluded as part of this study. An inspection and vibration test on the motor of Unit 1 was made in October 2017 that showed vibration issues with that unit were likely electrical in nature as the dominant frequency was two times the line frequency (120 Hz) [7]. This is outside the frequency range measured on the penstock and is not likely to be related.

Flow induced vibrations are very unlikely to be the source of excitation due to relatively small penstock velocities, low dominant frequencies observed during operation, and no noise associated with the vibration. Flow past features with discontinuities that could cause a flow disturbance, such as the new expansion joint or short branch for manhole access, would produce natural frequencies that are much higher than those measured during testing. Furthermore, such a flow-induced resonance is typically accompanied by either a “droning” sound or a pure tone (single frequency) which were not observed in this case.

Since penstock vibrations were not measured or observed before the old expansion joint was replaced it is difficult to predict what impact the new expansion joint now has on penstock vibrations. In late 2014 an inspection indicated the expansion joint was experiencing leakage, corrosion, and restricted motion. As such, replacement was recommended [8]. It is possible that the old joint had seized up, potentially stiffening the penstock considerably and restricting pipe movement, resulting in natural frequencies well above the blade passing frequency. Conversely the new expansion joint, which now allows pipe movement, may have a lower natural frequency that is closer to the running frequency of the units. Again, this cannot be confirmed and hence results from the stress testing are the only means of assessing the severity of penstock vibrations.

Stress Testing

Testing began with an empty penstock to reference the strain gage readings to a zero-stress condition (which zeroes dead loading due to weight of the penstock). Data were collected while the penstock filled and then began operation in generating mode starting one unit at a time. A minimum of ten minutes was given between each additional unit coming online to record stresses at each operating condition. After operating in generation mode with four units on-line, the plant was switched to pump mode with one unit and then all four. The test sequence is shown in Figure 13 which also displays measured stresses in the axial direction at six different locations.

Maximum axial stresses occurred at mid-span in the 40-ft section. Negative values indicate that the steel was in compression which was due to the weight of the water causing the penstock to sag. The absolute maximum was 1,620 psi with a peak-to-peak fluctuation of 300 psi. These values are within the normal range and well below the endurance limit.

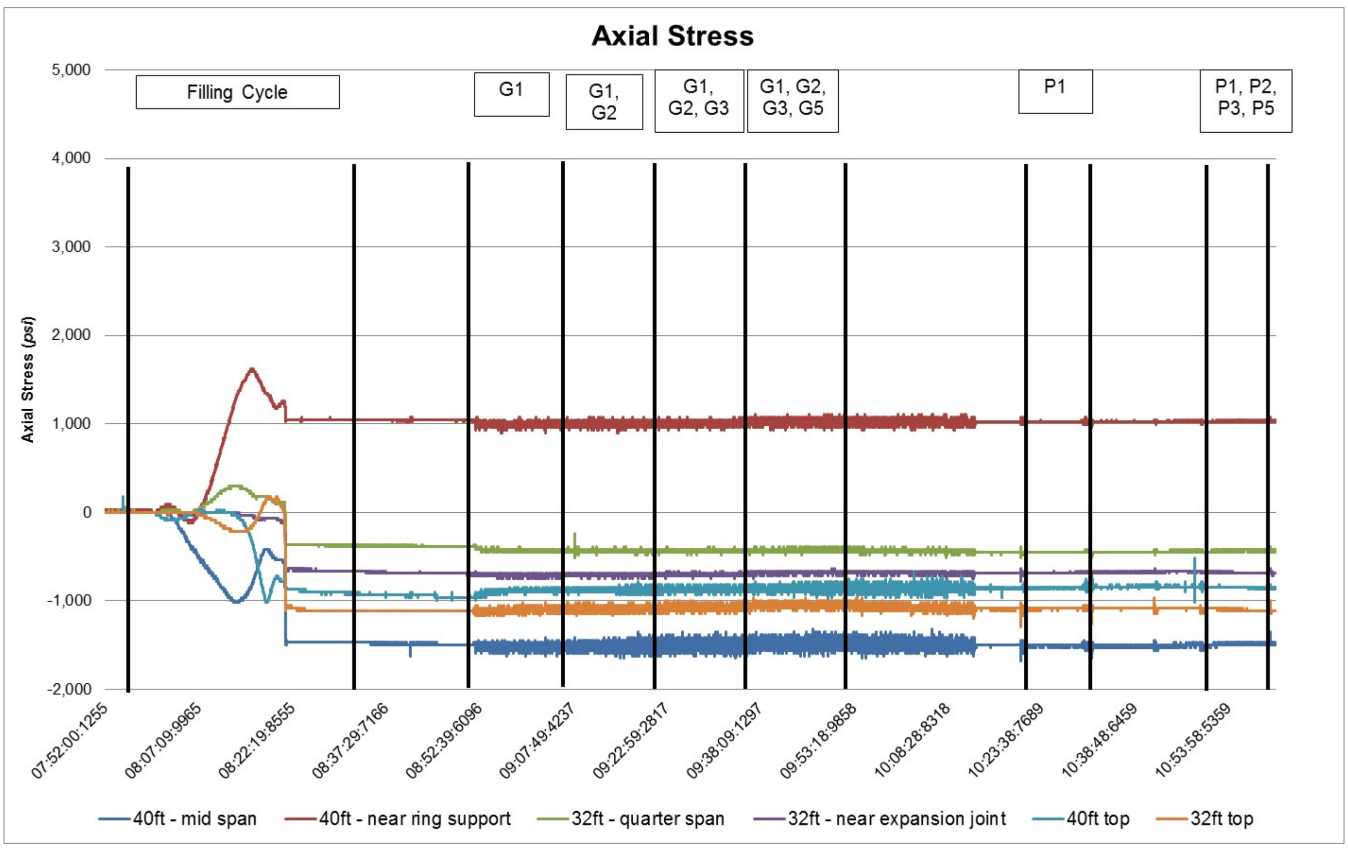

Figure 13. Plot of axial stress measurements for both generating and pumping modes.

Hoop stresses are shown for the test period in Figure 14. The largest stresses were measured at quarter-span of the 32-ft section with a maximum of 3,900 psi and peak-to-peak fluctuating stress of 510 psi. Again, this is well below the endurance limit, producing a safety factor of approximately 7. The theoretical hoop stress is approximately 2,900 psi based on Eq. (3) [5], which compares reasonably to measured hoop stresses given the uncertainty of the actual grade of steel and true penstock thickness and diameter.

$$t = \frac{pd}{2s} \qquad \text{Eq. (3)}$$

Where:

t = pipe wall thickness (0.5-inch)
p = pressure (estimated 23.9 psi from water surface elevations)
d = outside diameter (121-inch)
s = allowable design stress – estimated hoop stress (psi)

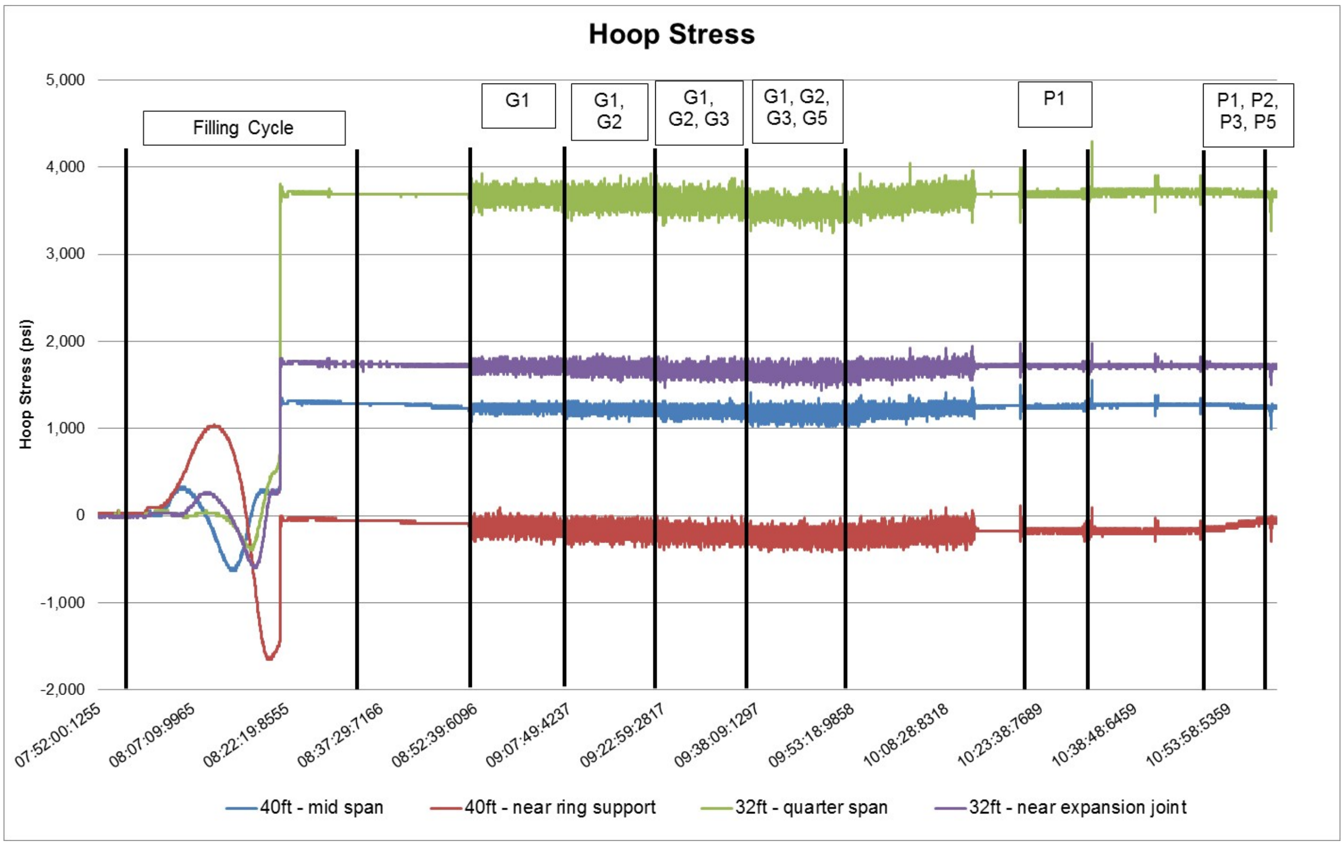

Figure 14. Plot of hoop stresses measured for both generation and pumping modes.

For generating mode there was not much difference in stress fluctuation with various units operating, especially with two or more units. Conversely, there were almost no fluctuations during pump mode operation regardless of the number of units in operation. Peaks shown in pumping mode coincide with bringing another pump online which causes a transient in the pipe. However, these are still well within the acceptable range.

It could not be determined why there is more vibration and fluctuating stresses in generating mode compared to pump mode. However, according to the Designer's Operating Criteria the units were designed to meet a rated efficiency when operating as pumps but no efficiency is warranted when operating as turbines [2]. Flow across turbine blades when operating away from the design or best efficiency point can cause poor hydraulics, a variety of secondary flow patterns, and other hydrodynamic conditions that may produce pressure fluctuations. While this cannot be confirmed from the current test data, it may be the reason for greater vibrations in generating mode.

Conclusions and Recommendations

Strain gage measurements indicated that the maximum stresses were well below the endurance limit of the penstock material suggesting future fatigue failure is not likely an issue. Furthermore, vibrations were prevalent during generating mode only and significantly decreased in pumping mode.

The specific cause of the vibrations could not be identified. However, dominant frequencies coincided with the runner frequencies of the units and were near the estimated natural frequency of the penstock. The new expansion joint may have reduced the stiffness of the penstock and affected the natural frequency resulting in more movement during operation. However, this cannot be confirmed without data or observations prior to expansion joint replacement. It is unlikely that the vibration, as observed, is caused by flow induced vibrations related to the expansion joint or other features of the penstock.

No modifications to the penstock or operational limitations appear necessary at this time as penstock stresses are well within the acceptable range and penstock movement is not considered excessive. However, regular (annual) inspections of the penstock and pertinent features should be a high priority. Special attention should be given to the ring supports, base plate assembly connections, and concrete pedestals for signs of cracking (See Figures 18 through 20 in Appendix A).

References

[1] Bureau of Reclamation, "Senator Wash Dam," [Online]. Available: https://www.usbr.gov/projects/index.php?id=328. [Accessed January 2018].

[2] Bureau of Reclamation, "Designer's Operating Criteria - Senator Wash Dam, Dikes, and Pumping-Generating Plan," Department of the Interior, Denver, 1968.

[3] J. D. Mortensen, "Travel to Senator Wash Pump-Generation Plant to observe penstock vibrations, TR-2017-01," Bureau of Reclamation, Denver, CO, 2017.

[4] J. E. Shigley, C. R. Mischke and R. G. Budynas, Mechanical Engineering Design, New York: McGraw-Hill, 2004.

[5] American Water Works Association, M11 Steel Pipe - A Guide for Design and Installation, Fourth Edition, American Water Works Association, 2004.

[6] J. C. Wachel, S. J. Morton and K. E. Atkins, "Pipe Vibration Analysis," Engineering Dynamics, Incorporated, San Antonio, TX.

[7] Sulzer EMS, "On-site Vibration Report, Imperial Irrigation District, Unit #1 Vibe Issue," 10/3/2017.

[8] K. Smith, "Travel Report, PRJ-20.00, Senator Wash Pumping Plant Site Visit and Penstock Analysis," Bureau of Reclamation, Denver, CO, May 2015.

APPENDIX A: Instrumentation and Penstock Drawings

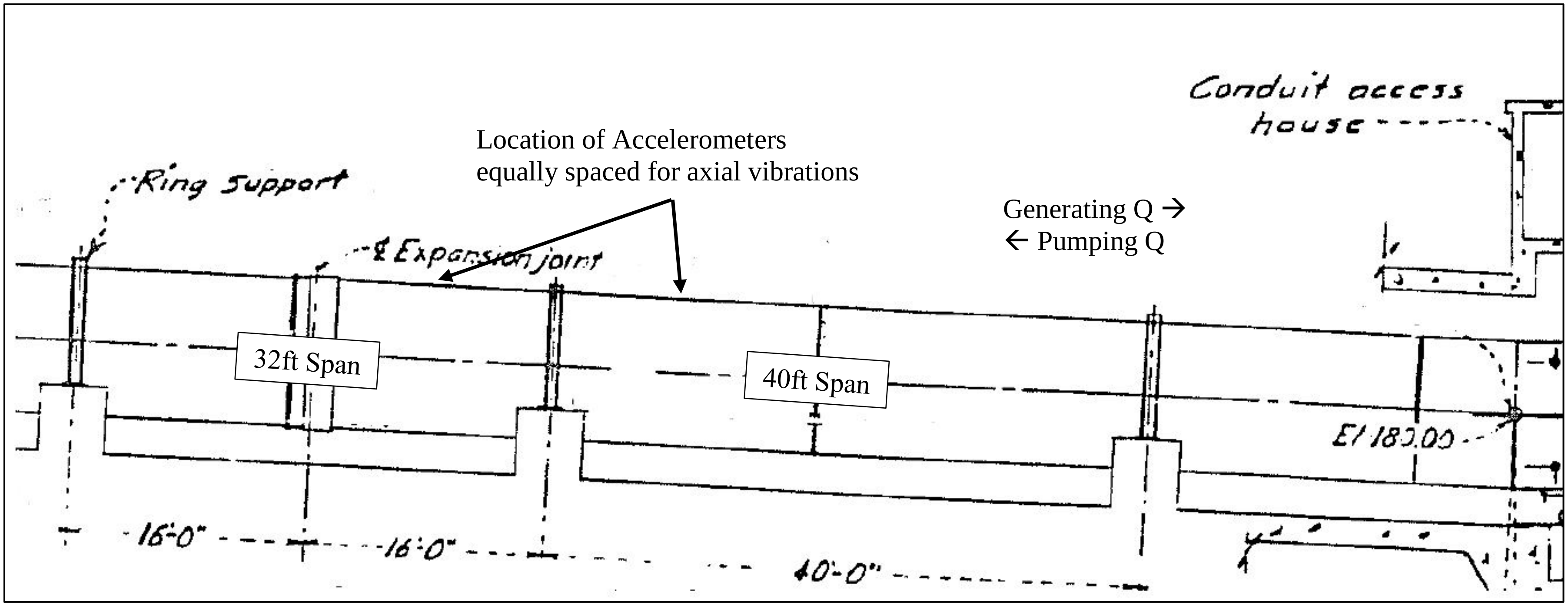

Figure 15. Drawing of Senator Wash Penstock showing 32-ft span (with expansion joint) and 40-ft span where test measurements were acquired.

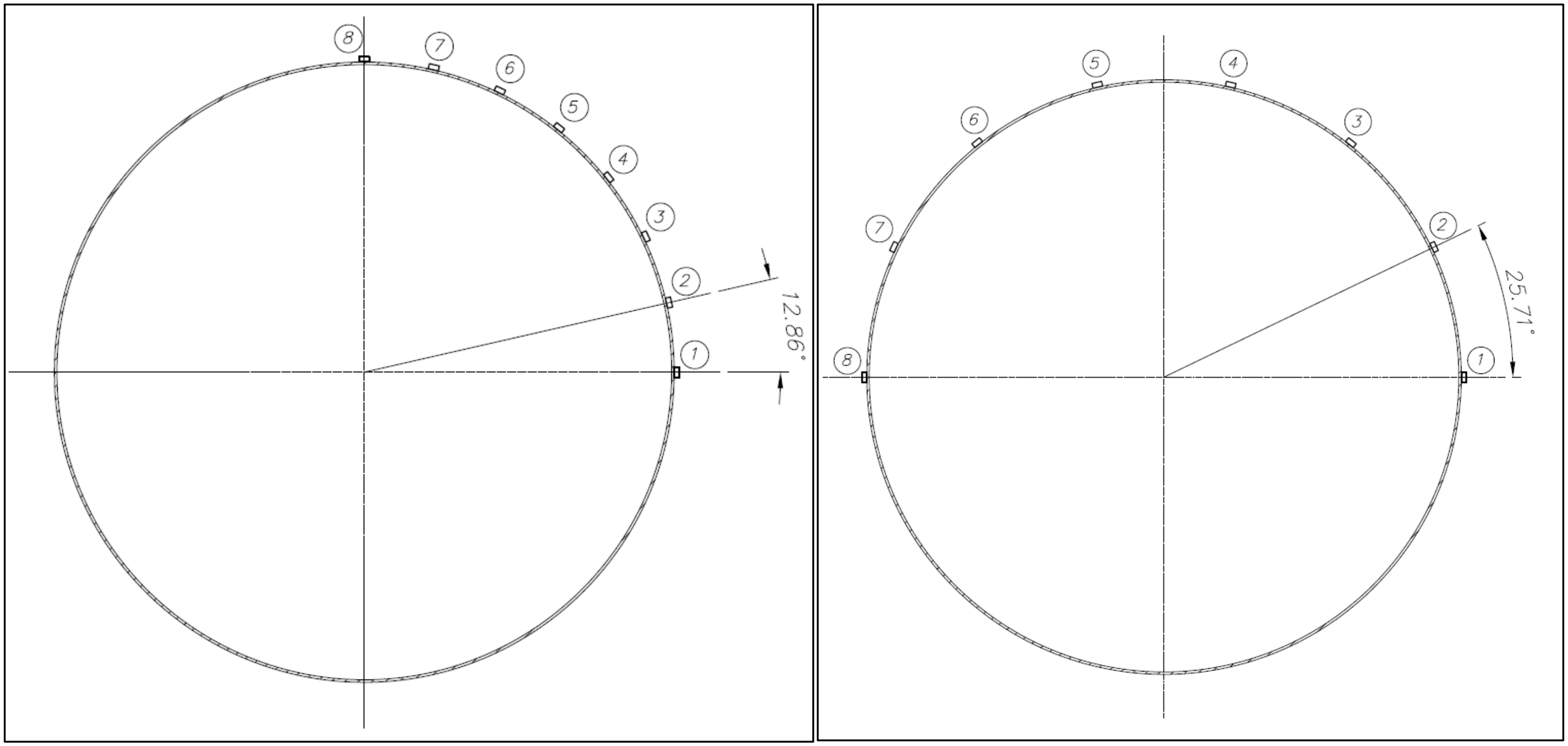

Figure 16. Accelerometer circumferential locations for the 90° (left) and 180° (right) for measuring shell vibrations. View is looking downstream in pumping mode.

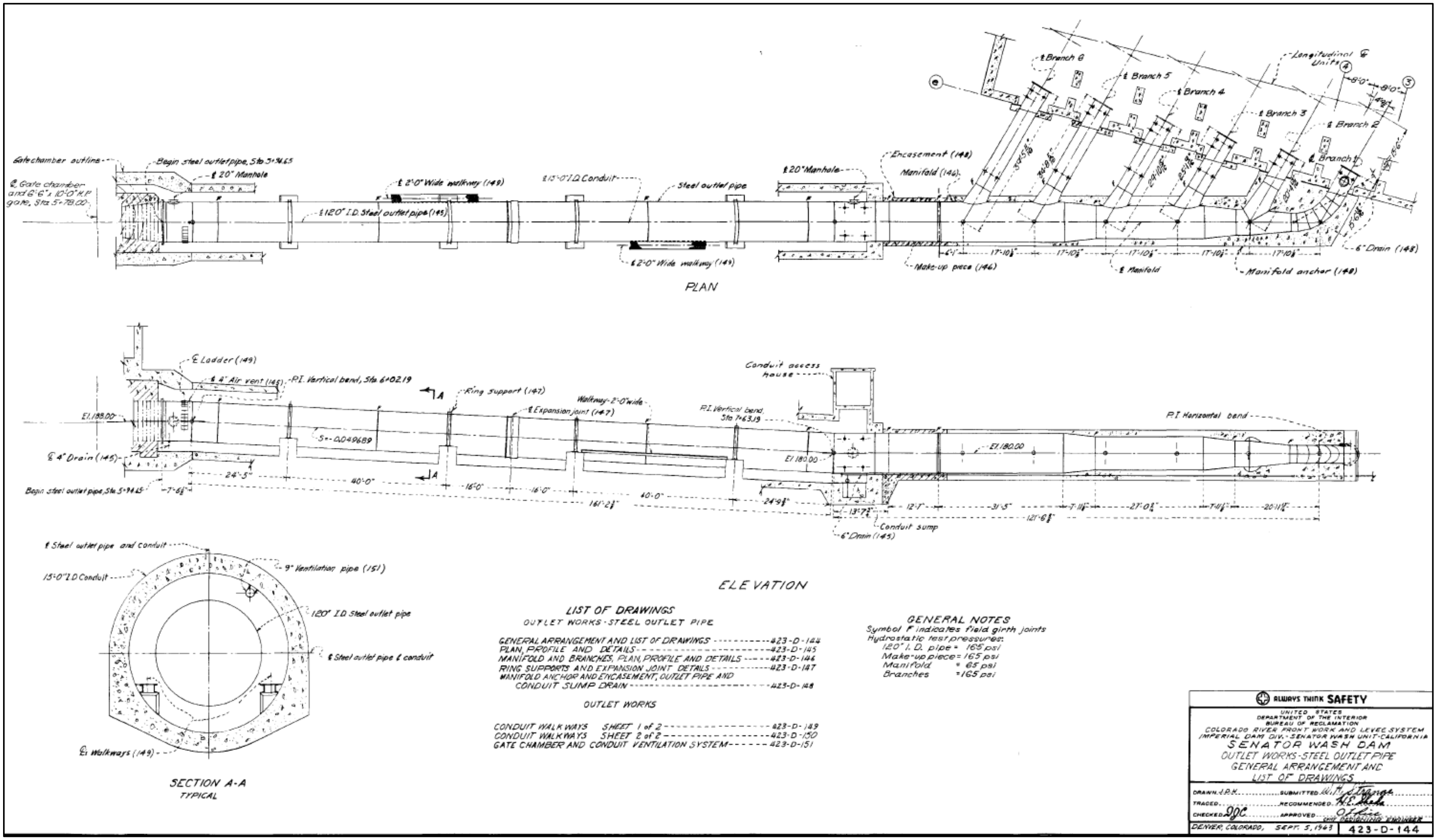

Figure 17. Drawing 423-D-144, showing Plan and Elevation views of Senator Wash Penstock.

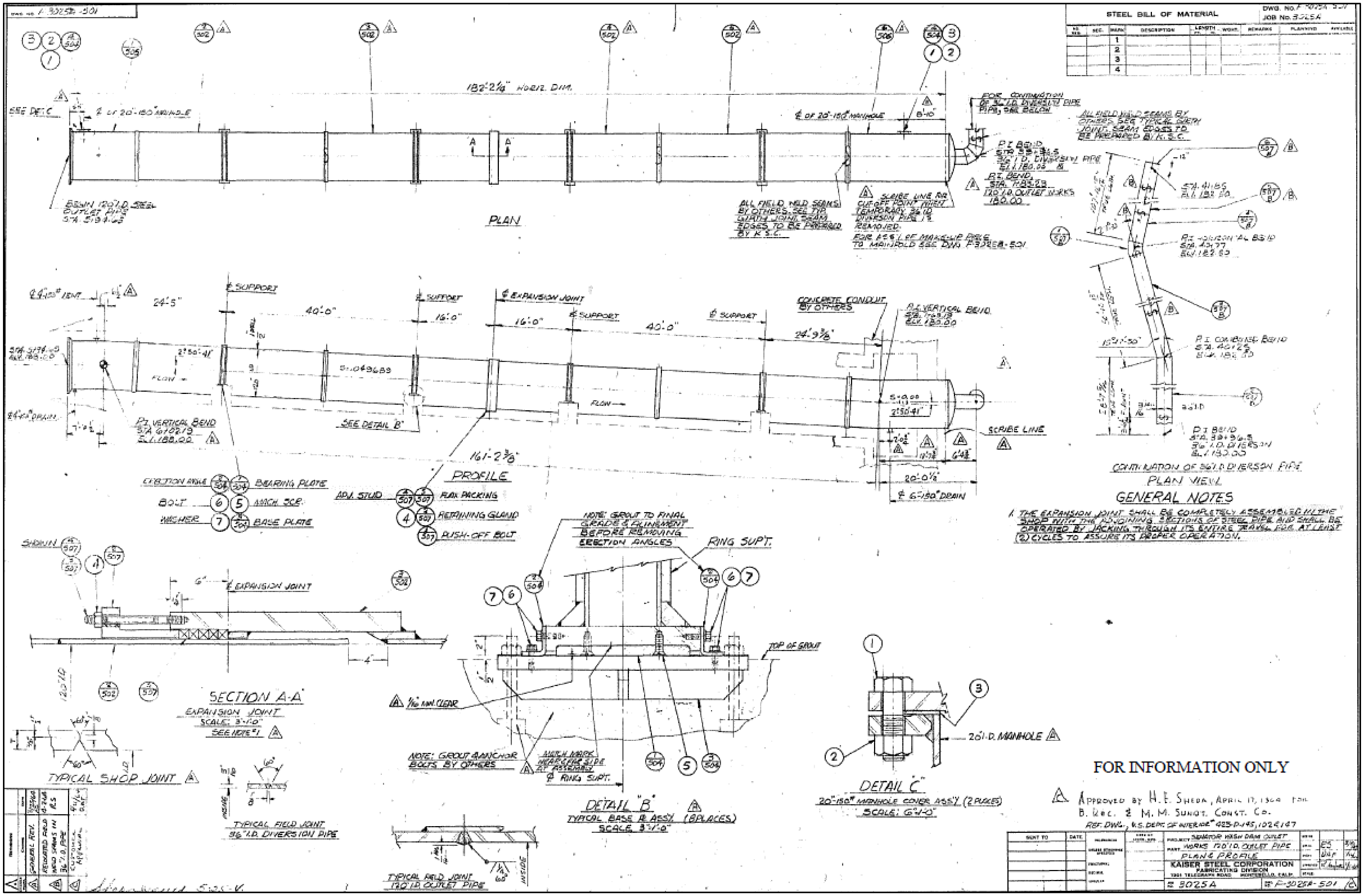

Figure 18. Drawing F-3025A-501, showing Detail B of the base plate assembly for connecting the penstock ring supports to the concrete pedestals.

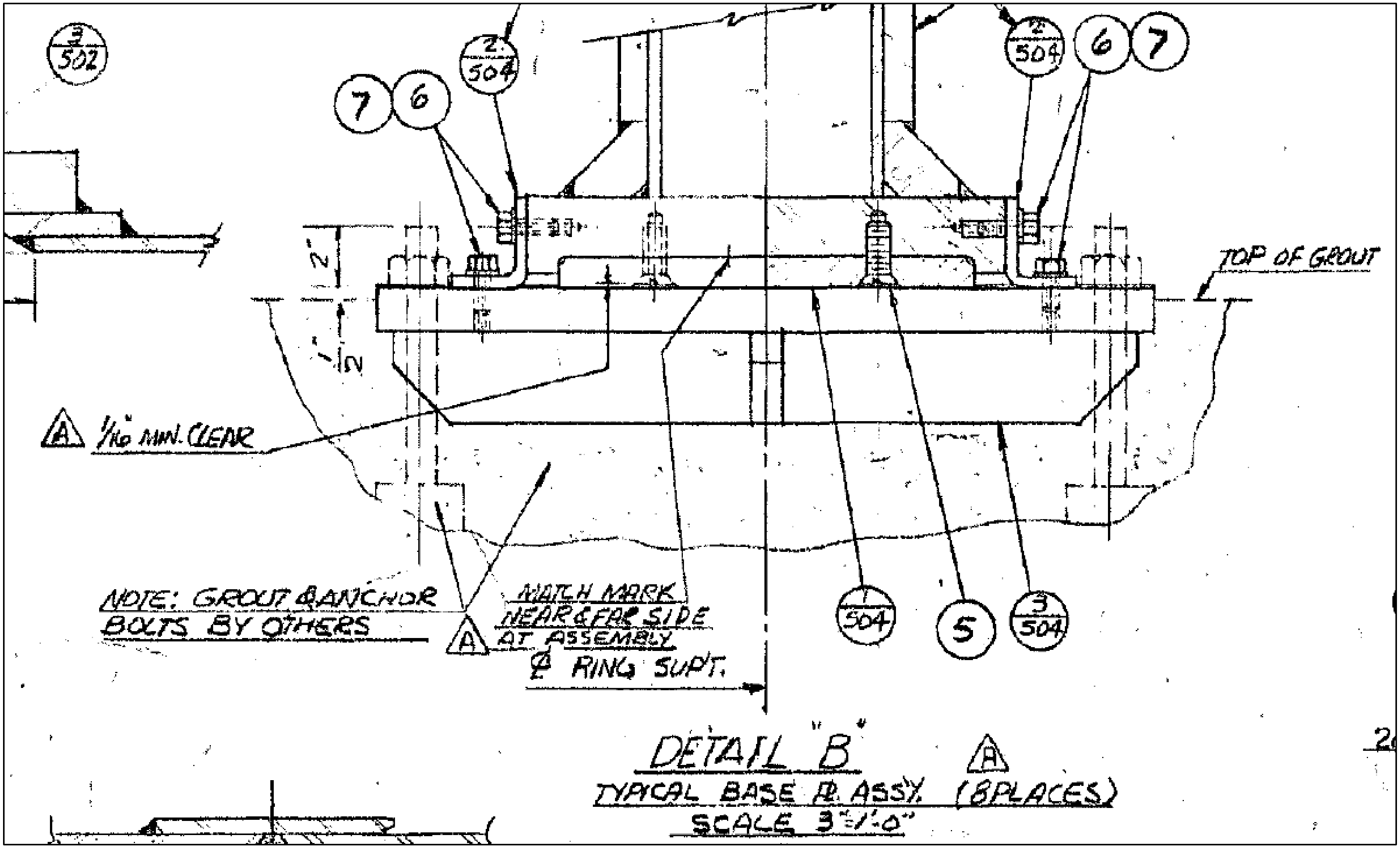

Figure 19. Close-up of Detail B from drawing F-3025A-501, showing the base plate assembly for connecting the penstock ring supports to the concrete pedestals.

Table 2. Strain gage locations and orientations.

Strain gage	**Orientation**	**Axial location**	**Circumferential location** (*deg from horizontal*)
1	Hoop	40-FT, mid span	25
2	Axial	40-FT, mid span	25
3	Axial	40-FT, 40% span	90 (top of pipe)
4	Axial	40-FT, ring support	25
5	Hoop	40-FT, ring support	25
6	Axial	32-FT, quarter span	25
7	Hoop	32-FT, quarter span	25
8	Axial	32-FT, quarter span	90 (top of pipe)
9	Hoop	32-FT, mid span	25
10	Axial	32-FT, mid span	25

APPENDIX B: Operational Vibration Test Data

Table 3. Test log for operational vibration testing on 11/14/2017.

Time	Senator Wash WSE	Sqaw Lake WSE	Discharge	Penstock Span	Accelerometer Arrangement	Operating Units	Notes
MT	*ft*	*ft*	*ft³/s*	*ft*	-	-	-
10:04			0	40	180° of circumference	0	baseline - generating mode
10:27			180	40	180° of circumference	G1	
10:32			180	40	90° of circumference	G1	
10:36			180	40	Axial - top	G1	
10:46			180	32	Axial - top	G1	
10:52			180	32	90° of circumference	G1	
10:55			180	32	90° of circumference	G1	
11:01	236.29		180	32	180° of circumference	G1	
11:12	236.25		360	32	180° of circumference	G1, G2	
11:15			360	32	90° of circumference	G1, G2	
11:20			360	40	90° of circumference	G1, G2	
11:25	236.22		360	40	180° of circumference	G1, G2	
11:35	236.22		538	40	180° of circumference	G1, G2, G5	
11:39			538	40	90° of circumference	G1, G2, G5	
11:46			538	32	90° of circumference	G1, G2, G5	
11:51	236.17		538	32	180° of circumference	G1, G2, G5	
12:18	236.17		lost signal	32	180° of circumference	G1, G2, G3, G5	Discharge signal intermittent, about 720 cfs
12:21			lost signal	32	90° of circumference	G1, G2, G3, G5	
12:25			lost signal	40	90° of circumference	G1, G2, G3, G5	
12:28			lost signal	40	180° of circumference	G1, G2, G3, G5	
12:36			lost signal	40	Axial - 45° from top	G1, G2, G3, G5	
12:47	236.02		lost signal	32	Axial - 45° from top	G1, G2, G3, G5	

Time	Senator Wash WSE	Sqaw Lake WSE	Discharge	Penstock Span	Accelerometer Arrangement	Operating Units	Notes
13:25			lost signal	32	Axial - 45° from top	G1, G2, G3, G5	Ramp down
16:00			0	32	180° of circumference	0	baseline - pumping mode
16:05	235.69	179.49	200	32	180° of circumference	P1	
16:08			200	32	90° of circumference	P1	
16:13			200	40	90° of circumference	P1	
16:16			200	40	180° of circumference	P1	
16:21	235.69	179.46	200	40	180° of circumference	P1	Ramp up from 1 to 4 operating units
16:36			lost signal	40	180° of circumference	P1, P2, P3, P5	
16:38	235.72	179.45	lost signal	40	90° of circumference	P1, P2, P3, P5	
16:46			lost signal	32	90° of circumference	P1, P2, P3, P5	
16:49			lost signal	32	180° of circumference	P1, P2, P3, P5	
16:52	235.76	179.44	lost signal	32	180° of circumference	P1, P2, P3, P5	Ramp down

Table 4. Dominant frequencies for each accelerometer during testing in generating mode.

Generating Mode - Dominant Frequencies from Accelerometers								Penstock Span	Accel. Orientation	Operation
1 [Hz]	*2 [Hz]*	*3 [Hz]*	*4 [Hz]*	*5 [Hz]*	*6 [Hz]*	*7 [Hz]*	*8 [Hz]*	*(ft)*	-	-
6.2	6.2	6.2	6.2	6.2	6.2	6.2	10.9	32	90° of circumference	G1
6.2	6.2	6.2	10.9	6.2	6.2	6.2	6.2	32	180° of circumference	G1
10.9	10.9	10.9	10.9	10.9	42.0	42.0	8.5	32	Axial - top	G1
12.0	12.0	6.2	9.5	9.5	11.7	11.7	9.5	40	90° of circumference	G1
12.0	6.2	6.2	11.7	9.5	6.1	9.5	11.7	40	180° of circumference	G1
8.5	8.5	8.5	8.5	8.5	8.5	8.5	8.5	40	Axial - top	G1
6.3	6.3	6.0	6.3	6.3	6.3	6.3	11.0	32	90° of circumference	G1, G2
5.8	5.8	5.8	10.9	6.2	6.2	5.8	5.8	32	180° of circumference	G1, G2
14.0	14.2	14.2	14.2	6.3	14.1	14.1	14.0	40	90° of circumference	G1, G2
14.1	14.1	14.1	14.1	14.1	14.1	14.1	14.1	40	180° of circumference	G1, G2
84.0	6.0	6.0	84.0	6.0	84.0	42.0	84.0	32	90° of circumference	G1, G2, G5
84.0	6.0	6.0	42.0	84.0	6.0	6.0	42.0	32	180° of circumference	G1, G2, G5
12.0	21.8	21.7	6.1	21.7	21.7	21.8	21.7	40	90° of circumference	G1, G2, G5
42.0	21.7	21.7	21.7	21.7	21.7	6.0	84.0	40	180° of circumference	G1, G2, G5
84.0	23.3	84.0	84.0	6.1	84.0	42.0	84.0	32	90° of circumference	G1, G2, G3, G5
84.0	6.0	6.0	42.0	42.0	6.0	6.0	84.0	32	180° of circumference	G1, G2, G3, G5
14.0	21.7	14.0	14.0	21.7	14.0	14.0	14.0	40	90° of circumference	G1, G2, G3, G5
126.0	14.1	78.0	14.1	14.1	21.6	14.1	126.0	40	180° of circumference	G1, G2, G3, G5
21.7	21.7	21.7	21.7	28.3	28.3	78.0	6.0	40	Axial - 45° from top	G1, G2, G3, G5

Table 5. Dominant frequencies for each accelerometer during testing in pumping mode.

Pumping Mode - Dominant Frequencies from Accelerometers								**Penstock Span**	**Accel. Orientation**	**Operation**
1 [Hz]	*2 [Hz]*	*3 [Hz]*	*4 [Hz]*	*5 [Hz]*	*6 [Hz]*	*7 [Hz]*	*8 [Hz]*	*(ft)*	-	-
18.0	18.0	18.0	6.0	6.0	6.0	18.0	18.0	32	90° of circumference	P1
18.0	6.0	6.0	18.0	18.0	6.0	18.0	18.0	32	180° of circumference	P1
18.0	18.0	18.0	6.0	6.0	6.0	8.3	18.0	40	90° of circumference	P1
18.0	6.0	6.0	8.5	8.5	6.0	6.0	8.5	40	180° of circumference	P1
84.0	84.0	84.0	84.0	6.0	84.0	6.0	84.0	32	90° of circumference	P1, P2, P3, P5
84.0	84.0	6.0	6.0	84.0	84.0	84.0	84.0	32	180° of circumference	P1, P2, P3, P5
84.0	6.0	84.0	42.0	6.0	84.0	84.0	84.0	40	90° of circumference	P1, P2, P3, P5
84.0	42.0	6.0	84.0	42.0	42.0	42.0	84.0	40	180° of circumference	P1, P2, P3, P5

Table 6. Generating mode pipe wall displacements.

Penstock span	Accel orientation	Operation	Avg p-p	Max p-p
(ft)	-	-	*[inch]*	*[inch]*
32	90° of circumference	G1	0.066	0.087
32	180° of circumference	G1	0.051	0.064
32	Axial - top	G1	0.036	0.076
40	90° of circumference	G1	0.104	0.151
40	180° of circumference	G1	0.102	0.143
40	Axial - top	G1	0.059	0.100
			0.070	**0.151**
32	90° of circumference	G1, G2	0.065	0.083
32	180° of circumference	G1, G2	0.060	0.066
40	90° of circumference	G1, G2	0.083	0.104
40	180° of circumference	G1, G2	0.114	0.163
			0.080	**0.163**
32	90° of circumference	G1, G2, G5	0.107	0.132
32	180° of circumference	G1, G2, G5	0.137	0.172
40	90° of circumference	G1, G2, G5	0.189	0.237
40	180° of circumference	G1, G2, G5	0.196	0.278
			0.157	**0.278**
32	90° of circumference	G1, G2, G3, G5	0.165	0.229
32	180° of circumference	G1, G2, G3, G5	0.135	0.148
32	Axial - 45° from top	G1, G2, G3, G5	0.109	0.155
40	90° of circumference	G1, G2, G3, G5	0.206	0.255
40	180° of circumference	G1, G2, G3, G5	0.156	0.186
40	Axial - 45° from top	G1, G2, G3, G5	0.128	0.191
			0.150	**0.255**

Table 7. Pumping mode pipe wall displacements.

PENSTOCK SPAN	Accel Orientation	Operation	Avg P-P	Max P-P
(ft)	-	-	*[inch]*	*[inch]*
32	90° of circumference	P1	0.032	0.042
32	180° of circumference	P1	0.028	0.035
40	90° of circumference	P1	0.059	0.083
40	180° of circumference	P1	0.054	0.080
			0.043	**0.083**
32	90° of circumference	P1, P2, P3, P5	0.042	0.068
32	180° of circumference	P1, P2, P3, P5	0.032	0.043
40	90° of circumference	P1, P2, P3, P5	0.049	0.070
40	180° of circumference	P1, P2, P3, P5	0.045	0.065
			0.042	**0.070**

APPENDIX C: Operational Stress Test Data

Table 8. Test log for operational penstock stress testing on 11/16/2017.

Time	Senator Wash WSE	Squaw Lake WSE	Discharge	Operating Units	Notes
MT	*ft*	*ft*	*ft³/s*	-	-
8:00	235.34	179.95	0	0	Start of filling, baseline for zero strain
8:52	235.32	179.90	140	G1	
9:07	235.3	179.89	282	G1, G2	
9:24	235.28	179.90	420	G1, G2, G5	
9:36	235.27	179.91	557	G1, G2, G3, G5	
10:15	235.17	179.92	0	0	
10:20	235.14	179.93	165	P1	
10:50	235.18	179.86	lost signal	P1, P2, P3, P5	

Table 9. Mean and maximum hoop stresses and fluctuations measured at the location of maximum displacement on the 32-ft section.

Worst case hoop stress - 32ft max deflection				
Operation	**Time sample**	**Mean**	**Max**	**Peak to peak**
		psi	*psi*	*psi*
Filled - No flow	8:39 - 8:46	3,690	3,690	30
G1	8:57 - 9:04	3,677	3,870	360
G1, G2	9:12 - 9:19	3,645	3,900	480
G1, G2, G3	9:28 - 9:35	3,605	3,870	510
G1, G2, G3, G5	9:41 - 9:48	3,540	3,750	480
P1	10:21 - 10:28	3,696	3,750	90
P1, P2, P3, P5	10:52 - 10:59	3,703	3,750	90