

Vibration Control using Vibration Dampers at Pipeline Stations

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Abstract

Liquid pipeline stations experience a variety of vibration issues due to the characteristics of large centrifugal pumps driven by variable speed drive (VFD) motors that are common selections for pumping hardware at the stations. The speed on the pumps continuously varies to support pipeline demand, which results in the vane pass frequency of the pump eventually coinciding with pipe natural frequencies at some speed. When this occurs, resonance on the piping will result in very high vibration and associated fatigue failures.

Various options are available to minimize the risk of failure on the small-bore branch lines including eliminating as many branch lines as possible, adding additional braces or stiffeners when possible (self-reinforcing back to the main pipe), and adding damping to reduce the resonant amplification/response.

This paper provides a review of the benefits of various options for reducing fatigue risk on small bore piping like those present in pipeline stations.

Pipeline Stations

The United States has more than 2.8 million miles of energy pipelines, including natural gas distribution mains, transmission lines, and hazardous liquid pathways. Many of these are in urban areas or passing through farms and neighborhoods, and either through or near sensitive environmental areas.

Figure 1 details the routes for hazardous liquid pipelines located in the USA along with a 2nd plot showing the locations of “accidents” on liquid pipelines. As noticed, there have been a large number of incidents of various criticality along most pipelines shown as detailed on <https://pvnpm.phmsa.dot.gov/PublicViewer/>.

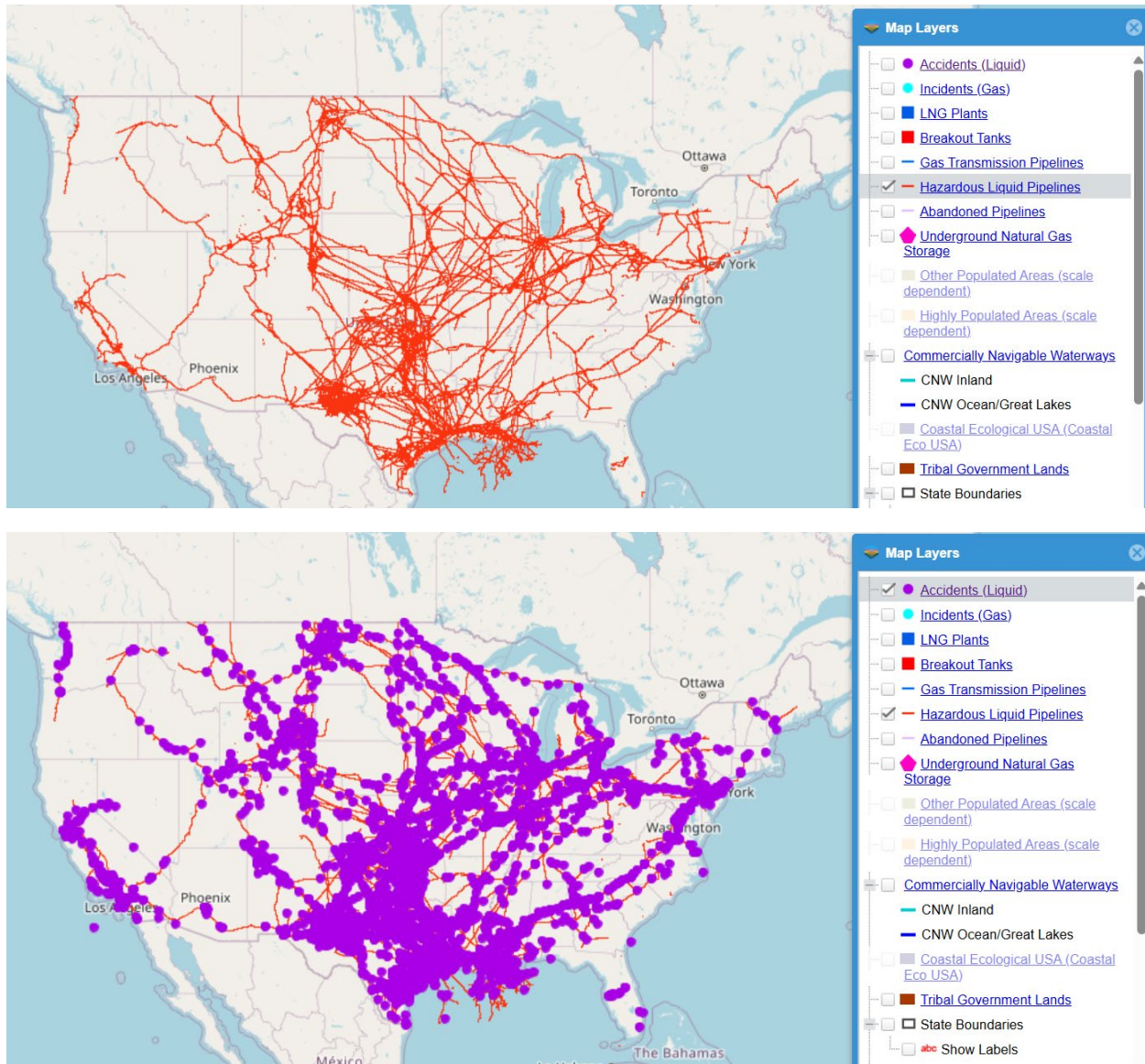


Figure 1 - Liquid Pipeline Locations and Incidents

It is obvious from Figure 1 that incidents on pipelines are common. Although I never knew it, there was an incident in 2017 that was 1000' from my house in which 10 barrels of hazardous hydrocarbon liquid leaked out due to “incorrect operation” as shown in Figure 2.

Some of the incidents noted are due to fatigue failures of piping where it has been suggested that vibration and fatigue can account for up to 20% of hydrocarbon releases, and a large portion of these are due to failure of small-bore connections. (1)

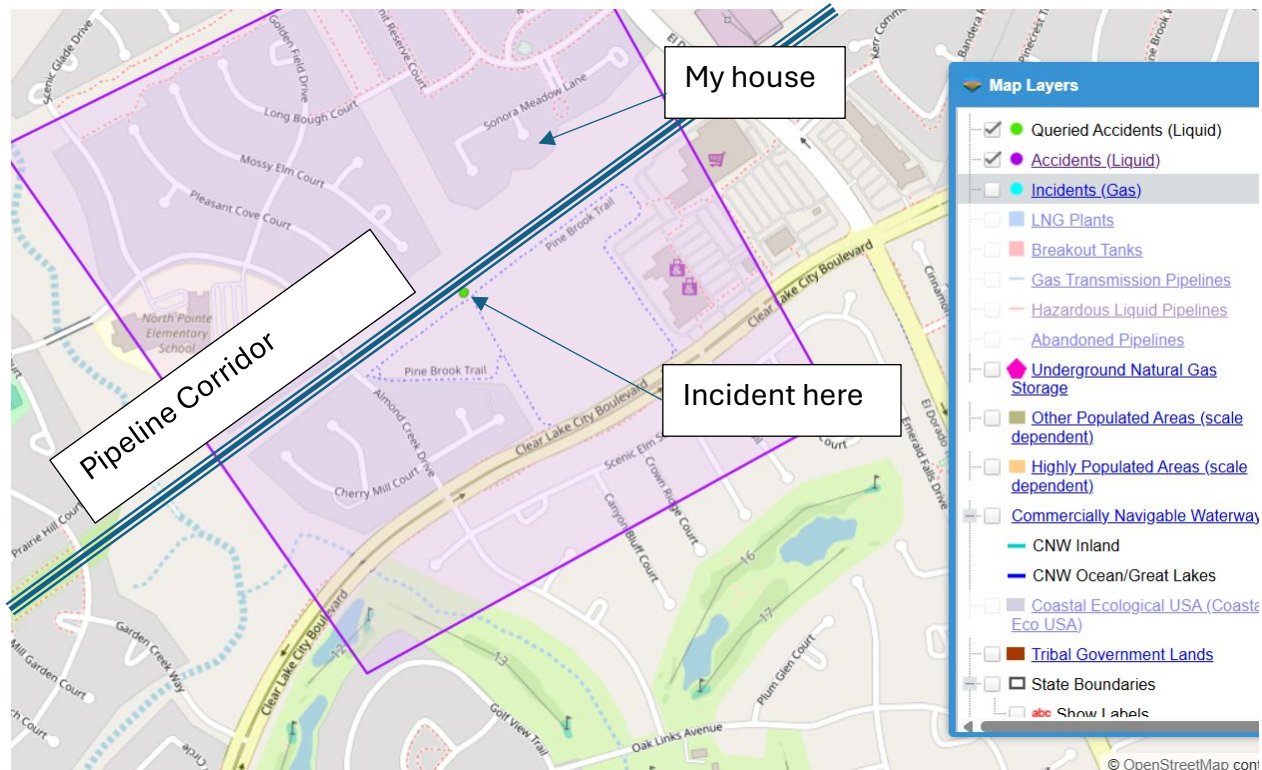


Figure 2 - "Incident" of 10 Barrel Release of Hazardous Liquid

Since hydrocarbon pipelines are pumping liquids for long distances, it is necessary to have pipeline pump stations about every 50-80 miles apart. Each of these pump stations will have multiple centrifugal pumps that are configured in various ways including in series or parallel as shown in figure 3. Both series and parallel operation provide operating challenges for the pumps for different reasons.

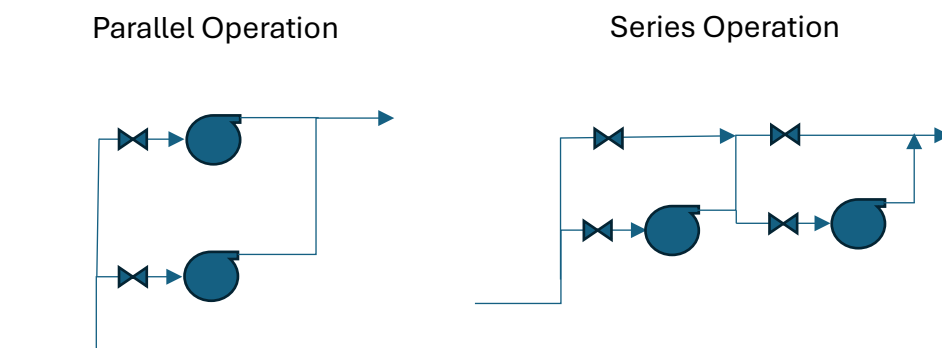


Figure 3 - Parallel and Series Operation

Pumps that are arranged in parallel have the advantage of a good layout for identical spare pumps. Normally one pump is operated while the other is a spare. If both pumps are operated at the same time, the combined flow rate would increase as would the differential pressure at the pumps. One disadvantage of parallel pumps is that there is often a

dominant pump with higher differential pressure (i.e. higher speed or better pump performance) that can overtake the 2nd parallel pump so that the flow on the 2nd pump is relatively low. High or low flow on a specific pump can cause the pumps to either cavitate or recirculate, respectively, causing higher general vibration near the pumps as well as poor reliability.

For the series operation, any pump in the series (or multiple pumps) can be operated while others can be bypassed around so that all the flow passes through all the in-service pumps. This arrangement prevents one pump overwhelming another and provides the ability to vary the differential pressure for the pump station while continuing to operate within the design flow rate for a single pump.

For either pump arrangement, the main vibration issue impacting the piping is due to vane pass frequency vibration that will generally increase when not operating near the BEP for the pump. A typical pump performance curve is shown in figure 4 with differential head vs flow rate.

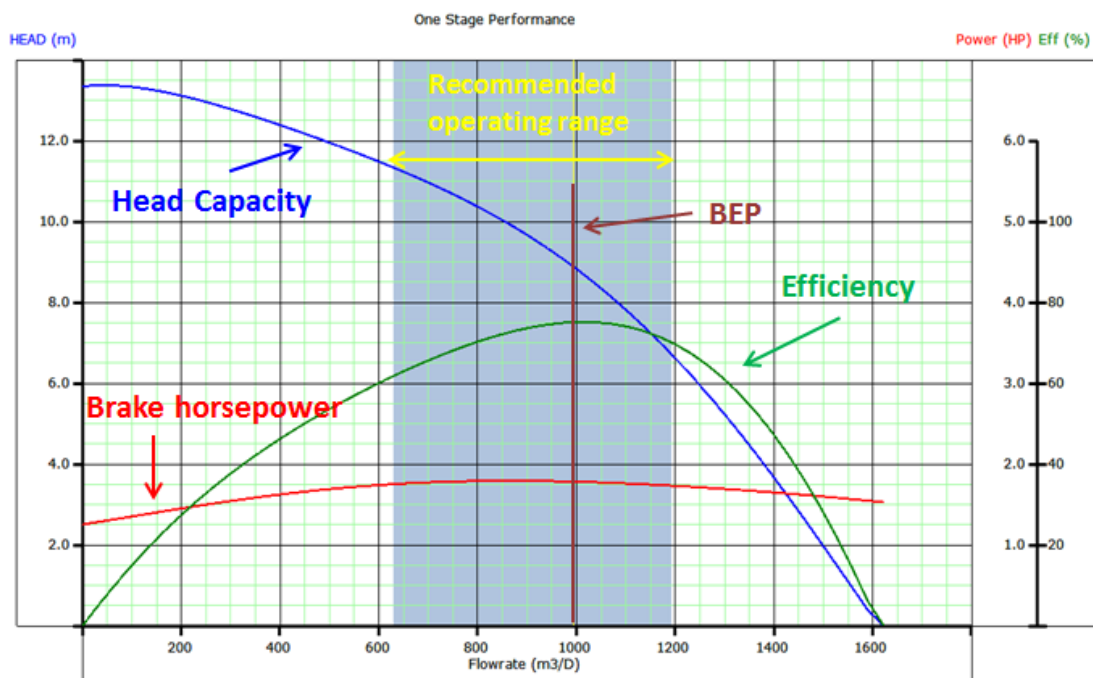


Figure 4 - Typical Pump Performance Curve

As the pump flow rate deviates away from the BEP (best efficiency point), vibration will generally increase as detailed in figure 5. At low flow rates, the issues driving higher vibration will be internal flow recirculation with increasing vane pass frequency vibration. At higher flow rates, the vibration is driven by pump cavitation as well as higher blade pass frequency vibration. For either higher or lower flow, the flow angles in the impeller become

less aligned with the impeller blades, causing local cavitation and inefficiency in pumping. Once the flow angles start to be misaligned with impeller blades, vane pass frequency will increase.

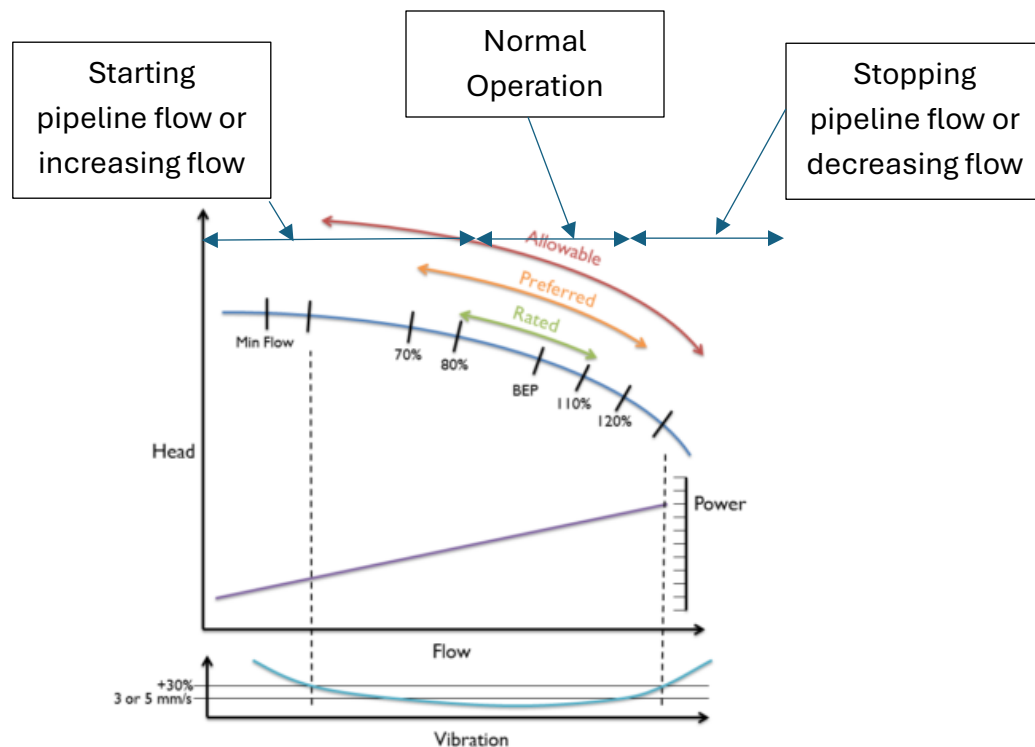


Figure 5 - Vibration vs. Flow Rate for Centrifugal Pump

Although vane pass frequency vibration can be reduced with a variety of pump modifications such as reducing impeller diameter, increasing A/B gaps and other adjustments, vane pass frequency vibration for operation well away from BEP will persist for all pumps.

Pipeline Operation

Since pipelines involve pushing liquid down a pipe for many miles, accelerating and decelerating the flow requires time to get the flow moving. When a pipeline is started, pump stations are turned on from the inlet end and progressively turned on as the line “packs” and pressure increases.

If there is no flow and the first pump is turned on, the pump will operate well to the left of BEP until the line begins to “pack” and the pump operating flow shifts towards the BEP. Each successive pump station will have the same challenge.

When the pipeline is stopped, there would be a similar but opposite procedure to allow the pipeline flow to gradually decrease as the line pressure drops. When this occurs, the pumps will operate well to the right of BEP, again causing high blade pass frequency vibration.

The result is that there are many operating conditions where blade pass frequency will be higher than at other times, with some conditions producing very high vibration.

To complicate all this, it is common practice to use variable frequency drives (VFD's) on the motors so that the pump speed can be varied from about 50-100% of rated speed to better control flow rate and pump mismatch as the flow on the pipeline is either increased or decreased. Since the desire would be to always operate near BEP to achieve the lowest vibration on the pumps (and improved reliability), the easiest way is to vary the pump speed continuously to match the pump better to the service.

The side effect of the continuous speed variation to control pump performance is that the vane pass frequency will continuously vary and WILL at some point excite a resonance of some piping components or sections.

The vane pass frequency vibration range tends to be in the range of vent/drain natural frequencies for many pipeline designs. As a result, many vent/drain lines are at high risk of fatigue failure at many pipeline stations.

Vent/Drain Vibration Risk

Operational requirements dictate that many vents and drains are provided near the pumps for product clearing and purging during or after maintenance activities. These vents/drains are commonly configured as cantilevered branches using smaller pipe (3/4" to 2" nominal pipe size) with a variety of details that are used to connect a valve to either vent gas from the line (top of main pipe) or drain liquid to clear the pump (bottom of main pipe). The outlet of the valve is frequently closed using a pipe plug (threaded) or a blind flange.

These vent/drain attachments will have natural frequencies that are often in the range of 80-300 Hz. The pumps either operate at 1800 RPM or 3600 RPM maximum speed and have a turndown capability of about 50% using a VFD on the motor. For a 3600 RPM pump with 5 impeller vanes, the vane pass frequency vibration would vary continuously from 150 Hz up to 300 Hz. Any small-bore pipe connected to the pump or the main pipe with a natural frequency in the 150 – 300 Hz range will resonate when the vane pass frequency equals the natural frequency.

The vibration of small-bore pipes will be excited either by acoustic pulsations in the main pipe or by transmitted vibration from the main pipe. Pipelines will have various main pipeline sizes with many crude oil and NGL lines operating with 16" to 30" pipes. Since the centrifugal pumps generate vane pass frequency pulsation and vibration that is transmitted to the main piping, the dominant source of vibration on the small-bore pipe is transmitted vibration from the main pipe.

Some specific issues can result in very high main pipeline vibration near pumps including:

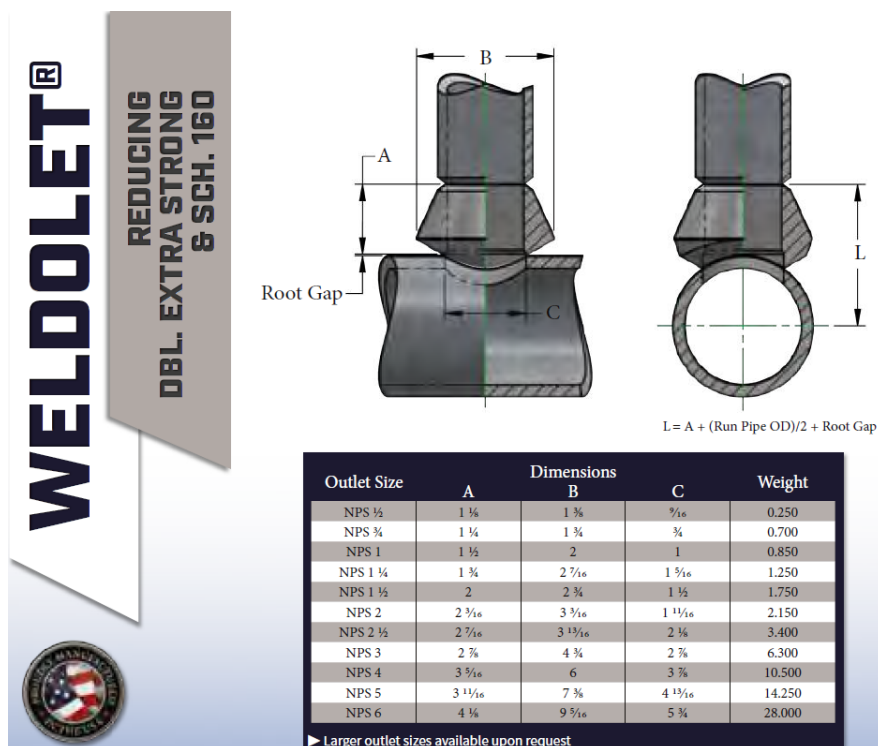
- Acoustic resonance internal to the pump (i.e. pump crossover length equals $\frac{1}{2}$ wave length for the pumped product)
- Acoustic resonance/reactions in the piping at elbows, filters, reducers, etc.
- Shell wall vibration on the main pipe

One or more of these contributors will frequently exist for current pipeline station designs.

Vent Design Options

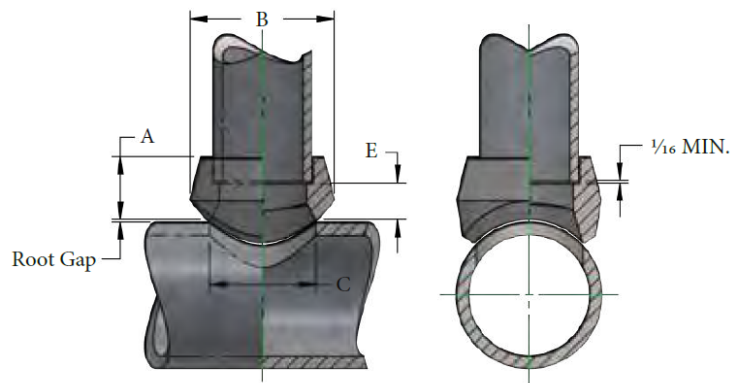
The vents are normally fabricated using some forged fitting that is attached to the main pipe such as a WeldOlet or SockOlet. Other forged fittings are available as well. One popular supplier for the forged branch fittings is Bonney Forge.

Typical Olet fittings are shown below:



SOCKOLET®

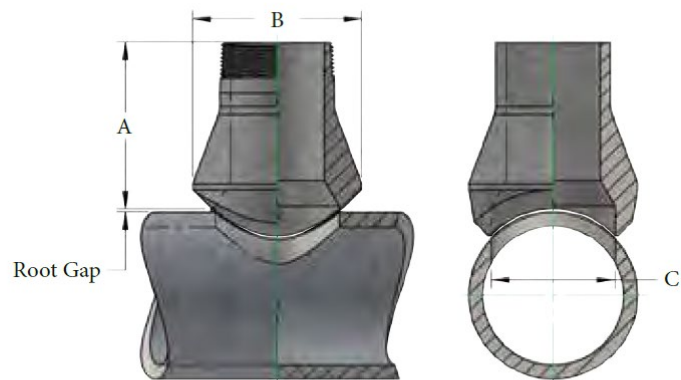
REDUCING
CLASS 3000, 6000
& 9000



	Outlet Size	Dimensions				Weight
		A	B	C	E	
CLASS 3000	NPS 1/4	3/4	1	3/4	7/16	0.100
	NPS 3/8	3/4	1	3/4	7/16	0.100
	NPS 1/2	1 1/16	1 1/4	3/4	7/16	0.200
	NPS 3/4	1 1/16	1 3/4	1 1/16	7/16	0.300
	NPS 1	1 1/16	2 1/4	1 7/16	1 1/16	0.600
	NPS 1 1/4	1 1/16	2 7/16	1 3/4	3/4	0.850
	NPS 1 1/2	1 3/4	2 7/8	2	3/4	1.000
	NPS 2	1 1/2	3 1/2	2 1/16	1 1/16	1.600
	NPS 2 1/2	1 13/16	4 1/16	3	3/4	2.750
	NPS 3	2	4 13/16	3 11/16	1 1/16	3.800
	NPS 3 1/2	2 1/4	5 3/8	4	1	4.300
	NPS 4	2 3/4	6	4 3/4	1 1/16	7.250
	NPS 5	2 3/4	7	5 1/16	1 3/8	12.000
	NPS 6	2 3/4	8 3/8	6 11/16	1 7/16	14.500

NIPOLET®

THREADED & SOCKET-WELD
EXTRA STRONG &
DBL. EXTRA STRONG



	Outlet Size	Dimensions		
		A	B	C
XS	NPS 1/2	SEE NOTES	1 3/8	1 1/16
	NPS 3/4		1 3/4	1 3/16
	NPS 1		2 1/8	1 7/16
	NPS 1 1/4		2 1/16	1 3/4
	NPS 1 1/2		2 3/8	2
	NPS 2		3 1/2	2 1/16
XXS	NPS 1/2	SEE NOTES	1 3/8	7/16
	NPS 3/4		1 3/4	3/4
	NPS 1		2	1
	NPS 1 1/4		2 7/16	1 3/16
	NPS 1 1/2		2 3/4	1 1/2
	NPS 2		3 1/4	1 11/16

- ▶ Available in threaded or plain end (male socket-weld)
- ▶ Standard lengths: 3 1/2", 4 1/2", 5 1/2", and 6 1/2"
- ▶ Non standard lengths available upon request

THREDOLET[®] REDUCING CLASS 3000 & 6000					
Outlet Size	A	Dimensions B	C	Weight	
NPS ¼	¾	1	¾	0.100	
NPS ½	¾	1	¾	0.100	
NPS ¾	1 ⅜	1 ¼	¾	0.200	
NPS 1	1	1 ⅞	1 ⅝	0.250	
NPS 1 ¼	1 ⅞	1 ¾	1 ⅝	0.350	
NPS 1 ½	1 ⅞	2 ¼	1 ⅞	0.600	
NPS 2	1 ⅞	2 ⅞	1 ¾	0.900	
NPS 2 ½	1 ¾	2 ⅞	2	1.000	
NPS 3	1 ½	3 ½	2 ⅞	1.750	
NPS 3 ½	1 ⅞	4 ⅞	3	3.000	
NPS 4	2	4 ⅞	3 ⅞	4.350	
NPS 5	2 ¼	5 ¼	4	5.750	
NPS 6	2 ¼	6	4 ¾	7.100	
NPS 8	2 ¾	7 ⅞	5 ⅞	12.000	
NPS 10	2 ¾	8 ¾	6 ⅞	15.300	

Once the fitting is selected for connection to the main pipe, a short piece of pipe is used to connect from the fitting to the valve. The details for the pipe extension (when used) will depend on whether the layout is using a valve that is flanged, socket welded, or threaded to connect to the main pipe.

The selected valve has a HUGE impact on fatigue risk since many of the selected valves are very heavy, particularly for higher pressure service. In general, the “best” option is to use as light a valve as possible.

Using ASME OM-2009 as a reference (2), a very heavy valve can reduce the allowable vibration on a branch line by a factor of 10. If the concentrated weight of valves and flanges is 25 times the weight of the pipe alone, the allowable vibration is 10% of what it would be without the added weight.

Since most pipelines have rather high rating pressures (1500 psig or higher), it is common to have flanges ratings of 600# and higher for flanges and valves. Using flanged valves will dramatically increase the weight of the vent.

One common layout is to use a weld in valve for a vent with a Socket Weld connection at the bottom of the valve with a threaded connection on the top of the valve. A typical selection for 800# gate valves is shown below:

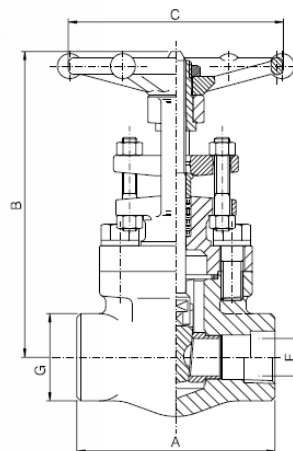
B[®] GATE VALVES- BOLTED BONNET- FULL & STANDARD PORT

BONNEY FORGE

800 LB.

1500 LB.

Design construction:
 API 602 - ASME B16.34 - BS 5352
 Testing according to API 598
 Marking MSS SP25
 Outside Screw and Yoke (OS&Y)
 Self aligning two piece packing gland
 Spiral-wound gasket
 Integral backseat
 Socket Weld Ends to ASME B16.11
 Screwed Ends (NPT) to ASME B1.20.1
 Butt Welding Ends to ASME B16.25
Ratings:
 - carbon steel class 800 1975 psig @ 100°F
 138 bar + 38°C
 - carbon steel class 1500 3705 psig @ 100°F
 255 bar + 38°C



FULL PORT - FIG. H 10											
SIZE	inch	1/4	3/8	1/2	3/4	1	1 1/4	1 1/2	2	2 1/2	3
	mm	6	10	15	20	25	32	40	50	63	75
A	inch	3.15	3.15	3.54	4.33	5	5	5.28	6	6	6
	mm	80	80	90	110	127	127	134	152	152	152
B open	inch	5.98	5.98	6.22	7.72	8.86	10.04	11.42	14.09	14.09	14.09
	mm	152	152	158	196	225	255	290	358	358	358
C	inch	3.46	3.46	3.46	3.82	5.43	5.43	5.43	6.77	6.77	6.77
	mm	88	88	88	97	138	138	138	172	172	172
F	inch	.31	.38	.55	.71	.94	1.18	1.48	1.83	1.83	1.83
	mm	8	9.6	14	18	24	30	36.5	46.5	46.5	46.5
G	inch	1.26	1.26	1.50	1.89	2.20	2.52	3.07	3.35	3.35	3.35
	mm	32	32	38	48	56	64	78	85	85	85
Weight	lb.	4.5	4.5	5	8.25	13	16.25	18.3	27.5	27.5	27.5
	kg	2.1	2.1	2.3	3.7	5.9	7.4	8.3	12.5	12.5	12.5
PACKING		BH2	BH2	BH2	BH4	BH5	BH6	BY5	BH8	BH8	BH8
GASKET		G2	G2	G2	G3	G4	G6	G11	G10	G10	G10

STANDARD PORT - FIG. H 10											
SIZE	inch	1/2	3/4	1	1 1/2	2	2 1/2	3	3 1/2	4	5
	mm	15	20	25	40	50	63	75	88	100	125
A	inch	3.15	3.54	4.33	5	5.28	6	6	6	6	6
	mm	80	90	110	127	134	152	152	152	152	152
B open	inch	5.98	6.22	7.72	10.04	11.42	14.09	14.09	14.09	14.09	14.09
	mm	152	158	196	255	290	358	358	358	358	358
C	inch	3.46	3.46	3.82	5.43	5.43	6.77	6.77	6.77	6.77	6.77
	mm	88	88	97	138	138	172	172	172	172	172
F	inch	.38	.55	.71	1.18	1.48	1.83	1.83	1.83	1.83	1.83
	mm	9.6	14	18	30	36.5	46.5	46.5	46.5	46.5	46.5
G	inch	1.26	1.5	1.89	2.52	3.07	3.35	3.35	3.35	3.35	3.35
	mm	32	38	48	64	78	85	85	85	85	85
Weight	lb.	4.25	5	7.75	16	16.75	38	38	38	38	38
	kg	1.9	2.3	3.6	7.3	7.6	17.2	17.2	17.2	17.2	17.2
PACKING		BH2	BH2	BH4	BH6	BY5	BH8	BH8	BH8	BH8	BH8
GASKET		G2	G2	G3	G6	G11	G10	G10	G10	G10	G10

This type of valve will have a threaded outlet on the top that is usually plugged using a round plug as shown below. Some call this plug a “bull plug”.

This arrangement does not provide a typical double block/bleed arrangement as some services require, and when that is needed, it may be necessary to use an integral double block/bleed valve such as shown in figure 6. The double block/bleed valve will normally require a flanged connection.

The “best” layout for minimal fatigue risk would be a design with the lowest stress concentration at the base (at the forged fitting where bending stress is highest) combined with the lowest added weight (valve and flanges).

The lowest stress concentration factor at the forged fitting will be achieved using a NipOLet fitting since the connection to the main pipe would be using a forged fitting with no weld on the branch connection until the flange or valve. This would be best since there would not be a weld at the common location where fatigue cracks occur and would allow use of higher allowable stress due to the lack of the weld. The NipOLet would require either a threaded or Socket Weld connection at the flange or valve which may end up the weak point from a fatigue perspective.

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A very typical vent/drain layout for a pipeline pump would involve a SockOLet on the main pipe, a short pipe nipple (4-6" long) using XS or heavier pipe, a Socket Weld gate valve with threaded outlet, and a bull plug to seal the top.

Reviewed Design

The design chosen for the review is as described above for a 1 ¼" pipe vent with a standard 6000# forged SockOLet fitting welded to a 14" Sch80 carbon steel pipe, a 4" long 1 ¼" XS pipe nipple (unthreaded), an 800# gate valve with Socket Weld at the base and Threaded at the top with a 2" long bull plug on top. The valve has a catalog weight of 16.25 Lb.

The model of the valve was simplified (removing bonnet details and valve handle) to match the total weight of the valve while maintaining the mechanical details of the socket weld at the base of the valve. The model of the vent as reviewed is shown in figure 7.

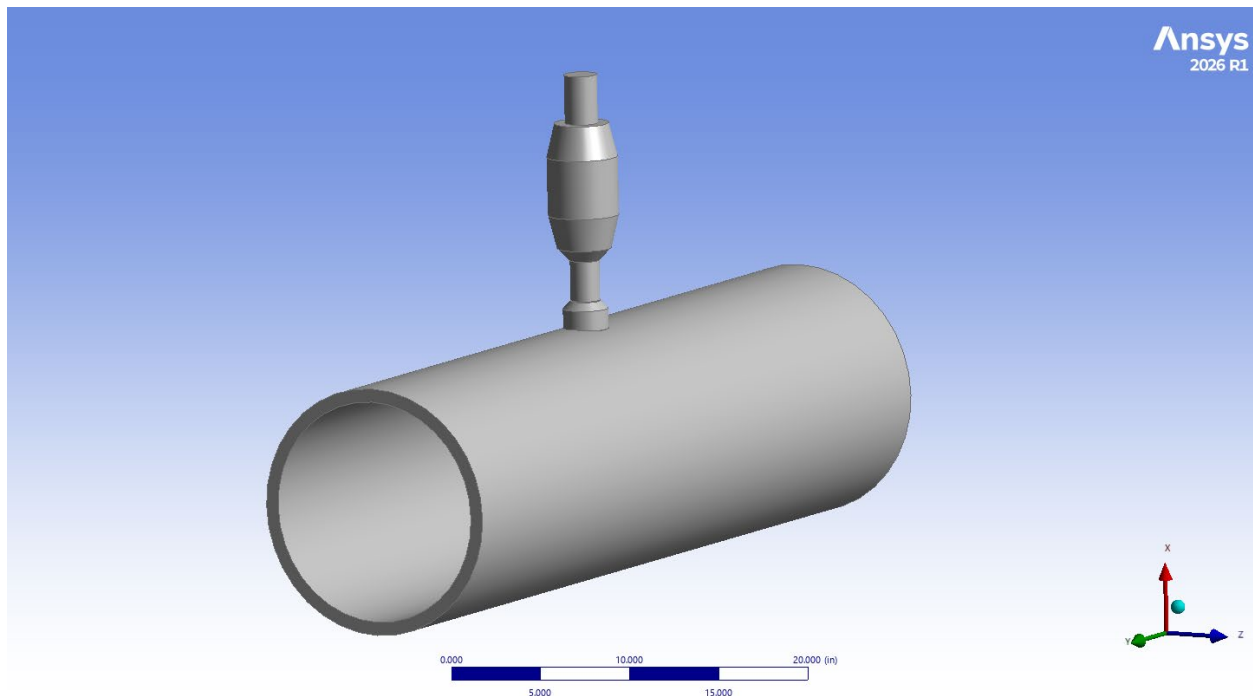


Figure 7 - Model layout

The vent as modeled was calculated to have a first natural frequency of 136 Hz in the direction perpendicular to the main pipe and 142 Hz as detailed in figure 8 with motion in the direction of the main pipe. The first mode is the one that is normally the cause of fatigue failures and is a typical cantilever mode with the free end of the vent vibrating either inline with the pipe or perpendicular to the main pipe. The 1st bending mode for both directions

will be similar in frequency but slightly separated depending on the flexibility of the connection of the branch pipe to the main pipe.

This mode will normally have the highest stress at the weld between the SockOLet and the pipe nipple as shown in figure 9. In some cases where the main pipe is rather thin wall (i.e. Sch10 pipe), it is possible that the main pipe wall is the most flexible component and the maximum stress and likely point of fatigue would shift to the main pipe near the weld between the SockOLet and the main pipe (or near the weld with the crack in the main pipe).

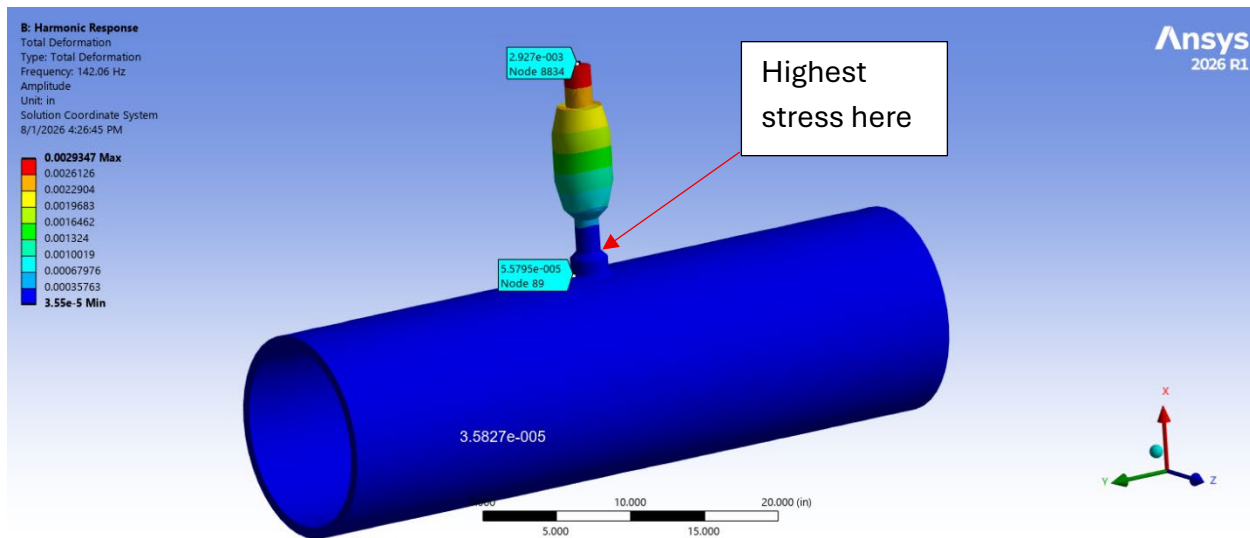


Figure 8 - 1st Mode Deflection

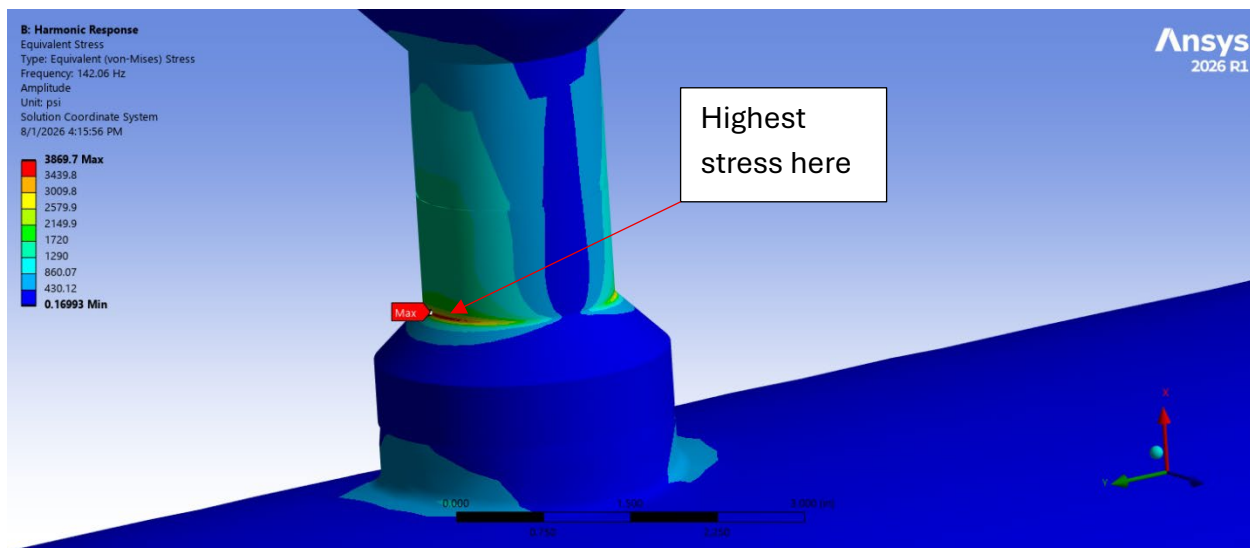


Figure 9 - Stress at Resonance

Allowable Vibration

The allowable vibration is defined by limits on fatigue stress at the weld as shown in figure 9. Although the API 579 Part 15 is expected to improve the rules for assessing small bore pipe fatigue, it has not yet been formally released at the time this paper was developed. Preliminary publications have suggested that the limiting fatigue stress will be 3,870 psi dynamic stress amplitude. Since the detailed FEA model should include an appropriate stress concentration for the weld geometry, the FEA stress should be directly comparable to the allowable limit.

The forced response conditions used to generate the stress plot in figure 9 included 0.021 in/sec pk vibration of the main pipe in the direction of normal flow at the natural frequency. This was used to evaluate forced response of the main pipe with the vent resonating at the natural frequency.

The calculated vibration velocity for the vent line with 1% modal damping is shown in figure 10. This was determined by increasing the vibration on the main pipe in the direction of flow until the stress at the weld reached the limit of 3,870 psi.

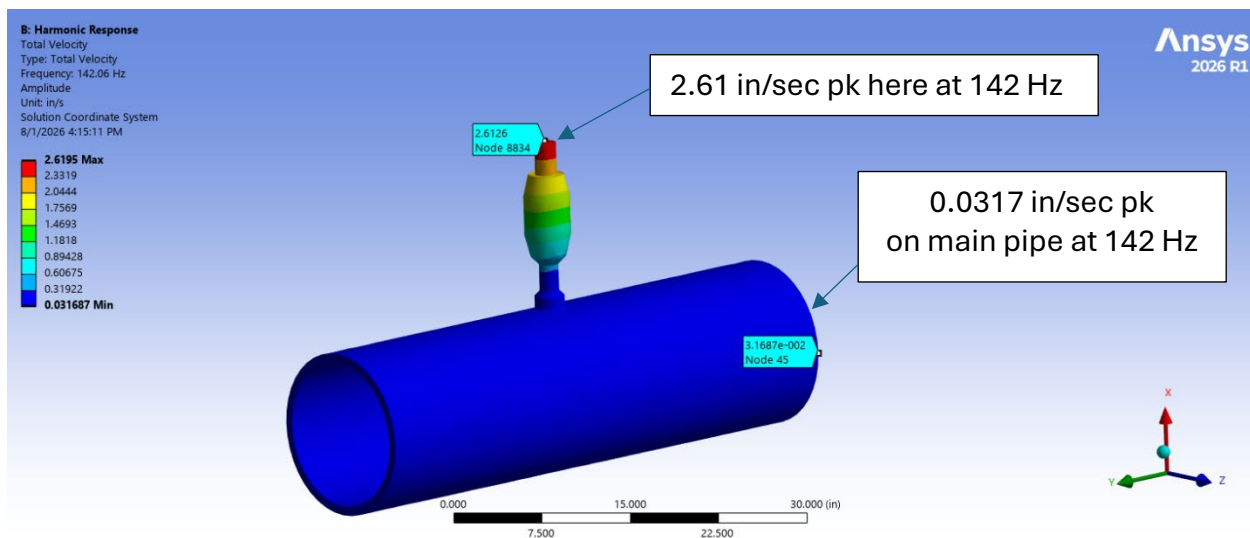


Figure 10 - Vibration Velocity on Branch When at the Fatigue Limit

Actual damping values on the vent arrangement can be determined from field testing. Author's experience suggests that typical damping levels will be from about 0.25% to 2% depending on a variety of factors. If the actual damping level was 0.25% instead of 1%, the vibration amplitude in figure 10 and the stress in figure 9 would be about 4 times higher.

Remediation Options

Once a pipeline is built, there are limited options for what to do to reduce the risk of fatigue. It is often not practical to weld additional bracing or stiffeners onto the branch connection without hydrotesting the line. The practical options that can be considered are:

- Remove the vent if possible
 - If flanged, the valve can be removed and replaced with a blind flange
 - This would obviously eliminate the function of the vent
- Add gussets to rigidize the vent to the main pipe
- Add some external stiffening to the vent by bracing back to the main pipe (self-reinforcement)
 - Various implementations are available online such as the AVT Reliability Inc Easy-Brace® product
 - ZetaQuest has a similar offering with a saddle brace
- Use the self-reinforced brace with viscous dampers
- Attach a ZetaQuest Rohr damper on the end

Removing the Vent

This is the most obvious remediation option with highest confidence of eliminating the risk of fatigue on the vent connection. How this is accomplished can impact the results. If it is a flanged valve connection, the valve could be removed and replaced with a blind flange. This still leaves the small-bore connection but would greatly reduce the overhung weight and increase the natural frequency significantly.

Adding Gussets

This is a common option at the original fabrication stage since it requires welding from the branch back to the main pipe. The specific design details may warrant a repad under the gusset on the main pipe of some sort so that the gusset is not directly welded onto the main pipe.

The layout modeled is shown in figure 11 and did not include any repad on the main pipe. This was generated with a typical gusset layout using 3/8" flat plate and typical weld details.

This arrangement shifts the natural frequency up from 142 Hz to 186 Hz. Using the same approach of increasing the main pipe vibration until the maximum stress reached the limit, the maximum allowable vibration velocity for the gusseted design would be 1.96 in/sec pk.

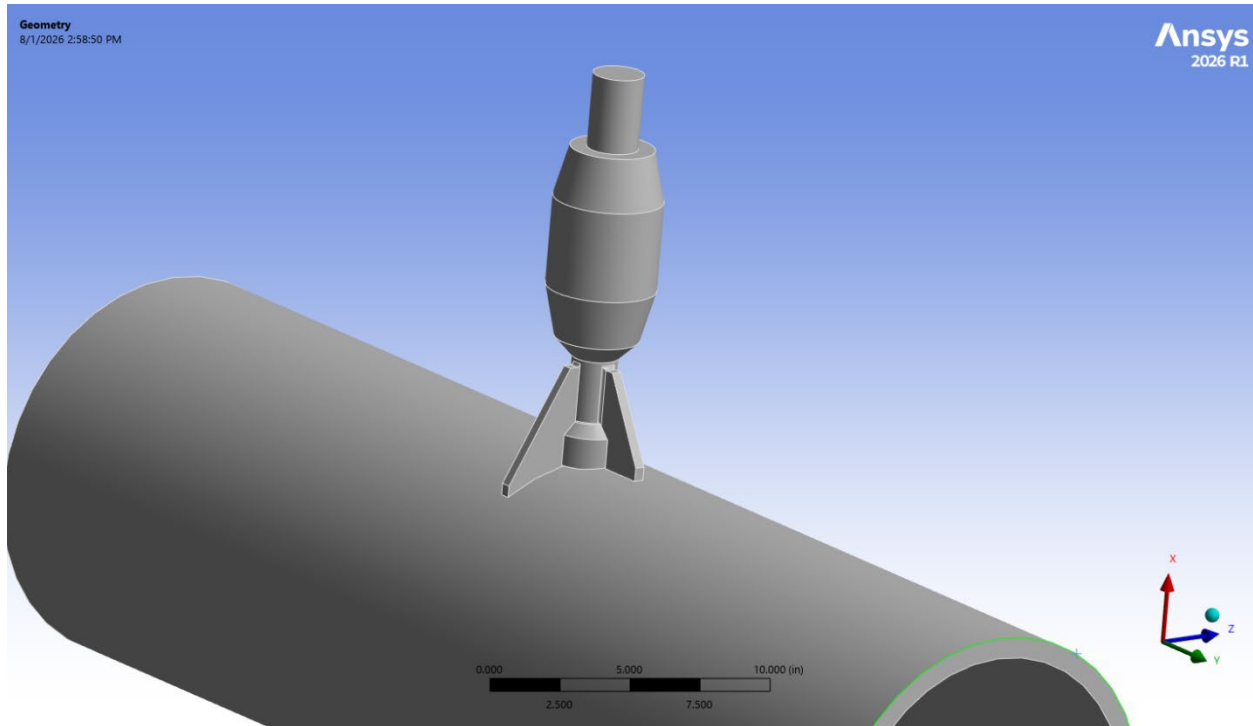


Figure 11 - Layout with Gussets

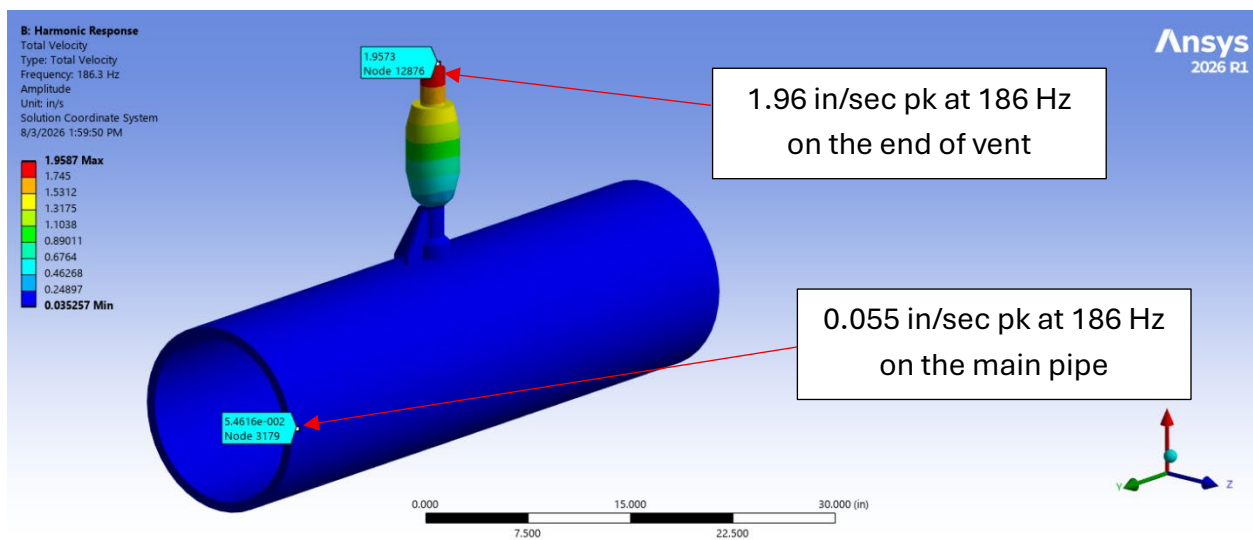


Figure 12 - Vibration Velocity on Gusseted Design at Fatigue Limit

The vibration velocity for the gusseted design is shown in figure 12 where the maximum vibration at the end of the vent was 1.96 in/sec pk while the main pipe vibration was at 0.055 in/sec pk all at 186.3 Hz (at resonance). It should be noted that the presence of the gussets allows the vibration on the main pipe to increase from 0.032 to 0.055 in/sec pk for the same weld stress compared to the design with no gussets.

This is a solid improvement and well within ASME/API code and standard guidance. This result validates the common practice of using 2-dimensional reinforcement on small bore branches when they are located near machinery that is prone to vibrate like reciprocating compressors or screw compressors.

The stress with the gusseted design shifts up to the top of the gusset at the connection to the pipe and away from the weld on the SockOlet as shown in figure 13.

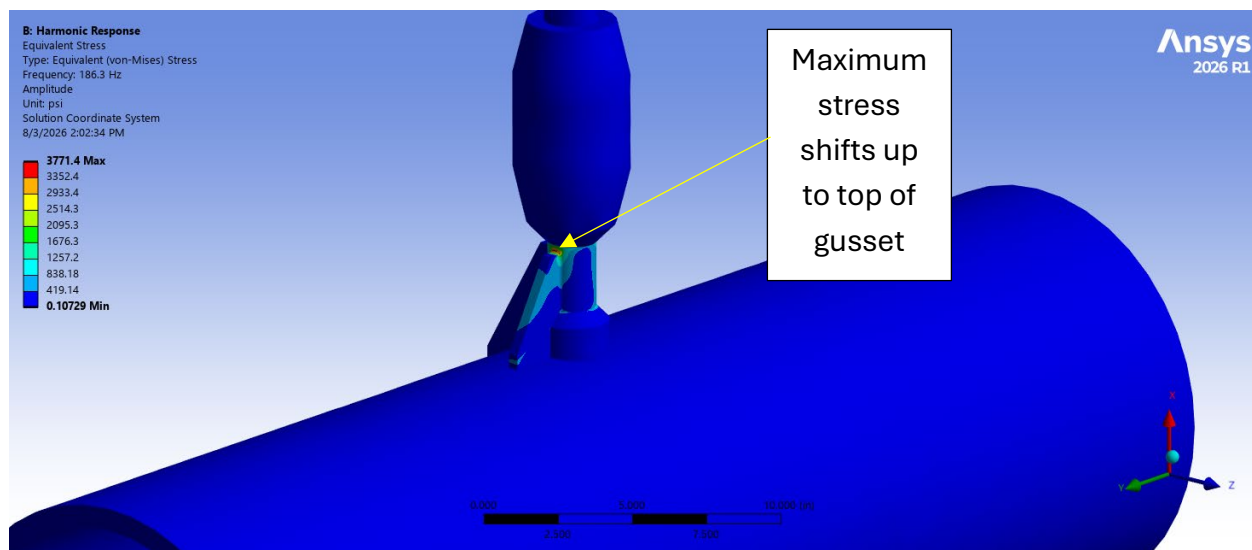


Figure 13 - Dynamic Stress for Gusseted Design

Self-Reinforcement

The self-reinforcement option includes connecting the top of the vent line back to the main pipe using 2 or more “rigid” links that are terminated in some form to the main pipe. There are a variety of concepts used for a self-reinforcement design including using Unistrut, U-clamps and a variety of other hardware to brace back to the main pipe. One custom option is offered by AVT Reliability Inc. as shown in Figure 14.

The bolt on self-reinforcement can be a very good option for remediation on an existing pipeline since it would not require welding modifications and could be installed with the line in service.



Figure 14 - AVT Reliability Inc Easy-Brace®

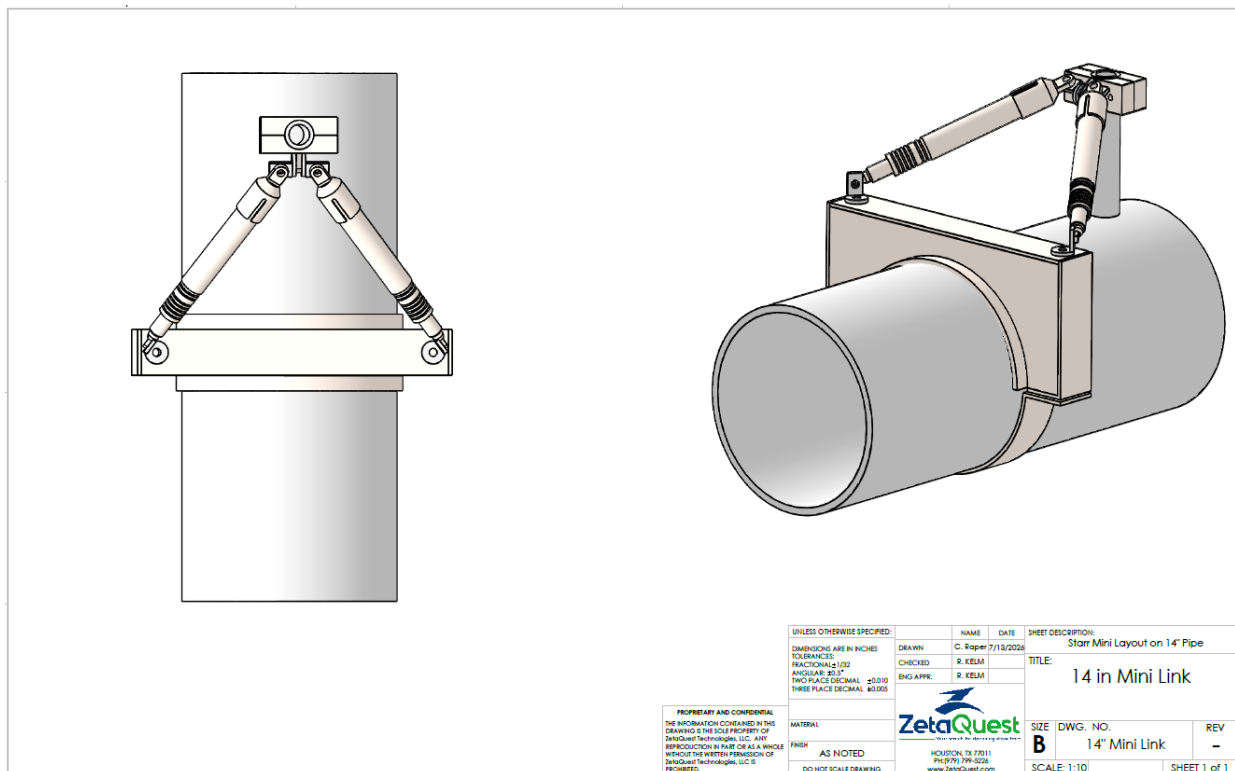


Figure 15 - ZetaQuest Saddle Clamp Design

ZetaQuest offers a similar functioning device that is called a Saddle brace. An example of the hardware is shown in figure 15. The ZetaQuest option uses a clamp on saddle for the main pipe with a platform above the top of the main pipe that can be used to attach either rigid links or Starr Mini viscous dampers.

The design that was modeled is shown in figure 15. For the rigid links, a standard $\frac{3}{4}$ " Sch 40 pipe was used.

With the rigid links attached from the saddle to the end of the branch above the valve (at the bull plug), the model is shown in figure 16. The “rigid” link is modeled using a differential spring with a stiffness calculated for the element used.

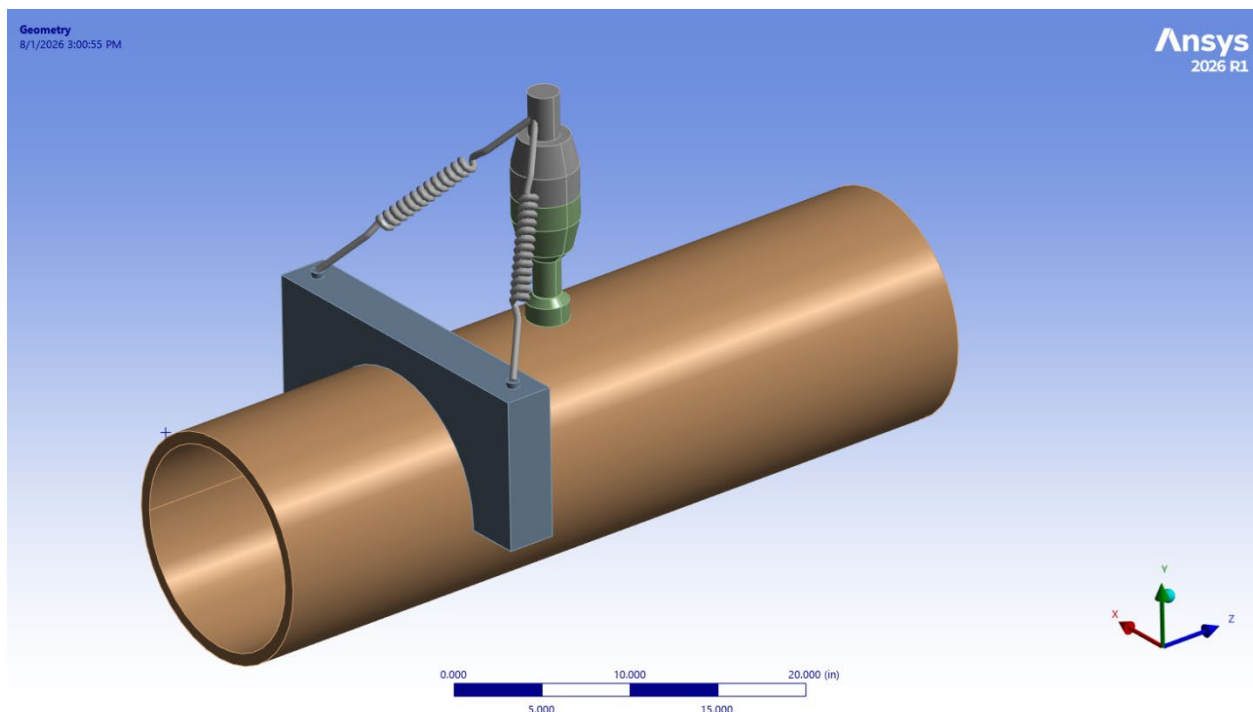


Figure 16 - ZetaQuest Saddle Geometry

The addition of the “rigid” brace back to the parent pipe increases the natural frequency dramatically up to a calculated value of 357 Hz. The shifting of the natural frequency up from 142 Hz could easily be adequate to eliminate a vibration issue. However, the same pumps that generate vane pass frequency also generate 2x vane pass frequency. The higher harmonics can again be excited if present on the main pipe.

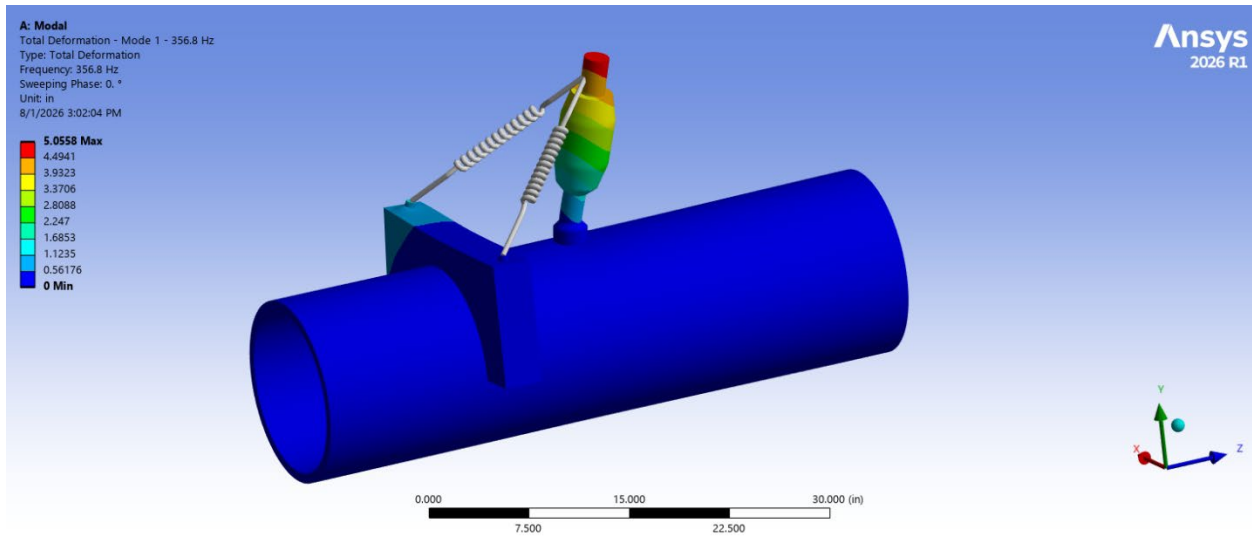


Figure 17 - Natural Frequency with 2D Brace

The vibration amplitude required to approach the same stress limit on the pipe was found as shown in figure 18 with the stress shown in figure 19.

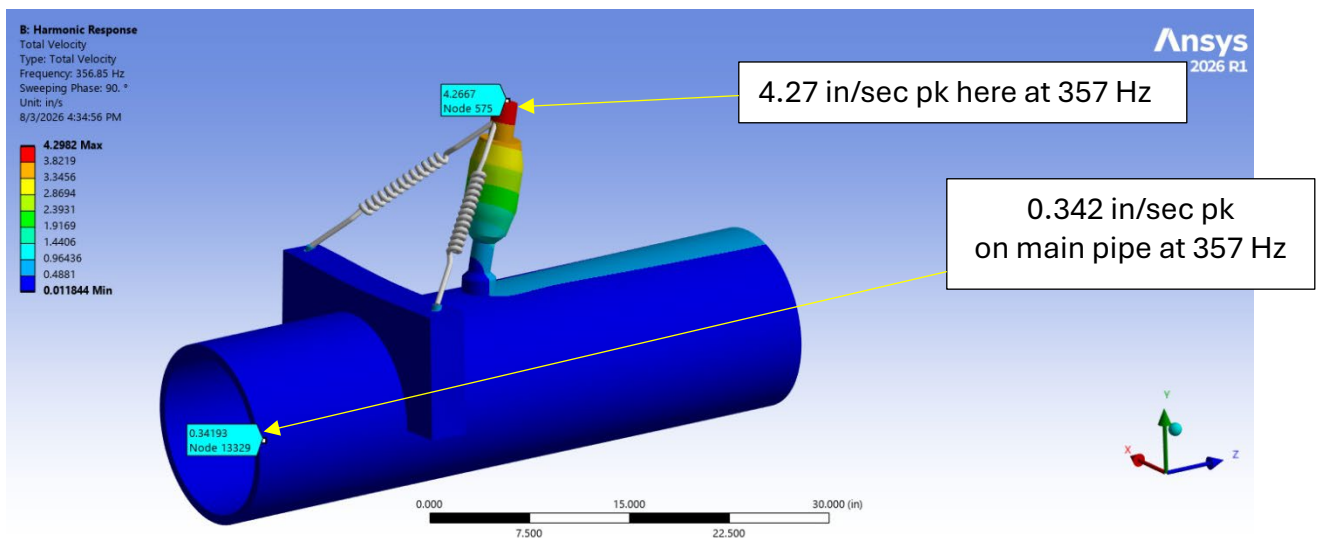


Figure 18 - Vibration Velocity with 2D Brace at Fatigue Limit

Although the vibration velocity values are significantly higher with the brace, the differential deflection between the pipe and the end of the small-bore pipe is about the same as the differential deflection for the original design as expected. The higher allowable vibration velocity is due to the frequency being so much higher than the original design. The stress on the weld is essentially dictated by differential deflection of the small-bore pipe compared to the main pipe.

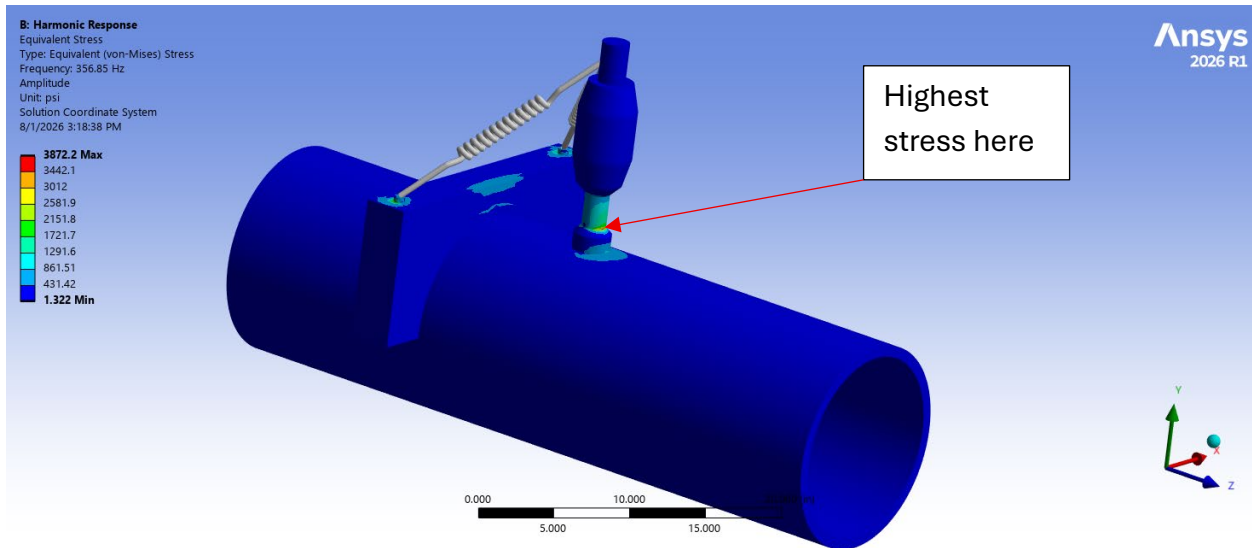


Figure 19 - Dynamic Stress on 2D Braced Design

ZetaQuest Starr Mini Option

The same model was used as the 2D brace option but with the rigid links replaced with ZetaQuest Starr Mini viscous dampers. The Starr Mini option is preferred over the rigid links since it not only provides some dynamic stiffness but also provides moderate damping to greatly reduce the vibration at the natural frequency that still exists.

Using again the same approach by increasing the main pipe vibration until the stress limit on the branch was reached, the maximum vibration is shown in figure 20. The associated stress plot is shown in figure 21.

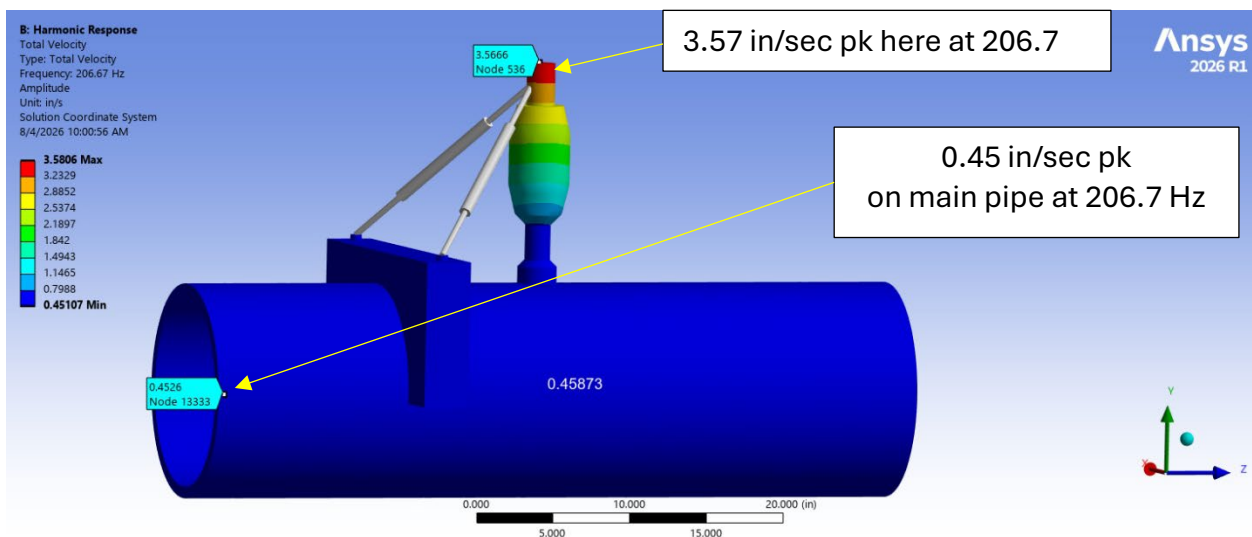


Figure 20 - ZetaQuest Starr Mini Velocity

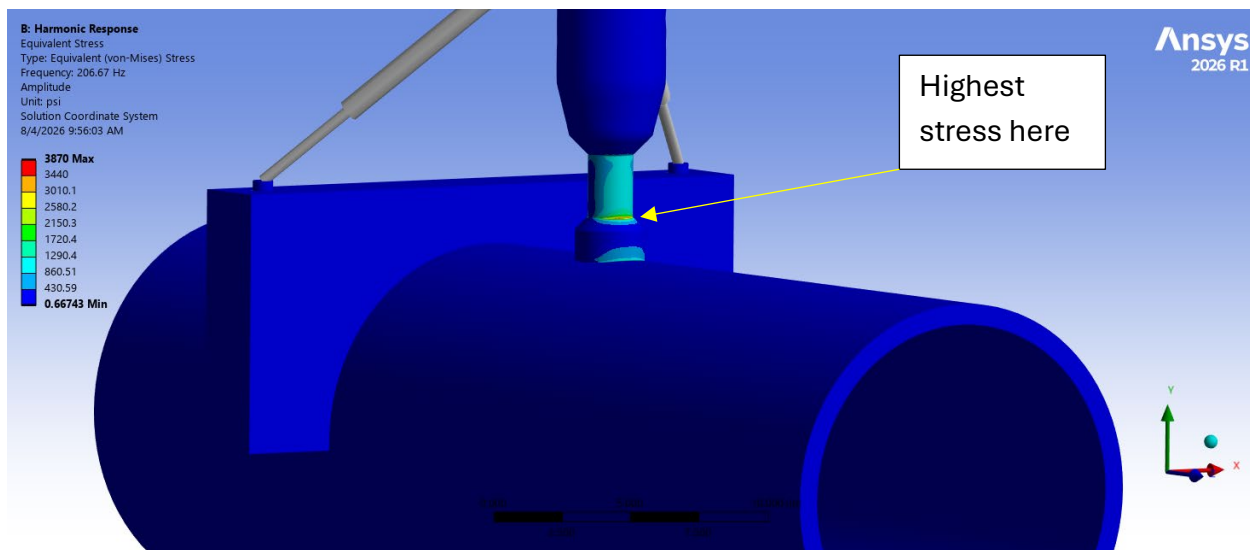


Figure 21 - Stress Plot for Starr Mini Results

The use of the viscous damper produces a design that can tolerate higher vibration on the main pipe before reaching the fatigue limit on the small-bore pipe.

ZetaQuest Rohr Damper

A ZetaQuest Rohr damper was also mounted to the end of the small-bore branch using a Rohr 1.5 damper designed for the natural frequency. The attachment is similar to that shown in figure 22.

The Rohr damper is normally selected with a size that provides 15% of the modal mass of the vent branch line and is an optimally damped tuned vibration absorber. The addition of the device will provide damping up to about 13.5% for the first cantilever mode on the vent attachment.

Using the same approach of increasing the vibration on the main pipe until the stress limit on the small-bore pipe was reached resulted in vibration as shown in figure 23 and stress shown in figure 24.

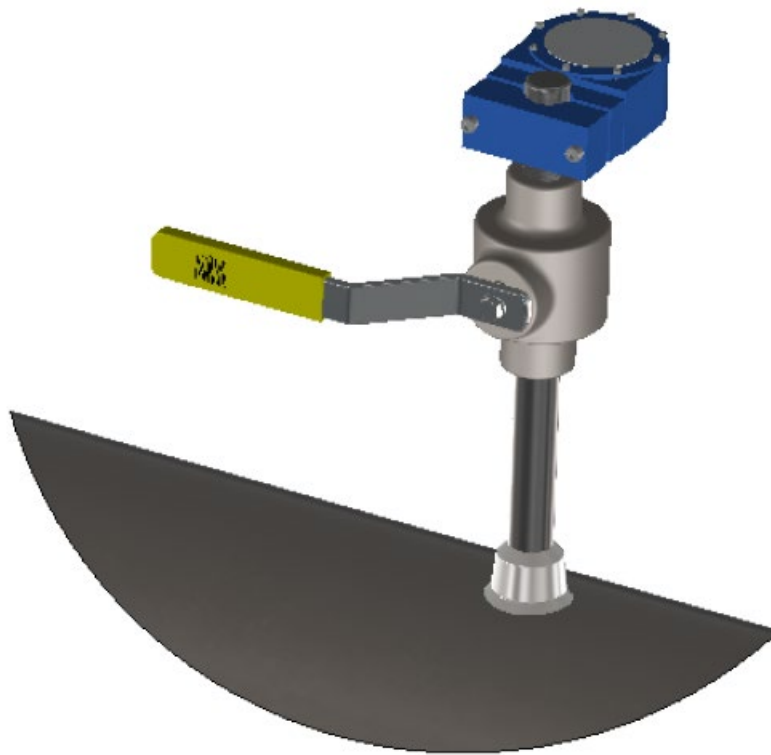


Figure 22 - ZetaQuest Rohr Damper Application

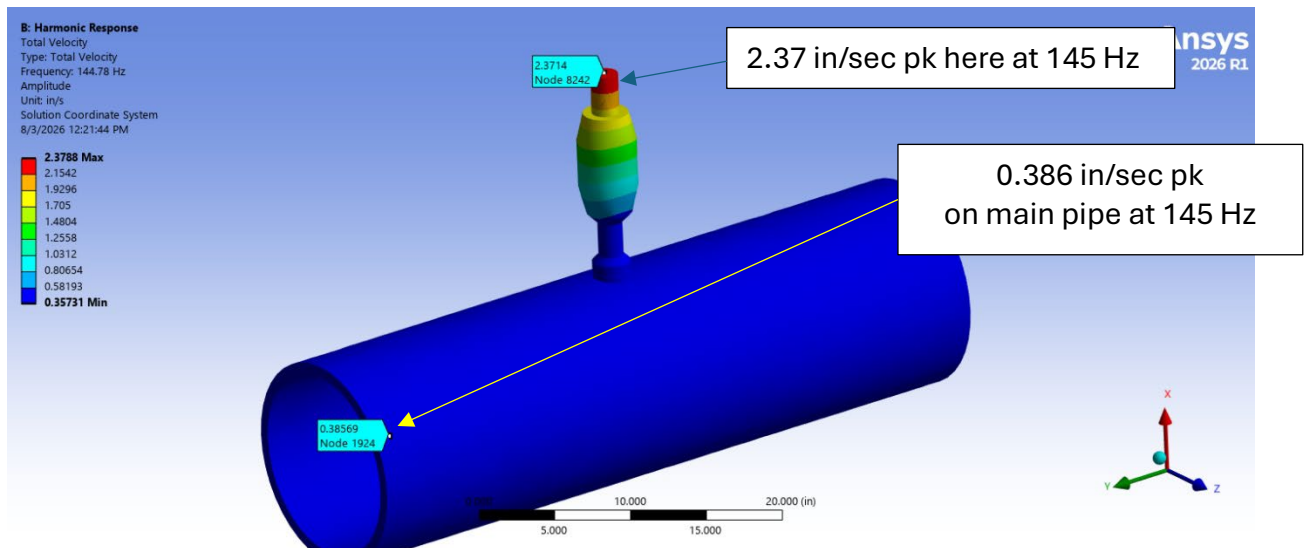


Figure 23 - Vibration Response with Rohr Damper

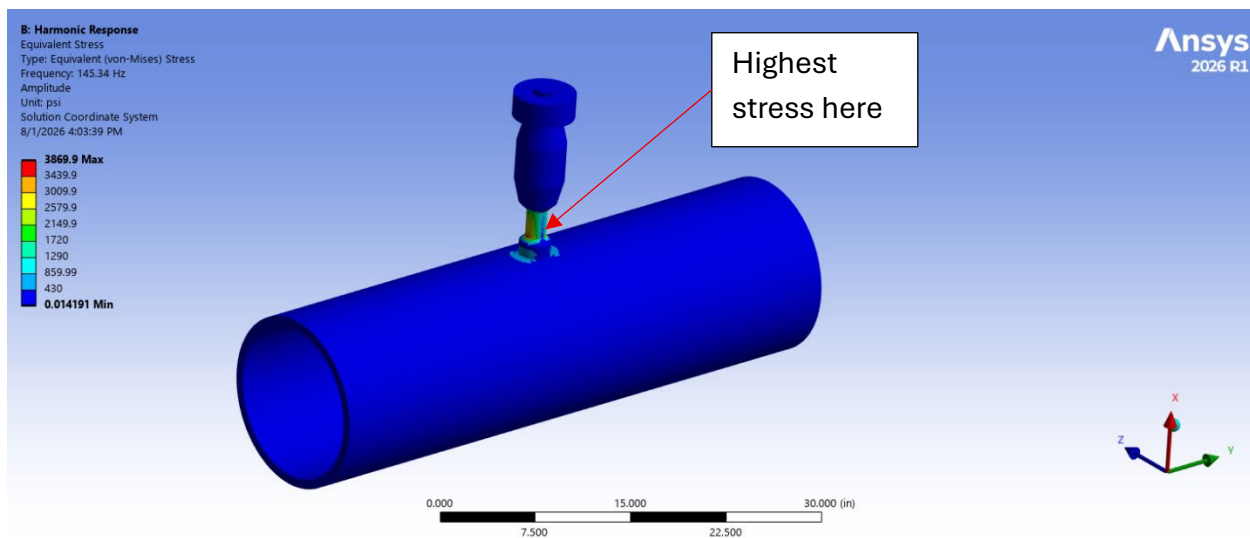


Figure 24 - Dynamic Stress for Rohr Damper Results

Summary

Based on finite element modeling for various methods of remediating elevated vibration on a small-bore vent line typical on pipeline pump applications, 5 options were evaluated to determine the relative sensitivity of each to fatigue risk.

The results are summarized in table 1 below with a focus on the allowable vibration on the main pipe for each design type. This is a key criterion because it would be preferable to have a vent branch design that could tolerate the highest vibration possible on the main pipe since the primary source for fatigue is transmitted vibration from the main pipe.

Table 1 - Results Summary

Description	Natural Frequency, Hz	Maximum Pipe Velocity, in/sec pk	Maximum small bore branch vibration, in/sec pk
Original design	142 Hz	0.032 in/sec pk	2.61 in/sec pk
Gusseted design	186 Hz	0.055 in/sec pk	1.96 in/sec pk
2D Brace	357 Hz	0.342 in/sec pk	4.27 in/sec pk
2D with Starr Mini	208 Hz	0.45 in/sec pk	3.57 in/sec pk
Rohr Damper	145 Hz	0.386 in/sec pk	2.37 in/sec pk

As observed above, there are obvious advantages to using some option other than the standard forged fitting to reduce the risk of fatigue on the small-bore connections.

Another way to present the results is shown in Table 2 where the vibration on the vent is shown as a percentage of the original design. That dictates the improvement that should be achieved with each remediation effort. The vibration amplitudes on the vent are presented for the same 0.032 in/sec pk vibration velocity on the main pipe.

Table 2 - Relative Vibration

Design	Maximum Vent Vibration	% of Original Vibration
Original Design	2.61 in/sec pk	100%
Gusseted Design	1.14 in/sec pk	44%
2D Brace	0.40 in/sec pk	15%
2D with Starr Mini	0.25 in/sec pk	10%
Rohr Damper	0.20 in/sec pk	8%

Conclusions

A typical vent line attachment that is common in the pipeline industry is at high risk of fatigue failure due to even relatively small vibration transmitted from the pipeline pumps to the suction and discharge piping at the pump station. This review suggests that main pipe vibration as low as 0.032 in/sec pk at vane pass frequency can produce excessive stress in the branch connections.

Using gussets on the small-bore branches can improve the acceptable vibration for the main pipe before the vent is at risk of fatiguing but does not eliminate the risk for moderate main pipe vibration.

Use of external self-reinforcement can greatly improve the tolerance to main pipe vibration, with a based self-reinforcement using “rigid” links achieving over 10 times the allowable main pipe vibration.

Replacing the “rigid” links with viscous dampers for the self-reinforced design further improves the allowable main pipe vibration by adding damping that reduces the resonant vibration response for the vent.

Using the Rohr damper approach provides the best vibration control for vent/drain lines as demonstrated in the review. Allowable vibration on the main pipe increases to over 10 times the value without the damper. However, it should be noted that elevated vibration can still create fatigue risk on small bore pipe even with good remediation efforts.

Considering that ASME fatigue data generally has about a factor of 2 built into the charts, operating with main pipe vibration in the range of 1 in/sec pk or higher near small bore

attachments will pose a real risk of fatigue failure even with the best remediation options implemented.

Works Cited

1. *Integrity Evaluation of Small Bore Connections (Branch Connections)*. **Chris B. Harper, P.Eng.** 2014, 9th Conference of the EFRC, Vienna.
2. *Operation and Maintenance of Nuclear Power Plants - ASME OM-2009*. **OM-2009, ASME.** s.l. : ASME, 2009.