ASIA TURBOMACHINERY & PUMP SYMPOSIUM
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Plunger Pump Station
Vibration-Induced Cracks in Piping

Presenter:

Paul Crowther, MSc, CEng,

Principal Consultant, Wood

wood.

TEES TURBOMACHINERY LABORATORY
TEXAS A&M ENGINEERING EXPERIMENT STATION

TEXAS A&M UNIVERSITY

Author bios

Kelly Eberle, Wood, BSc, PEng, Principal Consultant

Kelly is a Principal Consultant with Wood (formerly BETA Machinery Analysis) since 1988. Kelly graduated from the University of Saskatchewan with a Bachelor of Science in Mechanical Engineering in 1986. He has been a professional member of APEGA since 1991. He has accumulated a wide range of design and field experience, particularly in the area of pressure pulsation analysis and mechanical analysis of reciprocating compressor and pump installations. The scope of his experience includes pulsation simulations, thermal flexibility studies, dynamic finite element analysis, structural analysis and foundation analysis.

Mena Ghattas, BSc, PE,

Mena Ghattas worked with Wood's vibration integrity group until October 2017, which specializes in design, inspection, and consulting services related to static and transient piping vibration for onshore and offshore facilities. This includes requirements defined by the Energy Institute Guideline (Avoidance of Vibration Induced Fatigue Failure), as well as advanced analysis services such as dynamic pulsation analysis, dynamic structural analysis and rotating equipment analysis.

Presenter bio

Paul Crowther, Wood, MSc, CEng, Principal Consultant

Paul is a technical authority in Wood's piping vibration and integrity team, which involves design, inspection, and consulting services related to static and transient piping vibration for onshore and offshore facilities. This includes requirements defined by the Energy Institute guideline (Avoidance of Vibration Induced Fatigue Failure), as well as many different advanced analysis services.

Paul is a principal consultant, with over 11 years of experience in advanced engineering analysis in Europe and the Middle East, where he has supervised large scale piping vibration projects. He has carried out numerous investigations and studies covering most vibration excitation mechanisms found in modern process plant operations across the world, both topsides and subsea. He is actively involved in industry committees, research, supervision, and specialized engineering projects.

Abstract

Many plunger pumps are installed without adequate pulsation and vibration considerations at the design stage. Pressure pulsations from normal pump operating can cause high shaking forces throughout the entire piping system. These shaking forces typically occur at multiples of plunger passing frequencies. In fixed-speed, low power applications it may be acceptable to not complete an API 674 vibration study, however, in variable-speed applications above 50 HP (37 kW) the consequences can be costly. In many cases uncontrolled shaking forces can result in production downtime, costly fixes and high-risk fatigue failures.

This case study highlights how a simple station upgrade resulted in significant failures, field troubleshooting, non-destructive testing investigations, downtime and re-design. Both field data and design modeling are used to tell the story and showcase design elements that should be considered for all variable-speed plunger pump applications over 50 HP (37 kW).

Objective

Demonstrate the risks and costs of excluding a pulsation analysis from a pump installation, as well as ways to mitigate the pulsation and cavitation risks.

Analysis approach

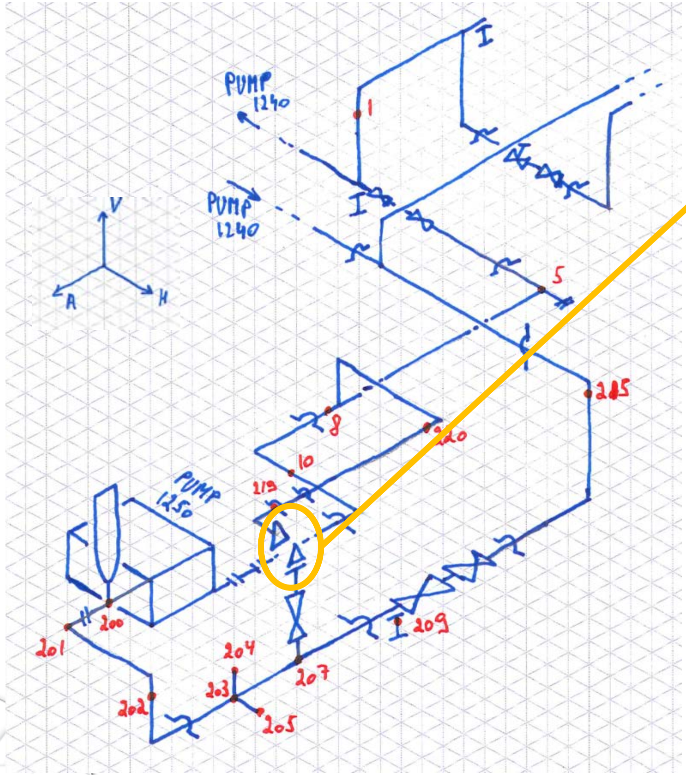
Field-measured pulsations and numerical simulations (1-D pulsation model).

Case study

- (2) Quintuplex pumps
- 297 HP
- Liquid propane 416 GPM @ 400 RPM
- Speed range 200-400 RPM

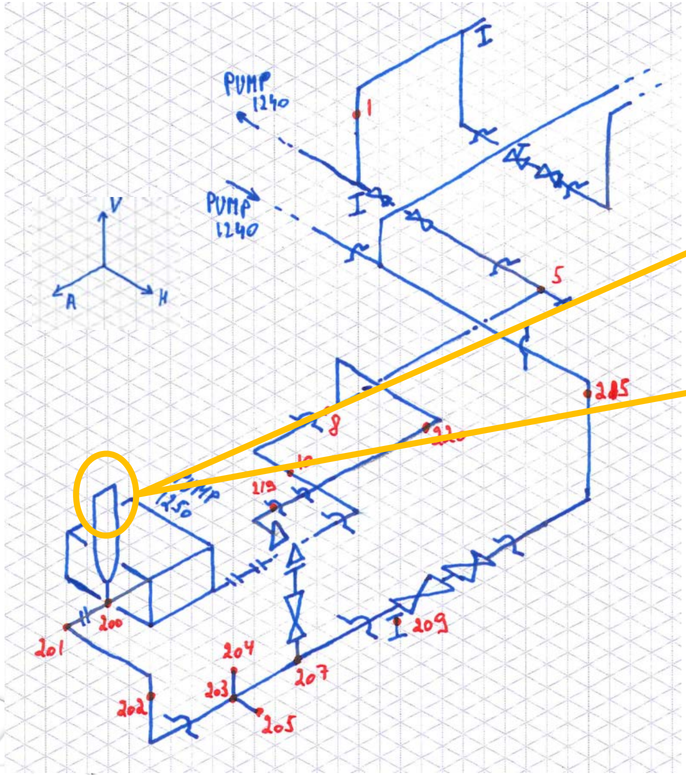
6 months in operation, very high vibrations

Site vibration measurements



Location	Dir'n	Percent Guideline
PSV top	A	1192
PSV top	H	883
220	V	693
Dampener top	A	628
220	H	579
1	H	539
small bore near 5	A	399
Dampener top	H	392
5	A	392
204	-H	322
203	H	314
220	A	287
8	H	272
207	H	270
215	A	264
209	H	223
202	A	217
10	A	215

Site vibration measurements



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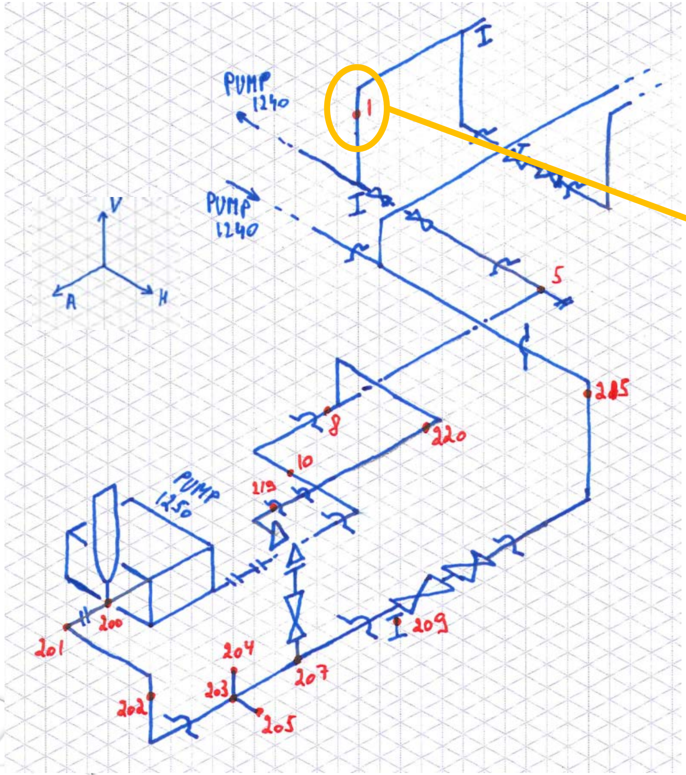

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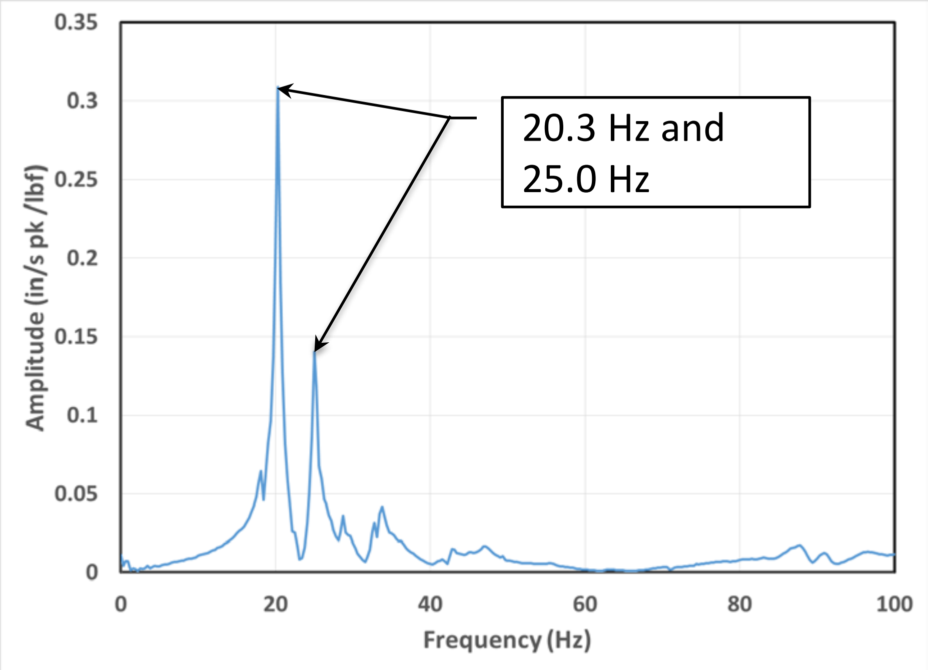
PSV site measurements

Mechanical natural frequency measurement



Vibration measurement

Location	Dir'n	Percent Guideline	Amplitude	Units	Freq. (Hz)
PSV top	A	**1192**	9.01	ips Pk	24.1
PSV top	H	**883**	6.85	ips Pk	24.7

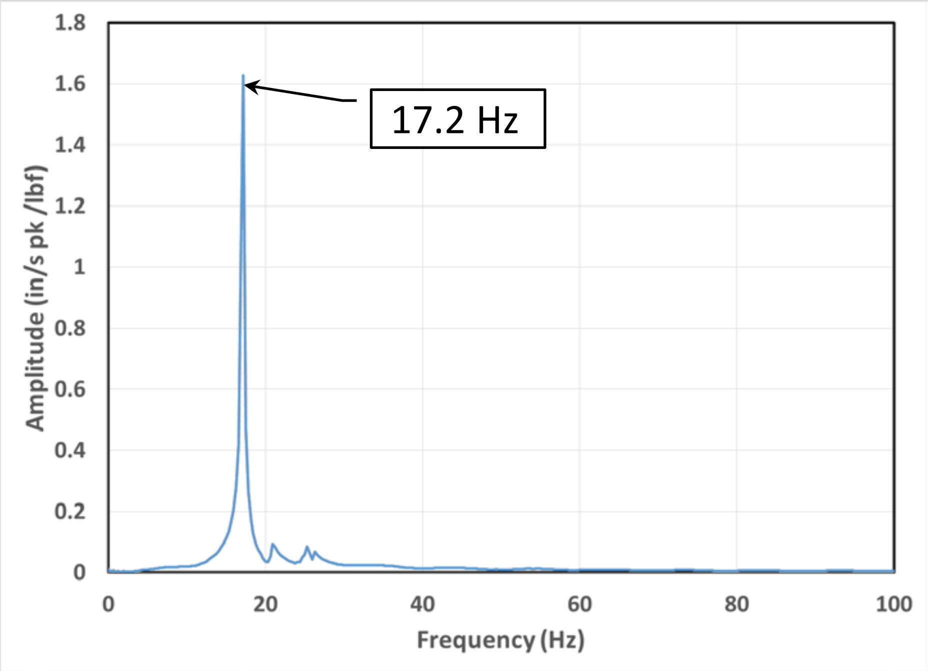
Maximum vibration is close the measured mechanical natural frequency (MNF).

<u>Conclusion:</u>

PSV is mechanically resonant

Dampener site measurements

Mechanical natural frequency measurement



Vibration measurement

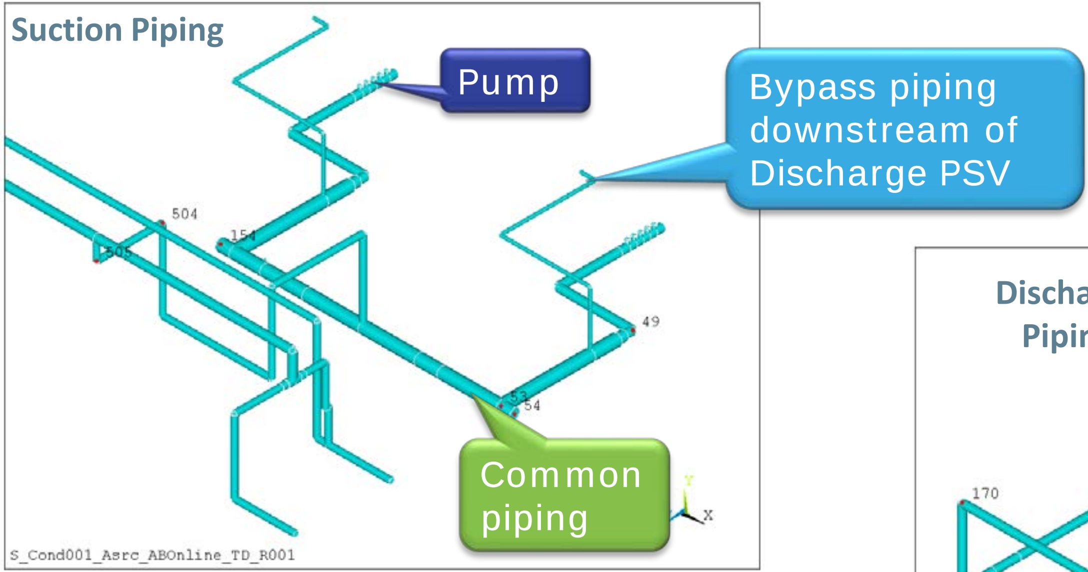
Location	Dir'n	Percent Guideline	Amplitude	Units	Freq. (Hz)
Dampener top	A	628	3.33	ips Pk	16.9
Dampener top	H	392	1.85	ips Pk	15

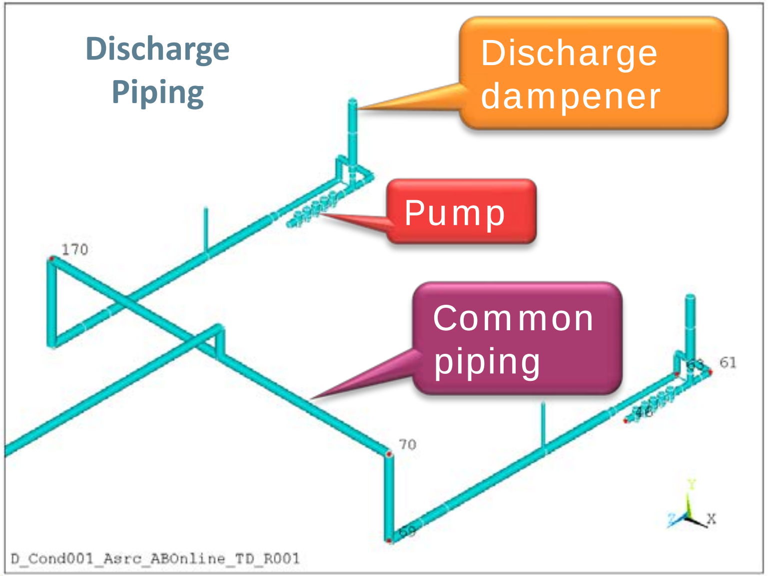
Maximum vibration is close the measured MNF.

Conclusion:

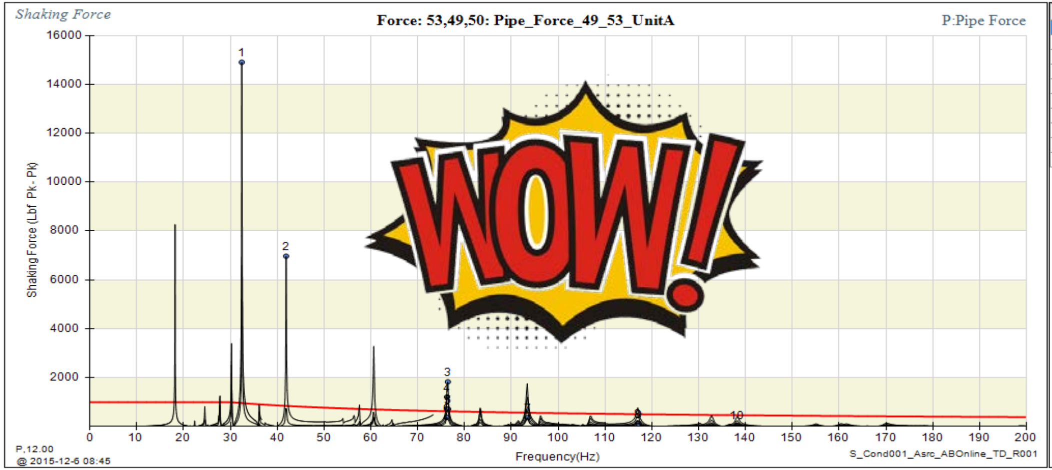
Dampener is mechanically resonant

Pulsation model





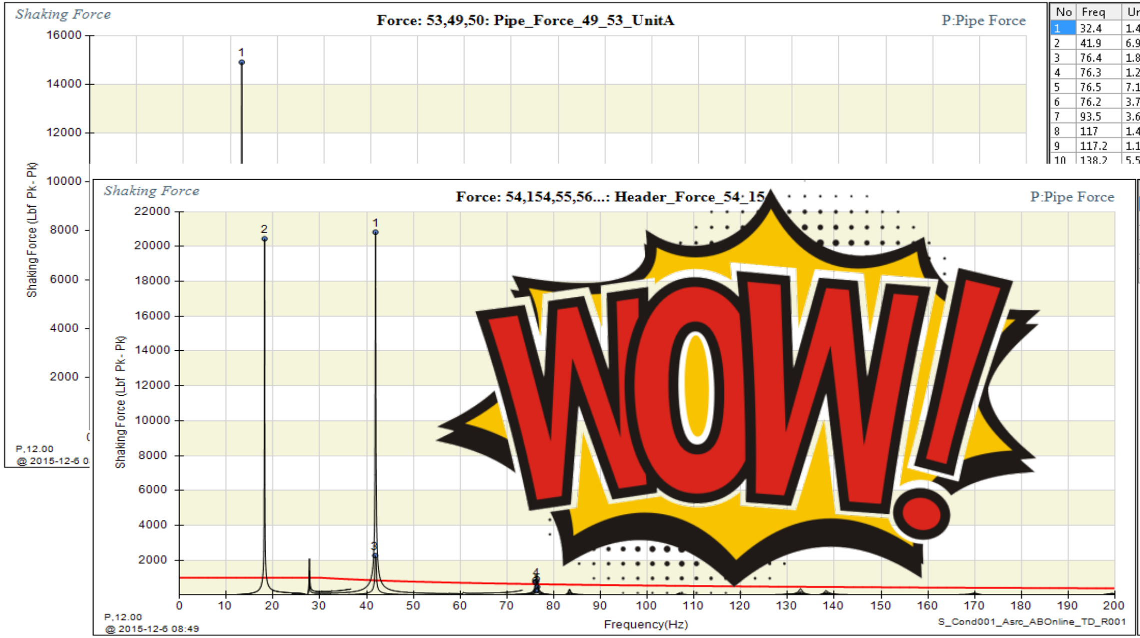
Pulsation model – shaking forces



Harmonics Peak Points :

No	Freq	Unbal_Forc
1	32.4	1.49E+04
2	41.9	6.97E+03
3	76.4	1.83E+03
4	76.3	1.21E+03
5	76.5	7.14E+02
6	76.2	3.78E+02
7	93.5	3.64E+02
8	117	1.40E+02
9	117.2	1.15E+02
10	138.2	5.59E+01

Pulsation model – shaking forces



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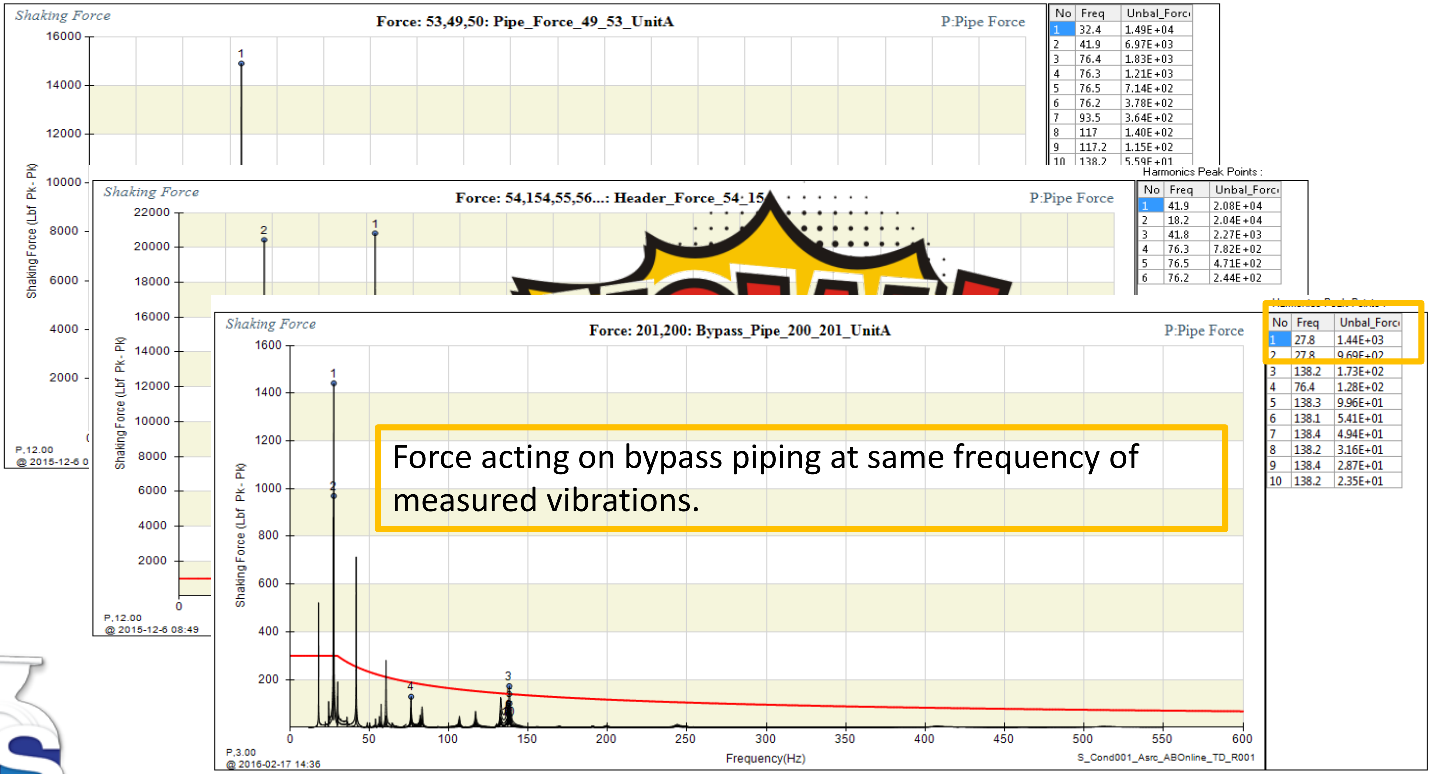
Harmonics Peak Points :

No	Freq	Unbal_Forc
1	41.9	2.08E+04
2	18.2	2.04E+04
3	41.8	2.27E+03
4	76.3	7.82E+02
5	76.5	4.71E+02
6	76.2	2.44E+02

Pulsation model – shaking forces



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6	76.2	2.44E+02

No	Freq	Unbal_Forc
1	27.8	1.44E+03
2	27.8	9.69E+02
3	138.2	1.73E+02
4	76.4	1.28E+02
5	138.3	9.96E+01
6	138.1	5.41E+01
7	138.4	4.94E+01
8	138.2	3.16E+01
9	138.4	2.87E+01
10	138.2	2.35E+01

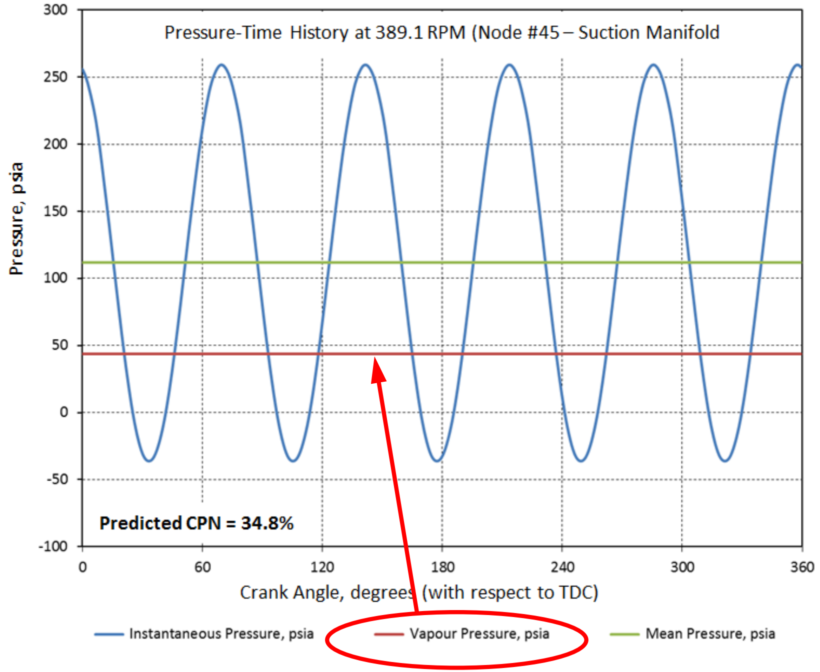
Pulsation model – shaking forces

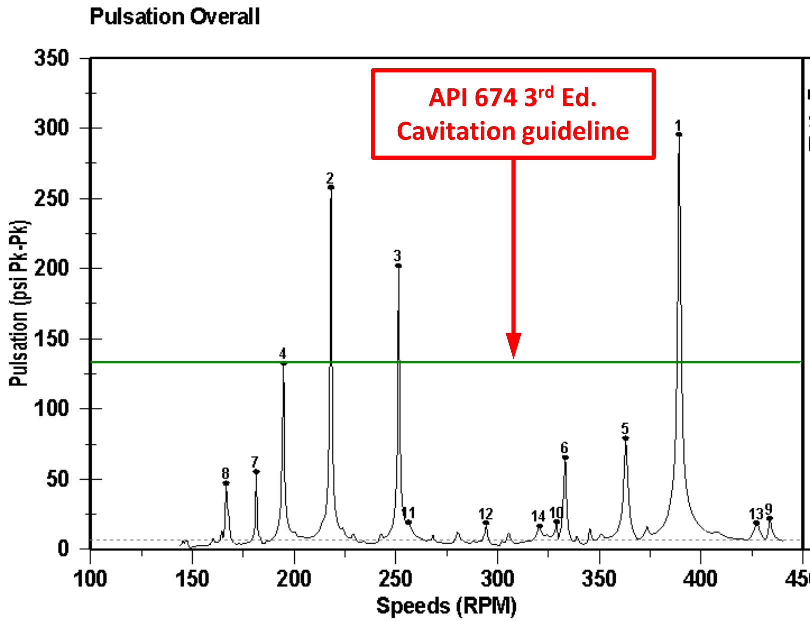
Pipe span	Existing (full speed 200 – 400 rpm) Lbf peak-peak	Existing (650 bbl/hr 200 – 218.5 rpm) Lbf peak-peak
45 – 48	5,100	1,020
48 - 49	7,220	1,580
49 - 53	14,900	4,920
200 - 201	1,440	309
201 - 202	6,980	139
54 - 154	20,800	12,100
57 - 500	2,410	524
500 - 501	3,180	695
501 - 502	2,150	473

Pipe span	Existing (full speed 200 – 400 rpm) Lbf peak-peak	Existing (650 bbl/hr 200 – 218.5 rpm) Lbf peak-peak
502 - 503	2,070	299
503 - 504	1,370	296
504 – 505	2,020	439
400 - 507	4,390	958
404 - 409	8,100	196
508 - 509	6,990	1,520
510 - 702	2,500	434
511 - 600	1,660	398
702 - 703	1,800	390

Shaking forces in other areas in the system.

Pulsation model

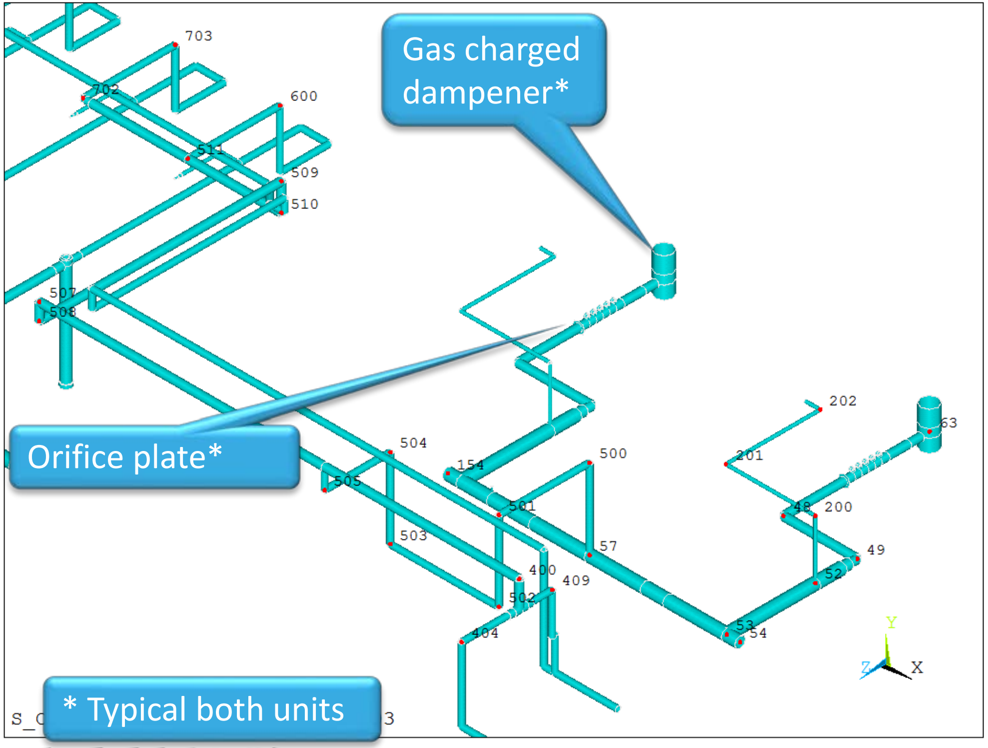




#	Speeds RPM	Pulsation psi Pk-Pk
1	389.1	2.95E+02
2	218.2	2.58E+02
3	251.4	2.02E+02
4	194.9	1.33E+02
5	363.0	7.92E+01
6	333.3	6.55E+01
7	181.4	5.53E+01
8	166.6	4.70E+01
9	433.6	2.21E+01
10	329.1	1.96E+01
11	256.3	1.92E+01
12	294.5	1.88E+01
13	427.3	1.87E+01
14	320.6	1.66E+01

Pulsations will lead to **CAVITATION** when operating at resonance

Short-term solution

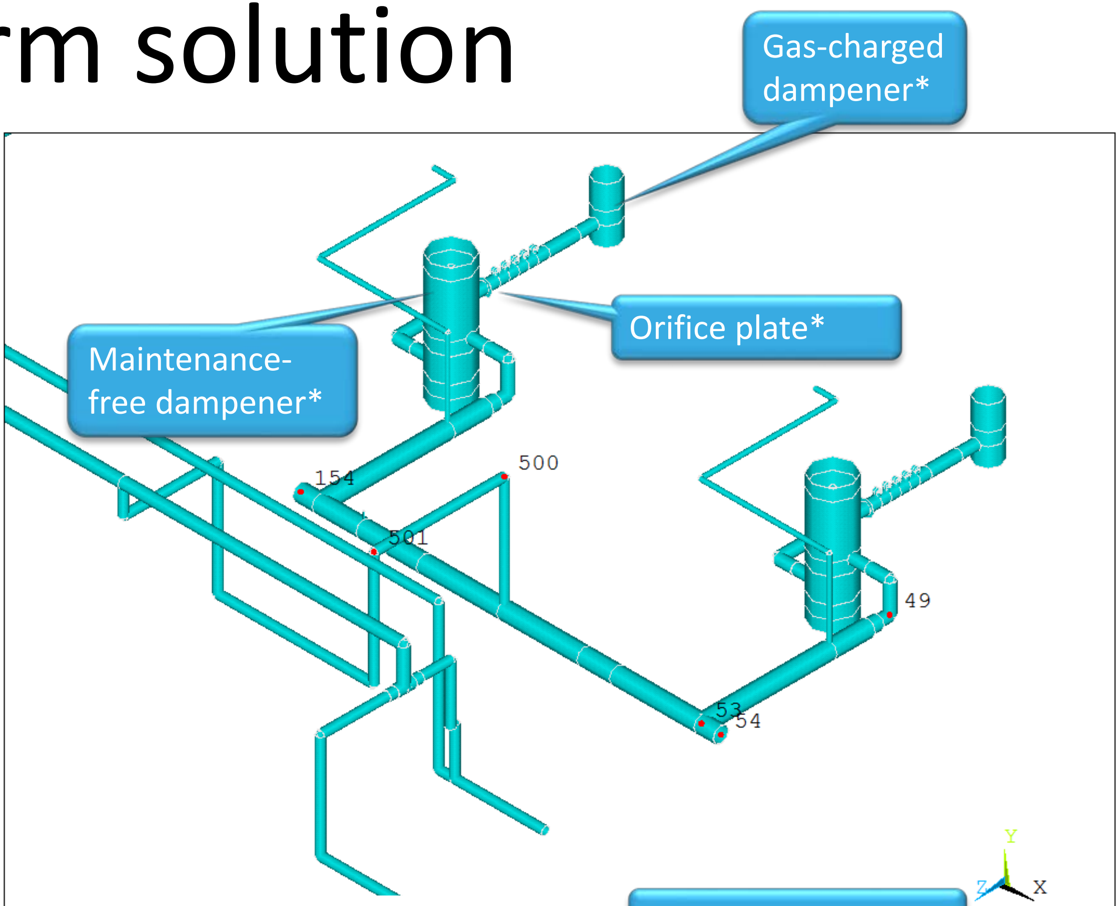


Shaking forces (lbf peak-peak)

Pipe span	Existing (full speed 200 – 400 rpm)	Existing (650 bbl/hr 200 – 218.5 rpm)	Short-Term (full speed 200 – 400 rpm)	Short-Term (650 bbl/hr 200 – 218.5 rpm)
45 – 48	5,100	1,020	1,820	253
48 - 49	7,220	1,580	1,880	163
49 - 53	14,900	4,920	2,970	410
52 - 200	1,140	275	214	35.1
200 - 201	1,440	309	187	47.5
201 - 202	6,980	139	309	53.8
54 - 154	20,800	12,100	2,890	462
57 - 500	2,410	524	474	43.3
500 - 501	3,180	695	807	55.9
501 - 502	2,150	473	323	45.1

Long-term solution

Allows for continuous operation over the complete speed range of 200-400 rpm



Suction system

Site inspection - NDT

- Predicted forces in pulsation study are well above acceptable limits. Operator decided to evaluate piping integrity.
- NDT completed on piping and found 20 cracks

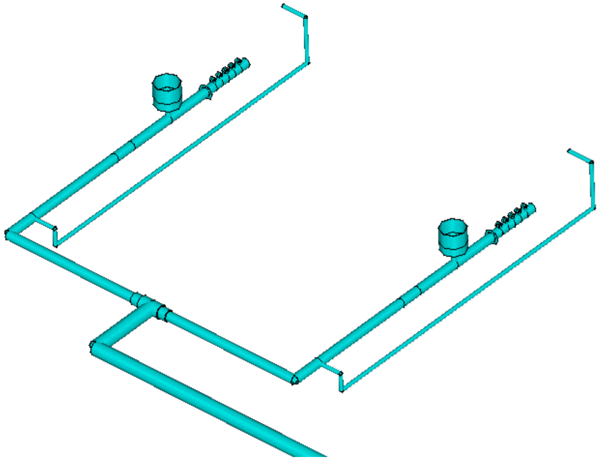
UNITS SHUT DOWN

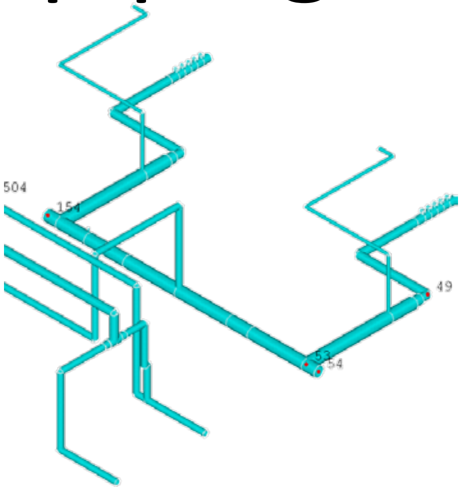
Replace and reroute piping

- Operator had to replace most piping due to cracks
- Improve piping layout and supporting
  - Reduce number of elbows
  - Avoid elevated piping (stiffness proportional to $height^3$)
  - Avoid unsupported elbows
  - Avoid redundant small-bore connections

Replace and reroute piping

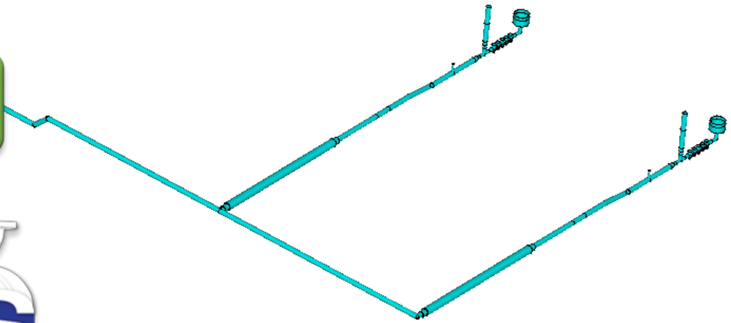
New suction layout

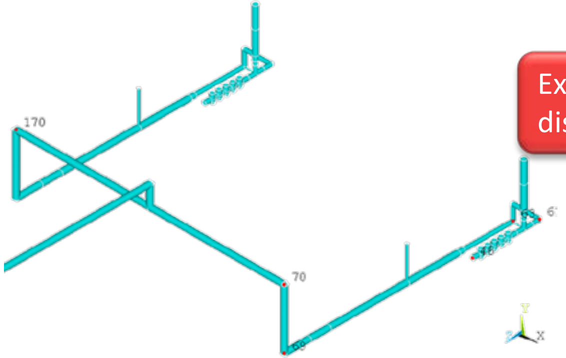




Existing suction layout

New discharge layout





Existing discharge layout

Recommendations with new layout

Suction

- 3.125” ID orifice plates (pump manifold inlets & vessel outlet)
- 20-gallon gas-charged dampeners
- Vessel 30” OD x 72” S/S acoustic filter

Discharge

- Secondary 5-gallon gas-charged dampeners
- 10” XS line expansion

Suction system forces (lbf peak-peak)			
Design #	**20 gal dampener**	**20 gal dampener + Orifice plates**	**20 gal dampener + orifice plates + vessel**
Lateral piping	15,000	2,600	1,200
Bypass piping	450	500	100
Pump header	19,000	3,500	500
Main header	1,800	1,800	400
Header pipe span	1,760	950	184

Pressure drop element	Orifice plate (manifold inlet)	Orifice plate (Vessel outlet)	Internal choke tube	External choke tube
Pressure drop (psi) @ 400 RPM	2.46	2.44	0.16	0.05

Design implementation

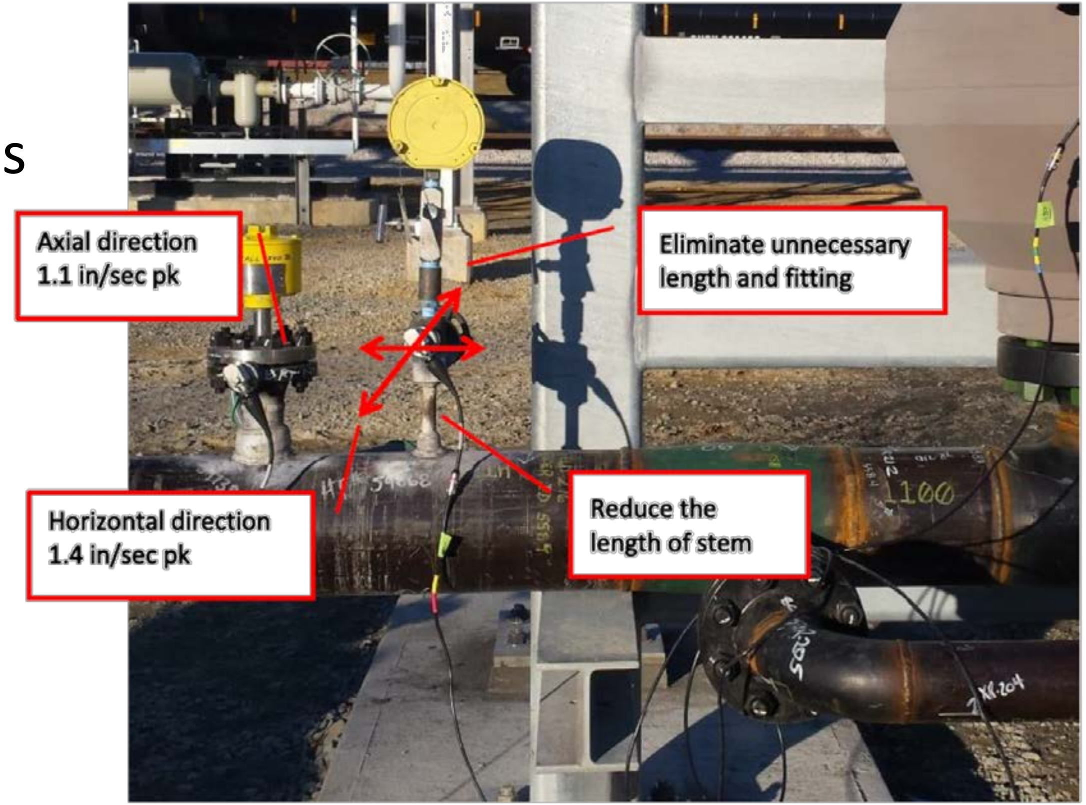
- Add suction gas charged dampener
- Add orifice plate at suction pump manifold
- Add discharge secondary gas charged dampener

Design implementation

- Add suction gas charged dampener
- Add orifice plate at suction pump manifold
- Add discharge secondary gas charged dampener
- Add proper support

Follow-up Vibration Measurement

- Main piping vibration is reduced to acceptable levels
- Small-bore and instrumentation piping required remedial work



Conclusions

- Pulsation study **<u>MUST</u>** be completed on pumps > 50 HP (37 kW)
- Orifice plates can be crucial to reducing pulsations and avoiding cavitation
- Interaction between multiple units should be considered
- Pipe routing and support is vital to mitigating vibrations
- Field test small bore piping and instrumentation

Questions