ASME STANDARDS TECHNOLOGY, LLC

STP-PT-093

DEVELOPMENT OF B31.8 CODE CONTENT TO ADDRESS COMPRESSOR PIPING VIBRATION AT THE DESIGN, STARTUP AND OPERATIONAL STAGES

STP-PT-093

DEVELOPMENT OF B31.8 CODE CONTENT TO ADDRESS COMPRESSOR PIPING VIBRATION AT THE DESIGN, STARTUP AND OPERATIONAL STAGES

Prepared by:

George Antaki, PE, Fellow ASME
Becht Engineering Co.

ASME STANDARDS TECHNOLOGY, LLC

Date of Issuance: July 31, 2021

This publication was prepared by ASME Standards Technology, LLC ("ASME ST-LLC") and sponsored by The American Society of Mechanical Engineers ("ASME").

Neither ASME, ASME ST-LLC, the author, nor others involved in the preparation or review of this publication, nor any of their respective employees, members, or persons acting on their behalf, makes any warranty, express or implied, or assumes any legal liability or responsibility for the accuracy, completeness, or usefulness of any information, apparatus, product or process disclosed, or represents that its use would not infringe upon privately owned rights.

Reference herein to any specific commercial product, process, or service by trade name, trademark, manufacturer, or otherwise does not necessarily constitute or imply its endorsement, recommendation, or favoring by ASME ST-LLC or others involved in the preparation or review of this publication, or any agency thereof. The views and opinions of the authors, contributors and reviewers of the publication expressed herein do not necessarily reflect those of ASME ST-LLC or others involved in the preparation or review of this publication, or any agency thereof.

ASME ST-LLC does not take any position with respect to the validity of any patent rights asserted in connection with any items mentioned in this document, and does not undertake to insure anyone utilizing a publication against liability for infringement of any applicable Letters Patent, nor assumes any such liability. Users of a publication are expressly advised that determination of the validity of any such patent rights, and the risk of infringement of such rights, is entirely their own responsibility.

Participation by federal agency representative(s) or person(s) affiliated with industry is not to be interpreted as government or industry endorsement of this publication.

ASME is the registered trademark of The American Society of Mechanical Engineers.

ASME Standards Technology, LLC
Two Park Avenue, New York, NY 10016-5990

ISBN No. 978-0-7918-7474-5

TABLE OF CONTENTS

Foreword vi
1 Project Objective 1
1.1 Background 1
1.2 Objective 1
1.3 Challenge 1
1.4 Report Outline 2
1.5 Steady-State Vibration 2
1.6 Gas Compressor Station Piping 3
2 Recommendations for B31.8 5
2.1 Option 1 – API-Compliant 5
2.1.1 Option 1 Recommendation 5
2.1.2 Option 1 Commentary 6
2.2 Option 2 – Cause Prevention-Based Design 7
2.2.1 Option 2 Recommendation 7
2.2.2 Option 2 Commentary 10
2.3 Option 3 – Fully Qualitative 10
2.3.1 Rules of Good Practice 10
2.3.2 Opinion 3 Commentary 11
2.4 Option 4 – Short and Long Term Approach 11
3 Current Piping Vibration Rules 12
3.1 Overview 12
3.1.1 Current Methods and Criteria 12
3.1.2 Sources of Knowledge 17
3.1.3 Industry Codes, Standards and Guidelines 18
3.2 ASME B31 Vibration Rules 18
3.2.1 Current B31 Vibration Stress-Based Rules 18
3.2.2 Current B31 Vibration Design and Fabrication Detail Restrictions 19
3.3 ASME O&M Vibration Rules 20
3.4 API Vibration Rules 21
3.5 Gas Machinery Research Council Vibration Rules 21
3.6 UK Energy Institute Vibration Rules 21
4 Sources of Vibration and Prevention in Design 22
4.1 Mechanical-Induced Excitation 23
4.1.1 Vibration Through Equipment Nozzle 23
4.1.2 Vibration Through Support Structure 24
4.1.3 Vibration of Branch Through Header 24
4.2 Flow-Induced Excitation (FIE) 25
4.2.1 Periodic Pressure Fluctuations 25
4.2.2 Broadband Pressure Fluctuations 28
4.3 Resonance 32
4.3.1 Acoustic Resonance 32
4.3.2 Structural Resonance 33
5 Vibration During Commissioning or Operation 35

5.1 Document the Observed Vibration 35
5.2 Document the Operating Conditions....................35
5.3 Develop and Implement a Monitoring Plan....................35
5.3.1 Hydraulic Monitoring 36
5.3.2 Structural Monitoring 36
5.3.3 Monitoring Report 36
5.4 Evaluate the Severity of the Observed Vibration....................37
5.4.1 Beam mode Vibration....................38
5.4.2 AIV Shell mode Vibration....................45
5.5 Determine the Potential Cause(s) of Vibration 47
5.6 Proposed Solutions to Piping Vibration....................47
5.6.1 Hydraulic Prevention Solutions 47
5.6.2 Mechanical Mitigation Solutions....................47
Annex A: Current B31 Vibration Rules....................49
A.1 B31.8 2016 Vibration Rules 50
A.2 B31.1 2016 Vibration Rules for Metallic Piping 51
A.3 B31.3 2016 Vibration Rules for Metallic Piping 54
A.4 B31.4 2016 Vibration Rules for Metallic Piping 57
Annex B: Current API Standards for Piping Vibration 60
B.1 Introduction....................61
B.2 API 618 Reciprocating Compressors....................61
B.2.1 Selection of the API 618 Design Approach....................61
B.2.2 Step-by-Step Vibration-Prevention Design 61
B.2.3 Foundation Analysis 63
B.3 API 674 Positive Displacement Pumps-Reciprocating....................64
B.4 API RP-688 Pulsation and Vibration Control....................65
B.5 API RP 1111 Offshore Pipelines....................65
Annex C: Bibliography of GMRC Publications Related to Vibration....................66
Annex D: B31.8 Paragraph 843.4 (2016) 72

LIST OF TABLES

Table 2-1: Design Approach as a Function of Discharge Pressure and Rated Power....................6
Table 4-1: Likelihood of Failure (LOF) from FIV for Positive Displacement Compressor....................26
Table 4-2: Likelihood of Failure (LOF) from Stalling of a Centrifugal Compressor 27
Table B-1: Design Approach as a Function of Discharge Pressure and Rated Power....................61

LIST OF FIGURES

Figure 1-1: Simplified Sketch of a Natural Gas Compressor Station 3
Figure 1-2: Analysis Model of a Gas Compressor Station 4
Figure 3-1: Piping Vibration at the Design Stage 12
Figure 3-2: Piping Vibration at the Startup (Commissioning) Stage 15
Figure 3-3: Piping Vibration at the Startup (Commissioning) Stage 17
Figure 4-1: Sources of Piping Vibration....................22
Figure 5-1: Current Vibration Velocity Amplitude Criteria, Superimposed, Color-Coded....................38

Figure 5-2: Reproduces Table 1 of Draft API 579-1/ASME FFS-1 Part 16, November 2019 42
Figure 5-3: Reproduces Figure 14 of Draft API 579-1/ASME FFS-1 Part 16, November 2019..........43
Figure 5-4: Carucci-Mueller Criterion for AIV 45
Figure 5-5: Eisinger-Francis Criterion for AIV 46

FOREWORD

The author acknowledges, with deep appreciation, the activities of ASME staff and volunteers who have provided valuable technical input, advice and assistance with review of, commenting on, and editing of, this document.

Established in 1880, the ASME is a professional not-for-profit organization with more than 100,000 members and volunteers promoting the art, science and practice of mechanical and multidisciplinary engineering and allied sciences. ASME develops codes and standards that enhance public safety, and provides lifelong learning and technical exchange opportunities benefiting the engineering and technology community. Visit https://www.asme.org/ for more information.

ASME ST-LLC is a not-for-profit Limited Liability Company, with ASME as the sole member, formed in 2004 to carry out work related to new and developing technology. ASME ST-LLC's mission includes meeting the needs of industry and government by providing new standards-related products and services, which advance the application of emerging and newly commercialized science and technology, and providing the research and technology development needed to establish and maintain the technical relevance of codes and standards. Visit http://asmestllc.org/ for more information.

1 PROJECT OBJECTIVE

1.1 Background

ASME B31.8 Record Number 10-974 was opened to address the following concern:

*"The current text of this Paragraph [843.4.1 Compressor Station Piping**] does not address current practice regarding acoustic analysis, mechanical studies, and fatigue/cyclic loading limitations in reciprocating compressor station piping. These design considerations are essential to quantify and evaluate the performance of a piping system that will be connected to a reciprocating machine."*

** Note: Para. 843.4.1 is reproduced in Annex D: of this report.

In order to address this shortcoming, ASME ST-LLC issued the following scope of work as ASME ST-LLC Project 0156 (Mr. Dan Andrei, ASME ST-LLC Project Manager):

"This project will provide the technical basis for incorporation into ASME's B31.8 Gas Transmission and Distribution Piping Systems code ("B31.8") requirements that can be used to identify and mitigate design and operational issues that if not addressed can lead to catastrophic failure of compressor station piping systems. The work resulting from this project will provide the basis for anticipated substantive technical improvement to B31.8 that address compressor station design, specifically fatigue issues in compressor station piping as well as capture current industry practices so that B31.8 can remain relevant."

1.2 Objective

The objective of this project is to propose to ASME B31.8 design rules to prevent damaging vibration of reciprocating and centrifugal gas compressor station piping.

As a result of the peer review, the scope added vibration assessment during the startup (commissioning) stage, and in operation.

In accordance with the scope of work of ASME ST-LLC Project 0156, the work resulting from this project is to be in the form of proposed inserts to ASME B31.8 Code for Gas Transmission and Distribution Piping Systems.

1.3 Challenge

There are challenges in a project like this one:

1) Achieving a synthesis of the multitude of Codes, Standards, and publications on the subject, which are summarized in Chapter 3 "Current Piping Vibration Rules". In other words, rather than develop new design rules to prevent piping vibration, build upon the current best practices:

- ASME
  - ASME B31 Pressure Piping Code, in particular B31.1 (power, 2016), B31.3 (process, 2016), B31.4 (liquid pipelines, 2016), and B31.8 (gas pipelines, 2016).
  - ASME Operation and Maintenance of Nuclear Power Plants, Part 3, Vibration Testing of Piping Systems (current edition is 2017). Referred to as "OM-3" in this report.

- API
  - API 617 Standard, Axial and Centrifugal Compressors and Expander-Compressors (current edition is 2014).
  - API 618 Standard, Reciprocating Compressors for Petroleum, Chemical, and Gas Industry Services (current edition is 2007 with 2009 and 2010 errata).
  - API RP-688 Recommended Practice, Pulsation and Vibration Control in Positive Machinery Systems for Petroleum, Petrochemical, and Natural Gas Industry Services (current edition is 2012).

- GMRC
  - Gas Machinery Research Council, Pipeline Research Council International, and Southwest Research Institute, Design Guideline for Small Diameter Branch Connections.
  - GMRC publications and research projects listed in Annex C:, provided for information.

- UK Energy Institute
  - An international guideline that has seen broad application across industries, is the "Guidelines for the Avoidance of Vibration Induced Fatigue Failure in Process Pipework" by the Energy Institute of the UK Health & Safety Executive. The second edition is dated January 2008, and is referred to as "EI Guideline" in this report.

Note: When referring to API Standards and Recommended Practices, ASME B31 Codes, and the EI Guideline, the editions listed above apply.

2) Proposing B31.8 rules that capture this vast knowledge and that are current, correct, clear, and concise, without either (1) being overwhelming in their complexity, or (2) being over-simplified to the point of becoming incomplete.

3) Identifying differences between current Codes and standards regarding the assessment of piping and pipeline vibration and, in case of differences, attempting to determine the best option.

4) While the main focus of this report is vibration prevention at the design stage, the report also addresses in Chapter 5 steps to take when vibration is identified during commissioning (start-up), or later in the operating life of the piping system.

1.4 Report Outline

The following outline covers the topics specified in ASME ST-LLC Project 0156.

Chapter 1 – Project Objective
Chapter 2 – Recommendations for B31.8
Chapter 3 – Current Piping Vibration Rules
Chapter 4 – Sources of Vibration and Prevention in Design
Chapter 5 – Vibration During Commissioning or Operation

Background and bibliographies are provided in Annexes.

1.5 Steady-State Vibration

This report addresses steady-state vibration. **Steady-state vibration** is vibration that occurs continuously under certain operating modes of the system. Steady-state vibration is a term used in contrast to **transient vibration** caused by short-duration events.

Transient vibration is defined in a consistent manner in draft Part 16 of API-579-1/ASME FFS-1 as well as in ASME Operation and Maintenance of Nuclear Power Plants (O&M), Part 3 "Vibration Testing of Piping Systems" as *"transient vibrations: vibrations that occur during relatively short periods of time and result in less than 106 stress cycles. Examples of transient sources of vibration are pump [and compressor] actuation and pump [and compressor] switching, rapid valve opening or closing, and safety relief valve operation."*

1.6 Gas Compressor Station Piping

Figure 1-1 (simplified illustration) and Figure 1-2 (numerical simulation model) provide the general arrangement of a natural gas compression station, either for gathering or for transmission pipelines.

The piping system in a natural gas compressor station includes the following main equipment and components:

- The main gas pipeline entering the transmission stations (typically 20 to 48 in. pipeline), or station yard piping for gathering stations (typically 6 to 20 in. piping)
- Scrubbers and filters to extract condensate and solid deposits and particulates
- Suction pulse bottles
- Compressors, in parallel or in series (stages). Gas-powered or electrically-powered
- Discharge pulse bottles
- Cooling to dissipate the heat from compression
- Drying to separate the liquid drop-out
- Ancillary piping for over-pressure protection (safety valves), bypass, lube oil, cooling water and other utilities, gas metering equipment, filtration systems, odorization system; small bore vents and drains; and instrumentation piping or tubing

All these pipes, main line and smaller pipes, and small bore connections, can be subjected to flow-induced or mechanically-induced vibration.

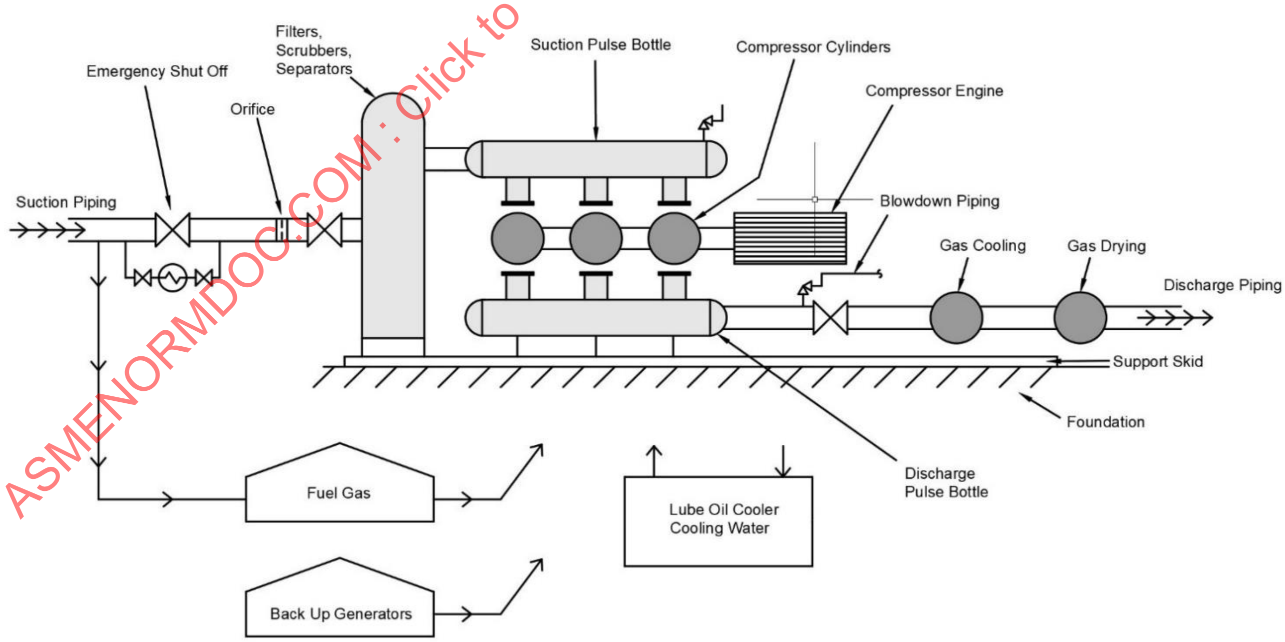

Figure 1-1: Simplified Sketch of a Natural Gas Compressor Station

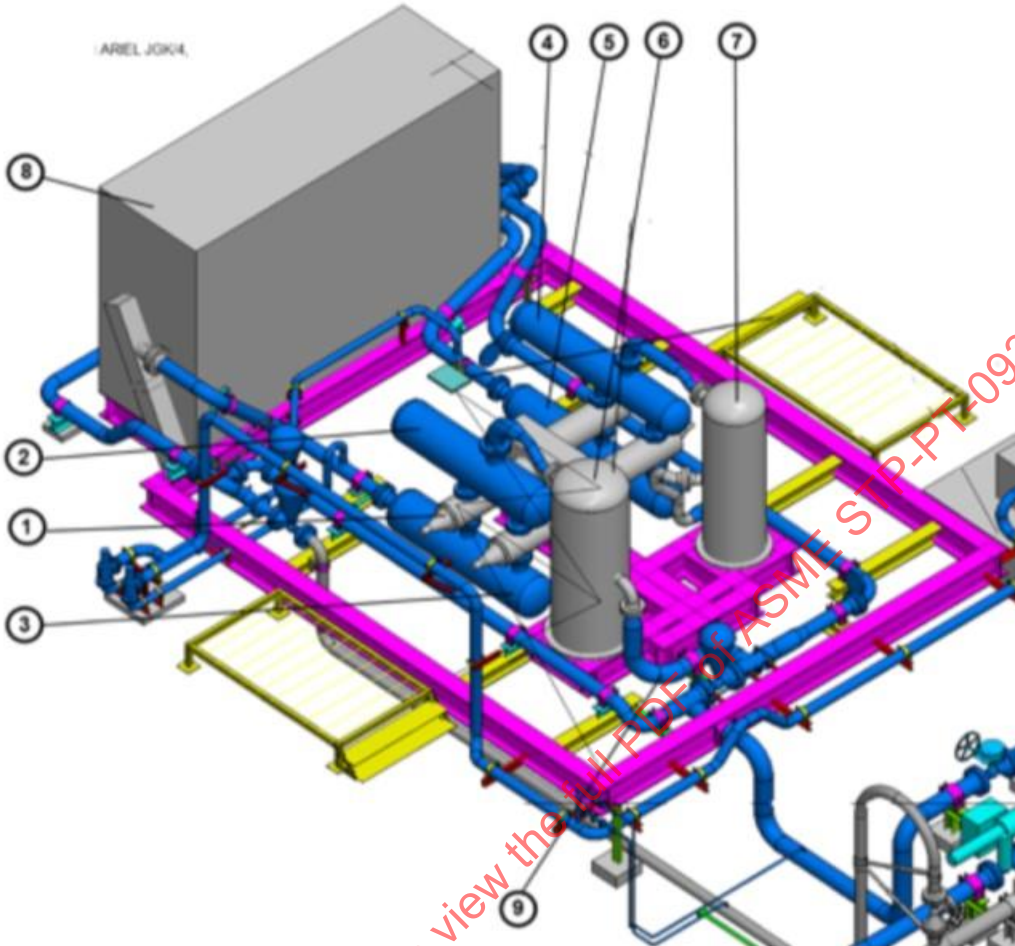

Figure 1-2: Analysis Model of a Gas Compressor Station

(1) Reciprocating compressor
(2) Anti-pulsation bottle suction side
(3) Anti-pulsation bottle discharge side
(4) Anti-pulsation bottle suction side
(5) Anti-pulsation bottle discharge side
(6) Gas filter, for the first stage of compression
(7) Gas filter, for the second stage of compression
(8) Gas coolers IC & AC – intercooler and aftercooler
(9) Pipelines

Source: ASME PVP2018-84565, Experiences in Developing a Practical Algorithm of Identification and Attenuation of Pressure Pulsation and Piping Vibration in Gas Reciprocating Compressor Plants and Other Systems, Maciej Rydlewicz and Wojciech Rydlewicz, July 2018.

2 RECOMMENDATIONS FOR B31.8

2.1 Option 1 – API-Compliant

2.1.1 Option 1 Recommendation

In Paragraph A842.2.5 "Design Against Fatigue", add the following:

Piping in gas compressor stations shall be designed to minimize vibration by following the rules of:

- *API Standard 617 "Axial and Centrifugal Compressors and Expander-Compressors".*
- *API Standard 618 "Reciprocating Compression for Petroleum, Chemical, and Gas Industry Services", in particular Chapter 7.*

API 618 for reciprocating compressors takes a graded approach to the design of compressor station piping ((DA-1 design by rule, DA-2 design with acoustic analysis, and DA-3 design with acoustic and mechanical analyses). While API 617 for centrifugal compressors does not address to the same level of detail the design of compressor piping systems, a graded approach similar to API 618, applies to centrifugal compressors, and is to be established by agreement between the designer and the owner.

The design analyses for compressor station piping shall address the various sources of excitation:

1) Flow-induced Excitation:
1.1 Periodic excitation:
- *a) Pressure pulsing from compressor piston or vane passing frequencies.*
- *b) Vortex shedding through flow restrictors or branch openings.*

1.2 Broadband excitation:
- *a) Flow turbulence from high kinetic energy.*
- *b) Acoustic-induced excitation of shell modes.*

2) Mechanical-induced excitation:
2.1 Excitation through nozzle.
2.2 Excitation through supports.
2.3 Excitation through header (small bore connections).

The design of compressor station piping should address the recommended practices of the following standard:

- *API Recommended Practice 688 "Pulsation and Vibration Control in Positive Displacement Machinery Systems for Petroleum, Petrochemical, and Natural Gas Industry Services", in particular Chapters 3 and 4.*

The designer shall provide adequate stiffness and/or damping of the piping system to reduce vibration while not exceeding allowable stresses in piping, supports, and equipment nozzle loads under the range of applicable operational and postulated loads. In addition, physical limitations and layout constraints may necessitate modifications to the pulsation control approach and pipe support plans. Therefore, the design of compressor station piping is an iterative process.

2.1.2 Option 1 Commentary

Option 1 has the advantage of linking B31.8 to API 618 and API 688, which are the current best practice in compressor piping vibration-prevention design, without being redundant. The two API standards would become referenced standards in B31.8 Mandatory Appendix A "References". Like other API references currently in B31.8 Appendix A (API 1102, 1104, 1111, etc.), when the B31.8 edition is updated, the latest edition of API 618 and API 688 would be reviewed and the decision made to update the B31.8 reference if appropriate.

Annex B: of this report summarizes the key vibration-related rules of API 618 and API 688. A key point is the **three-tier approach** to anti-vibration design in API 618, which is illustrated in the following Table:

Table 2-1: Design Approach as a Function of Discharge Pressure and Rated Power

Absolute Discharge Pressure	Rated Power per Cylinder		
	kW/cyl < 55 hp/cyl < 75	55 < kW/cyl < 220 75 < hp/cyl < 300	kW/cyl > 220 hp/cyl > 300
P < 35 bara P < 500 psia	DA-1	DA-2	DA-2
35 bara < P < 70 bara 500 psia < P < 1000 psia	DA-2	DA-2	DA-3
70 bara < P < 200 bara 1000 psia < P < 3000 psia	DA-2	DA-3	DA-3
200 bara < P < 350 bara 3000 psia < P < 5000 psia	DA-3	DA-3	DA-3

From API Standard 618, 5th edition, Table 6

- Design Approach 1 (DA-1) relies on the use of suction and discharge pulse bottles, or acoustical filters, and rules of good practice.
- Design Approach 2 (DA-2) calls for a compressor-piping acoustic analysis of the system to prevent acoustic resonance between pressure pulse frequencies and the acoustic natural frequencies of the piping system.
- Design Approach 3 (DA-3) calls for acoustic analysis as in DA-2 and, in addition, a structural analysis. API 618 calls for a two-tier approach to structural analysis at the design stage:

  (a) A modal analysis of the piping system to verify that the structural natural frequencies of the piping systems are not in resonance with the acoustic natural frequencies of the system.

  (b) A full displacement-stress response analysis which would be performed if (a) cannot be met, i.e. "*… When the excitation frequency separation margins or the shaking force amplitude guidelines for pulsation suppression devices cannot be met, a forced-mechanical-response analysis of the compressor mechanical model to the pulsation-induced forces and cylinder-gas forces shall be performed." (API 618 Section 7.9.4.2.4.3).*

In practice for pipeline compressor stations, Design Approach 1 and possibly Design Approach 2 are used at conceptual and front end engineering design (FEED) to approximate the size of the Pulsation Suppression Devices (PSDs) for estimating and preliminary layouts. For Procurement and final design, Design Approach 3 is common, unless it is a small compressor as shown in Table 2-1 for DA-1. The investment in the DA-3 study pays for itself compared to the costs of mitigation.

Acoustic resonance and structural resonance are addressed in Section 4.3.

Caution: The structural analyses, in either case (a) or (b), are a challenge because the structural modal analysis must correctly reflect the stiffness of the system, i.e. the correct stiffness of the supports and the proper treatment of the support-pipe gaps and frictions, which is addressed in Section 3.1.1.1. This imposes a close control on the quality of installation of the piping-supports system, and a final as-built reconciliation. Other factors that affect the structural modal analysis predictions are the flexibilities of bends and branch connections which must be consistent with the flexibility factors in Table E-1 of ASME B31.8, 2016.

API-617 is listed to include axial and centrifugal compressors. At this time, API-617 addresses in much detail shaft and casing vibration but does not address piping vibration. The reference is added in case, in the future, API-617 addresses piping vibration, in particular acoustic-induced vibration (AIV) in large D/t piping.

If this option is adopted, it will have to be kept up-to-date as the referenced API standards are updated.

The proposal includes a caution for the need to strike a balance between the competing needs of vibration prevention and attenuation, which require a stiffer and damped design, and other design loads, such as thermal expansion, which require more flexibility, and the iterative nature of the design process. Good guidance for a balanced design approach can be found in "Mechanical, Stress and Flow Considerations for Piping Design of Centrifugal Compressors", B.A. White et. al., 47th Turbomachinery and 34th Pump Symposia, Houston, TX, September 2018.

In addition to ASME B31.8 Code considerations, the design has to strike a balance between the permanent pressure drop added by the vessels, orifices, etc., that provide pulsation energy control and piping stiffness or flexibility with the capital and maintenance costs of such components.

This is my preferred and recommended Option.

2.2 Option 2 – Cause Prevention-Based Design

2.2.1 Option 2 Recommendation

Create a new B31.8 non-mandatory appendix in accordance with Chapter 4 of this report. The starting point for this B31.8 Non-Mandatory Appendix "X" would read as described hereunder. It should include Figure 4-1, and would be improved as it undergoes ASME Committee peer reviews.

Note, in the following proposed rules, we use "should" instead of "shall" because it is non-mandatory guidance. ASME B31.8 Committee may decide to convert it to a mandatory Appendix, in which case several of the "shoulds" would become "shalls". Some Code-Standard definitions of "should":

- Definition in API 618 Foreword *"Should: As used in a standard, "should" denotes a recommendation or that which is advised but not required in order to conform to the specification."*
- Definition in B31.8 Para. 805.2.7 *"should, should not, or it is recommended: used to indicate that a provision is not mandatory but recommended as good practice."*
- ASME B31.3 Para. 300.2 *"should: a term that indicates a provision is recommended as good practice but is not a Code requirement."*

B31.8 Non-Mandatory Appendix "X" Vibration Reduction Guidance for Gas Compressor Stations

The guidance provided in this Appendix is intended to minimize the likelihood of damaging vibration of compressor station piping systems and the main pipeline. In addition to this guidance, the prevention of damaging vibration includes the following considerations:

- *Hydraulic sizing of the compressor station equipment, piping, and components to achieve the expected flows at reasonable flow velocities.*
- *Selection, sizing, and installation of pulsation bottles or pulsation filters.*
- *Vibration monitoring during initial commissioning of the compressor station.*
- *Periodic monitoring and maintenance of the compressor assembly to prevent excessive machine vibration.*
- *Periodic monitoring of the piping systems to prevent damaging vibration caused by changes in flow condition or aging and degradation of the equipment.*

X.1 Reducing Mechanically-Induced Vibration

X.1.1 Equipment-Driven Vibration

In order to reduce the likelihood of mechanically-induced piping vibration caused by the vibration of compressor station equipment (such as the compressors, filters, scrubbers, separators, pulsation bottles, and cooling and drying units), the following guidance should be considered:

- *Equipment foundations should be designed to maintain structural vibration within the limits specified by the equipment manufacturer.*
- *The equipment anchorage to the concrete foundation, and the stiffness of the load path from the compressor to the foundation, should be designed in accordance with margins consistent with structural codes such as American Institute of Steel Construction (AISC) Manual of Steel Construction for the design of the steel support structure and American Concrete Institute (ACI) Appendix D "Anchoring to Concrete" for the concrete anchor bolts attaching the steel frames to the concrete walls and floors.*
- *The design should include a sufficient number of stiff pipe supports and stiff support structures to prevent the machine vibration to be amplified through the pipe supports and support structures. Two criteria provided in API 618 can be applied to achieve this objective (see Para. 4.2.1.1.5), and would apply to the beam mode frequencies as well as the shell mode frequencies:*
    - *The minimum mechanical natural frequency of any compressor or piping system element shall be designed to be greater than 2.4 times maximum rated speed.*
    - *The predicted mechanical natural frequencies shall be designed to be separated from significant excitation frequencies by at least 20%.*
- *In addition to design, the compressor should be pre-operationally tested and checked for acceptable vibration.*

X.1.2 Header-Driven Vibration

In order to reduce the likelihood of mechanically-induced vibration of small bore branch lines cantilevered off headers, such as vents and drains, the following guidance should be considered:

- *Threaded fittings should be avoided.*

- *Swage-type, and mechanically-joined fittings should be avoided, unless they have been qualified for vibration environment of the magnitude expected and have a stress intensification factor below 1.5.*
- *Pipe-on-pipe welded branch connections should be avoided. Instead, tees or, better, 45-degree tee connections with a smooth rounded interior profile should be used, with the applicable flexibility factor and stress intensity factor in design.*
- *The header pipe thickness should be sch.80 or thicker or be reinforced to achieve an equivalent thickness.*
- *The size of the small bore branch should not be less than NPS 1.*
- *Cantilevered vents and drains, without a tie-back to the header, should not exceed 3 ft and should have a natural structural frequency (including the weight contribution of in-line valves, flanges, and fittings) above the compressor first mode running frequency, and not within ±20% of the higher modes of the compressor running frequency.*

X.2 Reducing Flow-Induced Vibration

X.2.1 Periodic Pressure-Induced Vibration

X.2.1.1 Prevention of Acoustic Resonance: The compressor system should be acoustically analyzed to prevent acoustic natural frequencies within ±20% of the compressor pressure pulsing frequency harmonics $f_{\text{pressure pulsing}}$

$$f_{pressure\ pulsing} = n\frac{N\ B}{60}$$

Where $f_{pressure\ pulsing}$ *= frequency of pressure pulses (Hz); n = mode number (1, 2, 3, etc.); N = machine speed (RPM); B = number of vanes in a centrifugal machine, number of plungers or pistons in a reciprocating machine; 60 = factor to convert 1/minutes (RPM) to 1/seconds (Hz).*

Control valves and orifice plates should be designed to prevent vortex-shedding frequencies within ±20% of the acoustic natural frequencies of the system.

X.2.1.2 Prevention of Structural Resonance: The layout and support of the compressor suction and discharge piping should prevent natural structural frequencies equal to the fundamental and harmonic pressure pulsing frequencies $f_{pressure\ pulsing}$. *This design check would be by means of a modal structural analysis of the piping system to determine its structural frequencies and compare them to* $f_{pressure\ pulse}$. *The structural model should accurately reflect the piping configuration, the support stiffnesses, and should address pipe-to-support gaps, rattle points, and friction.*

Control valves and orifice plates should be designed to prevent vortex-shedding frequencies within ±20% of the structural frequencies of the system.

X.2.2 Broadband Pressure-Induced Vibration

The compressor-piping system should be designed to operate at kinetic energies below 20,000 kgm/m.sec^2.

If the kinetic energy of the flow in the pipe exceeds 5,000 kgm/m.sec^2*, the gas mixing points should be sweeping, with an intersecting angle near 45 degrees, rather than 90 degrees.*

2.2.2 Option 2 Commentary

Option 2 is new; it is based on the vibration-prevention rules of API 618 and API 688 and the UK EI Guideline, but it structures them on the basis of the root causes of the vibration in Figure 4-1, where they apply to gas (rather than liquid) service. This new approach is described in Chapter 4. It has the advantage that, for the first time, the root-causes of vibration are presented in a **complete and structured manner** (Figure 4-1) and, for each cause of vibration, design-stage prevention measures are provided. This structure of root causes is lacking in current standards.

The ±20% margin from the excitation frequency to avoid excessive vibration is from API 618 (2010) Para. 6.7.3, 7.1.2.10, 7.5.4.14, and 7.9.4.2.and the EI Guideline (second edition) Para. 2.2, 2.3.2, and 2.3.3.

If there is more than one significant structural mode of vibration, the natural frequencies should preferably be halfway between resonant frequencies, which will minimize, but not eliminate, resonance effects.

The 20,000 kgm/m.sec^2 and 5,000 kgm/m/sec^2 threshold are from the EI Guideline Table T1-1.

The disadvantage of this Option 2 is that it is new. It will therefore take time for peer reviews, improvements through constructive comments, consensus, and finalization. The advantage of such a structured approach, if we persevere, is evident and will be useful to the owner, the designer, the field engineer, and the regulator.

Note: If this effort is generalized to include liquid service, it could be a new vibration-prevention and analysis standard, for example B31V, that would apply to all B31 Code books. It would possibly be coordinated by the ASME B31 Mechanical Design Technical Committee (MDC), and in liaison with ASME O&M Part 3 Committee, which is currently the other standard addressing piping vibration within ASME; and possibly API and the UK EI Guideline.

Note: At the time of this writing, API 579-1/ASME FFS-1 is circulating a draft Part 16 Assessment of Piping Vibration which addresses the fitness-for-service evaluation of observed vibration in piping systems.

If this Option 2 is adopted, it will have to be kept up-to-date by incorporating feedback that will result from its implementation, and evolving industry knowledge.

2.3 Option 3 – Fully Qualitative

2.3.1 Rules of Good Practice

Create a new B31.8 non-mandatory appendix that compiles the design-stage vibration-prevention rules of good practice of B31.1, B31.3, B31.4, and B31.8. These are listed in Annex A: of this report, and the actionable guidance is listed in Section 3.2. Based on Section 3.2 of this report, a recommended Non-Mandatory Appendix "Y" may read as described hereunder. Because the guidance in B31 (compiled in Annex A: and Section 3.2) is weak, improvements have been added here. Paragraph numbers have been kept to trace the origin of the rule. Paragraphs starting with 1 are from B31.1; starting with 3 are B31.3; starting with 8 are B31.8.

B31.8 Non-Mandatory Appendix "Y" – Piping Design Precautions to Reduce Damage from Vibration in Compressor Stations

(a) On small bore lines, flexible metal hose assemblies may be used to isolate or control vibration, or to compensate for misalignment (105.4, with improvements). If flexible connections or hose are used, the maintenance program must include inspection, maintenance and replacement program to address fatigue limits of flexible connections.

(b) *Socket welded piping joints should not be used for piping larger than NPS 2. For small bore piping, the profile of the socket weld should be twice as long (along the pipe) than it is wide (along the fitting) (111.3.1, with improvements).*
(c) *Threaded joints should not be used. More generally, joints with a stress intensification factor larger than 1.5 should not be used (114.2.1 and 841.1.9, with improvements).*
(d) *The design and installation of inserted instrument, control, and sampling devices shall be adequate to withstand the vibratory effects of the fluid flow (114.2.3).*
(e) *Brazed and soldered joints should not be used (117.3).*
(f) *Backing rings should be removed and the internal joint face ground smooth (311.2.4).*
(g) *Expanded joints, and flared, flareless, and compression type tubing fittings should not be used unless qualified for vibration service and have a stress intensification factor equal to or less than 1.5 (313 and 315.1 and U305.6, with improvements).*
(h) *The use of controlled bolting procedures that prevent bolt relaxation should be implemented (para. F309.1).*
(i) *Augmented nondestructive examination of pipe and attachment welds should be implemented during construction (853.1.7, with improvements).*

2.3.2 Opinion 3 Commentary

Option 3, which uses solely qualitative guidance, is the status-quo option. The advantage of this option is that it does not impose new requirements. The shortcoming of this option is that it is insufficient, which prompted the need for this report. It is not recommended if adopted by itself. It may be added to Option 1 (API-compliant) as described in Option 4.

2.4 Option 4 – Short and Long Term Approach

- In the short-term, Option 4 would adopt Option 1 (API-compliant), possibly with the qualitative guidance of Option 3.
- In the long-term, Option 4 would adopt Option 2 (the new cause-based design approach) after it has undergone B31 Committee reviews.

3 CURRENT PIPING VIBRATION RULES

3.1 Overview

3.1.1 Current Methods and Criteria

The current **strategy for the prevention or mitigation** of piping and pipeline vibration is illustrated in Figure 3-1 and outlined here.

3.1.1.1 Piping Vibration at the Design Stage

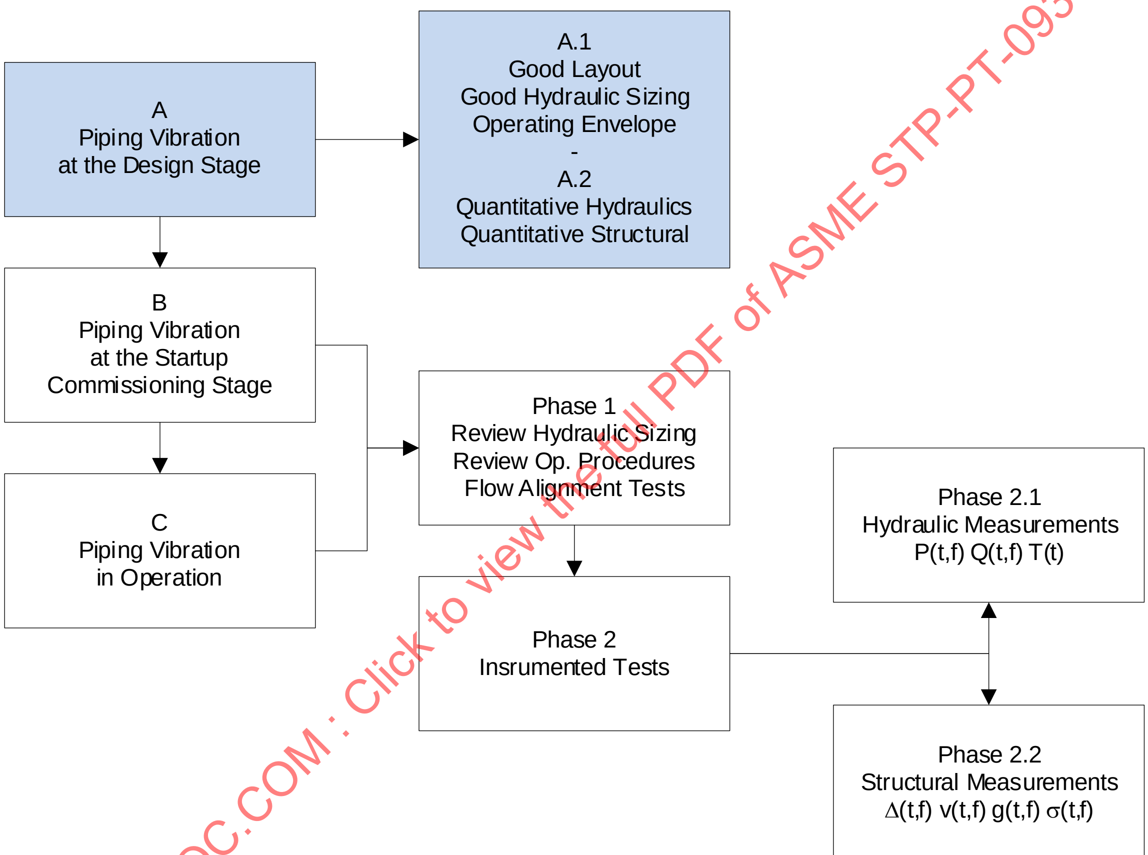

Figure 3-1: Piping Vibration at the Design Stage

Figure 3-1 **Block A**: For the prevention of piping vibration at the design stage, designers rely primarily on three design functions, listed in Figure 3-1 **Block A.1**:

1) **Good layout** of the piping system, which includes a multitude of design and layout decisions regarding the placement of equipment, in-line components, types of fittings, supports and structural dampers, pulsation dampers, slopes, drains and vents, instruments, etc. Layout is an art form not addressed in Codes or Standards, for which there are few written references, and which was perfected by experienced drafting staff at engineering companies. Over the last four decades the drafting profession has given way to a larger reliance on stress analysis and more recently on 3D modeling of the plant assets.

2) **Good hydraulic sizing**, which includes the thermohydraulic sizing of pumps, compressors, pipes, fittings, valves, heat exchangers, pulsation dampers, strainers, orifice plates, etc. to maintain flow rates at reasonable values, prevent cavitation, prevent slugging, prevent excessive vortex shedding, prevent excessive turbulence, etc. In other words, good thermohydraulic sizing to prevent the sources of vibration that will be described in Chapter 4.
3) Clear system thermohydraulic **operating envelopes** (limits on flow rates, pressures, and temperatures) which reflect the hydraulic sizing into the operating procedures for start-up, shutdown, normal and abnormal operation. In other words, the plant must be operated the way it was intended by the thermohydraulic design.

Figure 3-1 **Block A.2** points out that in a few cases the prevention of vibration is done at the design stage by numerical simulation. There are two aspects to numerical simulation:

1) Numerical hydraulic simulation to determine hydraulic flows, hydraulic forcing functions, acoustic natural frequencies, and potential acoustic resonance.
2) Numerical structural simulation using pipe stress analysis software, or general purpose finite element analysis (FEA) to determine stresses or strains for comparison to design limits.

Two industrial applications where numerical simulation is currently used at the design stage are:

1) Oil pipeline pumping stations and gas pipeline compression stations, where the rules of analysis are established primarily by API Standard 618, API Standard 674, API Recommended Practice (RP) 688, and API RP 1111.
2) Nuclear power plants reactor coolant system, including the reactor and steam generator vessel internals, where the rules of analysis are determined in part by the US Nuclear Regulatory Commission Standard Review Plan (NUREG-0800) Sections 3.9.2 and 3.9.5 for the reactor vessel internals, and Section 5.4.2.1 for the steam generator.

In both cases (pipelines and nuclear power plants), piping vibration has significant safety, environmental, and cost consequences, which justify the level of effort involved in hydraulic and structural numerical analysis at the design stage.

Apart from these two applications (pipelines and nuclear), there are two reasons why piping vibration is not currently numerically analyzed at the design stage (unlike other loads such as wind, seismic, or anticipated pressure transients which are part of a design analysis):

- The complexity of quantifying the possible sources of vibration and their magnitude and frequencies, as is evident in Chapter 4.
- The cost and complexity of detailed 3D computational shell and/or solid elements finite element analysis.

Users of modelling software must know and understand the limitations of the software and the impacts those limitations can have on the interpretation and subsequent use of the analysis results. Pipe stress analysis software (for beam mode analysis) and finite element analysis software (for beam mode and shell mode analysis) is not a replacement for good, experienced engineering.

An important consideration in the structural analysis of beam mode vibration of piping systems (Block A.2) is the adequacy of the modeling of the pipe supports in the piping system modal and stress analyses. This important question has been addressed in a GMRC project "Pipe Support Stiffness" dated 2015, and a related report "Shake, rattle and grow – empirical data on the effectiveness of vibration supports in a thermal growth environment" by M. Barabé et. al. (Wood), GMRC Gas Machinery Conference, 29 September-2

October 2019, San Antonio TX. There are some key points to keep in mind when modeling a piping-support system for modal analysis and subsequent stress-displacement analysis at the design stage:

1) The piping layout is usually well established in isometric or orthographic drawings, and in some cases three-dimensional solid models.
2) Pipe supports locations are either identified or the supporting structures are known so that the piping analyst can locate the pipe supports.
3) The stiffness of the pipe support assembly is a combination of three stiffnesses in series: (1) the pipe clamp stiffness, (2) the stiffness of the support members (rods, springs, struts, trunions, lugs, etc.), and (3) the stiffness of the backup structure (structural steel, embedment plate, etc.). For the purpose of performing a valid modal analysis the designer-analyst must enter in the model the correct value, or at least order of magnitude, of the combined stiffness of this support assembly. In modal analysis, it is incorrect to simply model the supports as “rigid”, which for most piping analysis software means they are modeled with a default stiffness in the order of 1E12 lb/in. API 618 5th edition, Para. P.3.2.1, provides an equation which “satisfies the minimum required support stiffness for all practical piping configurations with maximum acceptable spans”, i.e. the following stiffness will act as a vibration node point, i.e. zero displacement in the direction of action of the support:

$$min.K_s = C_{ks}\, A^{0.75}\, I^{0.25}\, {f_{nT}}^{1.5} \left(n - \frac{1}{n}\right)$$

   Where: C_{KS} is the constant dependent on support stiffness units (SI units: 1/130; USC units: 25); A is the pipe cross-sectional metal area in mm^2 (in.^{2}) = π/4 × (OD^2 – ID^2); I is the pipe cross-sectional area moment of inertia in mm^4 (in.^{4}) = π/64 × (OD^4 – ID^4); *OD* is the pipe outer diameter in mm (in.); *ID* is the pipe inner diameter in mm (in.); f_{nT} is the minimum transverse natural frequency in Hz; and n is the number of active supports (or n = 2 as a minimum). However, the Barabé et. al. paper (page 6) warns that this min.K_s is not sufficient.
4) The gaps (clearances) between the pipe and the support and along the support assembly along the load path. Typical pipe support gaps of 1/16 in. = 0.060 in. may be too large when be compared to the API limit of 20 mils = 0.020 in. peak-to-peak below 10 Hz for beam mode vibration. In this case, the vibration support may have to rely on clamps and support structures that are shimmed, and tightly bolted or welded. The designer must consider that bolted connections subjected to vibration can loosen over time, causing the vibration to become more severe.
5) The friction force and corresponding breakaway force between the pipe and the support, and along the support assembly, must be addressed in in the analysis model.
6) If the pipe-support contact area is not sufficiently wide, the potential for pivot-rotation of the pipe at the support must be addressed in the analysis model.

3.1.1.2 Piping Vibration at the Startup and Commissioning Stage

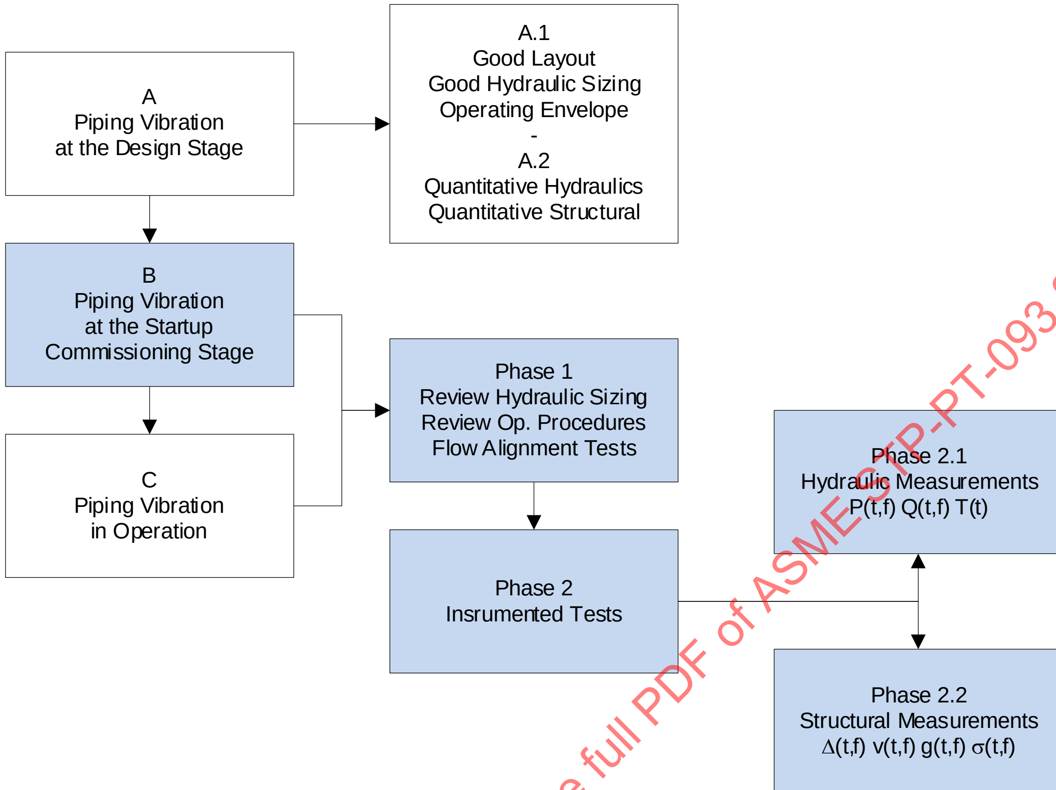

Figure 3-2: Piping Vibration at the Startup (Commissioning) Stage

Figure 3-2 **Block B**: It is common in the process, power, and pipeline industries to monitor a new piping system during startup and commissioning for evidence of vibration. In many cases a visual inspection during the various startup modes of operation is sufficient. In other cases, the lines are instrumented to measure hydraulic parameters (flow and pressure pulsations) and/or structural parameters (vibration displacement amplitude, velocity, and accelerations).

Vibration monitoring and troubleshooting at the commissioning and operational stage is addressed in Chapter 5.

3.1.1.3 Corrective Measures

Figure 3-2 Block **Phase 1**: If significant vibration is identified during startup, commissioning, or later in operation, the Phase 1 root-cause investigation consists of the following:

- Retrieve the hydraulic sizing calculations and review their correctness for the various operating modes.
- Review the operating procedures to verify that they correctly reflect the hydraulic design and are being followed correctly.
- Develop and implement a plan to explore different flow alignments and operating modes (valve alignments, pump-compressor operating sequences) to zero-in on the possible hydraulic causes of the observed vibration, which are described in Chapter 4.

3.1.1.4 Instrumented Testing

Figure 3-2 Block **Phase 2**: If the cause of vibration cannot be determined from the Phase 1 reviews and alignment tests, then an instrumented test becomes necessary, which will be Phase 2 of the root-cause investigation. There are two types of instrumented tests:

1) Figure 3-2 **Phase 2.1** Hydraulic measurements to determine flow rate and pressures versus time, using dynamic pressure transducers that can measure and record pressures at a rate sufficiently high to obtain an accurate waveform P(t), that can be converted to frequency domain P(f). The required recording frequency depends on the estimated pressure pulse frequency. Dynamic pressure sensors are available in the range of a few Hz to 250 kHz. Strain gages may also be used to indirectly measure the pressure pulse through the strain response in the pipe wall.

   This hydraulic data helps determine several parameters such as the location of maximum pressure and flow fluctuations, whether the vibration is broadband or periodic, the significant frequencies of the pressure signal, the presence of acoustic resonance, etc. This knowledge permits the engineer to determine the sources of vibration. Other useful instruments include a mechanic stethoscope to determine the sound at various locations along the piping systems, used with caution to avoid hearing damage, and video recordings with and without amplification.

2) Figure 3-2 **Phase 2.2** Structural measurements to obtain accelerations vs. time and frequency g(t,f), that can be integrated into velocities v(t,f) and displacements Δ(t,f). The structural measurements are compared to endurance limits, typically in the form of stress, displacement, or velocity limits. In some cases, strain gages are used to directly measure the pipe wall response to the shell mode excitation.

Where necessary, both hydraulic and structural data are recorded and correlated. CFD and FEA may or may not be required, depending on the complexity of the vibration and the consequence of vibration-induced fatigue failure. However, in practice, in many cases, Figure 3-1 Phase 2 has proceeded only with the structural data of Figure 3-2 Phase 2.2, without the benefit of the corresponding hydraulic data of Figure 3-1 Phase 2.1.

3.1.1.5 Piping Vibration in Operation

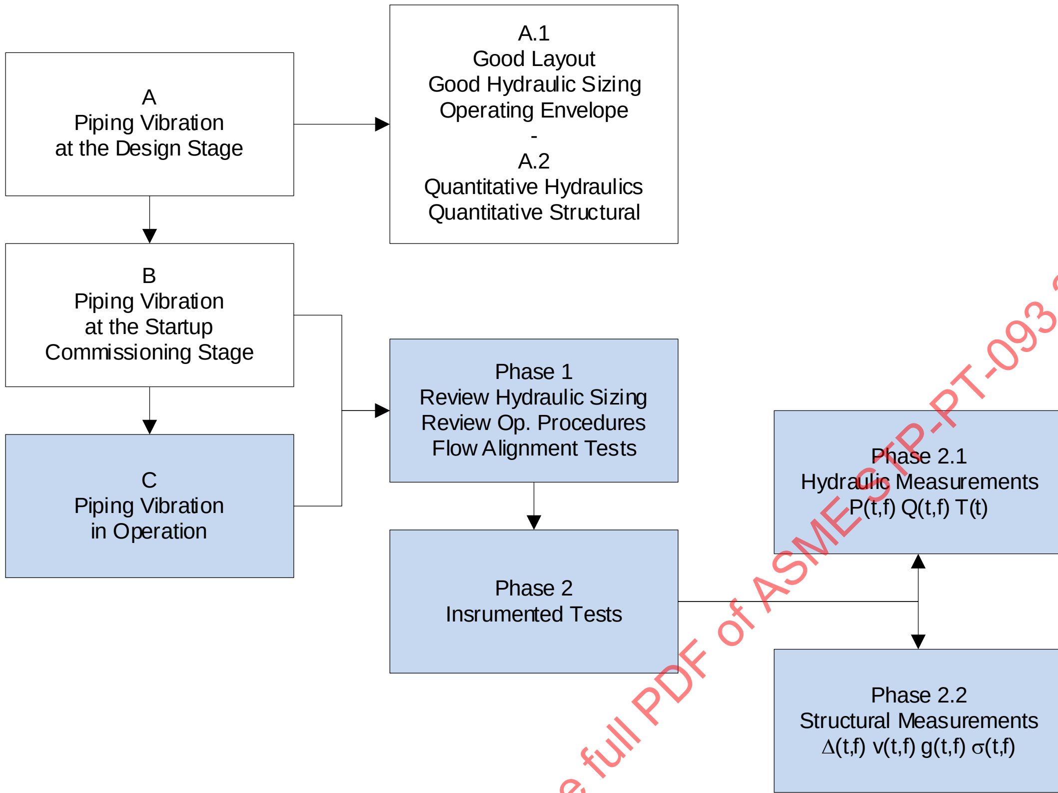

Figure 3-3: Piping Vibration at the Startup (Commissioning) Stage

Figure 3-3 **Block C**: This is the case, not uncommon, where piping spans that had not vibrated in the past start to vibrate later in the system's operating life. This is often due to wear and tear of component parts, for example a stem-to-disk link that wears-out causing the valve disk to flutter in the flow, causing vortex shedding; or wear of bolted or threaded support members and clamps that over time lose their grip on the pipe; or damage to valve internals. Another cause of late flow-induced vibration is caused by maintenance or repair modifications that alter the flow pattern.

The investigation of these piping vibrations later in the operating life is similar to that of piping vibration at the startup or commissioning stage, addressed above.

3.1.2 Sources of Knowledge

The current engineering knowledge in vibration of piping systems is documented in three primary forms:

1) Consensus Codes, standards, recommended practices, and guidelines published by committees and councils with professional memberships, such as the American Society of Mechanical Engineers (ASME), the American Petroleum Institute (API), the Gas Machinery Research Council (GMRC), or the Health and Safety Institute's Executive in the United Kingdom. In some industries, the application of these documents is required by a regulatory body.
2) Publications, such as textbooks, papers and articles presented at conferences and symposia, or published in professional journals. In the case of interest here, gas compressor station piping, one source of such publications is the Turbomachinery and Pump Symposia, and the ASME Pressure Vessels and Piping (PVP) conferences.

3) Procedures, methods, and criteria documents developed by operating companies, specialty services firms, and Research Institutes (such as the Southwest Research Institute).

3.1.3 Industry Codes, Standards and Guidelines

Currently, in the United States, the following documents provide vibration guidance and rules for piping systems, either for the prevention of vibration at the design stage, or for the evaluation of measured vibration in operation:

- ASME
  - ASME B31 Pressure Piping Code, in particular B31.1 (power), B31.3 (process), B31.4 (liquid pipelines), and B31.8 (gas pipelines).
  - OM-3: ASME Operation and Maintenance of Nuclear Power Plants, Part 3, Vibration Testing of Piping Systems (current edition is 2017).
- API
  - API 618 Standard, Reciprocating Compressors for Petroleum, Chemical, and Gas Industry Services (current edition is 2007 with 2009 and 2010 errata).
  - API 674 Positive Displacement Pumps-Reciprocating (current edition 2010).
  - API RP-688 Recommended Practice, Pulsation and Vibration Control in Positive Machinery Systems for Petroleum, Petrochemical, and Natural Gas Industry Services (current edition is 2012).
- GMRC
  - Gas Machinery Research Council, Pipeline Research Council International, and Southwest Research Institute, Design Guideline for Small Diameter Branch Connections.
- UK Energy Institute
  - UK EI Guideline: An international guideline that has seen broad application across industries is the "Guidelines for the Avoidance of Vibration Induced Fatigue Failure in Process Pipework" by the Energy Institute of the UK Health & Safety Executive.

The salient features of these codes, standards, and guidelines are described next.

3.2 ASME B31 Vibration Rules

The vibration rules in the 2016 edition (latest at the time of this writing) of B31.8, B31.1, B31.3, and B31.4 are listed in Annex A:. Of these rules, the most useful rules for the intent of this report are summarized here. As can be seen from this Section and from Annex A:, the current B31 Codes do not provide explicit quantitative methods and criteria for the prevention of piping vibration at the design stage.

3.2.1 Current B31 Vibration Stress-Based Rules

(a) B31.3

- For Category M Fluid Service, suitable dynamic analysis, such as computer simulation, shall be made where necessary to avoid or minimize conditions that lead to detrimental vibration, pulsation, or resonance effects in the piping (para. M301.5.4).
- Compression-type tube fittings may be used in vibratory service if the fitting has a stress intensity factor not larger than 1.5 (para. U305.6).
- Vibration loads on metallic bellows should be stated in the design (para. Z301.1.3).
- The new B31.3 Appendix W "High-Cycle Fatigue Assessment of Piping Systems" applies to cyclic loads including vibration. In particular, Appendix W defines significant stress cycles as follows:

  "A significant stress cycle is defined as a cycle with a computed stress range [emphasis added], *in accordance with para. 319, greater than 20.7 MPa (**3.0 ksi**) for ferritic steels and austenitic*

stainless steels. For other materials, or corrosive environments, all cycles shall be considered significant, unless otherwise documented in the engineering design."

For piping vibration, this implies that 3 ksi is an endurance limit for carbon steel and austenitic stainless steel. The 3 ksi limit (stress range iM_{range}/Z) differs from ASME O&M limits of **7 ksi** (stress amplitude $2iM_{amplitude}/Z$) carbon and low alloy steels, and **13.6 ksi** for austenitic stainless steel and Ni alloys, as will be described later.

(b) B31.8

- The stress range in unrestrained piping due to vibrational displacements shall be computed as $S_E = M_E/Z$ (para. 833.8).
- The design fatigue life, predicted by the Palmgren-Miner (S-N) methods, should be at least 10 times the service life for all components (API RP 1111, referenced in para. A842.2.5).

3.2.2 Current B31 Vibration Design and Fabrication Detail Restrictions

(a) B31.1

- Flexible metal hose assemblies may be used to provide flexibility in a piping system, to isolate or control vibration, or to compensate for misalignment (105.4).
- Special consideration should be given to restricting the use of socket welded piping joints where severe vibration is expected to occur (para. 111.3.1).
- Threaded joints are prohibited where vibration is expected to occur (114.2.1).
- The design and installation of insertion type instrument, control, and sampling devices shall be adequate to withstand the effects of the fluid characteristics, fluid flow, and vibration (114.2.3).
- Brazed and soldered joints shall not be used in piping subject to vibration (para. 117.3).
- Soldered fittings are not recommended where mechanical vibration is encountered (para. 122.3.2).

(b) B31.3

- In vibration service, backing rings should be removed and the internal joint face ground smooth (para. 311.2.4).
- Consideration shall be given to the tightness of expanded joints when subjected to vibration (para. 313).
- In selecting and applying flared, flareless, and compression type tubing fittings, the designer shall consider the possible adverse effects on the joints of vibration, shock, and thermal expansion and contraction (315.1).
- Compression-type tube fittings may be used in vibratory service if the fitting has a stress intensity factor not larger than 1.5 (para. U305.6).
- The use of controlled bolting procedures should be considered under conditions involving vibration to reduce the possibility of stress relaxation and loss of bolt tension (para. F309.1).

(c) B31.8

- Threaded reducing bushings should not be used in pressure/flow control facilities where they are subject to high frequency piping vibrations (para. 841.1.9).
- Monitoring Effects of Pulsation and Vibration (para. 853.1.7) Facilities exposed to the effects of vibration and pulsation induced by reciprocating compression as well as vibration induced by gas flow or discharge, may be susceptible to fatigue crack growth in fabrication and attachment welds. Susceptible facilities include (a) compressor station piping having an observed history of vibration; (b) blowdown piping; (c) pulsation bottles and manifolds; (d) piping not meeting the requirements of para. 833.7(a); such facilities may warrant engineering assessment and/or nondestructive examination for fatigue cracking in fabrication and attachment welds.

3.3 ASME O&M Vibration Rules

ASME operations and Maintenance of Nuclear Power Plants (O&M), Part 3 "Vibration Testing of Piping Systems" was originally developed to conduct the start-up vibration monitoring of nuclear power plants, including piping systems. Over the years, O&M Part 3 has proven to be useful to tackle piping vibration in service. Part 3 addresses steady-state vibration as well as transient vibration. OM-3 chapters and appendices address the following topics:

Chapters:
1 - Scope
2 - Definitions
3 - General Requirements
4 - Visual Inspection Method
5 - Simplified Method for Qualifying Piping Systems
6 - Rigorous Verification Method for Steady-State and Transient Vibration
7 - Instrumentation and Vibration Measurement Requirements
8 - Corrective Action

Appendices:
A - Instrumentation and Measurement Guidelines
B - Analysis Methods
C - Test/Analysis Correlation Methods
D - Velocity Criterion
E - Excitation Mechanisms, Responses, and Corrective Actions
F - Flowchart — Outline of Vibration Qualification of Piping Systems
G - Qualitative Evaluations
H - Guidance for Monitoring Piping Steady-State Vibration Per Vibration Monitoring Group 2
I - Acceleration Limits for Small Branch Piping

O&M Part 3, provides a limit on steady-state vibration stress amplitude S_{alt}:

$$S_{alt} = \frac{C_2 K_2}{Z} M_{ampl} = \frac{2i}{Z} M_{ampl} \leq S_{el}$$

C_2K_2 = stress indices (ASME III NB-3600).
i = stress intensification factor (from ASME B31, or ASME B31J).
M = maximum amplitude (zero to peak) dynamic moment loading due to vibration, in.kips.
S_{el} = fatigue stress (amplitude) endurance limit equal to S_A at 1E11 cycles from ASME BPV Code, Section III, Fig. I-9.1 or Fig. I-9.2, as applicable. The user shall consider the influence of temperature on the modulus of elasticity, ksi.
Z = section modulus of the pipe, in^3.

The values of S_{el} from ASME III Appendix I (2019) are:

- For carbon steel and low alloy steel, for metal temperatures not exceeding 700°F, the value of S_{el} = 7 ksi (amplitude) at 1E11 cycles.
- For austenitic steels, nickel–chromium–iron alloys, nickel–iron–chromium alloys, and nickel–copper alloys, for metal temperatures not exceeding 800°F, the value of S_{el} = **13.6 ksi** (amplitude) at 1E11 cycles.

ASME O&M Part 3 Nonmandatory Appendix D Velocity Criterion provides a piping vibration screening velocity criterion of 0.5 ips. This is a simple criterion and is usually used as a first screen. The vibration velocity limit of **0.5 ips** is based on simple span formulas, with built-in adjustment factors C_1, C_3, C_4 and C_5 for concentrated weights, weight of pipe contents and insulation, end support conditions of the span, and natural frequency of the span.

$$V_{measured}(ips) \leq \frac{C_1\, C_4}{C_3\, C_5} \frac{S_{el}}{C_2\, K_2}$$

O&M Part 3 also provides a limit on accelerations of small unsupported branch piping:

$$accel.(g) \leq \frac{S_{el}\, Z}{C_2 K_2\, W_T L_E}$$

Where W_T is the total weight of the branch piping and L_E its effective length.

3.4 API Vibration Rules

The vibration prevention guidance of API 618 (2010), API 674 (2010), and API 688 (2012) is presented in Annex B:.

- API Standard 618 applies to reciprocating compressors and is therefore of importance to this report on compressor station piping.
- API Standard 674 applies to positive displacement pumps and includes pressure pulsing limits for liquid service.
- API Recommended Practice 688 applies to gas and liquids and includes fundamental and practical concepts of vibration theory, and vibration-induced loads and pressure pulsing limits.

3.5 Gas Machinery Research Council Vibration Rules

As described on their website, *"The Gas Machinery Research Council (GMRC) is a community of proactive natural gas companies dedicated to investigating technical issues within the rapidly evolving gas machinery industry and uncovering innovative solutions that improve reliability, efficiency, and cost-effectiveness of mechanical and fluid systems."* Annex C: is a partial list of valuable GMRC publications that address compressor and compressor piping vibration.

3.6 UK Energy Institute Vibration Rules

The Guidelines for the Avoidance of Vibration Induced Fatigue Failure in Process Pipework, published by Energy Institute, UK (UK EI Guideline), is a comprehensive document that provides guidance for the prevention of piping vibration at the design phase, as well as the evaluation of piping vibration in the field. A partial overview of the Energy Institute's approach is provided in Chapter 5.

4 SOURCES OF VIBRATION AND PREVENTION IN DESIGN

Figure 4-1 provides a **structured description** of the sources of vibration in piping and tubing systems. Sources of vibration can be divided into two major groups, mechanical-induced excitation (MIE) and flow-induced excitation (FIE), which can in turn be subdivided into sub-categories as described in this Chapter.

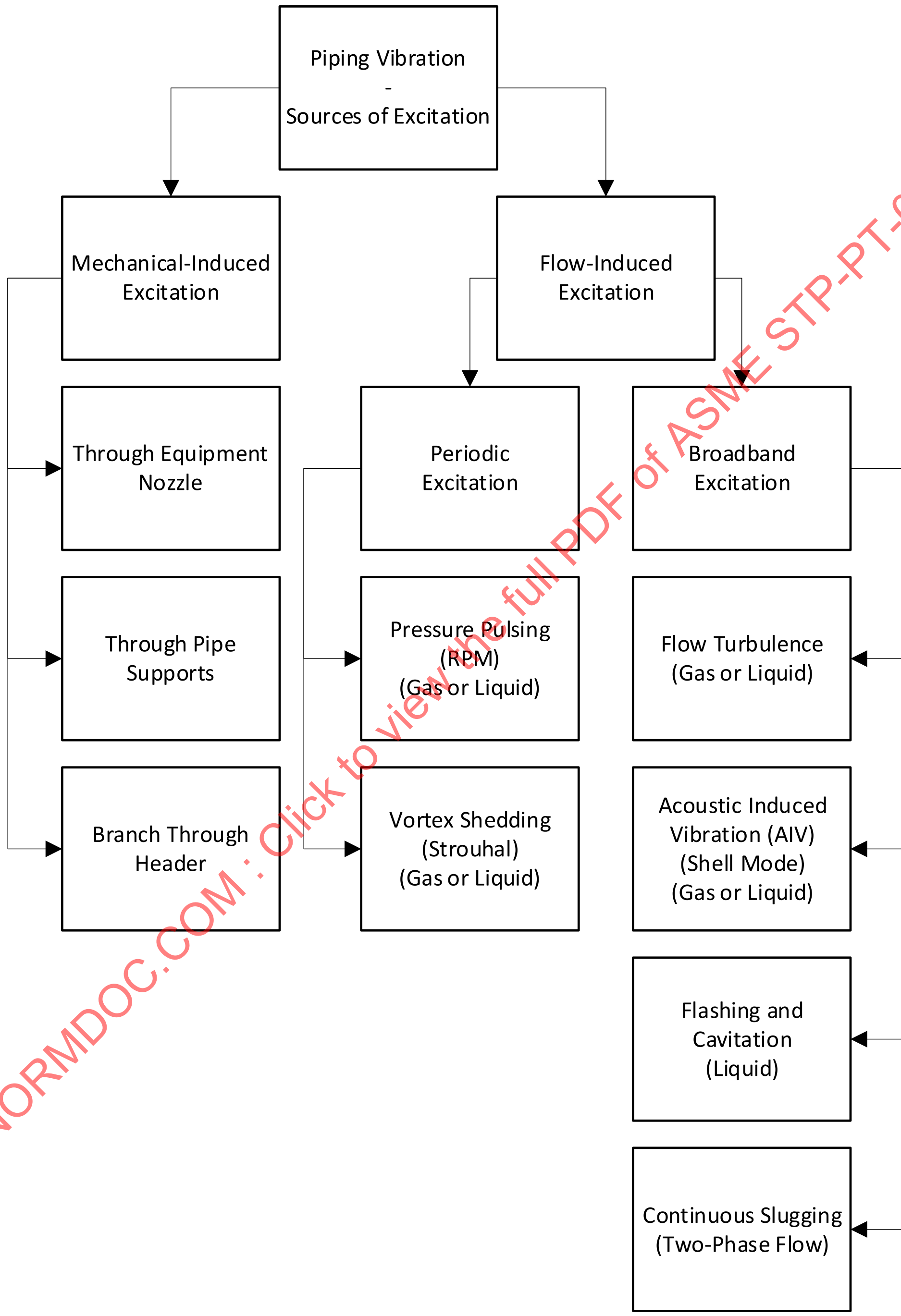

Figure 4-1: Sources of Piping Vibration

The clarity of the structured approach of Figure 4-1 is essential, as many existing codes, standards, and publications tend to classify and discuss the sources of vibration in a haphazard manner.

Sections 4.1 and 4.2 describe each of the sources of excitation resulting in vibration of piping systems and pipelines. Figure 4-1 is a structured presentation of the causes of excitation, either mechanical or flow-induced (hydraulic). For each source of vibration, we address the considerations that apply at the design stage to prevent, or at least minimize, piping and pipeline vibration. The amplifying effect of acoustic and/or structural resonances is addressed in Section 4.3.

Steady-state vibration is subdivided into two main categories, as described in this Chapter.

- **Mechanical-induced excitation** (MIE), resulting in mechanically-induced vibration (MIV): Vibration caused by an external mechanical or structural driving force rather than by the flow of fluid or pressure pulsing inside the pipe. MIE consists of three causes of vibration:
    - Structural vibration of the compressor driving the piping attached to the compressor nozzles to vibrate (Section 4.1.1).
    - Structural vibration of a skid or structure driving the supported piping to vibrate (Section 4.1.2).
    - Vibration of the header (either MIV or FIV) driving the attached small branch piping to vibrate (Section 4.1.3).

- **Flow-induced excitation** (FIE), resulting in flow-induced vibration (FIV): Vibration caused by the fluid flow or the fluid pressure pulses inside the pipe. The many forms of FIV are addressed in this Chapter. The driving pressure for FIV will be subdivided into periodic pressure fluctuations (Section 4.2.1) or random pressure fluctuations (Section 4.2.2).

Regarding the nomenclature MIE vs. MIV and FIE vs. FIV: During the peer review of this report an important comment was made, which prompted me to revise my 45 year-old use of MIV/FIV (a common nomenclature) to MIE/FIE in some cases. The comment is:

- When speaking of the mechanical or hydraulic **cause** of vibration, it is best to speak of "excitation", i.e. MIE and FIE.
- When speaking of the **resulting** vibratory movement, then it is best to refer to "vibration", i.e. using the usual MIV and FIV.

4.1 Mechanical-Induced Excitation

Mechanical-induced excitation (MIE) is the vibration of a piping or tubing system, or part of such systems, driven not by the flow or the pressure fluctuations of the fluid inside the line, but instead driven directly by the vibration of the supporting structure or the connected equipment, or the movement of the header pipe. In other words, the pipe follows (and sometimes amplifies) the vibratory movement of a structural driver such as a vibrating pump or compressor, a vibrating support structure such as foundations-floors-walls-ceilings-racks, or a vibrating header.

4.1.1 Vibration Through Equipment Nozzle

4.1.1.1 Description

This type of vibration is caused by the vibration of a machine (pump or compressor), at its running speed or harmonic multiples, which is transmitted to the connected piping or tubing through the pipe-to-equipment nozzle.

The UK EI Guideline provides the following qualitative estimate of the likelihood of failure through equipment nozzle (this section) or through the support structure (next section).

The likelihood of mechanically-induced vibration (MIV) of a main line depends on the type of driver (screw/gear or reciprocating), and the structural coupling between the pipe or pipe supports and the driver, per UK EI Guideline Section T1:

- If the piping is connected to a screw/gear compressor either directly (suction or discharge nozzle), or through supports mounted on the compressor skid/module/deck or within, or pipe spans within a distance equal to the maximum length of the skid (which in EI Guideline Table T1-2 item 3 is considered to be a pipe supported near the skid), the likelihood of MIE is classified as medium for this screen.
- If the piping is connected to a reciprocating compressor either directly (suction or discharge nozzle), or through supports mounted on the compressor skid/module/deck or within, or pipe spans within a distance equal to the maximum length of the skid, the likelihood of MIE is classified as high for this screen.

4.1.1.2 Prevention in Design

(a) Well-designed machinery (compressor) foundation to prevent rocking and translational vibration.
(b) Well-designed machine anchorage to the foundation.
(c) Well-designed machinery, with sufficient stiffness, including sufficient torsional stiffness, to prevent rocking, translational, or torsional vibration.
(d) Well-balanced machinery, verified during installation and initial testing.

Beyond the design stage, during the operating life, the facility's maintenance program will be relied upon to limit the machinery vibration through periodic vibration measurement and analysis, trending, diagnostics, and the resulting adjustments of the machine, the compressor in the case of a gas transmission pipeline.

4.1.2 Vibration Through Support Structure

4.1.2.1 Description

This type of vibration is caused by the vibratory motion of the machine (the compressor in the case of a gas pipeline), at its running speed or harmonic multiples, which is transmitted to the skid, the foundation, the support racks, and the pipe supports. This vibration of the supporting structure is transmitted to the pipe supports and from there to the piping or tubing itself.

4.1.2.2 Prevention in Design

(a) Same design provisions as for vibration through equipment nozzle (Section 4.1.1).
(b) Stiff pipe supports and support structures to prevent the small machine vibration to be amplified through the pipe supports and support structures.

4.1.3 Vibration of Branch Through Header

4.1.3.1 Description

This type of vibration occurs when a small bore branch line (a vent, a drain, a bypass) is attached to a main line header that vibrates. The vibration of the main line header may be flow-induced, but the attached small bore line mechanically follows and often amplifies the main line header movement, often due to resonance of the branch line.

4.1.3.2 Prevention in Design

The UK EI Guideline addresses this type of vibration of a branch line driven by the header vibration in its technical modules T3 "Quantitative Small Bore Connections (SBC) LOF (likelihood of failure) Assessment", T11 "SBC Corrective Actions", and Appendix C "SBC LOF Assessment Guideline". This approach is described in Section 5.4 of this report.

For the design of SBCs in gas compressor stations, we recommend guidance similar to the UK EI Guideline (Section 5.4 of this report), with some modifications.

In the case of cantilevered SBCs:

- Type of header attachment fitting: Use fittings with a low stress intensification factor.
- Length of SBC, tie-back if longer than 4 ft. without valve, and reduce this length based on the weight-equivalent length of pipe if there are valves on the SBC.
- The header cannot have a thickness less than sch.80, or the equivalent reinforcement for sch.40.
- The size of the SBC should not be less than NPS 1.

In case of a mechanically-supported SBCs:

- The span of the SBC between vibration-preventing supports must be selected to prevent structural resonance with the structural natural frequency of the header pipe spans near the SBC. The natural frequency of the SBC and the header are preferably established by modal analysis, or for simple geometries by equivalent beam formulas that reflect the moment of inertia of the pipes (a function of their diameter and thickness); the layout, spans and weights; the flexibility of fittings; and the location and stiffness of supports.

4.2 Flow-Induced Excitation (FIE)

4.2.1 Periodic Pressure Fluctuations

Periodic (tonal) pressure fluctuations are characterized by periodic pressure peaks in the time waveform (time-history) domain of the pressure signal. These convert into discrete, narrow, evenly-spaced (harmonic) peaks in the frequency domain. There are two principal sources of periodic pressure fluctuations: (1) Pressure pulsing (Section 4.2.1.1), and (2) vortex-shedding (Section 4.2.1.2). Both create periodic pressure fluctuations which are reacted at changes in flow area or changes in direction, causing periodic forces on the piping.

4.2.1.1 Pressure Pulsing

4.2.1.1.1 Description

By their very nature, the operation of pumps and compressors, either reciprocating or centrifugal, generate oscillating pressures (at the pump or compressor vane passing frequency or piston stroking frequency) that travel down the suction and discharge lines. The frequency of pressure pulses is:

$$f_{pressure\ pulsing} = n\,\frac{N\,B}{60}$$

Where $f_{pressure\ pulsing}$ = frequency of pressure pulses (Hz); n = mode number (1, 2, 3, etc.); N = machine speed (RPM); B = number of vanes in a centrifugal machine, number of plungers in a reciprocating machine, and to account for double-acting vs single-acting compressors; 60 = factor to convert 1/minutes (RPM) to 1/seconds (Hz).

The acoustic analysis should cover the full range of anticipated design and operating conditions. The analysis should include single acting operation if that is an anticipated compressor load step.

While the fundamental principles of dynamic resonance are not different, compressors pressure pulsing from reciprocating compressors tend to excite lower frequency beam modes of the piping system, whereas centrifugal compressors produce pressure pulses at vane passing frequencies, in the range of 10 kHz to 5 kHz that tend to excite shell modes (breathing modes). The shell modes excitation will be more pronounced for pipes with large D/t, as will be addressed in Section 5.4.2.2.

Pressure pulsing is a potential source of vibration in liquid or gas flow.

4.2.1.1.2 Pressure Pulsation Screen for Positive Displacement Compressor

According to the UK EI Guideline, Section T1 and Appendix D, the likelihood of FIV caused by pressure-pulsation depends on the type of positive displacement compressor:

- If the gas is compressed by a screw or gear type of compressor, the likelihood of FIV is classified as medium for this screen.
- If the gas is compressed by a reciprocating, positive displacement type of compressor, the likelihood of FIV is classified as high for this screen.

4.2.1.1.3 Pressure Pulsation Screen for Centrifugal Compressor

According to the UK EI Guideline, Section T1 and Appendix D, the likelihood of FIV caused by pressure-pulsation depends on the potential for a centrifugal compressor to stall:

- If the compressor has stall characteristics but has formal operational constraints in place to prevent stalling, the likelihood of FIV is classified as medium for this screen.
- If the compressor stall characteristics are unknown, or if the compressor has stall characteristics and has no formal operational constraints in place to prevent stalling, the likelihood of FIV is classified as high for this screen.

4.2.1.1.4 Prevention in Design

The EI Guideline rules estimate a judgement-based likelihood of failure (LOF) from FIV caused by pressure pulsation from a positive displacement compressor, as presented in Table 4-1. The LOF is a number less than or equal to 1.0, where LOF = 1.0 is the largest likelihood of failure. Failure in the EI Guideline refers to fatigue crack propagation through the wall of the piping.

Table 4-1: Likelihood of Failure (LOF) from FIV for Positive Displacement Compressor

Piping Connection	LOF
Compressor power less than 112 kW and discharge less than 35 bar	0.4
Piping design based on an API 618/674 acoustic analysis	0.4
Neither of the above conditions apply	1.0

(Based on EI Guideline (2008) Section T2.4.2)

The LOF from FIV caused by the potential for stalling of a centrifugal compressor are presented in Table 4-2.

Table 4-2: Likelihood of Failure (LOF) from Stalling of a Centrifugal Compressor

Piping Connection	LOF
The compressor does not display a rotating stall characteristic	0.2
The compressor does not operate at low flow, near the stall condition	0.4
The compressor operates at low flow, may stall	1.0

(Based on EI Guideline (2008) Section T2.5.2)

The proposed guideline would be as follows:

The layout and support of the compressor discharge piping should prevent natural structural frequencies equal to the fundamental and harmonic pressure pulsing frequencies $f_{pressure}$ pulsing. This design check would be by means of a **modal structural analysis** of the piping system to determine its structural frequencies and compare them to $f_{pressure}$ pulse. The following piping may be exempted from this guideline:

- The compressor power less than 112 kW and discharge less than 35 bar, or
- The piping design is based on an API 618/674 acoustic analysis.
- The compressor does not operate at low flow, near the stall condition, i.e. the compressor does not display a rotating stall characteristic.

4.2.1.1.5 Structural Decoupling by Separation of Frequencies

API 618 Para. 7.9.4.2.5.3.2 recommends the following separation margins "to avoid coincidence of excitation frequencies with mechanical natural frequencies of the compressor, pulsation suppression devices and piping system". While these rules are written for reciprocating compressors, the fundamental principles of dynamic resonance are not different for centrifugal compressors:

- The minimum mechanical natural frequency of any compressor or piping system element shall be designed to be greater than 2.4 times maximum rated speed.
- The predicted mechanical natural frequencies shall be designed to be separated from significant excitation frequencies by at least 20%.

4.2.1.2 Vortex Shedding

4.2.1.2.1 Description

Flow through a restriction such as an orifice plate or a control valve, or flow passed sharp edges such as sharp branch openings, or flow across an obstruction such as protruding instruments like thermowells, causes vortex shedding, also referred to as von Karman vortices. The shed vortices in turn cause pressure pulses which have a dominant frequency. For example, in the case of vortices formed downstream of an orifice plate, the pressure pulse frequency from vortex shedding is:

$$f_{vortex\ shed} = \frac{S\ v}{d}$$

Where $f_{vortex\ shed}$ = vortex shedding frequency, also referred to as Strouhal frequency (Hz); S = Strouhal number which depends on the shape and roughness of the opening edges and the Reynolds number; v = flow velocity through the opening (in/sec); and d = diameter of the opening or characteristic dimension of the obstruction (in.).

For a full treatment of vortex-shedding and resulting vibration across a thermowell, refer to ASME PTC 19.3 "Thermowell".

The likelihood of FIV caused by vortex shedding depends on the presence of orifice plates or elements protruding into the flow of gas (such as insert nozzles or thermowells): According to the UK EI Guideline Sections 2.3.4 and T2.7.1, if there are orifice plates or elements protruding in the gas flow, the likelihood of FIV is classified as high for this screen.

Vortex shedding is a potential source of vibration in liquid or gas flow.

4.2.1.2.2 Prevention in Design

- Design the piping system to pipe-on-pipe welded branch connections, using instead sweeping tee connections.
- Select and size orifice plates and control valve internals to minimize vortex shedding.
- Design the discharge piping to avoid acoustic natural frequencies and structural frequencies equal to the vortex shedding frequencies $f_{\text{vortex shed}}$ by a minimum of ±20%.

This design check would be by means of a **modal structural analysis** and an **acoustic natural frequency analysis** of the piping system to avoid $f_{\text{pressure pulse}}$ ±20%.

The exemption from acoustic or structural analysis would be in accordance with API Standard 618 5th edition Table 6, as follows, where DA-1 is an exemption from both acoustic and structural analysis, DA-2 is an exemption from structural analysis, and DA-3 does not exempt neither the acoustic nor the structural analyses.

Absolute Discharge Pressure	Rated Power per Cylinder		
	kW/cyl < 55 hp/cyl < 75	55 < kW/cyl < 220 75 < hp/cyl < 300	kW/cyl > 220 hp/cyl > 300
P < 35 bara P < 500 psia	DA-I	DA-2	DA-2
35 bara < P < 70 bara 500 psia < P < 1000 psia	DA-2	DA-2	DA-3
70 bara < P < 200 bara 1000 psia < P < 3000 psia	DA-2	DA-3	DA-3
200 bara < P < 350 bara 3000 psia < P < 5000 psia	DA-3	DA-3	DA-3

4.2.2 Broadband Pressure Fluctuations

Broadband pressure fluctuations are characterized by the lack of periodic peaks in the pressure time waveform (time-history) domain, and therefore the lack of discrete, narrow, often evenly-spaced (harmonic) pressure peaks in the frequency domain. Random pressure peaks in the frequency domain (for example a linear plot of y = velocity as a function of x = frequency), have wide skirts (tails) when plotted in log-y domain (i.e. logarithm of y = velocity against linear x = frequency).

4.2.2.1 Turbulent Flow

4.2.2.1.1 Description

Significant flow turbulence causes a broadband pressure excitation which is a common cause of piping vibration. Significant turbulence can occur as a result of inadequate system layout, such as using short radius elbows on high velocity flow, using 90-degree branch connections in place of wyes or smooth-contoured fittings such as o-lets where branch flow merges into a header, or using incorrectly sized control valves. The spectrum of pressure frequencies from flow turbulence is broad band. Turbulent excitation caused by high flow velocities exhibits broad band pressure fluctuations, broad band noise, and generally low energy. It tends to only drive the low natural frequencies of the pipe, often below 10 Hz.

Some engineers reserve the term flow-induced vibration (FIV) to this sole type of this turbulence-induced vibration. This exclusive use of the term FIV for turbulence-induced vibration can lead to misunderstandings.

Turbulent flow is a potential source of vibration in liquid or gas flow.

4.2.2.1.2 Prevention in Design

The kinetic energy limit of EI Guideline is proposed here for the prevention of turbulence-induced vibration. The KE screening criterion is defined in EI Guideline Table T1-1, as follows:

- If $\rho\ v^2 \geq 20{,}000$ kgm/m.sec^2 (13,400 lbm/ft.sec^2) the likelihood of FIE is high, and this condition should be avoided in design of compressor stations.

To illustrate this EI Guideline, consider the case of a methane pipeline operating at 1450 psi and 68°F (20°C). The gas density under these conditions is $\rho = 78.7$ kgm/m^3. The threshold of 20,000 kgm/m.sec^2 for turbulence-induced vibration is reached when the gas velocity is approximately 17 m/sec (56 fps, 38 mph).

There is an additional KE screening criterion at control valves, in EI Section T2.2.3.5, as follows:

$$\sqrt{\frac{\mu_{gas}}{10^{-3}}} \times \frac{\rho\ v^2}{F_v} \geq 480\ kPa\ (70\ psi)$$

Where μ_{gas} is the gas dynamic velocity (Pa.sec), ρ is the density of gas at the valve trim conditions (kg/m^3), v is the gas velocity exiting the valve trim (m/sec), F_v is factor which is a function of the pipe size (EI Guideline Section T2.2.3.4).

One more design criterion will be defined, based on experience with vibration in large flow rate systems:

- If $\rho\ v^2 > 5{,}000$ kgm/m.sec^2 (3,350 lbm/ft.sec^2), use sweeping tees, with an angle near 45 degrees, rather than mixing of gas streams at 90 degree branch connections.

4.2.2.1.3 UK EI Likelihood of Failure (LOF) Turbulent Flow

In the UK EI Guideline, the LOF from FIE caused by turbulent flow is given by:

$$LOF_{TF} = \frac{\rho\ v^2}{F_v}\ FVF$$

$$FVF = \sqrt{\mu}$$

Where LOF_{TF} = likelihood of failure caused by turbulent flow; F_v = turbulent flow factor, a function of the pipe size and natural frequencies; FVF = fluid viscosity factor, a function of the dynamic viscosity of the gas; v = gas flow velocity, m/sec; ρ = mass density of the gas, at concurrent pressure and temperature, kgm/m^3; ρv^2 = kinetic energy, calculated in the qualitative screen, kgm/m.sec; μ = dynamic viscosity of the gas, at concurrent pressure and temperature, cP (centipoise, force×time/area, 1 cP = 10^{-3} Pa.sec).

The flow-induced vibration factor F_v in the EI Guideline is an elaborate function of the piping natural frequency and the pipe size (diameter and thickness). The EI Guideline provides simplified, rough, yet elaborate formulas to estimate the natural frequencies of spans, to an accuracy of a few Hz, since the piping systems are to be classified as stiff (14 to 16 Hz), medium stiff (7 Hz), medium (4 Hz), and flexible (1 Hz), a difference of a few Hz changing the piping system classification. For example, a piping system is classified as flexible if its spans are longer than

$$L_{span} > -1.5968\ 10^{-5}\ D^2 + 0.033583\ D + 4.429$$

Where L_{span} = length of pipe span (m); and D = outside diameter of the main line (m).

Having classified the piping spans as stiff, medium, or flexible, the UK EI Guideline then provides elaborate relationships between the flow-induced vibration factor and the pipe size and natural frequencies, for which we have not yet found a satisfactory technical basis. For example, the turbulence-induced vibration factor F_v for NPS 10 and larger piping is given as:

$$F_v = \alpha \left(\frac{D}{t}\right)^{\beta}$$

where

$$\alpha = (41.21\ D + 49397) \times f_n^{\ 0.0001665\ D + 0.84615}$$

$$\beta = 0.0815 \times lnD - 1.3842 + 0.0191 \times (f_n - 1)$$

Where D = outside diameter of the main line (m); t = pipe wall thickness (mm), f_n = natural frequency of span (Hz).

Since the design of piping systems in gas compressor stations are typically based on a pipe stress analysis model, the designer can perform a modal analysis of the piping system and obtain a realistic estimate of the system natural frequencies. The designer can therefore preferably implement the modal analysis of the actual pipe configuration, without having to use the elaborate yet approximate estimates of natural frequencies from the EI Guideline.

4.2.2.2 Flashing and Cavitation

Flashing occurs when the liquid pressure drops below the saturation pressure at the operating temperature. This tends to happen at flow restrictions (control valve openings or orifice plates) where the flow velocity picks up while the pressure drops. In this case, the liquid flashes to small vapor bubbles which will collapse as the pressure recovers. This type of cavitation is sometimes accompanied by erosion as the liquid rushes into the collapsing cavity, at high pressure, against the metal wall.

Flashing and cavitation is a potential source of vibration in liquid flow only and is therefore not addressed here.

4.2.2.3 Acoustic-Induced Vibration (AIV)

4.2.2.3.1 Description

AIV is well described in the classic paper by V.A. Carucci and R.T. Muller ("Acoustically Induced Piping Vibration in High Capacity Pressure Reducing Systems". 82-WA/PVP-8, ASME), which is reproduced in part here: *"The level of acoustic energy (or aerodynamically generated noise) is a result of the shear forces and Reynolds stresses found within any turbulent flow. The magnitude of the turbulence and the internal forces/stress is a function of the flowing fluid and its velocity... When choked conditions occur across [an] orifice, the turbulent forces/stress will become so large that intense noise due to large pressure fluctuations will be generated. The generated noise is nonperiodic due to the randomness of the turbulent forces involved in its formation. The randomness causes the acoustic spectrum of a control valve to have a broad frequency range with a peak normally occurring in excess of 1000 Hertz."*

Another description is provided in the public domain publication "Differentiating Between Acoustic and Flow Induced Vibrations", by M. Jouahari et. al., Bechtel Virtual Technology Expo 2018, Nov. 5-16, 2018: *"AIV refers to structural vibration excited by intense acoustic pressure in a piping system with vapor flow. The acoustic pressure is usually created from pressure-reducing devices due to high pressure drops and mass flows of vapor services. These acoustic energies excite the pipe wall circumferentially due to high-frequency sound waves in the range of 500–2,500 Hz, where most of the energy is captured. The circumferential mode of vibration causes the pipe to displace radially, and this leads to fatigue failures where stress concentrations occur downstream, such as at pipe fittings and welded pipe supports."*

Another description of AIV is provided in the conference paper "Mechanical, Stress and Flow Considerations for Piping Design of Centrifugal Compressors", B.A. White et. al. (SwRI), 37th Turbomachinery and 34th Pump symposia, Houston, TX, September 17-20, 2018: *"Acoustic induced vibration is a vibration phenomenon due to excitation from valves with high pressure differentials combined with high mass flow rates. The flow is choked at the vena contracta creating a high energy turbulent jet that expands into the piping creating broadband turbulence. This creates pressure and velocity fluctuations in the piping system that can couple with piping acoustic and mechanical natural frequencies creating high cycle fatigue failures at high stress concentration points such as tee connections or welded supports. The excitation is typically low amplitude broadband excitation from approximately 200 Hz to 5,000 Hz, …".*

This flow-induced noise has frequencies typically in the range of a few hundred Hz to 2 kHz, within the range of sound detectable by the human ear (20 Hz to 20 kHz). The pipe tends to respond to such high excitation frequencies by vibrating in a shell (breathing) mode, rather than in a beam (bending) mode.

If AIV is amplified by acoustic resonance, then the noise is predominantly a pure tone at the acoustic natural frequency.

Some engineers treat AIV separately from FIV, whereas here AIV is listed under FIV, since AIV is at its origin generated by the flow inside the pipe, i.e. it is flow-induced. This segregation of AIV from FIV is not essential and does not alter the engineering treatment of the problem.

AIV is a potential source of vibration in liquid or gas flow.

4.2.2.3.2 Prevention in Design

According to the UK EI Guideline, the likelihood of acoustic-induced vibration (AIV) is a function of the potential for sonic flow, causing shock waves, and the sound power of the gas flow.

The second edition of the UK EI Guideline Section T2.7 High Frequency Acoustic Excitation, provides a step-by-step diagram to estimate the severity of AIV. If the sound power level at the source exceeds 155 dB limit, the EI procedure describes how to calculate the sound power level at the pipe discontinuities.

The 155 dB limit provided in the EI Guideline is consistent with the Carucci-Mueller 1982 paper (Figure 5-4) which plots an empirical "Recommended Design Limit" curve that slopes from approximately 170 dB for NPS 10 piping, down to approximately 158 dB for NPS 34 piping.

The sound power level at the source of the sound is estimated as follows (EI Guideline Table T2.7.3):

$$PWL_{source} = 10\ log_{10}\left[\left(\frac{P_{up} - P_{down}}{P_{up}}\right)^{3.6} W^2 \left(\frac{T_e}{M_w}\right)^{1.2}\right] + 125.1 + SFF$$

Where PWL_{source} = sound power level inside the pipe at the source of sound, dB; M_w = molecular weight, grams/mol; P_{up} and P_{down} = upstream and downstream pressure of pressure reducing device, Pa; SFF = sonic factor (6 if sonic flow, 0 if no sonic flow); T_e = upstream absolute temperature, deg.K.; W = mass flow rate kg/sec.

The PWL_{source} equation is shown here to highlight the parameters that can be reduced at the design stage to reduce the projected sound level: These are (1) the pressure drop across pressure reducing devices $P_{up} - P_{down}$ and (2) the gas mass flow rate.

Application of this formula to existing systems is addressed in Section 5.4.2.

4.2.2.4 Continuous Slugging

In two phase flow (gas and liquid), under certain flow conditions, the gas can propel slugs of liquids along the pipe. The slugs of liquid, as they impact changes in flow area (valves, orifice plates) or changes in direction (bends or branch connections) will create unbalanced forces that will cause the pipe to shake. If slugging is continuous, the shaking will also be continuous and will appear as continuous FIV. With vapor and liquid (for example steam and water) the pressure transients can be in the form of slugging (the water slugs propelled by the steam) or of cavitation (the steam bubbles collapsing as described above).

Continuous slugging is a potential source of vibration in liquid flow with entrapped gas and is therefore not addressed here.

4.3 Resonance

The various sources of vibration described above can cause piping or tubing systems to vibrate. Resonance is not a prerequisite to vibration: Piping and tubing systems **can vibrate without resonating**. Resonance, if it does happen, will amplify the vibration, and should therefore be avoided. There are two types of resonances to be considered: acoustic resonance (Section 4.3.1) and structural resonance (Section 4.3.2).

4.3.1 Acoustic Resonance

Acoustic resonance takes place when the flow-induced pressure fluctuating frequency is near an acoustic natural frequency of the piping system or connected vessel.

$$f_{pressure} \cong f_{acoustic}$$

For simple piping with a few straight segments, such as vents or drains, acoustic natural frequencies can be estimated by analytical closed-form solutions. The acoustic natural frequency of a pipe span is given by:

$$f_{acoustic} \cong n \frac{c}{k\,L}$$

Where f = natural acoustic frequency of a pipe span cavity (Hz); n = mode number (1, 2, 3, etc.); c = velocity of sound in the fluid (liquid or gas) (in/sec); k = factor that depends on the boundaries of the cavity (2 for open-open cylinder, 4 for open-closed cylinder with a pressure node near the header-branch entrance and a pressure anti-node at the closed end of the branch). The approximate sign ≅ is used because the actual length should be corrected for end conditions to be L + 0.8d and L + 0.4d for open-open and open-closed end conditions respectively, where d = inner diameter of the pipe (in).

For actual multi-span piping systems, with bends, valves, reducers, branch lines, etc. numerical simulation analysis may be necessary to compute the multiple piping system acoustic natural frequencies.

4.3.2 Structural Resonance

Structural resonance takes place when the flow-induced or mechanically-induced driving frequency is near a natural structural frequency of the piping system.

$$f_{pressure}\ or\ f_{structural\ driver} \cong f_{structural\ piping}$$

In the study of structural resonance in piping, two types of structural response must be addressed: (1) beam-type (bending) response of the pipe spans; and (2) shell-type (breathing) response of the pipe shell.

4.3.2.1 Beam Mode Resonance

For beam-type response, the pipe spans act as beams that deflect laterally. For simple piping with a few straight segments, such as vents or drains, beam-type structural frequencies can be estimated by analytical closed-form solutions. Closed-form solutions for lateral beam frequencies for single and multiple spans are tabulated in "Formulas for Natural Frequencies and Mode Shapes", R.D. Blevins. In the case of a uniform single span (without concentrated weights such as a valve) the beam-like lateral natural frequency is:

$$f_{structural\ piping\ (beam)} = \frac{\lambda}{2\,\pi}\sqrt{\frac{g\,E\,I}{\mu\,L^4}}$$

Where $f_{structural\ piping\ (beam)}$ = natural lateral (beam-like deflection) structural frequency of a pipe span (Hz); λ = frequency factor which is a function of the boundary conditions and the mode number; g = gravity (386 in/sec^2); E = modulus of elasticity of the metal at operating temperature (psi); I = moment of inertia of the pipe cross-section (in^4); μ = weight per unit length of the pipe including contents and insulation (lbf/in); L = length of the pipe span (in).

In the seminal paper "Piping Vibration Analysis" by J.C. Wachel, S.J. Morton, and K.E. Atkins (Proceedings of 19th Turbomachinery Symposium, Texas A&M University, 1990), the simple span formula is expanded to include adjustment factors for multi-spans, elbows, and in-line concentrated weights. A similar approach is followed in ASME O&M Part 3 to extend the prediction of structural natural frequencies for multiple spans by formulas.

For actual multi-span piping systems, with bends, valves, reducers, branch lines, etc. a pipe stress analysis software or finite element analysis may be necessary to compute the multiple piping system structural frequencies.

4.3.2.2 Shell Mode Resonance

For the high-frequency shell mode response, the radial (breathing) frequency of an infinitely long cylinder is given in Blevins (Formulas for Natural Frequency and Mode Shape, R.D. Blevins, Krieger Publishing, 1979, Table 12-1) as:

$$f_{structural\ piping\ (breathing)} = \frac{\lambda_i}{2\,\pi\,R}\sqrt{\frac{E}{\gamma\,(1-\nu^2)}}$$

For the extension modes (radial breathing in-out shape):

$$\lambda_i = \sqrt{1+i^2}$$

For the radial-circumferential flexural modes (cross-section ovalizing shape):

$$\lambda_i = \frac{1}{\sqrt{12}}\,\frac{t}{R}\,\frac{i\,(i^2-1)}{\sqrt{1+i^2}}$$

Where $f_{structural\ piping\ (breathing)}$ = pipe cross-section shell mode frequency of the pipe (Hz); λ_i = modal number; R = mean radius of the pipe (in); E = modulus of elasticity of the metal at operating temperature (psi); γ = mass density of the metal (lbm/in^3); ν = Possion ratio of the metal; t = pipe wall thickness (in); i = mode number (1, 2, 3, etc. for the extension modes, and 2, 3, 4, etc. for the flexural modes).

5 VIBRATION DURING COMMISSIONING OR OPERATION

It is not uncommon to identify piping vibration at the commissioning stage (start-up) or later in life during operation. This Chapter proposes a structured approach to:

1) Document the observed vibration (5.1)
2) Document the operating conditions (5.2)
3) Develop and implement a monitoring plan (5.3)
4) Evaluate the severity of the observed vibration (5.4)
5) Determine the potential cause(s) (5.5)
6) Propose solutions (5.6)

This six-step structured approach is recommended to be followed by compressor station staff when piping vibration is identified at the commissioning or operational stages.

5.1 Document the Observed Vibration

1) Assemble marked-up P&IDs showing the flow path and operating modes leading to the vibration.
2) Assemble isometrics showing the vibrating pipe sections. Scaled isometrics are preferred, if available.
3) Take photographs, and videos if possible, of the vibrating areas.
4) Gather and document, from existing reports and from interviews with operating-maintenance staff, the history of the observed vibration and their opinions on the possible causes.

5.2 Document the Operating Conditions

1) Retrieve the system hydraulic sizing calculations (pressure drops, flow rates along the piping system for various operating modes). Verify that the operating conditions (pressures, temperatures, and flow rates measured using existing station instrumentation) are within the design envelope set by the system hydraulic sizing calculations.
2) Confirm the compressor operating configuration (single or double acting, machine running speed, etc.).
3) Retrieve the component hydraulic sizing calculations (compressors, valves, filters, etc.) and verify that the operating conditions (pressures, temperatures, and flow rates using existing station instrumentation) are within the design envelope set by the component hydraulic sizing calculations.
4) Retrieve the maintenance records for the active components (compressors and valves) and verify that the diagnostics and operational tests have been performed satisfactorily, i.e. the components are operating as designed.
5) Visually walkdown the piping-supports system to verify its material condition, and its conformance to the design. Identify and fix non-conformances such as gaps between pipe and supports, loose or damaged supports, damaged concrete anchor bolts and foundations, etc.

5.3 Develop and Implement a Monitoring Plan

At this point, decide whether the monitoring will be hydraulic (pressures, flow rates, temperatures, sound), or structural (accelerations, velocities, displacements, strains, structural natural frequencies), or both, and decide on a selection of instruments, instrumented locations, and instrument accuracy.

Hydraulic monitoring is necessary if we intend to understand the hydraulic **root cause** of the vibration, as described in Chapter 3: Pressure Pulsing, Vortex Shedding, Flow Turbulence, Flashing and Cavitation, Acoustic Excitation, or Continuous Slugging, and help select hydraulic means of **prevention**.

Structural monitoring is primarily intended to determine the **severity** of the vibration, and therefore the likelihood of resulting fatigue failure, and as a result to help select the structural means of **mitigation**.

Unfiltered vibrations in complex structures, such as compressors and attached piping, equal the total vibration shown in a plotted vibration display. This vibration spectrum includes the discrete vibrations associated with every attached component of the structure and vibrations induced by fluid flows in the system. Filtered vibration data plots separate each vibration frequency into discrete identifiable frequencies and magnitudes, which need to be identified by a vibration analyst.

5.3.1 Hydraulic Monitoring

Monitoring and recording instrumentation to consider, and select as best applicable:

1) Portable strap-on flowmeters to record average flow velocities.
2) Dynamic pressure gages to record high-frequency pressure pulses (time-domain (time waveform) and frequency domain).
3) Thermocouples and/or infrared temperature sensors.
4) For acoustic-induced vibration (AIV), strain gages to record high frequency hoop stress fluctuations and convert to high frequency pressure fluctuations (time-domain (waveform) and frequency domain).
5) Sound recorders to determine dB level for AIV.
6) Non-intrusive optical infrared sensing (NIOIRS).
7) Photographs of each mounted monitor.
8) For AIV, there are two methods to obtain the sound power level (PWL):
   a) A decibel meter within 1 ft. or so of the pipe, to measure the sound pressure level (SPL), which is then converted into a sound pressure level at the pipe, and finally to a sound power level (PWL).
   b) From operating data (pressures, temperatures, and flow rates), obtaining the parameters of the PWL formula in Section 4.2.2.3.2.

5.3.2 Structural Monitoring

Monitoring and recording instrumentation to consider, and select as best applicable:

1) Video (with and without ODS i.e. amplified deflected shape capability).
2) Multi-channel accelerometers (each accelerometer location recording in 3 directions), with capability for hot pipe applications. With capability for integration to velocities and displacement, wave form and frequency domain. Also, capability for statistical analysis to differentiate periodic and random signals to help with root cause analysis.
3) Laser proximity sensors for non-contact recording of movement waveform and frequency.
4) Strain gages for strains and stresses at critical locations.
5) Bump test equipment for structural natural frequencies.
6) Photographs of each mounted monitor location.

5.3.3 Monitoring Report

1) Collect and store the monitoring data in a clear and retrievable manner.
2) Present and display the results in a clear manner, linked to the time of the recording, operating modes, location, photograph of the mounted instrument, isometric showing where the instrument was mounted.

3) Display the measurements in waveform (parameter vs time) and frequency domain (parameter vs frequency), with clear units and in each case whether the data represents peak values or root-mean-square values.
4) If possible, provide a statistical analysis of the data to determine whether the recorded signal is random or periodic, as this will help in the root-case assessment.
5) Allow time for review of the report by operations so they may confirm the operating modes that existed during data collection.

5.4 Evaluate the Severity of the Observed Vibration

While hydraulic monitoring is used to determine the hydraulic root cause of vibration, structural monitoring is used to determine the severity of the recorded vibration. At the time of this writing, several standards address the severity of recorded beam mode (not high-frequency shell mode AIV) vibration. The severity criteria of these standards will be summarized here, and some are plotted, superimposed, in Figure 5-1, as will be explained.

Since my recommendation in Section 2.1.2 is to espouse the API standards, the API 618 (green curve in Figure 5-1) would apply for beam mode vibration.

Vibrations are generally measured in terms of acceleration using accelerometers, but vibrations can also be expressed in terms of displacement, velocity, or acceleration. Software used by vibration analyzer equipment performs fast Fourier transforms (FFT) of the accelerations to determine unfiltered vibration frequencies. The vibration accelerations can then be converted to displacements or velocities. In general, velocities have been shown to be equivalent for different sizes of equipment, regardless of size. In other words, a 0.1 in. displacement is not the same for a 1/4 horsepower motor as it is for 500 horsepower motor. Similarly, a 1 g acceleration is not the same for 1/4 or 200 horsepower motors. Velocities are commonly assumed to be equivalent throughout a range of equipment sizes. Even so, the equipment geometry affects vibration velocities, where cantilevered sections may experience higher vibration velocities before damage occurs. In short, vibration velocity charts provide good guidance for troubleshooting and understanding vibration problems, but these charts do not provide the sole input to acceptance criteria for vibration.

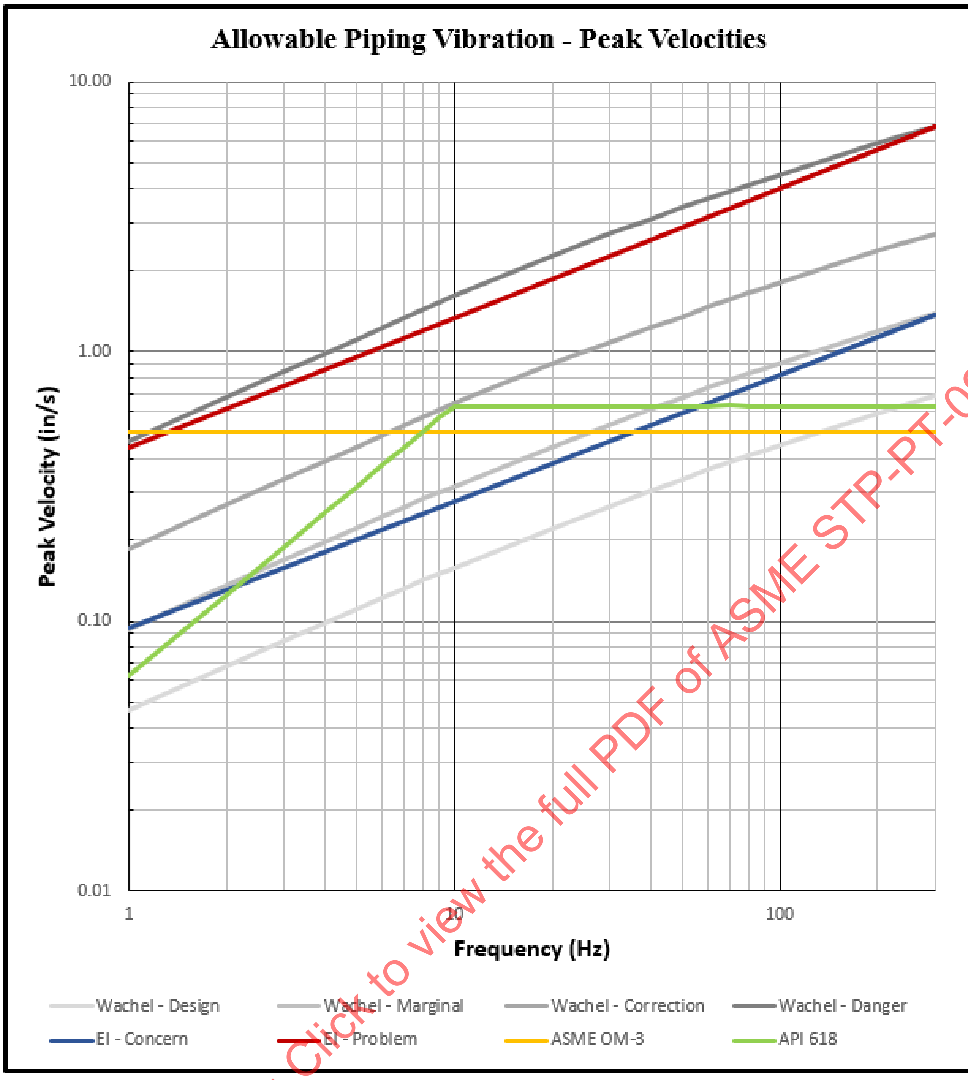

Figure 5-1: Current Vibration Velocity Amplitude Criteria, Superimposed, Color-Coded

From ASME OM-3 (0.5 in/sec, yellow), UK Energy Institute (Concern and Problem zones, blue and red), and Wachel 6-zone lines (black), converted to peak-to-peak for consistency. Note the cut-off at approximately 300 Hz for beam mode vibration as opposed to shell mode AIV (compiled by J. Lundquist, Becht)

5.4.1 Beam mode Vibration

5.4.1.1 ASME OM-3 Criteria

The ASME Operation and Maintenance of Nuclear Power Plants (O&M) Part 3 "Vibration testing of Piping Systems" (OM-3) standard has been used in the nuclear power industry since the 1970's to evaluate piping vibration of critical piping systems during start-up testing of nuclear power plants. OM-3 contains two acceptance criteria:

5.4.1.1.1 OM-3 Velocity-Based Criterion

The first OM-3 acceptance criterion is a simple, velocity-based limit of 0.5 ips peak velocity for carbon steel pipe and fittings. This simple screening criterion is developed in OM-3 Appendix D, based on conservative span configurations and end conditions, fatigue stress indices, and concentrated weights; and an allowable stress of 7 ksi for carbon steel. The 0.5 ips criterion is based on the relationship between vibration velocity and stress, similar to the Wachel approach that will be explained below. The OM-3 velocity-stress relationship is given as:

$$V_{allow\,peak}(ips) = \frac{C_1\,C_4}{C_3\,C_5}\,\frac{S_{el}(psi)}{C_2\,K_2}$$

The following parameters are provided in OM-3 Appendix D:

C_1 = correction factor that compensates for the effect of concentrated weights. If concentrated weight is less than 17 times the weight of the span for straight beams, L-bends, U-bends, and Z-bends, a conservative value of 0.15 can be used for screening purposes.
C_2K_2 = stress indices as defined in the ASME Code; $C_2K_2 = 2i$ where i is the B31 stress intensification factor; $C_2K_2 \leq 4$ for most piping systems.
C_3 = correction factor accounting for pipe contents and insulation; for contents and insulation equal to the weight of the pipe, the value would be 1.414.
C_4 = correction factor for end conditions different from fixed ends and for configurations different from straight spans; = 1.33 for cantilever and simply supported beam; = 0.74 for equal leg Z-bend; = 0.83 for equal leg U-bend; = 0.74 as conservative value for screening purposes.
C_5 = correction factor that is used when measured frequency differs from the first natural frequency of the piping span; for frequency ratios less than 1.0, the value is 1.0.
S_{el} = as defined in the next Section, 7 ksi stress amplitude for carbon steel.

$$V_{allow\,peak}(carbon\ steel) = \frac{(C_1 = 0.15)\,(C_4 = 0.74)}{(C_3 = 1.414)\,(C_5 = 1.0)}\,\frac{S_{el} = 7{,}000\,psi}{(C_2\,K_2 = 4.0)} = 0.5\,ips$$

5.4.1.1.2 OM-3 Stress-Based Criterion

The second OM-3 acceptance criterion is stress-based, as follows:

$$S_{alt} = \frac{C_2K_2}{Z}M < S_{el}$$

Where S_{alt} = maximum calculated stress intensity amplitude (ksi); C_2 and K_2 = secondary and peak stress indices from ASME III Div.1 NB-3600 (typically $C_2K_2 = 2i$, where i is the B31 stress intensification factor); Z = section modulus of the pipe (in^3); M = maximum (zero-to-peak) moment amplitude caused by the vibration (in.kips); S_{el} = allowable stress intensity from ASME III Div.1 Appendix I, at 1E11 cycles, which is equal to 7 ksi stress amplitude for carbon steel, and 13.6 ksi stress amplitude for austenitic stainless steel.

5.4.1.2 UK EI Guideline

Section T7.2.2 of the UK EI Guideline standard provides a log-log plot of RMS vibration velocity (mm/sec) against frequency. Two parallel lines subdivide the graph into three zones: “Acceptable”, “concern”, and “problem”. Above 300 Hz, the vibration is labeled “high frequency vibration – seek specialist advice”. This graphic criterion is shown in Figure 5-1, superimposed to the OM-3 vibration amplitude limit of 0.5 ips, and Wachel's 6-zone criterion.

5.4.1.3 Wachel Method

One of the earliest methods for the evaluation of piping vibration was published by J.C. (Buddy) Wachel and his colleagues. Mr. Wachel had joined Southwest Research Institute SwRI in 1961, and in 1982 co-founded Engineering Dynamics Inc. (EDI), San Antonio, TX, until his retirement in 1997.

In his paper "Piping Vibration and Stress" Proceedings of the Machinery Vibration and Analysis, Vibration Institute, April, 1981, Wachel introduced the 6-zone severity diagram, converted to peak velocities in Figure 5-1, superimposed to the OM-3 velocity criterion of 0.5 ips, and the UK EI Guideline 3-zone diagram. The 6-zone diagram encompasses vibration "perception" up to "danger" zones in the form of vibration amplitude peak-to-peak, vs. frequency, on log-log plot. The 6-zone diagram is described in Wachel's 1981 paper as *"Based on some 25 years of experience with piping vibration and failures, SwRI has developed vibration amplitude versus frequency criteria (Figure 1) in lieu of a more exact technique for estimating vibration dynamic stress in specific piping configurations."*

Caution: Because of its simplicity, the 6-zone diagram is reproduced and used in engineering assessments of vibration, without taking into consideration the many warnings Wachel mentioned in his 1981 paper, and repeated here:

- *"... vibration amplitude criteria for piping systems are likewise dangerous and, again, are fundamentally the wrong approach unless consideration is given to the configuration and dimensions of the piping being considered."*
- *"While they [the vibration amplitude diagrams] may be applicable in a statistical sense to average or typical piping, they are fundamentally incorrect because they do not consider the configuration involved".*
- *"While the statistical data from which the criterion was generated proves it works in most cases, the risk that it may not work for the next design should often dictate a more thorough analysis."*

In the same 1981 paper "Piping Vibration and Stress", Wachel develops a velocity-base criterion. The reader should refer to the 9-page 1981 paper for the derivations, but they can be summarized as follows. Seven pipe span configurations are considered, from a cantilever to the three-leg U-bend. The spans are assumed to be sinusoidally excited at their first natural frequency. The allowable vibration velocity is then given by:

$$V_{allow\ peak} = \frac{S_{all}}{\left(\frac{S}{v}\right)_{all} \times C_1 \times C_2 \times C_3 \times C_4 \times C_5 \times SF}$$

Where $V_{allow\ peak}$ = allowable vibration velocity amplitude (zero-to-peak) (ips); S_{all} = allowable alternating stress amplitude (zero-to-peak)(psi); $(S/v)_{all}$ = stress-to-velocity ratio, a function of the piping configuration and natural frequency (psi/ips); C_1 = correction factor for concentrated weights; C_2 = stress concentration factor; C_3 = correction factor for pipe contents and insulation; C_4 = correction factor for piping configuration and end conditions; C_5 = correction factor for vibration mode shapes other than the first mode; and SF = safety factor.

The allowable alternating stress amplitude selected in the 1981 paper is taken from the reference available at the time, the ASME B31.7 (the nuclear power plant piping Code) fatigue curve for carbon steel which went to 13 ksi stress amplitude at 1E6 cycles, and API Standard 618 "Reciprocating Compressors for

General Refinery Services" which had a stress limit of 26 ksi peak-to-peak, i.e. 13 ksi amplitude, same as B31.7. As can be seen above, this stress amplitude limit in today's OM-3 is down to 7 ksi amplitude for carbon steel.

Interestingly, in the 1981 paper Wachel takes the case of a fixed-fixed straight uniform beam vibrating at its natural frequency $(S/v)_{all}$ = 275; with a correction factor for concentrated weight C_1 = 8, a stress concentration factor C_2 = 4, and a safety factor SF = 2, which results in the same 0.5 ips vibration velocity amplitude (zero-to-peak) limit as OM-3:

$$V_{allow\ peak} = \frac{13{,}000\ psi}{(\left(\frac{S}{v}\right)_{all} = 275) \times (C_1 = 8) \times (C_2 = 4) \times (C_3 = 1) \times (C_4 = 1) \times (C_5 = 1) \times (SF = 2)}$$

$$V_{all} = 0.5\ ips$$

In a later paper ("Piping Vibration Analysis", J.C. Wachel, S.J. Morton, K.E. Atkins, Proceedings of the Nineteenth Turbomachinery Symposium, pp. 119-134, 1990), Wachel et. al. developed a proposed limit for vibration velocity:

$$V_a = \frac{S_a}{K_v \times SF \times SCF}$$

Where $V_{allow\ peak}$ = allowable peak vibration velocity in pipe span (ips); S_a = allowable endurance limit stress (psi) = 13,000 psi zero-to-peak attributed to API 618; K_v = velocity stress factor which are a function of simplified pipe configurations (cantilever, Z-bend, U-bend, etc.) and for a first mode structural resonance, K_v is tabulated in the 1990 paper and varies between 219 (single span) to 407 (double-Z configuration), with a value of 318 selected in the 1990 paper; SF = safety factor = 2; and SCF = stress concentration factor = 5; which leads to the allowable zero-to-peak pipe vibration velocity:

$$V_a = \frac{S_a = 13{,}000\ psi}{(K_v = 318) \times (SF = 2) \times (SCF = 5)} = 4\ ips\ zero.to.peak$$

Because the method is based on simple, empty pipe configurations, Wachel et. al. introduce adjustment factors for concentrated weights, mode-frequencies other than the first mode, and weight of contents and insulation. For this reason, the 4 ips should not be used generically, for all configurations.

5.4.1.4 Draft API 579-1/ASME FFS-1 Part 16

At the time of this writing, an API 579-1/ASME FFS-1 "Fitness for Service" Working Draft Part 16 "Assessment of Piping Vibration" dated November 26, 2019 (by Lyle Breaux et. al.) is being circulated for comments. Draft Part 16 consists of a three level approach (Level 1, Level 2, and Level 3), each Level with increasing complexity and reduced over-conservatism.

The Level 1 approach itself is subdivided into three methods:

1) Method A Screening Method
2) Method B Graphical Method
3) Method C Calculation Method

In addition to the three methods, Level 1 is based on the classification of the piping being evaluated into one of three groups:

1) Group A main line butt welded and with fittings and joints of low stress intensification factor.
2) Group B main line with fittings and joints of high stress intensification factors, such as threaded and fillet welded joints, and fabricated branch connections.
3) Group C small bore cantilever branch connections, not tied-back to the main line. Group C in turn is subdivided into 10 different categories.

In **Level 1 Method A Screening Method**, the overall vibration amplitude peak velocity V_{peak} or the overall root-mean-square vibration velocity V_{rms} are directly compared to allowable limits that are a function of the configuration of the piping system, as reproduced in Figure 5-2.

In **Level 1 Method B Graphical Method**, the overall RMS velocity V_{rms} is obtained from the vibration measurement. An effective frequency f_e is then calculated following a procedure outlined in draft API 579-1/ASME FFS-1 Part 16 Appendix B. Having V_{rms} and f_e the acceptance criteria are given in graphs that depend on (a) the Group of the pipe configuration, and (b) whether the vibration is random or periodic. An example of such graphs is reproduced in Figure 5-2.

In **Level 1 Method C Calculation Method**, the allowable RMS velocity is calculated as a function of the effective frequency, in the form:

$$V_{rms}{}^{a}(ips) = 0.0394 \times 10^{\left(\frac{log_{10} f_e + 0.48017}{2.127612}\right)}$$

The technical basis for this and other expressions, and for the acceptance graphs, is to be published in an upcoming API 579-1/ASME FFS-1 Part 16 Annex A "Technical Basis and Validation – Assessment of Piping Vibration", not issued at the time of this writing.

Piping System		Waveform Peak V^a_{pk}	Overall RMS V^a_{rms}
Group A	Main Line, no Unsupported Valves	1.41 in/s PK	0.39 in/s RMS
	Main Line, with Unsupported Valves	1.00 in/s PK	0.24 in/s RMS
Group B	Main Line, no Unsupported Valves	0.79 in/s PK	0.25 in/s RMS
	Main Line, with Unsupported Valves	0.57 in/s PK	0.18 in/s RMS
Group C	SBC(1), All Configurations not Listed Below	0.21 in/s PK	0.15 in/s RMS
	SBC, Weldolet fitting with blind flange	1.20 in/s PK	0.85 in/s RMS
	SBC, Weldolet fitting with single valve	0.45 in/s PK	0.32 in/s RMS
	SBC, Weldolet fitting with double valve	0.21 in/s PK	0.14 in/s RMS
	SBC, Weldolet fitting with multiple valves	*0.45 in/s PK*	*0.32 in/s RMS*
	SBC, Contoured fitting with blind flange	0.90 in/s PK	0.63 in/s RMS
	SBC, Contoured fitting with single valve	0.49 in/s PK	0.35 in/s RMS
	SBC, Contoured fitting with double valve	0.34 in/s PK	0.24 in/s RMS
	SBC, Short Contoured Body fitting with blind flange	4.40 in/s PK	3.14 in/s RMS
	SBC, Short Contoured Body fitting with single valve	2.90 in/s PK	2.07 in/s RMS
	SBC, Short Contoured Body fitting with double valve	1.80 in/s PK	1.25 in/s RMS
(1) SBC = Small-Bore Cantilever Branch Connection			

Figure 5-2: Reproduces Table 1 of Draft API 579-1/ASME FFS-1 Part 16, November 2019

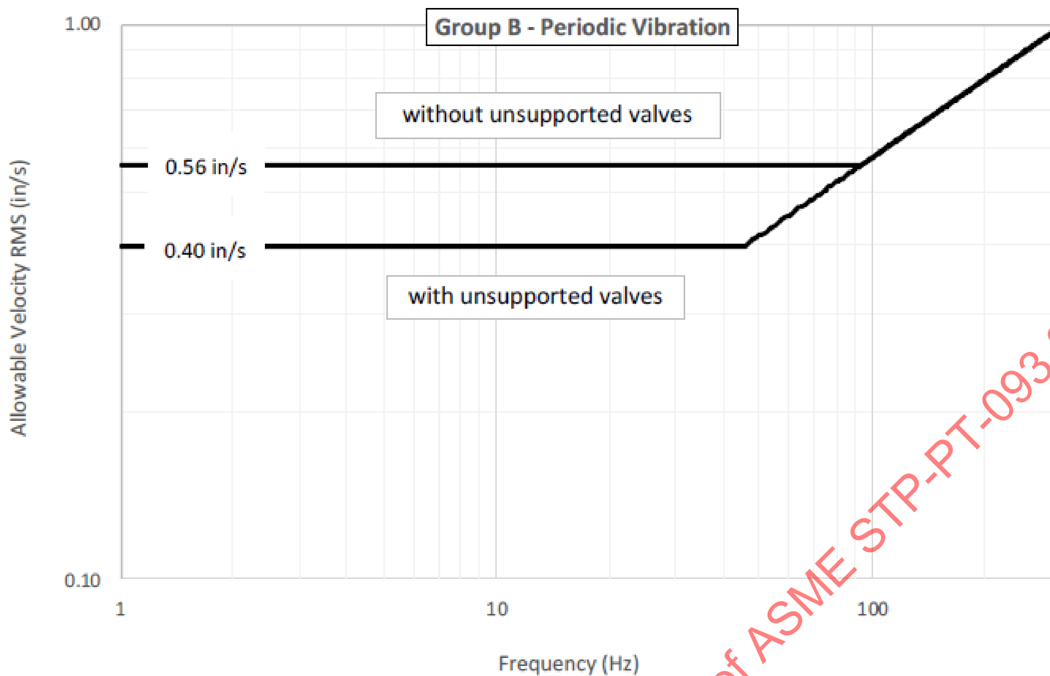

Figure 5-3: Reproduces Figure 14 of Draft API 579-1/ASME FFS-1 Part 16, November 2019

The draft API 579-1/ASME FFS-1 Part 16 **Level 2** procedure is more elaborate, and consists in calculating a fatigue endurance stress limit, and converting the endurance stress limit into an allowable velocity using a stress-to-velocity conversion K_v not unlike Wachel's approach. Two different approaches apply depending on whether the vibration is periodic or random.

The draft API 579-1/ASME FFS-1 Part 16 **Level 3** procedure presents the various options for detailed, layout-specific analyses of the observed vibration.

At the time of this writing, the draft API 579-1/ASME FFS-1 Part 16 Sections on "Remaining Life Assessment", "Remediation", and In-Service Monitoring" remain to be issued.

5.4.1.5 ASME B31.1 Appendix W High-Cycle Fatigue Assessment of Piping Systems

This is a relatively recent Appendix (2018), which describes many steady state vibration failures. The following excerpt from W300 Application defines a "significant stress cycle".

"This Appendix addresses the fatigue evaluation of Code piping subjected to cyclic loadings when the total number of significant stress cycles due to all causes exceeds 100,000. The Appendix may be used subject to the owner's approval. When it is used, the details shall be documented in the engineering design. A significant stress cycle is defined as a cycle with a computed ***stress range****, in accordance with para. 319, greater than 20.7 MPa (****3.0 ksi****) for ferritic steels and austenitic stainless steels. For other materials, or corrosive environments, all cycles shall be considered significant, unless otherwise documented in the engineering design. The allowable displacement stress range requirements of para. 302, using the computed stress range in accordance with para. 319, provide an acceptable method of evaluating piping systems for fatigue when the number of significant stress cycles is less than or equal to 100,000. The piping cyclic loadings may be due to thermal expansion, anchor motion, vibration, inertial loads, wave motion or other sources."*

5.4.1.6 API 618 Criteria

As described in Annex B: of this report, API 618-2008 Section 7.9.4.5.2.4 provides the following acceptance criteria for vibration:

- Allowable vibration displacement of 20 mils peak-to-peak for frequencies below 10 Hz.
- Allowable vibration velocity of approximately 1.25 ips peak-to-peak for frequencies between 10 and 200 Hz.
- A caution that these limits may be unconservative, without further elaboration.

5.4.1.7 API RP-688 Criteria

As described in Annex B: of this report, API RP-688-2012 places a limit on the vibration shaking force peak-to-peak of approximately equal to the pipe NPS divided by 4, in kips. For example, the vibration force from a 12 in. line would be limited to 12/4 = 3 kips.

5.4.1.8 Synthesis and Reconciliation for Beam Mode Vibration

5.4.1.8.1 Differing Criteria

Regarding acceptance criteria for beam mode vibration, we can now summarize the severity criteria provided in the various current standards:

1) Para. 5.4.1 ASME Operation and Maintenance of Nuclear Power Plants Part 3 (OM-3) = screening criterion of 0.5 ips (in/sec) peak velocity; or a stress limit 7 ksi amplitude CS and 13.6 ksi amplitude SS.
2) Para. 5.4.2 UK EI Guideline = velocity (RMS) vs frequency limit 3-zone diagram.
3) Para. 5.4.3 Wachel's velocity (peak) vs. frequency 6-zone diagram; stress-to-velocity conversion factors; correction factors; fixed-fixed beam at resonance with a safety factor of 2 leads to the same 0.5 ips as OM-3, based on 13 ksi stress amplitude limit.
4) Para. 5.4.4 API 579-1/ASME FFS-1 draft Part 16 = multitude of curves velocity (RMS) vs frequency depending on the configuration, three Level approach.
5) Para. 5.4.5 B31.3-2018 Ap.W (new appendix) = stress range of 3 ksi (CS and SS) is considered significant.
6) API 618 = 20 mils peak-to-peak below 10 Hz, and 1.25 ips between 10 Hz and 200 Hz.

As is evident, there are inconsistencies among stress limits and velocity limits which are difficult to reconcile. Two criteria are proposed for the evaluation of the severity of beam mode vibration, as follows.

5.4.1.8.2 The OM-3 Criteria

1) A screening velocity criterion of 0.5 ips amplitude (zero-to-peak).
2) If the screening criterion of 0.5 ips is not met, perform a stress analysis of the vibrating piping configuration, statically imposing the maximum deflected vibration shape, and check the maximum intensified stress amplitude 2iM/Z (refer to paragraph 5.4.1.1.2 for nomenclature) against 7 ksi for carbon steel and 13.6 ksi for stainless steel and Ni alloys.

5.4.1.8.3 The API 579-1/ASME FFS-1 Criteria

Alternatively, the 3-Level approach of draft API 579-1/ASME FFS-1 Part 16 can be applied when (1) Part 16 is finalized, and (2) Part 16 Annex A "Technical Basis and Validation – Assessment of Piping Vibration" has been reviewed and agreed upon.

5.4.2 AIV Shell mode Vibration

5.4.2.1 Carucci-Mueller Criterion

In their 1982 paper "Acoustically Induced PipingVibration in High Capacity Pressure Reducing Systems", 82-WA/PVP-8, 1982, V.A. Carucci and R.T. Mueller propose a sound power level PWL criterion vs. pipe size for carbon steel pipe, reproduced below in Figure 5-4. Carucci-Mueller compared their recommended design limit curve to acoustically-induced failures (black triangle), severe vibration without failures (black squares), and no abnormal behavior (black dot).

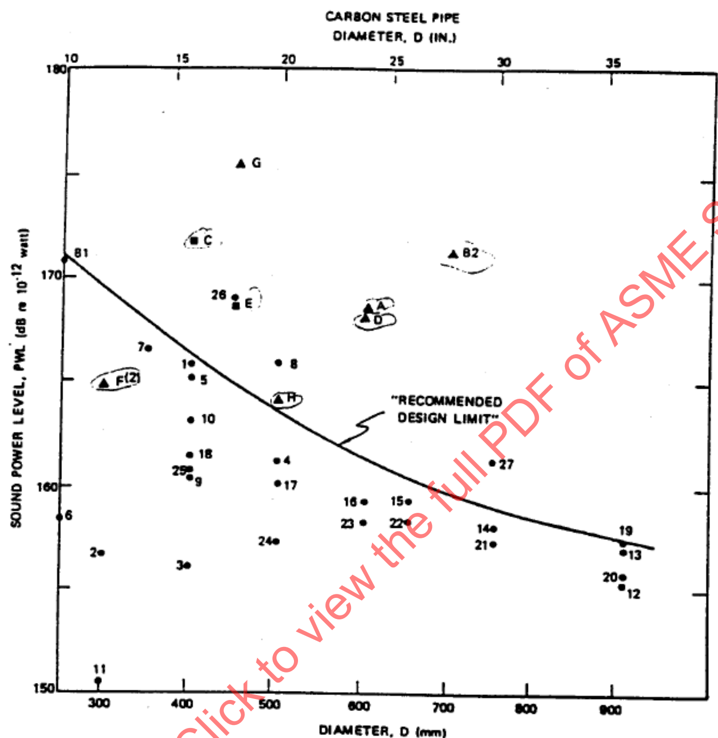

Figure 5-4: Carucci-Mueller Criterion for AIV

Source: "Acoustically Induced Piping Vibration in High Capacity Pressure Reducing Systems", 82-WA/PVP-8, 1982

5.4.2.2 Eisinger-Francis Criterion

In their 1999 paper "Acoustically Induced Structural Fatigue of Piping Systems" Transactions of the ASME Vol. 121, November 1999, F.L. Eisinger and J.T. Francis propose a fatigue failure diagram for piping exposed to acoustic loading that is a function of the input energy and the ratio of diameter over thickness D/t. In this paper, they conclude:

"It is shown that the fatigue limits decrease mildly up to about a value of D/t = 64. For greater values of D/t > 64, a rapid decrease of fatigue resistance occurs, based on available experimental data (Fig. 5 [reproduced here]*). The derived relationship clearly indicates that the fatigue resistance of a piping system can be easily maximized by selecting an appropriate wall thickness for a given pipe diameter."*

In his paper "Acoustic and Turbulence/Flow Induced Vibration in Piping Systems – A Real Problem for LNG Facilities", LNG18 Conference, 11-15 April 2016, Perth, Australia, J. Cowling voices some reservation regarding the Eisinger-Francis AIV fatigue failure criterion.

It is, however, logical to correlate AIV fatigue failure in the shell mode to (1) D/t or better the hoop stress PD/2t which drives the breathing effect, and (2) the presence of discontinuities around the circumference (such as a branch connection or a welded attachment) which cause stress concentrations resulting in fatigue crack formation and propagation.

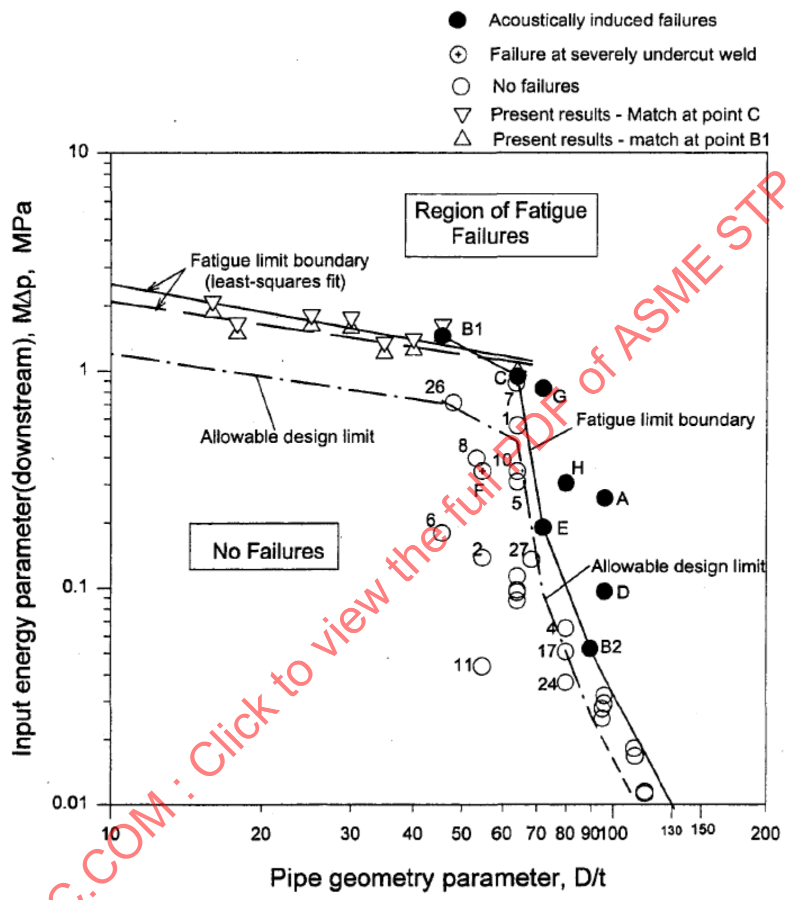

Fig. 5 Fatigue diagram for piping systems exposed to internal acoustic loading relating acoustic input energy parameter $M\Delta p$ to pipe geometry parameter D/t. Shown are originally published data (Eisinger, 1997) with superimposed present data for D/t < 64.

Figure 5-5: Eisinger-Francis Criterion for AIV

Source: "Acoustically Induced Structural Fatigue of Piping Systems" Transactions of the ASME Vol. 121, November 1999

5.4.2.3 AIV Criterion

The high-frequency excitation of shell modes by AIV can result in rapid failures at stress concentrations as described in 4.2.2.3.1. It is therefore important to assess occurrences of high noise in gas systems with a sense of urgency. The assessment procedure is as described in the EI standard, starting with the field measured sound pressure level at a certain distance, converted into sound pressure level at the pipe, and then converted into sound power level at the source. This procedure is described in the EI Standard Section T2.7. The sound power level at the source is then evaluated as described in Section 4.2.2.3.2.

5.5 Determine the Potential Cause(s) of Vibration

Based on the monitoring results of Section 5.3, and the root-cause diagram of Figure 4-1, determine the cause(s) of the observed vibration. The severity criteria outlined in Section 5.4 must guide the engineer in deciding the urgency of this assessment.

As described in Chapter 3, the pressure oscillations causing the vibration may be amplified acoustically and/or structurally by resonance that must be addressed in the assessment of the cause of vibration.

5.6 Proposed Solutions to Piping Vibration

5.6.1 Hydraulic Prevention Solutions

1) Size and install compressor suction and discharge anti-pulsation bottles.
2) Select compressors and pipe configuration combinations that avoid acoustic resonance. An acoustic resonance analysis is recommended in API 618 depending on compressor power and pressure.
3) If acoustic resonance is occurring, de-tune it by changing the pipe length, or by installing an acoustic absorber in the form of a ¼-wave side branch tuned to disrupt the acoustic wave.
4) Replace valves or valve trims to prevent excessive turbulence and noise.
5) Replace orifice plates; if necessary use multiple orifices instead of a single one.
6) Avoid sources of vortex shedding such as protrusions, sharp branch angles, and incorrectly sized orifice plates.
7) Verify the adequacy of the hydraulic sizing of the whole system, as well as the individual piping system components.
8) Verify that the system and the components are being operated within their original design envelope.
9) Verify that components (valves, filters, pulsation bottles, etc.) are inspected and maintained in good operating condition.
10) If high flow velocities cannot be avoided, prevent mixing at sharp 90-degree branch connections, use sweeping tees.
11) Avoid the operating modes (valve alignments, flow rates, pressures, temperatures, bypass) that cause vibration, modify operating procedures.
12) If necessary, install internal vanes to force a more laminar flow.
13) To reduce AIV, per UK EI Guideline Section T10.7.1.1, reduce the flow rate in the line so that the Mach number at valve outlets (ratio of gas velocity at the valve outlet over the sonic velocity at the outlet temperature) does not exceed 0.4 in continuous service, and 0.5 in intermittent service.
14) To reduce AIV, reduce the sound power level by avoiding large pressure drops through a single orifice or valve, using instead multiple orifices and valves properly sized to meet the recommended sound design limit.
15) To reduce AIV in safety valve vent piping, reduce the vent piping diameter by a series of concentric reducers to meet the recommended sound design limit.
16) Plan on vibration monitoring during commissioning, and periodic vibration inspections during operation. Provide operations and maintenance staff the criteria to judge vibration severity.

5.6.2 Mechanical Mitigation Solutions

1) Stiffen compressor foundations and skids, based on torsional analysis.
2) Tie-back to header overly flexible or cantilevered branch lines.
3) Optimize the contour of socket welds and fillet welds.
4) Do not use socket welds for piping larger than NPS 2.
5) Avoid threaded and friction-type joints in high vibration areas.
6) Avoid slip-on flanges in high vibration areas.
7) Do not used soldered or brazed joints.

8) Restore damaged and loose pipe supports.
9) If necessary, add anti-vibration springs.
10) If necessary, add mechanical dampers (viscous or elastomeric) on the larger lines.
11) If necessary, restrain the smaller lines, back to a rigid structure, and verify adequacy of load path and rigid structure.
12) If necessary, use cushioned clamps for vibrating instrument tubing.
13) If necessary, replace metal-to-metal pipe-support contacts by specialty anti-vibration clamps with high-damping elastomeric liner placed between the pipe and the clamp. This high damping elastomer is placed at the pipe-to-clamp top and bottom contact surfaces. Some of these specialty lined clamps have slotted configurations that permit sliding of the clamp assembly for thermal expansion.
14) Avoid layouts that lead to structural resonance. Acoustic and structural resonance analyses are recommended in API 618 depending on compressor power and pressure.
15) Avoid concrete expansion anchors in areas of vibration, preferably use cast-in-place concrete anchor bolts.
16) Plan on vibration monitoring during commissioning, and periodic vibration inspections during operation. Provide operations and maintenance staff the criteria to judge vibration severity.
17) To reduce high noise, where temperature permits, there are insulating materials that consist of a polyurethane foam decoupler lagging wrapped within a high density vinyl jacket.
18) Specify that design and fabrication of pulsation and filter bottle internals take into account the operating conditions these items are subject to, which may include high temperatures, thermal cycling, and vibration. Skip welding of internal choke tube supports should be avoided. Full penetration welds are less susceptible to failure than fillet welds.

ANNEX A: CURRENT B31 VIBRATION RULES

A.1 B31.8 2016 Vibration Rules

815 Equipment Specifications

Certain details of design and fabrication, however, necessarily refer to equipment, such as pipe hangers, vibration dampeners, electrical facilities, engines, compressors, etc. Partial specifications for such equipment items are given herein.

831.4 Reinforcement of Welded Branch Connections

831.4.1 General Requirements.

All welded branch connections shall meet the following requirements:

(a) When branch connections are made to pipe in the form of a single connection or in a header or manifold as a series of connections, the design must be adequate to control the stress levels in the pipe within safe limits. The construction shall accommodate the stresses in the remaining pipe wall due to the opening in the pipe or header, the shear stresses produced by the pressure acting on the area of the branch opening, and any external loadings due to thermal movement, weight, vibration, etc. The following paragraphs provide design rules for the usual combinations of the above loads, except for excessive external loads.

831.4.2 Special Requirements.

In addition to the requirements of para. 831.4.1, branch connections must meet the special requirements of the following paragraphs as given in Table 831.4.2-1: (d) Reinforcement calculations are not required for openings 2 in. (51 mm) and smaller in diameter; however, care should be taken to provide suitable protection against vibrations and other external forces to which these small openings are frequently subjected.

833.8 Flexibility Stresses and Stresses Due to Periodic or Cyclic Fatigue Loading

(a) The stress range in unrestrained piping due to thermal expansion and periodic, vibrational, or cyclic displacements or loads shall be computed as $S_E = M_E/Z$.

834 Supports and Anchorage for Exposed Piping

834.1 General

Piping and equipment shall be supported in a substantial and workmanlike manner, so as to prevent or reduce excessive vibration, and shall be anchored sufficiently to prevent undue strains on connected equipment.

840 Design, Installation, and Testing

840.1 General Provisions

(a) The design requirements of this Code are intended to be adequate for public safety under all conditions encountered in the gas industry. Conditions that may cause additional stress in any part of a line or its appurtenances shall be provided for, using good engineering practice. Examples of such conditions include long self-supported spans, unstable ground, mechanical or sonic vibration, weight of special attachments, earthquake-induced stresses, stresses caused by temperature differences, and the soil and temperature conditions found in the Arctic.

841.1.9 Additional Design Information or Instructions

(d) Design of Metering and Pressure/Flow Control

(1) All piping and piping components, up to and including the outlet stop valve(s) of individual meter and pressure/flow control runs, shall meet or exceed the maximum design pressure of the inlet piping system. Threaded reducing bushings should not be used in pressure/flow control facilities where they are subject to high frequency piping vibrations. The design requirements of para. 840.3 and Table 841.1.6-2 apply to the design requirements of this section.

845.5 Instrument, Control, and Sample Piping
845.5.2 Materials and Design
(i) The arrangement of piping and supports shall be designed to provide not only for safety under operating stresses, but also to provide protection for the piping against detrimental sagging, external mechanical injury, abuse, and damage due to unusual service conditions other than those connected with pressure, temperature, and service vibration.

853.1.7 Monitoring Effects of Pulsation and Vibration.
Facilities exposed to the effects of vibration and pulsation induced by reciprocating compression as well as vibration induced by gas flow or discharge, may be susceptible to fatigue crack growth in fabrication and attachment welds. Susceptible facilities include
(a) compressor station piping having an observed history of vibration
(b) blowdown piping
(c) pulsation bottles and manifolds
(d) piping not meeting the requirements of para. 833.7(a)
Such facilities may warrant engineering assessment and/or nondestructive examination for fatigue cracking in fabrication and attachment welds.

A842.2.5 Design Against Fatigue.
Stress fluctuations of sufficient magnitude and frequency to induce significant fatigue should be considered in design. Loadings that may affect fatigue include
(a) pipe vibration, such as that induced by vortex shedding
… Pipe and riser spans shall be designed so that vortex induced resonant vibrations are prevented, whenever practical. When doing so is impractical, the total resultant stresses shall be less than the allowable limits in para. A842.2.2, and such that fatigue failure should not result during the design life of the pipeline. Additional information for fatigue analysis can be found in API RP 1111, para. 4.5.

A.2 B31.1 2016 Vibration Rules for Metallic Piping

101.5.4 Vibration.
Piping shall be arranged and supported with consideration of vibration [see paras. 120.1(C) and 121.7.5].

101.6 Weight Effects
The following weight effects combined with loads and forces from other causes shall be taken into account in the design of piping. Piping shall be carried on adjustable hangers or properly leveled rigid hangers or supports, and suitable springs, sway bracing, vibration dampeners, etc., shall be provided where necessary.

105.4 Flexible Metal Hose Assembly
(A) Flexible metal hose assemblies may be used to provide flexibility in a piping system, to isolate or control vibration, or to compensate for misalignment.

111.3 Socket Welds
111.3.1 Restrictions on size of socket welded components are given in paras. 104.3.1(B.4), 122.1.1(H), and 122.8.2(C). Special consideration should be given to further restricting the use of socket welded piping joints where temperature or pressure cycling or severe vibration is expected to occur or where the service may accelerate crevice corrosion.

114 Threaded Joints
114.2.1
(A) Threaded joints are prohibited where any of the following conditions is expected to occur:
… *(A.5)* vibration

114 Threaded Joints
114.2.3 … The design and installation of insertion type instrument, control, and sampling devices shall be adequate to withstand the effects of the fluid characteristics, fluid flow, and vibration.

115 Flared, Flareless, and Compression Joints, and Unions
(B) A suitable quantity of the type, size, and material of the fittings to be used shall meet successful performance tests to determine the safety of the joint under simulated service conditions. When vibration, fatigue, cyclic conditions, low temperature, thermal expansion, or hydraulic shock are expected, the applicable conditions shall be incorporated in the test.

115.2 Pressure–Temperature Ratings
Fittings shall be used at pressure–temperature ratings not exceeding the recommendations of the manufacturer.
Unions shall comply with the applicable standards listed within Table 125.1 and shall be used within the specified pressure–temperature ratings. Service conditions, such as vibration and thermal cycling, shall be considered in the application.

117 Brazed and Soldered Joints
117.3 Limitations
(C) Soldered socket-type joints shall not be used in piping subject to shock or vibration.

120 Loads on Pipe-Supporting Elements
(C) Where the resonance with imposed vibration and/or shock occurs during operation, suitable dampeners, restraints, anchors, etc., shall be added to remove these effects.

121.7 Fixtures
121.7.5 Sway Braces. Sway braces or vibration dampeners shall be used to control the movement of piping due to vibration.

122.1 Boiler External Piping; in Accordance With
Para. 100.1.2(A) — Steam, Feedwater, Blowoff, and Drain Piping
(C) Provision shall be made for the expansion and contraction of piping connected to boilers to limit forces and moments transmitted to the boiler, by providing substantial anchorage at suitable points, so that there shall be no undue strain transmitted to the boiler. Steam reservoirs shall be used on steam mains when heavy pulsations of the steam currents cause vibration.

122.3.2 Instrument Piping
(A) Takeoff Connections
(A.1) Takeoff connections at the source, together with attachment bosses, nozzles, and adapters, shall be made of material at least equivalent to that of the pipe or vessel to which they are attached. The connections shall be designed to withstand the source design pressure and temperature and be capable of withstanding loadings induced by relative displacement and vibration. The nominal size of the takeoff connections shall not be less than NPS 1⁄2 (DN 15) for service conditions not in excess of either 900 psi (6 200 kPa) or 800°F (425°C), and NPS 3⁄4 (DN 20) (for adequate physical strength) for design conditions that exceed either of these limits. Where the size of the main is smaller than the limits given above, the takeoff connection shall not be less than the size of the main line.

122.3.2 Instrument Piping
(E) Fittings and Joints
(E.2.3) For pressures up to 175 psi (1 200 kPa) or temperatures up to 250°F (120°C), soldered type fittings may be used with water-filled or air-filled tubing under adjusted pressure–temperature ratings. These fittings are not recommended where mechanical vibration, hydraulic shock, or thermal shock are encountered.

122.8.1 Flammable Gas
(B.3.5) Consideration shall be given in the design to the lower strength and melting point of copper compared to steel. Adequate support and protection from high ambient temperatures and vibration shall be provided.

122.8.2 Toxic Fluids (Gas or Liquid)
(E) Tubing not larger than 5⁄8 in. (16 mm) O.D. with socket welding fittings may be used to connect instruments to the process line. An accessible root valve shall be provided at the process lines to permit isolating the tubing from the process piping. The layout and mounting of tubing shall minimize vibration and exposure to possible damage.

122.8.2 Toxic Fluids (Gas or Liquid)
(F) The provisions of para. 102.2.4 are not permitted. The simplified rules for analysis in para. 119.7.1 (A.3) are not permitted. The piping system shall be designed to minimize impact and shock loads. Suitable dynamic analysis shall be made where necessary to avoid or minimize vibration, pulsation, or resonance effects in the piping.

144 CPS Walkdowns
The Operating Company shall evaluate the effects of unexpected piping position changes, significant vibrations, and malfunctioning supports on the piping system's integrity and safety. Significant displacement variations from the expected design displacements shall be considered to assess the piping system's integrity.
Subsequent evaluations and corrective actions may necessitate activities such as detailed examinations of critical weldments and support adjustments, repairs, and replacement of individual supports and restraints.

Nonmandatory Appendix II
Rules for the Design of Safety Valve Installations
II-2.4 Other Mechanical Loads
Other design mechanical loads that must be considered by the piping designer include the following:
… II-2.4.2 Loads due to earthquake and/or piping system vibration (see para. II-3.4).

II-5.2.1 Locations of Safety Valve Installations.
Safety valve installations should be located at least eight pipe diameters (based on I.D.) downstream from any bend in a high velocity steam line to help prevent sonic vibrations. This distance should be increased if the direction of the change of the steam flow is from vertical upwards to horizontal in such a manner as to increase density of the flow in the area directly beneath the station nozzles. Similarly, safety valve installation should not be located closer than eight pipe diameters (based on I.D.) either upstream or downstream from fittings.

V-7 CPS Position History
V-7.2 Visual Survey
The CPS should be observed visually, as frequently as deemed necessary. Any unusual conditions should be brought to the attention of plant management personnel as prescribed in the procedures of para. V-3.1. Observations should include determination of interferences with or from other piping or equipment,

vibrations, and general condition of the piping system and supports, including but not limited to hangers, guides, restraints, anchors, supplementary steel, and attachments.

V-7.6 Recommendations
After complete examination of the records of observations made in accordance with para. V-7.5, recommendations
for necessary corrective actions should be made by a qualified individual. Evaluations, repairs, and/or modifications should be carried out by qualified personnel for all of the following discrepancies:
… *(C)* excessive piping vibration; valve operator shaking or movement.

A.3 B31.3 2016 Vibration Rules for Metallic Piping

301.5.4 Vibration.
Piping shall be designed, arranged, and supported so as to eliminate excessive and harmful effects of vibration that may arise from such sources as impact, pressure pulsation, turbulent flow vortices, resonance in compressors, and wind.

304.3.5 Additional Design Considerations.
(2) where repetitive stresses may be imposed on the connection by vibration, pulsating pressure, temperature cycling, etc.
In such cases, it is recommended that the design be conservative and that consideration be given to the use of tee fittings or complete encirclement types of reinforcement.

311.2.4 Backing Rings and Consumable Inserts
(a) If a backing ring is used where the resulting crevice is detrimental (e.g., subject to corrosion, vibration, or severe cyclic conditions), it should be removed and the internal joint face ground smooth. When it is impractical to remove the backing ring in such a case, consideration shall be given to welding without backing rings or to the use of consumable inserts or removable nonmetallic backing rings.
(b) Split backing rings shall not be used under severe cyclic conditions.

313 Expanded Joints
(b) Consideration shall be given to the tightness of expanded joints when subjected to vibration, differential expansion or contraction due to temperature cycling, or external mechanical loads.

315 Tubing Joints
315.1 General
In selecting and applying flared, flareless, and compression type tubing fittings, the designer shall consider the possible adverse effects on the joints of such factors as assembly and disassembly, cyclic loading, vibration, shock, and thermal expansion and contraction.

321 Piping Support
321.1 General
The design of support structures (not covered by this Code) and of supporting elements (see definitions of piping and pipe supporting elements in para. 300.2) shall be based on all concurrently acting loads transmitted into such supports. These loads, defined in para. 301, include weight effects, loads introduced by service pressures and temperatures, vibration, wind, earthquake, shock, and displacement strain (see para. 319.2.2).

321.1.1 Objectives.
The layout and design of piping and its supporting elements shall be directed toward preventing the following:

… *(e)* resonance with imposed or fluid-induced vibrations

321.1.4 Materials
… *(b)* Gray, ductile, and malleable iron may be used for rollers, roller bases, anchor bases, and other supporting
elements subject chiefly to compressive loading. Gray iron is not recommended if the piping may be subject to impact-type loading resulting from pulsation or vibration.

Chapter VIII
Piping for Category M Fluid Service
M301.5.4 Vibration.
Suitable dynamic analysis, such as computer simulation, shall be made where necessary to avoid or minimize conditions that lead to detrimental vibration, pulsation, or resonance effects in the piping.

The B31.3 restrictions for severe cyclic service should be considered for gas compressor station piping:

- Only fittings that are seamless, or welded with 100% RT of welds, or cast with factor $E_c \geq 0.90$ should be used.
- Fabricated branch connections should be the stub-in type.
- Flanges should be welding neck.
- High strength bolting should be used.
- Threaded joints should be prohibited, except for non-moment bearing connections.
- Socket welding should be prohibited above NPS 2.
- Tubing joints may be used only if safeguarded.
- Gray and malleable iron should be prohibited
- Welding procedure should provide a smooth, regular, fully penetrated inner surface.
- More stringent weld imperfection acceptance criteria should apply.
- Backing rings should be removed prior to service.

Chapter IX
High Pressure Piping
K301.5.4 Vibration.
Suitable dynamic analysis shall be made where necessary, to avoid or minimize conditions that lead to detrimental vibration, pulsation, or resonance effects in the piping.

Part 4
Fluid Service Requirements for Piping Joints
K310 General
Joints shall be suitable for the fluid handled, and for the pressure-temperature and other mechanical loadings expected in service.
Factors such as assembly and disassembly (if applicable), cyclic loading, vibration, shock, bending, and thermal expansion and contraction of joints shall be considered in the engineering design.

K314 Threaded Joints
K314.1 General
Except as provided in paras. K314.2 and K314.3, threaded joints are not permitted.
(a) Layout of piping shall be such as to minimize strain on threaded joints that could adversely affect sealing.
(b) Supports shall be designed to control or minimize strain and vibration on threaded joints and seals.

Chapter X
High Purity Piping
U306 Fittings, Bends, Miters, Laps, and Branch Connections
U305.6 Tube Fittings
(a) Tube fittings not listed in Table 325.1 or Appendix A shall meet the pressure design requirements described in para. 302.2.3 and the mechanical strength requirements described in para. 303.
(b) Compression-type tube fittings may be used in accordance with para. U315.2 provided that the type of fitting selected complies with the following:
(1) The gripping action of the fitting shall provide vibration resistance as demonstrated by exhibiting a stress intensity factor equal to or less than 1.5.

U315.1 General
In selecting and applying compression, face seal, and hygienic clamp-type tube fittings, the designer shall consider the possible adverse effects on the joints of such factors as assembly and disassembly, cyclic loading, vibration, shock, and thermal expansion and contraction. See para. FU315.

B31.3 Appendix F Guidance and Precautionary Considerations
F301.10.3 Severe Cyclic Conditions. Designating piping as being under severe cyclic conditions should be considered when piping is subjected to both a high stress range and many cycles. The phrase *many cycles* can be taken as when the stress range factor, *f*, is less than the maximum, *fm*. The phrase *high stress range* is normally taken as when the calculated stress range approaches the allowable stress range. Examples include piping associated with batch chemical reactors that cycle more frequently than once a day and piping that has a reasonable likelihood of vibrating.
Frequently, failures occur at small branch connections attached to main piping runs that do not have a high stress range. When experience shows that these small branch connections might be vulnerable to fatigue failure, consideration should be given to designating such piping as being under severe cyclic conditions. See the following references for guidance on the design of small branch connections to avoid fatigue failure:
(a) Guidelines for the Avoidance of Vibration Induced Fatigue Failure in Process Pipework, published by Energy Institute
(b) Design Guideline for Small Diameter Branch Connections, published jointly by Gas Machinery Research Council, Pipeline Research Council International, and Southwest Research Institute
More conservative approaches to designating piping as being under severe cyclic conditions should be taken when the fluid handled is toxic, flammable, or damaging to human tissue; when failure of the piping would be costly; and also when examination of the piping during operation or normal outages is impracticable.

F309 Bolting
F309.1 General
The use of controlled bolting procedures should be considered in high, low, and cycling temperature services,
and under conditions involving vibration or fatigue, to reduce
… *(b)* the possibility of stress relaxation and loss of bolt tension

Appendix G
Safeguarding
G300.3 Engineered Safeguards
Engineered safeguards that may be evaluated and selectively applied to provide added safeguarding include
(a) means to protect piping against possible failures, such as
… *(3)* damping or stabilization of process or fluid flow dynamics to eliminate or to minimize or protect against destructive loads (e.g., severe vibration pulsations, cyclic operating conditions)

Appendix X
Metallic Bellows Expansion Joints
X301.1.3 Other Loads. Other loads, including dynamic effects (such as wind, thermal shock, vibration, seismic forces, and hydraulic surge); and static loads, such as weight (insulation, snow, ice, etc.), shall be stated.

A.4 B31.4 2016 Vibration Rules for Metallic Piping

Chapter II Design
401.1.2 Sustained Loads.
Sustained loads are those arising from the intended use of the pipeline system and loads from other sources. The weight of the pipeline, including components, fluids, and slurries, and loads due to pressure are examples of sustained loads. Soil cover, external hydrostatic pressure, and vibration due to equipment are examples of sustained loads from other sources. Reaction forces at supports from sustained loads and loads due to sustained displacement or rotations of supports are also sustained loads.

401.2.3.5 Vibration.
Loads resulting from vibration (including Karmon vortex effect) and resonance shall be considered.

402 Calculation of Stresses
402.1 General
Circumferential, longitudinal, shear, and equivalent stresses shall be considered, taking into account stresses from all relevant sustained, occasional, construction, and transient loads, including vibration, resonance, and subsidence

403 Criteria for Pipelines
403.1 General
Design and installation analyses shall be based upon accepted engineering methods, material strengths, and applicable design conditions.
… the design shall provide reasonable protection to prevent damage to the pipeline from unusual external conditions that may be encountered in river crossings, offshore and inland coastal water areas, bridges, areas of heavy traffic, long self-supported spans, unstable ground, vibration, weight of special attachments, or forces resulting from abnormal thermal conditions.

403.5 Criteria to Prevent Fatigue
The pipeline shall be designed, installed, and operated to limit stress fluctuations to magnitudes and frequencies
that will not impair the serviceability of the pipeline. Loads that may cause fatigue include internal pressure variations, currents, and vibrations induced by vortex shedding. Pipeline spans shall be designed to prevent vortex-induced resonant vibrations when practical. When vibrations must be tolerated, the resulting stresses due to vibration shall be included in allowable stresses listed in para. 403.3.1. If alternative acceptance standards for girth welds in API 1104 are used, the cyclic stress analysis shall include the determination of a predicted fatigue spectrum to which the pipeline is exposed over its design life. See Chapter 2 of ASMEB31.3 for guidance.

404.3.3.4 Design.
When the design meets the limitations on geometry contained herein, the rules established are valid and meet the intent of the Code. These rules cover minimum requirements and are selected to ensure satisfactory performance of extruded headers subjected to pressure. Extruded headers shall be designed to withstand forces and moments applied to the branch by thermal expansion and contraction; by vibration; by deadweight of piping, valves, fittings, covering, and contents; and by earth settlement.

404.3.4 Welded Branch Connections.
Welded branch connections shall be as shown in Figs. 404.3.4-1, 404.3.4-2, and 404.3.4-3. Design shall meet the minimum requirements listed in Table 404.3.4-1 and described in paras. 404.3.4(a) through 404.3.4(d). Where reinforcement is required, paras. 404.3.4(e) and 404.3.4(f) shall apply.
… *(c)* Reinforcement for branch connections with hole cut NPS 2 or smaller is not required (see Fig. 404.3.4-3 for typical details); however, care shall be taken to provide suitable protection against vibrations and other external forces to which these small branch connections are frequently subjected.

404.3.5 Reinforcement of Single Openings
(a) When welded branch connections are made to pipe in the form of a single connection, or in a header or manifold as a series of connections, the design shall be adequate to control the stress levels in the pipe within safe limits. The construction shall take cognizance of the stresses in the remaining pipe wall due to the opening in the pipe or header, the shear stresses produced by the pressure acting on the area of the branch opening, and any external loading due to thermal movement, weight, vibration, etc., and shall meet the minimum requirements listed in Table 404.3.4-1.

404.8.4 Sleeve, Coupled, and Other Patented Joints.
… *(a)* a production joint has been subject to proof tests to determine the safety of the joints under simulated service conditions. When vibration, fatigue, cyclic conditions, low temperature, thermal expansion, or other severe conditions are anticipated, the applicable conditions shall be incorporated in the tests.

404.9.3 Braces.
Braces and damping devices may occasionally be required to prevent vibration of piping.

435.4 Manifolds
435.4.6 Final assembly of all components shall minimize locked-in stresses. The entire assembly shall be adequately supported to provide minimum unbalance and vibration.

435.5 Auxiliary Piping
435.5.2 All welded auxiliary lines shall be assembled in accordance with the requirements of this Code with special provisions as required for assembly to minimize locked-in stress, and for adequate support or restraint to minimize vibration.

Chapter IX
Offshore Liquid Pipeline Systems
A402.3.4 Strength Criteria During Installation and Testing
… *(c) Design Against Fatigue.* The pipeline shall be designed and installed to limit anticipated stress fluctuations to magnitudes and frequencies that will not impair the serviceability of the installed pipeline. Loads that may cause fatigue include wave action and vibrations induced by vortex shedding. Pipelines and riser spans shall be designed to prevent vortex-induced resonant vibrations, when practical. When vibrations must be tolerated, the resulting stresses due to vibration shall be considered. If alternative acceptance standards for girth welds in API 1104 are used, the cyclic stress analysis shall include the determination of a predicted fatigue spectrum to which the pipeline is exposed over its design life.

A402.3.5 Strength Criteria During Operations
… *(c) Design Against Fatigue.* The pipeline shall be designed and operated to limit anticipated stress fluctuations to magnitudes and frequencies that will not impair the serviceability of the pipeline. Loads that may cause fatigue include internal pressure variations, wave action, and pipe vibration, such as that induced by vortex