Guidelines for the Avoidance of Vibration Induced Fatigue in Process Pipework

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THE MARINE TECHNOLOGY
DIRECTORATE LIMITED

Guidelines
for the Avoidance
of Vibration Induced
Fatigue in Process Pipework

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Foreword

The project leading to these Guidelines was initiated in response to a growing number of pipework failures in process systems of both offshore and onshore installations. A database of actual and potential failures was compiled and, when analysed, indicated that various contributing factors were present, most importantly changes in process operating conditions and a greater use of thin-walled pipework. The Marine Technology Directorate Ltd (MTD) therefore initiated a Joint Industry Project to produce appropriate engineering Guidelines, with the aim of minimising the risk related to fatigue failure of process piping systems.

These Guidelines were prepared under contract to MTD, on behalf of the sponsors, by ATL Consulting Group Ltd and Mitsui Babcock - Technology Centre as the principal consultants, and with assistance from Shell Global Solutions. It was sponsored by the 16 organisations listed below and was managed for MTD by Mr R W Barrett.

Project Sponsors

AMEC Process and Energy

Amerada Hess

Chiyoda Corporation

Foster Wheeler Petroleum Development

Health & Safety Executive

Kvaerner Oil & Gas

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Norsk Hydro

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Shell UK Exploration & Production

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Toyo Engineering Corporation

Steering Group

A Steering Group comprising representatives of participants, MTD, and the Technical Services Contractors provided the forum for both verbal and written discussion of the content of these Guidelines during its preparation. During the period of the project, the following individuals served on the Steering Group which was chaired by Mr T McMahon (Mobil Technology Company) and latterly Mr C R Howard (Shell UK Exploration & Production):

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P. Tring	AMEC Process and Energy Ltd
R. Tyagi	Minerals Management Service (USA)
T. Williams	Scottish Enterprise
J. Young	McDermott Marine Construction

Technical Services Contractors

The preparation of these Guidelines was undertaken principally by the following organisations and the individuals who worked on its various elements were:

ATL Consulting Group Ltd	J. R. Baker W. J. Skailes B. M. H. Soper R. J. Swindell
Mitsui Babcock - Technology Centre	T. Grant W. J. G. Little C. McGlinchey D. H. Mitchell

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A further acknowledgement is made to The Health & Safety Executive for assisting with the database at an early stage of the project.

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1.0 Introduction

Process piping systems have traditionally been designed on the basis of a static analysis with little attention paid to vibration induced fatigue. Vibration, especially with regard to small bore connections, has normally been considered on an ad-hoc basis after the design stage or following a fatigue failure [1].

In recent years there have been a number of factors which have led to an increasing incidence of vibration related fatigue failures in piping systems both on offshore installations and on petrochemical plants. The most significant factors have been:

- increased flowrates as a result of debottlenecking and the relaxation of erosion velocity limits, with a correspondingly greater level of turbulent energy in process systems;
- a greater use of thin walled pipework made possible by the use of new higher strength materials (e.g. duplex stainless steel alloys), resulting in more flexible pipework and higher stress concentrations at small bore connections.

In response to the problem of vibration related fatigue in process piping systems, a Joint Industry Project was set up by the Marine Technology Directorate Limited (MTD). The result is this set of Guidelines which are designed to:

- provide awareness of the most common vibration excitation and response mechanisms;
- provide a qualitative assessment of the likelihood of failure due to vibration;
- identify the geometric features that can contribute to fatigue failures of small bore connections;
- suggest potential modifications to reduce the likelihood of failure associated with vibration related fatigue;
- identify suitable survey and assessment techniques which can be used to evaluate a piping system in its operational environment.

The Guidelines can be used at any time during the design stage from concept to detailed design. In addition they also apply to existing plants that are undergoing modification or when the process conditions in the plant are being changed.

Although the impetus for the Guidelines has come from the offshore industry, they have been designed to be equally applicable to onshore petrochemical plants and other process industries.

The Guidelines are based on four excitation mechanisms; flow induced turbulence, pulsation, high frequency acoustic excitation and mechanical excitation. However pulsation has not been analysed as part of this project, but limited guidance is given to enable problem areas to be recognised and followed up via references.

The Guidelines are divided into four major sections.

- The first (Section 2) outlines the different vibration mechanisms and their effect on pipework and small bore connections.
- The second (Section 3) is the methodology to assess the likelihood of failure of piping systems and small bore connections for the various excitation mechanisms.
- The third (Section 4) outlines possible design solutions or best practices for piping systems or small bore connections that are susceptible to vibration.
- The fourth (Section 5) outlines survey methods for vibration measurement and assessment of pipework fatigue in operating plant.

Where the type of excitation or complexity of response is outside the scope of these Guidelines, specialist advice should be sought.

Use of colour

To assist the user in working through the assessment methodology in these guidelines, a colour tint has been used as follows:

Stage 1	Identification of Excitation Mechanisms	colour
Stage 2	Detailed Screening of Main Pipe	colour
Stage 3	Detailed Screening of Small Bore Connections	colour

The detailed information necessary to evaluate the above stages is covered in Appendices A1, A2 and A3 and the same colours have been used in these appendices to aid identification with respect to the three stages.

The items that are required to be completed in each stage are shown in red in the worksheets in Appendix B.

2.0 Vibration Mechanisms

The purpose of this section is to give an overview of the different types of excitation and the accompanying response that will typically be encountered in offshore and onshore oil, gas and chemical plants. Before the discussion of each individual excitation mechanism, a general overview of pipework vibration normally encountered in such plant will be given.

The Guidelines will determine the likelihood of failure but not the consequence of failure. The consequence of failure will be different for every operator and thus its calculation is the operator's responsibility. Section 2.3 outlines how the likelihood of failure calculations from these Guidelines can be linked with a consequence of failure calculation to give the overall risk. Mitigation measures based on the overall risk are the responsibility of the operator.

2.1 Excitation Mechanisms and Response

The following section is a brief overview of the different excitation mechanisms covered in the Guidelines. Although not exhaustive, it covers the major sources of excitation. The different mechanisms covered are:

- flow induced turbulence;
- high frequency acoustic excitation;
- mechanical excitation;
- pulsation
  - reciprocating machinery
  - rotating stall
  - periodic flow induced excitation.

These Guidelines are for 'steady state' operation of plant. They do not consider:

- the response of the structure to impulse loading (slugflow, pressure transients as a result of rapid valve closure, impact loading etc);
- environmental excitation mechanisms such as inertial effects on pipework on ships and floating facilities;
- wave induced motion on risers and seismic excitation and topside vibration due to ship collision.

However, the design of small bore connections in line with the recommendations in these Guidelines will minimise the likelihood of failure due to these excitation mechanisms.

2.1.1 Introduction to Vibration

Vibration can be defined as the study of oscillatory motion; the ultimate goal is to determine the effect of vibration on the performance and safety of systems and to control its effects.

Consider a simple mass on a spring as illustrated in Figure 2-1.

As can be seen when the mass is pulled down and then released, the spring extends, then contracts and continues to oscillate over a period of time. The resulting frequency of oscillation is known as the ***natural frequency*** of the system, and is controlled by the system's mass and stiffness i.e.

$$\text{Natural frequency:} \quad f_n = \frac{1}{2\pi}\sqrt{\frac{\text{stiffness}}{\text{mass}}} \tag{1}$$

Very little energy is required to excite the natural frequency of a system, as the structure 'wants' to respond at this particular frequency. If damping is present then this will dissipate the dynamic energy and reduce the vibrational response. The resulting vibration can be defined in terms of:

- displacement
- velocity
- acceleration

The amplitude for all three parameters is dependent on frequency (see Figure 2-2). Displacement is frequency dependent in a manner which results in a large displacement at low frequencies and small displacements at high frequencies for the same amount of energy. Conversely acceleration is weighted such that the highest amplitude occurs at the highest frequency. Velocity gives a more uniform weighting over the required range, is most directly related to the resulting dynamic stress and is therefore most commonly used as the measurement of vibration. This is why the visual observation of pipework vibration (displacement) is not a fail safe method of assessing the severity of the problem.

Any structural system, such as a pipe, will exhibit a series of natural frequencies which depend on the distribution of mass and stiffness throughout the system. The mass and stiffness distribution are influenced by pipe diameter, material properties, wall thickness, location of lumped masses (such as valves) and pipe supports and also fluid density (liquid versus gas). It should be noted that pipe supports designed for static conditions may act differently under dynamic conditions.

Each natural frequency will have a unique deflection shape associated with it, which is called the ***mode shape***. The response of the pipework to an applied excitation is dependent upon the relationship between the frequency of excitation and the system's natural frequencies, and the location of the excitation relative to the mode shape.

Excitation can either be ***tonal*** i.e. energy is only input at discrete frequencies, or ***broad-band*** i.e. energy is input over a wide frequency range.

There are several different types of response that can exist depending on how the excitation frequencies match the system natural frequencies:

<u>Tonal Excitation - Resonant</u>

If the frequency of the excitation matches a natural frequency then a ***resonant*** condition is said to exist. In this situation, all the excitation energy is available to 'drive' the natural frequency of the system, and, as noted previously, only a small amount of excitation at a natural frequency is required to generate substantial levels of vibration, if the system damping is low. To avoid vibration due to tonal excitation, where there is interaction between the excitation and response, the excitation frequency should not be within ±20% of the system natural frequencies.

Tonal Excitation - Forced
If the frequency of the excitation does not match a natural frequency, then vibration will still be present at the excitation frequency, although at much lower levels than for the resonant case. This is known as ***forced vibration*** and can only lead to high levels of vibration if the excitation energy levels are high, relative to the stiffness of the system.

Broad-band Excitation
If the excitation is ***broad-band*** then there is a probability that some energy will be input at the system natural frequencies. Generally, response levels are lower than for the purely resonant vibration case described above because the excitation energy is spread over a wide frequency range.

In the majority of cases, the vibration generated in the pipework may lead to high cycle fatigue of components (such as small bore connections) or, in extreme cases, to failure at welds in the main line itself.

There are a variety of excitation mechanisms which can be present in a piping system; these are described in the next sections. For a more detailed introduction to vibration see references [2] and [3] and for applications to process piping systems see [4] and [6].

2.1.2 Flow Induced Turbulence

Turbulence will exist in most piping systems encountered in practice. In straight pipes it is generated by the turbulent boundary layer at the pipe wall, the severity of which depends upon the flow regime as defined by Reynolds number. However, for most cases experienced in practice the dominant sources of turbulence are major flow discontinuities in the system. Typical examples are process equipment, partially closed valves, short radius or mitred bends, tees or expanders.

This in turn generates potentially high levels of broad band kinetic energy which can propagate through the system (see Figure 2-3). Although the energy is distributed across a wide frequency range, the majority of the excitation is concentrated at low frequency (typically below 100 Hz); the lower the frequency, the higher the level of excitation from turbulence (Figure 2-4). This leads to excitation of the low frequency vibration modes of the pipework, in many cases causing visible motion of the pipe and, in some cases, the pipe supports.

2.1.3 High Frequency Acoustic Excitation

In a gas system, high levels of high frequency acoustic energy can be generated by a pressure reducing device such as a relief valve, control valve or orifice plate. Although a comparatively rare form of failure, acoustic fatigue is of particular concern as it tends to affect safety related (e.g. relief and blowdown) systems.

In addition, the time to failure is short (typically a few minutes or hours) due to the high frequency response. As well as giving rise to high tonal noise levels external to the pipe, this form of excitation can generate severe high frequency vibration of the pipe wall. The vibration takes the form of local pipe wall flexure (the shell flexural modes of vibration) resulting in potentially high dynamic stress levels at circumferential discontinuities on the pipe wall, such as small bore connections, fabricated tees or welded pipe supports.

The high noise levels are generated by high velocity fluid impingement on the pipe wall, turbulent mixing and, for choked flow, shockwaves downstream of the flow restriction. They are a function of the pressure drop across the pressure reducing device and the gas mass flow rate.

2.1.4 Mechanical Excitation

Most of the problems of this nature encountered have been associated with reciprocating compressors and pumps. In such machines, forces directly load the pipework connected to the machine or cause vibration of the support structure which in turn results in excitation of the pipework supported from the structure. Normally high levels of vibration and failures only occur where the pipework system has a natural frequency at the multiple of the running speed of the machine. As this type of equipment has many harmonics of the running speed with appreciable energy levels which can excite the system, the problem can occur at many orders of the running speed.

2.1.5 Pulsation

In the same way as structures exhibit natural frequencies the fluid within piping systems also exhibits acoustic natural frequencies. These are frequencies at which standing wave patterns are established in the liquid or gas. Acoustic natural frequencies can amplify low levels of pressure pulsation in a system to cause high amplitudes of pressure pulsation, which can lead to excessive shaking forces.

In the low frequency range (typically less than 100 Hz), acoustic natural frequencies are dependent on the length of the pipe between acoustic terminations and process parameters (e.g. molecular weight, density and temperature). Acoustic terminations can generally be designated as closed (e.g. a close valve) or open (e.g. entry to a vessel such as a knock out drum). In the high frequency range (typically 1 kHz and higher) the acoustic natural frequencies are generally associated with short sections of pipe and are largely dependent on pipe diameter and process parameters. If there is any change in process parameters (eg molecular weight or temperature) it is **critical** that the pipework's design is reassessed for pulsation.

Pressure pulsation is a tonal form of excitation whereby dynamic pressure fluctuations are generated in the process fluid at discrete frequencies. The pressure pulsation results in dynamic force being applied at bends, reducers and other changes of section. For pulsation to result in significant levels of vibration, the dynamic force must couple to the structural response of the pipework in both the frequency and spatial domains.

In the frequency domain (with reference to Figure 2-5), to experience high levels of vibration the frequency of the source of excitation (a) must correlate with the acoustic natural frequency (b) resulting in high levels of pulsation (c). This in turn must correlate with the structural natural frequency (d) to cause high levels of vibration (e), as shown in the figure at 40 Hz.

However, if the structural natural frequency (d) does not correlate with the pulsation (c), as shown in the figure at 60 Hz, then there will be pulsation but only a low level of forced vibration at 60 Hz (e). The amplitude of this forced vibration will be significantly lower than the resonant response. Furthermore, if the acoustic natural frequency (b) does not correlate with the excitation (a) then there will be little pulsation and therefore lower vibration levels (e), as shown in the figure at 20 Hz.

Therefore, for the most serious vibration problems the frequency of excitation, acoustic natural frequency and structural natural frequency must correlate (i.e. a resonant condition). However, high levels of vibration (non-resonant) can be experienced if there are significant levels of excitation present in the system.

In the spatial domain, it is the location and phase of the dynamic force relative to the structural mode shape (see section 2.1.1) that is important. The mode shape determines the pipework's receptance of dynamic force. This means that if the dynamic force occurs at a structural node of vibration (e.g. at a pipework anchor) then this will not result in vibration. However, if the dynamic force is located elsewhere, and if the force and deflection of the mode shape are in phase, high levels of vibration will result.

The predominant sources of low frequency pressure pulsation encountered in the oil and petrochemical industry are described below.

2.1.5.1 Reciprocating Pumps and Compressors

Reciprocating pumps and compressors generate oscillating pressure fluctuations in the process fluid simply by virtue of the way in which they operate.

The dominant excitation frequencies relate to pump operating speed or multiples thereof, and the resulting pressure fluctuations can be further amplified by acoustic natural frequencies of the system.

This in itself can lead to high levels of dynamic pressure (and hence shaking forces) which can cause a forced vibration problem. However extreme levels of vibration can be generated if coincidence occurs with a structural natural frequency of the piping system.

Detailed analyses are often undertaken by the manufacturers (or suppliers) of reciprocating compressors and pumps to predict the pressure pulsation levels in the system. This analysis is usually undertaken to meet the requirements of API 618 [7] (reciprocating compressors) and API 674 [8] (reciprocating pumps).

2.1.5.2 Periodic Flow Induced Excitation

Flow over a body causes vortices to be shed at specific frequencies according to the equation:

$$f = \frac{S\mathrm{v}}{d} \tag{2}$$

where v is the fluid velocity, d is the representative dimension of the component (see Figure 2-6 as an example) and S is the Strouhal number. Strouhal number is dependent on the shape of the component and the flow regime. Given the range of shapes and Reynolds numbers which can occur, Strouhal numbers can vary widely over the range 0.1 to 1.0 [3].

Periodic pressure disturbances in the low frequency range can occur at:

- flow past the end of a dead leg branch;

- flow past components inserted in the fluid stream or non-symmetrical flow at vessel outlets;

- thermowells are a special case of the previous point and are considered separately.

These mechanisms seldom cause failure on their own. In general there must be interaction with some other mechanism, such as correlation with a structural natural frequency or an acoustic natural frequency, before sufficient energy is generated to cause significant vibration. One feature of this form of excitation is lock-on between the excitation and response frequencies. For this reason separation of ±20% must be maintained over the flow regimes of interest.

<u>‘Dead Leg’ Branches</u>

Gas systems, at relatively high flow velocities, can exhibit another form of tonal excitation although this is comparatively rare. This is generated when flow past the end of a ‘dead leg’ branch generates an instability at the mouth of the branch connection (see Figure 2-6), similar to blowing across the top of a bottle. Process examples are a branch line with a closed end, such as a relief line or a recycle line with the valve shut. This leads to the generation of vortices at discrete frequencies which, if these frequencies coincide with an acoustic natural frequency of the branch, can generate high levels of pressure pulsation. The generation of the flow instability is heavily dependent on flow rate, and the highest flow rate may not be the worst case condition.

Flow over Components in Fluid Stream

Flow over bodies or across edges of components in the gas stream can cause vortex shedding in a similar way to the 'Dead Leg Branches'. These periodic disturbances in the flow pattern interact with the system acoustics to increase the levels of pulsation in the system. Because of the range of shapes and Reynolds numbers which can occur, Strouhal numbers can vary widely over the range 0.1 to 1.0. Each case should be assessed for the particular geometry, flow regime and possible acoustic modes. As a result this subject is outside the scope of these Guidelines and a separate assessment as to the potential high pulsation levels should be made.

Thermowells/Probes

In the case of thermowells or other probes inserted in the flow stream (e.g. chemical injection quils or flow measurement probes), the vortex shedding should not correlate with the structural natural frequency of the probe. When this does occur the thermowell/probe is excited like a tuning fork and fatigue failure of the thermowell/probe occurs in a relatively short time scale. The design of thermowells is normally carried out to ANSI/ASME PTC 19.3 temperature measurement [9], but it is known that this can be non-conservative in certain situations. It takes no account of the variation in Reynolds number with flow, the supporting arrangement provided by the thermowell upstand, the gasket stiffness or the schedule of the main pipe wall.

Therefore, if the connection being considered is a thermowell or other probe, then a dynamic assessment (calculation of the mechanical natural frequency and vortex shedding frequency) will be required for either a new installation or a change in process, to ensure that vortex shedding does not occur. This assessment is outwith the remit of these Guidelines.

2.1.5.3 Centrifugal Compressors (Rotating Stall)

Centrifugal compressors can generate tonal pressure pulsations at lower flow conditions [5]. Certain compressor designs can experience a flow instability caused by rotating stall, which leads to a tonal pressure component at a sub-synchronous frequency (typically 10 - 80% of rotor speed). While the level of this excitation is generally not high enough to lead to a rotor mechanical vibration problem, it can generate significant levels of pressure pulsation, particularly in the discharge piping, if it excites an acoustic natural frequency of the system (see Figure 2-7). The susceptibility to rotating stall is a function of wheel geometry, speed and process conditions which should be addressed by the compressor designer. Typically the last wheel in a stage is the most susceptible.

2.2 Failure Mode

Vibration of the pipework causes dynamic stresses (cyclic stresses) which, if above a critical level, can result in the initiation of a fatigue crack. Fatigue cracking, if unchecked, can lead to through thickness fracture and subsequent rupture. Fatigue life of the component can be relatively short (minutes or days). However, if the vibration is intermittent the fatigue life of the component can be longer, even years. Dynamic stress level is therefore the primary parameter which determines whether failure will occur by fatigue (Section 5).

2.3 Likelihood of Failure

The likelihood of failure (L.O.F.) is a conservative assessment to be used for screening purposes. The likelihood of failure is not an absolute probability of failure nor an absolute measure of failure. The calculations are based on simplified models to ensure ease of application and are necessarilyconservative.

The purpose of these Guidelines are to identify problem areas, assess their likelihood of failure and provide possible design solutions.

2.4 Determination of Overall Risk

These Guidelines do not purport to address the consequence of failure. The consequence of failure is the responsibility of the user. However, the likelihood of failure which results from these Guidelines can be used in combination with a consequence of failure calculation to determine the overall risk of a system or component. A typical criticality matrix is shown in Figure 2-8 where the likelihood of failure is on the vertical axis and the consequence of failure is on the horizontal axis. Mitigation measures, depending on the level of risk, are the responsibility of the user. However the design solutions in Section 4 of these Guidelines can be used to reduce the likelihood of failure of a specific system.

Consequence of failure calculations usually require the knowledge of the failure mode for the system. For vibration excitation mechanisms covered in these Guidelines the failure mechanism is fatigue cracking. Fatigue cracking, if unchecked, can lead to through thickness fracture or rupture.

Categorisation of the final failure mechanism (e.g. leak before break or rupture) then has an input into the consequence of failure assessment. This can be done by conducting an engineering critical assessment such as BS-7910, Guidance on methods for assessing the acceptability of flaws in fusion welded structures [10].

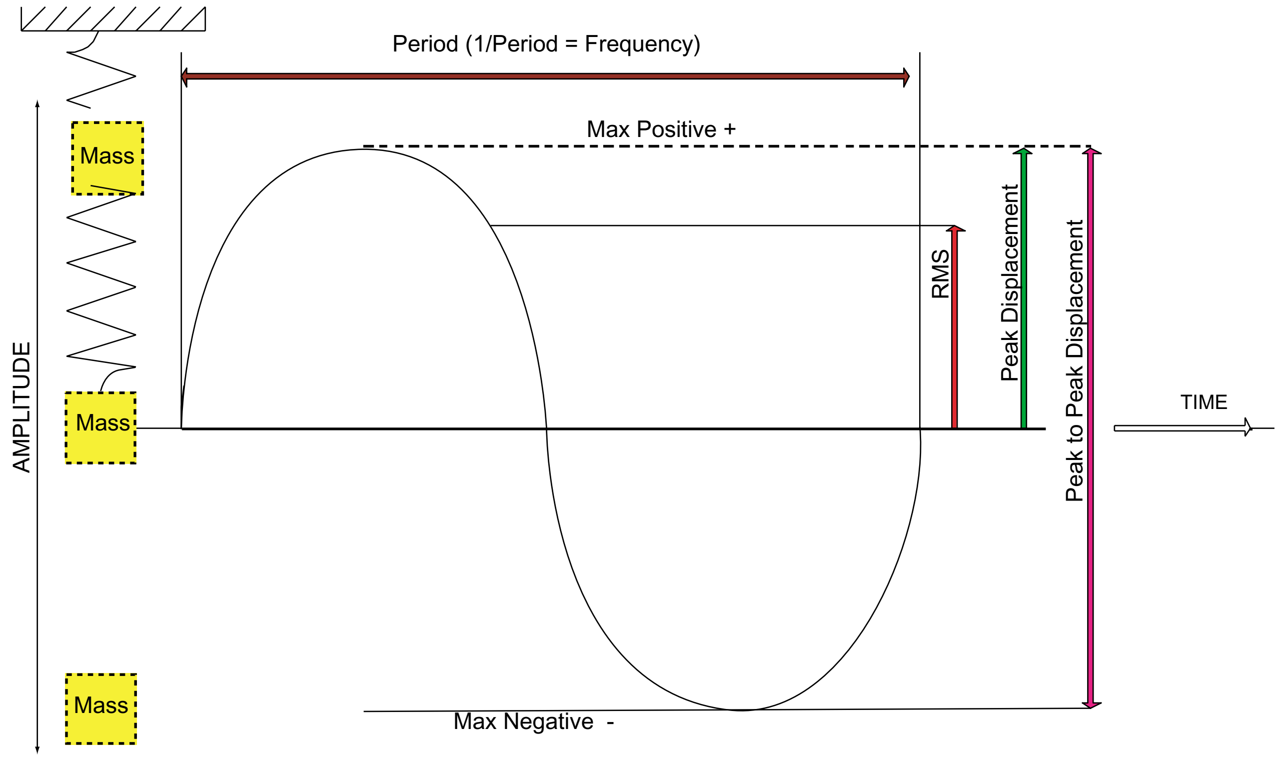


Figure 2-1 *Description of vibration using a simple spring-mass system (rms=root mean square)*

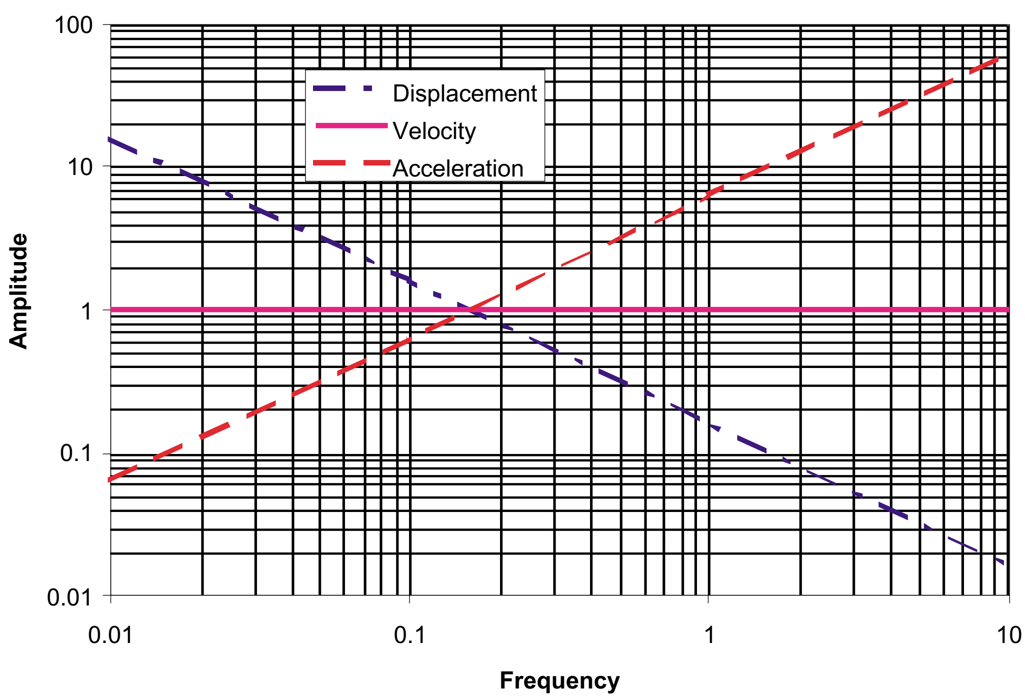


Figure 2-2 *Comparison of the amplitude of displacement, velocity, and acceleration as a function of frequency.*

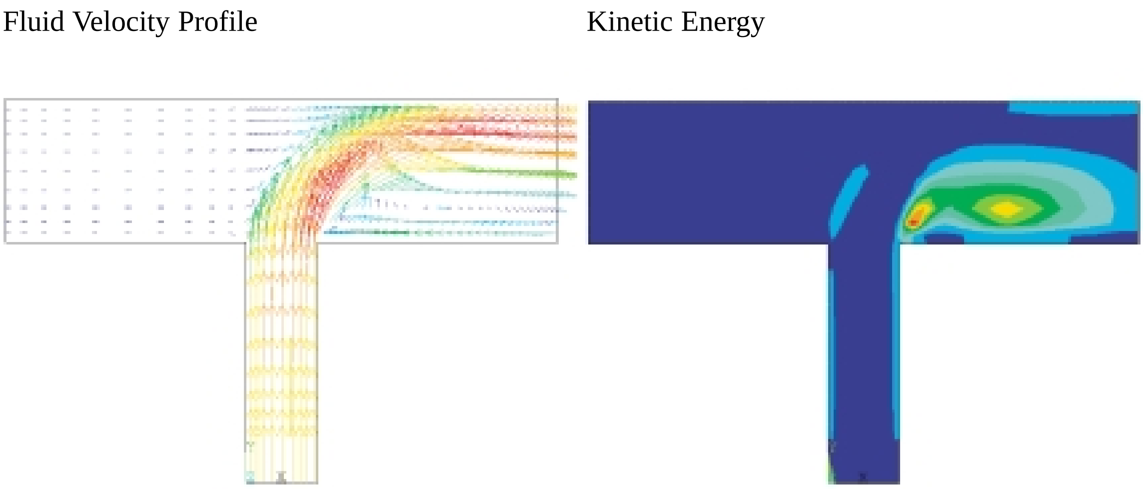


Figure 2-3 *An example of the distribution of kinetic energy due to turbulence generated by flow into a tee*

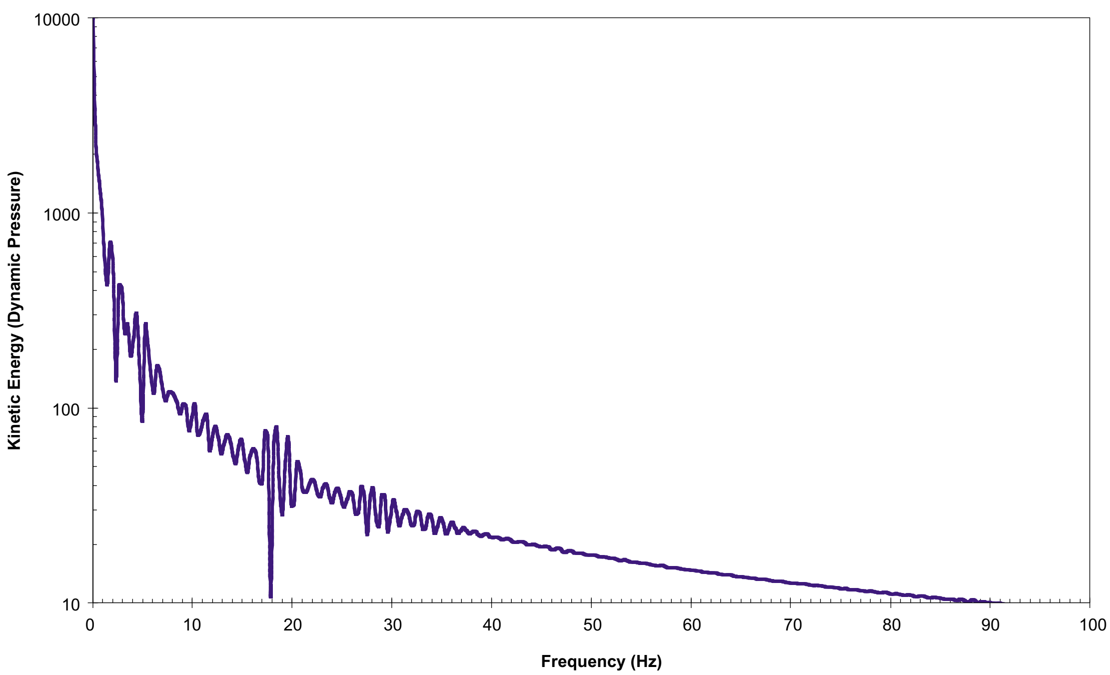


Figure 2-4 *Turbulent energy as a function of frequency*

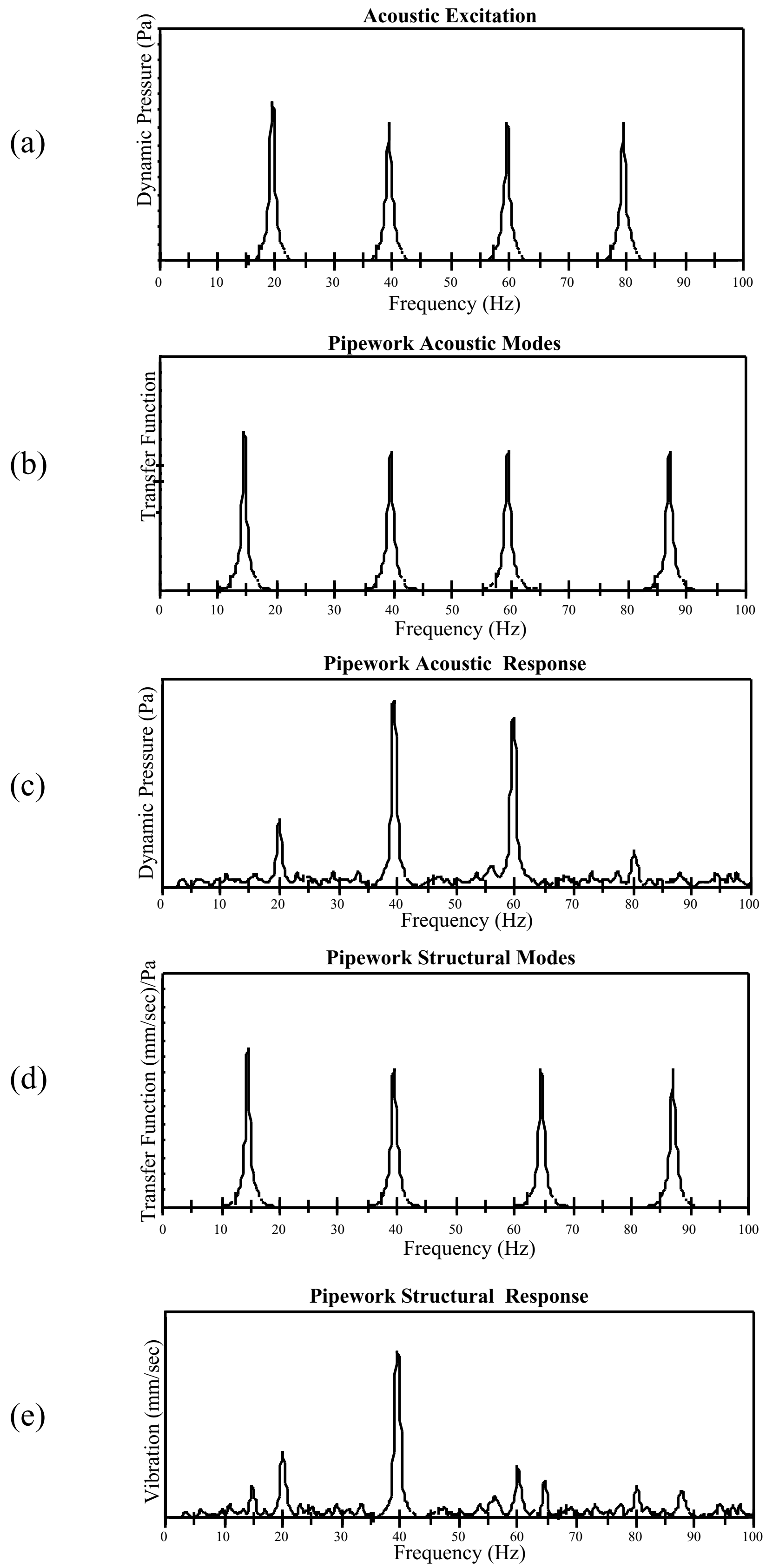


Figure 2-5 *Relationship between acoustic natural frequencies and structural response*

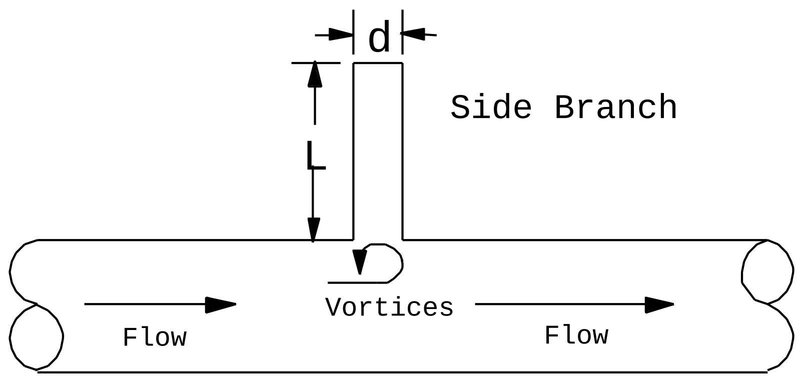


Figure 2-6 *An example of 'Dead Leg Branch'*

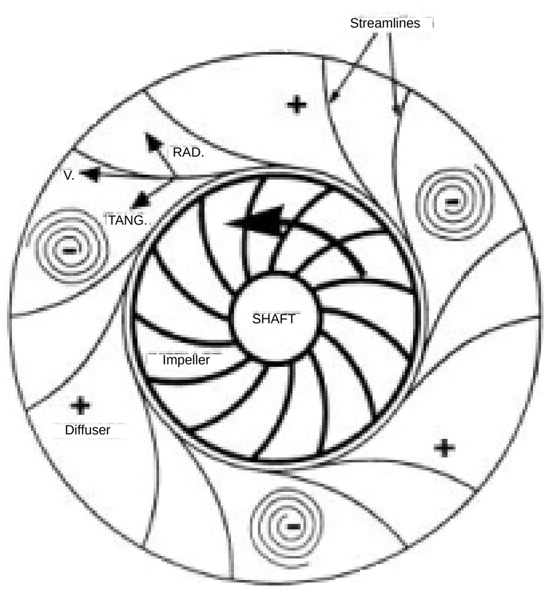


Figure 2-7 *An example of rotating stall [5]*

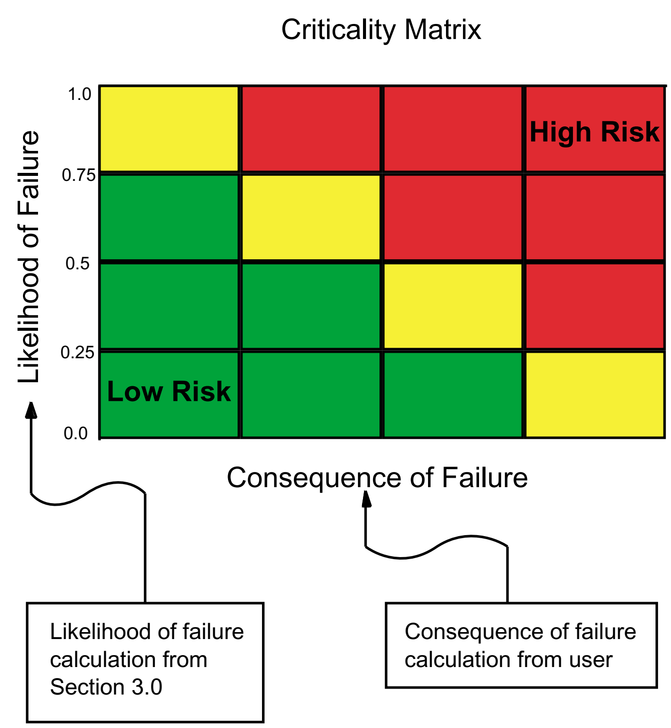


Figure 2-8 *Criticality matrix linking likelihood of failure calculation from these guidelines and consequence of failure from the user*

3.0 Assessment Methodology

3.1 Overview

These Guidelines have been structured to ensure:

- the straightforward identification of systems typically found in offshore platforms and onshore petrochemical plants;
- the identification of potential problem systems and their appropriate source(s) of excitation;
- that detailed screening is restricted to problem systems, to reduce cost and effort.

To meet these objectives the Guidelines have been split into three:

Stage 1. Identification of excitation mechanisms.

Stage 2. Detailed screening of main pipe.

Stage 3. Detailed screening of small bore connections.

Each stage will be discussed in turn.

- An overview of the assessment methodology is given in Figure 3-1.
- The complete methodology is illustrated in Figure 3-2.
- Figure 3-3 illustrates, in a flowchart, the major components and logic of Stages 1 and 2. After the completion of Stage 1 and Stage 2, **Worksheet 3-1** should be completed.
- Figure 3-4 describes Stage 3 in a similar manner. After the completion of Stage 3, **Worksheet 3-2** should be completed.
- **Worksheet 3-1** and **Worksheet 3-2** are the primary records of the assessment.

3.2 Stage 1 - Identification of Excitation Mechanisms

3.2.1 Purpose

The purpose of this section is to divide the plant into manageable systems and identify whether there is a potential problem for each system. If there is a potential problem this stage will identify what applicable excitation mechanisms are likely to be present.

3.2.2 Method

The systems are categorised by the fluid that is transported. The two divisions are:

a) Gas

b) Liquid/Multiphase

The possible excitation mechanisms for each system identified below are shown in Figure 3-3:

- flow induced turbulence;
- high frequency acoustic excitation;
- mechanical excitation;
- pulsation - reciprocating machinery
  - rotating stall
  - periodic flow induced excitation.

Note: The mechanism considered under pulsation "periodic flow induced excitation" is that due to flow past a branch (dead leg). Other mechanisms, such as flow over components and thermowells, are outwith the remit of these Guidelines.

These Guidelines are for 'steady state' operation of plant. They do not consider:

- the response of the structure to impulse loading (slugflow, pressure transients as a result of rapid valve closure, impact loading etc);
- environmental excitation mechanisms such as inertial effects on pipework on ships and floating facilities;
- wave induced motion on risers, seismic excitation and topside vibration due to ship collision.

The Guidelines do consider certain failure modes associated with non 'steady state' operations such as acoustic fatigue on relief and blowdown systems.

However the design of small bore connections in line with the recommendations in these Guidelines will minimise the likelihood of failure due to these excitation mechanisms.

The problem areas and their excitation mechanisms for each system are established by the questionnaire in Appendix A1.

There are two questionnaires; one for liquid/multiphase systems and one for gas systems. For each system use the appropriate questionnaire. The boxes in **Worksheet 3-1** must be completed with 'yes' or 'no' answers from the questionnaire to identify which mechanisms are present.

3.2.3 Recommended Action

A system that has been identified as a potential problem system should be analysed according to Stage 2.

3.3 Stage 2 - Detailed Screening of Main Pipe

3.3.1 Introduction

The basis of this stage, embodied in Figure 3-3, is to allow the user to develop the Likelihood Of Failure (L.O.F.) for the main line. This is determined for each of the mechanisms identified to be present by the Stage 1 assessment. Some of these are calculated from process parameters (density, flow rate, etc.) and the detailed calculation methods are given in Appendix A2 Stage 2 Detailed Screening of Main Pipe. Other parameters such as the L.O.F. due to mechanical excitation, or the default values if no process information is available, have been developed from experience. These values reflect the results from the database, a typical extract of which is shown in Figure 3-5. This demonstrates that the number of failures or potential failures in gas compression or separation areas are high, hence the high factors in the process L.O.F. inputs.

3.3.2 Purpose

The purpose of this stage is to determine the extent to which the potential excitation mechanism(s) defined in Stage 1 will cause vibration problems in pipework. This Stage will identify whether the design is intolerable or what is the likelihood of failure. Recommended actions are given depending on the likelihood of failure. This Stage will also identify those systems that require particular attention to the design of small bore connections.

3.3.3 Method

For all potential problem systems identified in Stage 1, conduct a detailed screening of the main pipe. The likelihood of failure is calculated using the procedures outlined in Appendix A2 and is illustrated in Figure 3-3. The resultant likelihood of failure values should be inserted in **Worksheet 3-1** as appropriate. Consequence of failure can also be used at this stage in conjunction with the L.O.F.values to determine the risk or criticality of the main pipe. However to ensure piping systems with acceptable vibration and fatigue characteristics, the full assessment as recommended by these Guidelines must be completed.

If the only excitation mechanism identified as a potential problem in gas systems from Stage 1 is high frequency acoustic excitation, then a Stage 3 analysis is not required. However the general guidance given on small bore connection design in Section 4.3 for the design of small bore connections is essential. A visual survey should also be conducted using the parameters in Figure 3-4 as a guide to assess the acceptability of the small bore connections.

3.3.4 Recommended Actions

The following actions are recommended as a result of the Stage 2 detailed screening of main pipe.

Intolerable - Likelihood of Failure ≥ 1.0

- The main pipe must be redesigned, resupported or a detailed analysis of the main line should be conducted as per Section 4.2 (Design solutions for main pipe).

- In addition the small bore connections should be assessed as per Stage 3.

1.0 > Likelihood of Failure ≥ 0.5

- Where possible and feasible, the main pipe should be redesigned, resupported or a detailed analysis should be conducted as per Section 4.2 (Design solutions for main pipe).

- In addition the small bore connections must be assessed according to Stage 3.

0.5 > Likelihood of Failure ≥ 0.3

- The principal area of concern is small bore connections and thus should be assessed according to Stage 3.

Likelihood of Failure < 0.3

- To ensure that design features of small bore connections are sound, refer to the guidance given in Section 4.3.

3.4 Stage 3 - Detailed Screening of Small Bore Connections

3.4.1 Introduction

The excitation energy which causes vibration of small bore connections comes from the main pipe. In the preceding section this excitation has been assessed and where it is predicted to be high, suggestions for methods to reduce main pipe vibration levels have been produced (Section 4.2). In this stage the features of the small bore connection and its connection to the parent pipe are reviewed and the contribution of each feature to the L.O.F. of the small bore connection identified.

To develop some of the small bore modifier factors a numerical (finite element method) approach has been used. The small bore connection modifiers have been developed such that lower small bore L.O.F. values reflect a higher natural frequency of the small bore connection and thus move it away from the prime excitation frequencies. Conversely a higher small bore connection modifier reflects a lower natural frequency of the small bore connection and thus moves it closer to the prime excitation frequencies. A parametric study of the effect of the various parameters on natural frequency has been completed and trends produced. Typical relationships are shown in Figure 3-6 and Figure 3-7. These trends have been used to formulate the small bore connection modifier values for parameters such as the effect of parent pipe schedule, small bore pipe diameter, overall branch length and type of fitting to the main pipe.

Other branch connection parameters such as the number of valves and location on the main pipe (near a bend, mid span, at a support etc.) were developed from experience and from review of the database. Typical results, from the database shown in Figure 3-8, shows cases where there are a number of valves or where the connection is at mid span of a pipe is causing a significant number of problems. It also confirms other high small bore connection modifier cases such as long unsupported connections.

3.4.2 Purpose

The purpose of Stage 3 is to define the likelihood of failure of small bore connections on a system exhibiting a higher L.O.F. (1.0>L.O.F.≥0.3) at Stage 2. Stage 3 will give recommended actions for the small bore connections of the particular system depending on the likelihood of failure.

3.4.3 Method

All small bore connections on systems from Stage 2 that have an 1.0>L.O.F.≥0.3 shall be assessed according to Appendix A3. The small bore connection modifier is calculated using the procedures outlined in Appendix A3 and is illustrated in Figure 3-4. The resultant L.O.F. values shall be inserted in **Worksheet 3-2**.

The likelihood of failure of the small bore connection is the minimum of:

- the Main Pipe L.O.F. from Stage 2 which accounts for the excitation mechanism(s);

- the small bore connection modifier from Appendix A3 which accounts for the geometry of the small bore connection and its location on the main pipe.

The minimum of the two inputs is required because both a badly placed/designed small bore connection and an excitation source need to be present for the small bore connection to have a higher likelihood of failure.

An illustration of this is contained in Appendix B.

3.4.4 Recommended Actions

The following are recommended actions as a result of the detailed screening of small bore connection analysis.

1.0 > Likelihood of Failure ≥ 0.7

- Modify the connection at the design stage or brace the small bore connection by means of suitable support. Remove unnecessary or redundant small bore connections. Further possible design solutions are contained in Section 4.3 (Design solutions for small bore connections).

0.7 > Likelihood of Failure ≥ 0.4

- Monitoring is required during commissioning to determine if bracing is required. In the event of bracing being required, design solutions are itemised in Section 4.3 (Design solutions for small bore connections). Alternatively modify the connection at the design stage, as above.

Likelihood of Failure < 0.4

- To ensure that design features of small bore connections are sound, refer to the guidance given in Section 4.3.

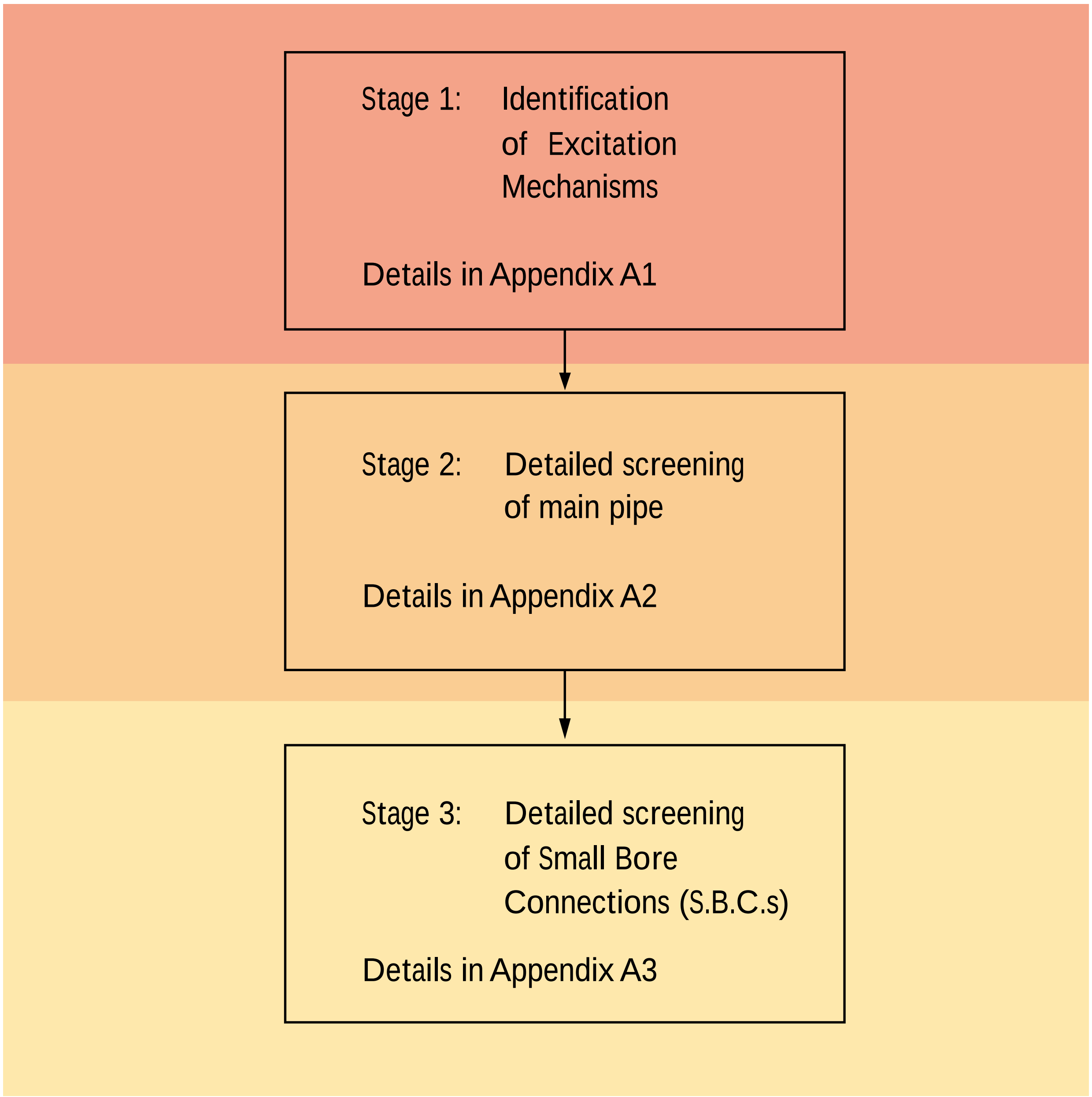


Figure 3-1 *Overview of assessment methodology*

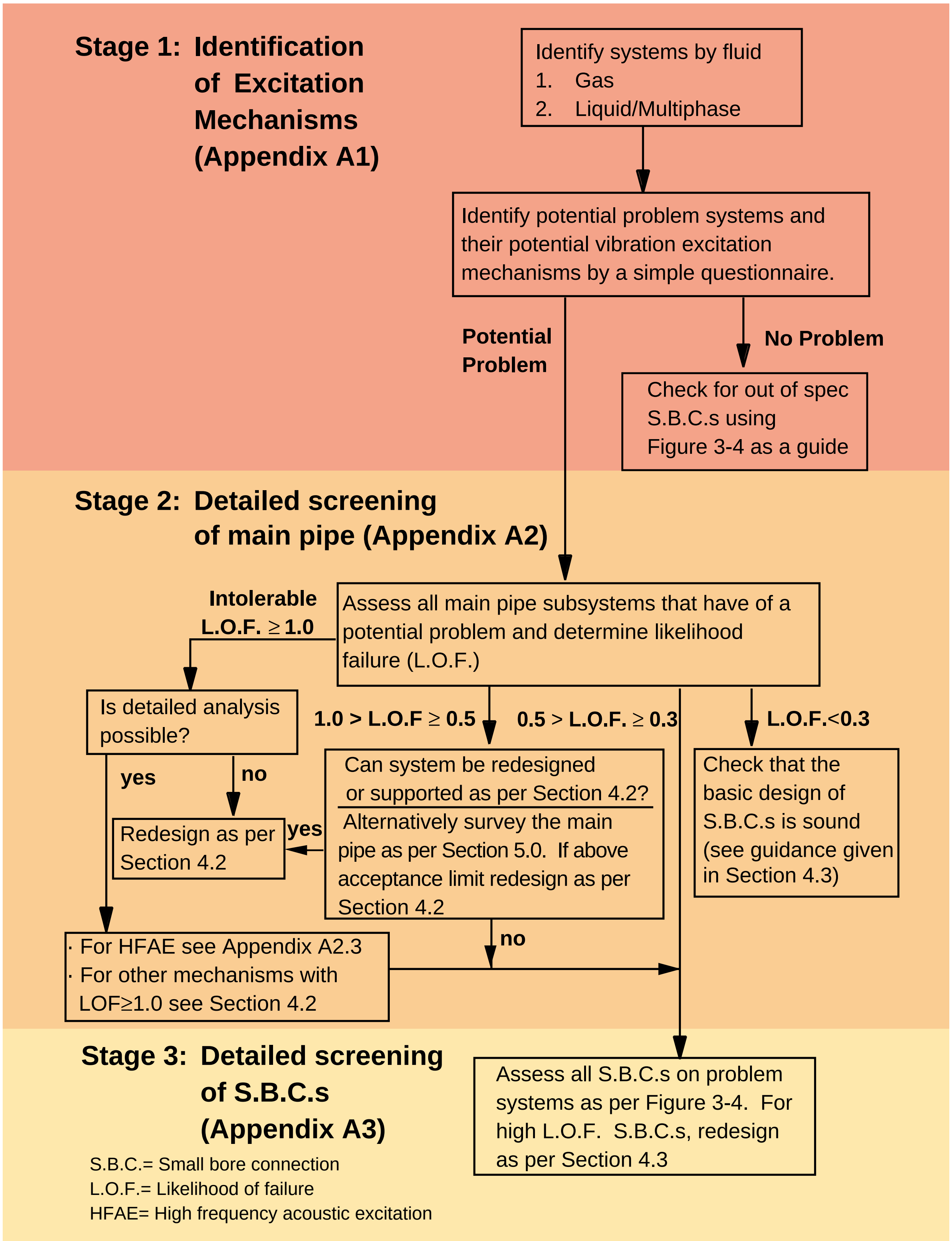


Figure 3-2 *Methodology for assessing vibration fatigue in process piping systems and small bore connections*

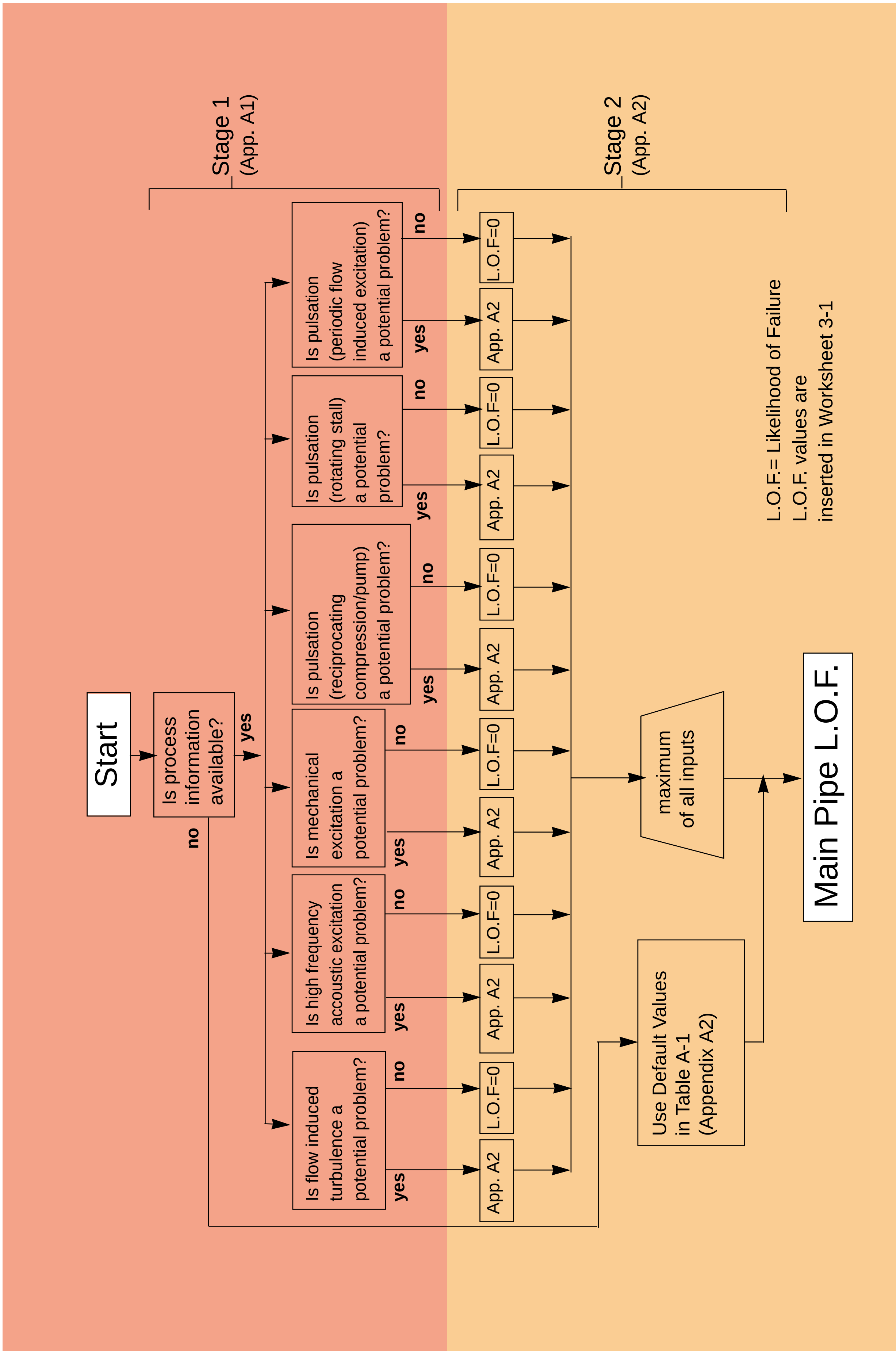


Figure 3-3 *Stage 1 (Identification of Excitation Mechanism) and Stage 2 (Detailed Screening of Main Pipe) Flowchart*

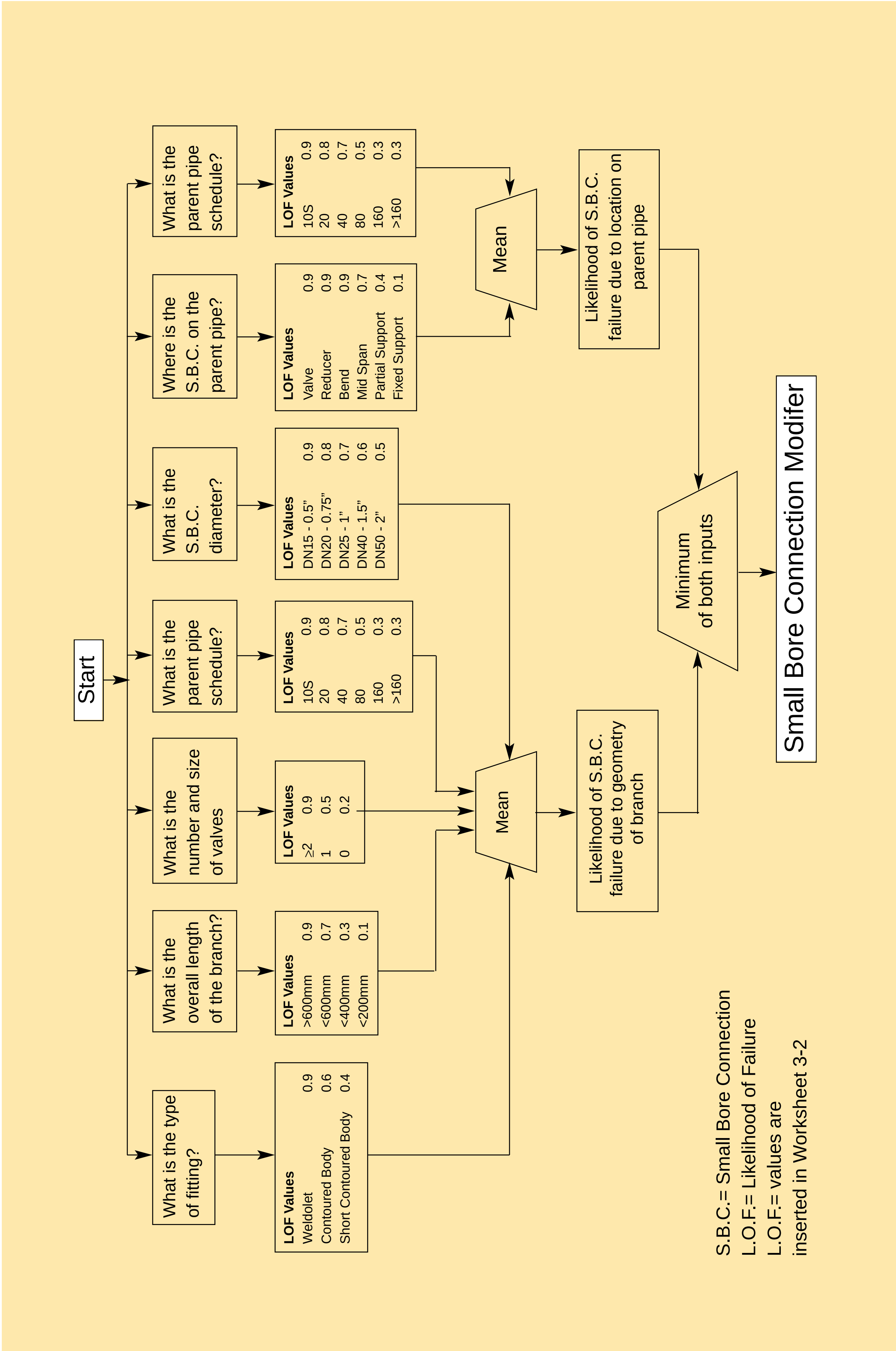
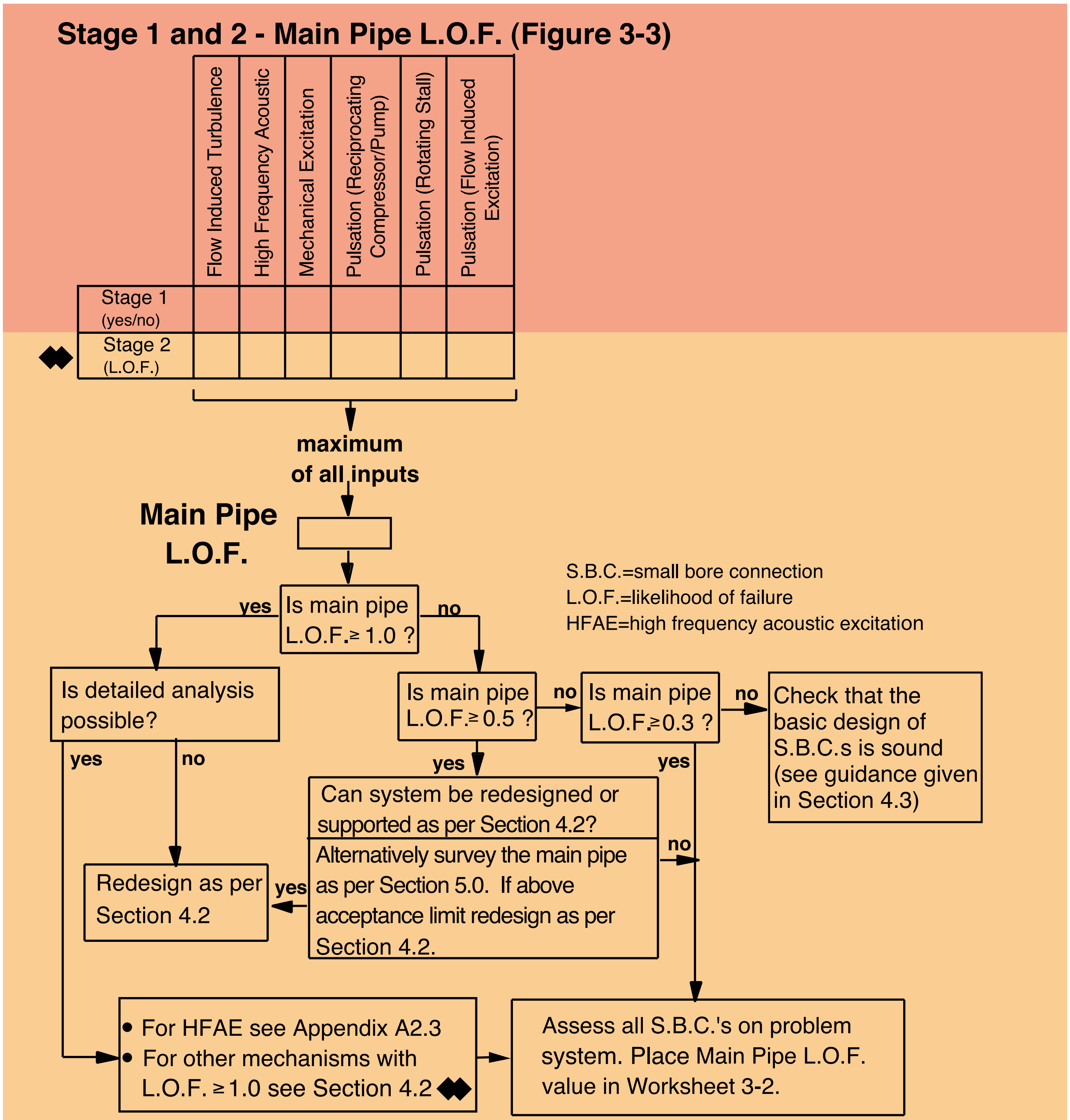


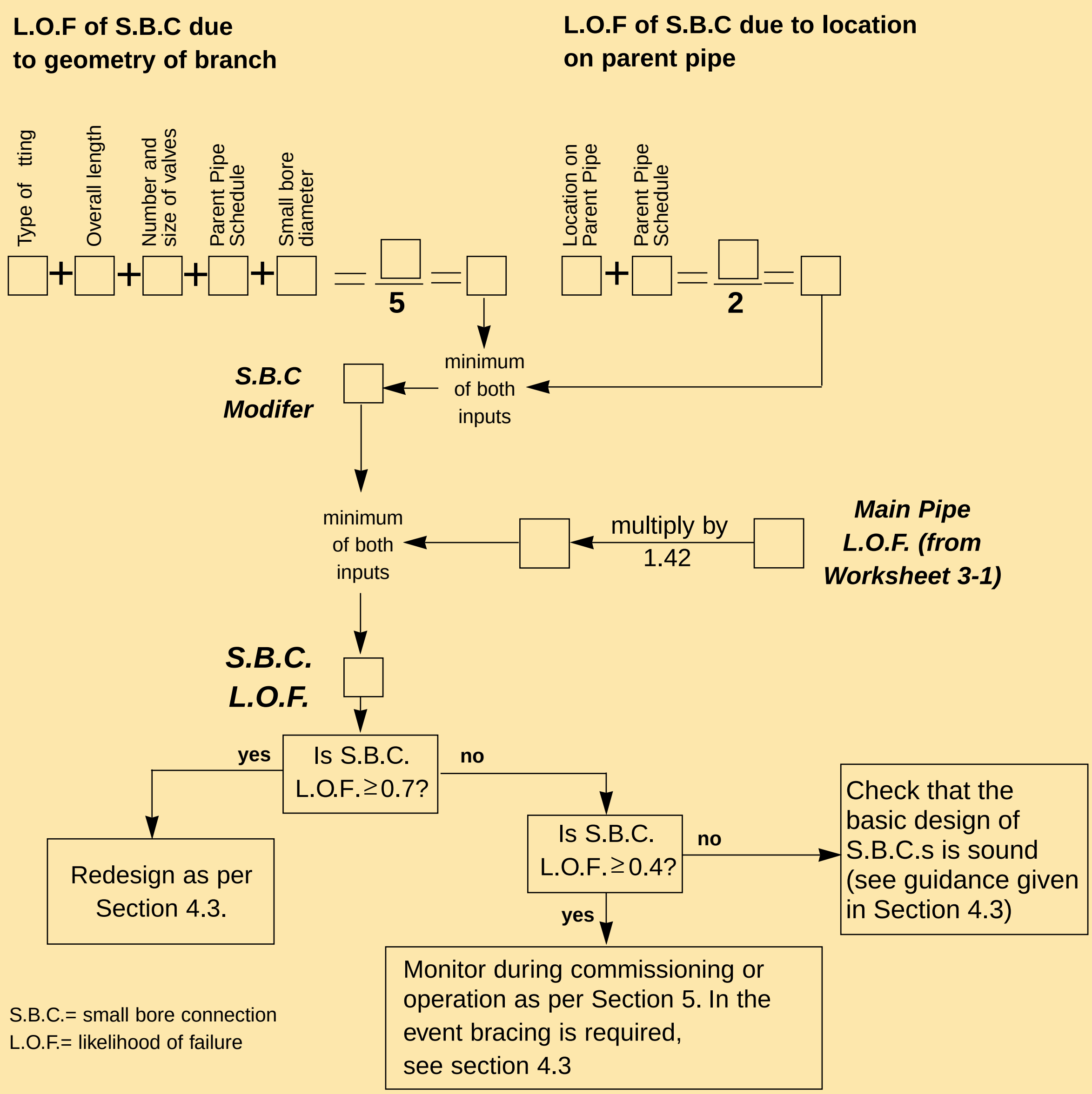
Figure 3-4 *Stage 3 (Detailed Screening of Small Bore Connection) Flowchart*



Project/Plant:		Line number:
System:		Main Pipe L.O.F.:
Subsystem:		Actions:
Assessed by:	Date:	
Ref. No.		

Worksheet 3-1 *Stages 1 and 2 of the Guidelines*

Stage 3 - Small Bore Connection L.O.F. (Figure 3-4)



Project/Plant:		Line number:
System:		S.B.C number:
Subsystem:		S.B.C. L.O.F.:
Assessed by:	Date:	Actions
Ref. No.		

Worksheet 3-2 *Stages 3 of the Guidelines*

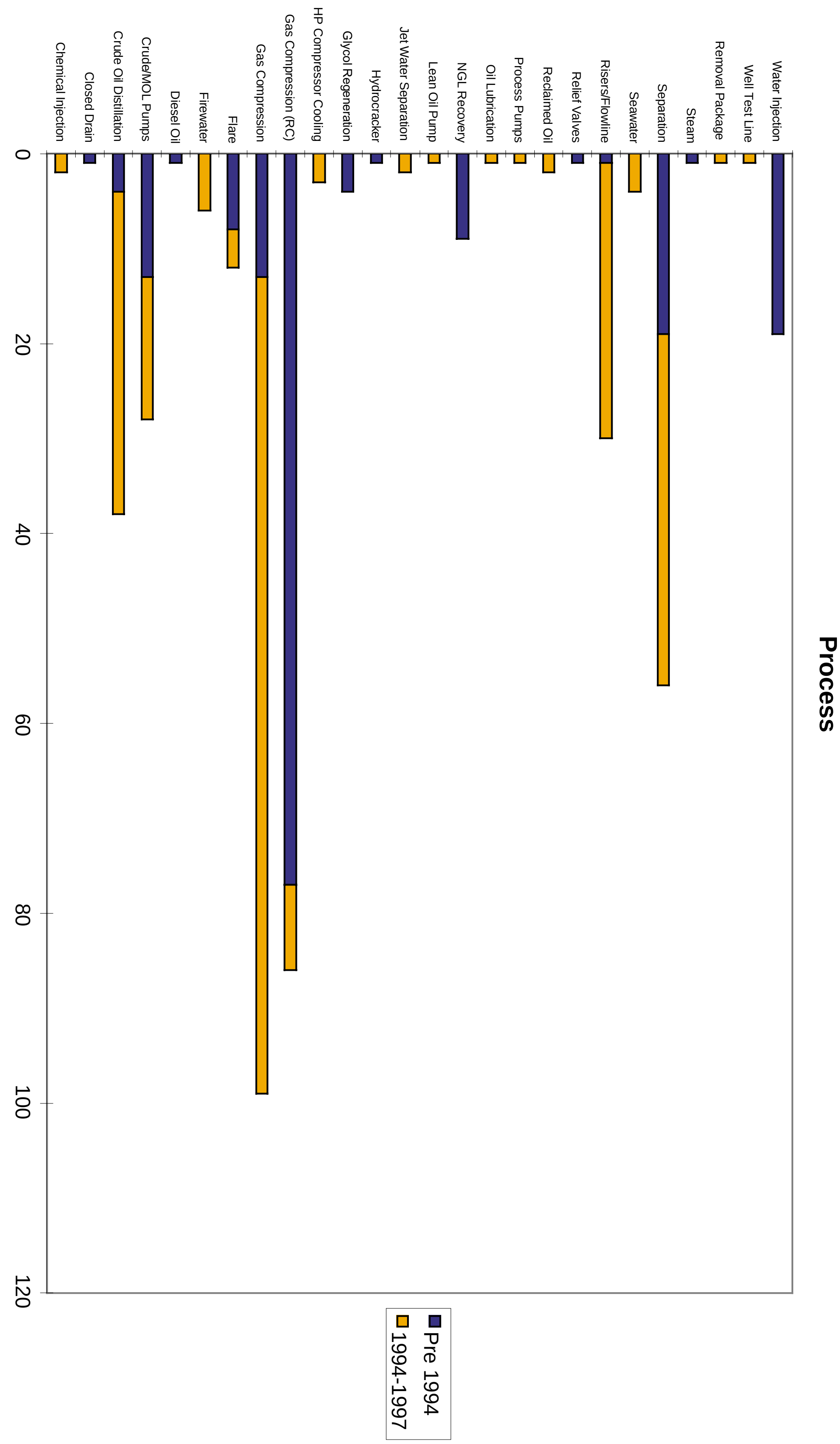


Figure 3-5 *The basis of default 'likelihood of failure' results for process area from database of failures*

DBB = Integral Double Block & Bleed Valve

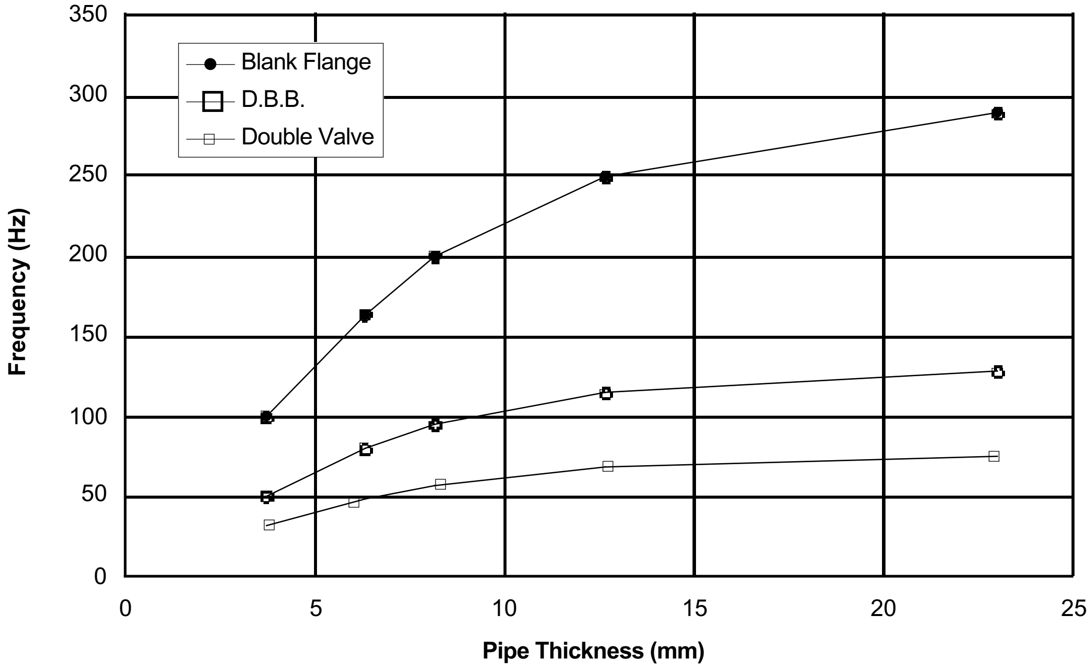


Figure 3-6 *The effect of pipe wall thickness and mass of valve on the natural frequency of the small bore connection*

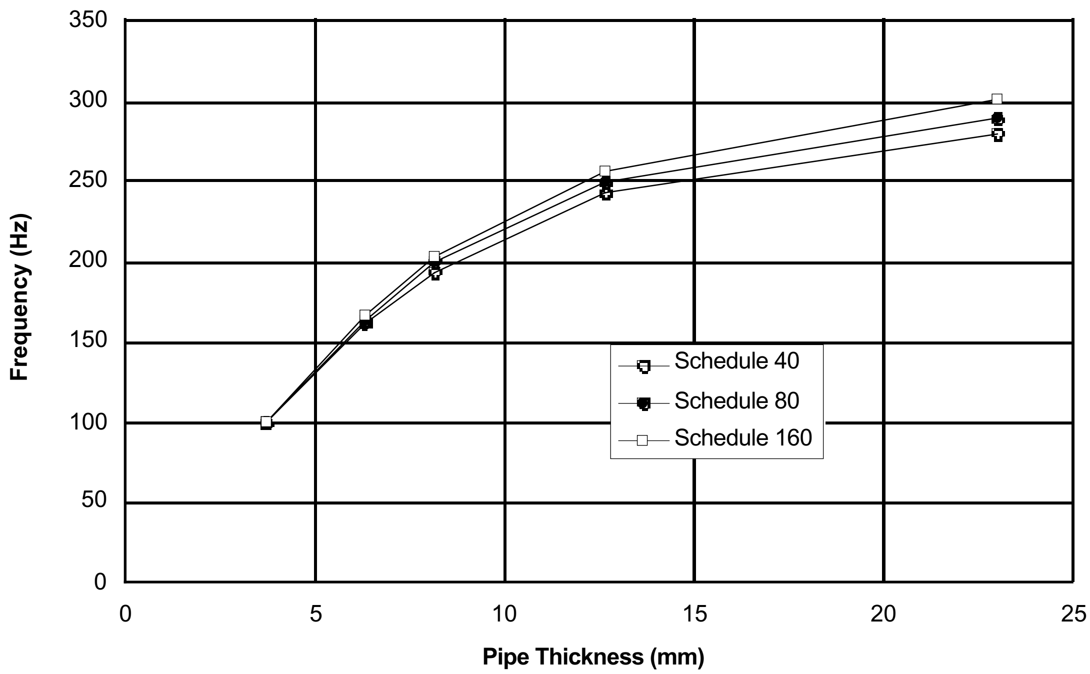


Figure 3-7 *The effect of small bore connection schedule (thickness) on the natural frequency of the small bore connection*

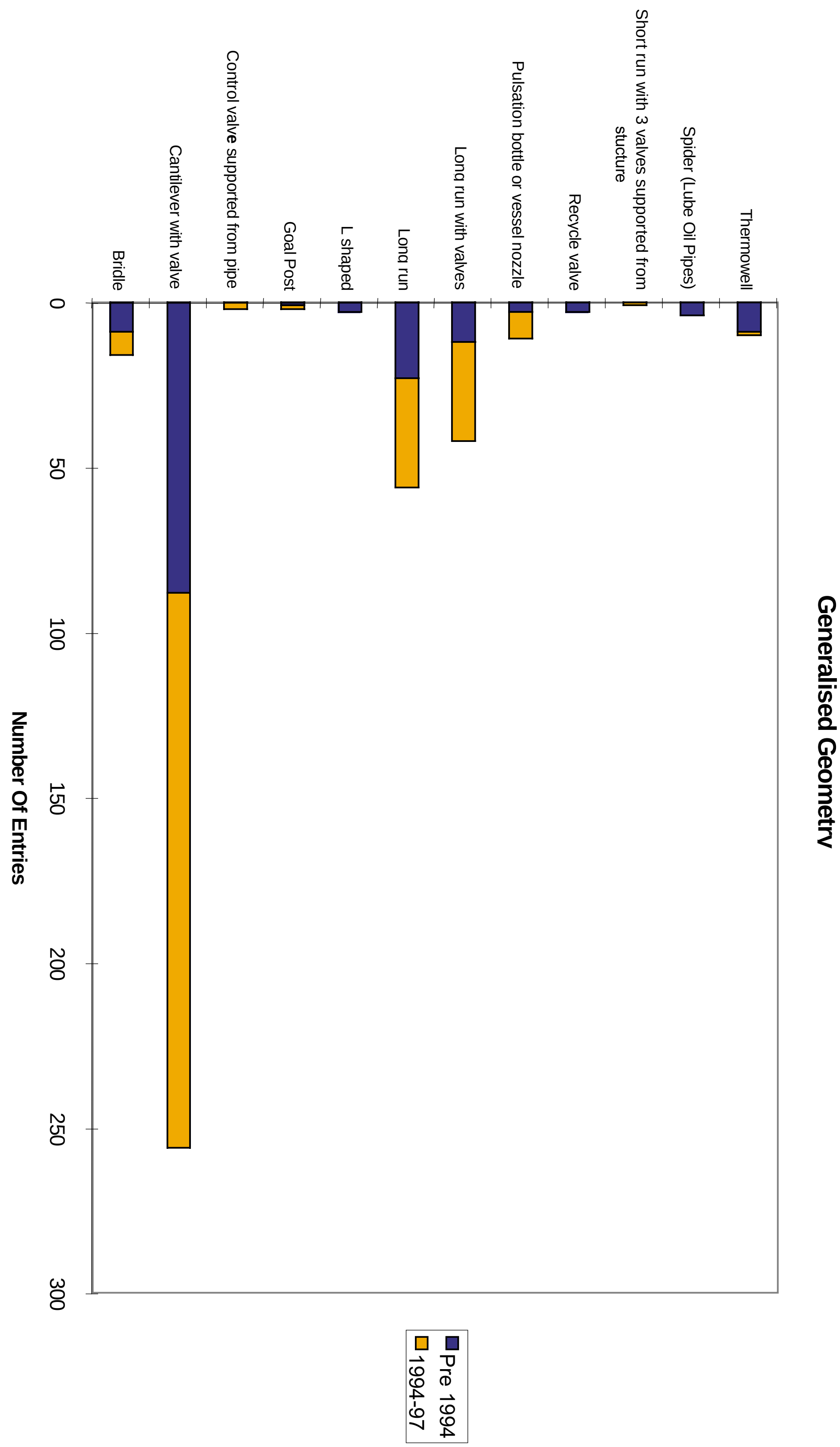


Figure 3-8 *Database extract showing high occurrence of problems where there is a high number of unsupported valves*

4.0 Design Solutions

4.1 Overview

The purpose of this section is to give possible design solutions, best practices or remedial action for existing plants. Where possible, recommendations for detailed analyses are given.

If the likelihood of failure is greater than 1.0, the main pipe design is intolerable and the user must modify the design or process conditions (e.g. reduce flow rate) or follow the possible design solutions in Section 4.2. If the likelihood of failure from Stage 2 is greater than 0.5, attempts should be made to modify the main pipe or consider main pipe design solutions covered in Section 4.2.

If the likelihood of failure from Stage 3, screening of small bore connections, is greater than 0.7 redesign is required (see Section 4.3). If the L.O.F. is less than 0.7 but greater than 0.4 monitoring is required during commissioning to determine if redesign is necessary (see Section 5.0). If the L.O.F. is less than 0.4 then a visual survey should be undertaken and the basic design of the S.B.C. should be checked to see if it is sound as given in Section 4.3. This will ensure that all small bore connections are meeting best practice.

4.2 Design Solutions for Main Pipe

4.2.1 Flow Induced Turbulence

If feasible, one of the simplest solutions is to decrease the flow velocity by increasing the diameter of the main pipe or running a second pipe in parallel. Increasing the pipe wall thickness can also have a beneficial effect.

For turbulent excitation, flow smoothing is one option which can be accomplished through the use of swept tees rather than 90° tees, minimising the number of bends in a system, the use of long radius bends, and the use of flow straighteners.

Stiffening of the main line and its supporting structure can also be beneficial. This is because the fundamental natural frequency of the pipe is then increased, and, as the level of turbulent energy falls off rapidly with frequency (see Figure 2-4), the resulting vibration level falls also.

However, stiffening is not always an appropriate approach. Thermal growth requirements may limit the amount of additional support that can be included, and for some systems, such as offshore wellhead flowlines, the piping system must retain high flexibility to accommodate riser movement. In these cases, use of specialist vibration dampers can prove effective as they allow relatively large quasi-static movement whilst providing damping of vibration. These units are different from the normal type of snubber and damper devices used in piping systems and thus specialist advice should be sought when considering vibration dampers.

Careful consideration should also be given to adequate support at sources of turbulence, for example valves and mitred bends, as this will help to reduce the coupling between the turbulent energy generated by the source and the piping.

In cases where the above solutions are not feasible, not easily quantifiable or there is uncertainty regarding their applicability in certain situations, seek specialist advice.

4.2.2 High Frequency Acoustic Excitation

Use of low noise trim in a control valve can help to reduce noise levels at source and therefore reduce the risk of an acoustic fatigue failure; this may also reduce the need for acoustic insulation on the exterior of the pipe which has direct benefits from a corrosion perspective. It should be noted that the converse is not true, i.e. the use of lagging will not have a significant influence on the high frequency response of the piping which leads to acoustic fatigue failure. However, the use of low noise trim is not always an option, especially for relief valves. While acoustic silencers are an alternative, their use is not recommended because the success rate and durability is limited. They are themselves exposed to high levels of acoustic energy which can result in fatigue failure of the silencer itself.

Another method of reducing noise levels at source is to reduce the mass flow rate through the valve, either by the use of multiple valves or extending the time taken to relieve or blowdown the system (by changing the valve orifice at the design stage).

Increasing the pipe wall thickness locally around a circumferential discontinuity such as a small bore connection is an option for a new design. This reduces the resulting dynamic stress levels at various connections; alternatively, full wraparound reinforcement can be used to achieve the same goal.

The use of localised circumferential stiffening rings has been found to be effective in some cases. These change the high frequency structural characteristics of the pipe wall, resulting in lower dynamic stress levels at sensitive connections to the main line. Because of sensitivity to positioning, specialist advice should be sought when considering stiffening rings. In addition, the use of forged rather than fabricated tees should be considered at sensitive locations as the resulting stress concentrations are much reduced. Providing additional support for the main line will have little or no impact on acoustic fatigue problems; it may even make the situation worse as it adds another circumferential discontinuity to the pipe wall.

In cases where the above solutions are not feasible, or there is uncertainty regarding their applicability, specialist advice should be sought.

4.2.3 Mechanical Excitation

In the case of mechanical excitation, the nature of the problem is generally a resonant response of the pipework and support system. This is caused when the excitation frequency is coincident with a structural natural frequency of the pipework and support system.

The natural frequencies of the pipework and support system should be determined using a three dimensional model. This analysis can be undertaken by most commercial pipework analysis software. As the structural natural frequencies are dependent on the distribution of the pipework's mass and stiffness (see section 2.1), it is essential that the mass of valves and the stiffness of supports are incorporated into the model. The results from the three dimensional model should, where possible, be verified for the actual installed system. This is because true support stiffness values are difficult to evaluate.

The structural natural frequencies of existing pipework can be determined experimentally by an impact test (i.e. applying a known excitation to the pipework using for example a force hammer and measuring the vibrational response). This should only be conducted by a specialist.

Having considered the source; the frequency content of the excitation should be determined. This will generally be the running speed of the machine and its harmonics i.e.:

$$f = \frac{\text{order*RPM}}{60} \ \text{(Hz)} \ \text{ where order} = 1,2,3 \ ... \qquad (3)$$

For mechanical excitation problems most of the excitation energy will be found at the lower orders . It is therefore usually only necessary to consider the first two orders of excitation.

To determine the presence or absence of coincidence, a band of ± 10% should be taken at each excitation frequency being considered (e.g. for 20 Hz running speed the bands would be 18 to 22 Hz for the first order, or 36 to 44 Hz for the second order, and so on). The purpose of this band is to allow for both the damping present in the system and for any inaccuracies in the model. Pipework structural natural frequencies **must no**t occur within these bands. In addition, where practical and feasible, the first structural natural frequency should occur above the excitation band associated with the running speed of the machine.

In the event that coincidence does occur, then the structural natural frequencies should be modified to occur outwith the bands of excitation, as it is usually impractical to change the excitation frequencies. In general, this will involve stiffening the pipework by the modification or addition of supports. Changing the speed of rotating machinery is possible in some cases, such as belt or gear drives, and has been successfully used to move the excitation frequency away from the structural natural frequency to avoid pipework resonance.

4.2.4 Pulsation

For pulsation related problems stiffening of the main line may be appropriate if a particular structural natural frequency of the pipe system has to be de-tuned to avoid a resonant condition. However, in some situations, stiffening may raise a structural natural frequency so that it becomes coincident with one of the higher order harmonics, therefore causing a resonant response. In such a case reducing the stiffness to lower the natural frequency may be appropriate.

Serious consideration should be given to undertake a pulsation analysis to meet the requirements of API 618 [7] (reciprocating compressors) or API 674 [8] (reciprocating pumps). Line length changes can also be considered for pressure pulsation problems if an acoustic resonance is the main issue. This is only likely to be beneficial for lines where the change of length greater than 25% of the original line length can be accommodated. Design checks should be undertaken to ensure the modified acoustic frequencies are not coincident with excitation frequencies or structural natural frequencies of the line.

For pulsation related problems a number of potential modifications exist. These include the use of pulsation bottles for reciprocating compressors, or precharged pulsation dampers for reciprocating pumps. One drawback with precharged units is that the precharge pressure must be maintained otherwise the dampers become ineffective. The frequency characteristics of the pulsation bottles should be checked to ensure the design provides the required attenuation.

Orifice plates can also provide significant damping of acoustic modes, providing they are placed at the correct position (i.e. at a dynamic pressure minimum). Their use must be carefully balanced against the pressure drop that they impose on the system. Specialist advice should be sought when considering orifice plates.

For ‘dead leg’ branches consideration should be given to undertaking a detailed analysis as described in [11]. It may be possible to detune the system by changing the branch line length as discussed previously. The problem can also be effectively dealt with by using tee connections that do not have a sharp joint (as would be the case with a fabricated tee), but which instead have a relatively high inside radius of curvature (such as that provided by a forged swept tee).

In cases where the above solutions are not feasible, or there is uncertainty regarding their applicability in certain situations, seek specialist advice.

4.3 Design Solutions for Small Bore Connections

Key recommendations for ensuring satisfactory small bore connection design are:

- the fitting and overall unsupported length should be as short as possible;
- the mass of unsupported valves/instrumentation should be minimised (e.g. by the use of lightweight double block and bleed valves or monoflange valves);
- any mass at the free end of a cantilever should be supported in both directions perpendicular to the axis of the small bore;
- it is essential that bracing supports should be from the main pipe, thus ensuring that the small bore connection moves with the main pipe;
- the diameter of small bore connections should be maximised - use of short contoured body fittings are preferable.

Bracing of small bore connections can be applied where the design options have been exhausted or as a modification to an existing system where problems have occurred. It is effective in reducing the dynamic stress levels at the welded connection for vibration induced by flow turbulence, pulsation and mechanical excitation.

Note: Bracing will **no**t be beneficial in the case of high frequency acoustic excitation.

Whilst not reducing the level of vibration in the main line, bracing reduces relative movement between the connection and its main pipe and hence reduces the dynamic stress. The bracing should be in two planes, connected between the small bore pipework and the main pipe, and the clamping carefully designed so that the connection is adequately supported. Under **no circumstances** should the connection be braced from a local structure such as steelwork, decks or bulkheads as this will only make the problem worse. Drawings of recommended small bore branch supports are contained in Appendix C.

5.0 Survey Methods

5.1 Introduction

Several survey methods exist which allow the assessment of pipework vibration on operational systems. All rely on the measurement of either pipework vibration velocity, or the direct measurement of dynamic strain. The following sections provide guidance on the use of these methods, and the interpretation of measured data.

Generally, the use of vibration measurement provides a simple method for screening a piping system for potential problems. However, it is not a fail safe assessment technique. For a full and robust assessment of the fatigue life of a critical piping system and its components then direct measurement of dynamic strain should be considered. This enables dynamic stress to be calculated, which is used to determine susceptibility to failure by fatigue.

5.2 Survey Methods

5.2.1 Vibration Based Survey Techniques

The use of vibration based survey techniques is limited to the assessment of low frequency vibration generated by flow induced turbulence, mechanical excitation and pulsation. Such techniques are not suitable for the assessment of vibration generated by high frequency acoustic excitation.

A method for assessing vibration of both the main pipe and small bore connections is given in Appendix D. This provides details of the instrumentation required, considerations for mounting the measurement transducer, and guidance on the interpretation of results.

5.2.2 Fatigue Direct Strain Measurement

The vibration measurements described above are able to provide a first screening of potential problem areas, but as highlighted, do not provide definitive answers as to whether fatigue will be a problem. However, the results can be used by specialists to identify those regions or locations on pipework systems where strain measurements should be taken to confirm that the connection will not fail in fatigue.

In all structures the weakest link from a fatigue viewpoint is the weld toe. This is because of the stress concentration at the weld toe, but also the metallurgical weakness at the HAZ (Heat Affected Zone) which has lower fatigue resistance. Comparison of the endurance limit (the dynamic stress range below which the components have an infinite fatigue life) for parent material (Class B) and a flush ground butt weld (Class C) in Figure 5-1 shows that the weld is weaker than the parent material. For other weld geometries (see Figure 5-1) the stress concentration at the weld toe reduces the tolerable stress range for infinite fatigue life.

For considering where to measure fatigue stresses, refer to the stress distribution (in two planes) at a typical small bore connection shown in Figure 5-2. It can be seen that there is an area remote from the weld where the stresses are classed as membrane or nominal. In the area close to the weld the stresses are classed as geometric, i.e. the stress distribution is influenced by the presence of the connection. Very close to the weld the stresses are influenced by the specific geometry of the weld toe and it is this area which will be most affected by stress reduction techniques such as grinding of the weld toe or shot peening.

The recommended method of fatigue life evaluation is that used by BS7608 [15] or BS5500 [16]. In these codes fatigue curves are generated for specific weld geometries as shown in Figure 5-1. The basis of the curves is test specimens which have been fatigued to failure. Axial strain gauges are placed close to the weld so that they predominantly measure the geometric strain - not the peak strain at the weld toe. The peak to peak strain levels are converted to stress using Youngs Modulus (i.e. the strains are assumed to be uniaxial). Since most fatigue modes involve bending of the connection, this is a reasonable assumption.

When taking measurements of dynamic strains on plant, it is therefore acceptable to place a small uniaxial strain gauge close to the weld toe. The gauge length should be less than 10 mm and the centre should be within 15 mm of the weld toe. Various methods of strain gauge attachment and measurement are available.

Gauges can be attached to the surface by bonding in line with procedures contained in [12]. This method is time consuming as it requires surface preparation, attachment of the gauge to the surface and wiring to an analyser. One gauge must be fixed to each location of interest, and the gauge can not be reused.

Alternatively, a press-on gauge can be used as described in [13]. This gauge is connected to a signal conditioning unit through to a spectrum analyser which displays the strain time-history. The time- history can be converted into the frequency domain to show the frequency content of the dynamic strains.

Following minimal surface preparation, the gauge is pressed on the surface at the location of interest, the time-history reviewed, stored in the analyser with an appropriate tag number identifying the location and the peak to peak strain range measured. Normally several time-histories are reviewed for each location over a period to ensure that maximum values have been captured.

Once the measurement is taken, the same gauge is moved to another position and another measurement taken. Many positions round a small bore connection can be surveyed to obtain the maximum strain range. A single branch connection can be surveyed, the results annotated and archived in approximately 15 minutes.

This press-on gauge has considerable benefits in rapidly assessing fatigue strain ranges on operational plant.

5.3 Interpretation of Fatigue Assessment

All design codes contain a section which considers fatigue assessment of the plant. Within the offshore industry and the onshore petrochemical industries the codes most frequently used for fatigue assessment are ASME III [17], ASME VIII [18], BS5500 and BS7608. The two ASME codes are based on the same methodology. The two British Standards are very similar and are based on a fundamentally different concept (for assessing welded connections) compared to the ASME codes.

In the ASME codes the relevant stress used to assess fatigue lives of components under high cycle fatigue loading is the maximum difference in the principal stresses called the Stress Intensity Range (the direction of the principal stresses is unlikely to change during the cycle). The Stress Intensity Range is divided by two (2) to give the value of Alternating Stress Intensity which is used as the input to the SN Curves provided by the code.

The code provides SN Curves for different categories of materials and within a category, the effect of material strength is considered. The curves are based on experimental tests on parent material and with the exception of high alloy/austenitic steels, the effect of the presence of weld metal on fatigue is not considered. As the SN curves provided are for parent material with no geometric discontinuities, the user is left to assess the Fatigue Strength Reduction Factors which are required to take account of the effect of geometry and the presence of any welds. Limited guidance is given as to what these factors should be, but a maximum Fatigue Strength Reduction of five (5) is specified. For fillet attachment welds, such as at supports, a recommended Fatigue Strength Reduction value of four (4) is given. The lack of specific recommendations for Fatigue Strength Reduction Factors for other geometries leaves the assessment open to question and can be the subject of debate, especially in marginal cases.

Both BS5500 and BS7608 use a different approach to that of ASME. One SN curve is used for ferritic, austenitic steels and aluminium alloys. The temperature limits of applicability are different for each material reflecting their various strength vs temperature characteristics. It was found by tests conducted for this Joint Industry Project that the SN curve for ferritic steels is equally applicable to duplex stainless steels.

The stress used in the assessment is the maximum peak-to-peak principal stress range in the parent material adjacent to the weld toe or discontinuity. This again assumes the direction of the principal stress does not change throughout the cycle (in the case of high cycle fatigue this is a reasonable assumption). The position of the relevant stress to be used in conjunction with the fatigue curves is that which takes account of the effect due to the additional membrane and bending stresses caused by the branch or due to the loads on the branch. The influence of the local stresses at the weld, due to the corner, is included in the fatigue curve. The various stress definitions are shown in Figure 5-2.

In both BS codes a range of geometries is defined and a letter is assigned to each geometry or structural arrangement. BS5500 tends to consider those geometries which are more relevant to pressure vessels whereas BS7608 concentrates on geometries which are found in structural steel fabrications. In both cases, however, the SN Curves are based on experimental tests **where components have been manufactured to the appropriate standard and quality.** The strain levels in the components have been measured using strain gauges adjacent to the weld, but sufficiently remote so as to ignore the localised stress concentration adjacent to the weld toe. The curves are presented both graphically and as equations. The data has been manipulated to produce curves based on mean (50% probability of survival), minus two standard deviations (97.7% probability of survival) or minus three standard deviations (99.7% probability of survival).

In the assessment of stresses in components which are in service, the endurance limit for a component is taken from the curves (Figure 5-1) which define the 97.7% probability of survival for that particular geometry or its nearest equivalent. If values of measured dynamic stress are found above this level, action must be taken immediately to rectify the problem. If levels above half of this level are found, remedial action is recommended as soon as possible to safeguard the plant.

Example

For small bore connections using BS7608 and a weld class F2 (Figure 5-1) action is required immediately if the dynamic stress range exceeds 35 MPa peak to peak. Consideration for remedial action is required if the dynamic stress range exceeds 17.5 MPa peak to peak

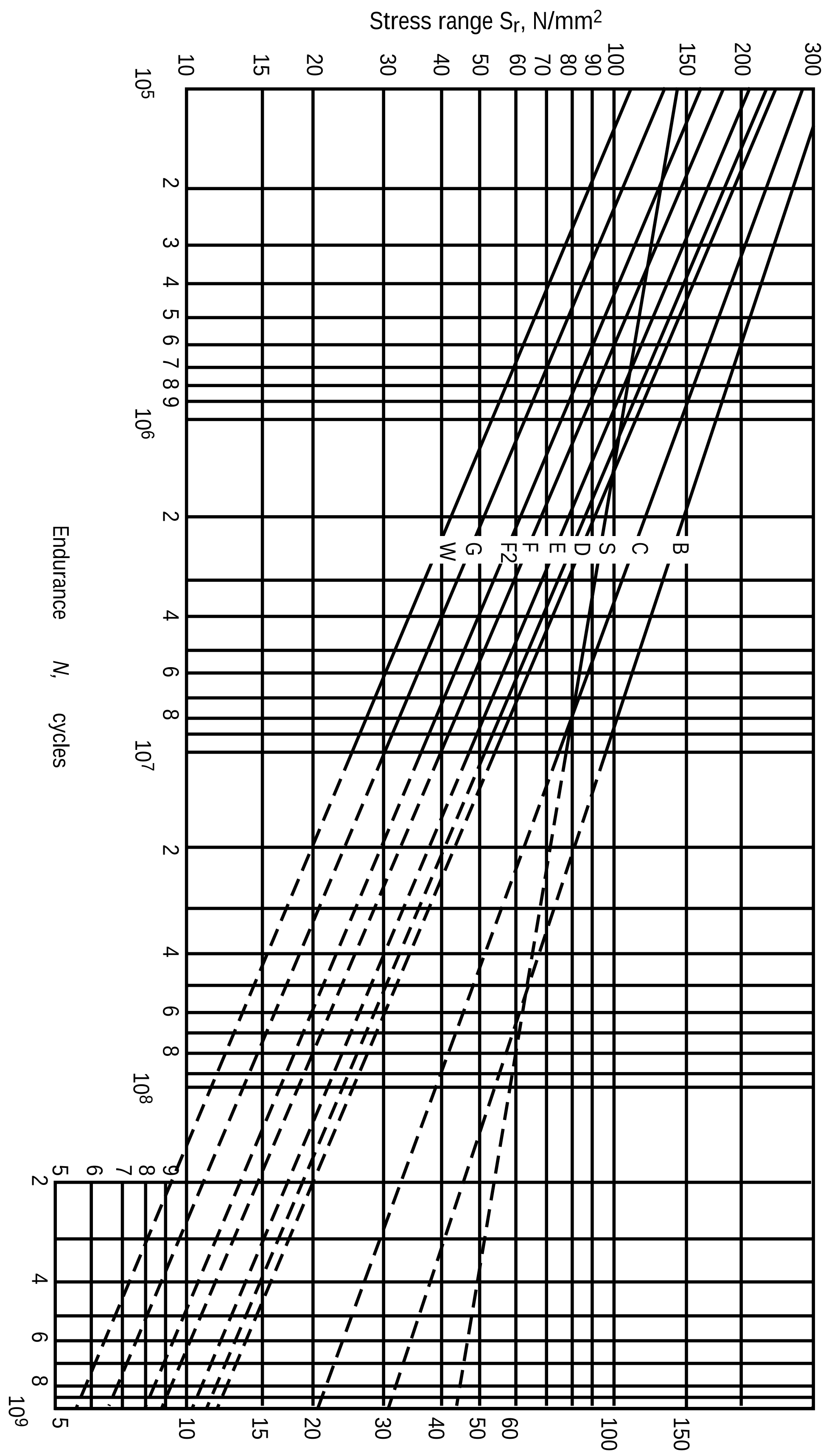


Figure 5-1 *S-N curves for different weld classes (courtesy BS7608) [15]*

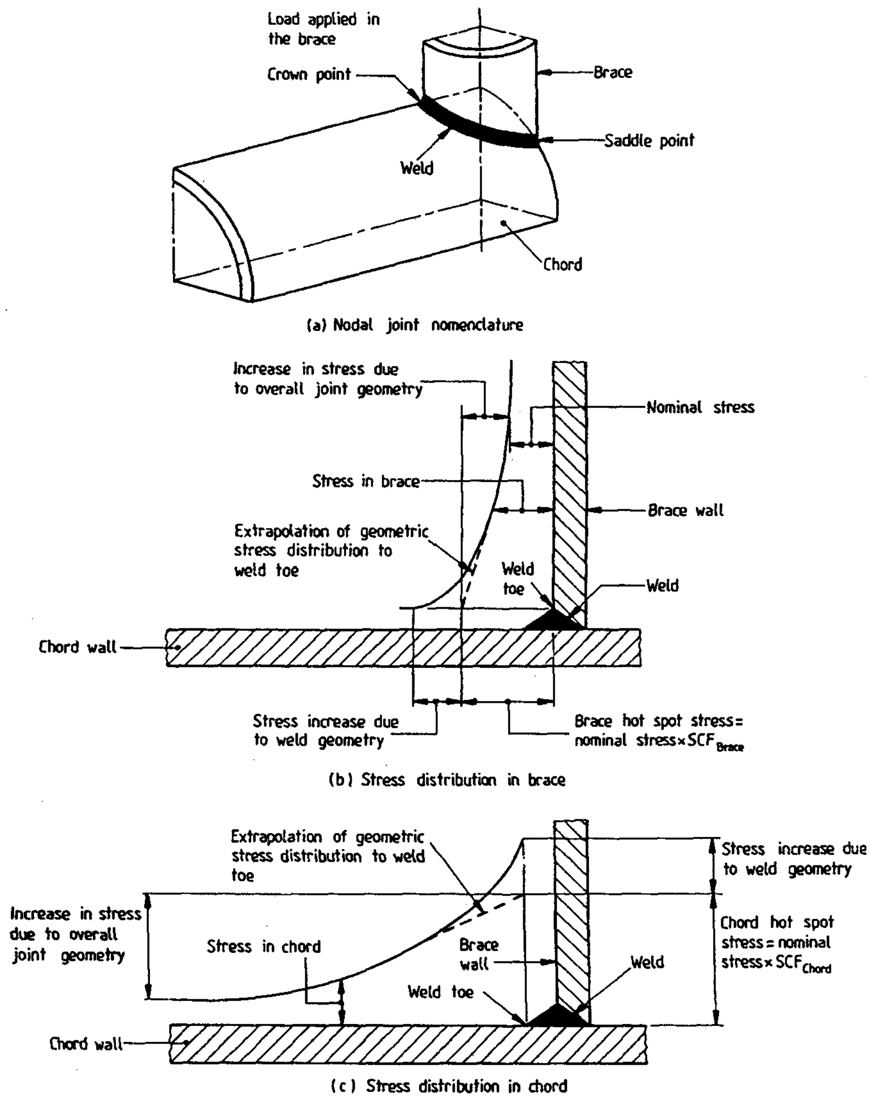


Figure 5-2 *Stress distribution at a typical connection (courtesy BS7608) [15]*

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6.0 References

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13 Little, W.J.G: "Fatigue assessment of pressure vessels, pipework and structures", *Sastech mechanical equipment engineering symposium,* South Africa (1992).

14 Holmes, E.: "*Handbook of Industrial Pipework Engineering*", McGraw Hill (1973).

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16 BS5500: "Code of Practice for Unfired Fusion Welded Pressure Vessels", British Standards Institution (1997).

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Appendix A1
Stage 1 - Identification of Excitation Mechanisms

1.0 Overview

The purpose of this appendix is to identify whether or not there is a potential problem for each system. If there is a potential problem this stage will identify what the excitation mechanisms are. The most common excitation mechanisms are identified below and described in more detail in Section 2.1 of the main text:

- flow induced turbulence;
- high frequency acoustic excitation;
- mechanical excitation;
- pulsation - reciprocating machinery
  - rotating stall
  - periodic flow induced excitation.

The mechanism considered under periodic flow induced excitation is flow past a branch (dead leg). Other mechanisms such as flow over components and thermowells are outwith the remit of these Guidelines.

These Guidelines are for 'steady state' operation of plant. They do not consider:

- the response of the structure to impulse loading (impact loading, slugflow, pressure transients as a result of rapid valve closure, impact loading etc);
- environmental excitation mechanisms such as inertial effects on pipework on ships and floating facilities;
- wave induced motion on risers, seismic excitation and topside vibration due to ship collision.

However, the design of small bore connections in line with the recommendations in these Guidelines will minimise the likelihood of failure due to these excitation mechanisms.

The identification of problem areas and their excitation mechanisms for each system are established by the questionnaires below. There are two questionnaires; Questionnaire 1 for liquid/multiphase systems and Questionnaire 2 for gas systems. For each system use the appropriate questionnaire. Stage 1 in Worksheet 3-1 must be completed with 'yes' or 'no' to identify which mechanisms are present.

2.0 Questionnaire 1 - Liquid/Multiphase fluid systems

<u>Q1.1 Flow induced turbulence</u>

Is the system operating under steady state conditions?

The following are examples of situations that are not considered as a flow induced vibration problem: relief and blowdown events and transients caused by valve closures.

yes - flow induced turbulence is a potential problem
no - no problem

Q1.2 High Frequency Acoustic Excitation

Does the system have choked flow?
no - High frequency acoustic excitation is not a problem with liquid/multiphase systems.

Q1.3 Mechanical Excitation

Is there reciprocating or rotating equipment in the system?

yes - mechanical excitation is a potential problem
no - no problem

Is the system close to a reciprocating compressor/pump on another system or any other large vibration source such as a diesel engine? note: The definition of close is not definitive but the following is a rule of thumb based on engineering experience. For offshore plants close is defined as being supported from the same module/deck (above or below). For onshore plants close is defined as a radius equal to twice the maximum length of the skid.

yes - mechanical excitation is a potential problem
no - no problem

Q1.4 Pulsation (Reciprocating machinery)

Is there a reciprocating pump in the system?

yes - pulsation (reciprocating machinery) is a potential problem
no - no problem

Q1.5 Pulsation (Rotating Stall)

If the system has a centrifugal compressor does it have a rotating stall characteristic?

no - Pulsation (Rotating Stall) is not a problem with liquid/multiphase systems.

Q1.6 Pulsation (Periodic Flow Induced Excitation)

Does the system have any dead leg branches longer than 1 m?

no - Pulsation due to flow induced excitation is not a problem with liquid/multiphase systems.

Recommended Action

If any of the above questions indicate a potential problem proceed to Stage 2.

If all of the above questions indicate no potential problems there could still be a problem with small bore connections, if other excitation mechanisms are present (eg..intermittent shock). To ensure that design features of small bore connections are acceptable a visual survey should be undertaken and the basic design of the S.B.C. should be checked to see if it is sound as given in Section 4.3.

3.0 Questionnaire 2 - Gas systems

Q2.1 Flow induced turbulence

Is the system operating under steady state conditions? The following are examples of situations that are not considered as a flow induced vibration problem: relief and blowdown events and transients caused by valve closures.

yes - flow induced turbulence is a potential problem
no - no problem

Q2.2 High Frequency Acoustic Excitation

Does the system have choked flow?

yes - high frequency acoustic excitation is a potential problem
no - no problem

Q2.3 Mechanical Excitation

Is there reciprocating or rotating equipment in the system?

yes - mechanical excitation is a potential problem
no - no problem

Is the system close to a reciprocating compressor/pump on another system or any other large vibration source such as a diesel engine? note: The definition of close is not definitive but the following is a rule of thumb based on engineering experience. For offshore plants close is defined as being supported from the same module/deck (above or below). For onshore plants close is defined as a radius equal to twice the maximum length of the skid.

yes - mechanical excitation is a potential problem
no - no problem

Q2.4 Pulsation (Reciprocating machinery)

Is there reciprocating compressor or pump in the system?

yes - pulsation (reciprocating machinery) is a potential problem
no - no problem

Q2.5 Pulsation (Rotating stall)

If the system has a centrifugal compressor does it have a rotating stall characteristic?

yes - pulsation (rotating stall) is a potential problem
no - no problem

Q2.6 Pulsation (Periodic Flow Induced Excitation)

Does the system have any dead leg branches longer than 1 m?

yes - pulsation (periodic flow) is a potential problem
no - no problem

Recommended Action

If any of the above questions indicate a potential problem, proceed to Stage 2.

If the only excitation mechanism identified as a potential problem in gas systems from Stage 1 is high frequency acoustic excitation then a Stage 3 analysis is not required.

If all of the above questions indicate no potential problems there could still be a problem with small bore connections, if other excitation mechanisms are present (eg.intermittent shock). To ensure that design features of small bore connections are acceptable a visual survey should be undertaken and the basic design of the S.B.C. should be checked to see if it is sound as given in Section 4.3.

Appendix A2
Stage 2 - Detailed Screening of Main Pipe

1.0 Overview

The purpose of this stage is to define clearly whether a potential problem system has a vibration problem. This appendix will identify whether the design is intolerable or what is the likelihood of failure. The flowchart for the likelihood of failure (L.O.F.) calculation is given in Figure A-1.

2.0 Method

2.1 Process Information

In the event of being unable to perform the Stage 2 calculations for the applicable excitation mechanism identified in Stage 1 the default process values shall be used. Further it is recognised that not all the process data may be available to carry out this assessment. If this is the case, then default values may be used. These default values are based mainly on the process areas relating to offshore oil and gas plants and are described below.

Process Area	Likelihood of Failure (L.O.F.)
Gas Compression	0.9
Separation	0.8
Risers / Flowlines	0.8
Glycol	0.8
Crude / MOL Pumps	0.4
Crude Oil Distillation	0.4
Water Injection	0.4
Flare	0.2
NGL Recovery	0.2

Table A-1 *Default Process Data*

If default values are used, the assessment must be revisited after process information has been obtained and the L.O.F. reassessed.

2.2 Excitation Mechanisms

Flow Induced Turbulence

Calculate the L.O.F. from Appendix A2.1

If process information is not available use default L.O.F. values in Table A-1. If default values have been used a flow induced turbulence calculation according to Appendix A2.1 must be revisited after process information is obtained and the L.O.F. reassessed.

High Frequency Acoustic Excitation

The Stage 2 screening requires the calculation of the acoustic sound power (PWL) acting on the main line due to the sound generated by the pressure reducing device.

Calculate the L.O.F. as detailed below.

If PWL ≥ 155 dB then LOF = 1.0

If PWL< 155 dB then LOF = 0.2

Where:

$$\text{PWL (source)} = 10\log_{10}\left[\left(\frac{p_1 - p_2}{p_1}\right)^{3.6} W^2 \left(\frac{t_1}{M}\right)^{1.2}\right] + 126.1$$

PWL = sound power level (dB) generated by the pressure reducing device causing choked flow
p_1 = pressure upstream of pressure reducing device (kPa absolute)
p_2 = pressure downstream of pressure reducing device (kPa absolute)
W = mass flow rate (kg/s)
t_1 = upstream temperature (K)
M = molecular weight of gas

In the absence of information, any line with choked flow, assume an L.O.F.=1.0.

Mechanical Excitation

The likelihood of failure is set to the values below. If a detailed structural dynamic analysis of the main line pipework and its supports has been conducted and coincidence with excitation forces from the rotating equipment has been eliminated, the L.O.F. for reciprocating compressor/pumps and diesel engines can be reduced to 0.4.

Pipework connected or adjacent to	Likelihood of Failure (L.O.F.)
Reciprocating Compressor/Pump	0.9
Diesel Engine	0.8
Centrifugal Pump	0.4
Centrifugal Compressor	0.2
Fan	0.2

Table-A2 *Mechanical Excitation values*

Pulsation (Reciprocating Compressor/Pump)

Calculate the likelihood of failure (L.O.F.) from below:

Is the power of the reciprocating compressor/pump less than 112 kilowatts and the discharge pressure less than 35 bar (equivalent to Level 1 of API 618)?

yes - L.O.F.= 0.4
no - Has an API 618/674 analysis been conducted?

yes - L.O.F.= 0.4
no - L.O.F.= 1.0

In the absence of specific information regarding reciprocating compressor/pump assume as L.O.F.=1.0.

Pulsation (Rotating Stall)

Calculate the likelihood of failure (L.O.F.) from below:

Is there a rotating stall characteristic in the centrifugal compressor?

no - L.O.F.= 0.2
yes - Is the centrifugal compressor operating at low flow conditions (i.e. around the rotating stall conditions)?

yes - L.O.F.= 1.0
no - L.O.F.= 0.4

In the absence of specific information regarding rotating stall assume an L.O.F.=1.0.

Pulsation (Periodic Flow Induced Excitation)

Calculate the critical side branch diameter d_{crit} using the following expression

$$d_{crit} = \left[\frac{400}{\pi\rho v^2}\right]^{0.5} \qquad (4)$$

where:

ρ is the density of the fluid (kg/m^3)
v is the velocity of the fluid (m/s)
d_{crit} is the critical side branch diameter (m)

NOTE: Units are in the SI system i.e. $\rho v^2 \equiv kg/(m\ s^2)$. Density and flow rate are actual values, not those at standard temperature and pressure.

If there are any dead leg branches on the main pipe with an internal diameter > d_{crit} then L.O.F. = 1.0, otherwise L.O.F. = 0.0.

If process information is not available use Default Process L.O.F. values in Table A-1. If default values have been used a dead leg branch calculation must be revisited after process information is obtained.

Note: It should be noted that the mechanism considered under pulsation – periodic flow induced excitation is that due to flow past a branch (dead leg). Other mechanisms such as flow over components and thermowells are outwith the remit of these Guidelines.

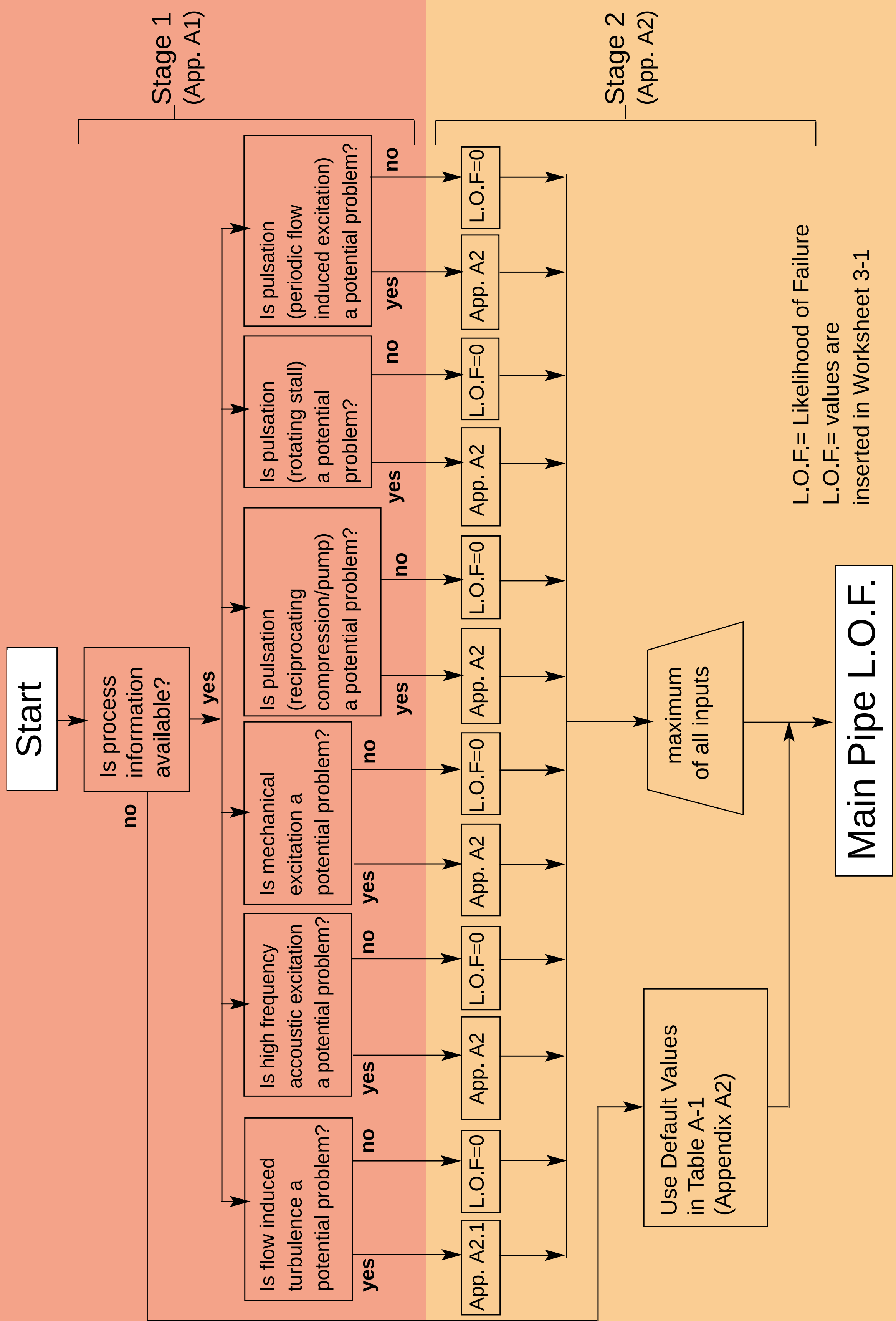

Figure A-1 *Stage 1 (Identification of Excitation Mechanism) and Stage 2 (Detailed Screening of Main Pipe) Flowchart*

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Appendix A2.1

Screening Method for Flow Induced Turbulence in Process Piping Systems

1.0 Support Arrangement

The screening method is designed for four support arrangements; stiff, medium stiff, medium and flexible, as detailed below. The principal response of the pipe to low frequency flow induced turbulence is associated with the low frequency bending modes of piping spans, either between supports or, if the supports are poorly designed, the supports should also be included.

‘Stiff Support Arrangement’: applicable to piping systems which are well supported (as per recommendations given in [14]). The fundamental natural frequency of the piping span is approximately 14 to 16 Hz.

‘Medium Stiff Support Arrangement’: applicable to piping systems which are well supported. The fundamental natural frequency of the piping span is approximately 7 Hz.

‘Medium Support Arrangement’: applicable to piping systems which are well supported. The fundamental natural frequency of the piping span is approximately 4 Hz.

‘Flexible Support Arrangement’: applicable to piping systems where long unsupported spans are encountered and the fundamental natural frequency of the piping span is approximately 1 Hz. An example of such a system is a wellhead flowline where increased flexibility is required to accommodate riser movement.

The selection of support arrangement can be simplified as follows (Figure A-2):

Support Arrangement	Span Length Criteria	Typical Natural Frequency
Stiff	$L \le -1.2346*10^{-5}D^2 + 0.02D + 2.0563$	14 to 16 Hz
Medium Stiff	$L > -1.2346*10^{-5}D^2 + 0.02D + 2.0563$ $L \le -1.1886*10^{-5}D^2 + 0.025262D + 3.3601$	7 Hz
Medium	$L > -1.1886*10^{-5}D^2 + 0.025262D + 3.3601$ $L \le -1.5968*10^{-5}D^2 + 0.033583D + 4.429$	4 Hz
Flexible	$L > -1.5968*10^{-5}D^2 + 0.033583D + 4.429$	1 Hz

L≡span length (m) D≡actual outside diameter (mm)

Table A-3 *Support Arrangement*

The likelihood of failure is then determined by the following equation:

$$L.O.F. = \frac{\rho v^2}{F_V} \qquad (5)$$

where ρ is the density of the fluid (kg/m^3) and *v* is the velocity of the fluid (m/s). F_V is a flow induced vibration number dependent on the actual outside diameter of the pipe D (mm), the wall thickness T (mm) and the support arrangement (see Table A-4).

2.0 Screening Method

Calculation of ρv^2

The first step is to calculate ρv^2 using the following equations depending on whether the fluid is single phase or multi-phase flow:

For single phase flow:

ρv^2 = (actual liquid or gas density) x (actual liquid or gas velocity)^2

For multi-phase flow: ρv^2 = (effective density) x (effective velocity)^2

where:

effective density = total mass flow rate/total volumetric flow rate (kg/m^3)
effective velocity = total volumetric flow rate/pipe internal cross sectional area A, (m /s)

and

total mass flow rate = ∑ (actual volumetric flow rate for each phase x phase density, kg/s)
total volumetric flow rate = ∑ (actual volumetric flow rate for each phase, m^3/ s)

NOTE: Units are in the SI system i.e. $\rho v^2 \equiv kg/(m\ s^2)$. Density and flow rate are actual values, not those at standard temperature and pressure.

Calculation of Fv

The second step is to calculate Fv.

Fv is defined in Table A-4 given on page A2.1-4 depending on actual outside diameter (D), wall thickness (T) and whether the support is a **stiff, medium stiff, medium,** or **flexible** support arrangement. The definition of stiff, medium stiff, medium, and flexible support arrangement has been expressed in terms of span length (L) and actual outside diameter (D).

Calculation of Likelihood of Failure (L.O.F.)

The third step is to calculate the Likelihood of Failure using equation (5).

	Range of Outside Diameter	F_v	α	β	Figure No
Stiff	60 mm to 762 mm	$\alpha(D/T)^{\beta}$	$446187.1+645.51D+9.166*10^{-4}D^{3}$	$0.1\ln(D)-1.3739$	A-3
Medium Stiff	60 mm to 762 mm	$\alpha(D/T)^{\beta}$	$283921+370.24D$	$0.1106\ln(D)-1.501$	A-4
Medium	273. mm to 762 mm	$\alpha(D/T)^{\beta}$	$150412+208.93D$	$0.0815\ln(D)-1.3269$	A-5
Medium	60 mm to 219 mm	$\exp\left[\alpha(D/T)^{\beta}\right]$	$13.1129-4.7455*10^{-3}D+1.4141*10^{-5}D^{2}$	$-0.132+2.282*10^{-4}D-3.7245*10^{-7}D^{2}$	A-6
Flexible	273 mm to 762 mm	$\alpha(D/T)^{\beta}$	$41.21D+49397$	$0.0815\ln(D)-1.3842$	A-7
Flexible	60 mm to 219 mm	$\exp\left[\alpha(D/T)^{\beta}\right]$	$1.3175*10^{-5}D^{2}-4.4213*10^{-3}D+12.217$	$-4.622*10^{-7}D^{2}+2.835*10^{-4}D-0.164$	A-8

Note: $\exp[z] = e^{z}$

Table A-4 *Method of calculating F_V*

NOTE:
For flexible pipes having a natural frequency greater than 1 Hz and less than or equal to 3 Hz, advanced screening should be carried out in accordance with Appendix A2.2. This is particularly relevant where the L.O.F. from flow induced turbulence is greater than or equal to 1.0. This is necessary because the L.O.F. for flexible pipes is very sensitive to its fundamental natural frequency.

Example 1

Input

Main line:	14" Schedule 10S D= 355.6mm (14.0"), T = 4.78mm (0.1
Flow rate:	375 kg/s
Fluid:	water
Support arrangement:	stiff (water injection)

Step 1:

$$v = \frac{\text{flow rate}}{(\text{density})(\text{pipe internal cross - sectional area})} = \frac{Q}{\rho A} = 3.987\,{}^{m}\!/\!_{s}$$

$$Q = 375\,{}^{kg}\!/\!_{s}$$

$$\rho = 1000\,{}^{kg}\!/\!_{m^3} \tag{6}$$

$$A = \frac{\pi(\text{ID})^2}{4} = \frac{\pi(0.3556 - 2(0.00478))^2}{4} = 0.0940\text{m}^2$$

$$\rho v^2 = (1000\,{}^{kg}\!/\!_{m^3})(3.987\,{}^{m}\!/\!_{s})^2 = 15899\,{}^{kg}\!/\!_{ms^2}$$

Step 2:

D/T = 74.4

Step 3:

α=716946.4, β = -0.7865
$F_v = \alpha(D/T)^{\beta} = 24181.8$
L.O.F.=15899/24181.8
L.O.F.=0.657

Example 2

Input

Main line:	8" Schedule 80, D=219mm(8.625"), T=12.7mm (0.5")
Flow rate:	265.7 m^3/hr (oil), 382.3m^3/hr (gas)
Fluid:	oil (density 920 kg/m^3) & gas (density 36.5 kg/m^3)
Support arrangement:	flexible (wellhead flowline)

Step 1:

$$\text{effective density} = \frac{\text{total mass flow rate}}{\text{total volumetric flow rate}} = \frac{m_{flow}}{Q} = 398.8\,{}^{\text{kg}}\!/\!_{\text{m}^3}$$

$$m_{flow} = \frac{265.7\,{}^{\text{m}^3}\!/\!_{\text{hr}}}{3600\,{}^{s}\!/\!_{hr}}920\,{}^{kg}\!/\!_{m^3} + \frac{382.3\,{}^{\text{m}^3}\!/\!_{\text{hr}}}{3600\,{}^{\text{s}}\!/\!_{\text{hr}}}36.5\,{}^{kg}\!/\!_{m^3} = 71.78\,{}^{kg}\!/\!_{s} \tag{7}$$

$$Q = \frac{265.7\,{}^{m^3}\!/\!_{hr}}{3600\,{}^{s}\!/\!_{hr}} + \frac{382.3\,{}^{m^3}\!/\!_{hr}}{3600\,{}^{s}\!/\!_{hr}} = 0.18\,{}^{m^3}\!/\!_{s}$$

$$\text{effective velocity} = \frac{\text{total volumetric flow rate}}{\text{pipe internal cross-sectional area}} = \frac{Q}{A} = 6.11\,{}^{m}\!/_{s}$$

$$A = \frac{\pi (ID)^2}{4} = \frac{\pi\left(0.219075 - 2(0.0127)\right)^2}{4} = 0.0294m^2 \tag{8}$$

$$Q = \frac{265.7\,{}^{m^3}\!/_{hr}}{3600\,{}^{s}\!/_{hr}} + \frac{382.3\,{}^{m^3}\!/_{hr}}{3600\,{}^{s}\!/_{hr}} = 0.18\,{}^{m^3}\!/_{s}$$

$$\rho v^2 = \left(398.8\,{}^{kg}\!/_{m^3}\right)\left(6.10\,{}^{m}\!/_{s}\right)^2 = 14909\,{}^{kg}\!/_{ms^2} \tag{9}$$

Step 2:

D/T = 17.2

Step 3:

α=11.8806, β=-0.1241
F_v=exp(α(D/T)$^{\beta}$)=4206.6
L.O.F.=14909/4206.6
L.O.F.=3.54
$\therefore$ L.O.F.>1.0

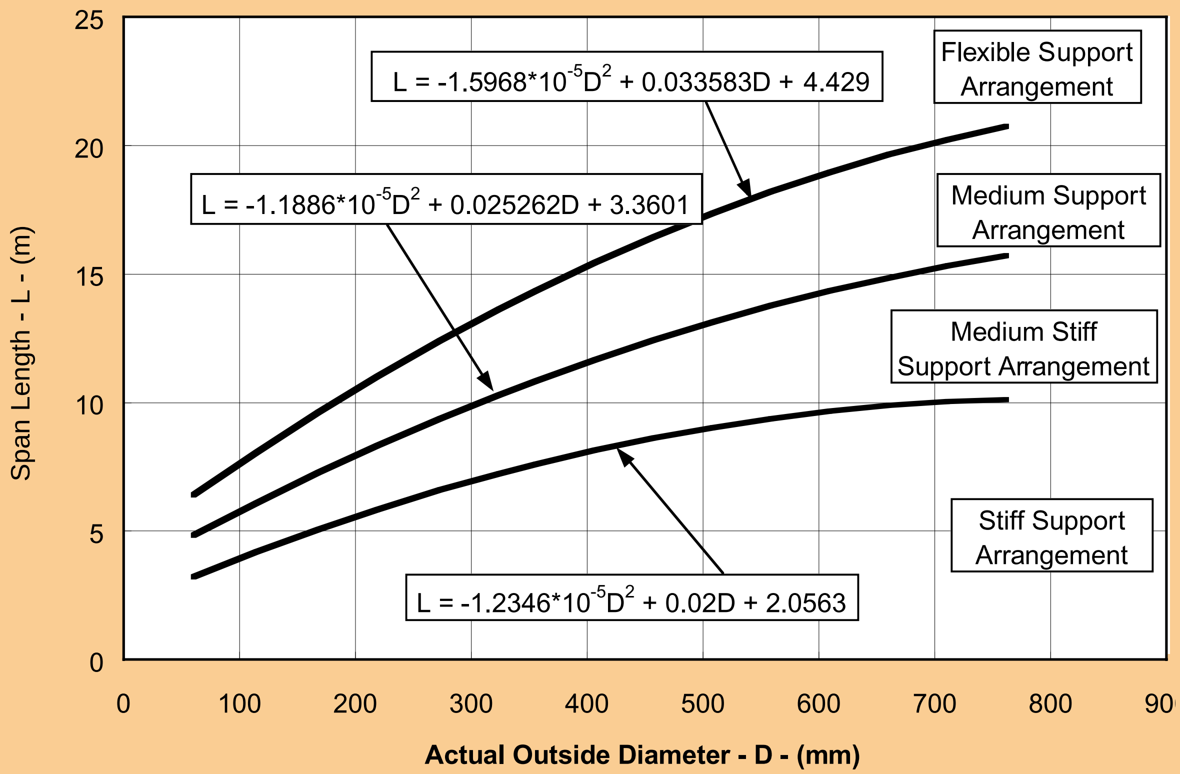

Figure A-2 *Different support arrangements as a function of span length and outside diameter*

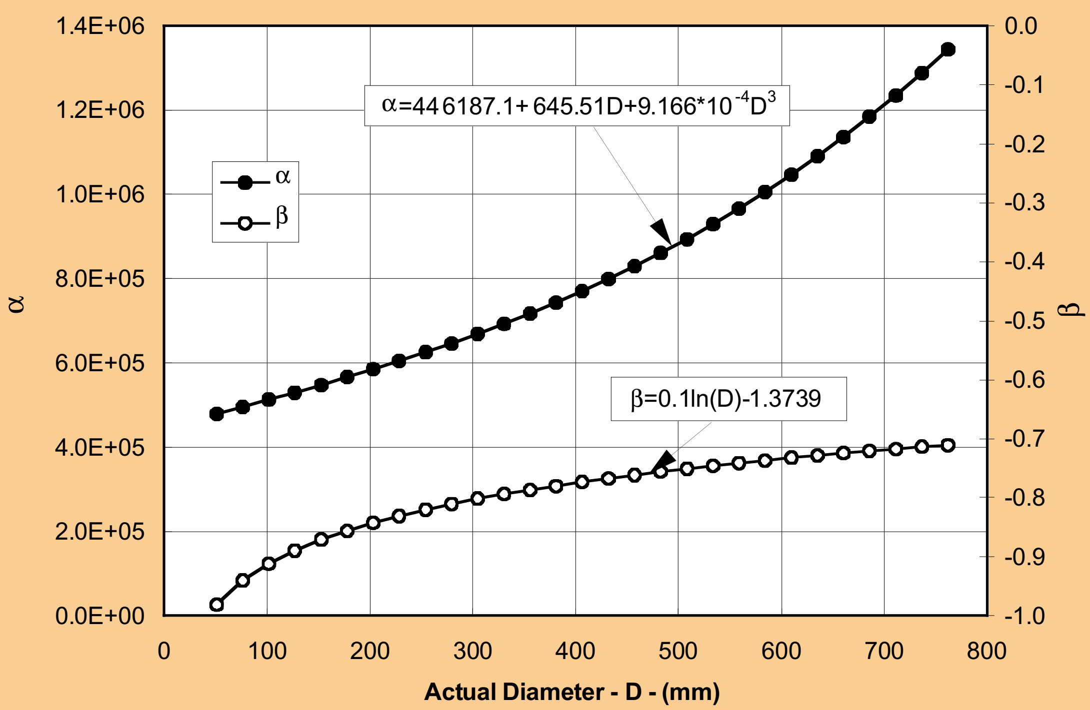

Figure A-3 *Fv Curve Fit - Stiff Support Arrangement – 60 mm<D<762 mm*

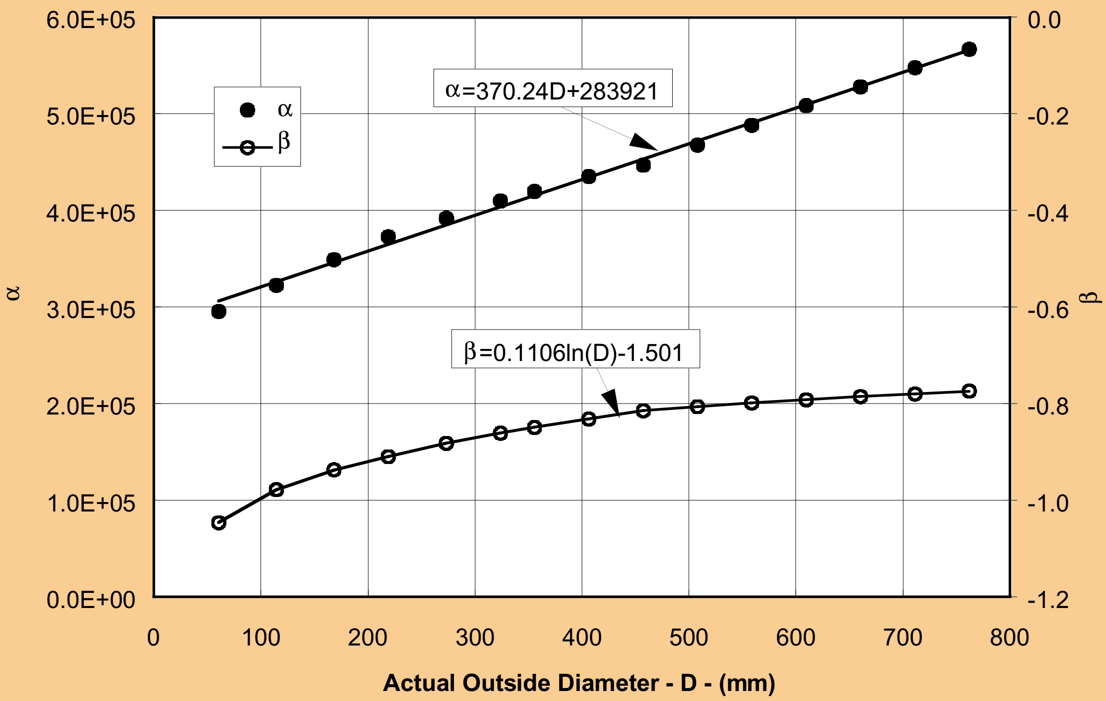

Figure A-4 *F_V Curve Fit – Medium Stiff Support Arrangement – 60 mm<D<762 mm*

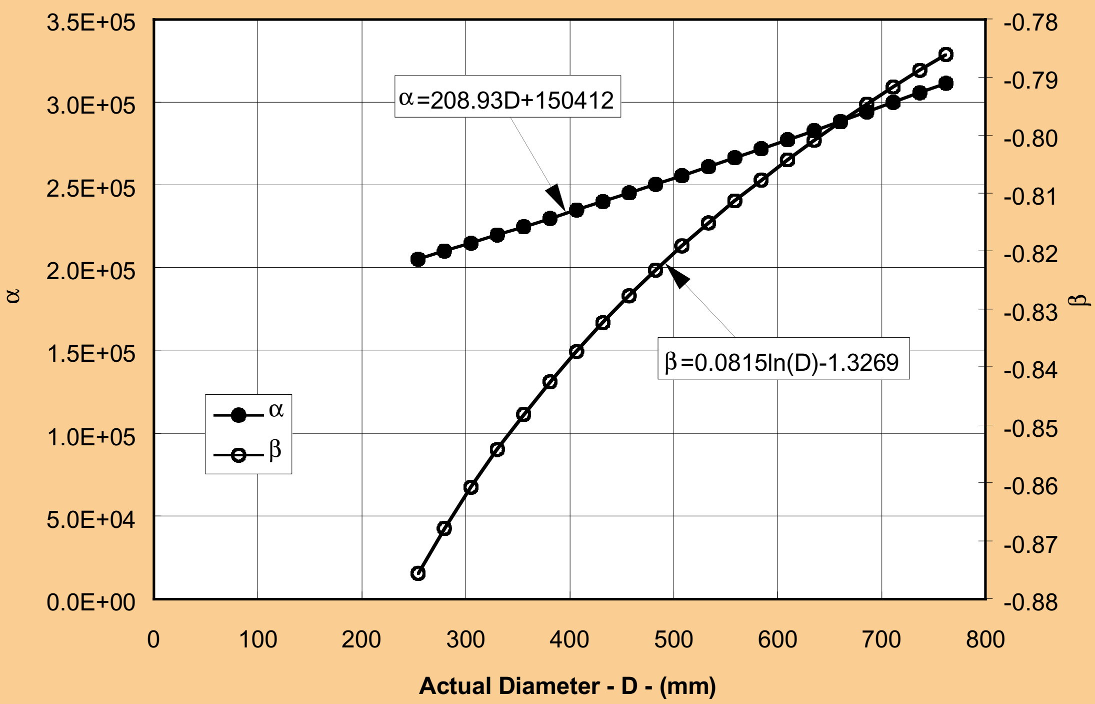

Figure A-5 *F_V Curve Fit - Medium Support Arrangement - 273 mm<D<762 mm*

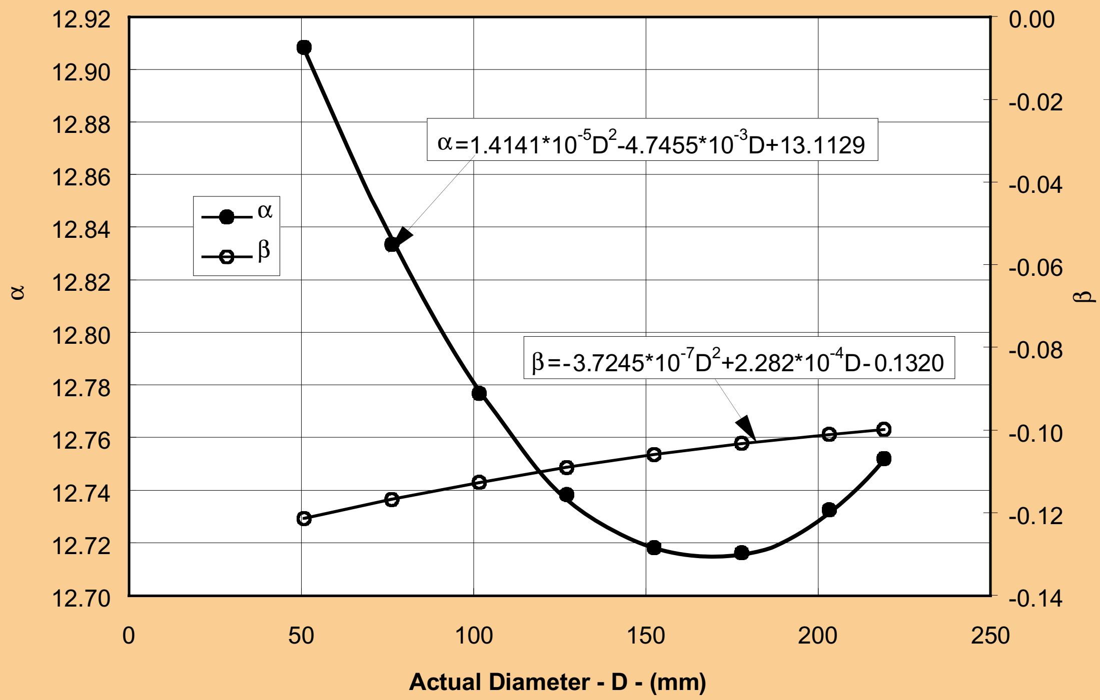

Figure A-6 *Fv Curve Fit - Medium Support Arrangement - 60 mm<D<219 mm*

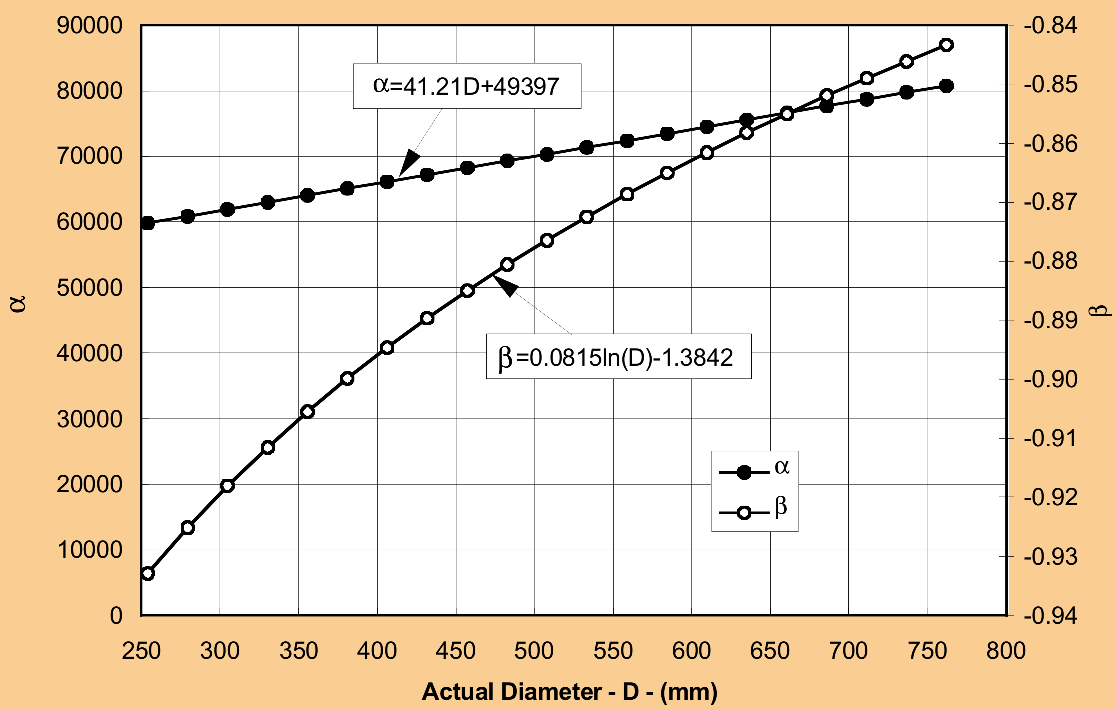

Figure A-7 *Fv Curve Fit - Flexible support arrangement – 273 mm<D<762 mm*

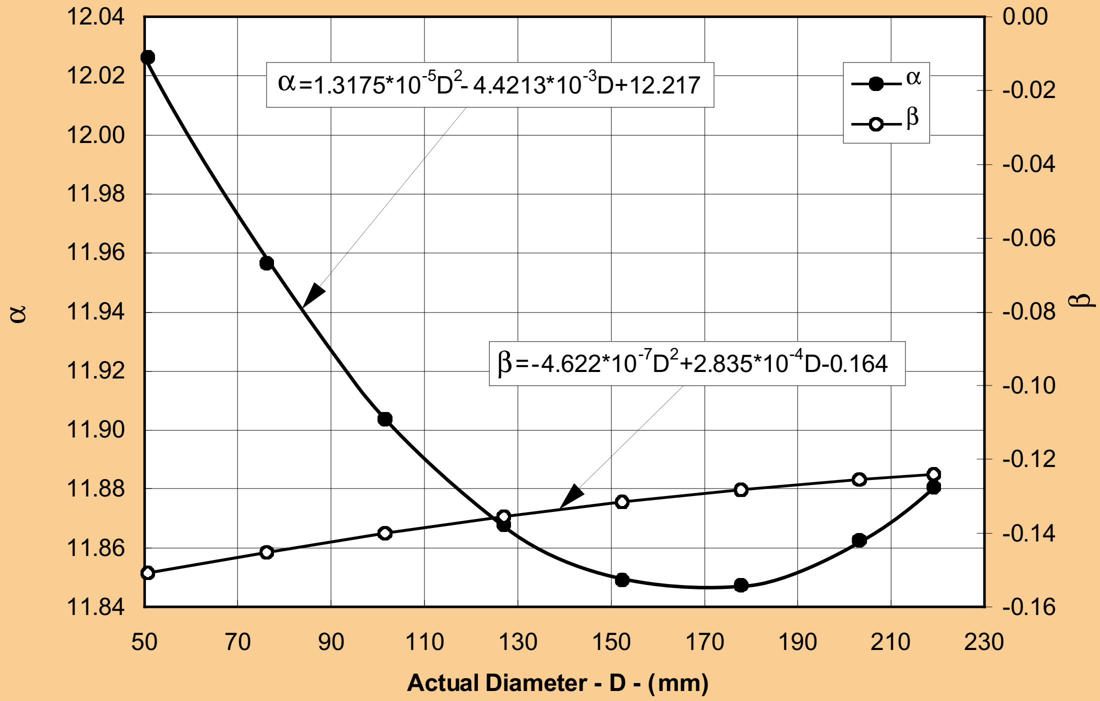

Figure A-8 *Fv Curve Fit- Flexible Support Arrangement – 60 mm<D<219 mm*

Appendix A2.2

Advanced Screening Method for Flow Induced Turbulence in Flexible Pipework

1.0 Overview

The method detailed in Appendix A2.1 for flexible pipework, assumes a fundamental natural frequency of the pipe span of 1 Hz.

In a number of cases, the actual fundamental natural frequency of a flexible pipe span may be significantly higher, and in such a situation the method given in Appendix A2.1 may be too conservative.

The method detailed in this Appendix provides a means of calculating the L.O.F. of flexible pipework, taking into account the fundamental natural frequency of the main pipe for a range of fundamental natural frequencies.

The method only applies to pipe spans with a fundamental natural frequency greater than 1 Hz and less than or equal to 3 Hz.

The method is a screening assessment and is therefore is based on basic process and structural parameters. In certain situations, depending on the local configuration of the pipe and its support arrangement, the method may not be conservative. In certain circumstances the pipe should be checked for natural frequency as described in section 4.0 of this Appendix. If there is any uncertainty regarding the application of this method then specialist advice should be sought.

2.0 Calculation Method

If a flexible line has been assessed using the method given in Appendix A2.1, and an L.O.F. ≥ 1.0 obtained, then consideration should be given to reassessing the pipe using the calculation method given below. The fundamental natural frequency of the pipe can be calculated once detailed isometric drawings are available.

The resulting L.O.F. value may then be substituted for the original L.O.F. value in Worksheet 3.1.

The likelihood of failure due to flow induced turbulence for main pipe is calculated using:

$$LOF = \frac{\rho v^2}{F_v}$$

where ρ is the density of the fluid (kg/m^3) and v is the velocity of the fluid (m/s). F_V is a flow induced vibration number dependent on the actual outside diameter of the pipe D (mm), the wall thickness T (mm) and the support arrangement.

The following is valid for flexible pipe spans having actual diameters 273 mm to 762 mm (i.e. greater than 10 inch nominal) and structural natural frequencies (f_n) ranging from 1 Hz to 3 Hz.

Flexible Support Arrangement for D > 273mm:

$$F_V = a\left({}^{D}\!/_{T}\right)^{b}$$
$$\alpha = (41.21\mathrm{D} + 49397)\left(f_n^{\,0.0001665\,\mathrm{D}+0.84615}\right) \qquad (10)$$
$$\beta = 0.0815\ln(\mathrm{D}) - 1.3842 + 0.0191(f_n - 1)$$

The following is valid for flexible pipe spans having actual diameters less than 219 mm (i.e. 8 inch nominal) and structural natural frequencies (f_n) ranging from 1 Hz to 3 Hz..

Flexible Support Arrangement for D > 219mm:

$$F_V = \exp\left[\alpha\left({}^{D}\!/_{T}\right)^{\beta}\right]$$
$$\alpha = \left(1.3175 * 10^{-5} D^2 - 4.4213 * 10^{-3} D + 12.217\right)\left(0.0529\ln(f_n) + 1\right) \qquad (11)$$
$$\beta = (-4.622*10^{-7}\mathrm{D}^2 + 2.835*10^{-4} D - 0.164)(-0.1407\ \ln(f_n) + 1)$$

3.0 Limitations

Extreme care needs to be taken with such an assessment because the method relies heavily on knowing the fundamental natural frequency of the pipe.

Once detailed isometric drawings are available then an initial assessment of the fundamental natural frequency of the line can be undertaken (e.g. using pipework analysis software).

The predicted fundamental natural frequency will depend on several aspects, including:

- The mass distribution of the pipe (including lagging, contained fluid and lumped masses such as valves etc).
- The stiffness of the pipe and its supports in particular.

One of the most difficult aspects to determine is the influence of the support arrangement. Pipe supports can act very differently dynamically compared with statically, so careful consideration must be given to how supports are represented in a pipework model. Often, for static analysis, supports are modelled simply by constraining the appropriate degrees of freedom on the pipe at the support location. However, this may be incorrect from a dynamic standpoint for two reasons:

(i) Certain degrees of freedom which may be released in a static model may be fixed for the dynamic case. An example of this is a guided support which allows (static) thermal growth in the axial direction, but which (due to friction between the pipe and the support) restrains the pipe dynamically in the axial direction unless the dynamic forces generated are so high that friction is overcome.

(ii) The support itself (and even the deck, piperack or structure to which the support is attached) may flex with the pipe, and therefore cause a lowering of the fundamental natural frequency of the line compared to the case where the support is assumed to be infinitely stiff.

If there are any doubts regarding this type of assessment then specialist advice should be sought.

4.0 Existing lines

Where piping systems are installed and filled with process fluid, the fundamental natural frequency can be measured as this will provide the most accurate means of assessing frequency. Specialist techniques will have to be applied to obtain meaningful data. The main benefit of measuring the natural frequency would be where screening is required for a future change in duty i.e. higher flow rate or fluid density change.

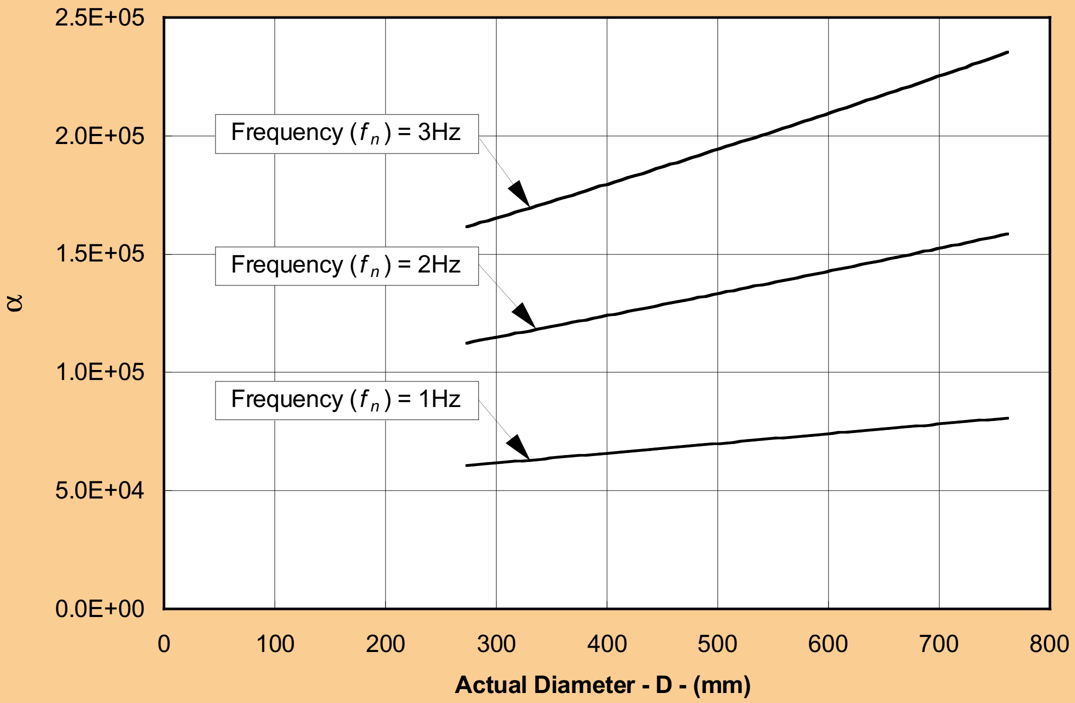

Figure A-9 *α curve fit - Flexible Support Arrangement - 273 mm<D<762 mm*

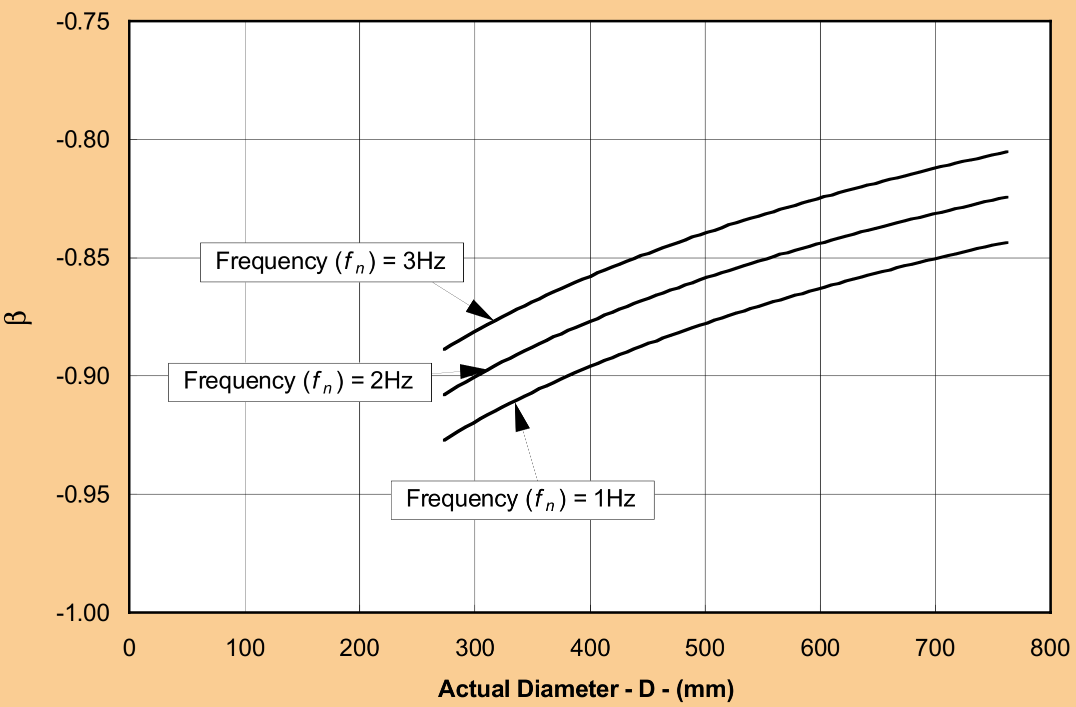

Figure A-10 *β curve fit - Flexible Support Arrangement - 273 mm<D<762 mm*

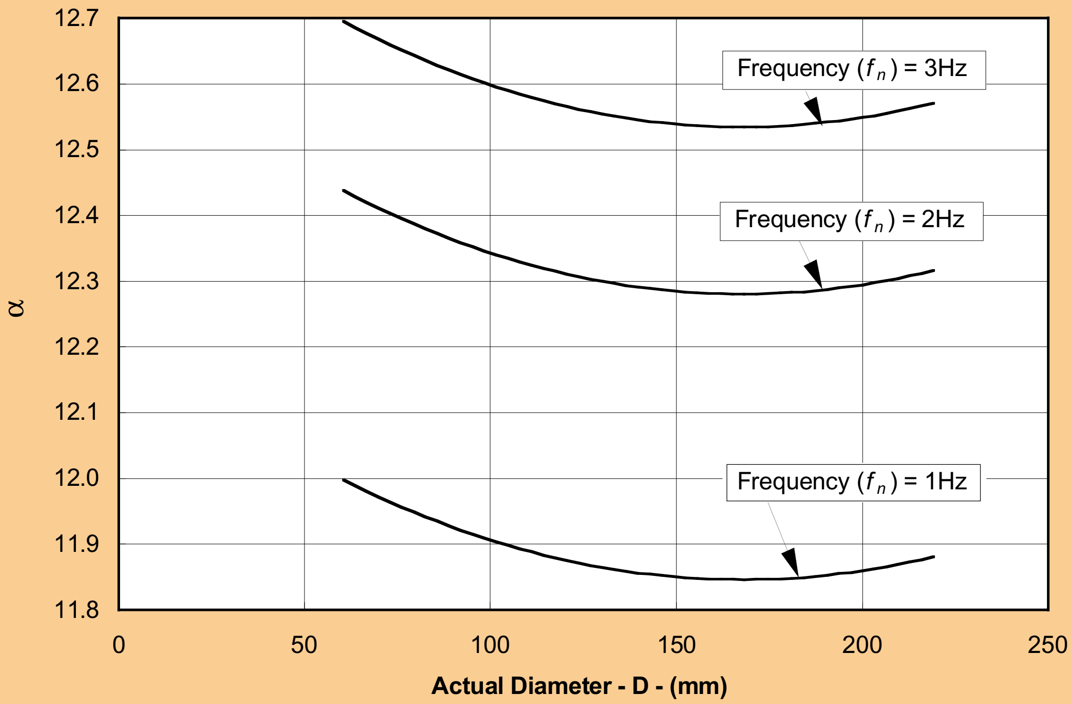

Figure A-11 *α curve fit - Flexible Support Arrangement - 60 mm<D<219 mm*

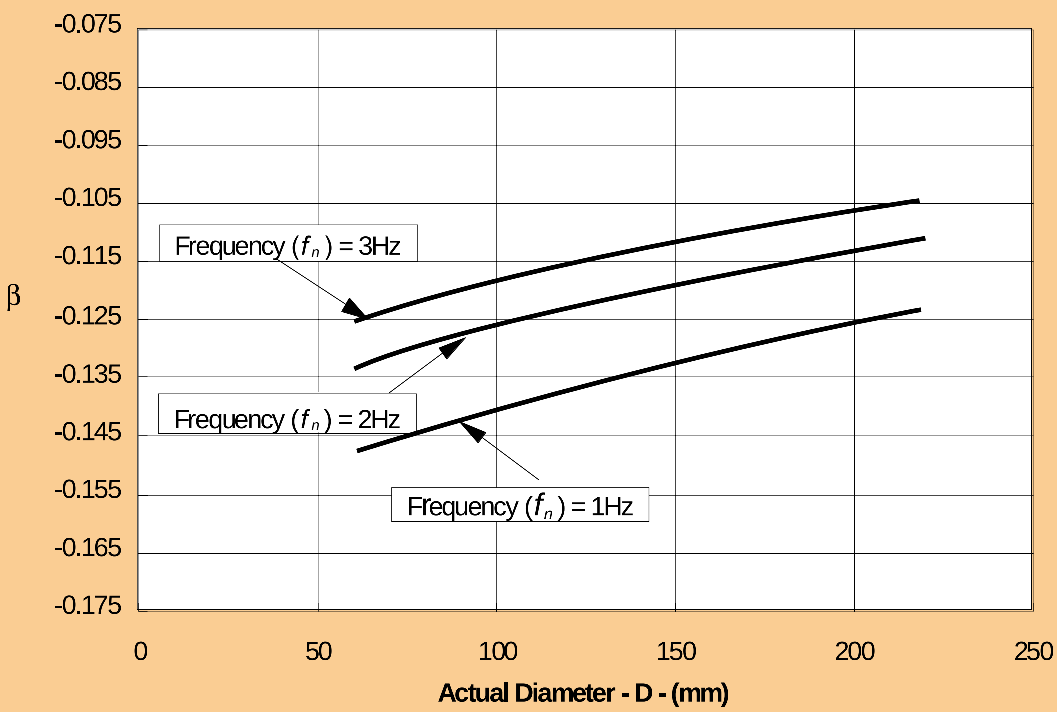

Figure A-12 *β curve fit - Flexible Support Arrangement - 60 mm<D<219 mm*

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Appendix A2.3

Screening Method for High Frequency Acoustic Excitation in Process Piping Systems

1.0 Overview

The method described in this Appendix allows the calculation of the likelihood of failure (LOF) value for connections downstream of a pressure reducing device when subjected to high frequency acoustic excitation. .

Typical dominant frequencies associated with high frequency acoustic fatigue are between 500 Hz and 2000 Hz. In view of the high frequency nature of this excitation mechanism, failure can be experienced in a short space of time.

As the pipe response is high frequency and involves local pipe wall flexure, the definition of a connection is any connection which causes a circumferential discontinuity in the pipe wall. Typical examples would include tees, branches and small bore connections.

The method provides an LOF rating for each connection on the main line downstream of the pressure reducing device.

If there is any uncertainty regarding the application of this method then specialist advice should be sought.

2.0 Method

The method for calculating the likelihood of failure (LOF) for this excitation mechanism is presented as a flowchart in figure A-13. The individual steps in this process are given below.

Step 1

The first step in the flowchart is to calculate the source sound power level (PWL) using the following equation:

$$\mathrm{PWL\ (source)} = 10\log_{10}\left[\left(\frac{p_1 - p_2}{p_1}\right)^{3.6} W^2 \left(\frac{t_1}{M}\right)^{1.2}\right] + 126.1 \qquad (12)$$

where:

PWL = sound power level (dB)

p_1 = pressure upstream of valve (kPa absolute)

p_2 = pressure downstream of valve (kPa absolute)

W = mass flow rate (kg/s)

t_1 = upstream temperature (deg K)

M = molecular weight of gas

Step 2

The second step in the flowchart is to calculate the sound power level in the main pipe at the branch [PWL (branch)] using the following equation:

$$\text{PWL (branch)} = \text{PWL (source)} - 60\frac{L}{D_{\text{int}}} \tag{13}$$

where:

L = distance downstream between source and branch (m)
D_{int} = internal diameter (mm) of main line

Step 3

If more than one valve is open then the total sound power level in the main pipe at the branch, PWL (branch, total), is calculated using the following equation:

$$\text{PWL (branch, total)} = 10\log_{10}\left[10^{\frac{PWL1(branch)}{10}} + 10^{\frac{PWL2(branch)}{10}} + ...\right] \tag{14}$$

where:
PWL1 (branch) = PWL in the main pipe of valve 1 at the branch
PWL2 (branch) = PWL in the main pipe of valve 2 at the branch

If a low noise trim is fitted then the PWL (branch, total) should be reduced in line with data supplied by the valve manufacturer. For example, if the low noise trim reduces the sound power level by 15dB, then this value should be subtracted from the calculated sound power level. When using this method, the source sound power level (PWL) supplied by the valve manufacturer must not be used.

Step 4

Enter Figure A-14 at the calculated D/T ratio of the main line at the calculated PWL (branch, total) to obtain an estimate of the fatigue life (i.e. number of cycles to failure N).

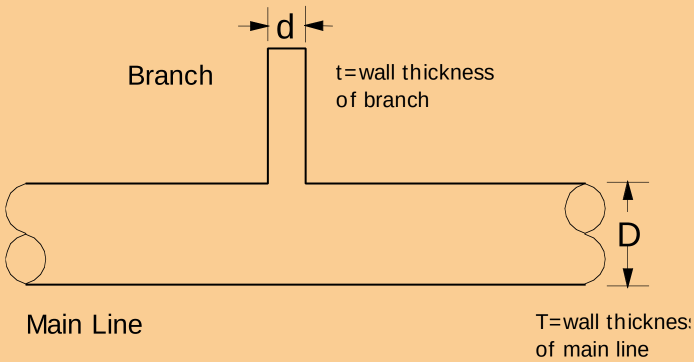
Alternatively, use the numerical representation of the graph to obtain N as follows:

$$\log_{10} N = 470711.5155 - 63075.1242(\log_{10}\text{B}) + \frac{183685.4368}{\sqrt{\text{B}}} - \frac{575094.3273}{\text{B}^{0.1}}$$

$$\text{B} = a(\text{PWL(branch)} - 0.112762(s) - 0.001812(s)^2 + 4.307277*10^{-5}(s)^3)$$

$$s = 91.9 - \frac{D}{T}$$

$$a = 3.28*10^{-7}\left(\frac{D}{T}\right)^3 - 8.503*10^{-5}\left(\frac{D}{T}\right)^2 + 7.063*10^{-3}\left(\frac{D}{T}\right) + 0.816$$



where:
D=external diameter of main line
T=wall thickness of main line
d=external diameter of branch
t=wall thickness of branch

Step 5

Enter Figure A-15 at the calculated D/d ratio to obtain the fatigue life multiplier (FLM), and use this to obtain the new estimate of N (i.e. $N_{new} = N_{old} \times FLM$).

Alternatively, use the numerical representation of the graph to obtain the FLM.

Step 6

If the connection is a Weldolet type fitting (including contour and short contour body fittings) then enter Figure A-16 at the calculated PWL (branch, total) value to obtain the fatigue life multiplier (FLM), and use this to obtain the new estimate of N (i.e. $N_{new} = N_{old} \times FLM$).

Alternatively, use the numerical representation of the graph to obtain the FLM.

Step 7

If the piping material is duplex or super duplex then enter Figure A-17 at the calculated PWL (branch, total) value to obtain the fatigue life multiplier (FLM), and use this to obtain the new estimate of N (i.e. $N_{new} = N_{old}$ x FLM).

Alternatively, use the numerical representation of the graph to obtain the FLM.

Step 8

Calculate the likelihood of failure as follows:

$$\text{L.O.F.} = -0.1303\ln(N) + 3.1 \tag{15}$$

$$\text{unless L.O.F.} < 0.0\text{, in which case take L.O.F.} = 0.0$$

$$\text{or L.O.F.} \geq 1.0\text{, in which case take L.O.F.} = 1.0$$

$$\text{where N} = \text{Number of cycles to failure.}$$

3.0 Worked Example

Single relief valve venting into a 6” line.

Main line	6” Schedule 10S D= 168.3mm (6.625”), T = 3.4mm (0.134”)
Material:	Duplex Stainless Steel
Connection:	1” NB weldolet, approximately 1m downstream of valve
Flow rate:	11.1 kg/s
Upstream pressure:	357 bar g = $(357 + 1)*10^2$ = 35800 KPa absolute
Downstream pressure:	5 bar g = $(5 + 1)*10^2$ = 600 KPa absolute
Upstream temperature:	111 deg F (317 K)
Gas molecular weight:	20

Step 1 Source PWL

Calculate the source PWL using Equation 12.
Source PWL = 161.1 dB

Step 2 Branch PWL

Calculate the branch PWL using Equation 13.
Branch PWL = 160.7 dB

Step 3 Total PWL

Calculate the total PWL at the branch using Equation 14.
(only one valve so PWL = 160.7dB)

Step 4 Initial Estimate of Fatigue Life

D/T = 49.5; Branch PWL = 160.7 dB
From Figure A-14 this gives N = 1×10^9

Step 5 Correction for D/d

In this case, D/d = 6. From Figure A-15:
Fatigue Life Multiplier = 0.9
Therefore, N = 0.9×10^9

Step 6 Correction for Weldolet Connection

As the connection is a Weldolet rather than a tee or swept tee, from Figure A-16:
PWL = 160.8 dB
Therefore, Fatigue Life Multiplier = 0.2
Therefore, N = 1.8×10^8

Step 7 Correction for Duplex Stainless Steel Alloy

As the material is duplex stainless steel, from Figure A-17:
PWL = 160.8 dB
From Figure A-11, Fatigue Life Multiplier = 0.17
Therefore, N = 3.1×10^7

Step 8 Calculate L.O.F.

L.O.F.=-0.1303ln(N)+3.1
L.O.F.=0.85

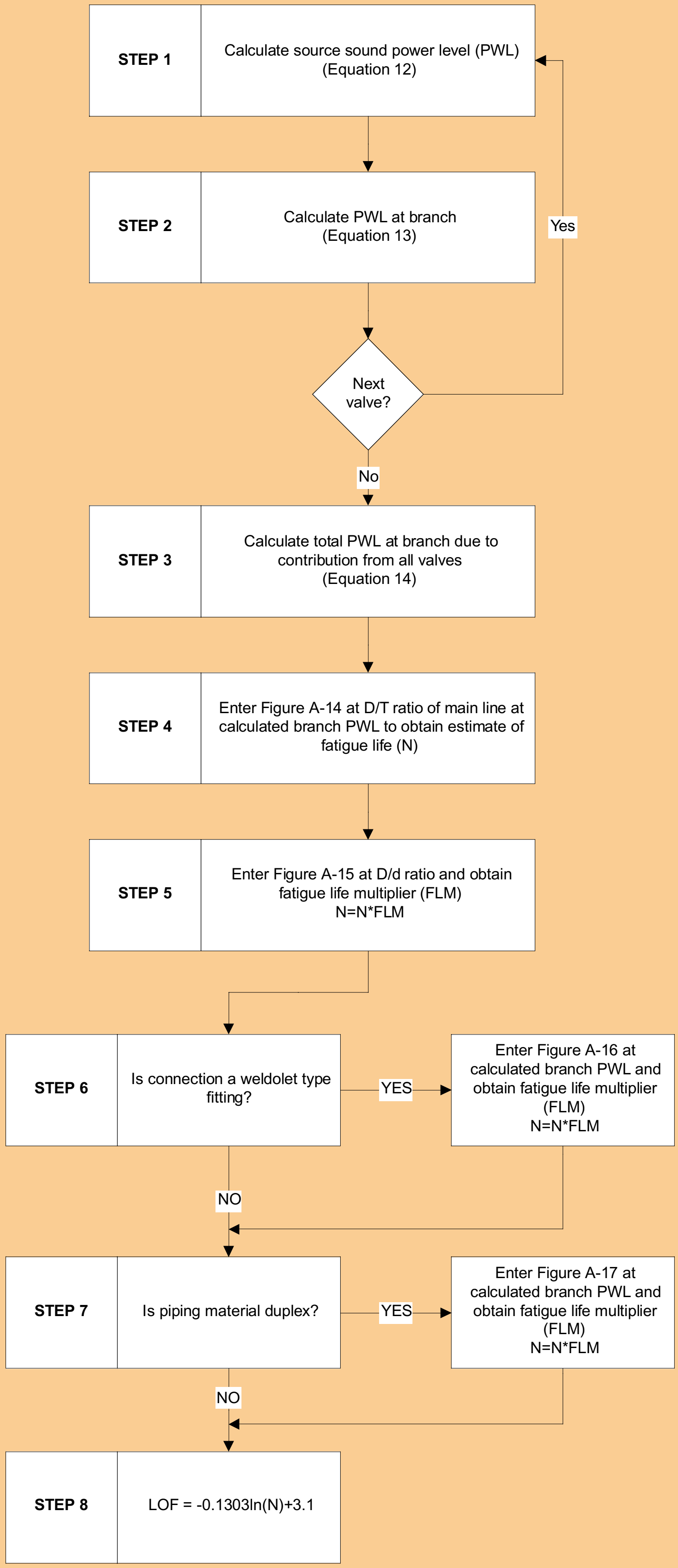


Figure A-13 *Flowchart for Screening High Frequency Acoustic Fatigue*

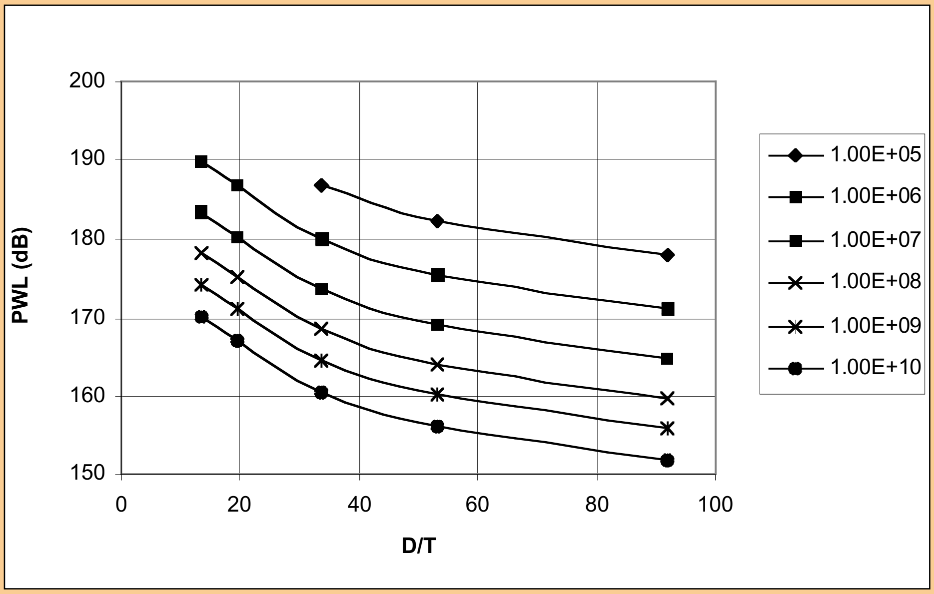


Figure A-14 *Number of Cycles to Failure*

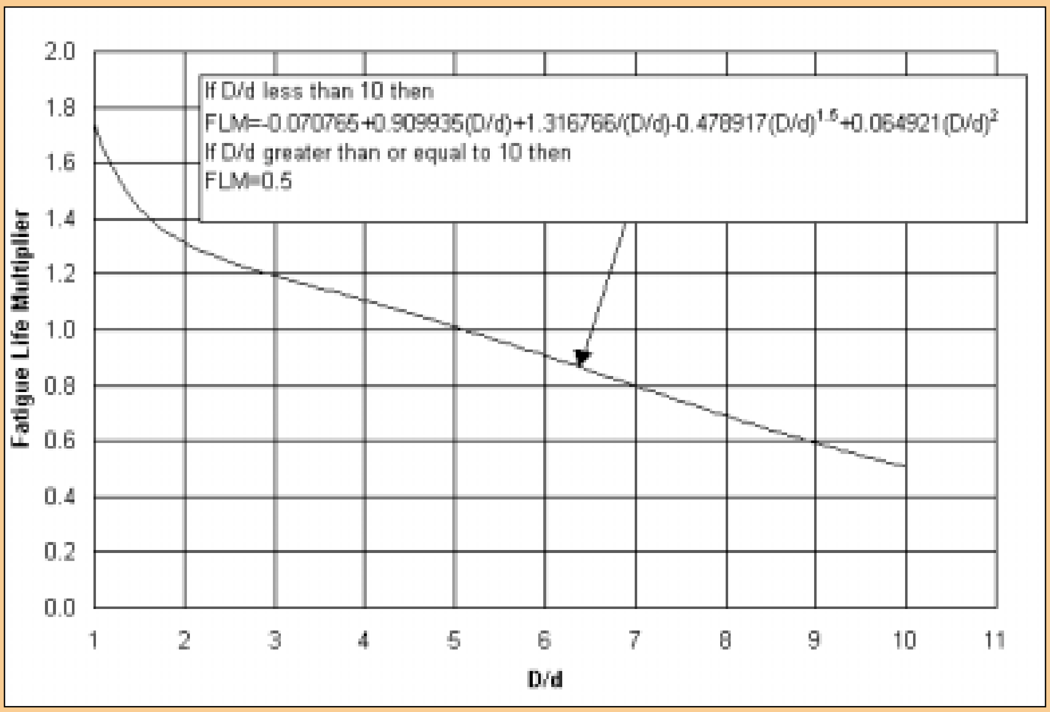
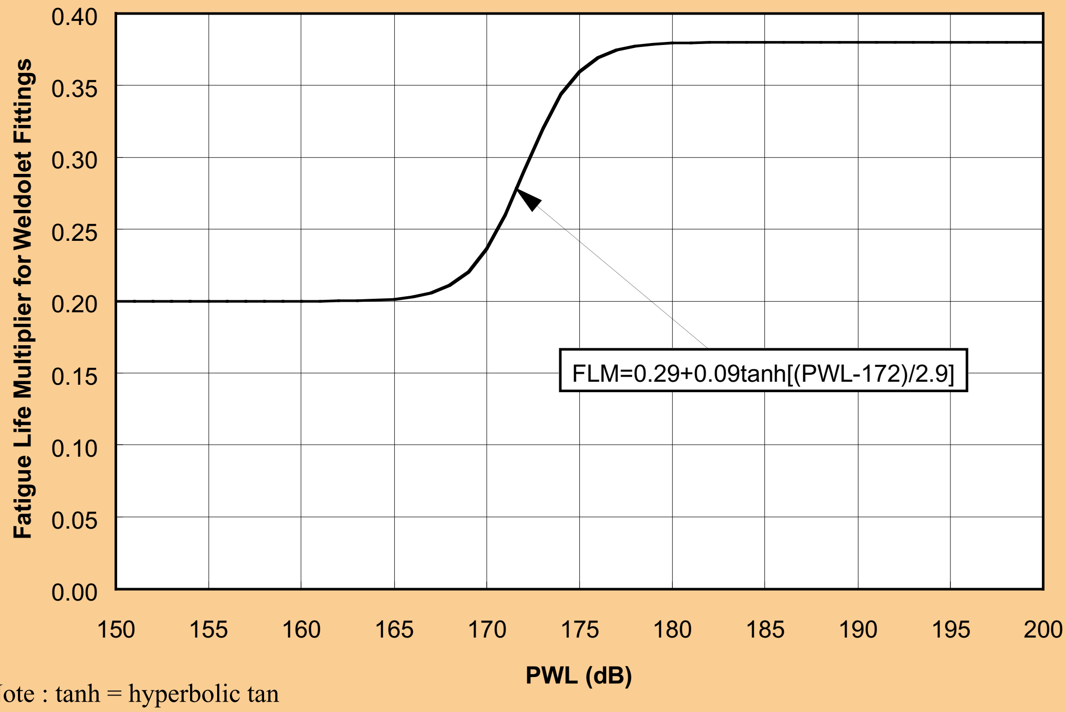
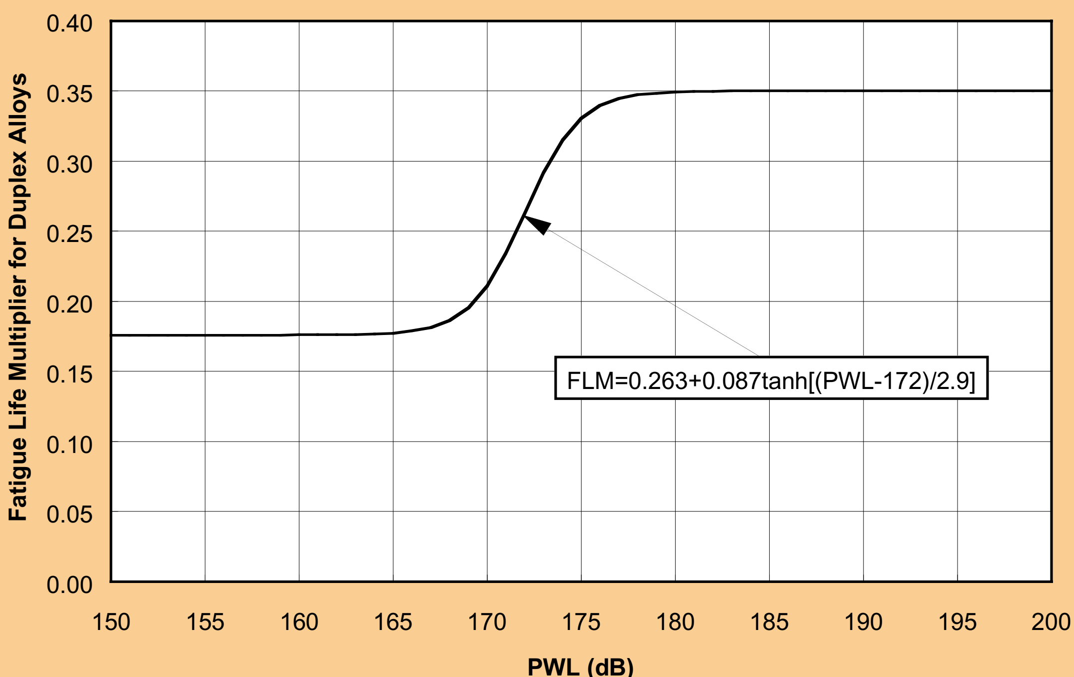


Figure A-15 *Fatigue Life Multiplier (FLM) for D/d Ratio*



Note : tanh = hyperbolic tan

Figure A-16 *Fatigue Life Multiplier (FLM) for Weldolet Fittings*



Note : tanh = hyperbolic tan

Figure A-17 *Fatigue Life Multiplier (FLM) for Duplex Stainless Steel Alloys*

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Appendix A3

Stage 3 - Detailed Screening of Small Bore Connections

1.0 Small Bore Connection Modifier

The calculation of the small bore connection modifier is categorised into two parts:

- Likelihood of failure in branch due to branch geometry
- Likelihood of failure due to main pipe geometry.

These are combined as shown in Figure A-18 to give the small bore connection modifier. The small bore connection modifier is the minimum of the likelihood of failure in branch due to branch geometry and the likelihood of failure due to main pipe geometry. The L.O.F. values shall be inserted in **Worksheet 3-2.**

2.0 Likelihood of Failure due to the Branch Geometry

The factors governing the likelihood of failure of the branch are:

- type of fitting;
- overall length of branch;
- number and size of valves;
- main pipe schedule;
- small bore pipe diameter.

The various factors are combined as shown in Figure A-18 to give an overall probability of failure in the small bore branch connection.

2.1 Type of Fitting

A weldolet involves two welds and hence (in comparison to a contoured body fitting or short contoured body fitting) has double the number of sites at welds for potential fatigue failures. Additionally contoured body fittings and short contoured body fitting have higher natural frequencies than weldolets.

Fitting	Likelihood of Failure (L.O.F.)
Weldolet	0.9
Contoured body fitting	0.6
Short contoured body fitting	0.4

2.2 Overall Length of Branch

The length also determines the natural frequency. Again a longer unsupported branch results in lower natural frequencies and hence greater likelihood of failure. Length is measured from the main pipe wall to the end of the branch assembly (including valve(s) if fitted).

Length	Likelihood of Failure (L.O.F.)
over 600mm	0.9
up to 600mm	0.7
up to 400mm	0.3
up to 200mm	0.1

2.3 Number and Size of Valves

This is the element of likelihood of failure associated with the unsupported mass. Higher mass results in lower natural frequencies and hence greater likelihood of failure.

Number of Valves	Likelihood of Failure (L.O.F.)
2 or more	0.9
1 or integral double block and bleed valve	0.5
0	0.2

2.4 Main Pipe Schedule

Thin walled main pipe is at higher likelihood of failure than the heavier schedules as its lower stiffness results in low natural frequencies and high levels of stress at the joint between the small bore branch and the main pipe.

Schedule	Likelihood of Failure (L.O.F.)
10S	0.9
20	0.8
40	0.7
80	0.5
160	0.3
>160	0.3

2.5 Small Bore Pipe Diameter

As the diameter of the small bore fitting increases the natural frequency will also increase and hence likelihood of failure will be reduced.

Fitting Diameter (Nominal Bore)		Likelihood of Failure (L.O.F.)
Inches	DN (mm)	
0.5	15	0.9
0.75	20	0.8
1	25	0.7
1.5	40	0.6
2	50	0.5

3.0 Likelihood of Failure due to Location on the Parent Pipe

The likelihood of failure of a connection due to the geometry of the main pipe is dependent on:

- pipe schedule;
- location of the connection on the main pipe.

3.1 Main Pipe Schedule

Thin walled main pipe has a higher likelihood of failure than the heavier schedules as its lower stiffness results in low natural frequencies and high levels of stress at the joint between the small bore branch and the main pipe.

Schedule	Likelihood of Failure (L.O.F.)
10S	0.9
20	0.8
40	0.7
80	0.5
160	0.3
>160	0.3

3.2 Location on Main Pipe

Small bore located at rigid supports for the main pipe is unlikely to vibrate as the support will force a node of vibration on the main pipe and as a result no forcing for the small bore branch. Conversely small bore branches located near bends, reducers or valves are more likely to experience high levels of excitation and therefore a higher likelihood of failure.

Location	Likelihood of Failure (L.O.F.)
Valve	0.9
Reducer	0.9
Bend	0.9
Mid span	0.7
Partially Fixed Support *	0.4
Fixed support**	0.1

* Braced in one direction : (1 translational degree of freedom perpendicular to the axis of the small bore is fixed and the remaining degrees of freedom are free)
** Braced in two directions : (two translational degrees of freedom perpendicular to the axis of the small bore are fixed (braced in two directions), please note this means no allowance for movement).

4.0 Small Bore Connection Modifier

The L.O.F. values are combined as shown in Figure A-18 to give the small bore connection modifier. The small bore connection modifier is the minimum of the likelihood of failure in the branch due to branch geometry and the likelihood of failure due to main pipe geometry.

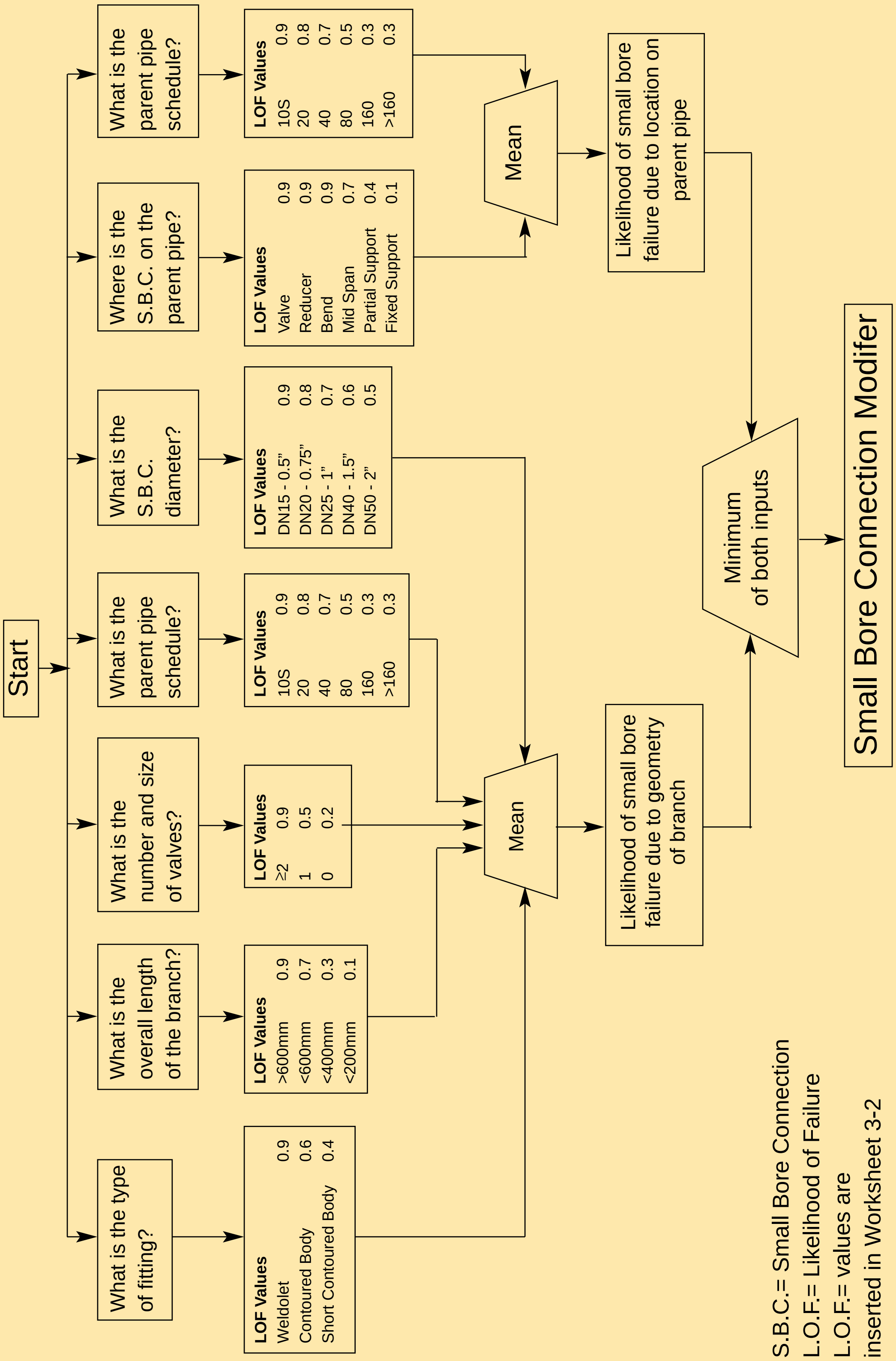

Figure A-18 *Stage 3 (Detailed Screening of Small Bore Connection) Flowchart*

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Appendix B

Worked Example

1.0 Introduction

A small bore connection is welded to a main run pipe coming out of a reciprocating compressor (discharge pressure 40 bar gauge). The reciprocating compressor is in an onshore plant and has been designed according to API-618 Design Approach 2. The main pipe is a 14 inch schedule 10S pipe transporting gas of a density equal to 39.5 kg/m^3 and a velocity of 18.68 m/s. The weldolet small bore connection is located mid-span with no valves, has a diameter of DN40 and is schedule XS, a length of 650 mm and is located 1 times the radius of the skid from the reciprocating compressor. The length of the pipe is approximately 5 m between major supports.

The items that are required to be completed in each stage are shown in red in the worksheets in Appendix B.

2.0 Stage 1 - Identification of Excitation Mechanisms

The system is identified as a gas system.

2.1 Gas systems

Address the issues in Questionnaire 2 (Appendix A1) for gas systems.

<u>Q2.1 Flow induced turbulence</u>

Is the system operating under steady state conditions? The following are examples of situations that are not considered as a flow induced vibration problem: relief and blowdown events and transients caused by valve closures.

yes - flow induced turbulence is a potential problem

<u>Q2.2 High Frequency Acoustic Excitation</u>

Does the system have choked flow?

no - no problem

<u>Q2.3 Mechanical Excitation</u>

Is there reciprocating or rotating equipment in the system?

yes - mechanical excitation is a potential problem

Is the system close to a reciprocating compressor/pump on another system or any other large vibration source such as a diesel pump? (The term “close” is not definitive but the following is a rule of thumb based on engineering experience. For offshore plants close is defined as being supported from the same module/deck (above or below). For onshore plants close is defined as a radius equal to twice the maximum length of the skid.)

yes - mechanical excitation is a potential problem

Q2.4 Pulsation (Reciprocating machinery)

Is there reciprocating compressor in the system?

yes - pulsation (reciprocating machinery) is a potential problem

Q2.5 Pulsation (Rotating stall)

Does the system have a centrifugal compressor?

no - no problem

Q2.6 Pulsation (Periodic Flow Induced Excitation)

Does the system have any dead leg branches longer than 1 m?

no - no problem

Figure B-1 shows how Worksheet 3-1 would look like after Stage 1 for this example

3.0 Stage 2 - Detailed Screening of Main Pipe

From Stage 1 the excitation mechanisms that are of concern are

- flow induced turbulence;
- mechanical excitation;
- pulsation (reciprocating compressor).

3.1 Flow Induced Turbulence

Using Appendix A2.1 the likelihood of failure due to flow induced turbulence is calculated as follows:

Input

Main line:	14” Schedule 10S D= 355.6mm (14.0”), T = 4.78mm (0.188”), L=5m
Flow rate:	18.68 m/s
Fluid:	gas - density of 39.6 kg/m^3
Support arrangement:	stiff

Step 1:

ρ=39.6 kg/m^3
v=18.68 m/s

$$\rho v^2 = \left(39.6 \, {}^{kg}\!/\!_{m^3}\right)\left(18.68 \, {}^{m}\!/\!_{s}\right)^2 = 13818 \, {}^{kg}\!/\!_{ms^2}$$

Step 2:

D/T=74.4

Step 3:

Stiff Support Arrangment :

$$\text{L.O.F.} = \frac{\rho v^2}{F_V} = \frac{\rho v^2}{\left(D/T\right)^{\beta}}$$

$$= 446187.1 + 645.51D + 9.166*10^{-4}D^3 = 716946$$

$$= 0.1\ln(D) - 1.3739 = -0.7865$$

$$\text{L.O.F.} = \frac{\rho v^2}{F_V} = \frac{\rho v^2}{\alpha\left(D/T\right)^{\beta}} = \frac{\left(39.6\,{}^{kg}/_{m^3}\right)\left(18.68\,{}^{m}/_{s}\right)^2}{\left(716946\right)\left(74.4\right)^{-0.7865}}$$

$$\text{L.O.F.} = 0.57$$

3.2 Mechanical Excitation

Reciprocating compressor - L.O.F. = 0.9

3.3 Pulsation (Reciprocating Compressor)

Reciprocating compressor with a discharge pressure greater than 35 bar and an API 618 analysis has been conducted - L.O.F. = 0.4

Figure B-2 shows how Worksheet 3-1 would look like after Stage 2 for this example.

3.4 Recommended Actions

The likelihood of failure for the main pipe is 0.9.

Where possible and feasible, the main pipe should be redesigned, re-supported or a detailed analysis should be conducted as per Section 4.2 (Design solutions for main pipe).

The reason for the high L.O.F. is mechanical excitation due to a reciprocating compressor. Since the reciprocating compressor has been designed to API 618 Design Approach 2, the L.O.F. relating to pulsation will be tolerable (0.4). However, there is potential for mechanical excitation of the small bore connection, hence the high L.O.F. Small bore connections must be assessed according to Stage 3.

4.0 Stage 3 - Detailed Screening of Small Bore Connections

The following are the details of the small bore connection on the main pipe with the likelihood of failure results from Appendix A3.

Factor	Value	Known/Assumed	Likelihood of Failure
Type of Fitting	Weldolet	Known	0.9
Length of Branch (mm)	650	Known	0.9
Number of Valves	0	Known	0.2
Main Pipe Schedule	10s	Known	0.9
Small Bore Diameter (mm)	DN40	Known	0.6
Location on Main Pipe	Mid Span	Known	0.7

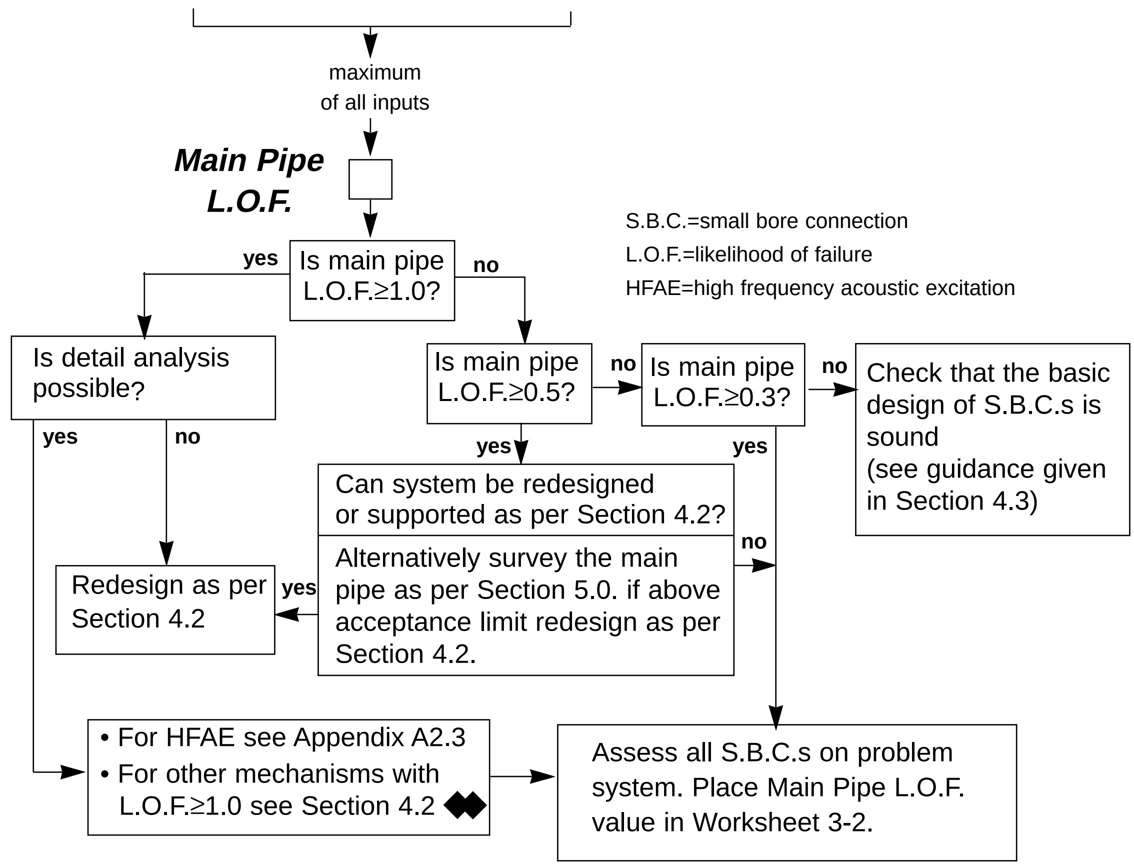
Recommended Actions

The likelihood of failure is 0.7 as determined from Figure B-3. The small bore connection must be redesigned in line with the design solutions itemised in Section 4.3 (Design Solutions for Small Bore Connections) or supported as per Appendix C.

Figure B-3 shows how Worksheet 3-2 would look like after Stage 3 for this example.

Stage 1 and 2 - Main Pipe L.O.F. (Figure 3-3)

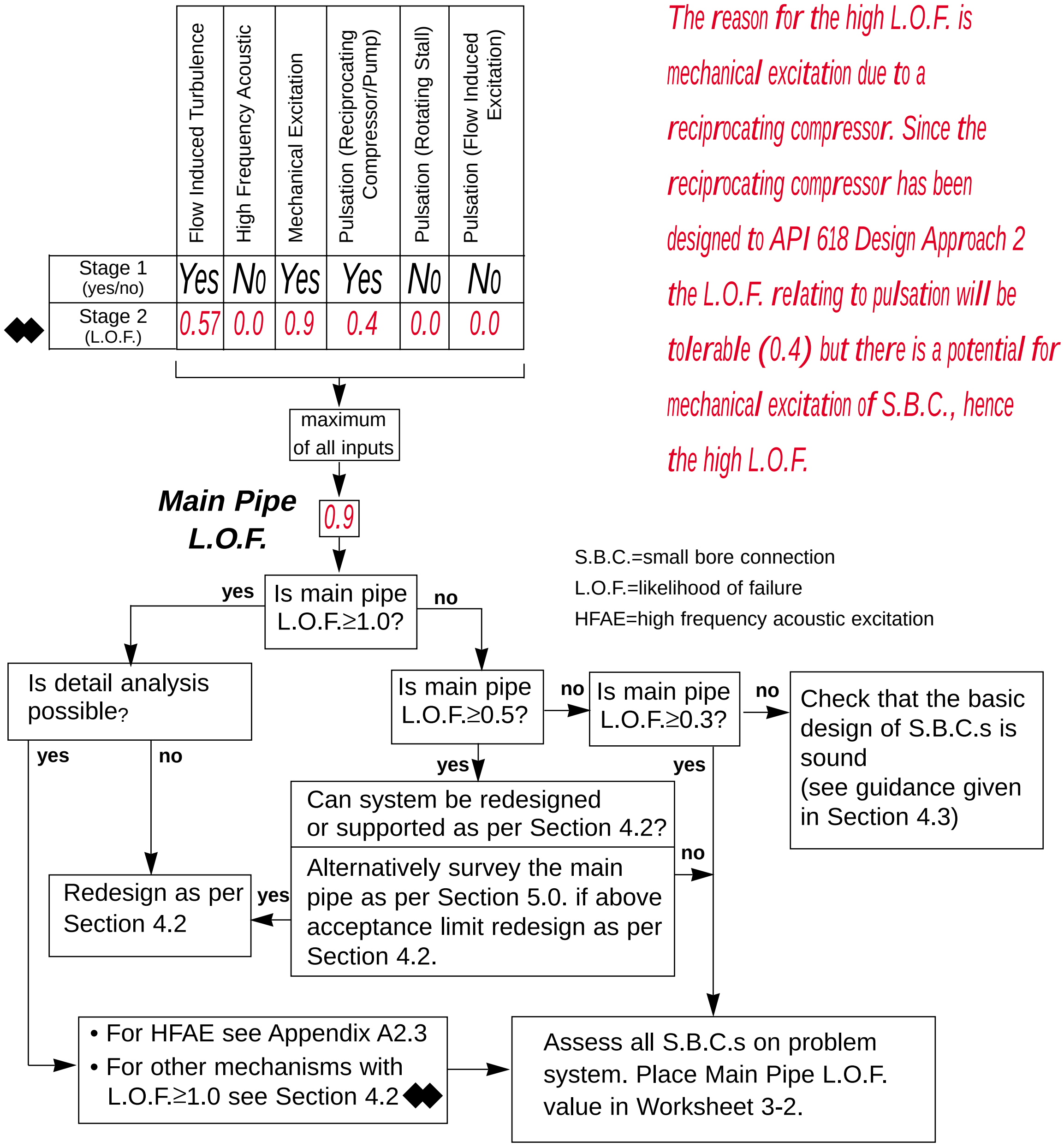
	Flow Induced	High Frequency Acoustic	Mechanical Excitation	Pulsation (Reciprocating Compressor/Pump)	Pulsation (Rotating Stall)	Pulsation (Flow Induced Excitation)
Stage 1 (yes/no)	Yes	No	Yes	Yes	No	No
◆◆ Stage 2 (L.O.F.)						



Project/Plant: Alpha		Line number:
System: Gas Compression		Main Pipe L.O.F.:
Subsystem: Discharge		Actions:
Assessed by:	Date	
Ref.No.		

Figure B-1 *Worksheet 3-1 example after completing Stage 1*

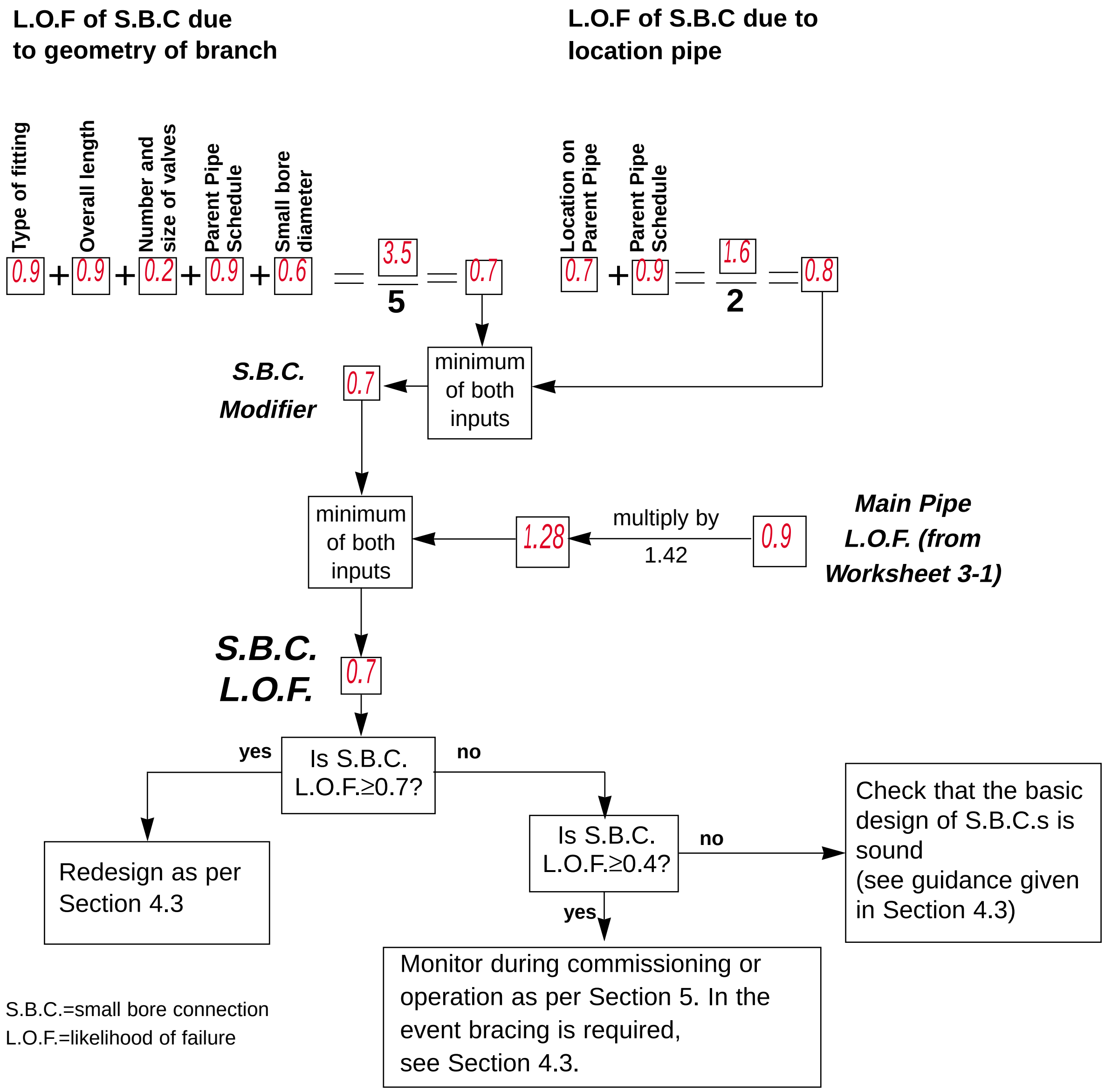
Stage 1 and 2 - Main Pipe L.O.F. (Figure 3-3)



Project/Plant:	*Alpha*	Line number:	
System:	*Gas Compression*	Main Pipe L.O.F.:	*0.9*
Subsystem:	*Discharge*	Actions:	*1) Mechanical Excitation (see above)*
Assessed by:	Date		*2) Flow Induced Turbulence*
Ref.No.			*MAY BE OF CONCERN*

Figure B-2 *Worksheet 3-1 example after completing Stage 2*

Stage 3 - Small Bore Connection L.O.F. (Figure 3-4)



Project/Plant: Alpha		Line number:
System: Gas Compression		Main Pipe L.O.F.:
Subsystem: Discharge		S.B.C. L.O.F: 0.7
Assessed by:	Date	Actions: Redesign as per Section 4.3
Ref.No.		

Figure B-3 Worksheet 3-2 example after completing Stage 3

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Appendix C

Recommended Small Bore Connection Supports

This section gives general guidance on the design of small bore connections. While typical examples of well designed small bore supports are presented as Figures C-1 to C-10, this is not intended to be a complete list of possible arrangements.

In general small bore pipework should be designed bearing in mind the following rules:

- the fitting and overall unsupported length should be as short as possible;
- the mass of valves and instrumentation should be minimised;
- any mass at the free end of a cantilever should be supported in both directions perpendicular to the axis of the small bore;
- supports should be from the main pipe, to ensure that the small bore connection moves with the main pipe during startup, thermal transients and vibration;
- any fastenings used must be designed to be effective under vibration;
- particular care must be taken when adopting small bore supports that are welded to the connection and its main pipe, as these welds provide additional potential sites for fatigue failure; dressing of welds by grinding will help.
- Clamp type supports rely on bolted arrangements. Periodic inspection will be required to ensure that no loosening occurs during years of operation.
- where possible remove any valves that are not required for plant operation (e.g. only required for hydrotesting or cleaning of lines) and replace with a blank flange.

Figures C-1 to C-4 are drawings for the clamp type of small bore support, and Figures C-5 to C-9 give examples of welded supports.

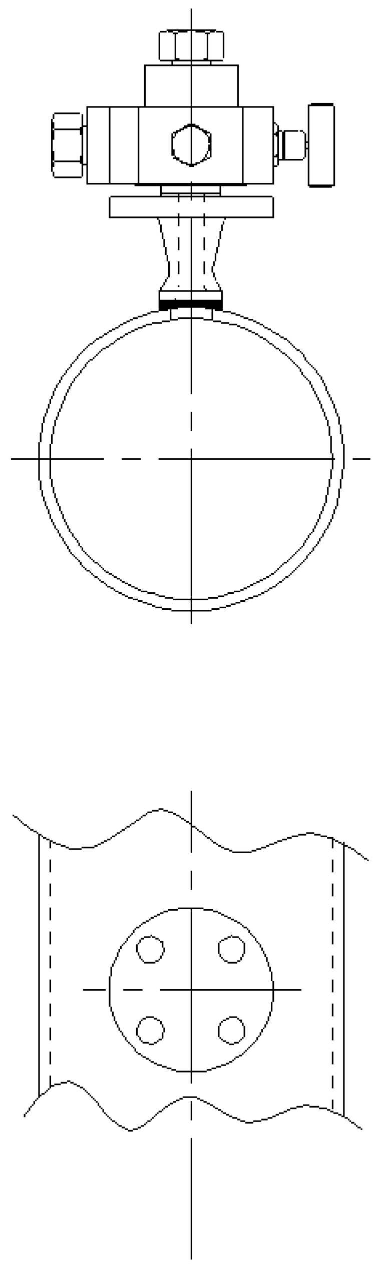


Figure C-1 *Preferred Small-bore Arrangement*

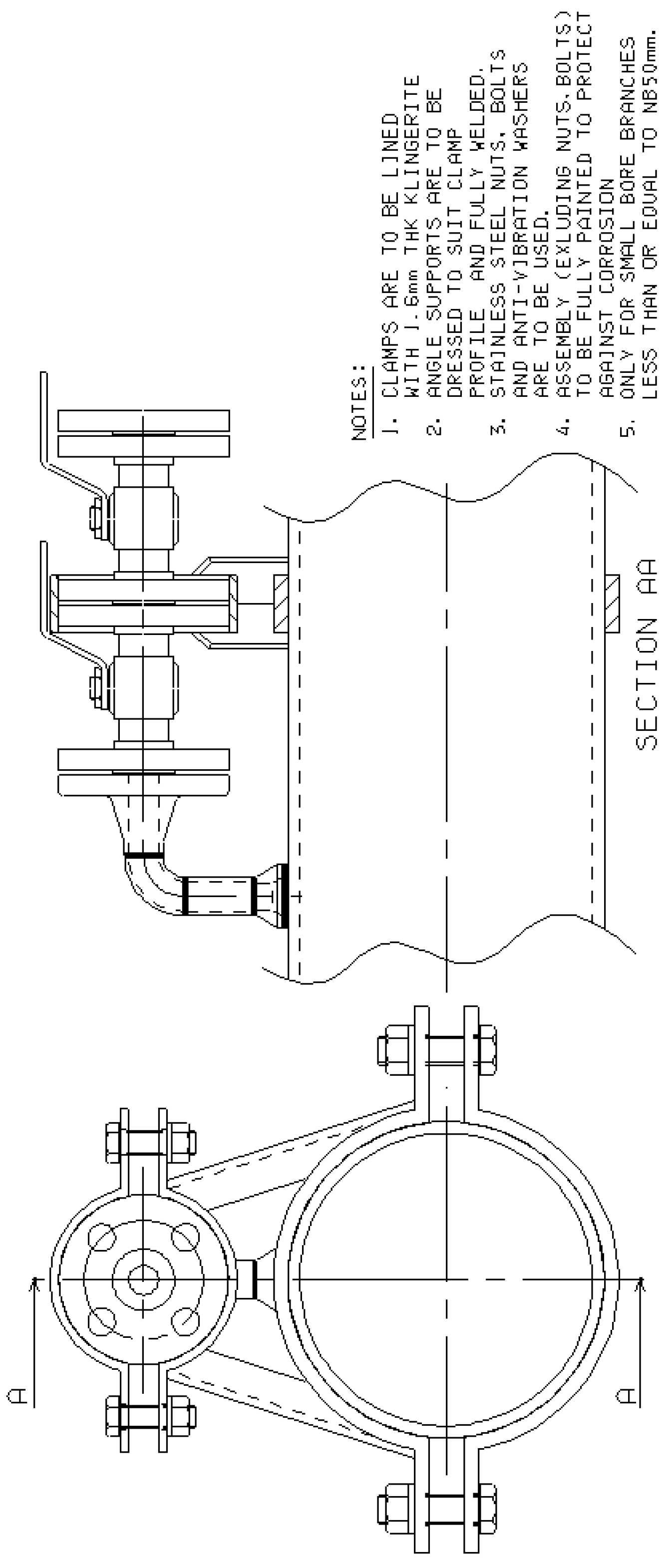


Figure C-2 *Clamp Type of Support*

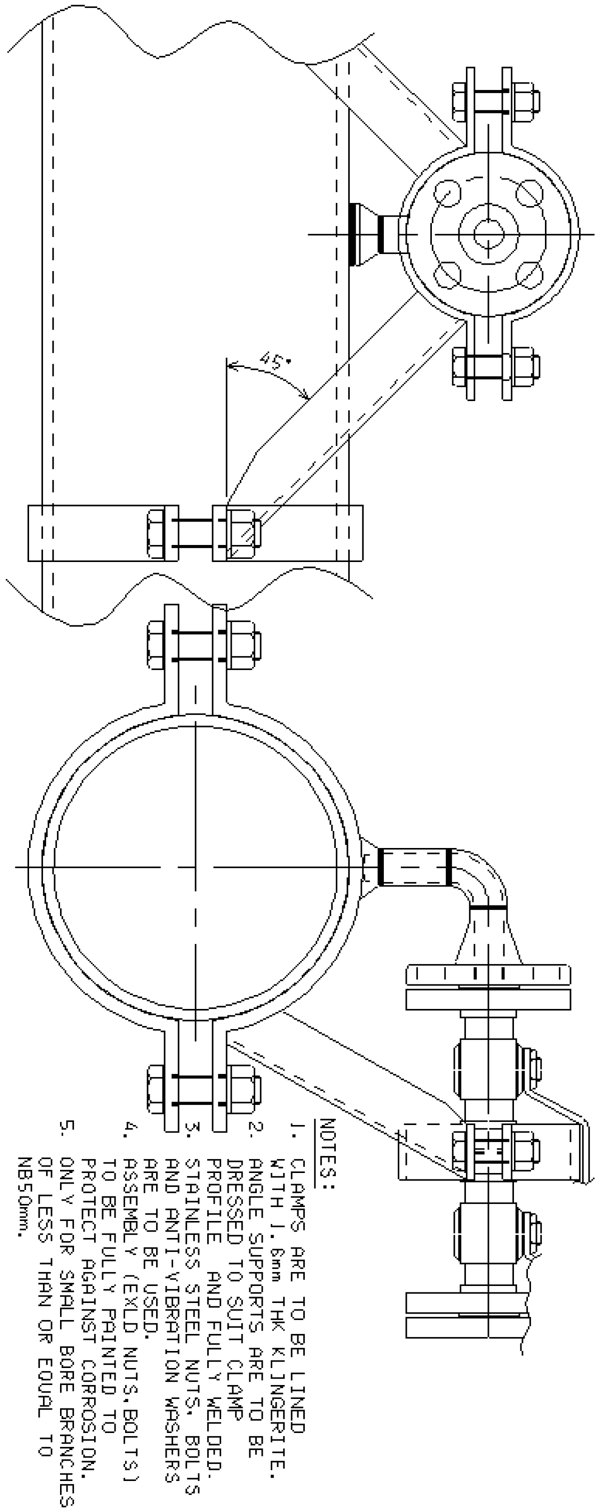


Figure C-3 *Clamp Type of Support*

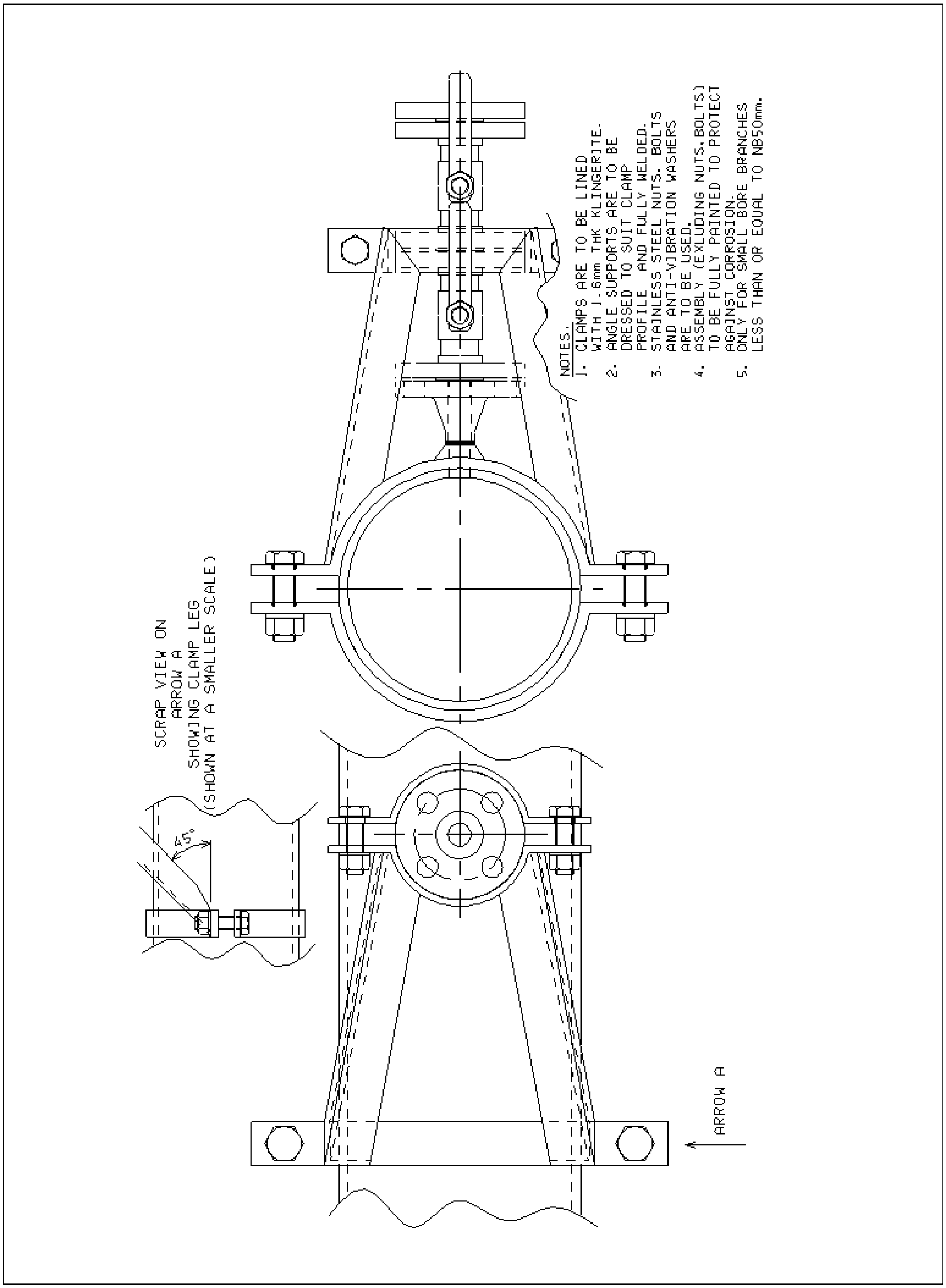


Figure C-4 *Clamp Type of Support*

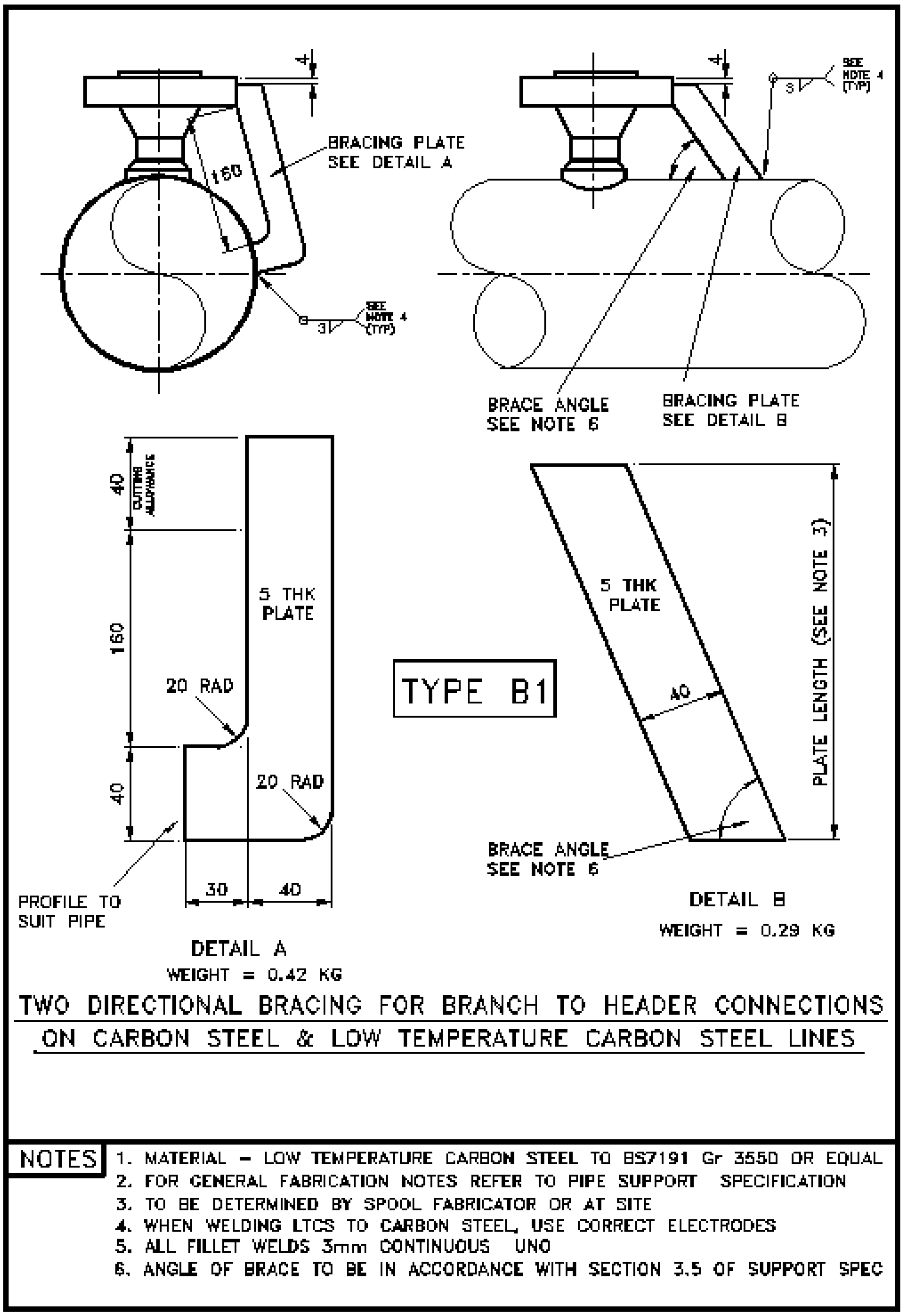


Figure C-5 *Welded Support*

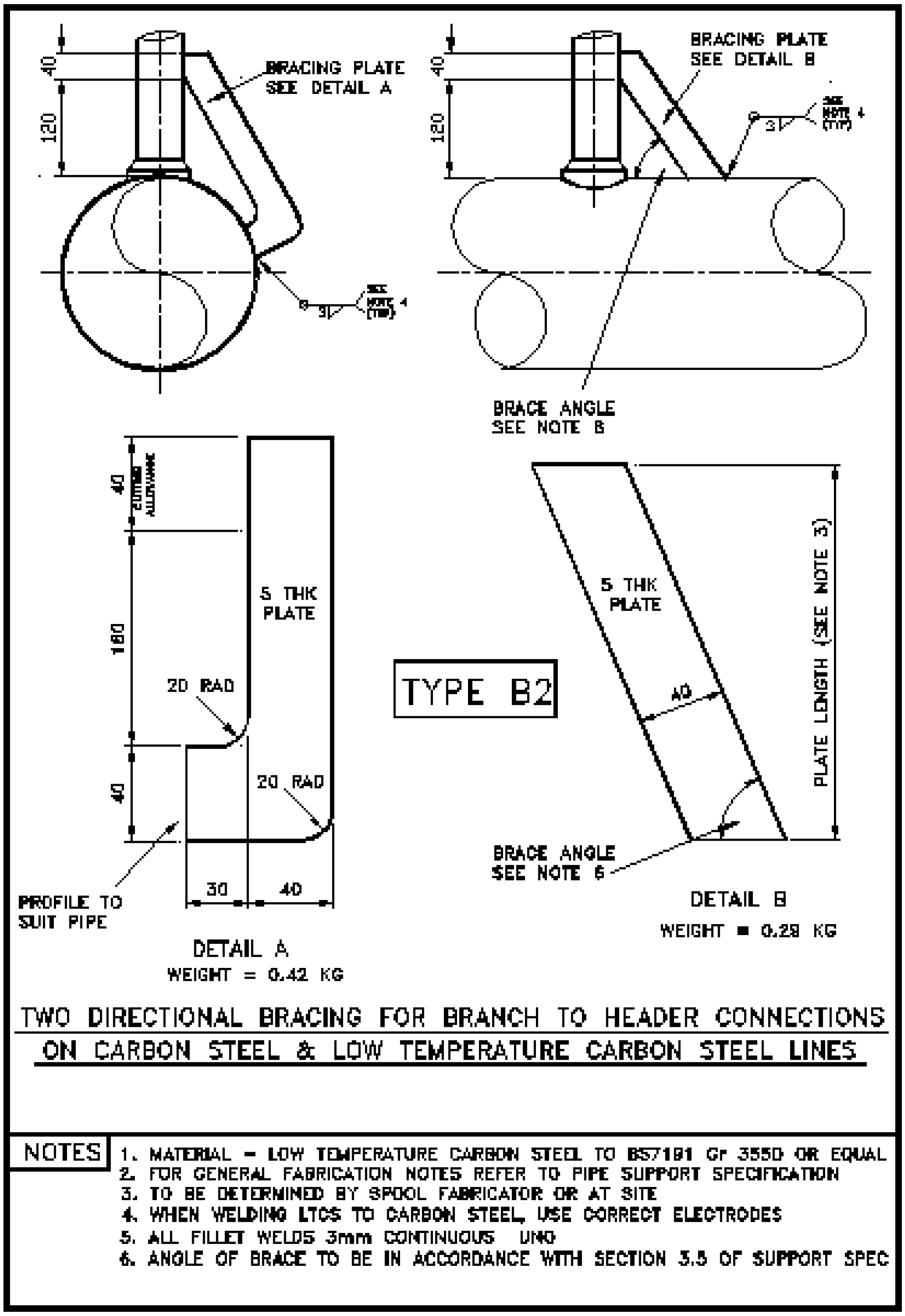


Figure C-6 *Welded Support*

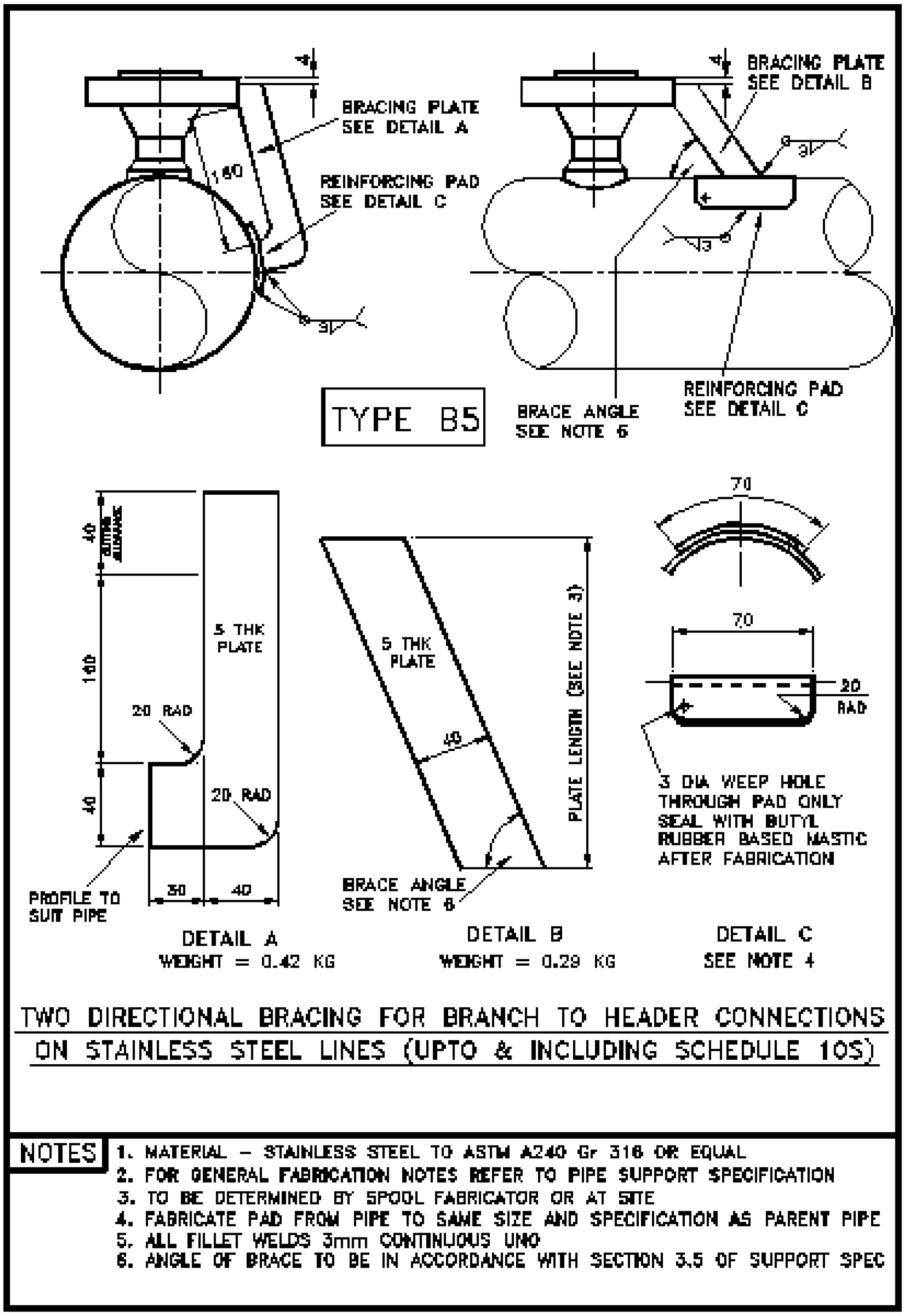


Figure C-7 *Welded Support*

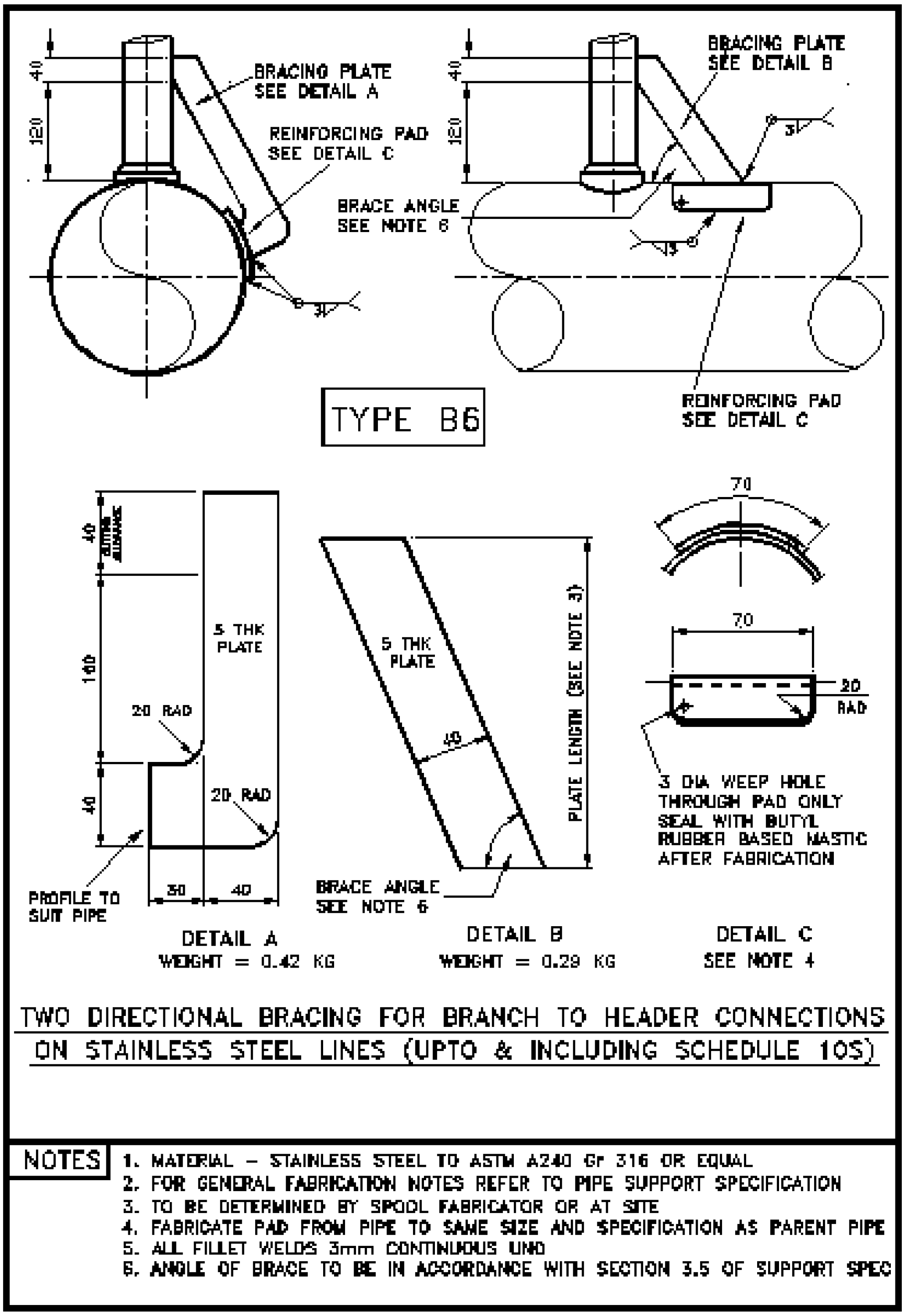


Figure C-8 *Welded Support*

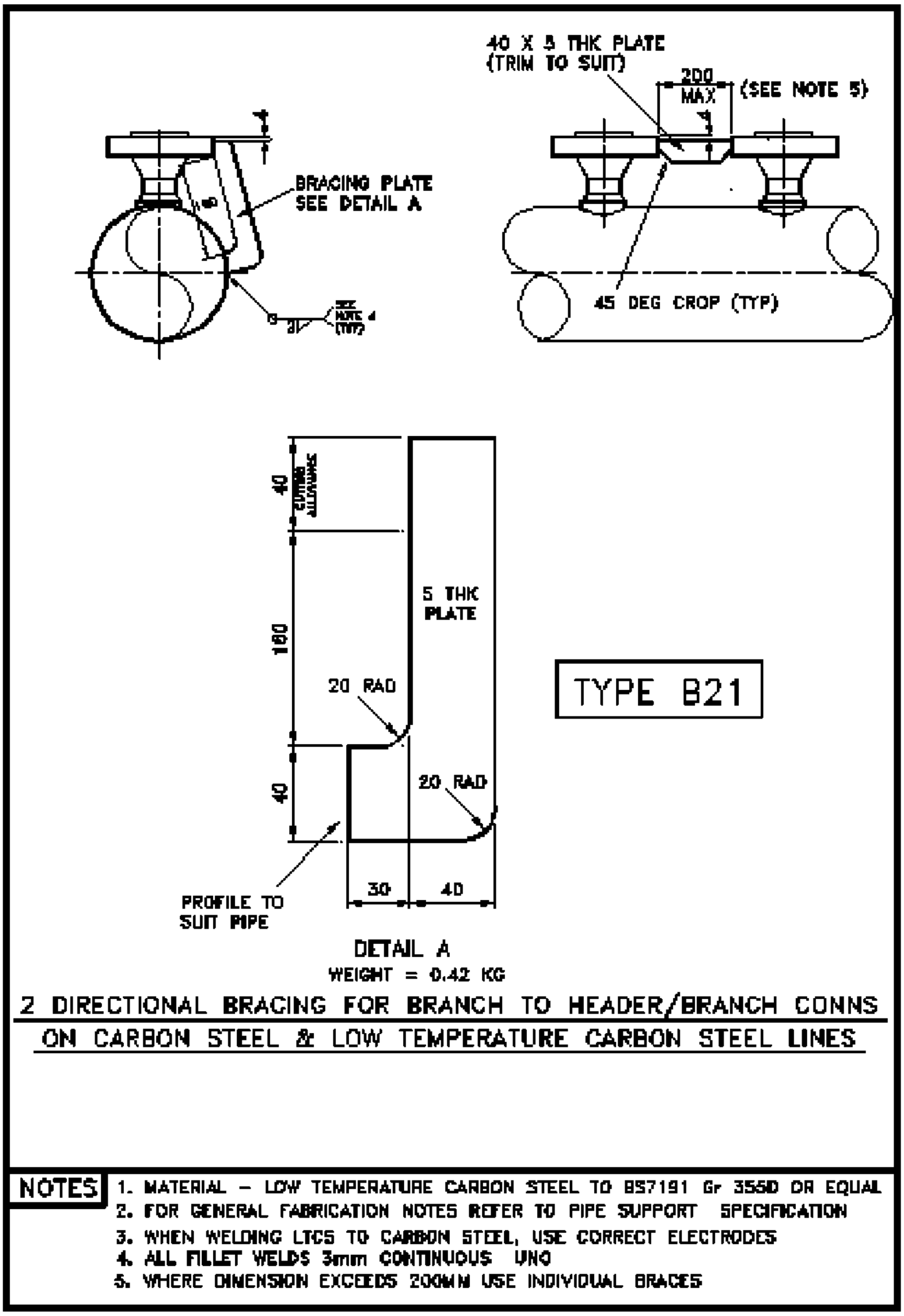


Figure C-9 *Welded Support*

Appendix D

Pipework Vibration Survey Method

1.0 Aim

To provide a simple means of assessing the vibration of small bore connections in main process pipework, using a single channel vibration data logger.

In addition, guidance is provided on assessing the vibration of the main pipe using previously published data [4].

2.0 Background

At present, two main approaches are employed to determine the risk of vibration-induced process pipework fatigue failure. These are as follows.

1. Use of vibration nomograms – the advantage of this scheme is the relative ease of obtaining vibration measurements, while the main disadvantage is that an estimate of the fatigue life cannot be derived directly from the measured data.

2. Direct dynamic strain measurements using either permanent or portable strain gauges – the advantage of this system is that a fatigue life can be determined, but with the disadvantages that specialist equipment is required.

Small bore connections are generally the primary source of piping system failures. The objective of this work therefore was to determine a simple assessment method based on the direct measurement of vibration levels of the small bore connection.

Note: Where dynamic strain/stress measurements are required this is outside the scope of these Guidelines and advice should be sought.

3.0 Method

The method is applicable to the following range of fittings.

1. Fitting type – contoured body, short contoured body and weldolet.

2. Fitting diameter – $^1/_2$" to 2".

3. Fitting schedule – 40 to 160.

4. Main Pipe diameter – 4" to 16".

5. Main Pipe schedule – 5S to 160.

6. Fitting termination – blank flange, single valve, double valve and multiple valve double block and bleed. (Note: single valve also covers an instrument type integral double block and bleed valve).

7. Flange rating – ANSI/ASME pressure class 150 to 2500.

4.0 Results and Analysis

A series of linear relationships has been determined between the ratio of the *maximum dynamic stress in the vicinity of the welded connection to the fitting end vibration velocity*, for each of the fitting geometric variables listed above. Employing these relationships and the infinite (non-propagating) fatigue life stress range for an F2 class weld ([15]), “Acceptance Limit” vibration velocity levels were determined as a function of frequency. In practice, ten different assessment criteria were determined to cover the various fitting types and terminations, and these are shown graphically in Figures D-1 to D-10.

The assessment criterion for vibration measurements obtained on the main pipe is shown in Figure D-11.

5.0 Assessment Technique

The assessment method consists of the following steps.

1. Using an appropriately configured vibration data logger, (Section 7.0), operating personnel would record vibration velocity spectra at the end of the small bore connection, in the two side-to-side directions.

   Figures D-1 to D-10 show the required vibration measurement locations and directions for each type of fitting.

2. The vibration velocity (mm/s rms) and its frequency (Hz) would then be entered onto the appropriate assessment curve to determine the vibration rating.

3. Where measurements can only be obtained on the main pipe, the vibration criterion shown in Figure D-11 should be used. The position of the transducer should be at the location exhibiting the highest level of vibration; the maximum vibration level obtained from measurements in the two axes perpendicular to the axis of the pipe should be used.

6.0 Interpretation

A vibration level in excess of the “Acceptance limit” in Figures D-1 to D-10 means that there is the potential for crack propagation, and relates to a dynamic stress range greater than 17.5 MN/m^2 peak to peak. If exceeded there is the possibility of fatigue failure and vibration control measures should be considered. In addition, direct dynamic strain measurement should be undertaken to accurately determine the likelihood of failure.

For the main line (Figure D-11), vibration levels in excess of the “Acceptance limit” indicate that there is the potential for fatigue failure of the main line itself. Dynamic stress measurements should be performed to accurately determine the likelihood of failure.

7.0 FFT Analyser/Data Logger Setup

The following list describes a typical analyser setup:

1. The FFT analyser/data logger should be set up to measure the root mean square (rms) vibration velocity amplitude in mm/s.

2. Set frequency range to 0 to 500 Hz.

3. Set resolution (i.e. number of spectral lines) to greater than 500:- typically 800 or 1600 (this will ensure a frequency resolution of better than 1 Hz).

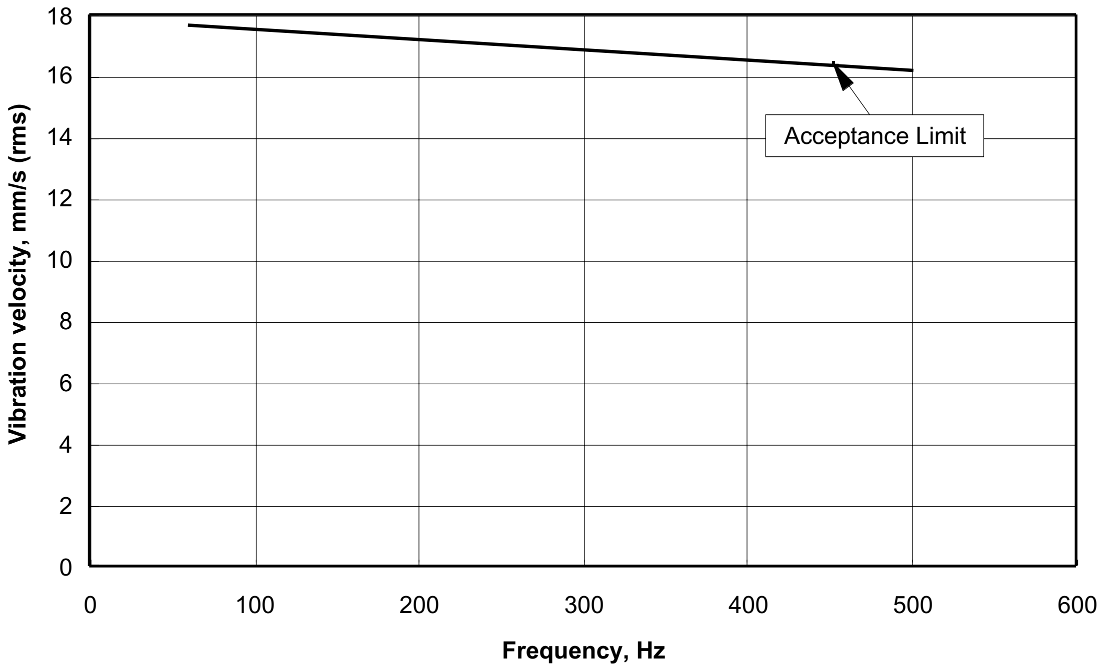
4. Use a “Hanning” window (a typical function on a datalogger).

5. Use at least 10 frequency averages.

6. Use a root mean square (rms).

7. If an acceleration transducer is employed, integrate the signal to velocity in the analyser. Alternatively a velocity transducer may be used. A displacement proximity transducer is not applicable.

Mount the accelerometer using a magnet, or for non-magnetic fittings consider adhesive or stud mounting (this maybe useful if a regular monitoring programme is to be established). Hand-held transducers should be avoided whenever possible, and should only be used as a last resort.



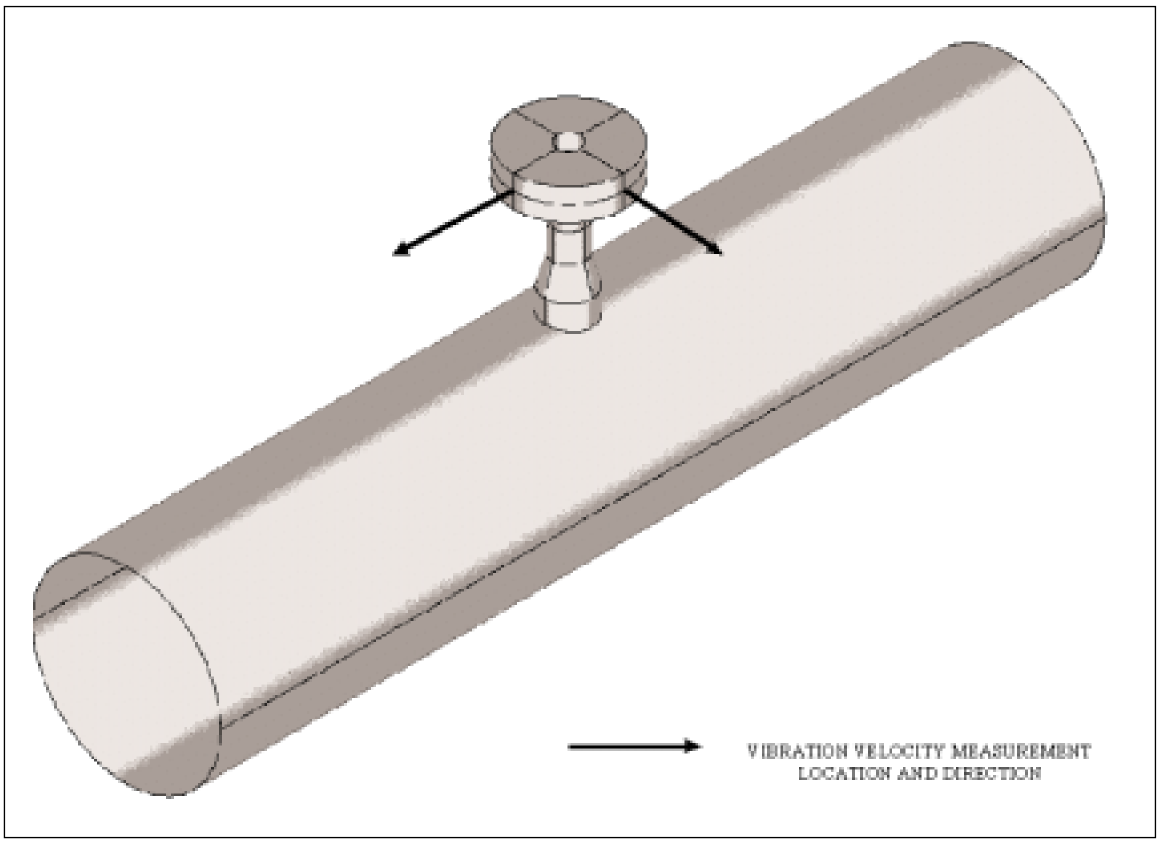
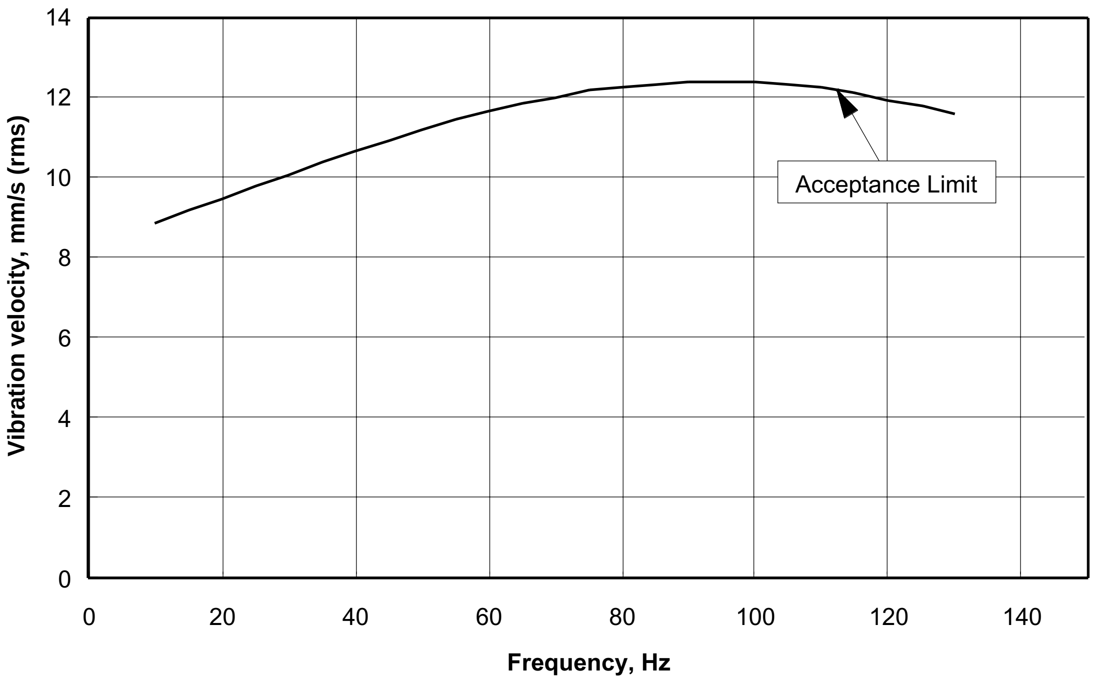


Figure D-1 *Contoured Fitting with ANSI/ASME B16.5 Flange*



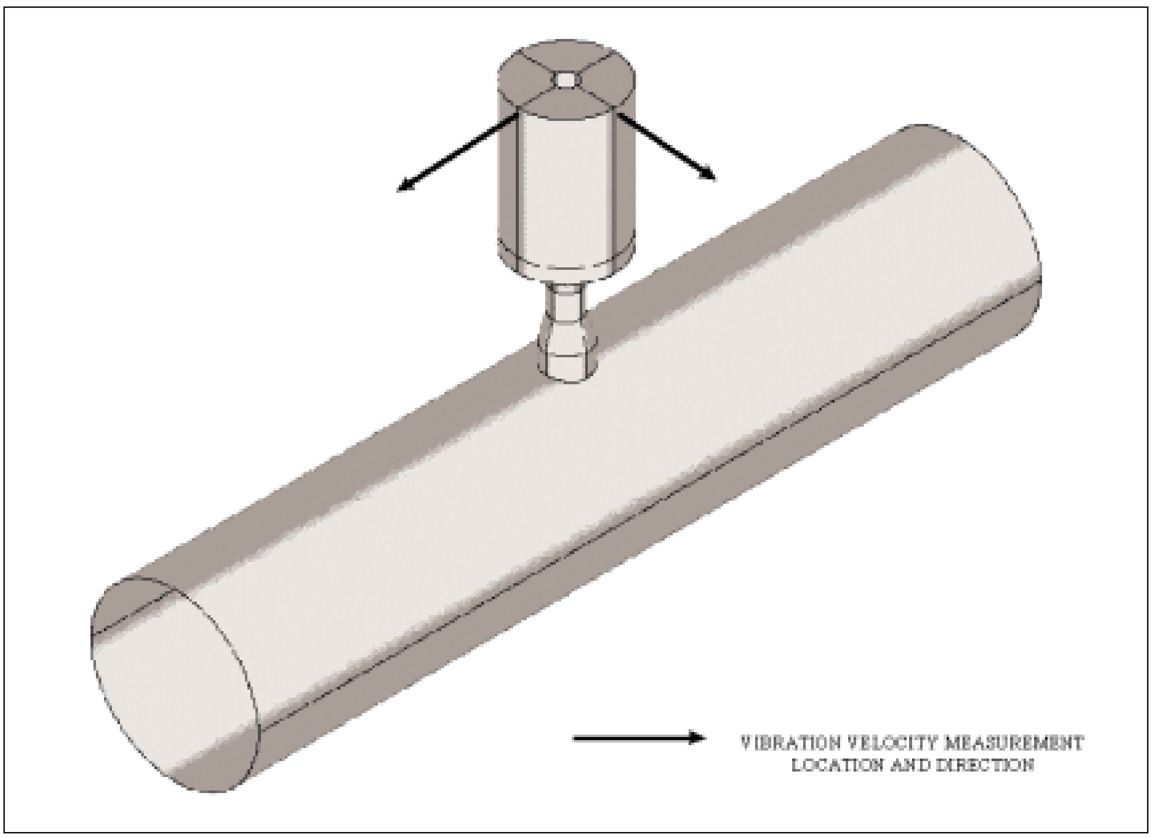
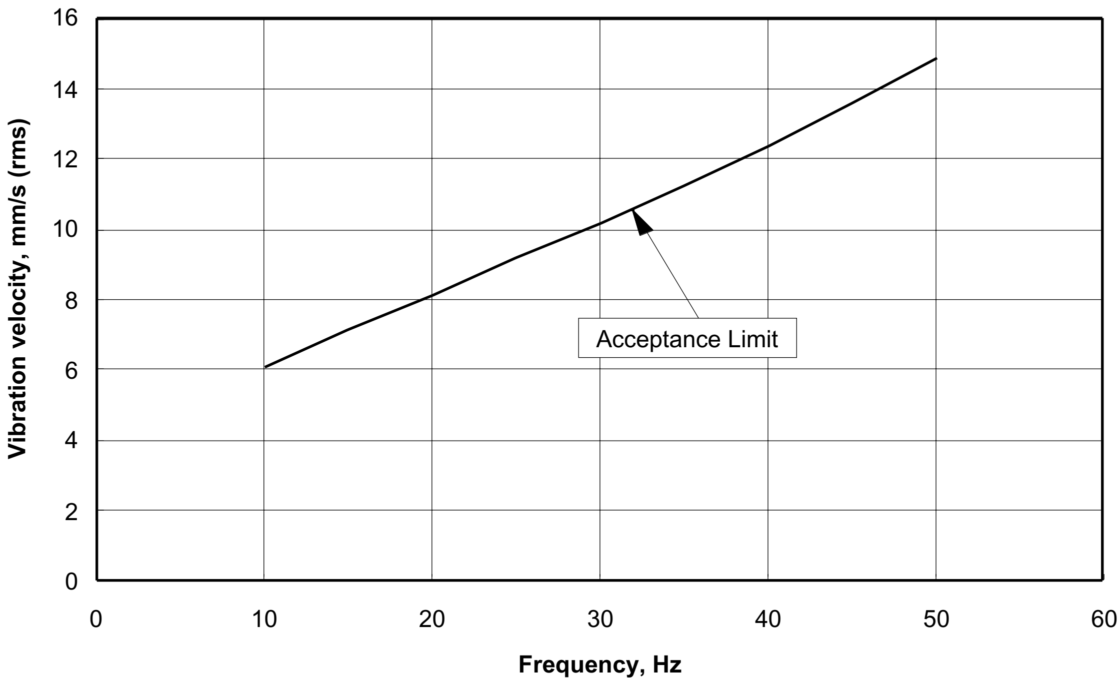


Figure D-2 *Contoured Fitting with Single Valve*



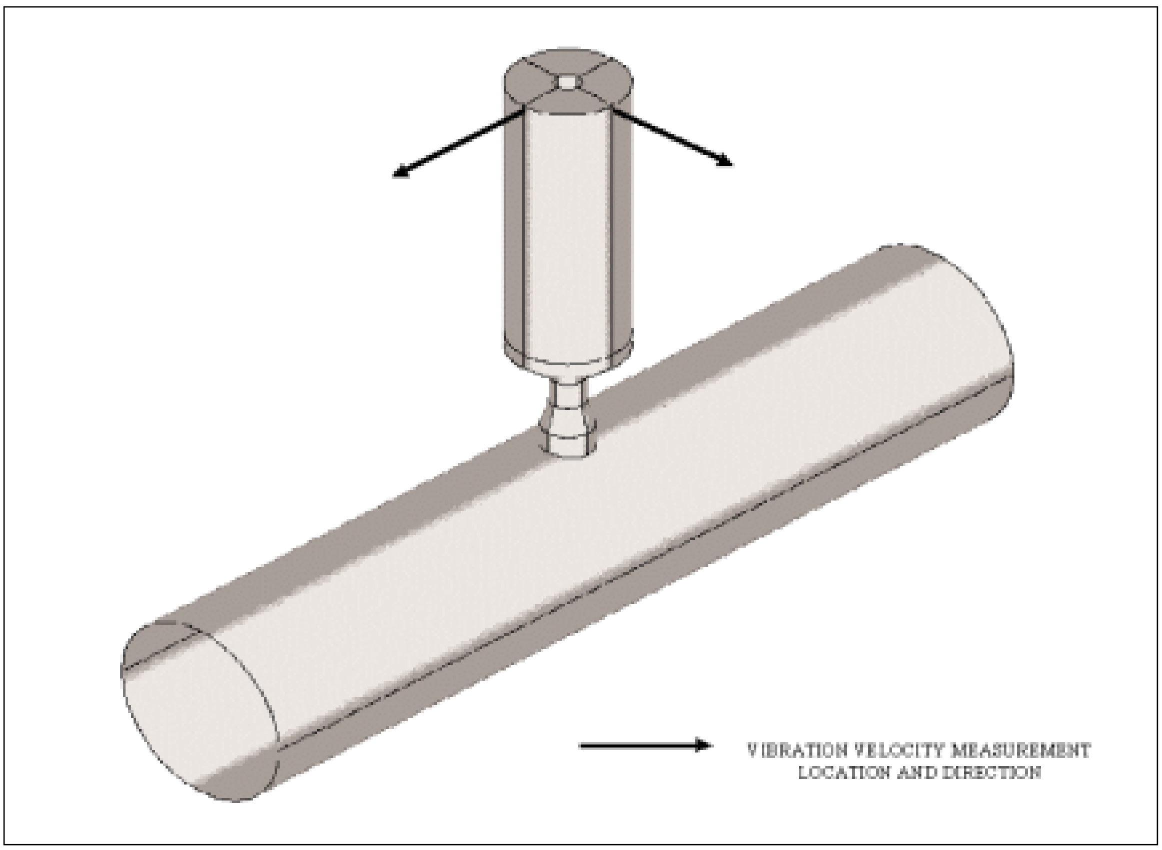
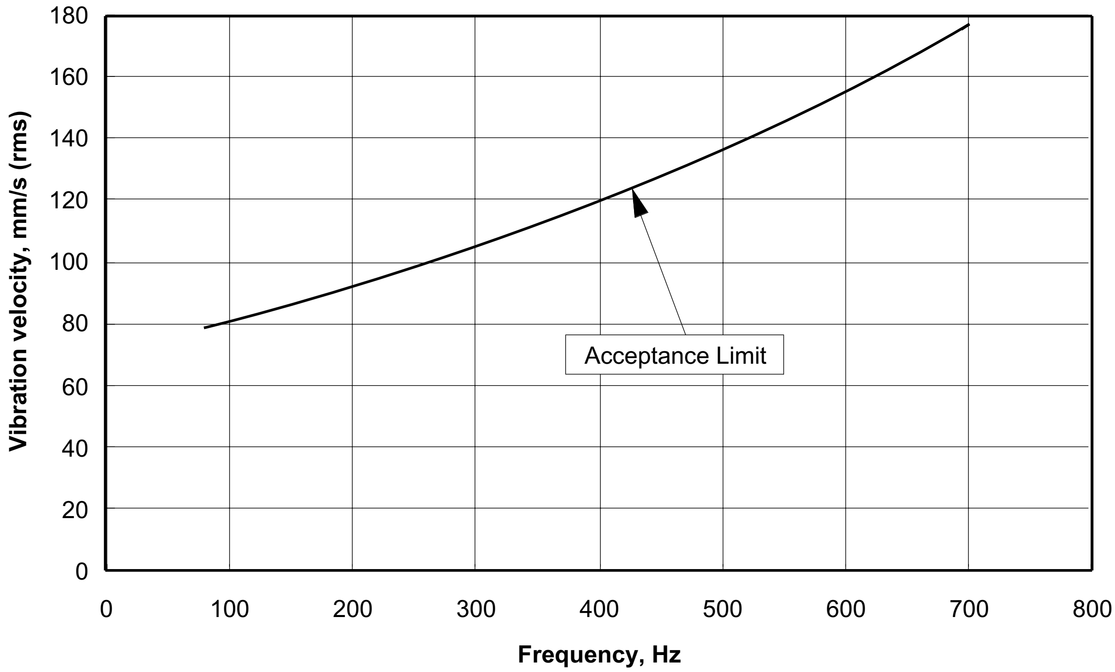


Figure D-3 *Contoured Fitting with Double Valve*



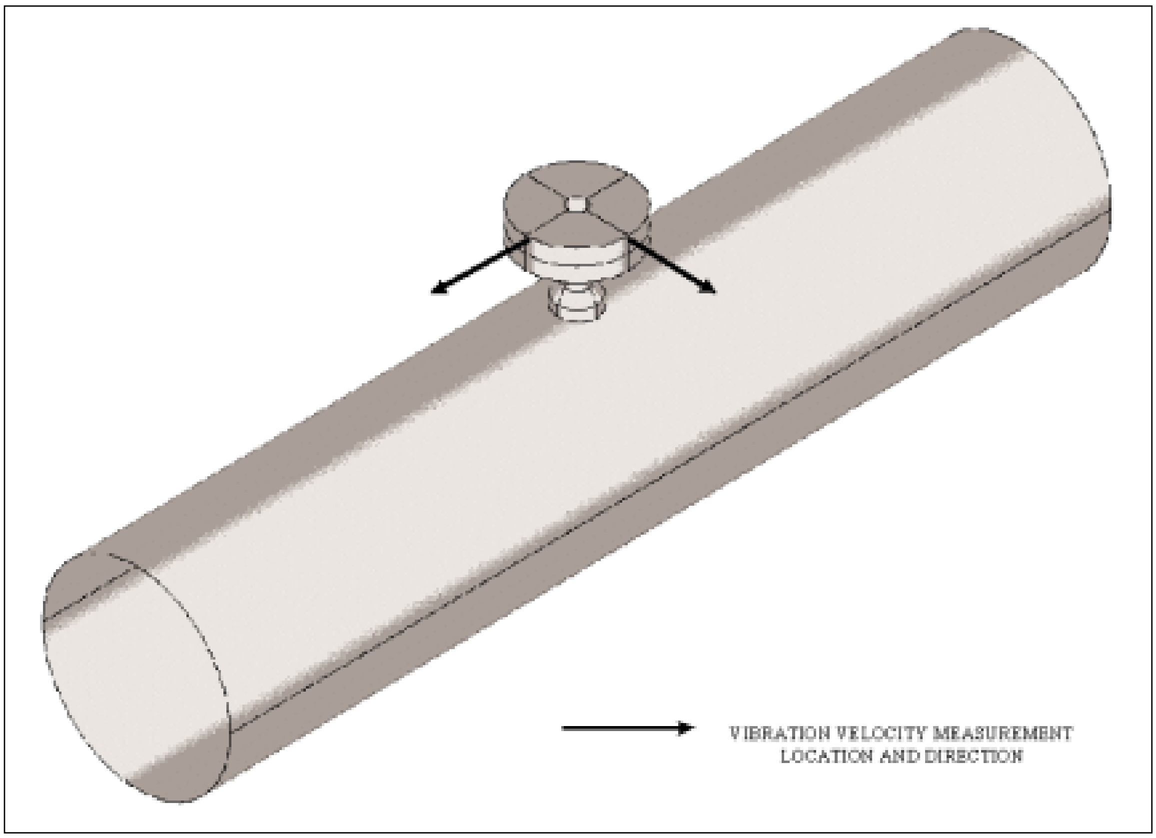
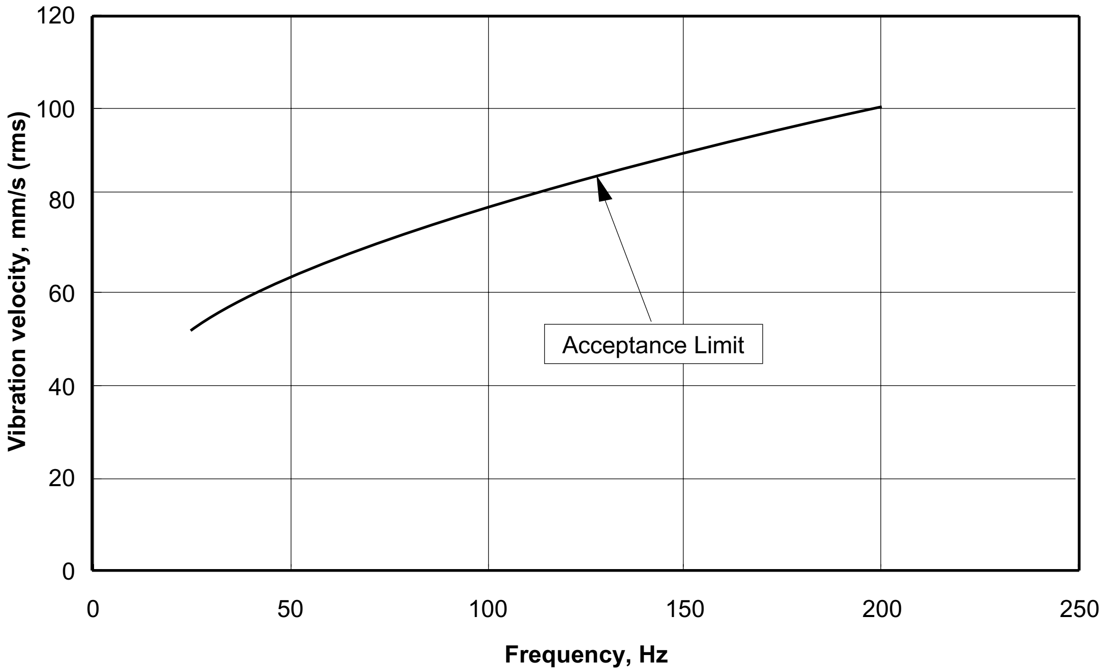


Figure D-4 *Short Contoured Body Fitting with ANSI/ASME B16.5 Flange*



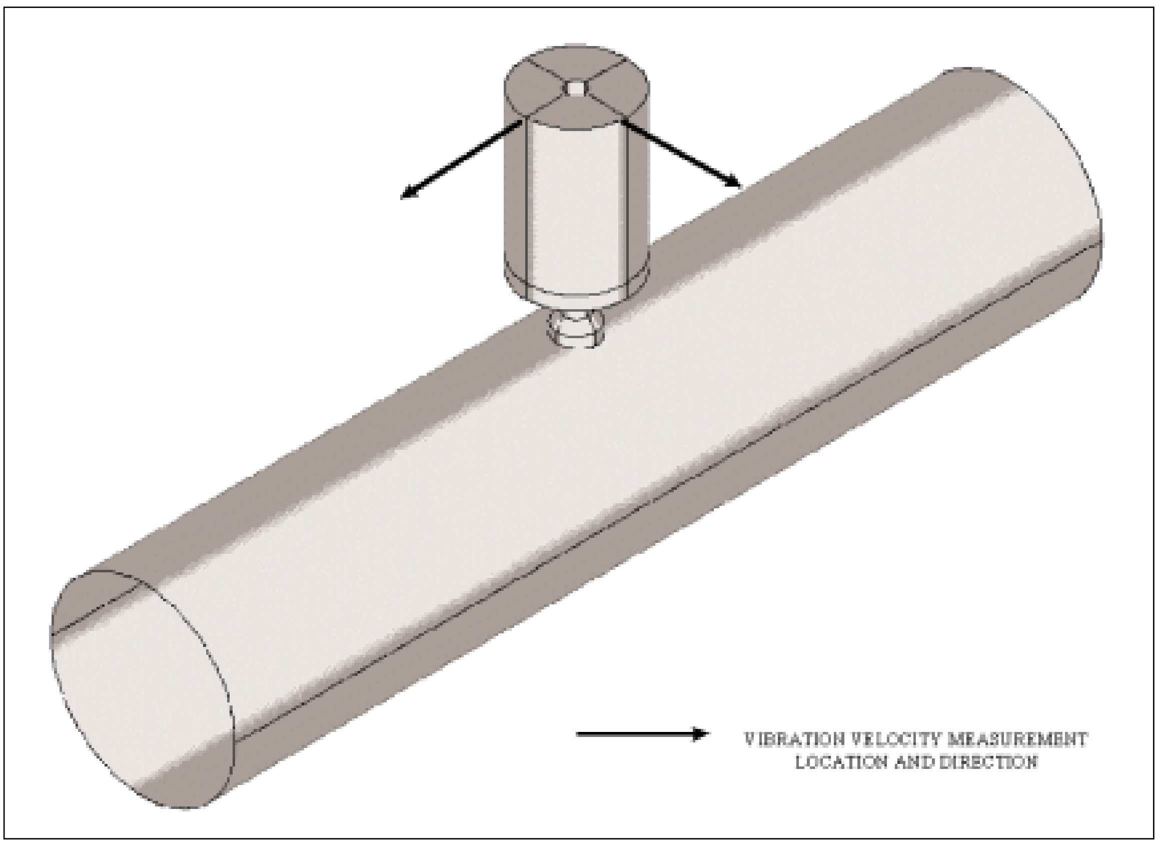
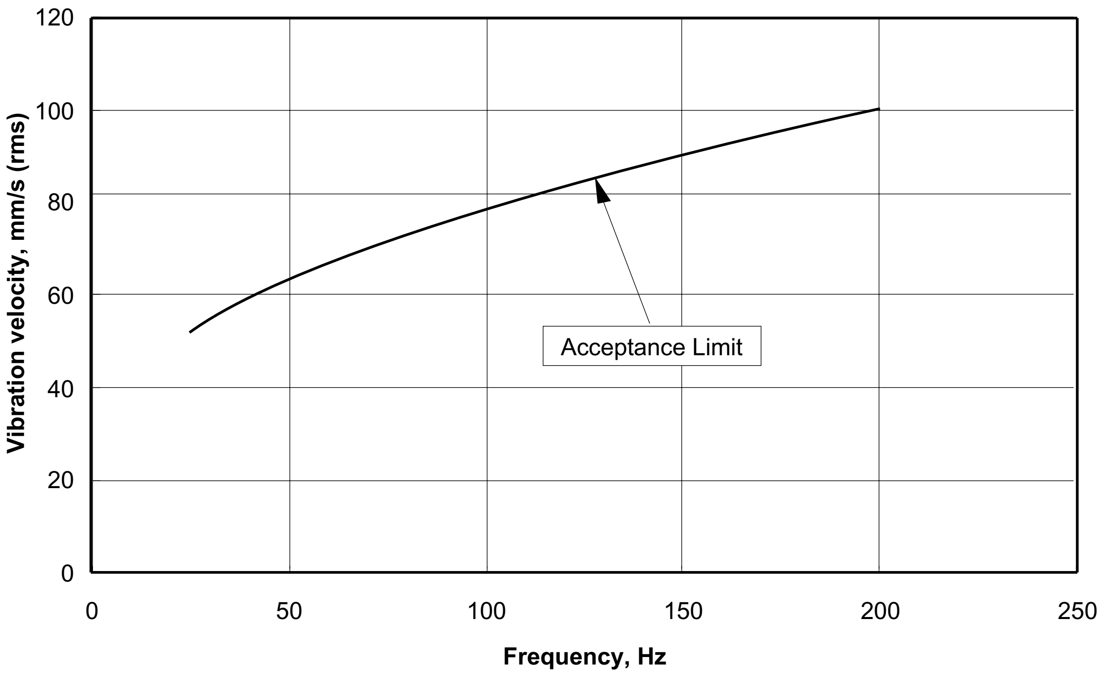


Figure D-5 *Short Contoured Body Fitting with Single Valve*



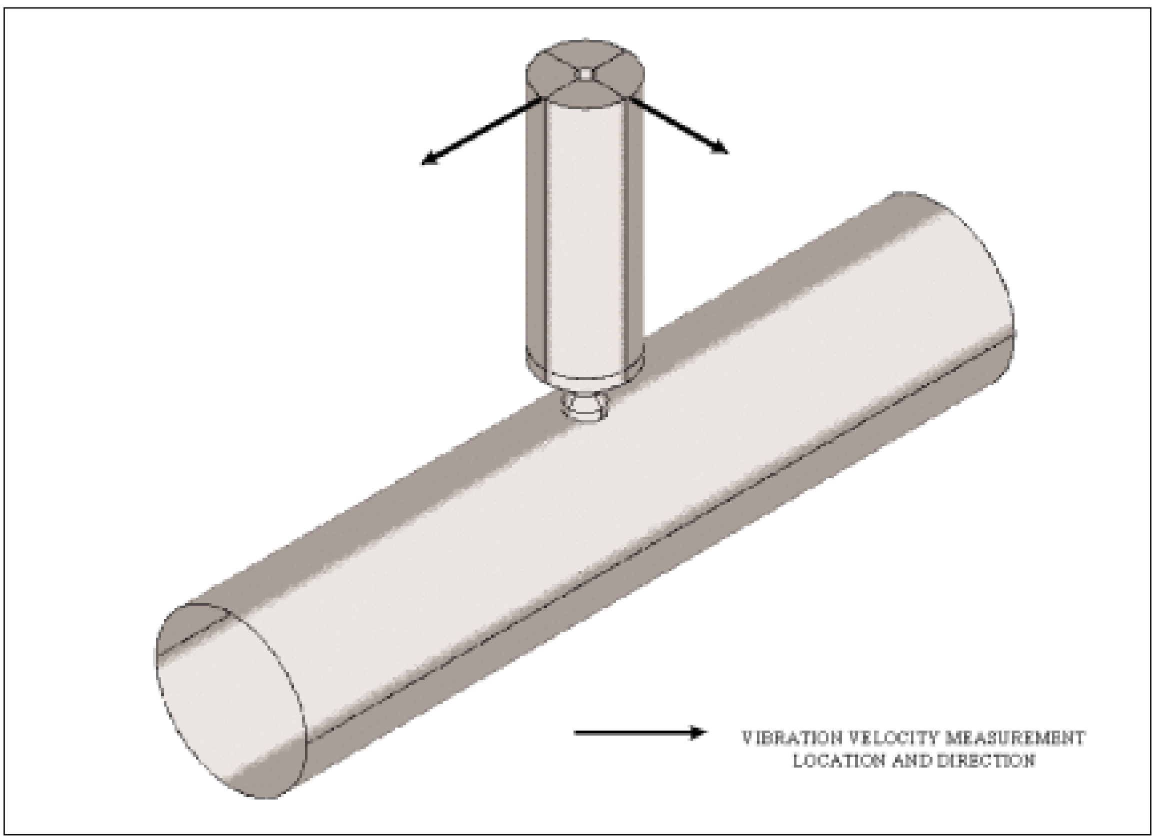
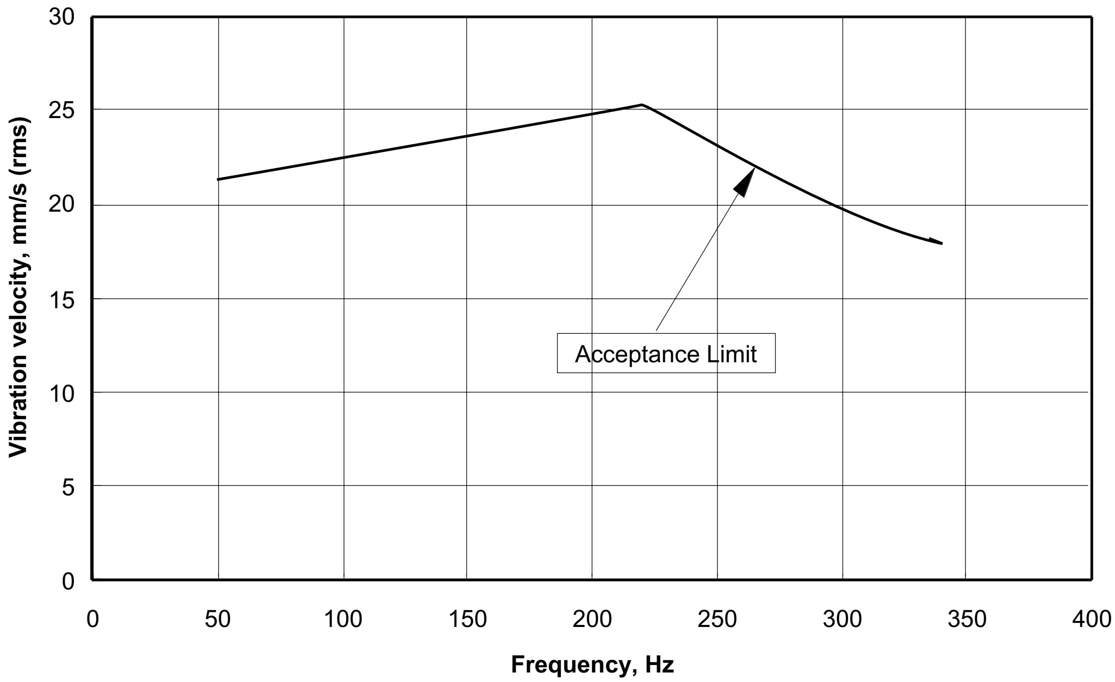


Figure D-6 *Short Contoured Body Fitting with Double Valve*



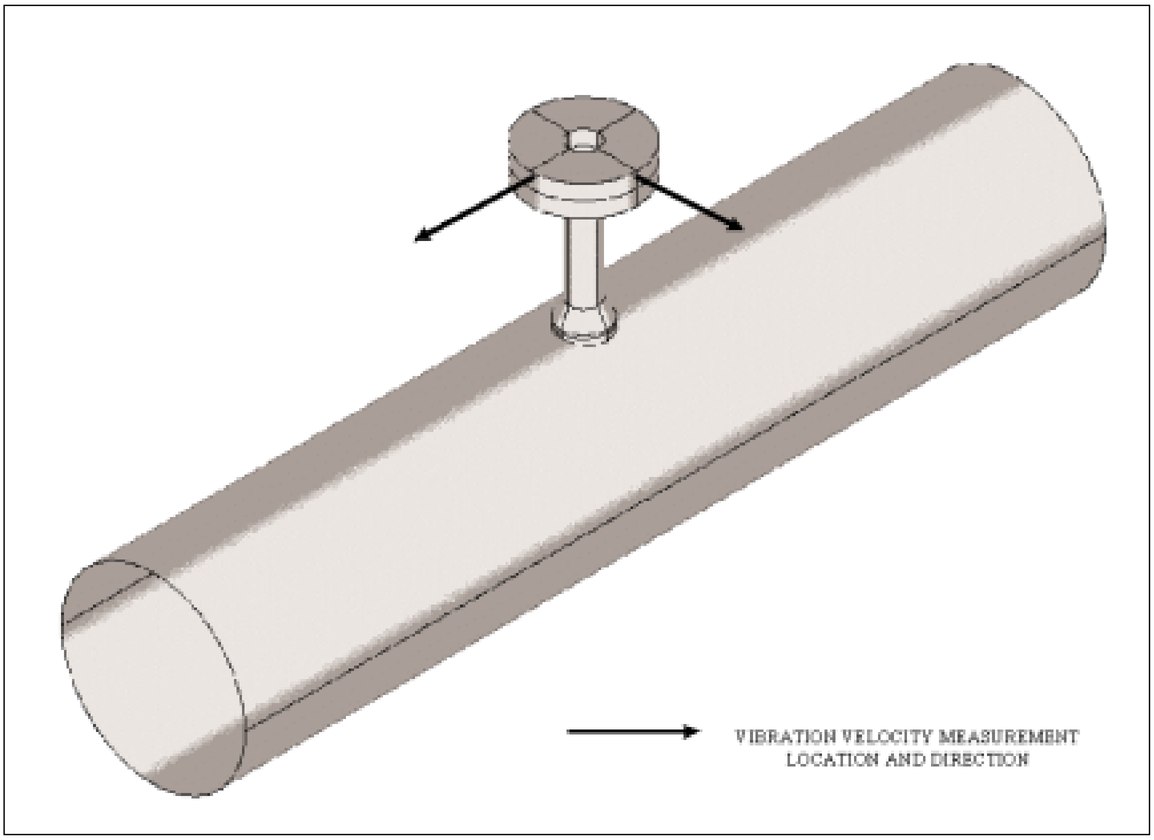
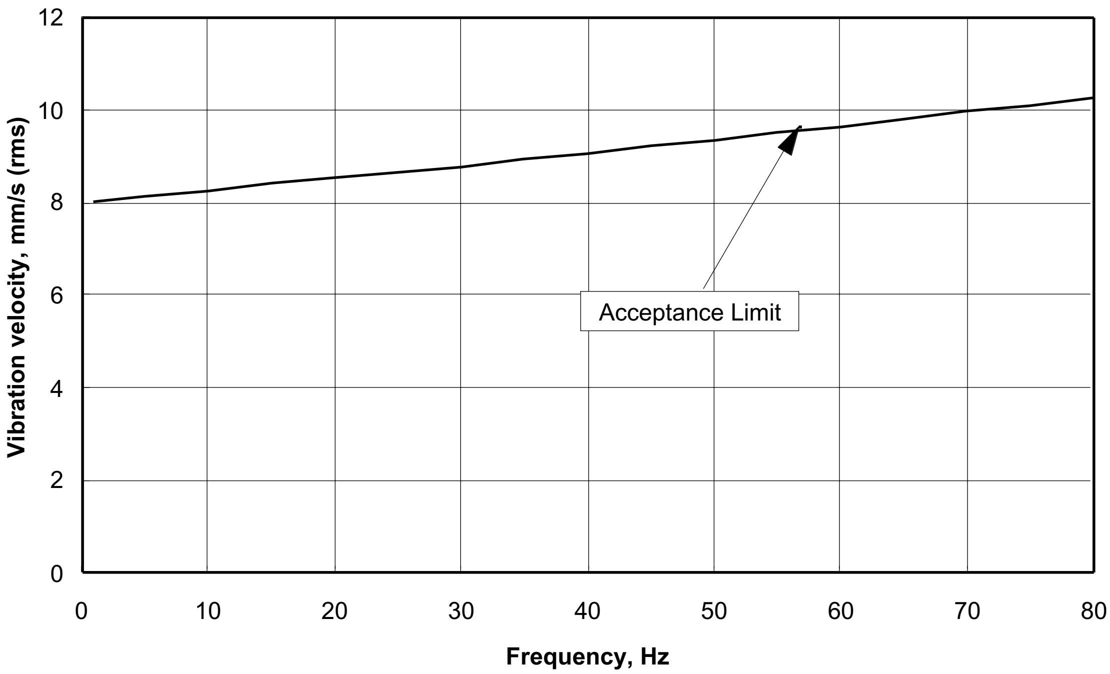


Figure D-7 *Weldolet Fitting with ANSI/ASME B16.5 Flange*



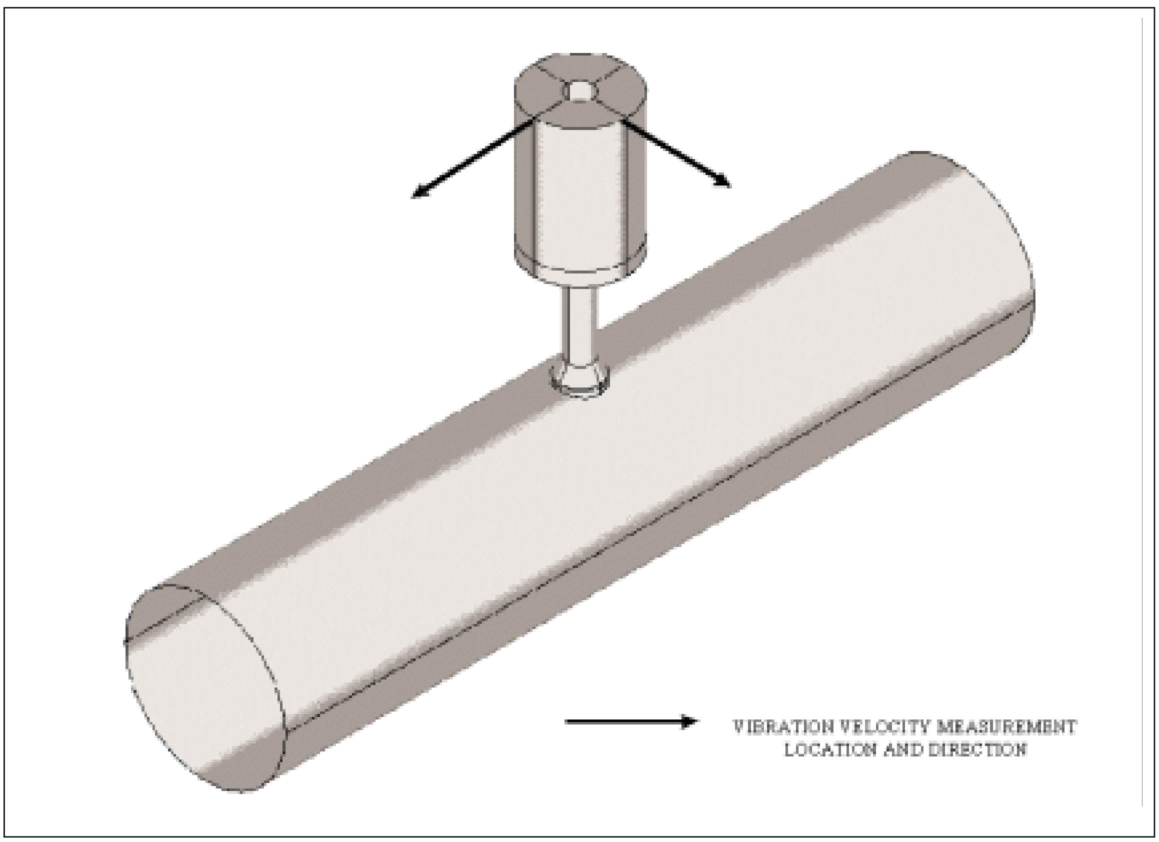
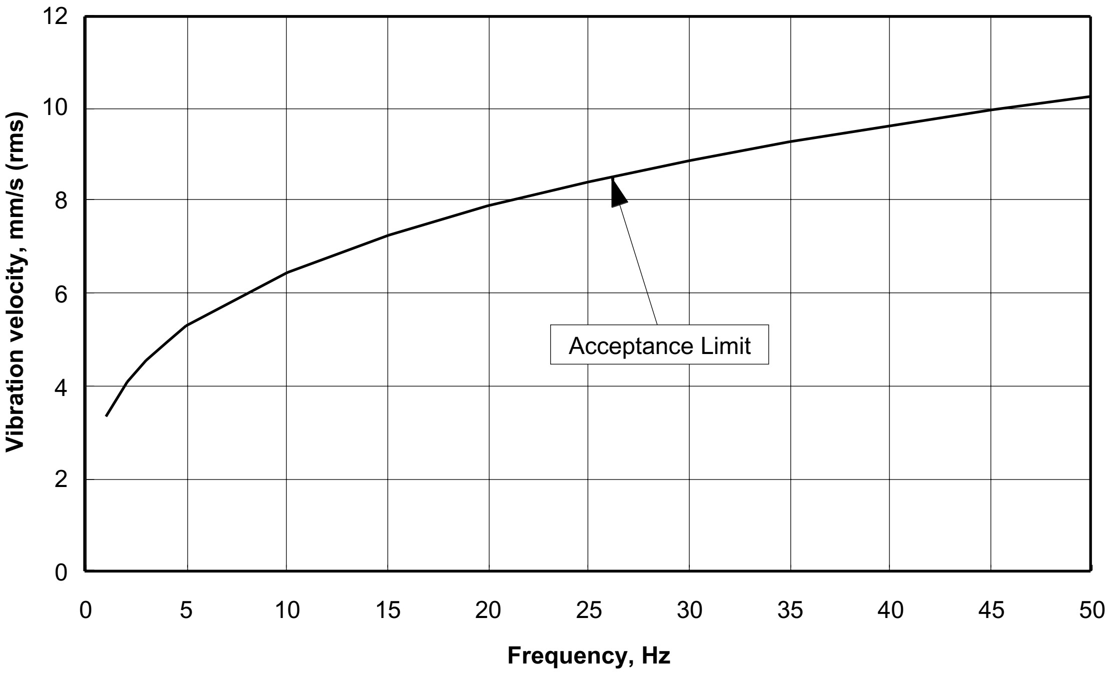


Figure D-8 *Weldolet Fitting with Single Valve*



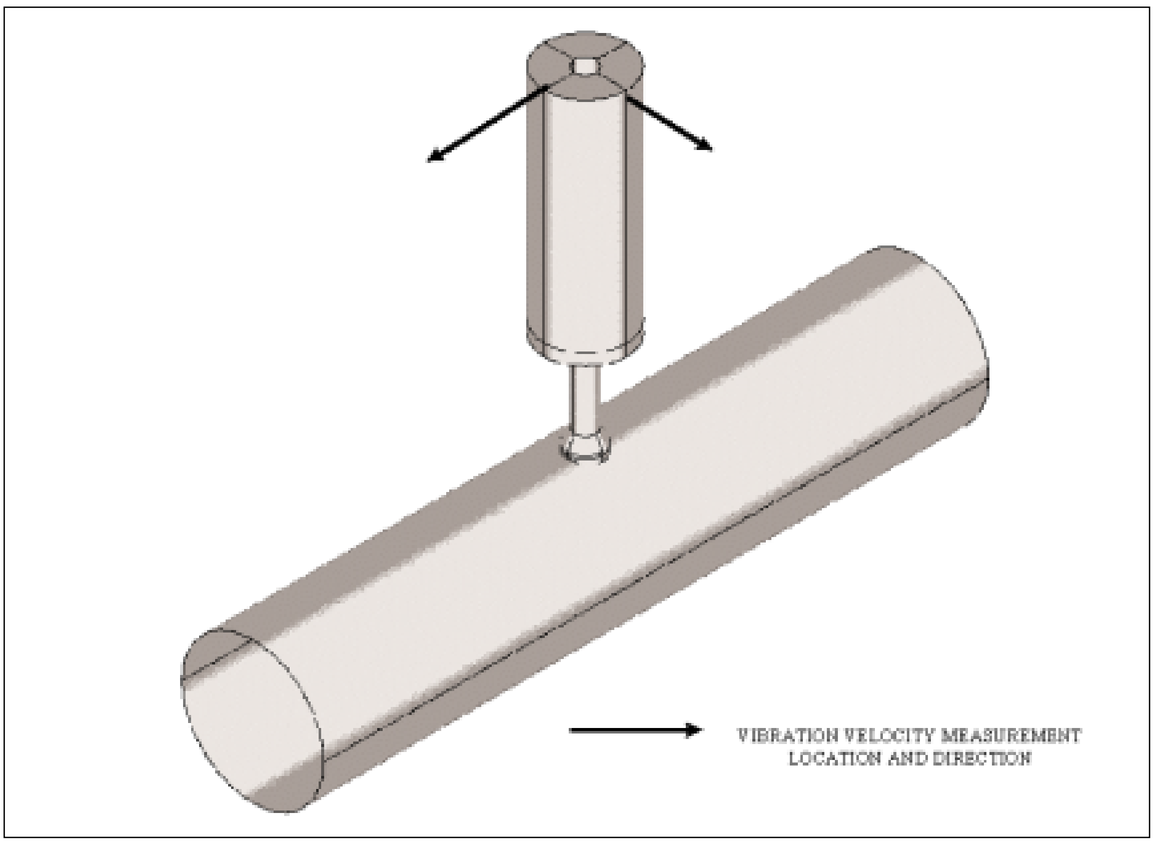
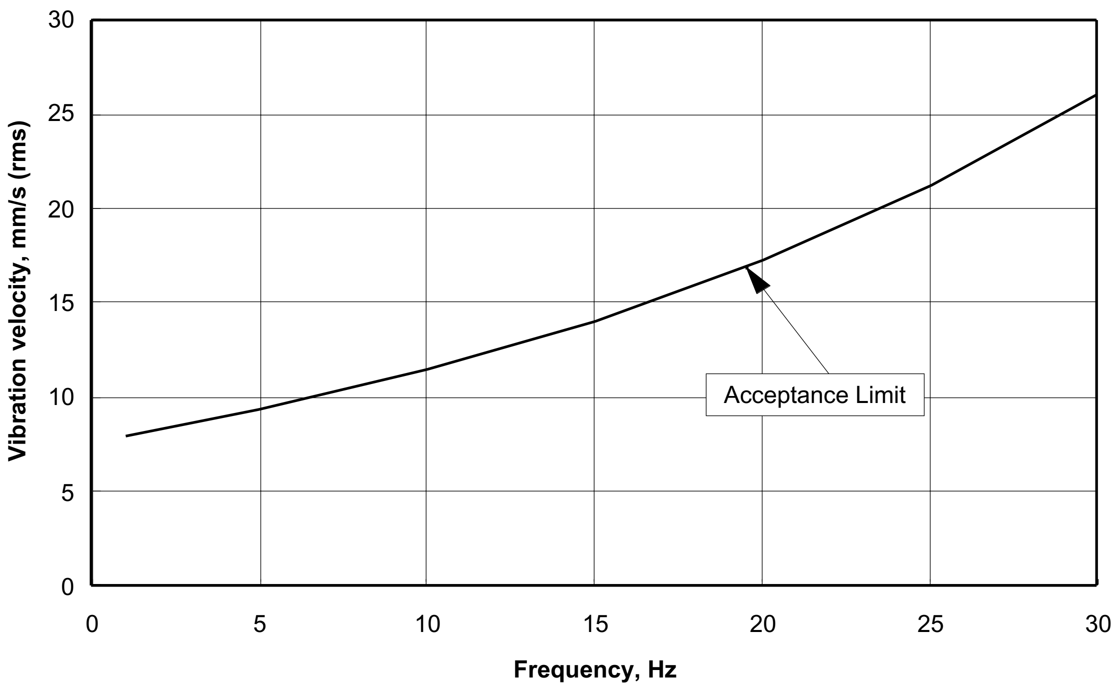


Figure D-9 *Weldolet Fitting with Double Valve*



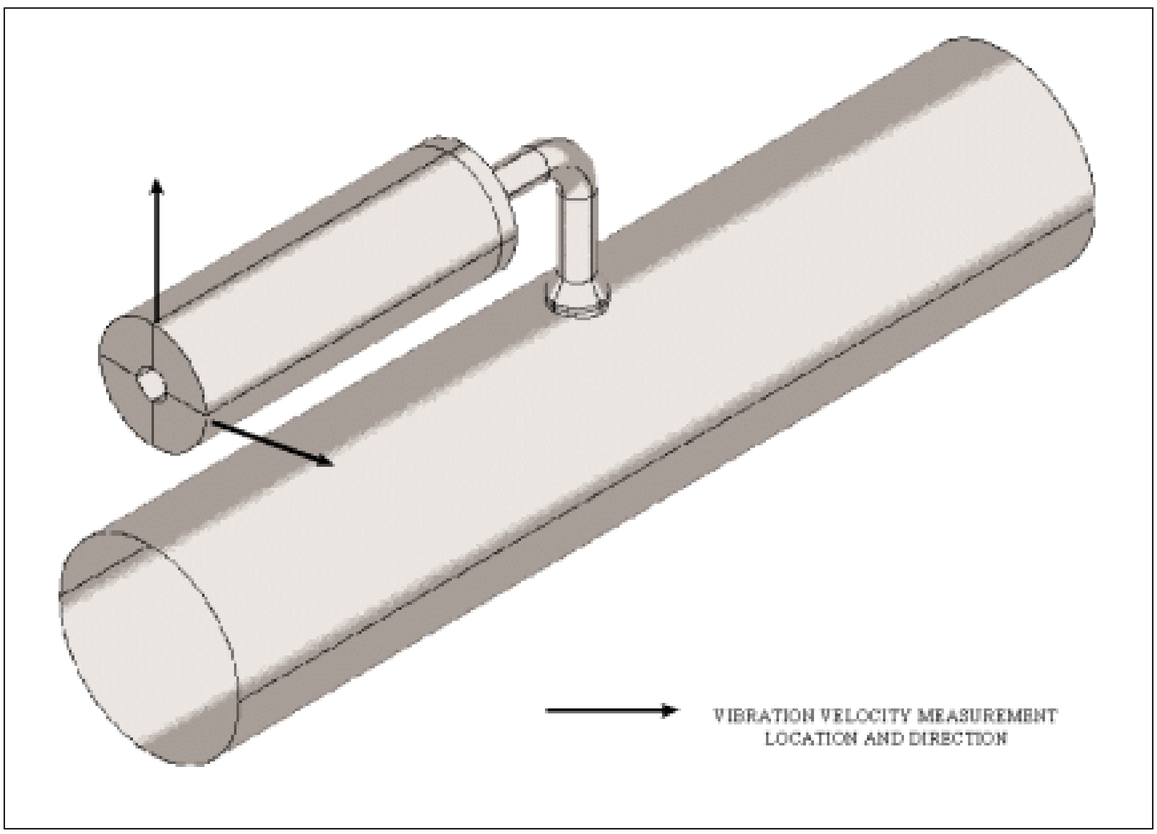


Figure D-10 *Weldolet Fitting with multiple Double Block and Bleed Valve*

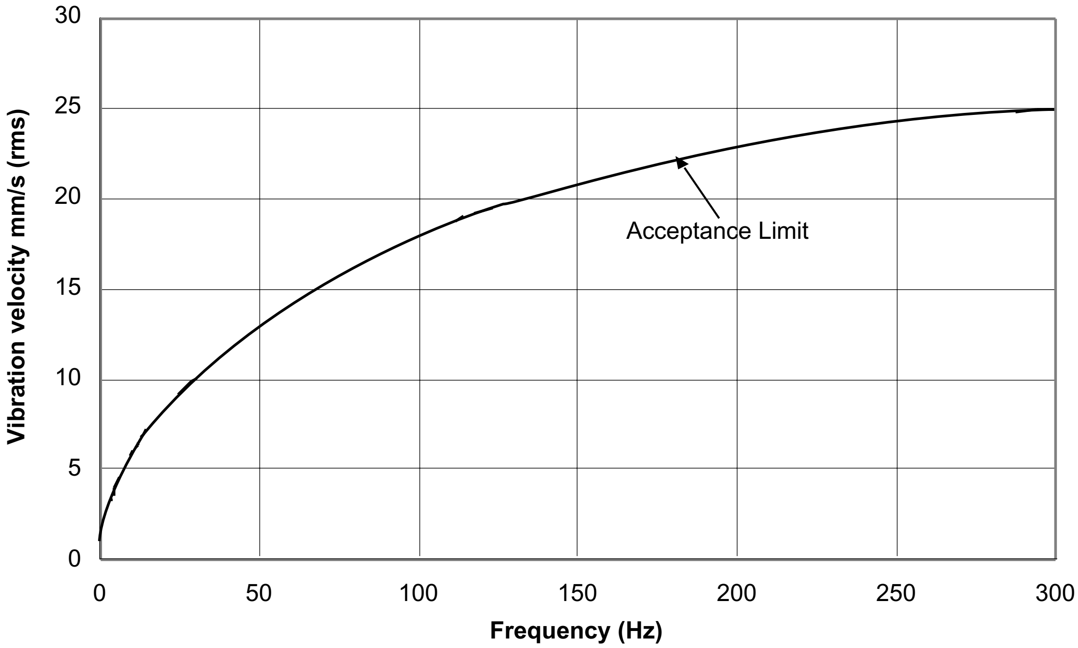


Figure D-11 *Acceptance Limit for Main Pipe*

Appendix E

Record Sheet for Piping Vibration Problems Experienced in Service

MTD would welcome any feedback from users of these Guidelines. In particular, we are interested in the ease of use and comprehension of the Guidelines and whether there are any points requiring clarification.

The Guidelines have already been tested in the field on different design details and process conditions, however, we are keen to receive additional experience to add to our database which in turn will enable further refinements and validation.

Your input will be greatly appreciated and please be assured that all responses will be treated in strict confidence.

1.0 Problem Details

1.1 Was the problem associated with:

a) a fatigue failure? If so, were any other degradation mechanisms present (e.g.corrosion/erosion)?
b) excessive vibration?

1.2 Was the problem associated with the main pipe or a small bore connection?

1.3 Was vibration/strain data measured? If so, please attach survey data, including:

a) type of measurement (vibration/strain etc)
b) measurement locations
c) measurement results (especially amplitude and frequency data).

1.4 Was a metallurgical examination conducted? If so, please attach details.

2.0 Process Details

Please attach the following information:

a) type of system and system description
b) type of fluid and phase
c) process data pertaining to the conditions at the time the problem was experienced, including:
   - ☐ date and time
   - ☐ pressure
   - ☐ temperature
   - ☐ fluid density
   - ☐ flow rate
d) P&ID’s (please record the drawing number here)
e) process flow diagram

3.0 Piping Geometry Details

Please attach the following information:

a) system isometric drawing(s) (please give the drawing number here)
b) pipe diameter (nominal), wall thickness (schedule), span length, material specification
c) details of small bore connection(s).

4.0 Contact Details

Please provide a contact name, address, telephone number, fax number and Email address (if available).

PLEASE RETURN TO:
The Executive Secretary,
c/o The Society For Underwater Technology
80 Coleman Street
London,
WC2R 5BJ
United Kingdom.

If you have any initial queries our email address is MTD.rwb@btinternet.com

Background information about MTD

The Marine Technology Directorate (MTD) was set up in 1976 by the then Science Research Council with the objective of facilitating co-operative projects between industry, academia and government in research, and education and training in marine technology. "Marine Technology " is defined as all aspects of science, engineering and technology relating to the exploration and exploitation of the sea, both above and below the seabed. More specifically, the research interests evolved in support of offshore oil and gas production, subsea technology, ships and transportation, ocean environment and non-hydrocarbon resources.

MTD advanced into an association of members with representatives from industry companies, universities, government departments and agencies, and in 1986 was privatised to become a company limited by guarantee and a Registered Charity. This particular 'brand' of partnership enabled new technology challenges to be defined and developed into research programmes that underpinned the operation of the stakeholders in their effectiveness in an increasingly competitive and global market.

In 1989 MTD absorbed UEG (the member-based research and information group for the offshore and underwater engineering industries) and enhanced its initiatives in transferring technology from other sectors of industry. An important aspect of transferring technology is the dissemination of the results from the research programmes and a new publication series was launched to achieve that objective. MTD has therefore gained recognition as a publisher of guidelines and other material emanating from its research programme.

Historically, MTD's Managed Programmes of research have been funded jointly by The Engineering and Physical Sciences Research Council and industry and at its peak represented an annual research spend of £8.6 million.

For publications enquiries please email Publications@mtd.org.uk

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