
Guidelines for the Avoidance of Vibration Induced Fatigue Failure in Process Pipework

2nd edition

GUIDELINES FOR THE AVOIDANCE OF VIBRATION INDUCED FATIGUE FAILURE IN PROCESS PIPEWORK

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FOREWORD

The first edition of the Guidelines for the Avoidance of Vibration Induced Fatigue in Process Pipework was published by the Marine Technology Directorate in 2000 [0-1]. The document was based on the outcome of a Joint Industry Project, which was initiated in response to a growing number of onshore and offshore process piping failures especially within systems deploying extensive use of duplex stainless steel.

The Guidelines were augmented in 2002 with the publication of a Health and Safety Executive document covering transient pipework excitation associated with fast acting valves [0-2].

During 2004, copyright for the original Guidelines was transferred to the Energy Institute.

The original publication was intended principally for use at the design stage and in the period since first issue, more experience has been gained in practical application, and a number of potential extensions and improvements were identified. A second Joint Industry Project was therefore initiated to improve and expand the scope of the first edition. This commenced in late 2005 and was project managed by the Energy Institute, with Doosan Babcock and Bureau Veritas as specialist contractors. The objectives were to:

- i. Improve the overall usability of the Guidelines;
- ii. Update the assessment methodology to include the experience gained;
- iii. Include intrusive elements and extend the scope to a greater range of small bore connection designs;
- iv. Include the Health & Safety Executive publication.

The second edition now provides a comprehensive approach to the “through life” management of pipework vibration-induced fatigue. Both qualitative and quantitative assessment methods are provided, following a similar philosophy to that outlined in API581 [0-3].

This publication has been compiled for guidance only and is intended to provide knowledge of good practice to assist operators develop their own management systems. While every reasonable care has been taken to ensure the accuracy and relevance of its contents, the Energy Institute, its sponsoring companies and other companies who have contributed to its preparation, cannot accept any responsibility for any action taken, or not taken, on the basis of this information. The Energy Institute shall not be liable to any person for any loss or damage which may arise from the use of any of the information contained in any of its publications.

These Guidelines may be reviewed from time to time and it would be of considerable assistance for any future revision if users would send comments or suggestions for improvements to:

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The Joint Industry Project was set up to also enable a Steering Group to be formed from expert representatives from the sponsoring companies. The Steering Group met on several occasions to permit discussion and agreement on the direction and format of the Guideline as it was being developed. The group also provided written comment and feedback on technical reports and document text out with the meetings. The Steering Group comprised the following members:

Keith Hart (JIP Manager & Chairman)	The Energy Institute
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Acknowledgement is also attributed to other key personnel at Doosan Babcock and BV especially Jonathan Baker, who provided valuable assistance to the principal authors.

SUMMARY

This document provides a public domain methodology to help minimise the risk of vibration induced fatigue of process piping. It is intended for use by engineers with no prerequisite knowledge of vibration.

Pipework vibration is only superficially covered by standard design codes, and hence awareness of the problem among plant designers and operators is limited (e.g. B31.1 **[0-4]**). It is intended that this document will address this issue.

These Guidelines can be used to assess (i) a new design, (ii) an existing plant, (iii) a change to an existing plant and (iv) a potential problem that has been identified on an operating system. They therefore offer a proactive approach to pipework vibration issues. This is in contrast to the highly reactive approach traditionally employed when vibration problems arise, e.g. during the commissioning or when operational changes are made.

These Guidelines provide a staged approach. Initially, a qualitative assessment is undertaken to (i) identify the potential excitation mechanisms that may exist and (ii) provide a means of rank ordering a number of process systems or units in order to prioritise the subsequent assessment. A quantitative assessment is then undertaken on the higher risk areas to determine the likelihood of a vibration induced piping failure. Details of onsite inspection and measurement survey techniques are provided to help refine the quantitative assessment for an as-built system. To reduce the risk to an acceptable level, example corrective actions are outlined.

It is recognised that there will always be some cases where the type of excitation or complexity of response is outside the scope of these Guidelines. In such cases specialist advice should be sought.

1 INTRODUCTION

1.1 OVERVIEW

Vibration induced fatigue failures of pipework are a major concern due to the associated issues with:

- safety, e.g. sudden release of pressurised fluid which is hazardous or flammable etc.,
- production down time,
- corrective action costs,
- environmental impact,

Therefore it is in the interest of the duty holder or operator to minimise this risk.

Process piping systems have traditionally been designed on the basis of a static analysis with little or no attention paid to vibration induced fatigue. This is principally because most piping design codes do not address the issue of vibration in any meaningful way. This results in piping vibration being considered on an adhoc or reactive basis.

Data published by the UK's Health & Safety Executive for the offshore industry have shown that in the UK Sector of the North Sea piping vibration and fatigue accounts for over 20% of all hydrocarbon releases **[1-1]**. Although overall statistics are not available for onshore facilities, data are available for individual plants which indicate that in Western Europe between 10% and 15% of pipework failures are caused by vibration induced fatigue.

There are several 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 flow rates as a result of debottlenecking and the relaxation of erosion velocity limits, resulting in higher flow velocities with a correspondingly greater level of turbulent energy in process systems.
- for new designs of offshore plant the greater use of thin walled pipework (e.g. duplex stainless steel alloys) results in more flexible pipework and higher stress concentrations particularly at small bore connections.

These Guidelines are designed to provide guidance, assessment methods and advice on control and mitigation measures for the following situations:

- i. When a new process system is being designed.
- ii. When undertaking an assessment of an existing plant or process system.
- iii. When changes to an existing plant or process system are being considered (such as operational, process or equipment changes).
- iv. When a vibration issue is identified on an existing plant.

Cases (i) to (iii) above constitute a proactive approach to the management of vibration induced fatigue, whilst case (iv) is, by its very nature, reactive. It is hoped, that by using the guidance given in this document, designers and operators will move towards a more

proactive approach to the “through life” management of vibration induced fatigue in process piping systems.

These Guidelines have been divided into two main parts:

1. A series of core sections (Chapters) which provide an introduction to piping vibration and how the Guidelines should be used in different situations.
2. A toolbox of methods (Technical Modules) encompassing ‘paper based’ assessment methods and visual inspection and measurement survey techniques; these are applied in different ways depending on the individual situation. Advice is also provided in terms of typical corrective actions which might be employed and good design practice.

In addition supplementary information is provided in the appendices.

These guidelines cover the most common excitation mechanisms which occur in process plant. However they do not cover environmental loading (e.g. wind, wave, seismic activity).

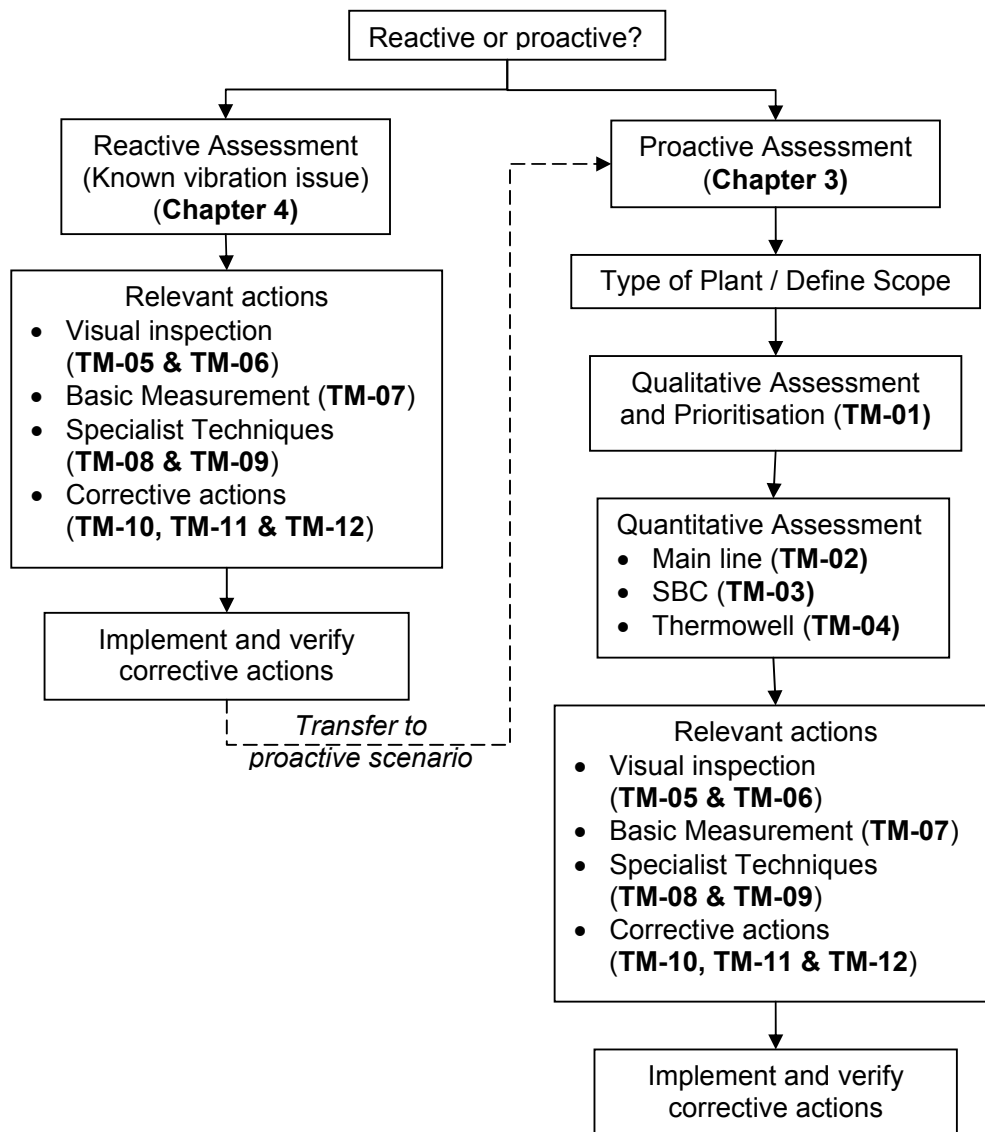
It should be noted that corrosion and erosion issues are likely to increase the susceptibility of pipework to vibration induced fatigue failures. The assessment approach assumes that the plant has been built to industry standard codes and procedures and is in a good condition. If this is not the case, a greater emphasis should be placed on the onsite inspection and measurement aspects.

1.2 HOW TO USE THESE GUIDELINES

An overview of piping vibration and various excitation mechanisms is provided in **Chapter 2**. **Chapter 3** details a proactive assessment methodology and how it is applied in different situations (i.e. a new plant, an existing plant or changes to an existing plant). Finally **Chapter 4** addresses the case where there is a known vibration issue, which results in a reactive assessment.

Details of specific elements of the assessment are provided in the technical modules (TM) and the appendices provide supplementary information and examples of how the assessment can be applied.

An overview of the assessment methodology is given in **Flowchart 1-1**.



Flowchart 1-1 Overview of Assessment Approach

1.2.1 Types of Assessment

1.2.1.1 Proactive Assessment (Chapter 3)

There are three different situations considered in these Guidelines:

- **New Plant:** New green/brownfield site or a new process module or unit. (refer to **Flowchart 3.1**)
Note: many common vibration issues can be addressed by incorporating good engineering practice at the design phase, refer to **TM-13** for general guidance.
- **Existing Plant:** Plant in current operation (refer to **Flowchart 3.2**)
- **Plant Change:** Process, piping or equipment change to an existing system (refer to **Flowchart 3.3**)

For each of the three situations there is an initial qualitative assessment (provided in **TM-01**) and subsequent quantitative assessments (provided in **TM-02**, **TM-03** and **TM-04**).

The primary difference between qualitative and quantitative assessments has been defined by API 581 [1-2] and relates to the level of resolution in the analysis. The qualitative procedure requires less detailed information about the facility and, consequently, its ability to discriminate is much more limited. The qualitative technique would normally be used to rank units or major portions of units at a plant site to determine priorities for quantitative studies or similar activities.

A quantitative analysis, on the other hand, will provide likelihood of failure values for main pipework, small bore connections (SBC) and intrusive elements. With this level of information, suitable actions can be identified including vibration measurements and corrective actions.

1.2.1.2 Reactive Assessment (Chapter 4)

The reactive assessment addresses the case of an existing plant where there are known vibration issues. Once these have been addressed a proactive strategy should be implemented.

1.2.2 Operating Conditions

The assessment will only be effective if the full operational envelope is considered.

1.2.3 Visual Inspection

Visual inspection is an important tool and is used to identify potential issues which cannot be identified by a “paper based” assessment (refer to **TM-05** and **TM-06**).

1.2.4 Implement and Verify Corrective Actions

To ensure that any corrective actions applied to a plant have reduced the risk of vibration induced fatigue to an acceptable level, a verification process is required.

2 OVERVIEW OF PIPING VIBRATION

2.1 OVERVIEW

The purpose of this section is to give an overview of the different types of excitation and the accompanying piping 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.

2.2 INTRODUCTION TO VIBRATION

Vibration is an oscillatory motion about an equilibrium position.

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

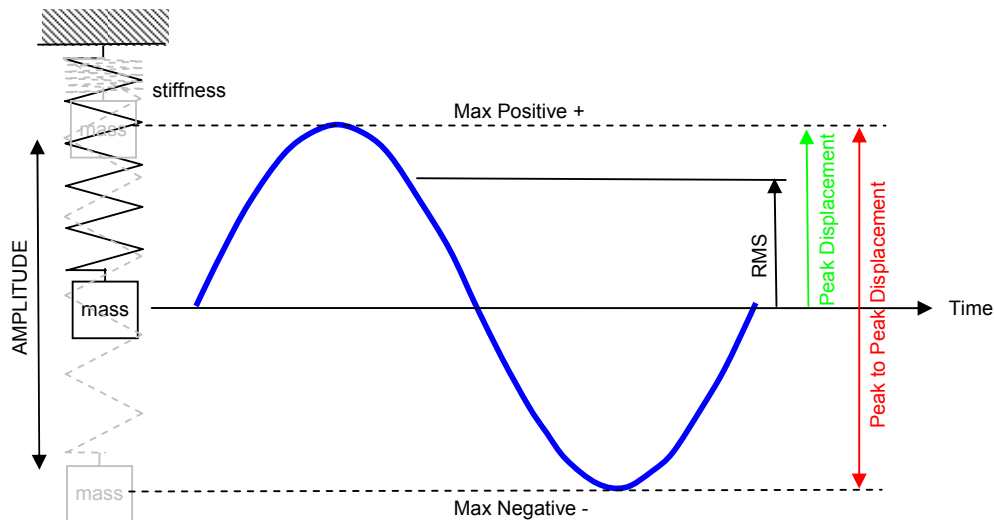


Figure 2-1 Description of vibration using a simple spring-mass system

Where RMS is root mean square

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.

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

Very little energy is required to excite the natural frequency of a system, as the system '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 (refer to **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 and 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 reliable method of assessing the severity of the problem.

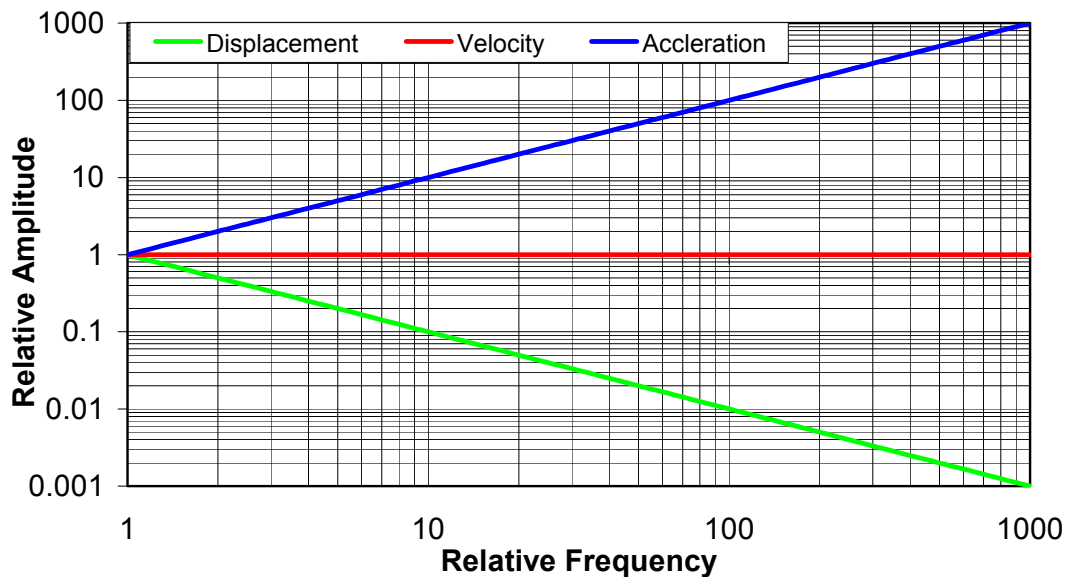


Figure 2-2 Comparison of the amplitude of displacement, velocity and acceleration as a function of frequency

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, which has locations of zero motion (nodes) and maximum motion (anti-nodes). 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 nodes and anti-nodes of the respective mode shapes.

Excitation can either be tonal i.e. energy is only input at discrete frequencies, or broadband 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's natural frequencies:

Tonal Excitation - Resonant

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 $\pm 20\%$ of the system's 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.

Broadband Excitation

If the excitation is broadband then there is a probability that some energy will be input at the system's 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.

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 [2-1] and [2-2] and for applications to process piping systems see [2-3] and [2-4].

2.3 COMMON CAUSES OF PIPING VIBRATION

2.3.1 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 the 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 reducers.

This in turn generates potentially high levels of broadband kinetic energy local to the turbulent source (refer to **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 (refer to **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.

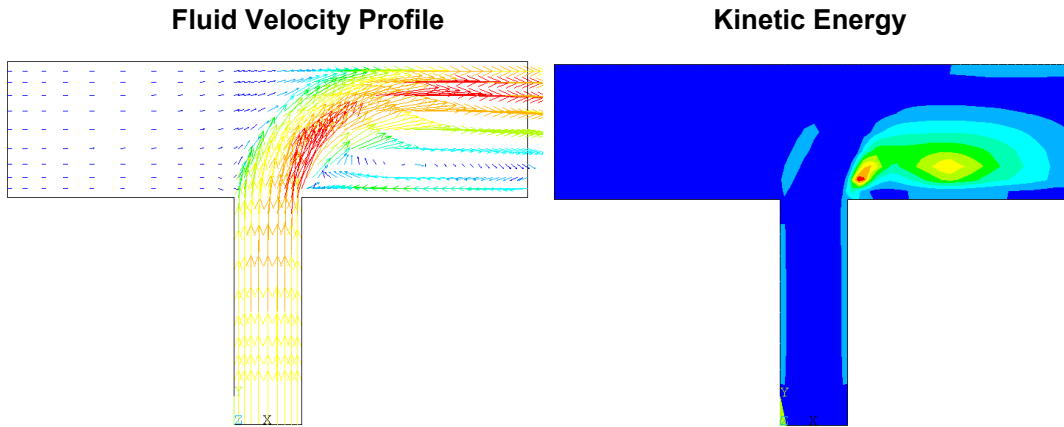


Figure 2-3 An example of the distribution of kinetic energy due to turbulence generated by flow into a tee

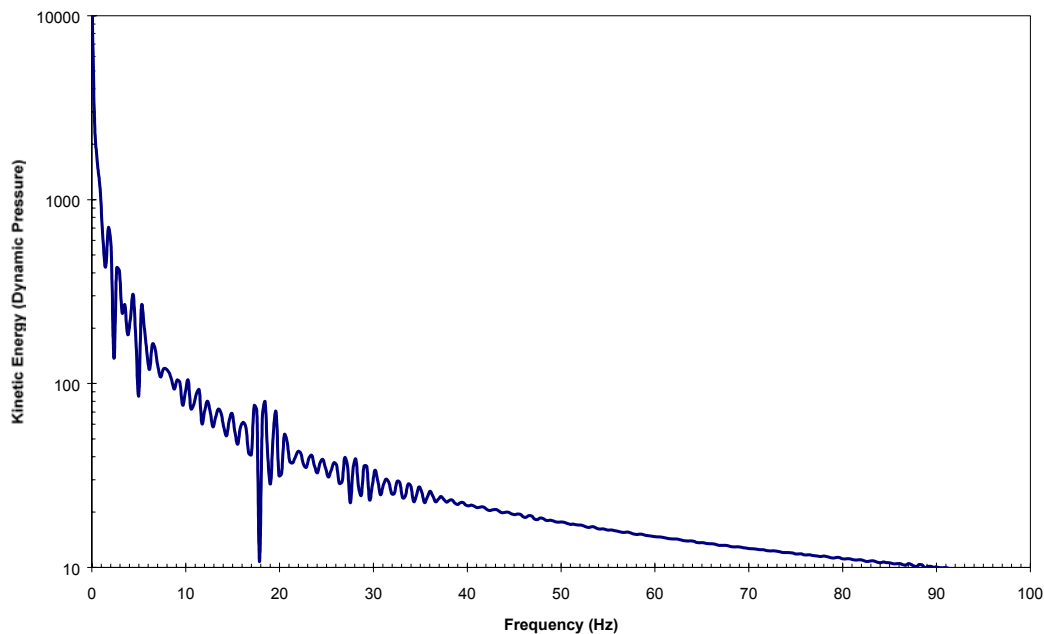


Figure 2-4 Turbulent energy as a function of frequency

2.3.2 Mechanical Excitation

Most of the problems of this nature encountered have been associated with reciprocating/positive displacement compressors and pumps. In such machines, the dynamic 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 a 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. To ensure that there is no coupling the excitation frequency(ies) (including harmonics) should not be within $\pm 20\%$ of the structural natural frequencies.

Problems can also occur on pipework which shares supports with either the machinery or associated pipework, but is not part of the system which involves the excitation.

2.3.3 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 closed valve) or open (e.g. entry to a vessel such as a knock out drum). In the high frequency range (typically above a few hundred Hertz) 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 (e.g. 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 (refer 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 non-resonant vibration can be experienced if there are significant levels of excitation present in the system.

To ensure that there is no coupling the excitation frequency(ies) (including harmonics) should not be within $\pm 20\%$ of the structural and acoustic natural frequencies.

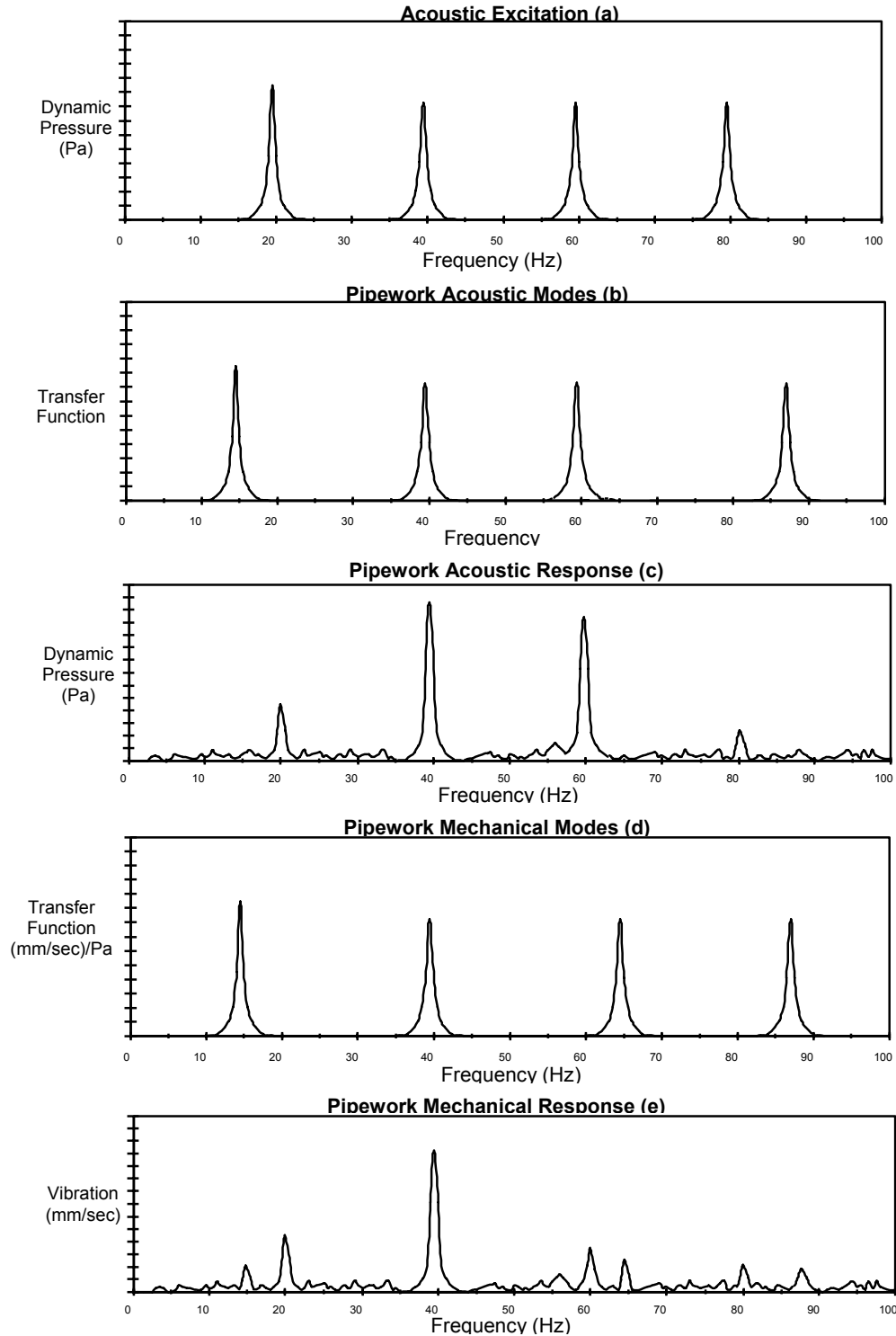


Figure 2-5 Relationship between acoustic natural frequencies and structural response

In the spatial domain, it is the location and phase of the dynamic force relative to the structural mode shape (refer to **Section 2.2**) that are 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.

2.3.3.1 Reciprocating/Positive Displacement Pumps and Compressors

Reciprocating/positive displacement 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/positive displacement compressors and pumps to predict the pressure pulsation levels in the system. This analysis is usually undertaken to meet the requirements of API 618 [2-5] (reciprocating compressors) and API 674 [2-6] (positive displacement-reciprocating pumps).

2.3.3.2 Centrifugal Compressors (Rotating Stall)

Centrifugal compressors can generate tonal pressure pulsations at low flow conditions [2-7]. 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). Even if 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. 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.

2.3.3.3 Periodic Flow Induced Excitation

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

$$f = \frac{Sv}{d} \quad (2)$$

where v is the fluid velocity, d is the representative dimension of the component 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, the Strouhal numbers can vary widely over the range 0.1 to 1.0 [2-2].

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

- flow past the end of a dead leg branch (e.g. a recycle line or relief line with the valve shut);
- 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 (refer to **TM-04**).

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 greater than $\pm 20\%$ should be maintained over the flow regimes of interest.

'Dead Leg' Branches

Gas systems, at relatively high flow velocities, can exhibit a form of tonal excitation which is generated when flow past the end of a 'dead leg' branch generates an instability at the mouth of the branch connection (refer to **Figure 2-6**), similar to blowing across the top of a bottle generating a tonal response. 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.

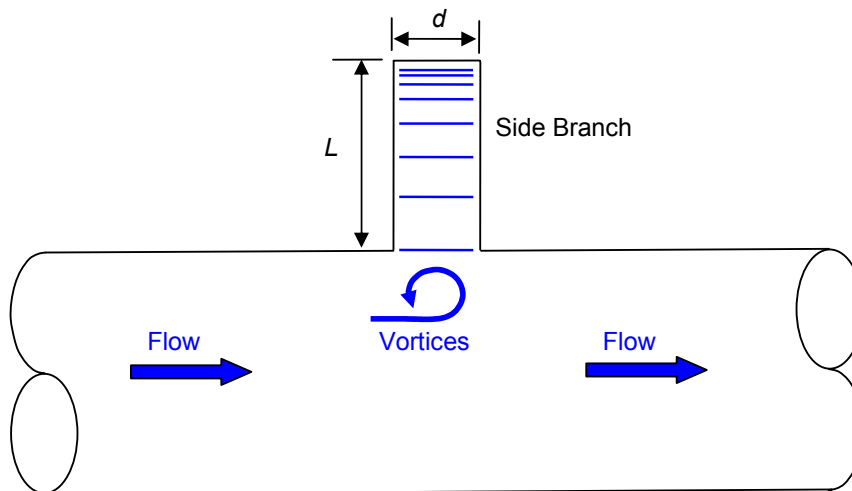


Figure 2-6 An example of a 'Dead Leg Branch'

Flow over Components in Fluid Stream

Flow over bodies or across edges of components in the gas stream can result in vortex shedding. 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 for the occurrence of 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 quills 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 frame. The design of thermowells is normally carried out to ANSI/ASME PTC 19.3 [2-8], but it is known that this can be non-conservative in certain situations.

2.3.4 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. 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.

Typical dominant frequencies associated with high frequency acoustic excitation are between 500 to 2000Hz.

2.3.5 Surge/Momentum Changes Due to Valve Operation

Surge (or water hammer, as it is commonly known) is a pressure wave caused by the kinetic energy of a fluid in motion when it is forced to stop or change direction suddenly. If the pipe is suddenly closed at the outlet (downstream) a pressure wave is generated which travels back upstream at the speed of sound in the liquid. This can give rise to high levels of transient pressure and associated forces acting on the pipework.

High transient forces can also be generated by the rapid change in fluid momentum caused by the sudden opening or closing of a valve, e.g. fast operating of a relief valve.

2.3.6 Cavitation

Cavitation is the dynamic process of formation of bubbles inside a liquid, which suddenly form and collapse. It can occur where there is a localised pressure drop within the process fluid (e.g. at centrifugal pumps, valves, orifice plates). When the vapour bubbles collapse, they create very high localised pressures which result in noise, damage to components, vibrations, and a loss of efficiency.

2.3.7 Flashing

In cases when the pressure within the pipe becomes less than the vapour pressure of the fluid, the fluid can suddenly change from liquid into vapour state, resulting in large forces. Flashing typically occurs where there is localised pressure drop within the process fluid (e.g.

at centrifugal pumps, valves, orifice plates) or where two fluid types mix (e.g. chemical injection, merging of process streams).

2.4 VIBRATION RELATED ISSUES

2.4.1 Piping Fatigue

Vibration of the pipework causes dynamic stresses which, if above a critical level, can result in the initiation and/or propagation of a fatigue crack. Fatigue cracking, if unchecked, can lead to through thickness fracture and subsequent rupture, refer to **Figure 2-7**. The fatigue life of the component can be relatively short (in some cases minutes or days). However, if the vibration is intermittent the fatigue life of the component can be much longer, depending on the dynamic stress amplitude and frequency of vibration.



Figure 2-7 An example of a fatigue crack, shown by dye penetrant testing

The most fatigue sensitive locations are welded joints associated with main lines and small bore connections. Typically, fatigue failure of small bore connections occurs at the connection with the parent pipe, refer to **Figure 2-7**. However, depending on the local configuration fatigue failures can occur at other weld locations, refer to **Figure 2-8**.

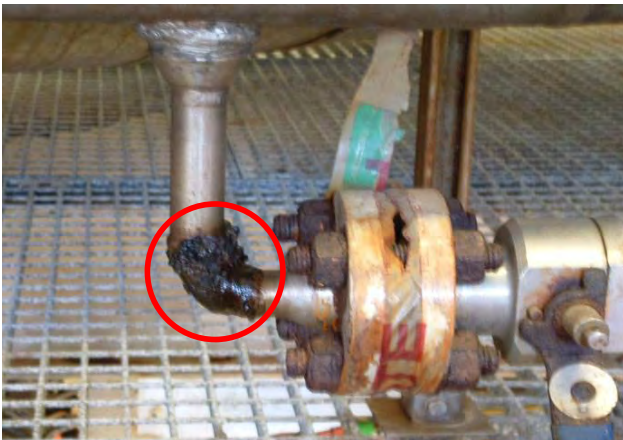


Figure 2-8 An example of a fatigue crack which did not occur at the connection to main line, resulting in a clear leak

2.4.2 Fretting

In addition to fatigue issues, vibration can result in fretting. Fretting occurs between two surfaces in contact subjected to cyclic relative motion, resulting in one or both of the surfaces being worn away, leading to a loss of containment.

3 UNDERTAKING A PROACTIVE ASSESSMENT

3.1 OVERVIEW

The three most common cases for which a proactive assessment is undertaken are:

- i. When a new process system is being designed.
- ii. When undertaking an assessment of an existing plant or process system.
- iii. When changes to an existing plant or process system are being considered (such as operational, process or equipment changes).

Whilst there are a number of common steps to be undertaken in all three cases, the order in which these steps are performed may vary. For example, in the case of a new design the initial emphasis is placed on a 'paper based' assessment during the design phase prior to construction. In this way potential issues are identified early enough such that mitigation measures can be incorporated easily. Other steps, such as visual inspection to identify as-built issues, are only possible once the plant is built.

Conversely, the assessment of an existing plant may start with a visual inspection (supported as necessary by targeted vibration measurements) to identify any immediate integrity threats due to vibration prior to undertaking a paper-based assessment to determine the risk of failure for the complete operating envelope.

The approach adopted for each case is outlined in the following sections as detailed below:

Type of Project	Example(s)	Flowchart
New design	New green/brownfield site or a new process module or unit	3-1
Existing plant	Plant in current operation	3-2
Change to existing plant	Process, piping or equipment change to an existing system	3-3

An overview of the main steps in the assessment process is given in **Section 3.3**.

3.2 RISK ASSESSMENT

3.2.1 Likelihood of Failure

The likelihood of failure (LOF) is a form of scoring 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 necessarily conservative.

The initial focus for the assessment should be those systems which are considered to be safety and/or business critical. Other areas of the plant should subsequently be subjected to an assessment to ensure all potential issues are identified and addressed. The definition of safety and/or business critical is not considered as part of these Guidelines.

3.2.2 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 3-1** 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 corrective actions in **TM-10**, **TM-11** and **TM-12** 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 the vibration excitation mechanisms covered in these Guidelines the failure mechanism is usually fatigue cracking, although failures due to fretting can occur. 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 using methods such as BS 7910, Guide to methods for assessing the acceptability of flaws in metallic structures [3-1].

3.3 MAIN STEPS

3.3.1 Qualitative Assessment (TM-01)

A qualitative assessment is undertaken to (i) identify the potential excitation mechanisms that may exist and (ii) provide a means of rank ordering a number of process systems or units in order to prioritise the subsequent quantitative assessment.

This assessment can be performed at any of the following levels:

- An operating unit
- A major area or functional section in an operating unit
- A system (a major piece of equipment/package or auxiliary equipment)

When working through each item in the qualitative assessment consideration should be given to the complete operating envelope of the plant or system under review. For example, in the case of a compression system several scenarios would typically be considered:

- Full flow (zero recycle)
- Full recycle
- Bypass
- Relief/blowdown

The qualitative assessment for new designs and existing plant provides a likelihood of failure ranking based on High, Medium and Low scores, which may be used with (user supplied) consequence scores to give an overall qualitative assessment of risk. Where any excitation factor results in a “High” or “Medium” score the corresponding excitation mechanisms should be subjected to a quantitative assessment, refer to **TM-02** and **TM-04**. In addition, irrespective of the qualitative assessment score, a visual inspection of the plant should be undertaken to capture any as-built issues, refer to **TM-05** and **TM-06**.

In certain cases (e.g. the design of a new process module which will be tied into an existing system) the effect of the new module on the existing facilities (e.g. in terms of changes to process and/or operating conditions) should also be assessed, refer to **Section 3.1.3**.

Key information required:

- P&IDs
- PFDs
- General knowledge of the plant operation
- Plant history (existing plant/plant change)
- Plant maintenance and corrosion management

3.3.2 Quantitative Main Line LOF Assessment (TM-02)

A quantitative assessment is undertaken for each of the excitation mechanisms identified from the qualitative assessment. This results in an LOF score for each main line in the system, for each identified excitation mechanism. As with the qualitative assessment consideration should be given to the complete operating envelope of the plant or system under review.

In addition, if there is any uncertainty regarding the type of excitation that may apply (including excitation mechanisms not explicitly covered in **TM-02**, e.g. slug flow, environmental loading) then the respective main line should be assigned an LOF=1.

The LOF score for some excitation mechanisms is pipe diameter and wall thickness dependent (e.g. flow induced turbulence). Therefore when working through a typical process system, as pipe diameters and specifications change, different LOF scores may be generated within the same system for the same excitation mechanism.

The typical output of the quantitative main line LOF assessment is therefore a listing of LOF score against line number for each excitation mechanism considered. This also provides a means of rank ordering main lines within a process system based on LOF score.

Note that if any main line has an LOF score greater than 0.5 then a check should be made for vibration transmission to neighbouring pipework, see **Section T2.3**.

The required actions based on main line LOF score are given in **Table 3-1**.

Key information required:

- P&IDs
- PFDs
- More detailed equipment and process information (e.g. valve data sheets, heat mass balance information containing information such as mass flow rates, fluid densities)
- Selected piping isometrics
- General knowledge of the plant operation

3.3.3 Quantitative SBC LOF Assessment (TM-03)

Depending on the main line LOF scores, refer to **Table 3-1**, a quantitative small bore connection LOF assessment may be required. This involves assessing each individual SBC on the main line based on key geometric and location information.

At the design stage there may be insufficient information available to undertake the SBC quantitative assessment, in which case it can only be undertaken once the pipework is fabricated. In addition some SBC pipework is site-run and therefore the only option may be to obtain the necessary geometric data by visual inspection.

Providing the information required is available (which will certainly be the case for an existing plant or at the construction stage of a new design) then each SBC is assigned an LOF value as shown in **Flowchart 3-4**. The main line LOF score is the maximum LOF score of all of the individual excitation mechanisms assessed in **Section 3.3.2**.

It is possible to perform an SBC LOF assessment without having first determined the main line LOF score (i.e. the SBC assessment can be undertaken in isolation); however it should be noted that in this case the main line LOF defaults to 1.0.

The required actions based on the SBC LOF score are given in **Table 3-2**.

In addition if an SBC is on a main line subjected to tonal excitation, coupling between a structural natural frequency of the SBC and the tonal excitation frequency(ies) should be avoided. Tonal excitation is generated by the following excitation mechanisms:

- Mechanical Excitation
- Pulsation: Reciprocating/Positive Displacement Pumps & Compressors
- Pulsation: Rotating Stall
- Pulsation: Flow Induced Excitation

The structural natural frequencies of the SBC should be determined by specialist measurement or predictive techniques, refer to **TM-08** and **TM-09**. Corrective actions where coupling between structural natural frequencies and excitation frequencies occurs are given in **TM-11**.

Key information required:

- Main line LOF from **TM-02** (or default to main line LOF = 1.0)
- SBC geometry and location

3.3.4 Quantitative Thermowell LOF Assessment (TM-04)

If the excitation of thermowells is identified as a potential issue from the qualitative assessment then a quantitative assessment shall be undertaken. The thermowell LOF score is obtained from **TM-04**.

The required actions based on the thermowell LOF score are given in **Table 3-3**.

Key information required:

- Process data
- Thermowell geometry
- Main line schedule

3.3.5 Visual Assessment (TM-05 Piping & TM-06 Tubing)

A visual inspection is required to be undertaken in line with **TM-05** and **TM-06** irrespective of the results of the qualitative and quantitative assessment in order to capture as-built issues and to ensure that any corrective actions have been implemented satisfactorily. For existing operational plant visual inspection also helps identify particular operating conditions of concern.

However, the results of the qualitative and quantitative assessments can be used to prioritise the order in which a visual assessment is undertaken.

3.3.6 Basic Piping Vibration Measurement Techniques (TM-07)

Basic piping vibration measurements provide a first level assessment of the severity of piping vibration for both main lines and SBCs. The methods and criteria given in **TM-07** allow a non-specialist to obtain an initial indication of whether a piping integrity threat exists.

In order to obtain representative data, measurements should be taken at the worst case operating condition identified.

Key information required:

- Process and operating information at time of survey

3.3.7 Specialist Techniques (TM-08 Measurement TM-09 Predictive)

In some situations specialist advice should be sought. There are a number of techniques that can be deployed, encompassing both measurement (**TM-08**) and prediction (**TM-09**).

Certain measurement techniques can be applied during construction or when the plant is not operating which will provide useful information that could not easily be obtained by other means. A typical example would be the determination of structural natural frequencies of pipework and connections that are to be subjected to tonal excitation when the plant is operational.

Other measurement techniques, such as dynamic strain measurement, can be deployed with the plant operational, and used to quantify more accurately whether a fatigue issue exists. Dynamic pressure (pulsation) measurements can quantify the level of excitation in the fluid system, while experimental modal and operating deflection shape analysis can help identify forced and resonant behaviour. Permanently installed monitoring systems can quantify transient vibration or changes to excitation and/or response levels with process or operational changes.

Predictive techniques can provide a further level of quantification of excitation and response levels, and can be used to explore potential modifications. Examples include structural and acoustic finite element analysis, pulsation and surge simulation, and computational fluid dynamics (CFD).

3.3.8 Corrective Actions (TM-10 Main Line, TM-11 SBC, TM-12 Thermowell)

The requirement for corrective actions can be identified from:

- The LOF scores determined for main lines, SBCs and thermowells
- The results of vibration measurements

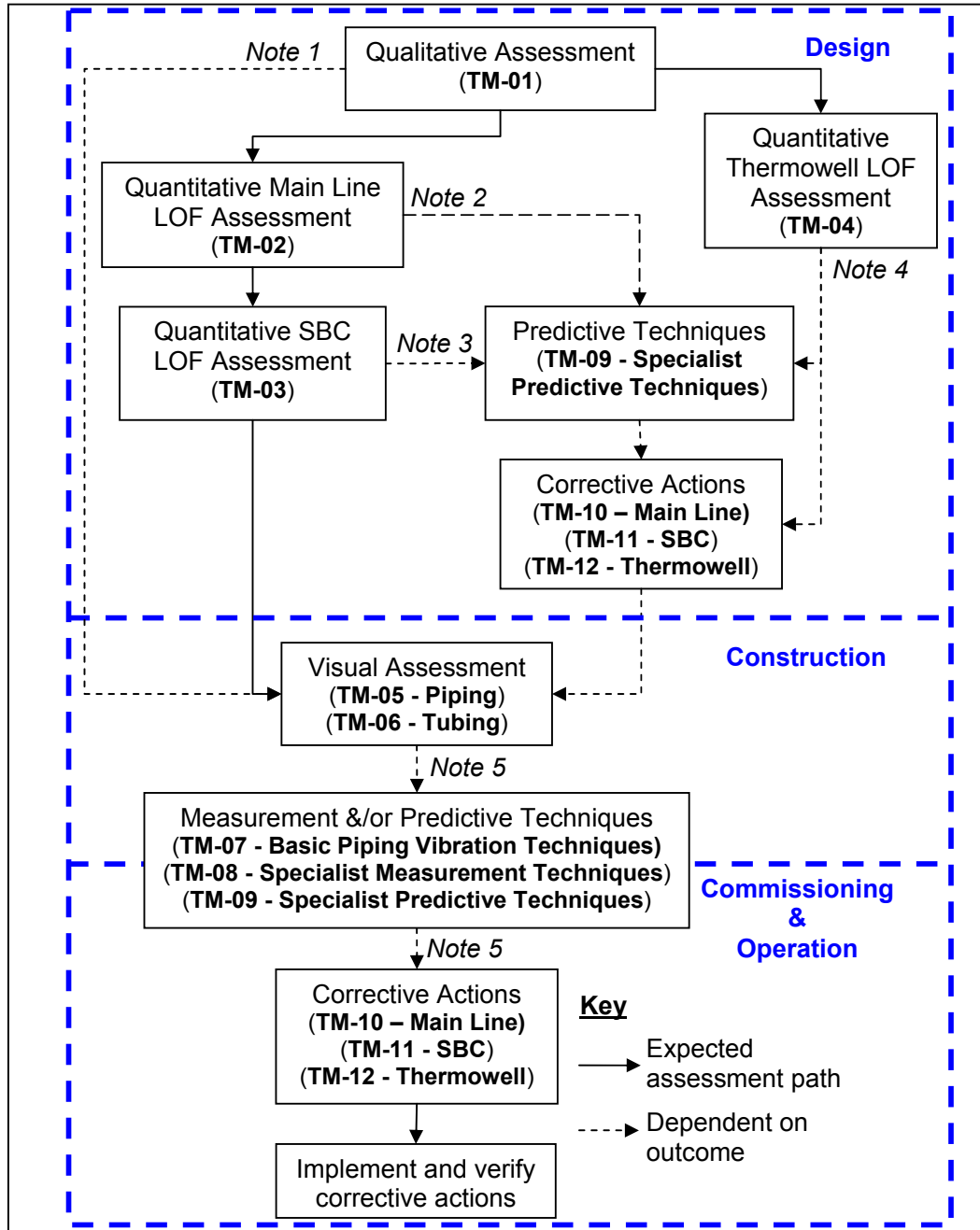
Corrective actions can take a variety of forms, and can affect excitation or response. In most cases it is preferable to reduce the level of excitation wherever practicable. The type of corrective action(s) to be deployed will depend on the dominant excitation mechanism(s) and the type of response. It is therefore important to gain an understanding (either from the quantitative LOF assessment or from direct measurement) of both excitation and response.

3.3.9 Implement and Verify Corrective Actions

The implementation of any corrective actions should be undertaken in a timely manner and verification of these implemented corrective actions should then be promptly undertaken.

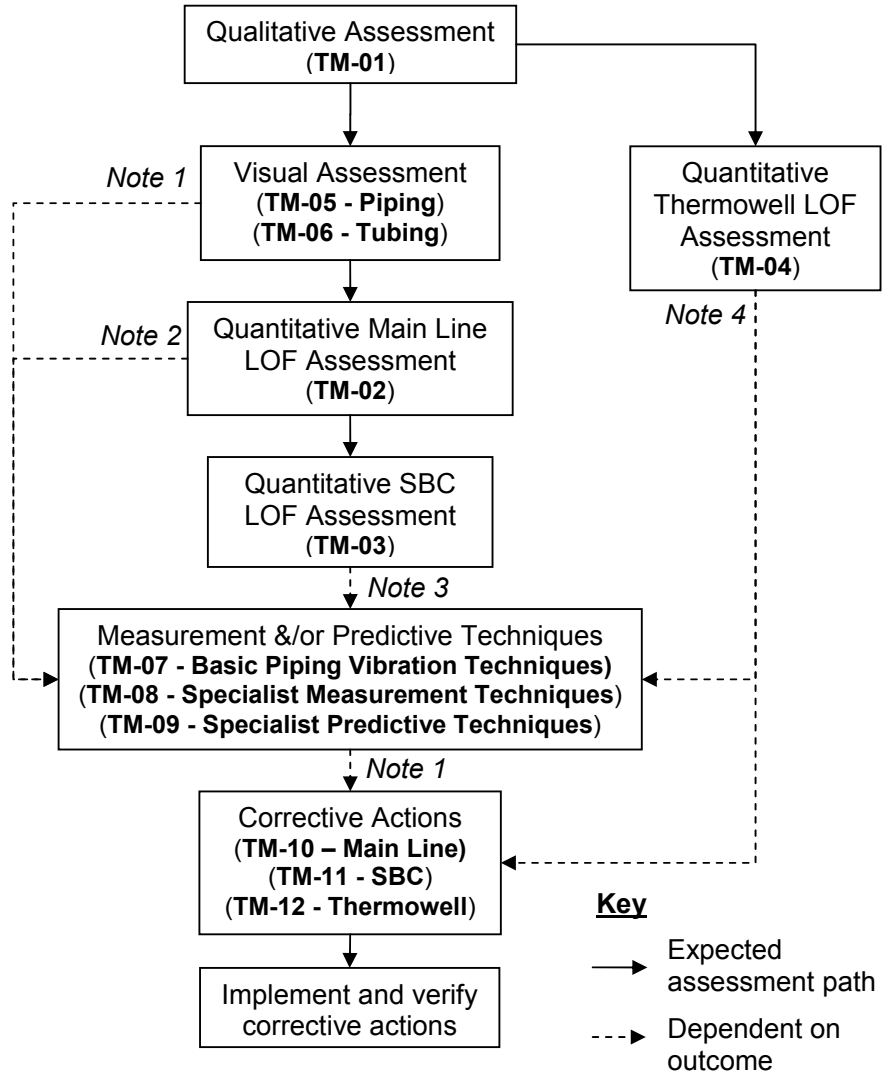
Implementing and verifying corrective actions is a key activity to ensure that any corrective actions have been correctly incorporated and that the resulting vibration levels are acceptable. Verifying activities can include both visual inspection (**TM-05 / TM-06**) and vibration measurements (**TM-07 / TM-08**).

In addition, certain corrective actions require ongoing inspection/maintenance (e.g. bolted braces, pre-charge pressure of gas filled pulsation dampeners) to ensure that they remain effective. This is best addressed by ensuring that such aspects are incorporated into the plant's inspection and maintenance strategy.



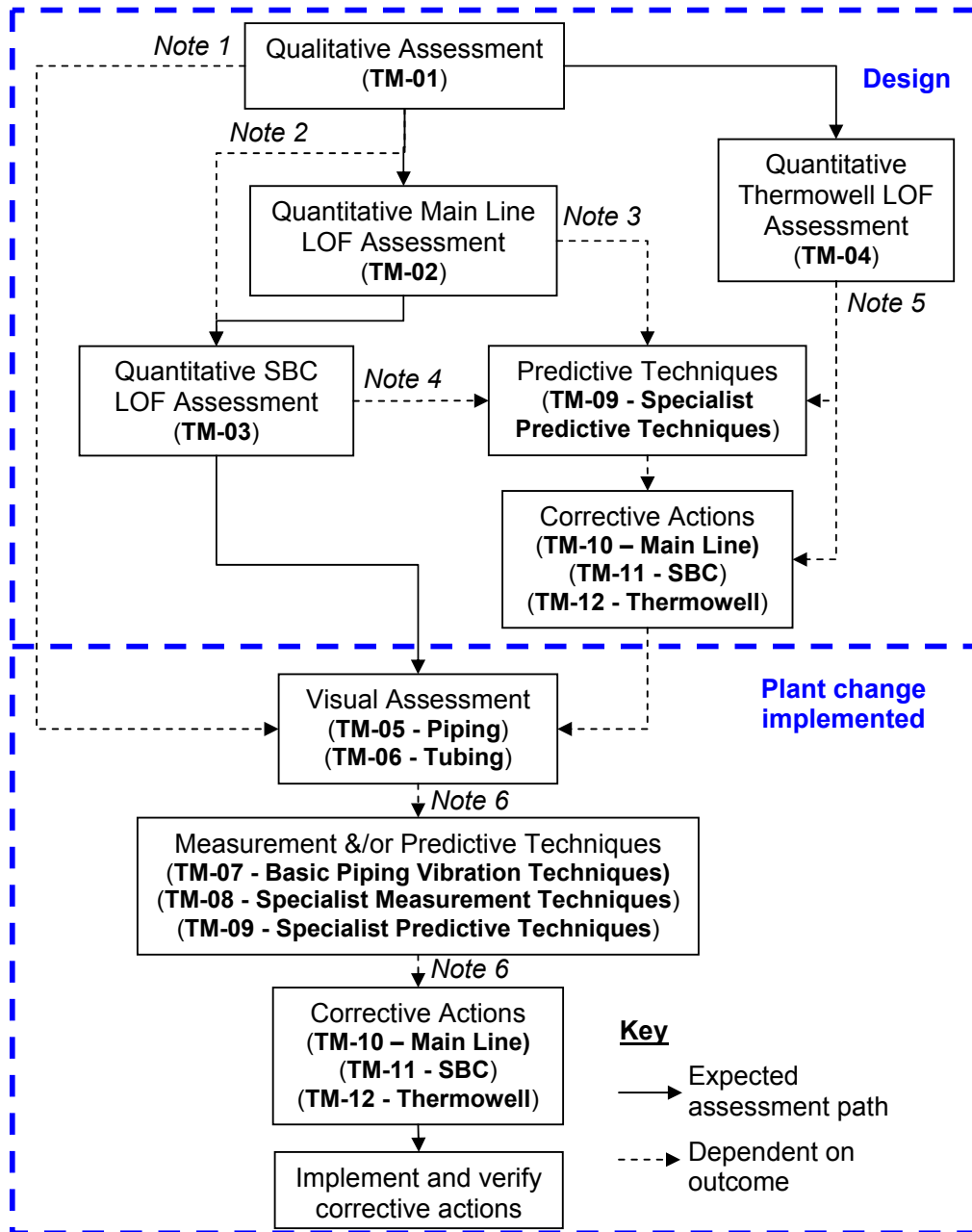
Flowchart 3-1 Proactive Methodology for a New Design

- Note 1** If the qualitative assessment does not indicate any high or medium scores
- Note 2** If the main line qualitative assessment results in a LOF score greater than 0.5
- Note 3** If the SBC qualitative assessment results in a LOF score greater than 0.4
- Note 4** If the thermowell qualitative assessment results in a LOF score of 1.0
- Note 5** If the location is identified to be of concern



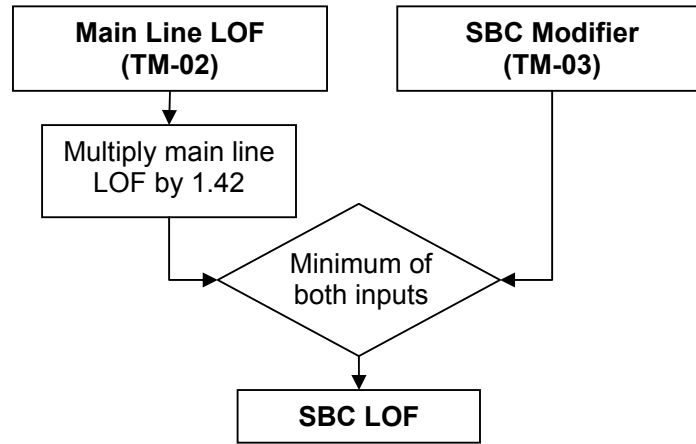
Flowchart 3-2 Proactive Methodology for an Existing Plant

- Note 1* If the location is identified to be of concern
- Note 2* If the main line qualitative assessment results in a LOF score greater than 0.5
- Note 3* If the SBC qualitative assessment results in a LOF score greater than 0.4
- Note 4* If the thermowell qualitative assessment results in a LOF score of 1.0



Flowchart 3-3 Proactive Methodology for Change to Existing Plant

- Note 1* If the qualitative assessment does not indicate any high or medium scores
Note 2 Change only occurs on SBCs
Note 3 If the main line qualitative assessment results in a LOF score greater than 0.5
Note 4 If the SBC qualitative assessment results in a LOF score greater than 0.4
Note 5 If the thermowell qualitative assessment results in a LOF score of 1.0
Note 6 If the location is identified to be of concern



Flowchart 3-4: Determining the SBC LOF Score

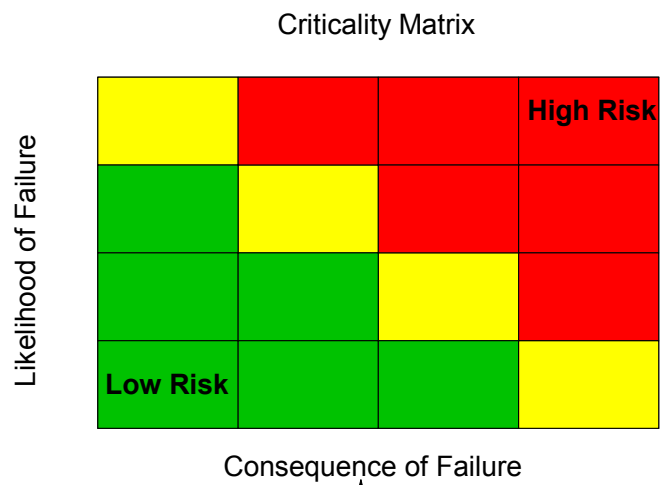


Figure 3-1 Criticality matrix linking likelihood of failure calculation from these Guidelines and consequence of failure from the user

Score	Action	Technical Module
LOF $\geq$ 1.0	The main line shall be redesigned, resupported or a detailed analysis of the main line shall be conducted, and vibration monitoring of the main line shall be undertaken (Note 1) Corrective actions shall be examined and applied as necessary Small bore connections on the main line shall be assessed. A visual survey shall be undertaken to check for poor construction and/or geometry and/or support for the main line and/or potential vibration transmission to neighbouring pipework.	TM-09 TM-07/TM-08 TM-10 TM-03 TM-05 TM-06
1.0 > LOF $\geq$ 0.5	The main line should be redesigned, resupported or a detailed analysis of the main line should be conducted, or vibration monitoring of the main line should be undertaken (Note 1) Corrective actions should be examined and applied as necessary Small bore connections on the main line shall be assessed. A visual survey shall be undertaken to check for poor construction and/or geometry and/or support for the main line and/or potential vibration transmission to neighbouring pipework.	TM-09 TM-07/TM-08 TM-10 TM-03 TM-05 TM-06
0.5 > LOF $\geq$ 0.3	Small bore connections on the main line should be assessed. A visual survey should be undertaken to check for poor construction and/or geometry and/or support for the main line and/or potential vibration transmission from other sources.	TM-03 TM-05 TM-06
LOF < 0.3	A visual survey should be undertaken to check for poor construction and/or geometry and/or support for the main line and/or potential vibration transmission from other sources.	TM-05 TM-06

Table 3-1: Main Line Actions

Note 1 For certain transient vibration mechanisms specialist measurement techniques may be required

Note 2 For the case of high frequency acoustic excitation, this mechanism affects only the main line. The small bore connections on the main line only require assessment if there are other excitation mechanisms affecting the main line

Score	Action	Technical Module
LOF ≥ 0.7	The SBC shall be redesigned, resupported or a detailed analysis shall be conducted, and vibration monitoring of the SBC shall be undertaken A visual survey shall be undertaken to check for poor construction and/or geometry for the SBC's and instrument tubing.	TM-11 TM-07/TM-08 TM-05/TM-06
$0.7 > \text{LOF} \geq 0.4$	Vibration monitoring of the SBC should be undertaken. Alternatively the SBC may be redesigned, resupported or a detailed analysis conducted. A visual survey should be undertaken to check for poor construction and/or geometry for the SBC's and instrument tubing.	TM-07/TM-08 TM-11 TM-05/TM-06
LOF < 0.4	A visual survey should be undertaken to check for poor construction and/or geometry for the SBC's and instrument tubing.	TM-05/TM-06

Table 3-2: SBC Actions

Score	Action	Technical Module
LOF = 1.0	Modify the thermowell or a detailed analysis shall be conducted.	TM-12
LOF = 0.29	No action required	N/A

Table 3-3: Thermowell Actions

4 TROUBLESHOOTING A VIBRATION ISSUE

4.1 IDENTIFYING A VIBRATION ISSUE

On an operating plant there are various signs and indicators that there may be a vibration issue. These include:

- Fatigue failure or damage to plant, on items such as main pipework, small bore connections, instrumentation, connections or braces
- Damage to supports, connections, electrical instruments
- Fretting of pipework and/or associated structures
- Weeping/leaking from instrument tubing
- Loosening of bolts
- Perceived high levels of noise and vibration
- Concern from issues identified on similar plants or units

4.2 APPROACH

When it is thought that there is a potential vibration issue the approach outlined in **Flowchart 4-1** should be followed. The main steps are summarised below.

4.2.1 Review History & Plant Operation

From a good review of the history of the problem and the plant operation a great deal of useful information can be obtained. As part of this process the following should be undertaken where possible:

- Identify location of failures and any similar susceptible locations
- Review failure investigation and/or metallurgical reports
- Correlate operating conditions with high vibration or failure history and identify under what conditions the vibration occurs (e.g. is it steady state, under certain operating conditions, transient in nature)
- Review previous design studies (e.g. compressor/pumps studies considering shaking forces from pulsation)
- Review previous investigations
- Review any available measurement data, considering the frequency content and amplitude

4.2.2 Walkdown

From the walkdown of the plant the following information is being sought:

- A subjective assessment of the type of vibration occurring. For example:
 - Steady state / Transient / Random in nature?
 - Exhibits tonal properties?
 - Is the response subjectively low frequency or high frequency (Note, low frequency vibration involves much greater displacements and often can be seen, whilst higher frequency vibration can be detected by touch)?
 - Are there impact type events?
 - Does the excitation result in high noise levels?
- Identifying where in the pipework system the vibration levels are at a maximum
- Note under which operating conditions maximum vibration occurs

- Consider excitation of connected items (e.g. SBC, instruments, tubing)
- Note condition of supports (e.g. damage, loosening, ineffective)

TM-05 (Visual Inspection - Piping) and **TM-06** (Visual Inspection - Tubing) provides guidance of items to consider during the walkdown.

4.2.2.1 Information From Plant Operators

Due to the effect that operating conditions of the plant have on the excitation mechanisms and subsequent vibration it is important to record the plant operating conditions to assist with assessing the potential vibration issue. Where appropriate it is also important to note the operating conditions when there is little or no vibration. Details of the information that should be collected are given in **Table 4-1**.

4.2.2.2 Perceived Vibration Levels

If at any time there is concern over the perceived vibration levels then basic vibration measurements should be undertaken when the vibration is relatively steady state. The line should be inspected under the range of operating conditions and the relevant information recorded as detailed in **Table 4-1**.

If the perceived vibration levels are not of concern then the pipework should be kept under regular review.

4.2.3 Basic Vibration Measurement/ Preliminary Acceptance Criteria

Details of basic measurement techniques and assessment criteria are given in **TM-07**. Measurements should be undertaken under the operating conditions for which the concern was noted.

If the vibration level is in excess of the “Problem” criterion then there is a high risk of fatigue damage occurring. In this case short term vibration control measures should be immediately implemented (refer to **Section 4.2.4**) and specialist advice sought.

A vibration level in excess of the “Concern” criterion means that there is the potential for fatigue damage occurring and therefore specialist advice should be sought.

If the vibration level lies in the “Acceptable” criterion the pipework should be periodically reviewed to ensure that under different operating conditions the vibration levels remain at an “Acceptable” level.

In the case of high frequency (typically greater than 300Hz) or transient (i.e. non steady state) vibration, the basic vibration measurement method given in **TM-07** is not appropriate and more sophisticated measurement techniques are required, refer to **TM-08**.

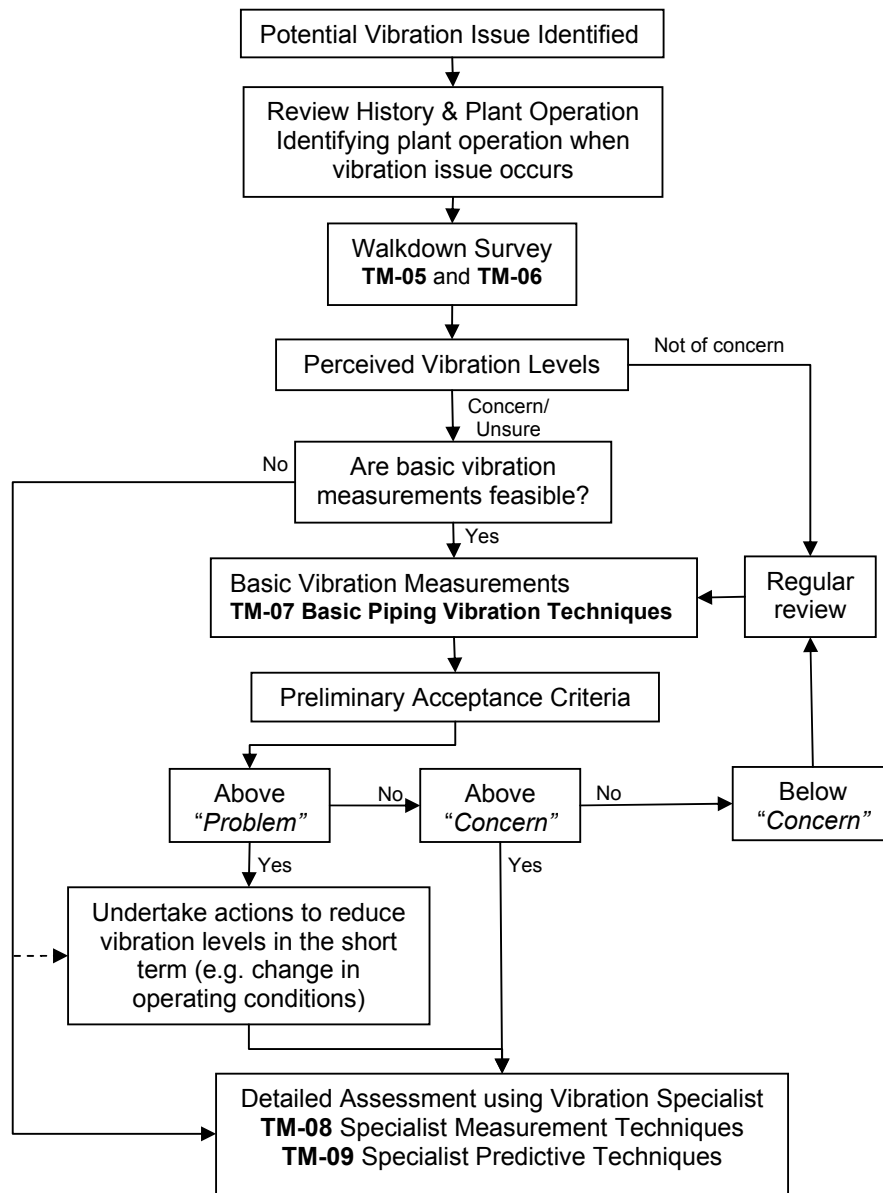
4.2.4 Short Term Measures to Reduce Vibration

From the review of the plant history and operational data the conditions at which the problem levels of vibration occur should be known. Using this information one short term measure is to reduce the level of vibration by altering the operation of the plant. In addition, if a serious problem exists, then consideration should be given to a more detailed assessment and the use of more specialist techniques (see **TM-08** and **TM-09**). An inspection of all supports should be undertaken, referring to **TM-05**, to ensure that they are all effective. In other cases installation of temporary supports can be of value, however the vibration response should be understood sufficiently to ensure that the modification will not result in further problems.

4.2.5 Regular Review

Many vibration excitation mechanisms are affected by the plant operating conditions. Therefore, at the time of inspection and/or measurement, the plant may not be exhibiting its worst vibration levels. Therefore, the locations where potential vibration issues have been identified should be kept under regular review to ensure the vibration condition remains in acceptable limits.

This can be undertaken either by routine visual inspection or routine measurement of vibration levels. Items to be noted are changes in amplitude, frequency and characteristic of vibration. Where changes occur details of the operating conditions should be made, refer to **Table 4-1**. As the response of pipework is often dominated by the changes in the process conditions, the pipework should be reviewed so the full operating envelope is considered.



Flowchart 4-1 Overview of piping vibration troubleshooting

Item	Description
By	• The person identifying the vibration
Details of Concern	• Description of concern • Photos of the area of interest
Location Identification	• Line number • P&ID number • Process Fluids
Operating Condition(s)	• Process data summary pertaining to the conditions in the line or system at the time the problem was experienced. Including: Pressure; Temperature; Fluid density; Flow rate; Machinery operations; Valve position/change. This information could come in the form of a print out from the process control system. • Date/Time (when vibration observed)
Historical Information	• Details of any previous failures/concerns raised on this system, where appropriate • Details of any previous work undertaken on this system, where appropriate • Details of any process changes on the system, where appropriate

Table 4-1 Information to be recorded about potential vibration issues

Technical module

T1 - QUALITATIVE ASSESSMENT

T1.1 GENERAL

This section describes a qualitative method for determining a likelihood of failure (LOF) to provide a basis for identifying potential threats and prioritising a more formal (quantitative) assessment. It provides:

- i. The identification of those excitation mechanisms which may give rise to a vibration induced fatigue failure and which should then be subjected to a quantitative assessment.
- ii. A means of prioritising the formal assessment of a process plant for a new design or an existing plant. This is particularly useful when a number of systems or process units are being assessed.
- iii. A method for identifying potential piping vibration issues which may arise when changes are being implemented on an existing plant.

The methodology is dependent on whether an assessment is to be undertaken on a new design, an existing plant, or as part of a change to an existing plant as follows:

For a new design or an existing plant the methodology takes into account an assessment of the possible sources of excitation and certain plant operation and condition factors. For a change to an existing plant the methodology focuses on identifying potential issues. For each of these types of project the methodology is slightly different and is explained in more detail in **Sections T1.2 to T1.4**.

Type of Project	Example(s)	Refer to Section
New design	New green/brownfield site or a new process module or unit	T1.2
Existing plant	Plant in current operation	T1.3
Change to existing plant	Process, piping or equipment change to an existing system	T1.4

This assessment can be performed at any of the following levels:

- An operating unit
- A major area or functional section in an operating unit
- A system (a major piece of equipment/package or auxiliary equipment)

T1.2 NEW DESIGN

This section addresses the situation of a new green/brownfield site or a new process module or unit.

Due consideration should be given to any previous work undertaken or experience gained on identical sister plants or on parallel process modules to determine any lessons learnt and the associated corrective actions that have been put in place.

The main focus of this assessment should be those systems which are considered to be safety and/or business critical. Other areas of the plant should subsequently be subjected to a similar assessment to ensure all potential problem areas are identified.

Items in **Table T1-1** identify the significant potential excitation factors, whilst the items in **Table T1-2** consider certain condition and operational factors which may have an influence with respect to vibration induced fatigue. Guidance notes for each item are included in **Table T1-3** and **T1-4**.

An overview of how the different factors are combined is given in **Flowchart T1-1**. The eleven excitation factors (each scoring “High”, “Medium” or “Low”) and the maximum of the condition and operational factors (resulting in a single score of “High”, “Medium” or “Low”) are added together to give a total number of “High”, “Medium” and “Low” scores (twelve in total). The final result is used in two ways:

- i. To identify the principal excitation factors of concern. Where any excitation factor results in a “High” or “Medium” score the corresponding excitation mechanisms should be subjected to a quantitative assessment, refer to **TM-02** and **TM-04**.
- ii. When a number of different operating units/major areas/systems are subjected to separate qualitative assessments, to prioritise the order in which the subsequent quantitative assessment should be undertaken.

T1.3 EXISTING PLANT

This section addresses the situation where an operator wishes to undertake a formal risk assessment for piping vibration on an existing plant to determine whether there is potential for a vibration related fatigue failure to occur.

Due consideration should be given to any previous work that has been undertaken to assess piping vibration issues and any corrective actions that have been put in place.

The approach for an existing plant is the same as that for a new design (refer to **Section T1.2**). The significant differences between the new design and an existing plant assessment are:

- for an existing plant a visual inspection is undertaken early in the assessment process to capture any as-built issues
- for an existing plant item 10 on **Table T1-1** considers the actual plant operating history.

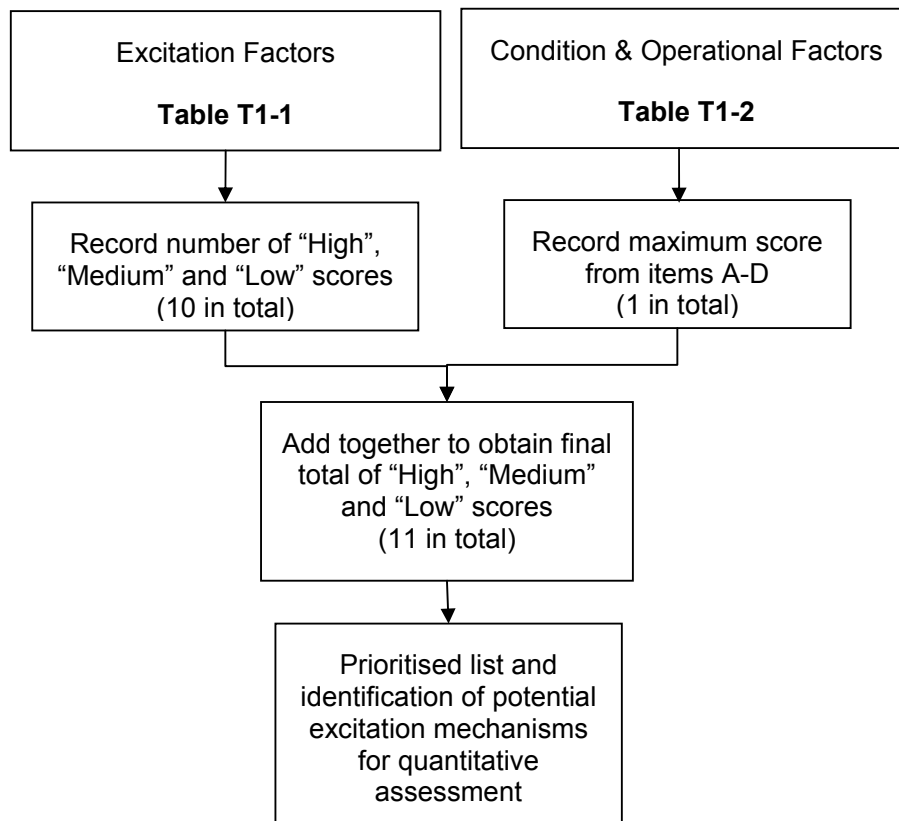
T1.4 CHANGE TO EXISTING PLANT

This section addresses the situation where there is a process, piping or equipment change to an existing system and can be used as part of the HAZID/HAZOP process.

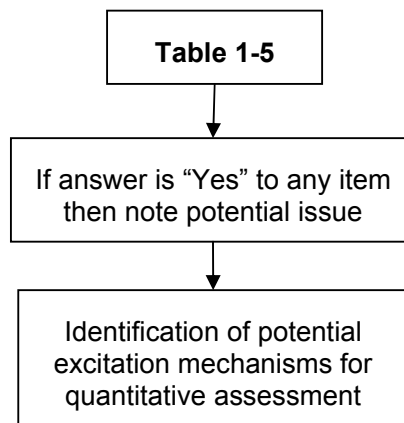
It is assumed that the existing pipework has already been assessed for vibration induced fatigue (**Section T1.3**) and that any existing vibration issues have already been addressed, with suitable mitigation measures in place.

The items in **Table T1-5** identify which process, piping or equipment changes require consideration with regard to vibration induced fatigue. Guidance notes for each question are included in **Table T1-6**.

An overview of the qualitative assessment procedure is given in **Flowchart T1-2**.



Flowchart T1-1 Qualitative Assessment for a New Design or an Existing Plant



Flowchart T1-2 Qualitative Assessment for Change to Existing Plant

T1 – QUALITATIVE ASSESSMENT

Item	Aspect	Applicable process fluid(s)	Likelihood Classification			Potential excitation mechanism(s)
			Low	Medium	High	
1	What is the maximum value of kinetic energy (ρv^2) of the process fluid within the system under consideration?	All	$\rho v^2 < 5,000 \text{ kg/m s}^2$	between $5,000 \leq \rho v^2 < 20,000 \text{ kg/m s}^2$	$\rho v^2 \geq 20,000 \text{ kg/m s}^2$	Flow induced turbulence (All fluids) refer to Section T2.2 Flow induced pulsation (Gases only) refer to Section T2.6
2	Is choked flow possible or are sonic flow velocities likely to be encountered?	Gas	No		Yes	High frequency acoustic excitation refer to Section T2.7
3	Is there any rotating or reciprocating machinery?	All	No	rotating equipment only	reciprocating equipment	Mechanical excitation refer to Section T2.3
4	Are there any positive displacement pumps or compressors?	All	No	Screw/gear type positive displacement machine	reciprocating type positive displacement machine	Pulsation - reciprocating refer to Section T2.4
5	Are there any centrifugal compressors which have the potential to operate under rotating stall conditions?	Gas	No	Compressor has stall characteristics but operational restraints in place to ensure that rotating stall is not encountered	Stall rotating condition unknown. Compressor has rotating stall characteristics and may operate at conditions that will give rise to stall conditions	Pulsation - rotating stall refer to Section T2.5

Table T1-1 Excitation Factors for a New Design or an Existing Plant (part 1 of 2)

T1 – QUALITATIVE ASSESSMENT

Item	Aspect	Applicable process fluid(s)	Likelihood Classification			Potential excitation mechanism(s)
			Low	Medium	High	
6	Are there any systems which may exhibit flashing or cavitation?	Liquid / Multiphase	No		Yes	Cavitation and Flashing refer to Section T2.9
7	Are there any systems with fast acting opening or closing valves?	All	No		Yes	Surge/ Momentum changes (refer to Section T2.8)
8	Are there intrusive elements in the process stream?	Gas/Liquid	No		Yes	Vortex shedding from intrusive elements to refer to TM-04
9	Is there a possibility of slug flow?	Multiphase	No		Yes	Slug flow - seek specialist advice
10	Is there a history of pipework vibration issues, or are there any systems which are similar to those on another plant which have a known history of pipework vibration issues?	All	No	Yes: however, suitable corrective action in place and validated for the complete operating envelope.	Yes	Known vibration refer to Chapter 4

Table T1-1 Excitation Factors for a New Design or an Existing Plant (part 2 of 2)

T1 – QUALITATIVE ASSESSMENT

Item	Aspect	Applicable process fluid(s)	Likelihood Classification			Contributory factor
			Low	Medium	High	
A	What is the quality of construction?	All	Better than industry standards	At industry standard	Below industry standards	Build quality
B	What is the effectiveness of the plant maintenance programme (including corrosion management)?	All	Better than industry standards	At industry standard	Below industry standards	Corrosion/ maintenance management
C	Are there any cyclical operations (e.g. batch operation)?	All	No		Yes	Cyclical loading
D	What is the number of unplanned process interruptions in an average year? (this is intended for normal continuous process operations)	All	0-1	2-8	9 or more	Process upsets

Table T1-2 Condition and Operational Factors for a New Design or an Existing Plant

Item	Guidance Notes
1	For gas, liquid or multiphase systems, higher fluid velocity and/or fluid density increases the level of turbulent energy in the system, and therefore increases the potential for a piping vibration issue. In addition, for a gas system, higher fluid velocity and/or fluid density increases the amplitude of the shaking forces generated by flow induced pulsations. For a liquid system, higher fluid velocity and/or fluid density increases the surge pressure likely to be experienced when a valve is shut. In some situations the highest value of ρv^2 may not be associated with any of the streams given in a Process Flow Diagram. For example, flow through a recycle, bypass or relief line, whilst not considered in the PFD, may give rise to high levels of process fluid kinetic energy. If there is any doubt (and particularly if none of the process streams given on the PFD have a value greater than 5000 kg/m.s²), then a check should be made on those systems which operate intermittently.
2	Choked flow and/or sonic velocities can result in high levels of high frequency acoustic excitation and the formation of shock waves downstream of the pressure reducing device. This can lead to high levels of high frequency piping vibration and stress (often referred to as “acoustic fatigue”).
3	Piping systems associated with, or in close proximity to, reciprocating and rotating machinery can experience piping vibration issues due to potentially high levels of mechanical excitation (particularly reciprocating machines). 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 the maximum length of the skid.
4	Positive displacement pump and compressor systems often experience piping vibration issues due to pulsation in the process fluid; pulsation issues are also sometimes experienced with screw type compressors.
5	Operating a centrifugal compressor at low flow conditions increases the possibility of inducing rotating stall. Rotating stall will induce pressure pulsations in the fluid system, leading to potentially high levels of piping vibration. Note: severe vibration may be generated by operating close to a compressor’s surge line, depending on how the anti-surge control has been configured. It is assumed here that the anti-surge control is effective in limiting the severity of any potential compressor surge condition.

Table T1-3 Guidance Notes for **Table T1-1** (part 1 of 2)

Item	Guidance Notes
6	Flashing will result in potentially high levels of unsteady transient vibration generated by the sudden volume change when a fluid changes from the liquid to vapour state. Similarly, cavitation will result in high levels of vibration due to the formation and instantaneous collapse of innumerable tiny voids or cavities within a liquid where the pressure rises above the vapour pressure of the liquid. Consideration should be given to systems where there are discrete pressure drops which may cause the system pressure to be close to the liquid vapour pressure (e.g. valves, orifice plates, pumps, different fluid streams which combine). In addition, consideration should be given to situations where the fluid temperature increases, which would increase the vapour pressure of the liquid and therefore make it more likely that flashing or cavitation could occur.
7	Fast closure of a valve on a liquid system may generate excessive surge pressures which can generate high levels of transient vibration and/or exceed the flange rating of the pipe. Fast opening valves (e.g. fast acting protection devices) can give rise to large changes in fluid momentum leading to high transient forces. All manually operated valves can be excluded. Typical automatic valves that need to be considered in the assessment include: Fast closing valves (liquid/multi-phase systems only): • Emergency Shut Down Valves (ESD) • Flow Control Valves (FCV) • Pressure Control Valve (PCV) Fast opening valves (gas/liquid and multi-phase systems): • Blow Down Valves (BDV) • Relief Valves (RV)
8	Intrusive elements, such as thermowells, can be a source of vortex induced vibration, leading to failure of the intrusive element.
9	Slug flow may result in potentially high levels of unsteady transient vibration. Due to the complexity of the issue it is recommended that specialist advice is sought and a SBC assessment is undertaken following the assessment method in TM-03 .
10	Are there any systems which are of similar design to others already in operation for which there is a history of fatigue failures and/or high vibration and noise noted previously? If such issues have been identified in the past then has an investigation been undertaken to identify the cause(s) and have corrective actions been recommended? If so, have these actions been implemented correctly and verified for the complete operating envelope? Have the lessons learnt been incorporated in the new design?

Table T1-3 Guidance Notes for **Table T1-1** (part 2 of 2)

Item	Guidance Notes
A	Poor quality construction can have a detrimental effect on the fatigue resistance of a piping system.
B	Poor corrosion management and/or poor maintenance practices can exacerbate vibration induced fatigue issues.
C	Will there be a repeating operation cycle (e.g. a batch process) that could lead to many repetitions of fluctuating flow or pressure? This may lead to periods of high amplitude dynamic loading of the pipework.
D	The number of planned or unplanned process interruptions in an average year? (this is intended for normal continuous process operations). This may lead to short duration but high amplitude dynamic loading of the pipework.

Table T1-4 Guidance Notes for **Table T1-2**

Item	Description	If Yes - Potential Issues
1	Will the modification result in one or more of the following: - An increase in flow velocities by more than 5% over previous operational experience? - An increase in fluid density by more than 10% over previous operational experience?	- Flow induced turbulence (all fluids), refer to Section T2.2 - Flow induced pulsation (gases systems only), refer to Section T2.6 - Vortex shedding from intrusive elements (all fluids), refer to TM-04 - Surge/Momentum Change refer to Section T2.8
2	For a gas system, will the modification result in one or more of the following: - A change in the molecular weight of the gas by more than $\pm 5\%$ from previous maximum/minimum operational experience? - A change to the temperature of the gas by more than $\pm 5\%$ from previous maximum/minimum operational experience? - A change to the ratio of specific heats (C_p/C_v) of the gas by more than $\pm 5\%$ from previous maximum/minimum operational experience?	For all systems: - Pulsation - Flow induced excitation, refer to Section T2.6 If there is a centrifugal compressor: - Pulsation - rotating stall (gas systems only) refer to Section T2.5 If there is a reciprocating compressor: - Pulsation – reciprocating compressor (gas systems only) refer to Section T2.4
3	For a liquid system incorporating a reciprocating / positive displacement pump, will the modification result in one or more of the following: - A change in the density of the liquid by more than $\pm 5\%$ from previous maximum/minimum operational experience? - A change to the bulk modulus of the liquid by more than $\pm 5\%$ from previous maximum/minimum operational experience?	Pulsation – reciprocating /positive displacement pump (liquid systems only) refer to Section T2.4

Table T1-5 Potential Issues for Changes to Existing Plant (part 1 of 3)

Item	Description	If Yes - Potential Issues
4	Will the modification result in a change to the operational configuration of a positive displacement compressor or pump which is outside existing operational experience e.g.: • The use of a second compressor/pump in tandem? • The use of compressor/pump recycle or partial unloading of the compressor?	• Pulsation – reciprocating /positive displacement compressor or pump (liquid and gas systems only) refer to Section T2.4
5	Will the modification result in a centrifugal compressor being operated at low flow conditions?	• Pulsation - rotating stall (gas systems only) refer to Section T2.5
6	Will the modification result in choked flow and/or sonic velocities in the pipework?	• High frequency acoustic excitation (gas systems only) refer to Section T2.7
7	Will the modification result in flashing or cavitation?	• Cavitation and Flashing refer to Section T2.9
8	Will the modification result in a change or addition to the existing pipework or associated equipment (valves, machinery or intrusive elements such as thermowells) which is not a like-for-like replacement?	For changes to valves (including change of valve type or changes to valve closing timings) check for: • Surge/Momentum Change refer to Section T2.8 For changes to machinery check for: • Mechanical excitation refer to Section T2.3 For changes to thermowells check for: • Vortex shedding from intrusive elements refer to TM-04 For changes to pipework, supports, small bore connections and tubing check for: • Poor geometry refer to TM-05 and TM-06

Table T1-5 Potential Issues for Changes to Existing Plant (part 2 of 3)

Item	Description	If Yes - Potential Issues
9	Will the modification result in slug flow?	• Slug flow - seek specialist advice

Table T1-5 Potential Issues for Changes to Existing Plant (part 3 of 3)

Item	Guidance Notes
1	For gas, liquid or multiphase systems, increasing fluid velocity and/or fluid density increases the level of turbulent energy in the system, and therefore increases the potential for a piping vibration issue. In addition, for a gas system, increasing the fluid velocity and/or fluid density increases the amplitude of the shaking forces generated by flow induced pulsations. For a liquid system, increasing the fluid velocity and/or fluid density increases the surge pressure likely to be experienced when a valve is shut. Increasing fluid velocities also potentially affect vortex induced vibration of intrusive elements.
2	Changes to gas temperature, molecular weight or ratio of specific heats will change the speed of sound in the gas. This will change the acoustic natural frequencies of the gas system, and may result in resonant behaviour leading to high levels of pressure pulsation.
3	Changes to liquid density or bulk modulus will change the speed of sound in the liquid. This will change the acoustic natural frequencies of the liquid system, and may result in resonant behaviour leading to high levels of pressure pulsation.
4	Changing the operational configuration of one or more positive displacement compressors or pumps can result in changes to the pressure pulsations in the system due to the changes in flow induced damping or the phasing between machines.
5	Operating a centrifugal compressor at low flow conditions increases the possibility of inducing surge and rotating stall. Rotating stall will induce pressure pulsations in the fluid system, leading to potentially high levels of piping vibration. Note: severe vibration may be generated by operating close to a compressor's surge line, depending on how the anti-surge control has been configured. It is assumed here that the anti-surge control is effective in limiting the severity of any potential compressor surge condition.
6	Choked flow and/or sonic velocities can result in high levels of high frequency acoustic excitation and the formation of shock waves downstream of the pressure reducing device. This can lead to high levels of high frequency piping vibration and stress (often referred to as "acoustic fatigue").
7	Flashing will result in potentially high levels of unsteady transient vibration generated by the sudden volume change when a fluid changes from the liquid to vapour state. Similarly, cavitation will result in high levels of vibration due to the formation and instantaneous collapse of innumerable tiny voids or cavities within a liquid where the pressure rises above the vapour pressure of the liquid. Consideration should be given to changes in systems which result in discrete pressure drops which may cause the system pressure to be close to the liquid vapour pressure (e.g. valves, orifice plates, pumps, different fluid streams which combine). In addition, consideration should be given to situations where the fluid temperature increases, which would increase the vapour pressure of the liquid and therefore make it more likely that flashing or cavitation could occur.

Table T1-6 Guidance Notes for **Table T1-5**

Item	Guidance Notes
8	A direct like-for-like replacement (e.g. of a pipe spool) would not be expected to give rise to a problem. However, changes to the piping (diameter, wall thickness), support arrangement, small bore connections, intrusive elements or equipment such as valves and machinery may affect the vibration excitation or response.
9	Slug flow may result in potentially high levels of unsteady transient vibration. Due to the complexity of the issue it is recommended that specialist advice is sought and a SBC assessment is undertaken following the assessment method in TM-03 .

Table T1-6 Guidance Notes for **Table T1-5**

Technical module

T2 - QUANTITATIVE MAIN LINE LOF ASSESSMENT

T2.1 GENERAL

For each of the excitation mechanisms identified as potentially being an issue (refer to **TM-01**) an LOF value is calculated using the methods detailed in the following sections:

Excitation Mechanism	Section
Flow Induced Turbulence	T2.2
Mechanical Excitation	T2.3
Pulsation: Reciprocating/Positive Displacement Pumps & Compressors	T2.4
Pulsation: Rotating Stall	T2.5
Pulsation: Flow Induced Excitation	T2.6
High Frequency Acoustic Excitation	T2.7
Surge/Momentum Changes Due to Valve Operation	T2.8
Cavitation and Flashing	T2.9

In each section advice is provided on the extent of the assessment and the LOF calculation.

Sample input parameters have been provided in **Appendix B**. These should be used if actual values cannot be easily obtained.

T2.2 FLOW INDUCED TURBULENCE

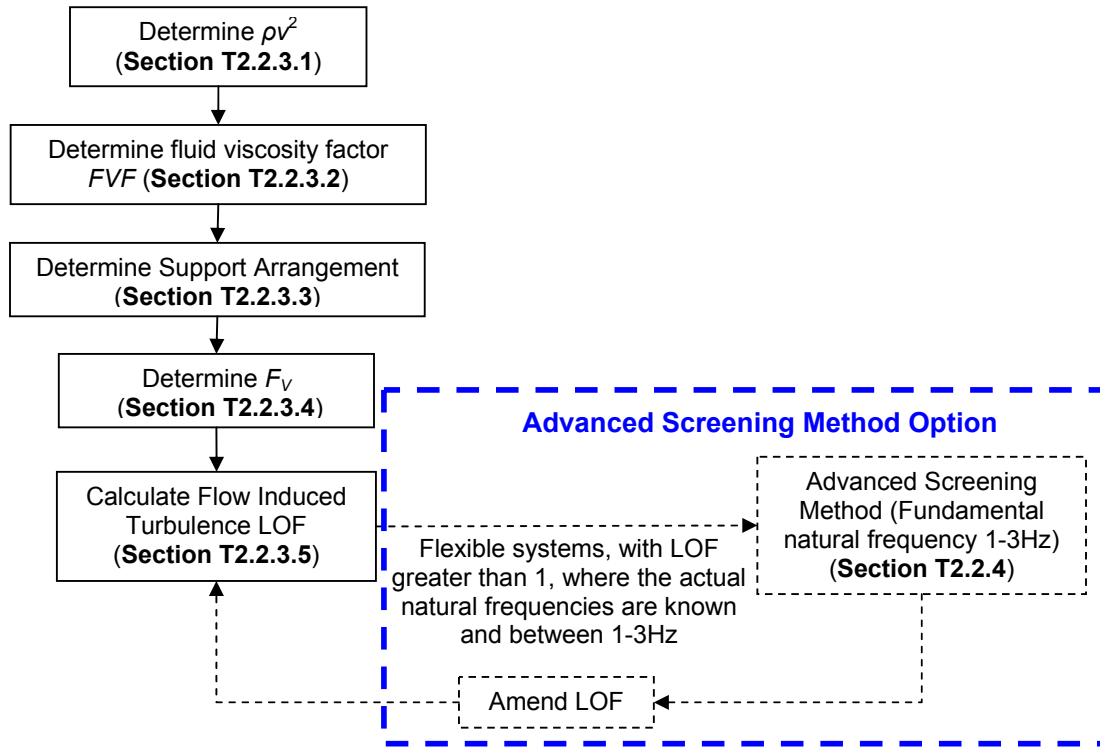
T2.2.1 Extent of Excitation

Turbulent energy is generated by fluid flow. Therefore, the extent of the assessment is limited to those main lines containing flowing fluid.

T2.2.2 Input

Input	Symbol	Units	Comment
External Pipe Diameter	D_{ext}	mm	
Structural natural frequencies	f_n	Hz	Used for “Advanced Screening Method” only
Maximum Span Length between supports on line of interest	L_{span}	m	Refer to Appendix B for definition
Wall thickness of main pipe	T	mm	
Fluid velocity	v	m/s	
Gas dynamic viscosity	μ_{gas}	Pa.s	Required for gas systems only (Refer to Appendix B for typical values)
Fluid Density	ρ	kg/m ³	

T2.2.3 Standard Assessment for Flow Induced Turbulence



Flowchart T2-1 Flow Induced Turbulence assessment for a given line

T2.2.3.1 Determining ρv^2

Calculate ρv^2 using the following equations depending on whether the fluid is single phase or multi-phase flow:

For single phase flow:

$$\rho v^2 = (\text{actual density}) \times (\text{actual velocity})^2 \quad (1)$$

For multi-phase flow:

$$\rho v^2 = (\text{effective density}) \times (\text{effective velocity})^2 \quad (2)$$

where:

$$\text{effective density} = \text{total mass flow rate} / \text{total volumetric flow rate} \quad (3)$$

$$\text{effective velocity} = \text{total volumetric flow rate} / \text{pipe internal cross sectional area} \quad (4)$$

And

$$\text{total mass flow rate} = \sum (\text{actual volumetric flow rate for each phase}) \times (\text{phase density}) \quad (5)$$

$$\text{total volumetric flow rate} = \sum (\text{actual volumetric flow rate for each phase}) \quad (6)$$

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

T2.2.3.2 Determining Fluid Viscosity Factor (FVF)

The amount of turbulent energy partially depends upon the fluid viscosity. This is taken into account by the Fluid Viscosity Factor (*FVF*).

For liquid and multi-phase fluids the *FVF* is equal to one.

To determine the *FVF* for a gas system the dynamic viscosity (μ_{gas}) is required. Examples of some common process gases under a pressure 500psi (35barg) of the dynamic viscosity (μ_{gas}) can be found in **Appendix B**.

The *FVF* for a gas system is calculated by:

$$FVF = \frac{\sqrt{\mu_{\text{gas}}}}{\sqrt{1 \times 10^{-3}}} \quad (7)$$

T2.2.3.3 Determining Support Arrangement

Support arrangement is determined using **Table T2-1**:

Support Arrangement	Span Length Criteria	Typical Fundamental Natural Frequency
Stiff	$L_{\text{span}} \leq -1.2346 * 10^{-5} D_{\text{ext}}^2 + 0.02 D_{\text{ext}} + 2.0563$	14 to 16 Hz
Medium Stiff	$L_{\text{span}} > -1.2346 * 10^{-5} D_{\text{ext}}^2 + 0.02 D_{\text{ext}} + 2.0563$ $L_{\text{span}} \leq -1.1886 * 10^{-5} D_{\text{ext}}^2 + 0.025262 D_{\text{ext}} + 3.3601$	7 Hz
Medium	$L_{\text{span}} > -1.1886 * 10^{-5} D_{\text{ext}}^2 + 0.025262 D_{\text{ext}} + 3.3601$ $L_{\text{span}} \leq -1.5968 * 10^{-5} D_{\text{ext}}^2 + 0.033583 D_{\text{ext}} + 4.429$	4 Hz
Flexible	$L_{\text{span}} > -1.5968 * 10^{-5} D_{\text{ext}}^2 + 0.033583 D_{\text{ext}} + 4.429$	1 Hz

Table T2-1 Support Arrangement

Details of how the maximum span length (L_{span}) is determined and other important aspects such as the significance of supports, are given in **Appendix B.1**.

Alternatively the fundamental natural frequency of the line can be assessed by analytical or measurement techniques to determine the support arrangement.

Note: 'Flexible Support Arrangement' is 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. In this case the Advanced Screening Method should be considered, refer to **Section T2.2.4**.

T2.2.3.4 Determining Flow Induced Vibration Factor F_v

The Flow Induced Vibration factor F_v is determined using **Table T2-2**:

Support Arrange-ment	Range of Outside Diameter	F_v	α	β
Stiff	60 mm to 762 mm	$\alpha \left(\frac{D_{ext}}{T} \right)^\beta$	$446187 + 646 D_{ext} + 9.17 \times 10^{-4} D_{ext}^3$	$0.1 \ln(D_{ext}) - 1.3739$
Medium Stiff	60 mm to 762 mm	$\alpha \left(\frac{D_{ext}}{T} \right)^\beta$	$283921 + 370 D_{ext}$	$0.1106 \ln(D_{ext}) - 1.501$
Medium	273. mm to 762 mm	$\alpha \left(\frac{D_{ext}}{T} \right)^\beta$	$150412 + 209 D_{ext}$	$0.0815 \ln(D_{ext}) - 1.3269$
Medium	60 mm to 219 mm	$\exp \left[\alpha \left(\frac{D_{ext}}{T} \right)^\beta \right]$	$13.1 - 4.75 \times 10^{-3} D_{ext} + 1.41 \times 10^{-5} D_{ext}^2$	$-0.132 + 2.28 \times 10^{-4} D_{ext} - 3.72 \times 10^{-7} D_{ext}^2$
Flexible	273 mm to 762 mm	$\alpha \left(\frac{D_{ext}}{T} \right)^\beta$	$41.21 D_{ext} + 49397$	$0.0815 \ln(D_{ext}) - 1.3842$
Flexible	60 mm to 219 mm	$\exp \left[\alpha \left(\frac{D_{ext}}{T} \right)^\beta \right]$	$1.32 \times 10^{-5} D_{ext}^2 - 4.42 \times 10^{-3} D_{ext} + 12.22$	$2.84 \times 10^{-4} D_{ext} - 4.62 \times 10^{-7} D_{ext}^2 - 0.164$

Table T2-2 Method of calculating F_v

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

T2.2.3.5 Calculation of Likelihood of Failure (LOF)

The likelihood of failure for flow induced turbulence is then determined by the following equation:

$$\text{Flow Induced Turbulence LOF} = \frac{\rho v^2}{F_v} FVF \quad (8)$$

where ρv^2 is determined in **Section T2.2.3.1**, Fluid Viscosity Factor (FVF) is 1.0 for liquid and multiphase fluids and calculated in **Section T2.2.3.2** for gas systems. The Flow Induced Vibration Factor F_v is defined in **Section T2.2.3.4**.

An additional check which can be undertaken on each control valve in the system is to assess the level of fluid kinetic energy at the trim exit. This should be 480 kPa or less for continuous service single phase fluids, and 275 kPa or less for multiphase fluids (where the kinetic energy in kPa is given by $\rho v^2 / 2000$, ρ is the fluid density in kg/m^3 , and v is the velocity of the fluid exiting the valve trim in m/s) **[T2-1]**.

T2.2.4 Advanced Screening Method for Flow Induced Turbulence

T2.2.4.1 Overview

This advanced screening approach is only relevant for pipes having a natural frequency greater than 1 Hz and less than or equal to 3 Hz. This is particularly relevant where the LOF from flow induced turbulence is greater than or equal to 1.0, as calculated using the standard assessment method described in **Section T2.2.3.5**. This is necessary because the flow induced turbulence LOF for flexible pipes is very sensitive to the fundamental natural frequency.

The method detailed above 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 **Section T2.2.3** may be too conservative.

In certain situations, depending on the local configuration of the pipe and its support arrangement, the method may not be conservative. If there is any uncertainty regarding the application of this method then specialist advice should be sought.

T2.2.4.2 Calculation Method

Determining Flow Induced Vibration Factor F_v

F_v is a flow induced vibration factor dependent on the actual outside diameter of the pipe D_{ext} (mm), the wall thickness T (mm) and the fundamental natural frequency f_n .

The following is valid for flexible pipe spans with structural natural frequencies (f_n) ranging from 1Hz to 3Hz.

For pipework with nominal bore between 273 mm to 762 mm (i.e. greater than or equal to 10 inch nominal)

$$F_v = \alpha \left(\frac{D_{ext}}{T} \right)^\beta \quad (9)$$

$$\text{where, } \alpha = (41.21 D_{ext} + 49397) \left(f_n^{0.0001665 D_{ext} + 0.84615} \right) \\ \beta = 0.0815 \ln(D_{ext}) - 1.3842 + 0.0191(f_n - 1)$$

For pipework with nominal bore less than 219 mm (i.e. between 2 into to 8 inch nominal)

$$F_v = \exp \left[\alpha \left(\frac{D_{ext}}{T} \right)^\beta \right] \quad (10)$$

$$\text{where, } \alpha = (1.3175 * 10^{-5} D_{ext}^2 - 4.4213 * 10^{-3} D_{ext} + 12.217) (0.0529 \ln(f_n) + 1) \\ \beta = (-4.622 * 10^{-7} D_{ext}^2 + 2.835 * 10^{-4} D_{ext} - 0.164) (-0.1407 \ln(f_n) + 1)$$

The fundamental natural frequency f_n of the pipe can be determined via site measurements on existing plant or calculated once detailed isometric drawings are available on a new design.

Determining the LOF

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

$$\text{Advanced Screening Flow Induced Turbulence } L.O.F. = \frac{\rho v^2}{F_v} FVF \quad (11)$$

where ρv^2 is determined in **Section T2.2.3.1**, FVF is 1.0 for liquid and multiphase fluids and calculated in **Section T2.2.3.2** for gas systems. The Flow Induced Vibration Factor F_v is defined in **Section T2.2.3.4**.

The resulting LOF value may then be substituted for the Standard Assessment LOF.

T2.2.4.3 Limitations of the Advanced Screening Method

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, refer to **TM-09**).

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 assessment (refer to **TM-08**).

T2.3 MECHANICAL EXCITATION

T2.3.1 Extent of Excitation

There are three cases to consider:

- i. Pipework which is directly attached to machinery (e.g. suction and discharge lines of a pump).
- ii. Pipework which does not form part of the piping system associated with a machine (e.g. (i) above) but is routed close to a machine and may therefore be subjected to mechanical excitation by transmission through the supporting structure.
- iii. Pipework which shares common supports (e.g. the same pipe rack) with another line which itself displays high vibration levels. This can only practically be covered by a visual inspection.

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 the maximum length of the skid.

T2.3.2 Calculation of Likelihood of Failure (LOF)

The likelihood of failure is set to the values below.

Pipework connected or adjacent to	Mechanical Excitation Likelihood of Failure (LOF)
Reciprocating/Positive Displacement Compressor/Pump	0.9
Diesel Engine / Gas Engine	0.8
Screw Compressor/Pump	0.6
Centrifugal Pump	0.4
Electric Motor/Alternator (15kW or greater)	0.4
Electric Motor/Alternator (below 15kW)	0.2
Centrifugal Compressor	0.2
Gas Turbine	0.2
Fan	0.2
Other pipework with an LOF ≥ 0.5	Equal to adjacent pipework LOF

Table T2-3 Mechanical Excitation values

If a detailed structural dynamic analysis of the main line pipework and its supports has been conducted (refer to **TM-09**) to establish that there will be no coincidence with excitation frequencies from reciprocating/positive displacement pumps or compressors or diesel engines then the LOF can be reduced to 0.4.

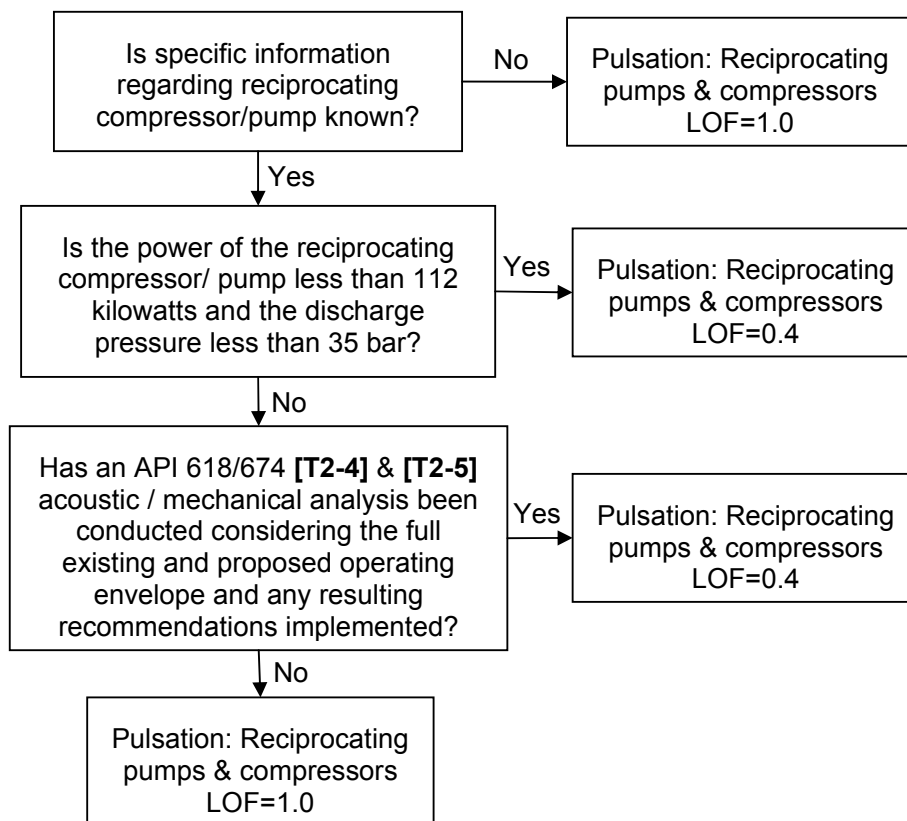
T2.4 PULSATION: RECIPROCATING/POSITIVE DISPLACEMENT PUMPS & COMPRESSORS

T2.4.1 Extent of Excitation

The pulsations caused by reciprocating/positive displacement pumps and compressors affect the pipework upstream and downstream to the first major vessel.

The excitation characteristics can change under certain operations (e.g. recycling, change in speed, running trains in parallel) and the acoustic modes are affected by changes in pressure, temperatures and molecular weight or fluid density. Therefore the full range of operating conditions should be considered as part of the assessment.

T2.4.2 Calculation of Likelihood of Failure (LOF)



Flowchart T2-2
assessment

Pulsation: Reciprocating/positive displacement pumps & compressors

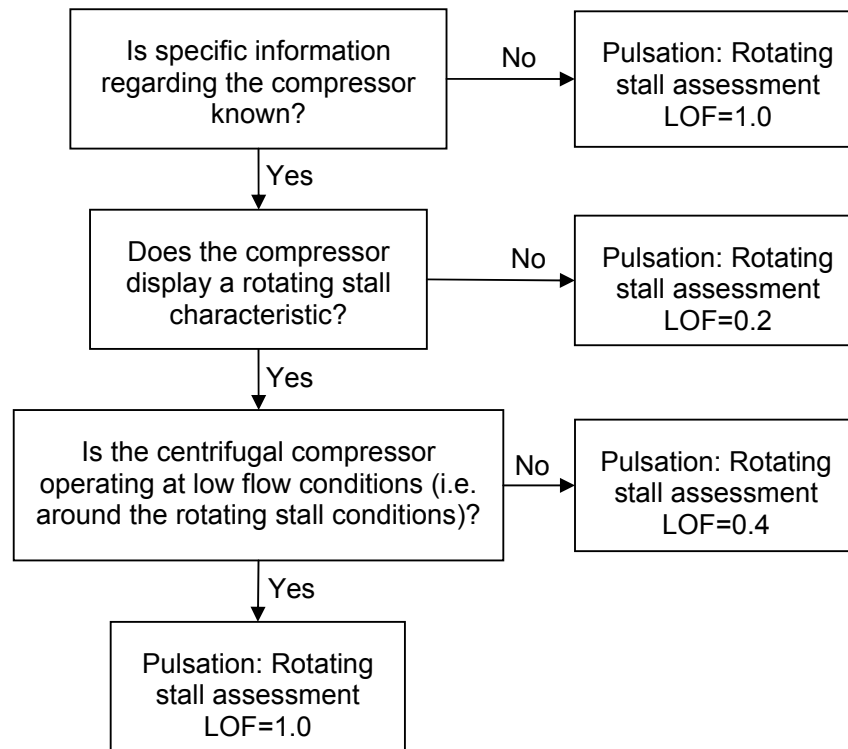
T2.5 PULSATION: ROTATING STALL

T2.5.1 Extent of Excitation

The pulsations caused by rotating stall affect the pipework upstream and downstream to the first major vessel.

The excitation characteristics can change under certain operations (e.g. recycling, change in speed, running trains in parallel) and the acoustic modes are affected by changes in pressure, temperatures and molecular weight. Therefore the range of operating conditions should be considered as part of the assessment.

T2.5.2 Calculation of Likelihood of Failure (LOF)



Flowchart T2-3 Pulsation: Rotating stall assessment

T2.6 PULSATION: FLOW INDUCED EXCITATION

T2.6.1 Extent of Excitation

The mechanism considered is that due to flow past a branch with a closed end (a deadleg branch off the main line).

The pulsations caused can propagate upstream and downstream from the sidebranch to the first major change in main pipe diameter.

Note: A major change is defined as a pipe diameter change by a factor of 2 or more (e.g. a vessel or significant expansion/reduction).

The excitation characteristics can change under certain operations (e.g. flowrate) and the acoustic modes are affected by changes in pressure, temperatures and molecular weight. Therefore the anticipated range of operating conditions should be considered as part of the assessment.

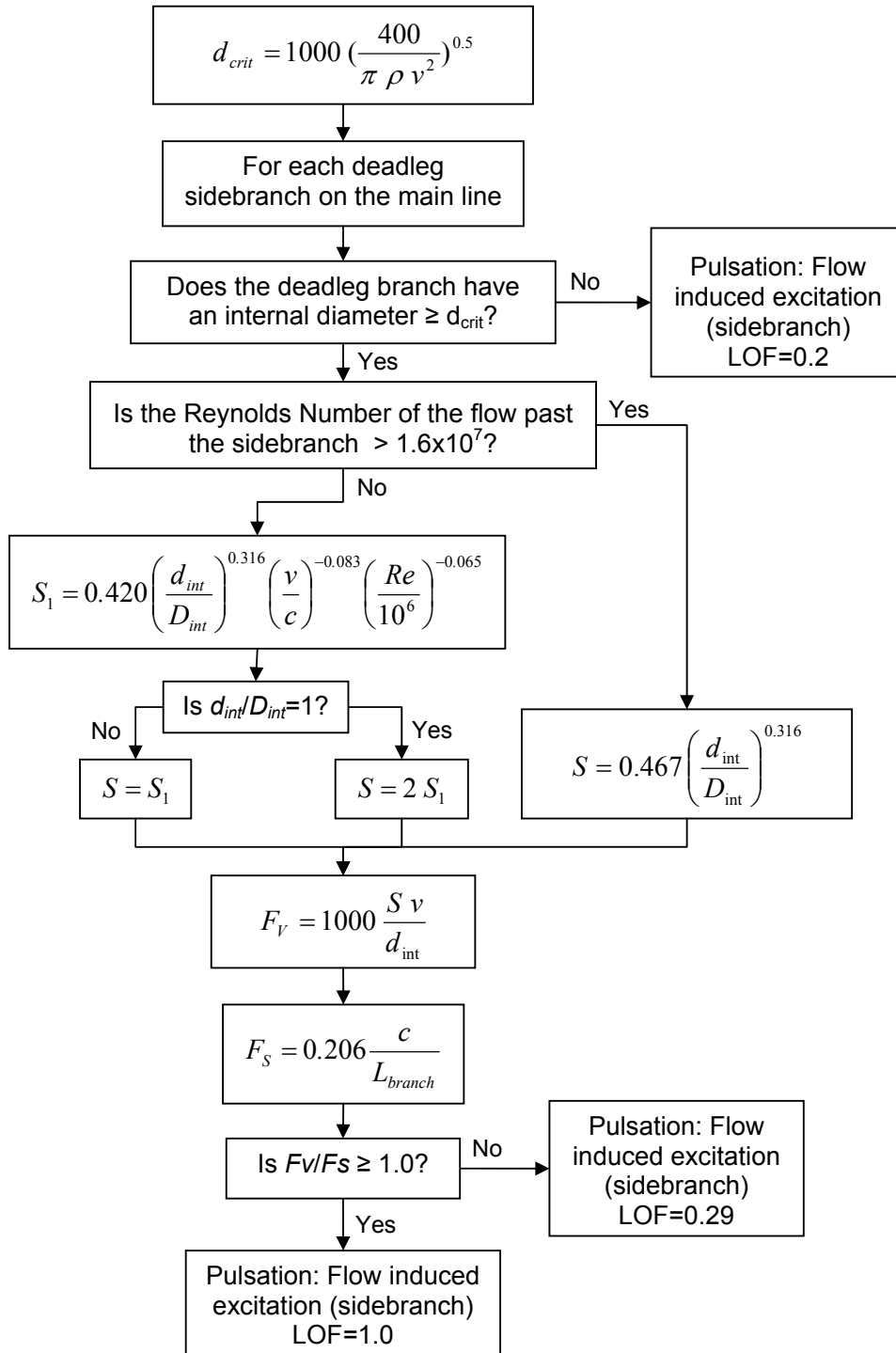
T2.6.2 Input

Input	Symbol	Units	Comment
Speed of sound in gas	c	m/s	
Internal diameter of branch	d_{int}	mm	
Internal diameter of main line	D_{int}	mm	
Length of sidebranch	L_{branch}	m	
Reynolds Number	Re		Refer to Appendix B for definition of characteristic dimension and calculation method
Mean fluid velocity in main pipe	v	m/s	
Gas density	ρ	kg/m ³	

T2.6.3 Calculation of Likelihood of Failure (LOF)

The assessment method allocates a main line LOF score for each sidebranch on the main line. The highest LOF score from all the sidebranches on the main line should then be used as the representative LOF score for the main line itself.

The simplified screening analysis given in **Flowchart T2-4** does not strictly apply if the sidebranch geometry is complex (i.e. the sidebranch itself is not a single line from the main line to the closed end). A typical example would be a relief line that divides to feed two or more relief valves. In such cases a detailed analysis **[T2-2]** should be conducted to accurately determine the acoustic natural frequencies of the sidebranch (i.e. F_s).



Flowchart T2-4 Pulsation: Flow induced excitation assessment

Note: For each sidebranch that scores an LOF = 1 it is recommended that a more detailed analysis as described in [T2-2] is undertaken.

T2.7 HIGH FREQUENCY ACOUSTIC EXCITATION

T2.7.1 Extent of Excitation

The response caused by high frequency acoustic excitation affects the pipework downstream of the source to the first major vessel, e.g. separator, KO drum.

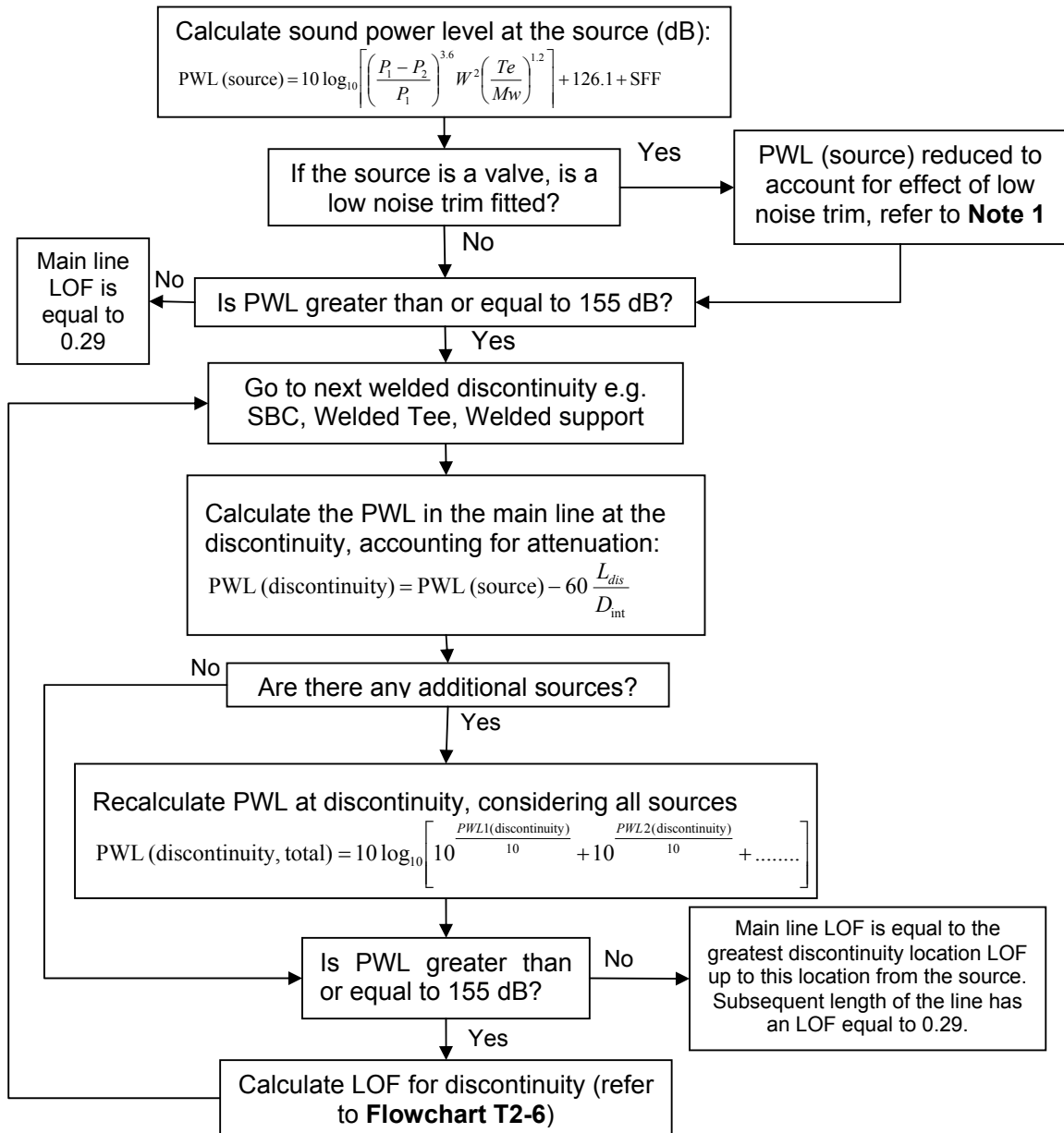
The assessment generates a main line LOF value at each welded discontinuity, e.g. SBC, Welded Tee, Welded support. It is at the discontinuities with an LOF equal to one where corrective actions are required.

The sources of high frequency acoustic excitation are pressure reducing devices such as control / relief valves, restriction orifices, or branch connections.

T2.7.2 Input

Input	Symbol	Units	Comment
External diameter of the main line	D_{ext}	mm	
External diameter of the branch	d_{ext}	mm	
Internal diameter of the main line	D_{int}	mm	
Distance between source and the welded discontinuity	L_{dis}	m	
Molecular weight of gas	M_w	grams/mol	Refer to Appendix B for typical values
Pressure upstream of pressure reducing device	P_1	Pa absolute	
Pressure downstream of pressure reducing device	P_2	Pa absolute	
Wall thickness of the main line	T	mm	
Wall thickness of the branch	t	mm	
Upstream temperature	T_e	K	
Mass flow rate	W	kg/s	

T2.7.3 Calculation of Likelihood of Failure (LOF)



Flowchart T2-5 High frequency acoustic fatigue assessment,

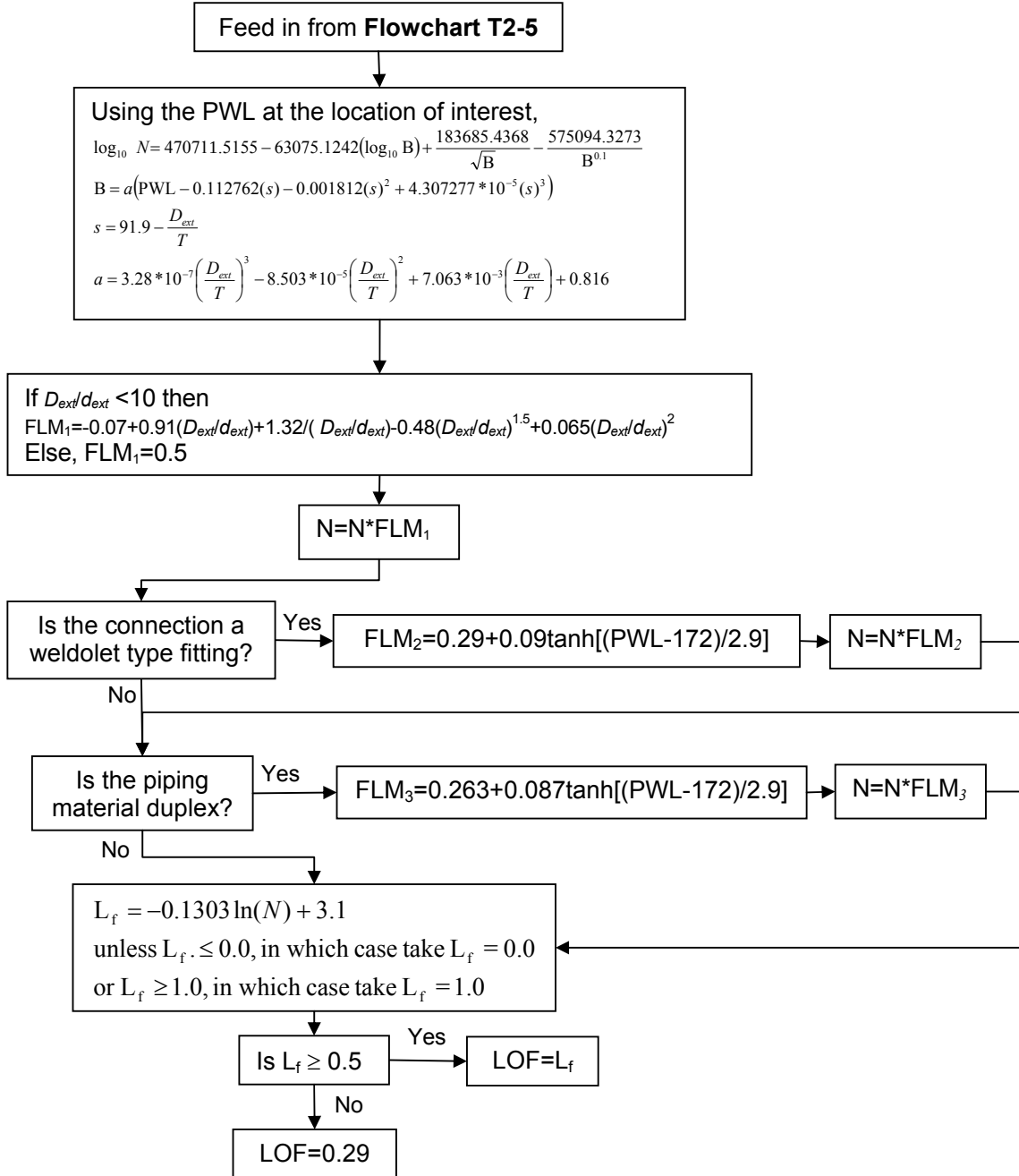
Where, PWL is the sound power level

PWL1 (discontinuity) = PWL at the discontinuity due to source 1

PWL2 (discontinuity) = PWL at the discontinuity due to source 2

SFF is a correction factor to account for sonic flow. If sonic conditions exist then SFF=6; otherwise SFF = 0.

Note 1: If the source is a valve and a low noise trim is fitted then the PWL (source) 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.**



Where, N is the number of cycles to failure,
 FLM_i is the fatigue life multiplier for stage i
 PWL is $PWL_{(discontinuity, total)}$ calculated in **Flowchart T2-5**

T2.8 SURGE/MOMENTUM CHANGES DUE TO VALVE OPERATION

T2.8.1 Overview

The first step assessment process involves identifying all the significant valves on a particular line. Excitation due to surge and momentum changes is only considered for fast acting valves [T2-3], which excludes all manually operated valves. Typical automatic valves that need to be considered in the assessment include:

- Emergency Shut Down Valves (ESD)
- Flow Control Valves (FCV)
- Pressure Control Valve (PCV)
- Blow Down Valves (BDV)
- Relief Valves (RV)

The assessment of excitation due to surge and momentum changes can be split into the three following operational cases:

- Dry Gas valve operation valve opening
- Liquid or Multiphase valve closure
- Liquid or Multiphase valve opening

For a dry gas any potential surge pressure due to a rapid valve closure is taken up via compression of the gas, hence the likelihood of failure due to a gas valve closing is considered negligible. Therefore the Likelihood of Failure for this operation is zero.

The assumption is made that the line is adequately supported for any reaction loads and that any anchors have significant strength.

T2.8.1.1 Extent of Excitation: Liquid or Multiphase valve closure

The main line LOF value predicted below should be applied to the entire main line length upstream of the valve, up to the next major vessel or significant pipe diameter change (“L” in **Table T2-5**) and up to two partial or full pipe supports downstream of the valve, (not spring hangers or constant load supports).

T2.8.1.2 Extent of Excitation: Liquid or Multiphase valve Opening

The main line LOF value predicted below should be applied to up to two partial or full pipe supports both upstream and downstream of the valve (not spring hangers or constant load supports).

During this type of valve operation there is a likelihood of Cavitation and Flashing and assessments detailed in **Section T2.9** and **T2.10** respectively are required.

T2.8.1.3 Extent of Excitation: Dry Gas Rapid Valve Opening

The main line LOF value predicted below should be applied to up to two partial or full pipe supports both upstream and downstream of the valve (not spring hangers or constant load supports).

T2.8.2 Information Requirements

The following table lists the information required for analysis of the excitation due to surge and momentum changes of different valve operations.

Proposed values of some of the input parameters listed are presented in **Appendix B** for some typical fluid types encountered in process systems. These are marked in the “comment” section of the table.

Piping Information	Symbol	Units	Comment	Liquid or Multiphase		Gas
				Valve Closure	Valve Opening	Valve Opening
Fluid density	ρ	kg/m ³		*	*	
Ratio of Specific Heat Capacities	γ		(C_p/C_v) Refer to Appendix B for Sample Input Parameter Values			*
Speed of sound	c	m/s	Refer to Appendix B for Sample Input Parameter Values	*		
External Main Line Diameter	D_{ext}	mm		*		*
Internal Main Line Diameter	D_{int}	mm		*		*
Young's Modulus of the main line material	E_{ml}	N/m ²		*		
Fluid Bulk Modulus	K	N/m ²		*		
Upstream Pipe Length	L_{up}	m	From valve to next major vessel or change in pipe diameter (greater than 2:1 diameter change) If the length is greater than 100m than a detailed surge analysis is required. Refer to Appendix B for Sample Input Parameter Values	*		
Molecular Weight	Mw	grams/mol	Refer to Appendix B for Sample Input Parameter Values			*
Upstream Static Pressure	P_1	Pa		*	*	
Pump head at zero flow	$P_{shut-in}$	Pa		*		
Vapour Pressure	P_v	Pa	Refer to Appendix B for Sample Input Parameter Values			
Static Pressure drop	ΔP	Pa				
Universal Gas Constant	R	J/K.kmol	Value of 8314			*
Main line Wall Thickness	T	mm		*	*	*
Valve Closing Time	T_{close}	sec	Refer to Appendix B for Sample Input Parameter Values	*		
Upstream Temperature	Te	K				*
Steady State Fluid Velocity	v	m/s		*		
Mass Flow Rate	W	kg/s			*	*
Pipe Support Type			Refer to Section T2.2.3.3	*	*	*
Main line Wall Thickness for Schedule 40 Piping		mm		*	*	*
Valve Type			Refer to Appendix B for Sample Input Parameter Values	*		

Table T2-4 Information Requirements

Also,

$$\Psi = \frac{\text{Actual pipe wall thickness}}{\text{Schedule 40 pipe wall thickness}}$$

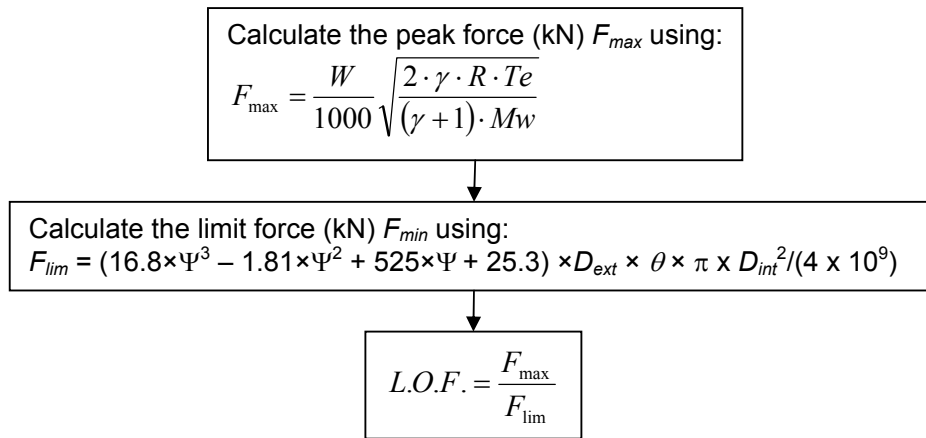
θ is the correction for the support type, refer to **Section T2.2.3.3** for definition of support type:

Support Type	Stiff	Medium Stiff	Medium	Flexible
θ	4	2	1	0.5

T2.8.3 Calculation of Likelihood of Failure (LOF)

T2.8.3.1 Dry Gas Rapid Valve Opening

For a rapid opening of a gas valve the transient forces are due to the sudden change in momentum.

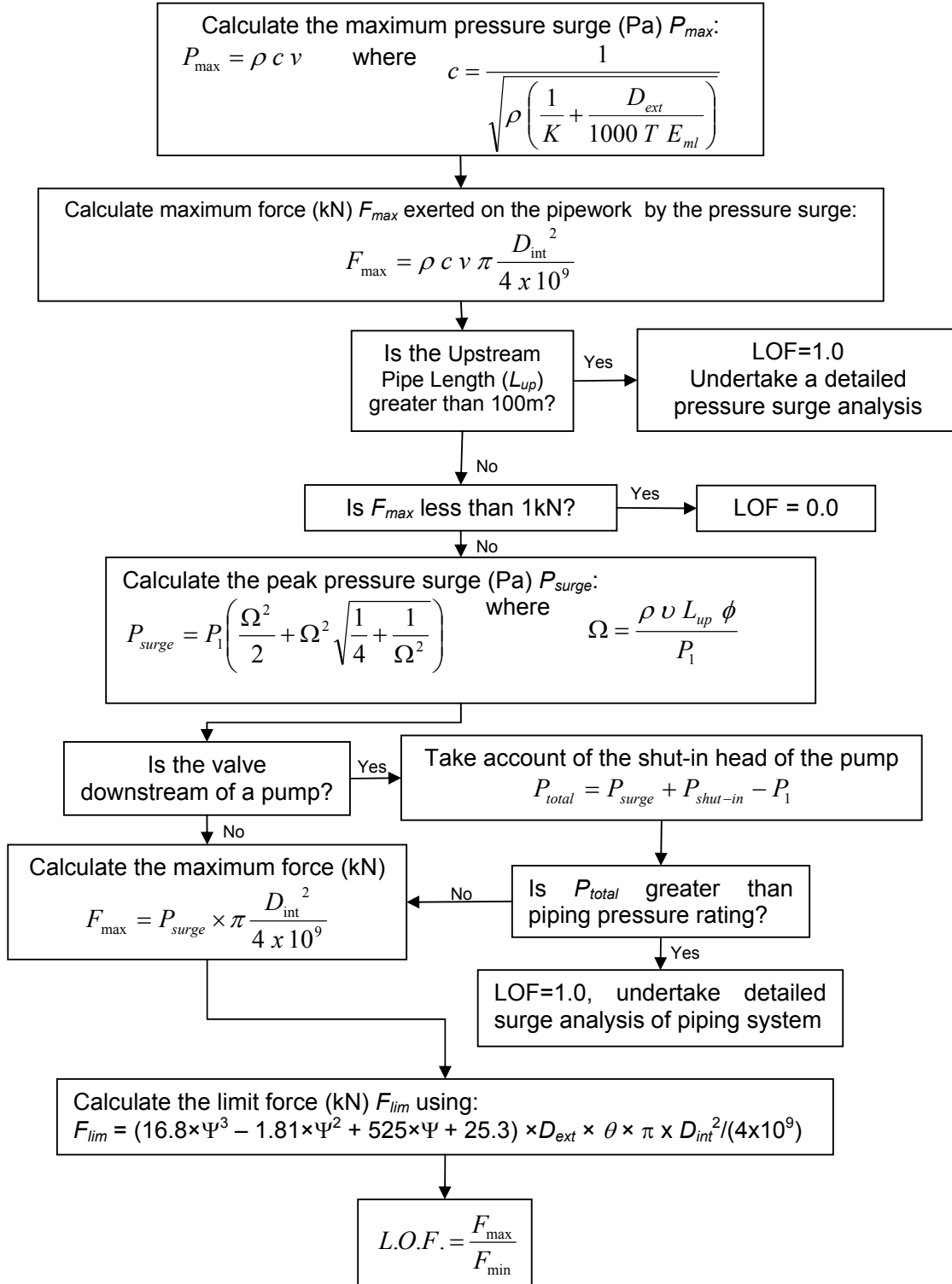


Flowchart T2-6 Dry gas rapid valve opening assessment

T2.8.3.2 Liquid or Multiphase Valve Closure

The peak pressure surge (P_{max}) generated during a valve opening or closure should remain within the design pressure rating for the line. If this is not the case a detailed surge analysis should be carried out in addition to the following assessment (refer to **TM-09**).

This initial assessment considers the worst case event of a 'sudden' valve closure, and the effect of the pressure surge on the pipe. If this is considered acceptable then no further analysis is required as pressure surge is unlikely to affect the integrity of the pipe. Sudden valve closure is defined as a valve closure time that is less than $(2L_{up}/c)$.



Flowchart T2-7 Liquid or Multiphase valve closure assessment

where, ϕ is the function defining the flow area of the valve as a function of time. The function can be simplified for specific valve types by assuming that the peak pressure surge occurs at the point when the valve is closed, at a time T_{close} . The

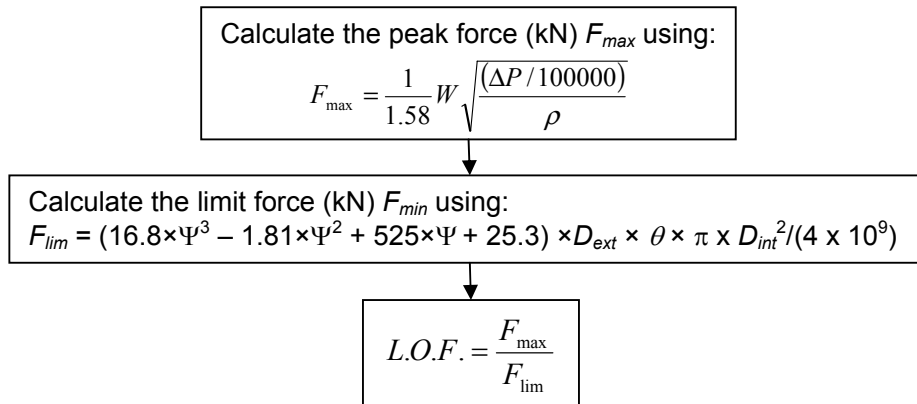
following table summarises the resulting α functions for different valve types. These are valid for valve closure times of up to 30 seconds.

Valve Type	ϕ
Full bore ball	$\frac{-1.281}{T_{close}} - 0.27$
Reduced bore ball	$\frac{-1.268}{T_{close}} - 0.362$
Butterfly	$\frac{-2.877}{T_{close}} - 0.275$
Globe	$\frac{-2.266}{T_{close}} - 0.32$
Gate	$\frac{-3.41}{T_{close}} - 0.315$

Note, If the type and/or closing time of the valve are not known then assume a globe valve and a valve closing type of 1 second per inch of pipe diameter.

T2.8.3.3 Liquid or Multiphase Valve Opening

High dynamic forces due to the rapid change in momentum, considering the valve opening scenario in a liquid or multiphase system, is outlined in the steps below. Note: for this case cavitation and flashing need to be taken into account using the approach outlined in **Sections T2.9** and **T2.10**, respectively.



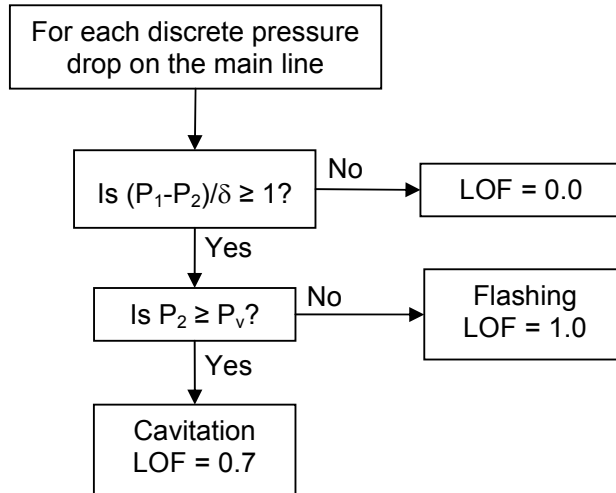
Flowchart T2-8 Liquid or Multiphase valve opening assessment

T2.9 CAVITATION AND FLASHING

T2.9.1 Extent of Excitation

Cavitation and flashing are relatively localised effects. However the energy generated can be transmitted along the pipework and the main line LOF value predicted below should be applied to the pipework up to two partial or full pipe supports both upstream and downstream of the flow discontinuity (not spring hangers or constant load supports).

T2.9.2 Calculation of Likelihood of Failure (LOF)



Flowchart T2-9 Cavitation and flashing assessment

Note: $\delta = P_1 - P_v$ unless the pressure drop is caused by a valve, in which case

$$\delta = F_L^2 (P_1 - 0.96 \times P_v)$$

Where P_1 = pressure upstream of discrete pressure drop (Pa)

P_2 = pressure downstream of discrete pressure drop (Pa)

P_v = liquid vapour pressure at upstream temperature (Pa)

F_L = liquid pressure recovery factor (typical values are given below)

Valve Type	F_L
Ball	0.6
Butterfly	0.62
Globe	0.9
Gate	0.6

Technical module

T3 - QUANTITATIVE SBC LOF ASSESSMENT

T3.1 OVERALL ASSESSMENT METHODOLOGY

Small bore connections (SBC) on all main lines that have been scored with a main line LOF ≥ 0.3 should be assessed using the methods given in **Section T3-2**. In addition, if there is any uncertainty regarding the type of excitation that may apply (including excitation mechanisms not explicitly covered in **TM-02**, e.g. slug flow, environmental loading) then the respective main line should be assigned an LOF=1 and a quantitative SBC LOF assessment undertaken. This will result in a conservative assessment.

It is possible to perform a quantitative SBC LOF assessment in isolation, without having first determined the main line LOF score (i.e. the SBC assessment can be undertaken in isolation); however it should be noted that in this case the main line LOF defaults to 1.0. This will result in a conservative assessment.

Guidance on undertaking the assessment detailed in this technical module is provided in **Appendix C**.

In addition, if an SBC is on a main line subjected to tonal excitation, coupling between a structural natural frequency of the SBC and the tonal excitation frequency(ies) should be avoided. Tonal excitation is generated by the following excitation mechanisms:

- Mechanical Excitation
- Pulsation: Reciprocating /Positive Displacement Pumps & Compressors
- Pulsation: Rotating Stall
- Pulsation: Flow Induced Excitation

In this case, as well as undertaking the assessment given in this Technical Module, the structural natural frequencies of the SBC should be determined by specialist measurement or predictive techniques, refer to **TM-08** and **TM-09**.

SBC which are already braced should still be assessed using one or more of the techniques described below, as determined by the main line quantitative assessment, **TM-02**, and the SBC Visual Inspection, **TM-05**. This will indicate whether there is still a residual concern, e.g. whether the bracing is fit for purpose.

If the SBC is subjected to an excitation greater than 300Hz specialist advice should be sought.

There are two stages to the SBC assessment:

- Geometric LOF: which takes account of the physical make up of the SBC to assess the connection's fundamental natural frequency and susceptibility to stress levels causing damage.
- Location LOF: which takes account of the SBC location on the main line.

The minimum of the Geometric LOF and Location LOF results in the overall **SBC Modifier**.

T3.2 GEOMETRY ASSESSMENT METHODOLOGY

The four types of SBC design considered are listed below:

- Type 1: Cantilever type
- Type 2: Continuous - In and out same main line
- Type 3: Continuous – With intermediate supports
- Type 4: Continuous - Between Different Main Lines (with no intermediate supports)
(Note: Type 4 connections are not common and not recommended since they are likely to experience significant static stress, due to the differential movement of the lines at either end.)

The geometric assessment is dependent on the type of SBC being assessed. **Flowchart T3-1** gives an overview of the assessment process, while **Flowcharts T3-2 to T3-8** detail the specific assessment steps for each SBC type, as discussed in the following sections.

T3.2.1 Type 1: Cantilever Type Connection

Details of the Type 1 “Cantilever type” SBC assessment methodology are presented in **Flowchart T3-2**.

T3.2.2 Type 2: Continuous – In and Out Same Main Line

Details of the Type 2 SBC assessment methodology for a connection in and out of the same main line are presented in **Flowchart T3-4**. If there is a support to the deck or structural steelwork on the SBC it should be assessed as if it was a Type 3 SBC with Intermediate Supports.

To take account of the mass on the SBC (e.g. valve or flange), the connection should be split into two Type 1 cantilever type connections (refer to **Section C.1.11**) about the midspan point. Assess both sides as if the free end was the last mass on each half of the line and determine $LOF_{GEOM(A)}$ and $LOF_{GEOM(B)}$.

T3.2.3 Type 3: Continuous – With Intermediate Supports

An overview of the Type 3 SBC assessment for a connection off a main line with intermediate supports is presented in **Flowchart T3-5**. This encompasses the assessment for the first span length in **Flowchart T3-6** and for subsequent span lengths in **Flowchart T3-7**.

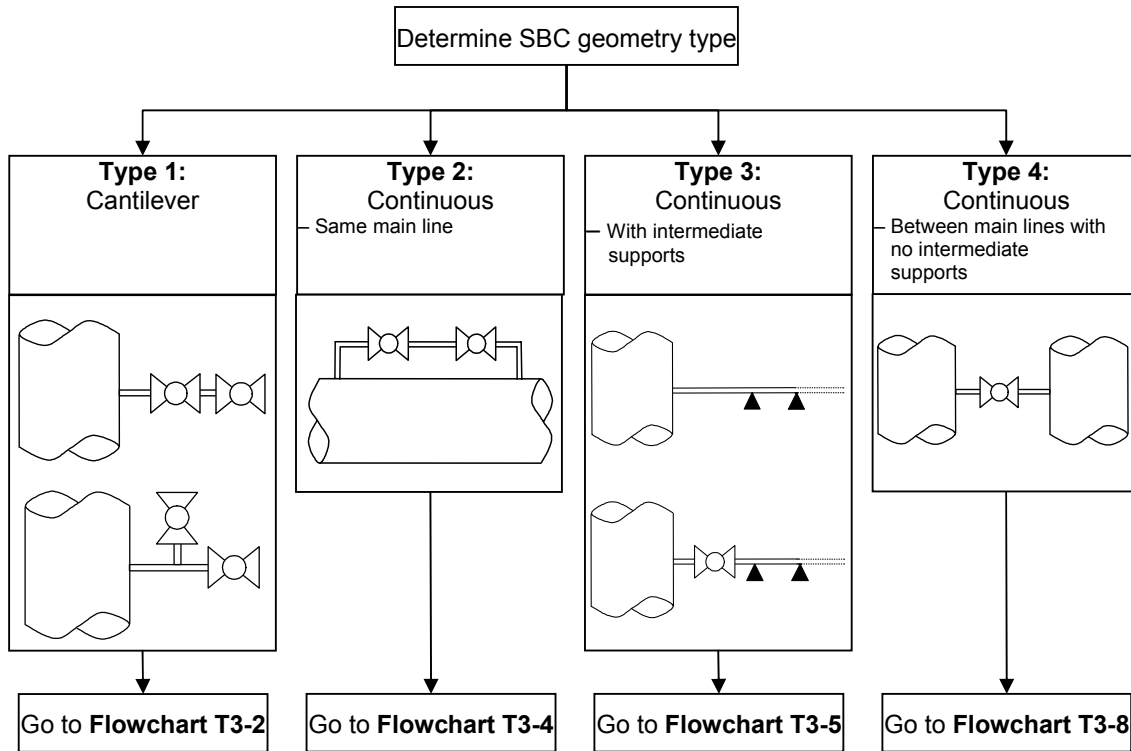
T3.2.4 Type 4: Continuous – Between Different Main Lines

Details of the Type 4 SBC assessment for a connection running between two different main lines are presented in **Flowchart T3-8**. If there is a support to the deck or structural steelwork on the SBC it should be assessed as if it was a Type 3 SBC with Intermediate Supports.

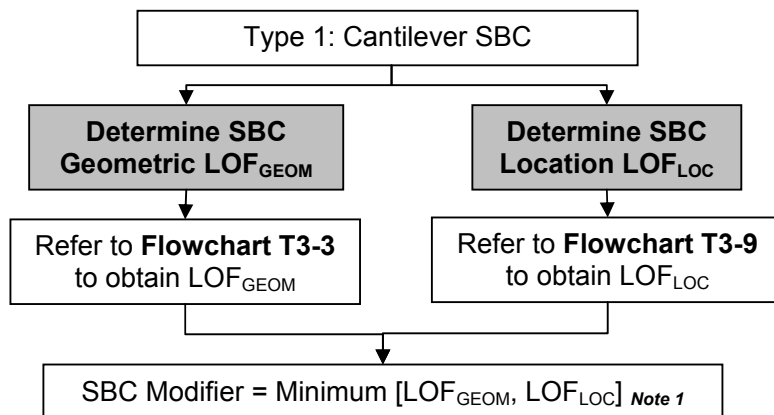
To take account of the mass on the SBC (e.g. valve or flange), the connection should be split into two Type 1 (refer to **Section C.1.11**) cantilever type connections about the midspan point. Assess both sides as if the free end was the last mass on each half of the line.

T3.3 LOCATION ASSESSMENT METHODOLOGY

Details of the Location LOF assessment are presented in **Flowchart T3-9**.

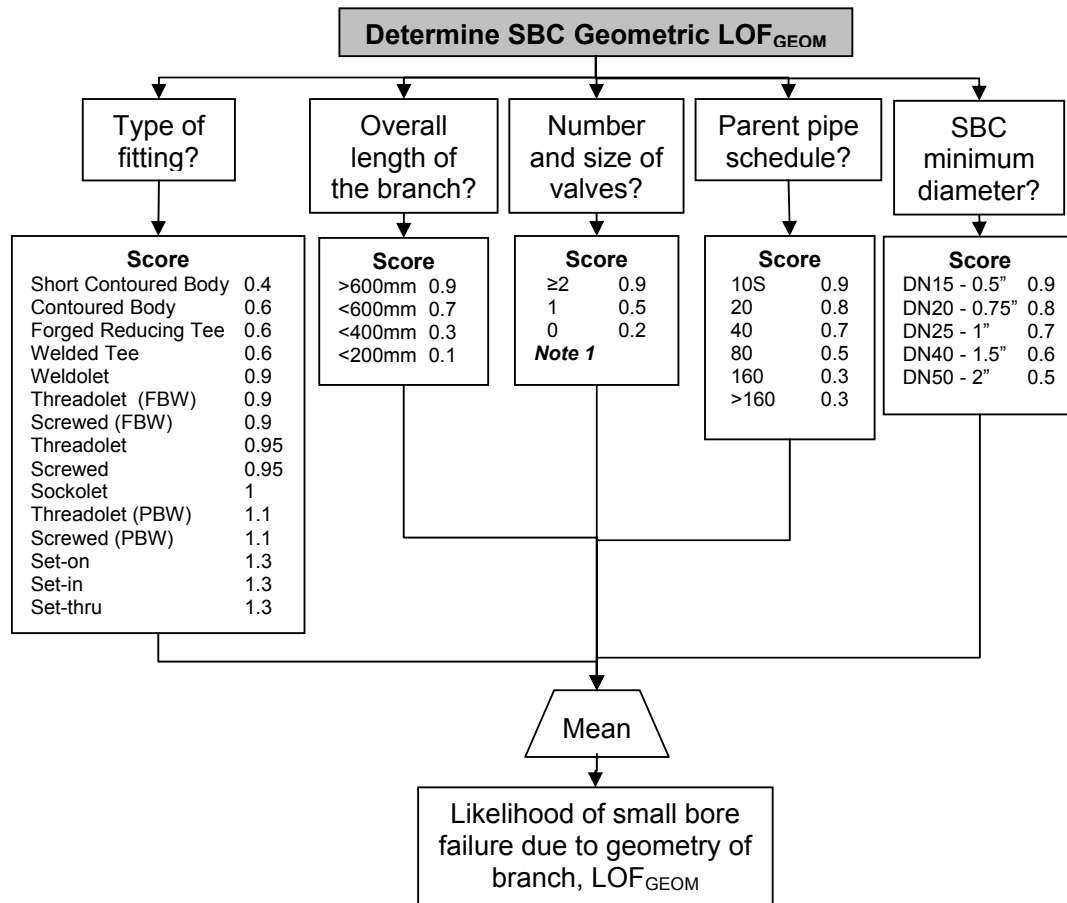


Flowchart T3-1 Overview of SBC Assessment Methodology



Flowchart T3-2 Type 1 SBC Assessment Methodology

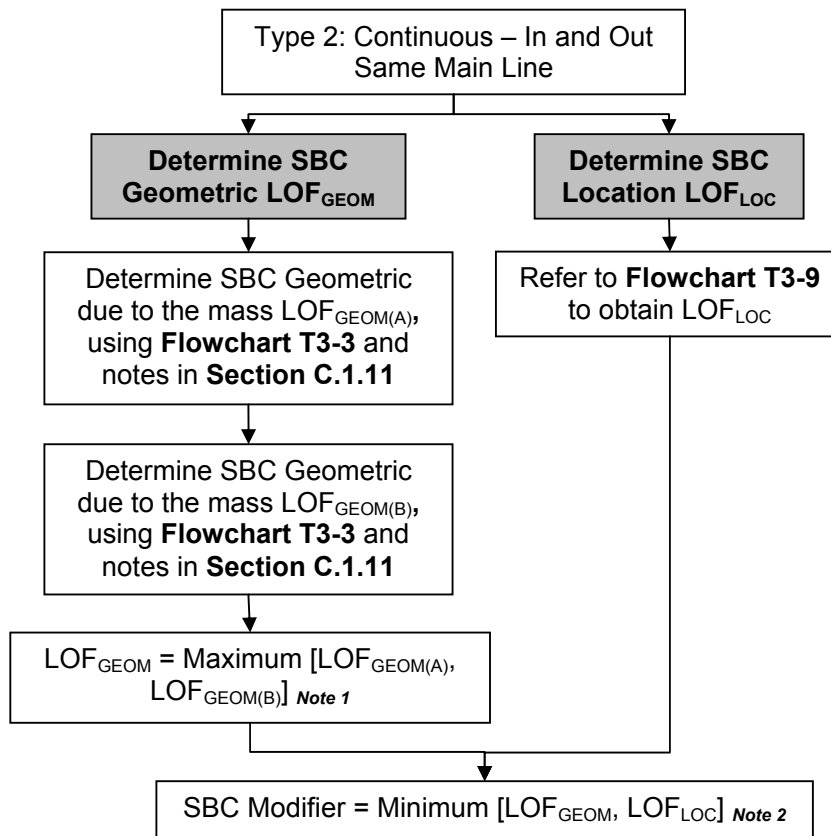
Note 1, the minimum of the two inputs (LOF_{GEOM} and LOF_{LOC}) is required because both a poorly placed and poorly designed SBC need to be present for the SBC to have a high LOF.



Note: FBW – Fully Backwelded
PBW – Partially Backwelded

Note 1: This applies for flange and/or valve ratings below ANSI 900. Where the flange and/or valve rating is ANSI 900 or greater, refer to **Section C.1.2**.

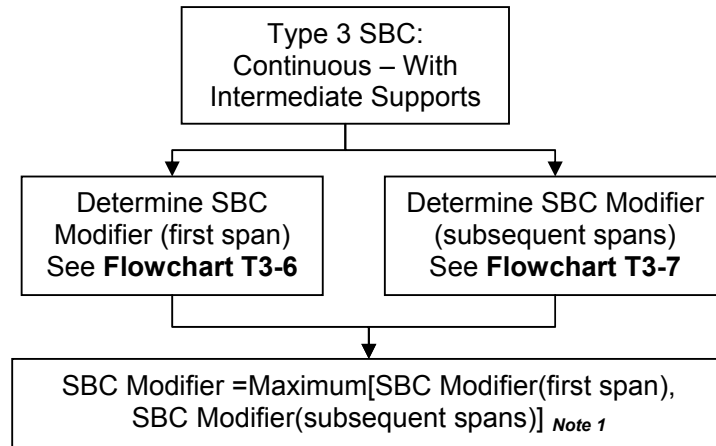
Flowchart T3-3 Type 1 SBC Assessment Methodology



Flowchart T3-4 Type 2 SBC Assessment Methodology

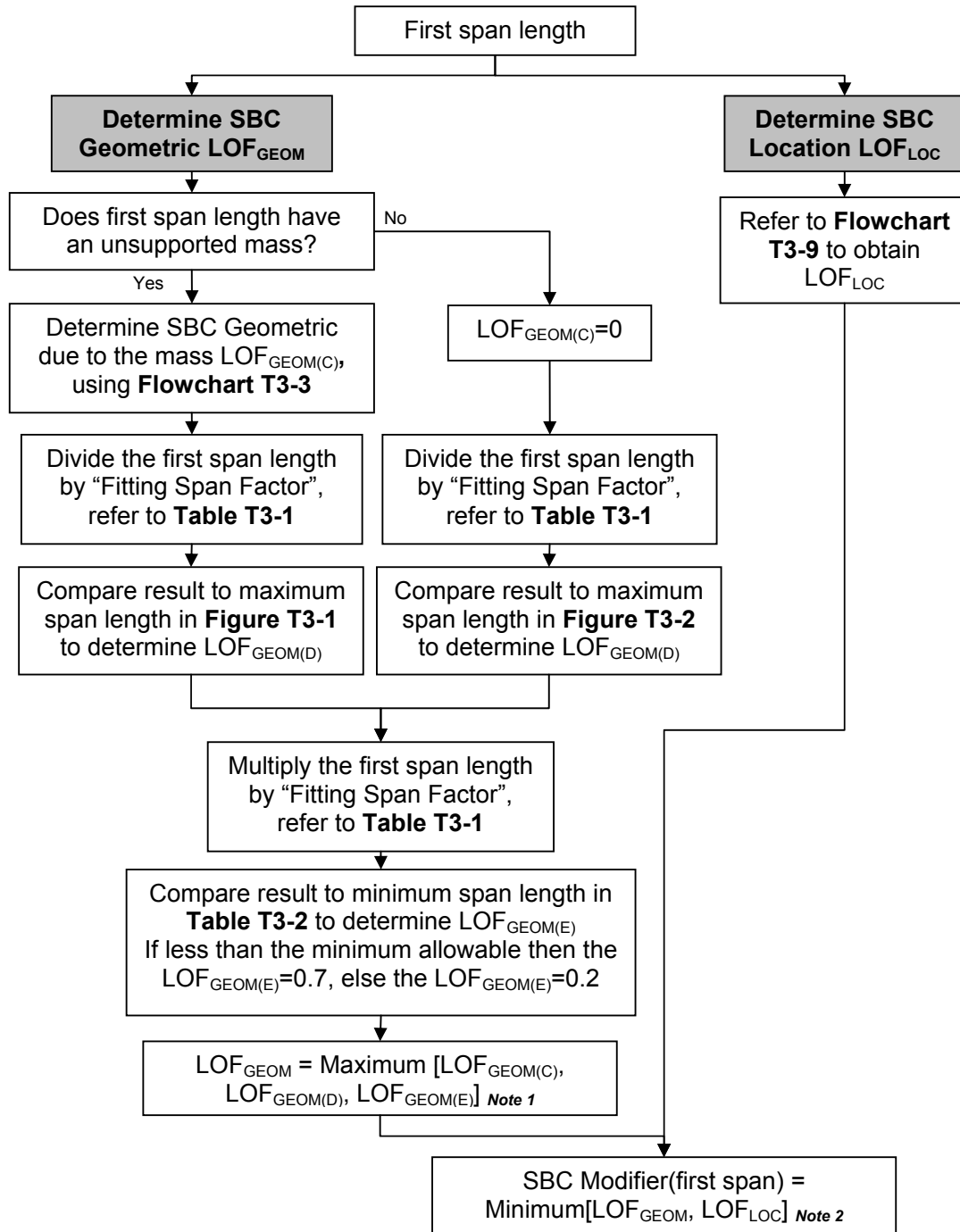
Note 1, the maximum of the two inputs [LOF_{GEOM(A)}, LOF_{GEOM(B)}] is required because the characteristic with the greatest geometric LOF is required.

Note 2, the minimum of the two inputs (LOF_{GEOM} and LOF_{LOC}) is required because both a poorly placed and poorly designed SBC need to be present for the SBC to have a high LOF.



Flowchart T3-5 Type 3 SBC Assessment Methodology – Overview

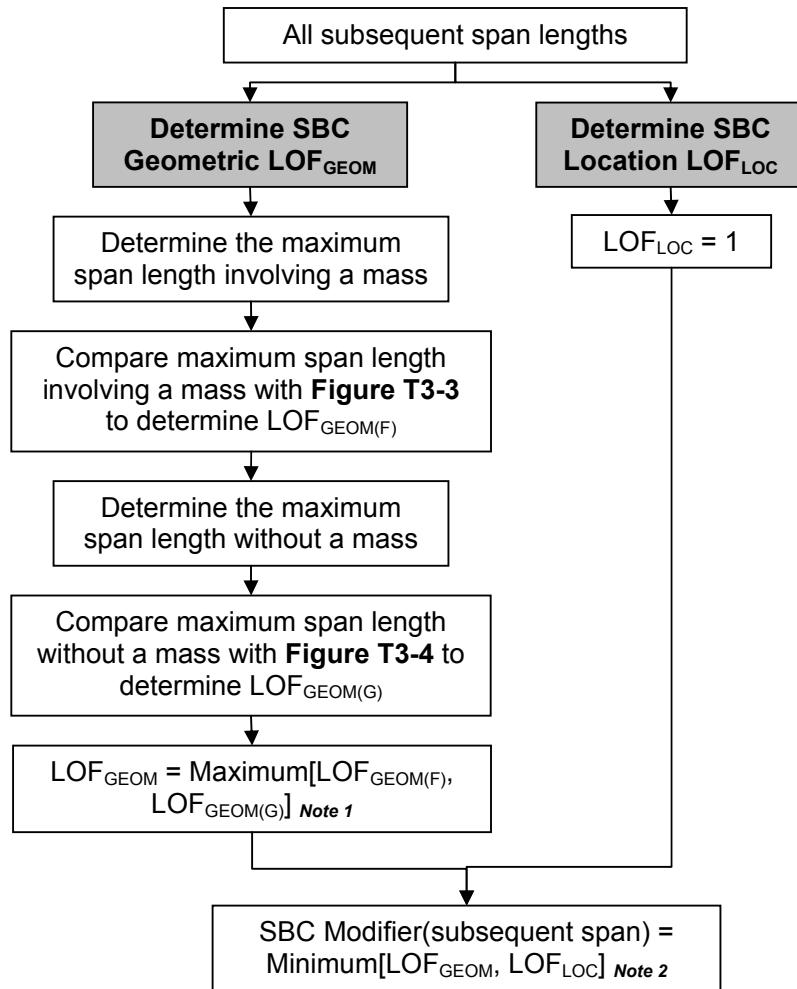
Note 1, the maximum of the two inputs [SBC Modifier(first span), SBC Modifier(subsequent spans)] is required because the characteristic with the greatest geometric LOF is required.



Flowchart T3-6 Type 3 SBC Assessment Methodology - First Span

Note 1, the maximum of the three inputs [LOF_{GEOM(C)}, LOF_{GEOM(D)}, LOF_{GEOM(E)}] is required because the characteristic with the greatest geometric LOF is required.

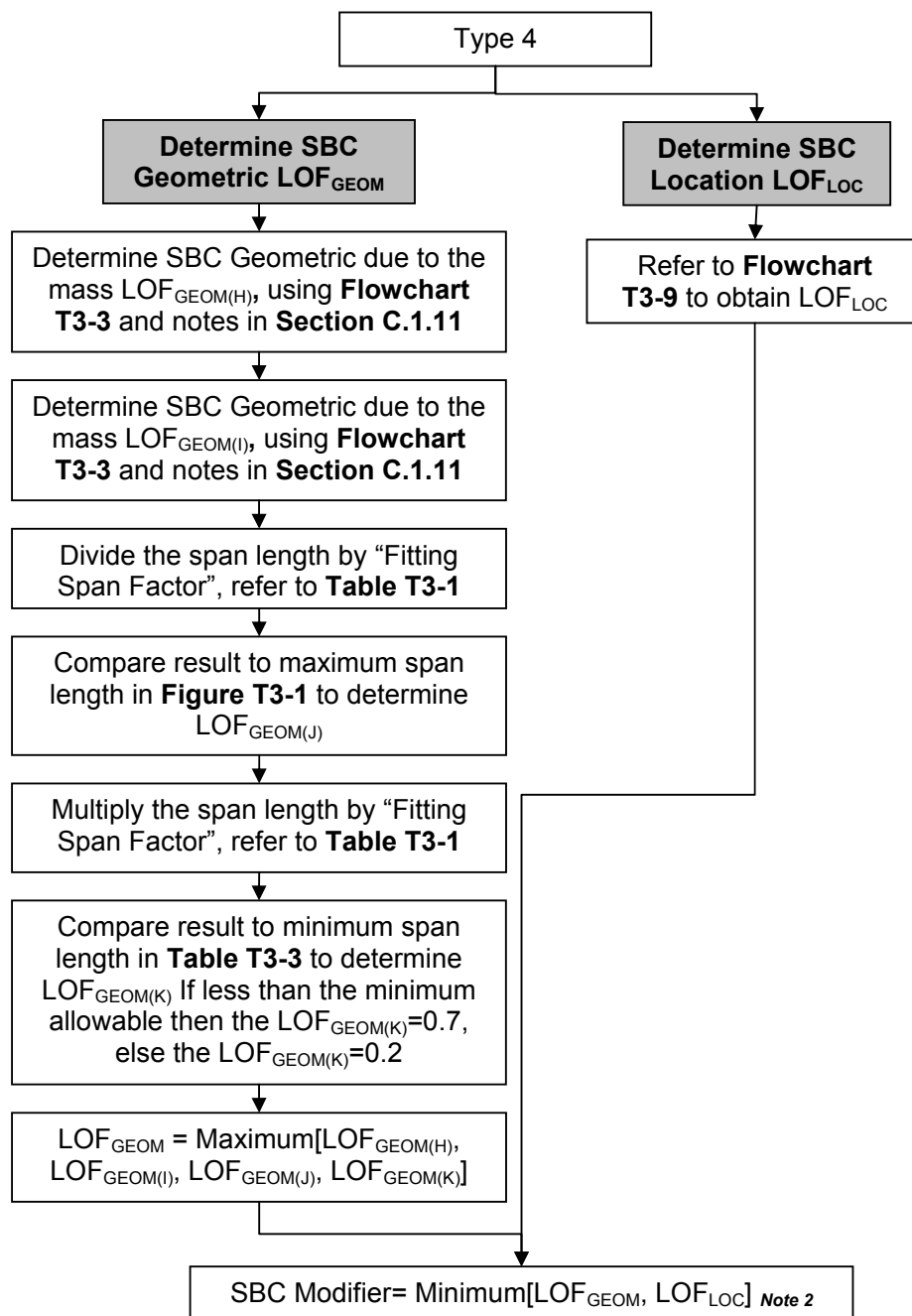
Note 2, the minimum of the two inputs (LOF_{GEOM} and LOF_{LOC}) is required because both a poorly placed and poorly designed SBC need to be present for the SBC to have a high LOF.



Flowchart T3-7 Type 3 SBC Assessment Methodology - Subsequent Spans

Note 1, the maximum of the two inputs [$LOF_{GEOM(F)}$, $LOF_{GEOM(G)}$] is required because the characteristic with the greatest geometric LOF is required.

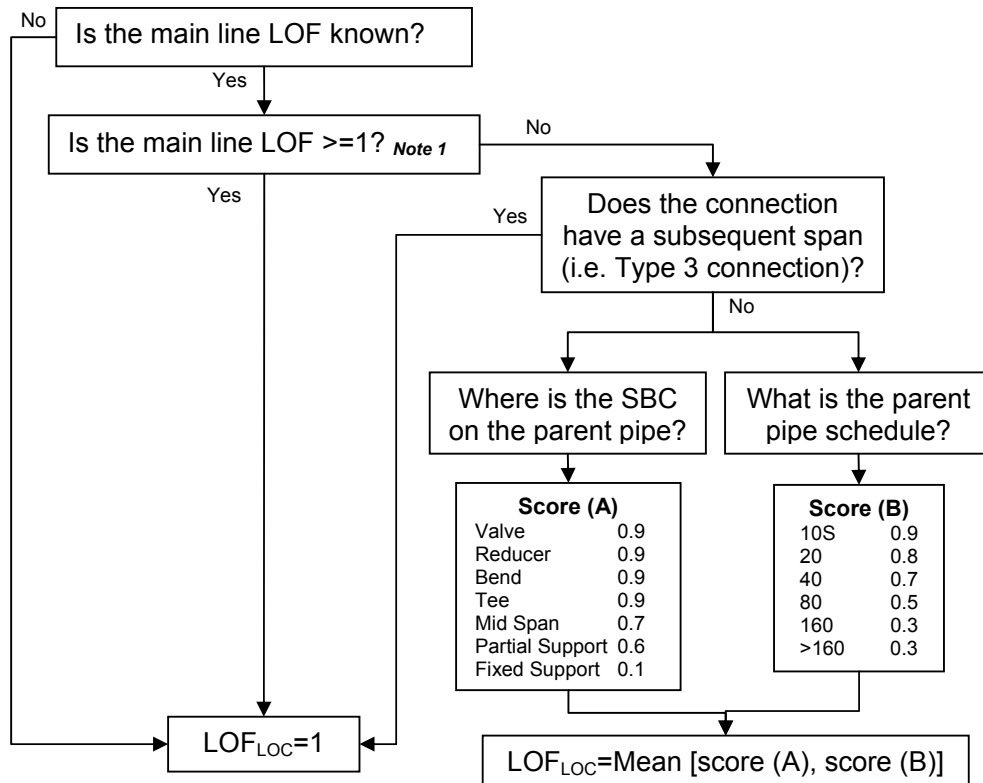
Note 2, the minimum of the two inputs (LOF_{GEOM} and LOF_{LOC}) is required because both a poorly placed and poorly designed SBC need to be present for the SBC to have a high LOF.



Flowchart T3-8 Type 4 SBC Assessment Methodology

Note 1, the maximum of the four inputs [LOF_{GEOM(H)}, LOF_{GEOM(I)}, LOF_{GEOM(J)}, LOF_{GEOM(K)}] is required because the characteristic with the greatest geometric LOF is required.

Note 2, the minimum of the two inputs (LOF_{GEOM} and LOF_{LOC}) is required because both a poorly placed and poorly designed SBC need to be present for the SBC to have a high LOF.



Flowchart T3-9 Location Assessment Methodology

Note 1, if there is a high main line LOF (i.e. greater or equal to 1, identifying there is a high excitation source) the LOF_{LOC} defaults to 1, which means the SBC LOF is dominated by the SBC geometry.

Fitting type	Fitting Span Factor
Short Contoured Body	1.00
Contoured Body	0.85
Forged Reducing Tee	0.85
Welded Tee	0.85
Weldolet	0.70
Threadolet - Fully back welded (no exposed threads)	0.70
Screwed - Fully back welded (no exposed threads)	0.70
Threadolet	0.65
Screwed	0.65
Sockolet	0.65
Threadolet - Partially back welded (exposed threads)	0.60
Screwed - Partially back welded (exposed threads)	0.60
Set-on	0.55
Set-in	0.55
Set-thru	0.55

Table T3-1 Fitting Span Factor

SBC Size(in)	Minimum allowable first span length (m)
$\frac{1}{4}$	0.7
$\frac{3}{8}$	0.8
$\frac{1}{2}$	0.8
$\frac{3}{4}$	0.9
1	1.1
$1\frac{1}{4}$	1.2
$1\frac{1}{2}$	1.3
2	1.4

Table T3-2 Minimum span length for SBC connected to deck or structural steelwork

SBC Size(in)	Min span length (m)
$\frac{1}{4}$	1.0
$\frac{3}{8}$	1.1
$\frac{1}{2}$	1.1
$\frac{3}{4}$	1.3
1	1.6
$1\frac{1}{4}$	1.7
$1\frac{1}{2}$	1.8
2	2.0

Table T3-3 Minimum span length for SBC connected between two main lines

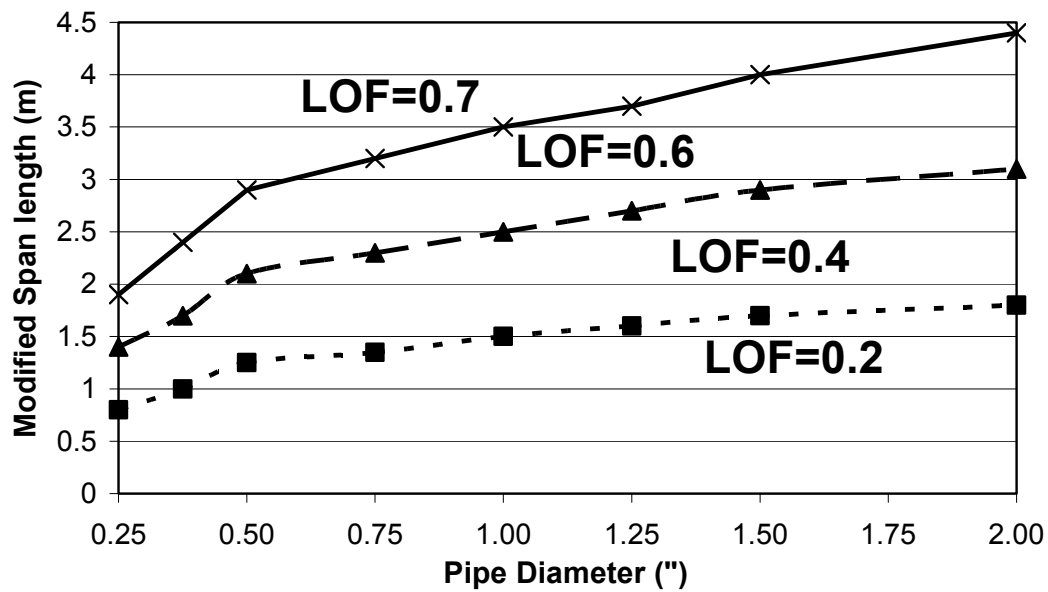


Figure T3-1 Maximum Span connected to main line and involving mass

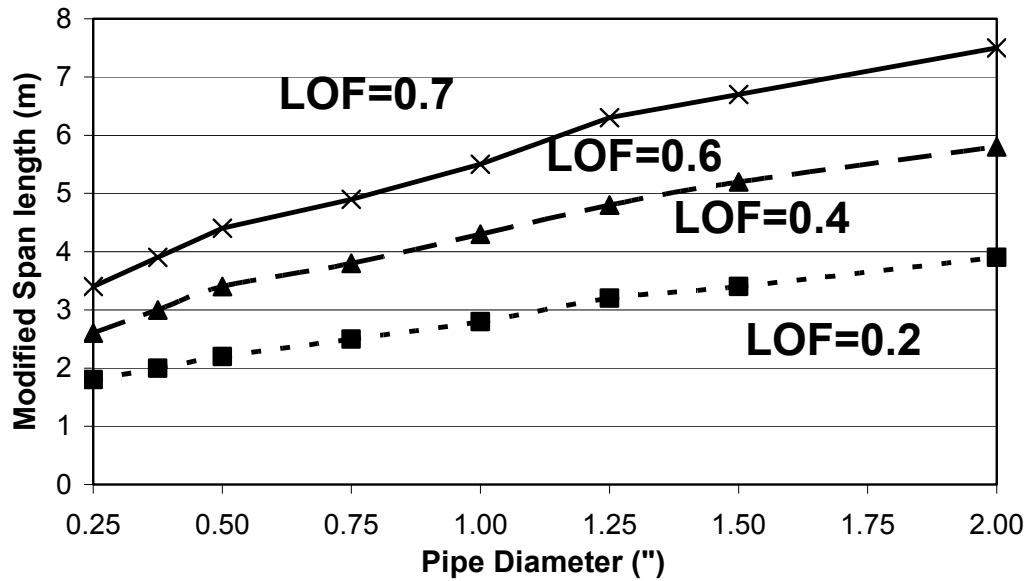


Figure T3-2 Maximum span length connected to main line and with no additional mass

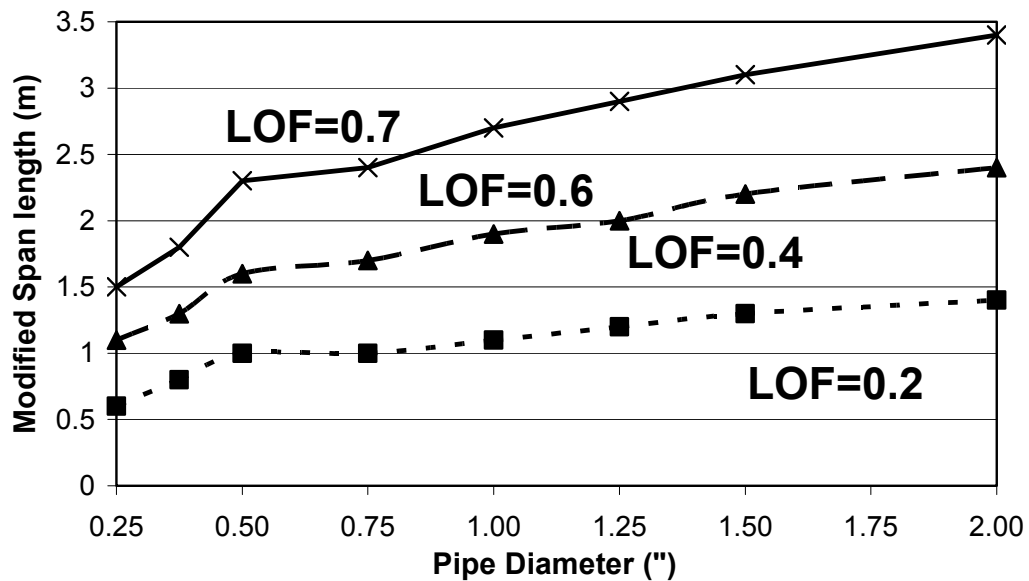


Figure T3-3 Maximum span length for subsequent spans and involving mass

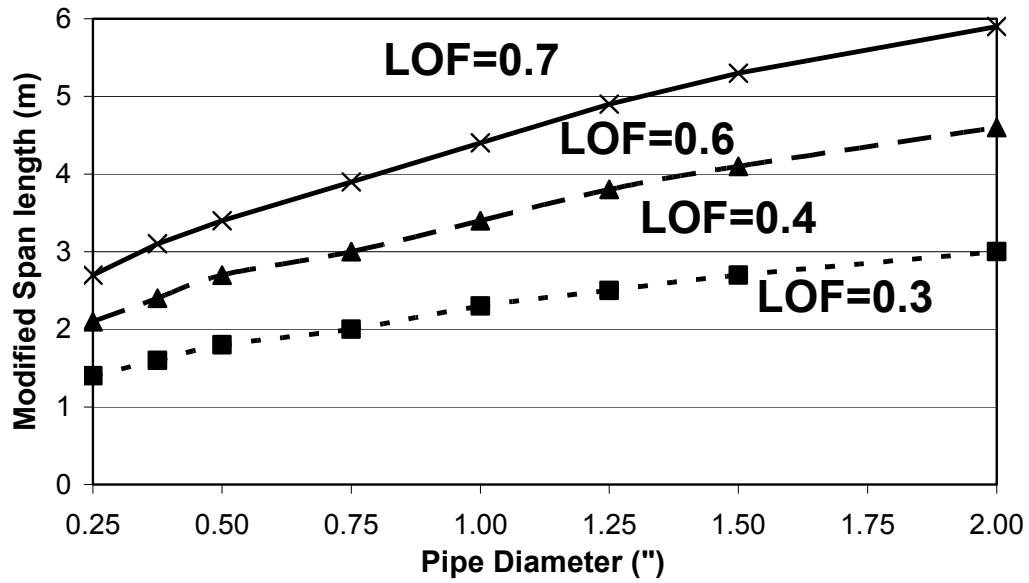


Figure T3-4 Maximum span length for subsequent spans and with no additional mass

Technical module

T4 - QUANTITATIVE THERMOWELL LOF ASSESSMENT

T4.1 INTRODUCTION

This Technical Module considers the excitation of thermowells by vortex shedding.

This technical module is specifically focused on thermowells, with three different geometries (i) straight, (ii) tapered and (iii) stepped, see **Figure T4-1**.

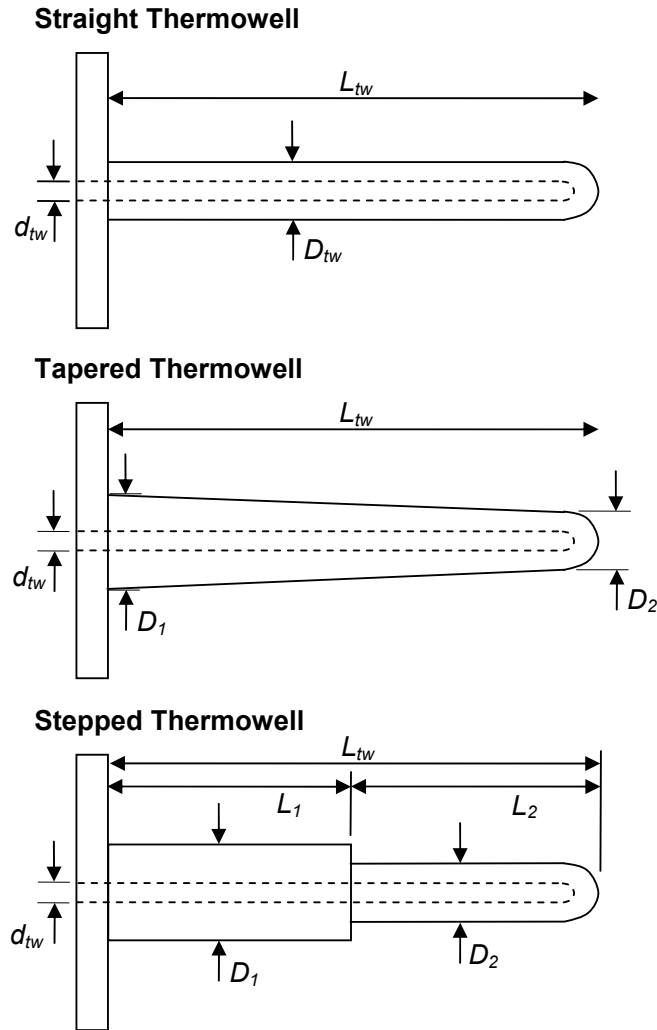


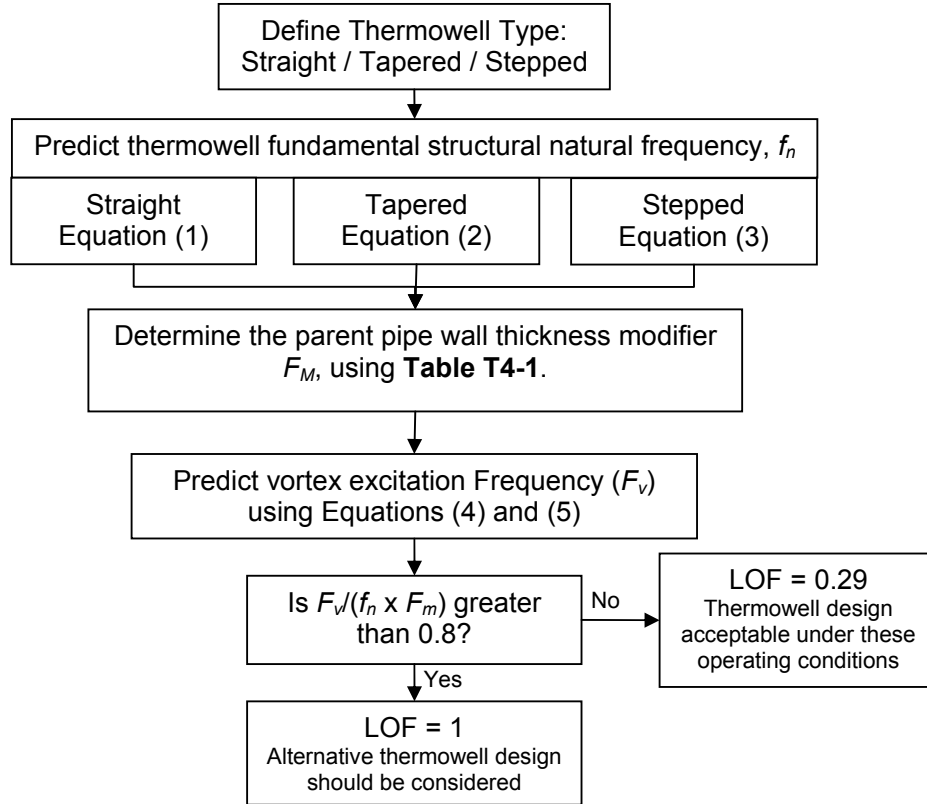
Figure T4-1 Different Geometries of Thermowell

The underlying approach described in this technical module, of considering lock-on for the natural frequency and vortex shedding frequency, is valid for all intrusive elements with similar geometries to those outlined above.

If there are pulsations within the process fluid, or mechanical excitation from nearby equipment, then there is a further possibility that the thermowell could be excited at one of its structural natural frequencies. In this case specialist advice should be sought.

T4.2 QUANTITATIVE THERMOWELL ASSESSMENT

The overview of the thermowell assessment is shown in **Flowchart T4-1**.



Flowchart T4-1 Quantitative Thermowell Assessment

T4.2.1 Thermowell Structural Natural Frequency

The fundamental structural natural frequency of the three types of thermowell is predicted by the following:

Straight Thermowell

$$f_n = \frac{3.516}{2 \pi L_{tw}^2} \sqrt{\frac{E_{tw} I}{\rho A}} \quad (1)$$

Tapered Thermowell [T4-1]

$$f_n = \frac{1.12 D_1}{1000 L_{tw}^2} \sqrt{\frac{E_{tw}}{\rho} \left(\frac{k^4 + 5k^3 + 15k^2 + 35k + 70 - 126\delta^4}{5353k^2 + 2142k + 513 - 8008\delta^2} \right)} \quad (2)$$

Stepped Thermowell

$$f_n = \frac{0.14 D_A}{1000 L_{tw}^2} \sqrt{\frac{E_{tw} (1 + \delta_A^2)}{\rho}} \quad (3)$$

Where:

f_n is the thermowell fundamental structural natural frequency in Hz

D_{tw} is the outside diameter of a straight thermowell in mm

D_1 is the outside diameter at the base in mm

D_2 is the outside diameter at the tip in mm

D_A is the average outside diameter in mm, i.e. $D_A = \left[\frac{D_1 L_1}{L_1 + L_2} + \frac{D_2 L_2}{L_1 + L_2} \right]$

δ is d_{tw}/D_1 where d_{tw} is the internal bore in mm

δ_A is d_{tw}/D_A where d_{tw} is the internal bore in mm

k is D_2/D_1

E_{tw} is the Young's Modulus of the thermowell material in Pa

I is the second moment of area in m^4 , i.e. $I = \frac{\pi}{10^{12}} \left[\frac{D_{tw}^4 - d_{tw}^4}{64} \right]$

A is the cross-sectional area in m^2 , i.e. $A = \pi \frac{D_{tw}^2 - d_{tw}^2}{4 \times 10^6}$

L_{tw} is the length from the support point to the tip of the thermowell in m

L_1 is the length of the largest diameter section on the stepped thermowell in m

L_2 is the length of the smaller diameter section on the stepped thermowell in m

ρ is the density of the thermowell material in kg/m^3

T4.2.2 Parent Pipework Wall Thickness Modifier

The wall thickness of the parent pipe affects the fundamental structural natural frequency of the thermowell, as the connection supporting the thermowell cannot be considered to be infinitely stiff, especially on thin-walled pipe. However, if the connection is locally stiffened using suitable welded gusset plates at 90 degree intervals around the connection then the value of F_M can be increased. The value for the parent pipework wall thickness modifier, F_M , is determined from **Table T4-1**:

Parent Pipe Schedule	Wall thickness modifier, F_M	Wall thickness modifier, F_M , with 4-way welded gussets
Schedule 160 or greater	0.96	0.98
Schedule 80 to less than Schedule 160	0.93	0.96
Schedule 40 to less than Schedule 80	0.85	0.93
Less than Schedule 40	0.42	0.85

Table T4-1 Wall thickness modifier to account for the effect on thermowell structural natural frequency

T4.2.3 Strouhal Number

The Strouhal Number (S) is determined by calculating the Reynolds Number (Re) and using:

$$S = 0.184 + 0.012 \log_{10}(Re) \quad (4)$$

The Reynolds number (Re) is calculated using the approach described in **Section B.9**.

T4.2.4 Vortex Excitation Frequency

The vortex excitation frequency, F_v , is predicted by:

$$\text{Vortex Excitation Frequency, Hz [T4-2]} \quad F_v = \frac{1000 \times S \times v}{D_{Char}} \quad (5)$$

Where:

- F_v is the vortex excitation frequency in Hz
- S is the Strouhal number.
- v is the fluid velocity in m/s
- D_{Char} is the characteristic dimension (mm). For the straight thermowell D_{Char} is D_{tw} and for tapered and stepped thermowells D_{Char} is D_2 .

If there are a number of thermowells in close proximity to each other (within $10 \times D_{Char}$), there is a potential for the vortices generated from the upstream thermowell to excite thermowells downstream. In this case specialist help should be sought.

T4.2.5 LOF Score

For a vortex excitation frequency greater than 80% of the thermowell fundamental structural natural frequency, the LOF is 1. For a vortex excitation frequency less than 80% of the thermowell structural natural frequency, the LOF is 0.29.

Technical module

T5 - VISUAL INSPECTION - PIPING

T5.1 General

The objective of this Technical Module is to provide guidance for the visual inspection of process pipework, i.e. main lines and small bore connections, with specific regard to vibration induced fatigue. Tubing is considered in **TM-06**.

Visual inspection plays an important part in the identification of potential piping vibration issues, either by identifying as-built issues or subjectively high vibration under certain operating conditions.

It is recommended that a visual inspection is undertaken at different operating conditions due to variation in piping vibration with plant operation.

T5.2 Main Process Pipework and Small Bore Connections

T5.2.1 Method

Table T5-1 lists factors to be considered during a visual inspection.

T5.2.2 Users

This technical module has been designed to be used by inspection and/or operations personnel who are familiar with the plant.

T5.2.3 Visual Inspection

It should be noted that some forms of piping vibration are heavily dependent on how the process plant is being operated. The absence of high noise and/or vibration levels during the visual survey should not be taken as necessarily being indicative of there being a low risk from vibration induced fatigue.

Table T5-1 attempts to capture specific aspects associated with the geometry and maintenance of the pipework, and associated elements, which are indicative of potentially fatigue sensitive locations should sufficient levels of excitation be present.

Item	Guidance
1	High vibration/noise? There are three aspects to consider. Each should be considered separately. • Can the pipework be seen to be vibrating? This is indicative of low frequency vibration • Can the pipework be felt to be vibrating? This is indicative of low to medium frequency vibration • Is there high noise from the pipework? This is indicative of high frequency vibration If high vibration and/or noise is identified, then the process and operating conditions of the system under which the vibration and/or noise is apparent should be noted, especially if the problem is intermittent in nature. This would typically include operating pressures and temperatures, the operating regime of nearby equipment (e.g. the position of valves) the load on compressors, the machine running speed etc, and the system throughput including flow rates and fluid densities where feasible. Ideally vibration and/or noise levels should be quantified using an appropriate measurement survey. Details of recommended measurement procedures are given in TM-07.
2	Fretting damage? (refer to Examples T5-1) Fretting occurs when there is contact and relative movement between two surfaces. This movement can be small but can result in significant localised loss of pipe wall thickness. Typical locations to be considered include: • U-bolt pipe clamps, particularly where there is no resilient layer (e.g. tico pad) (refer to Example T5-1a) • Resting supports (refer to Example T5-1b) • Deck penetrations (refer to Example T5-1c) • Loose insulation cladding (refer to Example T5-1d) • Contact between pipes (partial clash) (refer to Example T5-1e) • Pipework in contact with other equipment items (e.g. cable racks, handrails, other fittings, etc) (refer to Example T5-1f) • Temporary supports (e.g. scaffold poles, chain blocks etc.) Where fretting is identified, the items in contact should be separated and appropriate inspection performed to quantify any damage which has been sustained.

Table T5-1 Visual Inspection Guidance (part 1 of 4)

Item	Guidance							
3	Pipe geometry There are several aspects that affect how susceptible a main line is to vibration issues. These are: - As pipework becomes more complex (e.g. greater density of bends, valves, etc) then higher levels of turbulent energy are likely, resulting in higher levels of vibration. - Wall thickness: the thinner the pipe wall thickness, the more susceptible it will be to fatigue. - Sources of turbulent excitation, such as valves, should be suitably supported. - Also be aware of any plant change that has resulted in changes to the pipework (e.g. removal of a section of line resulting in a 'dead leg' which is not supported or poorly supported).							
4	SBC geometry (refer to Examples T5-2) There are several aspects that affect how susceptible a connection is to fatigue damage. These are: - Type of fitting: this determines the stress concentration at the fatigue sensitive location. <table border="1"> <tr> <td rowspan="6"> Good  Poor </td> <td>Short contoured body</td> </tr> <tr> <td>Contoured body / welded tee / forged reducing tee</td> </tr> <tr> <td>Weldolet / threadolet fully back welded / screwed fully backed welded</td> </tr> <tr> <td>Threadolet / screwed</td> </tr> <tr> <td>Threadolet partially back welded / screwed partially back welded</td> </tr> <tr> <td>Set-on / set-in / set-through</td> </tr> </table> - Length of fitting: the longer the fitting from the connection to the parent pipe to any unsupported mass (e.g. valves, flanges, etc) on the connection, the more susceptible the fitting will be to fatigue. (refer to Example T5-2a) - Mass loading on end of connection: the larger the mass, the more susceptible the fitting will be to fatigue. - Diameter of fitting: the smaller the diameter, the more susceptible the fitting will be to fatigue. Note that some connections will reduce down in diameter along the length of the small bore connection and therefore the most fatigue sensitive location may not be at the connection to the parent pipe. (refer to Example T5-2b) - Parent pipe schedule: the thinner the parent pipe wall thickness, the more susceptible the fitting will be to fatigue. Note that the use of duplex alloys often results in a thinner pipe wall than for the equivalent carbon steel section. - Location of connection on parent pipe: if the small bore connection is located at or close to an anchor location on the parent pipe then the connection will be less susceptible to fatigue than if it is located at mid span or close to discrete sources of energy in the pipework (e.g. control valves, orifice plates, etc).	Good  Poor	Short contoured body	Contoured body / welded tee / forged reducing tee	Weldolet / threadolet fully back welded / screwed fully backed welded	Threadolet / screwed	Threadolet partially back welded / screwed partially back welded	Set-on / set-in / set-through
Good  Poor	Short contoured body							
	Contoured body / welded tee / forged reducing tee							
	Weldolet / threadolet fully back welded / screwed fully backed welded							
	Threadolet / screwed							
	Threadolet partially back welded / screwed partially back welded							
	Set-on / set-in / set-through							

Table T5-1 Visual Inspection Guidance (part 2 of 4)

Item	Guidance
5	Pipe supports (refer to Examples T5-3) In the majority of cases, the more effectively a pipe run is supported the less susceptible it will be to vibration. Aspects to consider are: • Damaged or missing supports. (refer to Example T5-3a) • Temporary supports (wood chocks, slings, 'temporary' load out supports still in place, etc). (refer to Example T5-3b) • Pipe supports insufficiently stiff relative to the supported pipe (e.g. goalpost supports with small section and no cross bracing, pipe racks with poor lateral or transverse stiffness, etc). (refer to Example T5-3c) • Pipe supports not acting as intended (e.g. pipe lifted off resting support due to thermal growth or poor design/constructions, spring hangers incorrectly set or sized, etc). (refer to Example T5-3d) • Fretting between pipe and supports (refer to Item 2 above and Examples T5-1)
6	Bracing of SBCs (refer to Examples T5-4) There are several aspects to consider which can lead to potential fatigue issues even when a connection has been braced or clamped: • Unsuitable bracing applied (e.g. wood blocks, rope, cable ties, etc) (refer to Example T5-4a) • Brace/clamp not stiff enough to provide adequate support (refer to Example T5-4b) • Brace/clamp not supporting free mass on end of connection (refer to Example T5-4c) • Brace/clamp protecting first weld only (refer to Example T5-4d) • Brace/clamp not completely restraining connection (e.g. braced in only one plane) (refer to Example T5-4e) • Connection braced to deck, neighbouring structure or adjacent pipework rather than back to parent pipe (refer to Example T5-4f) • Use of welded gusset plates on pipework without reinforcing plates (potential punch through issue), with particular reference to thin walled pipes. • Damaged or missing braces/clamps (including missing bolts, corrosion, etc) (refer to Example T5-4g) • Regular checks should be made to ensure that bolts remain tight on bolted clamps.

Table T5-1 Visual Inspection Guidance (part 3 of 4)

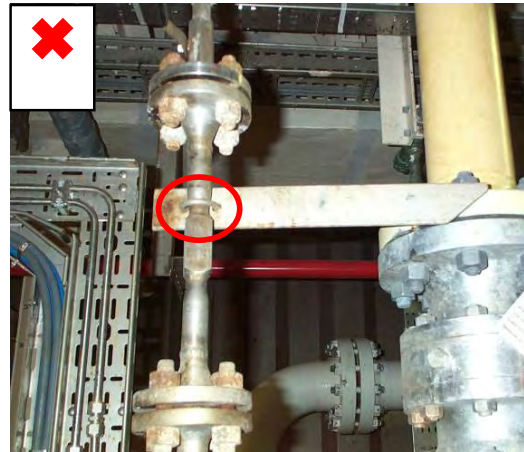
Item	Guidance
7	Other vibration control measures (refer to Examples T5-5) There are a number of vibration control measures that can be applied to pipework. Key aspects in terms of ensuring the control measures (if fitted) are fit for purpose include: • Gas filled pulsation dampeners: check use of correct pre-charge pressure (refer to Example T5-5a) • Viscous dampers: check for lock-up of damper; dirt ingress due to damaged or incorrectly fitted cover (refer to Example T5-5b) • Hydraulic vibration snubbers: ensure that they are functioning as designed and have not slackened/seized.
8	Vibration transmission to other pipework Vibration transmission can occur from high energy systems to other pipework not directly associated with that system, i.e. • Through shared supports e.g. pipe rack • From machinery skids to neighbouring pipework
9	Other considerations There are several additional aspects to be aware of which can have a detrimental effect on the vibration induced fatigue resistance of the pipework. These include: • Corrosion • Erosion • Poor weld quality and profile • Mechanical damage There also excitation mechanisms that should be considered when undertaking a visual inspection as they may not be identified otherwise. These include: • Environmental loading (e.g. wind, wave, seismic) • Slugging

Table 5-1 Visual Inspection Guidance (part 4 of 4)

EXAMPLE T5-1 FRETTING



Lining provides protection to line from fretting at the U-Bolt

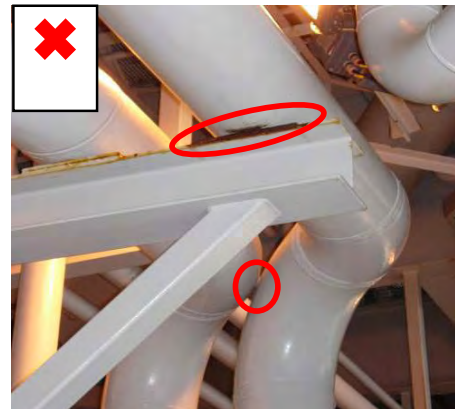


U-bolt is attached to the connection on a reducer section and is not lined and susceptible to fretting damage

Example T5-1a Fretting, good & poor practice at U-bolt pipe clamps



Reinforcement plate at rest support to resist fretting damage on pipe.



Fretting damage to main pipe; no resilient pad between support and pipe; also pipe clash

Example T5-1b Fretting, good & poor practice at rest supports



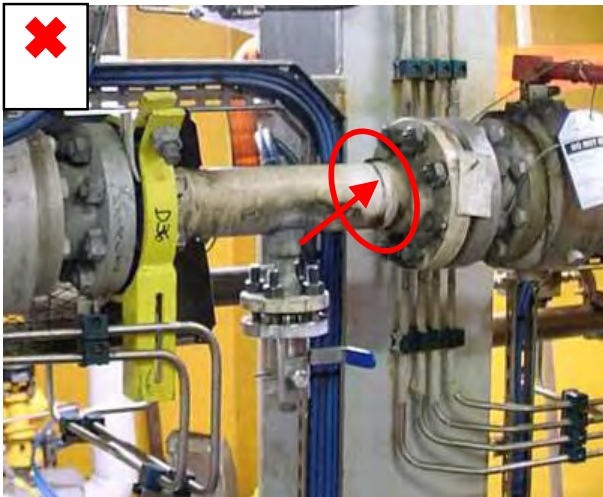
Resilient pad between support and pipe protects against fretting damage



Fretting damage to pipe caused by pipework vibrating relative to deck penetration cover

Example T5-1c

Fretting, good & poor practice at deck penetrations



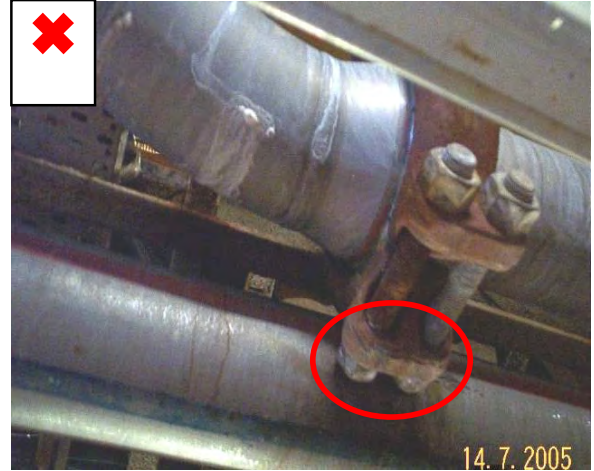
Fretting due to loose cladding and damage caused by knife edge contact at insulation end cap (existing cladding has been removed)

Example T5-1d

Fretting, poor practice due to loose insulation cladding



Clash between pipes resulting in fretting damage



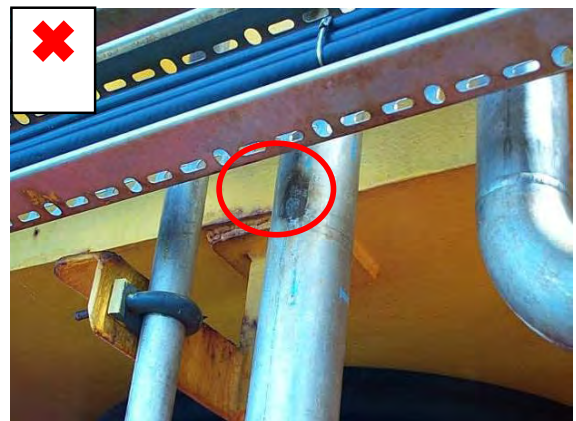
Fretting damage due to contact between well flow line and greylock fitting on adjacent flow line

Example T5-1e

Fretting, poor practice due to contact between pipework



Pipeline contact to cable rack resulting in fretting damage



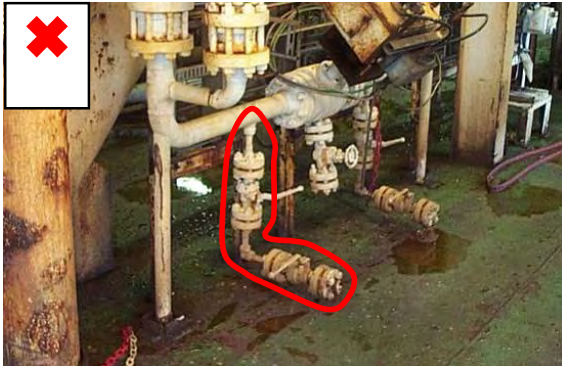
Fretting damage between pipeline and cable tray



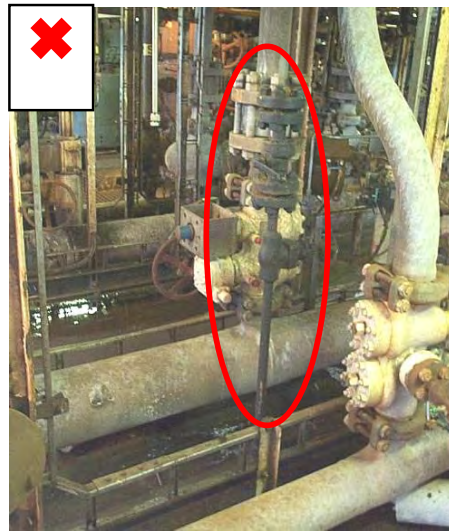
The screw and nut used to mount a temp gauge in contact with pipe, resulting in penetration of sch. 160 pipe

Example T5-1f Fretting, poor practice due to contact with other equipment items

EXAMPLE T5-2 SBC GEOMETRY

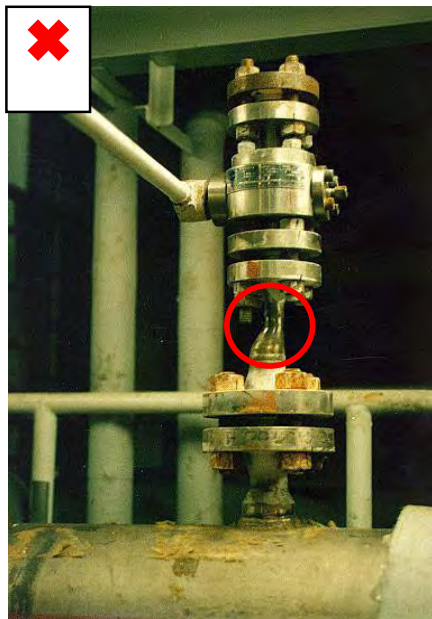


Large cantilevered mass with poor geometry



Long straight connection, example of a cantilevered mass

Example T5-2a SBC geometry, poor practice of cantilevered mass



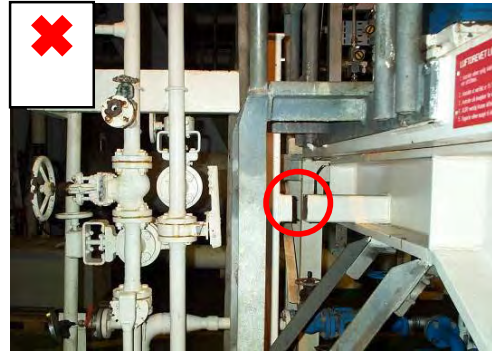
Necked down connection and large cantilevered mass

Example T5-2b SBC geometry, poor practice with a necked down connection

EXAMPLE T5-3 MAIN LINE SUPPORTING



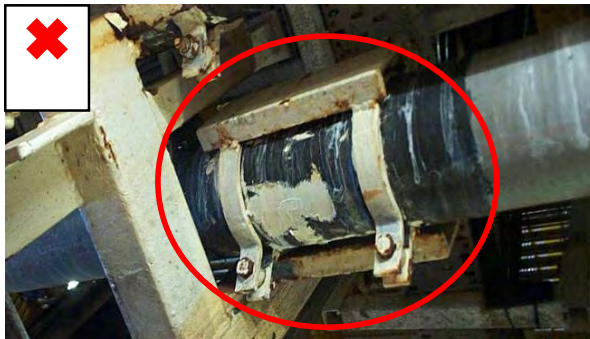
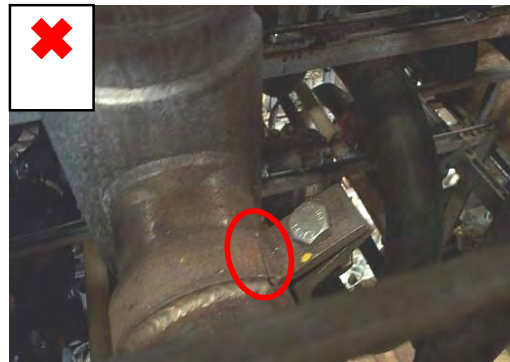
Concrete pipe support plinth detached from ground



Part of structural support has been removed. Pipework is flexible and insufficiently stiff.



Support cracked

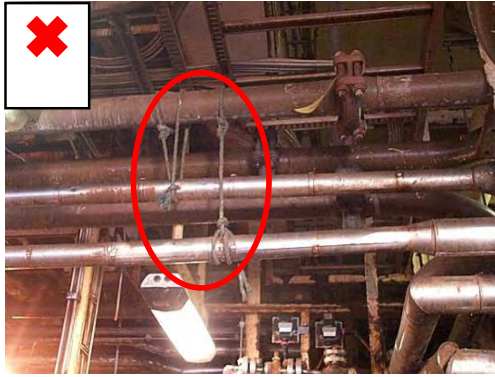


Pipework guide support slid off hanger, allowing the pipework to vibrate

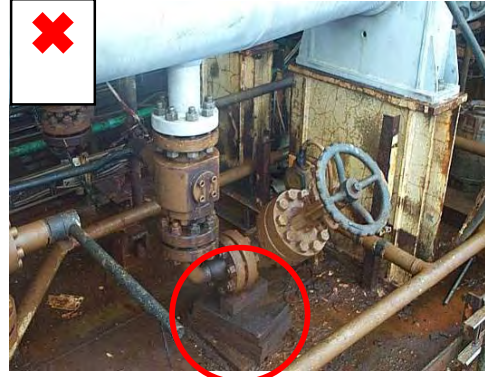


Conductor riser guide – Minimise gap and use low friction pads where necessary

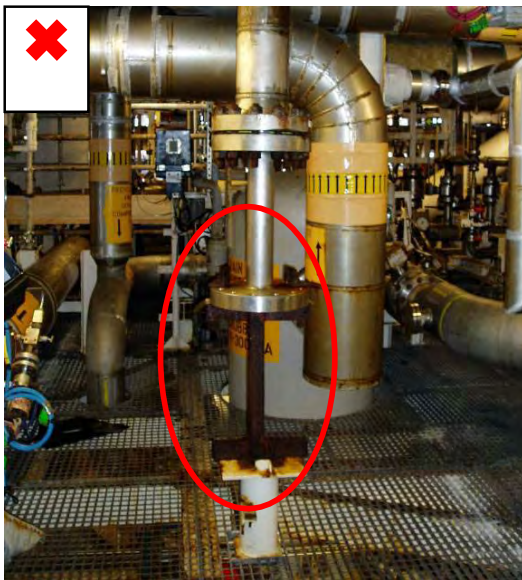
Example T5-3a Main line supports, poor practice of damaged or missing supports



Rope used to support pipework. In addition pipe suspended from another pipe.



Wooden blocks used to support SBC

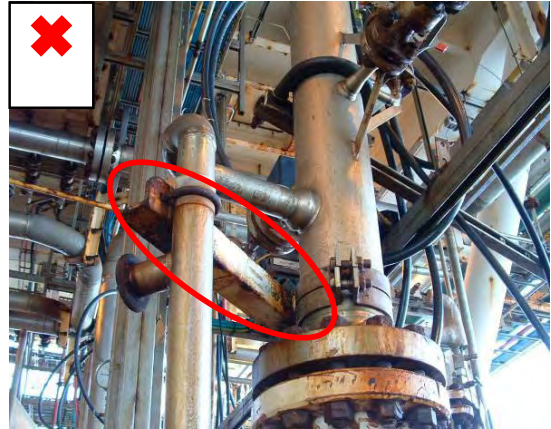


Use of temporary support on end of dead-leg pipe which provides little lateral stiffness

Example T5-3b Main line supports, poor practice in the use of temporary supports



U-Bolt providing little or no restraint to vibration; no lining between U-Bolt and pipe



Attempt to restrain main pipework by use of U-Bolt and strut (little/no lateral support)

Example T5-3c Main line supports, poor practice of supports insufficiently stiff



Pipework clear of resting support, due to thermal growth (air gap)



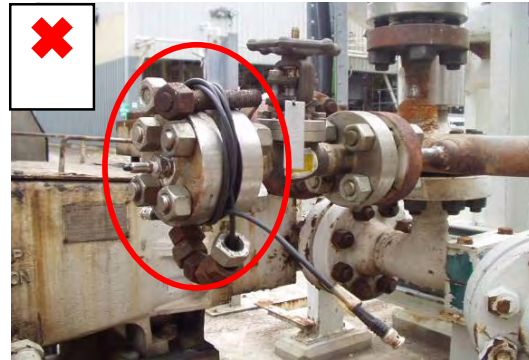
Pipework clear of resting support (air gap)

Example T5-3d Main line supports, poor practice of supports not acting as intended

EXAMPLE T5-4 BRACING OF SBCS

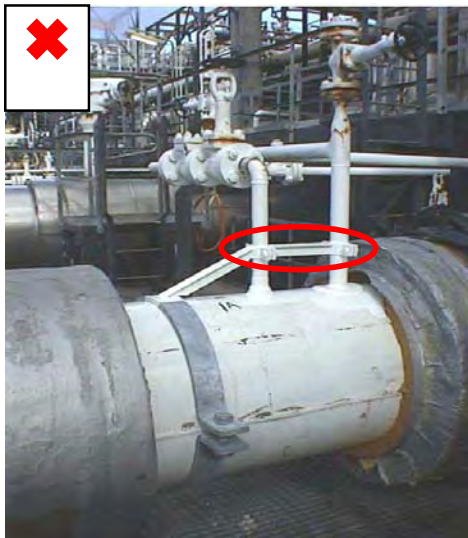


Rope used to support cable tray

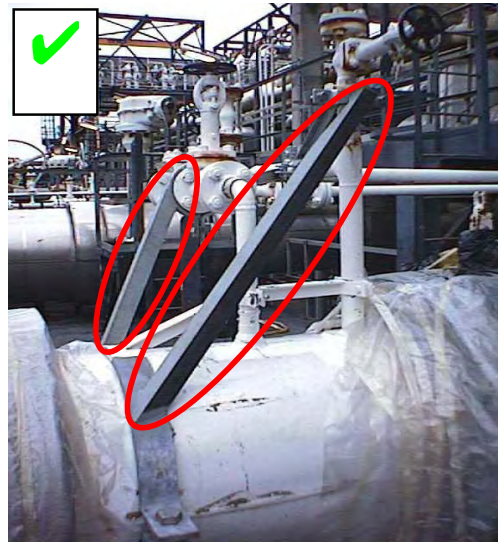


'Temporary' fix of "mass loading" to detune a structural resonance still in place some time later

Example: T5-4a Bracing of SBCs, poor practice of bracing

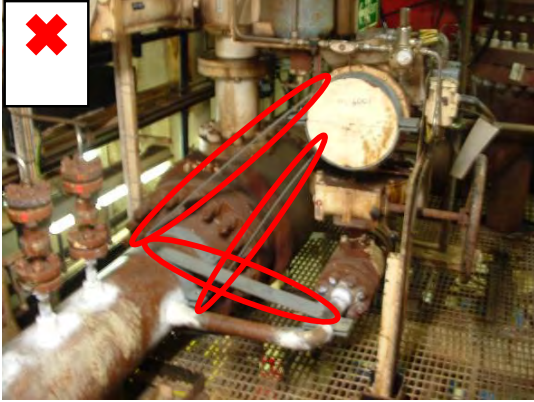


Bracing insufficiently stiff; single plane only; only protecting weld to parent pipe

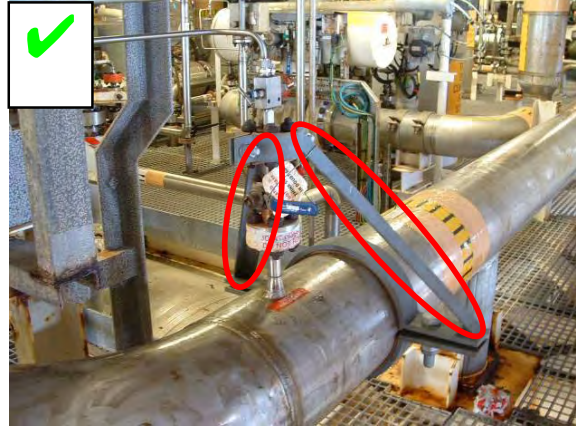


Bracing stiffness increased; diagonal brace protects in two planes; valves now supported

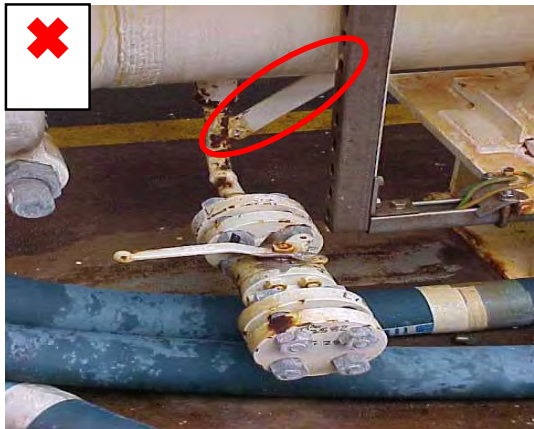
Example: T5-4b Bracing of SBCs, poor practice of brace with insufficient stiffness



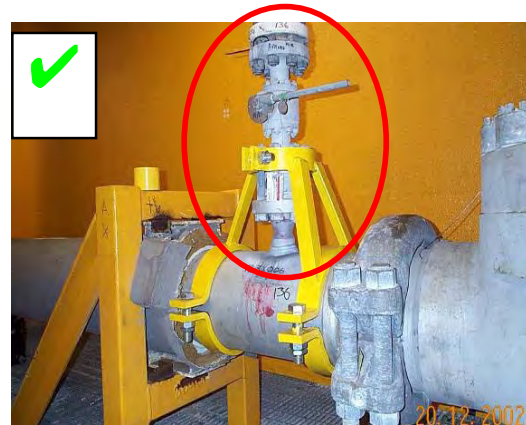
Flat bar used as support, poor tangential stiffness



Example of a brace which provides good two plane support to free mass by use of diagonal members



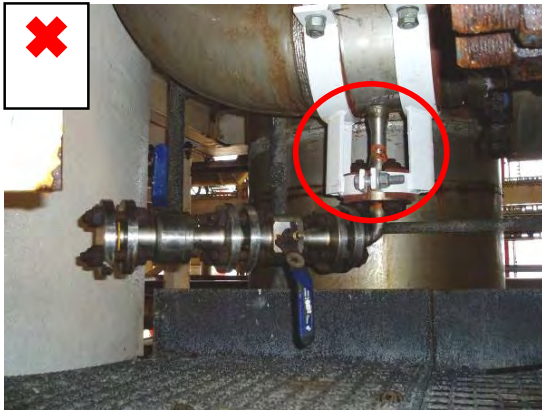
Connection braced at small bore pipe using flat bar, no support provided to the valve and potential punch through threat.



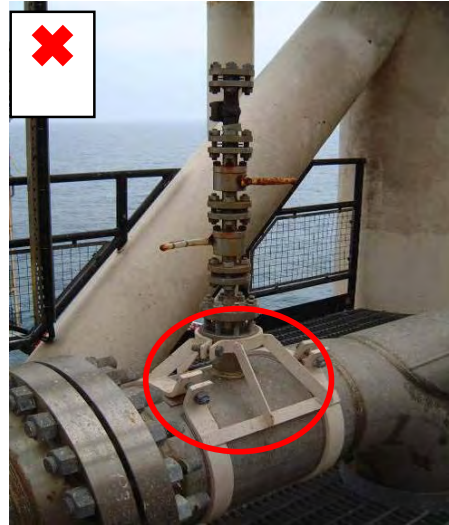
Good support of cantilevered mass

Example: T5-4c

Bracing of SBCs, poor practice of not supporting free mass on end of connection

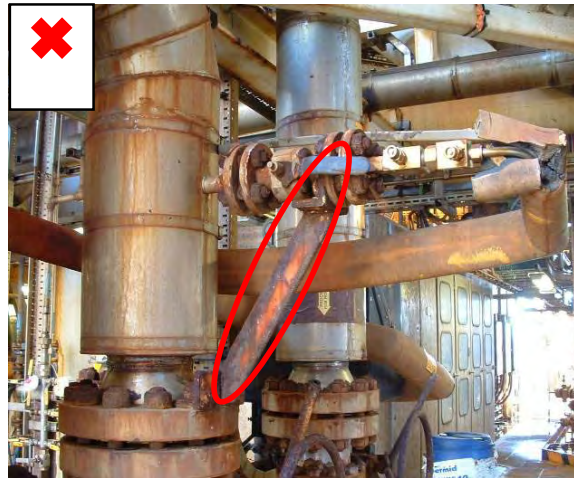
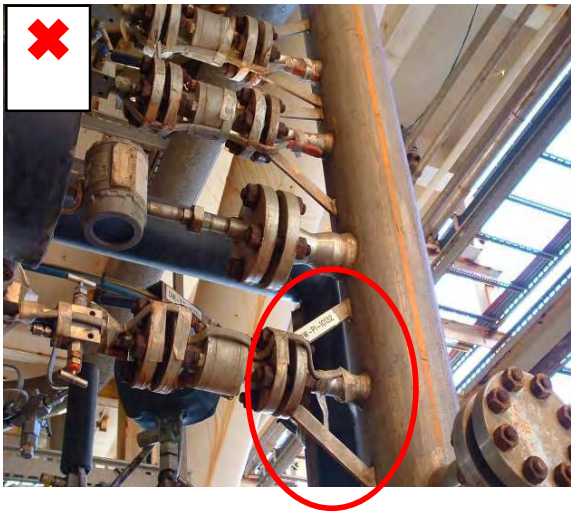


Brace only protects welded connection to parent pipe. Down stream elbow welded connection unprotected



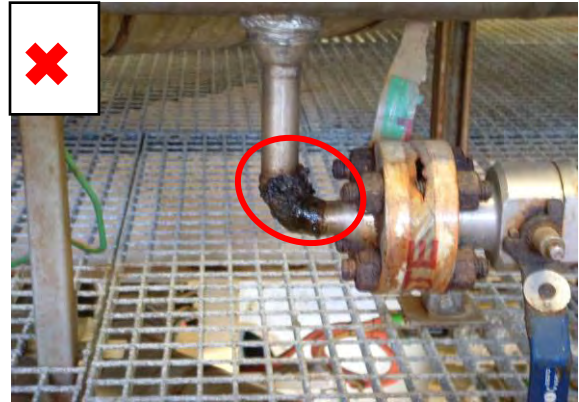
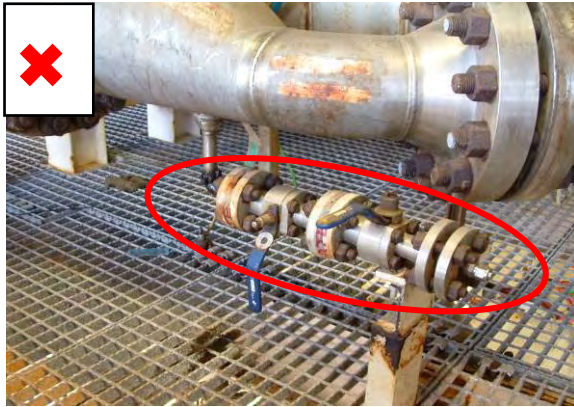
Brace only protects first weld

Example: T5-4d Bracing of SBCs, poor practice of protecting first weld only

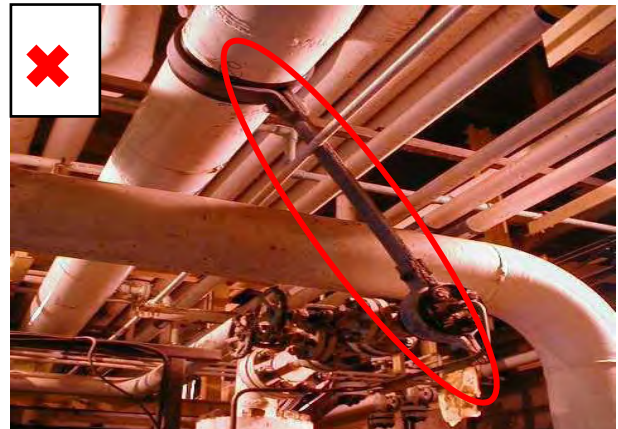
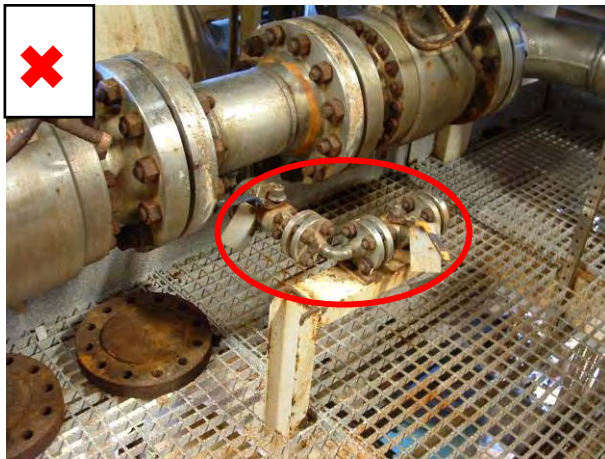


Connections braced in one plane only using flat bar – little lateral support and potential punch through threat.

Example: T5-4e Bracing of SBCs, poor practice of not completely restraining connection



Connection braced to deck. Combination of static (axial) loading and vibration leading to failure



Connection on resting support back to deck,
Friction between support and connection
means that connection is effectively "clamped"

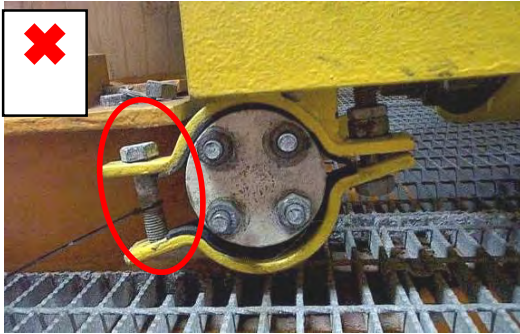
Connection handcuffed to adjacent pipe rather
than parent pipe



SBC supported to deck

Example: T5-4f
structure

Bracing of SBCs, poor practice of bracing to deck or neighboring



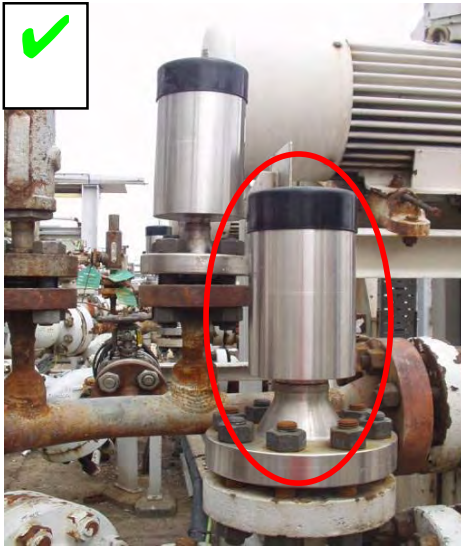
Example of clamp not been re-instated correctly after intervention work on line



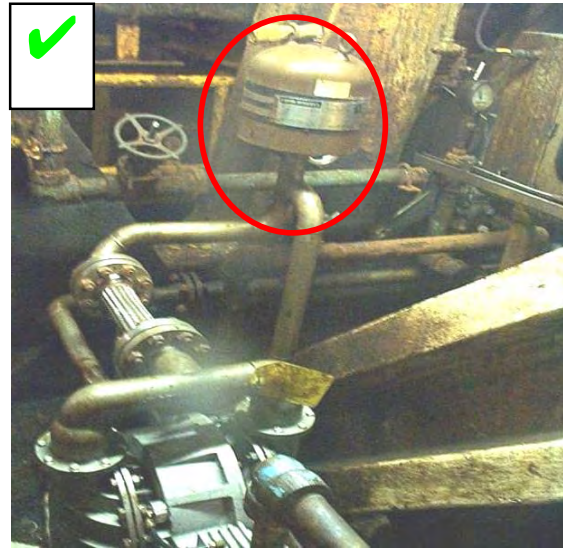
Good SBC bracing to parent pipe

Example: T5-4g Bracing of SBCs, poor practice of damaged or missing braces

EXAMPLE T5-5 OTHER VIBRATION CONTROL METHODS



Pre-charged pulsation dampeners



Pre-charged pulsation dampeners

Example: T5-5a Other vibration control methods, good practice of pulsation dampeners



Viscous damper

Example: T5-5b Other vibration control methods, good practice of viscous dampers

Technical module

T6 - VISUAL INSPECTION - TUBING

T6.1 GENERAL

Small bore tubing systems are extensively used in industrial processes and historically they are known to be a major contributor to the incidence of process and hydraulic fluid releases. The mechanical characteristics of these systems make them economically attractive because of their ease of installation and they can, by design, provide the necessary integrity over the installation life cycle. Tubing and connectors range in size from 1/8" to 2" diameter. Their geometry is often complex involving use of many in-line junction connectors and fittings.

To prevent the loss of integrity of the instrument tubing it is essential that it is regularly inspected to ensure that there is no damage, either in the form of broken or ineffective supports, or onset of corrosion or tube distortion.

Section T6.3 overviews commonly encountered tubing damage mechanisms, and general good practice in addressing them. **Table T6-1** lists factors to be considered as part of the visual inspection.

T6.2 MODE OF FAILURE

T6.2.1 Mechanical Damage at Instrument Tubing Connector or Support

T6.2.1.1 Damage Mechanism

Mechanical failure of tubing systems occurs predominantly at connections. Due to the tubing not involving welded connections, sections of tubing have a greater allowable level of dynamic stress before fatigue cracking will be initiated. If the dynamic loading/stress is sufficiently high, failure typically occurs at the connection, where damage can initially be caused during construction/re-assembly.

In addition to fatigue cracking, excessive displacement can result in localised plastic deformation in the form of creasing and buckling.

T6.2.1.2 Location of Damage

The damage occurs on the tubing at the point it enters the connector or support.

T6.2.1.3 Good Practice

Minimise vibration: It is the relative displacement between the pipework and instrument tubing that results in the damage at the tubing connection. By reducing the main line vibration levels the relative displacement will be reduced.

Where possible the instrument should be connected directly to the main line rather than to neighbouring structure, therefore removing the relative displacement issues.

In addition, tubing which is poorly supported is also susceptible to vibration damage.

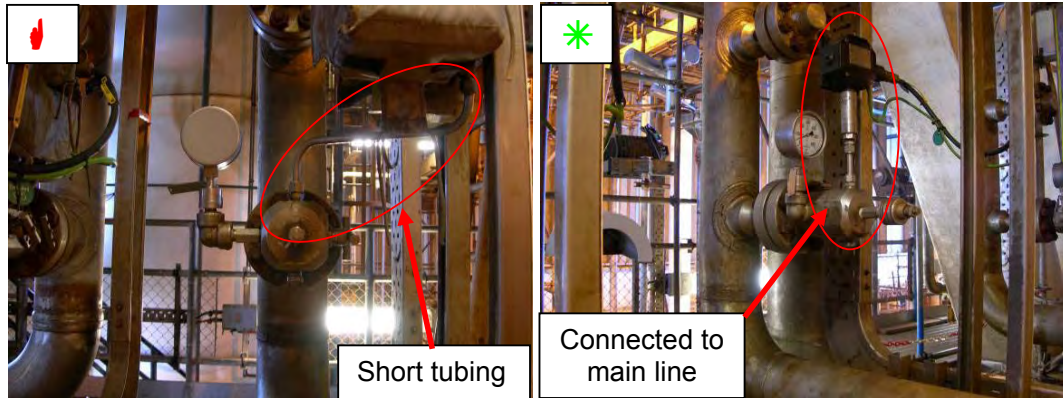


Figure T6-1 Example of instrument being connected to structure and directly to the main line

In addition, tubing which is poorly supported is also susceptible to vibration damage.

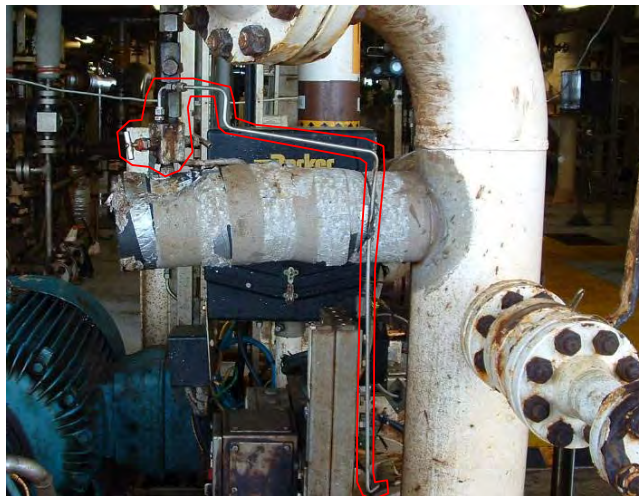


Figure T6-2 Example of poorly supported tubing

Design: The design of the tubing should allow differential movement of the two connecting items, i.e. there should be no direct tubing connection between two points.

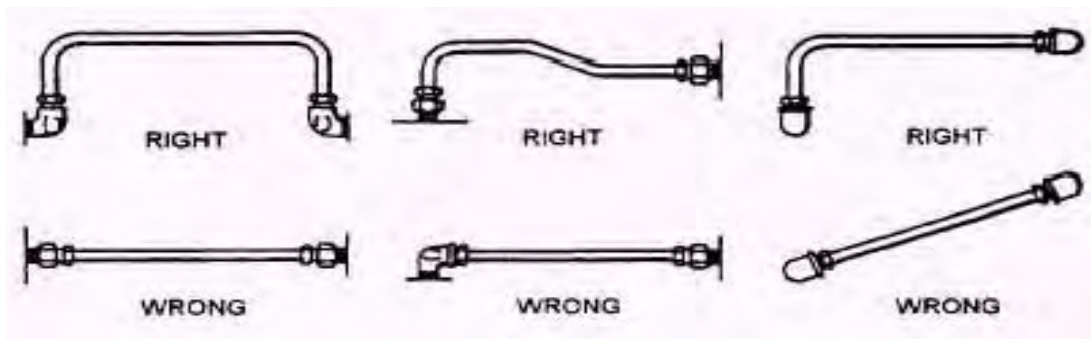


Figure T6-3 Correct and incorrect methods of installing tubing

Pigtails: Are used to allow greater differential movement between two connecting items, such as control instrument tubing off a flowline. There should be a minimum of 2½ loose turns on a pigtail. The pigtail should be located close to a support/termination point, therefore supporting the additional mass concentration.

Stress Raisers: As part of the construction or re-assembly of instrument tubing connections damage can occur; this will act as a stress concentrator.

T6.2.2 Fretting

T6.2.2.1 Damage Mechanism

Fretting wear occurs between tight-fitting surfaces subjected to cyclic relative motion, typically of extremely small amplitudes, resulting in one or both of the surfaces being worn away. This can occur in instrument tubing if there is contact with external structures, or if supports are ineffective and allow movement.

T6.2.2.2 Location of Damage

The fretting damage occurs at the point of contact with the external structure or at ineffective supports.

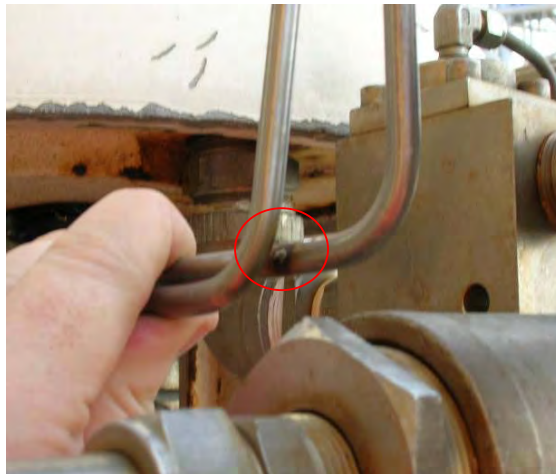


Figure T6-4 Example of fretting

T6.2.2.3 Good Practice

Ineffective supports and mountings: During visual inspection look for supports which have become loose and thus ineffective. Damage due to poor routing of the tubing tends to result in loosening off of the mountings, or damage to the connections.

Fretting at Supports: Where there is a poorly designed support, which allows motion, there is a risk of fretting.

Minimise vibration: The greater the level of vibration the greater the likelihood of fretting damage if there is contact with other structures or loose supports.

T6.2.3 Loosening at Connector Fittings

T6.2.3.1 Damage Mechanism

Any differential movement can cause the tubing connection to loosen, allowing weeping/leaking of the connection. In cases where the vibration level is sufficiently high and/or the construction is poor, the tubing can be ripped from the connections.



Figure T6-5 Example of weeping at the tubing connection

T6.2.3.2 Location of Damage

The damage will occur at the interface with the connector and instrument tubing.

T6.2.3.3 Good Practice

The corrective actions are the same for Mechanical Damage at Instrument Tubing Connector or Support, refer to **Section T6.2.1.3**.

T6.2.4 General Good Practice

Support Mass: Any mass upon the tubing, such as valves, gauges and instruments, should be supported. Any pigtails which have a significant number of turns, and therefore localised mass, should be located close to a support.

Disconnected Tubing: Tubing that is disconnected should be removed or suitably supported. The increased flexibility of the disconnected tubing will make the connection fitting more susceptible to vibration induced issues.



Figure T6-6 Example of disconnected tubing which has not been supported

Good construction & Maintenance: It is important that the instrument tubing has been designed, constructed and maintained to a suitable standard and appropriate components have been used, e.g. do not mix fittings of different types, ensure correct assembly. Details can be found in the "Guidelines For The Management, Design, Installation & Maintenance Of Small Bore Tubing Systems" [T6-1].

Flexible hose: Flexible hose is an alternative connection type for instrumentation in cases where there is significant main line movement and should be considered as a replacement where appropriate. Details on flexible hosing are found in UKOOA's Flexible Hose Management Guidelines [T6-2].

T6.3 ASSESSMENT

T6.3.1 Measurement

There is no appropriate measurement technique for non-specialists to assess the tubing condition and assessment is made against good practice, via a visual inspection.

T6.3.2 Visual inspections

All instrument tubing should be visually inspected to ensure that the installation follows the good practice outlined in this document. As the likelihood of damage to the instrument tubing is affected by the vibration level of the main line to which it is connected, the main line LOF should be used to prioritise the order in which the tubing is inspected.

For a given instrument tubing run the questions in **Table T6-1** should be considered as part of the visual inspection. Where any of the outcomes are "yes" the relevant actions should be considered.

No.	Consideration	Response “Yes” - Action
1	Is the main line subject to vibration?	Review the design to ensure it is suitable.
2	Are there insufficient bends or pigtails, making the tubing inflexible and unable to accommodate the main line movement?	Consider replacing the tubing with a more suitable design. Check the connector interface for signs of weeping/leaking/ damage.
3	Is there evidence of damage at the point where the tubing enters a connector?	Replace the existing tubing and/or connection and if appropriate, alter the design taking into account the good practice guidelines.
4	Any there any signs of weeping/leaking?	Replace the existing tubing and/or connection and if appropriate, alter the design taking into account the good practice guidelines.
5	Is there any evidence of damage at the tubing supports?	Replace the tubing and consider alternative support arrangements, taking into account the good practice guidelines
6	Are the supports ineffective or loose?	Replace the tubing if there are any signs of damage. Install effective supports.
7	Is there any contact with other structures along its span?	Replace the tubing if there are any signs of damage. Reroute the tubing to avoid contacts.
8	Are any of the masses unsupported?	Install additional supports.
9	Is any disconnected tubing unsupported?	Remove, support or minimise the tubing length.
10	Does the tubing involve long unsupported runs, leading to excessive vibration?	Install additional supports.

Table T6-1 Considerations During Visual Inspection

Technical module

T7 - BASIC PIPING VIBRATION MEASUREMENT TECHNIQUES

T7.1 GENERAL

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.

Two main survey techniques are commonly employed to determine the risk of vibration-induced process pipework fatigue failure. These are as follows:

- Use of vibration velocity measurements. Generally, the use of vibration velocity measurements provides a simple method for screening a piping system for potential problems. However, it is not a fail-safe assessment technique. The major advantage is the relative ease of obtaining the measurements, while the main disadvantage is that an estimate of the fatigue life cannot be derived directly from the measured data
- Direct dynamic strain measurements using either permanent or portable strain gauges – This provides a full and robust assessment of the likelihood of a fatigue failure of a critical piping system and its components. It enables dynamic stress to be calculated, which is used to determine susceptibility to failure by fatigue. The main disadvantages are that more specialist equipment is required and the location of the strain gauges is critical to obtaining a representative stress measurement. Note: where dynamic strain/stress measurements are required this is outside the scope of these Guidelines and specialist advice should be sought (refer to **TM-08**)

This technical module provides guidance on the use of vibration velocity measurements, and the interpretation of measured data.

T7.2 VIBRATION

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.

It is essential that the operating conditions of the plant are considered at the time of the survey and that the measurements are made during the most onerous operating conditions. Where more than one operating condition is believed to result in significant vibration levels, measurements should be made at each of these conditions.

The level of vibration provides an indication of the risk of damage. However it does not provide a direct measure of dynamic stress.

T7.2.1 Assessment Technique

The assessment method consists of the following steps:

- Use an appropriately configured vibration data logger, (refer to **Section T7.2.1.3**) to record vibration velocity spectra.

- Vibration measurements should ideally be made at a number of locations, to ensure that the maximum value is captured, including those locations which subjectively appear to have the highest amplitude.
- For main lines, the position of the transducer should be at the location exhibiting the highest level of vibration; typically mid span or at unsupported locations.
- The maximum vibration level obtained from measurements in three axes should be used.
- For small bore connections, measurements should be performed at the end flange of the cantilever arrangement. If the SBC arrangement consists of more than one valve, then measurements should be performed at the furthest flange from the connection to the main pipe, as illustrated in **Figure T7-1**.

The vibration velocity spectra are then assessed against the criteria given in **Figure T7-2**.

Measurement Location -  (Note, measurements should be made in all three perpendicular directions)

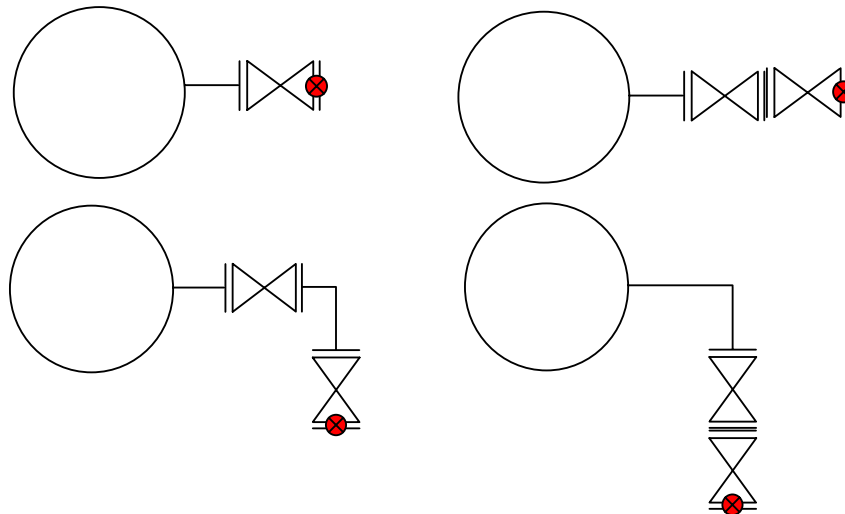


Figure T7-1 Location for vibration measurement on a SBC

T7.2.1.1 Selecting an Accelerometer

Many commonly available accelerometers have a relatively low maximum operating temperature (up to approximately 120 degrees C). Therefore, when measurements are being considered on high temperature pipework it should be ensured that the accelerometer to be used is appropriate for these conditions.

On most pipework the mass of the accelerometer and mounting block is insignificant compared to the mass of the pipework. However, if this is not the case, and the mass of the accelerometer and mounting block does become significant, then this can invalidate the measured data.

Care should also be taken to ensure the accelerometer has a flat frequency response over the frequency range of interest and has a suitable sensitivity.

T7.2.1.2 Attaching Accelerometers to the Pipework

There are a number of ways of connecting the accelerometer to the pipework to ensure that a consistent and representative measurement is obtained.

Magnetic Pipework: The magnet should have a wedged dual-rail base for mounting on curved surfaces, such as pipes, flanges and small bore connections. The magnet should be positioned such that the rails are aligned parallel to the pipe or SBC.

Ensure that the accelerometer is firmly secured to the magnet and that the whole assembly is not able to 'rock' in any direction.

Non-magnetic Pipework - Metallic Washer: A metallic washer should be glued to the required location using a suitable epoxy. Care is required to ensure that the pipe surface is clean prior to gluing. Once the glue has fully cured, the accelerometer can be mounted using the magnetic accelerometer mount.

Consideration is required to ensure the epoxy glue is applicable to the temperature range considered.

The washer and glue should be removed after the measurement has been performed.

Non-magnetic Pipework – Banding: Stainless steel banding can be used to secure the accelerometer arrangement to the pipework. The banding should be sized to the particular pipe or flange diameter of interest. The banding is typically secured using a ratchet or screw locks.

Non-magnetic Pipework – Stud: for non-magnetic fittings consider adhesive or stud mounting (this may be useful if a regular monitoring programme is to be established).

T7.2.1.3 FFT Analyser/Data Logger Setup

The following list describes a typical analyser setup:

- The FFT analyser/data logger should be set up to measure the root mean square (rms) vibration velocity amplitude in mm/s.
- Set frequency range to 0 to 300 Hz, or next highest available range.
- Set resolution (i.e. number of spectral lines) to greater than 300:- typically 800 or 1600 (this will ensure a frequency resolution of better than 1 Hz).
- Use a "Hanning" window (a typical function on a data logger).
- Use at typically least 10 frequency averages.
- Use a root mean square (rms) average.
- If an accelerometer is employed, integrate the signal to velocity in the analyser. A displacement proximity transducer is not acceptable.

T7.2.2 Vibration Assessment Criteria

The vibration assessment criteria for both the main pipe and small bore connections are given in **Figure T7-2**, using the measured RMS levels and peak frequency of the measured response.

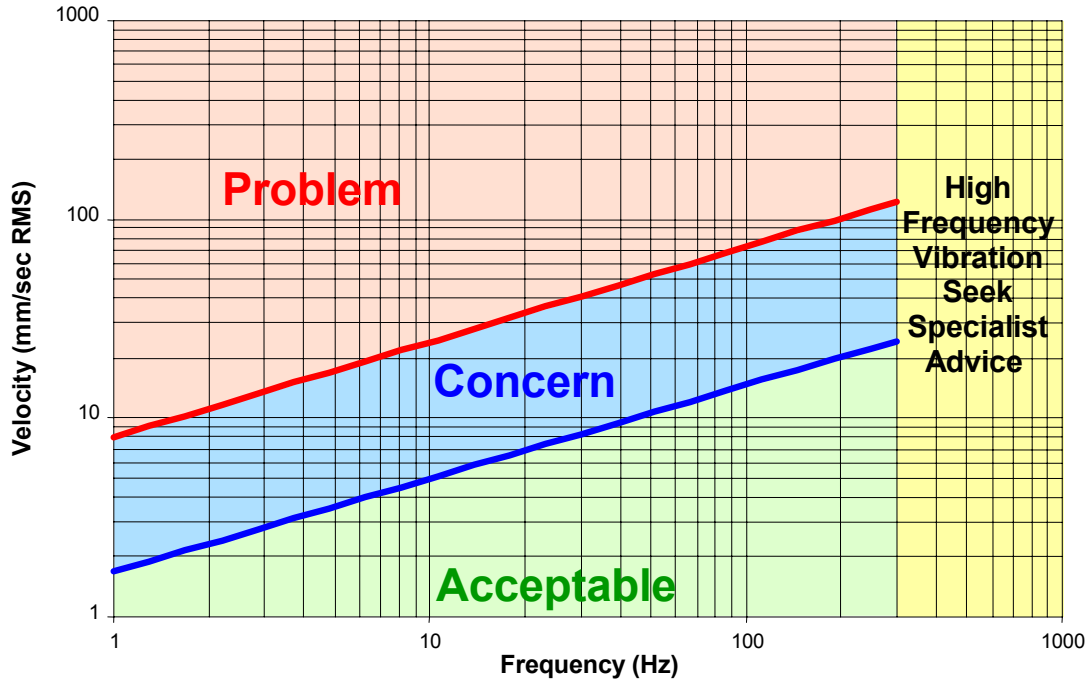


Figure T7-2 Pipework vibration criteria [T7-1]

The “Concern” “Problem” and criteria can be calculated from the following:

$$Concern\ Vibration \geq 10^{\frac{(\log(f) + 0.48017)}{2.127612}}$$

$$Problem\ Vibration \geq 10^{\frac{(\log(f) + 1.871083)}{2.084547}}$$

Where f is the dominant peak frequency in Hz

If the vibration level is in excess of the “Problem” criteria in **Figure T7-2** there is a high risk of fatigue damage occurring. In this case vibration control measures should be immediately implemented and/or direct dynamic strain measurement should be undertaken immediately to accurately determine the likelihood of failure. Checks should be performed immediately on relevant welds non destructively to ensure fatigue crack has not initiated.

A vibration level in excess of the “Concern” criteria in **Figure T7-2** means that there is the potential for fatigue damage to occur. In this case vibration control measures should be implemented and/or direct dynamic strain measurement should be undertaken to accurately determine the likelihood of failure. Checks should be performed on relevant welds non destructively to ensure fatigue crack has not initiated.

If the vibration level falls within the “Acceptable” criteria in **Figure T7-2** the location should be kept under review to ensure that the measured values are representative of the most onerous conditions.

High frequency vibration (typically greater than 300Hz) involves pipework shell modes or complex modes which have more localised responses, therefore the curves presented in **Figure T7-2** are not appropriate. Hence, specialist measurement techniques should be considered, refer to **TM-08**.

Similarly for transient responses, such as surge or slugging, a means of recording the time history of the vibration response is required, refer to **TM-08**.

Technical module

T8 - SPECIALIST MEASUREMENT TECHNIQUES

T8.1 GENERAL

There are a number of specialist measurement techniques that can be deployed to provide information not available from basic vibration measurements. This module describes some of the more common specialist techniques and their use.

T8.2 DYNAMIC STRAIN MEASUREMENT

The basic vibration measurement technique described in **TM-07** is able to provide a first screening of potential problem areas, but does not provide definitive answers as to whether fatigue will be a problem. Dynamic strain measurement, however, allows a direct assessment as to whether fatigue failure is likely.

When taking measurements of dynamic strains on plant, it is usual 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 either by bonding in line with procedures contained in **[T8-1]**, or weldable gauges are available for high temperature applications. This method is time consuming as it requires surface preparation, attachment of the gauge to the surface and associated wiring. One gauge is required to be fixed to each location of interest, and the gauge cannot be reused.
- Alternatively, a press-on gauge can be used as described in **[T8-2]**. This gauge is connected through a signal conditioning unit 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. The press-on gauge has considerable benefits in rapidly assessing fatigue strain ranges on operational plant.

The peak to peak strain levels are converted to stress using Young's Modulus (i.e. the strains are assumed to be uniaxial). Since most fatigue modes involve bending of the connection, this is a reasonable assumption.

The recommended method of fatigue life evaluation is that used by BS7608 **[T8-3]** or PD 5500 **[T8-4]**. In these codes fatigue curves are generated for specific weld geometries as shown in **Figure T8-1**. The basis of the curves is test specimens which have been fatigued to failure.

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. In the assessment of stresses in components which are in service, the endurance limit (usually taken as 10^7 cycles) for a component is taken from the S-N curves which uses design curve (mean minus two standard deviations) for that particular geometry or its nearest equivalent. If values of measured dynamic stress are found above this level, action is required to 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. For example, for a weld of class

F2 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.

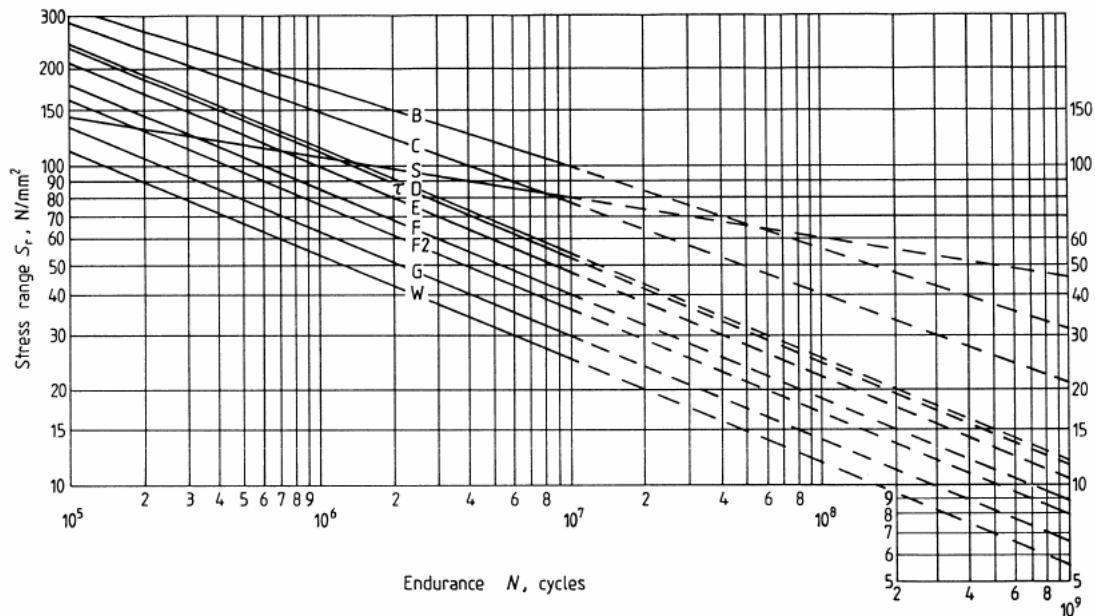


Figure T8-1 Fatigue design S-N curves for different weld classes (courtesy BS7608 [T8-3])

T8.3 EXPERIMENTAL MODAL ANALYSIS

Experimental modal analysis is based on the principle of exciting the pipework or SBC with a known input force (applied using an electrodynamic shaker or, more usually, a load hammer) and measuring the resulting vibration response [T8-5]. The resulting frequency response function (i.e. the vibration response / input force as a function of frequency) provides key information on the free vibration characteristics of the pipework:

- Structural natural frequencies
- Structural damping
- Mode shapes (available if measurements are made at a number of locations)

Such data can be used to verify the results of finite element predictions and also provide information (e.g. damping estimates) for input to a finite element model.

To obtain good quality data the background vibration levels during a test should be as low as possible.

T8.4 OPERATING DEFLECTION SHAPE ANALYSIS

Operating deflection shape analysis (or 'running mode' analysis) is a useful tool to characterise the vibration amplitudes and dynamic motion of a piping system or SBC in its operating environment.

Providing the vibration is relatively steady state then simultaneous vibration measurements at a number of locations can be used to obtain relative amplitude and phase information against a fixed reference. These data, once analysed, give a clear indication of the dynamic motion (or operating deflection shape) at any frequency of interest.

T8.5 DYNAMIC PRESSURE (PULSATION) MEASUREMENT

The measurement of dynamic pressure (or pulsation) is a very useful tool to quantify pulsation amplitudes and frequencies.

Pressure transducers designed for static pressure measurements do not usually have a fast enough response time to allow an accurate measurement of dynamic pressure to be made, particularly as the pulsation frequency increases.

Dynamic pressure transducers are available which are based on either strain gauge or piezoelectric technology and which allow the measurement of pressure pulsation over a wide frequency bandwidth. One of the principal issues associated with dynamic pressure measurement is how to introduce the transducer into the fluid stream, which is often achieved by using available isolated instrumentation tappings. One aspect to consider is that if the available tapping is too long then local acoustic resonances of the resulting 'dead leg' will interfere with the measurement of pressure pulsations in the main line. Where possible dynamic pressure should be made at several locations on the same line. This avoids the problem of a single measurement at or near a pressure node, at a particular frequency, which would not be representative of the maximum dynamic pressure in the line.

Pressure pulsation criteria are available for certain applications (e.g. reciprocating/positive displacement compressors [T8-6] and pumps [T8-7]). However, it should be appreciated that the measurement of pulsation at a limited number of locations may not give a true indication of the maximum pulsation amplitude in the piping system as the position of the anti-nodes in the standing wave in the fluid may not coincide with the available measurement locations.

T8.6 MEASUREMENT OF TRANSIENT VIBRATION

The measurement of transient vibration requires some form of continuous data recording to allow the capture of transient time histories. Digital recording and analysis systems allow a large volume of data to be captured across a large channel count which can then be subsequently analysed in the time and frequency domains as required.

Technical module

T9 - SPECIALIST PREDICTIVE TECHNIQUES

T9.1 GENERAL

There are a number of specialist predictive techniques that can be deployed to provide a more detailed assessment of piping excitation and response, either at the design stage once a potential issue has been identified from the quantitative LOF assessment, or in support of troubleshooting a known vibration problem. In both cases the techniques can be used to explore the theoretical effectiveness of possible corrective actions.

This module provides an overview of some of the most common techniques that may be used and some of the assumptions that may be used in the modelling process.

The table below identifies the applicable predictive techniques for the different excitation mechanisms, both in terms of the excitation itself and the response of the pipework.

Excitation Mechanism	Structural finite element analysis	Acoustic finite element analysis	Computational fluid dynamics (CFD)	Pulsation analysis	Surge analysis	Valve sizing calculations
Flow Induced Turbulence	*		*			
Mechanical Excitation	*					
Pulsation: Reciprocating /Positive Displacement Pumps & Compressors	*	*		*		
Pulsation: Rotating Stall	*	*		*		
Pulsation: Flow Induced Excitation	*	*		*		
High Frequency Acoustic Excitation	*					
Surge/Momentum Changes Due to Valve Operation	*				*	*
Cavitation and Flashing	*		*			*

T9.2 STRUCTURAL FINITE ELEMENT ANALYSIS

Structural finite element analysis is a commonly used tool which is used to predict the dynamic response of structures [T9-1] including piping systems and components. A number of different analyses can be undertaken, including:

- the prediction of free vibration characteristics (natural frequencies and mode shapes)
- the prediction of steady state and transient forced vibration amplitudes (displacements, velocities, accelerations and stresses)

The type of modelling will depend on the application, for example:

- for low frequency flexural modes of the main pipework 3D beam elements (or pipe elements derived from beam elements) are suitable, refer to **Figure T9-1**.
- for high frequency shell modes of the main pipework then 8-node shell elements are recommended, refer to **Figure T9-2**.
- for modelling of a SBC a combination of shell and solid brick elements is required, refer to **Figure T9-3**.

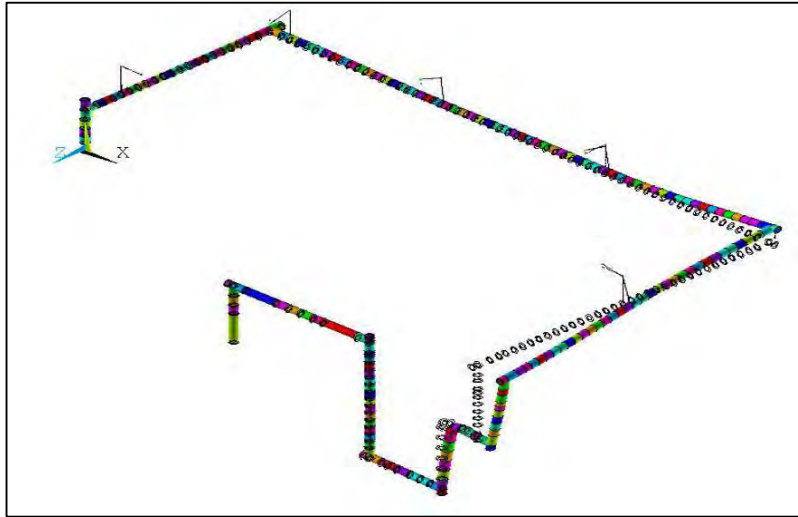


Figure T9-1 Low frequency flexural modes of the main pipework using 3D beam elements

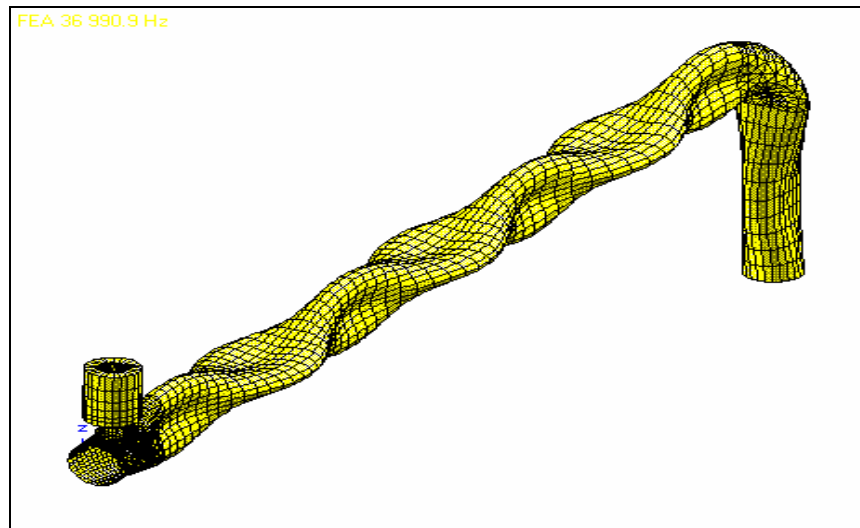


Figure T9-2 High frequency shell modes of the main pipework using shell elements

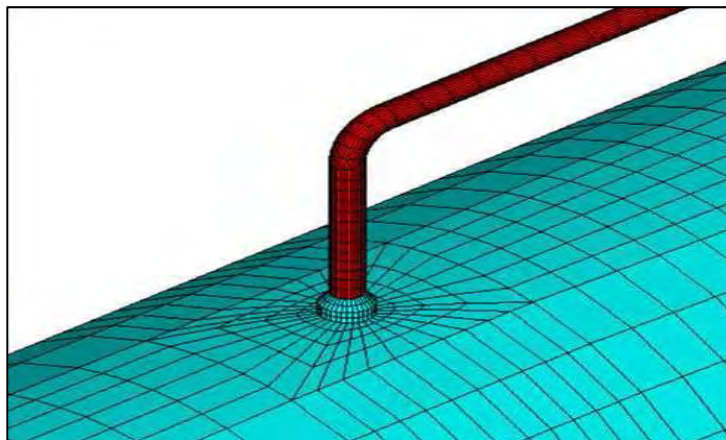


Figure T9-3 SBC modes using a combination of shell and solid brick elements

The accuracy of the predicted pipework natural frequencies 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 their static behaviour, so careful consideration should 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:

- 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.
- 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.

The accuracy of the prediction of forced response levels depends on estimating (i) the dynamic force levels acting on the pipework, and (ii) the structural damping. Structural damping of piping systems is often estimated at between 1-2% of critical; however, this will vary considerably and the use of experimental modal analysis techniques (refer to **Section T8.3**) can be used to provide more accurate damping estimates for a particular configuration.

T9.3 ACOUSTIC FINITE ELEMENT ANALYSIS

Acoustic finite element analysis is used to predict the dynamic response of contained fluids in a piping system and associated volumes (e.g. vessels) **[T9-2]**. A number of different analyses can be undertaken, including:

- the prediction of modal characteristics (natural frequencies and mode shapes)

- the prediction of steady state and transient response levels (dynamic pressures and associated forces)

Acoustic finite element analysis is well suited to predict both the low and high frequency modal behaviour of the contained fluid as (depending on finite element mesh density) both axial and cross modes can be predicted.

T9.4 COMPUTATIONAL FLUID DYNAMICS

Computational fluid dynamics (CFD) is a modelling technique which can be used to predict the flow patterns within pipework and associated components (e.g. valves and orifice plates). This can lead to a better understanding of flow related issues which can give rise to piping vibration related problems [T9-3].

T9.5 PULSATION ANALYSIS

Pulsation analysis is an acoustic simulation of the fluid contained in a piping system, which results in the prediction of acoustic natural frequencies and mode shapes of the fluid system. Predictions of the forced response of the fluid to excitation from a reciprocating/positive displacement compressor or pump [T9-4] [T9-5], or flow induced pulsation [T9-6], can also be undertaken.

There are obvious similarities between this form of acoustic simulation (often using transfer matrix methods) and acoustic finite element analysis. One difference is that the transfer matrix method is limited to plane wave transmission in the fluid system and so is not able to predict cross mode behaviour. However, the transfer matrix method is generally better suited to the modelling of piping system components such as valves and orifice plates and the acoustic damping provided by fluid flow.

T9.6 SURGE ANALYSIS

Transient flow (surge) analysis is used to predict the dynamic pressures and forces generated in a piping system caused by a transient event (e.g. sudden valve closure or pump start-up or shut-down) [T9-7]. Predictions are undertaken in the time domain and results are available in terms of dynamic pressures and forces as a function of time [T9-8]. Analyses can also be undertaken which model the characteristics of valve and pump control systems.

T9.7 VALVE SIZING CALCULATIONS

Comprehensive valve sizing calculations can be used to determine the suitability of a specific valve, particularly with respect to flashing and cavitation [T9-9].

Technical module

T10 - MAIN LINE CORRECTIVE ACTIONS

T10.1 GENERAL

The purpose of this module is to give possible design solutions, best practices or remedial action for new and existing plants. Where possible, recommendations for detailed analyses are given. The corrective actions have been categorised by excitation mechanism.

Excitation Mechanism	Section
Flow Induced Turbulence	T10.2
Mechanical Excitation	T10.3
Pulsation: Reciprocating/Positive Displacement Pumps & Compressors	T10.4
Pulsation: Rotating Stall	T10.5
Pulsation: Flow Induced Excitation	T10.6
High Frequency Acoustic Excitation	T10.7
Surge/Momentum Changes Associated with Valves	T10.8
Cavitation and Flashing	T10.9

There are two general areas in which corrective actions can be grouped: those which affect the excitation mechanism, and those affecting the response mechanism. Where possible it is preferable to address the excitation mechanism, as this will either remove or reduce the excitation energy. Alternatively, by targeting the response mechanism the levels of vibration and dynamic stress can be managed. However, if the corrective action becomes ineffective damage can still occur on the pipework, or the excitation energy could result in other issues.

Where there is more than one excitation mechanism of concern, the applied corrective action(s) should ensure that all the excitation mechanisms are addressed.

The use of detailed predictive techniques (**TM-09**) may be required in order to fully quantify the effectiveness of different potential modifications prior to implementation. Specialist measurement techniques (**TM-08**) can also provide useful information to either validate the predictions or verify the corrective action(s). In certain circumstances this may require specialist advice.

T10.1.1 General Corrective Actions Affecting Pipework Response

T10.1.1.1 Tighten up clearance on supports

Tightening up clearances on supports has a similar effect to adding additional supports with the potential to stiffen the pipework and change the natural frequency. If an existing support has a clearance that allows the pipework to move it can be the cause of excessive vibration. By tightening the clearance on supports the pipework fundamental natural frequency is then increased, and, as typically the levels of energy fall with frequency, the resulting vibration level falls also. However, this is not always an appropriate approach as thermal growth requirements may limit the amount of additional support that can be included.

Care is required when changing the support effectiveness when there is a tonal excitation, as there is a possibility that by changing the stiffness of the pipework a structural natural frequency could become coincident with the excitation frequency.

T10.1.1.2 Avoiding metal to metal contact

Where pipework is moving and in metal to metal contact with other pipework, rest supports or structural members there is a risk from fretting. If the contact is necessary (e.g. a pipe support) then a wear resistant or compliant layer should be inserted between the surfaces. Otherwise the two items should be separated to ensure there is no contact.

T10.2 FLOW INDUCED TURBULENCE

T10.2.1 Main Line Excitation

T10.2.1.1 Reduction in fluid velocity

If feasible, one of the simplest solutions to deal with a flow induced turbulence issue is to decrease the flow velocity. This will reduce the amount of excitation energy and therefore the response of the pipework. This can be achieved by:

- Increasing the diameter of the main pipe
- Running a second pipe in parallel
- Changing the operating conditions.

T10.2.1.2 Flow Smoothing

Flow smoothing can be accomplished through the use of swept tees rather than 90° tees, minimising the number of bends in a system (and ensuring, where practicable, that bends are separated by a distance of at least 10 pipe diameters), the use of long radius bends, and the use of flow straighteners. This will reduce the level of turbulence in the fluid flow and reduce the excitation level. This is most effective when turbulence is occurring at a single location, such as a U-bend, resulting in minimal modifications to be carried out.

Other sources of turbulence within the flow can be intrusive elements. However, in many cases excitation can be from multiple sources and removal of an individual intrusive element may not result in a significant reduction.

T10.2.1.3 Change valve type

Valves which display a high recovery factor dissipate relatively little flow stream energy due to the streamlined internal contours. Therefore, the pressure downstream of the valve vena contracta recovers to a high percentage of its inlet value, giving rise to lower levels of flow turbulence. High recovery factor valves are identifiable by a relatively clear or straight through flow path; examples are most rotary control valves, such as the eccentric plug, butterfly, and ball valve.

T10.2.1.4 Change valve trim

Changing the trim of a control valve can help reduce the level of turbulent energy. As a first approximation the fluid kinetic energy, at the trim exit, should be 480 kPa or less for continuous service single phase fluids, and 275 kPa or less for multiphase fluids (where the kinetic energy in kPa is given by $\rho v^2/2000$, ρ is the fluid density in kg/m³, and v is the velocity of the fluid exiting the valve trim in m/s) [T10-1].

T10.2.2 Main Line Response

T10.2.2.1 Pipework Supports

Stiffening of the main line and its supporting structure can also be beneficial. This is because the fundamental natural frequency is then increased, and, as the level of turbulent energy falls off rapidly with frequency (see **Figure T10-1**), the resulting vibration level falls also. Tightening up on clearance on supports has a similar effect to adding additional supports with the potential to stiffen the pipework and increase the natural frequencies.

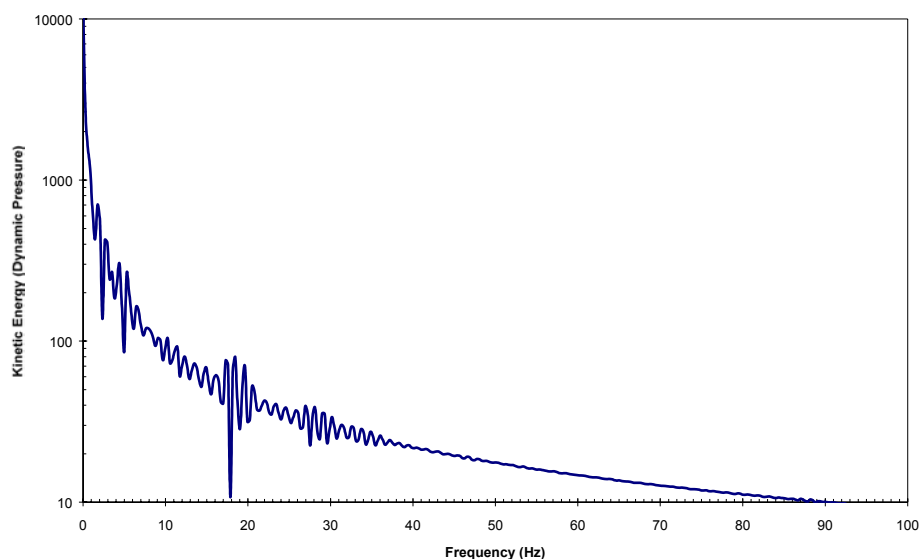


Figure T10-1: Turbulent energy as a function of frequency

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.

T10.2.2.2 Viscous Dampers

Stiffening is not always an appropriate approach. Thermal growth requirements may limit the amount of additional support that can be included. 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. It should also be noted that they are ineffective at frequencies over 30Hz or for narrow band excitation.

When correctly installed viscous dampers have a significant effect on the response of the pipework over a range of frequencies (see **Figure T10-2**)

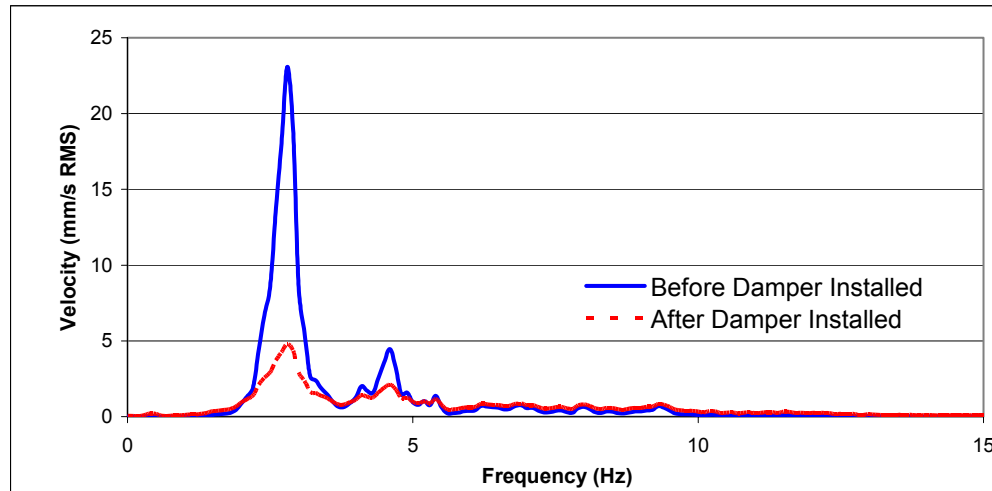


Figure T10-2: Effect of installing a Viscous Damper on pipework response

The key aspects to ensure before installing a viscous damper are:

- The connecting structure is sufficiently stiff
- The damper is not located at a nodal position in the pipework's response, where the pipework's dynamic response is at a minimum.
- Thermal growth/displacement of the line is considered to ensure suitable sizing
- The viscous damper has a suitable level of damping
- Under high temperature applications there is sufficient thermal isolation

T10.2.2.3 Shock Arrestor / Absorber/ Snubber

Thermal growth requirements may limit the amount of additional support that can be included. In these cases, use of a shock arrestor may prove effective as they allow low velocity movement such as thermal growth but provide resistance to sudden movements caused by forced vibration.

It should be noted that the shock arrestor “locks” in position at a certain level of vibration which can result in high loads being applied to the pipework during events such as slugging.

Maintenance is required after installation to ensure that they are functioning as designed and have not slackened/seized.

It should be noted that it is difficult to design and install snubbers effectively for vibration problems, as they are only suitable in certain circumstances. Incorrect installation can make pipework stresses higher and it is recommended that specialist advice is sought.

T10.2.2.4 Composite Pipework Wraps

Applying a composite wrap can have a beneficial effect on the main line by stiffening it. However, the effect of increased mass could counter the gain in increased stiffness. Pipework wraps also increase the damping levels, which may help to reduce the response.

The use of composite wraps in pipework repairs for dynamic issues should always be approached with caution due to the current lack of in-depth knowledge in the area, with particular reference to their high-cycle fatigue resistance. It is difficult to quantify the

effectiveness of composite wraps when applied to vibration/cyclic stress over the long term as the mechanical integrity after extended periods of vibration is unknown. Therefore, if pipework wraps are being applied to areas which have been subjected to vibration/fatigue damage additional corrective actions should be applied to reduce the level of excitation.

Note, composite pipework wraps should be used with caution on safety critical lines because of their fire resistant properties.

T10.2.2.5 Changes in section - wall thickness

For a given pipe diameter, increasing the wall thickness of the pipe can have a beneficial effect, principally due to the increase in structural inertance (i.e. acceleration response for a unity force input), resulting in lower dynamic stress levels for a given level of excitation. It should be noted that for a given length of pipe and pipe diameter, increasing the pipe wall thickness does not affect the low order natural frequencies significantly as the change affects both mass and stiffness.

Note, that an increase in wall thickness will increase the flow velocity and hence the turbulent excitation, however, this is far outweighed by the benefits.

T10.3 MECHANICAL EXCITATION

T10.3.1 Main Line Excitation

T10.3.1.1 Change of operation

In the event that coincidence does occur between the excitation frequency and structural natural frequency of the pipework, 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. Care is required to ensure that altering the machine speed will not excite different structural natural frequencies and cause other problems.

T10.3.1.2 Isolation of Vibration Source

Anti-vibration mounts isolate the source of excitation from the rest of the system. They can be very effective in isolating large structures such as decks or skids and require little maintenance.

They are more suitable for installation during the design stage as they are often difficult to install without any major modifications. Achieving confidence in a predicted solution can be difficult. For an isolation mount to work effectively the foundation on which it is mounted (i.e. the structure on the 'isolated' side of the mount) should display a high level of dynamic stiffness relative to the mount stiffness. As a first approximation the mobility (i.e. the velocity/force as a function of frequency) of the foundation should be 100 times that of the mount. The mobility of a support foundation can be determined either from test (using experimental modal analysis techniques) or by finite element modelling, although for the latter case the modal damping will be required, refer to **TM-08** and **TM-09**.

Care is required to ensure that all transmission paths are considered (e.g. pipework connections) to ensure that the isolation system is effective.

T10.3.1.3 Bellows

By decoupling the pipework via bellows the transmission path of the vibration is impeded. If bellows are installed, new stress calculations need to be carried out with a potential redesign of supports required. The presence of bellows in pipework can also introduce a greater pressure drop over the pipe and any obstruction to the flow caused by the bellows is a possible cause of turbulence.

T10.3.2 Main Line Response

T10.3.2.1 De-tuning pipework – changing mass and stiffness

If high levels of vibration are caused by a discrete excitation frequency or one of its harmonics coinciding with a structural natural frequency then where practical and feasible, the first structural natural frequency should be moved above the excitation band associated with the running speed of the machine. In any case, the pipework structural natural frequencies should be outwith $\pm 20\%$ (this is based on site experience and should be a minimum limit) of the excitation frequency.

As it is often impractical to change the excitation frequencies or pipework geometries, mass or stiffness tuning can be used to alter the structural natural frequencies with no alterations to pipe geometry. In general, this will involve modifying the structural response of the pipe by the addition of stiffness, mass or damping. The most effective location at which to make the modification is where the pipe is vibrating the most.

- If it is required to move the structural natural frequency above the excitation frequency, then the pipework should be stiffened (e.g. addition of clamps or additional supports)
- If it is required to move the structural natural frequency below the excitation frequency, then mass should be added to the pipework.
- Alternatively, if the system is at resonance, and if it is impracticable to move the structural natural frequencies, then addition of damping will reduce the structural response.

Where a mass has been applied to decouple the system a suitable inspection strategy is required to ensure that the mass remains in the correct location. If the mass is removed or positioned in an incorrect location the pipework could become excited at its natural frequency again.

Care should be taken when modifying structural natural frequencies (using stiffness or mass changes) to ensure that the modified natural frequencies are not coincident with one of the order harmonics, therefore causing a resonant response.

Caution should also be applied if the system is subject to varying excitation frequencies (non-constant speed pump) as any frequency changes could result in coincidence reoccurring and the system becoming resonant.

T10.3.2.2 De-tuning pipework - changing piping parameters (span length and diameter)

Changing certain piping parameters can also move the structural natural frequencies away from a problem excitation frequency. There are two important parameters which have a major influence in determining the fundamental structural natural frequency of a pipe. These are:

- The outside diameter of the pipe.

- The 'effective span' of the pipe (the length of pipe between the locations where the pipe is effectively constrained).

The dependence of the fundamental mode of a simply supported pipe span on both diameter and span length are shown in **Figure T10-3**.

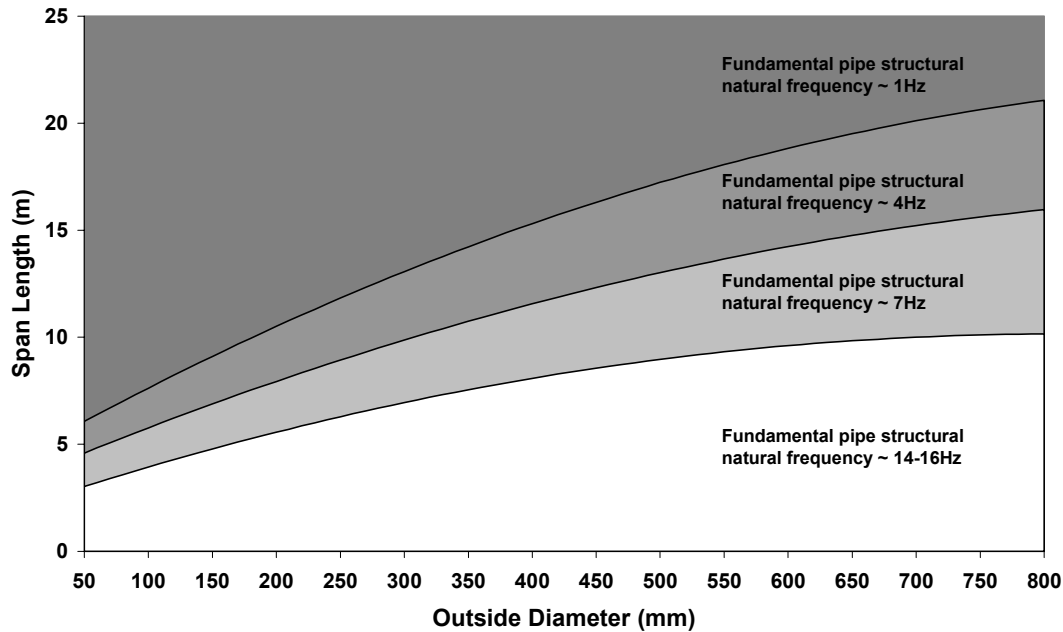


Figure T10-3: Variation of pipe fundamental natural frequency

It should be noted that for a given length of pipe and pipe diameter, increasing the pipe wall thickness does not affect the low order natural frequencies significantly as the change affects both mass and stiffness.

T10.4 PULSATION – RECIPROCATING/POSITIVE DISPLACEMENT PUMPS AND COMPRESSORS

T10.4.1 Main Line Excitation

T10.4.1.1 Change in operation

Acoustic standing waves are present in all pipe fluid systems. To avoid coincidence of standing acoustic waves with the excitation from a reciprocating/positive displacement pump or compressor, the compressor speed should not be within $\pm 20\%$ of the nearest acoustic natural frequency (this is based on site experience and should be a minimum limit).

This is often very difficult to achieve as in practice acoustic modes in complex pipework are often closely spaced together. Therefore any change in operating speed of the reciprocating/ positive displacement pump or compressor could lead to coincidence with a new acoustic natural frequency.

Where two or more reciprocating/positive displacement pumps or compressors are working in parallel the relative phasing between the machines can have a significant bearing in terms of the resulting levels of pulsation in the common manifold and pipework. In certain

situations it may be possible to “phase-lock” the machines so that the resulting pulsation levels are minimised. It should also be noted that in some situations, operating a pump or compressor at lower flow rates can give rise to higher pulsation levels than operating at maximum flow – this is typically caused by the reduction in acoustic damping at lower flow velocities.

T10.4.1.2 Changing line length

A change in line length will change the acoustic natural frequencies – increasing the line length (i.e. the length of the column of fluid) will reduce the acoustic natural frequencies, while conversely reducing the line length will increase the acoustic natural frequencies. This is effective with side branches or small bore connections experiencing “quarter wave” acoustic resonances. However caution should be taken to ensure the new pipework geometry does not result in coincidence with another standing wave (or pipework structural natural frequency).

For complex geometries a pulsation model of the system may need to be generated in order to predict the acoustic natural frequencies, which is achieved using specialist pulsation software, refer to **Section T9.5**. For this specialist help should be considered.

T10.4.1.3 Smoothing Flow

High ratio reducers and tight geometries can sometimes cause partial reflections of pressure waves which result in the formation of acoustic standing waves. Removing these, or using more gradual transition pieces can help to eliminate problem standing waves.

T10.4.1.4 Pulsation Bottles

A potential corrective action is the use of pulsation bottles for reciprocating compressors, or nitrogen precharged pulsation dampers for reciprocating/positive displacement pumps. One drawback with precharged units is that the precharge pressure should be maintained to the manufacturer’s recommended level (usually set as a percentage of the static line pressure, typically 70%-80%) otherwise the dampers become ineffective.

The frequency characteristics of the pulsation bottles should be checked to ensure the design provides the required attenuation, as incorrect design and/or installation of dampener bottles can make the vibration levels worse. This can be undertaken using specialist pulsation software (refer to **Section T9.5**), although care should be exercised when the lateral dimension of the vessel is large enough that ‘cross’ acoustic modes may be present; in this case acoustic finite element modelling is a more appropriate tool.

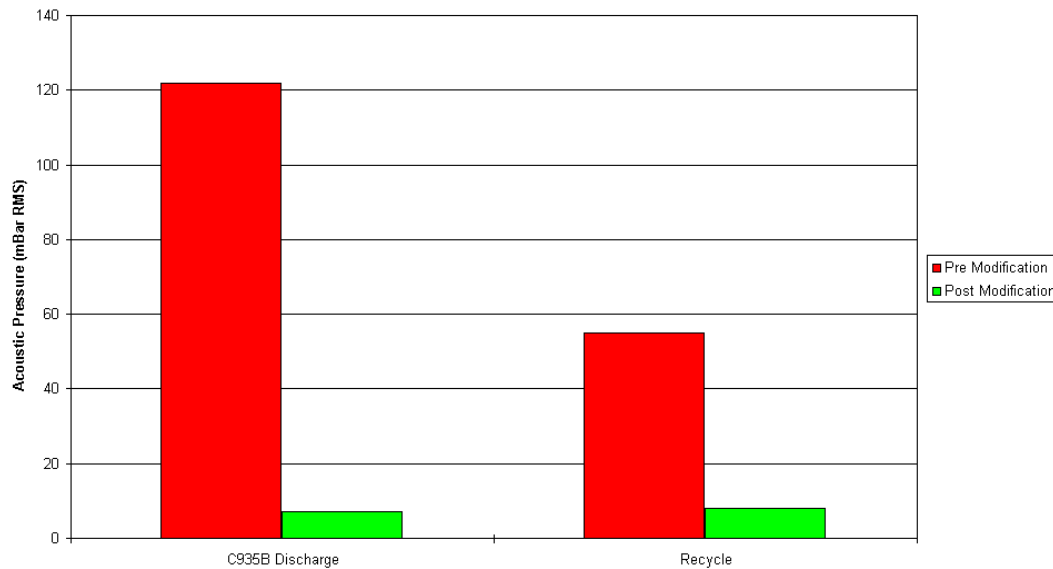


Figure T10-4: The Effect of installation of a pulsation bottle on dynamic pressure

T10.4.1.5 Orifice plates

Orifice plates can also provide significant damping of acoustic modes. The most effective location for an orifice plate is at a dynamic pressure minimum (or node), although other locations between pressure nodes and anti-nodes may also give some benefit. A detailed pulsation simulation can be used to quantify the expected benefit (refer to **Section T9.5**). Their use should be carefully balanced against the pressure drop that they impose on the system.

T10.4.1.6 Acoustic absorbers

In some situations (where there is a single, fixed, acoustic natural frequency which is leading to a resonant condition) it is possible to design an acoustic absorber to reduce the total energy in the system at the problem frequency. This typically takes the form of a $\frac{1}{4}$ wave side branch which is specifically designed and tuned for the purpose.

T10.4.2 Main Line Response

T10.4.2.1 De-tuning of pipework

Where the excitation frequency matches one of the structural natural frequencies of the pipework consideration should be given to de-tuning the resulting structural resonance. Refer to **Section T10.3.2**.

T10.4.2.2 Changes in pipe geometry

Fluid pressure pulsation can excite pipework predominantly as a result of the unbalanced shaking forces that are developed due to the dynamic pressure reacting against bends or abrupt changes in section. Minimising the number of bends in the system, and avoiding abrupt changes in section, will help decouple the pulsation from the pipework and minimise the opportunity for high shaking forces to be developed.

T10.5 PULSATION – ROTATING STALL

T10.5.1 Main Line Excitation

T10.5.1.1 Change in operation

Pulsation excitation from compressor rotating stall will occur where the rotating stall frequency coincides with one of the acoustic modes of the pipework. Rotating stall tends to occur at low flow conditions and is heavily dependent on the geometry of the impellers and diffusers.

If rotating stall is experienced then changing the operational configuration is one short term solution (e.g. operating the compressor at higher flow by using more recycle).

T10.5.1.2 Change in line length

The principal cause of high vibration levels in compressor system pipework when rotating stall is experienced is due to the excitation of an acoustic resonance in the system. Therefore, changing the line length to ensure that there are no acoustic natural frequencies within $\pm 20\%$ (this is based on site experience and should be a minimum limit) of the stall frequency is one modification that can be made at the design stage (refer to **Section T10.4.1.2**).

T10.5.1.3 Orifice plates

If it is impractical to modify the pipework to change the acoustic natural frequencies then an orifice plate can be considered to damp the problem response, refer to **Section T10.4.1.5**.

T10.5.1.4 Acoustic absorbers

Refer to **Section T10.4.1.6**.

T10.5.2 Main Line Response

The corrective actions which can reduce the main line response to pulsations resulting from rotating stall are similar to those detailed in **Section T10.4.2**.

T10.6 PULSATION – FLOW INDUCED EXCITATION

T10.6.1 Affecting Main Line Excitation

T10.6.1.1 Change in operation

Vortices which form over obstructions in the flow, or the instabilities that occur at the mouth of a 'dead leg' branch, only occur at certain fluid velocities. If these vortices are in tune with an acoustic resonance of the pipework, flow induced pulsation can occur. These vortices can be prevented from forming by changing the fluid velocity to ensure that the resulting excitation frequencies are outwith $\pm 20\%$ (this is based on site experience and should be a minimum limit) of the acoustic natural frequencies of the fluid system. This can be achieved by changing the operating conditions or the pipe diameter which will result in a change in flow velocity.

T10.6.1.2 Change in line length - Acoustic frequency

A change in line length can detune any acoustic standing waves which are likely to be excited by flow induced vortices or flow instabilities, refer to **Section T10.4.1.2**. It should be noted that this will not eliminate the flow induced vortices, only their ability to excite the pipework's acoustic resonance.

T10.6.2 Main Line Response

The corrective actions which affect the main line response for the pulsations resulting from flow induced effects are similar to those detailed in **Section T10.4.2**.

T10.7 HIGH FREQUENCY ACOUSTIC EXCITATION**T10.7.1 Main Line Excitation****T10.7.1.1 Reduction in mass flow rate**

An effective 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 blow down the system. General guidance indicates that limiting the valve outlet Mach number (i.e. the ratio of the fluid velocity at the valve outlet, to the sonic velocity in the fluid at the given temperature) to between 0.4 (continuously operating systems) and 0.5 (intermittently operating systems) should result in relatively low levels of acoustic energy, although this may be difficult to achieve in practice for some relief systems.

T10.7.1.2 Change of valve trim

Use of multi stage pressure drop internal trim in a control valve can help to reduce noise levels at source and therefore reduce the risk of an acoustic fatigue failure. However, information should be sought from the valve manufacturer to confirm the reduction in sound pressure level that might be expected if the valve is fitted with a "low noise" trim, e.g. typical examples holed cage and labyrinth cage technology. 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.

T10.7.1.3 Change in line length - Attenuation with distance

Line length changes can also be considered at the design stage. A typical figure for the attenuation of sound power with distance is 3dB per 50 pipe diameters downstream, and therefore by increasing the pipe length between the valve and high risk locations downstream of the valve it may be possible to reduce the acoustic energy to an acceptable level. There are obviously additional cost and weight implications associated with this type of modification, although this approach has been used in some situations.

T10.7.1.4 Acoustic silencers

Acoustic silencers can be considered when it is not possible to reduce the level of high frequency acoustic energy at source. While acoustic silencers are an alternative, their use is not generally recommended because the success rate and durability is limited. A silencer

will be exposed to high levels of acoustic energy which can result in fatigue failure of the silencer itself.

T10.7.2 Main Line Response

T10.7.2.1 Change in wall thickness

Increasing the local pipe wall thickness is an option for a new design as this reduces the resulting high frequency dynamic stress levels at circumferential discontinuities; alternatively, full wrap around reinforcement can be used to achieve the same goal. Partial reinforcement should not be used. Reducing the diameter to pipe wall thickness ratio at fatigue sensitive locations is an effective and, in most cases, a practical approach at the design stage.

T10.7.2.2 Removal of circumferential discontinuities

Wherever practical, circumferential discontinuities (such as small bore connections) should be designed out or removed as these will be the main fatigue sensitive locations, or alternatively changed into an axisymmetric discontinuity (for example, by using a full wrap around reinforcement as outlined previously). Alternatively, the use of connections such as forged or extruded tees can be considered as an alternative to more fatigue sensitive geometries such as weldolets or welded/stabbing tees.

It should be noted that no acoustic fatigue failures of a plain section of pipe without a circumferential discontinuity have been reported to date. Therefore for pipework without any form of circumferential discontinuity the only precaution is to ensure good quality full penetration welds with no undercut.

T10.7.2.3 Use of circumferential stiffening rings

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. The location of stiffening rings will be determined by the local geometry and should be checked by some form of detailed analysis (e.g. finite element methods to predict the change in likely response levels), although as an initial guide they should be placed approximately $2D$ upstream and downstream of the connection (where D = diameter of the connection).

T10.8 SURGE/MOMENTUM CHANGES ASSOCIATED WITH VALVES

T10.8.1 Main Line Excitation

T10.8.1.1 Change in operation

Rapid changes in fluid velocity occur when valves are opened/closed. The resulting forces on the pipework caused by the pressure wave (or surge) travelling back upstream from the closing valve can be reduced by either reducing the mean fluid velocity or slowing down the time taken to close the valve.

Pump start-up and shut-down can also induce rapid changes in fluid velocity resulting in surge problems. The use of a 'soft start' pump can help reduce the resulting surge pressures in the system.

T10.8.1.2 Surge Pressure Relief

Fast acting specialist relief valves are available to reduce the surge pressure in a system. Alternatively bursting discs are an alternative option, with the relieved fluid sent to a separate holding vessel or tank.

T10.8.1.3 Surge Tank or Arrestor

A specially designed tank can be used to decelerate the fluid gradually and hence reduce the levels of surge that are experienced.

T10.8.2 Main Line Response**T10.8.2.1 Reduction in bends/reducers**

The effect of rapid changes in fluid momentum caused by transient flow can be reduced by minimising the number of bends in a system and the use of long radius bends. This will result in less energy being transmitted from the fluid to the pipework.

T10.8.2.2 Viscous dampers

Installation of a viscous damper can provide resistance against the forced movement caused by the rapid changes in fluid velocity in the line. This forced vibration is broadband in nature which often excites one of the lower structural natural frequencies of the pipework, making it suitable for damper installation. The installation of a damper should be considered in cases where extra supporting of the line and changes in process condition are not possible. Refer to **Section T10.2.2.2**.

T10.9 CAVITATION AND FLASHING**T10.9.1 Main Line Excitation****T10.9.1.1 Change in operation**

Reducing the flow through the affected system will reduce the pressure drop and subsequently reduce/eliminate the cavitation and/or flashing.

By reducing the fluid temperature sufficiently (i.e. reducing the fluid's vapour pressure) the effects of cavitation and/or flashing can be addressed.

Alternatively for a valve, both inlet and outlet pressures can be increased (e.g. locating the valve at a lower elevation in a piping system) which results in an increase in the critical pressure drop (i.e. δ in **Section T2.9.2**).

T10.9.1.2 Change valve type

Ball valves only allow the fluid to be controlled without cavitation and/or flashing at relatively small pressure ratios. Butterfly valves and rotary plug valves are slightly better, whereas linear valves allow control with very little cavitation and/or flashing even at high pressure ratios providing the plug is correctly designed.

T10.9.1.3 Change valve trim

Changing the trim of a control valve can help reduce the level of turbulent energy. As a first approximation the fluid kinetic energy, at the trim exit, for cavitating flow should be 275 kPa or less (where the kinetic energy in kPa is given by $\rho v^2/2000$, ρ is the fluid density in kg/m^3 , and v is the velocity of the fluid exiting the valve trim in m/s) [T10-1]. Control valves can, in some cases, be fitted with anti-cavitation trims and multi-stage axial plugs.

T10.9.1.4 Staging The Pressure Drop

Staging the required pressure drop to occur across a number of valves, orifice plates will reduce the individual stage pressure drop and subsequently reduce and/or eliminate the cavitation and/or flashing.

T10.9.1.5 Flow smoothing

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. This will reduce the level of pressure drop over these components resulting in a reduced possibility of cavitation and/or flashing occurring.

T10.9.2 Main Line Response**T10.9.2.1 Supporting**

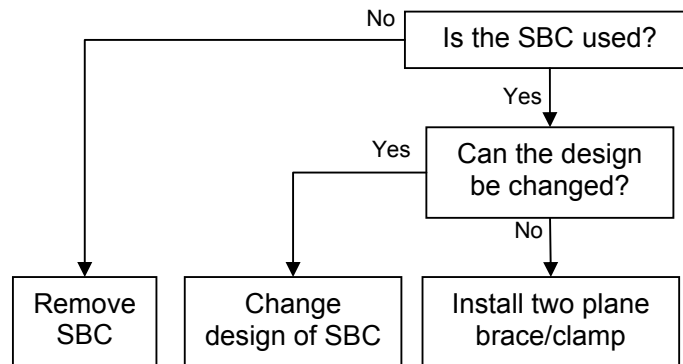
Because the cavitation and flashing effect only extends a limited distance downstream of the valve, bracing the downstream main line and small bore connections will help to minimise the induced pipework vibration. However it should be noted that this will not prevent pitting damage to the pipework and valves associated with the cavitation effect and action should preferably be taken to eliminate this first. Where this may be considered a suitable application is when cavitation is present only during the opening/closing of a valve and is causing excessive vibration. Here additional clamping would be effective if the cavitation and/or flashing was deemed to be of an acceptably low level.

Technical module

T11 - SMALL BORE CONNECTION CORRECTIVE ACTIONS

T11.1 GENERAL SBC CORRECTIVE ACTIONS

All the corrective actions described in this Technical Module are aimed at improving the response of the SBC, rather than reducing the excitation (which tends to come from the main line). Therefore undertaking the main line corrective actions (**TM-10**) will reduce the SBC excitation levels, and hence have a beneficial effect on the SBC fatigue life.



Flowchart T11-1 Corrective actions methodology for SBCs

There are various approaches to reduce the response of the small bore connections to vibration excitation.

T11.1.1 Remove SBC

The preferred method is to, wherever possible, remove the small bore connection.

T11.1.2 Change design of SBC

The secondary approach would be to alter the design to make it more robust with respect to vibration. This can be achieved using one or more of the following:

- the mass of unsupported valves/instrumentation should be minimised, for example by removal of existing valves and replacement with lightweight double block and bleed valves, monoflange valves or blank flanges. 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
- the fitting and overall unsupported length should be made as short as possible
- the diameter of the small bore connection should be maximised
- if the small bore connection is being replaced, use of a short contoured body fitting is preferable
- where existing threaded fittings are used they should be fully back welded, ensuring there are no exposed threads

T11.1.3 Install Brace/Clamp

The final corrective action for an SBC is bracing and this should only be applied where the other options have been exhausted. 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 design of the support should ensure the following:

- any mass at the free end of a cantilever should be supported in both directions perpendicular to the axis of the small bore connection
- the bracing should be in two planes, connected between the small bore pipework and the main pipe
- clamping should be designed so that the SBC is adequately supported. Note, it is not just the first weld that can be susceptible to vibration induced fatigue, subsequent welds can be an issue and should be suitably supported
- it is essential that bracing should be from the main pipe, thus ensuring that the small bore connection moves with the main pipe. Under **no circumstances** should the connection be braced from local structure such as steelwork, decks or bulkheads
- any applied supports should be sufficiently stiff in the direction of interest - if the support is not stiff it will have little effect on the response. As a general rule of thumb the support should be at least as stiff as the connection to be of any effect
- in the case where the small bore connection has a geometry making it difficult to support, it should be re-routed to allow easy support
- any fastenings used should be designed to be effective under vibration (e.g. bolted clamps include anti-vibration washers/lock nuts)

When considering installing a brace/clamp to a parent pipe of small diameter (i.e. typically less than 6") the effect of the added mass could affect the response of the parent pipe (i.e. the additional mass if significant to the mass of the pipe could reduce the natural frequency of the parent pipe itself).

For low frequency excitation (typically <50Hz) bolted clamps/braces are suitable. For bolted clamps/braces periodic inspection will be required to ensure that no loosening occurs during years of operation and if the brace/clamp has been removed for maintenance purposes it has been correctly re-instated. At higher frequencies bolted clamps/braces become less effective and are not recommended. For higher frequency excitation (>50Hz) welded gusset plate clamps/braces are recommended. The higher the frequency the thicker/stiffer the gusset plate required. Particular care should 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 and re-enforcement plates will help. It should be noted that when installing welded braces on existing pipe the weld process needs to take account of the service requirements, e.g. PWHT. **Figures T11-2 to T11-4** are drawings for the clamp type of small bore support, suitable if the excitation is less than 50Hz, while **Figures T11-5 to T11-7** give examples of welded supports.

Where anti-vibration clamps are installed it is recommended that a clamp inspection plan is incorporated into the overall inspection strategy, particularly if the clamps involve bolted connections. This would include regular visual surveys of critical locations following shutdown activities to ensure clamps have been reinstated correctly, and that clamps are still fit for purpose. A clamp register should be used to control this activity where each clamp is given a unique serial number and tagged accordingly.

T11.2 SBC CORRECTIVE ACTIONS FOR TONAL EXCITATION

If there is coupling between the excitation frequency(ies) and the structural natural frequency(ies) of the SBC there are two ways to de-couple the system, either by changing the mass or changing the stiffness of the SBC. Increasing the mass on the SBC will reduce the structural natural frequency and increasing the stiffness will increase the structural natural frequency (e.g. install brace/clamp, shorten connection).

It is strongly recommended that in this case experimental modal analysis (see **TM-08**) is used to ensure that the structural natural frequencies of the modified SBC are well removed from the excitation frequencies.

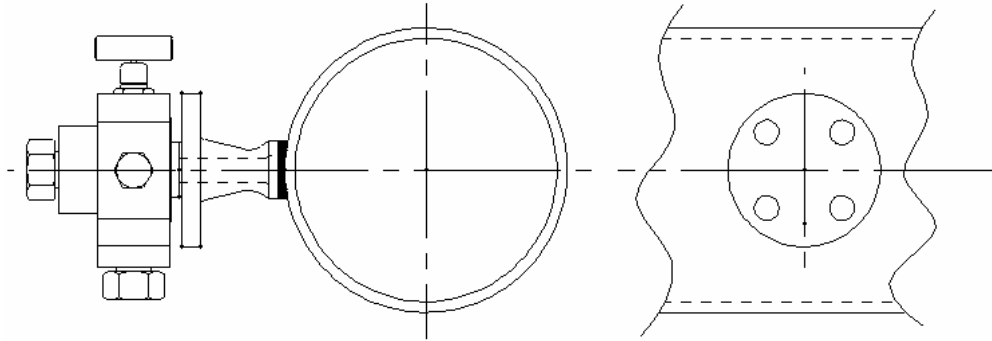


Figure T11-1 Preferred Small-bore Arrangement

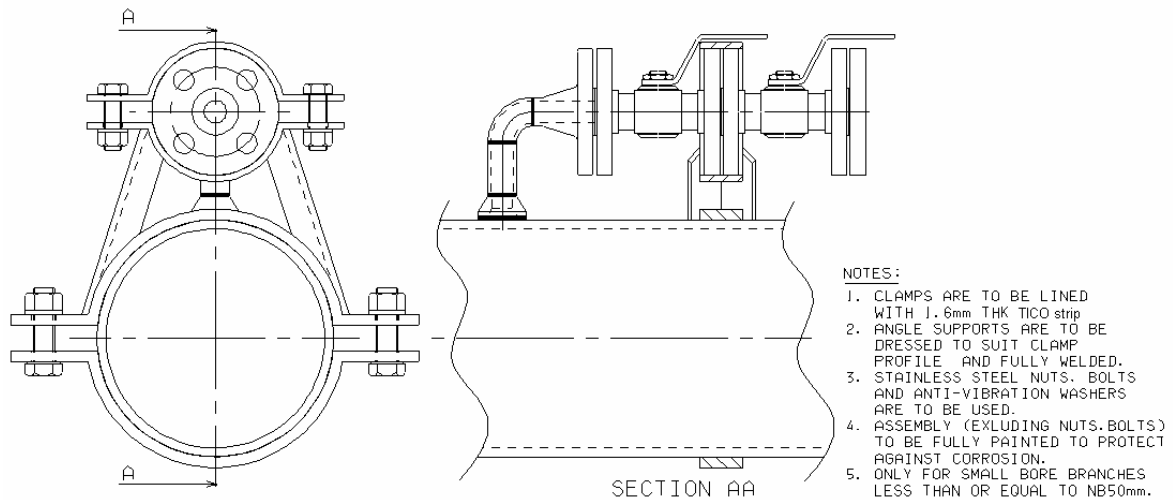


Figure T11-2 Clamp Type of Support

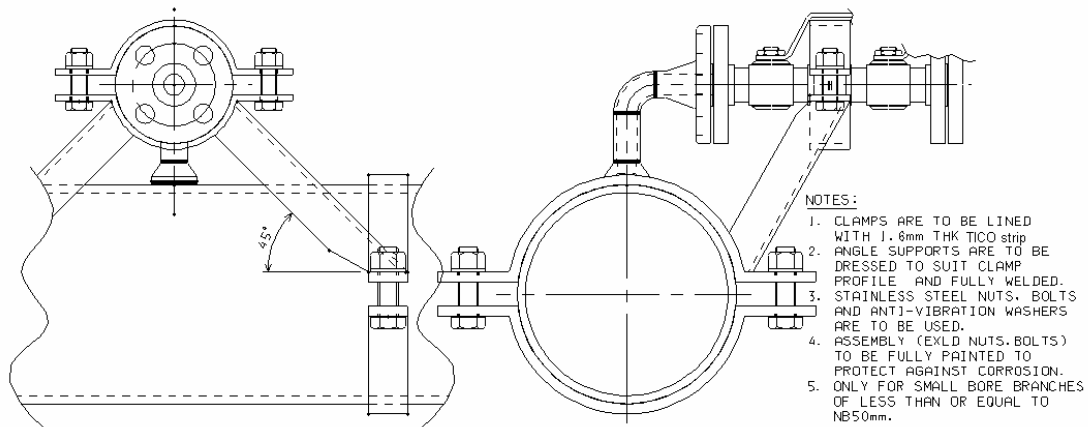


Figure T11-3 Clamp Type of Support

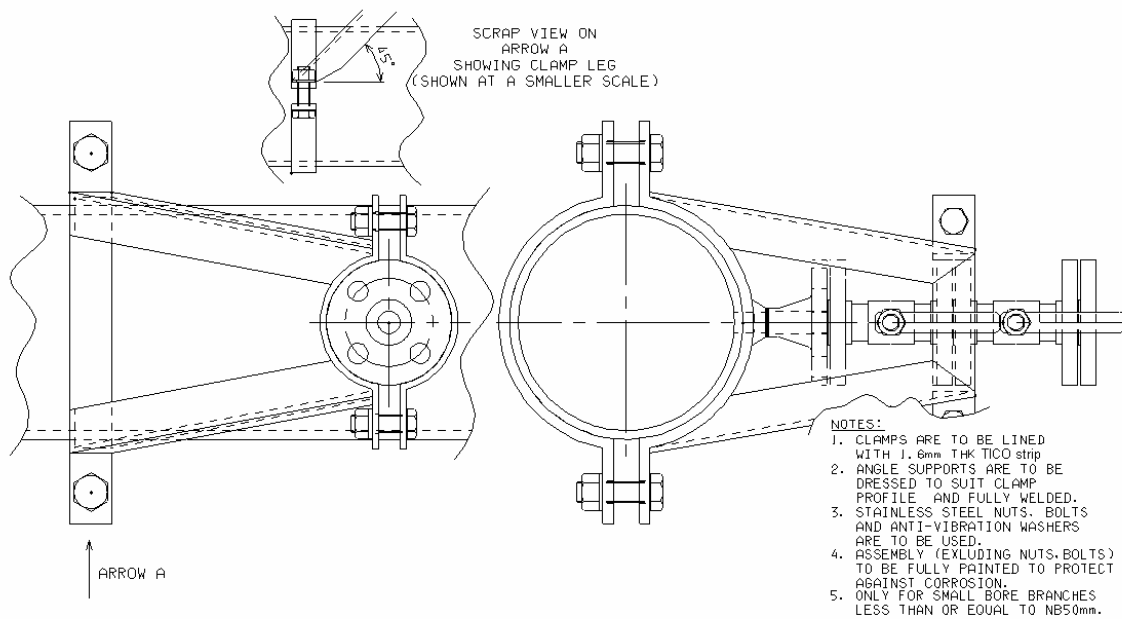
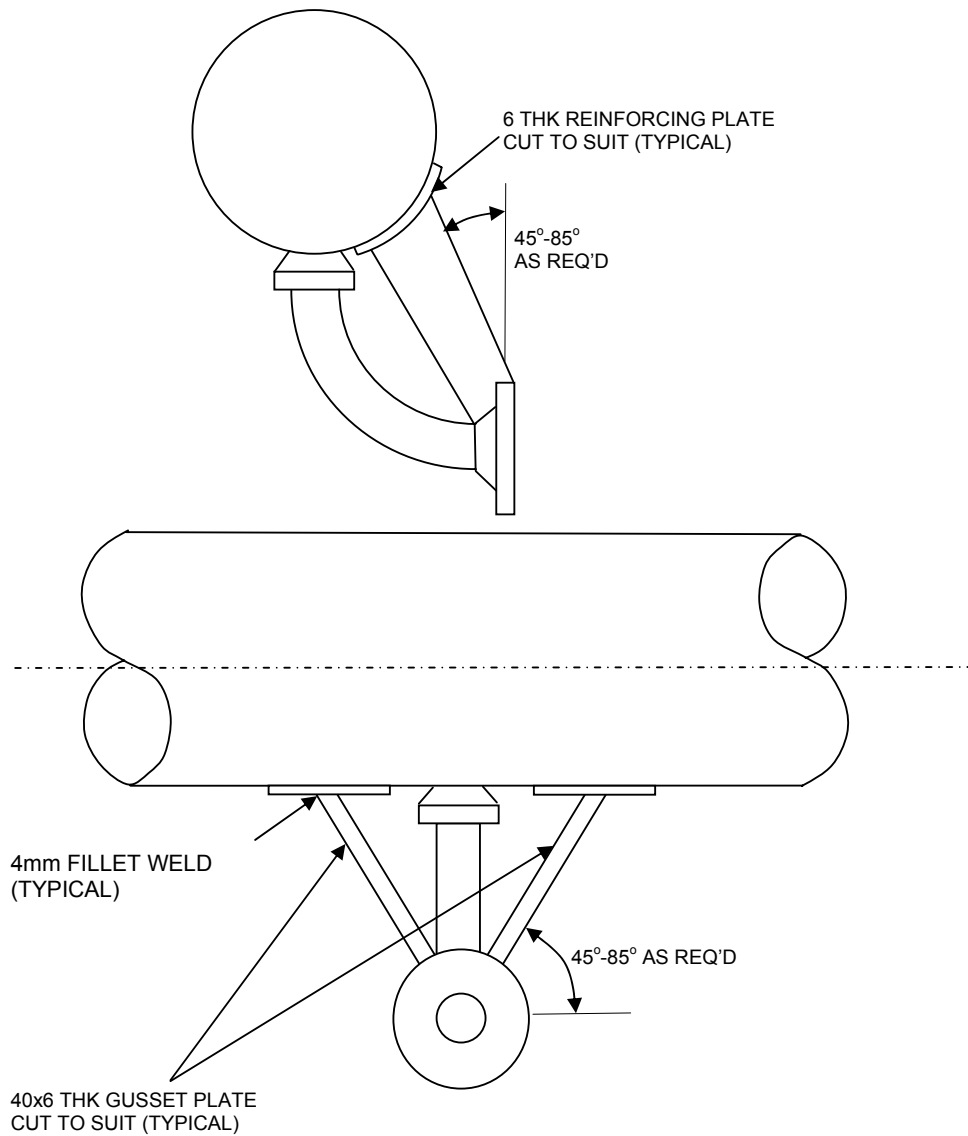


Figure T11-4 Clamp Type of Support



Note: Bracing material to be compatible with parent pipe.

Figure T11-5 Two way welded gusset plates support on unconnected SBC (2" & below)

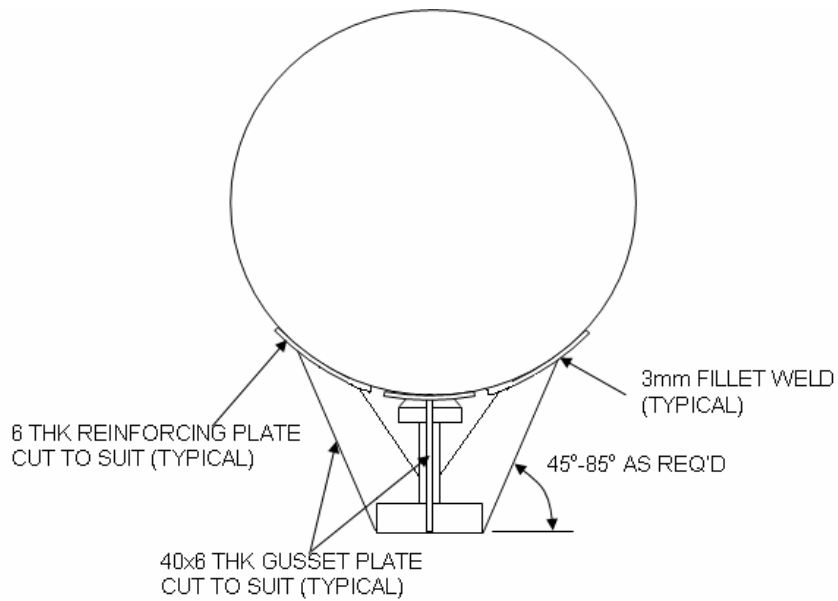


Figure T11-6 Four way welded gusset plates support on unconnected SBC (2" & below)

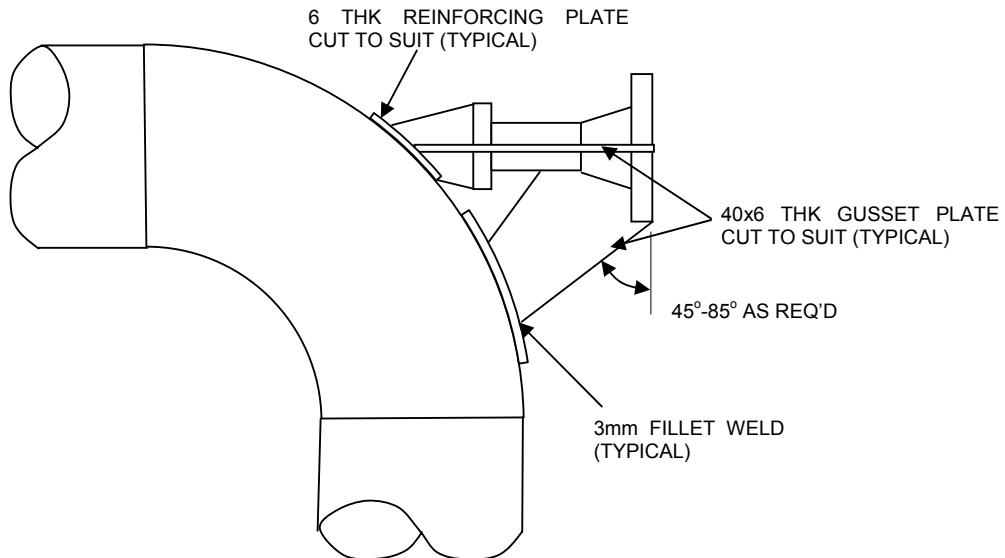


Figure T11-7 Three way welded gusset plates support on unconnected SBC (2" & below)

Technical module

T12 - THERMOWELL CORRECTIVE ACTIONS

T12.1 CORRECTIVE ACTIONS

The purpose of this Technical Module is to give possible design solutions, best practices or remedial actions for thermowells, where an LOF of 1 for the thermowell has been determined from **TM-04**.

T12.1.1 Re-design or Replacement of Thermowell

The thermowell can be re-designed or replaced with a thermowell with a higher fundamental structural natural frequency which meets the criteria in **TM-04**.

T12.1.2 Reduction in Fluid Velocity

The vortex shedding frequency is proportional to the velocity of the fluid flow. If feasible the main line fluid velocity can be reduced sufficiently so there is no longer lock-on between the vortex shedding frequency and the thermowell fundamental natural frequency.

T12.1.3 Finite Element Modelling

The approach in **TM-04** provides an estimation of the fundamental structural natural frequency of the thermowell. Using finite element modelling (see **TM-08**) a more accurate prediction of the thermowell's natural frequencies can be made; in addition dynamic stress levels can be estimated.

T12.1.4 Velocity Collars

Velocity collars are used to provide support at the pipe wall where the thermowell enters the flow stream. The principle is to reduce the unsupported length and therefore increase the thermowell natural frequency. However it is difficult to ensure sufficient contact with the main line pipework and the velocity collar and when there is no contact the natural frequency is unaltered and in the worst case reduced due to the mass of the velocity collar. Therefore, velocity collars should not be used as the primary means to address any issue.

T12.1.5 Dynamic Strain Measurements

Dynamic strain can be measured on thermowells using bonded strain gauges (see **TM-09**). These are usually difficult to install during operation and this is therefore a specialist technique.

T12.1.6 Vibration Velocity Measurements at Thermowell Tip

Where the diameter of the internal bore of the thermowell and the operating temperature allow there are specialist techniques to measure the velocity down the internal bore of the thermowell at the tip. This will provide a measure of the thermowell dynamic motion and help to identify if it is being excited by vortex shedding as the flow velocity increases.

T12.1.7 Supported via a 4-way welded gusset

For thin walled pipework applying 4-way welded gusset plates between the parent pipe and the connection which supports the thermowell increases the fundamental structural natural frequency of the thermowell. The effect of the increased stiffness can be predicted using the wall thickness modifier, F_M with 4-way welded gussets in **Table T4-1**.

Technical module

T13 - GOOD DESIGN PRACTICE

T13.1 GENERAL

This module gives a summary of good design practice for piping systems with respect to vibration induced fatigue. Examples of good and poor practice are given in **TM-05** and **TM-06**.

T13.2 MAIN LINE

The following should be considered as part of the design process for main lines:

- The piping layout should contain adequate guides and line stops where practicable.
- Long sections vulnerable to large transient deflections should be avoided.
- As many bends as possible should be eliminated and supports added as close to the bend as possible.
- Bends should be separated by at least 10 pipe diameters.
- Use of long radius bends rather than short radius or mitred bend.
- The stiffness of clamps and supports should be adequate to restrain the piping.
- Pipe supports should be added at all heavy masses such as valves.
- The span between supports should be carefully assessed, to minimise long unsupported lengths.
- Spring hangers should be avoided or their number minimised.
- A wear resistant or compliant layer should be inserted between the pipe and supports.

T13.3 SMALL BORE CONNECTIONS

The following should be considered as part of the design process for SBCs:

- 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 the cantilever should be supported in both directions perpendicular to the axis of the connection.
- Any bracing should be from the parent pipe, not from any surrounding structure.
- The diameter of small bore connections should be maximised.
- Use of short body contoured fittings (i.e. one piece forgings rather than weldolet and nipple) is preferred.
- Threaded connections should not be used.
- Bolted clamps designed to be effective under vibration (e.g. bolted clamps include anti-vibration washers/lock nuts)

T13.4 TUBING

The following should be considered as part of the design process for instrument tubing:

- Sufficient bends or pigtails are incorporated to allow the tubing to accommodate the main line movement
- Any mass upon the tubing, such as valves, gauges and instruments, is well supported.
- Ensuring that all supports are effective.
- The instrument tubing has been designed to a suitable standard and appropriate components have been used, e.g. do not mix fittings of different types, ensure correct assembly.

Details can be found in the "Guidelines For The Management, Design, Installation & Maintenance Of Small Bore Tubing Systems" **[T13-1]**.

Appendix A

CHANGES TO APPROACH FROM MTD GUIDELINES

A.1 GENERAL

This Appendix provides a summary of the principal modifications to the original MTD document "Guidelines for the Avoidance of Vibration Induced Fatigue in Process Pipework" [A-1].

The modifications have been categorised as follows:

- Changes to the overall methodology
- Changes to the assessment methodology (MTD Section 3)
- Changes to design solutions (MTD Section 4)
- Changes to survey methods (MTD Section 5)
- Changes to examples (MTD Appendix B)

A.2 CHANGES TO THE OVERALL METHODOLOGY

Addition / Change	Refer
The original MTD document principally addressed the assessment of a new design. This document covers the application to (i) a new design, (ii) existing plant, and (iii) changes to existing plant.	Chapter 3
Addition of a new Chapter on troubleshooting piping vibration issues on an operational plant.	Chapter 4

A.3 CHANGES TO THE ASSESSMENT METHODOLOGY (MTD SECTION 3)

A.3.1 MTD STAGE 1 (Identification of Excitation Mechanisms)

Addition / Change	Refer
Replacement of the MTD Stage 1 with a new qualitative assessment procedure to identify potential excitation mechanisms and obtain a rank order to prioritise the subsequent actions.	TM-01

A.3.2 MTD STAGE 2 (Detailed Screening of Main Pipe)

Addition / Change	Refer
Addition of screening methodology for surge/momentum changes to valve operation, cavitation and flashing.	T2.8, T2.9
For each excitation mechanism guidance regarding the extent of the pipework to be considered is now provided.	TM-02
Sample input parameters are now provided.	Appendix B
Change to the flow induced turbulence screening method for gas systems to account for the dynamic viscosity of the gas which reduces the degree of conservatism in the original method.	T2.2.3
Change to the mechanical excitation categories (based on machine types and power rating) and the significance of structural transmission.	T2.3
Change to the method for pulsation (flow induced pulsation) to provide a next level assessment based on coincidence between the fundamental acoustic natural frequency and the fundamental Strouhal excitation frequency.	T2.6

A.3.3 MTD STAGE 3 (Detailed Screening of Small Bore Connections)

Addition / Change	Refer
Addition of new SBC types (Types 2, 3 and 4) to the existing simple cantilevered connection	T3.2
Addition of further guidance regarding the assessment of SBCs.	Appendix C
Inclusion of a wider variety of fitting types in addition to the existing weldolet / contoured body / short contoured body fittings.	TM-03
Change to the SBC location assessment methodology to account for cases where there is a high level of energy in the parent pipe or the main line LOF is not known.	T3.3

A.3.4 ADDITION OF THERMOWELL ASSESSMENT METHODOLOGY

Addition / Change	Refer
Addition of a screening methodology for straight, tapered and stepped thermowells.	TM-04

A.4 CHANGES TO DESIGN SOLUTIONS (MTD SECTION 4)

Addition / Change	Refer
Modifications and additions to the corrective actions for main lines and SBCs.	TM-10, TM-11
Addition of corrective actions for thermowells.	TM-12
Addition of a review of specialist predictive techniques	TM-09

A.5 CHANGES TO SURVEY METHODS (MTD SECTION 5)

Addition / Change	Refer
Replacement of the separate vibration acceptance criteria (D1-D11) with a single criterion which covers all geometries, Refer to Figure A-1 .	TM-07
Addition of a review of specialist measurement techniques	TM-08
Addition of more comprehensive guidance on the visual inspection of pipework and instrument tubing, including examples of good and poor practice.	TM-05, TM-06

A.6 CHANGES TO WORKED EXAMPLE (MTD APPENDIX B)

Addition / Change	Refer
Changes to examples to demonstrate the revised assessment methodology.	Appendix D

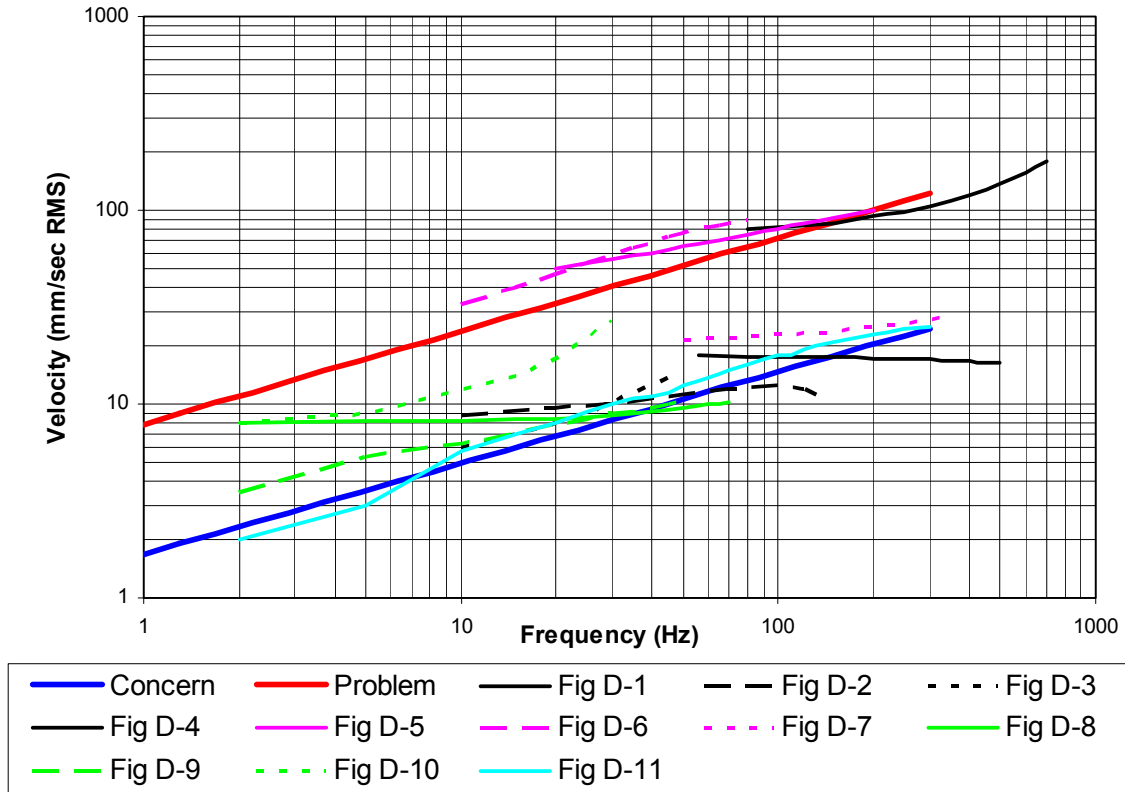


Figure A-1 Previous vibration classifications (**Figures D-1 to D-11** in **[A-1]**) compared to the new “Concern” and “Problem” vibration classifications

Appendix B

SAMPLE PARAMETERS

The data contained within this Appendix are to be used to assist with undertaking the assessments. They will typically result in a more conservative assessment than using actual data. Where possible actual data should be used.

Item	Section	Usage
Main Line Support	B.1	Flow Induced Turbulence main line LOF Section T2.2 – Support Arrangements Surge/Momentum Changes Due to Valve Operation main line LOF Section T2.8 – Support Arrangements
Dynamic viscosity	B.2	Flow Induced Turbulence main line LOF Section T2.2 – Fluid Viscosity Factor
Specific Heat Ratio (C_p/C_v)	B.3	Surge/Momentum Changes Due to Valve Operation main line LOF Section T2.8 – Gas rapid valve opening
Molecular Weights	B.4	High Frequency Acoustic main line LOF Section T2.7 – Sound pressure level calculation Surge/Momentum Changes Due to Valve Operation main line LOF Section T2.8 – Gas rapid valve opening
Vapour Pressure	B.5	Surge/Momentum Changes Due to Valve Operation main line LOF Section T2.8 – Liquid or multi-phase valve opening
Valve Closing Assumptions	B.6	Surge/Momentum Changes Due to Valve Operation main line LOF Section T2.8 – Liquid or multi-phase valve closure
Upstream Pipe Length	B.7	Surge/Momentum Changes Due to Valve Operation main line LOF Section T2.8 – Liquid or multi-phase valve closure
Speed of Sound	B.8	Surge/Momentum Changes Due to Valve Operation main line LOF Section T2.8 – Liquid or multi-phase valve closure
Reynolds Number	B.9	Pulsation – Flow Induced Excitation Section T2.6.3 Thermowell TM-04 – Determination of Strouhal Number

B.1 MAIN LINE SUPPORT

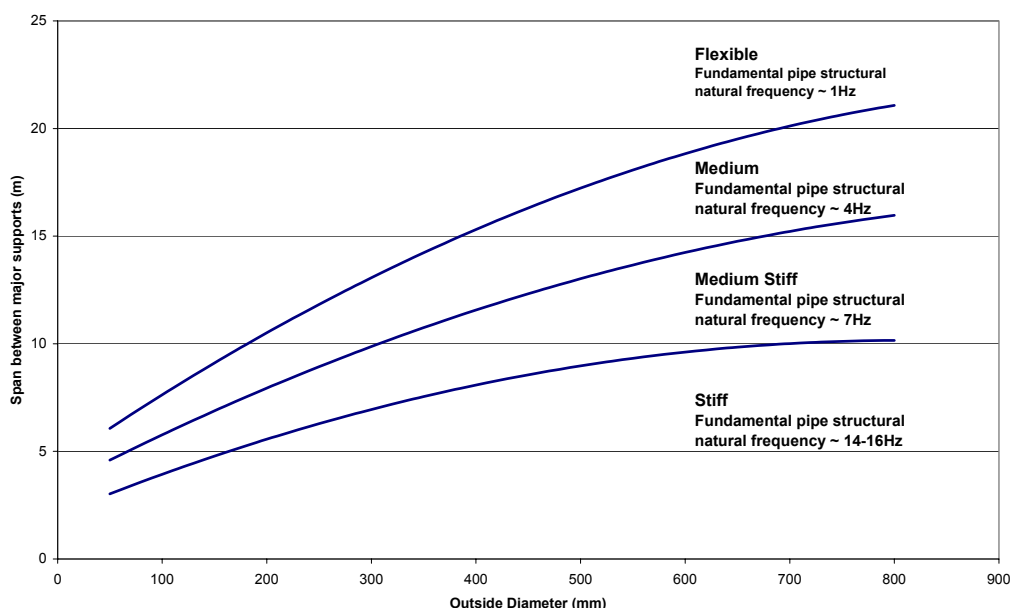
The span length is the distance between effective supports (i.e. between Fixed Support and/or Partially Fixed Support). For a Fixed Support 3 translational degrees of freedom of the main pipe are fixed (i.e. a pipe anchor) and for a Partially Fixed Support 1 or 2 translational degrees of freedom of the main pipe are fixed and the remaining degrees of freedom are free (e.g. sliding shoe, goal post, rest support, guide).

The assumption is made that the structure that the support is connected to is effectively rigid. For example, the use of long goal post type frameworks may lead in some situations to a far less effective support.

Items which are not considered as pipe supports include: spring hangers, shock arrestors, snubbers, viscous dampers, constant effort supports, rods.

It should be noted that main line supports can be difficult to inspect in some locations, such as at height, and it can be difficult to verify if there is good contact and the support is effective (e.g. that the line has not lifted from the support). If there is a question regarding the effectiveness of the support the line should be assessed as if the support was not present.

The equations in **Table T2-1** which use the span length to determine the support arrangement can be presented by the following:



The fundamental natural frequency can also be assessed using piping structural predictive software or modal testing.

B.2 DYNAMIC VISCOSITY

For some common process gases under a pressure 500psi (35barg) the dynamic viscosity (μ_{gas}) can be found from **Figure B-1**. Note: if the pressure is greater than 500psi (35barg) then the gas dynamic viscosity should be determined by other methods.

B.3 SPECIFIC HEAT RATIO (C_P/C_V)

Figures B-2 to B-5 show typical estimates for the specific heat capacity ratios at different temperatures and pressures for Methane, Chlorine, Air and Steam, (if in doubt use the lowest applicable value).

B.4 MOLECULAR WEIGHTS

Substance	Molecular weight grams/mol
Air	29.0
Chlorine	70.9
Methane	16.0
Natural Gas ^{Note}	19.5
Steam	18.0

Note, the molecular weight of natural gas is dependent upon its actual composition.

B.5 VAPOUR PRESSURE

Typical vapour pressures for water are shown in **Figure B-6** below

For oil, glycol and condensate systems it is not possible to list typical values due to variations in the composition of the fluid encountered in different systems. Therefore if the vapour pressures are not known then a Likelihood of Failure (LOF) of 1 should automatically be assigned to the line.

B.6 VALVE CLOSING ASSUMPTIONS

If detailed information on the valves is not available the following conservative assumptions may be applied to the transient analysis:

Valve Type – Globe Valve

Valve Closing Time – 1 second per inch of pipe diameter

B.7 UPSTREAM PIPE LENGTH

When dealing with Surge/Momentum Changes Due to Valve Operation main line LOF (Liquid or multi-phase valve closure), if detailed information on the upstream pipe length is not available, a value of one hundred metres is a conservative assumption

B.8 SPEED OF SOUND**B.8.1 Gases**

The speed of sound (c) in gases can be calculated using the following:

$$c = \sqrt{\frac{\gamma R T_e}{M_w}}$$

where,

- γ is the ratio of specific heat capacities (C_p/C_v) (refer to **Section B.3**)
- R is the universal gas constant, 8314J/K.kmol
- T_e is the gas temperature in Kelvin
- M_w Molecular weight of the gas in grams/mol (refer to **Section B.4**)

B.8.2 Liquids

The speed of sound in some of the common liquids is given in the following table:

Fluid	Speed of sound m/s at 20°C
Benzene	1321
Crude Oil	1385
Ethanol	1180
Ethyl ether	1008
Gasoline	1166
Heptene	1082
Hexane	1203
Hydraulic oil	1280
Kerosene	1315
Methanol	1123
Naphtha	1225
Nonane	1248
Octane	1192
Pentane	1008
Sea water	1481

B.9 REYNOLDS NUMBER

The Reynolds number (Re) is calculated using the following

$$Re = \frac{\rho v D_{Char}}{1000 \mu}$$

where,

- ρ is the density of the fluid in kg/m³
- v is the mean fluid velocity in m/s
- D_{Char} is the characteristic dimension in mm
 - for Pulsation – Flow Induced Excitation (**Section T2.6.3**) D_{Char} is internal diameter of mainline
 - for Thermowells (**TM-04**) D_{Char} is the tip diameter of the thermowell (D_1 for straight thermowells and D_2 for tapered or stepped thermowells)
- μ is the dynamic viscosity in Pa.s (refer to **Section B.2**)

APPENDIX B – SAMPLE PARAMETERS

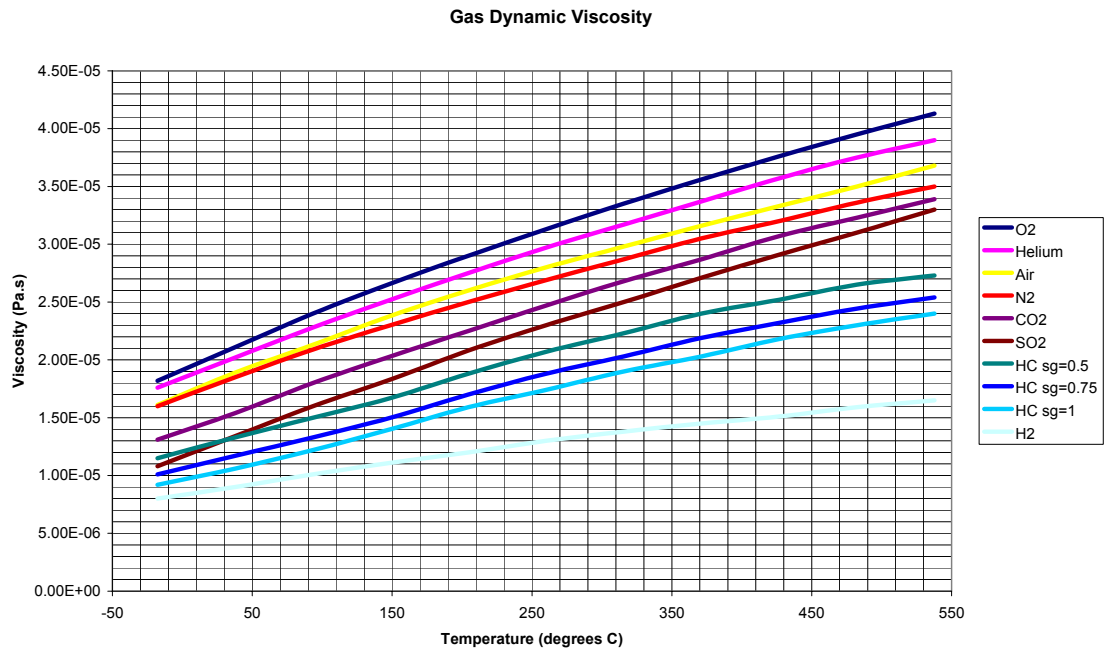


Figure B-1 Variation of gas dynamic viscosity with temperature [B-1]

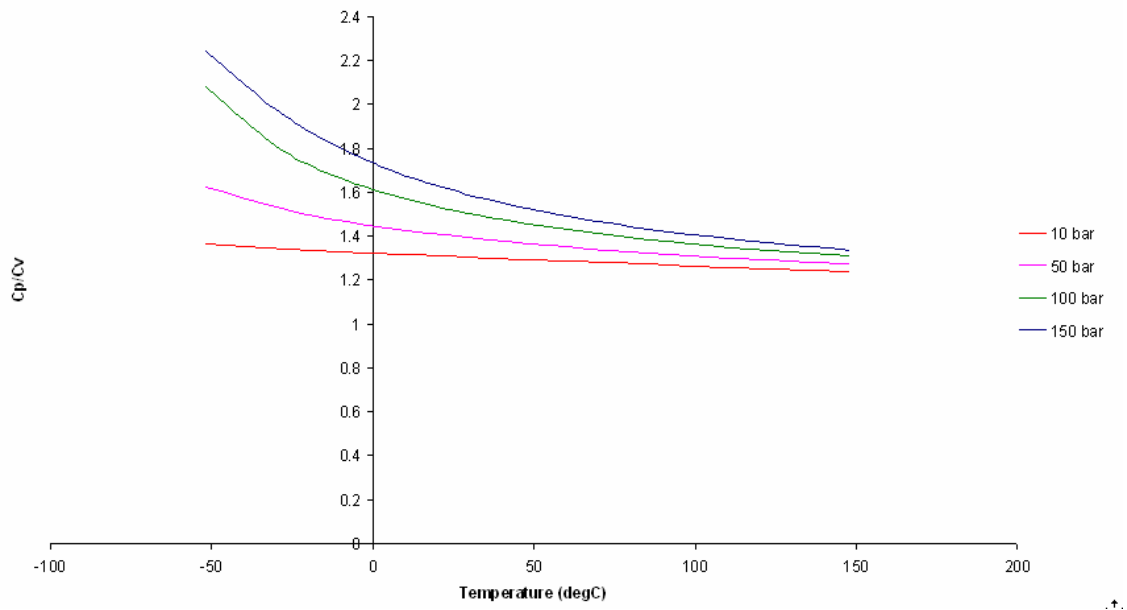


Figure B-2 Specific Heat Ratio - Methane

APPENDIX B – SAMPLE PARAMETERS

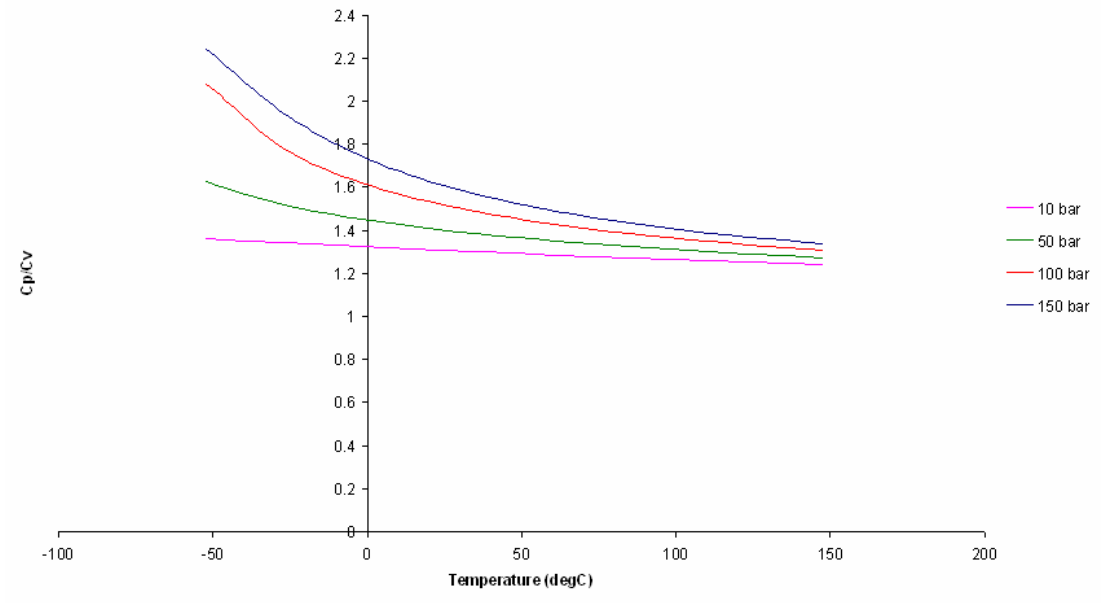


Figure B-3 Specific Heat Ratio – Chlorine

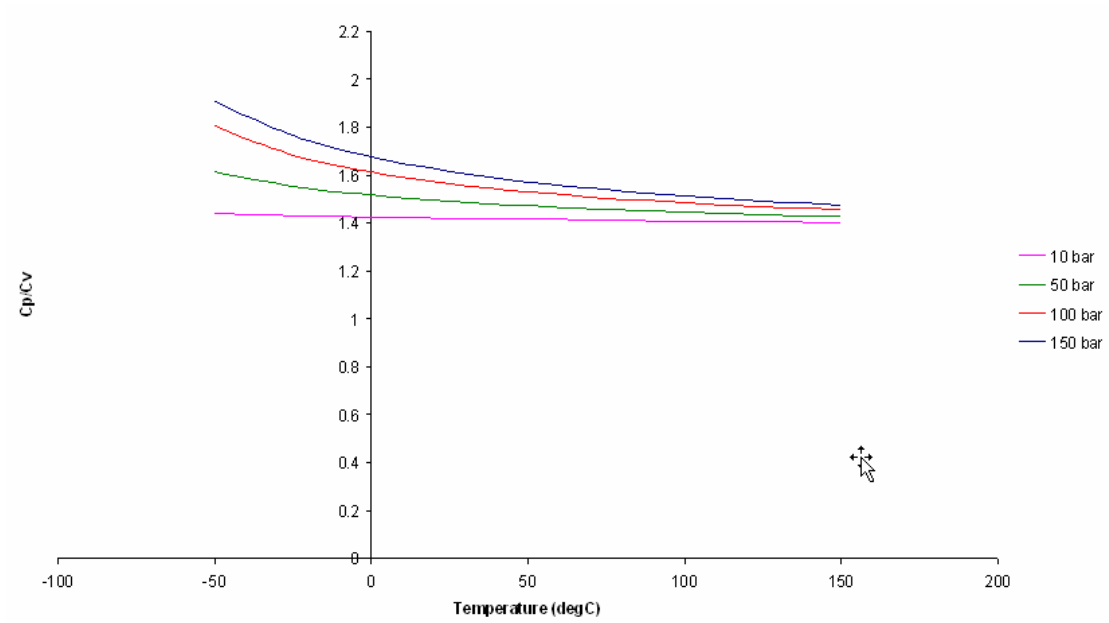


Figure B-4 Specific Heat Ratio – Air

APPENDIX B – SAMPLE PARAMETERS

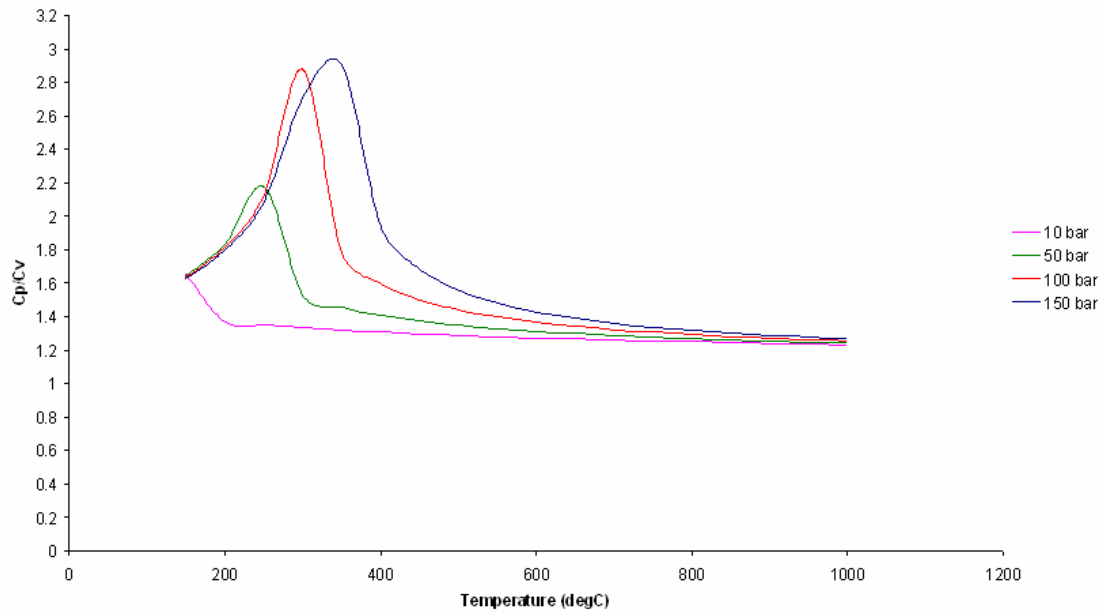


Figure B-5 Specific Heat Ratio – Steam

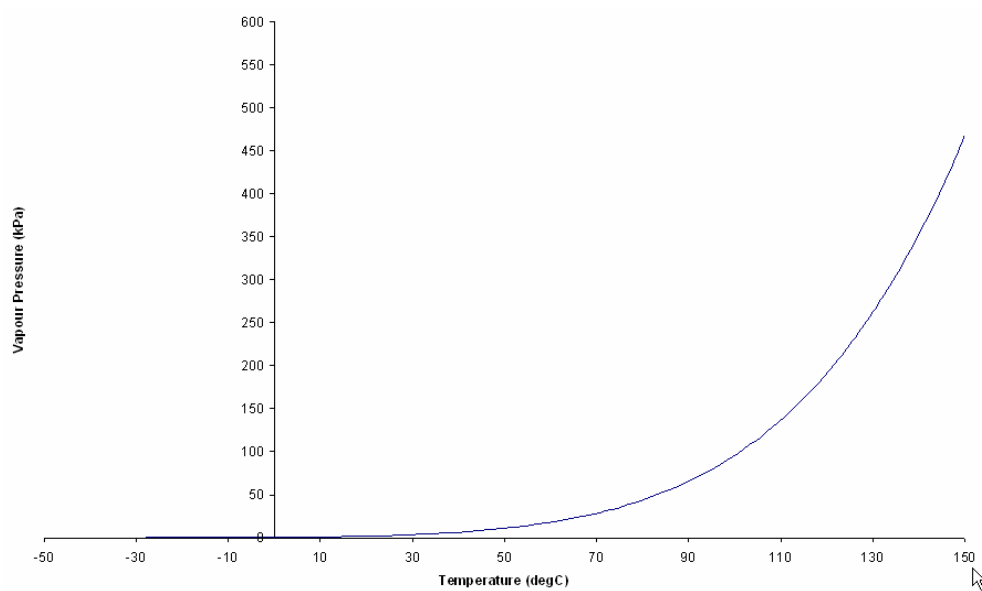


Figure B-6 Vapour Pressure for Water

Appendix C

SBC LOF ASSESSMENT GUIDANCE

C.1 GENERAL

The data given in this Appendix are to help the assessment in **TM-03**. Each section relates to a certain assessment type as detailed in the table below.

Section	Description	Type 1	Type 2	Type 3	Type 4	Location Assessment Methodology
C.1.1	Length of Branch	*	*	*	*	
C.1.2	Number of valves	*	*	*	*	
C.1.3	Diameter of SBC	*	*	*	*	
C.1.4	Type of Fitting	*	*	*	*	
C.1.5	Fitting Span Factor		*	*	*	
C.1.6	Supported mass on first span			*		
C.1.7	Unsupported mass on first span			*		
C.1.8	Determining if mass is present			*		
C.1.9	Parent Pipe Schedule					*
C.1.10	Location on Parent Pipe					*
C.1.11	Splitting line into two Type 1 SBC		*		*	

C.1.1 Length of Branch

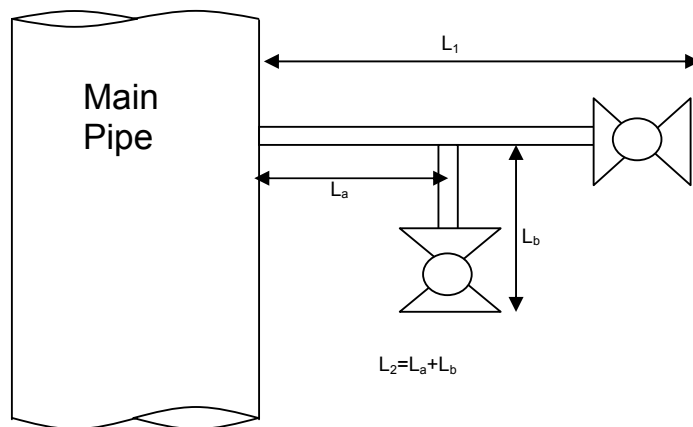
The length of the connection is one of the key parameters that determines the fundamental natural frequency. A longer unsupported branch results in lower natural frequencies and hence greater likelihood of failure. Length is measured from the main pipe wall to the end of the branch assembly (including valve(s) if fitted).

Length	Score
over 600mm	0.9
up to 600mm	0.7
up to 400mm	0.3
up to 200mm	0.1

The overall length of the connection for a simple small bore connection, e.g. a high point vent or low point drain, should be taken as the total distance from the wall of the parent pipe to the end of the branch assembly. If there is any extension to the connection with negligible mass and stiffness, e.g. instrument tubing / impulse line, then this can be ignored from the length assessment.

If the length of the connection is less than 600 mm, then the length should be estimated to within + 100 mm for assessment purposes, i.e. the length estimated should be conservative.

For the case where the SBC contains a branch the length from the main line connection point to the tip of each branch should be considered. The length of the longest branch should be used for the assessment (i.e. the greater of L_1 or L_2).



C.1.2 Number 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. This applies for flange and/or valve ratings below ANSI 900.

Number of Valves	Score
2 or more	0.9
1 or integral double block and bleed valve	0.5
Flange only	0.2

The assessment is made on the basis of the number of valves located at the end of the 'overall length' of the connection. If a lightweight integral double block and bleed valve is used then this is treated as a single valve.

Where the flange and/or valve rating is ANSI 900 or greater the following applies:

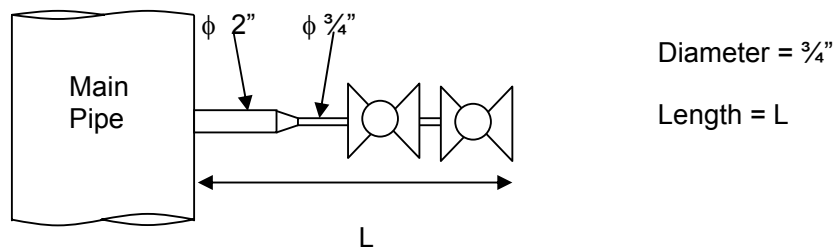
Number of Valves (ANSI 900 or greater)	Score
1 or more	0.9
Flange only	0.5

C.1.3 Diameter of Small Bore Connection

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)		Score
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

Where there is a necked section on the SBC, the smaller diameter and the longest length should be considered - this will result in a conservative assessment.



C.1.4 Type of Fitting

By considering the susceptibility to fatigue, stress intensity factor, and natural frequencies of the fittings, the score for the fitting can be characterised. Fittings with higher natural frequencies, low stress intensity factors and low susceptibility to fatigue, such as Short Contoured Body type, therefore have lower likelihood of failure.

An example of each of these fittings is given in **Table C-1**.

If there is doubt as to which type of welded fitting is used in a particular application then the fitting designation with the higher likelihood of failure should be assumed, as this will give a conservative assessment.

The method assumes fully welded out connections, if this is not the case more detailed analysis/modelling is required to determine the effect of partially welded out fitting on the stress concentration factor.

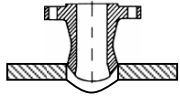
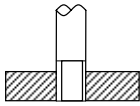
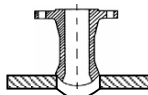
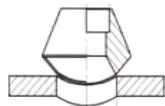
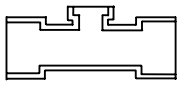
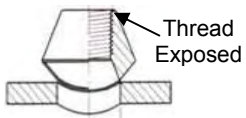
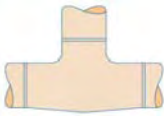
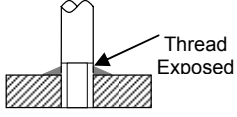
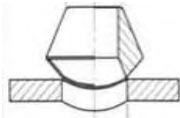
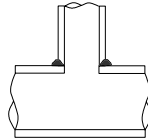
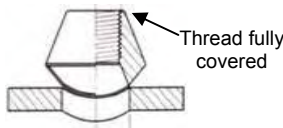
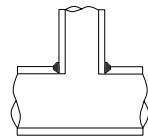
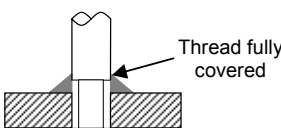
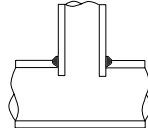
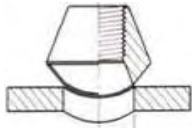
Type of fitting	Sketch	Type of fitting	Sketch
Short Contoured Body		Screwed	
Contoured Body		Sockolet	
Forged Reducing Tee		Thredoilet (Back welded) (Thread Exposed)	
Welded Tee		Screwed Back welded) (Thread Exposed)	
Weldoilet		Set-on	
Thredoilet (Back welded) (Thread fully covered)		Set-in	
Screwed (Back welded) (Thread fully covered)		Set-thro'	
Thredoilet			

Table C-1 Fitting Types – example drawings

C.1.5 Fitting Span Factor

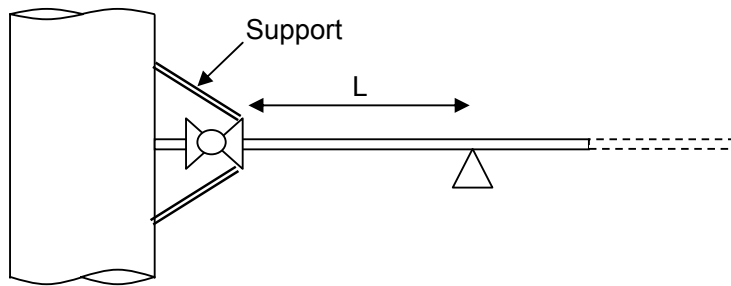
The fitting span factor is determined by identifying the fitting type at the connection with the main line and the SBC (refer to **Table C-1**) and selecting a value from **Table T3-1**. The fitting span factor considers the susceptibility to fatigue, stress intensity factor, and natural

frequencies of the fittings types and it is used to adjust the minimum and maximum span length accordingly (refer to **Tables T3-2** and **T3-3** and **Figure T3-1** to **T3-4**).

In the case where there are different fittings at each end of the connection, use the smaller of the two Fitting Span Factors.

C.1.6 Supported mass on first span length

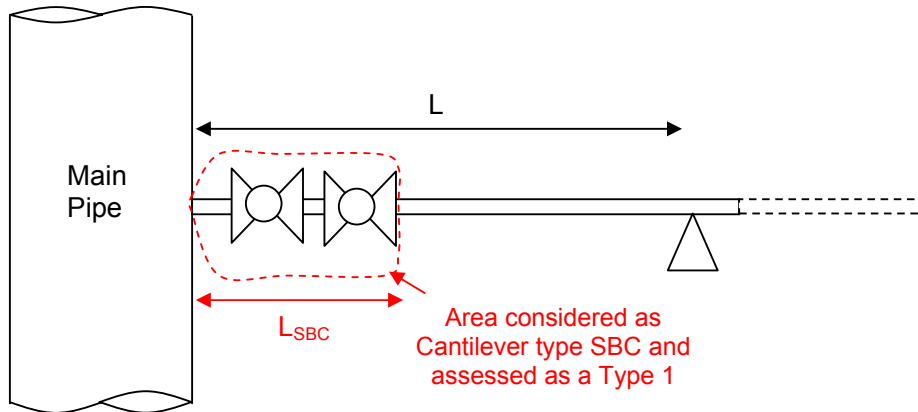
If any valve or flange at the point of connection to the main pipe is braced to the main pipe, the span length is taken from after this support to the first support to deck, with the span assessed as having no added masses. The brace should be sufficiently stiff in order to restrain the mass in all directions of movement.



C.1.7 Unsupported mass on first span

If there is an unsupported mass, i.e. a valve or flange, between the main line and the first support, then the assessment is done in three parts:

1. Undertake as if the SBC was terminated at the final mass element, and modelled as a Type 1 cantilever SBC ($LOF_{GEOM(C)}$).
2. Compare the span length with the maximum span length to determine $LOF_{GEOM(D)}$
3. Compare the span length with the minimum span length to determine $LOF_{GEOM(E)}$



C.1.8 Determining if Mass is Present

A span is defined as involving a mass if it contains any form of additional weight other than a straight run of pipe, e.g. involving a valve or flange.

C.1.9 Parent Pipe Schedule

This is the pipe schedule (or wall thickness) of the parent pipe at the connection. 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	Score
10S	0.9
20	0.8
40	0.7
80	0.5
160	0.3
>160	0.3

If the actual parent pipe schedule lies between two of the 'standard' pipe schedules listed, then the lower 'standard' schedule of the two should be chosen for assessment purposes.

C.1.10 Location on Parent Pipe

Small bore connections located at rigid supports on the main pipe are unlikely to vibrate as the support will force a node of vibration on the main pipe, and as a result little or 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.

The location score is based on the connection being close to certain key locations on the parent pipe ('close to' is defined in the following table). In order of decreasing importance these are:

- If close to a fixed support on the parent pipe (i.e. within ± 2 main pipe diameters) the Fixed Support Score applies
- If one or more of the other locations (i.e. Valve, Reducer, Bend, Tee or Partially Fixed Support) apply then the highest score applies.
- If no other location applies then the Mid Span score should be used.

For example, if the connection is close to a bend and mid span between supports, then the assessment would be '*bend*'. If, however, the connection was close to a valve, but also close to a fixed support, then the assessment would be '*fixed support*'.

Location	Score	Close to definition
Valve	0.9	Within ± 10 main pipe diameters
Reducer	0.9	Within ± 10 main pipe diameters
Bend	0.9	Within ± 10 main pipe diameters
Tee	0.9	Within ± 10 main pipe diameters
Mid span	0.7	If none of others apply
Partially Fixed Support *	0.6	Within ± 2 main pipe diameters
Fixed support**	0.1	Within ± 2 main pipe diameters

* 1 or 2 translational degrees of freedom of the main pipe are fixed and the remaining degrees of freedom are free, e.g. sliding shoe, goal post, rest support, guide

** 3 translational degrees of freedom of the main pipe are fixed, i.e. a pipe anchor. If uncertain assume Partially Fixed Support.

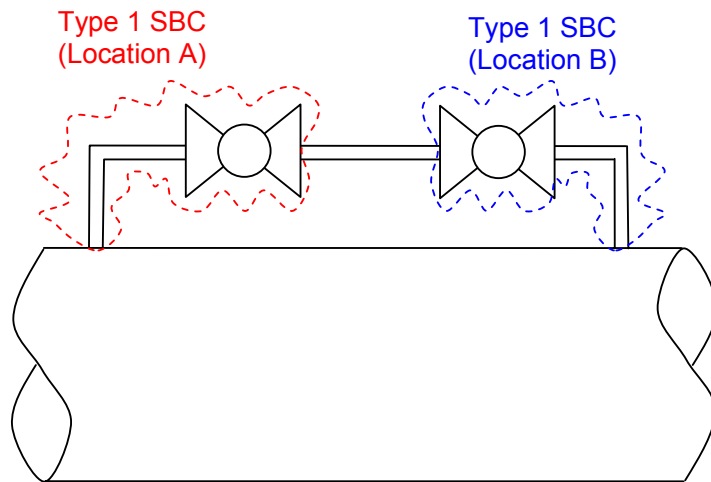
Items which are not considered as pipe supports include: spring hangers, shock arrestors, snubbers, viscous dampers, constant effort supports and rods.

It should be noted that main line supports can be difficult to inspect in some locations, such as at height, and it can be difficult to verify if there is good contact and the support is effective, e.g. that the line has not lifted from the support. If there is a question regarding the effectiveness of the support it should be assessed as if the support was not present.

The $\pm$ main pipe diameters for a Valve, Reducer, Bend and Tee are based upon empirical data, where the decay of turbulent excitation reaches a low level within 10 main line diameters of the source. For Partially Fixed Support and Fixed Support the distance is based upon site experience.

C.1.11 Splitting line into two Type 1 SBC

To take account of the mass on the SBC (e.g. valve or flange), the connection should be split into two Type 1 (refer to **Section T3.2.2.1**) cantilever type connections about the midspan point. Assess both sides as if the free end was the last mass on each half of the line and determine $LOF_{GEOM(A)}$ and $LOF_{GEOM(B)}$.



If one of the masses is located near the mid span of the line it should be considered on the Type 1 SBC assessment for both sides of the SBC. If there is a change in section consider the smallest diameter.

Appendix D: WORKED EXAMPLES

This Appendix contains several worked examples to illustrate the use of the various assessment methodologies.

Example	Description
D1	Gas compression system: main line qualitative assessment
D2	Gas compression system: main line quantitative assessment
D3	Separation system: main line qualitative assessment
D4	Separation system: main line quantitative assessment
D5	Type 1 SBC assessment
D6	Type 2 SBC assessment
D7	Type 3 SBC assessment
D8	Type 4 SBC assessment

D.1 EXAMPLE D1: GAS COMPRESSION SYSTEM: QUALITATIVE ASSESSMENT

This example is based on the assessment of a new design for a gas compression system. The Process Flow Diagram is shown in **Figure D-1**, with the relevant stream data shown in **Table D-1**. The Piping and Instrumentation Diagram is shown in **Figure D-2**.

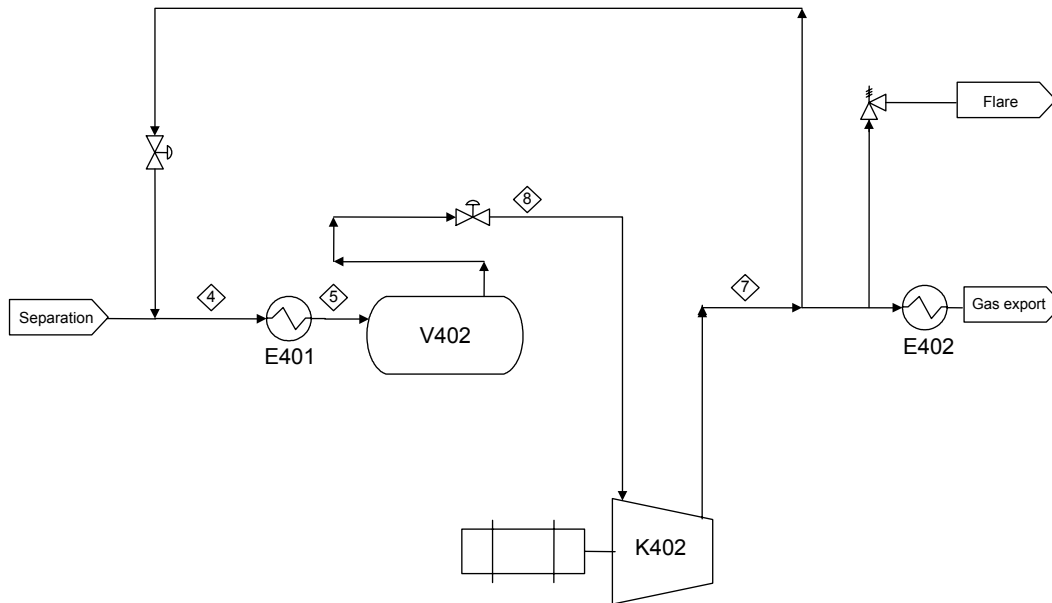


Figure D-1: Example D1: Process Flow Diagram

Stream	4	5	6	7
Vapour Fraction	1	1	1	1
Temperature deg C	141.9	30	30	136.7
Pressure Bar g	25	23.5	23.5	87
Density kg/m ³	18	26	23	62
Viscosity cP	0.02	0.01	0.01	0.02
Flow BPD/MMSCFD	51.24	59348	49.29	49.29
Mass flow kg/hr	59358	59358	53482	53482
Mass heat capacity kj/kg-degC	2.49	2.25	2.23	2.74
Molecular weight	23.22	23.22	21.75	21.75
Compressibility	0.96	0.96	0.91	0.91
C_p/C_v	1.22	1.3	1.34	1.33
Heat of vaporisation kj/kg	162	163	151	92

Table D-1: Example D1: Stream data

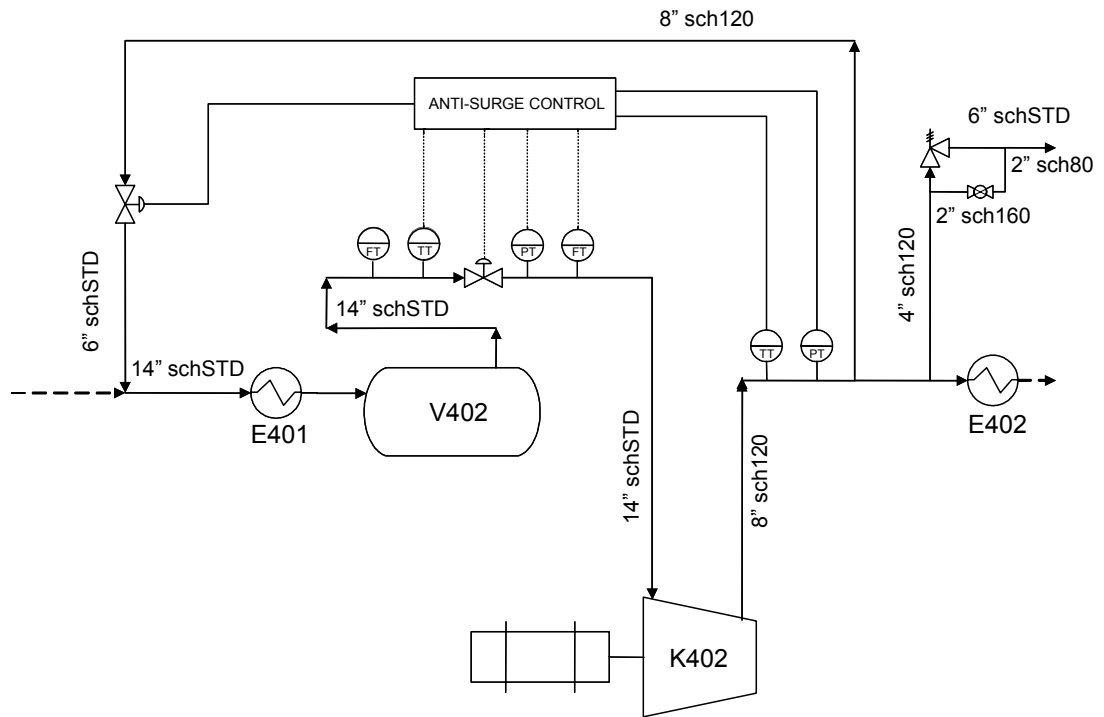
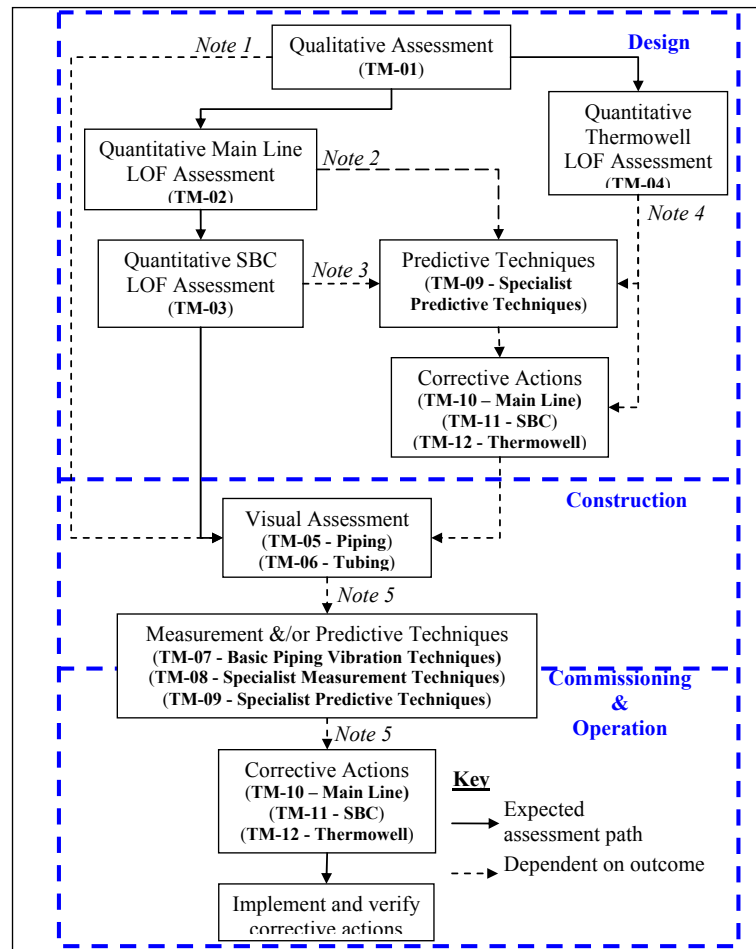


Figure D-2: Example D1: Piping & Instrumentation Diagram

For the case of a new design, the approach given in **Flowchart 3-1** should be followed.



The first step is to undertake a qualitative assessment as described in **Technical Module TM-01**. This should be undertaken with process and/or operations engineers to ensure that all relevant operational cases are identified and taken into account in the assessment.

The qualitative assessment is undertaken by answering each of the questions in **Table T1-1** in turn, considering the various operational scenarios that may occur.

Item 1: Kinetic energy

Item	Aspect	Applicable process fluid(s)	Likelihood Classification			Potential excitation mechanism(s)
			Low	Medium	High	
1	Is there a high level of kinetic energy (ρv^2) of the process fluid?	All	$\rho v^2 < 5,000 \text{ kg/m s}^2$	between $5,000 = \rho v^2$ $< 20,000 \text{ kg/m s}^2$	$\rho v^2 > 20,000 \text{ kg/m s}^2$	Flow induced turbulence (All fluids) refer to Section T2.2 Flow induced pulsation (Gases only) refer to Section T2.6

The kinetic energy (ρv^2) for each process stream is calculated and the maximum value obtained is compared with the limits given. This requires knowledge of the stream data (mass flow rate and fluid density) and also the main line internal diameter. In this case on the suction side of K402 (streams 4, 5, 6) the pipework is

14" schedule STD, whilst on the discharge side (stream 7) the pipework is 8" schedule 120.

Stream	4	5	6	7
Calculated ρv^2 (kg/m.s ²)	1909	1321	1213	5191

In this example the maximum value is 5191 kg/m.s², which, when compared to the limits, results in a 'Medium' classification.

Note: in some situations the highest value of ρv^2 may not be associated with any of the streams given in a Process Flow Diagram. For example, flow through a recycle, bypass or relief line, whilst not considered in the PFD, may give rise to high levels of process fluid kinetic energy. If there is any doubt (and particularly if none of the process streams given on the PFD have a value greater than 5000 kg/m.s²), then a check should be made on those systems which operate intermittently.

In this case, both flow induced turbulence and flow induced pulsation should be considered.

Item 2: Choked flow / sonic velocity

Item	Aspect	Applicable process fluid(s)	Likelihood Classification			Potential excitation mechanism(s)
			Low	Medium	High	
2	Is choked flow possible or are sonic flow velocities likely to be encountered?	Gas	No		Yes	High frequency acoustic excitation refer to Section T2.7

In this case choked flow is possible under two scenarios: either when (i) the recycle valve is just open or (ii) when the relief valve lifts. This results in a 'High' classification. High frequency acoustic excitation must therefore be considered.

Item 3: Machinery

Item	Aspect	Applicable process fluid(s)	Likelihood Classification			Potential excitation mechanism(s)
			Low	Medium	High	
3	Is there any rotating or reciprocating machinery?	All	No	rotating equipment only	reciprocating equipment	Mechanical excitation refer to Section T2.3

The only rotating machinery is the electric motor driven centrifugal compressor K402. This results in a 'Medium' classification. Mechanical excitation must therefore be considered.

Item 4: Positive displacement pumps / compressors

Item	Aspect	Applicable process fluid(s)	Likelihood Classification			Potential excitation mechanism(s)
			Low	Medium	High	
4	Are there any positive displacement pumps or compressors?	All	No	Screw/gear type positive displacement machine	reciprocating type positive displacement machine	Pulsation - reciprocating refer to Section T2.4

There are no reciprocating or positive displacement pumps or compressors. This results in a 'Low' classification.

Item 5: Rotating stall

Item	Aspect	Applicable process fluid(s)	Likelihood Classification			Potential excitation mechanism(s)
			Low	Medium	High	
5	Are there any centrifugal compressors which have the potential to operate under rotating stall conditions?	Gas	No	Compressor has stall characteristics but operational restraints in place to ensure that rotating stall is not encountered	Stall rotating condition unknown. Compressor has rotating stall characteristics and may operate at conditions that will give rise to stall conditions	Pulsation - rotating stall refer to Section T2.5

In this example the compressor is known not to exhibit a rotating stall characteristic.

Item 6: Flashing / cavitation

Item	Aspect	Applicable process fluid(s)	Likelihood Classification			Potential excitation mechanism(s)
			Low	Medium	High	
6	Are there any systems which may exhibit flashing or cavitation	Liquid / Multiphase	No		Yes	Cavitation and Flashing refer to Section T2.9

As this is a gas system this excitation mechanism does not apply.

Item 7: Fast acting valves

Item	Aspect	Applicable process fluid(s)	Likelihood Classification			Potential excitation mechanism(s)
			Low	Medium	High	
7	Are there any systems with fast acting opening or closing valves?	All	No		Yes	Surge/ Momentum changes (refer to Section T2.8)

There is only one fast acting opening valve on the system which is the relief valve. This results in a 'High' classification.

Item 8: Intrusive elements

Item	Aspect	Applicable process fluid(s)	Likelihood Classification			Potential excitation mechanism(s)
			Low	Medium	High	
8	Are there intrusive elements in the process stream?	All	No		Yes	Vortex shedding from intrusive elements to refer TM-04

There is one thermowell on the system. Therefore the resulting classification is 'High'.

Item 9: Slug flow

Item	Aspect	Applicable process fluid(s)	Likelihood Classification			Potential excitation mechanism(s)
			Low	Medium	High	
9	Is there a possibility of slug flow?	Multiphase	No		Yes	Slug flow - seek specialist advice

As this is a gas system this excitation mechanism does not apply.

Item 10: History of pipework vibration

Item	Aspect	Applicable process fluid(s)	Likelihood Classification			Potential excitation mechanism(s)
			Low	Medium	High	
10	Is there a history of pipework vibration issues, or are there any systems which are similar to those on another plant which have a known history of pipework vibration issues?	All	No	Yes: however, suitable corrective action in place and validated for the complete operating envelope.	Yes	Known vibration refer to Chapter 4

As this is a new design there is no history of issues. Providing a check is made and there is also no record of piping vibration problems on a similar (operational) plant then a 'Low' classification can be made.

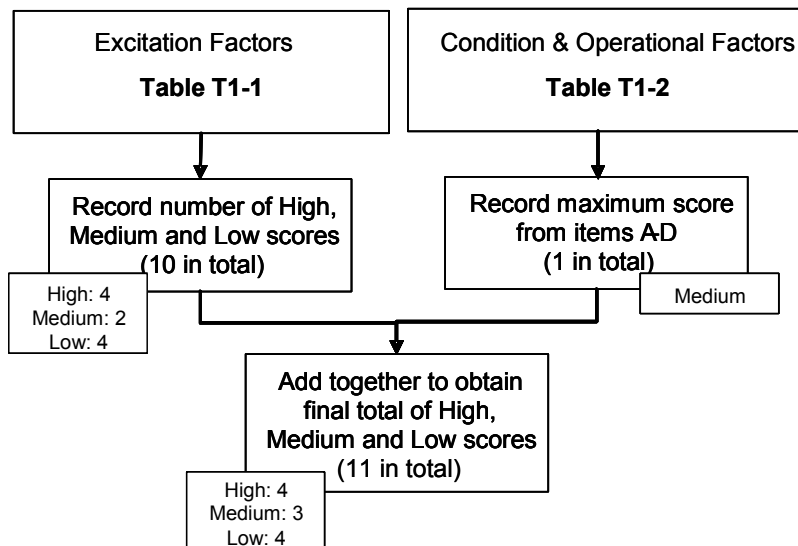
Items A-D: Condition and Operational Factors

Item	Aspect	Applicable process fluid(s)	Likelihood Classification			Contributory factor
			Low	Medium	High	
A	What is the quality of construction?	All	Better than industry standards	At Industry standard	Below industry standards	Build quality
B	What is the effectiveness of the plant maintenance programme (including corrosion management)?	All	Better than industry standards	At industry standard	Below industry standards	Corrosion/ maintenance management
C	Are there any cyclical operations (e.g. batch operation)?	All	No		Yes	Cyclical loading
D	What is the number of unplanned process interruptions in an average year? (this is intended for normal continuous process operations)	All	0-1	2-8	9 or more	Process upsets

As this is a new design items A and B have been assessed as being 'at industry standard'. There is no cyclical operations and a low number of unplanned process interruptions.

Combination of factors

Flowchart T1-1 is used to combine the various factors and to provide a final 'score' for this particular system.



This provides the score for the one system under consideration. If several separate systems had been assessed then each would be individually scored; comparison of the individual system scores would then provide a rank ordering to prioritise the subsequent quantitative assessment.

D.2 EXAMPLE D2: GAS COMPRESSION SYSTEM: QUANTITATIVE ASSESSMENT

This example follows on directly from the qualitative assessment undertaken in **Example D1**. From the results obtained in the qualitative assessment the following excitation mechanisms (based on those scoring Medium or High) should be considered for a quantitative assessment using the relevant methods given in Technical Modules **TM-02** and **TM-04**:

Excitation Mechanism	Technical Module	Section
Flow induced turbulence	TM-02	T2.2
Flow induced pulsation	TM-02	T2.6
High frequency acoustic excitation	TM-02	T2.7
Mechanical excitation	TM-02	T2.3
Surge / momentum changes	TM-02	T2.8
Vortex shedding from intrusive elements	TM-04	

Each of these will be addressed in turn.

D.2.1 Flow Induced Turbulence (see T2.2)

Step 1: Determine ρv^2 (see T2.2.3.1)

For single phase flow $\rho v^2 = (\text{actual density}) \times (\text{actual velocity})^2$

The stream data give the mass flow and density data for the normal full flow condition (stream numbers 4-7). The values of ρv^2 have already been calculated for the qualitative assessment for these stream numbers and are summarised below:

Stream	4	5	6	7
Calculated ρv^2 (kg/m.s ²)	1909	1321	1213	5191

However, there are two further operational cases that need to be considered: (i) recycle operation and (ii) relief conditions. The data for these cases are not usually given in the overall stream data, and must therefore be obtained from other sources.

- (i) Recycle: in the absence of specific information as to the maximum mass flow rate that could be achieved through the recycle line, the maximum compressor discharge flow rate should be used. This is likely to result in a conservative assessment which can then be modified if specific data become available. There are two line sizes to consider:
 - On the compressor discharge side of the recycle valve the recycle line is 8" schedule 120 which gives an internal pipe diameter of 182.4mm. Assuming that the recycle line experiences a maximum flow of 53482 kg/hr with a density of 62 kg/m³ then this would give a value of ρv^2 of 5191 kg/m.s².
 - On the compressor suction side of the recycle valve the recycle line is 6" schedule STD which gives an internal pipe diameter of 154.1mm. Taking a conservative approach and assuming that the gas density is the same as stream 4 (18 kg/m³) with a maximum flow of 53482 kg/hr then this would give a value of ρv^2 of 35294 kg/m.s².
- (ii) Relief: an extract from the valve data sheet is shown below, and gives a flowrate of 49.29 MMscfd once the valve opens, which equates to a mass flow rate of 53482 kg/hr.

APPENDIX D – WORKED EXAMPLES

GENERAL	1	Tag Number	PSV -2223	
	2	Service	MP CMFR DSCH	
	3	PID No.	XX-4530-00	
	4	Line Number	Equipment Number	XX-1110
	5	Area Classification	IEC Zone 2, Group IIA, T3	
	6	Pipe Size / Pipe Specifications	EN X BN	
	7	Compliance to PED Directive 97 / 23 / EC	Required	
	8			
PROCESS CONDITIONS	9	Fluid	State	HC Vapour Gas
	10	Required Capacity	49.29 SMt/d	
	11	Molecular Weight	21.75	
	12	Oper. Pressure	Set Pressure	87 bar-g 98 bar-g
	13	Oper. Temperature	Relieving Temperature	137 °C 137 °C
	14		Constant	0 bar-g
	15	Back Pressure	Variable	10 bar-g
	16		Total	11 bar-g
	17	% Allowable Overpressure	10	
	18	Overpressure Factor		
	19	Compressibility Factor	0.910	
	20	Latent Heat of Vaporization	kJ/kg	
	21	Ratio of Specific Heats	1.33	
	22	Relief Viscosity	cP	
	23	Barometric Pressure		
	24	Built Up Back Pressure	1.0 Bar-g	

Note, that this flow rate is the required capacity. The actual installed capacity may be higher when the actual valve is selected. A check should be made when these data become available.

As with the recycle line there are two line sizes / process conditions to consider:

- Upstream of the PSV the relief line is 4" schedule 120 which gives an internal pipe diameter of 92.1mm. The relief line experiences a maximum flow of 53482 kg/hr with a fluid density of 62 kg/m³. This would give a value of ρv^2 of 80203 kg/m.s².
- Downstream of the PSV the line is 6" schedule STD which gives an internal pipe diameter of 154.1mm. The gas density (obtained from a relief system process simulation model) is 4.0 kg/m³ which, with a maximum flow of 53482 kg/hr, gives a value of ρv^2 of 158823 kg/m.s².

A summary of the various ρv^2 values is given below.

Stream	4	5	6	7	Recycle line (compressor discharge)	Recycle line (compressor suction)	Relief line (upstream of PSV)	Relief line (downstream of PSV)
Calculated ρv^2 (kg/m.s ²)	1909	1321	1213	5191	5191	35294	80203	158823

Step 2: Determine Fluid Viscosity Factor (see T2.2.3.2)

As the fluid in this example is gas the fluid viscosity factor (*FVF*) must be calculated; this requires the gas dynamic viscosity (μ_{gas}). This can either be determined from **Figure B-1**, or in this case from the available stream data. Where data are not available (i.e. the recycle and relief lines) then values have been assumed.

Note: that the units required are in Pa.s, whilst often (as in this case) the units for dynamic viscosity are given as cP. To convert from cP to Pa.s multiply by 10⁻³.

The *FVF* factor is then calculated for each case using

$$Fluid\ Viscosity\ Factor = \frac{\sqrt{\mu_{gas}}}{\sqrt{1 \times 10^{-3}}}$$

Stream	4	5	6	7	Recycle line (compressor discharge)	Recycle line (compressor suction)	Relief line (upstream of PSV)	Relief line (downstream of PSV)
Dynamic viscosity (cP)	0.02	0.01	0.01	0.02	0.02	0.02	0.02	0.01
FVF	0.141	0.1	0.1	0.141	0.141	0.141	0.141	0.1

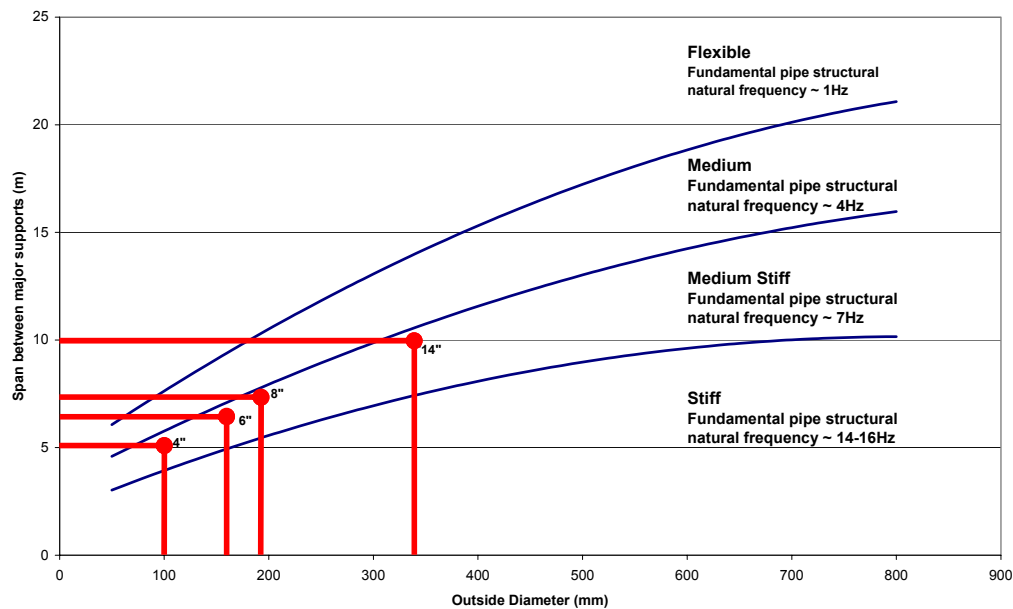
Step 3: Determine Support Arrangement (see T2.2.3.3)

The pipe support arrangement must now be determined. This requires the maximum span lengths between supports to be identified (see guidance in **Appendix B**) and compared with the criteria given in **Table T2-1**. This can be done by (i) working through system isometrics, (ii) walking the lines (on an existing system), or (iii) basing the maximum span length on industry guidance or a particular piping standard or code.

In this example the maximum span lengths have been taken from the project piping standard and are shown below.

Nominal diameter (m)	4"	6"	8"	14"
Maximum span (m)	5.2	6.4	7.3	9.9

These values are then compared with the criteria given in **Table T2-1** (shown graphically below).



In all cases the support classification is "Medium Stiff" (7 Hz).

Step 4: Determine Flow Induced Vibration Factor F_v (see T2.2.3.4)

F_v is determined from the expressions given in **Table T2-2** for the relevant pipe outside diameter and support arrangement. The results are summarised below.

Pipe diameter (schedule)	4" (sch 120)	6" (sch STD)	8" (sch 120)	14" (sch STD)
α	326212	346183	364978	415493
β	-0.9769	-0.9341	-0.9049	-0.8514
F_v	33433	18022	38483	19061

Step 5: Calculation of Likelihood of Failure (LOF) (see T2.2.3.5)

Finally the LOF for each line is calculated using:

$$\text{Flow Induced Turbulence LOF} = \frac{\rho v^2}{F_v} FVF$$

The results are summarised below.

Stream (Sub system) Pipe dimensions	4 (supply to cooler) 14" (sch STD)	5 (supply to suction scrubber) 14" (sch STD)	6 (compressor suction) 14" (sch STD)	7 (compressor discharge) 8" (sch 120)	Recycle line (compressor discharge) 8" (sch 120)	Recycle line (compressor suction) 6" (sch STD)	Relief line (upstream of PSV) 4" (sch 120)	Relief line (downstream of PSV) 6" (sch STD)
ρv^2 (kg/m.s ²)	1909	1321	1213	5191	5191	35294	80203	158823
FVF	0.141	0.1	0.1	0.141	0.141	0.141	0.141	0.1
F_v	19061	19061	19061	38483	38483	18022	33433	18022
LOF	0.014	0.007	0.006	0.019	0.019	0.276	0.338	0.881

D.2.2 Flow Induced Pulsation (see T2.6)

The assessment procedure is shown in **Flowchart T2-4**.

Step 1: Determine critical side branch diameter

$$d_{crit} = 1000 \left(\frac{400}{\pi \rho v^2} \right)^{0.5}$$

This requires the values of ρv^2 calculated previously.

Stream (Sub system) Pipe dimensions	4 (supply to cooler) 14" (sch STD)	5 (supply to suction scrubber) 14" (sch STD)	6 (compressor suction) 14" (sch STD)	7 (compressor discharge) 8" (sch 120)	Recycle line (compressor discharge) 8" (sch 120)	Recycle line (compressor suction) 6" (sch STD)	Relief line (upstream of PSV) 4" (sch 120)	Relief line (downstream of PSV) 6" (sch STD)
Calculated ρv^2 (kg/m.s ²)	1909	1321	1213	5191	5191	35294	80203	158823
d_{crit} (mm)	258	310	324	157	157	60	40	28

Step 2: Identify side branches on each main line with ID $\geq d_{crit}$

Once the critical side branch diameter is calculated then any side branches with an internal diameter greater or equal to d_{crit} must be identified.

- Supply to cooler: the only side branch is the recycle line (which would act as a deadleg if the recycle valve is shut). However, the internal diameter of the recycle line is 154mm (6" sch STD) which is less than d_{crit} .
- Supply to suction scrubber: no side branches exist with a d_{crit} greater than 310mm.
- Compressor suction: although there are connections for the flow and pressure instruments these are all 2" nominal bore or less. There are no side branches with a d_{crit} greater than 324mm.
- Compressor discharge: the d_{crit} is 157mm; the recycle line (8" sch 120) has an internal diameter of 183mm and is therefore a potential problem. The internal diameter of the relief line is 92.1mm which is below the d_{crit} threshold.
- Under recycle conditions there is flow through the recycle line, and therefore any side branches off the recycle line need to be identified. However, in this case, there are none.
- Under relief conditions there is flow through the relief line. The deadleg side branch caused by the 2" bypass around the PSV with the 2" valve locked closed has an internal diameter of 43mm (upstream of the PSV) and 49mm (downstream of the PSV). Both of these are greater than the relevant d_{crit} (40mm and 28mm respectively) and are therefore potential issues.

Step 3: Determine Reynolds Number

For the remaining two side branches the Reynolds Number of the flow in the main line is calculated using:

$$Re = \frac{\rho v D_{Char}}{1000 \mu}$$

Where D_{Char} is internal diameter of main line

Side branch	1	2	3
Description	Recycle line (8" sch 120)	2" PSV bypass (2" sch 160)	2" PSV bypass (2" sch 80)
Main line	Compressor discharge (8" sch 120)	Relief line (4" sch 120)	6" (sch STD)
Fluid density (kg/m ³)	62	62.0	4.0
Fluid velocity in main line (m/s)	9.2	36.0	199.3
D_{int} (mm)	183	92.1	154
Dynamic viscosity (Pa.s)	2e-5	2e-5	1e-5
Re	5.19e6	1.03e7	1.23e7

In all cases the Reynolds Number is below 1.6×10^7 and therefore S_1 needs to be calculated.

Step 4: Calculate Strouhal Number and Excitation Frequency

$$S_1 = 0.420 \left(\frac{d_{int}}{D_{int}} \right)^{0.316} \left(\frac{v}{c} \right)^{-0.083} \left(\frac{Re}{10^6} \right)^{-0.065}$$

Where c is the speed of sound in the gas, given by (see **Appendix B**):

$$c = \sqrt{\frac{\gamma R T_e}{M_w}}$$

The temperature of the gas drops across the PSV and so the speed of sound must be calculated for both the upstream and downstream cases as follows:

Side branch	1	2	3
Description	Recycle line (8" sch 120)	2" PSV bypass (2" sch 160)	2" PSV bypass (2" sch 80)
Main line	Compressor discharge (8" sch 120)	Relief line (4" sch 120)	6" (sch STD)
Molecular weight M_w	21.75	21.75	21.75
γ (Cp/Cv)	1.33	1.33	1.33
R (J/K.kmol)	8314	8314	8314
T_e (deg C)	136.7	136.7	88.0*
c (m/s)	456.4	456.4	428.4

*from relief system process simulation

Side branch	1	2	3
Description	Recycle line (8" sch 120)	2" PSV bypass (2" sch 160)	2" PSV bypass (2" sch 80)
Main line	Compressor discharge (8" sch 120)	Relief line (4" sch 120)	6" (sch STD)
Side branch internal diameter (d_{int}) (mm)	183	43	49
Main line internal diameter (D_{int}) (mm)	183	92	154
c (m/s)	456.4	456.4	428.4
Fluid velocity in main line (m/s)	9.2	36.0	199.3
Re	5.19e6	1.03e7	1.23e7
S_1	0.522	0.350	0.265
Ratio d_{int}/D_{int}	1.0	0.47	0.32
Strouhal Number, S	1.044	0.350	0.265

Note, that for side branch 1 the S_1 value is multiplied by 2 due to the d_{int}/D_{int} ratio.

The next step is to calculate the fundamental Strouhal Number (S) and the fundamental excitation frequency (F_v) for each sidebranch, using:

$$F_v = \frac{Sv}{d_{int}}$$

Sidebranch	1	2	3
Description	Recycle line (8" sch 120)	2" PSV bypass (2" sch 160)	2" PSV bypass (2" sch 80)
Main line	Compressor discharge (8" sch 120)	Relief line (4" sch 120)	6" (sch STD)
Sidebranch internal diameter (d_{int}) (mm)	183	43	49
Main line internal diameter (D_{int}) (mm)	183	92	154
Strouhal Number	1.044	0.350	0.265
F_v (Hz)	52.2	293.4	1076.8

Step 5: Calculate fundamental acoustic natural frequency of side branch

The next step is to calculate the fundamental acoustic natural frequency of the branch using:

$$F_s = 0.206 \frac{c}{L_{branch}}$$

Where L_{branch} is the length of the side branch and

The length of the two side branches are determined from the system isometrics. In each case the length is the total distance between the connection to the main pipe at one end and the closed valve at the other.

Side branch	1	2	3
Description	Recycle line (8" sch 120)	2" PSV bypass (2" sch 160)	2" PSV bypass (2" sch 80)
Main line	Compressor discharge (8" sch 120)	Relief line (4" sch 120)	6" (sch STD)
L_{branch} sidebranch length (m)	5.1	0.3	1.1
c (m/s)	456.4	456.4	428.4
F_s	18.4	313.4	80.2

Step 6: Obtain LOF score

Finally, the ratio of F_v/F_s is calculated:

Side branch	1	2	3
Description	Recycle line (8" sch 120)	2" PSV bypass (2" sch 160)	2" PSV bypass (2" sch 80)
Main line	Compressor discharge (8" sch 120)	Relief line (4" sch 120)	6" (sch STD)
F_v	52.2	293.4	1076.8
F_s	18.4	313.4	80.2
F_v/F_s	2.83	0.94	13.42

Therefore sidebranch 2 scores an LOF of 0.29, while sidebranches 1 and 3 score an LOF of 1.

D.2.3 High Frequency Acoustic Excitation (see T2.7)

There are two cases to consider:

- (i) When the relief valve lifts
- (ii) When the recycle valve opens

The assessment method is shown in **Flowchart T-2-5**. Note that a more comprehensive acoustic fatigue assessment is shown in **Example D-3**.

Step 1: PWL Calculation

The first step is the calculation of the sound power level (PWL) using:

$$PWL \text{ (source)} = 10 \log_{10} \left[\left(\frac{P_1 - P_s}{P_1} \right)^{3.6} W^2 \left(\frac{Te}{Mw} \right)^{1.2} \right] + 126.1 + SFF$$

In both cases sonic flow does not exist and therefore SFF=0.

The results of the calculation of PWL are given below. For the relief valve the relevant data are taken from the valve data sheet with a worst case assumption made that the downstream pressure is 1 Bar absolute; for the recycle valve the worst case condition is taken which assumes the highest mass flowrate combined with the maximum pressure drop across the valve.

Valve	Relief	Recycle
P_1 (Bar g)	98	87
P_2 (Bar g)	0	25
W (kg/hr)	53482	53482
T_e (deg C)	137	136.7
Mw	21.75	21.75

Converting to appropriate units and calculating the PWL for each valve:

Valve	Relief	Recycle
P_1 (Pa absolute)	9 900,000	8 800,000
P_2 (Pa absolute)	100	2600
W (kg/s)	14.86	14.86
T_e (K)	410	410
Mw	21.75	21.75
PWL (dB)	164.7	159.4

The source sound power levels of both sources is above 155 dB.

Examination of the recycle valve data sheet (below) shows the valve is fitted with a multi-path, multi-stage trim which, according to the valve manufacturer, gives a reduction in external sound pressure level of approximately 30dB (118 dB-88.2dB).

Outlet Type Match Number			
System Noise with Stages/Turns Supplied			
*System Noise	dBA		
	81.3		
System Noise versus Trim Type			
Ported Cage	118		
Trim with 16 Stages/Turns	88.2		
Notes:			
1. Sound Pressure Levels shown has a +0, -5 dBA tolerance band.			
2. System Noise is the noise produced by the fluid exiting the valve trim and the valve outlet			

If this reduction is applied to the PWL then the PWL of the recycle valve falls below 155dB and therefore the main line LOF for the recycle line for high frequency acoustic excitation is set to 0.29 as shown in **Flowchart T2-5**.

Conversely the relief valve has no low noise trim and therefore the PWL remains unaltered at 164.7dB and the methodology given in **Flowchart T2-5** is followed:

Step 2: LOF Calculation

- Go to next welded discontinuity (e.g. SBC, welded tee, welded support)

From inspection of the system drawings the first welded discontinuity downstream of the source is the 2" bypass line. This is a weldolet connection.

- Calculate the PWL in the main line at the discontinuity accounting for attenuation

PWL at the discontinuity is calculated using:

$$\text{PWL (discontinuity)} = \text{PWL (source)} - 60 \frac{L_{dis}}{D_{int}}$$

In this case the connection is 0.8m downstream of the source, so :

2" SBC connection	Value
L_{dis} (m)	0.8
D_{int} (mm)	154
$60 \times L_{dis} / D_{int}$	0.312
PWL source (dB)	164.7
PWL discontinuity (dB)	164.4

- Are there any additional sources? In this case, no.
- Is $\text{PWL} > 155\text{dB}$? In this case, yes. Continue to **Flowchart T2-6**.
- Calculate N :

2" SBC connection	Value
D_{ext} (mm)	168.3
T (mm)	7.11
A	0.93989
S	68.229
B	152.207
$\text{Log}_{10}N$	9.9026
N	7.99E9

- Calculate D_{ext}/d_{ext} ($= 168.3 / 60.3$) $= 2.791$
- As this is < 10 calculate FLM_1 ($= 1.2133$)
- Calculate new N ($= 7.99\text{E}9 \times 1.2133$) $= 9.70\text{E}9$
- As the connection is a weldolet, calculate $\text{FLM}_2 = 0.2009$
- Calculate new N ($= 9.70\text{E}9 \times 0.2009$) $= 1.95\text{E}9$
- The piping material is not duplex therefore calculate $L_f = 0.31$
- As $L_f < 0.5$ then $\text{LOF} = 0.29$

The assessment would then return to **Flowchart T2-5** and move to the next discontinuity on the line downstream.

D.2.4 Mechanical Excitation (see T2.3)

The LOF value is dependent on the maximum LOF from **Table T2-3**. The only source is the electric motor driven centrifugal compressor. The compressor would score an LOF of 0.2 while the electric motor (which is $> 15\text{kW}$) scores 0.4. Therefore the overall LOF value to be used is 0.4.

In this case this LOF value would be applied to the suction line (as far as the suction scrubber) and the discharge line (as far as the cooler). Note that as the recycle line is connected to the discharge line before the cooler (and hence potentially subject to vibration transmission from the discharge line) then the recycle line would also score an LOF of 0.4 for mechanical excitation.

D.2.5 Surge / Momentum Changes (see T2.8)

As the system fluid is gas, only one of the mechanisms (dry gas rapid valve opening) applies – see **T2.8.3.1**. There is only one fast opening valve on the system (the relief valve).

Step 1: Peak Force Calculation

For each valve the peak force is calculated using:

$$F_{\max} = \frac{W}{1000} \sqrt{\frac{2 \cdot \gamma \cdot R \cdot T_e}{(\gamma + 1) \cdot M_w}}$$

Valve	Relief valve
W (kg/s)	14.86
γ (Cp/Cv)	1.33
R (J/K.kmol)	8314
T_e (deg K)	410
M_w	21.75
$F_{\max}$ (kN)	6.28

Step 2: Limit Force Calculation

The next step is to calculate the limit force using:

$$F_{\lim} = (16.8 \times \Psi^3 - 1.81 \times \Psi^2 + 525 \times \Psi + 25.3) \times D_{\text{ext}} \times \theta \times \pi \times D_{\text{int}}^2 / (4 \times 10^9) \text{ (kN)}$$

Valve	Relief valve	
	4" sch 120	6" sch STD
T (mm)	11.1	7.11
T sch 40 (mm)	6.02	7.11
Ψ	1.843	1
D_{ext} (mm)	114.3	168.3
D_{int} (mm)	92.1	154.1
θ (medium stiff support)	2	2
$F_{\lim}$ (kN)	1.67	3.55

Step 3: LOF Calculation

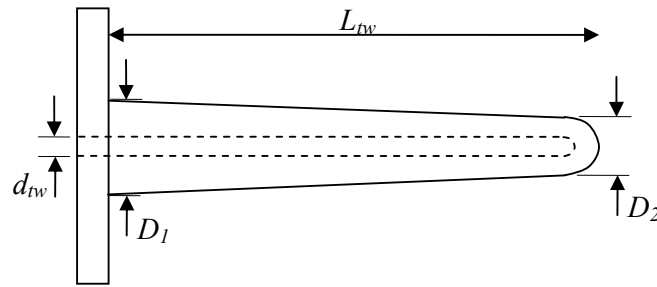
Finally, the LOF value is calculated using $F_{\max} / F_{\lim}$

Valve	Relief valve	
	4" sch 120	6" sch STD
$F_{\max}$ (kN)	6.28	6.28
$F_{\lim}$ (kN)	1.67	3.55
LOF	3.77	1.77

D.2.6 Vortex Shedding from Intrusive Elements (TM-04)

There is a single thermowell associated with TT-001. Dimensions (taken from the manufacturer's drawings) are shown below:

Tapered Thermowell



$$\begin{aligned} d_{tw} &: 8\text{mm} \\ D_1 &: 26.5\text{mm} \\ D_2 &: 18.0\text{mm} \\ L_{tw} &: 225\text{mm} \\ E_{tw} &: 207\text{E}9 \text{ N/m}^2 \\ \rho &: 7850 \text{ kg/m}^3 \end{aligned}$$

The main steps in the assessment are as follows:

Step 1: Predict Thermowell Structural Natural Frequency

This is calculated using the following expression for a tapered thermowell:

$$f_n = \frac{1.12 D_1}{1000 L_{tw}^2} \sqrt{\frac{E_{tw}}{\rho} \left(\frac{k^4 + 5k^3 + 15k^2 + 35k + 70 - 126\delta^4}{5353k^2 + 2142k + 513 - 8008\delta^2} \right)} \quad (\text{Hz})$$

Thermowell	Value
d_{tw} (mm)	8
D_1 (mm)	26.5
D_2 (mm)	18
L_{tw} (m)	0.225
E_{tw} (N/m ²)	207E9
ρ (kg/m ³)	7850
K	0.679245
δ	0.301887
f_n (Hz)	497.9

Step 2: Parent Pipework Wall Thickness Modifier

The parent pipe is 14" schedule STD. The pipe wall thickness (9.5mm) is less than schedule 40 (11.1 mm) and therefore a wall thickness modifier (F_M) of 0.42 is selected (assuming the connection does not have 4 way welded gusset plates).

Step 3: Strouhal Number

Firstly, the Reynolds Number is calculated, using

$$\text{Re} = \frac{\rho v D_{Char}}{1000 \mu}$$

where the characteristic dimension D_{Char} is D_2

Thermowell	Value
Fluid density (kg/m ³)	23
Fluid velocity in main line (m/s)	7.26
D_2 (m)	0.018
Dynamic viscosity (Pa.s)	1e-5
Re	3.01E5

A Reynolds Number of 3.01E5 gives a conservative value of 0.25 for the Strouhal Number (see **Section T4.2.3**)

Step 4: Vortex Excitation Frequency

The vortex excitation frequency is calculated using:

$$F_v = \frac{1000 \times S \times v}{D_{Char}} \quad (\text{Hz})$$

With D_{Char} taken as D_2 (18mm). This gives $F_v = 100.8$ Hz.

Step 5: LOF Calculation

The value of $F_v/(f_n \times F_m)$ is calculated (= 0.48). This is <0.8 and therefore the LOF is set to 0.29.

D.2.7 Summary and Interpretation of Main Line LOF Scores

Stream (Sub system) Pipe dimensions	4 (supply to cooler) 14" (sch STD)	5 (supply to suction scrubber) 14" (sch STD)	6 (compressor suction) 14" (sch STD)	7 (compressor discharge) 8" (sch 120)	Recycle line (compressor discharge) 8" (sch 120)	Recycle line (compressor suction) 6" (sch STD)	Relief line (upstream of PSV) 4" (sch 120)	Relief line (downstream of PSV) 6" (sch STD)
Flow induced turbulence	0.01	0.01	0.01	0.02	0.02	0.28	0.34	0.88
Flow induced pulsation	0.2	0.2	0.2	1.0 (recycle line)	n/a	n/a	0.29	1.0 (2" bypass)
High frequency acoustic excitation	n/a	n/a	n/a	n/a	0.29	0.29	n/a	0.29
Mechanical excitation	n/a	n/a	0.4	0.4	0.4	n/a	0.4	n/a
Surge / momentum changes	n/a	n/a	0.22	n/a	n/a	n/a	3.77	1.77

Main Line LOF ≥ 1.0

A summary of the actions required for a main line LOF score ≥ 1.0 are given below.

Score	Action	Technical Module
LOF ≥ 1.0	The main line shall be redesigned, resupported or a detailed analysis of the main line shall be conducted, and vibration monitoring of the main line shall be undertaken (Note 1)	TM-09 TM-07/TM-08
	Corrective actions shall be examined and applied as necessary	TM-10
	Small bore connections on the main line shall be assessed.	TM-03
	A visual survey shall be undertaken to check for poor construction and/or geometry and/or support for the main line and/or potential vibration transmission to neighbouring pipework.	TM-05 TM-06

There are several lines that score an LOF ≥ 1.0 :

- The compressor discharge line scores an LOF of 1.0 for flow induced pulsation. This is due to the recycle line acting as a deadleg when the recycle valve is shut. The screening method could be used to see whether varying the length of the recycle line deadleg could reduce the LOF score. Alternatively, a more detailed calculation could now be performed using an acoustic simulation of the recycle line and compressor discharge pipework to accurately predict the acoustic natural frequencies and the excitation frequencies of the recycle line (see **Section T9.5** and reference **[T9-6]** in particular). This would identify whether coincidence will occur for the range of flow rates anticipated, and whether the resulting shaking forces are unacceptable.

Remedial measures should be investigated as outlined in **T10.6**. For example, at the design stage it may be feasible to shorten the length of the recycle line between the discharge line and the recycle valve by re-locating the recycle valve. This would have the effect of increasing the acoustic natural frequencies of the recycle line such that any predicted coincidence no longer occurs. Vibration monitoring of the recycle and discharge line should also be considered during operation.

Finally, all small bore connections on the discharge line and the 'deadleg' recycle line should also be assessed.

- Similarly the relief line downstream of the PSV scores an LOF of 1.0 due to the 2" branch downstream of the PSV acting as a deadleg when the PSV lifts. In this case the screening method could be used to see whether varying the length of the 2" deadleg could reduce the LOF score. Alternatively, as with the recycle line above, a more detailed assessment of the 2" branch pipework could be undertaken to identify the range of coincidence between the fundamental excitation frequency and the acoustic natural frequencies of the 2" deadleg (see **Section T9.5**).

It might be argued that in this case the high flow velocities which give rise to this potential mechanism will only occur for a short time period until the system is depressurised, and therefore the possibility of a fatigue failure may be limited. However, in this case it would be prudent to adopt a conservative approach (particularly for such a safety critical system).

- The relief line upstream and downstream of the relief valve scores an LOF greater than 1.0 due to the forces generated when the relief valve lifts. In this case some form of detailed assessment should be undertaken to ensure that the pipework and associated supports can withstand the associated dynamic loads. Such an analysis is beyond the scope of this document.

Main Line LOF ≥ 0.5 and < 1.0

A summary of the actions required for a main line LOF score ≥ 0.5 are given below.

Score	Action	Technical Module
1.0 > LOF ≥ 0.5	The main line should be redesigned, resupported or a detailed analysis of the main line should be conducted, or vibration monitoring of the main line should be undertaken (Note 1)	TM-09 TM-07/TM-08
	Corrective actions should be examined and applied as necessary	TM-10
	Small bore connections on the main line shall be assessed.	TM-03
	A visual survey shall be undertaken to check for poor construction and/or geometry and/or support for the main line and/or potential vibration transmission to neighbouring pipework.	TM-05 TM-06

There is one main line that scores an LOF ≥ 0.5 :

- The relief line downstream of the PSV has an LOF of 0.88. This is due to the relatively high flow velocity through the line giving rise to a high level of turbulent energy. In this case it may be feasible to increase the stiffness of the piping (at present it is assessed as 'medium stiff' – changing the assessment to 'stiff' would reduce the LOF value). Small bore connections on this line should also be assessed.

Main Line LOF ≥ 0.3 and < 0.5

A summary of the actions required for a main line LOF score ≥ 0.3 are given below.

Score	Action	Technical Module
0.5 > LOF ≥ 0.3	Small bore connections on the main line should be assessed.	TM-03
	A visual survey should be undertaken to check for poor construction and/or geometry and/or support for the main line and/or potential vibration transmission from other sources.	TM-05
		TM-06

There are several lines that score an LOF ≥ 0.3:

- The relief line upstream of the PSV has an LOF of 0.34 due to flow induced turbulence. No main line issues are anticipated, but a small bore connection assessment should be undertaken.
- Mechanical excitation potentially affects the compressor suction, discharge and the recycle line, and the relief line downstream of the compressor. Again, no main line issues are anticipated, but a small bore connection assessment should be undertaken.

Main Line LOF < 0.3

A summary of the actions required for a main line LOF score < 0.3 are given below.

Score	Action	Technical Module
LOF < 0.3	A visual survey should be undertaken to check for poor construction and/or geometry and/or support for the main line and/or potential vibration transmission from other sources.	TM-05 TM-06

For all main lines a walkdown should be conducted during the construction phase to ensure that the as-built arrangement is fit for purpose, using the guidance given in **TM-06** and **TM-07**.

D.2.8 Summary and Interpretation of Thermowell LOF Score

Stream (Sub system) Pipe dimensions	⁴ (supply to cooler) 14" (sch STD)	⁵ (supply to suction scrubber) 14" (sch STD)	⁶ (compressor suction) 14" (sch STD)	⁷ (compressor discharge) 8" (sch 120)	Recycle line (compressor discharge) 8" (sch 120)	Recycle line (compressor suction) 6" (sch STD)	Relief line (upstream of PSV) 4" (sch 120)	Relief line (downstream of PSV) 6" (sch STD)
Vortex shedding from intrusive elements	n/a	n/a	0.29	n/a	n/a	n/a	n/a	n/a

No issues are anticipated.

D.3 EXAMPLE D3: SEPARATION SYSTEM: QUALITATIVE ASSESSMENT

This example is based on the assessment of an existing separation system where a large increase in water production is being considered. The Process Flow Diagram is shown in **Figure D-3**, with the relevant stream data shown in **Table D-2** (for the original case) and **Table D-3** for the revised case. The Piping and Instrumentation Diagram is shown in **Figure D-4**.

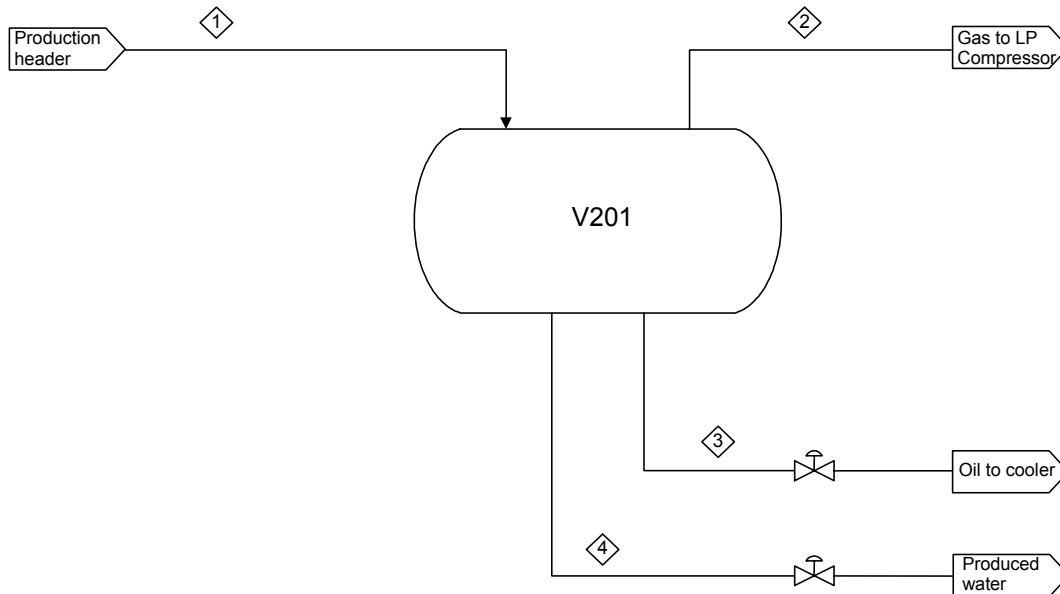
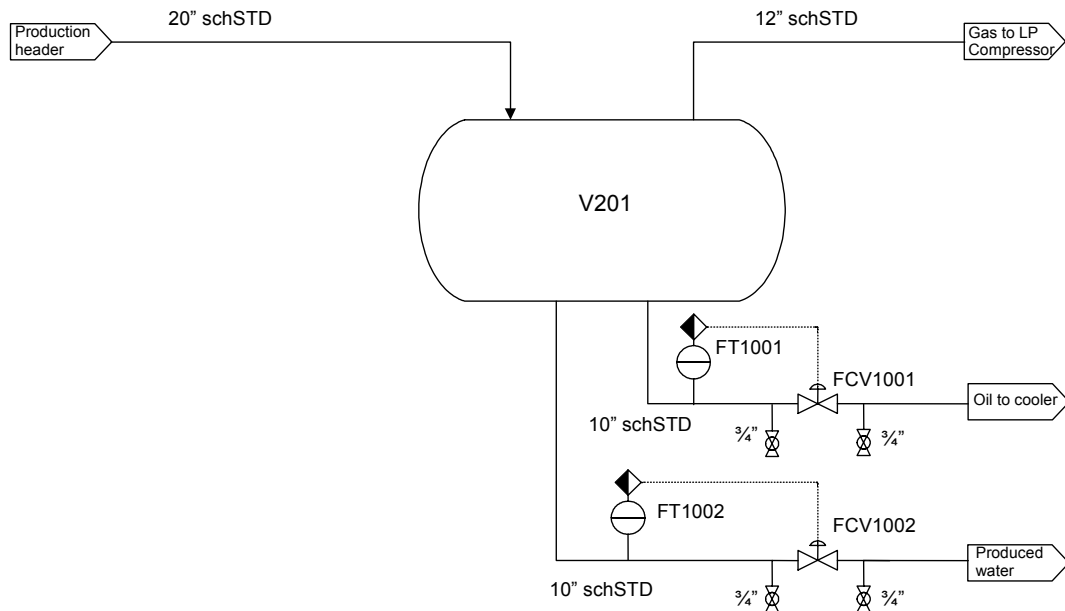


Figure D-3: Example D3: Process Flow Diagram

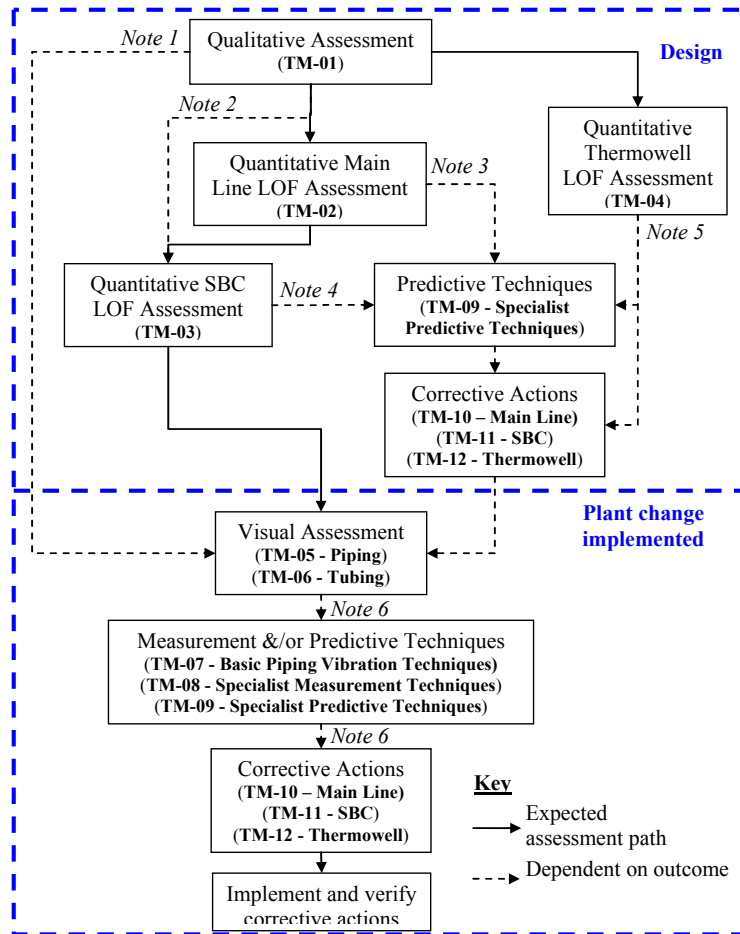
Stream	1	2	3	4
Temperature deg C	50	50	50	50
Pressure Bar g	5.5	5.5	5.5	5.5
Gas flow m ³ /hr	5436	5436		
Gas density kg/m ³	5	5		
Oil flow m ³ /hr	326		326	
Oil density kg/m ³	930		930	
Water flow m ³ /hr	133			133
Water density kg/m ³	988			988
Viscosity cP		0.01	39.35	0.55
Mass flow kg/hr	461764	27180	303180	131404
Molecular weight	28.09	21.99	38.29	18.05
Compressibility		0.98		
Cp/Cv	1.01	1.24	1.08	1.16

Table D-3: Example D3: Stream data (original)

Stream	1	2	3	4
Temperature deg C	50	50	50	50
Pressure Bar g	5.5	5.5	5.5	5.5
Gas flow m ³ /hr	5436	5436		
Gas density kg/m ³	5	5		
Oil flow m ³ /hr	326		326	
Oil density kg/m ³	930		930	
Water flow m ³ /hr	532			532
Water density kg/m ³	988			988
Viscosity cP		0.01	39.35	0.55
Mass flow kg/hr	855976	27180	303180	525616
Molecular weight	28.09	21.99	38.29	18.05
Compressibility		0.98		
Cp/Cv	1.01	1.24	1.08	1.16

Table D-4: Example D3: Stream data (revised – increased water cut)**Figure D-4: Example D3: Piping & Instrumentation Diagram**

For the case of a change to an existing plant the approach given in **Flowchart 3-3** should be followed.



The first step is to undertake a qualitative assessment as described in **Technical Module TM-01**. This should be undertaken with process and/or operations engineers to ensure that all relevant operational cases are identified and taken into account in the assessment.

Note: For this example it is assumed that the existing pipework and process conditions have already been assessed for vibration induced fatigue, and that any existing vibration issues have been addressed, with suitable mitigation measures in place.

The qualitative assessment is undertaken by answering each of the questions in **Table T1-5** in turn, considering the various operational scenarios that may occur.

Item 1: Increase in flow velocities and/or fluid densities

Item	Description	If Yes - Potential Issues
1	Will the modification result in one or more of the following •An increase in flow velocities by more than 5% over previous operational experience? •An increase in fluid density by more than 10% over previous operational experience?	•Flow induced turbulence (all fluids), refer to Section T2.2 •Flow induced pulsation (gases systems only), refer to Section T2.6 •Vortex shedding from intrusive elements (all fluids), refer to TM-04 •Surge/Momentum Change refer to Section T2.8

In this example the only change is the increase in the water mass flow, which affects streams 1 and 4. There are no changes to fluid densities. The increase in fluid velocity is proportional to the increase in volumetric flow rate.

Stream	1	4
Volumetric flow rate (original case) m ³ /hr	5895 (summation of oil, gas and water)	133
Volumetric flow rate (revised case) m ³ /hr	6294 (summation of oil, gas and water)	532
% increase	6.8	300.0

In this example both streams will experience an increase in fluid velocity of over 5%. There are no intrusive elements in the system and therefore only the following potential issues need to be considered for these two process streams:

- Flow induced turbulence
- Surge/momentum changes

Item 2: Change in gas properties

Item	Description	If Yes - Potential Issues
2	For a gas system, will the modification result in one or more of the following: •A change in the molecular weight of the gas by more than $\pm 5\%$ from previous maximum/minimum operational experience? •A change to the temperature of the gas by more than $\pm 5\%$ from previous maximum/minimum operational experience? •A change to the ratio of specific heats (C_p/C_v) of the gas by more than $\pm 5\%$ from previous maximum/minimum operational experience?	For all systems: •Pulsation - Flow induced excitation, refer to Section T2.6 If there is a centrifugal compressor: •Pulsation - rotating stall (gas systems only) refer to Section T2.5 If there is a reciprocating compressor: •Pulsation – reciprocating compressor (gas systems only) refer to Section T2.4

No changes are made to the gas properties and therefore no potential issues are identified.

Item 3: Change in liquid properties

Item	Description	If Yes - Potential Issues
3	For a liquid system incorporating a reciprocating pump, will the modification result in one or more of the following: •A change in the density of the liquid by more than $\pm 5\%$ from previous maximum/minimum operational experience? •A change to the bulk modulus of the liquid by more than $\pm 5\%$ from previous maximum/minimum operational experience?	•Pulsation – reciprocating pump (liquid systems only) refer to Section T2.4

There are no reciprocating pumps in the system and therefore no potential issues are identified.

Item 4: Change to operational configuration of positive displacement compressor or pump

Item	Description	If Yes - Potential Issues
4	Will the modification result in a change to the operational configuration of a positive displacement compressor or pump which is outside existing operational experience e.g.: •The use of a second compressor/pump in tandem? •The use of compressor/pump recycle or partial unloading of the compressor?	•Pulsation – reciprocating compressor or pump (liquid and gas systems only) refer to Section T2.4

There are no reciprocating/positive displacement compressors or pumps in the system and therefore no potential issues are identified.

Item 5: Change to centrifugal compressor operational configuration

Item	Description	If Yes - Potential Issues
5	Will the modification result in a centrifugal compressor being operated at low flow conditions?	•Pulsation - rotating stall (gas systems only) refer to Section T2.5

There are no centrifugal compressors in the system and therefore no potential issues are identified.

Item 6: Choked flow and/or sonic velocities

Item	Description	If Yes - Potential Issues
6	Will the modification result in choked flow and/or sonic velocities in the pipework?	•High frequency acoustic excitation (gas systems only) refer to Section T2.7

Choked flow and/or sonic velocities will not occur and therefore no potential issues are identified.

Item 7: Flashing or cavitation

Item	Description	If Yes - Potential Issues
7	Will the modification result in flashing or cavitation?	•Cavitation and Flashing refer to Section T2.9

No issues are anticipated. If the modification had resulted in an increased pressure drop in the system or an increase in liquid temperature then flashing or cavitation could become an issue – however, that is not the case in this example.

Item 8: Change or addition to existing pipework or associated equipment

Item	Description	If Yes - Potential Issues
8	Will the modification result in a change or addition to the existing pipework or associated equipment (valves, machinery or intrusive elements such as thermowells) which is not a like-for-like replacement?	For changes to valves (including change of valve type or changes to valve closing timings) check for: •Surge/Momentum Change refer to Section T2.8 For changes to machinery check for: •Mechanical excitation refer to Section T2.3 For changes to thermowells check for: •Vortex shedding from intrusive elements refer to TM-04 For changes to pipework, supports, small bore connections and tubing check for: •Poor geometry refer to TM-05 and TM-06

No changes are being made to the existing pipework or associated equipment and therefore no potential issues are identified.

D.4 EXAMPLE D4: SEPARATION SYSTEM: QUANTITATIVE ASSESSMENT

This example follows on directly from the qualitative assessment undertaken in **Example D3**. From the results obtained in the qualitative assessment the following excitation mechanisms (based on those identified) should be considered for a quantitative assessment using the relevant methods given in Technical Module **TM-02**:

Excitation Mechanism	Technical Module	Section
Flow induced turbulence	TM-02	T2.2
Surge/momentum changes	TM-02	T2.8

Each of these will be addressed in turn.

D.4.1 Flow Induced Turbulence (see T2.2)

Step 1: Determine ρv^2 (see T2.2.3.1)

Stream 1 is multiphase, therefore:

$$\rho v^2 = (\text{effective density}) \times (\text{effective velocity})^2$$

$$\text{Effective density} = \text{total mass flow rate} / \text{total volumetric flowrate}$$

$$= 855976 / (5436+326+532)$$

$$= 136 \text{ kg/m}^3$$

$$\text{Pipe diameter (20")} = 508\text{mm}$$

$$\text{Wall thickness (STD)} = 9.525\text{mm}$$

$$\text{Effective velocity} = \text{total volumetric flow rate} / \text{pipe internal area}$$

$$= ((5436+326+532)/3600) / 0.188$$

$$= 9.3\text{m/s}$$

Stream 4 is single phase.

$$\text{Pipe diameter (10")} = 273\text{mm}$$

$$\text{Wall thickness (STD)} = 9.271\text{mm}$$

$$\text{Velocity} = (532/3600)/0.051$$

$$= 2.9 \text{ m/s}$$

The values of ρv^2 for streams 1 and 4 are summarised below:

Stream	1	4
Velocity (m/s)	9.3	2.9
Density (kg/m ³)	136	988
Calculated ρv^2 (kg/m.s ²)	11763	8309

Step 2: Determine Fluid Viscosity Factor (see T2.2.3.2)

Streams 1 and 4 are multiphase and liquid respectively and therefore the fluid viscosity factor (FVF) = 1.

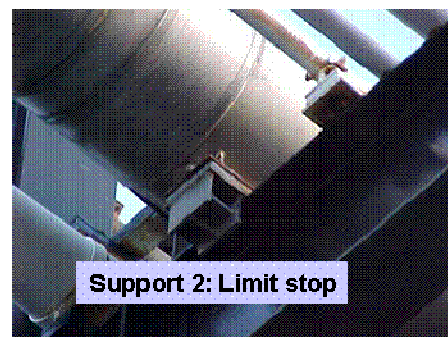
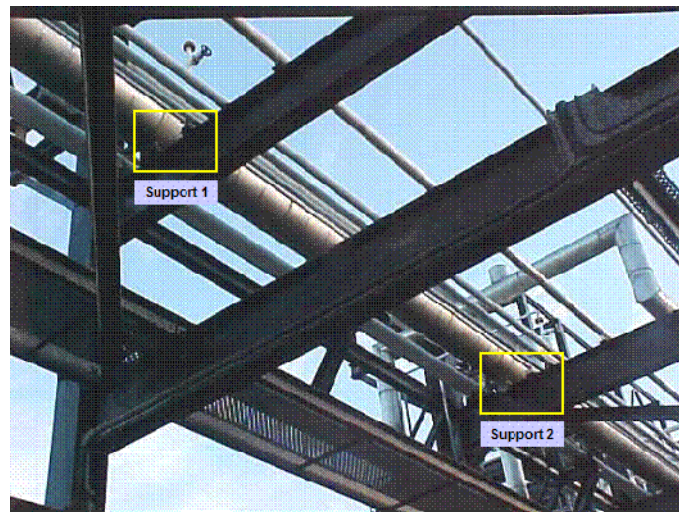
Step 3: Determine Support Arrangement (see T2.2.3.3)

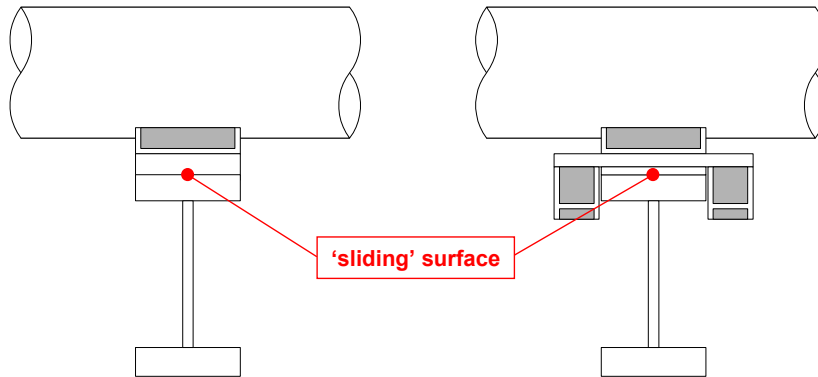
The pipe support arrangement must now be determined. This requires the maximum span lengths between supports to be identified (see guidance in **Appendix B**) and compared with the criteria given in **Table T2-1**. This can be done by working through system isometrics, or as in this case, walking the lines. Alternatively the fundamental natural frequency could be calculated (refer to **Section B.1**) or measuring using modal testing techniques (refer to **Section T8.3**).

The following assessment uses the method given in **Appendix B**.

Separator Inlet

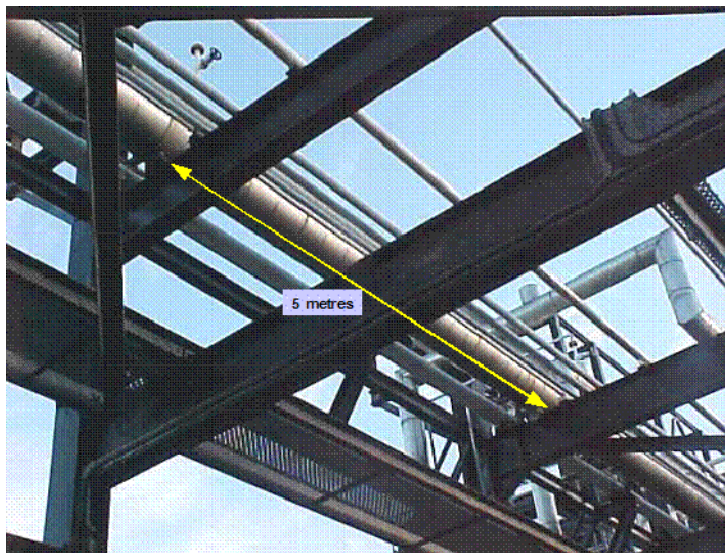
The separator inlet is a 20" NB pipe (actual outside diameter 508mm), supported at regular intervals on a pipe rack. At the left end there is a rest support (**Support 1**), which is designed to support the pipe vertically. The support at the right end (**Support 2**) is a limit stop that will allow the pipe to move from side to side. In both cases the support is attached to a substantial 'I' girder, which forms part of the pipe rack.



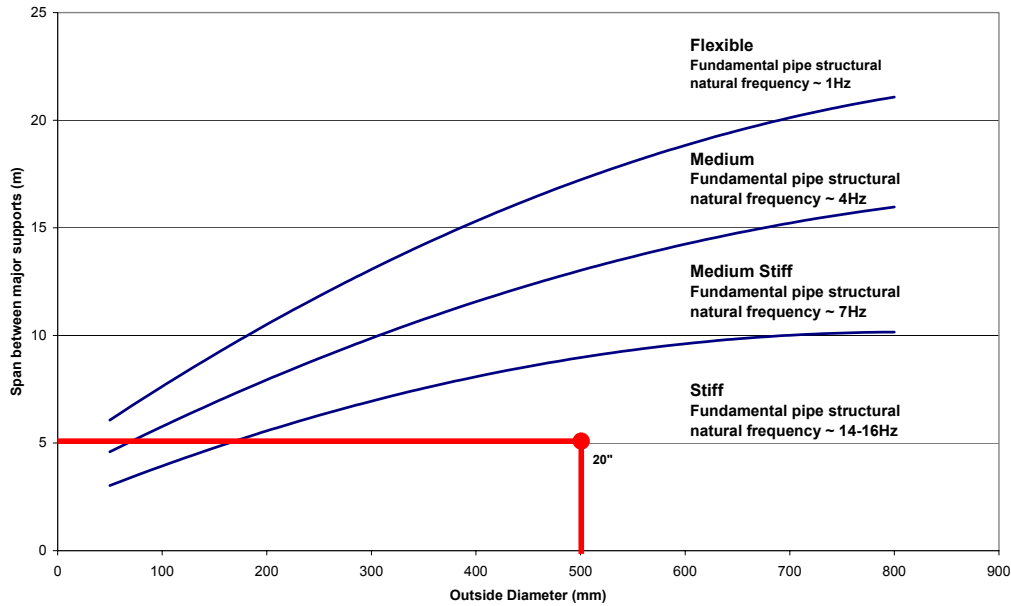


Due to the nature of this type of support, and the self-weight of the pipe, there will be a significant amount of friction between the sliding surfaces. Whilst this friction will not constrain the pipe when subjected to high static loads (e.g. thermal growth), it is usually the case that the friction is sufficient to restrain the pipe when the pipe vibrates.

Both the supports can be considered 'effective', and both have a substantial 'I' girder which forms the primary foundation. The span length can therefore be taken as the distance between these two supports: in this case, approximately 5 metres.



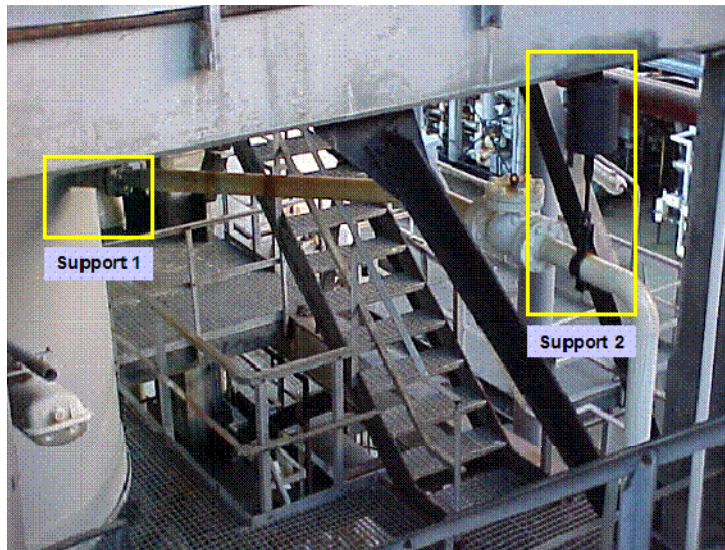
This span length can then be used to determine the support classification as shown below.



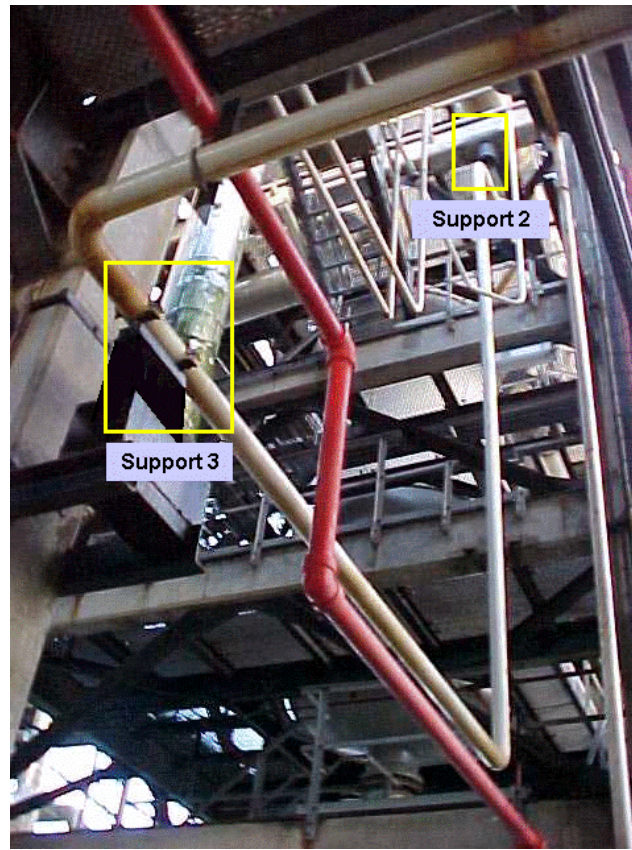
This results in a 'stiff support arrangement' classification.

Produced Water Outlet

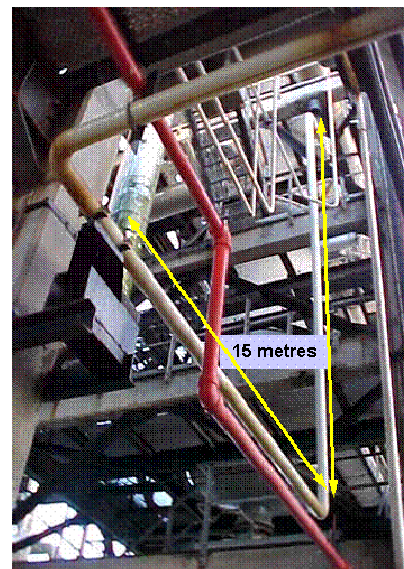
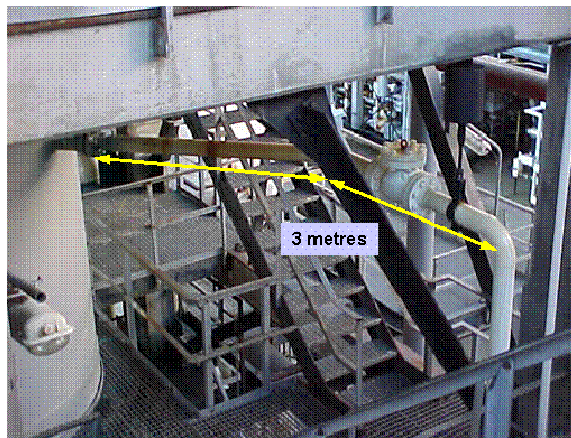
The separator inlet is a 10" NB pipe (actual outside diameter 273mm); the longest span is immediately downstream of the vessel as shown below.



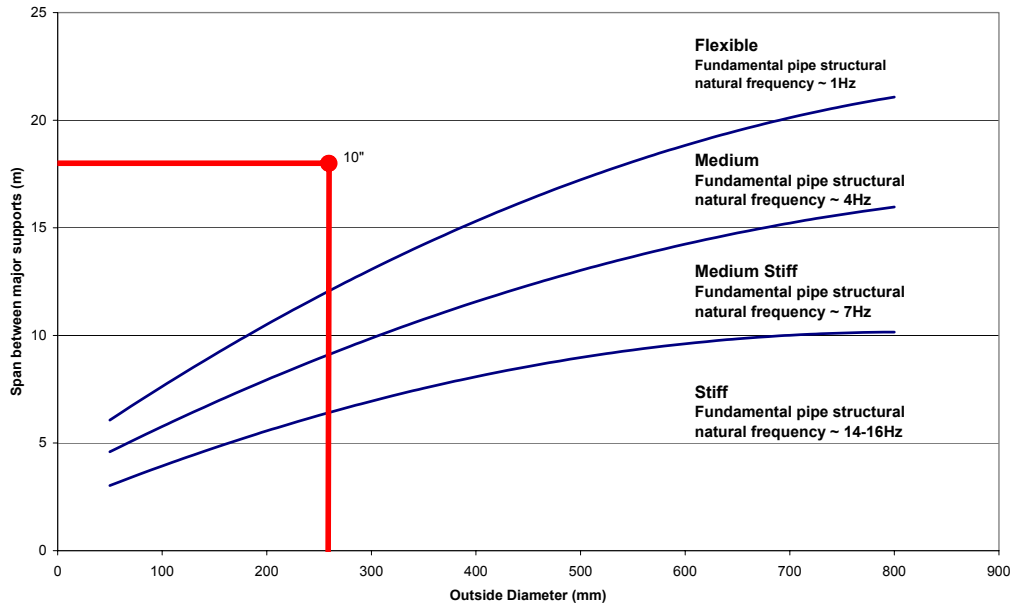
The first support (**Support 1**) is the vessel nozzle, and constitutes a stiff termination point for the pipe. The next support (**Support 2**) is a variable spring hanger with an extended support rod between the spring and the pipe and is therefore not considered an 'effective' support..



The final support (**Support 3**) is a saddle which itself is well supported and can therefore be considered an 'effective' support. The span length is therefore the length between **Support 1** and **Support 3**. This gives a total span length of 18 metres.



This span length can then be used to determine the support classification:



This results in a “Flexible” classification.

Step 4: Determine Flow Induced Vibration Factor F_v (see T2.2.3.4)

F_v is determined from the expressions given in **Table T2-2** for the relevant pipe outside diameter and support arrangement. The results are summarised below.

Pipe diameter (schedule)	20" (sch STD)	10" (sch STD)
α	894571	60647
β	-0.75085	-0.92703
F_v	45175	2636

Step 5: Calculation of Likelihood of Failure (LOF) (see T2.2.3.5)

Finally the LOF for each line is calculated using:

$$\text{Flow Induced Turbulence L.O.F.} = \frac{\rho v^2}{F_v} FVF$$

The results are summarised below.

Stream (Sub system) Pipe dimensions	1 (supply to separator) 20" (sch STD)	4 (produced water) 10" (sch STD)
ρv^2 (kg/m.s ²)	11763	8309
FVF	1	1
F_v	45175	2636
LOF	0.26	3.15

D.4.2 Surge / Momentum Changes (see T2.8)

Surge and momentum changes only apply when there is an automatically actuated valve. In this example the only valve that meets this criterion on process streams 1 and 4 is FCV1002 on the produced water discharge line from the separator.

[**Note:** although the oil system line from the separator also has a flow control valve (FCV1001), this particular line has not been identified from the previous qualitative assessment, as there is no change to the oil flow rate].

The assessment should cover liquid or multiphase valve closing (i.e. when the flow control valve shuts in and has the potential to generate a pressure surge transient). Liquid or multiphase valve opening does not apply as the FCV is not a fast acting valve (such as a relief valve).

The procedure shown in **Flowchart T2-7** should be followed:

Step 1: Determine surge pressure (P_{max}) and maximum force (F_{max})

$$P_{max} = \rho c v \quad \text{where } c = \frac{1}{\sqrt{\rho \left(\frac{1}{K} + \frac{D_{ext}}{1000 T E_{ml}} \right)}}$$

In this case:

ρ (fluid density)	= 1000 kg/m ³
K (fluid bulk modulus)	= 2.19×10^9 N/m ²
D_{ext}	= 273mm
T (main line wall thickness)	= 9.271mm
E_{ml} (Youngs Modulus of pipe material)	= 207×10^9 N/m ²
Therefore $c = 1293$ m/s.	

The maximum fluid velocity (v) was calculated previously as 2.9 m/s (**Section D4.1**).

Therefore $P_{max} = 1000 \times 1293 \times 2.9 = 3749589$ (N/m²).

$$F_{max} = \rho c v \pi \frac{D_{int}^2}{4 \times 10^9} = 190.7 \text{ kN}$$

The upstream length between the valve and the separator (the first large volume) is approximately 2 metres. As $F_{max} > 1 \text{ kN}$ then the next step is to take into account the valve closure time.

Step 2: Effect of valve closure time

The surge pressure is calculated as follows

$$P_{surge} = P_1 \left(\frac{\Omega^2}{2} + \Omega^2 \sqrt{\frac{1}{4} + \frac{1}{\Omega^2}} \right) \quad \text{where} \quad \Omega = \frac{\rho v L_{up} \phi}{P_1}$$

P_1 (static pressure)	= 5.5 barg = 6.5×10^5 N/m ²
v (fluid velocity)	= 2.9 m/s

$$\begin{aligned}
 L_{up} \text{ (upstream line length)} &= 2\text{m} \\
 \rho \text{ (fluid density)} &= 1000 \text{ kg/m}^3 \\
 \text{Valve closure time} &= 5 \text{ seconds} \\
 \text{The valve is a globe valve, and therefore } \phi &= -2.266/T_{close} - 0.32 = -0.7732
 \end{aligned}$$

Therefore $\Omega = -0.0069$

$$\text{and } P_{surge} = 54084 \text{ N/m}^2$$

Which in turn gives an F_{max} of 2.75 kN

Step 3: Limit Force Calculation

The next step is to calculate the limit force using:

$$F_{lim} = (16.8 \times \Psi^3 - 1.81 \times \Psi^2 + 525 \times \Psi + 25.3) \times D_{ext} \times \theta \times \pi \times D_{int}^2 / (4 \times 10^9) \quad (\text{kN})$$

Line	10" sch STD
T (mm)	9.271
T sch 40 (mm)	9.271
Ψ	1
D_{ext} (mm)	273
D_{int} (mm)	254.5
θ (flexible support)	0.5
F_{lim} (kN)	3.92

Step 4: LOF Calculation

Finally, the LOF value is calculated using F_{max} / F_{lim}

Line	10" sch STD
F_{max} (kN)	2.75
F_{lim} (kN)	3.92
LOF	0.70

D.4.3 Summary and Interpretation of Main Line LOF Scores

Stream (Sub system) Pipe dimensions	1 (supply to separator) 20" (sch STD)	4 (produced water) 10" (sch STD)
Flow induced turbulence	0.26	3.15
Surge / momentum	n/a	0.70

Main Line LOF ≥ 1.0

A summary of the actions required for a main line LOF score ≥ 1.0 are given below.

Score	Action	Technical Module
LOF ≥ 1.0	The main line shall be redesigned, resupported or a detailed analysis of the main line shall be conducted, and vibration monitoring of the main line shall be undertaken (Note 1)	TM-09 TM-07/TM-08
	Corrective actions shall be examined and applied as necessary	TM-10
	Small bore connections on the main line shall be assessed.	TM-03
	A visual survey shall be undertaken to check for poor construction and/or geometry and/or support for the main line and/or potential vibration transmission to neighbouring pipework.	TM-05 TM-06

- The produced water outlet line scores an LOF of 3.15 for flow induced turbulence due to the combination of a high value of ρv^2 combined with a flexible support arrangement (i.e. a low fundamental structural natural frequency). Changes to the way the pipe is supported – to increase the fundamental structural natural frequency by reducing the long unsupported span – would be one way of reducing the LOF score. This could potentially be achieved by introducing an intermediate support from the lower of the two horizontal deck beams.

In addition, all small bore connections on the line should also be assessed.

Main Line LOF ≥ 0.5 and < 1.0

A summary of the actions required for a main line LOF score ≥ 0.5 are given below.

Score	Action	Technical Module
1.0 > LOF ≥ 0.5	The main line should be redesigned, resupported or a detailed analysis of the main line should be conducted, or vibration monitoring of the main line should be undertaken (Note 1)	TM-09 TM-07/TM-08
	Corrective actions should be examined and applied as necessary	TM-10
	Small bore connections on the main line shall be assessed.	TM-03
	A visual survey shall be undertaken to check for poor construction and/or geometry and/or support for the main line and/or potential vibration transmission to neighbouring pipework.	TM-05 TM-06

There is one main line that scores an LOF ≥ 0.5:

- The produced water outlet line scores an LOF of 0.70 for pressure surge. In this case any changes to the pipe support arrangement considered for the flow induced turbulence issue (see above) would also be beneficial in terms of reducing the LOF score. A surge analysis (see **Section T9.6**) taking into account the true valve closure characteristics (i.e. valve flow coefficient (C_v) against percentage closure) might also be considered.

Main Line LOF ≥ 0.3 and < 0.5

There are no lines that score an LOF ≥ 0.3 and < 0.5.

Main Line LOF < 0.3

A summary of the actions required for a main line LOF score < 0.3 are given below.

Score	Action	Technical Module
LOF < 0.3	A visual survey should be undertaken to check for poor construction and/or geometry and/or support for the main line and/or potential vibration transmission from other sources.	TM-05 TM-06

For all main lines a walkdown should be conducted to ensure that the as-built arrangement is fit for purpose, and that no changes have been introduced in the period since the original assessment was undertaken, using the guidance given in **TM-06** and **TM-07**.

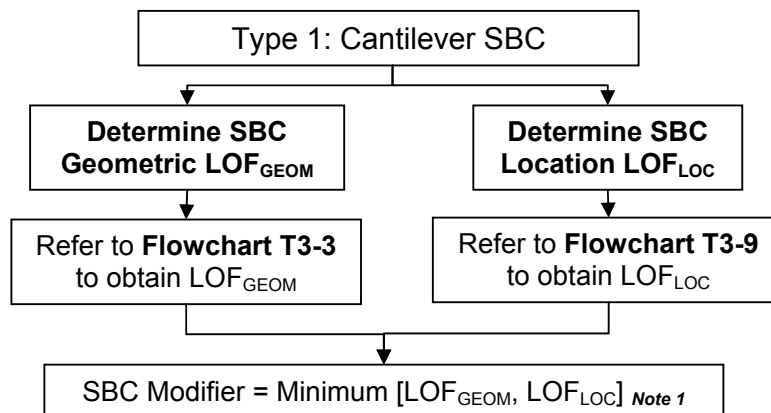
D.5 EXAMPLE D5: SMALL BORE CONNECTION (TYPE 1)

This example is a connection on the compressor discharge line assessed in **Example D2**, which scored a main line LOF of 1.0.



The connection shown is a pressure tapping. There is a single isolation valve, with an instrument line from the flange at the top of the valve. The connection is 1" NB, and the parent pipe schedule is Sch 120. The parent pipe is lagged. The connection is located close to mid span on the parent pipe (i.e. approximately halfway between parent pipe supports).

For a Type 1 SBC **Flowchart T3-2** applies:

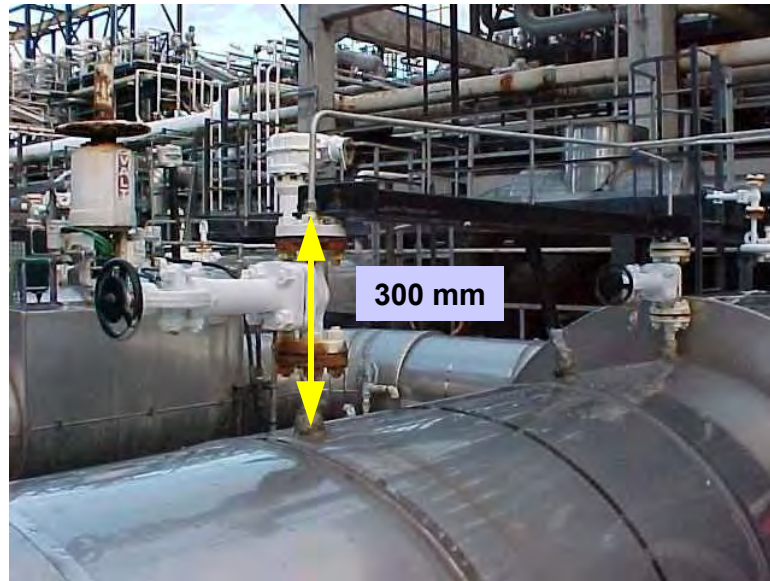


Step 1: Determine LOF_{GEOM} (Flowchart T3-3)

Type of fitting: the lagging also makes it difficult to identify the type of fitting. In this case, the isometric of the parent pipe identified this connection as a weldolet fitting. If no information had been available then the fitting type would have been scored as a 'set on' to provide a conservative assessment.

Overall length of branch: the visible length of the connection is 300 mm. However, an additional length must be included to account for the thickness of the lagging in order to give a 'true' length of the fitting from the wall of the parent pipe to the end of the valve. In this case the lagging is approximately 200 mm deep, so the total length is 500 mm.

Number and size of valves: there is one valve (valve ratings below ANSI 900).



Parent pipe schedule: the parent pipe is Schedule 120. As this is not on the list given in **Flowchart T3-3** then the next lowest 'standard' Schedule is 80 – this is used for the assessment.

SBC minimum diameter: the minimum SBC diameter is 1" NB.

A summary of the scores and the calculated LOF_{GEOM} is given below:

Geometric item	Value	Score
Type of fitting	Weldolet	0.9
Overall length of branch	500mm	0.7
Number & size of valves	1	0.5
Parent pipe schedule	120 (assessed as 80)	0.5
SBC minimum diameter	1" NB	0.7
LOF_{GEOM}		0.66

Step 2: Determine LOF_{LOC} (Flowchart T3-9)

In this case the main line LOF is known and is equal to 1.0. Therefore the LOF_{LOC} defaults to 1.0.

Step 3: Determine SBC Modifier (Flowchart T3-2)

The minimum of the LOF_{GEOM} and LOF_{LOC} scores is taken. In this case this results in a SBC Modifier score of 0.66.

Step 4: Determine SBC LOF (Flowchart 3-4)

The main line LOF is multiplied by 1.42. In this case this results in $1.0 \times 1.42 = 1.42$.

The minimum of this value (1.42) and the SBC Modifier (0.66) is then obtained to give the SBC LOF (0.66).

Score	Action	Technical Module
LOF ≥ 0.7	The SBC shall be redesigned, resupported or a detailed analysis shall be conducted, and vibration monitoring of the SBC shall be undertaken A visual survey shall be undertaken to check for poor construction and/or geometry for the SBC's and instrument tubing.	TM-11 TM-07/TM-08 TM-05/TM-06

Note: in this example the main line assessment (which is detailed in Example D2) has been identified as being associated with tonal excitation from a dead leg branch (the recycle line) and also from mechanical excitation from the compressor. If the excitation frequencies are known then the structural natural frequencies of the SBC should also be determined by specialist measurement or predictive techniques (see **Chapter 3, Section 3.3.3**).

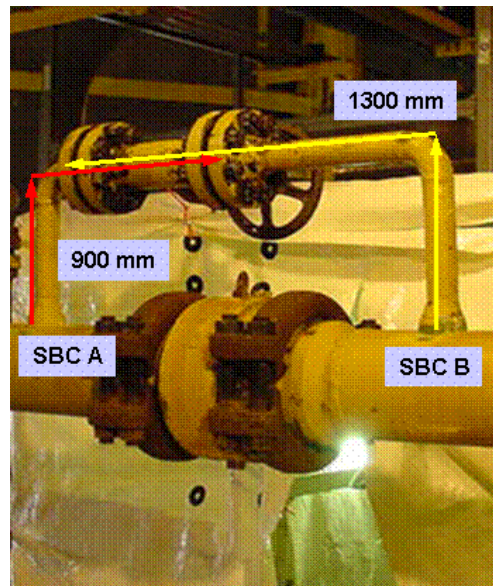
D.6 EXAMPLE D6: SMALL BORE CONNECTION (TYPE 2)

This example is a bypass which exits and enters the same main line.



There is a single valve and the connection is 2" NB, and the parent pipe schedule is Sch 40. The connection is located just downstream of a 90 degree bend in the parent pipe. It is assumed for the case of this example that the mainline LOF has previously been assessed with an LOF of 0.49.

For a Type 2 SBC **Flowchart T3-4** applies. In this case the connection is divided into two (see **C.1.11**) as shown below, each with different total lengths.



Step 1: Determine LOF_{GEOM} (Flowchart T3-4)

Type of fitting: Both connections (A and B) are weldolet fittings.

Overall length of branch: for SBC A the length of the connection is 900 mm. The length of SBC B is 1300mm.

Number and size of valves: for both SBC A and B there is one valve (valve ratings below ANSI 900).

Parent pipe schedule: the parent pipe is Schedule 40.

SBC minimum diameter: the minimum SBC diameter is 2" NB.

Using the method given in **Flowchart T3.3** the following values for $LOF_{GEOM(A)}$ and $LOF_{GEOM(B)}$ are given below:

Geometric item	SBC A		SBC B	
	Value	Score	Value	Score
Type of fitting	Weldolet	0.9	Weldolet	0.9
Overall length of branch	>600mm	0.9	>600mm	0.9
Number & size of valves	1	0.5	1	0.5
Parent pipe schedule	40	0.7	40	0.7
SBC minimum diameter	2" NB	0.5	2" NB	0.5
LOF_{GEOM}		0.7		0.7

The final LOF_{GEOM} applied to the complete connection is the maximum of the LOF_{GEOM} score for SBC A and B which in this case is 0.7.

Step 2: Determine LOF_{LOC} (Flowchart T3-9)

In this case the main line LOF is known and is equal to 0.49. The connection is just downstream of a bend and the parent pipe schedule is 40. This gives values of 0.9 and 0.7 respectively, giving an LOF_{LOC} of 0.8.

Step 3: Determine SBC Modifier (Flowchart T3-4)

The minimum of the LOF_{GEOM} and LOF_{LOC} scores is taken. In this case this results in a SBC Modifier score of 0.7.

Step 4: Determine SBC LOF (Flowchart 3-4)

The main line LOF is multiplied by 1.42. In this case this results in $0.49 \times 1.42 = 0.696$.

The minimum of this value (0.696) and the SBC Modifier (0.7) is then obtained to give the SBC LOF (0.696).

Score	Action	Technical Module
LOF $\geq$ 0.7	The SBC shall be redesigned, resupported or a detailed analysis shall be conducted, and vibration monitoring of the SBC shall be undertaken A visual survey shall be undertaken to check for poor construction and/or geometry for the SBC's and instrument tubing.	TM-11 TM-07/TM-08 TM-05/TM-06
0.7 > LOF $\geq$ 0.4	Vibration monitoring of the SBC should be undertaken. Alternatively the SBC may be redesigned, resupported or a detailed analysis conducted. A visual survey should be undertaken to check for poor construction and/or geometry for the SBC's and instrument tubing.	TM-07/TM-08 TM-11 TM-05/TM-06

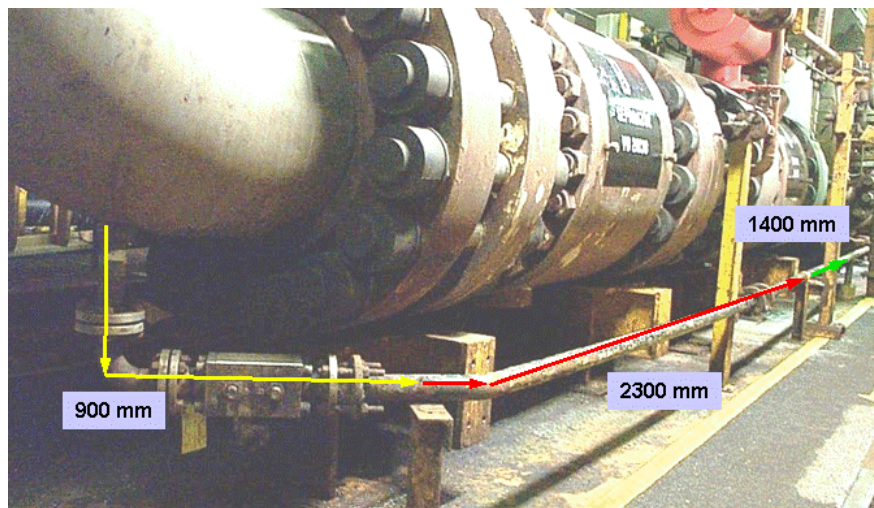
The final result is borderline (i.e. just below 0.7). Consideration should therefore be given to applying some form of modification – in this case bracing back to the main line is a practical option. See **TM-11** for potential options.

D.7 EXAMPLE D7: SMALL BORE CONNECTION (TYPE 3)

This example is a long chemical injection line.



There is a single valve between the parent pipe and the first resting support on the connection. This first resting support is 900mm from the connection to the parent pipe. The span length to the next support on the connection is approximately 2300mm. The connection is 1.5" NB, and the parent pipe schedule is Sch 120. The connection is located just downstream of a 90 degree bend in the parent pipe and close to a fixed anchor on the parent pipe. It is assumed for the case of this example that the mainline LOF is unknown.



For a Type 3 SBC **Flowchart T3-5** applies. In this case the SBC modifier must be obtained for (i) the first span and (ii) the subsequent spans.

First Span**Step 1: Determine LOF_{GEOM} (Flowchart T3-6)**

$LOF_{GEOM(C)}$ is obtained as follows:

As there is a mass associated with the first span $LOF_{GEOM(C)}$ is obtained using **Flowchart T3-3** as follows:

Type of fitting: The connection to the parent pipe is a weldolet fitting.

Overall length of branch: the length of the connection to the end of the valve is approximately 700mm.

Number and size of valves: there is one valve (valve ratings below ANSI 900).

Parent pipe schedule: the parent pipe is Schedule 120.

SBC minimum diameter: the minimum SBC diameter is 1.5" NB.

Using the method given in **Flowchart T3-3** the following value for $LOF_{GEOM(C)}$ is given below:

Geometric item	Value	Score
Type of fitting	Weldolet	0.9
Overall length of branch	700mm	0.9
Number & size of valves	1	0.5
Parent pipe schedule	120 (assessed as 80)	0.5
SBC minimum diameter	1.5" NB	0.6
$LOF_{GEOM(C)}$		0.68

$LOF_{GEOM(D)}$ is obtained as follows:

The first span length (900mm to the first support) is divided by the fitting span factor from **Table 3-1**. For a weldolet fitting the fitting span factor is 0.7; the span length divided by the fitting span factor is therefore $900\text{mm}/0.7 = 1286\text{mm}$.

This value is then assessed against the criteria given in **Figure T3-1**. For the combination of modified span length (1286mm) and connection diameter (1.5") this results in an $LOF_{GEOM(D)}$ of 0.2.

$LOF_{GEOM(E)}$ is obtained as follows:

This time the first span length (900mm to the first support) is multiplied by the fitting span factor from **Table 3-1**. For a weldolet fitting the fitting span factor is 0.7; the span length multiplied by the fitting span factor is therefore $900\text{mm} \times 0.7 = 630\text{mm}$.

The minimum allowable span length from **Table 3-2** is 1.3m for a 1.5" NB connection. This is greater than 630mm and therefore there is a potential issue with respect to the connection being supported from the deck too close to the parent pipe. The $LOF_{GEOM(E)}$ is therefore set to 0.7.

The LOF_{GEOM} is then set to the maximum of $LOF_{GEOM(C)}$, $LOF_{GEOM(D)}$ and $LOF_{GEOM(E)}$, which results in an LOF_{GEOM} of 0.7.

Step 2: Determine LOF_{LOC} (Flowchart T3-9)

In this case the main line LOF is not known and therefore the LOF_{LOC} is set to 1.0.

Step 3: Determine SBC Modifier – first span (Flowchart T3-6)

The minimum of the LOF_{GEOM} and LOF_{LOC} scores is taken. In this case this results in a SBC Modifier score of 0.7.

Subsequent Spans**Step 1: Determine LOF_{GEOM} (Flowchart T3-7)**

$LOF_{GEOM(F)}$ is obtained as follows:

The maximum span length associated with a mass is 2300mm. For the combination of span length (2300mm) and connection diameter (1.5”) this results in an $LOF_{GEOM(F)}$ of 0.6 from **Figure T3-3**.

$LOF_{GEOM(G)}$ is obtained as follows:

The maximum span length without a mass is 1400mm. For the combination of span length (1400mm) and connection diameter (1.5”) this results in an $LOF_{GEOM(G)}$ of 0.2 from **Figure T3-4**.

The LOF_{GEOM} is then set to the maximum of $LOF_{GEOM(F)}$ and $LOF_{GEOM(G)}$, which results in an LOF_{GEOM} of 0.6.

Step 2: Determine LOF_{LOC} (Flowchart T3-7)

LOF_{LOC} is set to 1.0.

Step 3: Determine SBC Modifier – subsequent spans (Flowchart T3-6)

The minimum of the LOF_{GEOM} and LOF_{LOC} scores is taken. In this case this results in a SBC Modifier score of 0.6.

Overall Connection SBC Modifier**Step 1: Determine SBC Modifier (Flowchart T3-5)**

The overall SBC Modifier is the maximum of the SBC Modifier [first span] (0.7) and the SBC Modifier [subsequent spans] (0.6). This results in a final overall SBC Modifier for this connection of 0.7.

Step 2: Determine SBC LOF (Flowchart 3-4)

The main line LOF is multiplied by 1.42. In this case this results in $1.0 \times 1.42 = 1.42$.

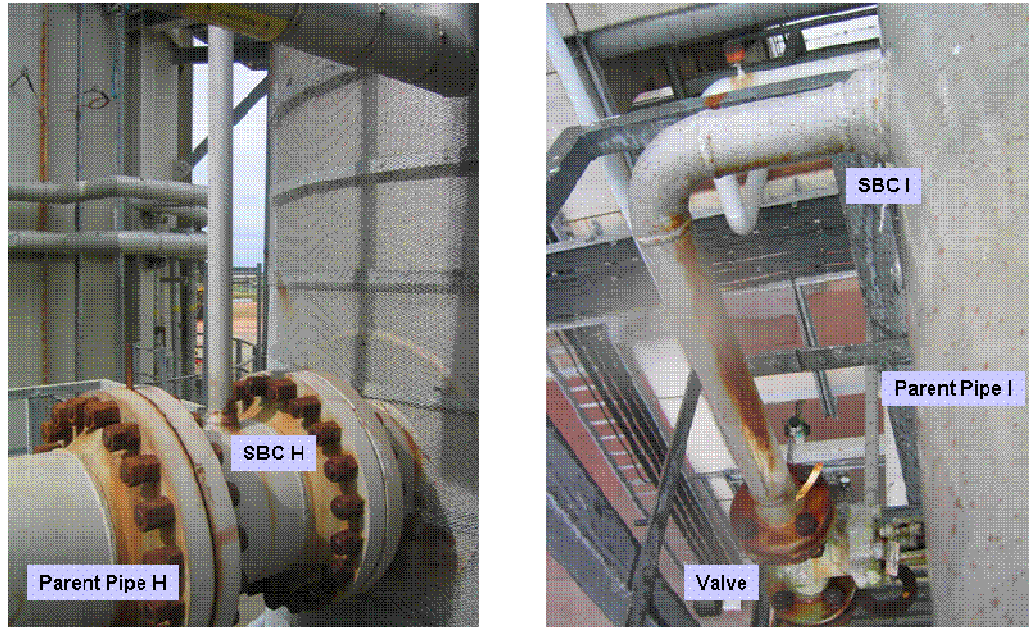
The minimum of this value (1.42) and the SBC Modifier (0.7) is then obtained to give the SBC LOF (0.7).

Score	Action	Technical Module
LOF $\geq$ 0.7	The SBC shall be redesigned, resupported or a detailed analysis shall be conducted, and vibration monitoring of the SBC shall be undertaken A visual survey shall be undertaken to check for poor construction and/or geometry for the SBC's and instrument tubing.	TM-11 TM-07/TM-08 TM-05/TM-06

The final result indicates that some form of modification is required. From the assessment process the main issue which gives rise to the high score is that the first support to the deck is too close to the parent pipe and therefore this support needs to be relocated. Note that the assessment is based on a main line LOF = 1.0 as no main line assessment has been made, and will be conservative.

D.8 EXAMPLE D8: SMALL BORE CONNECTION (TYPE 4)

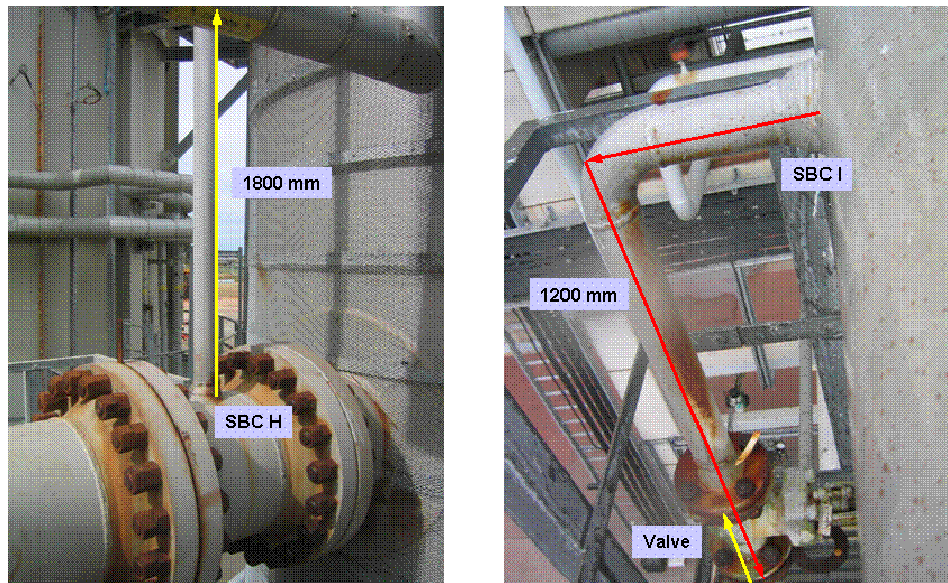
This example is a 2" NB connection which spans between two separate parent pipes. There is no intermediate support on the connection, but there is a valve which isolates one parent main line from the other.



The left hand picture shows the connection from SBC H rising vertically from its parent pipe. The 2" line is then connected to an isolation valve (right hand picture) before turning through 90 degrees and connecting to the other parent pipe at SBC I.

The parent pipe schedule in both cases is Sch 160. SBCs H and I are both located close to fixed supports on their respective parent pipes. It is assumed for the case of this example that the main line LOF has previously been assessed with an LOF of 0.3. The fittings type in both cases is a weldolet.

For a Type 4 SBC **Flowchart T3-8** applies. In this case the connection is divided into two (see C.1.11) as shown below, each with different total lengths. The conservative approach is to assume that although the valve is located closer to SBC H than SBC I both SBCs ‘see’ the valve.



Step 1: Determine LOF_{GEOM} (Flowchart T3-8)

$LOF_{GEOM(H)}$ is obtained using **Flowchart T3-3** as follows:

Type of fitting: The connection to the parent pipe is a weldolet fitting.

Overall length of branch: the length of the connection to the end of the valve is approximately 1800mm.

Number and size of valves: there is one valve.

Parent pipe schedule: the parent pipe is Schedule 160.

SBC minimum diameter: the minimum SBC diameter is 2" NB.

Using the method given in **Flowchart T3.3** the following value for $LOF_{GEOM(H)}$ is given below:

Geometric item	Value	Score
Type of fitting	Weldolet	0.9
Overall length of branch	1800mm	0.9
Number & size of valves	1	0.5
Parent pipe schedule	160	0.3
SBC minimum diameter	2" NB	0.5
$LOF_{GEOM(H)}$		0.62

$LOF_{GEOM(I)}$ is obtained using **Flowchart T3-3** as follows:

Type of fitting: The connection to the parent pipe is a weldolet fitting.

Overall length of branch: the length of the connection to the end of the valve is approximately 1200mm.

Number and size of valves: there is one valve (valve ratings below ANSI 900).

Parent pipe schedule: the parent pipe is Schedule 160.

SBC minimum diameter: the minimum SBC diameter is 2" NB.

Using the method given in **Flowchart T3.3** the following value for $LOF_{GEOM(I)}$ is given below:

Geometric item	Value	Score
Type of fitting	Weldolet	0.9
Overall length of branch	1200mm	0.9
Number & size of valves	1	0.5
Parent pipe schedule	160	0.3
SBC minimum diameter	2" NB	0.5
$LOF_{GEOM(I)}$		0.62

$LOF_{GEOM(J)}$ is obtained as follows:

The overall span length ($1800 + 1200 \text{ mm} = 3000\text{mm}$) is divided by the fitting span factor from **Table 3-1**. For a weldolet fitting the fitting span factor is 0.7; the span length divided by the fitting span factor is therefore $3000\text{mm}/0.7 = 4286\text{mm}$.

This value is then assessed against the criteria given in **Figure T3-1**. For the combination of modified span length (4286mm) and connection diameter (2") this results in an $LOF_{GEOM(J)}$ of 0.6.

$LOF_{GEOM(K)}$ is obtained as follows:

This time the overall span length (3000mm) is multiplied by the fitting span factor from **Table 3-1**. For a weldolet fitting the fitting span factor is 0.7; the span length multiplied by the fitting span factor is therefore $3000\text{mm} \times 0.7 = 2100\text{mm}$.

The minimum allowable span length from **Table 3-3** is 2m for a 2" NB connection. This is less than 2100mm and therefore there are no potential issues with respect to the connection being too short to accommodate relative movement between the two parent pipes. The $LOF_{GEOM(K)}$ is therefore set to 0.2.

The LOF_{GEOM} is then set to the maximum of $LOF_{GEOM(H)}$, $LOF_{GEOM(I)}$, $LOF_{GEOM(J)}$ and $LOF_{GEOM(K)}$, which results in an LOF_{GEOM} of 0.62.

Step 2: Determine LOF_{LOC} (Flowchart T3-9)

In this case the main line LOF is 0.3. Both connections are near fixed supports on the parent pipe(s) and the parent pipe schedule is 160. This gives values of 0.1 and 0.3 respectively, giving an LOF_{LOC} of 0.2 for both SBC H and SBC I.

Note: if the two different SBCs had scored different values of LOF_{LOC} then the maximum of the two scores would be taken.

Step 3: Determine SBC Modifier (Flowchart T3-8)

The minimum of the LOF_{GEOM} and LOF_{LOC} scores is taken. In this case this results in a SBC Modifier score of 0.2.

Step 4: Determine SBC LOF (Flowchart 3-4)

The main line LOF is multiplied by 1.42. In this case this results in $0.3 \times 1.42 = 0.426$.

The minimum of this value (0.426) and the SBC Modifier (0.2) is then obtained to give the SBC LOF (0.2).

Score	Action	Technical Module
LOF < 0.4	A visual survey should be undertaken to check for poor construction and/or geometry for the SBC's and instrument tubing.	TM-05/TM-06

A walkdown should be conducted of the SBC to ensure that the as-built arrangement is fit for purpose, using the guidance given in **TM-06** and **TM-07**.

Appendix E

TERMS

Term	Description
API	American Petroleum Institute
Broadband	Energy is input over a wide frequency range
Choked Flow	Choked flow occurs when the gas velocity is the same as the speed of a pressure wave through the fluid and the maximum mass flow rate is achieved.
Dynamic viscosity	Ratio of shear stress to the associated strain rate
Forced vibration	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.
HAZID	Hazard Identification
HAZOP	Hazard and Operability
KE	Kinetic Energy
LOF	Likelihood of failure
Mode shape	The relative displacement of all points on a vibrating structure at a given natural frequency.
Natural Frequency	The frequency of free vibration of a system.
Node	A point or line on a vibrating structure that remains stationary.
PWHT	Post weld heat treatment
Ratio of specific heats (C_p/C_v)	The ratio of molar heat capacity at constant pressure to molar heat capacity at constant volume.
Resonance	The resonant frequency of a system is defined as the natural frequency for which the response of the system is a maximum. If the excitation frequency is either increased or decreased the amplitude of response will decrease.
RMS	Root mean squared
SBC	Small Bore Connection
SI	International System of Units
S-N diagram	A plot showing the relationship of stress, S, and the number of cycles, N, before fracture in fatigue testing.

APPENDIX E - TERMS

Term	Description
Stress-intensity factor	A factor, to describe the intensification of applied stress at the tip of a crack.
Tonal	Energy is only input at discrete frequencies
Vapour pressure	The partial pressure of a gas in equilibrium with a condensed form of the same substance.
Vena contractor	The point at which the minimum cross-sectional area of the flow stream occurs

Appendix F

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