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Vibration Troubleshooting  
of  
Existing Piping Systems

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# Vibration Troubleshooting of Existing Piping Systems

## 1. Overall Approach

When a vibration problem is reported, the vibration analyst/engineer must determine whether or not the high vibrations represent a problem. The evaluation of whether or not the high vibrations represent a problem has to be based primarily on the vibratory stresses introduced into the piping, however, the vibration levels can be high based on the psychological effect that it has on the operators. In addition, many times apparently high vibrations may not cause excessive stresses in the piping, but could cause excessive stresses to piping systems or appurtenances that are attached to the vibrating pipe.

The vibrations are obviously too high if there has been a piping fatigue failure or if the piping has extremely high vibrations. Since it requires approximately ten million cycles of stresses in excess of the endurance limit for a pipe to fail, it is desirable to correct high vibrations before failures occur, if at all possible. As a rule of thumb, excessive vibrations at 5 Hz require 20 days to reach ten million cycles, therefore, for low frequency vibrations failures may not occur for a month. If the vibrations are at 100 Hz, it would only require one day of operation at the excessive levels. This means that extreme care should be exercised in evaluating piping systems in low speed reciprocating machinery systems or any system which has low frequency vibrations.

In addition, vibrations are sensitive to engine speed and loading conditions, therefore, the vibrations that are present during the initial survey may not be the highest that will occur. The point of maximum vibration is sometimes difficult to establish in an initial survey and some safety factor is necessary to allow for these factors.

If the high vibration levels have been occurring for many months without failures, the vibrations may not be so severe that failure is imminent but may need to be reduced to increase the margin of safety.

This paper will discuss the types of problems that occur in typical piping systems and will try to direct the analyst to determine the cause of the problems and give input into possible solutions. A set of diagnostic charts has been developed to aid in the assessment of the severity of the vibrations and to help evaluate possible solutions. The following steps will help the vibration analyst/engineer diagnose and solve the problem.

1. *Walkdown/Survey the system.* Physically examine the piping system to determine the severity of the vibration problem. Note the locations where the vibrations are the maximum since these will be proper points to make vibration measurements for use with the

screening criteria.

2. *Review the operating history and vibration data.* Try to correlate operating conditions to the incident of high vibrations or failures. Analyze any vibration data for frequency content. Review any background design analyses reports which may give clues as to the excitation sources in the system. As an example, many compressor and pump piping installations are studied in the design stage to predict the pulsation amplitudes and shaking forces. These design studies are called acoustical pulsation analyses and are performed on an analog or by a digital computer analysis. These reports could give clues as to the potential sources of excitation.
3. *Measure the vibrations and determine its characteristics.* Try to characterize the types of vibration as to whether they are:
  - (a) low frequency modes of piping or compressor manifold systems
  - (b) high frequency modes of piping, such as shell wall resonances excited by high pressure drop across valves, or acoustically excited by high flow rates, blade passing frequency pulsations, etc.
  - (c) transient phenomenon, such as surge, waterhammer, cavitation, slug flow, etc.
4. *Compare to appropriate screening criteria.* The severity of the measured vibrations can be evaluated by comparing the amplitudes to the appropriate screening criteria presented in this paper.
5. *Determine the source/cause.* An important part of the troubleshooting procedure is to determine if the excessive piping vibrations are a result of a mechanical or acoustic resonance (or both). A variety of measurement procedures can aid in defining the cause of the problem.
6. *Evaluate possible solutions.* Once the cause of the problem is defined, the solution is a matter of developing a practical modification which will eliminate the cause or reduce its effects. Treatment of vibration and pulsation problems are discussed later.

In order to assist the analyst in the troubleshooting process, a series of Diagnostic Charts have been developed. Table 1 gives a summary of the charts and their functions. The Diagnostic Charts given in Tables 2 to 5 define step by step procedures to guide the analyst/engineer through a logical approach in troubleshooting a piping vibration problem. The charts suggest particular data to be taken, specific questions to be asked, and gives the path to be taken, depending upon the type of vibration problem. By following the steps, along with the simplified analyses presented in the text, the analyst/engineer should be able to define the probable sources which cause the vibration problem. Modifications to detune mechanical and acoustical resonances and correct piping vibration problems are discussed later in this paper.

## 2. Walkdown/Survey The System

The first step in the determination of the severity of the vibration problem is to make an initial survey or walkdown of the piping system. The purpose of the walkdown is to determine if the reported high vibrations are a real problem or not. During the walkdown, watch for the following common symptoms of piping vibrations which indicate potential problems.

1. failed piping with fatigue cracks
2. high visual vibrations/line movements
3. damaged or ineffective supports and restraints
4. high vibration of connected appurtenances
5. high impact or flow excited noise

In addition to the above symptoms, the analyst/engineer should make the following observations regarding the character of the vibrations:

1. frequency of vibration (low frequency less than 300 Hz or high frequency)
2. amplitude of vibration
3. location of highest vibration
4. mode shape or vibration pattern
5. steady state, transient or random vibrations

### 2.1 Failed Piping with Fatigue Cracks

If a pipe fails and the failure surface has classical fatigue markings, an obvious vibration problem exists. Fatigue failures typically have beach marks, which are centered around a common point that corresponds to the fatigue-crack origin. Also called clamshell, conchoidal and arrest marks, beach marks are perhaps the most important characteristic feature in identifying fatigue failures. Beach marks can occur as a result of changes in loading or frequency or by oxidation of the fracture surface during periods of crack arrest from intermittent service of the part or component. Figure 1 gives the characteristics of typical fatigue failures for a variety of loadings [2]. As indicated, fatigue failures occur most often at points of high stress concentration, such as at branch connections and welded joints. Other likely points would be near supports or restraints.

If failures occur repeatedly near the same location in the piping system, this is normally an indication that there is a mechanical or acoustical resonance, or both. If there are no resonances present there must be an excessive excitation force in the system.

The fatigue failure surface should be carefully examined since the direction of the crack can give clues to the mode shape of the vibrations. The location and orientation of the failure should be fully documented with photographs or sketches so that if it is necessary to perform tests, strain gages can be placed in the appropriate locations.

For example, if the fatigue failure occurred as a circumferential crack, the failure was probably caused by a bending mode. If the fatigue crack occurred on a 45 degree spiral, the failure may have been caused by a torsional mode, such as the low mode of vibration of the compressor manifold system. Cracks in the longitudinal direction could be a result of a high frequency shell resonance.

If the failure occurred at the piping reinforcement weld, the quality of the weld may be at fault. The location of a fatigue crack along the weld line usually indicates a high stress concentration factor. The stress concentration factor for a good weld can be as low as 1.6; however, for a poor weld, the stress concentration factor can be 5 or greater. Other locations of high stress concentration include all weld joints, restraints, hangers, and other geometrical discontinuities, such as changes in piping wall thicknesses.

Since the function of the piping system is to contain the fluid, bolt failures which cause a leak in flanges, or at valves, etc. are considered to constitute a failure of the piping system. Therefore, check the bolted joints in the piping system to ensure that they are not loose and have no failures. This is very critical on high pressure let-down valves which may have sonic flow. Often the high level turbulence will excite the lateral and axial modes of the piping and cause bolt failures.

## 2.2 Visual Vibrations/Line Movements

During the initial survey, look for parts of the piping system that are obviously vibrating excessively. Tests have shown that, with training, a person can judge the approximate amplitude and can distinguish low frequency from high frequency vibrations, usually those less than 50 Hz. This judgement is enhanced when the sense of feeling is used, such as feeling the vibrations with the fingers or by a coin held in the fingers. It should be noted that extreme caution should be observed before touching a pipe with your finger since it could be at an elevated temperature and severe burns could occur. If the piping is experiencing high vibrations or movements, then the approximate location of the maximum vibrations should be noted so that vibration measurements can be made at that location when detailed testing is performed. Many times individual components, such as gage lines, pressure gages, thermowells, instrument lines, etc., will vibrate and result in failures. Particular attention should therefore be paid to these small connections and other appurtenances during this initial survey.

In liquid piping systems, a sudden movement of the pipe may indicate waterhammer. This can be caused by sudden opening or closing of a valve which may produce a pressure wave which moves down the pipe. As it reaches elbows or other changes in direction or cross section, a large low frequency force can shock the pipe and cause large motion.

Careful attention should be paid to determine if the vibrations are steady state or transient

in nature. Also, note whether the vibrations may have a varying amplitude or "beating" which is typical of a piping vibration in which excitation is occurring from two sources at slightly different frequencies.

### 2.3 Damaged or Ineffective Supports and Restraints

During the initial survey, careful attention should be paid to the condition of supports and restraints, and particularly the bolts. Loose restraints, supports, clamps, or bolts that have been bent or broken can change the low frequency resonant frequencies of piping spans and can be the cause of a vibration problem by causing the mechanical natural to occur at frequencies of high excitation. Loose or broken supports/restraints can be a result of resonant vibrations, high dynamic shaking forces, inadequate design, or excessive thermal loads. If the support does not have sufficient stiffness to control the static and dynamic loads, it may bend due to the large forces and movements. For example, U-bolt clamps have very little stiffness and are ineffective restraints and in piping systems with large shaking forces, they will often fail first. Thermal pipe guides should be investigated since they are designed to keep the pipe in position, but may not provide adequate vibration control. Long unsupported spans with low support stiffness could result in low mechanical natural frequencies. Piping spans with unsupported block valves or other heavy masses can also have low response frequencies.

Note where the maximum vibrations occur and determine the nearest effective support location. The lowest lateral natural frequency of a span between supports is a function of the square of the span length. The stiffness of a piping span is a function of the span length between supports cubed. Moving the supports closer together will shift the natural frequency and increase the stiffness and may significantly reduce the vibrations for those systems with low frequencies of less than 300 Hz. Determine possible locations for restraints which could be added to detune the resonant frequency.

In systems which experience transient vibrations, the snubbers may have to be checked to ensure that they are behaving in the proper manner. This is particularly important if large displacements at the snubbers and supports have been experienced.

### 2.4 High Vibration of Connected Appurtenances

Many of the failures that occur in piping systems are in the smaller diameter piping connected to larger piping which has vibrations. Vents, drains, pressure gage lines, instrument lines, thermowells, etc. are most sensitive to this kind of failures. The vibrations of the appurtenance are caused by a base excitation whereby direct mechanical coupling between the vibrating pipe or machinery causes the side branch to vibrate at its mechanical natural frequency. Therefore all appurtenances near points of high vibrations should be examined. The large pipe can be excited by machinery unbalance or by pulsations. Examples of the mechanical coupling between a pump and its piping which caused vent and drain valve failures are discussed in References [3,4].



Failures of appurtenances can also occur in high energy fluid flow systems near valves with high pressure drops and especially if the flow is sonic. The high energy will excite the low frequency natural frequencies, and will also excite the shell wall resonances in the axial and circumferential directions on the large and small diameter pipes. These shell wall resonances can excite excessive stresses at points of high stress concentration [5].

If high vibrations occur at a particular location, note where the closest point of coupling occurs. Points of acoustic coupling include the closed end of headers, pulsation bottles, restrictions in the piping such as partially closed valves, orifices, piping elbows, and any changes in piping cross section. A branch pipe from the suction and discharge lines to a closed bypass valve may be the point of coupling for machinery piping systems. Blind ends of headers, tees, and manifolds most often serve as points of acoustic coupling.

## 2.5 High Impact or Flow Excited Noise

During the initial walkdown survey, listen for impact noises that may be created by the piping vibrating and impacting loose clamps and supports. The type of noise created due to the vibration will give clues as to the nature of the vibrations. If the noise is a pure tone with a constant amplitude, the vibrations will most likely be steady state and caused by a constant amplitude shaking force at or near a mechanical natural frequency. If the noise is broadband and varies in amplitude, or is intermittent, the vibrations may be caused by high flow excitation or a result of some process transient.

In piping systems with high flow velocities, such as systems with pressure letdown valves, bypass lines, flare lines or fluid transfer lines, the major indication of a potential vibration problem is the presence of high noise levels. The high energy broadband noise created by a large pressure drop across a letdown valve may be amplified by mechanical shell wall resonances. If acoustical resonant natural frequencies are present in the range of the broadband energy, they will be excited and the noise spectra will be influenced by these resonances [5].

In addition to the broadband energy created by the flow, the high velocity flow past a closed branch can generate turbulence and vortices which excite an acoustical resonance frequency in the side branch. In centrifugal equipment piping systems, the noise may be caused by an acoustical resonance in the compressor case or internals or be amplified by an acoustical resonance of the piping system.

It is especially important to determine whether the noise is predominantly a pure tone or broad band frequency. Pure tones are usually associated with acoustical or mechanical resonances, whereas broad band noise is generally indicative of high energy flow velocity excitation.

After it is decided that the vibrations are high and additional analysis is needed to determine the characteristics, cause, and solution to the problem, it is necessary to develop a test plan for measuring the vibrations. To ensure that the test includes the conditions that were present during the high vibration event, it is necessary to study the operating conditions and the past data.

### 3. Review Operating History

Once it is determined that the vibrations are a definite problem, and a general feeling of the severity, frequency range, and patterns of vibration are noted, a preliminary vibration analysis test plan needs to be developed. Prior to testing, it is desirable to define the test parameters which may have a strong influence on the vibrations so that the vibration testing will lead to the exact excitation mechanism. One step that has been helpful in defining the test plan is to review the prior vibration data and the operation history of the unit. The following items should be investigated to determine possible influencing parameters.

1. Review the past vibration data, if available. Carefully note the operating condition at the time of the failures. This includes speed, pressures, temperatures, gas composition and loading. If a chronic problem exists, check for a possible correlation between operating variables.  
Review frequency analysis data previously obtained on the vibrations or pulsations. Similar data could be taken for a direct comparison to determine if the problem has worsened.
2. If the unit has run successfully for a significant length of time, try to determine the most significant changes from the previous logged data.
3. Review the normal speed and operating conditions for the recent past. Small changes in speed can have a strong effect when acoustical or mechanical resonances are involved.
4. Check the normal loading since changing the loading on a reciprocating compressor from double-acting to single-acting by the use of unloaders can cause the odd harmonics to increase, which could excite resonances near the odd harmonics. In some instances, a lightly loaded compressor may cause more severe piping vibration and stress problems than a heavily loaded compressor.
5. Review the acoustical analysis of the compressor piping system if one was performed. The pulsation analysis may give expected amplitudes of pulsations and unbalanced shaking forces and their frequencies in the vicinity of the high vibrations. The anticipated acoustical resonances in the piping should be determined by noting the change in pressure pulsations as the operating speed is changed.
6. Review the pump starts and stops that may have caused transient vibrations due to a waterhammer type of excitation. Rapid valve closure of the pump block valve (if motor operated) can cause this type of transient. Measure the valve closing time to compare with the design value for the system.

#### 4. Preliminary Vibration Measurements

After the prior operating history has been studied and a test plan developed, the dynamic vibration characteristics need to be measured to provide sufficient information to define the cause of the problem. Measurements of vibration amplitudes with portable hand-held instruments may be sufficient for some problems; others may require modal testing with amplitude and phase data to accurately determine mode shapes. Measurement techniques and guidelines for instrumentation to use for various vibration problems are discussed. Typical piping vibration mode shapes are illustrated in Figure 2.

In general, the important factors which define the characteristics of dynamic movement are similar for vibration, pulsation, stress or noise. The measurements should define the following factors:

1. peak to peak amplitude
2. major frequency components
3. points of maximum amplitude
4. mode shape for each major frequency component

Pressure pulsation and noise measurements can be compared to applicable codes, such as API-618 Reciprocating Compressors and the Occupational Safety and Health Administration (OSHA) codes [4,5] that relate to allowable sound pressure levels and exposures. However, vibration and stress amplitudes are more directly related to failure risk. The following section presents screening criteria for vibration and stress to aid the analyst/engineer in making his judgement regarding whether the vibration is acceptable or if modifications are required.

## 5. Identify Vibration Characteristics

Based on the vibration measurements, it is important to identify the vibration characteristics. For ease of discussion, vibration of piping can be separated into three categories as presented in Table 2:

- Low frequency vibrations (less than 300 Hz) which are characteristic of overall movement of the piping in lateral beam modes or rigid body modes of the compressor manifold system. Low frequency vibrations may involve the resonance of one or more piping spans and are usually associated with low speed pumps and reciprocating compressors.
- High frequency vibrations are more commonly caused by piping systems with high flow rates (near restrictions) and centrifugal equipment. The vibrations usually involve piping shell wall resonances, or may involve individual components, such as a vent, drain or instrumentation lines and valves, or bolts and attachments to the valves, etc. Observation of this type of vibration in insulated piping can be difficult because the component with the largest vibration may be covered by insulation. It may be necessary to remove sections of the insulation downstream of a letdown valve so that the noise and vibrations can be measured.
- Transient events such as surge, waterhammer, cavitation, slug flow, sudden valve opening or closing, etc. which occur over relatively short periods of time. These types of vibrations typically have to be measured in the time domain.

## 6. Appropriate Screening Criteria

Whenever a vibration occurs, it must be determined whether the vibrations are excessive and whether the unit must be shut down immediately to ensure the safety of the installation. With the high cost of downtime, it is imperative that the analyst/engineer have some "rules of thumb" that can be used to judge the severity of the vibrations. While it is possible to gather complete data on the overall and spectral content of the piping vibrations, this could be very time consuming due to the large quantity of piping that may be involved in a vibration problem. Failures could actually occur before the analysis was completed.

There are several facts that can help to ease the pressure of making the decisions relative to acceptable vibration levels. First, most systems with failures do involve a resonance; therefore, systems which have no failures usually have fairly high safety factors. Typical mechanical amplification factors are between 10–20; however, for some systems such as cantilevered vents and drains with stainless steel pipe, it can be as high as 50. Typical acoustical amplification factors are 20–30 with nominal velocity flows. The major damping for acoustical resonances is the flow loss resistance; therefore, for zero flow elements such as vents, drains, bypass lines the amplification factors can be as high as 100. If the mechanical and acoustical resonances are coincident, the combined amplification factor could be as high as 600. Although, the actual measured amplification factor will typically be less than 50.

Another fact that helps in the development of a screening criteria is that an endurance limit stress does exist for piping materials. If the dynamic stress can be kept below a certain stress level, the piping can withstand infinite cycles of the stress without failure. This stress level is called the endurance stress level.

The dynamic stress introduced into a piping span by vibration is a function of the diameter and the inverse of the square of the piping span length. The natural frequency is also a function of the diameter divided by the span length squared. Based on this, screening criteria have been developed to eliminate the necessity to make a comprehensive analysis of every piping span in the piping system.

For transient events, the evaluation of the acceptability of the vibrations has to be determined by the application of cumulative fatigue theory.

### 6.1 Vibration Induced Stress

The development of the vibration induced stress for the lateral vibration bending modes of piping systems is developed in reference 11. The three methods that can be used are based on experience, vibration amplitude, and vibration velocity. The details of these criteria should be studied in order to correctly apply them to piping systems. This section will present a summary of these criteria.

**6.1.1 Preliminary Screening Based on Vibration Amplitude and Frequency**

Acceptable vibration levels versus frequency for both lateral piping bending modes and compressor manifold modes can be obtained by using the category lines given in Figure 3 [1]. This chart can be used as a screening criteria for low frequency vibrations of a piping system (frequency less than 300 Hz).

To use this chart, the measured vibration amplitudes at specific frequencies are compared to the allowable amplitudes given in Figure 3. If the vibrations are at the Design Level or lower, the system should be acceptable. If the levels are at the Danger Level, the chances of a fatigue failure are high and the vibrations must be reduced.

There are no readily acceptable charts for transient vibration amplitudes excited by surge or waterhammer; however, if the vibrations are higher than the danger level on Figure 3, the number of transient events should be limited.

**6.1.2 Based on Vibration Amplitude**

The relationship of vibration amplitude to stress for the lateral beam modes of a piping span is discussed in reference 11 and summarized in Figure 4 for a variety of piping configurations. The allowable vibration amplitude for a piping span vibrating at resonance can be estimated by the “rule of thumb” [1], and can be used as a screening criteria.

$$y_a = \frac{L^2}{D}$$

where:

- $y_a$  = allowable vibration amplitude between vibration nodes, mils peak to peak
- L = piping span length or distance between vibration nodes, feet
- D = outside diameter of pipe, inches

The vibration measurements must be made at the point of maximum vibration for the first mechanical natural frequency of the span in a lateral mode. The maximum stress occurs at the node (fixed end of the pipe) and a stress concentration factor of 4.33 and a safety factor of 2 are assumed. If the vibrations exceed the screening criteria, the vibration induced stresses are not necessarily excessive, but more detailed calculations using the methods described in reference 11 are required.

This criteria is not applicable to vibration problems other than lateral bending piping modes. It should not be applied to the compressor manifold system.

For high frequency vibrations (greater than 300 Hz), the criteria described above is still applicable if the mode is a lateral bending mode. For shell wall resonances, it is very difficult

to define an acceptable vibration amplitude, and equally difficult to accurately measure the vibrations. The sound pressure criteria discussed in reference 11 can be used as a screening criteria in many cases.

### 6.1.3 Based on Vibrational Velocity

The allowable vibrational velocity for the lowest natural frequency (lowest bending mode) of a piping span is discussed in reference 11 and a screening criteria of 2 ips was developed for a piping span with a typical weight. For a piping span which does not have a weight in the span, the allowable criteria is 4 ips. The stress concentration factor was assumed to be 4.33, the safety factor was 2, and the concentrated weight correction factor was 2.5. The vibration velocity must be measured at the point of the maximum vibration between the vibration nodes. As in the case of stress versus the vibration amplitude, extreme care should be taken to ensure that the frequency matches the natural frequency of the span between the vibration nodes. This is accomplished by using the methods presented in references 1 and 11 to calculate the lowest mechanical frequency.

$$V_a = 2 \text{ inches per second, Spans with weights}$$

The measurements must be made at the point of the maximum vibration between two vibration nodes. If the vibrations are at a higher mode between two supports, the allowable can be applied if the span between the nodes is vibrating at its mechanical natural frequency. Again, it should be noted that this criteria is for identifying those piping spans that are obviously safe. The screening criteria is very conservative for most piping systems, especially cantilevers. Most of the time, application of the criteria will eliminate the necessity of making detailed measurements of piping systems that have safe vibrations.

If the vibration velocity exceeds the screening criteria, the stresses are not necessarily excessive, however, additional calculations as outlined in reference 11 should be applied to evaluate in more detail the actual dynamic stress and safety factor of the piping system.

### 6.1.4 Dynamic Strain Criteria

The allowable vibration amplitude is based on the dynamic vibration induced stress and for high cycle fatigue, the dynamic stress must be below the endurance limit for the piping material.

For typical piping with an ultimate tensile strength of less than 80000 psi, the endurance limit from ASME B31.7 is 26000 psi, peak to peak [8]. Since the stress is equal to the dynamic strain times the modulus of elasticity, the allowable strain would be  $866 \times 10^{-6}$  in/in. If a stress concentration factor of 4.33 and safety factor of 2 is used, it can be shown that a safe allowable strain reading for a gage mounted near the area of high stress concentration would

be  $100 \times 10^{-6}$  in/in or 100 microstrain. The applicability of this strain criteria has been field verified [10].

In cases where piping failures have occurred, it may be necessary to measure strain levels in the piping to determine the safety and reliability of the rebuilt system. The strain criteria is based on placing the strain gage near, but not in, the area where the high stress risers are located. The guidelines for the interpretation of the strains is listed below.

#### Rules of Thumb for Strain Measurements

Strain:  $\epsilon < 100\mu\epsilon$  p-p probably ok  
Strain:  $100\mu\epsilon < \epsilon < 200\mu\epsilon$  p-p marginal  
Strain:  $\epsilon > 200\mu\epsilon$  p-p failure possible



## 7. Establish Need for Detailed Evaluation

All criteria are based upon the ASME fatigue allowable stress criteria [8], therefore all the discussed vibration criteria should result in the same allowable vibrations. The most accurate criteria to use is the strain criteria, but it requires the installation of strain gages and is more complicated than the measurement of vibrations.

When vibration measurements are used, the stress versus vibration displacement is the more accurate and is less likely to be misused. The velocity method is slightly less detailed; however, there are more limitations as to its use and therefore it can be misapplied.

If the vibrations exceed the appropriate criteria, the need for further detailed evaluation must be assessed. The fact that the vibrations exceed the screening criteria does not mean that the stresses are definitely excessive, but means that the piping system needs to be evaluated in more detail. The screening criteria has built in safety factors and is useful in eliminating those piping systems with safe stresses from further analysis. For example, if the stress concentration factor is less than 4.3, correction to the calculated stresses can be made and higher allowable vibrations would be acceptable.

In order to make an assessment, it must be decided if the piping system can be analyzed accurately with one of the simple models that are discussed in reference 11, or if additional modeling on a finite element computer program is needed. More accurate modeling can be accomplished by the use ANSYS, STARDYNE, NASTRAN, etc. If computer modeling is needed, it may be necessary to make additional vibration measurements using more sophisticated instrumentation, such as modal analysis systems.

Before detailed calculations are attempted, it is usually advantageous to attempt to reduce the vibrations by the installation of braces, supports, clamps, etc. so that the system is obviously safe. Various methods that can be used to increase the stiffness and mechanical natural frequency of piping systems. Many times the added clamp or brace will add damping to the system and will reduce the vibrations without a significant increase in the natural frequency. After the braces are added, the vibrations should be checked again and compared to the stress criteria. If the stresses are acceptable, the system vibrations are acceptable and no further analyses are needed.

It should be noted that the analysis and solution of complex vibration problems can sometimes require the services of vibration specialists within your organization or, in some cases, outside consultants.

## 8. Determination of Most Likely Cause of Vibration

Excessive piping vibrations are usually a result of a mechanical resonance, an acoustical resonance, or both. Diagnostic charts are presented in Tables 3, 4 and 5 to aid the analyst/engineer in determining the potential sources of the mechanical and acoustical resonances. The most likely excitation forces for steady state piping vibration are given in Figure 5. Figure 6 gives the observed responses and the probable causes.

When vibration and pulsation data are obtained, there are several types of patterns of amplitude versus speed that may be observed which can provide the information needed to determine whether the problem is caused by an acoustical or mechanical resonance.

If the pulsations do not change as the speed of the compressor changes but the vibrations show a definite resonance (as in Figure 7), this type of response pattern definitely indicates a mechanical resonance. The constant pressure pulsation amplitude eliminates an acoustic resonance as a factor. Another clue that reinforces the hypothesis of a mechanical resonance is the amplification factor of approximately 10–20, which is typical for mechanical resonances.

If both the vibration and pulsation amplitudes show a buildup as the speed is varied (Figure 7), an acoustical resonance is present, however a mechanical resonance may also be present. The amplification factor of an acoustic resonance is typically from 20 to 40, although for some branches without flow can be as high as 300.

When a mechanical and an acoustic resonance both occur, but at different frequencies, modifications must be carefully evaluated. In developing a solution, one must be careful not to shift the mechanical natural frequency where it would be coincident with the pulsation resonant frequency which could cause a large increase in the vibrations.

If the mechanical resonance is at a higher frequency as in Figure 7, one must be careful not to lower the mechanical natural frequency where it would be coincident with the pulsation resonant frequency. This may occur if weight were added to a system, such as filling up a skid with concrete to “mass” damp out the vibrations. If the acoustic resonance occurs at a higher frequency than the mechanical resonance as in Figure 7, the problem could be aggravated if a brace were added which would raise the mechanical natural frequency near the pulsation resonant frequency.

If the basic cause of the problem is determined to be a mechanical natural frequency of the piping system, improvements can probably be made by the addition of braces to stiffen the system or removal of stiffness to lower the natural frequency. Before installation, each modification should be evaluated based on the type of resonance involved. Braces (wedges, jacks, etc.) may be temporarily added to evaluate the effectiveness of additional supports and to determine if the mechanical resonance can be sufficiently changed. Sometimes the natural frequency cannot be moved sufficiently away from the excitation frequency; however, the amplitudes may be reduced by the addition of effective damping.

If the major cause of the problem appears to be an acoustical resonance, then potential acoustical modifications should be evaluated. These include orifice plates, acoustical filters, and

changes in piping size or configuration. In diagnosing the causes of piping vibration problems, the most important task is to determine if the large increases in vibrations are caused by an acoustical or mechanical resonance, or both. Vibrations should be measured over a range of speeds and operating conditions and at a sufficient number of points in the system to define the vibration pattern of the resonant mode. In addition, the pulsations in the system should be measured at a sufficient number of points so that the energy causing the vibrations can be determined.

### 8.1 High Vibrations Caused by Mechanical Resonances

If the high piping vibrations are determined to be caused by a mechanical resonance, the solution techniques to eliminate mechanical resonances should be used. The typical piping mechanical resonances that will likely be found are listed below.

1. *piping lateral beam modes*
2. *compressor manifold vibrations*
3. *pipe shell wall vibrations*
4. *instrument, vent or drain line vibrations*
5. *structural vibrations*
6. *equipment case or skid vibrations*
7. *vibrations of heat exchangers, scrubbers, etc*

These mechanical resonances can be excited by both steady state and transient forces. The description of the various excitation mechanisms are discussed in references 1 and 11 and are summarized in Figure 5.

### 8.2 High Vibrations Caused by Acoustic Resonances

If the high piping vibrations are determined to be caused by an acoustic resonance, the solution techniques for eliminating or reducing acoustical resonances should be used. Several typical acoustic resonances that will likely be found that will cause high piping vibrations are listed below.

1. *Half-wave length resonances and multiples*
2. *Quarter-wave length resonances and odd order multiples*
3. *Helmholtz resonances of piping systems*
4. *Complex acoustical modes due to interaction of complex piping*

### 5. *High frequency cross wall resonances and multiples (Bessel Functions)*

The sources of pulsations are given in Figure 5. Pulsation energy can be generated by reciprocating machinery, centrifugal machinery, or by flow excited mechanisms, such as Strouhal vortex shedding or pressure letdown devices. Transient vibrations are typically excited by the flow excited phenomenon mechanisms, such as waterhammer, slug flow, opening and closing of valves, etc.

## 8.3 Evaluate Possible Solutions

To evaluate possible solution determine whether some temporary modifications can be tried to reduce the vibrations.

### 8.3.1 Mechanical Resonance

Piping which has a mechanical resonance can be improved by:

- reducing or eliminating the energy causing the dynamic shaking forces
- detuning mechanical resonances to reduce the amplification of the energy
- strengthening the restraint system to absorb or withstand the dynamic forces while allowing acceptable vibration amplitudes.

It is important to try possible field modifications to detune the resonance. These include the addition of stiffness, weight, damping, or possibly incorporation of a dynamic absorber.

Since a mechanical resonance amplification factor for piping is approximately 10–20 for most carbon steel piping, small changes in the mechanical support system can result in a significant reduction in vibration amplitude. The mechanical amplification factor for stainless steel piping can be significantly higher and values as high as 50 have been measured [3].

Stiffness added at the location of maximum vibration will be the most effective. In the field, hydraulic jacks, come-alongs, etc. can be used to try to reduce the vibrations. Temporary stiffening may be used to determine the strength of a brace required to detune the system. Pipe clamps can add damping as well as stiffness, particularly if a visco-elastic material is used between the pipe and the clamp. In addition to reducing the vibration amplitude, the visco-elastic material will minimize pipe wear, and provide extra resiliency for thermal movement. The bracing and structural supports added to improve the system can also add weight which tends to lower the natural frequency and can offset the stiffening added.

The addition of weights can be used to detune the resonances; however, this method is generally limited due to the amount of weight that has to be added to cause a significant shift in the resonant frequency and the fact that the added weight lowers the natural frequency which moves it nearer to lower harmonics which most often have the highest energy level.

A dynamic absorber can be used as a temporary solution if it is properly designed and installed. For piping systems it is usually better to change the support spacing or add support stiffness to the system to solve the problem.

If a mechanical resonance is the cause and temporary modifications can not be tried, the analyst should investigate possible permanent solutions. This may involve developing simple piping models or complex finite element models, depending upon the system and the economics of the system.

### 8.3.2 Acoustical Resonances

For piping systems which have acoustic resonances, it is desirable to determine if the pulsations can be reduced. Listed below are several possible modifications to the system that can be used to reduce steady state pressure pulsations.

1. orifice plates
2. piping configuration change (length, routing)
3. large volumes
4. pulsation bottles (Helmholtz filter)
5. acoustic velocity change (gas composition)
6. tuned side-branch resonators

Orifices can be an effective method of adding damping or resistance to the acoustic system. Although the locations that they can be easily added is usually limited, orifices can be located at the flanges near the compressor suction and discharge. Orifices can change an acoustic resonance by locating it near a dynamic pressure minimum (open end) such as the entrance to a vessel. A ratio of the orifice diameter to the pipe inside diameter should be approximately 0.5 or less to have an appreciable effect. In this case the orifice changes the end condition from open to partially closed.

When the length resonance can be positively identified and pipe routing can be readily changed, the pipe could be shortened to increase the resonant frequency. An alternate detuning technique that has been effective on resonant pipe lengths would be to increase the pipe diameter of half of the length, leaving one-half of the length with the smaller diameter. The change in cross section at the center would break up the acoustic resonance of the original length and replace it with two shorter lengths with higher frequencies. To be effective, the larger diameter should provide a 2:1 increase in flow area.

For transient type pulsations, such as from high pressure drops across valves, etc., multiple downstream orifices in series with the valve can reduce the pressure drop across the valve and significant reduction in the pulsation energy can be accomplished. Other modifications include spargers and perforated diffusers.

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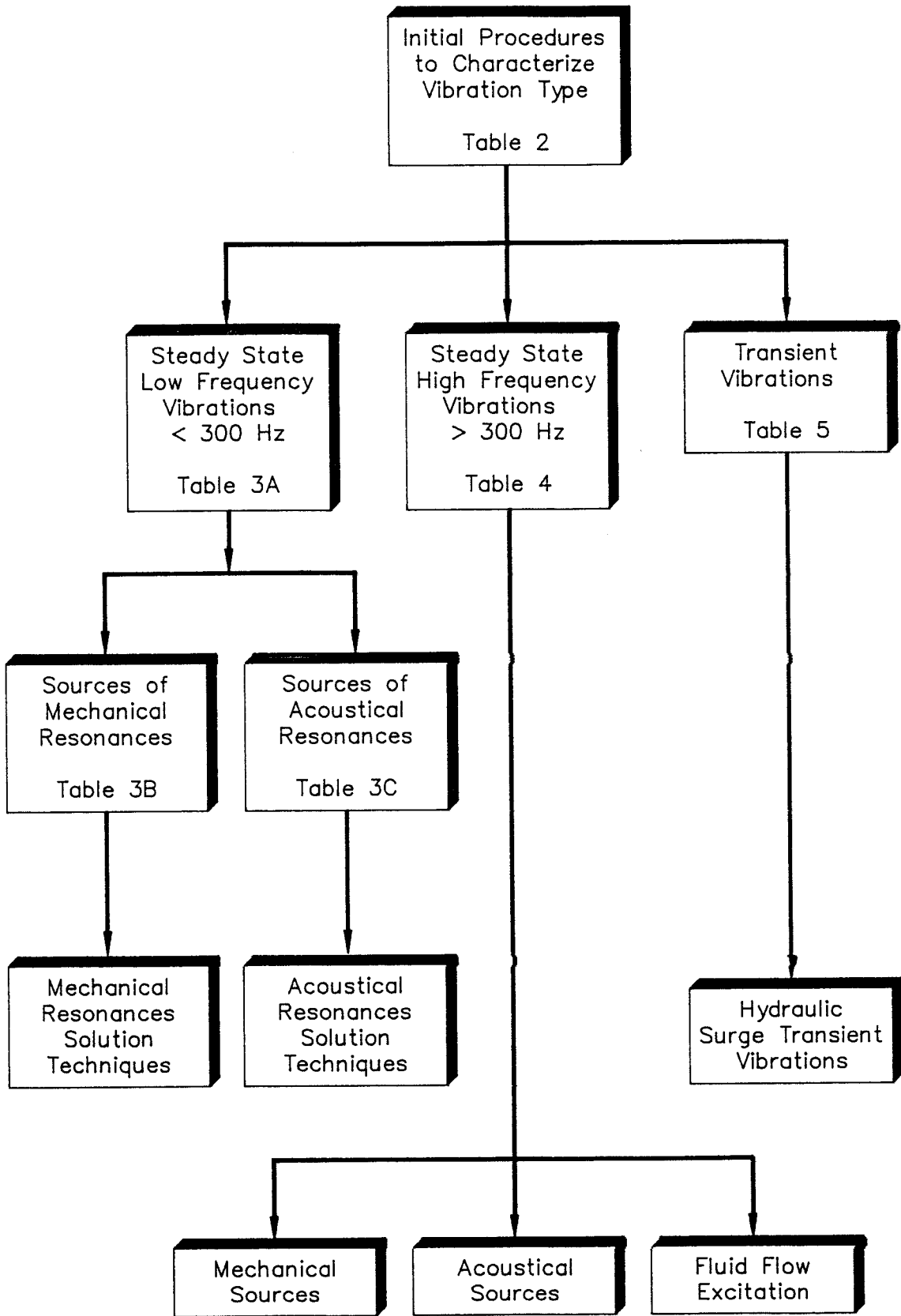


Table 1 Application Table for Diagnostic Charts

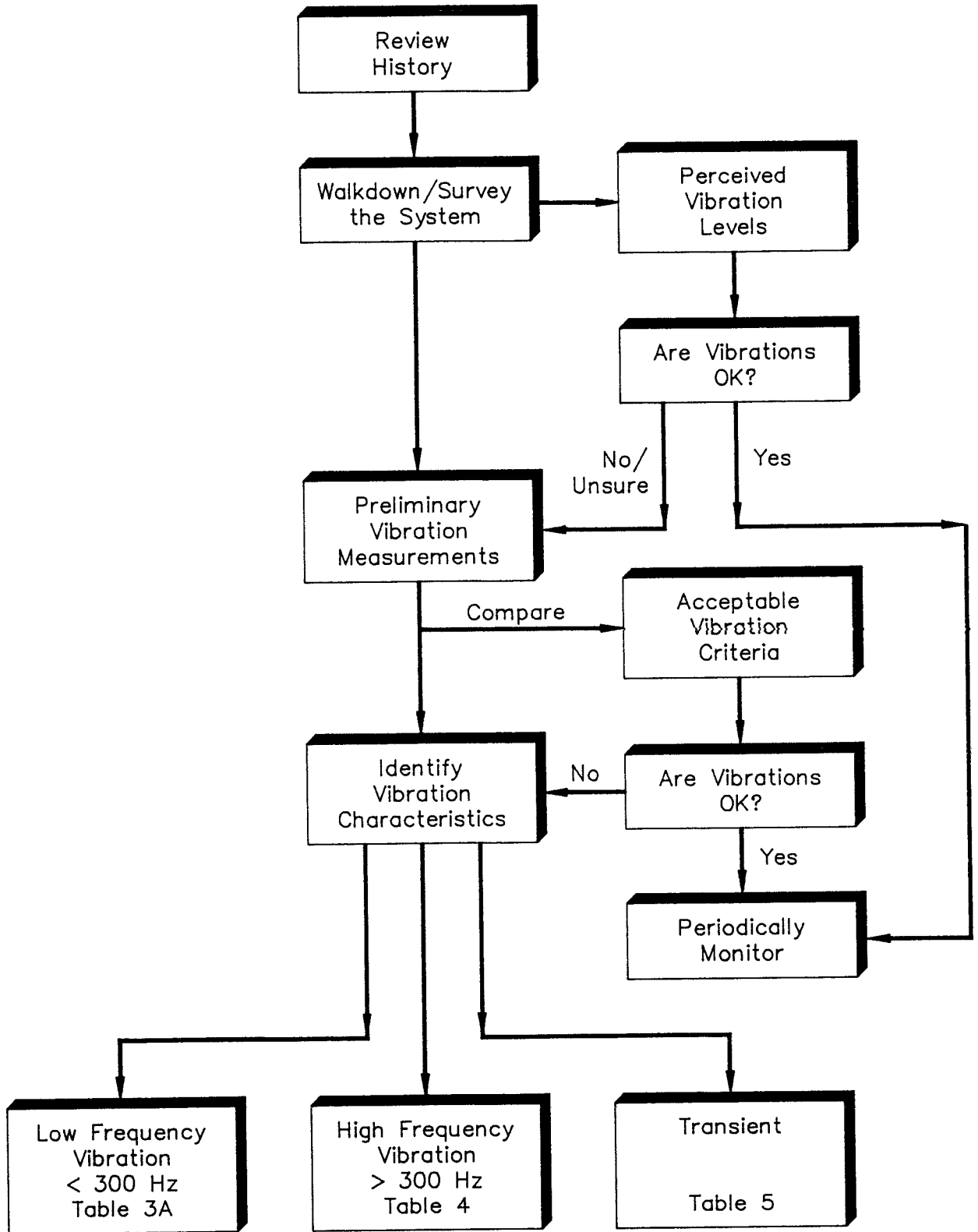


Table 2 Procedures for Investigating Vibration Problems



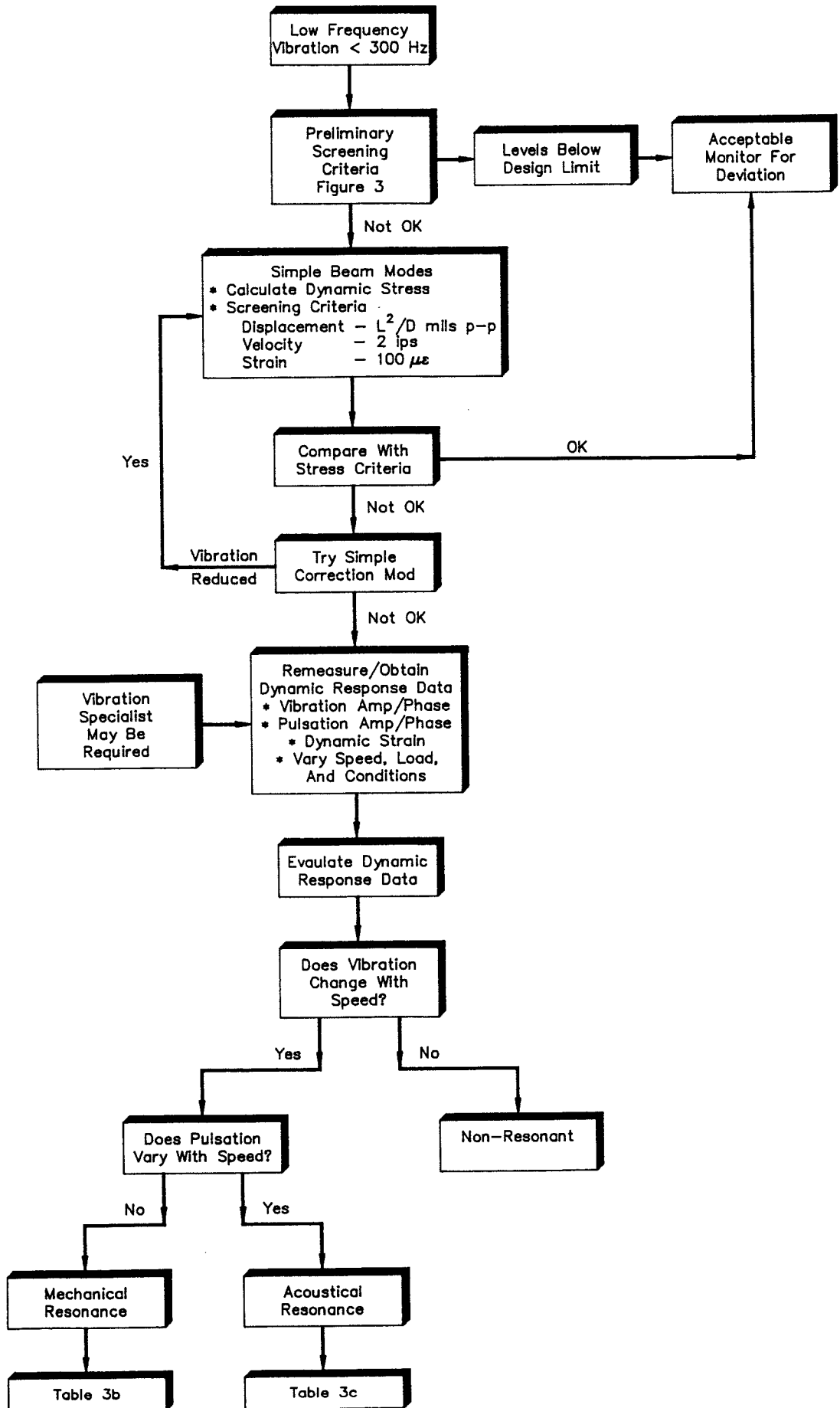


Table 3A Procedures for Investigating Low Frequency Vibration Problems

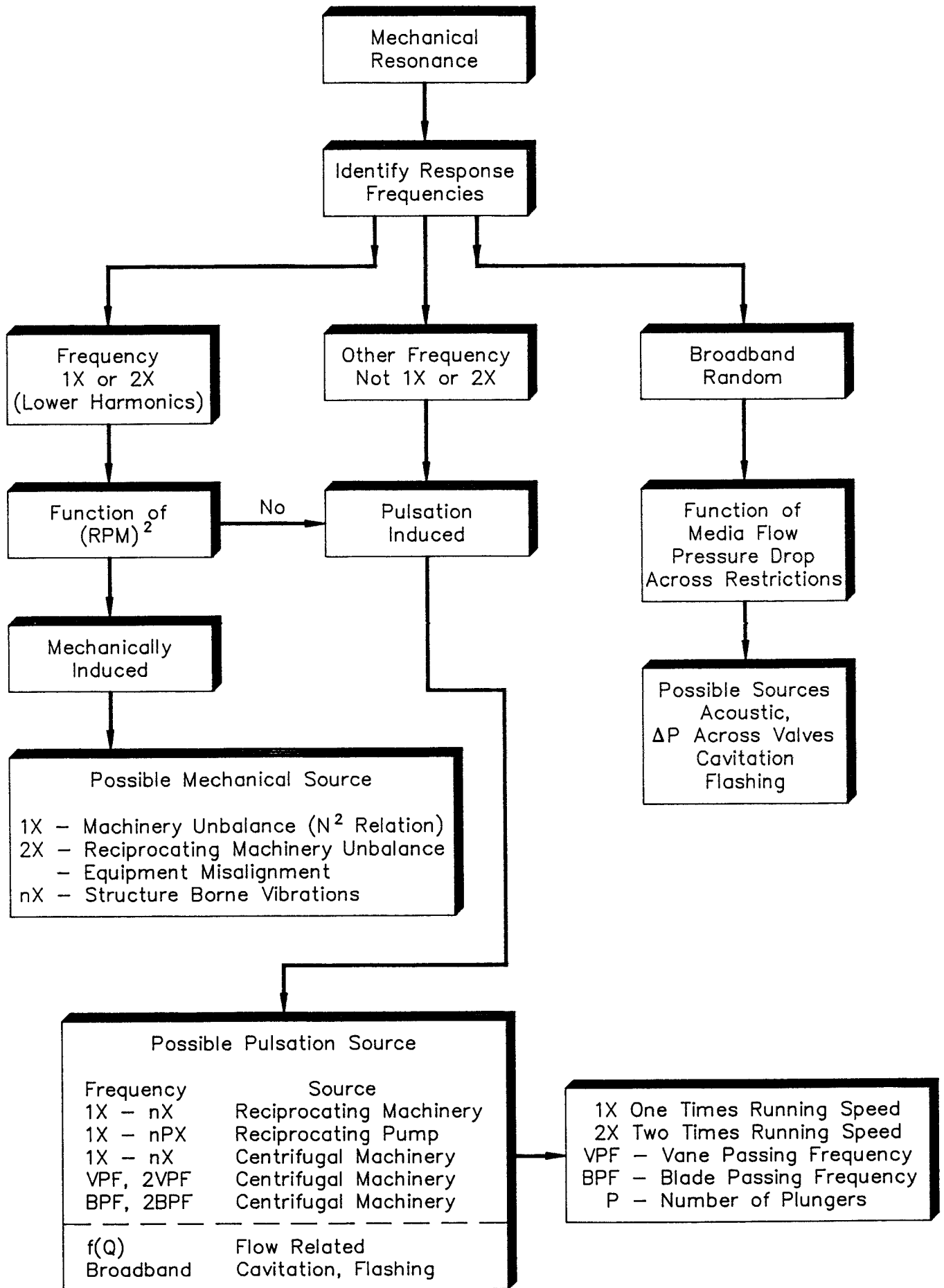


Table 3B Determination of the Source of Mechanical Resonances

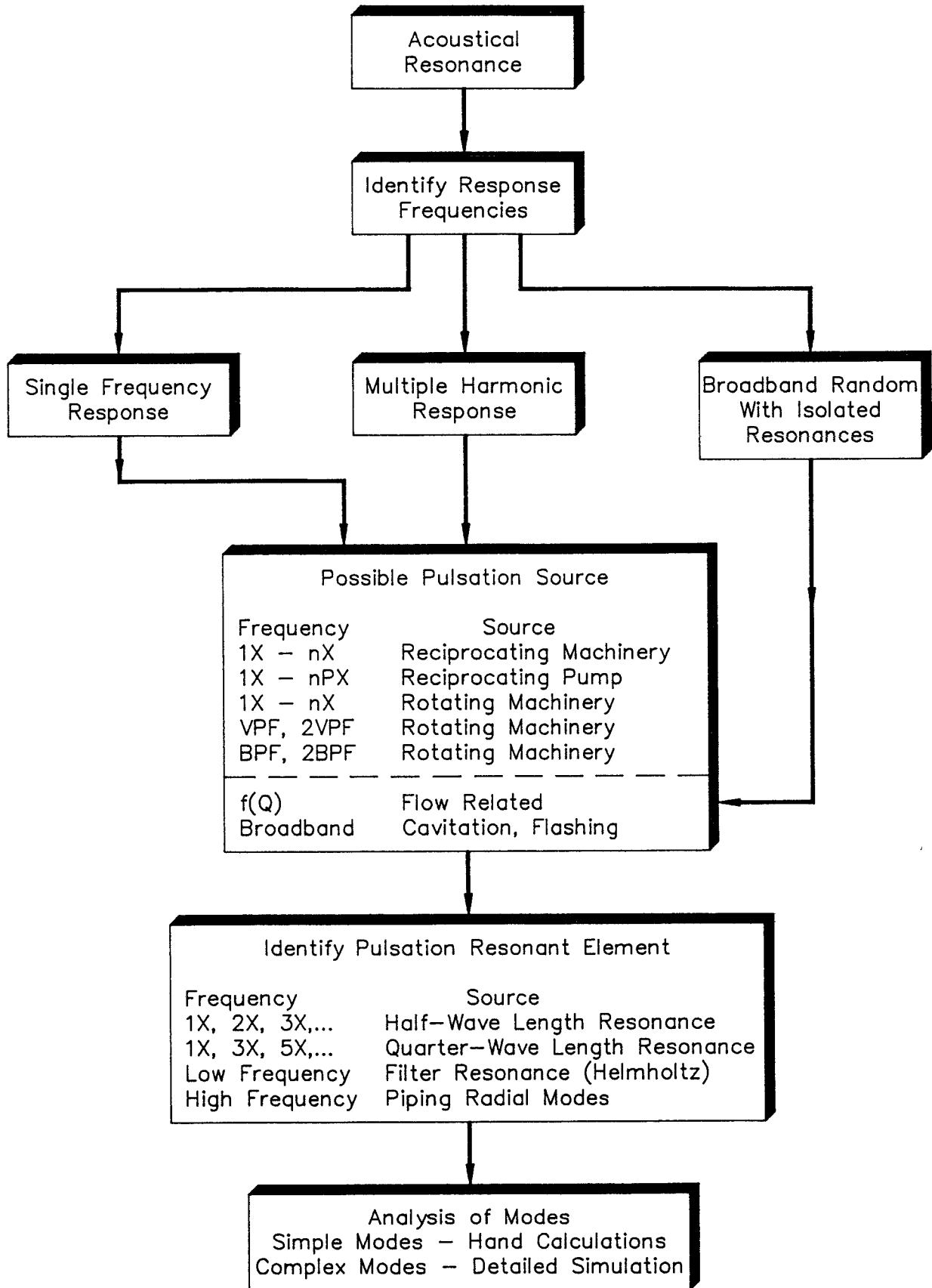


Table 3C Determination of the Source of Acoustical Resonances

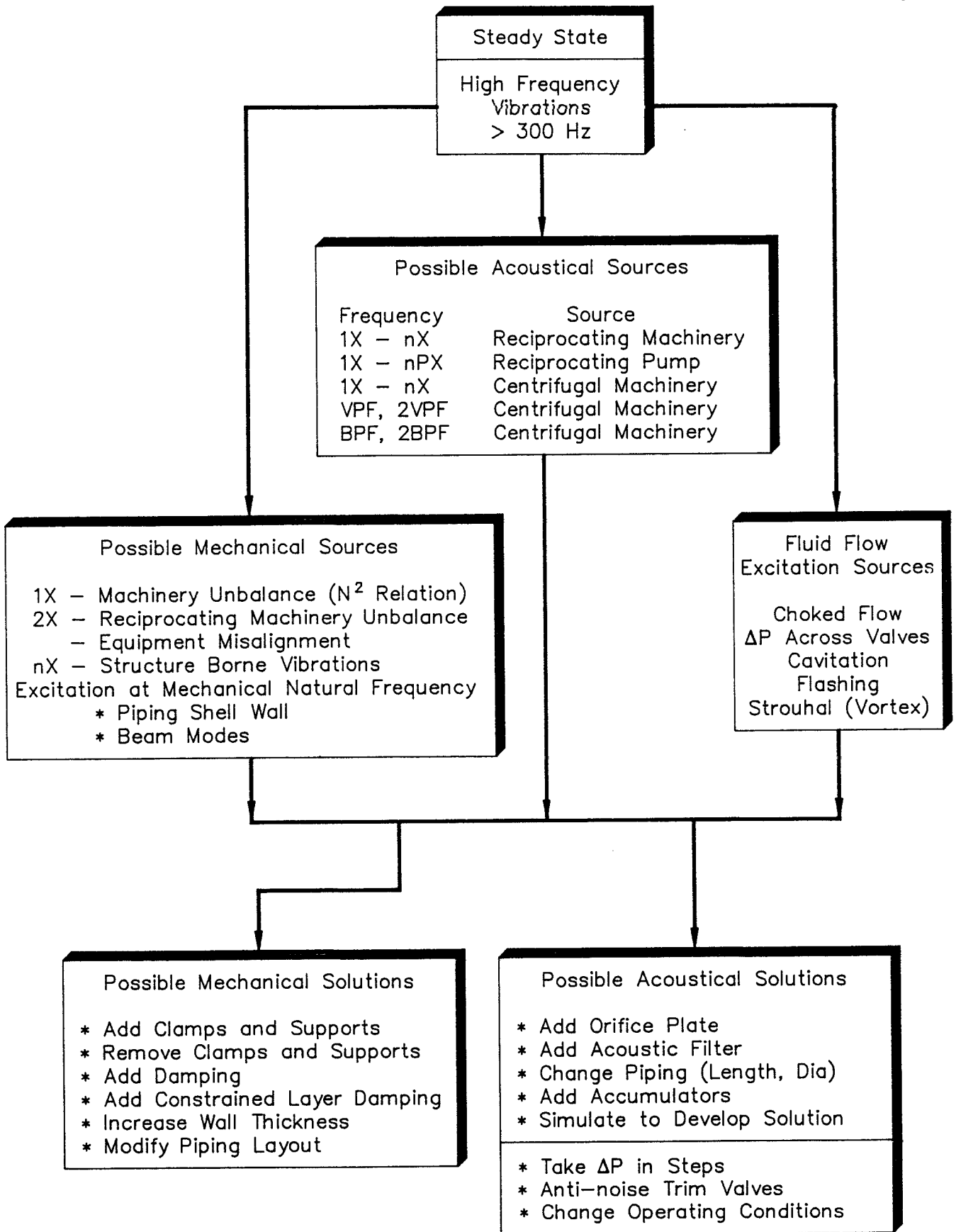


Table 4 Procedures for Investigating High Frequency Vibration Problems For Frequencies Greater Than 300 Hz

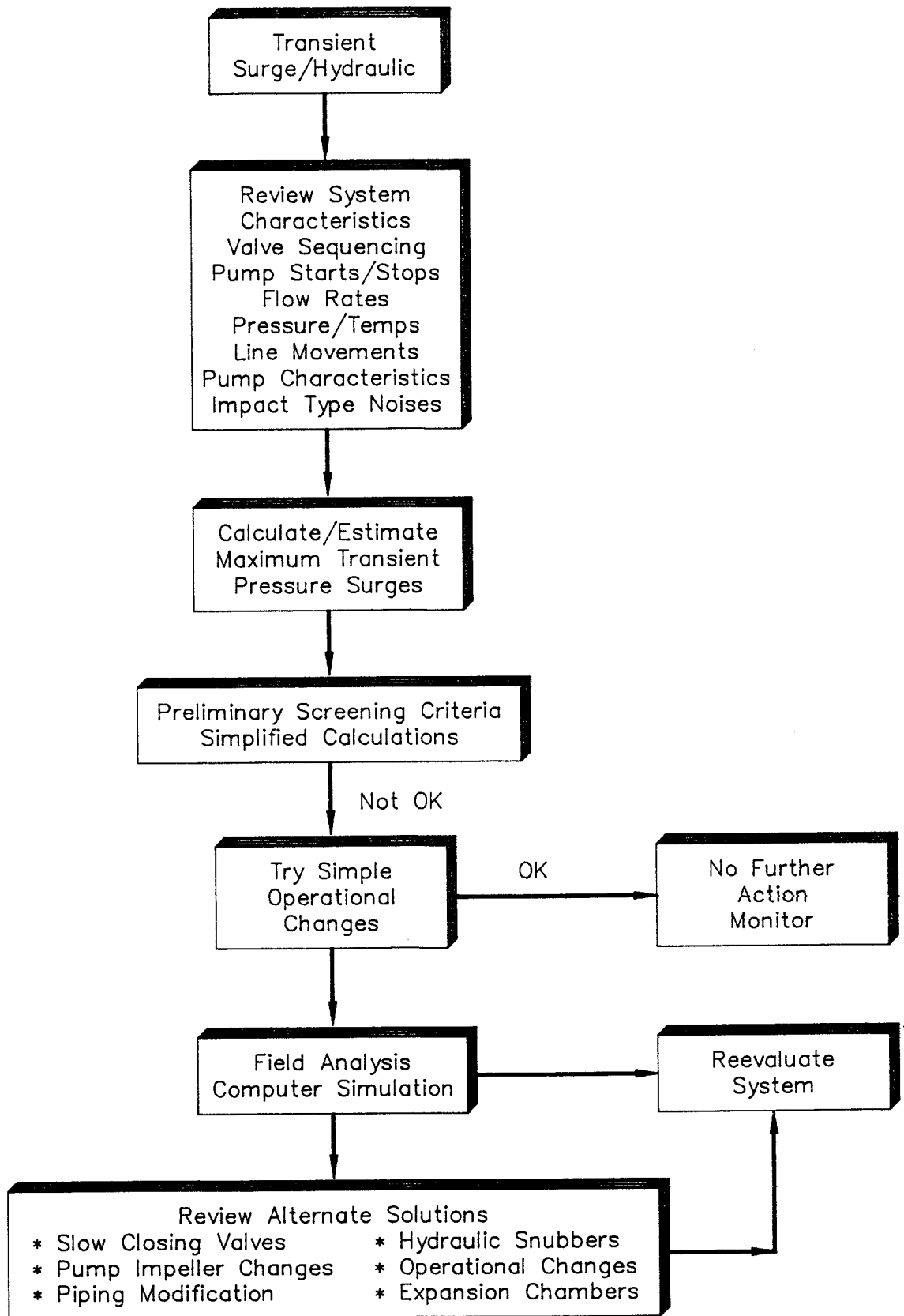


Table 5 Procedures for Investigating Transient Vibration Problems

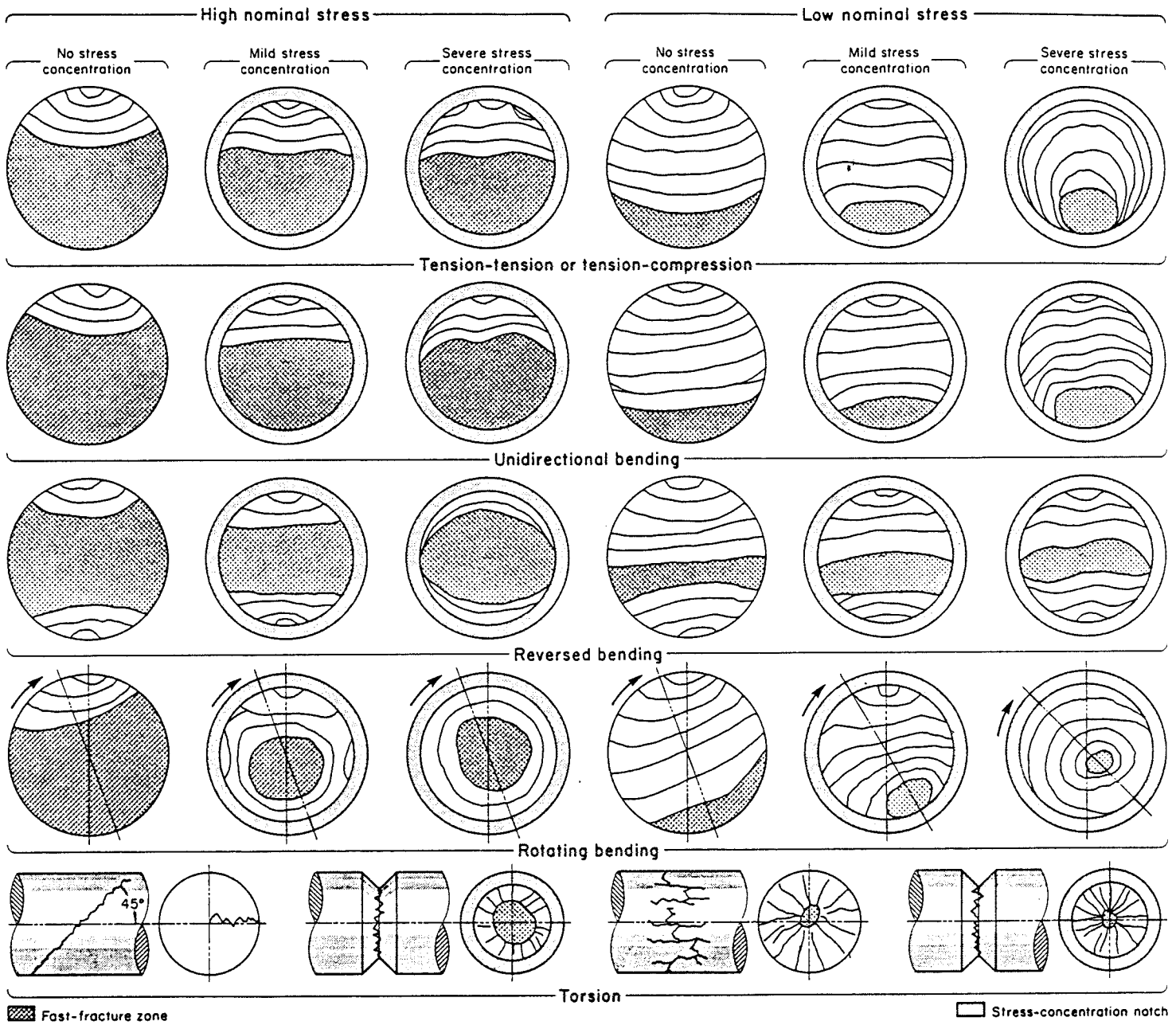
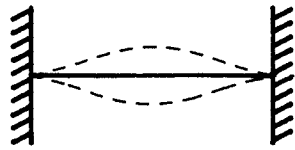
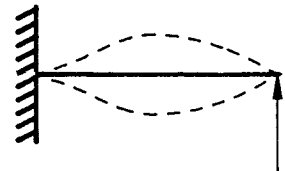


Figure 1

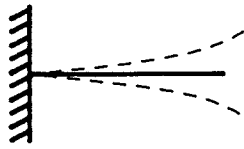
Schematic representation of marks on surfaces of fatigue fractures produced in smooth and notched components. [2]



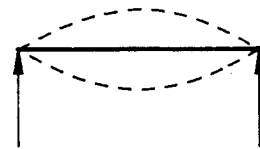
Fixed - Fixed



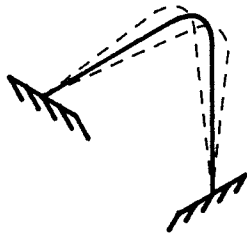
Fixed - Supported



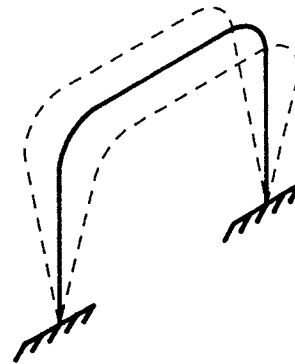
Cantilever



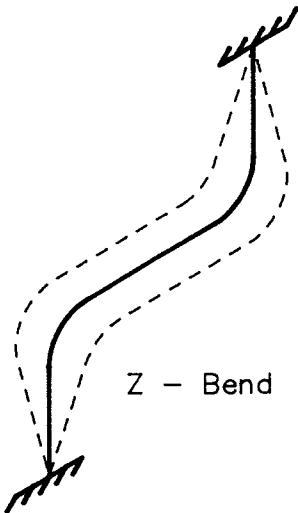
Simply Supported



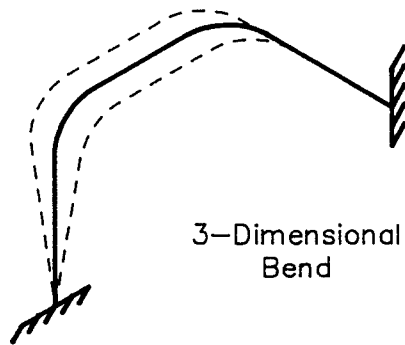
L - Bend



U - Bend



Z - Bend



3-Dimensional  
Bend

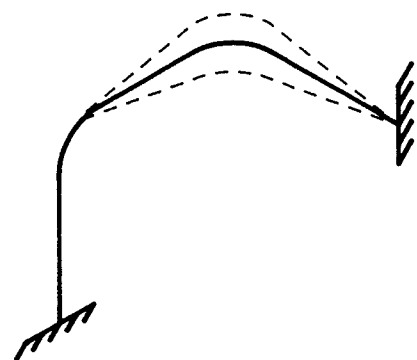
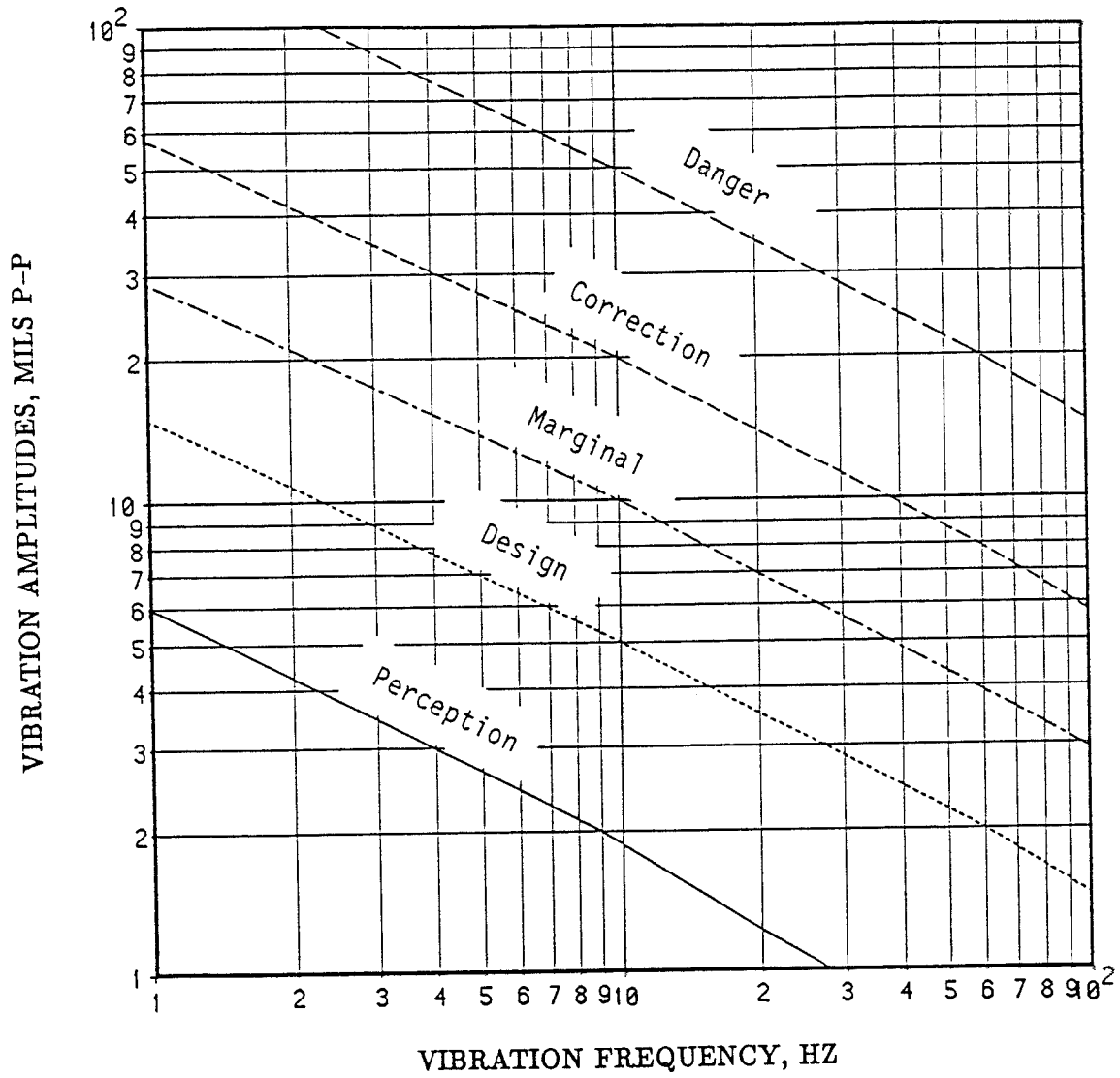
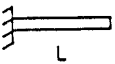
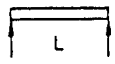
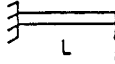
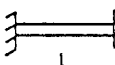
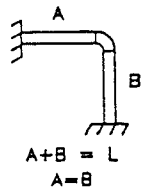
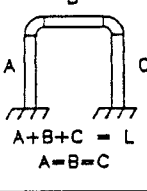
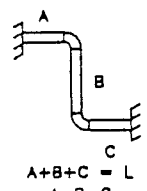
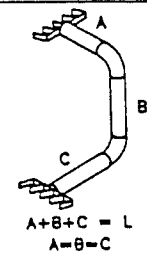


Figure 2 Vibration Mode Shapes

Figure 3 Allowable Piping Vibration Levels Versus Frequency





	Piping Configuration	Frequency Factor		Deflection Stress* Factor		Velocity Stress* Factor	
		1st	2nd	1st	2nd	1st	2nd
	Fixed-Free	3.52	22.4	366	2295	219	219
	Simply Supported	9.87	39.5	1028	4112	219	219
	Fixed - Supported	15.4	50.0	2128	6884	290	290
	Fixed-Fixed	22.4	61.7	2935	8534	275	290
 $A+B = L$ $A=B$	L-Bend Out	16.5	97.6	1889	13996	241	301
	L-Bend In	59.4	75.5	7798	9575	276	266
 $A+B+C = L$ $A=B=C$	U-Bend Out	18.7	111.6	2794	14511	314	273
	U-Bend In	23.7	95.8	3751	8722	332	191
 $A+B+C = L$ $A=B=C$	Z-Bend Out	23.4	34.2	3522	4133	317	254
	Z-Bend In	22.4	96.8	3524	8933	331	194
 $A+B+C = L$ $A=B=C$	3-D Bend	20.6	27.8	3987	4752	407	359
	Formula	$f = \frac{\lambda}{2\pi} \sqrt{\frac{gEI}{\mu l^4}}$		$S = K_d y \frac{D}{L^2} SCF$		$S = K_v V SCF$	

\*Steel Piping ( $E = 30 \times 10^6$  psi,  $\rho = 0.283$  lb/in<sup>3</sup>)

Figure 4 Frequency Factors and Stress Factors for Uniform Steel Pipe Configurations.

# VIBRATION EXCITATION SOURCES

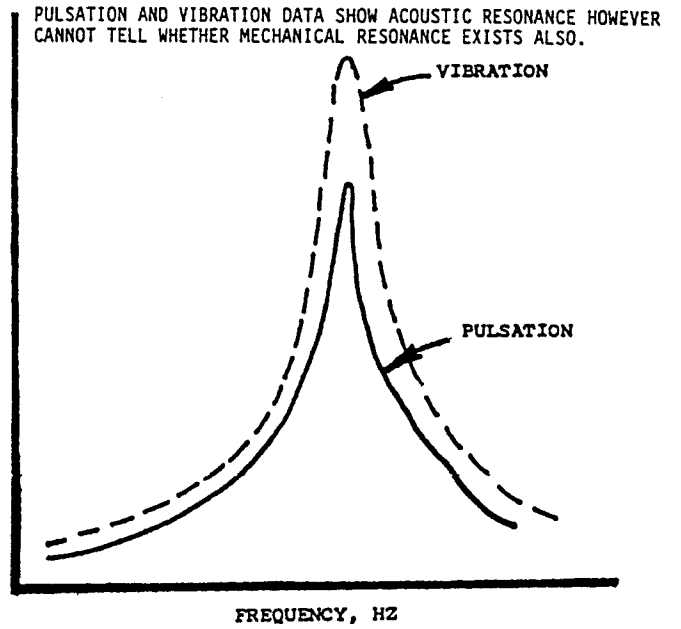
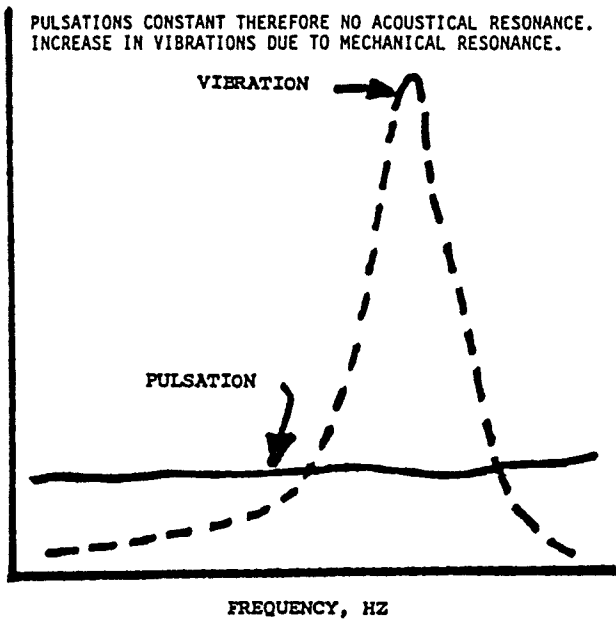
Generation Mechanism	Description of Excitation Forces	Excitation Frequencies		Piping Response	Typical Problems
<b>1. MECHANICAL INDUCED</b>					
A. Machinery Unbalanced Forces & Moments	High Level, Low Frequency	$f_1 = \frac{1N}{60}$ $f_2 = \frac{2N}{60}$		Mechanical and/or Piping Resonance of Piping System	Foundation Resonances
B. Structure — Bourne	Low Level	$f = \frac{N}{60}$			Vents & Drains Instrumentation Lines
<b>2. PULSATION INDUCED</b>					
A. Reciprocating Compressors	High Pressure Pulsations, Low Frequency	$f = \frac{nN}{60}$	$n = 1, 2, 3, \dots$ (modes) $N = \text{Speed, rpm}$	Mechanical and/or Acoustic Resonance of Piping System	Piping System Fatigue Failures, Excessive Loads to Rotating Equipment, Damaged Supports/ Restraints
B. Reciprocating Pumps	High Pressure Pulsations, Low Frequency	$f = \frac{nNP}{60}$	$P = \text{Number of Pump Plungers}$	Mechanical and/or Acoustic Resonance of Piping Systems	Cavitation on Suction Piping Fatigue Failures
C. Centrifugal Compressors & Pumps	Low Pressure Pulsations, High Frequency	$f = \frac{nN}{60}$  $f = \frac{nBN}{60}$ $f = \frac{nvN}{60}$	$B = \text{Number of Blades}$ $v = \text{Number of Volute or Diffuser Vanes}$	Complex Vibration Modes	High Acoustic Energies (Noise) Piping System Failures, Excessive Loads to Rotating Equipment, Small Branch Connection Failures
<b>3. GASEOUS FLOW EXCITED</b>					
A. Flow Through Pressure Letdown Valves or Restrictions/Obstructions	High Acoustic Energy, Mid to High Broad Band Frequencies	$f = S \frac{V}{D}$	$S = \text{Strouhal Number } 0.2 - 0.5$ $V = \text{Flow Velocity ft/sec}$ $D = \text{Restriction Diameter, ft.}$	Complex Vibration Modes in Both Longitudinal and Circumferential Directions	Fatigue Failures of Large Diameter Piping Downstream of High Capacity Pressure Letdown Valves, Small Branch Connection Failures, Flange Leakage
B. Flow Past Stubs	Moderate Acoustic Energy Mid to High Frequencies	$f = S \frac{V}{D}$	$S = 0.2 - 0.5$ $D = \text{Stub Diameter, ft.}$	Acoustic Resonance of Short Stubs	Fatigue Failure of Stub Connection to Main Run, Valve Chatter
<b>4. LIQUID (OR MIXED PHASE) FLOW EXCITED</b>					
A. Flow Turbulence Due to Quasi Steady Flow (e.g. Fluid Solids Lines)	Random Vibrations, Low Frequency	$f = 0 - 30 \text{ Hz}$	(Typically)	Low Frequency Line Movements at Mechanical Natural Frequencies	Excessive Loads on Piping Supports and Restraints
B. Cavitation and Flashing	High Acoustic Energy, Mid to High Frequencies	Broad Band		Complex Vibration Modes in Both Longitudinal and Circumferential Directions	Fatigue Failures, Small Branch Connection Failures
<b>5. PRESSURE SURGE/ HYDRAULIC HAMMER</b>					
	Transient Shock Loading	Discrete Events		High Impact Loads to Piping and Restraints	Excessive Piping/Structure Loads Due to Quick Valve Closures or Rapid Pump Starts/Stops

Figure 5  
Piping Vibration Excitation Sources.

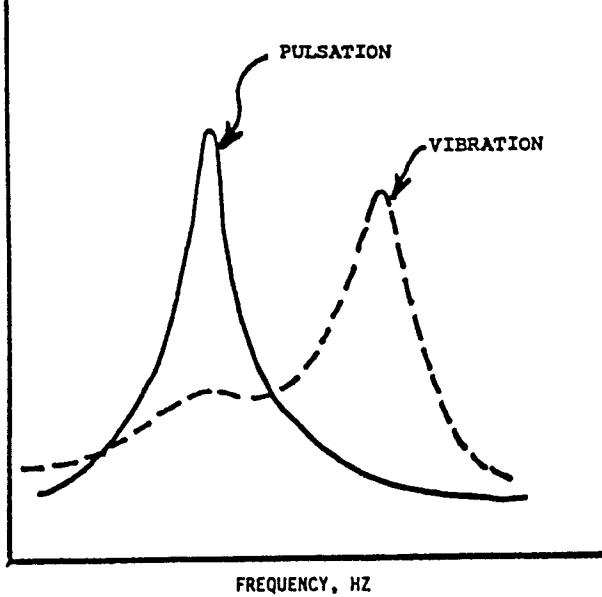
Figure 6 Summary of Observed Responses

Observed Response/ Symptoms	Probable Cause/ Source	Examples/ Corrections
Piping Lateral Beam Modes	Pulsation, Mechanical Unbalance	Brace, Add Damping Reduce Pulsations
Compressor Manifold System	Pulsations Cylinder Stretch Mechanical Unbalance	Suction & Dsch Bottles Tie down Cylinder Braces Reduce Pulsations
Pipe Shell Wall Resonances	High Pressure Drop Flow, Centrifugal BPF	Drag Valve Orifice Plates Increase Wall Thickness
Instrument, Vent, Drain Lines	Base Excitation Pulsations	Brace To Main Line
Structural Vibrations (Exciting Piping)	Mechanical Unbalance Misalignment Pulsations	Brace, Add Damping Reduce Pulsation
Equipment Case Or Skid Vibration (Exciting Piping)	Mechanical Unbalance Misalignment Pulsations	Brace Balance and Align
Heat Exchanges Scrubbers	Fluid Flow, Strouhal Pulsations	Detune Reduce Flow Brace

Figure 7  
 VIBRATION AND PULSATION PATTERNS



ACOUSTICAL RESONANCE OCCURS BELOW MECHANICAL RESONANCE THEREFORE  
 ADDITION OF MASS MAY INCREASE VIBRATIONS.



MECHANICAL RESONANCE OCCURS BELOW PULSATION RESONANCE,  
 THEREFORE ADDITION OF STIFFNESS MAY INCREASE VIBRATIONS

