COMPRESSOR STATION PIPING NOISE:

NOISE MECHANISMS AND PREDICTION METHODS

INTERIM REPORT

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1.0 INTRODUCTION

The intention of this report is to provide a more thorough understanding of the mechanisms which create audible noise from within the compressor station piping and available methods of predicting pipe flow noise apart from the machinery produced noise at a station. This report provides several noise prediction methods for flow-induced noise due to turbulent flow, noise related to pulsations, and excitation of piping resonances and noise due to the pipe wall motion from related vibrations and structure borne noise. These methods can help to better control noise from station piping and avoid operational problems through proper piping design.

At present, noise measurements and analysis of the compressor station is an “after the fact” scenario. Acoustic barriers, piping insulation, buried piping, diffusers, and silencers are used to reduce noise. Often, the noise issue is addressed by the operating company after start-up, although this could be part of the compressor station design effort. Engine and compressor original equipment manufacturers are held to a certain design specification (such as 85 dBA at a 10 foot distance) but the combined station noise levels are significantly higher than the OEM specification due to the additional contributions from the station piping and accessory machinery. As part of a station design project, it should be possible to avoid major noise issues through resonance avoidance, design mitigation for reducing turbulent, flow-induced excitation, and vibration controls in specific areas.

Field experience indicates that many noise issues are caused by a single source point within the piping (apart from the well characterized machinery level noise). The source point creates noise through one of the mechanisms described herein and leads to excessively high observed noise levels. Often times, pinpointing and reducing the dominant cause of the observed noise can help to reduce noise to operationally acceptable levels.

1.1 Objectives

This report is written as a general handbook for understanding and diagnosing compressor station piping noise issues. Its focus is solely on the mechanisms which can create noise within a piping system. It is limited to flow-induced noise, pulsation-related noise, and piping wall vibration noise (either from the flow or structure-borne) and provides a methodology for predicting noise levels related to these specific noise mechanisms. The report does not provide a design methodology or requirements for machinery, control valves, or noise controls (such as insulation, acoustic barriers, and silencers).

In addition, the methods presented herein will predict audible noise due to the sources within the pipe and the piping wall itself. External noise (outside the pipe) at the compressor station will be a combination of other noise sources at the compressor station and not composed solely of station piping noise. Noise will travel externally based on the location of reflection sources (buildings, walls, and other structures) and the buried and unburied pipe. Any methodology for station noise predictions should be applied within this framework.

1.2 Definitions

Acoustic Efficiency or Conversion Efficiency: The ratio of acoustic power to mechanical input power, expressed as an efficiency of conversion of mechanical to acoustical power.

A-Weighting Scale: The frequency weighting scale for noise measurements which weights the frequencies near the human hearing response frequency range most heavily.

Cutoff Frequency: The lowest frequency (first acoustic mode) for sound wave propagation across the diameter of the pipe within the fluid. Cross modes cannot propagate below the fundamental first mode of the cutoff frequency. The cutoff frequency is defined as the speed of sound in the gas/fluid within the pipe divided by the product of 2*pipe diameter, f = SOS / (2D).

Decibel: A measure of the amount of energy in an acoustic signal, expressed as a ratio compared to a certain baseline reference measurement depending on the use of the A, B, C, or D decibel scale. The mathematical formulation of the dB is as the common logarithm of the ratio of the measured sound pressure to that of a signal that is barely audible.

Pure Tone: A sound for which a waveform can be represented by a sine wave at a single frequency.

Radiation Ratio: The amount of energy radiated in the form of pressure fluctuations based on the amount of pipe wall acceleration for a given pipe

Ring Frequency: The frequency for which the longitudinal wavelength is equal to the pipe circumference, to define the upper limit for noise transmission within the pipe material. This is the upper limit for transmission of sound through the pipe wall, beyond which transmission is minimal. The ring frequency is defined as the speed of sound in the pipe wall material divided by the product of π*pipe diameter, , $f_r = c_L / (\pi D)$.

Sound Intensity: The continuous flow of power carried by a sound wave through an area in space, commonly expressed in Watts per square meter. Sound intensity is directional associated with the direction of the sound wave compared to the plane in space.

Sound Power: The amount of power radiated by a sound source or the energy radiated per unit time by a sound source, typically defined by a spherical boundary surrounding the sound source.

Sound Power Level (L_w, dB): A decibel expression of the sound power. The reference sound power is 10^{-12} watts.

Sound Pressure: The variation in pressure above and below atmospheric pressure, normally expressed as root-mean-square (rms) value of the time averaged fluctuations. This is a direct measure of the pressure variation.

Sound Pressure Level (L_p, dB): A decibel expression of the sound pressure. The reference sound pressure is 2 x10^{-5} Pa, which is the threshold of human hearing at 1,000 Hz.

Transmission Loss: The difference between the incident and transmitted sound power levels for the candidate noise control option. Similar to *Insertion Loss*, but in some cases introduction of the noise control option actually increases the sound power incident on itself.

2.0 BACKGROUND

The intention of this section is provide the reader with the basic foundation to understand the concepts regarding noise transmission, attenuation, and magnification. This background section covers basic sound theory, gas flows and station piping noise mechanisms, acoustic modes, shell modes, and transmission loss. Techniques for noise prediction and related equations are presented in later sections. However, noise and especially interaction between surfaces and fluids is a complex topic, and it is not the intent of this report to be an exhaustive resource on this topic.

2.1 Sound Theory

Sound is a compression (mechanical) wave in a gas or liquid. Significant noise can be induced by a vibrating solid in either a gas or liquid system, typically through bending waves. Conversely, noise in a fluid can induce vibrations in a solid. The most obvious examples are speakers and microphones. However, a similar interaction occurs when a pipe wall or enclosure wall vibrates. Station piping noise will be a combination of how the pressure waves within the fluid (produced by turbulent flow or a compressor excitation) reflect upon the internal piping structure and how these waves induce vibrations in the solid pipe wall.

Attributes of mechanical pressure waves such as frequency, speed, and damping are affected by the media in which they travel (such as air). The frequency, speed, and wavelength are related to each other by Equation 2-1. In addition, the amplitude (loudness or intensity) of the sound is a function of how the pressure wave is reflected and allowed to resonate. The intensity of the sound is important attribute associated with the origin of the sound.

$$c = \lambda \cdot f \qquad \textit{(Equation 2-1)}$$

where:

c = speed of sound for fluid
λ = wavelength, length of pressure or velocity wave
f = frequency, typically expressed as Hz or number of cycles per second

Sound produced by unsteady gas flows and by interactions of gas flows with solid piping components and the pipe wall at turbulent conditions is often termed *aerodynamic sound*. The same gas flows will excite structural modes of vibration in surfaces, but these are distinctly different sound sources, regarded as *structure-borne sound*.

For aerodynamic sound sources, there are three basic aerodynamic source types: monopole, dipole, and quadrupole. Monopole radiation is produced by the unsteady introduction of mass or heat into a fluid. For a compressor station, this type of sound is produced by a reciprocating compressor from its cylinder valves or pulsed flow through a check valve. This is also the type of sound radiation occurring for strouhal excitation, where vortex shedding off a blunt body or pipe dead leg can produce pulsing flows. Dipole sources occur when unsteady flow interacts with surfaces or bodies when the dipole strength is equal to the force on the body or surface. This source type is found in compressors where turbulence flow impinges on stators, rotor blades, and other control surfaces. Quadrupole source radiation is produced by the Reynolds stresses in a turbulent gas in the absence of obstacles. This is the type of noise characteristic of turbulent flow in straight pipe.

The acoustical source radiation will change direction if the source moves relative to the fluid as opposed to a stationary noise source. The frequency and intensity of the sound are increased ahead of the source and decreased in the reverse direction.

Pressure fluctuations will occur in turbulent gas flows at the boundary layer, causing both aerodynamic sound radiation and sound from vibrations of the pipe wall as the pressure wave contacts the solid surface. Sound from the solid/gas contact is produced as diffraction of the pressure wave, as it makes contact with the pipe wall or flows past other solid surfaces in the flow stream. These additional noise sources produced by the pipe wall vibration are essentially structural modes excited by the boundary layer pressures.

2.2 Range of Human Hearing

When discussing human hearing it is important to note that not all sounds can be heard by the human ear (due to the frequency range). Many sounds that can be heard will vary in "loudness." These two aspects of hearing represent two important criteria: the first is the tone or frequency range of the sound; the second is the "loudness" or intensity of the particular sound.

The "loudness" or intensity of a sound is directly related to the amplitude of the pressure wave of the sound: the larger the amplitude of the pressure wave, the louder the sound. In typical natural gas applications, normal pulsations amplitudes (peak to peak) can range from 1.0 to 100.0 psi peak to peak. In high pressure applications or situations with extremely high undamped responses, pulsations may exceed these amplitudes. However, these amplitudes are surprisingly much larger than the intensity range acceptable for a human ear. A sound such as jet taking off 200 ft away is extremely loud to the human ear, although such a noise is about 120 dB or 20 Pascals, rms (or 0.0082 psi pk-pk). The minimum threshold of hearing is considered to be 20x10-6 Pascals, rms (8.2x10-9 psi pk-pk).

These pressure levels are quite small when compared to typical pulsations within compressor station piping. By comparison, the large pulsation amplitudes experienced at reciprocating compressor stations or the smaller velocity/pressure waves caused by turbulent flow can easily project from inside the pipe to the atmosphere and cause a noise problem. Table 2-1 below provides example magnitudes of various noises within the range of human hearing.

Table 2-1. Qualitative Scale for Various Noise Levels

Noise Level dBA	Qualitative Scale
140	Jet take-off at 80 ft, threshold of pain.
130	Painfully loud.
120	Jet take-off at 200 ft.
110	Car horn at 3 ft.
100	Shouting into an ear.
90	Heavy truck at 50 ft.
80	Pneumatic drill at 50 ft.
70	Road traffic at 50 ft.
60	Room air-conditioner at 20 ft.
50	Normal conversation at 10 ft.
40	Background wind noise.
30	Soft whisper at 13 ft.
0	Threshold of hearing.

The tone or pitch of any particular noise is directly related to its frequency. The range of human hearing has been reported to be as wide as 20 to 20,000 Hz, with the most sensitive band being between the 500 to 10,000 Hz range. Long exposures to loud noise or aging can reduce the sensitivity of the human ear and the frequency range that a particular individual can hear.

2.3 Mechanical to Acoustic Energy Conversion

In analyzing noise sources and the potential amplitudes associated with pressure wave fluctuations, it is important to consider the conversion efficiency. Although some mechanical sources of noise have much higher overall power ratings, the conversion efficiency is low which means little mechanical energy is converted into acoustic energy (pressure wave fluctuations in space). This is the case with many compressors and engines at gas pipeline stations, which have high overall power ratings but less effective coupling with the ambient air, producing less overall noise. Gas flows at higher pressure and flow velocities will produce more noise in many cases than the station operating equipment. As such, the highest or most problematic noise sources which are often left as a by-product of the overall station design tend to come from station piping noise mechanisms, not compression equipment. This is due to the good conversion efficiency between high pressure gas flows and the coupling of the fluid with the pipe wall. The coupling effects are taken into account by the transmission loss factor and the radiation ratio.

2.4 Occupational Limits

The Occupational Safety and Health Act (OSHA) of 1970 used study results for human hearing loss to determine permissible limits for daily exposure time, designed to avoid permanent hearing loss. The limits are intended to protect human hearing in the range of frequencies necessary for understanding speech for the majority 85% of the population. They are stated below in Figure 2-1. Below a limit of 90 dBA, no hearing damage is possible and no limit is prescribed. These limits also assume that the remainder of an individual's eight hour work day is essentially spent unexposed to other noises above 90 dBA. Recreational noise sources can be severe and should also be monitored from an individual hearing loss perspective. Other studies and hearing loss sources recommend limiting non-occupational noise at 85 dBA to less than 2 hours and noise at 75 dBA to less than 8 hours.

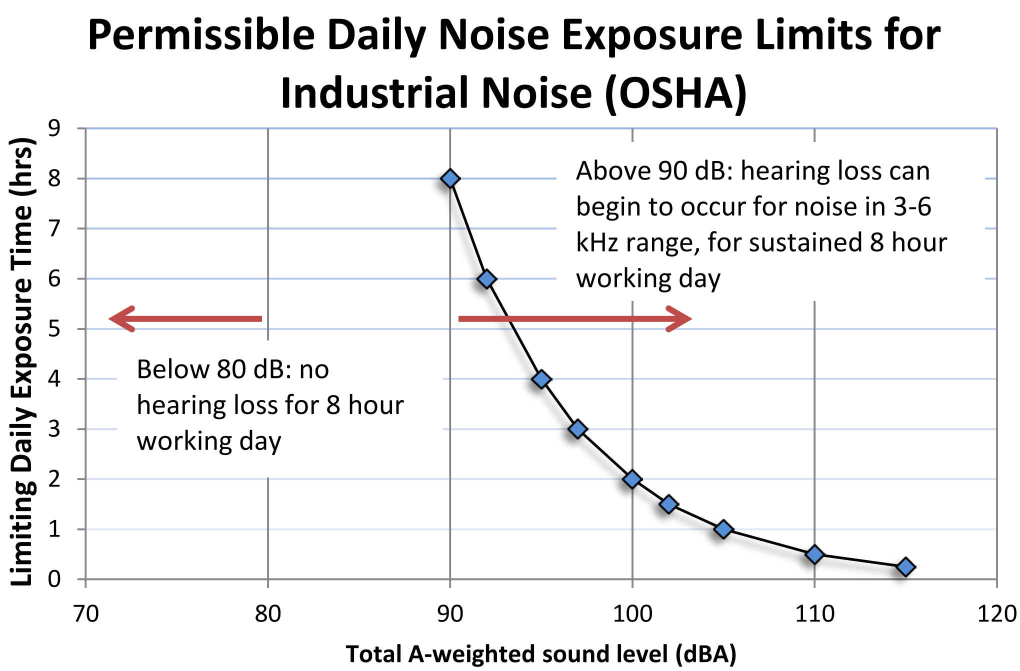

Figure 2-1. OSHA Defined Noise Exposure Limits, Starting at 90 dBA

2.5 Decibel Scale

The motivation to use the decibel scale for sound amplitude stems from the wide range of amplitudes observable to humans (20 µPa to 200 Pa). The decibel scale is a relative scale which requires a reference value. For noise and sound, the reference value is universally recognized as 20 µPa, which is the lower limit threshold of human hearing. The decibel scale is a log based scale which provides a more convenient way to plot/view the large range of observable amplitudes.

Equation 2-2 below shows how sound pressure level is calculated from known pressure fluctuations, calculated as a root mean square (RMS). The pressure is squared since the decibel scale is derived for power relationships. Pressure squared is proportional to sound power. With the log-based scaling for decibel levels of sound pressure, a basic equivalency rule is that an increase of 3 dB (sound pressure) will effectively double the sound power level.

$$L_p = 10 * \log_{10} \frac{p^2}{p_{ref}^2} = 20 * \log_{10} \frac{p}{p_{ref}} \qquad (Equation\ 2\text{-}2)$$

where:

Lp = Sound Pressure Level
p = pressure wave fluctuation, expressed as root mean square, RMS value
Pref = reference pressure (20 μPa)

2.5.1 SPL Scale with A-Weighting

The human ear is not sensitive to all frequencies within the observable range. For instance, sound power at a frequency of 1,000 Hz is potentially more damaging than sound at a frequency of 50 Hz with the same power. In order to take this effect into account, various scales or "weighting" have been created to adjust sound measurements. Four of the most common weighting scales are the A, B, C, and D scales. The one most commonly used in industry is the A-weighted scale which emphasizes the frequency range between 500 to 10,000 Hz. Figure 2-2 below provides a plot of the values added to a measured noise to provide an A-weighted decibel value. Noise values that have been A-weighted are typically denoted as dBA.

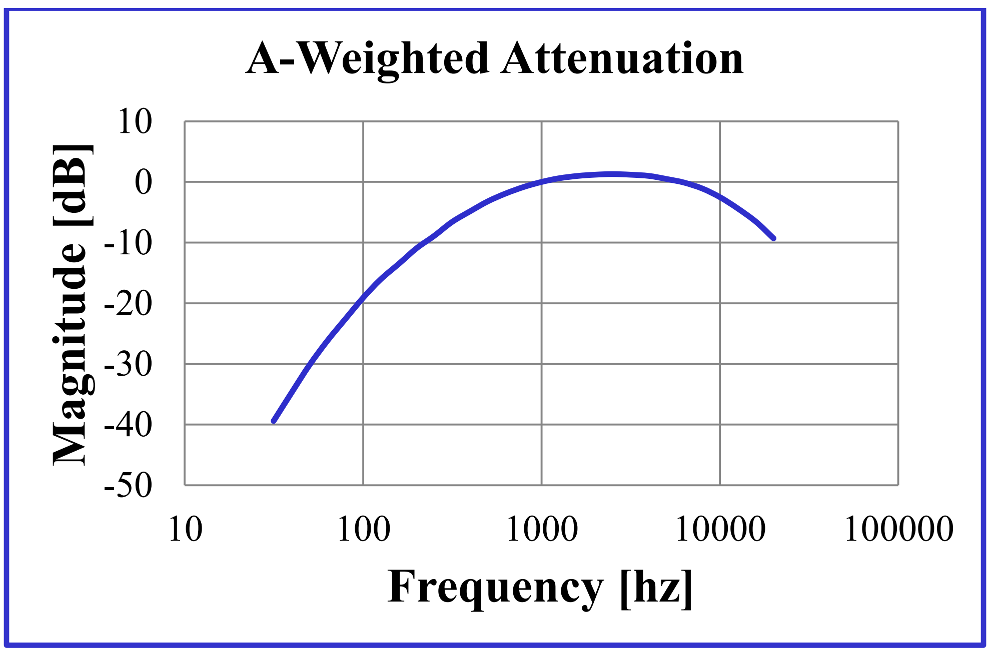

Figure 2-2. Decibel A-Weighted (dBA) Scaling for Sound Pressure Level Noise Measurement

2.6 Sound Pressure Level Measurements

Typical measurements of noise include sound power level (SWL), sound intensity, and sound pressure level (SPL). Since the pressure amplitudes decrease with distance away from a source, sound power level is useful for describing and comparing noise sources. Sound pressure level is measured in watts, but it can also be expressed in decibels using the reference power of 10-12 watts. Measuring sound power directly can be difficult and assumptions are typically made in most field conditions. It should be noted that SWL in decibels is not the same as sound intensity or sound pressure level in decibels.

Sound intensity is the flux of sound energy and has the units of watts per square meter. Sound intensity has a vector component; in other words it has a magnitude and direction. The most basic measure of sound intensity would be a combination of sound pressure and particle direction and velocity (pressure multiplied by velocity has the same units of W/m2). The advantages of measuring sound intensity are twofold: first, noise sources can be identified by using the direction aspect of the measurement, and second, the power of the source can be determined. Typical sound intensity measuring systems actually use two microphones and data processing methods to determine sound and direction. The source power can be determined by properly setting up a measurement and integrating sound intensity over an arbitrary area.

Sound pressure levels are the easiest measurement to take and also are directly related to the response of the human ear. Microphones are typically used to measure pressure pulsations (noise). This method is quite similar to the human ear where the eardrum responds to pressure pulsations. There is no direction aspect to this measurement and typically not a good way to determine source power. However, measuring SPL does indicate what noise level a human will experience at the measurement location.

2.7 Station Piping Noise Producing Mechanisms

Noise radiates from sources in the compressor station piping and piping flow disturbances which enact pressure wave reflections. Sound is produced by fully developed flow at high velocities through turbulence and may also be caused by reflections from partially developed flow from elbows, tees, valves, and other typical piping variations. The flow will produce a certain level of noise from turbulence but will also cause the pipe wall to vibrate and produce additional noise. Noise will be amplified at resonant conditions. The most common noise source is due to turbulent flow in a straight run of pipe. This tends to be relatively low in magnitude but will still manifest itself as ambient noise, as a function of the noise transmission through the pipe wall.

In acoustics, transmission loss is commonly defined as the loss in acoustic energy as pressure waves are reflected and transmitted across a medium. In this case, the medium is typically steel pipe surrounded on either side by two fluids: pressurized gas on the inside of the pipe and ambient pressure/temperature air on the outside of the pipe. The ratio of the intensities of the reflected and transmitted pressure waves depend on the acoustic impedances, the speed of sound and the angles of incidence. The transmission loss is a more difficult quantity to characterize when the acoustic impedances on either side of the pipe wall differ, as in this case, and when the internal fluid’s impedance varies throughout the piping network.

Apart from fully-developed turbulent flow noise (Mechanism #1), station piping will produce other noise due to the following:

- Mechanism #2: Turbulent flow past non-straight pipe or in-pipe disturbances, causing additional pressure wave reflections and possible noise amplification.

- Mechanism #3: Acoustic and mechanical resonances, which may be excited by the flow stream or pulsations in the flow stream.

- Mechanism #4: Pipe wall vibrations caused by turbulent flow, resonances, or structural born noise.

(Additional description based on these four noise mechanisms is provided in Section 3.)

Noise due to turbulent flow pipe noise and Mechanism 2 (flow past non-straight pipe and piping disturbances) can be predicted based on characterizing the internal pipe noise and transmission loss through the pipe wall, followed by noise source characterizations and superposition for a given position at the compressors station (distance from noise sources).

2.8 Transmission Loss Models: Cut-off Frequency and Ring Frequency

For pipe gas flows, the noise transmission level is frequency dependent, based on the gas flow cut-off frequency and the ring frequency (a function of pipe diameter). The particular model or correlation used to relate the frequency to the transmission loss through the piping will affect the results of any prediction model. Based on the literature review of industrial piping noise studies, there is considerable disagreement between existing correlations on the transmission loss.

The NASA "Reduced-Noise Gas Flow Design Guide" published in 2005 recommends a correlation which uses three frequency regions, divided by the cut-off frequency (defined as the first fundamental mode for wavelength transmission across the pipe) and the ring frequency (defined as the mechanical limit for natural structural frequency). Near the cutoff-frequency, noise transmission will be high. As the frequency increases, the transmission will be less due to increased transmission loss from the higher frequency modes approaching the ring frequency of the pipe. The ring frequency is an upper limit such that frequencies above the ring frequency will not couple well with the pipe wall and the radiation ratio is severely diminished.

Compressible gas flows tend to have much "louder" piping systems, due to the high potential for broadband noise and the relatively good conversion of mechanical to acoustic energy. This relationship between the cutoff frequency and the ring frequency effectively defines a frequency band of high noise transmission, where the transmission loss through the pipe will be relatively low.

The Norton method of 1997 recommends two equations for the transmission loss, above and below the pipe ring frequency. These equations are provided below as Equation 2-3 and 2-4. The Norton method does not show a low point for high noise transmission around the cutoff frequency, but the severe negative slope for low frequencies is still consistent with the NASA guideline. The NASA model and the Norton method are illustrated in Figure 2-3 for a typical 10-inch diameter steel pipe.

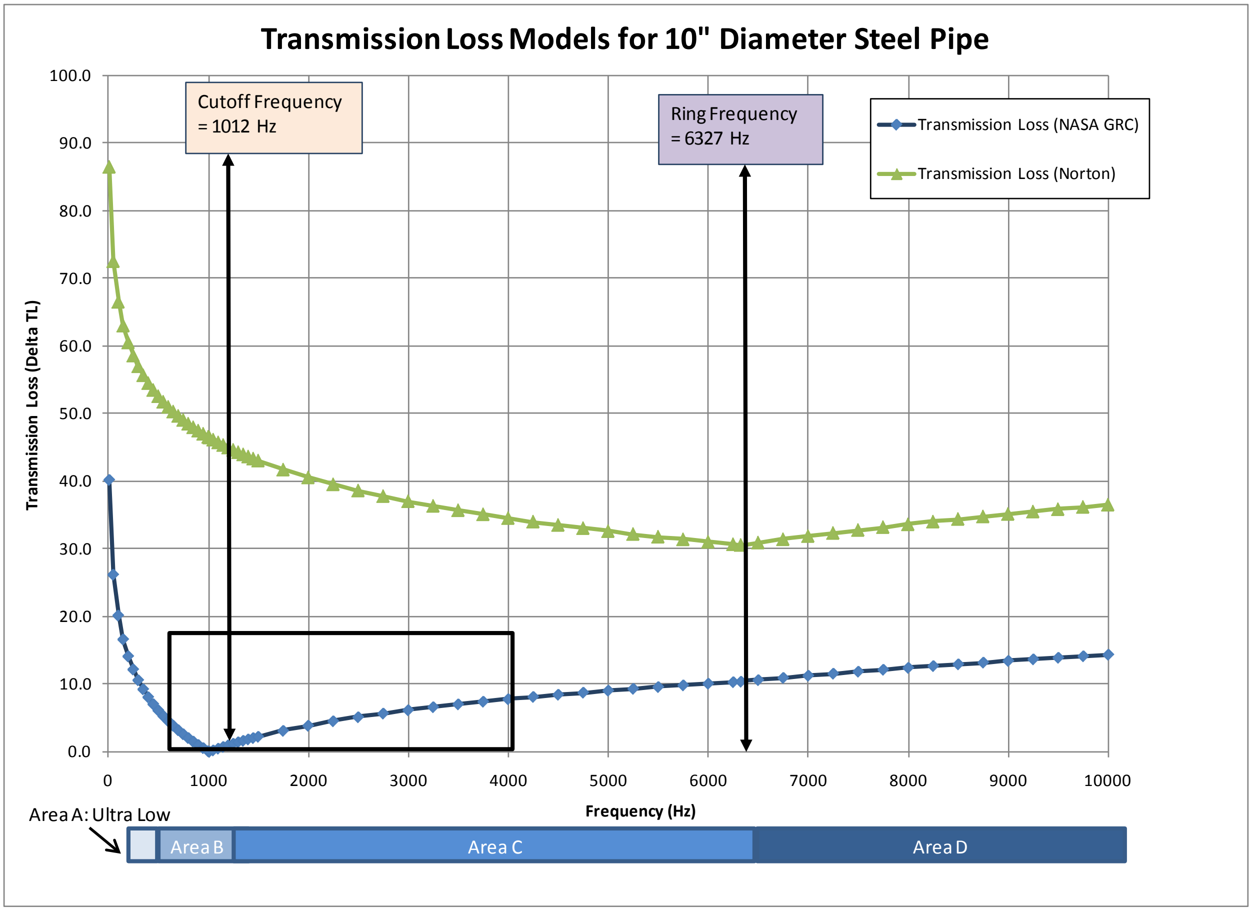

Figure 2-3. Transmission Loss as a Function of Ring Frequency and Cutoff Frequency

According to Norton, in general, for steel pipe and gaseous flows, the transmission loss (TL) factor will fall into one of two categories [1] :

Below the ring frequency:

$$For\ f < f_r : TL = 10 * log\left[\frac{\rho_s c_s t}{\rho c d}\right] - 20 * log\left[\frac{f}{f_r}\right] + 16dB \qquad (Equation\ 2\text{-}3)$$

Above the ring frequency:

$$For\ f \geq f_r : TL = 10 * log\left[\frac{\rho_s c_s t}{\rho c d}\right] + 30 * log\left[\frac{f}{f_r}\right] + 16dB \qquad (Equation\ 2\text{-}4)$$

where:

f = frequency of interest for noise calculation, source noise frequency

[1] Reference: Norton, M.P. and Pruiti, A., "*Universal Prediction Schemes for Estimating Flow-Induced Industrial Pipeline Noise and Vibration*," Applied Acoustics, volume 33, pp. 330-331, 1991.

f_r = ring frequency of pipe, defined as the pipe wall speed of sound divided by pi*pipe diameter, ($f_r = c_L/\pi D$)
ρ_s = density of steel pipe
c_s = speed of sound in the pipe material (steel)
t =thickness of pipe wall
ρ = density of fluid in pipe
c = speed of sound of fluid in pipe
d = diameter of pipe wall

These transmission loss groupings effectively define the expected level of ambient noise for fully developed turbulent flow, apart from other in-pipe noise or structural borne noise generation. Turbulent flow noise will propagate with the velocity of the fluid and will decay. Alternatively, acoustic fields and noise created from acoustic resonances will propagate at the speed of sound and decay less with distance, compared to turbulent flow noise.

Other guidelines such as the International Electrotechnical Commission (IEC) for control valve noise have shown a dependency on the cutoff frequency and first coincidence pipe frequency. The IEC Technical Monograph 41 describes a transmission loss model where very low frequencies below the cutoff frequency are not even considered due to high transmission loss. This is contradictory to some of the data taken in the GMRC research at reciprocating compressor stations where a significant contribution is made by the low acoustic excitation of wavelengths in the axial direction of the pipe. However, the IEC technical monograph also shows a low point for transmission loss (high noise transmission) around the first coincidence of the pipe frequency. Control valve noise is typically above the ring frequency of the pipe and assumed to be the most significant noise source, calculated in the IEC standard by the strouhal correlation and valve diameter.

2.9 Acoustic and Pipe Vibration Noise Modeling

Noise due to Mechanism #3 requires a forced response analysis and a characterization of the acoustic and mechanical resonances in the system. This should only be performed when a resonance is possible due to a transient excitation source (like a reciprocating compressor or vortex shedding, which creates "pulsing" outputs of the pressure wave within the pipe). Resonances may be identified as high amplitude, distinct peaks which stand out in station noise survey data or based on dynamic pressure data taken within the fluid. Resonance may be avoided through careful design and a pulsation and FEA mechanical study in the design stage.

Noise due to Mechanism #4 is predicted based on the pipe wall vibration levels, which are strongly linked to radiation ratio. Radiation ratio has a frequency based dependency for structure borne noise. Maximum radiation (zero transmission loss) occurs when a strong coupling exists between the surface vibration and the gas flow particle velocities at the surface. The radiation ratio is also used to define the amount of noise propagation from the internal pipe flow to the outside of the pipe wall. Pipe wall vibrations are directly related to the radiation ratio, which can be estimated relatively well based on frequency content alone.

Pipe wall acceleration, *a,* is directly related to the square of internal pressure wave fluctuations (p^2) the pipe wall geometry and fluid content and the radiation ratio, by the following formula:

$$p^2 = \sigma {\rho_o}^2 {c_o}^2 \left(\frac{d}{2r}\right) \frac{a^2}{\omega^2} \quad \textit{(Equation 2-5)}$$

where:

p = pressure wave as root mean square (rms) value
ρ_o = density of fluid medium external to pipe (in this case, air)
σ = radiation ratio
ω = radian frequency
c_o = speed of sound of air external to pipe
d = mean diameter of pipe
r = radial distance from center of pipe to location where pressure wave is incident

2.10 Radiation Ratio

Radiation ratio is also a frequency dependent value and used to determine the amount of noise radiation from within the pipe. There are three frequency groupings which should be used to estimate radiation ratio depending on whether the noise source of interest is above or below the cutoff frequency for the fluid within the pipe and above or below the ring frequency of the pipe itself (materially defined):

Below the cutoff frequency of the pipe:

$$f < f_c: 10 * log[\sigma] = 10 * log\left[\frac{f}{f_c}\right] + 3dB \quad \textit{(Equation 2-6)}$$

Above cutoff and below ring frequency:

$$f_c \le f \le f_r: 10 * log[\sigma] = 3dB \quad \textit{(Equation 2-7)}$$

Above the ring frequency:

$$f > f_r: 10 * log[\sigma] = 10 * log\left[\frac{f_r}{f}\right] + 3dB \quad \textit{(Equation 2-8)}$$

where:

r = radiation ratio
f_c= cutoff
f_r = ring frequency

2.11 Acoustic Modes

Acoustic modes exist within any system as one, two, or three-dimensional modes for incidence and reflection of pressure waves. In a piping network, these are evident as one-dimensional planar waves traveling along the length of the pipe span and as transverse waves traveling across the pipe. For larger bodies, such as the internal passages of large compressors, scrubbers, and other vessels, other modes exist for the pressure waves, which are a combination of traverse and planar waves.

2.11.1 Planar Waves

Sound in general travels away from a source and decreases in magnitude. The decrease in magnitude is primarily associated with the wave covering a larger area, reducing the energy at any given point further away. For instance, a wave from a point source will have a given amount of energy one foot away from the source. The energy will be distributed through an area surrounding the point source (area of a sphere in an open environment). As the sound moves away, the same energy is distributed through a much larger area reducing the amplitude. Therefore, the sound amplitude at any point 40 ft away is much smaller than at a distance of 1 ft simply due to geometry. In addition, the amplitude of the wave is reduced by the damping in the media, but this effect tends to be much less important in a media such as air.

When the wave is guided through a pipe, the effect of distance away from the source is different than in open air conditions. Instead of spreading out through the environment, the wave is contained in the pipe. Therefore, the amplitude of the wave does not diminish strictly due to distance away from the source. The amplitude is reduced through damping in the media and friction losses in the pipe. Due to these effects, pressure wave fluctuations in a pipe, due to various sources, can travel a long distance without much loss in amplitude. Planar waves can travel in a wide range of frequencies with little restriction.

The other important aspect of a planar wave in a pipe is that the pulse does not directly contact the pipe wall except at bends and reducers or other obstructions. This aspect makes it difficult for a planar wave to couple with a pipe wall or shell mode of a pipe. A planar wave can cause pipe walls to expand and contract due to the change in pressure between the peaks of the pulsation, which will cause the pipe to exhibit a forced response at a defined frequency corresponding to the planar wave.

2.11.2 Transverse Waves

Transverse waves are different than planar waves in that they tend reflect from one side of the pipe wall to the other. These reflections tend to oscillate in patterns, with each pattern associated with a specific frequency. These internal acoustic modes can be predicted based on the internal fluid properties and the diameter of the pipe. Figure 2-4 below shows the first ten internal acoustic modes of a pipe. Transverse modes are more likely than planar waves to excite shell modes because the transverse waves are constantly making direct contact with the pipe walls. However, because the acoustic modes are associated with particular frequencies, these modes cannot travel down the pipe in frequencies lower than those associated with the modes.

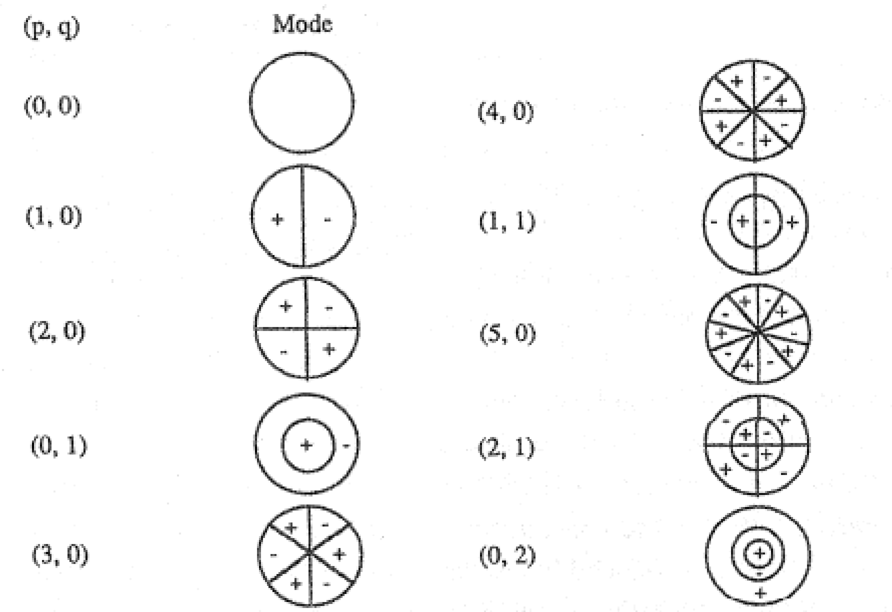

Figure 2-4. Internal Acoustic Modes for Pipe

Internal acoustic modes of a pipe are associated with turbulent flow within the pipe. For any given turbulent flow, acoustic modes will be present. These internal modes can increase in amplitude when excited by flow past a disturbance. The resulting increase in amplitude due to flow past a disturbance is quite difficult to predict. In fact predicting the internal flow field in general is a complex and difficult task. Empirical studies have found that piping disturbances causing increased amplitudes include flow into tee connections, flow through valves, flow through mitered bends, and flow past compressor blades. The same investigation found that radius bends do not tend to increase the amplitude of transverse waves.

The discussion of transverse waves has been limited to flow inside pipes. It is quite possible for acoustic modes to be set up in other configurations such as square ducts, within enclosures, and other geometry. However, these geometries are outside the scope of this report.

2.12 Shell Modes & Noise Coupling

Noise (fluid vibrations) can cause an object such as a pipe wall to vibrate. This vibration can be a forced response which tends to vibrate at relatively low amplitudes or the vibration can manifest itself as a resonant response, in which the noise related vibration of the pipe wall can be much louder. The sections below discuss the modes in which a pipe or panel can be excited at a natural frequency.

2.12.1 Pipes

Pipes have both circumferential and axial modes which can be excited. Circumferential modes distort the cross section of the pipe and are in general independent of pipe length. Examples of circumferential pipe modes are shown in Figure 2-5. Circumferential modes tend to have a frequency response that is quite higher than axial modes. Circumferential modes tend to be excited by internal acoustic modes. In fact, if one compares the modes in Figure 2-5 with those in Figure 2-4, it is easy to see how the pressure variations in the internal acoustic modes line up with the pipe wall deflections of the circumferential pipe modes. Internal acoustic modes will resonate and cause noise due to the pressure wave itself, while the circumferential pipe modes will couple with the internal acoustic modes to cause noise from the pipe wall vibrating. (The “p” mode in Figure 2-4 can couple with the “n” mode in Figure 2-5, for example.) However, for a coincidence to occur between the “p” and “n” modes, the frequency of the two modes must also be similar. For instance, if the 3rd “p” mode has a response at 1,000 Hz, but the “n” mode is at 2,000 Hz, the resulting response is expected to be quite low.

Axial modes extend down the length of a pipe segment and are highly dependent on the pipe length or clamp spacing. For long spans of pipe, the axial mode can have a low frequency response (< 100 Hz). The axial mode is not excited by transverse modes. However, the axial and circumferential modes are not completely independent and pipe segment length or clamp spacing can have an effect on the frequency at which the circumferential modes respond. In general, the response can be simplified by calculating the circumferential modes when estimating the frequencies at which a pipe wall may tend to vibrate.

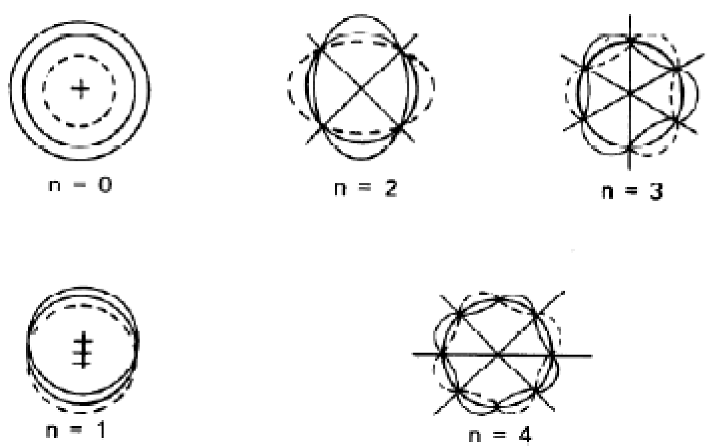

Figure 2-5. Circumferential Node Patterns of a Pipe

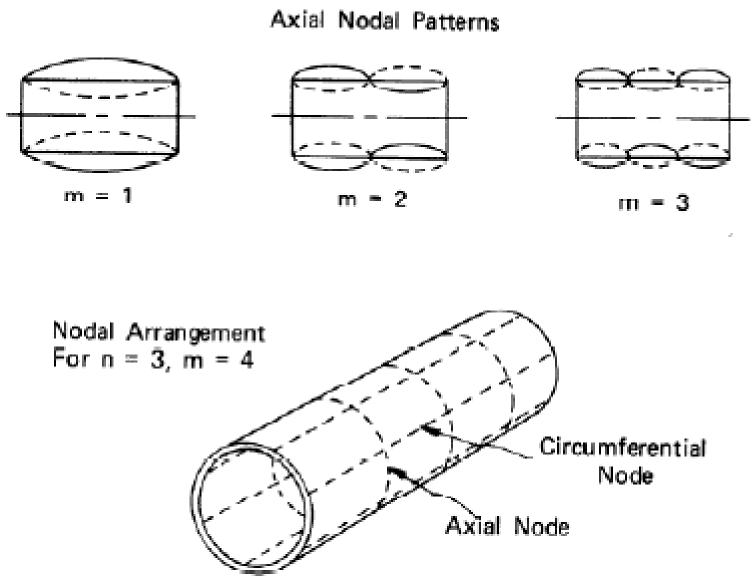

Figure 2-6. Axial Node Patterns of a Pipe

2.12.2 Panels

Panels such as walls of metal enclosures, walkways, skid plating, and other metal sheeting or plates common at compressor stations also have shell modes. The response of a panel and how it produces noise depends on whether an acoustic or a mechanical resonance has been excited. Mechanical excitation will produce a radiated sound related to the resonant panel modes. The frequency of excitation can be above or below the critical frequency. If acoustic excitation occurs and sound waves interact with the panel, the resonant frequencies will transmit noise above the critical frequency. Below the critical frequency, if sound waves are incident on the panel, the non-resonant modes of the panel will transmit the majority of the sound. Sound transmission characteristics depend on the material stiffness, damping, and mass unless the acoustic impedance on either side of the panel is distinctly different. In that case, the impedance mismatch between the two fluids dictates the sound transmission.

2.13 Transmission Loss

Modeling acoustic modes, pulsations, shell modes, mechanical damping, and structure/fluid coupling would be required to predict both noise due to structural movement and structural movement due to noise. This modeling effort is incredibly complex and often not very practical. To overcome these limitations, the idea of transmission loss (TL) has been introduced. Transmission loss is a simplification of the dynamics that occur in fluid structure interaction that can be reasonably applied to specific cases to predict noise. These TL models can be based on statistical methods or empirical data. Transmission loss models are used to determine the drop in power of the noise from one side of a surface to the other. Therefore, to predict the noise on the outside of a pipe (or panel), the noise inside the pipe must already be known. Typically TL models are used in industry to predict noise from piping.

2.13.1 Pipes

Transmission loss predictions are based on the diameter of the pipe, pipe thickness, pipe material, and fluid inside and outside of the pipe. The internal pulsations (transverse modes) must be predicted prior to using a pipe TL model. Transmission loss models for pipes also tend to include the "ring frequency" which is typically defined as the first mode (breathing/hoop mode). In most TL models, the lowest transmission loss occurs at or close to the ring frequency. A low TL means that more noise can escape through the pipe wall and into the environment. The TL loss will tend to increase for frequencies larger than the ring frequency and those smaller than the ring frequency.

2.13.2 Panels

Transmission loss through a panel is dependent on the frequency range of interest and the material composition of the panel. Fundamentally, the transmission loss is related to the ratio of transmitted to incident sound intensity. The characteristics will change if the panel is double-walled or if the material of the panel is not homogenous. Transmission loss will be controlled more by stiffness of the panel at low frequencies. Above the first fundamental frequency of the panel, the transmission is controlled by the resonances but mass changes will help to increase the transmission loss.

3.0 NOISE MECHANISMS

The basic noise mechanisms for compressor station piping noise are described below.

1) Straight Pipe, Turbulent Flow (Noise Mechanism #1): Turbulent flow in straight runs of pipe will produce characteristic broadband noise at frequencies between 500-10,000 Hz.

2) Added Turbulent Noise (Noise Mechanism #2): Disturbances such as elbows, bends, tees, and valves (gate and butterfly in particular) will cause more noise and specific frequency ranges to be excited compared to straight fully developed pipe flow. In this area, considerable research has been done to characterize the general sound pressure spectra inside the pipe from typical disturbances.

3) Excitation of Mechanical and Acoustic Resonances (Noise Mechanism #3): This is most commonly turbulence excitation of mechanical resonances or excitation from a reciprocating compressor coincident with an acoustic resonance. Broad high frequency (500-10,000 Hz) energy from turbulent flow often excites acoustic or mechanical resonances in a piping system. The resonant condition will cause one frequency to stand out and will enact a more distinct "tonal" pitch to the noise. The noise level can be very high (over 100 dB) at a single frequency. Acoustic resonances can be caused by propagation of either a length mode (caused by planar waves) or transverse waves across the pipe, at higher orders. Mechanical resonances are numerous and depend on the extent of tees, pipe fittings, couplings, and the types and locations of pipe restraints. Acoustic-induced vibration issues with pressure relief valves and station blowdown lines will often experience this type of noise issue.

For compressors with pulsating flows, the compressor will excite certain length resonances and noise will travel as planar waves in the piping. These are different from turbulent excitations. Acoustic energy will be created by the mechanical work of the gas at a resonant condition. This noise will also be transmitted through the pipe wall, similar to turbulent flow or flow-induced acoustic resonances. However, the excitation mechanism is in this case from the compressor or other forcing function. This type of noise will be at lower frequencies typical of compressor primary excitation orders (5-200 Hz).

Excitation of resonances may also occur due to the vortex shedding caused by strouhal effects, which will occur as flow passes by closed ends or piping stubs or inserts in the flow stream. Although the excitation mechanism has changed, the noise creation mechanism is the same (excitation of an acoustic or mechanical resonance).

4) Pipe Wall Vibration (Noise Mechanism #4): The vibration of the pipe wall will generate a separate noise level apart from the flow in the pipe. The magnitude will be based on the movement and frequency (for example, 1 inch per second at 1,000 Hz will generate more noise than 1 inch per second at 500 Hz.) Piping will tend to vibrate based on the level of flow or due to compressor excitation of a mechanical resonance. The typical range of frequencies for these types of resonances are 100-500 Hz.

However, structural born sound, transmitted from the structure itself, such as restraints or skid frames can be a secondary source of vibration noise. Structural born sound will introduce additional noise which is often at the frequency of the excitation mechanism (such as cooler or pump operating speeds) and transmitted due to the structural connections to the pipe.

3.1 Pipe Noise: Turbulent Flow Induced Noise

The following section provides a summary of Noise Mechanisms #1 and #2. These two noise mechanisms are strictly related to the noise inside the pipe and how it is transferred to the external surface. Noise Mechanisms #1 and #2 characterize noise created by turbulent flow in piping systems, apart from any excitation of resonances or structural noise.

3.1.1 Straight Fully Developed Flow

Broadband noise is created by fully developed turbulent flow in a pipe. The noise level is directly related to the level of flow turbulence, creating pressure fluctuations along the pipe wall in the boundary layer. As such, flow-induced noise will increase with flow velocity and turbulence level. Although this is simplistically regarded as noise which propagates in the flow direction, there are actually two pressure fields along the wall of the pipe – the turbulent boundary layer which will move with the fluid and the acoustic field produced by the fluid fluctuations, which propagates at a higher speed based on the speed of sound in the fluid (and decays less rapidly than noise produced by turbulent pressure fluctuations). Both noise sources rarely cause a "noise problem" for a compressor station, considering only the straight pipe with constant area. It is the area reductions, bends, elbows, and tees which will add noise and turbulence, often exceeding the 90 dBA, 8-hour working limit.

3.1.2 Minor vs. Major Disturbances

Disturbances in the flow stream, area contractions, and expansions and bends in the piping system will greatly increase the acoustic pressure fluctuations and the hydrodynamic effects. These added internal noise mechanisms can also lead to additional impingements on the pipe wall, leading to higher wall vibration noise. So, the added noise contribution due to disturbances is not just a contribution of the internal pressure fluctuation within the pipe. Characterizing the noise (amplitudes and frequency content) for each distinct type of noise source is one of the most critical steps in accurately predicting the overall noise content from piping systems. Most of this design data used in noise prediction methodologies, such as the Norton method, is based on empirical relationships extracted from experimental data. A brief summary is provided in this section.

3.1.2.1 Area Expansions/Contractions

Area reductions in the piping system will cause the wall flows to accelerate and create higher shear velocities. Most of the turbulent effects will decay rapidly and not propagate more than 10-20 pipe diameters. However, there is an additional acoustic field which is produced by the turbulence, which can propagate further distances and lead to higher external noise caused by area expansions and contractions far upstream. Area expansions can be severe as well in terms of the resulting pressure fluctuations in the boundary layer.

3.1.2.1.1 "Rules of Thumb" for Design

The pressure fluctuation (rms) of wall pressure is related directly to the area enlargement, where values of d/D greater than 0.3 ($d/D > 0.3$) tend to produce more severe pressure fluctuations within the fluid and resulting external noise, especially close by the disturbance. As a rule of thumb, within four pipe diameters of an area change, the sound level will be dominated by the turbulence produced by the area change. After just four pipe diameters, sound will be dominated by planar sound waves in most cases. Area expansions tend to cause more internal noise than area contractions.

3.1.2.2 Pipe Elbows

Piping elbows tend to produce a much larger level of turbulence and resulting wall pressure fluctuations for mitred bends compared to radiused bends. For normal radiused bends at 45 or 90 degrees, the effect of additional internal pressure fluctuations is very small compared to undisturbed pipe flow. Mitred bends of either 45 or 90 degrees or other geometries producing similar flow patterns will produce notably higher levels of internal noise. Additional pipe wall noise (vibration levels and radiation from the pipe wall) are also more pronounced for 90 degree mitred bends.

3.1.2.2.1 "Rules of Thumb" for Design

The effect of increased turbulence and wall pressure fluctuations is more pronounced at low frequencies and for 90 degree bends (greater than 20 dB increase) compared to 45 degree bends (about a 4-10 dB increase). Frequency range of concern for the higher dB levels will shift as a function of pipe diameter based on the strouhal equation. A strouhal value of below 4 or 5 is a good demarcation for increased db levels ($4,5 = St = a*w/U$) where a = pipe diameter, w = radian frequency, U = pipe velocity, and St = Strouhal number. For example, for velocities of 50 ft/sec, the higher dB levels can be expected below the target frequency $f\ (Hz) = 380 / a$, where pipe diameter is in inches. This means that smaller pipe diameters will cause more increase in sound due to mitred bends.

3.1.2.3 Valves with Internal Obstructions

Ball valves or other valve geometries with very little flow obstruction will not add to the internal sound levels. Butterfly valves and gate valves with notably different flow paths compared to straight pipe are the two exceptions for valves.

Also in general, for any obstruction (small fittings and flow-through areas within the valve passage ways), flow can become choked which will prevent propagation of a pressure wave from the downstream to the upstream side. This is effectively and anechoic termination point. Downstream noise levels will be produced by planar waves and downstream piping vibrations resulting from acoustic waves, but these will be completely unrelated to the upstream system.

3.1.2.3.1 "Rules of Thumb" for Design

Of all valve types, butterfly valves or similar valve geometry types will cause the most significant increase in pipe wall acceleration and acoustic power radiation, as much as 30 to 40 dB more than undisturbed turbulent pipe flow. These estimates are for a butterfly valve in a fully open position. Gate valves at 2/3 open or fully open will have a 10-20 dB increase in the primary frequency range. Valve noise will tend to concentrate in the range of 1,000-3,000 Hz but 10 dB increases below 1,000 Hz are also possible. The closed or partially closed position for either gate valves or butterfly valves will cause additional radiation and noise increases beyond the stated estimates above.

3.1.3 Role of Transverse Waves

Turbulent flow induced noise is initiated by high order acoustic modes within the pipe. These higher order modes occur in the cross sectional plane of the pipe, creating pressure differences on the pipe wall surface (changes in pressure occur between pipe walls). In order for these modes to propagate down the pipe, the frequency must be above the cut-off frequency of the mode. High order acoustic modes are different from plane waves which vary in pressure down the length of the pipe. (Pulsation studies are typically exclusively concerned with plane waves as compared to traverse waves which are not within the range of reciprocating compressor fundamental excitation orders.)

High order acoustic modes can excite circumferential modes in the pipe wall (coincidence). Both the mode shapes and frequencies of the internal acoustic modes and pipe shell modes must be similar for a coincidence to occur. Frequencies for the acoustic modes depend on pipe size and fluid properties, while the shell mode frequencies depend on pipe size, thickness, and material. The coincidence of the acoustic and pipe modes determine the level of noise outside the pipe.

3.1.4 Role of Planar Waves

Planar waves and the additional noise created by excitation of these are actually not as often induced by turbulent flow due to the lack of frequency correspondence. Planar waves tend to occur at longer wavelengths so the frequency content is often below 100 Hz for typical piping systems with piping lengths on the order of 10 ft or more. However, some piping systems will create turbulent flow fields which create acoustic fields in the process, allowing planar waves to propagate. These tend to be at lower overall sound levels however. This effect combined with the acoustic field for the planar waves will complicate the noise spectra and add uncertainty to any internal pipe noise modeling process which does not consider both the turbulent flow field and the planar wave reflections. Planar waves will be a noise source when a resonance is excited by a compressor or other transient condition.

3.2 Excitation of Mechanical/Acoustic Resonances

Apart from noise produced by the incident pressure waves within the turbulent boundary layer, standing waves within the fluid or pipe walls may be excited, causing a resonance phenomenon and leading to higher energy extraction and noise levels than the flow would normally be capable of producing.

3.2.1 Turbulent Excitation of Acoustic/Mechanical Resonances

Turbulent flow induced noise can be caused by any compression machinery reciprocating compressors, centrifugal compressors, and screw compressors. The velocity of the fluid, level of turbulence, and the diameter of the pipe are most important factors when calculating turbulence induced noise. When the turbulence creates excitation frequencies in the range of acoustic or mechanical resonances, an acoustic field is created within the fluid, allowing for the reflection of a standing wave, either within the fluid or nearby solid structures. Often times, this noise effect is also responsible for "acoustic induced vibration" effects where the turbulent flow excites acoustic or structural resonances, which can lead to fatigue failures due to high vibrations. This is a difficult noise level to predict since turbulent excitation is time-varying and will propagate with the fluid over a broad frequency range. Turbulent excitation will be most pronounced in the range of 500-2,000 Hz. This mechanism will almost always excite a traverse wave acoustically (not standing planar waves) due to the frequency correspondence. Mechanical excitation in this range is not uncommon for internal passages in valves or other disturbances with small mechanical components with resonances in the range of 300-800 Hz.

3.2.2 Role of Vortex Shedding

Vortex shedding will occur in any geometry where high flows pass over closed ends or stubs in the piping system or when gas particles encounter an obstruction in the flow stream and start to produce vortices directly. The vortex shedding will occur at the strouhal frequency and will then excite either acoustic resonances (planar waves typical of a closed end connection) associated with the stub connection or mechanical resonances. Mechanical resonance excitation from vortex shedding has been found to occur on baffled plates in heat exchangers, orifice plates, flow conditioning devices, and other insertions in the flow stream. The frequency range for mechanical excitation is usually from 100-400 Hz. Acoustic excitation from vortex shedding will be slightly lower, from 10-200 Hz.

3.2.3 Compressor Excitation of Resonance (Pulsation Related Noise)

The final mechanism for excitation of resonances is in the form of compression machinery which introduces a known pulsation into the fluid through the compression process. This type of excitation specifically occurs for reciprocating compressors and screw compressors. The pulsation related noise will be amplified when the compressor excitation corresponds to an acoustic resonance in the piping system. At resonance, the pressure wave frequency is exactly matched to the acoustic lengths within a chamber or piping segment. The wave reflects off the open or closed ends and amplifies in strength compared to the original pressure wave due to the resonant condition. Depending upon the level of damping and the number of reflections and pipe wall vibrations which result, the resonance can create loud audible noise which does not easily dissipate due to its low frequency content. This can be avoided through careful design of the acoustics within the system. The frequency range of concern tends to be less than 300 Hz for reciprocating compressors and between 100 to 500 Hz for screw compressors.

3.3 Vibration Induced Noise

Noise outside of the pipe can also be induced by the movement of the pipe wall. The pipe wall may be vibrating due to the flow inside of the pipe or excitation from mechanical sources. Regardless, the vibrations of the pipe wall themselves cause additional noise in the system. This noise component can be equal to or greater than the noise produced by the fluid alone. The amplitude of the pressure wave (noise) is directly related to the acceleration of the pipe wall through Equation 2-5 – provided previously, which requires an accurate value for the radiation ratio. Radiation ratio may be calculated from empirical relations such as those provided in Equations 2-6 through 2-8, which are frequency range dependent.

Determining the acceleration of the pipe wall due to internal acoustic modes is complex (if no experimental data is available). Acceleration levels depend on coincidence of acoustic and mechanical modes, damping, and fluid properties. In addition, the first longitudinal mode (ring frequency) in the pipe affects the efficiency of the transmission of noise through the pipe wall. Typically a transmission loss model is used to predict the amount of noise that escapes from inside the pipe to the external environment.

Vibration noise is fundamentally the noise due to the pipe wall acceleration, but this may be due to mechanical excitation (a pump or compressor nearby or mechanically coupled to the pipe), from small boundary layer pressure fluctuations within the fluid due to turbulence, or from shell mode vibrations for larger structures, typically produced by traverse waves. The frequency of the vibration affects the amount of noise transmitted externally. This noise generation/transmission effect is more pronounced on a weighted dB scale, where the higher frequencies near 1,000 Hz have more effect on the overall sound power level on the decibel scale, as shown in Figure 3-1.

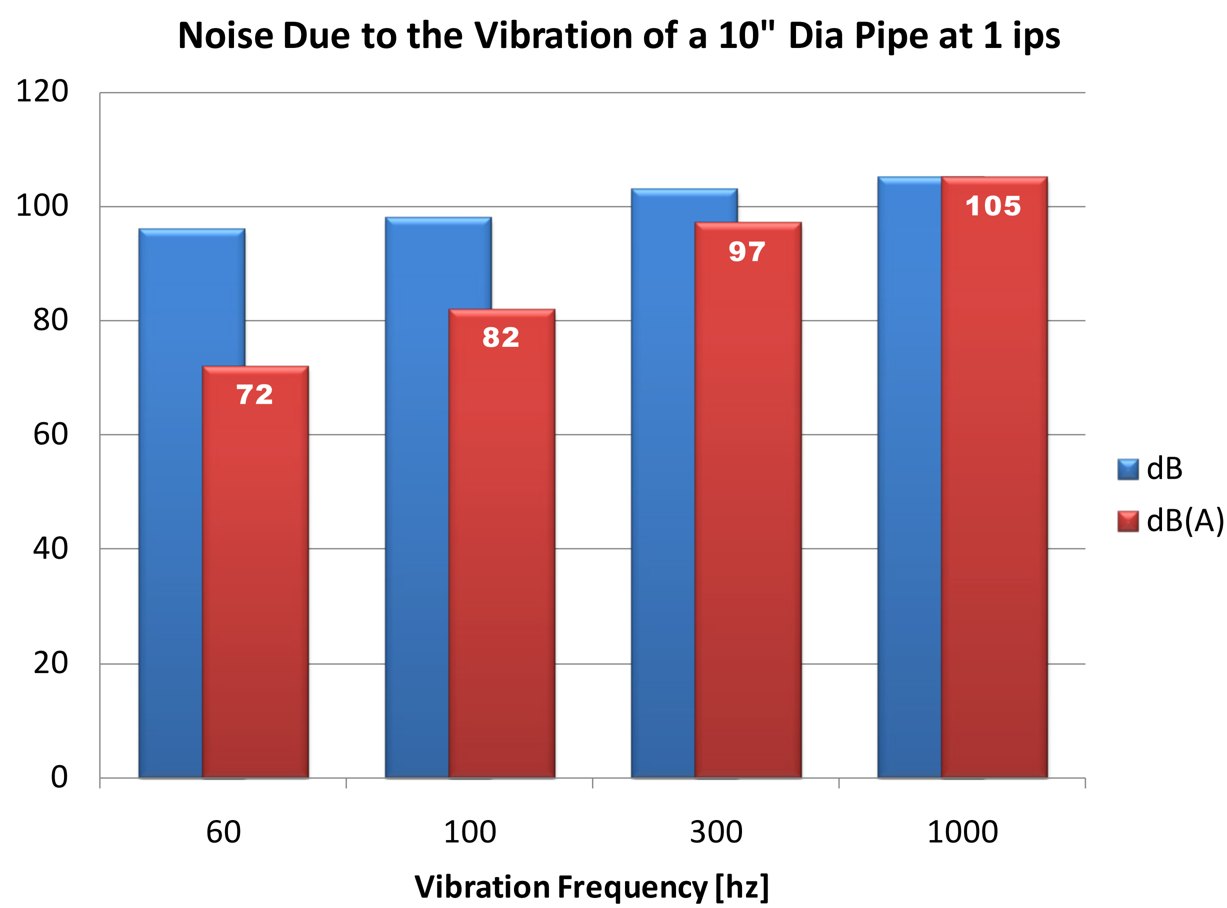

Figure 3-1. Noise due to Vibration of a 10" Diameter Pipe, Vibrating at 1 inch-per-second

4.0 NOISE PREDICTION METHODS

4.1 General Approach

The following subsection outlines the general approach that must be taken to quantify the noise associated from in-pipe flow due either to turbulence, vortex-shedding, or pulsating flow. Once the noise in the pipe is characterized throughout the piping system, a projection and superposition of noise sources must be made for the "in-pipe" region. The transmission loss is then considered through the pipe wall and should account for gas flows through pipe to ambient air.

Other methods exist for combining steps 1 and 2 to determine noise levels from non-empirical data. Specifically, pipe wall acceleration may be calculated by applying experimental strain gage or accelerometer data or through use of finite element methods to find pipe wall vibration values. Vibration levels may be combined with radiation ratio in order to calculate internal pipe noise (refer to Norton's Method 1 in Norton and Pruiti, reference 12). Another method uses an adaptation of these methods but estimates transmission loss through the pipe to modify the typical transmission loss using noise survey data. This method, Norton Method 3, is found to be least accurate due to a number of assumptions.

The method outlined herein is referred to by Norton as Method 2 and is the recommended approach for design stage analysis or analysis without experimental data available. Accuracy of this method for 500-10,000 Hz was found to be within 5 dB of measured valves for a gas turbine compressor station.

Note: this method has not been applied and combined with pulsating flow data to estimate noise at a reciprocating compressor station. This would be a necessary validation step to apply the method with confidence for noise very near a reciprocating compressor (within the manifold area) or in a case of acoustic/mechanical resonance. The method is applicable for piping outside the manifold area of a reciprocating compressor station. The flow chart of the three methods shows the similarity in the basic methodology in Figure 4-1 below.

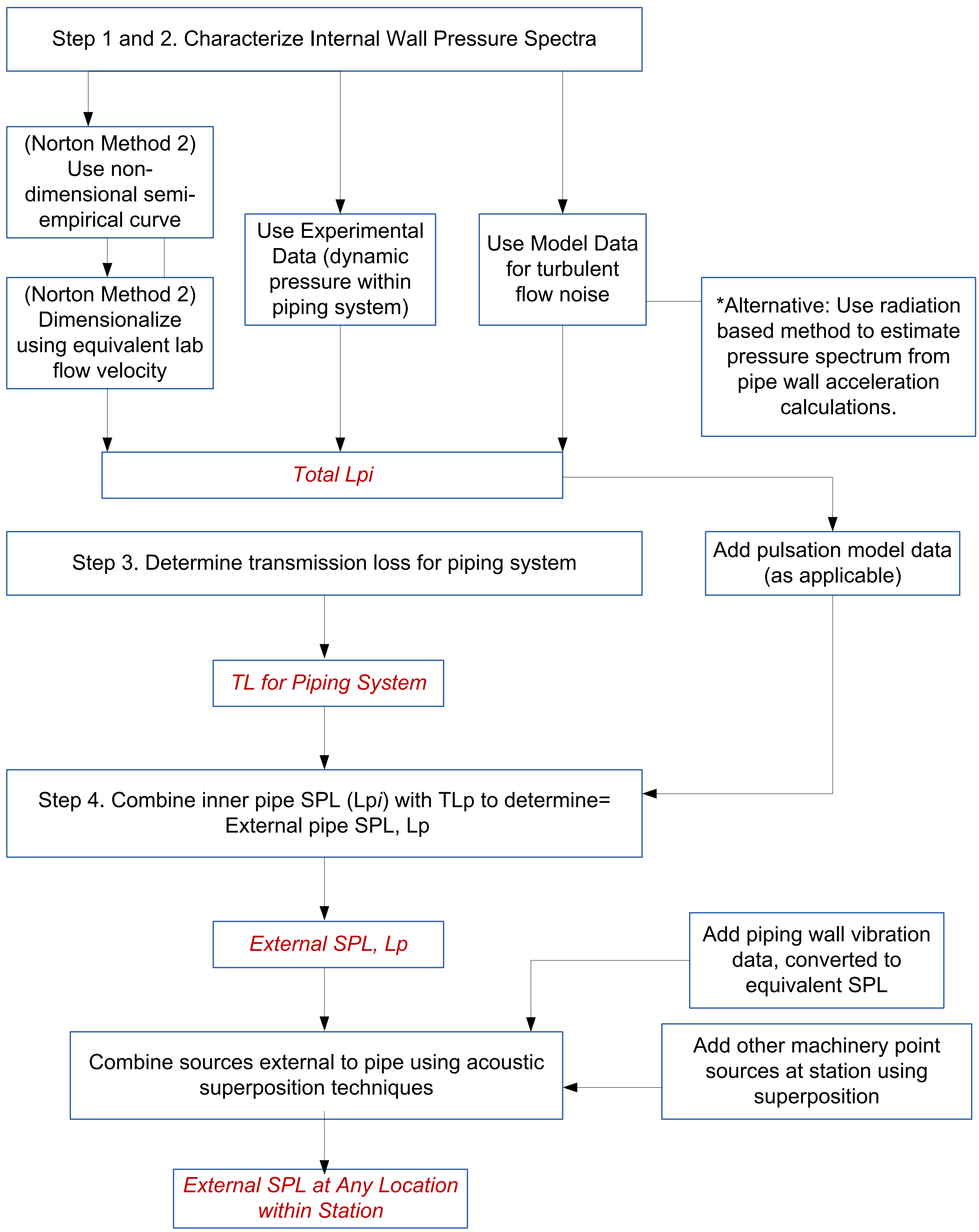

Figure 4-1. Noise Prediction Methods Methodology Flow Chart

4.2 The Norton Method 2, Semi-Empirical Data Approach

The Norton Method 2 approach uses semi-empirical data and can be combined with pulsation models for additional noise contributions in order to estimate in-pipe source noises as the first step. The transmission loss model is then applied to determine noise at the external surface of the pipe. Other superposition schemes are then used to determine overall station noise levels and identify high noise contributors.

4.2.1 Step 1. Selection of Appropriate Internal Noise Spectrum, Semi-Empirical

The first step in predicting the external noise of a pipe due to flow is to predict the internal noise field. This is a difficult task due to complexity and the randomness of turbulent flow. The Norton method sidesteps these difficulties by using a collection of empirical data to predict internal noise in the field based on the characteristic elements in the piping system.

The empirical data is from multiple flow velocities, pipe diameters, and noise sources. The noise sources investigated include mitered bends, radius bends, straight pipes, valves, and tees. The method is only applicable to steel pipes and gas flow.

Data from tests was converted to non-dimensional curves representing the internal pressure spectrum and converted to strouhal numbers and wall pressure spectra – see Figure 4-2 below. The Strouhal number is calculated using pipe diameter and internal flow speed, using the measured response frequency, based on Equation 4-1:

$$St = f * \frac{a}{U} \qquad \text{(Equation 4-1)}$$

where:

St = Strouhal number
f = frequency of pressure fluctuation
a = pipe diameter
U = pipe velocity

The wall pressure spectrum is calculated using internal flow (cubed relation), fluid density (squared relation), and pipe diameter.

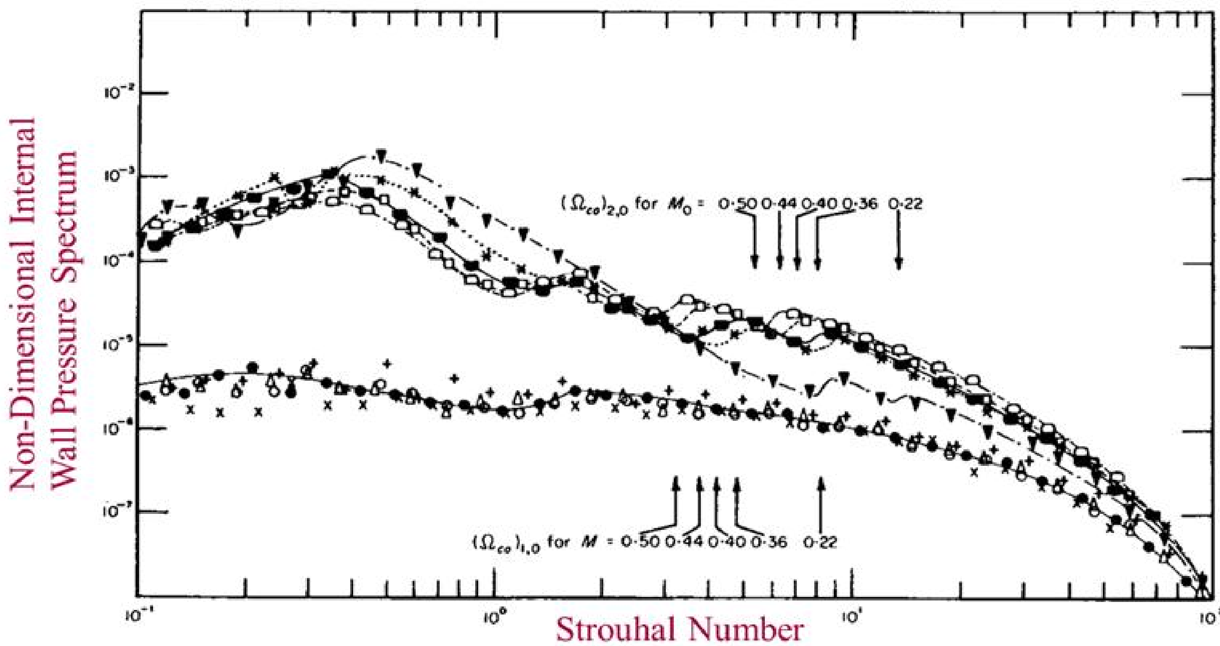

Figure 4-2. Example Non-Dimensional Plot Used by the Norton Method

The upper limit curve should be used for high noise configurations within 30 diameters of major noise sources from equipment or for other extreme piping system disturbances such as mitred bends, flow into a tee, through a valve, or through compressor blades. The internal acoustic modes are initiated and intensified by disturbances in the flow due to fittings, tee's, valves, and other discontinuities in the flow. Pressure spectra due to mitred bends and specific pipe fittings was found to be the highest. Fittings that cause significant disturbances include pipe tee's where incoming flow is split between the outgoing paths (but not past a "tee"), specific valve geometries, and compressor blades.

The lower limit curve should be used for other low noise configurations which do not fall into any of the above categories. The lower limit curve includes 90 degree elbows. Flow past a 90 degree radius bend was not found to significantly increase the magnitude of the wall pressure spectrum. It was also found that the magnitude of disturbances began to decrease after 10 diameters from the disturbance, and by 50 diameters the magnitude had decreased to a level that would transmit for long distances. Hence after 50 diameters downstream of major noise sources, the lower curve may be applied. Refer to Norton's paper, reference 12 for more data.

Note on appropriate selection of non-dimensional internal pressure spectrum: Two sets of curves are used in the Norton method. Disturbances tend to decay quickly dropping to a pressure level that can propagate for long distances. The proper internal wall pressure spectrum curve must be selected for the region(s) of interest for the calculation.

4.2.2 Step 2. Dimensionalize Pressure Spectrum for Specific Piping System

The non-dimensional internal wall pressure for the lab conditions is found using the curves in Figure 4-2. It must be converted to a "field conditions, dimensionalized" wall pressure by re-dimensionalizing. First, the equivalent lab flow velocities and densities are found using the equivalency equation.

The equivalent lab velocity is calculated using field and lab data. Equation 4-2 below provides the equation to calculate the equivalent lab velocity.

$$U_L = U_F \left(\frac{a_F}{a_L}\right)^{1/3} \left(\frac{\rho_F}{\rho_L}\right)^{2/3} \qquad \textit{(Equation 4-2)}$$

where:

UL = Equivalent lab velocity
UF = Field velocity
aL = Lab internal radius
aF = Field internal radius
ρL = Lab density
ρF = Field density

Using the equivalent lab velocity, actual lab density and actual laboratory radius, the field conditions dimensional wall pressure may be solved for each frequency point. Equation 4-3 shows the solved equation for the dimensional wall pressure spectra based on the equivalent lab velocity.

$$G_P = \Phi_p * \rho^2 * U^3 * \frac{a}{4} \qquad \textit{(Equation 4-3)}$$

where:

Φp = Non-dimensional wall pressure
Gp = Dimensional wall pressure
ρ = Internal density
U_L = Fluid velocity, calculated from equivalency Equation 4-3 above
a = Internal radius

Strouhal values should be dimensionalized from Figure 4-2 as well, using the equivalent laboratory velocity and the actual field geometry for the pipe diameter values, as shown below in Equation 4-4. At the completion of this step, the internal noise spectrum is predicted. This should be based on selection of the appropriate non-dimensional curve and dimensionalization based on equivalent lab velocities for the pipe.

$$St = f * \frac{a}{U_L} \qquad \textit{(Equation 4-4)}$$

where:

St = Strouhal number
f = frequency of pressure fluctuation
a = pipe diameter
U_L = pipe velocity for equivalent laboratory conditions

Prior to moving to the next step, if additional noise data is available for pulsating flows or vortex-shedding at specific frequencies based on a pulsation model or field data, this should be used here as well to increase the internal wall pressure spectra levels at appropriate frequencies. This data may be combined with the internal wall pressure spectra generated in the Norton method, which are mainly generated from empirical data for turbulent flow related noise.

4.2.3 Step 3. Calculation of Pipe Transmission Loss

At this point any number of transmission models can be used to predict the noise external to the pipe. However, the semi-empirical transmission loss (TL) presented in the Norton paper was found to match independent transmission loss data more closely than other available methods. This equation uses a linear relationship from the characteristic TL levels from industrial pipes:

For frequencies less than the ring frequency ($f < f_r$):

$$TL = 10 * log\left[\frac{\rho_s * c_s * t}{\rho * d * c}\right] - 20 * log\left[\frac{f}{f_r}\right] + 16dB \qquad \textit{(Equation 4-5)}$$

For frequencies greater than the ring frequency ($f \geq f_r$):

$$TL = 10 * log\left[\frac{\rho_s * c_s * t}{\rho * d * c}\right] + 30 * log\left[\frac{f}{f_r}\right] + 16dB \qquad \textit{(Equation 4-6)}$$

where:

TL = transmission loss
ρ_s = density of steel pipe
c_s = speed of sound in steel pipe
t = thickness of pipe
ρ = density of gas
d = pipe wall diameter
c = speed of sound in gas

Other transmission loss models calculate TL based on ring frequency and the first natural acoustic cross mode frequency. The transmission loss tends to be shaped like a "V" shape with smallest loss at or near the ring frequency. Pipe thickness, internal and external speed of sound, and internal and external density are all required for this calculation.

Transmission loss from the acoustic intensity definition above must be converted to a steel pipe specific TL based on the acoustic impedance difference:

$$TL_P \sim TL - 10 * log\left[\frac{\rho_o * c_o}{\rho * c}\right] \quad \textit{(Equation 4-7)}$$

where:

TL_P = Transmission loss for industrial pipe
TL = Typical Transmission loss from model Equations 4-5 and 4-6
ρ_o = density of pipe material
c_o = speed of sound of pipe wall material
ρ = density of internal fluid
c = speed of sound of internal fluid

After this step, the transmission loss and the internal sound power level and pressure fluctuation is known at each point within the piping system. As discussed previously in this report, other frequency based methods of predicting transmission loss specific to a piping geometry and given speed of sound may also be used for this step. These two parameters are combined in the final step.

4.2.4 Step 4. Prediction of Noise External to Pipe

The internal, dimensional wall pressure and transmission loss are combined to predict noise at the pipe surface. Additional loss due to propagation away from the pipe can then be calculated with conventional methods. Superposition methods may also be used to combine other significant station noise sources at various locations into an effective station noise map.

The equation for external radiated sound pressure level from the internal sound pressure and the transmission loss from the pipe is as follows:

$$L_P = L_{Pi} - TL_P + 10 * log\left[\frac{a_o}{r}\right] \quad \textit{(Equation 4-8)}$$

where:

L_p = External radiated sound pressure level
L_{pi} = internal sound pressure level
TL_p= transmission loss for pipe
a_o= external pipe radius
r = internal pipe radius

The method presented here will serve to estimate flow related turbulence noise produced by in-pipe flow. It should be combined with other fluid solutions to provide a complete depiction of the noise levels related to mechanical and acoustic resonances or vortex-shedding phenomena.

Conveniently, the method provides for these additional data inputs as a total superposition of noise sources within the "in-pipe" system at the second step. Transmission loss models can then be applied to predict external pipe flow noise with the Equation 4-8 provided above. As is noted in other discussions by Norton, the method works reasonably well to predict noise levels within 5 dB of measured sound pressure levels for 500-10,000 Hz. Additional data on the reciprocating compressor and screw compressor applications is warranted, particularly for the 0-500 Hz frequency range which is common to pulsation/mechanical resonances.

5.0 EXAMPLE COMPRESSOR STATION NOISE DATA

The following section provides examples of typical noise data recorded at various compressor stations and piping systems to illustrate the report's key points. The data shown is a collection of published work in the open literature and the GMRC noise research field test data. The general trend in most noise data from piping systems without strong characteristic resonances will follow a classic "v-shape" which is an inversion of the transmission loss model shown previously (Figure 2-3). A certain maximum noise level will be reached around 1,000-3,000 Hz, which corresponds to the minimum transmission loss.

This type of inverted v-curve is shown in Figure 5-1 and Figure 5-2. The data was recorded by Norton (reference 12) in assessing the validity of the method 2 approach to noise prediction. Overall, the method shows a good correlation with the field data. The data also corresponds well to typical noise spectra for noise due to turbulent flow effects, since the data was taken at a gas turbine compressor station where pipe flow velocities would be relatively high and pulsations in the flow stream would be small.

Departures between the measured and predicted data are notable at:

- **Low frequencies (0-300 Hz)**. Primarily due to Norton method basis in turbulent flow related noise. The actual measured noise levels are higher due to possible mechanical resonances in this region. For a reciprocating compressor application, the method will also tend to under predict the sound pressure levels, unless the inner pipe pressure spectra is compensated for pulsation related noise.

- **Distinct peaks at 800 and 1,200 Hz**. These more distinct measured values are likely acoustic resonances excited by flow induced effects. The Norton method does not include high noise inflations due to resonance effects but can be adapted to include these by superposition of this noise within the inner pipe region, prior to application of a transmission loss model.

- **High frequencies (above 8,000 Hz)**. In this region, the curve tends to fall off because most all noise will be above the ring frequency of the pipe. The discrepancy between actual and predicted noise could be due to inaccuracies in the transmission loss model or the semi-empirical curves recorded for general cases. The variation is fairly small (within 10 dB) and not likely to affect overall noise level since the curve declines in this high frequency region.

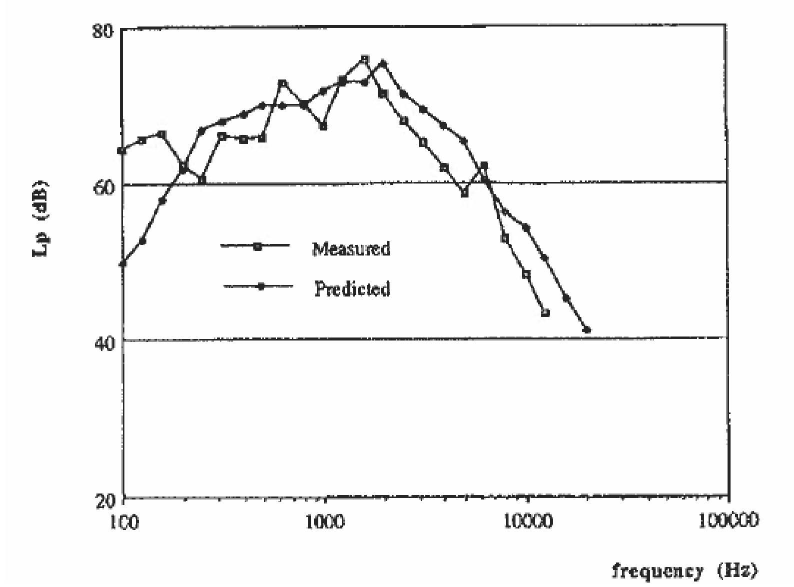

Figure 5-1. Gas Turbine Compressor Station Test Point 1: Measured vs Predicted Data from Norton (Ref.12), Predicted Data Utilized Norton Method No. 2

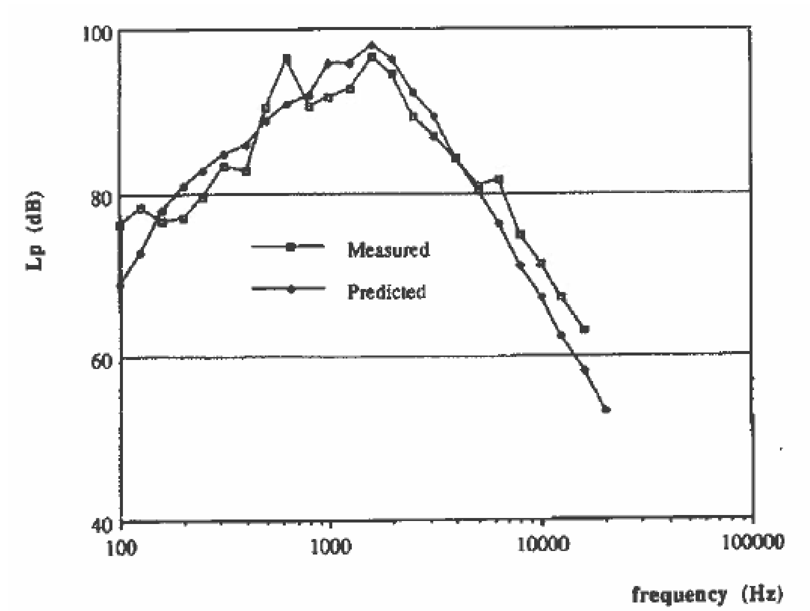

Figure 5-2. Gas Turbine Compressor Station Test Point 2: Measured vs Predicted Data from Norton (Ref.12), Predicted Data Utilized Norton Method No. 2

In contrast to the gas turbine compressor station data which was dominated by the turbulent flow type noise, the reciprocating compressor station piping noise is composed of significantly higher noise from the lower frequencies. For the reciprocating compressor station, the acoustic and mechanical resonances and piping vibration noise tends to dominate the noise spectrum instead of flow turbulent noise.

Figure 5-3 shows the data recorded at a high speed (800-1000 RPM) reciprocating compressor station. The engine driven (3550 hp) compressor was operating in single stage mode at the high end of the speed range for the noise data recording. The pulsation-induced noise shows the highest amplitudes of around 94 dB. Other peaks are notable in the range from 0-500 Hz. In this range, the noise peaks are most likely caused by pipe vibrations due to excitation of pipe axial and shell modes. After 500 Hz, there is a general decline towards 50 dB levels with other minor peaks throughout. The turbulent flow noise in this region is notably smaller in SPL than the Figure 5-1 and Figure 5-2 data due to reduced flow velocities.

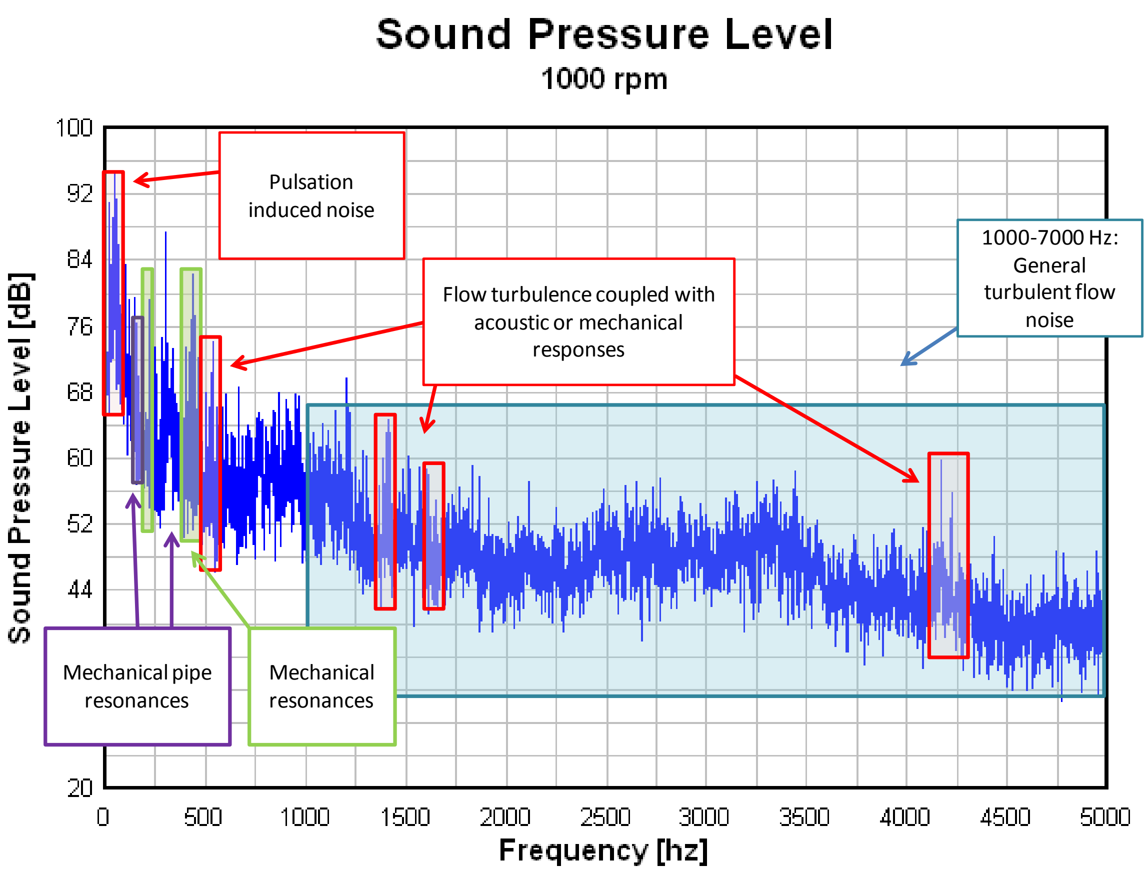

Figure 5-3. Reciprocating Compressor Station: Measured Noise Data near Discharge Piping System

The final example shows SPL level readings for a blow-down valve and downstream piping system where noise data was recorded at varying distances from the valve. The overall decibel levels were close to 120 dBc – see Figure 5-4. However, frequency spectrum data showed the dominant noise source to be at a single frequency of 593 Hz. Unlike the general turbulent flow noise where the noise band was broadly distributed, the data from the valve indicated flow excitation of a strong acoustic resonance. Further acoustic and mechanical models revealed that the turbulence levels had excited a higher order acoustic mode in the piping system which was coupled to a mechanical mode. The noise was thus a result of pressure wave reflections at resonance and pipe wall vibrations due to the mechanical response. Unlike the previous two examples, this noise level could be easily reduced through attenuation of the acoustic response and decoupling of the mechanical response at the problem frequency.

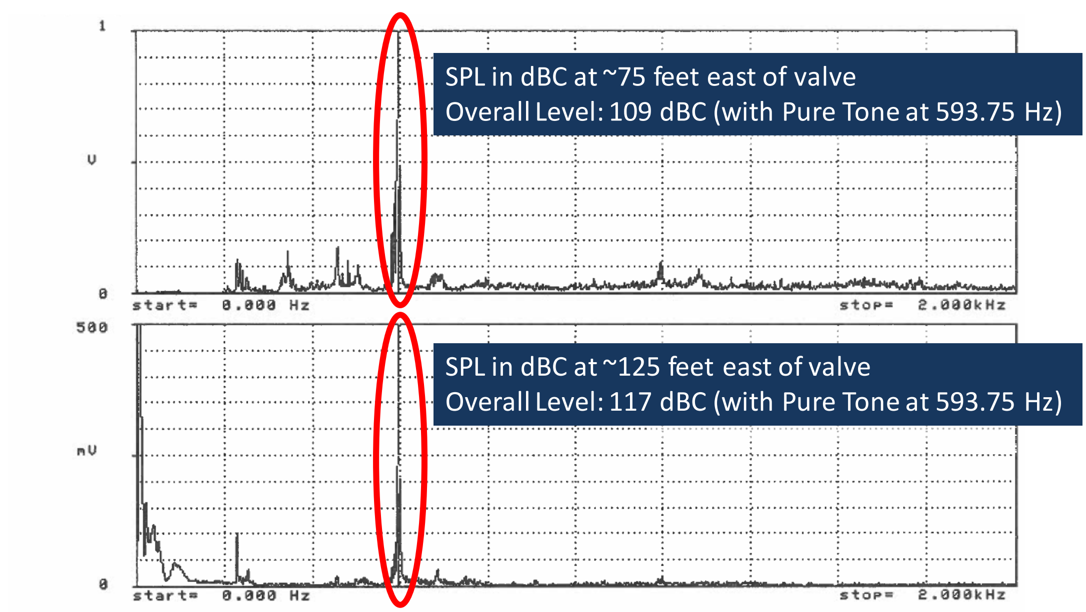

Figure 5-4. Example of Noise Due to Acoustic/Mechanical Resonance in Piping System

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