Solving Acoustic-Induced Vibration Problems in the Design Stage

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Failures of piping in the hydrocarbon industry represent a potential for catastrophic accidents in terms of both lost lives and dollars. It has been reported that more than 20% of the piping failures in the UK sector of the North Sea were due to piping vibration and fatigue failures. It is clear that acoustic-induced vibration is a serious risk since a single failure will often shut down a facility for hours or days, resulting in lost production at a minimum. In the 1980s, nine AIV failures were documented and used to develop criteria. One resulting criteria system plots the sound power level versus the diameter of the pipe; another plots the sound power level versus the ratio of the nominal diameter to the wall thickness of the pipe. Safe design curves are then drawn through the data. One of the simplified approaches seems to have misinterpreted the original data and now has a safe design curve where a failure has occurred. Since the 1980s, there have been many more failures, but the documentation of these failures has not been shared with the engineering community. This article discusses the existing criteria, presents CSTI's past approach to the challenge of designing to avoid potential AIV problems, identifies a new failure curve as a function of the ratio of the mean diameter to the thickness squared, and proposes a new safe design curve.

In 2010, Marsh Energy Practice[1] presented a summary of the property damage losses in the hydrocarbon industry. There is no record that any of these losses were actually caused by fatigue due to acoustic-induced vibration (AIV); however, it is highly probable that pipe fatigue was the cause of some losses. The total dollar value of these losses, adjusted for inflation, is greater than 10.7 billion USD (2009). Tragically, more than 200 people were killed from 1975 to 2009.

Earlier, the Energy Institute discussed the link between pipe fatigue and the release of hydrocarbons from some North Sea platforms:[2]

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

This article summarizes an approach to identifying and solving acoustic-induced vibration problems in the design stage for the oil and gas industry.

Background

Pressure-reducing devices, such as relief valves, control valves, and orifice plates, can generate high levels of high-frequency acoustical energy downstream of the valve. The sound power level is a function of the pressure drop across the device, the upstream pressure, the mass flow through it, the molecular weight, and the temperature. This acoustical energy propagates downstream of the valve where the resulting vibration has caused failures due to fatigue, sometimes in just a few hours of operation.

The sound power level L_w of the valve or orifice plate can be calculated as follows:

$$L_w = 10\,log\left[M^2 * \left(\frac{P_1 - P_2}{P_1}\right)^{3.6} * \left(\frac{T}{W}\right)^{1.2}\right] + 126.1 + K \qquad (1)$$

where:

Based on a paper presented at Inter-Noise 12, the 41st International Congress & Exposition on Noise Control Engineering, New York, NY, August 2012.

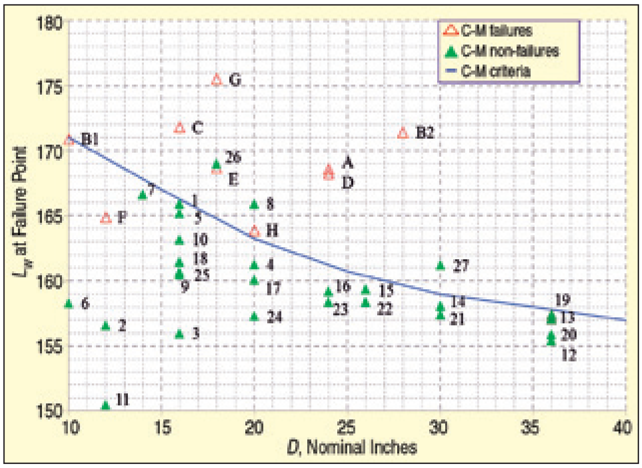

Figure 1. Carucci and Mueller data and criteria curve.

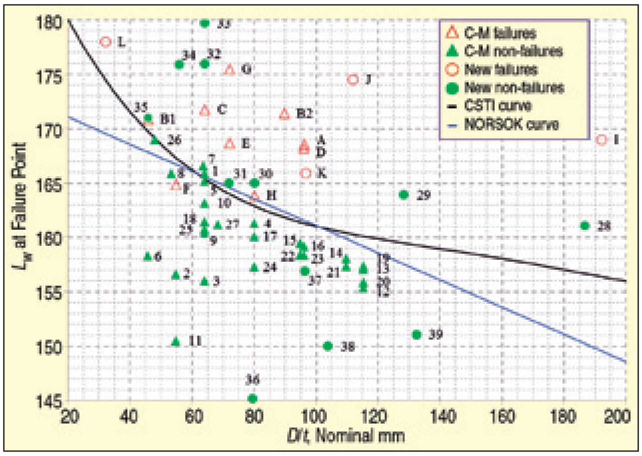

Figure 2. AIV data and criteria curves.

L_w = sound power level in dB re 10^{-12} watts
M = mass flow in kg/sec
P_1 = upstream pressure in kPa absolute
P_2 = downstream pressure in kPa absolute
T = temperature in Kelvin
W = molecular weight
K = zero for nonsonic flow and +6 for sonic flow conditions

History of Pipe Failures Due to AIV

In the 1980s, Carucci and Mueller[3] (C-M) investigated failures of thin-walled piping. They reported nine failures. Their work also lists 27 situations that did not fail. Figure 1 presents the C-M data with the reported L_w plotted as a function of the nominal diameter D. The nine failures are noted with letters A-H, and the 27 non-failures with numbers 1-27. The blue curve is their safe design criteria curve and is valid for non-continuous operation for a total of not more than 12 hours.

Eisinger[4] plotted the C-M data as a function of the ratio of the nominal diameter to the wall thickness D/t rather than D. Although Eisinger indicated in the text that internal diameter was used, the table clearly shows the use of the same nominal diameter as noted by C-M. Also, Eisinger plotted points H and 27 at substantially higher L_w than the original C-M values (perhaps due to calculating the power at the valve rather than at the point of failure). McMahon[5] has confirmed that point H is plotted correctly in the graph

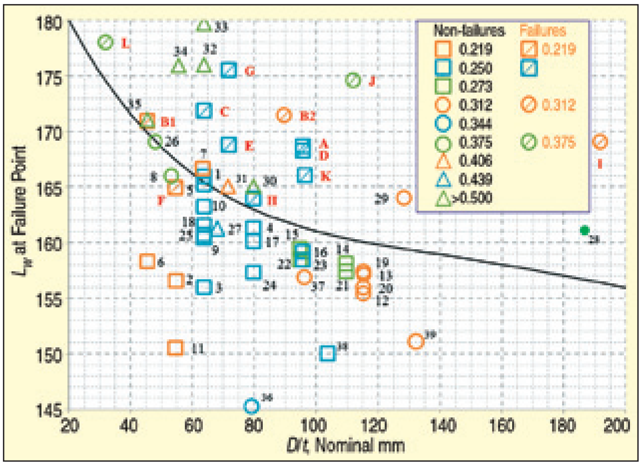

Figure 3. AIV data from Figure 2, with CSTI curve and pipe wall thicknesses.

in the C-M paper.

A number of oil companies have shared some additional failure and non-failure data, which are plotted in Figure 2 as a function of the nominal diameter divided by the thickness along with the original C-M data. The only information on the new data points is the L_w and the D/t ratio. The black curve is the CSTI design criteria curve:

$$L_w = 4.2914\times10^{-8}\left(\frac{D}{t}\right)^4 - 2.8195\times10^{-5}\left(\frac{D}{t}\right)^3 + 0.006781\left(\frac{D}{t}\right)^2 - 0.7549\left(\frac{D}{t}\right) + 192.6125 \quad (2)$$

This equation, patterned after C-M and developed by Riegel,[6] is shaped to plot below the known failures, which can be attributed to AIV. Point F of the C-M data, a failure at $L_w = 165$ and $D/t = 55$, was the result of an undercut weld, which had no further issues after repair. Therefore, it is not an AIV failure, but is more properly attributed to construction technique and was ignored for the purpose of fitting the CSTI curve. The new reported failures are labeled I, J, K, and L, and the additional reported non-failures are labeled 28-39. Point L ($L_w = 178$ and $D/t = 31$) operated for more than 12 hours and therefore would not be a failure of the criteria curve.

The Eisinger curve, which has been incorporated into the NORSOK Standard L-002 Edition 3, July 2009, is presented as the straight line in Figure 2. The formula for this line is:

$$L_w = 173.6 - 0.125 D_i / t \quad (3)$$

where t is the thickness in mm, and D_i is identified as the internal diameter, Eisinger seems to have plotted his figures using the nominal diameters. Although it purports to be a *safe* design curve, there has been at least one failure on this curve, namely point H, which falls on the line when plotted with the original L_w.

There are no AIV failures below the Riegel/C-M curve. Plotting L_w against D/t (rather than against D) continues to serve as the design approach for several major companies, using a criteria curve that is similar to the black curve shown in Figure 2.

The number of failures at each pipe thickness is summarized in Table 1. For the C-M data, the pipe diameter and thickness are given. For the new data, we have only D/t; neither the diameters nor the thicknesses were specified. Using standard piping size schedules, we have evaluated the D/t for each pipe diameter. There are at least two possibilities for diameters and thicknesses for data points I, J, and K, and three possibilities for data point L. Figure 3 presents all of the data, with the thickness of the pipe identified by the color of the circle, square, or triangle around the data point and with failures indicated by a slash through the circle or square. While we have not seen any reports of failures with pipe walls thicker than 0.375 inches, this is not proof that pipes will not fail above this thickness. For the four new data points for failures, we have plotted a single value of D/t^2, with thicknesses as follows: I – 0.312 inch, J – 0.375 inch, K – 0.250 inch, and L – 0.375 inch.

Table 1. Pipe wall thicknesses for documented AIV failures.

Pipe Wall Thickness, Inches	Total C-M Failures	C-M Failures	New Failures
0.219	2	B1, F	
0.250	6	A, C, D, E, G, H	K
0.312	1	B2	I
0.375	0		J, L
>0.375	0		

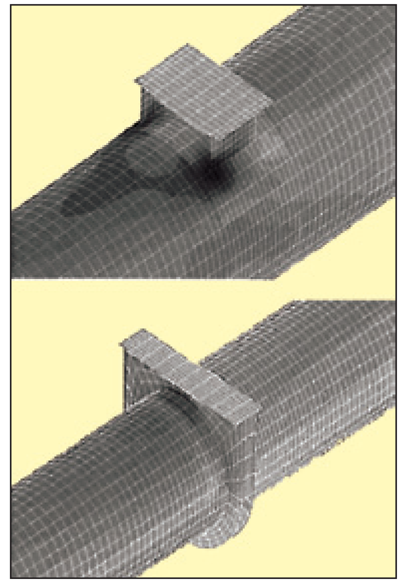

Figure 4. Finite-element modeling of stiffener rings.

Data points 28-39, representing the new non-failure data, are more difficult to classify into relevant diameters and thicknesses. For some data points (30, 32, 33, 34, 35, and 37), there are two or more combinations of diameters and thicknesses that give the relevant D/t. For data point 28, we could find no possible combination of standard diameter and thickness to give the appropriate D/t. For data points 29, 31, 36, 38, and 39, there is a single diameter and thickness that gives the appropriate D/t. These new data points are coded with the following thicknesses (in inches) in Figure 3: 29 – 0.312, 30 – 0.500, 31 – 0.406, 32 – 0.500, 33 – 0.500, 34 – 0.500, 35 – 0.656, 36 – 0.344, 37 – 0.312, 38 – 0.250, and 39 – 0.312.

CSTI Acoustics Approach

Valves produce sound power inside the downstream line. The sound is radiated downstream through the piping system and out through the walls of the line, causing the walls to vibrate. Most of the energy stays inside the line with very little attenuation over distance.

In sections where the sound power level exceeds the criteria, the pipe is vibrating significantly due to the high internal sound power levels. Failure points arise with AIV when there is an asymmetric junction or attachment. The movement of the pipe can cause high stresses at the joint and an eventual failure. For lines with sound power levels above the criteria, the asymmetric discontinuities in the pipe wall are the potential failure points. These include branch connections, tie-backs, support saddles, vents, drains, and any welded connections to the line.

Depending on how much the internal sound power level exceeds the criteria, the piping design is revised depending on established guidelines. Table 2 shows the criteria used at several petrochemical firms.

CSTI's approach is to calculate the L_w of the valve using Eq. 1 and propagate the sound power down the line, comparing the L_w with the criteria along the line. If the criteria are exceeded, we see if source controls will be allowed, e.g.:

- Use more valves, reducing L_w
- Use different valves, reducing L_w
- Use multi-staged restriction orifices, reducing L_w
- Use an in-line silencer, reducing L_w

Quiet valves are available and have been used to control AIV. We are unaware of any in-line silencers having been used for AIV problems and no manufacturers responded to a recent bid request for them.

Next, we consider the possibility of using damping and stiffener rings. Hayashi *et al.*[7] reported reduction of stress by 43% in a finite-element model that calculated the effect of stiffeners on pipe wall stresses. Figure 4 shows the differences in the stresses in the pipe wall for a pipe shoe on the left vs. a combined pipe shoe/stiffener ring on the right. Table 3 gives the stress for the pipe shoe and the combined pipe shoe and stiffener ring. Although this is encouraging information, since we have not seen any quantitative data for an application in the field, we are reluctant to incorporate stiffener rings into field applications. Neither do we know of any successful applications of damping techniques successfully applied to prevent or remedy AIV failures.

Once the source and path treatments have been considered, CSTI then moves to the two methods of treatment most often used:

- Increasing pipe wall thickness, which allows higher sound power levels, or
- Eliminating asymmetric discontinuities which reduces the risk, as Carucci and Mueller reported:[3]

Based on past experience, asymmetric discontinuities in the pipe wall, such as branch connections, support saddles, and restraint attachments are potential fatigue failure points. Fatigue failures are caused by the peak cyclic stresses that occur at these details where the vibrating pipe wall is abruptly restrained by an asymmetric discontinuity.

Axisymmetric discontinuities in the pipe wall, such as at flanges and stiffener rings, have been found not to be potential fatigue failure points. This is because the pipe wall vibration amplitudes damp out gradually as they approach an axisymmetric discontinuity due to the cylindrical shell stiffening effect. In this way, the shell vibration cyclic stresses are minimized. It is also interesting to note that an acoustically induced fatigue failure has not occurred in a section of plain unstiffened pipe. Therefore, the only recommended precautions to be taken for axisymmetric discontinuities are to assure good quality full penetration welds with no undercut in flange, stiffener ring, or pipe walls.

Figure 5 shows the AIV criteria that CSTI has used on previous projects. If the L_w at a location exceeded the design curve by less than 5 dB, the pipe wall thickness was increased until the L_w was less than the criteria, or the piping with this L_w had full wrap encirclements for all connections. If the L_w at a location was between 5 and 10 dB above the criteria, then the pipe wall thickness was increased to the lesser of 0.500 inches or just enough so that the point fell within 5 dB of the criteria, and full wrap encirclement was used. If the L_w exceeded the criteria by 10 dB or more, then the pipe wall thickness was increased until the L_w was not greater than the criteria +5 dB, and full wrap encirclement was used.

Table 2. Comparison of different AIV guidelines.

Above criteria by:	Guidelines/Recommendations for up to 12 Hrs of Non-Continuous Operation — Guideline A	Guideline B
0 to 5 dBA	Full wrap encirclement, forged tees, etc.	13 mm (0.500") wall thickness with welding tees, full wrap, etc.
5 to 10 dBA	13 mm (0.500") wall thickness with full wrap, forged tees, etc.	13 mm (0.500") wall thickness with welding tees, full wrap, etc.
10 to 15 dBA	Redesign system	16 mm (0.625") wall thickness with welding tees, full wrap, etc.
Above 15 dBA	Redesign system	Redesign system

Table 3. Maximum stress (MPa) with and without stiffener ring.

Item	Load Due to Thrust	Load Due to AIV
Pipe shoe	17	91
Pipe shoe/stiffener ring	14	52

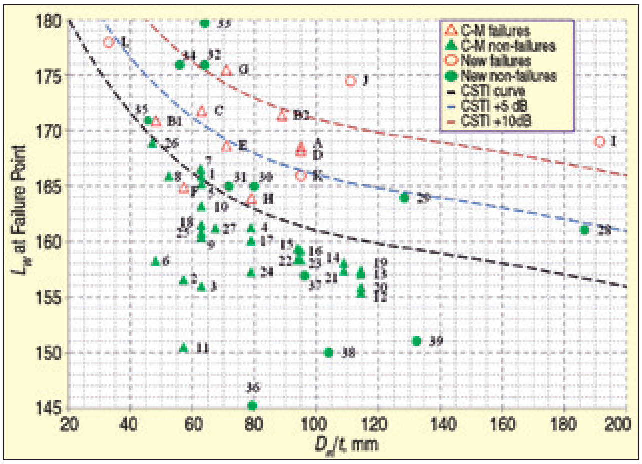

Figure 5. Previous CSTI AIV criteria curves.

All lines exceeding the criteria received treatment at all asymmetric locations. This treatment required:

- Eliminating all weldolets
- Full-wrap encirclement, forged tees, or sweepolets for all connecting lines 2 inches and above
- Eliminating small vents, drains, and other connections smaller than 2 inches or replacing with minimum 2-inch connections treated as discussed above

Criteria Uncertainty

To create a valid AIV criteria curve, there are two basic approaches:

- Using theoretical equations to determine fatigue based on the piping design and conditions
- Making a judgment based on historic data

With the theoretical approach, the necessary equations are well known. However, field conditions almost never match ideal laboratory conditions. Data from comprehensive laboratory testing of actual piping configurations combined with extensive field experience is needed to certify or revise the theoretical analyses. However, such data are extremely scarce in the public domain, and we know of no criteria curves derived therefrom. We suggest that major players have a vested interest in funding objective studies on AIV whose findings would be in the public domain.

When using historic data, the problem is an insufficient quantity of fully detailed data. Only the original C-M data set has enough detail for thorough analysis. As mentioned previously, the newly reported AIV points have only two known parameters: L_w and D/t. We don't know the pipe wall thickness or any of the flow parameters for the new data, and we cannot even be certain which diameter was used for the ratio – internal, external, or nominal. Furthermore, the uncertainty range of the sound power for any of the valves could easily be ±3 dB or worse. Finally, for all the data, there are no details on any downstream asymmetric discontinuities. With such limited knowledge, it is remarkable that the design curves in use have been so successful to date.

We need more data with at least the level of detail as provided by Carucci and Mueller, along with agreement to standardize on reporting the actual pipe dimensions rather than the nominal diameter. Knowledge of the flow conditions is critical, especially whether or not sonic flow exists. It is important to know if asymmetric discontinuities were present at the points of failure along with the use (or lack) of any reinforcements, stiffening, or damping. The result of lacking detailed information for the new data is a rather large uncertainty when formulating experience-based criteria.

With so much current uncertainty in both methods of establishing criteria curves, it is prudent to add a safety factor to any criteria curve obtained by either method. But just how much of a factor should be added? Standard engineering practice suggests that this buffer should be not less than 3 dB below the failure curve in view of the potential for extreme financial loss with AIV failures. Given that AIV failures can result in personal injuries (even death in the

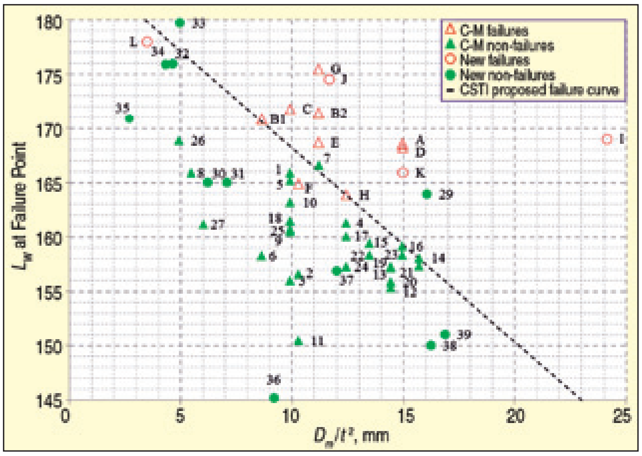

Figure 6. AIV failure curve: $L_w = 186.07 - 1.7857(D_m/t^2)$.

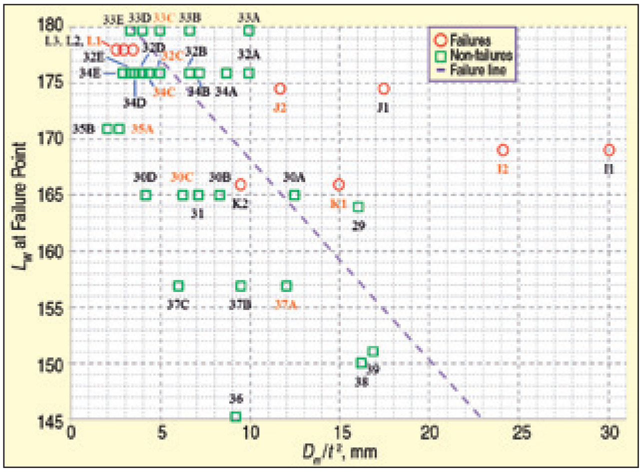

Figure 7. Possible diameter and thickness combinations, new data points.

extreme cases), one might wish to be more cautious and use a +5 dB safety factor. Here again, more data would be beneficial for making sound judgments.

Non-Linear Excitation Response

Maximum vibration of the pipe wall will occur when the pipe structural modes coincide with the propagating acoustic modes. Norton & Karczub[8] have noted that pipe wall vibrational response is a function of the non-dimensional pipe wall thickness parameter β a parameter that is derived from the wall thickness and mean pipe radius:

$$\beta = t / (2\sqrt{3}R_m) \tag{4}$$

where t is the pipe wall thickness and R_m is the mean pipe radius. First, there is a direct effect that is inversely proportional to β^2. It is important to note that this direct effect is non-linear. Let V be the total vibrational response, and let k be the lumped parameter for all the other effects, with D_m being the mean diameter:

$$V = k / \beta^2 = k(12R_m / t^2) = 6k(2R_m / t^2) = 6k(D_m / t^2) \tag{5}$$

Second, variations in β produce significant variations in the possible number of wave number coincidences, and the number of coincidences is essentially independent of non-dimensional length and is generally unaffected by flow speed and varies inversely with β.

So the direct affect says that vibration magnitude at coincidence is proportional to D_m/t^2, and the indirect effect says that increasing diameter relative to thickness or decreasing thickness relative to diameter increases the likelihood of coincidence.

These thoughts led us to plot the AIV data as a function of D_m/t^2 as shown in Figure 6. When L_w is plotted as a function of D_m/t^2, a straight-line failure curve can be drawn according to Eq. 6.

$$\text{Failure line: } L_w = 186.07 - 1.7857\left(D_m / t^2\right) \tag{6}$$

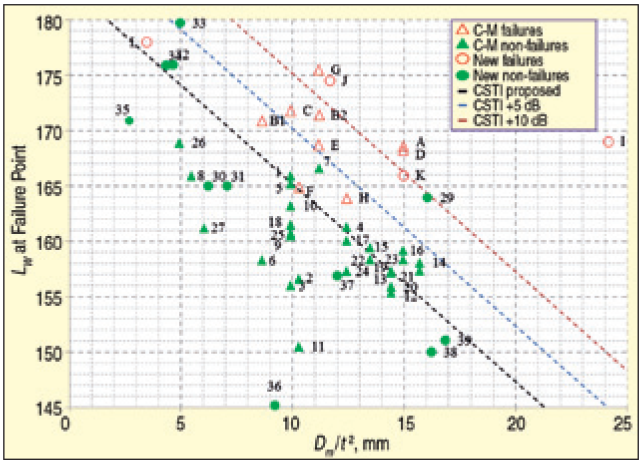

Figure 8. CSTI proposed design curves: $L_w = 183.07 - 1.7857(D_m/t^2)$

Figure 7 plots the new data points along with the failure line (excluding point 28, for which no diameter and thickness combination could be deduced). For each data point, there may be a number of combinations of diameter and thickness that give the appropriate D/t. For point 33, for example, there are five different possibilities that are identified as 33A-33E. The points in orange, e.g. 33C, are the most likely combinations. Note that points I1, I2, J1, J2, and K1 are above the failure line. K2 is below it, leading us to speculate that the dimensions of K1 are more likely correct than those of K2. L1, L2 and L3 are beneath the failure curve, but it was operated for more than 12 hours. Points 30-39 either had or could have had dimensions that would place them below the failure line. In addition, these lines could have had other treatments such as full wrap encirclements that would explain why they did not fail. Only points 29 and 33 of the non-failures are above the failure line.

A criteria curve could be drawn 3 dB below that failure line, as shown in Figure 8, according to Eq. 7:

$$\text{Criteria line: } L_w = 183.07 - 1.7857\left(D_m / t^2\right) \tag{7}$$

The new data identified in orange in Figure 7 are shown in Figure 8 along with the proposed criteria and treatment lines (criteria +5 and criteria +10). Again, note that point *F* was an undercut weld, and *L* was operated for more than 12 hours. It would be useful to know if these points (29-39) have failed in subsequent operation, how long they have ever operated, or if they have untreated asymmetric connections.

Conclusions

We have examined historical data and criteria curves along with the historical methods of designing to avoid or remedy AIV failures. CSTI's historical approach has been detailed and a new criteria curve has been proposed based on the non-linear nature of vibration excitation and using the ratio of D_m/t^2. The lack of practical experience with damping and stiffener rings has also been noted. By working together to improve the criteria, based on both experience and solid scientific investigation, we can hope to prevent future failures of piping due to acoustic-induced vibration in the hydrocarbon industry.

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