Fixed Equipment Newsletter

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ASME Section VIII, Division 1 - Appendix 46

Flow Induced Vibrations in Heat Exchangers

Risk-Based Inspection

Trouble Shooting Leaking Joints

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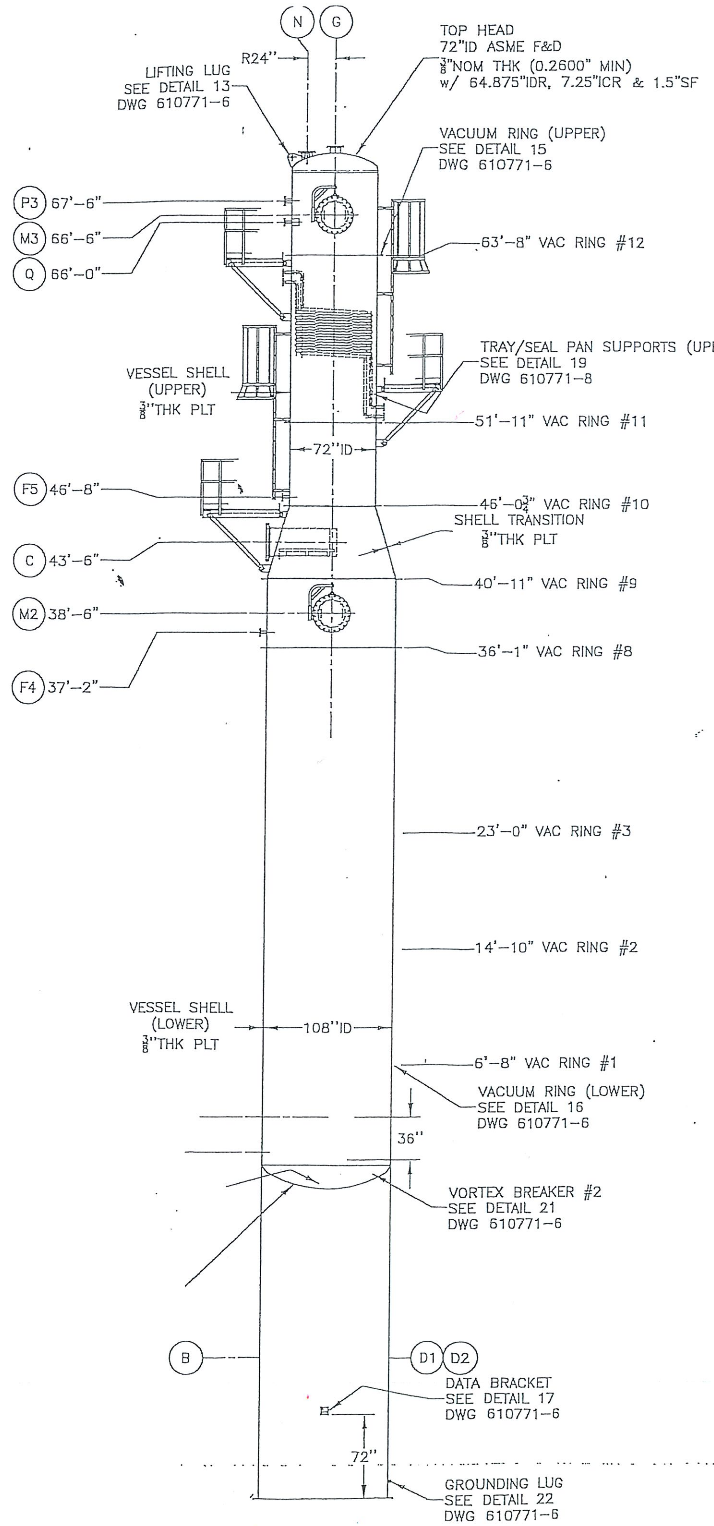

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From The Editor's Desk:

All industrial equipment require periodic maintenance to ensure the unit is in proper functioning condition. This is especially true with pressure vessels as the operating conditions in which these vessels are subject to can generate wear and other concerns under normal conditions.

Maintenance is not confined to corrective measures or to correct operating deficiencies. All safety precautions and maintenance tips described below should be observed to avoid harm when using pressure vessels.

Maximize Awareness: It is essential that maintenance personnel get a complete briefing on the contents of each pressure vessel.

Replace When Necessary: Some industries practice a "replace when needed" maintenance program which allows wear to exceed 75% or more before replacing the part. A better approach is "replace when necessary" which allows wear of up to only 50%. This prevents failures before they occur.

Inspect it Anyways: Each inspection should be as thorough as possible. Even if it was just inspected, inspect it anyways. Pressure vessels operate under dynamic conditions and loads, and while a previous inspection did not reveal any concerns, the situation may have changed.

Protective Device Installation: Installing protective devices helps ensure that the pressure vessel meets the challenges of daily use.

Follow All Safety Protocols: During the maintenance cycle, all rules and protocols that also take into account the execution of follow up measures must be followed. All safety features on the equipment must be inspected and tested to make sure they are in functioning order.

A proper inspection regimen with rigorous maintenance schedule will not only save your company money over the life of the pressure vessel but it will also go a long way to reducing any potential liability during the operation of the pressure vessel.

Ramesh K Tiwari
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In this issue…

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ASME SECTION VIII, DIVISION 1 - APPENDIX 46

When is Appendix 46 of ASME Section VIII, Division 1 applicable?

Appendix 46 is applicable when using Division 2 to establish the thickness and other design details of a component for a Section VIII, Division 1 pressure vessel.

What are the allowable stress values and the weld joint efficiencies used when applying Appendix 46?

ASME Section VIII, Division 2 allows the use of "Design by Rules" and/or "Design by Analysis" for design and construction of pressure vessels. Design by Rules requires adhering to Part 4 of the Code and Design by Analysis requires adhering to Part 5 of the Code. Therefore, when applying Appendix 46, the designer has choice of using Part 4 and/or Part 5 of the Code.

a. When Part 4, Design by Rule is used, the design shall be in accordance with paragraph 46-3.
    i. The allowable stress shall be in accordance with UG-23, except that the maximum allowable compressive stress shall be limited as prescribed in Division 2, paragraph 4.4.12 in lieu of the rules of UG-23(b).
    ii. The weld joint efficiency shall be established in accordance with UW-11 and UW-12.

b. When Part 5, Design by Analysis is used, the design shall be in accordance with paragraph 46-4.
    i. The allowable tensile stress shall be in accordance with UG-23.
    ii. The weld joint efficiency shall be established in accordance with the full radiography requirements of UW-11 and UW-12.

DESIGN BY RULE

When "Design by Rule" requirements in Division 2, Part 4 are used to design the components for a Division 1 pressure vessel, the following conditions shall be met:

a. If the thickness of a shell section or formed head is determined using the design rules in Division 2, paragraph 4.3 or 4.4
    1. For design of nozzles, any nozzle and its reinforcement attached to that shell section or formed head shall be designed in accordance with Division 2, paragraph 4.5.
    2. For conical transitions, each component comprising the cylinder-to-cone junction shall be designed in accordance with Division 2, paragraph 4.3 or 4.4.
    3. Material impact test requirements shall be in accordance with the rules of Division 1, except that the required thickness used in calculating the coincident ratio under the rules of UCS-66(b) or UCS-66(i) shall be calculated in accordance with the rules of Division 2.

b. The fabrication tolerances specified in Division 2, paragraphs 4.3.2 and 4.4.4 (as applicable) shall be satisfied. The provision of Division 2, paragraph 4.14, Evaluation of Vessels Outside of Tolerance, is not permitted.

c. When using the rules of Division 2, Part 4, the full set of design loads and load case combinations in paragraph 4.1.5.3 are not required except when necessary to satisfy the requirements of UG-22. When the design load combinations of Division 2, Table 4.1.2 are used, the allowable stress increase of UG-23(d) is not permitted.

1. The factors present in Division 2, Table 4.1.1 for wind loading, W, are based on ASCE/SEI 7-10 wind maps and probability of occurrence. If a different recognized standard for wind loading is used, the User shall inform the Manufacturer of the standard to be applied and provide suitable load factors if different from ASCE/SEI 7-10.
2. The factors present in Division 2, Table 4.1.1 for earthquake loading, E, are based on ASCE/SEI 7-10. If a different recognized standard for earthquake loading is used, the User shall inform the Manufacturer of the standard to be applied and provide suitable load factors if different from ASCE/SEI 7-10.

d. Evaluation of the stresses during the test condition of Division 2, paragraph 4.1.6.2 is not required. However, such calculations may form the basis of a calculated test pressure as described in UG-99(c) or UG-100(b).

e. The fatigue screening criterion of Division 2, paragraph 4.1.1.4 is not required. However, it may be used when required by UG-22.

f. Weld joint details shall be in accordance with Division 2, paragraph 4.2, with the exclusion of Category E welds.

DESIGN BY ANALYSIS

When “Design by Analysis” requirements in Division 2, Part 5 are used to design the components for a Division 1 pressure vessel, the following conditions shall be met:

a. Division 2, Part 5 shall not be used in lieu of the design thickness requirements of Division 1 or Division 2, Part 4.

b. The allowable stress increase of UG-23(d) is not permitted.

c. All of the failure modes listed in Division 2, Part 5 shall be considered.

1. When demonstrating protection against plastic collapse in Division 2, paragraph 5.2, the load case combinations of Division 2 shall be considered in addition to any other combinations defined by the User. In evaluating load cases involving internal and external specified design pressure, P, additional cases with P equal to zero shall be considered.
   - i. When applying the elastic stress analysis method in Division 2, paragraph 5.2.2, the allowable stress, S, shall be as per 46-2(a).
   - ii. When applying the limit-load analysis method in Division 2, paragraph 5.2.3, the yield strength defining the plastic limit shall equal $1.5S$, where S shall be as per 46-2(a).
   - iii. When applying the elastic‑plastic stress analysis method in Division 2, paragraph 5.2.4, in conjunction with Division 2, Table 5.5, β shall equal 3.5.
   - iv. Evaluation of the test condition is not required [see 46-3(d)].
2. When demonstrating protection against local failure in Division 2, paragraph 5.3, the load case combination of Division 2 shall be considered. The exemption provided in Division 2, paragraph 5.3.1.1 is applicable to weld details in Division 2, Part 4 only. There exist weld details in Division 1 that are not permitted in Division 2 [subject to the provision in 46-3(f)]; those details are not exempt from evaluation of protection against local failure.

3. When demonstrating protection against collapse from buckling in Division 2, paragraph 5.4, the design margin of the Division 2 assessment procedure shall be used.
4. When demonstrating protection against failure from cyclic loading: ratcheting in Division 2, paragraph 5.5.6 or 5.5.7, the design margin of the Division 2 assessment procedure shall be used, except that where it is used, the allowable stress, S, shall be per 46-2(a).

   The assessment procedure for this failure mode requires the use of operating load ranges. This requires information on the operating load cycle, which shall be provided by the User. In the absence of such information, the Designer may use his or her judgment to determine the worst case, taking into consideration the vector nature of certain loads and their potential reversal, as well as pressure cycles that may include vacuum conditions.
5. When demonstrating protection against failure from cyclic loading: fatigue in Division 2, paragraphs 5.5.3, 5.5.4, or 5.5.5, the design margin of the Division 2 assessment procedure shall be used, except that where it is used, the allowable stress, S, shall be per 46-2(a).

   The assessment procedure for this failure mode requires the use of operating load ranges. This requires information on the operating load cycle, which shall be provided by the User. In the absence of such information, the Designer may use his or her judgment to determine the worst case, taking into consideration the vector nature of certain loads and their potential reversal, as well as pressure cycles that may include vacuum conditions.

References:

ASME Boiler and Pressure Vessel, Section VIII, Division 1 - 2019 Edition

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FLOW INDUCED VIBRATION IN HEAT EXCHANGERS

INTRODUCTION

Along with heat transfer and pressure drop, tube vibration is one of the primary factors controlling the design of shell-and-tube heat exchangers. Tubes are usually the most flexible part of an exchanger and most susceptible to vibration. They vibrate at their natural frequencies as a result of shell-side fluid flowing past them. In most exchangers, the intensity of vibration is so low that it does not create a problem. Vibration becomes a problem when the intensity increases to the point that it causes some part of the exchanger to fail mechanically - the vibrating tubes impact with baffles, with the heat exchanger shell, or with one another. Extreme tube vibrations lead to leaks in the tubes or tube joints. The result is that the exchanger must be taken out of service for repair and modification.

Tubes vibrate at discrete frequencies depending primarily on their geometry, means of support, and material of construction. The lowest frequency at which tubes vibrate is called their fundamental natural frequency. The intensity of this vibration is evidenced by the amount of periodic movement of the tube, with the largest movement usually at midspan between supports. The extent of this peak-to-peak movement about the at-rest centerline is termed the amplitude of vibration. Energy must be fed to the tubes to excite them into vibration. This energy is provided by flow parallel to or across the tubes. Internal and external damping dissipate the energy. Prolonged tube vibration with large amplitudes leads to the mechanical failure of tubes, which then permits leakage between the shellside and tube-side fluids. Mechanical failure usually occurs by one or more of the following:

Collision Damage

If the amplitude of vibration is sufficiently large, adjacent tubes impact with one another or against the shell and emit a loud noise. The tube walls can be worn thin with time and eventually split open. Tube impact produces characteristic diamond-shaped wear patterns at the midspans of the tubes,

Baffle Damage

Baffle holes are made larger than tubes to facilitate exchanger bundle assembly, and thus the tubes are free to move at the baffles. The walls of vibrating tubes can be cut by the baffle, especially when the baffles are thin and made from a harder material than the tubes. The cutting action reduces the tube wall until leakage occurs.

Fatigue

Repeated bending of the tubes can lead to a breakdown of the tube material if the stresses are high enough and the vibration continues for a sufficiently long time. Fatigue failure can be accelerated by corrosion and erosion. This type of failure results in cracks or in pieces of the tube wall actually breaking off.

Tube Joint Leakage

The joint between the tube and tubesheet can fail due to vibration whether welded or expanded.

In addition, vibration failure results from the cutting action of the edge of the tube holes against the tube wall where the tubes emerge from the tubesheet.

Although tube vibration failures have been reported in different parts of an exchanger, regions of long spans and high local velocities are particularly susceptible. The largest number of reported tube failures occurs in tubes that are within two or three tube rows of the tips of baffles, where the flow velocity is usually very high.

TUBE BUNDLE VIBRATION CHARACTERISTICS

Every component of a shell-and-tube heat exchanger vibrates at its own unique natural frequency. Tubes are the most flexible and thus the most easily excited. Tubes can and do vibrate at different frequencies. The lowest natural frequency is called the fundamental or first mode. The higher natural frequencies are referred to as the second mode, third mode, and so on. For conservative design purposes the fundamental frequency will be emphasized and will be called simply the natural frequency of the tubes.

The natural frequency of tubes like that of a simple beam depends on the way the ends are fixed (clamped or simply supported), the type of intermediate supports (simply supported, pinned, or clamped), the tube cross-sectional geometry, the number of spans, the materials of construction, and the length of spans. Although the natural frequency of the tubes can be measured experimentally, predictive methods are used to give approximate values of the natural frequency.

Tubes are rigidly fastened to the tubesheets and supported at intermediate points along their length by baffles or support plates. Some tubes in the center of a bundle may be supported by every baffle, whereas tubes that pass through baffle windows may be supported only by every second or third baffle. Furthermore, end baffle spacings are often longer than central baffle spacings to accommodate the nozzles and the shell flanges. Thus, the lengths of the spans are seldom uniform along the length of a tube, and not all tubes in a bundle are supported by the same number of baffles. This results in tubes with different natural frequencies in the same exchanger.

STRAIGHT-TUBE NATURAL FREQUENCIES

Several different approaches can be used to predict the natural frequency of straight tubes. Most start with the assumption of a uniform beam clamped at least at one end with intermediate supports along its length. The rigorous approach considers spans of unequal length between supports by writing the governing differential equation of motion for each span separately. The solution is obtained through the use of the boundary conditions at the ends of the tube and by relating the deflections and slopes at each intermediate support. This yields a set of linear homogeneous equations in the constants. The determinant of the coefficients is set equal to zero to yield the characteristic or frequency equation from which the natural frequency can be derived. The coefficients and graphic solutions are available for determining the natural frequency of tubes with various combinations of spans with two different lengths. The computer-plotted graphs for the shape of the various span configurations of maximum deflection are also available. Computer programs, using the finite element approach, can provide accurate determination of the natural frequencies and mode shapes for tubes with unequal spans and varying degrees of intermediate restraint. Figure 1 is an example of the results from such a detailed analysis.

A somewhat less accurate, but normally adequate approach assumes that all spans are of equal length L. The natural frequency f_n is given by the equation

$$f_n = 73.03 \frac{C_n}{L^2} \sqrt{\frac{EI}{M_e}}$$

where, E = Modulus of elasticity of tube material,

I = Area moment of inertia, and

M_e = Effective mass per unit length.

M_e is made up of three parts the mass per unit length of the tube, the mass of the fluid inside the tube, and the virtual mass per unit length of tube for the shell-side fluid displaced by the tube.

The frequency constant C_n depends on the way the ends of the tube are fastened, the number of spans, and the type of intermediate supports. Values of C_n, for tubes clamped at the ends with simple intermediate supports are listed in Table 1. In addition to the values for the fundamental natural frequency (mode 1), the values for the higher natural frequencies (modes 2, 3, 4, etc.) are shown. Notice that with more than four spans the difference in C_n, between successively higher modes becomes small. Further, the separation between the fundamental and the several higher-mode natural frequencies becomes small once the number of spans exceeds eight. Thus, for the design of most exchangers, only the lowest natural frequency of the tubes needs to be considered.

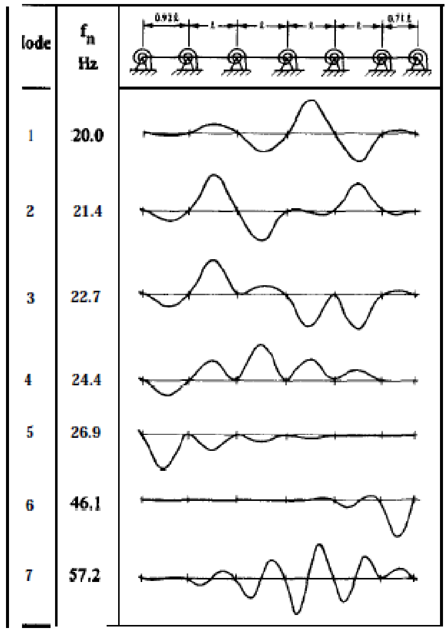

Figure 1: Natural Frequencies and Mode Shapes for a Tube with Uneven Length and Different Rotational Stiffness at Supports

Table 1: Frequency Constant C_n for Uniform Beams of Equal Span Length Simply Supported with Extreme Ends Clamped

Number of Spans	Mode				
	1	2	3	4	5
1	12.36	198.34	388.75	642.63	959.98
2	49.59	12.36	160.66	198.34	335.20
3	40.52	59.56	72.36	143.98	178.25
4	37.02	49.59	63.99	72.36	137.30
5	34.99	44.19	55.29	66.72	72.36
6	34.32	40.52	49.59	59.56	67.65
7	33.67	38.40	45.70	53.63	62.20
8	33.02	37.02	42.70	49.59	56.98
9	33.02	35.66	40.52	46.46	52.81
10	33.02	34.99	39.10	44.19	49.59
11	32.37	34.32	37.70	41.97	47.23
12	32.37	34.32	37.02	40.52	44.94

The number of spans depends on the baffle arrangements and location of the tubes in the bundle. Most heat exchangers have their longest unsupported spans passing through the baffle windows and/or in the end zones.

Segmental and double-segmental baffle arrangements result in window spans supported by every other baffle, whereas triple-segmental baffle arrangements are supported by every third baffle. Cross-flow exchangers and no-tubes-in-window exchangers provide support at each baffle and support plate. In the same bundle the number of spans may differ by one or two and the associated longest span lengths will vary. Since the natural frequency depends on both the length of span and the number of spans (see Table 1), all combinations in the bundle should be checked to determine the one that produces the lowest natural frequency.

a. Tube-to-baffle hole clearance effect on natural frequencies

Normal tube-to-baffle hole manufacturing clearances result in the baffles acting like simple supports. It has been found experimentally that the clearance has to be reduced to nearly a pressed fit before there was any appreciable change in the natural frequency. For most exchangers, the tube-to-baffle hole clearance is not a significant parameter for controlling the natural frequency, but it may be important in determining damping and tube wear.

b. Axial stress effect on natural frequencies

The tubes in a heat exchanger bundle may be under a tension or compression axial loading due to manufacturing procedures and/or to operating temperatures and pressures. Assuming that the axial force P_a, is known, the following equation can be used to adjust the natural frequency:

$$(f_n)_s = f_n\sqrt{1 + \frac{P_a L^2}{EI\pi^2}}$$

The sign for axial force P_a, is minus when it is compressive and plus when it is tensile. Thus, tension increases the natural frequency, whereas compression decreases it. The natural frequency due to axial loading in a typical heat exchanger varies by as much as ±40%. Not knowing or considering the axial loading can lead to serious inaccuracy of natural frequency predictions.

c. Finned-tube natural frequencies

To calculate the natural frequency of finned tubes, the area moment of inertia must be calculated using an effective diameter for the outside diameter. For low-finned tubes typically used in shell-and-tube exchangers, a value of about 8% thicker wall for the finned section should be used along with the actual inside diameter under the finned section to calculate the area moment of inertia. The mass of the tube per unit length term in the effective mass M_e, should be the actual mass for the finned section. The fins on tubes should not alter the assumed simple support constraint at the baffles.

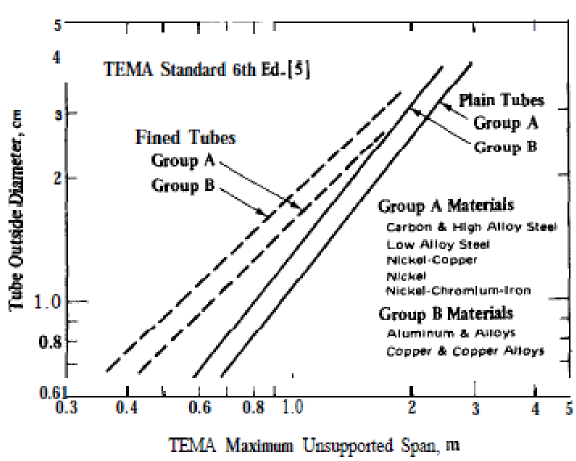

Figure 2: TEMA Maximum Unsupported Length for Plain Tubes and Finned Tubes

d. Span lengths

Of the variables that affect the natural frequencies, none is as influential as the length of the unsupported spans. The natural frequency varies as the reciprocal of the span length squared. Early in the 1940s the TEMA Standards somewhat arbitrarily placed maximum allowable unsupported spans for different tube sizes and tube materials. Figure 2 is a plot of these maximum lengths. By coincidence, these lengths have prevented many serious tube vibration problems. However, the current trends toward higher velocities to improve heat transfer and minimize fouling make the TEMA maximum spans marginal for many exchangers. The current trend is to limit maximum spans to 80% of the TEMA values, which increases the natural frequency by more than 50%.

NATURAL FREQUENCIES OF U-TUBE BENDS

The vibration characteristics of a U tube are more difficult to predict than those of a straight tube. A simplified approach is used that considers the straight portions and the bend separately. The straight portion is handled the same as straight tubes. The bend portion uses a method that is included in the TEMA Standards. Assuming simple supports for the first-mode U-tube out-of-plane, natural frequency can be calculated by the following:

$$f_{nu} = 59.55\frac{C_u}{R^2}\sqrt{\frac{EI}{M_e}}$$

where C_u, is the first-mode U-tube constant as shown in Figure 3, and R is the tube bend radius. This simplified approach does not consider intermediate supports around the bend for U tubes that materially increase the natural frequencies by stiffening the overhung section of the bundle. However, this complicates the prediction of the natural frequencies as the extent of their effectiveness is uncertain.

AMPLITUDE OF TUBE VIBRATION

The natural frequencies are characteristics of the tubes themselves and are independent of the way they are excited or the amplitude of vibration. On the other hand, the amplitude depends on the natural frequency of the tubes, the mode shape, the frequency and the strength of the exciting mechanism, and the internal and external system damping.

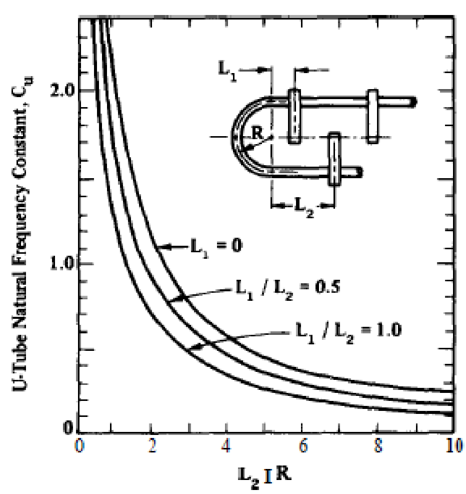

Figure 3: U-tube Natural Frequency Constant for First-mode Out-of-plane Vibration

The tubes in many shell-and-tube heat exchangers vibrate, but the amplitude is so small that the vibration is not a problem. For the amplitude to become appreciable the system damping must be small and the energy input large. For forced vibration of lightly damped structures when the ratio of the forcing frequency to the natural frequency approaches unity, resonance occurs with an attendant amplification of the amplitude. Thus, if the forcing frequency can be kept appreciably below the natural frequency, the amplitude of vibration will be small.

Amplitudes as large as half the gap distance between adjacent tubes lead to impact collisions. Lesser amplitudes can cause tube cutting by the baffles or failure of the tube joints.

SYSTEM DAMPING

System damping has a strong influence on the amplitude of vibration. In any real system, once the energy exciting the vibration ceases, the amplitude of vibration decays with time. The rate at which the vibration dampens out is often exponential. The logarithm of the difference in successive amplitude peaks is called the log decrement δ_o and is an indication of the damping. The higher the value of the log decrement, the more heavily damped the system is. Most tubes in heat exchangers are very lightly damped structures with low values of the log decrement.

Damping depends on the mechanical properties of the tube material, the geometry of the intermediate supports, and the physical properties of the shell-side fluid. Tight tube-to-baffle clearances and thick baffles increase damping, as does a very viscous shell-side fluid. The log decrements for copper-nickel finned tubes of 0.032 in still air have been measured. No method is available for predicting the log decrement, although the range of values most probably lies between 0.01 and 0.17 for tubes in heat exchanger bundles.

In the flow-induced vibration situation, if the energy input cannot be dissipated through damping, the amplitude of vibration will increase with time leading to a runaway condition. If there is a balance, the amplitude remains constant. If the damping exceeds the energy input, the amplitude diminishes with time.

SHELL-SIDE VELOCITIES IN SHELL-AND-TUBE HEAT EXCHANGERS

All predictive methods for flow-induced vibration contain a velocity term. The stream analysis method provides a procedure for determining the fraction of the total that flows in each of five shell-side streams. These streams are the main cross flow, the bundle-to-shell leakage, the baffle-to-shell leakage, the tube-to-baffle leakage, and the bypass stream in any lane due to tube-pass partitions parallel to the crossflow velocity. In most industrial heat exchangers, the main cross flow is considerably less than the total flow. Further, as the shell-side fluid flows through a bundle of tubes, the velocity is constantly changing in magnitude and direction.

CROSS-FLOW VELOCITIES

The definition for cross-flow velocity used by most investigators of flow-induced vibration is based on the minimum flow area through a tube row perpendicular to the primary direction of flow. For an ideal tube bank the selected velocity is well defined. However, for a shell-and-tube exchanger it is somewhat ambiguous, as the number of tubes in each row varies from baffle tip to baffle tip. In order to be consistent, the crossflow velocity for shell-and-tube exchanger vibration prediction will be based on an integrated average area between the maximum and minimum number of tubes in the rows between baffle tips, on the gaps between adjacent tubes in a tube row, and on the cross-flow fraction of the total flow.

PARALLEL-FLOW VELOCITIES

The flow components parallel to the tubes include baffle-to-shell leakage streams, tube-to-baffle hole leakage streams, and most important, the flow through the baffle windows. The parallel-flow velocity is usually assumed to be the total cross flow plus the bundle-to-shell leakage stream flowing through the net window area. The parallel-flow velocities for multi-segmental baffle exchangers are particularly difficult to define.

LEAKAGE AND BYPASS STREAM VELOCITIES

The leakage and bypass streams in a shell-and-tube heat exchanger are often disregarded, as they can be predicted only by a technique such as a stream analysis method. These streams can have velocities as much as 10 times higher than the main cross-flow velocity. Many of the tubes that have failed were in the proximity of one of these secondary flow streams. Heavy fouling can block the leakage through the tube-to-baffle clearances.

ENTRANCE AND ESCAPE VELOCITIES

The fluid entering the shell through the inlet nozzle impinges directly on the first row of tubes under the nozzle unless some type of impingement device is present. This device can be a row of dummy tubes, a solid plate, a perforated plate, or some type of bar grid. With an impingement device the bundle entrance velocity should be limited. This may require the removal of several rows of tubes to provide sufficient area between the bottom of the nozzle and the impingement device. The TEMA Standards specifies that the bundle entrance ρu^2 should not exceed 6,000 kg/ms^2. With impingement plates a region of high velocities may be found near the first rows of tubes below the impingement plate in the bundle-to-shell clearance areas. The flow in the end zone between the inlet and the fast baffle window is very difficult to analyze because of the large number of possible flow paths. There can be areas where the flow is anything from highly turbulent to nearly stagnant.

At the exit of the exchanger a sufficient escape area must be provided to prevent velocities from being high enough to initiate vibration in this region. These potentially troublesome areas are not easily analyzed and are often overlooked in a check for vibration problems.

TURNAROUND VELOCITIES

It is important to understand the flows in the baffle window turnaround regions, as most of the tubes that fail pass through them. Here the flow makes a reversal of direction between each cross pass. Velocity is continually changing in both magnitude and direction, Even though the flow at any point can be separated into its cross-flow and parallel-flow components, it is uncertain how to use these in the various predictive methods.

FLOW INDUCED VIBRATION PHENOMENA

There are several recognized phenomena that are associated with flow-induced vibration. These include vortex shedding, fluid elastic instability, turbulent buffeting, parallel-flow eddy formation, and acoustic vibration. Since any one of these can produce a flow-induced vibration problem, each must be considered in any comprehensive vibration analysis of a shell-and-tube heat exchanger.

VORTEX SHEDDING

Flow across a tube produces a series of vortices in the downstream wake formed as the flow separates alternately from the opposite sides of the tube as shown in Figure 4. This shedding of vortices produces alternating forces, which occur more frequently as the velocity of flow increases. For a single cylinder the tube diameter, the flow velocity, and the frequency of vortex shedding f_{vs} can be described by the dimensionless Strouhal number Sr:

$$f_{vs} = \frac{Sr\,.\,u_c}{D_o}$$

For single cylinders, the vortex shedding Strouhal number is a constant with a value of about 0.2. Vortex shedding occurs for the range of Reynolds number 100 < Re < 5 X 10^5, and >2 X 10^6, whereas it dies out in between. The gap is due to a shift of the flow separation point in vortices in the intermediate trans-critical Reynolds number range.

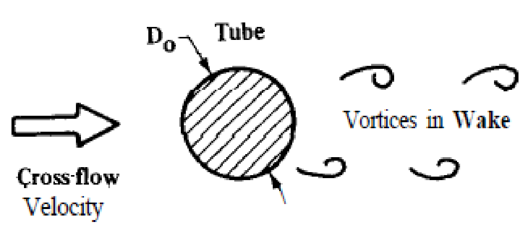

Figure 4: Sketch of Vortex Shedding Resulting from Flow across a Tube

Vortex shedding has also been observed for flow across ideal tube banks. The Strouhal number is no longer a constant, but varies with the arrangement and spacing of the tubes. Vortex shedding is fluid mechanical in nature and does not depend on any movement of the tubes. For a given arrangement and tube size, the frequency of the vortex shedding for non-vibrating tubes increases as the velocity increases. The vortex shedding can excite tube vibration when it matches the natural frequency of the tubes. The vortex shedding frequency can become locked in to the natural frequency of a vibrating tube even when the flow velocity is increased. The movement of the tube organizes the separation of the vortexes leaving the vibrating tube.

The values of the Strouhal number for ideal tube banks are nearly constant for a wide range of Reynolds numbers, but vary considerably with longitudinal and transverse tube spacing.

TURBULENT BUFFETING

Extremely turbulent flow of the shell-side fluid contains a wide spectrum of frequencies distributed around a central dominant frequency, which increases as the cross-flow velocity increases. This turbulence buffets the tubes, which selectively extract energy from the turbulence at their natural frequency from the spectrum of frequencies present. This is an extremely complex form of excitation. The dominant frequency for turbulent buffeting f_{tb} is as follows:

$$f_{tb} = \left[0.28 + 3.05\ \frac{u_c D_o}{l_t}\left(1 - \frac{D_o}{t}\right)^2\right]$$

About the central dominant frequency for turbulent buffeting, there is a band of frequencies of lesser energy content. The turbulent energy spectrum can be represented by a square band bounded by frequencies ±9 Hz about the central frequency.

When the dominant frequency for turbulent buffeting nearly matches the natural frequency, a considerable transfer of energy is possible leading to significant vibration amplitudes.

The above equation is not recommended when the shellside fluid is a liquid. This is not to say that turbulent buffeting cannot occur for liquids, but rather that the empirical predictions equation is based only on tests using gases as the shell-side fluid.

FLUID ELASTIC WHIRLING

Fluid elastic whirling is evidenced by tubes vibrating in an orbital motion. This motion is produced by flow across the tubes causing a combination of lift and drag displacements of the tubes at their natural frequencies. Typically, once fluid elastic whirling commences, it can lead to a runaway condition if the energy fed to the tubes exceeds that which can be dissipated by damping. The critical cross-flow velocity u_{crit} above which fluid elastic whirling can develop is given by the following equation:

$$u_{crit} = \beta f_n D_o \sqrt{\frac{M_e \delta_o}{\rho_s D_o^2}}$$

where β is the fluid elastic instability threshold constant and δ_o is the log decrement for the tube bundle in the shell-side fluid under static (no-flow) conditions.

Figure 5 indicates that β is a function of the tube field layout. The effect of both tube field layout angle and tube pitch-to-tube diameter ratio on β for banks of tubes is shown in Figure 6. These preliminary results also suggest that as the pitch-to-diameter ratio decreases, the value of β decreases.

For the critical velocity method to be fully effective, it is necessary to predict the log decrement and the variation of β with tube type, layout angle, and tube pitch.

PARALLEL-FLOW EDDY FORMATION

Vibration due to axial or parallel flow results from the development of eddies along the tube. Nuclear reactors, and occasionally their associated heat exchangers, have experienced this type of vibration. Typically, they had long, unsupported tube spans, relatively narrow shell side flow passages, and very high axial velocities. The turbulent eddy frequency is initiated by flow parallel to the tubes, which can excite the tubes into vibration at their natural frequency. In most industrial shell-and-tube heat exchangers, axial-flow-induced vibration is not a problem, as the axial flow in baffled exchangers is considerably below the critical velocity.

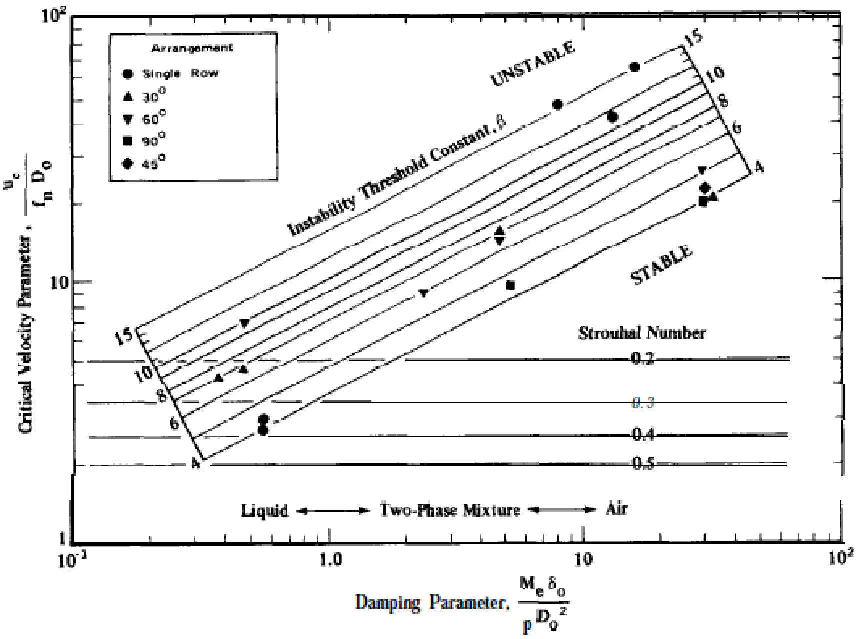

Figure 5: Stability Diagram for Tube Arrays

ACOUSTIC VIBRATION

Acoustic vibration occurs only when the shell-side fluid is a vapor or a gas. This type of vibration is related to sound generated in an organ pipe. The characteristic frequency of the acoustic vibration in a heat exchanger depends on some length, usually the shell diameter d, and the velocity of sound in the shell-side fluid u_{sound}. The acoustic frequency f_a, can be predicted by the following equation:

$$f_a = \frac{m u_{sound}}{2d}$$

where m is the mode number, which is a dimensionless integer. The lowest acoustic frequency is achieved when m = 1 and the characteristic length is the shell diameter. This is called the fundamental tone, and the higher overtones vibrate at acoustic frequencies two, three, or four times the fundamental (m = 2, 3, or 4). The first two overtones are illustrated in Figure 7. Reports of third, fourth, or higher overtones in heat exchangers are rare.

The acoustic frequencies of an exchanger can be excited by either vortex shedding or turbulent buffeting. So long as the exciting frequencies are within 20% of an acoustic frequency, a loud noise may be produced. This acoustic vibration becomes destructive when it is in resonance with some component of the exchanger. Good designs ensure that the natural frequencies of the tubes differ from the acoustic frequencies of the exchanger shell. The acoustic frequencies of the shell can be changed by inserting a detuning plate parallel to the direction of cross flow to alter the characteristic length. This does not materially alter either the heat transfer or the pressure drop.

PROCEDURE FOR VIBRATION PREDICTIONS

PRIMARY CHECK

A four-step primary check is suggested for shell-and-tube heat exchanger designs to minimize the probability of flow-induced vibration problems. The steps can be taken in any order and should be repeated for the inlet, central, and outlet regions. For a high probability of not having vibration, vibration must not be predicted at any step. The following are the four steps in the primary check procedure:

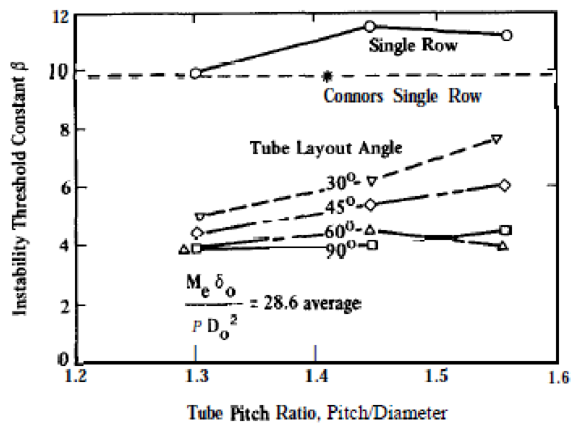

Figure 6: Instability Threshold Constants

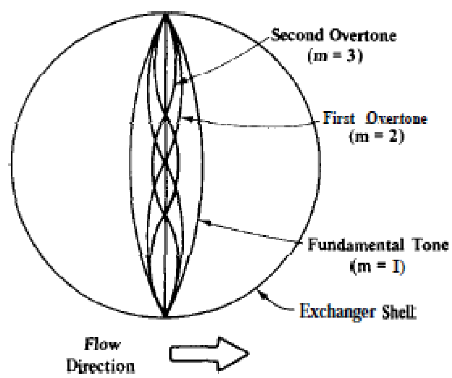

Figure 7: Most Probable Location of Standing Acoustic Waves for Fundamental and First Two Higher Overtones

Step 1.

For a gas or vapor as the shell-side fluid, compare the vortex shedding or turbulent buffeting frequency based on the cross-flow velocity to the acoustic frequency. If within 20%, acoustic vibration is probable.

Step 2.

For either gas or liquid as the shell-side fluid, compare the cross-flow velocity u_c, to the critical velocity to initiate fluid elastic instability u_{crit}. Vibration and damage are probable when u_c, is greater than u_{crit}.

Step 3.

For either gas or liquid as the shell-side fluid, compare the vortex shedding frequency to the lowest natural frequency of the tubes. Because of the number of uncertainties, vibration (not necessarily damage) is possible when the ratio f_{vs}/f_n, is greater than some value less than 1, often 0.5.

Step 4.

For gas or vapor as the shell-side fluid, compare the turbulent buffeting frequency to the lowest natural frequency of the tubes. Because of the number of uncertainties, vibration (not necessarily damage) is possible when the ratio of f_{tb}/f_n, is greater than some value less than 1, often 0.5.

Notice that the primary check procedure predicts the probability of vibration and not damage. Certainly, preventing vibration from occurring precludes damage from vibration. On the other hand, the presence of vibration does not automatically infer damage.

SECONDARY CHECK

The primary check procedure examines only the most obvious causes of tube vibration. There are additional secondary checks that can be made. These include the velocities in the region of the inlet and outlet nozzles, high values of the cross flow ρu^2 , and the parallel-flow and cross-flow amplitudes. The leakage and bypass streams can be checked to see if they are high and may lead to local vibration problems. A check can be made to ensure that the acoustic frequency and tube natural frequencies are not in resonance.

VIBRATION DAMAGE RELATIONSHIP

Vibration cannot be equated to tube damage. Many exchangers vibrate but do not experience tube failure. Damage is known to result from fatigue, tube-to-tube collision, and baffle-to-tube cutting. Fatigue is better understood than the other two mechanisms. Tests that simulate the motion, forces, tube materials, and support materials, etc., found in real exchangers are needed to answer questions about low-amplitude (low-stress) vibration for a very large number of cycles.

DESIGN CONSIDERATIONS

At the design phase there are a number of changes that can be made to reduce the vibration problem. Most require a compromise to satisfy all of the requirements with added cost a likely result. When an exchanger already in service has a serious problem, there are still some improvements that can be made in the field.

POSSIBLE DESIGN CHANGES TO CORRECT ANTICIPATED VIBRATION PROBLEMS

If, as the result of a vibration analysis, it is found that vibration is probable, design elements can be reconsidered from the standpoint of vibration. Assuming that the process conditions cannot be changed, the following changes should be considered:

Reduce the shell-side velocities

If the flow rate of the shell-side fluid is fixed, velocities can be reduced by increasing the tube pitch or using TEMA X or J shell styles. This is particularly attractive when the design is pressure drop limiting, but may result in a larger shell.

Increase the tube natural frequency

The most effective way to increase the tube natural frequency is to reduce the longest unsupported span length. Reducing the span length by 8% increases the natural frequency by more than 5%. Secondary effects can be produced by changing tube materials and increasing tube wall thickness, but neither of these will greatly increase the natural frequency within industrially acceptable limits. The natural frequency can be increased by lacing or by driving wedges between tubes to prevent movement. This technique has been particularly effective in controlling vibration in the bend region of U-tube bundles.

Reduce nozzle velocities

The nozzle sizes can be increased to reduce nozzle velocities. If an impingement plate is provided to prevent erosion, be sure that the velocities leaving the edges of the impingement plate are not excessively high. A support plate at the centerline of the nozzles gives added tube support near the mid-span at the end zone. Annular distributors and vapor belts are effective ways of reducing the velocity of the shell-side fluid entering the bundle.

Change baffle type

The no-tubes-in-window (NTIW) type bundle can have its baffles widely spaced, as every tube is supported by every baffle. Further, the NTIW construction permits intermediate support plates between each baffle without materially affecting the heat transfer or pressure drop performance. A change to a multi-segmental baffle will reduce the velocity. Although not common practice, the rod baffle type bundle has been used in low-pressure-drop exchangers.

Add detuning baffles

Acoustic vibration problems can easily be corrected in the design phase by the use of detuning baffles to reduce the characteristic length.

Reduce wear at baffles

Although the use of tighter tube-to-baffle clearance and thicker baffles do not materially change the tube natural frequency, they do reduce damage due to cutting and increase the system damping. A change of baffle material can sometimes reduce tube damage if originally the material was considerably harder than the tube material.

POSSIBLE SOLUTIONS FOR OBSERVED VIBRATION PROBLEMS

Once an operating heat exchanger is found to have a vibration problem, it should be determined if the vibration is transmitted to the exchanger from some external source. If it is ascertained that the vibration is flow induced, there are a number of possible solutions to be considered:

Plug leaking tubes

A temporary fix for leaking tubes, whatever the cause, is to drive plugs into the tube ends to seal off the leaking tube. Replacing a tube in the field is generally not practical.

Reduce shell-side flow rate

Flow-induced vibration often can be stopped by lowering the shell-side flow rate. This will be acceptable only if it can be tolerated within the operating requirements of the plant.

Insert bars, lacing, or wedges to stiffen the bundle

The natural frequencies of existing exchangers can be increased by inserting lacing or driving wedges between the tubes to restrict movement. This can result in altered thermal and pressure drop performance. One region where lacing and wedging is often used is in the U bend.

Remove tubes to create bypass lanes

This procedure can reduce the cross-flow velocities enough so that vibration can be controlled. However, this must be done with care, as it can significantly reduce the overall heat transfer and pressure drop performance.

Replace the tube bundle

As a last resort, redesign and install a new bundle that circumvents the vibration problem.

Vibration problems are best prevented rather than corrected. A careful analysis at the design stage can greatly reduce future vibration problems.

References:

Heat Exchanger Design Handbook – Spalding and Taborek

RISK BASED INSPECTION (As applicable to pressure vessels and storage tanks)

Risk-Based Inspection (RBI) is an analysis methodology and process that, as opposed to condition-based inspection, requires qualitative or quantitative assessment of the probability of failure (PoF) and the consequence of failure (CoF) associated with each equipment item, piping circuits included, in a particular process unit. A properly-implemented RBI program categorizes individual pieces of equipment by their risks and prioritizes inspection efforts based on this categorization.

Probability of Failure

Probability of Failure (POF) is likelihood that a piece of equipment will fail at a given time and an important part of effective risk analyses. POF is half of the equation when determining risk as part of Risk Based Inspection (RBI) methodology. The POF, calculated together with the Consequence of Failure (COF), helps operators establish the risk level for a particular piece of equipment and set inspection intervals based on the calculated risk.

POF is calculated for individual pieces of equipment by looking at the potential damage mechanisms it could be susceptible to, a general frequency of failures, and management system factors. More details on POF are provided in the American Petroleum Institute's Recommended Practice 580 - Risk Based Inspection (RBI), which contains directions on developing, implementing and maintaining an effective RBI program.

Consequence of Failure

Consequence of Failure (COF) is one part of the equation to determine risk as part of Risk Based Inspection (RBI) methodology. COF is calculated by reviewing and ranking the potential consequences for the equipment, personnel, environment, etc. in the event of equipment failure. The assessment shall consider all those consequences that may take place as a result of fluid release. So the consequences that could be expected are:

- Fire
- Explosion
- Release of poisonous substances
- Health impacts
- Environmental impacts and pollution

More details on COF are given in API RP 580 - Risk Based Inspection (RBI), which contains directions on developing, implementing and maintaining an effective RBI program.

WHY USE RBI?

RBI is used to identify and understand risk, risk drivers, and where equipment is in its lifecycle. RBI can indicate whether inspection is needed; however this requires additional data that is extremely targeted to reduce the underlying uncertainties associated with the risks about the current and future predicted damage state of the equipment. RBI should not be used to recommend any inspection when it will not improve knowledge about the damage state. In those cases, where PoF is driving the risk, RBI should point to other mitigation options such as replacement, repair, or other actions that satisfy the risk criteria.

RBI can be used to prioritize inspection-related activities, usually by means of nondestructive testing (NDT), in order to reduce the uncertainties around the true damage state of the equipment and the dynamics leading to such. The resulting inspection plan may outline the type and scheduling of inspection for an asset. In addition to NDE, additional risk mitigation activities identified by an RBI assessment might include a change in material of construction, installation of corrosion resistant liners, operating condition changes, injection of corrosion inhibition chemicals, etc.

Consistency and repeatability of analysis are critical to producing an effective RBI program, as RBI is based on relative risks. Caution should be used when mixing RBI platforms (e.g., using a qualitative method to perform the initial screening and quantitative methods to conduct the final analysis). Complementary methodologies must be calibrated against one another to ensure valid cut sets are achieved.

OVERVIEW OF RBI METHODOLOGY

RBI methodology uses equipment history and the likely consequences of equipment failure to determine Inspection regimes focused on actual risks, so as to prevent unsafe incidents from occurring. The RBI method adopted by most petroleum oil refineries is based on API 581 base resource document which involved representatives from a number of major oil and petrochemical companies, and a comprehensive statistical analysis of petrochemical facilities over a number of years. Due to the complexity of this RBI method, a computer-based assessment is commonly used.

Each piece of equipment is assigned a risk ranking based on the probability and consequence of failure. Figure 1 shows a typical risk matrix. Red boxes (no. 1 to 5) indicate high risk. Pink boxes (no. 6 to 12) indicate medium-high risk. Yellow boxes (no. 13-19) indicate medium risk. Green boxes (no. 20 to 25) indicate low risk.

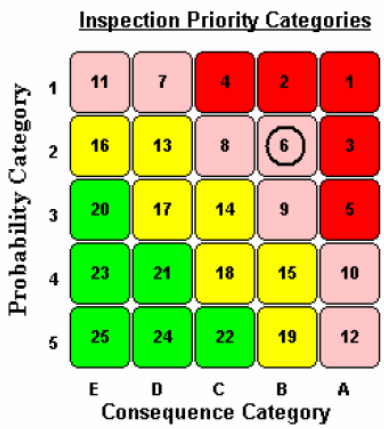

Figure 1: Typical Risk Matrix

Based on this risk ranking, one can decide whether to:

- Remove equipment from service and conduct inspection (if equipment is showing high risk and there are no suitable on-line inspection methods available that can help reduce the risk level).
- Apply appropriate on-line inspection methods for equipment to help reduce the risk level.
- Add to equipment inspection scope during turnaround to aid future risk assessments.
- Leave equipment on-line inspection and/or turnaround inspection scope at current level.
- Reduce equipment on-line inspection and/or turnaround inspection scope from current level.

A clear understanding of expected and possible damage mechanisms for equipment is required to conduct the risk assessment and apply suitable inspection methods to mitigate risk posed by them. A consistent approach is required for the reporting and assessment of damage mechanisms. API RP 571 provides valuable guidance in identifying and understanding relevant damage mechanisms in process equipment.

APPLICATION OF RBI TO CRUDE DISTILLATION COLUMN

Conducting RBI on a crude distillation column requires three basic processes:

1. Data collection and condition review
2. Criticality assessment and stakeholders input
3. Inspection planning and implementation

Data Collection and Condition Review

The RBI process begins by collecting accurate data on the equipment with regards to its process conditions, operating parameters, design, construction & installation considerations, drawings, maintenance activities, reliability, inspection history, modifications to the equipment/process, safety incidents, etc. The data collected is usually entered into a RBI software application that facilitates the RBI process for ease of computation, data management, inspection coordination and consistency between risk assessments.

Identifying the relevant damage mechanisms and their susceptible locations are particularly important for managing the risk of equipment failure. For a physically large piece of equipment such as a crude distillation column containing various process fluids, consideration is given to dividing the equipment into sub-components for this exercise. The column is divided into three sections:

i. Top section - consisting of the top head and top section of the distillation column with light fractions;

ii. Middle section - representing a large portion of the distillation column's cylindrical body containing medium fractions;

iii. Bottom section - including the bottom head and bottom section of the distillation column with heavy fractions.

Refer to Table 1 for damage mechanisms identified at these three sections.

Table 1: Damage Mechanisms in Crude Distillation Column, Measures for their Prevention/Mitigation and Inspection/Monitoring

Column Location	Damage Mechanisms	Prevention/Mitigation	Inspection/Monitoring
Top	Ammonium Chloride Corrosion	Use Ni and Ti based alloys, but some may suffer pitting corrosion Limit chlorides in tower feed through desalting and/or caustic addition to desalted crude Flush salt deposits on overhead line with water wash Control corrosion with filming amine inhibitors	Look for localized accumulation of ammonium chloride salts Monitor thickness loss - RT, UT Monitor ammonia and chlorides in feed streams Look for drop in thermal performance and pressure drop of downstream heat exchangers Use corrosion probes or coupons but deposits must be on them to work
	HCl Corrosion	Reduce chloride in feed by optimizing crude oil tank water separation & withdrawal, and crude desalting operation Upgrade carbon steel to Ni or Ti based alloys Water wash to quench overhead stream and help dilute condensing HCl concentration Inject caustic downstream of desalter to reduce amount of HCl going overhead. Use	Monitor general thinning on carbon steel, look for highly localized thinning where a water phase is condensing Look for signs of serious corrosion at mix points where dry chloride containing streams mix with streams containing free water or where water saturated streams are cooled below the dew point

		proper design and operating methods to avoid caustic SCC and fouling in feed preheat train. Inject various combinations of ammonia, neutralizing amines and filming amines in overhead line before water dew point.	Use UT scanning or profile RT to identify locally thinned areas Use an effective process & corrosion monitoring program Regularly check the water pH in the boot of overhead accumulator, also check chloride and iron content. Use strategically placed corrosion probes/coupons
Middle	Sulfidation	Upgrade to higher chromium alloy Use solid or clad 300 Series SS or 400 Series SS Use aluminum diffusion treatment of low alloy steel components	Monitor process conditions for increasing temperatures and/or changing Sulphur levels Use thermocouples and/or IR thermography to monitor temperature Check for thic kness loss with UT and RT Verify alloy used with PMI, prevent alloy mix-ups
	Naphthenic Acid Corrosion (NAC)	Change or blend crudes, upgrade metallurgy, utilize chemical inhibitors. Use alloys with higher molybdenum content, 317L SS Use high temperature NAC inhibitors	Use RT as primary detection method for localized erosion, followed by UT Monitor Total Acid Number (TAN) and Sulphur content of crude change and side streams Use corrosion probes/coupons, H2 probes Monitor streams for Fe and Ni content
	Erosion/ Erosion-Corrosion (E/EC)	Improve design on shape, geometry and materials selection Increasing substrate hardness using harder alloys or surface-hardening treatments Use corrosion-resistant alloys , alter process environment	Visual inspection, UT or RT - look for troublesome areas and detect thickness loss Specialized corrosion coupons, on-line corrosion monitoring electrical resistance probes On-line IR scans
	Chloride Stress Corrosion Cracking (Cl SCC)	Use resistant materials, properly apply coatings under insulation and design out stagnant regions to avoid chloride accumulation. High temperature stress relief 300 Series SS after fabrication considering the effects of sensitization. Hydrotest with low chloride content water, rinse & dry out thoroughly and quickly.	Visual inspection, PT, UT, phase analysis EC techniques For PT, may need polishing or high-pressure water blast RT may not be sufficiently sensitive to detect cracks except in advanced stages (network of cracks)
Bottom	Sulfidation	Refer to ”Sulfidation” above	Refer to ”Sulfidation” above
	NAC	Refer to ”NAC” above	Refer to ”NAC” above
	E/EC	Refer to ”E/EC” above	Refer to ”E/EC” above
	Cl SCC	Refer to “Cl SCC” above	Refer to “Cl SCC” above
	885°F (475°C) embrittlement	Use low ferrite or non-ferritic alloys or avoid material exposure to embrittling range Reverse embrittlement by heat treating to 1100°F followed by rapid cooling(1)	Impact or bend test sample removed from service Look for cracking formed during turnarounds, startup or shutdown when material is below ~200°F

			Check for increase in hardness

Note: (1) If the de-embrittled component is exposed to the same service conditions it will re-embrittle faster than it did initially.

Reviewing the data collected before using them for the criticality assessment underpins an accurate outcome; after all, there is a saying of "garbage in, garbage out". The RBI analyst should review the equipment inspection history being mindful of the relevant damage mechanisms, their effects on the equipment and effectiveness of inspections done in the past. Communication with stakeholders such as production, engineering, maintenance and inspection personnel is necessary to form a complete picture about the equipment and gain insight about it that is not often readily documented.

Equipment data and information relevant to the risk assessment are entered into the RBI software application before criticality assessment. A simple calculation of corrosion rates (e.g. external and internal) is often required and their entry into the RBI application should be accompanied by a clear explanation of their basis.

Criticality Assessment and Stakeholders Input

Criticality can be assessed for each three sections of the crude distillation column using section-specific information to determine the consequence and probability of failure. The consequence of failure usually considers flammability, toxicity and production loss, while the probability of failure usually considers the effects of internal & external corrosion, environmental cracking and other damage mechanisms. Other considerations for consequence and probability may be included depending on the configuration of RBI software application and user requirements. For the criticality analysis on each column section, a risk level can be evaluated for the section's current state and risk levels can be evaluated for the section's future states, depending on the number of equipment run length intervals being considered, or the level of conservatism used in the criticality analysis (e.g. Comparing risk using a more severe corrosion rate versus a less severe corrosion rate to reflect various repair or operating strategies). Figure 2 shows a typical criticality assessment for the top head of a crude distillation column.

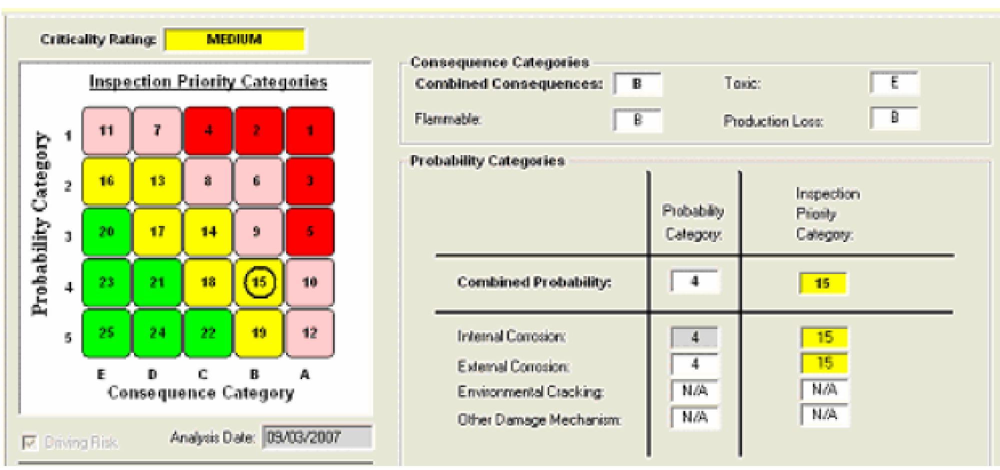

Figure 2: Typical Critical Assessment for the Top Head of Crude Distillation Column (Courtesy: Capstone RBMI, an RBI Software Application)

Stakeholders input from operations, engineering, environmental, safety and maintenance departments may be required to accurately reflect the consequence of failure. The RBI analyst must carefully interpret inspection records and seek input from experienced inspectors when addressing the probability of failure. A range of personnel are involved in equipment integrity assurance and reliability maintenance, effectively using the input from various stakeholders will help produce accurate criticality outputs for the equipment's current and future states.

Inspection Planning and Implementation

An inspection plan is developed based on the criticality evaluated in the current state and future states of the crude distillation column for its three sections, and the acceptable level of risk. For example, following equipment turnaround with an effective inspection scope (and repairs if required), one would not expect the equipment to still show high criticality. If a high criticality is evaluated for its future state depending on its run length, design, inspection history, relevant damage mechanisms, environment, operating & process conditions, and consequence parameters, then one or a combination of these must be addressed in order to reduce the criticality to a more acceptable level (e.g. medium or low).

For a crude distillation column, the inspection and monitoring measures presented in Table 1 should be included in the inspection plan along with regular on-line visual external inspection. The inspection scrutiny for each type of damage mechanism should be matched with the equipment history, as well as process and operating conditions.

APPLICATION OF RBI TO CRUDE STORAGE TANK

Application of RBI to a crude storage tank follows a similar process to the crude distillation column with the following exceptions:

Data Collection and Condition Review

The tank is divided into three sections:

1. Bottom - consisting of the annular plates and floor island plates. Sometimes, these are divided into two sub-sections for analysis;
2. Shell - the tank shell strakes; and
3. Roof - the roof plates.

Relevant damage mechanisms are:

- HCl Corrosion - related to crude heating
- Mechanical Fatigue - at shell corner joint with regular tank filling and emptying
- Atmospheric Corrosion - tank external components
- Corrosion under Insulation - for insulated crude tanks with heavy crude
- Microbiologically induced Corrosion - crude with hydrocarbon contaminants and water present
- Soil Corrosion - underside of tank bottom

Criticality Assessment and Stakeholders Input

Criticalities are assessed for the tank bottom, shell and roof representing the equipment's current state and future states.

Inspection Planning and Implementation

An inspection plan is developed based on the criticalities evaluated in the current state and future states of the tank for its three sections, and the acceptable level of risk. Implementing this plan and re-assessing this plan is crucial to keeping the risk ALARP (as low as reasonably possible).

CODES AND STANDARDS

International engineering standards and practices that relate to risk-based inspection include, but are not limited to, API RP 580 and 581, ASME PCC-3, and RIMAP. API RP 580 sets out the minimum guidelines for implementing an effective, credible RBI program. API RP 581 details the procedures and methodology of RBI.

CONCLUSIONS

With RBI, the equipment risk is managed by:

- Understanding the failure mechanism, susceptible locations, analyzing the risk of subcomponents, identify and address the highest risk, planning inspection and addressing the risk with stakeholders to prevent all undesired failures.
- Monitoring process variables and recognizing changes in the process such as upset conditions, change in process composition and operating limits. A management of change system is required.
- Knowing the level of inspection confidence (data representative of equipment's true condition and location) and inspection quality. Inspection confidence reflects how accurately inspection data represents the true condition of equipment, it relates to the damage type identified, inspection method used for its detection and location inspected. Inspection quality reflects the repeatability of the inspection process in terms of equipment access, instruments, operator, environment and process.

When applied and implemented properly, value is obtained from RBI by reducing equipment risk and cost.

References:

Application of Risk Based Inspection to Pressure Vessels and Aboveground Storage Tanks in Petroleum Fuel Refineries – Jenny Simpson

Overview of Risk Based Inspection - Inspectioneering

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TROUBLE SHOOTING LEAKING JOINTS

One of the best available tools to aid in determining the cause of leakage is a careful examination of the gasket in use when leakage occurred.

OBSERVATION	POSSIBLE REMEDIES
Gasket badly corroded.	Select replacement material with improved corrosion resistance.
Gasket extruded excessively.	Select replacement material with better cold flow properties, select material replacement with better load bearing capability - I.e., more dense.
Gasket grossly crushed.	Select material replacement with better load bearing capability, provide means to prevent crushing the gasket by use of a stop ring or re-design of flanges.
Gasket mechanically damaged due to overhang of raised face or flanged bore.	Review gasket dimensions to ensure gaskets are proper size. Make certain gaskets are properly centered in joint.
No apparent gasket compression achieved.	Select softer gasket material. Select thicker gasket material. Reduce gasket area to allow higher unit seating load.
Gasket substantially thinner on OD than ID.	Indicative of excessive "flange rotation' or bending. Alter gasket dimensions to move gasket reaction closer to bolts to minimize bending moment. Provide stiffness to flange by means of back-up rings. Select softer gasket material to lower required seating stresses. Reduce gasket area to lower seating stresses.
Gasket unevenly compressed around circumference.	Improper bolting up procedures followed. Make certain proper sequential bolt up procedures are followed.
Gasket thickness varies periodically around circumference.	Indicative of flange "bridging" between bolts or warped flanges. Provide reinforcing rings for flanges to better distribute bolt load. Select gasket material with lower seating stress. Provide additional bolts, if possible, to obtain better load distribution. If flanges are warped, re-machine or use softer gasket material.

References:

ASME Boiler and Pressure Vessel Code, Section VIII, Division 1 – 2019 Edition

BUILDING A BETTER TOMMORROW

It is becoming less practical for many companies to maintain in-house engineering staff. That is where we come in – whenever you need us, either for one-time projects, or for recurring engineering services. We understand the codes and standards, and can offer a range of services related to pressure vessels, tanks and heat exchangers.

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Pressure Vessels ● Heat Exchangers ● Storage Tanks

Oil & Gas ● Petrochemical ● Chemical ● Power ● Fertilizer