Improved Ferrule and Tube Design for Prevention of Film Boiling in SRU Waste Heat Boiler
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Background

The tube and ferrule design referred to in this paper is installed in the CITGO Corpus Christi Refinery East Plant SRU unit. This unit consists of two parallel trains with a combined original design capacity of 165 LTPD. The units were designed by Ford Bacon and Davis with the original startup being around 1976. The trains use a boiler condenser design with the thermal reactor effluent and the reactor condenser passes contained in a common shell, producing a nominal 50 psig steam. The basic process flow is shown below. Oxygen enrichment was added in the mid 1980s.

Feed to the SRU consists of amine regenerator acid gas derived primarily from a gas oil hydrotreater unit and fuel gas treating, and also Sour Water Stripper Gas. Over time, the refinery crude slate became progressively heavier and of higher sulfur content, resulting in higher sulfur production requirements for the SRU trains.

The combined waste heat boiler/condenser is the usual firetube boiler. The first pass tubes (waste heat boiler tubes) were originally 70 carbon steel tubes of 2 ¼-inch diameter, 0.109-inch wall thickness, and were welded into a carbon steel tubesheet of approximately 1-inch thickness. These were later changed to 55 carbon steel 3-inch tubes after the first failures. Ceramic ferrules of conventional design were installed in the first pass tube inlets. The first pass are the lowest tubes in the common boiler shell, with the second, third and subsequent condenser pass tubes above the first pass. Typical process values are given in Table 1, page 12.

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1 US Patent 7574981

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Description of Failures

The two trains operated successfully (at least with typical SRU reliability) until early 1996, at which time a series of tube failures began to occur. The dates of the recurring tube failures are given in Table 2, attached page 12.

The series of tube failures all occurred at the outlet of the first pass tube ferrules. Within several inches of the ferrule outlet, the tube wall became thin and collapsed inwards. The observed damage occurred within about 2 to 3 tube internal diameters of the outlet end of the ferrule.

Examination of failed tubes showed that there was a heavy sulfide scale on the inside of the tube, and that the carbide of the normal proeutectoid ferrite and pearlite microstructure had become partially spheroidized. The exposure time over which the individual failures developed could not be determined, although one of the failures occurred within about two weeks of the bundle having been inspected, indicating that the failure could develop over a relatively short period of time. These observations indicate an elevated tube temperature in the damaged area, most likely to have been in excess of 1000F. See Figures 4 through 9, pages 13, 14, and 15.

Film Boiling, Departure from Nucleate Boiling

Because the normal thermal profile across the boiler tube wall results in its temperature being within a few degrees Fahrenheit of the boiler water temperature, we concluded that the most likely cause of the locally elevated temperature of the boiler tubes was the result of film boiling at this location. During film boiling, the convective heat transfer from the tube to the boiler water drops drastically. The individual failure events are believed to have occurred in the following way: film boiling develops, the tube temperature increases to a high value, sulfidation occurs on the inside of the tube, the tube wall becomes thin, and the tube collapses when it becomes sufficiently thin.

Attempts to identify any particular or individual process events that preceded and initiated the individual failures were unsuccessful. This effort included calculating and trending the mass flux, heat transfer, oxygen usage, gas temperature, and steam make. Examination of the boiler water pressure and level did not identify any initiating events either. In the absence of any identifiable events, we concluded that the increasingly higher loads on the unit had resulted in a tendency for film boiling to occur. To our knowledge, the mass and heat flux on this unit were larger than typically recommended values. As of 2017, process data is only available beginning as of 1992. But using the available data starting in 1992, the mass flow, the oxygen content of the combustion air, and the dates of the film boiling failures are given in Figure 10, page 16.

Film boiling occurs when a critical heat flux is exceeded. To attempt to describe the physical process, steam is generated at the tube surface faster than boiler water can be re-supplied, which then results in the tube being insulated from the boiler water by a layer of steam. The heat transfer from the tube to the steam decreases, with the result that the tube wall temperature increases.

In normal operation, with convection by nucleate boiling heat transfer, the external convective coefficient of the boiling water is very large (1000 BTU/hr/ft² +) compared to the convection coefficient of the internal process gas flow (about 20 BTU/hr/ft² for this
The resulting thermal profile is one in which the tube wall temperature is very close to the boiler water temperature. But if the film boiling condition develops, the external convection coefficient drops to a value closer to the internal convection, which then causes the tube wall temperature to increase significantly.

The pressure of the steam is significant in determining the critical heat flux for the onset of film boiling. As the steam pressure increases, the critical heat flux for film boiling also increases. The nominal 50 psig boiler pressure is unfavorable for this boiler, making it more susceptible to the development of film boiling than a higher pressure waste heat boiler.

The heat flux through the tube wall is primarily related to the result of internal convection. And the internal convection is proportional to the mass flow through the tube (actually proportional to the 0.8 power of the mass flux using the Sieder Tate or other convective correlations), as well as the temperature difference between the process gas and the tube, which increased due to oxygen enrichment. These factors led us to conclude that the onset of the film boiling failures was related to the progressively higher loads on the unit, or the result of capacity creep.

Critical Heat Flux

![Predicted Critical Heat Flux](image)

Figure 2. Critical Heat Flux for Pool Boiling and for Tube Bundle

The critical heat flux for the tube bundle is reduced from the “theoretical” value for a submerged single surface, but is still significantly higher than the nominal value in the WHB tube. Turbulence at the tube exit is apparently the cause.

The critical heat flux for the development of film boiling on a single surface submerged under pool boiling conditions is given above in Figure 2 as calculated by three available correlations. The three sets of calculated data are plotted as the top “curve” in Figure 2, in which it can be seen that the three sets of data match each other fairly well.

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2 The several equations, Zuber, Mostinski, and Rohsenow Griffith, for the critical heat flux for a single surface are attached as Appendix 1.
The critical heat flux is dependent upon the steam pressure (the critical heat flux is basically proportional to the square root of the steam density).

In a tube bundle, the critical heat flux is reduced compared to the critical flux value for an isolated surface, presumably by the resistance of the tubes to allow steam to escape and for water to resupply by natural convection. Two estimates of the critical heat flux using a modified Zuber\textsuperscript{5} equation with the 3-inch diameter tubes are given also, first using the count of 55 tubes and then using 110 tubes (twice the WHB pass tubes) to attempt to account for the effect of the condenser tubes in the passes above the WHB pass. The resulting estimates of the critical heat flux for the 55-tube bundle with 50 psig steam are depicted in Figure 2 and are on the order of magnitude of about 200,000 to 350,000 BTU/hr/ft\textsuperscript{2}. Recognizing that there is uncertainty in these calculated values, one can conclude that the expected value of the heat flux to cause film boiling is clearly well above the nominal heat flux in the tube bundle.

Average Heat Flux

During the time period in which the failures occurred, the calculated nominal heat flux in the tube (for well developed flow downstream of the ferrule outlet region) varied with operating conditions but ranged from about 40,000 BTU/hr/ft\textsuperscript{2} to 55,000 BTU/hr/ft\textsuperscript{2}, based on the sum of convective and radiant transfer. The convective flux and internal convective coefficient are given by:

$$\dot{Q}/A = h(T_g - T_{wall})$$

where

$$h = \frac{k}{d} * 0.023Re_d^{0.8}Pr^{0.3}$$

Convection dominates the heat transfer with radiation from the gas contributing no more than 10%. These heat flux values are believed to be reasonably accurate because they were confirmed by calculating the heat loss down the length of the tube and using the measured outlet temperature of the process gas to verify the thus calculated tube outlet temperature.

Role of Turbulence at Exit of Tube

The average heat flux in the tube (40,000 to 55,000 BTU/hr/ft\textsuperscript{2}) is much lower than the estimated flux value to cause film boiling (200,000 to 350,000 BTU/hr/ft\textsuperscript{2}). This is confirmed by the fact that most of the length of the tube operated without damage, indicating heat transfer by nucleate boiling conditions, with only the first few inches of the tube at the outlet of the ferrule being damaged. Because the critical heat flux to cause film boiling is estimated to be so much larger than the nominal or average heat flux, it is logical to conclude that there must be a factor operating at the exit of the ferrule to cause higher heat flux in this area.

The primary factor causing the higher heat flux at the outlet of the ferrule is identified as the discontinuity or step change in the diameter of the flow channel at this location. Residual turbulence from the flow entering the ferrule could also be a minor or secondary factor, depending upon the length of the ferrule. But the major factor is as follows: the flow exits the ferrule and expands to the larger diameter of the tube, flow eddies occur, which increase the local convection as well as cause higher fluid flow pressure losses from friction between the flow and the tube wall. Using the geometry of the original 2 ¼-inch diameter tubes from 1996, the magnitude of this local magnification is estimated to be a factor on the order of about 500%, as follows:

\textsuperscript{5} The modified Zuber equation is in Appendix 1.
Figure 3. Turbulence at ferrule exit, 1996 tube, 2-1/4 inch diameter.
This sketch shows the geometry of the 1996 design tube and ferrule. Overheating damage to the tube was observed within about 3 diameters of the end of the ferrule, so it can be inferred that the higher convective transfer rates occurred within this zone, called here the “discharge zone”. Downstream of the discharge zone, the flow is “well developed,” with convection given by the normal convective calculations (Sieder-Tate\textsuperscript{4} or other.)

Because fluid friction and convective heat transfer are caused by the same physical processes, in turbulent flow, the friction and the heat transfer can be correlated. The development below basically attempts to estimate the heat transfer at the tube outlet by determining the pressure drop associated with the discontinuity at the tube exit, and then determining a friction factor for the exit region by assuming that the pressure drop occurs over the distance of the observed overheating. This uses the Reynolds analogy\textsuperscript{5} between pressure drop and convective heat transfer in a tube, which is:

\[
\frac{Nu}{RePr} = \frac{f}{8}
\]

Rearranging:

\[
Nu = \frac{RePrf}{8}
\]

\[
hd \quad RePrf
\]

\[
k = \frac{RePrf}{8}
\]

\[
h = \frac{kRePrf}{8d}
\]

Using the subscript “fo” for ferrule discharge zone (ferrule outlet), the convective heat transfer in the discharge region immediately downstream of the ferrule outlet is then:

\[
h_{fo} = \frac{k_{fo}Re_{fo}Pr_{fo}f_{fo}}{8d_{fo}}
\]

\textsuperscript{4} Holman, page 178.
\textsuperscript{5} Holman, page 160.

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Likewise, using the subscript “df” for developed flow, the convective heat transfer in the region downstream of the ferrule outlet is (this is the region of nominal heat transfer in the tube, which is characterized by about 40,000 to 55,000 BTU/hr/ft² as calculated using a Sieder Tate correlation for the subject WHB pass):

$$h_{df} = \frac{k_{df} Re_{df} Pr_{df} f_{df}}{8d_{df}}$$

The ratio of the convective heat transfer in the ferrule outlet region to the well developed flow is:

$$\frac{h_{fo}}{h_{df}} = \frac{k_{fo} Re_{fo} Pr_{fo} f_{fo}}{k_{df} Re_{df} Pr_{df} f_{df}}$$

The thermal conductivity of the process gas, the Reynolds number of the flow, the Prandtl number of the process gas, and the diameter of the flow channel are identical or very similar between the region immediately downstream of the ferrule outlet and the developed flow. This allows these quantities to be removed. The ratio of the convection in the ferrule outlet to the developed flow is thus simplified to:

$$\frac{h_{fo}}{h_{df}} = \frac{f_{fo}}{f_{df}}$$

The friction factor for the well developed flow can be obtained from the standard Moody friction factor charts, and is taken to be about 0.018. The friction factor in the ferrule discharge outlet region is not as easily available, but is estimated by making an equivalence between the head loss in the discharge region using the two forms of the head loss equation, (1) the discharge coefficient equation and (2) the Darcy friction factor equation.

The discharge coefficient form of the head loss equation is:

$$head loss_{ferrule discharge} = K \frac{v^2}{2g}$$

For the ferrule in use with the original 2 ¼ inch tubes, the ferrule inside diameter was 1.385-inches, and the tube inside diameter was 2.03-inches. The discharge coefficient for this combination of ferrule and tube is determined⁶ to be K=0.29, using the following equation:

$$K = \left[1 - \frac{d_{ferrule}}{d_{tube}}\right]^2 = \left[1 - \frac{1.385}{2.03}\right]^2 = 0.29$$

If the head loss at the ferrule outlet is considered to take place over some short distance of tube downstream of the ferrule, then it can be alternately calculated using the Darcy equation as:

$$head loss_{ferrule discharge} = f_{fo} \frac{L_{fo} v^2}{d_{fo} 2g}$$

Comparing the discharge coefficient and the friction factor forms of the head loss equations, the discharge coefficient is equivalent to the friction factor times the ratio of the length to diameter:

---

\[
\frac{f_{fo}}{L_{fo}} = K \\
\frac{f_{fo}}{d_{fo}} = K \frac{d_{fo}}{L_{fo}}
\]

Inserting this discharge zone friction factor into the previous equation:

\[
\frac{h_{fo}}{h_{df}} = \frac{K \frac{d_{fo}}{L_{fo}}}{f_{df}}
\]

\[
\frac{h_{fo}}{h_{df}} = \frac{K \frac{L_{fo}}{f_{df} d_{fo}}}{f_{df}}
\]

The ratio between the convection in the discharge zone to the downstream zone can now be determined using known quantities. The discharge coefficient is 0.29, the friction factor is 0.018, and the length of the discharge zone is estimated to be equal to the length of the observed overheating damage in the tube, which for the 2-1/4 inch tube was \(L_{\text{damaged zone}} \approx \frac{d}{3}\). Substituting these values yields:

\[
\frac{h_{fo}}{h_{df}} = \frac{0.29}{0.018 \times 3} = 5.4
\]

The convective heat transfer in the ferrule outlet zone is thus estimated to be about 5.4 times as great as the convective heat transfer in the zone of well developed flow, for the 1996 tube and ferrule geometry. Using the geometry of the 3-inch tube shown in Figure 7, page 14, in which the region of outlet damage is about 2 diameters, yields a similar result.

Since the heat flux in the well developed downstream zone is known to be about 40,000 BTU/hr/ft² to 55,000 BTU/hr/ft², the corresponding heat flux in the ferrule outlet zone is expected to be in the range of 220,000 BTU/hr/ft² to 300,000 BTU/hr/ft². This value of heat flux is on the same order of magnitude as the expected critical heat flux to cause film boiling (as per the modified Zuber equation, see Figure 2), thus indicating that the discontinuity in the flow channel at the ferrule outlet is the likely cause of the observed film boiling, or at least the primary cause. It also implies that correction of the flow discontinuity will yield a correspondingly large reduction in the heat flux at the exit of the ferrule.

Attempts at Correction (all of which were unsuccessful)

Before installing the upgraded tube and ferrule design, there were several unsuccessful attempts to mitigate the problem, which are mentioned here for completeness. The first thing that was done was to change the first pass tube bundle from 70 tubes of 2 1/4-inch outer diameter to 55 tubes of 3-inch outer diameter. The goal was to reduce the mass flux from about 5 lbm/sec/ft² to 3.5 lbm/sec/ft² and thus the heat transfer. Internally Alonized tubes were also installed as an additional enhancement.
We also tried several different ferrule designs, including increasing the length and using ferrules with an outlet taper. (The remaining tapered end of a ferrule can be seen in Figure 6, page 14.) For a conventional non-tapered ferrule, the minimum discontinuity in the flow channel is the combination of the thickness of the ferrule insulation plus the ferrule wall thickness at the exit. But even a ferrule with an outlet taper (Figure 6) still results in a discontinuity in the flow channel because of the fact that the ferrule has to have a minimum wall thickness (the ferrule cannot taper to zero thickness) and the length of the taper is limited.

Although our attempts to identify the combination of process factors that led to the individual failure events had been unsuccessful, we also placed limits on several process variables, including mass flux, calculated heat flux, percent oxygen, first pass outlet temperature, and steam production rate.

The combination of the larger 3-inch Alonized tubes, the tapered outlet ferrules, and the limits on process variables apparently made some improvement, because there was a period of about 4 to 5 years between 1998 and 2003 with no failures. But both trains again experienced at least failure each in 2003 and 2004.

Improved Tube and Ferrule design

Following the failures of 2003 and 2004, we developed a design, shown in Figure 11, page 17, to avoid the discontinuity in the flow channel at the outlet of the ferrule. This design used an internal weld overlay to transition between the end of the ferrule and the tube. It provides a continuous flow channel without any discontinuities in the flow channel. The overlay contained a shoulder to receive and center the ferrule, and it also was machined and tapered so that there was no discontinuity to cause turbulence and increase the convective heat transfer.

The flow channel through the ferrule was designed to emulate a converging and diverging nozzle. This also minimizes the overall pressure drop through the ferrule.

High temperature corrosion resistance

The internal geometry of the tube wall at the outlet of the ferrule was achieved by making an internal weld overlay of the approximate profile and then machining it to more accurate dimensions. The overlay was of Fe-22% Cr-5.5% Al alloy composition. This alloy was chosen on the basis of its expected resistance to high temperature oxidation and sulfidation. Thus the weld overlay not only provides the geometry for a smooth flow channel to minimize the heat transfer and the potential for film boiling, but it also provides some ability for the tube wall to tolerate the film boiling condition, should it occur.

Ferrule

The high alumina ferrule was produced\(^7\) using an injection molded freeze casting process, which provided the ability for precise tolerances and a smooth surface. The ferrule is of solid hex head design and is wrapped with the conventional ceramic paper insulation. The outlet end of the ferrule fits into the bore of the overlay (just the ceramic, not the paper insulation). This positively locates the ferrule in the center of the tube and provides for a continuous and smooth flow channel transition between the bore of the ferrule and the bore

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\(^7\) Blasch Precision Ceramics
of the tube. The flow channel through the ferrule consists of a rounded entry, with a converging section, a constant bore throat, and then a diverging outlet section.

Performance

The improved tube and ferrule design was installed into one boiler condenser in November 2005 (B-train). The second boiler condenser was retrofitted in April 2007 (A-train). Since that time, the units have operated successfully with no boiler tube film boiling failures. The ferrules were removed to permit inspection of the tubes in B-train in 2013 and A-train 2016. The tubes were found to be in good condition during those inspections, so new ferrules were installed and the units returned to service.

Comments

We consider that the successful performance of the improved tube and ferrule design for the 12 years since 2005 indicates that the peak heat flux has been lowered enough to avoid the previously observed film boiling problem. We also made a water side improvement, not described here, at the same time as the installation of the improved tube and ferrule design, which complicates the strict assessment of the benefit of the tube and ferrule itself. But removing the large enhancement of the heat transfer at the discontinuity in the flow suggests that the new internal tube overlay and matching ferrule is the major contributor to the improved performance.

After installation of the improved tube and ferrules in the mid 2000’s, we maintained the then existing process limits. In 2016, we increased the process limits by about 10%. As of August 2017, because of permit limits, there is no further incentive for any additional rate increases. But considerations of the heat transfer suggest that the tube and ferrule combination is capable of significantly more rate at least from the standpoint of avoiding the film boiling. The ultimate capacity is unknown at this time, and would require operations testing or some type of analysis (CFD, for example) or both.

Potential applications for this are waste heat boiler tubes in which the capacity is limited by the potential for film boiling or in which there is a need for the best reliability attainable.

References

Appendix 1. Critical Heat Flux Equations

Zuber\(^8\):

\[
\left( \frac{q}{A} \right)_{\text{crit}} = 0.13 h_f \rho_g \left[ \frac{g \sigma (\rho_f - \rho_g)}{\rho_g^2} \right]^{\frac{1}{4}}
\]

Where:

- \( \left( \frac{q}{A} \right)_{\text{crit}} \) = critical heat flux, kW/m\(^2\)
- \( h_f \) = latent heat of vaporization water to steam, kJ/kg
- \( \rho_g \) = density of steam, kg/m\(^3\)
- \( \rho_f \) = density of water, kg/m\(^3\)
- \( g \) = acceleration of gravity, 9.8 m/sec
- \( \sigma \) = surface tension of water in contact with steam, N/m

Convert using 317 BTU/hr/ft\(^2\) per kW/m\(^2\)

Mostinski\(^9\):

\[
\left( \frac{q}{A} \right)_{\text{crit}} = 803 p_c \left( \frac{p}{p_c} \right)^{0.35} \left( 1 - \frac{p}{p_c} \right)^{0.9}
\]

Where:

- \( \left( \frac{q}{A} \right)_{\text{crit}} \) = critical heat flux, BTU/hr/ft\(^2\)
- \( p_c \) = critical pressure of steam, 3208 psia
- \( p \) = pressure of steam in boiler, psia

Rohsenow Griffith\(^10\):

\[
\left( \frac{q}{A} \right)_{\text{crit}} = C h_f \rho_g \left[ \frac{g}{g_c} \right]^{\frac{1}{4}} \left( \frac{\rho_f - \rho_g}{\rho_g} \right)^{0.6}
\]

Where:

- \( \left( \frac{q}{A} \right)_{\text{crit}} \) = critical heat flux, kW/m2
- \( C = 0.012 \) m/sec
- \( h_f \) = latent heat of vaporization water to steam, kJ/kg
- \( \rho_g \) = density of steam, kg/m\(^3\)
- \( \rho_f \) = density of water, kg/m\(^3\)
- \( g \) = local acceleration of gravity, 9.8 m/sec
- \( g_c \) = standard acceleration of gravity, 9.8 m/sec

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\(^8\) Sandlikar, page 1072, with an algebraic rearrangement for \( \rho_g \). Perry and Chilton page 10-21 give a similar equation they term Kutateladze Zuber but with a different coefficient of 0.18.

\(^9\) Perry and Chilton, page 10-21

\(^10\) Sandlikar, page 1071
Modified Zuber\(^{11}\) (to account for tube bundle)

\[
\left(\frac{q}{A}\right)_{\text{crit}} = K_b \frac{p_t}{d_o} \frac{h_{fg}}{\sqrt{N}} \left[ g \sigma (\rho_f - \rho_g) \rho_g \right]^{\frac{1}{2}}
\]

Where: \(\left(\frac{q}{A}\right)_{\text{crit}}\) = critical heat flux, kW/m\(^2\)

- \(K_b\) = constant, 0.41 for triangular pitch, 0.44 for square pitch
- \(p_t\) = tube pitch, m or inches
- \(d_o\) = tube diameter, m or inches (just be consistent with \(p_t\))
- \(N\) = number of tubes in bundle
- \(h_{fg}\) = latent heat of vaporization water to steam, kJ/kg
- \(\rho_g\) = density of steam, kg/m\(^3\)
- \(\rho_f\) = density of water, kg/m\(^3\)
- \(g\) = acceleration of gravity, 9.8 m/sec
- \(\sigma\) = surface tension of water in steam, N/m

Convert using 317 BTU/hr/ft\(^2\) per kW/m\(^2\)

\(^{11}\) Sinnott, page 751. Perry and Chilton page10-21 give another form of the equation, Palen Small, for conventional British units directly.
Table 1 Process Conditions (for one train, during period of tube failures)

<table>
<thead>
<tr>
<th>Process Variable</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Acid Gas, typical</td>
<td>100 MSCFH</td>
</tr>
<tr>
<td>SWS Gas, typical</td>
<td>50 MSCFH</td>
</tr>
<tr>
<td>Air, typical</td>
<td>185 MSCFH</td>
</tr>
<tr>
<td>Oxygen, typical</td>
<td>16 MSCFH</td>
</tr>
<tr>
<td>Furnace Temperature</td>
<td>2100-2350F measured</td>
</tr>
<tr>
<td></td>
<td>2500-2850F predicted</td>
</tr>
<tr>
<td>Mass Flow, maximum</td>
<td>8 to 9 lbm/sec</td>
</tr>
<tr>
<td>Mass Flux, max (2-1/4 inch tubes)</td>
<td>5</td>
</tr>
<tr>
<td>Mass Flux, max (3-inch tubes)</td>
<td>3.5</td>
</tr>
</tbody>
</table>

Table 2. Dates of Tube Failures by Film Boiling at Ferrule Exit

<table>
<thead>
<tr>
<th>Date of Failure</th>
<th>Train</th>
</tr>
</thead>
<tbody>
<tr>
<td>February 4, 1996</td>
<td>A train</td>
</tr>
<tr>
<td>February 27, 1996</td>
<td>B train</td>
</tr>
<tr>
<td>July 7, 1996</td>
<td>B train</td>
</tr>
<tr>
<td>August 21, 1996</td>
<td>A train</td>
</tr>
<tr>
<td>December 14, 1998</td>
<td>A train</td>
</tr>
<tr>
<td>June 7, 2003</td>
<td>A train</td>
</tr>
<tr>
<td>February 29, 2004</td>
<td>B train</td>
</tr>
<tr>
<td>April 15, 2004</td>
<td>A train</td>
</tr>
</tbody>
</table>
Figure 4. Tube bundle with failures in 2 ¼-inch diameter tubes of August 1996. The tubesheet was replaced at this time to install 3-inch tubes.

Figure 5. Tube failure of 3-inch diameter tubes in February 2004. The tubes had been inspected and found to be in good condition two weeks prior to this failure.
Figure 6. **Sectioned tube showing broken end of ferrule and damage.**

*Flow entered at the right in this photograph. Note the end of the ferrule still in place in the tube.*

Figure 7. **Split tube, 3-inch diameter, showing thinning and collapse.**

*The ferrule was to the right; flow entered from the right. Note the thick sulfide scale downstream from the thin area. Sulfide scale is presumed to have been present everywhere downstream of the ferrule end but has fractured and been removed during either the tube failure or the tube removal and cutting. The damage occurs within two tube diameters of the ferrule outlet.*
Figure 8. Sulfide scale present at inner surface of tube.

Figure 9. Microstructure of tube at interface with scale.

*The deterioration of the carbide lamella in the pearlite microstructure indicates overheating.* (Magnification not recorded.)
Figure 10. Mass Flow, oxygen content of the combustion air, and film boiling failure occurrences.

(Mass flow points are plotted in blue and are generally the upper set of points on both graphs; percent oxygen is plotted in green and are generally the lower set of points on the graphs.) The mass flow is calculated using the averaged daily PI data from 1992 through 2017 for the acid gas, SWS gas, combustion air, and supplemental oxygen. The composition of the acid gas was taken as 0.85 H2S, 0.1 H2O, and 0.05 CO2. The SWS gas was taken as 0.46 H2S, 0.22 NH3, and 0.32 H2O. At the higher values of mass flow, about 9 lbm/sec, the corresponding mass flux for the original 2-1/4 tubes is 5.0 lbm/ft2/sec. With 3-inch tubes, the mass flux is reduced to 3.5 lbm/ft2/sec.
Figure 11. Improved Tube and Ferrule Design, Sectional View.

The internal weld overlay contains a machined shoulder to receive the end of the ferrule. This locates the ferrule to be concentric with the bore of the tube, and assures a smooth transition for the flow at the exit of the ferrule. The bore of the ferrule itself initially converges into a constant bore “throat”, followed by a diverging section. The slope of the diverging section of the ferrule matches the slope of the diverging weld overlay. The weld overlay essentially tapers to zero thickness, so there is no step change or discontinuity of the flow channel.