Pressure Vessel Newsletter

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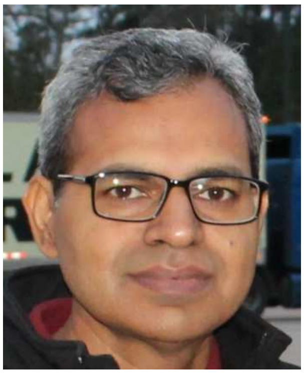
From The Editor's Desk:

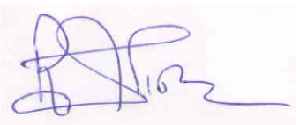


This issue features an article that describes an accident at a refinery and the lessons learnt from that accident. While we in the industry do everything possible to prevent such accidents, they still happen because we either can't compensate for all the failure *scenarios* or we don't know enough about all the failure *mechanisms*. As we learn from each incident, we increase our knowledge base and that lowers the probability that similar incidents will occur in the future. At the same time, we are pushing the envelope, operating equipment at higher pressures and temperatures, and pushing into areas where our knowledge base is sketchy at best. So it is unlikely that we will ever reach a stage where our industry will be incident-free.

Compensating for all failure scenarios is no easy task. We are used to working in silos and try to sort out the interface issues between the silos by conducting regular meetings until we have a good designat hand. Then we pad our design by employing various safety devices just in case we overlooked something. But even this cannot guarantee that there will be no incidents because we can only plan for things that we know about. Unfortunately, there is a lot we don't know about – the failure mechanisms that we either don't know exist or don't know enough about and therefore can't protect against them.

Let us say that we have designed the system for a particular pressure and temperature. We have looked at all the scenarios, and ensured that none of the scenarios result in a higher pressure or temperature. Then we have gone a step further and installed the safety devices that open if the pressure and temperature exceed their design limits. What can go wrong now? Have we overlooked anything? May be not, but what if we are working with a process we don't know much about which results in a damage mechanism thatis unknown to us. Therefore we can't design preventive measures to mitigate this damage mechanism. Measures that would have prevented failure at pressures much below the design pressure.

API RP 579/ ASME FFS is a very important living document that helps address many of the failure *mechanisms* that we will ever encounter, and helps bring the number of incidents such as that described in this Issue to a minimum. However, we would do well to stack the odds in our favor by adhering to a tried and tested adage – that of keeping things simple and minimizing complexities. The simplest of designs are very often the best ones.



[The picture on the cover is courtesy of Fourinox in Green Bay, Wisconsin.]

In this issue…

DESIGN OF SADDLE SUPPORTS

The problem of horizontal pressure vessels supported on saddles is considered as a beam on two single supports. The stresses exerted in the pressure vessel are:

1. Longitudinal stresses within the shell by the overall bending of the vessel
2. Shear stresses on the supports by the transmission of the loads on the supports
3. Circumferential stresses within the shell

The stress calculation method in ASME Section VIII, Division 2 (ASME VIII-2) is based on the work of Zick and covers modes of failure by excessive deformation and elastic instability. Figure 1 show a typical horizontal vessel supported on two saddles.

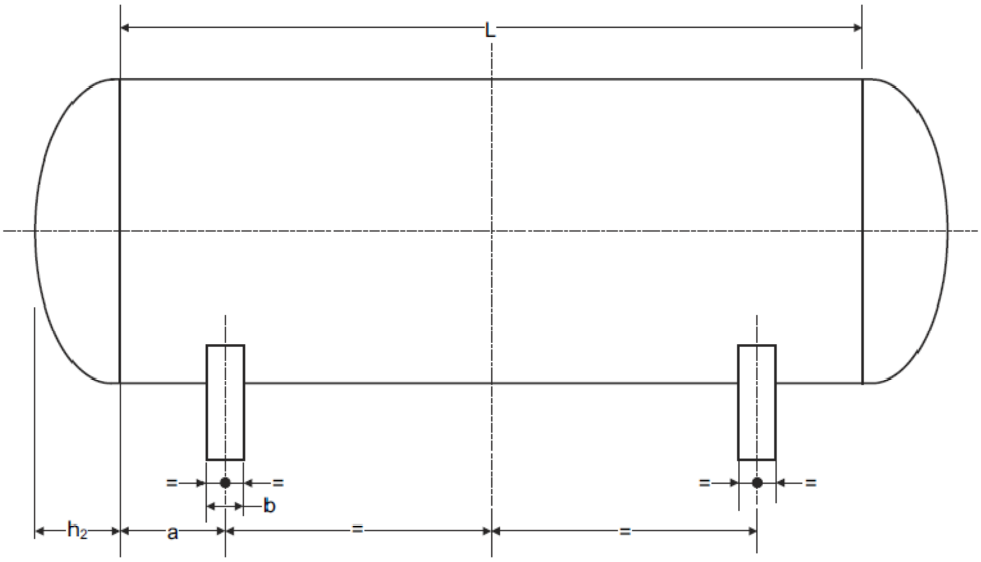


Figure 1: Typical Horizontal Vessel Supported on Two Saddles

In selecting the location of saddle supports, it is preferable to make dimension ‘a’ less than the dimension ‘R’, the inside radius of the shell, in order to take advantage of the stiffening effect of the head. Dimension ‘a’ is often selected so that a = 0.4R. In any case, ‘a’ should never exceed 20% of the dimension ‘R’. Further, saddle supports should be designed to provide continuous support for at least one-third of the shell circumference, or θ > 120 degrees.

A cylindrical vessel with dished closure at ends may be treated as an equivalent cylinder having a length equal to $(\mathrm{L} + 4/3\mathrm{h}_2)$. This approximation assumes that the weight of the head and the fluid contained in it is equal to 2/3rd of the weight of cylinder of length h_2 and the fluid contained in it. This approximation is valid for hemispherical heads and for elliptical heads.

The weight of the fluid and the vessel may be considered to be a uniform load equal to the total weight divided by the equivalent length.

$$\mathrm{w} = \frac{2\mathrm{Q}}{\mathrm{L}+\frac{4}{3}\mathrm{h}_2}$$

Q = Total load per saddle, lb

= Total weight divided by two

w = Uniform load, lb per feet

In the loaded condition, the shell, over the distance L, behaves as a uniformly loaded beam. The load of the heads introduces a shear load at the junction of the head and the shell equal to 2/3 h_2w. This load produces a vertical coupling acting at a distance of 3/8 h_2 from the point of tangency, and a horizontal couple acting with a lever arm of R/4.

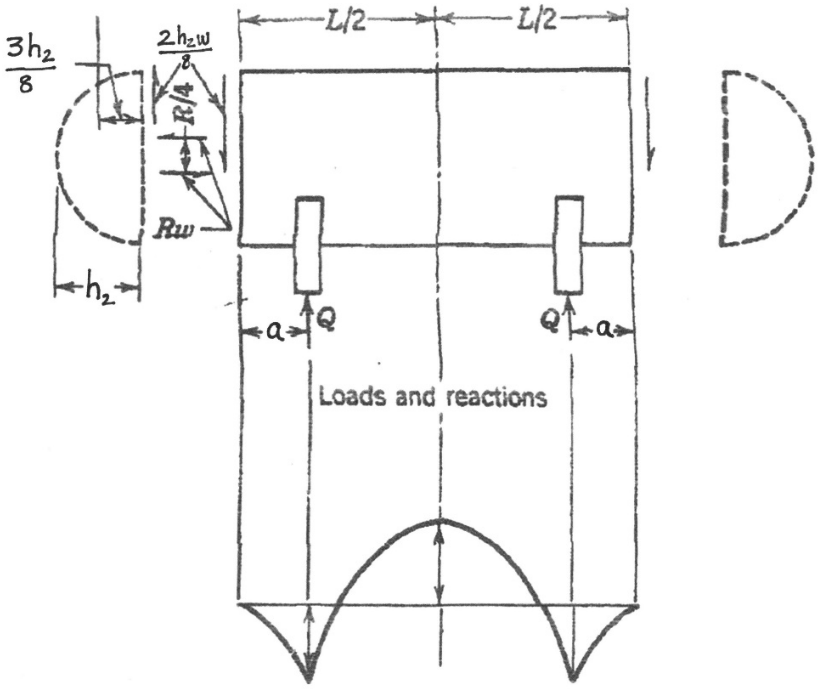


Figure 2: Cylindrical Shell Acting as a Beam over Supports, According to Zick

Reinforcing Plates

If a reinforcing plate is included in the design to reduce the stresses in the shell at the saddle support, then the width of the reinforcing plate, b_1, shall satisfy the following equation:

$$b_1 = \min\left[\left(b + 1.56\sqrt{R_m t}\right), 2a\right]$$

The supporting arc length must satisfy the following equation:

$$\theta_1 = \theta + \frac{\theta}{12}$$

A typical reinforcing plate arrangement is shown in Figure 3.

In the equations above, R_m is the mean radius of the shell, and t is the thickness of the shell. The other terms are as shown in Figures 1 and 3.

Stiffening Rings

Stiffening rings may be provided at the saddle support location, on either the inside or the outside of the shell. The stiffening rings may be mounted in the plane of the saddle as shown in Figure 4, or they be mounted on each side of saddle support equidistant from the saddle support as shown in Figure 5.

If stiffening rings are mounted on each side of the saddle support, the spacing between the two stiffening rings, h, cannot be greater than R_m. If $h \le 1.56\sqrt{R_m t}$ (as shown in Figure 4, sketch (c)), then both of the stiffening rings shall be considered as a single stiffening ring situated in the plane of the saddle in the stress calculations.

Moment and Shear Force

As in the case of overhanging beam with two supports, two maximum bending moments exist in the longitudinal direction of the vessel. One maximum occurs over the saddle supports, and the other maximum occurs in the center of the vessel span. If the vessel is composed of a cylindrical shell with a formed head at each end and 'a' ≤ 0.25L, then

Maximum moment over the supports, M_1 is given as:

$$M_1 = -Qa\left[1 - \frac{1-\frac{a}{L}+\frac{R_m^2-h_2^2}{2aL}}{1+\frac{4h_2}{3\,L}}\right]$$

Maximum bending moment at the center of the span, M_2 is given as:

$$M_2 = \frac{QL}{4}\left[\frac{1+2\frac{R_m^2-h_2^2}{L^2}}{1+\frac{4h_2}{3\,L}} - \frac{4a}{L}\right]$$

Shear force at the saddle, T is given as:

$$T = \frac{Q(L-2a)}{L+\frac{4h_2}{3}}$$

If the vessel supports are not symmetric, or more than two supports are provided, then the highest moment in the vessel, and the moment and shear force at each saddle location must be evaluated. The moment and shear force may be determined using (i.e. beam analysis with a shear and moment diagram). If the vessel is supported by more than two supports, then differential settlement should be considered in the design.

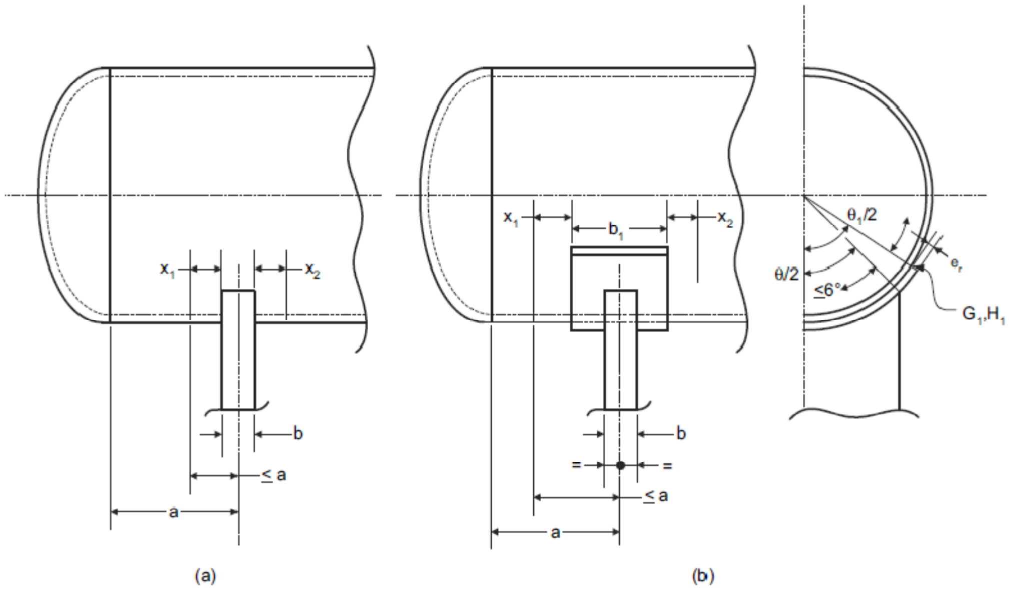


Figure 3: Typical Reinforcing Plate Arrangement

Longitudinal Stress

The longitudinal membrane plus bending stress in the shell between the supports are given by the following equation:

$$\sigma_1 = \frac{PR_m}{2t} - \frac{M_2}{\pi R_m^2 t} \qquad \text{(Top of the shell)}$$

$$\sigma_2 = \frac{PR_m}{2t} + \frac{M_2}{\pi R_m^2 t} \qquad \text{(Bottom of the shell)}$$

The shell may be considered as suitably stiffened if

a. It incorporates stiffening rings at or on both sides of the saddle support; OR

b. The support is sufficiently close, defined as a ≤ 0.5 R_m, to the torispherical or ellipsoidal head, flat cover or tubesheet

The value of longitudinal stress will depend on the rigidity of the shell at saddle support.

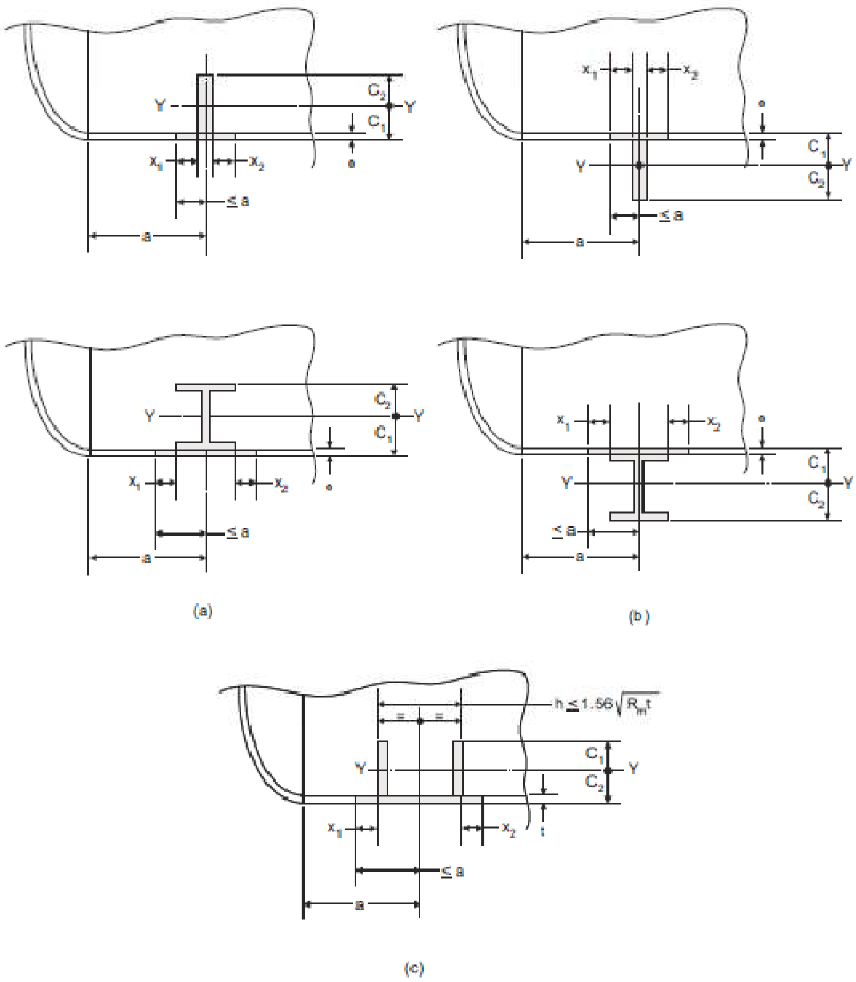


Figure 4: Horizontal Shell with Stiffening Rings in the Plane of Saddle

<u>Stiffened Shell</u>

The maximum values of longitudinal membrane plus bending stress at the saddle support are given as:

$$\sigma_3 = \frac{PR_m}{2t} - \frac{M_1}{\pi R_m^2 t} \qquad \text{(Top of the shell)}$$

$$\sigma_4 = \frac{PR_m}{2t} + \frac{M_1}{\pi R_m^2 t} \qquad \text{(Bottom of the shell)}$$

Unstiffened Shell

Above each saddle support, circumferential bending moments are produced which permit the unstiffened upper portion to deform. The deformation makes this portion of the cylindrical shell ineffective as a beam, and reduces the effective cross-section in the same manner as if a horizontal section were cut from the vessel some distance above the saddle.

The maximum value of longitudinal membrane plus bending stress at the saddle support is given as:

$$\sigma_3^* = \frac{PR_m}{2t} - \frac{M_1}{K_1 \pi R_m^2 t} \qquad \text{(Points A and B in Figure 6)}$$

$$K_1 = \frac{\Delta + \sin\Delta.\cos\Delta - \frac{2\sin^2\Delta}{\Delta}}{\pi\left(\frac{\sin\Delta}{\Delta} - \cos\Delta\right)}$$

$$\Delta = \frac{\pi}{6} + \frac{5\theta}{12}$$

$$\sigma_4^* = \frac{PR_m}{2t} + \frac{M_1}{K_1^* \pi R_m^2 t} \qquad \text{(Bottom of shell)}$$

$$K_1^* = \frac{\Delta + \sin\Delta.\cos\Delta - \frac{2\sin^2\Delta}{\Delta}}{\pi\left(1 - \frac{\sin\Delta}{\Delta}\right)}$$

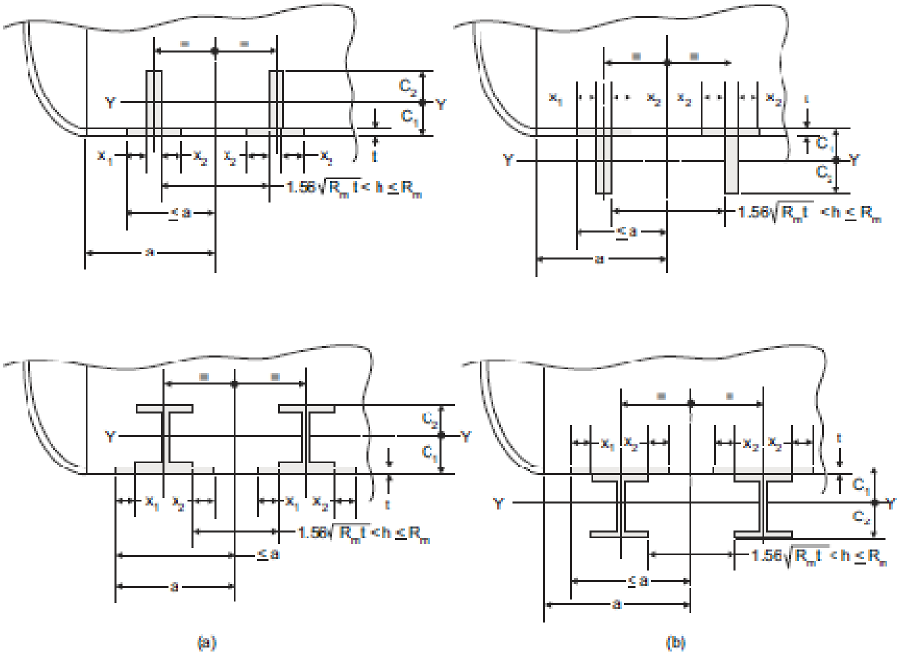


Figure 5: Horizontal Shell with Stiffening Rings on both Sides of Saddle

Acceptance Criteria

The absolute value of σ_1, σ_2, and σ_3, σ_4 or σ_3^*, σ_4^*, as applicable shall not exceed SE. If any of the stresses above are negative, then the absolute value of stresses shall not exceed S_c as given below where K = 1.0 for normal operating conditions and K=1.35 for exceptional operating or hydrotest conditions.

$$S_c = \frac{KtE_y}{16R_m} \qquad E_y = \text{Modulus of Elasticity}$$

Shear Stresses

Shell with Stiffening Ring in the Plane of Saddle Support

The shear stress is maximum at points C and D of Figure 6, Sketch (b) and is given by the following equation:

$$\tau_1 = \frac{T}{\pi R_m t}$$

The absolute value of τ_1 shall not exceed 0.8S for ferritic materials and 0.6S for all other materials. S is the allowable stress for the shell material at the design temperature.

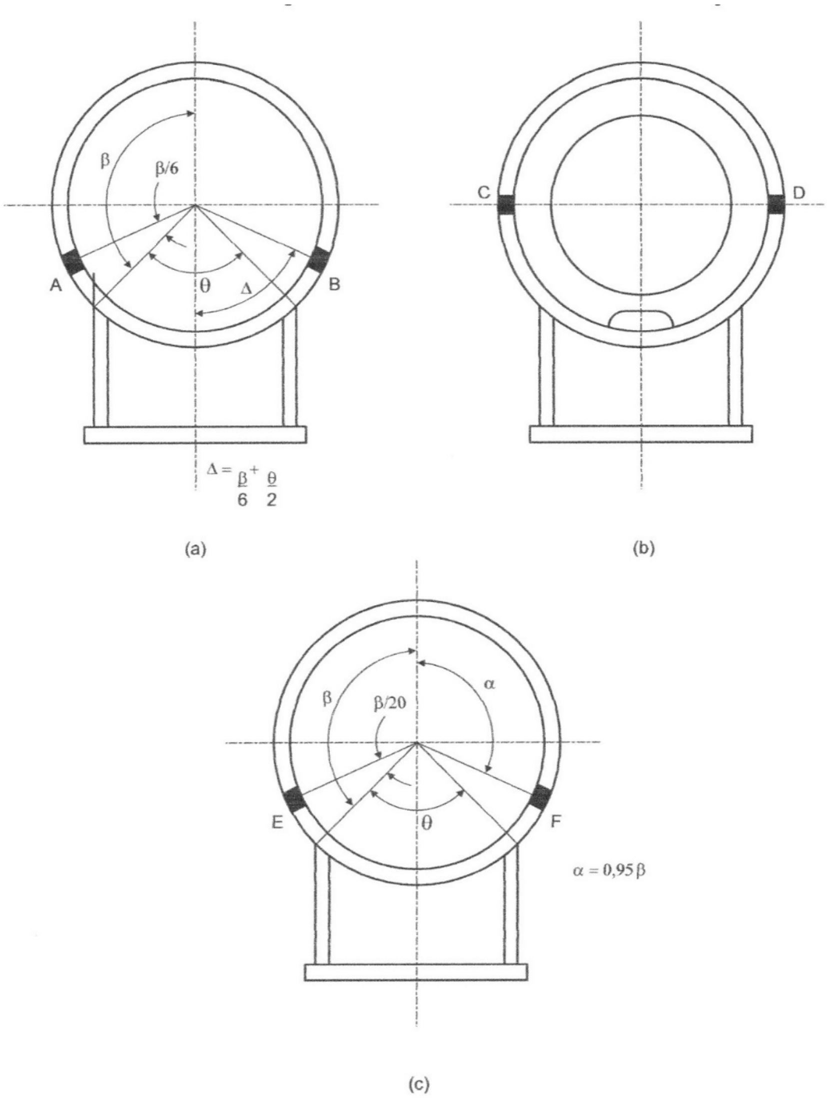


Figure 6: Locations of Maximum Longitudinal Normal Stress and Shear Stress in the Shell

Shell with Stiffening Ring on Both Sides of Saddle Support

The shear stress is maximum at points E and F of Figure 6, Sketch (c), and is given by the following equation:

$$\tau_2 = \frac{K_2 T}{\pi R_m t}$$

$$K_2 = \frac{\sin\alpha}{\pi - \alpha + \sin\alpha.\cos\alpha}$$

$$\alpha = 0.95\left(\pi - \frac{\theta}{2}\right)$$

The absolute value of τ_2 shall not exceed 0.8S for ferritic materials and 0.6S for all other materials.

Shell without Stiffening Ring(s) That is Not Stiffened by Formed Head, Flat Cover, or tubesheet ($a > 0.5R_m$)

The shear stress is maximum at points E and F of Figure 6, Sketch (c) and is given by the same equation as for τ_2 above.

The shear stress shall not exceed 0.8S for ferritic materials and 0.6S for all other materials.

Shell without Stiffening Ring(s) and Stiffened by Torispherical or Elliptical Head, Flat Cover, or Tubesheet ($a \leq 0.5R_m$)

The shear stress is maximum at points E and F of Figure 6, Sketch (c), and is given by the equations below. In addition to the shear stress, the membrane stress in the formed head, if applicable, shall also be computed using the equations shown below.

Shear Stress

$$\tau_3 = \frac{K_3 Q}{R_m t}$$ (In the cylindrical shell)

$$\tau_3^* = \frac{K_3 Q}{R_m t_h}$$ (In the formed head)

$$K_3 = \left(\frac{\sin\alpha}{\pi}\right)\left(\frac{\alpha - \sin\alpha.\cos\alpha}{\pi - \alpha + \sin\alpha.\cos\alpha}\right)$$

t_h = Thickness of head

The absolute value of τ_3 shall not exceed 0.8S for ferritic materials and 0.6S for all other materials. The absolute value of τ_3^* shall not exceed $0.8S_h$ for ferritic materials and $0.6S_h$ for all other materials. S_h is the allowable stress for the head material at the design temperature.

Membrane Stress in Torispherical or Elliptical Head Acting As a Stiffener

$$\sigma_5 = \frac{K_4 Q}{R_m t_h} + \frac{PR_i}{2t_h}$$ (Torispherical head)

$$\sigma_5 = \frac{K_4 Q}{R_m t_h} + \frac{PR_i}{2t_h}\left(\frac{R_i}{h_2}\right)$$ (Elliptical head)

$$\sigma_5 = 0$$ (Flat cover)

$$K_4 = \frac{3}{8}\left(\frac{\sin^2\alpha}{\pi - \alpha + \sin\alpha.\cos\alpha}\right)$$

The absolute value of σ_5 shall not exceed $1.25S_h$.R_i is the inside radius of spherical dome or torispherical head.

Circumferential Stress

Maximum circumferential bending moment

The distribution of circumferential bending moment at the saddle support is dependent on the use of stiffeners at the saddle location.

Shell without a stiffening ring or with stiffening ring in the plane of the saddle

The maximum circumferential bending moment is shown in Figure 7, Sketch (a), and is given by the following equation:

$$M_\beta = K_7 Q R_m$$

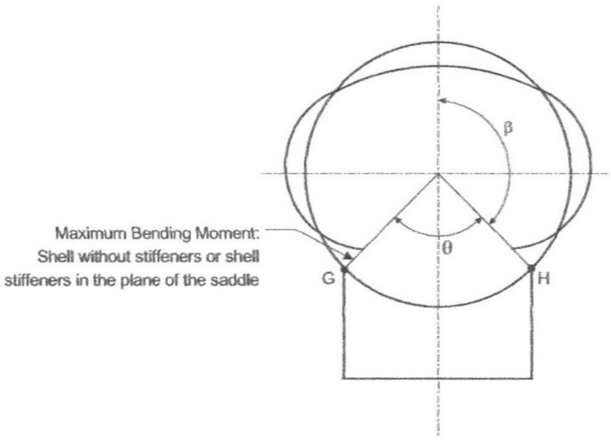
$K_7 = \frac{K_6}{4}$ when $\frac{a}{R_m} \leq 0.5$

$K_7 = \frac{3}{2}K_6\left(\frac{a}{R_m}\right) - \frac{1}{2}K_6$ when $0.5 < \frac{a}{R_m} < 1$

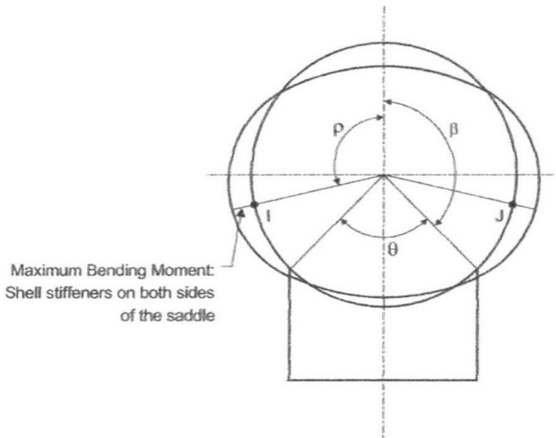
$K_7 = K_6$ when $\frac{a}{R_m} \geq 1$

$$K_6 = \frac{\frac{3\cos\beta}{4}\left(\frac{\sin\beta}{\beta}\right)^2 - \frac{5\sin\beta\cos^2\beta}{4\beta} + \frac{\cos^3\beta}{2} - \frac{\sin\beta}{4\beta} + \frac{\cos\beta}{4} - \beta\sin\beta\left[\left(\frac{\sin\beta}{\beta}\right)^2 - \frac{1}{2} - \frac{\sin2\beta}{4\beta}\right]}{2\pi\left[\left(\frac{\sin\beta}{\beta}\right)^2 - \frac{1}{2} - \frac{\sin2\beta}{4\beta}\right]}$$

$\beta = \pi - \frac{\theta}{2}$



(a)



(b)

Figure 7: Locations of Maximum Circumferential Normal Stresses in the Cylinder

Shell with stiffening rings on both sides of the saddle

The maximum circumferential bending moment is shown in Figure 7, Sketch (b), and is given by the following equation:

$M_\beta = K_{10}QR_m$

$$K_{10} = \frac{1}{2\pi}\left\{\rho\sin\rho + \cos\rho\left[\frac{3}{2} + (\pi - \beta)\cot\beta\right] - \frac{(\pi-\beta)}{\sin\beta}\right\}$$

$$\rho = \tan\rho[0.5 + (\pi - \beta)\cot\beta]$$

Values for ρ for a specified θ are shown in the table below:

θ	120°	130°	140°	150°	160°	170°	180°
ρ	93.667°	91.133°	87.833°	84.167°	79.667°	74°	66.933°

GENERAL NOTE:

$\rho = -158.58 + 7.86680\theta - 8.8037(10)^{-2}\theta^2 + 4.3011(10)^{-4}\theta^3 - 8.0644(10)^{-7}\theta^4$ for all values of θ that satisfy $120^o \leq \theta \leq 180^o$. This curve fit provides ρ in degrees

Width of cylindrical shell

The width of the cylindrical shell that contributes to the strength of the shell at the saddle location is given by the equation below.

$$x_1, x_2 \leq 0.78\sqrt{R_m t}$$

If the width x_1 extends beyond the limits in Figures 3, 4 or 5, as applicable, then the width x_1 shall be reduced so as not to exceed the limit given by above equation.

Circumferential stresses in the shell without stiffening ring(s)

1. The maximum compressive circumferential membrane stress in the shell at the base of the saddle support is given by the following equation:

$$\sigma_6 = -\frac{K_5 Qk}{t(b+x_1+x_2)}$$

$$K_5 = \frac{1+\cos\alpha}{\pi - a + \sin\alpha.\cos\alpha}$$

k = Factor to account for vessel support condition; k = 1 when vessel is resting on the support and k = 0.1 when the vessel is welded to the support

The stress σ_6 can be reduced by adding a reinforcement or a wear plate at the saddle location that is welded to the cylindrical shell in accordance with requirements discussed earlier in this module. The reduced stress can be expressed by the following equation:

$$\sigma_{6,r} = -\frac{K_5 Qk}{b_1(t+\eta t_r)}$$

b_1 = Width of reinforcing plate welded to the shell at saddle location

t_r = Reinforcing plate thickness

η = $\min\left[\frac{S_r}{S}, 1.0\right]$

The absolute value of σ_6 or $\sigma_{6,r}$ shall not exceed S.

2. The circumferential compressive membrane plus bending stress at points G and H of Figure 7, Sketch (a) is given as follows:

a. If $L \geq 8R_m$,

$$\sigma_7 = -\frac{Q}{4t(b+x_1+x_2)} - \frac{3K_7 Q}{2t^2}$$

b. If $L < 8R_m$,

$$\sigma_7^* = -\frac{Q}{4t(b+x_1+x_2)} - \frac{12K_7QR_m}{Lt^2}$$

These stresses can be reduced by adding a reinforcement or a wear plate at the saddle location that is welded to the cylindrical shell in accordance with requirements discussed earlier in this module. The reduced stresses can be expressed by the following equations:

$$\sigma_{7,r} = -\frac{Q}{4(t+\eta t_r)b_1} - \frac{3K_7Q}{2(t+\eta t_r)^2}$$

$$\sigma_{7,r}^* = -\frac{Q}{4(t+\eta t_r)b_1} - \frac{12K_7R_m}{L(t+\eta t_r)^2}$$

The absolute value of σ_7, σ^*_7, $\sigma_{7,r}$, or $\sigma^*_{7,r}$ shall not exceed 1.25S.

3. If $t_r > 2t$, then the compressive membrane plus bending stress at the ends of reinforcing plate (points G1 and H1 of Figure 3, Sketch (b) is given as follows:

a. If $L \geq 8R_m$, $\sigma_{7,1} = -\frac{Q}{4t(b+x_1+x_2)} - \frac{3K_{7,1}Q}{2t^2}$

b. If $L < 8R_m$, $\sigma_{7,1}^* = -\frac{Q}{4t(b+x_1+x_2)} - \frac{12K_{7,1}QR_m}{Lt^2}$

Coefficient $K_{7,1}$ is computed using the equation for K_7 evaluated at the angle θ_1. The absolute value of $\sigma_{7,1}$ and $\sigma^*_{7,1}$ shall not exceed 1.25S.

Circumferential stresses in the shell with a stiffening ring in the plane of the saddle support

1. The maximum compressive circumferential membrane stress in the shell is given by the following equation:

$$\sigma_6^* = -\frac{K_5Qk}{A}$$

A = Cross sectional area of the stiffening ring(s) and the associated shell width used in stress calculation

The absolute value of σ^*_6 shall not exceed $\min[S, S_r]$.

2. The circumferential compressive membrane plus bending stress at points G and H of Figure 7, Sketch (a) for stiffening rings located on the inside of the shell is given as follows:

$$\sigma_8 = -\frac{K_8Q}{A} - \frac{K_6QR_mC_1}{I} \quad \text{(Stress in shell)}$$

$$\sigma_9 = -\frac{K_8Q}{A} - \frac{K_6QR_mC_2}{I} \quad \text{(Stress in the stiffening ring)}$$

$$K_8 = \frac{\cos\beta\left[1-\frac{\cos 2\beta}{4}+\frac{9\sin\beta.\cos\beta}{4\beta}-3\left(\frac{\sin\beta}{\beta}\right)^2\right]}{2\pi\left[\left(\frac{\sin\beta}{\beta}\right)^2-\frac{1}{2}-\frac{\sin 2\beta}{4\beta}\right]} + \frac{\beta\sin\beta}{2\pi}$$

I = Moment of inertia of cross sectional area A in relation to its neutral axis that is parallel to the axis of the shell

C_1, C_2

= Distance to the extreme axis of the shell-stiffener cross section to the neutral axis of the shell-stiffener cross section

The absolute value of σ_8 shall not exceed 1.25S, and that of σ_9 shall not exceed 1.25 S_s where S_s is the allowable stress for the stiffener material at design temperature.

3. The circumferential compressive membrane plus bending stress at points G and H of Figure 7, Sketch (a) for stiffening rings located on the outside of the shell is given as follows:

$$\sigma_8^* = -\frac{K_8 Q}{A} + \frac{K_6 Q R_m C_1}{I} \qquad \text{(Stress in shell)}$$

$$\sigma_9^* = -\frac{K_8 Q}{A} - \frac{K_6 Q R_m C_2}{I} \qquad \text{(Stress in the stiffening ring)}$$

The absolute value of σ^*_8 shall not exceed 1.25S, and that of σ^*_9 shall not exceed 1.25 S_s where S_s is the allowable stress for the stiffener material at design temperature.

<u>Circumferential stresses in the shell with stiffening rings on both sides of the saddle support</u>

1. The maximum compressive circumferential membrane stress in the shell is given by the following equation:

$$\sigma_6 = -\frac{K_5 Q k}{t(b+2x_2)}$$

The absolute value of σ_6 shall not exceed S.

2. The circumferential compressive membrane plus bending stress at points I and J of Figure 7, Sketch (b) for stiffening rings located on the inside of the shell is given as follows:

$$\sigma_{10} = -\frac{K_9 Q}{A} + \frac{K_{10} Q R_m C_1}{I} \qquad \text{(Stress in shell)}$$

$$\sigma_{11} = -\frac{K_9 Q}{A} - \frac{K_{10} Q R_m C_2}{I} \qquad \text{(Stress in the stiffening ring)}$$

$$K_9 = \frac{1}{2\pi}\left\{\left[-\frac{1}{2} + (\pi - \beta)\cot\beta\right]\cos\rho + \rho\sin\rho\right\}$$

The absolute value of σ_{10} shall not exceed 1.25S, and that of σ_{11} shall not exceed 1.25 S_s.

3. The circumferential compressive membrane plus bending stress at points I and J of Figure 7, Sketch (b) for stiffening rings located on the outside of the shell is given as follows:

$$\sigma_{10}^* = -\frac{K_9 Q}{A} - \frac{K_{10} Q R_m C_1}{I} \qquad \text{(Stress in shell)}$$

$$\sigma_{11}^* = -\frac{K_9 Q}{A} + \frac{K_{10} Q R_m C_2}{I} \qquad \text{(Stress in the stiffening ring)}$$

The absolute value of σ^*_{10} shall not exceed 1.25S, and that of σ^*_{11} shall not exceed 1.25 S_s.

Saddle Support

The horizontal force at the minimum section at the low point of the saddle is given by the following equation:

$$F_h = Q\left(\frac{1+\cos\beta-0.5\sin^2\beta}{\pi-\beta+\sin\beta.\cos\beta}\right)$$

Saddle should be designed to resist this force.

Source: ASME Boiler and Pressure Vessel Code, Section VIII, Division 2, Edition 2015

Jawad, Maan H. and James R. Farr, <u>Structural Analysis and Design of Process Equipment</u>

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CATASTROPHIC RUPTURE OF HEAT EXCHANGER

I recently came across a post on my LinkedIn webpage that showed an animation of an explosion of a shell-and-tube heat exchanger at Tesoro's Anacortes refinery in Washington. Animation is available on YouTube at https://www.youtube.com/watch?v=8vPaQYM-tWs. I did google search on this incident and came across the investigation report by US Chemical and Safety Hazard Investigation Board on this catastrophic failure that left seven dead. This article has been extracted from that report.

On April 2, 2010, the Tesoro petroleum refinery in Anacortes, Washington experienced a catastrophic rupture of a heat exchanger in the catalytic reformer/ NapthaHydrotreater unit. The rupture caused an explosion and an intense fire that burned for more than three hours. It fatally injured seven employees who were working in the immediate vicinity of the heat exchanger at the time of the incident. The US Chemical Safety and Hazard Investigation Board (CSB) conducted an investigation of the rupture and produced a report in May 2014 detailing the key findings and providing recommendations to all stakeholders. The full report is 160 pages long; this article is a summary of the report.

Tesoro is an independent refiner and marketer of petroleum products, headquartered in San Antonio, Texas. It operates six refineries in the western United States with a combined rated crude oil capacity of approximately 875,000 barrels per day. The Anacortes refinery is located about 70 miles north of Seattle and on the Puget Sound and has a total crude oil capacity of 120,000 bpd. It receives crude feedstock via pipeline from Canada, by rail from North Dakota and the Central US, and by tanker form Alaska and foreign sources. The refinery supplies gasoline, jet fuel and diesel fuel to markets in Washington and Oregon through a third party pipeline system. The refinery has been in operation since 1955; it was acquired by Tesoro in 1988 from Shell.

The April 2, 2010 incident occurred in the Catalytic Reformer/ NapthaHydrotreater unit ("the NHT unit") which includes a napthahydrotreating process unit. Hydrotreating is a process that removes sulfur, nitrogen and oxygen impurities from the petroleum feedstock and the intermediate products by reacting with hydrogen in the presence of a catalyst. Hydrotreating serves two purposes:

1. It improves the quality and the environmental impact of the product, especially the quality specifications mandated by law.
2. It protects sensitive and costly downstream catalysts from contamination.

The Tesoro NHT unit was originally constructed in 1972 with a rated capacity of 24,800 bpd. Modifications and upgrades resulted in a rated capacity at the time of incident of 40,550 bpd, a 64% capacity increase.

Catalytic Reformer

Catalytic reforming is a chemical process used to convert petroleum refinery naptha, typically having low octane ratings, into high octane liquid products called reformates. The reformate product is sent to gasoline component storage for use in fuel blending. The reforming reaction generates hydrogen which is used in NHT.

NapthaHydrotreater – A/B/C & D/E/F Feed/ Product Heat Exchangers

The removal of sulfur, nitrogen and oxygen impurities in the NHT unit requires heating the naptha to over 600^{o}F at greater than 600 psig, and mixing it with hydrogen. The initial portion of this heating took place in the NHT unit's E-6600 A/B/C and D/E/F feed and product heat exchangers as depicted in Figure 1.

The cool NHT liquid naptha feed is pumped from storage and/or other active units and mixed with a stream of hydrogen rich gas, becoming a combined liquid and gas feed stream. The resulting liquid-gas mixture is then fed to the tube side of the two parallel banks of NHT heat exchangers A/B/C and D/E/F to be heated by the shell-side fluid. As the liquid-gas mixture inside the tubes is heated, the liquid portion vaporizes completely. Now liquid-free, the naptha and hydrogen vapors enter a furnace where they are heated and fed to the NHT reactor.

The reactions to remove sulfur, nitrogen and oxygen take place in this reactor. The hot reactor effluent is then fed through the shell-side of the heat exchangers to preheat the incoming tube side feed. The impurity free naptha is then fed to the other processes in the refinery.

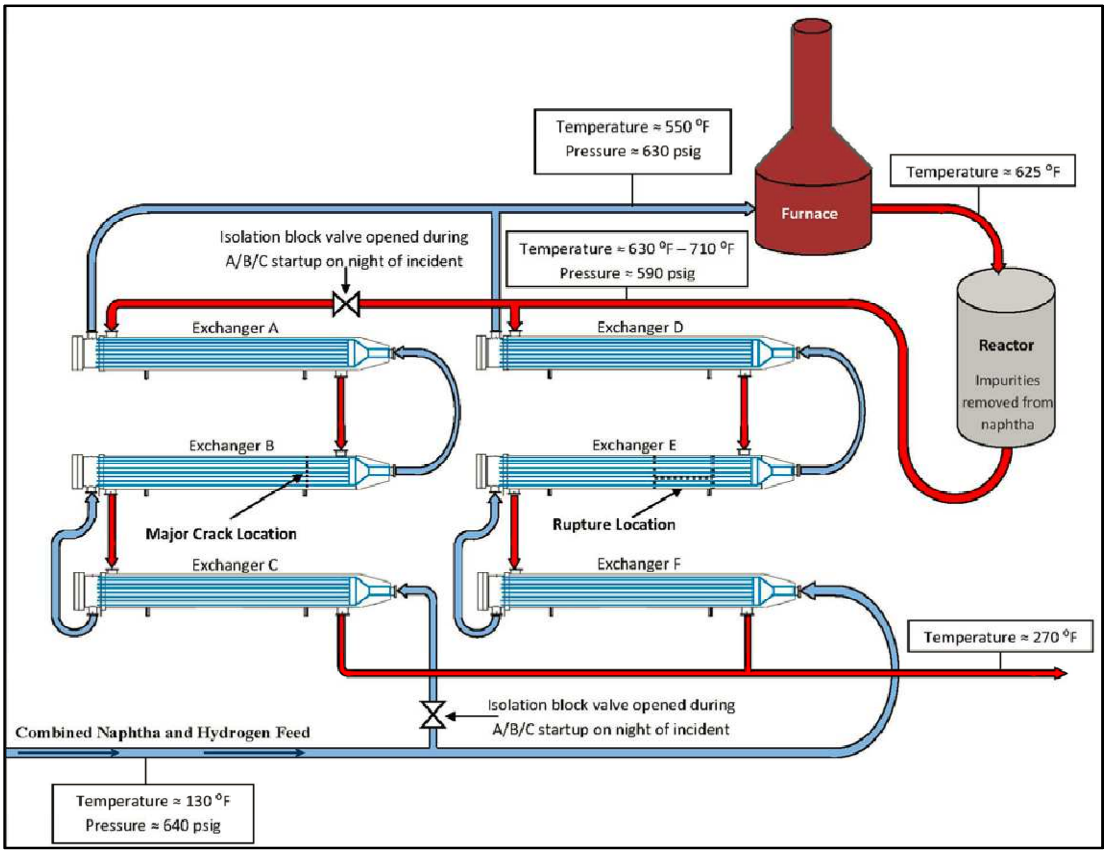

Figure 1: Process Flow of NHT Unit

Pre-incident Operations

During normal operation at the Anacortes refinery, the A/B/C and D/E/F heat exchangers were all in use. Because of the original shell design and the process operating conditions, the heat exchangers would foul during operation; i.e. they would both develop a build-up of process containment byproducts both inside and outside of the tubes, as illustrated in the Figure 2. The fouling inhibited heat transfer between the tube side and the shell side fluid, thus reducing the heat transfer efficiency.

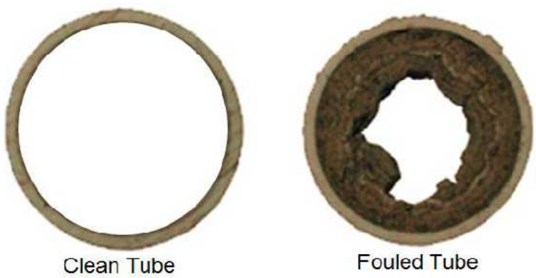

Figure 2: Example of Fouling Deposits on the Inside of Tubes

Because the heat exchangers fouled, they required periodic cleaning so that process temperature requirements could be maintained. Cleaning was typically required after about six months of continuous operation. During cleaning operation, one bank of heat exchangers was taken out of service while the other bank continued operating. The cleaned heat exchangers would be placed back into service by slowly introducing the hot naptha and hydrogen feed into the heat exchangers. Because of a long history of frequent leaks and occasional fires when putting these heat exchangers back into service, startup, shutdown and cleaning activities were hazardous non-routine operations. By employing thus non-routine operation, Shell and later Tesoro avoided a total shutdown of the NHT unit.

On March 28, 2010, five days before the incident, the A/B/C heat exchanger bank was taken offline. The D/E/F heat exchanger bank and the rest of the NHT unit remained in operation. On March 31, the three day maintenance cleaning activity was completed and the equipment was reassembled and prepared for operation.

Night of the Incident

On the evening of April 1, 2010, Tesoro initiated startup of A/B/C heat exchanger bank. The NHT unit was staffed in a typical manner, with one inside operator who monitored the console and one outside operator. the aerial view of the unit is shown in Figure 3.

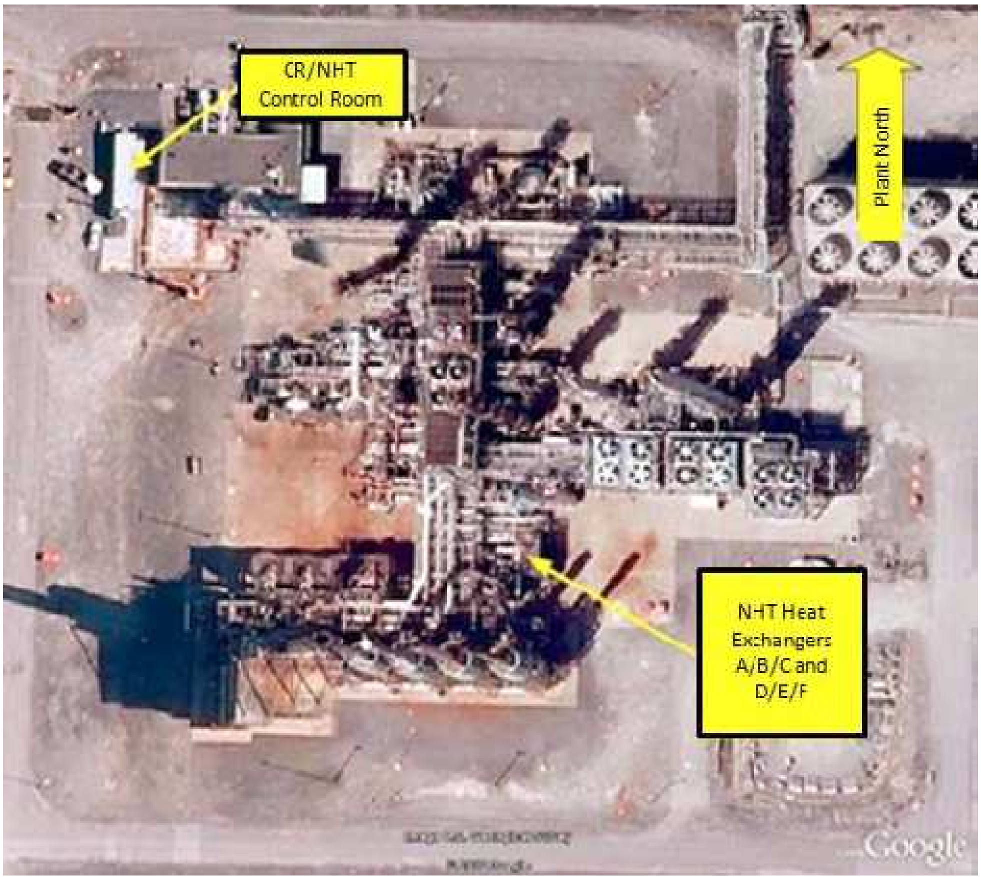

Figure 3: Aerial View of CR/NHT Unit

The inside NHT operator and the outside NHT operator began the process of placing the heat exchangers back into the service. The inside operator used a check list for the startup process; physically checking off the steps on a hard copy of the procedure while maintaining radio communication with the outside operator. Interviews conducted by CSB indicate that the startup of heat exchangers was a very difficult assignment for only a single outside operator. The startup procedure required manipulation of several isolation block valves as illustrated in Figure 4, which necessitated a significant amount of manual effort to open.

Figure 4: CSB Animation of Operator Opening a Long-Winded Valve on the Night of Incident

These valves had to be gradually and concurrently opened, so the operator could not simply stay by each valve until it was fully open or closed. Also, four steam lances were staged and ready for use during the startup to mitigate any leaks or fires that might occur. These valves and steam lances were located at different positions in the vicinity of A/B/C and D/E/F heat exchangers. At approximately 10:30 pm, six additional Tesoro employees (five operators and one supervisor) joined the outside operator at request of the supervisor to assist in bringing the A/B/C heat exchangers back online. The startup procedure did not specify defined roles for these six additional personnel.

The Incident

The operators continued the A/B/C heat exchanger bank startups as planned. At 12:30 am on April 2nd, while the seven outside personnel were still performing the startup operation, the E exchanger on the adjacent, in-service bank, catastrophically ruptured. The pressure containing “shell” of the heat exchanger separated at weld seams as depicted in Figure 5, expelling a large volume of very hot hydrogen and naptha. The naptha and hydrogen like auto-ignited upon release into the atmosphere, creating a large fireball as depicted in Figure 6. All seven outside personnel were badly burned, and within 22 days of the incident, all succumbed to their injuries.

Technical Analysis

Post-incident metallurgical analysis determined that the carbon steel E heat exchanger ruptured because it was in a highly weakened state because of high temperature hydrogen attack (HTHA). The HTHA damage mechanism occurs when steel equipment is exposed to hydrogen at high temperatures and partial pressures. Atomic hydrogen diffuses into the steel walls of the process equipment and reacts with the carbon to produce methane gas. Methane, a much larger molecule than atomic hydrogen, cannot diffuse out of the steel. Rather it accumulates inside the vessel walls exerting force on the surrounding steel. As more methane gas is formed, the methane pressure increases. The very high pressure exerted by the methane gas inside the steel can form fissures, as illustrated in Figure 7, or blisters in the steel as shown in Figure 8.

As more fissures are formed, they can link, forming micro-cracks in the steel. Micro-cracks can also link to form larger cracks which greatly weaken the steel and can lead to the rupture of the vessel. This process occurred in the E heat exchanger at the Tesoro Anacortes refinery.

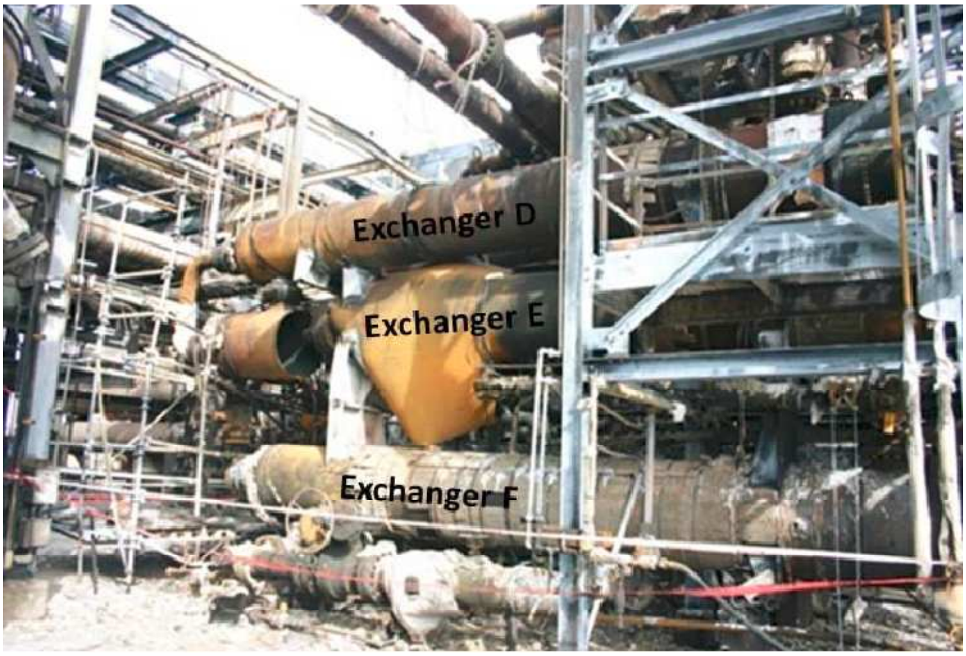

Figure 5: Post Incident View of the D/E/F NHT Heat Exchanger Bank

Figure 6: CSB Animation of the Fire Following the NHT Heat Exchanger Failure

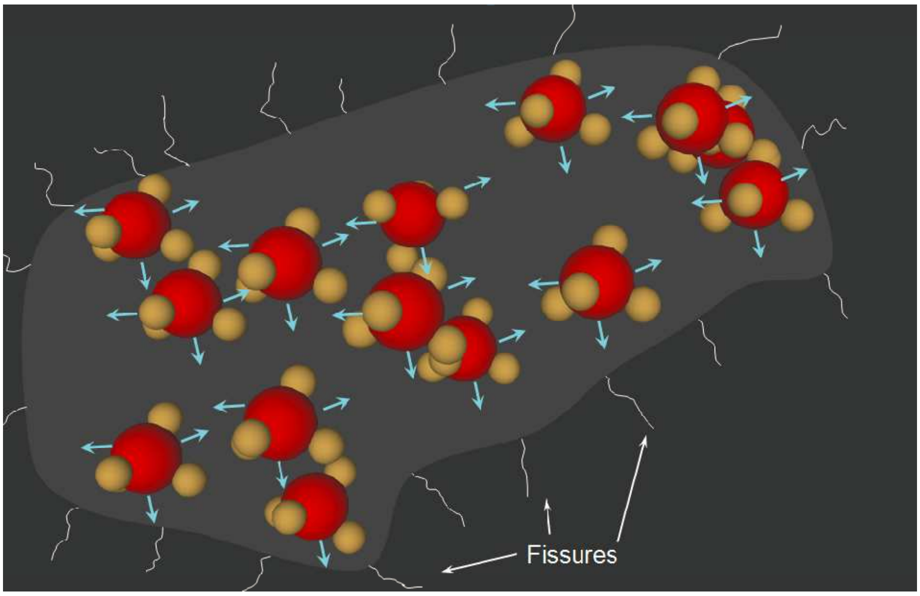

Figure 7: Methane Fissures

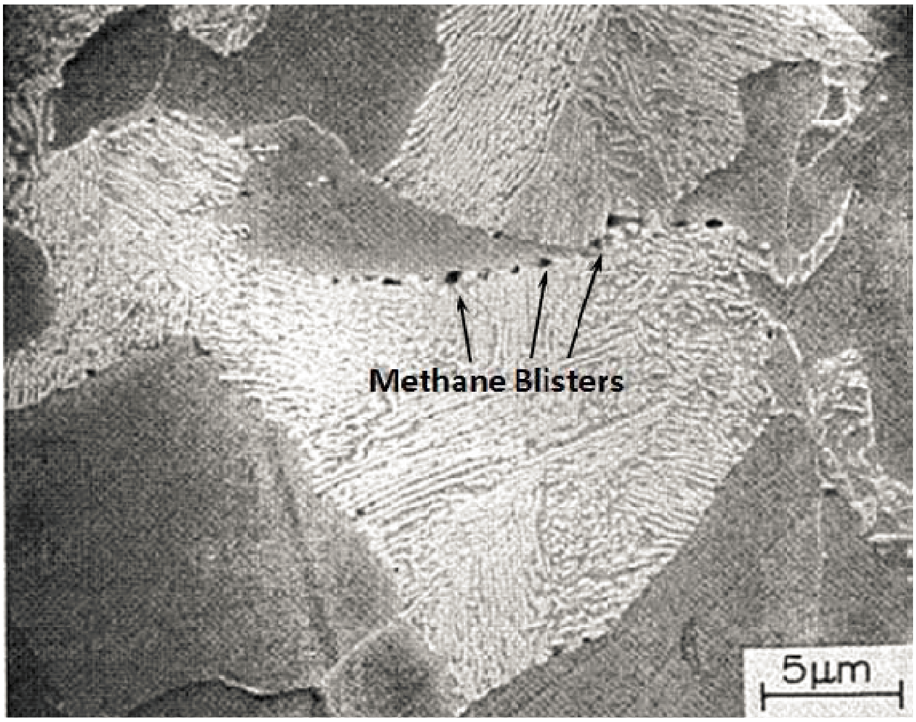

Figure 8: Methane Blisters

Predicting the Occurrence of HTHA

Industry relies on a graph in API RP 941 _Steels for Hydrogen Service at Elevated Temperatures and Pressures in Petroleum Refineries and Petrochemical Plants_ to predict the occurrence of HTHA in various steels. The lines in that graph are known as Nelson Curves; these are based on observed industry experience with HTHA. The most recent version of API RP 941 Nelson Curves is shown in Figure 9. Industry uses these curves as a line of demarcation to predict HTHA.

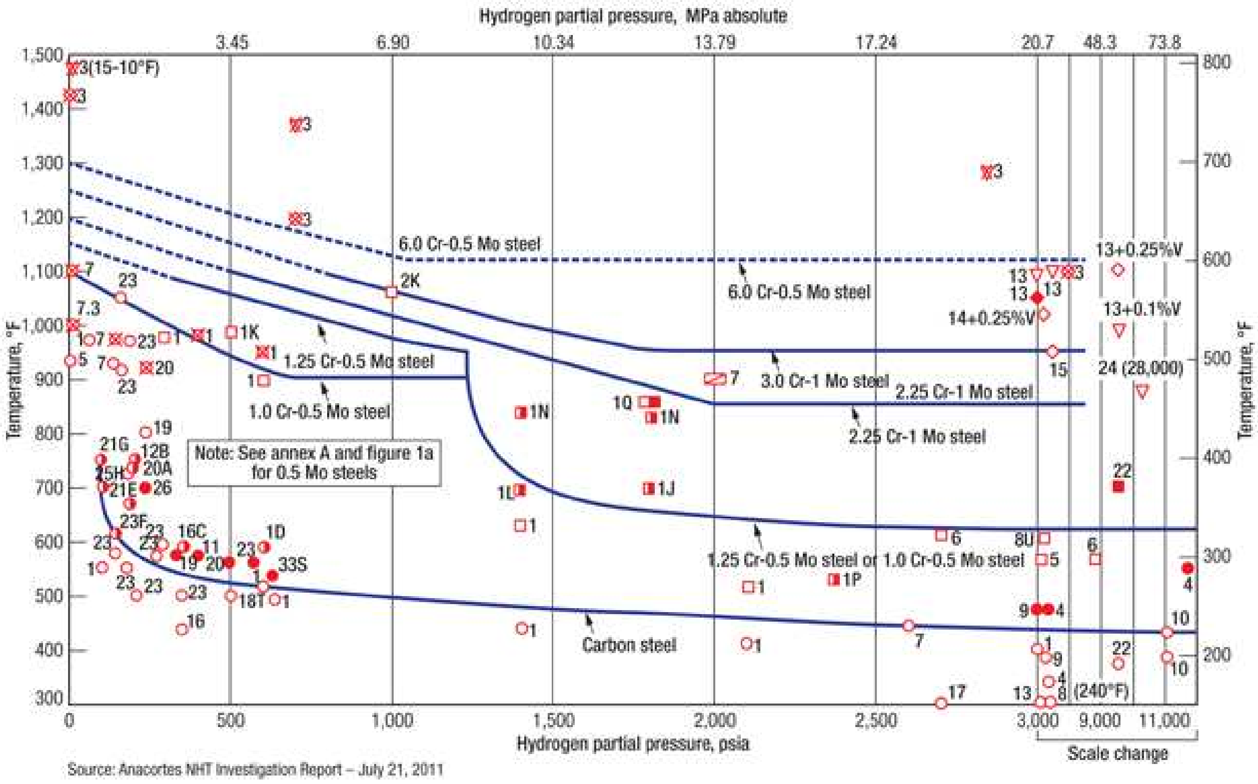

Figure 9: Nelson Curves (*From Hydrocarbon Processing*)

At temperatures above each curve, HTHA is possible for that material of construction, and at temperatures below the curve, the prediction is that HTHA will not occur for that material. Carbon steel is represented by the lowest curve indicating that this material is most susceptible to HTHA when compared to other material of construction shown in Figure 9.

Conditions That Increase HTHA Susceptibility

Welding performed on steel process vessels creates additional HTHA risk factors, such as residual stress. Post weld heat treatment is a method that can reduce the stress in steel that was generated from the welding process. The carbon steel shells of B and E heat exchangers were not post weld heat treated, and therefore the steel surrounding the welds may have been high stress areas. HTHA was found in areas near welds in both B and E heat exchangers.

TESORO'S REPLACEMENT HEAT EXCHANGERS

Since April 2010 incident, Tesoro has installed new NHT heat exchangers, incorporating aspects of an inherently safe design. The material of construction of the two heat exchangers has been upgraded to significantly reduce the potential of HTHA. In addition, an advanced process control system is in place to minimize fouling. The entire NHT unit must now be shut down for cleaning, eliminating the hazards of online switching and creating a much safer approach for maintenance. The new heat exchangers also incorporate additional instrumentation to allow the monitoring of each heat exchanger for fouling and decrease the likelihood of operation in HTHA susceptible conditions.

Source: US Chemical Safety and Hazard Investigation Board Investigation Report, "*Catastrophic Rupture of Heat Exchanger (Seven Fatalities), Tesoro Anacortes Refinery, Anacortes, Washington, April 2, 2010*"

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HOW TO IMPROVE YOUR MEMORY

Wouldn’t it be nice to just look at a page and never forget what was on there? What if you could never again forget a friend’s birthday? The bad news is, not everyone has a photographic memory, otherwise known as eidetic memory. Only a few actually have it, the rest rely on mnemonic devices. The good news, however, is that everyone can take steps to improve their memory, and with time and practice most people can gain the ability to memorize seemingly impossible amounts of information. Whether you want to win the World Memory Championships, ace your history test, or simply remember where you put your keys, this article can get you started.

1. Convince yourself that you have a good memory that will improve. Too many people get stuck here and convince themselves that their memory is bad, that they are just not good with names, that numbers just slip out of their minds for some reason. Erase those thoughts and vow to improve your memory. Commit yourself to the task and bask in your achievements -- it's hard to keep motivated if you beat yourself down every time you make a little bit of progress.

2. Keep your brain active. The brain is not a muscle, but regularly “exercising" the brain actually does keep it growing and spurs the development of new nerve connections that can help improve memory. By developing new mental skills—especially complex ones such as learning a new language or learning to play a new musical instrument—and challenging your brain with puzzles and games you can keep your brain active and improve its physiological functioning.

3. Exercise daily. Regular aerobic exercise improves circulation and efficiency throughout the body, including in the brain, and can help ward off the memory loss that comes with aging. Exercise also makes you more alert and relaxed, and can thereby improve your memory uptake, allowing you to take better mental “pictures."

4. Reduce stress. Chronic stress, although it does not physically damage the brain, can make remembering much more difficult. Even temporary stresses can make it more difficult to effectively focus on concepts and observe things. Try to relax, regularly practice yoga or other stretching exercises, and see a doctor if you have severe chronic stress.

5. Eat well and eat right. A healthy diet contributes to a healthy brain, and foods containing antioxidants—broccoli, blueberries, spinach, and berries, for example—and Omega-3 fatty acids appear to promote healthy brain functioning. Feed your brain with such supplements as Thiamine, Vitamin E, Niacin and Vitamin B-6. Grazing, eating 5 or 6 small meals throughout the day instead of 3 large meals, also seems to improve mental functioning (including memory) by limiting dips in blood sugar, which may negatively affect the brain.

6. Take better pictures. Often we forget things not because our memory is bad, but rather because our observational skills need work. One common situation where this occurs (and which almost everyone can relate to) is meeting new people. Often we don’t really learn people’s names at first because we aren’t really concentrating on remembering them. You’ll find that if you make a conscious effort to remember such things, you’ll do much better. One way to train yourself to be more observant is to look at an unfamiliar photograph for a few seconds and then turn the photograph over and describe or write down as many details as you can about the photograph. Try closing your eyes and picturing the photo in your mind. Use a new photograph each time you try this exercise, and with regular practice you will find you’re able to remember more details with even shorter glimpses of the photos.

7. Give yourself time to form a memory. Memories are very fragile in the short-term, and distractions can make you quickly forget something as simple as a phone number. The key to avoid losing memories before you can even form them is to be able to focus on the thing to be remembered for a while without

thinking about other things, so when you're trying to remember something, avoid distractions and complicated tasks for a few minutes.

8. Create vivid, memorable images. You remember information more easily if you can visualize it. If you want to associate a child with a book, try not to visualize the child reading the book – that's too simple and forgettable. Instead, come up with something more jarring, something that sticks, like the book chasing the child, or the child eating the book. It's your mind – make the images as shocking and emotional as possible to keep the associations strong.
9. Repeat things you need to learn. The more times you hear, see, or think about something, the more surely you'll remember it, right? It's a no-brainer. When you want to remember something, be it your new coworker's name or your best friend's birthday, repeat it, either out loud or silently. Try writing it down; think about it.
10. Group things you need to remember. Random lists of things (a shopping list, for example) can be especially difficult to remember. To make it easier, try categorizing the individual things from the list. If you can remember that, among other things, you wanted to buy four different kinds of vegetables, you'll find it easier to remember all four.
11. Organize your life. Keep items that you frequently need, such as keys and eyeglasses, in the same place every time. Use an electronic organizer or daily planner to keep track of appointments, due dates for bills, and other tasks. Keep phone numbers and addresses in an address book or enter them into your computer or cell phone. Improved organization can help free up your powers of concentration so that you can remember less routine things. Even if being organized doesn't improve your memory, you'll receive a lot of the same benefits (i.e. you won't have to search for your keys anymore).
12. Try meditation. Research now suggests that people who regularly practice meditation are able to focus better and may have better memories. Studies at Massachusetts General Hospital show that regular meditation thickens the cerebral cortex in the brain by increasing the blood flow to that region. Some researchers believe this can enhance attention span, focus, and memory.
13. Sleep well. The amount of sleep we get affects the brain's ability to recall recently learned information. Getting a good night's sleep – a minimum of seven hours a night – may improve your short-term memory and long-term relational memory, according to studies conducted at the Harvard Medical School.
14. Venture out and learn from your mistakes. Go ahead and take a stab at memorizing the first one hundred digits of pi, or, if you've done that already, the first one thousand. Memorize the names of countries and their capitals and heads of state, or your grocery list through visualization. Through diligent effort you will eventually master the art of memorization.

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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 ● Tanks

Oil & Gas ● Power ● Chemical ● Petrochemical ● Fertilizer