Pressure Vessel Newsletter

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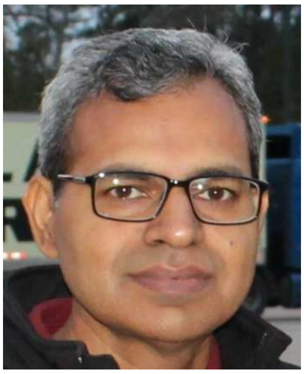
From The Editor's Desk:



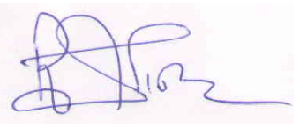
Sometime last month, I floated a question in pressure vessel community to find out what percentage of pressure vessels worldwide are built to ASME Section VIII, Division 2. Other than a few suggestions pointing to the data being available at National Board, I didn't get any helpful response. But I think it wouldn't be a stretch to surmise that the percentage is very small – perhaps less than 10%.

It is no secret that the ASME VIII-2 analysis results in more sound design of pressure vessels. After all, it is based on theory that is better suited to widely used pressure vessel materials than Division 1. Why then has it not received greater acceptance in pressure vessel community?

One of the major differences between the design process of Division 1 and Division 2 pressure vessels is that while a Division 1 pressure vessel can be fully designed by a pressure vessel engineer, a Division 2 pressure vessel also requires involvement of an FEA expert. In most cases, the pressure vessel engineers do not have the expertise to use the finite element software. The end result is that FEA is done by someone other than the pressure vessel engineer; however, the FEA results are analyzed for acceptance to the Division 2 code by the pressure vessel engineer.

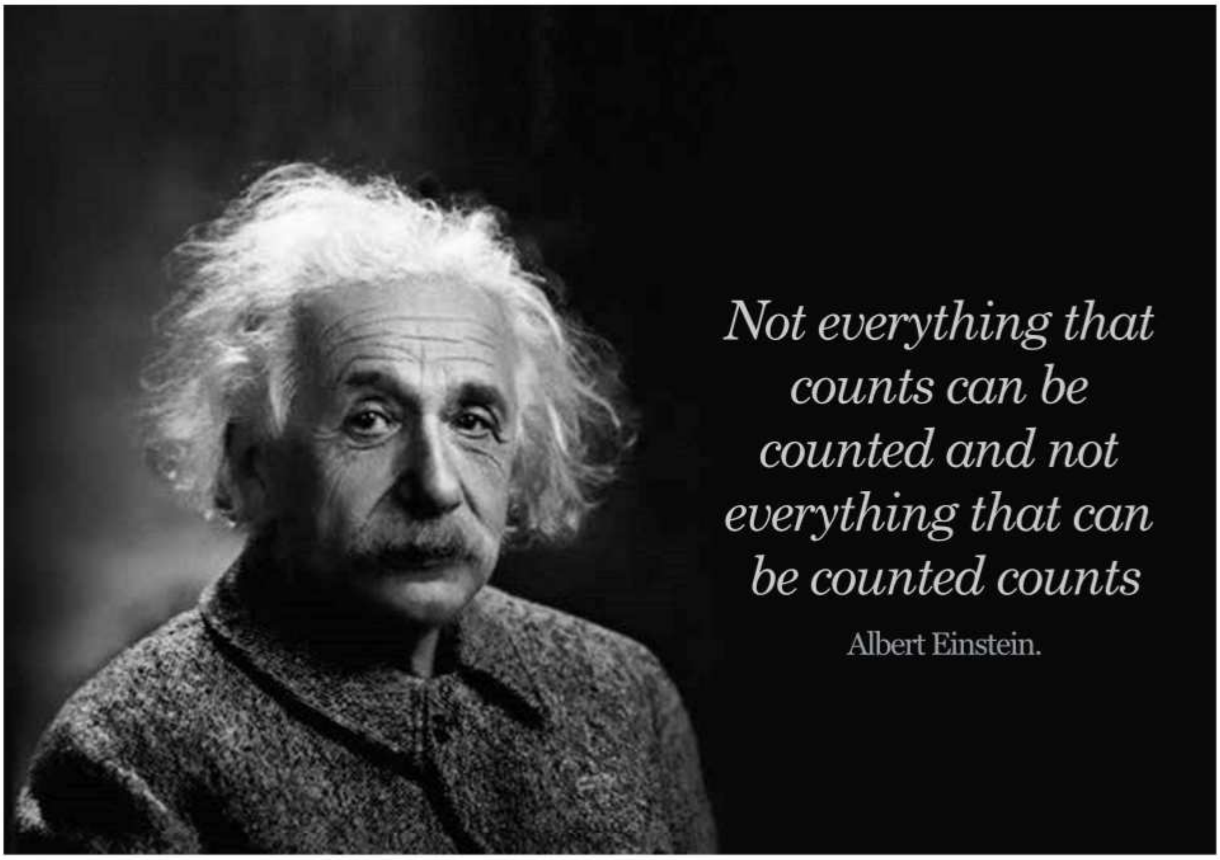
What can be done to increase the acceptance of ASME VIII-2? I think development of FEA software specially designed for pressure vessels that is easy enough to use by pressure vessel engineers will go a long way. It will give pressure vessel engineers the full control and ownership of the design. The makers of the pressure vessel software would have to take a lead in the development of such software.

For the time being, though, it appears that Division 1 will remain in wide use, and the use of two different ASME codes for the design of pressure vessels will remain in place.



[The picture on the cover is courtesy of Cheema Boilers in Chandigarh, India.]

In this issue…



Math of real world seldom adds up due to intangible variables that cannot be easily captured. For final tally, we know we don't just have to win contracts, we must earn customer confidence too. Our main focus is customer delight achieved due to & through positive interactions, quality alertness, proactive involvement and personalized service for varying situations & requirements.

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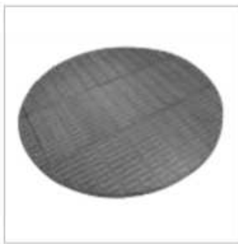

Mist Eliminators

STRESSES IN PRESSURE VESSELS

Stress analysis is the determination of the relationship between external forces applied to a vessel and the corresponding stress. In design of pressure vessels, the components are analyzed so as to arrive at an economical and safe design. This is done by analyzing stresses where necessary to determine thickness of the material and size of the members. The starting place for stress analysis is to determine all the design conditions for a given problem and then determine all the external forces. The external forces are then related to the vessel parts which must resist them by generating the corresponding stresses.

Of the many theories developed to predict elastic failure, the three most commonly used are:

- Maximum principal stress theory
- Maximum shear stress theory
- Distortion energy theory

Maximum Principal Stress Theory

This theory considers failure to have occurred when any one of the three principal stresses has reached a value equal to the elastic limit as determined from a uni-axial tension or compression test. This theory is the oldest, most widely used and simplest to apply. This theory is used for biaxial states of stress assumed in thin-walled pressure vessels, and ASME Code Section VIII-1 uses this theory as the basis for design. While the maximum principal stress theory does accurately predict failure in brittle materials, it is not always accurate for ductile materials.

Maximum Shear Stress Theory

This theory (also called the Tresca criterion) considers failure to have occurred when the maximum shear stress equals the shear stress at the elastic limit as determined by a pure shear test. The maximum shear stress is defined as one-half the algebraic difference between the largest and smallest of the three principal stresses. ASME Code Section VIII-2 (prior to 2007 Edition) utilized the maximum shear stress criterion. This theory closely approximates experimental results and is also easy to use.

Distortion Energy Theory

This theory (also called the von Mises criterion) considers failure to have occurred when the distortion energy accumulated in the part under stress reaches the elastic limit as determined by the distortion energy in a uniaxial tension or compression test. ASME Code Section VIII-2 (from 2007 Edition) utilizes the distortion energy criterion.

The ever increasing use of pressure vessels for storage, industrial processing and power generation under extreme conditions of pressure, temperature and environment has given special emphasis to analytical and experimental methods for determining their operating stresses. Therefore it is necessary to understand the meaning and significance of these stresses. This includes the means of determining the value and extent of the stresses and strains, establishing the behavior of the material involved and evaluating the compatibility of these two factors in the media or the environment to which they are subjected. Knowledge of material behavior is required not only to avoid failures, but equally to permit maximum economy of material choice and amount used.

The stresses produce changes in their dimensions known as strains. The determination of the relationship between the external forces (P) applied to a vessel, and the stresses (σ) and strains (e) within the vessel form the basis of this field of stress analysis. Engineering materials commonly used in the design of structures and pressure vessels have an initial stress-strain relationship which, for practical purposes, may be assumed linear, indicating that stress is directly proportional to strain and is represented by the equation:

$$E = \sigma / e \qquad (1)$$

This is known as Hooke's law. The value of *E* is called the modulus of elasticity, or Young's modulus. It is a measure of material stiffness. A material has high stiffness when its deformation in the elastic range is relatively small. Referring to the equation (1), the accompanying strain will be less for a material with a high *E* value than for one with a low *E* value. This property of stiffness is very important in designs where deformation must be kept small, as, for example, gasket joints, and the control rod portions of nuclear reactor vessels. Approximate room temperature values of this modulus for some engineering materials are:

Steel & steel alloys	30×10^6 psi (20.7×10^7kPa)
Aluminum alloys	10×10^6 psi (68.9×10^6kPa)
Magnesium alloys	6.5×10^6 psi (44.8×10^6kPa)
Copper	16×10^6 psi (11×10^7kPa)

An examination of the stress-strain diagram of Figure 1 shows that they are made of two general parts – an initial elastic range for which Hooke's law generally applies, and a following plastic range where the strains become large and this law no longer applies.

DEFINITIONS

Elasticity is the property of a material to return to its original shape after removal of the load.

Proportional limit is the greatest stress that a material can withstand without deviating from the direct proportionality of stress to strain.

Elastic limit is the maximum stress which a material is capable of withstanding without permanent deformation upon complete release of the stress. Determination of the elastic limit is very difficult; however it closely approximates the more readily determined value of the proportional limit.

Yield point is the stress at which there occurs a marked increase in strain without an increase in stress.

Ultimate strength is the maximum stress that the material can withstand. This stress equals the maximum load divided by the original cross sectional area of the specimen.

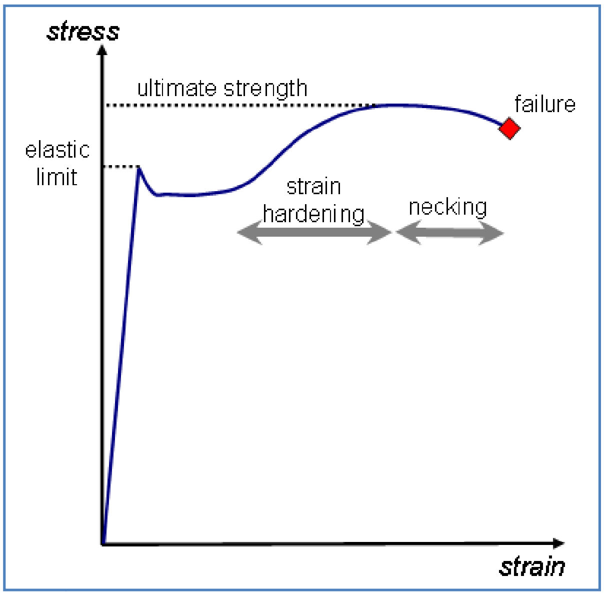

Figure 1: Stress-strain Diagram

Ductility is the property of a material to undergo deformation. It is an important property both from a design and a fabrication viewpoint. It acts as built-in excessive stress adjuster for localized stresses that were not considered or not contemplated in the design. In pressure vessels at joints, nozzles, openings, etc, high stresses develop locally which are subsequently reduced by plastic flow of the material. Ductility is also an important material property in fabrication and processing, such as rolling, forging, drawing and extruding. If the ductility is not adequate, the large deformations produced in these operations result in the rupture of the material. For instance, the steel plates used to fabricate the shells and heads of the vessels must have sufficient ductility to permit them to be bent from a flat plate to their final curvatures without cracking. Frequently it is necessary to heat the material to high temperatures to increase its ductility during the forming operation.

Toughness is the ability of a material to absorb energy during plastic deformation. It is often measured by the energy absorbed per unit of volume in stressing to rupture and is called the modulus of toughness. One convenient method that is employed for many ductile materials is to use an approximate measure of the area under the stress-strain diagram, the product of ultimate stress times the strain at rupture. Materials of high toughness have high strength, as well as large ductility. Brittle materials have low toughness since they only have small plastic deformations before rupture. Toughness is a most desirable property in vessels subject to mechanical or thermal shock.

STRESS SIGNIFICANCE

Analytical formulas for the evaluation of stresses are usually based on elastic theory and elastic behavior of the material, i.e., material which conform to Hooke's law, and at first glance, it may appear that materials which follow this behavior right up to the breaking point would be the most desirable for use. This is not the case, however, for instance, plaster of paris has a perfectly straight stress-strain curve up to the breaking point but, of course, is not a suitable material for pressure vessels, because it is totally elastic and not partially plastic in its behavior. It is this plastic property of the material, with its ability to give or yield under high peak or local stress and so accommodate the applied loading by a more favorable distribution of internal stress that is the most important property of a pressure vessel material. The elastically computed or actual strength of most members, considering the pressure vessel as a whole, would be considerably reduced if it were not accompanied by plastic deformation at various relatively small portions of the member where high local stresses occur.

Determination of stress significance requires knowledge of:

1. The type and nature of the applied loading, and the resulting stress distribution within the member. For instance, is the applied loading mechanical or thermal, of a steady (static) or unsteady (variable or cyclic) nature, and is the resulting stress pattern uniform or does it have high peak values?
2. The ductile and plastic properties of the material. For instance, are the properties of the material such that internal yielding or readjustment of strain can reduce the effects of local stress concentrations?
3. The toughness or adaptability of the material under adverse working conditions or environments. For instance, are the properties of the material sufficient to absorb applied impact or shock loadings?

The strength of a member does not depend only on the value of the maximum stress or strain in the member, but also on the external shape readjustment that the member itself can make to one more favorable than that assumed in the design, and on the plastic property of the material to permit internal stress adjustment.

TYPES OF LOADINGS AND THE STRESS PATTERN

Pressure vessels are subject to three types of loadings: namely, steady or static, unsteady (variable or cyclic) and impact. Although practically all vessels encounter variable or cyclic loadings, most pressure vessels may be assumed to be statically loaded without introducing serious error. These vessels made of ductile materials and subject to static loads fail by gross yielding. The ductility of the material allows a redistribution of stresses by plastic flow to attenuate points of high local values toward a pattern more favorable to maximum resistance.

However, when the loading is such that the member is subjected to a considerable number of stress cycles, even though the material is ductile, appreciable error can be introduced by considering a static loading condition to exist in appraising integrity on the basis of simple elastic formulas. Under such conditions, failure occurs due to a condition known as fatigue. In fatigue, the failure is due to highly localized stresses which cause a minute fracture that gradually spreads until the member is ruptured.

Impact or shock loadings can be imposed on pressure vessels by earthquake, explosions or collision of mobile equipment. This requires design considerations to accomplish transfer of the kinetic energy throughout the vessel, absorption of this energy within the vessel and the associated structure, and use of materials of adequate toughness. **Stresses form the dynamic (impact) loads are much higher in intensity than stresses from static loads of the same magnitude. A load is dynamic if the time of its application is smaller than the largest natural period of vibration of the body.**

The stress distribution near the point of load applications, such as the support brackets on a vessel, can vary greatly from the assumed pattern on which the ordinary equations are based, and these local stresses may be relatively high. Even though the material is ductile and a measure of stress redistribution can occur, these local stresses can be significant ones and are frequently responsible for failures. In relatively brittle materials, or in ductile material subjected to cyclic loading, stresses at point of load application may control the strength of the member rather than the stresses given by the ordinary equations. This is particularly important in vessels which are designed as membranes and cannot resist large bending moment perpendicular to their surface, and yet for practical purposes must have support brackets, lifting lugs, nozzles, etc., attached to them.

INITIAL OR RESIDUAL STRESS

The basic equations for determining stresses are based on the assumption that the stresses in a member are caused only by external loads and the residual stresses set up in the fabrication of construction processes, such as weld shrinkage, casting cooling, metal heat treatment, etc., are not considered. Although these stresses are secondary, since their value is self-limiting (they are not produced by unrelenting external loads), they may be of great importance in brittle materials, and even in ductile materials when the material is subjected to fatigue loading. Equally important is the danger of creating, in conjunction with the applied loading stresses, a three dimensional stress pattern in thick sections that is restrictive to the redistribution of high localized peak stresses through yielding. It is for this reason that stress relieving of thick vessels is much more important than thin ones in which the state of stress is essentially two-dimensional.

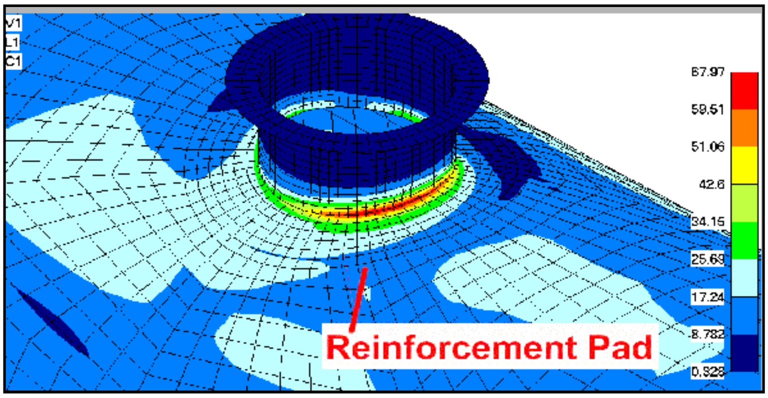

Figure 2: Discontinuities in Pressure Vessels

SHAPE OF MEMBER

The basic assumption for continuity of action in a member on which ordinary formulas for direct stress and bending stress are based, requiring that a plane section remain plane after bending, cannot hold near points of abrupt changes in section due to the restraining influence of this stiffer portion on adjacent sections. Figure 2 illustrates this condition. The stresses in the region influenced by these geometrical shape discontinuities are higher than that predicted by the assumed mathematical law of distribution on which the ordinary stress formulas are based, and are known as localized or concentration stresses. The errors introduced by the use of ordinary formulas for the design of members with abrupt changes of section are generally not serious if the load is static and the material ductile so as permit a slight measure of plastic flow; hence, the member acts more nearly as assumed.

These localized stresses are, however, most important in brittle materials, even under static loads, since under such conditions a redistribution or transfer of stress from the highly overstressed material to the adjacent lower stressed material does not take place and the rupture of member results. They are equally significant when cyclic loading is involved, even when the material is ductile, since the region of high stress acts as a focal point from which fatigue failure can stem.

The problem of evaluating localized stresses in vessels has assumed major importance lately as the engineering advances have placed great pressure, temperature, and environmental demands on pressure vessels. The petroleum and chemical processes require operating pressures in the range of 5,000 psi to 10,000 psi range. The rapidly expanding cryogenics industry has introduced low temperature conditions to minus 425^{o}F. The nuclear power industry has given rise to high pressure, high temperature and special cyclic and material irradiation operating conditions. All these requirements have focused considerable attention on stress analysis, materials of construction, and economics of design of vessels for these services.

MEMBRANE STRESS ANALYSIS

Pressure vessels commonly have the form of spheres, cylinders, cones, ellipsoids, tori or composites of these. When the thickness is small in comparison with other dimensions ($R_m/t > 10$), vessels are referred to as membranes and the associated stresses resulting from the contained pressure are called membrane stresses. These membrane stresses are average tension or compression stresses. They are assumed to be uniform across the vessel wall and act tangentially to its surface. The membrane or wall is assumed to offer no resistance to bending. When the wall offers resistance to bending, bending stresses occur in addition to the membrane stresses.

TYPES OF STRESS

The following list of stresses describes types of stress without regard to their effect on vessel or component.

Tensile	Thermal
Compressive	Tangential
Shear	Load Induced
Bending	Stress Induced
Bearing	Circumferential
Axial	Longitudinal
Discontinuity	Radial
Membrane	Normal
Principal	

Types of stresses that are present in pressure vessels are separated into various *classes* in accordance with the types of loads that produced them, and the hazard they represent to the vessel. Each class of stress must be maintained below an acceptable level, and the combined total stress must be kept under another acceptable level. The combined stresses due to a combination of loads acting simultaneously are called *stress categories*.

CLASSES OF STRESS

The pressure vessel codes define three important 'classes' of stress according to the types of loading which produced them and the hazard they represent to the vessel.

1. Primary Stress
    a. General
        - Primary general membrane stress, P_m
        - Primary general bending stress, P_b
    b. Local Primary Stress, P_L
2. Secondary Stress
    a. Secondary membrane stress, Q_m
    b. Secondary bending stress, Q_b
3. Peak Stress, F

Primary Stress

Primary stress is related to mechanical loading directly and is the most hazardous of all types of loads. These stresses act over a full cross section of the vessel. Any yielding through the entire shell thickness will not distribute the stress, but will result in gross distortions, often carried to failure. The main characteristic of primary stresses is that they are not self-limiting, i.e., they are not reduced in magnitude by the deformation they produce. Primary stresses are generally due to internal or external pressure or produced by sustained external forces and moments. **Thermal stresses are never classified as primary stresses.**

<u>Primary general stresses</u> are divided into membrane and bending stresses. The need for dividing primary general stress into membrane and bending is that the calculated value of the primary general bending stress may be allowed to go higher than that of primary general membrane stress.

Primary general membrane stress, P_m, occurs across the entire cross section of the vessel due tomechanical loads such as internal pressure, dead weight, orwind loads. It is remote from discontinuities such as head-shell intersections, cone-cylinder intersections, nozzles and supports. It is sodistributed in the structure that no redistribution of loadoccurs as a result of yielding.ASME Section VIII-1 establishes allowable stresses by stating that the maximum general primary membrane stress must be less than allowable stresses provided in the material sections. Examples of such stresses are a) circumferential and longitudinal stress due to pressure, b) compressive and tensile axial stress due to wind, c) longitudinal stress due to the bending of the horizontal vessel over saddles, d) membrane stress in the center of flat head, e) membrane stress in the nozzle wall with in the area of reinforcement due to pressure or external loads, and f) axial compression due to weight.

Primary general bending stresses, P_b, are due to sustained loads and are capable of causing collapse of the vessel. ASME Section VIII-1 states that the maximum primary membrane stress *plus* primary bending stress may not exceed 1.5 times the allowable stress provided in the material sections. There are relatively few areas where primary bending occurs: a) bending stress in the center of flat head or in the crown of a dished head, b) bending stress in a shallow conical head, and c) bending stress in the ligaments of closely spaced openings.

<u>Local primary stresses</u> are not technically a classification of stress but a stress category, since it is a combination of two stresses. The combination it represents is the primary membrane stress, P_m, plus secondary membrane stress produced from sustained loads. These have been grouped together in order to limit the allowable stress for this combination to a level lower than allowed for other primary and secondary stress combinations. It is felt that local stress from sustained (unrelenting) loads presents a great enough hazard for the combination to be classified as a primary stress.

A local primary stress is produced either by design pressure alone or by other mechanical loads. Local primary stresses have some self-limiting characteristics like secondary stresses. Since they are localized, once the yield strength of the material is reached, the load is redistributed to stiffer portions of the vessel. However, since any deformation associated with yielding would be unacceptable, an allowable stress lower than secondary stress is assigned. The basic difference between a primary local stress and a secondary stress is that a primary local stress is produced by a load that is unrelenting; the stress is just redistributed. In secondary stress, yielding relaxes the load and is truly self-limiting. The ability of primary local stresses to redistribute themselves after the yield strength is attained locally provides a safety valve effect. Thus, the higher allowable stress applies only to a local area.

Primary local membrane stresses are a combination of membrane stresses only. Thus only the membrane stresses from a local load are combined with primary general membrane stresses, not the bending stresses. The bending stresses associated with a local loading are secondary stresses. Therefore, the membrane stresses from a WRC-107 type analysis must be broken out separately and combined with primary general stresses. The same is true for discontinuity membrane stresses at head-shell junctures, cone-cylinder junctures, and nozzle-shell junctures. The bending stresses would be secondary stresses.

Therefore, $P_L = P_m + Q_m$ where Q_m is a local stress from a sustained or unrelenting load. Examples of primary local membrane stresses are:

a. P_m + membrane stresses at local discontinuities (Head-shell juncture, Cone-cylinder juncture, Nozzle-shell juncture, Shell-flange juncture, Head-skirt juncture, Shell-stiffening ring juncture)

b. P_m + membrane stresses from local sustained loads (Support lugs, Nozzle loads, Beam supports, Major attachments)

Secondary Stress

Secondary stresses are those arising from geometric discontinuities or stress concentrations. If a member is subjected to a stress attributable to a thermal expansion load, such as bending stresses in shell at a nozzle connection under thermal expansion of the piping, a slight, permanent, local deformation in the shell wall will produce relaxation in the expansion forces causing the stress. The basic characteristic of a secondary stress is that it is self-limiting;that is, once the yield point has been passed locally around the stress concentration, the direct relationship between load and stress is broken, due to the reduced post-yield stiffness of the material.. This means that local yielding and minor distortions can satisfy the conditions which caused the stress to occur. The most important self-limiting stresses in the design of pressure vessels are the stresses produced by thermal expansion and by internal pressure at shell structure discontinuities. Application of a secondary stress can not cause structural failure due to the restraints offered by the body to which the part is attached.

Secondary stresses are divided into two additional groups, membrane and bending. Examples of secondary membrane stresses, Q_m, are a) axial stress at the juncture of a flange and the hub of the flange, b) thermal stresses, c) membrane stress in the knuckle area of the head, and d) membrane stress due to local relenting loads. Examples of secondary bending stresses, Q_b, are a) bending stress at a gross structural discontinuity – nozzles, lugs etc. (relenting loads only), and b) discontinuity stresses at stiffening or support rings.

The practical difference between primary and secondary stresses is obvious and the criteria used to evaluate the safety of primary stresses should not be applied to the calculated values of stresses produced by self-limiting loads. Also, this is only true for ductile materials. In brittle materials, there would be no difference between primary and secondary stresses. If the material cannot yield to reduce the load, then the definition of secondary stress does not apply!

In a region away from any discontinuities, only primary stress will arise. The secondary stress cannot arise alone however - at a discontinuity, the secondary stress will be superimposed on the underlying primary stress.When we say *away from discontinuities* we are talking aboutareas of high local stresses such as nozzle-to-shell or cone-to-cylinderjunctions [see sketch below]. It is recognized inSection VIII-1 that high localized discontinuity stresses mayexist in vessels constructed to this standard. These stresses arenot directly calculated but are controlled to a safe level consistentwith experience through design rules and mandatoryfabrication details [e.g., opening reinforcement calculations,minimum 3:1 transition taper at head-to-shell joints, minimumweld sizes for nozzle attachments].

Peak Stress

Peak stresses are the additional stresses due to stress intensification in highly localized areas. They apply to both sustained loads and self-limiting loads. There are no significant distortions associated with peak stresses. Peak stresses are additive to primary and secondary stresses present at the point of stress concentration. Peak stresses are only significant in fatigue conditions or brittle materials. Peak stresses are sources of fatigue cracks and apply to membrane, bending and shear stresses. Examples are a) stress at the corner of a discontinuity, b) thermal stress in a wall caused by a sudden change in the surface temperature, c) thermal stresses in cladding or weld overlay, and d) stress die to notch effect (stress concentration).

STRESSES IN PRESSURE VESSELS

In any pressure vessel subjected to internal or external pressure, stresses are set up in the shell wall. The state of the stress is triaxial and the three principal stresses are 1) longitudinal/ meridional stress, 2) Circumferential/ latitudinal stress, and 3) radial stress. The radial stress is a direct stress, which is the result of the pressure acting directly on the wall, and causes a compressive stress equal to the pressure. In thin-walled vessels, this stress is so small compared to the other principal stresses that it is generally ignored. Thus we assume for the purposes of analysis that the state of stress is biaxial. For thick-walled vessels ($R_m/t < 10$), the radial stress cannot be ignored and the formulas are quite different from those used in finding "membrane stresses" in thin shells.

CATEGORIES OF STRESS

Once the various stresses of a component are calculated, they must be combined and this final result compared to an allowable stress. The combined classes of stress due to a combination of loads acting at the same time are stress categories. Each category has assigned limits of stress based on the hazard it represents to the vessel. The following is derived from ASME VIII-2, simplified for application to ASME VIII-1 vessels and allowable stresses. It should be used as a guideline only because ASME VIII-1 recognizes only two categories of stress – primary membrane stress and primary bending stress.

Stress Classification or Category	Allowable Stress
General primary membrane, P_m	SE
General primary bending, P_b	$1.5SE < 0.9F_y$
Local primary membrane, P_L ($P_L = P_m + Q_{ms}$)	$1.5SE < 0.9F_y$
Secondary membrane, Q_m	$1.5SE < 0.9F_y$
Secondary bending, Q_b	$3SE < 2F_y < UTS$
Peak, F	$2S_a$
$P_m + P_b + Q_m + Q_b$	$3SE < 2F_y < UTS$
$P_L + P_b$	$1.5SE < 0.9F_y$
$P_L + P_b + Q_m + Q_b$	$3SE < 2F_y < UTS$
$P_L + P_b + Q_m + Q_b + F$	$2S_a$

Notes: Q_{ms} = membrane stresses from sustained loads

Q_m = membrane stresses from relenting, self-limiting loads

S = allowable stress per ASME VIII-1 at design temperature

F_y = minimum specified yield strength at design temperature

UTS = minimum specified tensile strength

S_a = allowable stress for any given number of cycles from design fatigue curves

THERMAL STRESSES

Whenever the expansion or contraction that would occur normally as a result of heating or cooling an object is prevented, thermal stresses are developed. The stress is always caused by some form of mechanical restraint. Thermal stresses are “secondary” stresses because they are self-limiting, that is, yielding or deformation of the part relaxes the stress. Thermal stresses will not cause failure by rupture in ductile materials except by fatigue over repeated applications. They can, however, cause failure due to excessive deformations.

Mechanical restraints are either internal or external. An example of external restraint occurs when piping expands into a vessel nozzle creating a radial load on the vessel shell. Internal restraint occurs when the temperature through an object is not uniform. Stresses from a “thermal gradient” are due to internal restraint. Stress is caused by a thermal gradient whenever the temperature distribution or variation within a member creates a differential expansion such that the natural growth of one fiber is influenced by the different growth requirements of the adjacent fibers. The result is distortion or warpage.

Thermal stresses cause creep in the material at high temperatures and are very troublesome. The elastic limit or the yield stress of the material is very low at high temperatures. Hence the material gets into the plastic range to relieve the stresses. Furthermore, if the pressure fluctuates, thermal ratcheting takes place where each loading cycle results in incremental strain, which is highly unacceptable situation.

DISCONTINUITY STRESSES

Vessel sections of different thickness, material, diameter and change in direction would all have different displacements if allowed to expand freely. However, since they are connected in a continuous structure, they must deflect and rotate together. The stresses in the respective part at or near the junctures are called discontinuity stresses. Discontinuity stresses are necessary to satisfy the compatibility of deformation in the region. They are local in extent but can be of very high magnitude. Discontinuity stresses are also secondary stresses and are self-limiting. That is, once the structure has yielded, the stresses are reduced. In average applications, they will not lead to failure. Discontinuity stresses do become an important factor in fatigue design where cyclic loading is a consideration. Design of the juncture of the two parts is a major consideration in reducing discontinuity stresses.

In order to find the state of stress in a pressure vessel, it is necessary to find both the membrane stresses and the discontinuity stresses. From superposition of these two states of stress, the total stresses are obtained. Generally when combined, a higher allowable stress is permitted. One example specifically addressed by the ASME VIII-1 Code is discontinuity stresses at cone-cylinder intersections where the included angle is greater than 60^o. Paragraph 1-5(g) recommends limiting combined stresses (membrane + discontinuity) in the longitudinal direction to 3SE and in the circumferential direction to 1.5SE.

ALLOWABLE STRESSES

The Code-allowable stresses are determined by the ASME Subcommittee on Materials and are listed in ASME Section II, Part D of the B&PV Code. The basic rules for acceptance of new materials are contained in the “Guideline on the Approval of New Materials under the ASME Boiler and Pressure Vessel Code” (found in Section II, Part D, Appendix 5).

The allowable stresses of carbon steel material are based on properties data provided to the Subcommittee from at least three heats of the material. The properties that must be included are the tensile and yield strengths at 100ºF (38ºC) intervals from room temperature to 100ºF (38ºC) above the maximum intended use temperature. Also, if the material is expected to be used in the time-dependent temperature range (that is, creep), creep rate and stress rupture data must be included starting at approximately 50ºF (10ºC) below the temperature at which the time-dependent properties might govern to 100ºF (38ºC) above the maximum use temperature. Duration of at least 6000 hours is required for the creep rupture tests.

The basis for the allowable stresses can vary in different Codes, although the bases are generally the same for most power plant applications. Recent changes to the safety factor in the B&PV Code have resulted in increased allowable stresses (the safety factor based on tensile strength was reduced from 4 to 3.5). Although different Codes might have different requirements for the allowable stresses, the criteria used to establish the allowable stress for the Code's Tables 1A and 1B are shown in Table 1-100 of Appendix 1 of ASME Section II, Part D.

These criteria follow:

- (1/3.5) x the tensile strength at temperature (2YS/3)
- (2/3) x the yield strength at temperature (TS/3.5)
- A percentage of the creep rupture strength dependent on the testing period.

The data are used to develop trend curves. Each of these values (TS/3.5, 2YS/3, and the creep strength value) is plotted against the temperature, and the lowest value is the allowable stress for that material and that temperature. See Figure 1 below for an example plot for SA-516 Gr. 65.

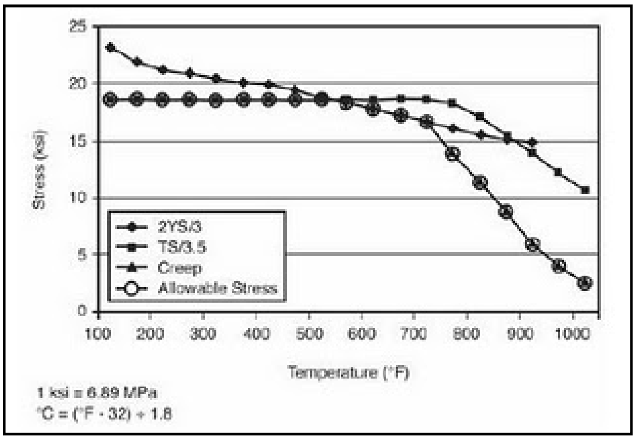

Figure 1: Trend Curve for SA 516 Gr. 65

Allowable stresses must be obtained from the applicable Code. The allowable stresses are subject to change because they are a function of the safety factor used in the applicable Code and of the properties of the material specification (which are also subject to change). There are also differences in the temperature limits for the materials. Due to the fact that the strength requirements and the pressure-temperature tables of the standards are subject to change, particular attention should be paid to the edition reference of the material specification or referenced standard. Prior to referencing a later edition, the Code committees review these changes and adjust the allowable stresses accordingly.

MAXIMUM ALLOWABLE TENSILE STRESSES

The maximum allowable stress is the maximum unit stress permitted in a given material used in the vessel. The maximum allowable tensile stress values permitted for different materials are given in ASME Section II-D. A listing of these materials is given in the following tables, which are included in Subsection C of ASME Section VIII-1. Subsection C contains the requirements pertaining to classes of materials. For materials identified as meeting more than one material specification and/or grade, the maximum allowable tensile stress value for either material specification and/or grade may be used provided all requirements and limitations for the material specification and grade are met for the maximum allowable tensile stress value chosen.

Table UCS-23	Carbon and Low Alloy Steel (stress values in Section II-D, Table 3 for bolting, and Table 1A for other carbon steels)
Table UNF-23	Nonferrous Metals (stress values in Section II-D, Table 3 for bolting, and Table 1B for other nonferrous metals)
Table UHA-23	High Alloy Steels (stress values in Section II-D, Table 3 for bolting, and Table 1A for other high alloy steels)
Table UCI-23	Maximum Allowable Stress Values in Tension for Cast Iron
Table UCD-23	Maximum Allowable Stress Values in Tension for Cast Ductile Iron
Table UHT-23	Ferritic Steels with Properties Enhanced by Heat Treatment (stress values in Section II-D, Table 1A
Table ULT-23	Maximum Allowable Stress Values in Tension for 5%, 8%, and 9% Nickel Steels and 5083-0 Aluminum Alloy at Cryogenic Temperatures for Welded and Nonwelded Construction

MAXIMUM ALLOWABLE COMPRESSIVE STRESSES

The maximum allowable longitudinal compressive stress to be used in the design of cylindrical shells or tubes, either seamless or butt welded, subjected to loadings that produce longitudinal compression in the shell or tube shall be the smaller of the following values:

1. The maximum allowable tensile stress value
2. The value of Factor *B* determined by the following procedure where

E = modulus of elasticity of material at design temperature. The modulus of elasticity to be used shall be taken from the applicable materials chart in Section II-D, Subpart 3 (Interpolation may be made between lines for intermediate temperatures)

R_o = outside radius of cylindrical shell or tube

t = the minimum required thickness of the cylindrical shell or tube

The joint efficiency for butt welded joints shall be taken as one. The value of *B* shall be determined as follows:

Step 1. Using the selected values of *t* and *R*, calculate the value of Factor *A* using the following formula:

$$A = \frac{0.125}{\left(R_o/t\right)}$$

Step 2. Using the value of *A* calculated in Step 1, enter the applicable material chart in Section II-D, Subpart 3 for the material under consideration. Move vertically to an intersection with the material/temperature line for the design temperature. Interpolation may be used between lines for intermediate temperatures. If the tabular values in Subpart 3 of Section II-D are used, linear interpolation or any other rational interpolation method may be used to determine a *B* value that lies between the two adjacent tabular values for a specific temperature. Such interpolation may also be used to determine a *B* value at an intermediate temperature that lies between two sets of tabular values, after first determining *B* values for each set of tabular values.

In cases where the value at *A* falls to the right of the end of the material/temperature line, assume an intersection with the horizontal projection of the upper end of the material/temperature line. If tabular values are used, the last (maximum) tabulated value shall be used. For values of *A* falling to the left of the material/temperature line, see Step 4.

Step 3. From the intersection obtained in Step 2, move horizontally to the right and read the value of Factor *B*. This is the maximum allowable compressive stress for the values of *t* and R_o used in Step 1.

Step 4. For values of *A* falling to the left of the applicable material/temperature line, the value of *B* shall be calculated using the following formula:

$$B = \frac{AE}{2}$$

If tabulated values are used, determine *B* as in Step 2 and apply it to equation in Step 4.

Step 5. Compare the value of B determined in Steps 3 or 4 with the computed longitudinal compressive stress in the cylindrical shell or tube, using the selected values of t and R_o. If the value of B is smaller than the computed compressive stress, a greater value of t must be selected and the design procedure repeated until a value of *B* is obtained that is greater than the compressive stress computed for the loading on the cylindrical shell or tube.

WALL THICKNESS

The wall thickness of a vessel shall be determined such that, for any combinations of loadings listed in UG-22 that induce primary stress and are expected to occur simultaneously during normal operation of the vessel, the induced maximum general primary membrane stress does not exceed the maximum allowable stress value in tension (See exception for combination of earthquake loading, or wind loading with other loadings). Except where limited by special rules, such as those for cast iron in flanged joints, the above loads shall not induce a combined maximum primary membrane stress plus primary bending stress across the thickness that exceeds 1½ times the maximum allowable stress value in tension. It is recognized that high localized discontinuity stresses may exist in vessels.

The primary plus secondary stresses at discontinuities shall be limited to S_{PS}, where S_{PS} = 3S, and S is the maximum allowable stress of the material at the temperature. In lieu of using S_{PS} = 3S, a value of S_{PS} = 2S_Y may be used where S_Y is the yield strength at the temperature, provided the following three conditions are met:

1. The allowable stress of the material is not governed by time-dependent properties as provided in Table 1A or 1B of Section II-D;
2. The room temperature ratio of the specified minimum yield strength to specified minimum tensile strength for the material does not exceed 0.7; and

The value of S_Y at temperature can be obtained from Table Y-1 of Section II-D.

For the combination of earthquake loading, or wind loading with other loadings, the wall thickness of a vessel shall be determined such that the general primary membrane stress shall not exceed 1.2 times the maximum allowable stress permitted. This rule is applicable to stresses caused by internal pressure, external pressure, and axial compressive load on a cylinder. Earthquake loading and wind loading need not be considered to act simultaneously.

Source: ASME Boiler and Pressure Vessel Code, Section VIII, Division 1, Edition 2015

Harvey, John F., Theory and Design of Modern Pressure Vessels

Jawad, Maan H. and James R. Farr, Structural Analysis and Design of Process Equipment

Moss, Dennis R., Pressure Vessel Design Manual

Bednar, Henry, Pressure Vessel Design Handbook

TANK INSPECTION, REPAIRS AND RECONSTRUCTION

Introduction

The trend in the domain of public and legislative bodies is to view aboveground storage tanks (ASTs) as environmentally hazardous equipment. Several recent AST accidents have contributed to this viewpoint and have resulted in the current AST regulatory atmosphere that exists on local, state and federal levels. The stringent regulations are, however, corrective rather than prescriptive or preventive measures. *Corrective* means that rather than the prevention of AST leaks and spills being addressed through design and inspection, the issues are addressed by requirements for secondary containment and other measures that go into force only after a spill or leak has occurred. API 653 was introduced to address the areas that were not covered by the existing regulations.

When designing and constructing new tanks, the engineer has several industrial standards that will provide a state-of-art tank design. Among these standards are API 650, API 620 etc. that are based upon accumulated industrial experience and will provide a tank that is reasonably free from the chance of failure when placed into service. However, the standards were not intended to eliminate maintenance and inspection procedures that can result in failure from long term service effects such as corrosion, foundation settlement, changes of service, or improper structural modifications made to the tank. API 653 was created to fill this void, and it applies to tanks that have already been placed into service.

API 653 is a comprehensive prescriptive approach to the problem of AST leaks and spills. It is the first line of defense against various tank failure modes. It is used in conjunction with other API publications like:

API RP 570	Piping Inspection
API RP 651	Cathodic Protection for Aboveground Petroleum Storage Tanks
API RP 652	Lining of Aboveground Petroleum Storage Tanks
API Standard 650	Welded Steel Tanks for Oil Storage
API Standard 620	Design and Construction of large, Welded Low Pressure Storage Tanks
API Standard 2000	Venting Atmospheric and Low Pressure Storage Tanks
API RP 2003	Protection against Ignitions Arising out of Static, Lightening, and Strong Currents
API Publication 2015	Cleaning Petroleum Storage Tanks
API Publication 2207	Preparing Tank Bottoms for Hot Work
API Publication 2217	Guidelines for Continued Space Work in the Petroleum Industry

Intent of API Standard 653

API 653 outlines a program of minimum requirements for maintaining the integrity of welded and riveted, non-refrigerated, atmospheric storage tanks. Most ASTs including the foundation, bottom, shell, structure, roof, appurtenances, and nozzles fall within its scope. This standard covers carbon and low alloy steel tanks built to API standard 650 and its predecessor API 12C. it covers the maintenance inspection, repair, alteration, relocation, and reconstruction of such tanks.

How is API 653 standard organized?

The fifth edition API 653 issued in 2014 is organized as follows:

1. Scope
2. References

3. Definitions
4. Suitability for Service
5. Brittle Fracture Considerations
6. Inspection
7. Materials
8. Design Considerations for Reconstructed Tanks
9. Tank Repair and Alteration
10. Dismantling and Reconstruction
11. Welding
12. Examination and Testing
13. Marking and Recordkeeping

There are 12 Appendices:

Appendix A: Background of Past Editions of API Welded Storage Tank Standards
Appendix B: Evaluation of Tank Bottom Settlement
Appendix C: Checklists for Tank Inspection
Appendix D: Authorized Inspector Certification
Appendix E: *Left Blank*
Appendix F: NDE Requirements Summary
Appendix G: Qualification of Tank Bottom Examination Procedures and Personnel
Appendix H: Similar Service Assessment
Appendix I: Inquiries and Suggestions for Change
Appendix S: Austenitic Stainless Steel Storage Tanks
Appendix SC: Stainless and Carbon Steel Mixed Material Storage Tanks
Appendix X: Duplex Stainless Steel Storage Tanks

How does API 653 Prevent Tank Failures?

API 653 provides three mechanisms to prevent potential storage tank failures:

<u>Assessment</u>: API 653 places great emphasis on engineering experience and evaluation of as AST's suitability for service. It also requires that this evaluation be conducted by "organizations that maintain or have access to engineering and inspection personnel technically trained and experienced in tank design, fabrication, repairs, construction and inspection". Suitability for service does not necessarily involve a physical change to the tank. One example would be a change in service temperature. Following are typical cases where an evaluation should be performed:

- Storage of fluids that are incompatible with the storage tank materials of construction, which could lead to pitting, unpredictable rates of corrosion, stress corrosion cracking etc.
- A change in stored product density
- Distortion of the shell, roof or bottom
- An observable change or movement in shell distortions
- Very high fluid transfer rates into or out of the tank
- High, low or variances in service temperatures
- Local thin areas in the shell
- The presence of cracks
- Brittle fracture considerations
- Foundation problems

Inspection: The philosophy of the standard is to gather data and to perform a thorough initial inspection in order to establish a baseline for each tank against which future inspections may be used to determine the rate of corrosion or changes that might affect its suitability for service. Key to the philosophy of the standard is the observation of changes and rates of change that affect the physical tank. From these data, the experienced tank engineer may be able to judge the suitability for continued service or the need for repairs.

Repair, Alteration and Reconstruction Guidelines: This standard provides guidelines for many of the common repairs and alterations that are done to tanks, including:

- Alteration of existing nozzles
- Patch plates
- Bulges and out-of-round conditions
- Bottom repairs
- Replacement of the bottom
- Roof repairs
- Floating roof seal repairs
- Hot taps
- Repair of defective welds

Responsibility and Compliance

The owner has the ultimate responsibility for complying with or not complying with the provisions of API 653. In some cases, the tank owner may have requirements more stringent than those described in API 653, while in some others, the owner may not endorse certain practices in API 653.

Is compliance really mandatory? In some states, the answer is clear-cut yes because the compliance with API 653 is legislated. However, for most facilities, the answer is not so clear. A standard in itself is rarely mandatory but is frequently made so by reference to industrial standards and good engineering practices. Sometimes, compliance to this standard may be mandatory merely because nothing better exists. An example of this is EPA's Spill Prevention Control and Countermeasures (SPPC) regulations. This regulation, which is applicable to facilities near navigable waterways, requires that tanks be inspected at regularly scheduled intervals and that the inspections be documented. EPA's SPPC program does not mandate the use of API 653. However, because API 653 is the only recognized industrial inspection standard for ASTs, it would seem that the compliance with the standard is, by default, required, unless the owner is already doing all the things outlined in API 653.

In many ways, the responsibility for compliance to API 653 is like an insurance policy. It increases the operating costs associated with ASTs, but in the long run these costs can be more than recouped. Many facilities have paid the costs of site remediation resulting from AST leaks and spills and the heavy fines levied by EPA costing much more than the preventive costs to implement and maintain the API 653 program. Many of the notable AST catastrophes would not have occurred if they had been on a FFS program such as outlined in API 653.

How long will it take to implement the API 653 program?

Because of the amount of work associated with compliance for a large facility with many tanks, the issue of how long the compliance should take is left undefined in the standard. Since the standard requires that internal inspections be performed at intervals not exceeding 10 years (greater periods are an exception), it would seem that a facility should have scheduled all its ASTs for the first comprehensive, internal inspection not more than 10 years from now. Many companies are proactive in their plans to comply within a 3- to 5-year timeframe. Tank failures are devastating to the owner company's public image and raise jurisdictional concerns about why the tank was not in compliance with an existing industry standard. So, the sooner you comply, the better your chances for reducing possible leaks and failures, public image problems, environmental civil and possibly even criminal penalties and liabilities.

API 653 and Costs

API 653 can have an impact on new tank designs in achieving the lowest capital and operating costs desired. Since the interval between internal inspections is governed by such factors as the use of a liner, amount of corrosion allowance, cathodic protection, and leak detection, these factors should be taken into account when a new tank is being designed. The cost of implementing API 563 can be impacted by the up-front planning and thinking that go into setting up such a program. These costs are expected to fall into three broad categories:

Costs for internal inspection

These costs are very significant and can easily run into millions of dollars per year for a large integrated oil company. Most of the costs can be attributed to the cost of preparing the tank for an internal inspection and interrupting its in-service operations. API 653 lists various methods of increasing the time interval between internal inspections that can result in interval to range from just a few years to as many as 20 years.

Costs based on inspection findings

In assessing suitability for service, there is an opportunity to save costs. For example, either an existing tank that has many violations of current standards can be brought up to date without regard to costs, or an engineering evaluation can be done which determines just what has to be brought up to date to make the tank fit for service. The cost difference between these two approaches can be significant.

Costs associated with recordkeeping

One of the goals of setting up and maintaining a record keeping system should be to develop and standardize a system that would optimally include standardized software to minimize overall costs to the owner.

In-house versus Contract Inspection

A large facility such as an oil refinery usually can afford to staff full-time API 653 certified inspectors. The advantage of running an in-house inspection program is that it can be controlled, it provides uniform results, and it has the potential to be the lowest cost system for maintaining tank integrity. However, there is a potential for conflict of interest. If operations is pressured not to take tanks out of service, then the owner's tank inspectors will be pressured not to take tanks out of service as well. API 653 attempts to resolve this by making a statement in the standard that "such inspectors shall have the necessary authority and organization freedom to perform their duties.

Doing the work, on the other hand, by contracting it has the advantage of contractors who specialize in this work, and who have the expertise and equipment to perform the tasks efficiently. The disadvantage to contracting the work is that the inspection agencies usually perform the recommended repairs as well, generating an inherent conflict of interest. The contracted inspection agencies also tend to be more conservative in their recommendations as a result of product liability issues.

Many companies have developed a hybrid of these options, contracting for the data collection efforts by contractors who have the necessary inspection equipment and performing the engineering and design assessments on their own.

Source: Aboveground Storage Tanks *by* Philip E. Meyers

API 653 Fifth Edition – November 2014

10 TIPS FOR A HAPPIER, HEALTHIER LIFE

By Nutritionist Dr. John Briffa

1) Eat "primally'. Common sense dictates that the best diet is one based on foods we have been eating the longest in terms of our time on this planet. These are the foods that we have evolved to eat and are best adapted to. Studies show that a "primal" diet made up of fruits, vegetables, nuts and seeds, as well as meat, fish and eggs, is best for health and improvement in risk markers for illnesses, such as heart disease and diabetes.

2) Keep hydrated. Water makes up two-thirds of the body and performs a plethora of functions, including acting as a solvent, carrier of nutrients, temperature regulator and body detoxifier. Maintaining hydration can have a profound influence on our vitality and energy levels, including mental alertness. Aim to drink enough water to keep your urine a pale yellow color throughout the course of the day.

3) Eat mindfully. In our fast-paced world, there can be a tendency to eat while distracted and shovel in more food than we need and, at the same time, miss out on culinary pleasure. Many of us will benefit from eating mindfully. Some things to think about are avoiding eating when distracted, eating more slowly, and taking time to taste food properly. One particular thing to focus on is chewing your food thoroughly – not only does this help us savor food, more importantly it also assists the digestive process.

4) Get plenty of sunlight in summer... Sunlight, and the vitamin D this can make in the skin, is associated with a wide spectrum of benefits for the body including a reduced risk of several forms of cancer, heart disease, multiple sclerosis and osteoporosis, as well as improved immune function. As a rule of thumb, vitamin D is made when our shadow is shorter than our body length, i.e., when the sun is high in the sky. While burning is to be avoided, get as much sunlight exposure as possible for optimal health.

5) …and in winter. Low levels of sunlight in winter can cause our mood to darken. Even when it is cold outside, it pays to get some external light exposure in the winter, say during lunchtime.

6) Get enough sleep. Sleep has the ability to optimize mental and physical energy, and optimal levels of sleep (about eight hours a night) are linked with reduced risk of chronic disease and improved longevity. One simple strategy that can help ensure you get optimal amounts of sleep is to go to bed earlier. Getting into bed by 10 pm or 10:30 pm is a potential useful investment in terms of your short- and long-term health and wellbeing. Shutting down the computer or turning off the TV early in the evening is often all it takes to create the time and space for earlier sleep.

7) Walk regularly. Aerobic exercise, including something as uncomplicated and low-impact as walking, is associated with a variety of benefits for the body and the brain, including a reduced risk of chronic diseases, anti-anxiety and mood-enhancing effects. Aim for a total of about 30 minutes of brisk walking every day.

8) Engage in some resistance exercise. Resistance exercise helps to maintain muscle mass and strengthens the body. This has particular relevance as we age, as it reduces the risk of disability and falls. Many highly useful exercises can be done at home, such as press-ups, sit-ups and squats. Invest in dumbbells to extend your home routine to other exercises, too.

9) Practice random acts of kindness: Random acts of kindness are good for givers and receivers alike. It could be a quick call or text to someone you care about or have lost touch with, or showing a fellow motorist some consideration, or giving up your seat on a train or bus, or buying someone lunch or giving a spontaneous bunch of flowers.

10) Practice the art of appreciation. Modern-day living tends to be aspirational and we can easily find ourselves chasing an ever-growing list of goals, many of which can be material. Some of us could do with more time focusing not on what we don't have, but on what we do. Our mood can be lifted by giving thanks for anything from our friends and family to a beautiful landscape or sunset.

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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