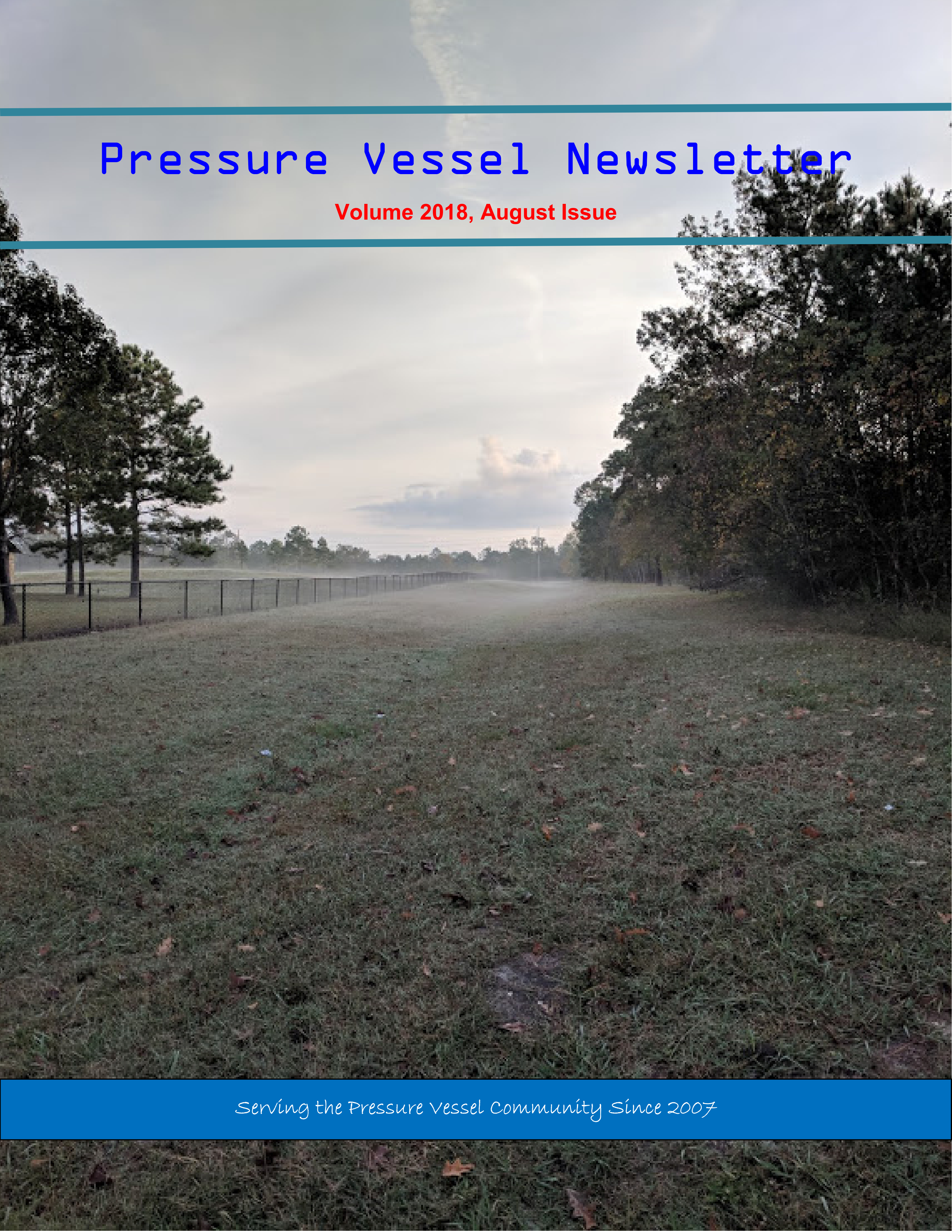
Pressure Vessel Newsletter

Volume 2018, August Issue



Serving the Pressure Vessel Community Since 2007

From The Editor's Desk:

Most engineering companies in Houston area, and I suspect elsewhere in US as well, have switched to either a four day work week or nine days every two weeks schedule. They have done so because this saves the companies one day a week (or every two weeks) with resultant savings on utilities and other overheads.

Regardless of the work schedule, every company treats 40 hours work week as a sacred concept that cannot be tampered with. So eight hour a day is no longer the norm. Nine hour days, or even 10 hour days have become the norm, and this is where I would like to focus the discussion today.

In the last 25 years, technology has ushered big changes in the ways employees perform their work. Software has made tedious hand calculations obsolete – a design that used to take days to complete can now be finished in an hour or less. Whether it is accounting or engineering or construction, most tasks are now automated or performed with help of software. The net result is that the productivity has improved multifold and it is now possible to complete projects much faster than ever before.

So clearly we are getting much more done in a day with an 8 hour day now. How much more work do we want to accomplish in a single day? If we can accomplish all the work in less than 8 hours, should we still require the employees to work 8 hours to earn a full day's pay? Studies have shown that working extended hours reduces effectiveness. Elsewhere in the world, many countries are experimenting with 35-hour work week and reporting encouraging results.

I think it is time for US companies to get on board as well and get away from the long practiced 40-hour work week model. There is not expected to be any loss of productivity, the employees will get to spend more time with the family/friends and will be thankful for it. As the saying goes, a happy employee is a productive employee.

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In this issue…

COMPARISON OF THE ASME, BS AND CEN FATIGUE DESIGN RULES FOR PRESSURE VESSELS

This paper was presented at IMechE Seminar, 'Which Code for Pressure Vessels – ASME, BS or CEN?' by Stephen Maddox on October 1, 2003 in London, UK. It has been reproduced here unaltered – only the formatting has been changed.

Please note that the ASME Code has undergone several revisions since this article was published and many of the points raised here may no longer be valid.

INTRODUCTION

Revision of the fatigue design rules in BS 5500 (now PD 5500), to bring them more into line with those in British Standard specifications for other structures, was one of the most radical changes in recent years. Subsequently, similar rules for welded vessels, but allied more closely with those in Eurocode 3 and the IIW fatigue design recommendations, were included in the new CEN pressure vessel rules, EN 13445. Meanwhile, ASME VIII, which also contains fatigue design rules, continues to be used. This paper compares the fatigue design rules in these three codes, identifies the main differences and considers them in the light of experimental evidence. It is known that the rules in ASME VIII are currently under review and that major changes are likely to be made in the near future. In view of this situation, rather less attention is paid to ASME than the other two codes.

FATIGUE DESIGN METHODS

Basis

All three codes base fatigue design on S-N curves used in conjunction with Miner's rule to assess variable amplitude loading. However, the basis of the design curves, their final forms and the ways in which they are used differ significantly. This partly reflects differences in emphasis but also in design philosophy.

The ASME rules date back to the early 1960s. They are based on the concept that the fatigue life of any component or structure can be estimated from the S-N curve for the material concerned, as obtained from fatigue tests on small polished specimens, by applying appropriate fatigue strength reduction factors (K_f). Initially, the design curves were obtained from low-cycle fatigue tests conducted under strain control and they covered life up to only 10^6 cycles. Later, they were extended to very long life 10^{11} cycles, presumably to enable potential fatigue damage from vibration to be assessed. Even so, the overall design approach is clearly directed mainly at high-strain low-cycle fatigue conditions. Furthermore, little attention is paid to weld details as sources of fatigue, implying that non-welded features, such as crotch corners in nozzles, are expected to be the most critical locations.

In contrast, both BS and CEN place particular emphasis on the fatigue assessment of weld details. Furthermore, they recognize the fact that the original ASME concept, that the plain material S-N curve can be simply factored to produce a design curve for a structural detail, is not applicable to weld details. Consequently, they provide completely different design data for assessing plain material and weld details, with the latter based on those developed for designing other welded structures (notably bridges and offshore structures). The data used to derive the design curves for weld details relate more to high-cycle than low-cycle fatigue, but it has proved possible to adapt them to cover the full range of endurances relevant to pressure vessels.

Design Procedures

All three design codes provide 'screening tests' to enable further fatigue analysis to be avoided, generally based on a limit to the number of stress cycles expected during the design life of the vessel. In addition, they all provide simplified fatigue analysis procedures based on assumptions about the loading and fatigue resistance. Finally, they all provide a detailed fatigue analysis procedure. These approaches are compared later.

Only ASME provides guidance on the use of special fatigue testing to prove a particular vessel or part, as a substitute for design. Plans are in hand to provide a similar route in the CEN rules. Meanwhile, guidance in the CEN and BS codes is confined to the use of special fatigue tests to validate or change a design curve for a particular detail. In all cases, the results of the testing are analyzed in such a way that they embody the same level of safety as the official design curves.

Design Curves

The methods for deriving the design curves from fatigue test data differ between the codes. In the case of ASME and the curves for assessing plain steel in the CEN rules, selected safety factors were applied to the mean curve fitted to the data. These factors were 2 on stress or 20 on life in the case of ASME compared with 1.5 on stress and 10 on life in the CEN rules. One consequence is that the design curves are non-linear even when plotted on a log-log basis.

The design curves for weld details in the BS and CEN rules are statistical lower bounds to test data. Therefore, in contrast to the curves for assessing plain materials, they embody known probabilities of survival. This is more in keeping with the design data in most modern fatigue design rules than the use of arbitrary safety factors. The probability of survival is higher in the CEN rules than in the BS. A further difference compared with the curves for plain materials is that they are linear on a log-log basis, extending down to a fatigue limit for constant amplitude loading at which point they become horizontal. For the usual practical case of variable amplitude loading, discussed later, this fatigue limit is ignored and the curves are extrapolated linearly at a shallower slope. In the case of CEN, the curves then terminate with an absolute fatigue limit at $N = 10^8$ cycles.

Material

The fatigue design rules in all cases cover all the materials specified in the relevant code. ASME is the most comprehensive in this respect, including steels, nickel and copper alloys, but not aluminum alloys. The British Standard covers steel and aluminum alloys, but the CEN rules are confined to steels. However, there is no doubt that the main background information in all cases comes from fatigue testing of steels, mainly ferritic. The author is not aware on any experimental evidence obtained specifically from pressure vessels to validate the design rules for any material other than steels.

Environment

All the rules require correction of the design curves for operation at elevated temperature, with rather similar values on the upper limit temperatures to which they apply (all below the creep regime). However, this is achieved simply on the basis of the reduction in elastic modulus in the ASME and BS rules. Higher factors, resulting in larger reductions in the design curves, are required by the CEN rules.

Although all the codes draw attention to the deleterious effect of a corrosive environment, none of them provides specific design data. Practical guidance in the BS on the operation of vessels in corrosive conditions has been included in the CEN code. That code also refers to the need to maintain the magnetite layer in water conducting parts in non-austenitic materials at elevated temperature.

Section Thickness

Both the British Standard and the CEN rules recognize that fatigue strength tends to decrease with increasing section thickness, especially in the case of weld details. Thus, correction factors are applied to the design curves when the section thickness exceeds a specified reference value (25mm in CEN, 22mm in the BS). No such correction is required by ASME.

Complex Loading

All three codes contain essentially the same guidance on the derivation of the required stress amplitude or range for multi-axial or combined loading. However, recent research has shown that the method for considering non-proportional loading, where the principal stress direction changes during a cycle, can be unsafe. [9] Alternative design methods are being developed and revision to the codes is likely in future.

Elastic-Plastic Conditions

All three codes introduce corrections to the estimated stress that effectively lower the design curve, if the range exceeds twice yield. The correction procedure is the same in the CEN and British Standard rules, but a different procedure is given in ASME. There are doubts about the validity of both procedures and they are likely to be reviewed in future.

FATIGUE ASSESSMENT OF PLAIN MATERIAL

Design Rules

Both ASME and CEN provide S-N curves for steels related to their tensile strengths (UTS), the assumption being that fatigue resistance increases with increase in tensile strength. In the case of Cu-Ni alloys in ASME, the distinction is based on yield strength. The CEN design curves for steels come from the German AD-Merkblatt code. They are higher than the corresponding curves in ASME, but in practice would usually be lowered to allow for surface finish and for mean stress if this is not zero, whereas the ASME curves would not. The CEN rules also provide a single lower design curve for steels, for use with the simplified assessment method. This is used independently of UTS, surface finish and mean stress. All the design curves were derived from fatigue test results generated from small-scale polished specimens, under stain control in the case of low-cycle fatigue data.

The ASME and CEN curves are used in conjunction with equivalent stresses, the stress amplitude in ASME and the stress range (twice the amplitude) in CEN. Both codes refer specifically to that based on the Tresca yield criterion (i.e., the maximum shear stress, although twice its value, referred to as the 'stress intensity', is actually used in ASME). However, any equivalent stress that 'produces the same fatigue damage as the applied multi-axial stress' is allowed by CEN. However, an implicit assumption is that this is not the maximum principal stress and therefore that the direction in which it acts is not known.

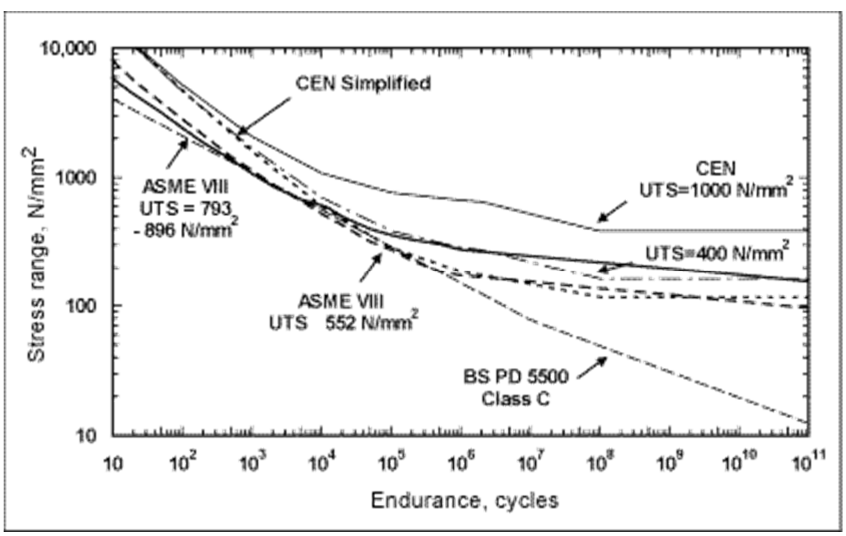

Figure 1: Comparison of Pressure Vessel Design Curves for Plain Steels

The BS provides a single design curve for assessing plain material, independent of UTS, surface finish and mean stress. In contrast to the other codes, it is used in conjunction with the maximum principal stress range. This curve was originally derived from fatigue test results obtained from welded specimens and it is used in other British Standards to assess longitudinal welds as well as plain steel.

Validation

The various design curves are compared in Figure 1. In practice, the CEN curves will usually be lowered as a result of the application of correction factors, but still there are large differences between the curves, chiefly due to the influence of the UTS. A recent review of experimental data obtained from fatigue tests on actual pressure vessels that failed in plain steel revealed not only that this variation was not justified, but also that some of the design curves were too high. The relevant data are shown in Figure 2. Whilst the lower design curves are consistent with this database, the ASME curve for high strength steels and all the CEN curves for use in the detailed assessment procedure are not. There is a clear need for review of the CEN design procedure for assessing plain steel.

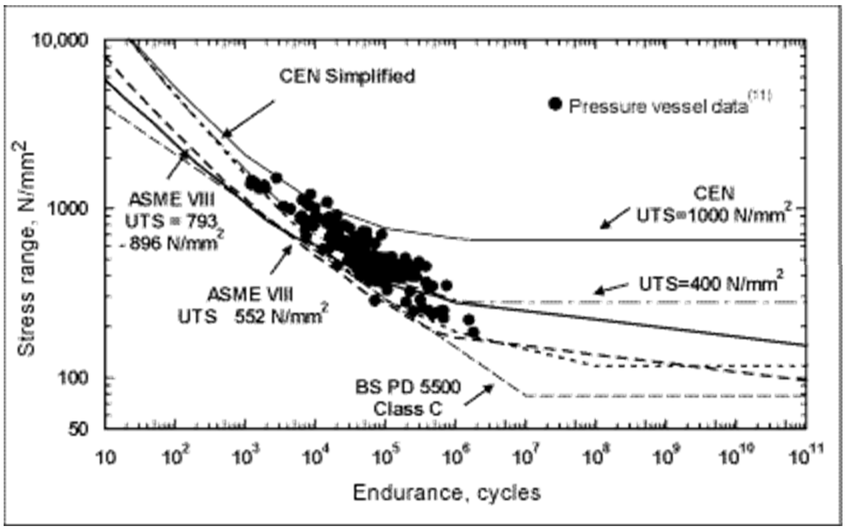

Figure 2: Comparison of Constant Amplitude Design Curves for Plain Steels and Fatigue Data Obtained from Pressure Vessels Failing in Plain Steel (Crotch Corner or Dished End)

FATIGUE ASSESSMENT OF WELD DETAILS

Design Rules

ASME provides very little guidance on the assessment of welds, recommending just one value of the fatigue strength reduction factor, namely 4 for fillet welds.

In contrast, both CEN and BS offer very detailed rules, reflecting the view that most vessels will be welded and the weld details will tend to be the most critical locations for potential fatigue cracking. A major difference between the American and European codes is the method of assessment. It has been known for decades that the fatigue performance of a weld detail is not related to the S-N curve for the parent metal obtained from tests on small polished specimens by a simple fatigue strength reduction factor, as assumed by ASME. Indeed, such an assumption can be very misleading and result in unsafe fatigue life estimates for weld details, notably because it infers a beneficial effect of increased tensile strength when in fact no such benefit actually exists. [8] The fundamental problem is that the fatigue life of the polished specimen is dominated by fatigue crack initiation process, whereas that in most weld details is dominated by fatigue crack growth from some pre-existing discontinuity. Thus, in common with most other design rules for welded structures, the CEN and BS rules are based on fatigue test results obtained from actual welded specimens. One consequence is that there are several design curves provided to cover the full range of weld details relevant to pressure vessels. A difference between the CEN and British Standard design curve is that the former are at least 3 standard deviations of log N below the mean S-N curve, while the latter are approximately 2 standard deviations of log N below the mean. Thus, the CEN curves embody a higher probability of survival than the BS curves and, therefore, for a given weld detail they are usually lower. Since the test data used to derive the design curves included the local stress concentration effect of the weld itself, the design curves are used in conjunction with the structural stress in the vicinity of the weld detail, neglecting the local stress concentration effect of the weld itself.

Each design curve refers to particular weld details, modes of fatigue failure and directions of loading. Tables are provided with sketches linking these features and the appropriate design curve or Class. However, an important difference is that the CEN rules favor use of the equivalent structural stress range, whereas the relevant principal stress range (i.e. that acting normal to the plane of potential fatigue cracking) is used in the BS. The CEN rules offer this as an option with the corresponding classification table in a separate annex. The advantage of the use of the principal stress is that account can be taken of the weld orientation. This is important because the fatigue strength of a weld detail usually varies with the direction of loading. Thus, more precise fatigue design is possible using the principal stress. However, it seems that industries in some European countries still prefer use of the equivalent stress range since this is the stress used for static design of the vessel.

Another difference between the CEN and BS rules concerns the form of the design curve in the high-cycle regime. The CEN design curves are based on those on Eurocode 3 and they embody the same assumption that the constant amplitude fatigue limit corresponds to an endurance of $5x10^6$ cycles. In contrast, the BS adopts the fatigue limit used in other British Standards, which corresponds to an endurance of 10^7 cycles and is therefore lower. There is plenty of experimental evidence to show that some weld details will fail at stresses below that corresponding to $5x10^6$ cycles on the S-N curves. Thus, the BS design curves are considered to be more realistic than CEN in the high-cycle regime.

Two other features of the European rules that contrast with ASME are:

- The inclusion of recommendations for improving the fatigue strength of some weld details, by weld toe grinding;
- Recognition of misalignment as a major source of stress concentration due to the introduction of local secondary bending when the misaligned joint is loaded.

With regard to the latter, the manufacturing rules in all the codes limit the allowable extent of misalignment. However, both the BS and CEN rules note that even allowable misalignment can reduce the fatigue performance of a weld detail to a level below the design curve. Guidance is given on the calculation of the secondary bending stress due to the various types of misalignment relevant to pressure vessels. In contrast, an implicit assumption in the ASME rules is that allowable misalignment will not reduce the fatigue performance of a weld detail below that estimated using the rules.

The guidance on misalignment is essentially an application of the so-called fitness-for-purpose philosophy, whereby an imperfection in the structure can be considered to be acceptable as long as it does not reduce the strength of the structure below that required. The BS encourages use of this same philosophy for assessing the significance of welding flaws in general, making direct reference to BS7910 that provides guidance on the application of the approach.

Validation

Validation of the design method for assessing welds on the basis of fatigue data obtained from actual pressure vessels was an important step in the adoption of the method in the British Standard. The original validation exercise was repeated recently and was extended to consider the CEN rules. Again, the BS rules were validated, but limited evidence suggested that some small changes should be made to the CEN rules.

FATIGUE ASSESSMENT OF BOLTS

Design Rules

ASME and BS provide essentially the same rules for assessing bolts. The corresponding design curves are expressed in terms of the nominal axial stress on the minimum bolt cross-section due to applied tension and bending. They are based on fatigue data obtained from polished specimens and therefore they must be used in conjunction with a fatigue strength reduction factor, 4 unless a lower value can be justified, to allow for the stress concentration effect of the thread root. In contrast, the design curve in BS7608, lowered to correspond to mean - 3 standard deviations of log N, was adopted for CEN. This was based directly on fatigue data obtained from actual (steel) bolts. Thus, in this case, the design curve already incorporates the stress concentration effect of the thread root. Therefore, it is used in conjunction with the nominal stress. In all cases, it is assumed that the fatigue strength of a bolt increases with increase in material UTS, up to specified limits.

Validation

Recent fatigue test data obtained from steel bolts show that the fatigue lives of bolt threads are in fact independent of the tensile strength of the steel. Therefore, there is no justification for assuming a higher design curve for higher strength bolts. However, the new data also suggest that all the present design curves are over-conservative, as seen in Figure 3. A preliminary analysis indicates that a single design curve, independent of steel strength, could be provided.

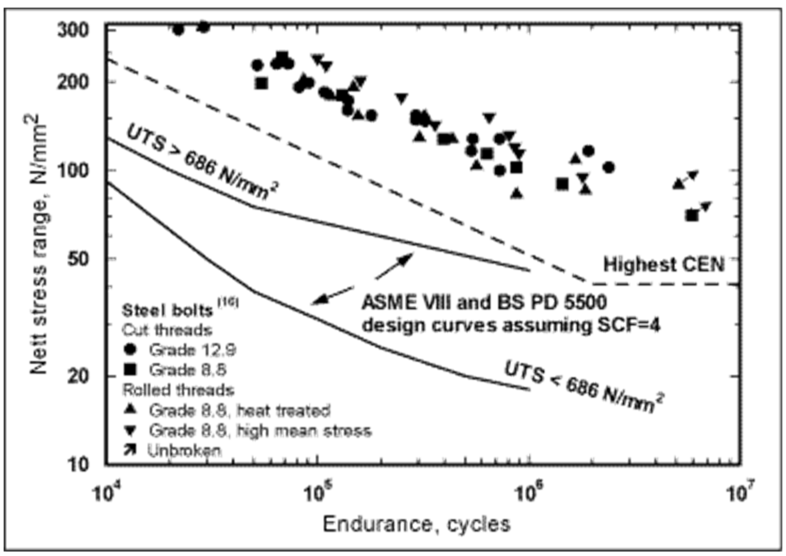

Figure 3: Comparison of Fatigue Test Results Obtained From Steel Bolts in Tension and Pressure Vessels Design Curves

PRACTICAL APPLICATION OF THE FATIGUE RULES

Exemption from Fatigue Analysis

Both ASME and CEN include simple 'screening test' criteria for exemption from fatigue analysis, based on specified numbers of stress cycles (1000 in ASME, 500 in CEN), together with some restrictions on the type of vessel concerned. However, a difference is that ASME allows consideration of both pressure and thermal cycles, whereas CEN is restricted to vessels that only experience pressure cycling.

BS provides a more comprehensive approach that limits a combination of the number of cycles from any source of loading and the design stress to a value that would lie on a relatively low S-N curve. The designer has the option to reduce the allowable design stress for the design of the vessel as a whole to meet the criterion if required.

Simplified Fatigue Design Method

All three codes provide simplified methods that can make use of conservative estimates of cyclic stresses. However, again the BS method is the most comprehensive. In ASME, the various sources of fatigue loading (pressure, temperature and mechanical) are identified and methods for estimating the resulting stress are given. However, it is acknowledged that some of these are non-conservative. The resulting stresses are then compared with the design curve. The corresponding number of cycles from the design curve at that stress must not exceed the number of cycles expected in service from the same load source. Clearly, a further non-conservative feature of this approach is that the possible combined effect of more than one load source, which is always more damaging than that due to the sum of the damage due to the separate load sources, is neglected.

Neither of the non-conservative aspects of the ASME method is present in the BS and CEN approaches, which are similar. However, as in the case of the 'screening tests', the CEN method is restricted to vessels experiencing only pressure loading. The basic method is to make conservative estimates of the cyclic stresses due to the various load sources (only pressure in CEN), perform a simplified cycle counting procedure (which combines load sources in the case of BS), and apply Miner's rule in conjunction with the appropriate design curve, or a specified low curve if this is not known. The BS rules give conservative estimates of the stresses due to pressure and thermal loading, with the basic assumption that details will be located in regions of structural stress concentration with an SCF of 3. However, the user has the option to perform analysis to produce more accurate stresses if required. In this context, a valuable feature of the CEN rules is the inclusion of SCFs (called 'stress factors') for a wide range of structural details.

Cumulative Damage Calculations

As noted earlier, all the rules recommend the use of Miner's rule to assess the fatigue damage introduced under variable amplitude loading. However, only the CEN and BS rules provide guidance on the analysis of fatigue loading, both referring to the use of the Reservoir cycle counting methods for converting complex stress spectra into recognizable cycles. Furthermore, they both acknowledge the need to modify the design S-N curve in the high-cycle regime to allow for the fact that stresses below the original constant amplitude fatigue limit become damaging once a fatigue crack has initiated under higher stresses in the spectrum. They both adopt the commonly used approach of extrapolating the S-N curve beyond the fatigue limit at a shallower slope. The CEN design curves then introduce an absolute cut-off fatigue limit for any loading conditions at 10^8 cycles. Since the ASME curves do not include a sharp cut off at the fatigue limit, in a sense they are already suitable for cumulative damage calculations.

It should be mentioned that there is now an extensive body of experimental data that throw doubt on the validity of Miner's rule and the method of allowing for the damaging effect of stress ranges below the fatigue limit. This is the subject of some research projects and revisions to the current cumulative damage approach may be introduced in future.

Stress Analysis

Although all three codes provide details of the stresses used in pressure vessel design, a particular effort was made in the CEN rules to include clear descriptions of the stresses used specifically in fatigue assessments. Even so, there is still a growing need for clearer guidance on the extraction of the relevant stresses from finite element (FE) outputs.

In the context of detailed stress analysis by methods such as FE, an important development that is likely to influence all the pressure vessel design rules in future is use of the so-called structural hot-spot stress for designing weld details from the viewpoint of potential fatigue failure from the weld toe. Preliminary guidance on the use of hot spot stress is given in both the BS and CEN rules, the latter being particularly detailed. However, the method of deriving the hot-spot stress given in the BS needs revision in the light of more recent research. Similarly, the guidance contained in the CEN document does not go far enough in applying the approach, in that the hot-spot stress could be used with higher design curves in some cases. It is understood that the approach is currently being developed to allow major revision of the ASME fatigue rules for weld details. Meanwhile, of the three design codes considered here, at present the most advanced guidance on the use of the hot-spot stress is that contained in the CEN rules.

FUTURE NEEDS

- Review of rules for assessing plain materials
- Revision of rules for threads and bolts
- Development of hot-spot stress approach
- Review of treatment of complex loading and elastic-plastic fatigue
- Clearer link between FEA and fatigue design data
- Procedure for use of experimental methods in design

UNDERSTANDING ATMOSPHERIC STORAGE TANKS

[THIS IS PART ONE OF TWO PART SERIES ON THIS TOPIC. THE SECOND PART WILL COVER INSTRUMENTATION, STRUCTURAL ACCESSORIES, SPILL CONTROL, ANS LAYOUT AND DESIGN]

Atmospheric storage tanks have been around for a very long time. They are used in various sizes to store liquids throughout the process industries. This article provides a basic understanding of tanks and the related requirements. Lack of sound engineering often results in high costs, shortened equipment life, ineffective inspection programs, environmental damage, or accidents and injuries as well as the threats of more legislation. The purpose of the article is to introduce appropriate information that will make tank facilities safer and more reliable.

NFPA CLASSIFICATIONS OF FLAMMABLE AND COMBUSTIBLE LIQUIDS

The focus here is on hydrocarbons, which deserve particular care because of their flammable or combustible properties. Following are some facts about flammable and combustible liquids:

- Flammable and combustible liquids ignite easily and burn with extreme rapidity.
- Flammability is determined by the flash point of a material.
- Flash point is the minimum temperature at which a liquid forms a vapor above its surface in sufficient concentration that it can be ignited.
- Flammable liquids have a flash point of less than 100°F. Liquids with lower flash points ignite easier.
- Combustible liquids have a flashpoint at or above 100°F.
- The vapor burns, not the liquid itself. The rate at which a liquid produces flammable vapors depends upon its vapor pressure.
- The vaporization rate increases as the temperature increases. Therefore, flammable and combustible liquids are more hazardous at elevated temperatures than at room temperature.

National Fire Protection Association (NFPA) hazard classifications for flammable and combustible liquids are listed in Table 1 below:

Table 1: Hazard Classifications for Flammable and Combustible Liquids

Class	Flash point	Boiling point	Examples
Hazard classification for flammable liquids			
I-A	Below 73°F (23°C)	Below 100°F (38°C)	diethyl ether, pentane, ligroin, petroleum ether
I-B	Below 73°F (23°C)	at or above 100°F (38°C)	acetone, benzene, cyclohexane, ethanol
I-C	73-100°F (24-38°C)	----	p-xylene
Hazard classification for combustible liquids			
II	101-140°F (39-60°C)	----	diesel fuel, motor oil, kerosene, cleaning solvents

Class	Flash point	Boiling point	Examples
III-A	141-199°F (61-93°C)	----	paints (oil base), linseed oil, mineral oil
III-B	200°F (93°C) or above	----	paints (oil base), neatsfoot oil

The classification system is based primarily on the flash point of the liquid; that is, the minimum temperature at which sufficient vapor is given off the liquid to form an ignitable mixture with air. Flammable liquids are classified as Class I, and have flash points below 100ºF. Combustible liquids are classified as Class II and Class III, and have flash points of 100ºF or more. From fire safety standpoint, Class I liquids are most hazardous while Class IIIB liquids are the least hazardous.

TANK CLASSIFICATIONS

Atmospheric storage tanks are defined as those tanks that are designed to operate at pressures between atmospheric and 15 psi, as measured at top of the tank. Such tanks are built in two basic designs – the cone-roof design where the roof remains fixed, and the floating-roof design where the roof floats on top of the liquid and rises and falls with the liquid level.

Fixed-Roof Design

Fixed roof tanks consist of a cylindrical shell with a permanently welded roof that can be flat, conical or dome-shaped. Such tanks are used to store materials with a true vapor pressure of less than 1.5 psi absolute.

External-Floating Roof Design

In floating-roof storage tanks, the roof is made to rest on the stored liquid and is free to move with the level of the liquid. These tanks reduce evaporation losses and control breathing losses while filling. They are preferred for storage of petroleum products with a true vapor pressure of 1.5 psi to 11 psi absolute. There are principally three different types of external floating roofs and an internal floating-roof tank.

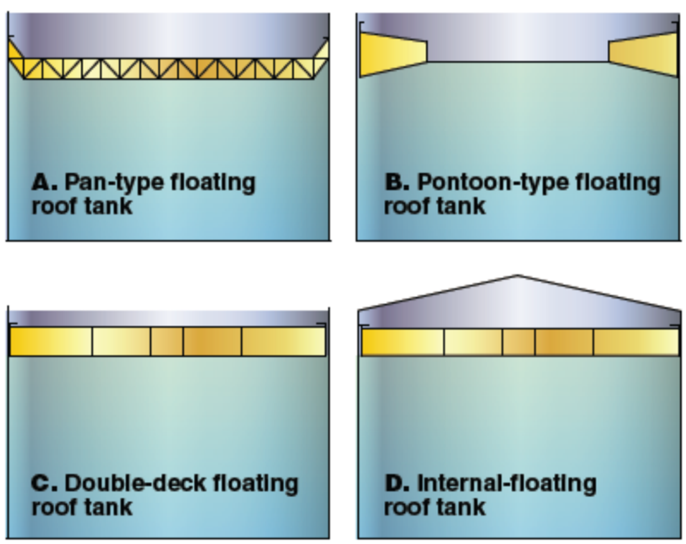

Figure 1: Types of Floating-Roof Tanks

A. Pan-Type Roof

This is a single-deck roof and has following characteristics:

- Full contact with liquid surface

- Has a deck, hence any leak through the deck will cause it to sink
- Has no buoyancy other than that provided by the deck
- Rain or snow may cause deformation
- Is the least expensive of the floating roofs

B. Pontoon-Type Roof

This is a significant improvement over the pan roof and has following characteristics:

- Increased buoyancy and stability
- Pontoons occupy about 20-40% of roof area

C. Double-Deck Roof

This roof comprises upper and lower decks separated by bulkheads and trusses. These roofs have the following characteristics:

- The space between the decks is separated into liquid-tight compartments
- Superior loading capacity
- Recommended for tank diameters below 30 feet and above 200 feet.

Internal-Floating Roof Design

These tanks (Figure 1.D) have an inside floating deck above which there is a permanently attached roof. Such tanks are preferred in areas of heavy snowfall where accumulation of snow or water on the floating roof may affect buoyancy. In such tanks, the vapor space is normally blanketed with an inert gas.

PHYSICAL CRITERIA

Tank Capacity

Three types of tank capacity are defined – nominal, gross and net capacity.

For fixed-roof tanks, the nominal capacity is the geometric volume from bottom of the tank up to the curb angle which is a metallic angle that is welded along the periphery at the top of the cylindrical portion of the tank. In case of floating-roof tanks, the nominal capacity is defined as the volume from the underside of the roof deck up to the maximum floating position of the roof.

The gross capacity (sometimes referred to as the total capacity) is the volume from bottom of the tank up to its maximum, safe filling height.

The net capacity is the volume of the tank contents between the low-liquid level (LLL) and the high-liquid level (HLL).

Tank Dimensions

In general, tank heights do not exceed one and a half times the tank diameter. In cases where availability of land is not a constraint, it is justifiable to go for larger diameters in preference to height. As the tank height increases, wall thickness plays a more important role. Higher tanks put a greater load on the soil. If the pressure becomes more than soil-allowable bearing pressure, pile-supported foundations become necessary and are expensive. This concern is particularly applicable for poor soils. In general, tanks that are higher than 15m are not commonly used in the industry.

TANK BLANKETING REQUIREMENTS

In many instances, the vapor space of tanks is blanketed with an inert gas. This may be needed when the liquid's vapors are harmful to health or when contact with air could lead to the formation of hazardous compounds or product degradation.

To achieve an inert atmosphere in a tank, a blanketing valve senses the pressure in the vapor space of the tank and controls the flow of inert gas (usually nitrogen) into the vapor space to maintain the tank pressure within the desired limit. Blanketing pressures are typically in the range of 8-10 inches H_2O.

When the liquid is moved out of a tank or if the temperature decreases, a tank can experience vacuum conditions. In this case, the blanketing valve provides vacuum relief to the tank by opening to allow gas flow and then resealing when the pressure has increased sufficiently. Secondary vacuum relief is provided by pressure/vacuum vents. Figure 2 illustrates a typical P&ID for a blanketed tank that contains a hydrocarbon mixture.

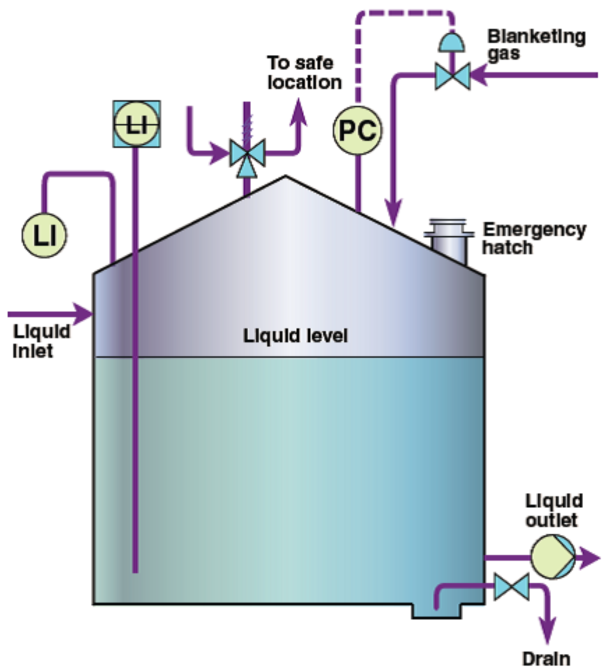

Figure 2: Typical P&ID for Tank Blanketed with Inert Gas

TANK VENTING

In designing, operating or maintaining storage tanks, consideration must be given to the concept of venting to relieve excessive build-up of internal pressures or vacuum.

Several conditions that subject a fixed-roof storage tank to venting include the following:

- Inbreathing due to liquid movement out of tank
- Inbreathing due to contraction or condensation of vapors caused by a decrease in the atmospheric temperature (also called thermal inbreathing)
- Outbreathing due to liquid movement into the tank
- Outbreathing due to expansion or vaporization caused by an increase in the atmospheric temperature (also called thermal outbreathing)
- Outbreathing resulting from external fire

Even if the liquid level does not change, there will be the need to vent a storage tank for both thermal inbreathing and outbreathing. Thermal inbreathing and outbreathing are caused by a change in the temperature of the vapor in the tank. The temperature of the vapor in the tank can be affected by several factors:

- Heat gain by direct radiation of the sun during the day
- Radiation losses during the night
- Convective heat gain by an ambient air temperature that is greater than the tank vapor temperature
- Convective heat loss by an ambient air temperature that is less than the tank vapor temperature
- Convective heat gain or loss due to the temperature difference between the tank vapor and the stored liquid temperature
- Other effects such as the quenching of internal temperature caused by rainfall

For flammable liquids, the various industrial and federal codes in US generally require compliance with API Standard 2000. However, the same principle should be applied to the venting of materials that are non-flammable in order to prevent damage to the tank.

Inbreathing

The venting capacity for maximum liquid movement out of a tank should be equivalent to 5.6 ft^3/hr of air for each 5.6 ft^3/hr of the maximum emptying rate of liquids. This holds for liquids of any flash point. There are also requirements for thermal inbreathing. The API furnishes these requirements as a function of tank capacity in the forms of tables. This information can also be expressed as an equation wherein the thermal venting is expressed as a function of tank capacity. The total venting capacity can be expressed as a sum of liquid movement and thermal inbreathing.

Outbreathing

Whereas venting due to inbreathing is independent of flash point, the requirements for outbreathing differ with flash point. For liquids with flash point above 100^oF or a normal boiling point above 300^oF, the required venting capacity for maximum liquid movement into a tank should be equivalent to 6 ft^3/hr of air for each 5.6 ft^3/hr of maximum filling rate. There are also requirements for thermal outbreathing - for those liquids with high flash point, the thermal outbreathing is roughly 60% of the thermal inbreathing requirement. The reason for this is that the roof and shell temperatures cannot rise and fall as rapidly as they can fall, for example, during a sudden rain shower. Liquids with a lower flash point, below 100^oF, or a normal boiling point below 300^oF, the venting capacity for maximum liquid movement into a tank should be equivalent to 12 standard ft^3/hr of air for each 5.6 ft^3/hr of maximum filling rate.

Emergency Venting on Fire Exposure

When storage tanks are exposed to fire, the venting rate of the vapor may exceed the inbreathing or outbreathing rate that results from a combination of thermal effects and liquid movement. Consideration must be given to vent this vapor, and it is termed emergency venting. The typical means of venting for fire conditions are the use of a frangible roof construction and the use of additional or larger emergency venting valves than required for normal venting. Table 2 provides the required emergency venting rate. It is based upon the area of tank exposed to the fire and a heat input rate that is empirically based.

Venting Floating-Roof Tanks

Since operating floating-roof tanks have no vapor space below the roof, there is normally no need for venting thermal breathing, filling and emptying losses. Venting does occur on initial filling until the roof floats, however. The space between the floating roof and the shell is called the *rim space*, and this volume is relatively small. However, rim space vents are installed to allow this volume to breath.

Table 2: Emergency Venting Requirements

Wetted Area, ft^2	Venting Requirement, ft^3/h free air	Wetted Area, ft^2	Venting Requirement, ft^3/h free air
20	21,100	350	288,000
30	31,600	400	312,000
40	42,100	500	354,000
50	52,700	600	392,000
60	63,200	700	428,000
70	73,700	800	462,000
80	84,200	900	493,000
90	94,800	1000	524,000
100	105,000	1200	557,000
120	126,000	1400	587,000
140	147,000	1600	614,000
160	168,000	1800	639,000
180	190,000	2000	662,000
200	211,000	2400	704,000
250	239,000	2800	742,000
300	265,000	>2800	---

Another problem that can develop with floating roof tanks is the boiling or vaporization under the roof for stocks with highly volatile components. If the roof is large and flexible, it will bulge into a spherical shape, trapping the vapors under the tank. This is not desirable because fatigue of the welds is possible, roof drainage is impaired, and there is always the possibility of damaging the seals. Pan-roofs and single-deck roofs are subject to this type of problem while the more rigid double-deck roof is less likely to distort. The vapor bleeds out from the periphery of the tank and through the seals. The solution to these problems is to select the proper type of tank or roof. In some circumstances, a PV valve can be used to bleed off the gassing that occurs under a floating roof.

The conventional manner of handling filling losses until the roof is able to float is to use landing actuated vent valves. Figure 3 shows the details. These devices operate when the roof leg touches the bottom. Some local regulations restrict the use of open vents on floating–roof tanks that have landed roofs. This rules out the mechanical landing-actuated vent valves discussed above. Instead one solution has been to use pressure vacuum (PV) valves on floating roofs

Following design guidance should be used for venting floating-roof tanks:

1. The provisions of API 2000 for determining venting requirements for fixed-roof tanks should be applied to floating-roof tanks using the landed roof condition. Apply the principles of venting to both flammable and non-flammable liquids.
2. Determine the true vapor pressure of liquid to be stored and tank size to assess whether the local, state or federal regulations limit the design to a particular tank configuration.

3. Store liquids in floating roof tanks only if the true vapor pressure is 11 psia or less. For liquids above this vapor pressure, there are special engineering considerations such as storage in pressure vessels or the use of fixed-roof tanks with vapor recovery systems.
4. Double-deck roofs reduce oiling losses and handle vapor build up under the roof better than the single-deck roofs. However, they usually cost more.

Open Vents

Tanks that store harmless or non-toxic liquids, such as fire water or service water, are vented to the atmosphere. These tanks operate at atmospheric pressure and the venting is called open venting. While being filled, the tank breathes out through the vent. When liquid is pumped out, the tank breathes in through the vent. To prevent rain or snow from entering, the vent pipe is usually provided with a weather hood, or alternatively, the pipe itself is shaped in the form of a goose neck.

As per API 2000, open vents without flame arrestors may be used for venting under the following circumstances:

- For storage of petroleum or petroleum products with a flash point of 100^{o}F or above
- For tanks holding petroleum or petroleum products at a temperature below that of the flash point
- For storage of any product in tanks with a capacity of less than 335 ft^3

Flame arrestors need to be used with open venting of tanks that store petroleum or petroleum products that have a flash point below 100^{o}F.

PV Valves

Pressure vacuum vent valves are usually employed to protect blanketed tanks. In situation where the blanketing valve fails and gets stuck in the open position, the tank can be pressurized by the continuous inflow of inert gas. A pressure vent will open to protect the tank from rupture. Conversely, in situations where a tank is being emptied and the blanketing valve fails, the tank can reach vacuum conditions. A vacuum valve will open, thus protecting the tank from collapse. Pressure vacuum vent valves are also known as *breather valves* or *conservation valves*. See Figure 3.

Figure 3: Typical PV Valve

PV valves are the workhorse of the industry. They have a number of very useful characteristics that have made them standard apparatus on storage tanks:

1. They protect tanks against over- and under-pressure.
2. They reduce evaporation losses compared to open vents.

3. They can double as flame arrestors, and they eliminate the need for flame arrestors in some cases.
4. Because the atmospheric oxygen concentration is apt to be lower in a tank using a PV valve than in a tank with open venting, the internal corrosion in the vapor space will often be reduced.
5. PV valves are generally required by EPA, OSHA, NFPA etc.

API Bulletin 2521 spells out the basic operations as well as design and requirements for PV valves. This bulletin addresses all types of PV valves including:

1. Solid pallet (hard pallet to hard seat)
2. Diaphragm pallet
3. Liquid seal valve

The pressure setting of the vent is kept slightly above the tank blanketing pressure but below the maximum pressure the tank can withstand. Similarly, the vacuum setting is kept higher than the normal operating vacuum, but at a vacuum level that is below the maximum vacuum that the tank can withstand.

Because these vents are designed to remain closed until they must open in order to protect the tanks, another advantage is that evaporation losses and fugitive emissions can be minimized by PV valves. This is achieved by preventing the release of vapors that would otherwise occur during minor variations in temperature, pressure or level.

References:

Understanding Atmospheric Storage Tanks *by* Siddhartha Mukherjee (Lurgi India)

Aboveground Storage Tanks *by Philip Myers*

VACUUM BOX TEST PROCEDURE

Vacuum box test is used to locate leaks in welds due to through thickness discontinuities. It is accomplished by applying a solution to a weld and creating a differential pressure across the weld causing formation of bubbles as leakage gas passes through the solution. This testing is usually performed prior to any main vessel or tank testing following the completion of all welding.

TEST MATERIAL – BUBBLE SOLUTION

It is required for the bubble forming solution to produce a film that does not break away from the area to be tested, and for the bubbles to not break rapidly due to air drying or low surface tension. The number of bubbles contained in the solution should be minimized to reduce the problem of discriminating between existing bubbles and those caused by leakage.

The bubble forming solution is one of the following:

1. For carbon steels, a commercially prepared leak testing may be used, or a solution with a ratio of one part Joy, Ivory liquid soap, or similar soap product to one hundred parts of the following:
    a. Water – for testing welds with surface temperatures between 40^{o}F and 125^{o}F; or
    b. A solution containing approximately 30% alcohol and 70% soapy water will be used for testing welds when the surface temperature is less than 40^{o}F.
2. For stainless and nickel steels and for carbon steels when required by contract, a commercially prepared leak testing solution with low chloride, halide and sulfur content, such as Leak Tec 277NE, Bubble Emission Leak Detector.

Commercial solutions shall be used only within the temperature ranges recommended by the manufacturer.

TEST EQUIPMENT

Figure 1: Vacuum Box Test Equipment

The vacuum box used shall be of convenient size and contain a window in the side opposite the open bottom. The open bottom edge shall be equipped with a suitable gasket to form a seal against the test surface. Suitable connections, valves, lighting and gauge shall be provided. The gauge should preferably have a range of 0-15 psi or a range of 0-30 inches of mercury.

A vacuum pump source shall be used; preferably a variable one which will achieve the required vacuum pressure.

PREPARATION FOR THE TEST

All welding shall be completed before the vacuum box examination is to begin; this shall include, but not be limited to cleaning all welds of slag, spatter dirt, oil, grease, paint or other contaminants that might mask a leak. The welder and/or welding machine operator is responsible for cleaning their own welds prior to vacuum box testing. Testing shall not be performed on wet surfaces because the liquid may tend to dilute the leak detection solution. The temperature of the weld surface to be examined shall be between 40°F and 125°F, unless the film solution is proven to work at temperatures outside these limits, either by testing or manufacturer recommendations. Local heating or cooling is permitted.

TEST PROCEDURE

The bubble forming solution is applied to the surface to be tested by flowing, spraying or brushing the solution over the examination area before placement of the vacuum box. The solution is applied at most one minute prior to testing so that evaporation or freezing of the solution does not occur. A partial vacuum of 3 to 5 psia is used for the test. If specified by the Purchaser, a second partial vacuum test of 8 to 10 psia can be performed for the detection of very small leaks. The required partial vacuum (differential pressure) should be maintained for at least 10 seconds of examination time. The examination time should be the duration of time necessary to visually scan the entire increment under examination during initial pressurization and stabilization, as well as any additional time required for depressurization and repressurization needed to resolve standing bubbles. An overlap of at least 2 inches of previously viewed surface must be used for each subsequent examination to assure complete coverage. A minimum light intensity of 1000 Lux at the point of examination is required during the application of the examination and evaluation of leaks; this can be provided by a LED head lamp at 2' or a 1000 watt halogen light at 6'.

ACCEPTANCE CRITERIA

The presence of continuous bubble formation or growth on the surface being examined indicates leakage through an orifice passage(s) in the area under examination. Any indicated leakage shall be considered unacceptable. Some large leaks may not be detected by bubble formation because the strong stream of air may break the bubble film as soon as it forms. To avoid missing this type of leak, the pressure may be monitored for a variation (decrease).

References:

Brown-Minneapolis Tank

15 WAYS TO INCREASE PRODUCTIVITY AT WORK

There are only so many hours in a day, so making the most of your time is critical. There are two ways increase your output - either put in more hours or work smarter. I don't know about you, but I prefer the latter.

Being more productive at work isn't rocket science, but it does require being more deliberate about how you manage your time. This post will walk you through 15 simple but effective strategies for increasing your productivity at work.

1. **Track and limit how much time you're spending on tasks.**

   You may think you're pretty good at gauging how much time you're spending on various tasks. However, some research suggests only around 17 percent of people are able to accurately estimate the passage of time. A tool like Rescue Time can help by letting you know exactly how much time you spend on daily tasks, including social media, email, word processing, and apps.

2. **Take regular breaks.**

   It sounds counterintuitive, but taking scheduled breaks can actually help improve concentration. Some research has shown that taking short breaks during long tasks helps you to maintain a constant level of performance; while working at a task without breaks leads to a steady decline in performance.

3. **Set self-imposed deadlines.**

   While we usually think of a stress as a bad thing, a manageable level of self-imposed stress can actually be helpful in terms of giving us focus and helping us meet our goals. For open-ended tasks or projects, try giving yourself a deadline, and then stick to it. You may be surprised to discover just how focused and productive you can be when you're watching the clock.

4. **Follow the "two-minute rule."**

   Entrepreneur Steve Olenski recommends implementing the "two-minute rule" to make the most of small windows of time that you have at work. The idea is this: If you see a task or action that you know can be done in two minutes or less, do it immediately. According to Olenski, completing the task right away actually takes less time than having to get back to it later. Implementing this has made him one of the most influential content strategists online.

5. **Just say no to meetings.**

   Meetings are one of the biggest time-sucks around, yet somehow we continue to unquestioningly book them, attend them and, inevitably, complain about them. According to Atlassian, the average office worker spends over 31 hours each month in unproductive meetings. Before booking your next meeting, ask yourself whether you can accomplish the same goals or tasks via email, phone, or Web-based meeting (which may be slightly more productive).

6. **Hold standing meetings.**

   If you absolutely must have a meeting, there's some evidence that standing meetings (they're just what they sound like - everyone stands) can result in increased performance.

7. **Quit multitasking.**

   While we tend to think of the ability to multitask as an important skill for increasing efficiency, the opposite may in fact be true. Psychologists have found attempting to do several tasks at once can result in lost time and productivity. Instead, make a habit of committing to a single task before moving on to your next project.

8. **Take advantage of your commute.**

   This goes for any unexpected "bonus" time you may find on your hands suggests author Miranda Marquit. Instead of Candy-Crushing or Facebooking, use that time to pound out some emails, create your daily to-do list, or do some brainstorming.

9. **Give up on the illusion of perfection.**

   It's common for entrepreneurs to get hung up on attempting to perfect a task--the reality is nothing is ever perfect. Rather than wasting time chasing after this illusion, bang out your task to the best of your ability and move on. It's better to complete the task and move it off your plate; if need be, you can always come back and adjust or improve it later.

10. **Take exercise breaks.**

    Using work time to exercise may actually help improve productivity, according to a study published in the *Journal of Occupational and Environmental Medicine*. If possible, build in set times during the week for taking a walk or going to the gym. Getting your blood pumping could be just what's needed to clear your head and get your focus back.

11. **Be proactive, not reactive.**

    Allowing incoming phone calls and emails to dictate how you spend your day will mean you do a great job of putting out fires--but that may be all you get accomplished. My friend and business partner Peter Daisyme from free hosting company Hostt says, "Set aside time for responding to emails, but don't let them determine what your day is going to look like. Have a plan of attack at the start of each day, and then do your best to stick to it."

12. **Turn off notifications.**

    No one can be expected to resist the allure of an email, voicemail, or text notification. During work hours, turn off your notifications, and instead build in time to check email and messages. This is all part of being proactive rather than reactive (see number 11).

13. **Work in 90-minute intervals.**

    Researchers at Florida State University have found elite performers (athletes, chess players, musicians, etc.) who work in intervals of no more than 90 minutes are more productive than those who work 90 minutes-plus. They also found that top performing subjects tend to work no more than 4.5 hours per day. Sounds good to me!

14. **Give yourself something nice to look at.**

    It may sound unlikely, but some research shows outfitting an office with aesthetically pleasing elements--like plants--can increase productivity by up to 15 percent. Jazz up your office space with pictures, candles, flowers, or anything else that puts a smile on your face.

15. **Minimize interruptions (to the best of your ability).**

    Having a colleague pop her head into your office to chat may seem innocuous, but even brief interruptions appear to produce a change in work pattern and a corresponding drop in productivity. Minimizing interruptions may mean setting office hours, keeping your door closed, or working from home for time-sensitive projects.

If you feel the need to increase your productivity at work, resist the temptation put in longer hours or pack more into your already-full calendar. Instead, take a step back, and think about ways you can work *smarter*, not harder. Here are some bonus tips to increase your productivity:

- Continuing on the subject of meetings, if a meeting is absolutely required then, in addition to 5) and 6) above, make sure that you a) set an agenda for the meeting, b) put a limit on each topic, and c) invite only those people that absolutely need to be there.
- Learn to say “no”. People often say “yes” when they should say “no”. Remember this – whenever you say “yes” to a request, you say “no” to something else (worth repeating the opening sentence of this article: there are only so many hours in a day). Of course, there might be some people you can’t say “no” to, such as a boss. If that is the case, practice the “yes, but” method. For example, you can say. “Yes, I’d be happy to do [requested item], but that will put me behind on [another important item]. Would you prefer that I do [requested item] first, or would it be better for me to focus on [the other important item] instead?” Keep in mind that saying “no” isn’t rude, and there are many ways to say “no” without using the word “no”. For instance, you can say something like, “Thanks so much for considering me for this fantastic opportunity, but I don’t have the bandwidth to do it justice right now.”
- Eliminate inefficiencies. There are many tasks you simply don’t need to do, and it may be possible to minimize others. For instance, a housewife may feel the need to vacuum every day, even if doing so once or twice a week is sufficient. In other cases, a task needs to be done but not necessarily by you. Using vacuuming as an example, the mom can perhaps delegate that task to one of her older children. If you are a business owner, or in management position, focus on tasks that can only be done by you and delegate the rest to others. Many tasks can be automated. For instance, you can set up email filters to automatically delete certain types of emails.
- But certainly not the least; take care of yourself. Getting enough sleep and making exercise part of your routine are just two of the things you need to do every day to be at your best and most productive. I bet you know the rest already. Eat a healthy diet. Drink lots of water. Get rid of your bad habits, whether they be smoking or hanging around toxic people. And be nice to yourself as well as to other people. Take time for yourself and do whatever (healthy) thing recharges and refreshes you. The healthier you are, the more productive you’ll be – and the more productive you are with your work, the more time you’ll have to spend however you like.

References:

15 Ways to Increase Productivity at Work *by* John Rampton

How to Increase Workplace Productivity *by* Susan Ward

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BUILDING A BETTER TOMMORROW

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