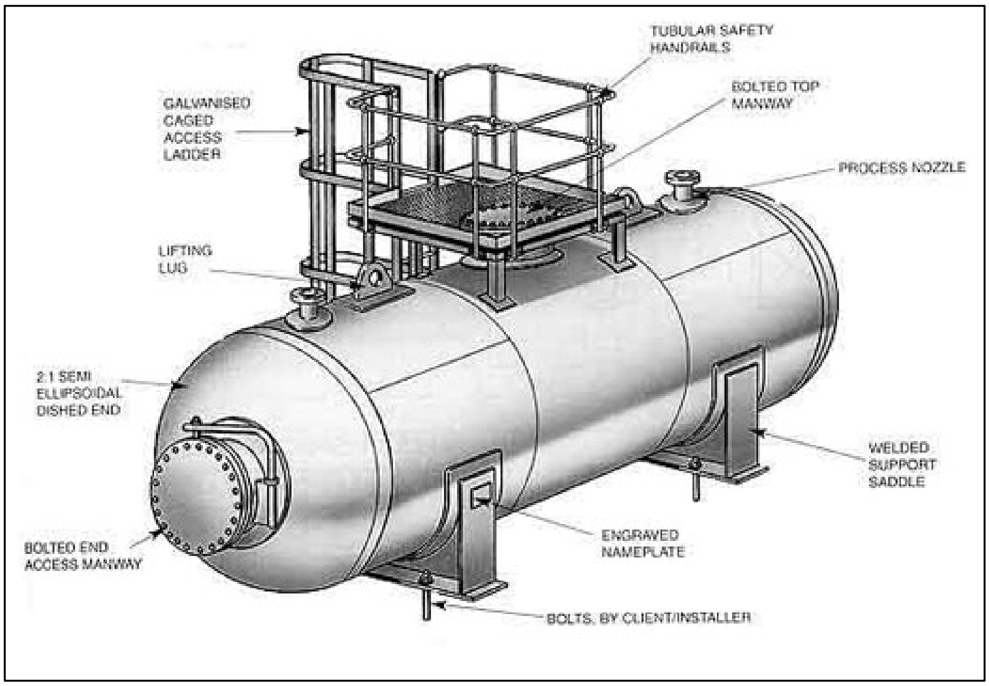
Pressure Vessel Newsletter

Volume 2014, February Issue



IN ASSOCIATION WITH

INTERGRAPH®

Serving the Pressure Vessel Community Since 2007

PV ELITE®

Pressure Vessel Analysis

PV ELITE 2014 JUST RELEASED

Benefit from a complete solution for vessel and heat exchanger design, analysis and evaluation. Users of PV Elite have designed equipment for the most extreme uses and have done so quickly, accurately and profitably.

www.intergraph.com/go/pvelite

PART OF HEXAGON

INTERGRAPH®

In this issue…

For any queries regarding the newsletter, please write to rtiwari123@gmail.com or call at +91 98109 33550

Photograph on the cover is courtesy of freedigitalphoto.net

From The Editor's Desk:

Welcome to the February 2014 issue of the newsletter. I hope you liked the new format presented in the January issue; this is an interim step towards establishing this newsletter as the mainstream magazine of choice for professionals in the pressure vessel and heat exchanger industry. To serve this objective, I would like to get feedbacks from you on, among other things, the quality of the articles and the overall presentation of the newsletter.

We will continue with the initiatives we started in the January issue of the newsletter.

The first of these initiatives is tailored for Micro, Small and Medium Enterprises (MSME) who would like to showcase themselves to our readers via the newsletter. Our definition of MSME is those companies that are engaged in pressure vessel related activities and have employee strength of 50 or less. If your organization qualifies, then you may send a small write up (no pictures please) of no more than half a page. For every issue of the newsletter, we will randomly select one submission and display the write up in the newsletter.

The second initiative is to create a directory of companies and individual consultants that are engaged in pressure vessel related activities. Such directories are not so easily available in all parts of the world. Elsewhere in this issue, we have listed the information to be provided for inclusion in the pressure vessel directory (*This information was to be provided in the January issue, but was overlooked*). If you would like your company or yourself to be listed in the directory, please supply the requested information. This exercise will continue until the end of the year.

The third initiative is to display the images supplied by the readers on the cover page of the newsletter. We only require that the images be original and be pertinent to the pressure vessels and heat exchanger industry. If your images are displayed, it will be accompanied by a proper credit on the inside pages.

Through these initiatives, we strive to facilitate greater interaction with the readers, and in the process improve the experience as you go through the pages of the newsletter.

LOCAL STRESSES IN PRESSURE VESSELS DUE TO INTERNAL PRESSURE AND NOZZLE LOADINGS

Raymond Chao

The subject of local stresses in the vicinity of nozzles in pressure vessels has been investigated for more than forty years. Indeed, the nozzle-to-shell intersection has been one of the most researched areas of pressure vessels. As a result of this effort, several practical approaches to this problem have evolved which enable the vessel designers to check the adequacy of nozzle designs in pressure vessels. However, very little direction has been given on the calculation of local stresses due to combined internal pressure and external nozzle loadings. This article will address this problem and provide guidance to the vessel designers in the correct application of the available simplified calculation methods for local stresses in pressure vessels.

One of the most widely used methods has been that detailed in the Welding Research Council (WRC) Bulletin 107 published in 1965. In 1989, WRC Bulletin 297 was published as a supplement to WRC Bulletin 107.Together they provide a simplified approach to the calculations of local stresses due to the combined internal pressure and external nozzle loadings.

Local stresses, however, also occur in the vicinity of nozzles due to internal pressure. Therefore, a complete evaluation would require that these stresses be accounted for, in addition to those due to external loadings. The calculations of the total local stresses due to the combined internal pressure and external loadings have not, in general, been done correctly. It appears that the following two approaches have often been taken but neither of them will give correct answers.

1. The nozzle is reinforced in accordance with the ASME Code, Section VIII, Division 1 based on the internal design pressure. This has been taken as being sufficient to nullify the effect of nozzle opening and only the general membrane stresses in the vessel due to the internal pressure are calculated and superimposed to those due to external loadings.
2. The internal pressure in the nozzle is converted into a radial outward thrust force on the nozzle and this is combined with the other nozzle loadings which are then used in the calculations of local stresses in the vicinity of nozzle using the WRC Bulletins 107 and 297.

The first approach ignores the local stresses due to internal pressure which will result in an underestimate of the total local stresses due to the combined internal pressure and nozzle loadings. The second approach, on the other hand, will result in an overestimate of the local stresses as has been shown by an FEM analysis of a large nozzle-to-cylinder shell junction by Doug Stelling in a Carmagen report in 1996. It should be noted that WRC Bulletins 107 and 297 are intended for external nozzle loadings only and should not be used for pressure thrust loads on nozzles.

In 1991, WRC Bulletin 368 was published to fulfill the need for the determination of local stresses at nozzles due to internal pressure. The design formulas presented in this Bulletin were based on the results of a parametric study performed using the computer program FAST2. Using these formulas, the maximum membrane and surface stresses in the vessel shell and nozzle at the nozzle-to-shell intersection may be computed. The resulting stresses due to internal stresses may then be combined with those due to external nozzle loadings by superposition.

A final evaluation of acceptability of the design requires that the computed stresses be compared to an allowable stress basis. Also a calculated value of stress means little until it is associated with its location and distribution in the structure and with the type of loading which produces it. ASME Code, Section VIII, Division 1 provides no guidance on the evaluation of local stresses in pressure vessels. Therefore, local stresses due to nozzle loadings are often calculated and evaluated using the guidelines given in Division 2, as it provides detailed guidance on the classification of stresses and also provides associated stress limits.

Source: *Raymond Chao is a mechanical engineer at Carmagen Engineering, and is a member of PVRC Committee on Elevated Temperature Design.*

TRAINING ANNOUNCEMENT

DESIGN & FABRICATION OF PRESSURE VESSELS: ASME SECTION VIII, DIVISION 1

Pressure vessels, along with tanks, are the workhorses for storage and processing applications in the chemical, petroleum, petrochemical, power, pharmaceutical, food and paper industries. ASME BPV, Section VIII, Div. 1 Code is used as a standard for the design and fabrication of pressure vessels by most companies across the world.

We would like to announce training course for "Design and Fabrication of Pressure Vessels: ASME Section VIII, Div. 1" on April 7-9, 2014 at Chennai, and on April 10-12 at Coimbatore. This course provides the information that will help you understand the ASME requirements for the design and fabrication of pressure vessels. The course material follows the contents of 2010 edition of the code, and is replete with worked examples covering important aspects of pressure vessel construction. This hands-on learning will allow you to master in 3 days what would otherwise take up to a year or more of on-job training.

The contents of the training course will be as follows:

- Introduction to Boiler and Pressure Vessel Code
- Materials of Construction
- Low Temperature Operation
- Joint Efficiencies
- Design of Components
- Openings and Reinforcements
- Fabrication, Inspection and Tests
- Markings and Reports
- Tall Towers and Pressure Vessel Supports
- Nozzle Loads
- Fatigue Analysis
- Introduction to ASME Section VIII, Division 2

The instructor, Ramesh Tiwari, is internationally recognized specialist in the area of pressure vessels, heat exchangers, materials, and codes and standards. He holds Bachelor's and Master's degrees in mechanical engineering from universities in India and United States. He is also a registered Professional Engineer in the State of Maryland in the United States. Mr. Tiwari is a member of ASME Boiler & Pressure Vessel, Section VIII Subgroup on Heat Transfer Equipment, and a member of ASME International Working Group on B31.1 for Power Piping in India. In this capacity, he has made invaluable contribution in resolving technical issues in compliance with the ASME codes for Code users. Mr. Tiwari has over 24 years of design engineering experience on a variety of projects spanning industries such as oil & gas, power, nuclear, chemical, petrochemical, pharmaceutical, food etc. He has provided engineering advice and code interpretations to senior management and guidance to several companies he has worked for in the US, India and Germany. He has initiated and implemented numerous innovative ideas to improve working process and quality, and developed and conducted training programs for peers as well as clients. Mr. Tiwari is an approved pressure vessel instructor at NTPC, a premier thermal power generating company in India and at several other companies, both public and private.

Registration fee for the training course is Rs. 25,300 for professionals and Rs 16,000 for students (inclusive of service tax). Early bird discount of 15% is available until March 7, 2014. Additional discount of 15% is also available for group registration of 3 or more participants. Registration fee includes training, handbook on design and fabrication of pressure vessels, copy of the presentation, certificate from CoDesign Engineering, and beverages and lunch on all days. It excludes travel to and from New Delhi, accommodation, and meals and beverages other than those provided during the course. We invite you to make nominations.

In case of any queries, including the registration process, please email at learning@codesignengg.com, or call at +91 98109 33550.

NEW STANDARDS FOR QUALITY CONTROL AND ASSURANCE OF BOLTED FLANGES

Neil Ferguson

If necessity is the mother of invention, then the American Society of Mechanical Engineers (ASME), the Occupational Safety and Health Administration (OSHA), and the Environmental Protection Agency (EPA) are the parents of guidelines, standards, and regulations that help keep industrial operations safe for humans and the environment.

While the great majority of global business managers are already seriously committed to conducting safe practices to avoid mishaps and reduce risk in their construction and operational activities, the adoption of standards and guidelines can assist mangers when hiring and training their personnel, staying within regulatory compliance, and documenting safety procedures.

To support such endeavors, ASME periodically publishes new series of guidelines and standards—some of which are specifically for bolted flange and joint assemblies.

Historically, the management of bolted flanges has been regulated by relatively limited guidelines. Yet, recent events have illustrated the need for new standards. In fact, in 2013, the Environmental Protection Agency reported that 32% of all volatile carbon emissions came from bolted flanges.

For example, several years ago an offshore platform in the North Sea experienced a large and costly fire. The investigation showed that a carbon steel ring had been inserted between two stainless steel flanges through a weak positive material verification program and passed the helium pressure test. Three years later the carbon steel ring eroded away from galvanic corrosion and leaked.

Eventually, the process release was enough to cause a fire that spread across the platform and upon investigation, inspectors determined that the incident occurred, in part, due to a lack of standards and procedures for bolted joints, although a plethora of standards exist to govern welded joints. In fact, according to ASME, practically no requirements for bolted joints existed, compared to welded joints, even though bolted joints hold back the same process conditions and therefore pose a similar risk.

Treat the bolt as if it were a weld

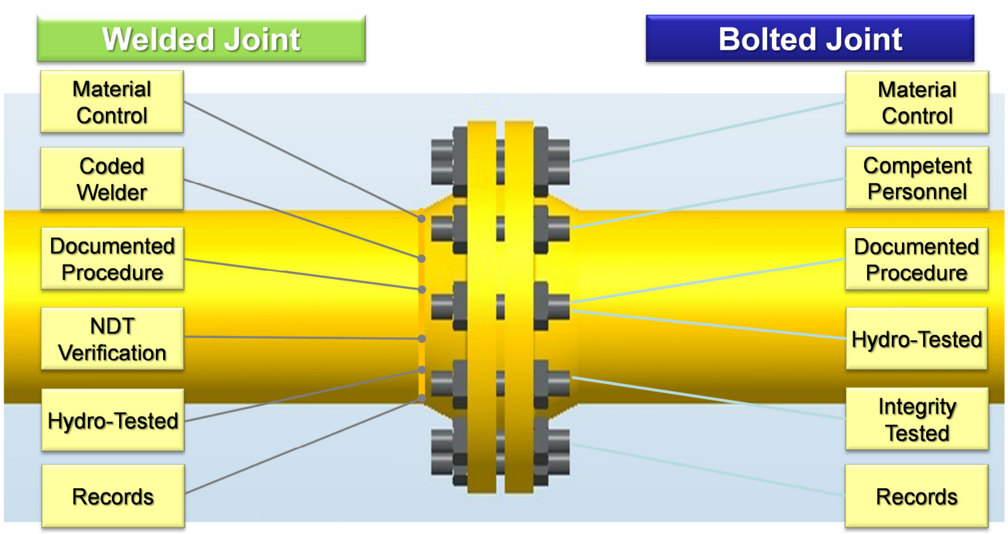


Figure 1: Comparison of Welded Joint and Bolted Joint Safety Standards

As a result, industry managers are encouraged to treat bolted joints with the same standard of detail and safety as a welded joint, (see Figure 1) and evaluate the training and competency of bolting technicians much the same as coded welders.

Recently, as a result of similar events, ASME and Comité Européen de Normalisation (CEN), a European committee for standardization, each published major updated standards concerned with defining the requirements for evaluating competent bolting personnel.

ASME's publication is an update to the agency's 2010 PCC-1 Guidelines for Pressure Boundary Bolted Flange Joint assembly, which includes an appendix defining the requirements for training and qualification of bolted joint personnel.

CEN's publication is an update of its EN1591 Part 4, with modifications of its standards, now entitled "Flanges and Their Joints Part 4: Qualification of Personnel Competency in the Assembly of the Bolted Connection of Critical Service Pressurized Systems."

The ASME guidelines' underpinning philosophy is: "To understand the importance of bolted joint assembler training, it is worthwhile to compare a bolted joint with the current practices for other pressure boundary joints: the welded joint."

The publication of new standards for flange assembly and management with more stringent requirements is an attempt to help engineers understand the importance of bolted joint assembler training, among other issues. The new guidelines include grades and types of experience for technician competency to allow technicians to achieve qualifications. These new guidelines will impact the industry markets as well~~s~~ as operating contractors.

Specifically, ASME categorizes assembly personnel according to their experience and training. Personnel with six months constant work experience ~~but~~ with ~~no~~ formal training with a qualified organization are categorized as qualified bolting specialists. A senior qualified bolting specialist must prove at least two or more years of field experience. A qualified senior bolting instructor must have four years of experience. And a subject matter expert must have an engineering degree and at least four years of experience.

With these standards, ASME has effectively offered the industry an opportunity to assemble bolted joints to ensure that bolted joints meet the same standards as welded joints. The standards contain advice and best practices on virtually every aspect of flange assembly and management, such as recommendations for three levels of joint assembly recordkeeping, including short, medium, and long term. The type of recordkeeping depends upon a number of factors such as process criticality, history, and referral criteria.

Also, similar to the standards for welded joints, personnel will have to prove their competency on a regular basis (every three years) and document their activities using pre-approved procedures and traceable bolt loads. Such personnel must maintain permanent records for future reference. When rigorously followed, the industry can expect significant paybacks via reduced leaks, improved safety performance, and new construction and turnarounds completed on time and within budgets.

Industry executives and managers have two options to take advantage of the new standards to improve their operations. One option is to develop and conduct training in-house. Such program development would also include record-keeping, testing, and certification programs to ensure compliance with the new standards.

A second option is to outsource bolt flange assembly and inspection to third-party providers who specialize in the past, present, and future operations. As a core competency, such third-party providers often include processes and strategies to meet or exceed future standards and guidance from ASME and other agencies and associations, and can do so in a more cost-effective manner.

With either option, industry operators should keep in mind that, many times, the guidance and standards issued from associations such as ASME can become government regulatory codes at a later date, so being prepared in advance for compliance is often the most efficient and economical strategy to ensure safe and successful operations.

Source: *Neil Ferguson is a Joint Integrity Leader: Hydratight North and South America at Actuant Corporation.*

TAKING ON ASME SECTION VIII, DIV. 1, PRESSURE VESSEL EFFICIENCY

This article by staff member Robert D. Schueler Jr. was originally published in the summer 2006 National Board BULLETIN. *It has been edited for space. Some code requirements may have changed because of advances in material technology and/or actual experience. The reader is cautioned to refer to the latest edition and addenda of the* National Board Inspection Code *and* ASME Boiler and Pressure Vessel Code *for current requirements.*

Presented here is a list of questions and corresponding answers addressing common inquiries about the 2004 Edition with 2005 Addenda of ASME Code Section VIII, Div. 1. While the answers are meant to be helpful, they are merely the author's explanation of the more complex rules found in the code book itself.

Q. ***Where do the requirements for pressure part efficiency begin?***

A. Look at the formulas given for each pressure part where the term "E" denotes efficiency. The nomenclature will refer to the rules in UW-12 for joint efficiency. Paragraph UW-12 includes subparagraphs (a) through (f), which refer to UW-11(a) and UW-11(a)(5). For the condition applicable to no radiographic examination, the path from the formula to UW-12 and then to UW-12(c) is correct. Unfortunately for the other plans, this does not direct the user to the true starting point, which can be found in UG-116(e). Paragraph UG-116(e)(1) through (4) provides the definitions of each of the radiographic plans and sends the user along the proper path.

Q. ***What is the difference between an RT-1 and an RT-2 vessel?***

A. The definitions for the RT-1 and RT-2 are provided in paragraph UG-116(e) and, by reference, UW-11(a). Paragraph UW-11(a) defines both plans as full radiography. The RT-1 plan requires all butt-welded joints be fully radiographed over their entire length using the criteria in paragraph UW-51. The RT-2 plan requires all category A and D butt-welded joints be radiographed over their entire length using the criteria in paragraph UW-51. All category B and C butt-welded joints must be spot radiographed per UW-11(a)(5)(b) using the criteria in paragraph UW-52. Depending on the welded joint type employed for welded components, the efficiency will normally be established by a category A or D butt-welded joint (UG-27 footnote 15). A vessel complying with either plan will be 100 percent efficient for both components having type 1 welded joints (Table UW-12 column [a]) and seamless head or shell sections (UW-12[d]).

Q. ***Can RT-2 be used to satisfy the radiographic requirements of special service lethal construction or must an RT-1 plan be used?***

A. RT-1 must be applied. This is a function of the rule provided in paragraph UW-2(a), which requires compliance with paragraph UW-11(a)(4). Paragraph UW-11(a)(4) ties in the rules in UW-11(a)(1) and UW-11(a)(3) which sets the condition RT-1 as defined in paragraph UG-116(e)(1). Paragraph UW-11(a)(5) was not part of this set of requirements and is therefore not applicable to special service lethal constructions.

Q. ***The vessel has a number of longitudinal and circumferential welded joints along with a category D butt-welded joint, all affecting a single cylinder shell section of the vessel. With each of these joints having its own welded joint efficiency, how do you determine what value of "E" is to be used in the formula in UG-27?***

A. The definition of the term "E" in UG-27(b) refers to UW-12 for welded joint efficiency. Based on the requirements for each joint, making contact with the cylindrical shell being considered, a list of all such welded joints and their corresponding joint efficiencies must be compiled. The joint efficiency must be expressed in terms of **equivalent longitudinal efficiency** (see UG-27 footnote 15) for each joint to permit the selection of the controlling item (most severe case).

Example:

The vessel is to be stamped RT-4. The cylinder has a type 1, fully radiographed longitudinal joint in accordance with UW-51. A nozzle conforming to Figure UW-16.1 sketch (f-4) is installed in the cylinder using a type 1 joint which is spot examined per UW-11(b). Seamless 2:1 ellipsoidal heads are attached at both ends and are type 1 butt-welded joints, spot examined per UW-11(a)(5)(b) (also see UW-52[b][4] for limitations). There are no ligament conditions on the cylinder.

Expressed in terms of **equivalent longitudinal efficiency**:
Ligament efficiency
– UG-53 not applicable to this example
Longitudinal cylinder joint
– Table UW-12 column (a) = 1.0
Circumferential joints
– Table UW-12 column (c) = 0.70 x 2 = 1.4
Nozzle joint
– Table UW-12 column (b) = 0.85

Based on this, the lowest value of "E" used in the equation will be 0.85 resulting from the nozzle joint.

Q. ***Given a seamless head or shell section, other than a hemispherical head (see UG-32), what is the design efficiency of the seamless section?***

A. Paragraph UW-12(d) answers this question with a question, as follows: Was the weld(s) joining the seamless head or seamless shell spot examined per the rules given in UW-11(a)(5)(b)? If yes, the seamless head or shell efficiency is set at 100 percent. If no, the seamless head or shell efficiency will be set at 85 percent.

Q. ***When following an RT-3 plan per UG-116(e)(3), can seamless head or shell sections have an efficiency of 100 percent?***

A. No, RT-3 complies with the rule in UW-11(b). The requirement that would permit a higher efficiency is found in paragraph UW-11(a) and is not applicable to a UW-11(b) spot radiographic plan. Therefore, the rule in UW-12(d) will set the efficiency at 85 percent. Note: UW-11(a)(5)(b) cannot be applied with RT-3 (see UW-52[b][4]).

Q. ***If the answer to the previous question is no, what would be required to permit a higher efficiency for seamless head and shell sections?***

A. It will be necessary to select an RT-1, RT-2, or RT-4 plan in which the requirements of UW-11(a)(5)(b) will be satisfied.

Q. ***Can a nonradiographed vessel have aligned vessel longitudinal joints between courses?***

A. No, with a nonradiographed construction, the rule in UW-9(d) takes on a different meaning and must be read as mandating the joints be staggered a distance greater than five times the plate thickness.

Q. ***How can one determine the applicable RT number from the data listed on the Manufacturer's Data Report?***

A. Based on the information provided, with the exceptions of an RT-4 and nonradiographed vessel, the RT level cannot be determined from the data report. Only a limited amount of weld joint efficiency and degree of radiographic examination information is required on the report. The actual RT number only appears on the vessel stamping (see UG-116[e]).

ASME IMPACT TEST REQUIREMENTS

This article provides information about impact test requirements in pressure vessel design and construction. The ASME Code exempts the pressure vessel material from impact testing when certain requirements are satisfied. These requirements are assessed in four steps that are carried out in a sequence. If the pressure vessel material is found to be exempt from impact testing at any of the interim steps, there is no need to assess the exemption criteria in the next step. However, if the vessel material does not meet the exemption criteria at any of the four steps, then impact testing will have to be carried out. Impact testing is very expensive, so manufacturers try to exempt the pressure vessel from this costly test.

Basic Concept

Carbon steels and low alloy steels exhibit a drastic change in their room temperature ductility at sub-zero service temperatures. There is a sudden, phenomenal drop in their notch-toughness properties below the "transition" range of temperature. Above transition temperature range, impact specimens fracture in a "ductile" manner, absorbing relatively large amounts of energy. At lower temperatures, i.e. below the transition temperature range, the impact test specimens are found to fracture in a brittle (cleavage) manner, absorbing less energy. And within the transition temperature range, the fracture is a mixture of ductile and brittle nature.

A material would be invulnerable to a sudden drop in notch-toughness at the lowest specified service (or design) temperature, if it is proved by conducting Charpy V-notch Impact tests on representative test samples, at reference (the lowest service) temperature.

Grain refined carbon steel forgings and wrought materials (thoroughly worked and normalized) generally exhibit good notch toughness.

ASME Code Section VIII Div 1 Exemption Rules for ASME Impact Test Requirement

One needs to follow the following clauses to make exemption assessment for ASME impact test requirement:

UG-20(f) →→→ UCS-66(a) →→→ UCS-66(b) →→→ UCS-68(c)

UG-20(f) has five clauses. If all five clauses are satisfied, then the pressure vessel material is exempted from impact testing, and there is no need to proceed any further.

If the pressure vessel material is not exempted by UG-20(f), then one needs to proceed to paragraph UCS-66(a). If this paragraph exempts the material, there is no need for more assessment.

The next criterion to asses is paragraph UCS-66(b). If this paragraph exempts the pressure vessel material from impact testing, there is no need for further assessment; otherwise, one needs to proceed to paragraph UCS-68(c), and again if still not exempted, then impact testing of the material needs to be carried out.

Paragraph UG-20(f)

If the pressure vessel material satisfies all the five criteria listed below, then that material is exempted from impact testing.

1. If the material is categorized as P-No. 1 or 2, and thickness does not exceed that given in (a) or (b) below.
    a. ½ in for materials listed in Curve A of Figure UCS-66
    b. 1 in for materials listed in Curve B of Figure UCS-66

All Carbon and Low Alloy Steels permitted in the construction of ASME Section VIII, Division 1 vessels are distributed in four groups (Curves) of materials.

Materials listed in curve D have the best toughness property, better than the materials listed in curve C. Similarly materials listed in curve C have better toughness properties compared to materials listed in curve B and materials listed in Curve B have better toughness than materials listed in Curve A.

2. Completed vessel is hydrostatically tested after completion.

3. Design temperature is between 650°F and -20°F.
4. Thermal or mechanical shock loadings are not controlling design requirements.
5. Cyclic loading is not a controlling design requirement.

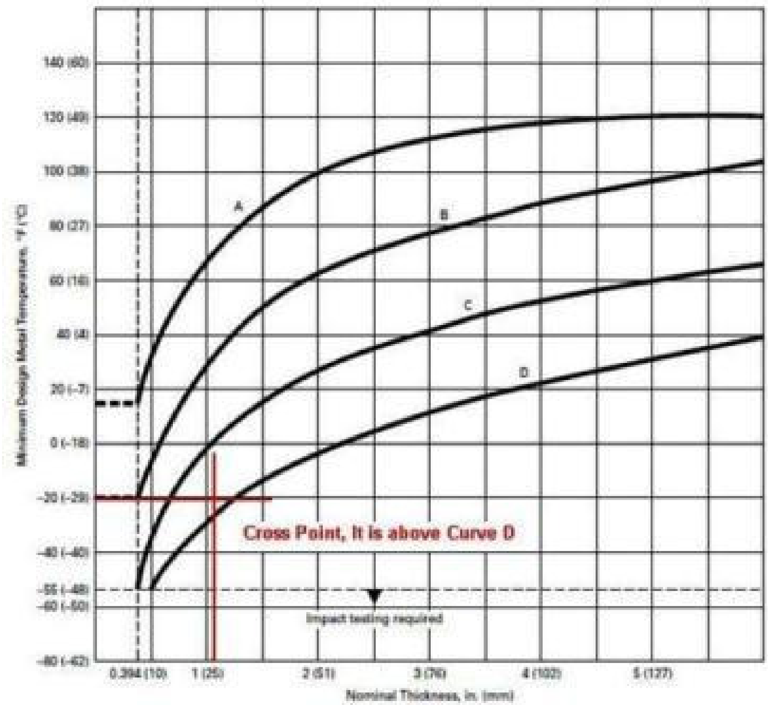


Figure 1: Impact Test Exemption Curves

For example, if the material is a normalized SA 516 Gr.70 with 0.75 inch thickness, then it will be exempted from ASME impact test requirement. Thickness, in this example, is 0.75 inch, and is listed in curve D which is permitted by UG-20(f) up to 1 inch. If other requirements of UG-20(f) are satisfied, then this material will be exempted from impact testing.

Paragraph UCS-66(a)

Let us assume that the material thickness is 1.125 inch instead of 0.75 inch. Now, the material will not be exempted from impact testing by UG-20(f) and assessment per the next step needs to be made, i.e., paragraph UCS-66(a). For this assessment, we need to know the pressure vessel MDMT (Minimum Design Metal Temperature). Assume that the MDMT is -20°F.

In the Figure UCS-66, locate 1.125 inch on the horizontal axis and draw a vertical line. In a similar manner, locate -20°F in the vertical axis and draw a horizontal line. These two lines will cross each other. In Figure 1 above, the lines are identified in red.

If the cross point falls above the curve D (because the material is listed in curve D), this material is exempted from impact testing. If not, then the assessment per the next step needs to be made. For the current example, we are above the curve D and therefore the material is exempted from impact testing. The lowest temperature that the material can be exposed to without the need for impact testing can be obtained from drawing a horizontal line from the intersection of the vertical line with the curve D. For our example, the lowest temperature will be -26°F.

Paragraph UCS-66(b)

Let us explain this clause with the above example. The material is normalized SA 516 Gr.70 listed in curve D, new MDMT is -27°F, and the nominal thickness is 1.125 inch,. Now the material is not exempted by UCS-66(a) because of the MDMT. So assessment of the material according to paragraph UCS-66(b) needs to be done. For this assessment, we need to calculate the following ratio:

Ratio= t_r.E / (t_n –c)

t_r is the required design thickness of the material for all applicable loading. We assume for our example, t_r is 0.95 inch. E is the joint efficiency, and we assume for this vessel it is 1. C is corrosion allowance, and we assume it is 0.125 inches; so let calculate:

Ratio = 0.95x1/(1.125 – 0.125)

Ratio= 0.95

See the following figure.

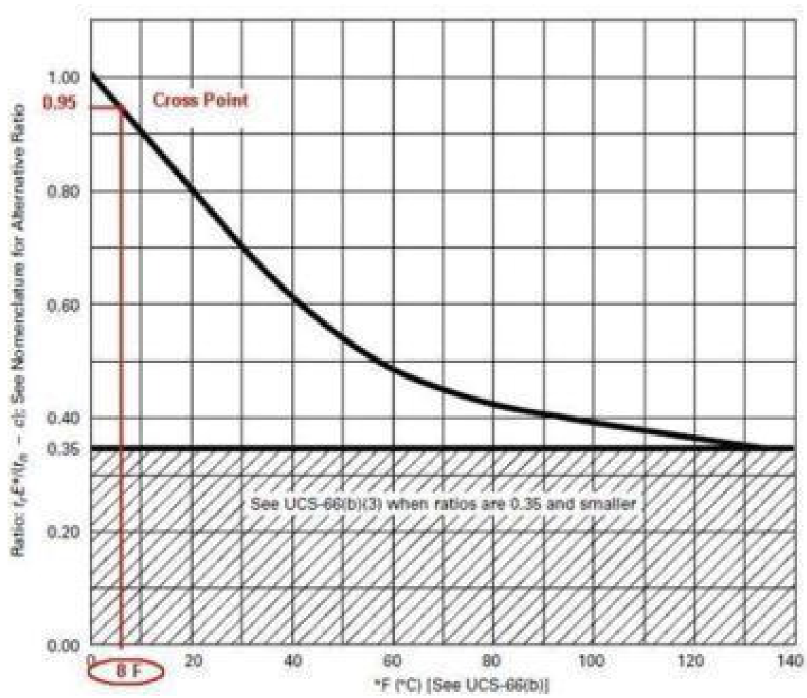


Figure 2: Reduction in MDMT without Impact Testing

We locate Ratio on the vertical axis and draw a horizontal line. Then we locate the cross point with the graph and draw a vertical line to cross the horizontal axis. The vertical line crosses the horizontal axis at 8°F. Which means that the MDMT can be further reduced by 8°F without the need for impact testing.

In our example, the MDMT is -27°F, and the lowest permissible temperature without impact testing designated -26°F. So with this clause, we can reduce the permissible lowest temperature to -34°F (-26 -8 = -34). The MDMT is -27°F, so the material is exempted from impact testing with this clause.

Paragraph UCS-68(c)

Let us change one variable in the above example. Let's assume that we need to have -45°F for MDMT. Other variable are the same; it means that normalized SA 516 Gr.70 listed in curve D with thickness of 1.125 inch will not be exempted from impact testing by UCS-66(b). This is because the minimum permissible temperature is -34°F, but the MDMT is now -45°F. Paragraph UCS-68(c) might be helpful.

It says that if post weld heat treatment is not a code requirement and the P-No is 1 and we carry out post weld heat treatment, then a 30°F bonus will be granted to reduce the minimum permissible temperature in table UCS-66.

So when is post weld heat treatment a code requirement?

It is code requirement when the service is lethal and when thickness for P-No. 1 is greater than 1.5 inch. In our example, the service is not lethal, material P-No. is 1, and the thickness is 1.125. Therefore, post weld heat treatment is not code requirement.

In this scenario, if we now carry out post heat treatment, a $30^{\circ}F$ bonus will be granted by this clause. For this example, our lowest permissible temperature would now be $-64^{\circ}F$, and the MDMT is $-45^{\circ}F$, so this material is exempted from impact testing.

Now let us look at a worse case: in the above example, assume that we need to have $-70^{\circ}F$ for MDMT; it can be seen that with this new condition, we cannot exempt the material even by UCS-68(c), and therefore impact testing will have to be carried out.

Source: *Inspection 4 Industry LLC*

NEWS AND EVENTS

Intergraph® Acquires GT STRUDL® from the Georgia Tech Research Corporation

Intergraph® Process, Power & Marine, part of Hexagon and the world's leading provider of enterprise engineering software to the process, power and marine industries, announced today the acquisition of GT STRUDL®, a leading computer-aided structural engineering (CAE) software system, from the Georgia Tech Research Corporation of Atlanta, Ga. As part of the acquisition, the 10 skilled staff and management team members of the Computer-Aided Structural Engineering Center (CASE Center) have joined Intergraph. GT STRUDL is widely used in a variety of industries such as nuclear power and nuclear defense industries, conventional power generation, general plant structures, offshore structures, marine applications, general civil engineering and infrastructure structures.

In the United States nuclear industry, GT STRUDL is widely used by major companies in the design, maintenance and upgrading of safety-critical structures such as turbine buildings, boiler buildings, equipment support structures, pipe support systems and other related civil engineering structures. The acquisition of GT STRUDL will strengthen Intergraph's existing suite of engineering analysis solutions for the power, process and offshore industries.

Developed by the CASE Center within the School of Civil and Environmental Engineering at Georgia Institute of Technology, GT STRUDL uniquely integrates graphical modeling, frame and finite element linear and nonlinear static and dynamic analysis, structural frame design, graphical analysis and design result display and structural database management all into a powerful, menu-driven information processing system.

To view this press release, please visit the Intergraph press room at:
www.intergraph.com/assets/pressreleases/2014/02-10-2014

18th Annual IPEIA Conference

Banff Center, Banff, Alberta (CANADA), February 19-21, 2014
International Pressure Equipment Integrity Association

Offshore Technology Conference 2014

Reliance Center, May 5 – 8, 2014
Houston, Texas

2014 ASME Pressure Vessel and Piping Conference

Hyatt Regency Orange County, July 20 -24, 2014
Annaheim, California

,

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 for pressure vessels, and can offer a range of services related to them.

Training & Development

Consulting Services

CoDesign Engineering

Pressure Vessels ● Heat Exchangers ● Piping Systems ● Welding

Oil & Gas ● Power ● Chemical ● Petrochemical ● Fertilizer ●Solar ● Biogas