STP-PT-007

COMPARISON OF PRESSURE VESSEL CODES ASME SECTION VIII AND EN13445

Technical, Commercial, and Usage Comparison

Design Fatigue Life Comparison

ASME STANDARDS TECHNOLOGY, LLC

Date of Issuance: December 12, 2006

This report was prepared as an account of work sponsored by ASME Pressure Technology Codes and Standards and the ASME Standards Technology, LLC (ASME ST-LLC).

Neither ASME, ASME ST-LLC, the authors, nor others involved in the preparation or review of this report, nor any of their respective employees, members, or persons acting on their behalf, makes any warranty, express or implied, or assumes any legal liability or responsibility for the accuracy, completeness, or usefulness of any information, apparatus, product, or process disclosed, or represents that its use would not infringe upon privately owned rights.

This report was prepared for ASME ST-LLC by independent consultants and is based in part on information not within the control of either ASME ST-LLC or the consultants. Neither ASME ST-LLC nor the consultants have made an analysis, verified, or rendered an independent judgment of the validity of the information provided by others. While it is believed that the information contained herein, will be reliable under the conditions and subject to the limitations set forth herein, neither ASME ST-LLC nor the consultants guarantee the accuracy thereof. Use of this report or any information contained therein shall constitute a release and agreement to defend and indemnify ASME ST-LLC and such consultants from and against any liability (including but not limited to liability for special, indirect or consequential damages) in connection with such use. Such release from and indemnification against liability shall apply in contract, tort (including negligence of such party, whether active, passive, joint or concurrent), strict liability, or other theory of legal liability; provided, however such release, limitation and indemnity provisions shall be effective to, and only to, the maximum extent, scope or amount allowable by law.

Reference herein to any specific commercial product, process, or service by trade name, trademark, manufacturer, or otherwise does not necessarily constitute or imply its endorsement, recommendation, or favoring by ASME ST-LLC or others involved in the preparation or review of this report, or any agency thereof. The views and opinions of the authors, contributors, reviewers of the report expressed herein do not necessarily reflect those of ASME ST-LLC or others involved in the preparation or review of this report, or any agency thereof.

ASME ST-LLC does not take any position with respect to the validity of any patent rights asserted in connection with any items mentioned in this document, and does not undertake to insure anyone utilizing a publication against liability for infringement of any applicable Letters Patent, nor assumes any such liability. Users of a publication are expressly advised that determination of the validity of any such patent rights, and the risk of infringement of such rights, is entirely their own responsibility.

Participation by federal agency representative(s) or person(s) affiliated with industry is not to be interpreted as government or industry endorsement of this publication.

ASME is the registered trademark of The American Society of Mechanical Engineers.

ASME Standards Technology, LLC
Three Park Avenue, New York, NY 10016-5990

ISBN No. 0-7918-3093-4

TABLE OF CONTENTS

FOREWORD

This report presents two papers presented during the 2006 ASME Pressure Vessels and Piping Division Conference held July 23-27, 2006, in Vancouver, BC, Canada. The papers have also been published by ASME along with the Proceedings of PVP2006-ICPVT-11. The papers resulted from projects sponsored by ASME in response to the "Comparative Study on Pressure Equipment Standards", published in June 2004 by the European Commission, Enterprise Directorate-General.

The American Society of Mechanical Engineers (ASME) is a not-for-profit professional organization promoting the art, science and practice of mechanical and multidisciplinary engineering and allied sciences. ASME develops codes and standards that enhance public safety, and provides lifelong learning and technical exchange opportunities benefiting the engineering and technology community. Visit www.asme.org.

The ASME Standards Technology, LLC (ASME ST-LLC) is a not-for-profit Limited Liability Company, with ASME as the sole member, formed in 2004 to carry out work related to newly commercialized technology, expanding upon the former role of ASME's Codes and Standards Technology Institute (CSTI). The ASME ST-LLC mission includes meeting the needs of industry and government by providing new standards-related products and services, which advance the application of emerging and newly commercialized science and technology and providing the research and technology development needed to establish and maintain the technical relevance of codes and standards. Visit www.stllc.asme.org for more information.

ABSTRACT

Part I of this report includes paper PVP2006-ICPVT11-94010, "Comparison of Pressure Vessel Codes ASME Section VIII and EN13445." This paper consists of a comparative study of the primary technical, commercial, and usage differences between the American Society of Mechanical Engineers (ASME) Boiler and Pressure Vessel Code Section VIII and the European Pressure Vessel Code EN13445 (EN). This study includes a review of "Comparative Study on Pressure Equipment Standards" published by the European Commission (EC) and provides technical comparisons between the code design requirements, material properties, fabrication, and contributing effects on overall cost. This study is intended to provide a broad viewpoint on the major differences and factors to consider when choosing the most appropriate vessel design code to use.

Part II of this report includes paper PVP2006-ICPVT11-93059, "Design Fatigue Life Comparison of ASME Section VIII and EN 13445 Vessels with Welded Joints." The "Comparative Study on Pressure Equipment Standards" performed by the EC included a comparison of design fatigue life of welded vessels allowed by the ASME Boiler and Pressure Vessel Code (B&PVC) Section VIII with that of the European Standard EN 13445. The allowable number cycles of the ASME Code was reported to be much larger than that of EN 13445, and, therefore, the ASME Code was regarded as unconservative for welded regions. This paper investigates the reason for the reported discrepancy between the two design codes, identifies errors in the EC calculation, recalculates the allowable cycles according to ASME Code rules and concludes that they are comparable with those of EN 13445.

INTENTIONALLY LEFT BLANK

PART I - PVP2006-ICPVT11-94010: Comparison on Pressure Vessel Codes ASME Section VIII and EN13445

Prepared by:

Leslie P. Antalffy
Senior Mechanical Engineering Director
Fluor, Corp.
Houston, Texas

J.J. Hajovsky
Principal Design Engineer
Fluor, Corp.
Houston, Texas

George A. Miller III
Principal Design Engineer
Fluor, Corp.
Houston, Texas

Barry J. Millet
Principal Design Engineer
Fluor, Corp.
Houston, Texas

Jeffrey A. Pfeifer
Principal Design Engineer
Fluor, Corp.
Houston, Texas

George T. West
Design Supervisor
Fluor, Corp.
Houston, Texas

ABSTRACT

This paper consists of a comparative study of the primary technical, commercial, and usage differences between the American Society of Mechanical Engineers (ASME) Boiler and Pressure Vessel Code Section VIII and the European Pressure Vessel Code EN13445 (EN). This study includes a review of "Comparative Study on Pressure Equipment Standards" (hereby referenced by the "EC Study") [see REF-1] and provides technical comparisons between the code design requirements, material properties, fabrication, and contributing effects on overall cost. This study is intended to provide a broad viewpoint on the major differences and factors to consider when choosing the most appropriate vessel design code to use.

1 INTRODUCTION

This paper covers four main topics, evaluation of the original EC Study, code parameter comparisons, cost structure breakdown and post EC survey.

Review of the EC Study - This section addresses the information covered in the EC Study and determines what factors contribute to the EC conclusions.

Code Parameter Comparisons - This section will evaluate the differences between the ASME and EN pressure vessel codes which includes a comparison of design requirements, material properties, and fabrication requirements.

Cost Structure Breakdown - This section considers the variables used in determining the total cost of the -vessel.

Survey Analysis - This section lists the results of a survey that was taken specifically for gathering general information from owners/users, material suppliers and fabricators around the world.

2 REVIEW OF "COMPARATIVE STUDY EN 13445 AND ASME SECTION VIII, DIV. 1 AND 2"

The EC Study provides a start at examining the economic differences between the two codes but is limited first by the scope of vessel manufacturers, second by the range of vessels used in the study and third by the statistical method used for normalizing the cost data.

Italian, French, German, and Austrian manufacturers provided cost estimates on the vessels for the study. These manufacturers represent, by gross vessel weight, a small percentage of the total vessels produce in the global market. The majority of pressure vessels manufactured in the world come from Japan, Korea and the U.S.A.

At the time of the EC Study, a small number of pressure vessels have been manufactured in accordance with the EC Code. A comprehensive knowledge base of this code, in comparison to existing codes, does not presently exist with many fabricators.

The size and quantity distribution of vessels used in the EC Study is generally not representative of typical chemical, petrochemical or petroleum process facilities. The vessels in the EC Study were not representative of the total pressure vessel market in distribution of size, thickness and quantity. For example, on a typical project for a process plant, the greater part of the total cost of pressure vessels is attributed to only a relatively small number of the higher end pressure vessels. These high end pressure vessel (reactors, large towers, etc.) costs dominate the overall global pressure vessel market. For example the "Hydrogen Reactor" used in the study had a low (below 454°C) design temperature and did not include a more stringent service specification such as API RP 934. Also ASME Code Case 2514 allows the use of ASME B&PV Code Section VIII Div. 3 which reduces wall thickness and cost by up to 15 percent over present Division 2 requirements.

The use of "relative averages" in the EC Study provided ambiguous cost results. The vessel cost for each example should have been normalized across all manufacturers providing a complete pricing picture for each example and not a pricing picture per manufacturer. Pressure vessels are always purchased based on the lowest cost between manufacturers not a "relative average."

3 CODE PARAMETER COMPARISONS

3.1 Material Properties

As can be seen in the detailed cost evaluations that follow, one of the greatest cost factors associated with vessel fabrication is material. If all other cost factors are considered equal, a thinner vessel requiring less material will be less expensive than a thicker vessel requiring more material. Of course, the primary driver in determining the minimum thickness of a vessel is the allowable stress used for design. Thus, the allowable stress/design margin philosophy employed by each of these Codes has a significant impact on a vessel's ultimate cost.

An evaluation of the allowable stress bases used by the ASME and EN Codes reveals some similarities as well as some significant differences. For purposes of discussion, these evaluations look at material allowables below the creep range and are focused on the two types of materials that are commonly used in vessel construction – carbon and low alloy ferritic steels, and austenitic stainless steels.

3.2 Carbon and Low Alloy Ferritic Steels

For ferritic steels, each of the Codes establishes allowable stresses which consider both the minimum yield strength and ultimate tensile strength of a material. Table 1 illustrates the specific allowable stress bases for each Code.

Table 1 – Allowable Stress Basis for Ferritic Steels

Design Code	Allowable Stress
ASME Section VIII Division 1	Lesser of $\frac{F_{u_T}}{3.5}$ and $\frac{F_{y_{0.2T}}}{1.5}$
ASME Section VIII Division 2	Lesser of $\frac{F_{u_T}}{3.0}$ and $\frac{F_{y_{0.2T}}}{1.5}$
EN 13445	Lesser of $\frac{F_{u_{68}}}{2.4}$ and $\frac{F_{y_{0.2T}}}{1.5}$

F_{u_T} = Ultimate Tensile Strength at Design Temperature

$F_{y_{0.2T}}$ = 0.2% Offset Yield Strength at Design Temperature

$F_{u_{68}}$ = Ultimate Tensile Strength at 68°F (20°C)

As can be seen from Table 1, each of the Codes utilizes a 1.5 design margin on material yield strength. However, the design margin on ultimate tensile strength becomes progressively smaller for each of the Codes – 3.5 for Division 1, 3.0 for Division 2, and 2.4 for EN 13445. This difference is generally the key factor in explaining why the highest allowable stresses are found in the EN Code, and the lowest are found in ASME Section VIII Division 1.

Another important point in this comparison is that the EN 13445 Code only considers the ultimate tensile strength of the material at ambient temperature (68°F, or 20°C), whereas the criteria used for allowable stresses in both of the ASME Codes consider the tensile strength at temperature. This difference can be significant for materials which have yield strengths that remain relatively high at elevated temperatures, such as Cr-Mo steels. When combined with the higher design margins on tensile strength in the ASME Codes, this approach can create a situation where an allowable in the EN Code is based on yield strength, yet the corresponding allowable in both of the ASME Codes at

the same temperature is based on tensile strength. In such cases, the EN Code's yield based allowables significantly exceed the ASME's tensile based values. This is indicated in Figures 1 through 6.

3.3 Austenitic Stainless Steels

When determining allowable stresses, EN 13445 separates austenitic stainless steels into two groups: 1) materials with a tensile elongation property between 30% and 35%; and 2) materials with a tensile elongation greater than 35%. To simplify this brief comparison, only the second group of materials will be discussed, which generally encompasses the 300 series group of stainless steel materials.

For austenitic steels that have a minimum tensile elongation property greater than 35%, each of the Codes establishes allowable stresses considering both the minimum yield strength and ultimate tensile strength of a material. However, the relationship between these two properties that is used to establish allowable stresses differs significantly from the EN Code to ASME. Table 2 below illustrates the specific allowable stress bases for each Code.

Table 2 – Allowable Stress Basis for Austenitic Steels

Design Code	Allowable Stress
ASME Section VIII Division 1	Case 1: Lesser of $\frac{F_{u_T}}{3.5}$ and $\frac{F_{y_{0.2T}}}{1.5}$ Case 2: Lesser of $\frac{F_{u_T}}{3.5}$ and $0.90 \times F_{y_{0.2T}}$
ASME Section VIII Division 2	Case 1: Lesser of $\frac{F_{u_T}}{3.0}$ and $\frac{F_{y_{0.2T}}}{1.5}$ Case 2: Lesser of $\frac{F_{u_T}}{3.0}$ and $0.90 \times F_{y_{0.2T}}$
EN 13445	$\max\left\{\frac{F_{y_{1.0T}}}{1.5}, \min\left[\frac{F_{y_{1.0T}}}{1.2}, \frac{F_{u_T}}{3.0}\right]\right\}$

F_{u_T} = Ultimate Tensile Strength at Design Temperature

$F_{y_{0.2T}}$ = 0.2% Offset Yield Strength at Design Temperature

$F_{yu_{1.0T}}$ = 1.0% Offset Yield Strength at Design Temperature

As noted in Appendices 1 and 2 of ASME Section II Part D, it is recommended that the higher stresses shown by Case 2 be used only where slightly higher deformation is not in itself objectionable, and are not recommended for the design of flanges or other strain sensitive applications.

There are two significant factors in the EN 13445 Code that produce higher allowable stresses. First, the yield strengths used for establishing the austenitic steel material properties are based on a 1.0% strain offset. SA-370 of ASME Section II Part A requires the yield strength testing of materials to be based on a 0.2% offset. A review of the material yield strength properties published in EN 10028-7:2000 for stainless steels indicates that the 1.0% yield strength is anywhere from 30% to 40% higher than the 0.2% yield strength. This higher material yield strength basis leads directly to higher allowables.

The second significant factor that contributes to the higher allowable stresses in the EN Code is the comparison basis between the yield and tensile that is used for establishing the allowables. Table 2 shows that the EN 13445 allowable stresses are a function of the greater of two values, whereas in the ASME Code the allowable is always based on the lesser of two values. (See Table 2) When combined with the fact that the value of the material yield strength used for these comparisons is always greater under the EN Code philosophy, the EN 13445 allowable stresses for austenitic materials will typically be higher than those specified by ASME. The exception to this general observation would be for applications where slightly higher deformations are not detrimental to the equipment design (see Case 2 criteria in Table 2). Figure 5 illustrates this case, where the ASME allowable stress is based on a value that does not exceed 90% of the minimum specified yield strength of the material.

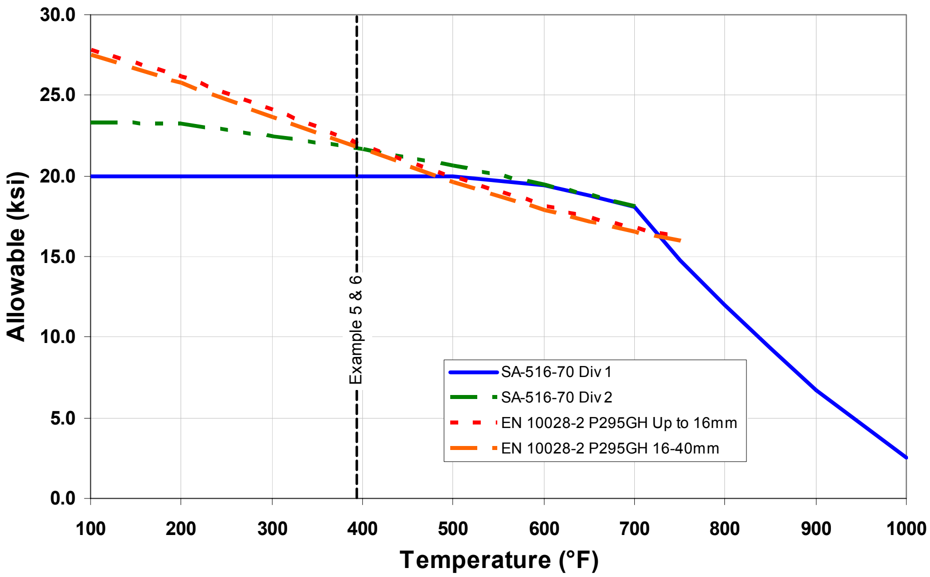


Figure 1 – Allowable vs Temperature for Carbon Steel

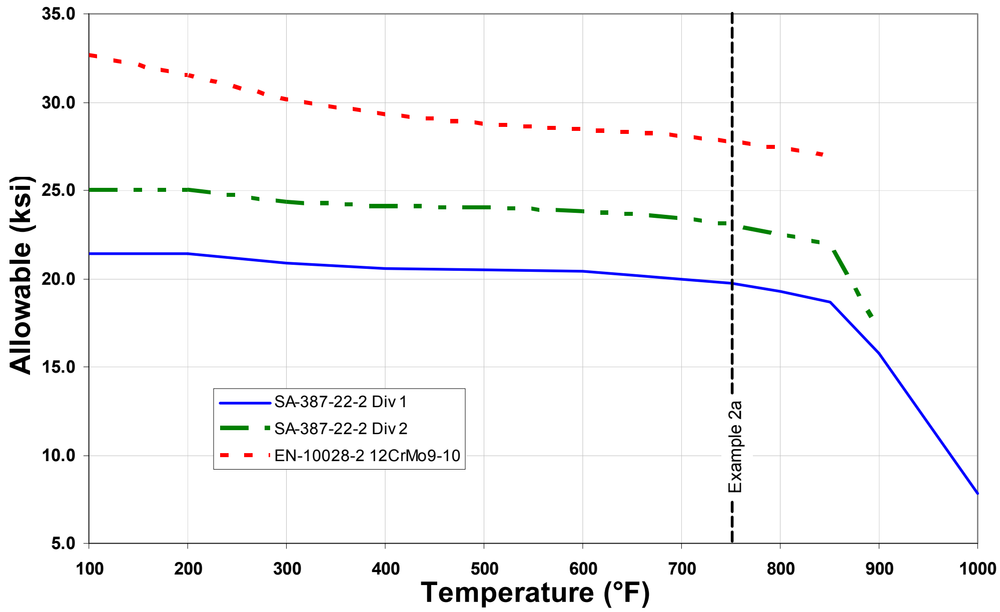


Figure 2 – Allowable vs Temperature for 2 ¼ Cr - 1 Mo Plate

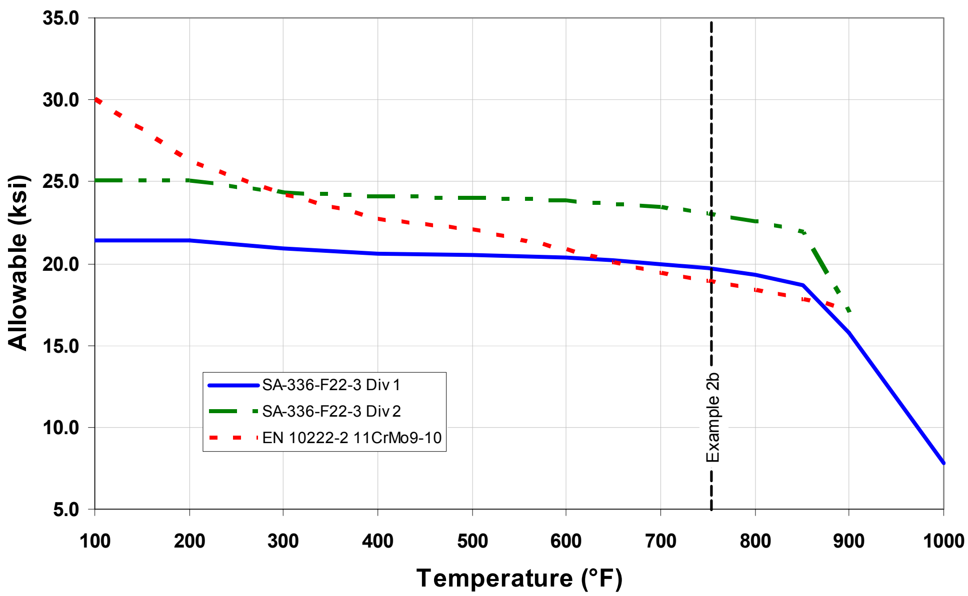


Figure 3 – Allowable vs Temperature for 2 ¼ Cr - 1 Mo Forging

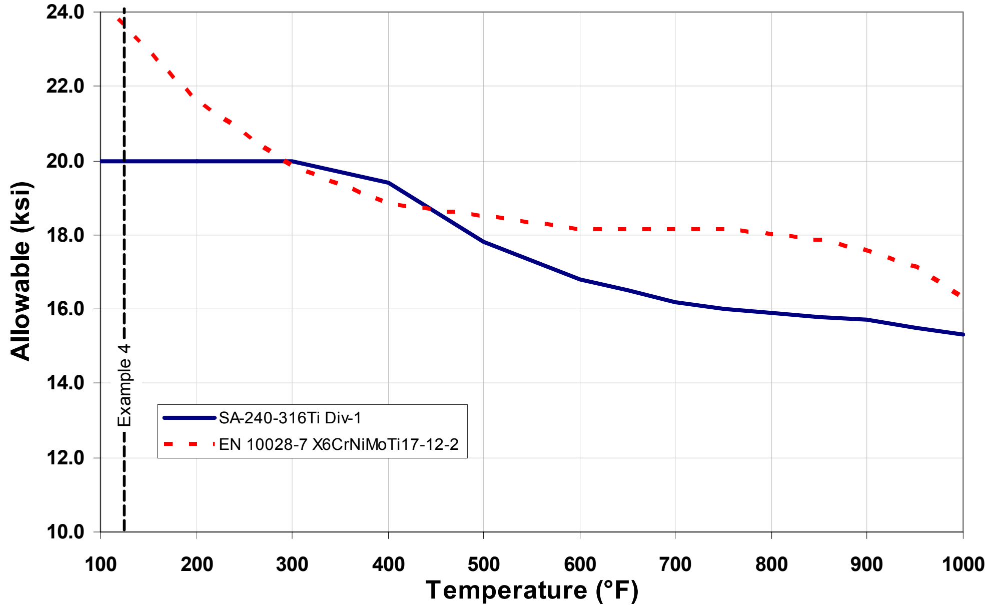


Figure 4 – Allowable vs. Temperature for 16Cr – 12 Ni – 2Mo – Ti

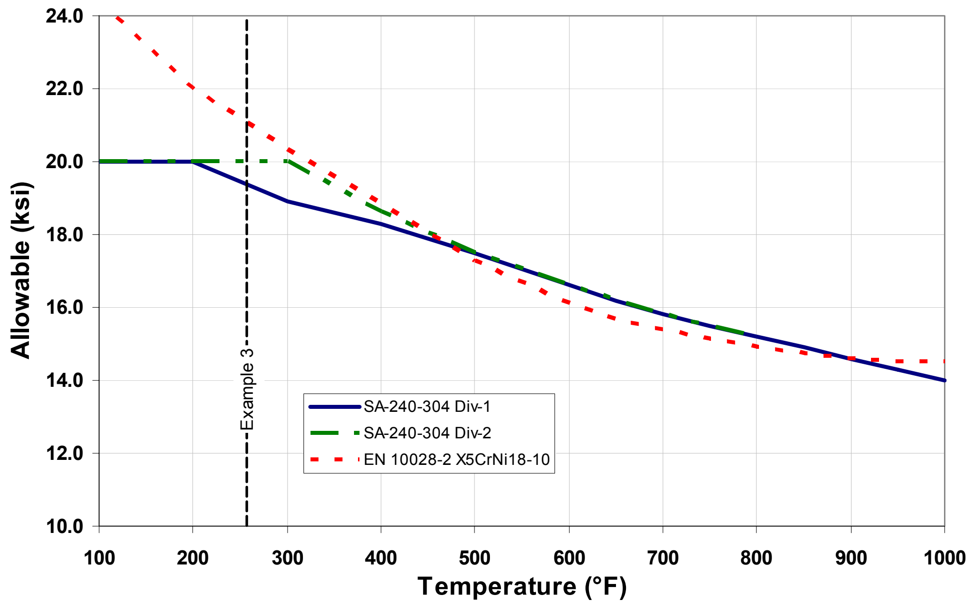


Figure 5 – Allowable vs Temperature for 18 Cr – 8 Ni

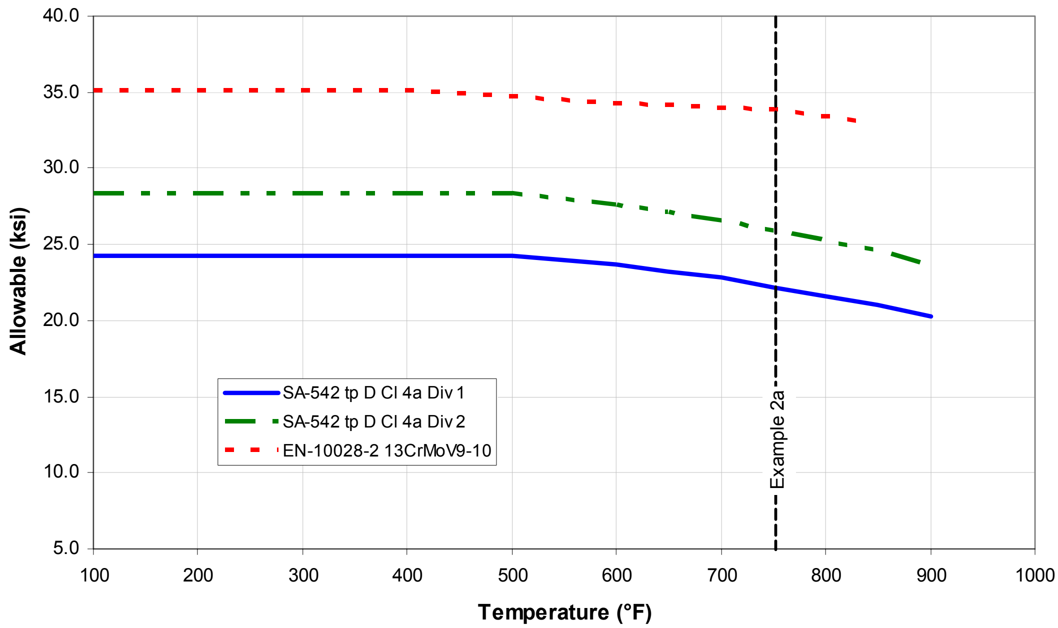


Figure 6 – Allowable vs Temperature for 13Cr – Mo – V

3.4 Design Rules

The ASME Section VIII (ASME) and the EN 13445 (EN) Codes have similar requirements for design and the rule based equations seem to be identical for both codes. The EN is a combined Code including design by rules and design by analysis. The advantage of this approach is greater consistency of design requirements when a combination of rules and design by analysis are employed. The design by analysis rules provides two options in the EN Code. The first is stress categorization including linearization similar to the present methods provided in Section VIII Division 2. However, the EN Code provides for a second option which is called the "Direct Route' which provides a method more attuned to finite element results.

It seems that the primary difference between the design by rules method of the ASME and EN Codes are the additional requirements and limitations associated with the non destructive weld inspection. The EN Code sorts equipment based on "Test Groups" which define the required NDE and other limitations. The ASME joint efficiency is only tied to the radiographic requirements. A review of EN 13445, Table 6.6.1-1 shows the test groups and requirements of the EN Code. The various Testing Groups in Table 6.6.1.1 of EN 13445 assume a significant degree of sophistication and expertise on the part of the design engineer. Table 6.6.1.1 permits joint efficiencies of 1.0, 0.85, and 0.70 for certain materials, with the related NDE requirements in Table 6.2.1.1. It is not immediately clear which joint efficiency / NDE / material combinations actually result in more cost effective designs. The ASME Codes are more straight forward. ASME Section VIII, Division 1 permits joint efficiencies of 1.0, 0.85, and 0.7. Designs with lesser joint efficiencies require less examination, but result in thicker vessels. ASME Section VIII, Division 2 only permits joint efficiency of 1.0 and requires 100% NDE of welds.

As an example, for those vessels designed using a weld joint coefficient (joint efficiency in ASME) of 0.7, the EN Code limits the materials, the thickness and the design temperature. No such limits are placed on an ASME vessel using this joint efficiency. Also, those vessels designed with in Test Group 4 are not intended to be in cyclic service for the EN code. The direct impact on the cost is not known, but the additional limitations and requirements for the EN Code could impact cost especially

(in the authors' experience) in the food and pharmaceutical industries which design vessels using the low joint efficiency and are typically in cyclic service.

Currently, the EN Code does not provide rules for operating temperatures in the creep range, but the rules are in development. Division 1 of the ASME Code however, provides allowable stresses at elevated temperatures well above the onset of the creep range.

3.5 Heat Treatments

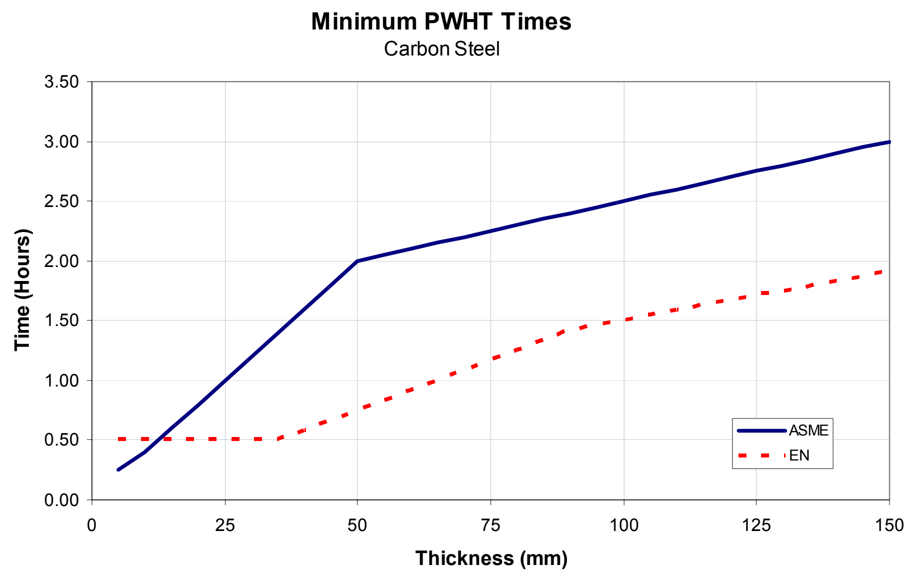


Figure 7 – Minimum PWHT Times For Carbon Steel

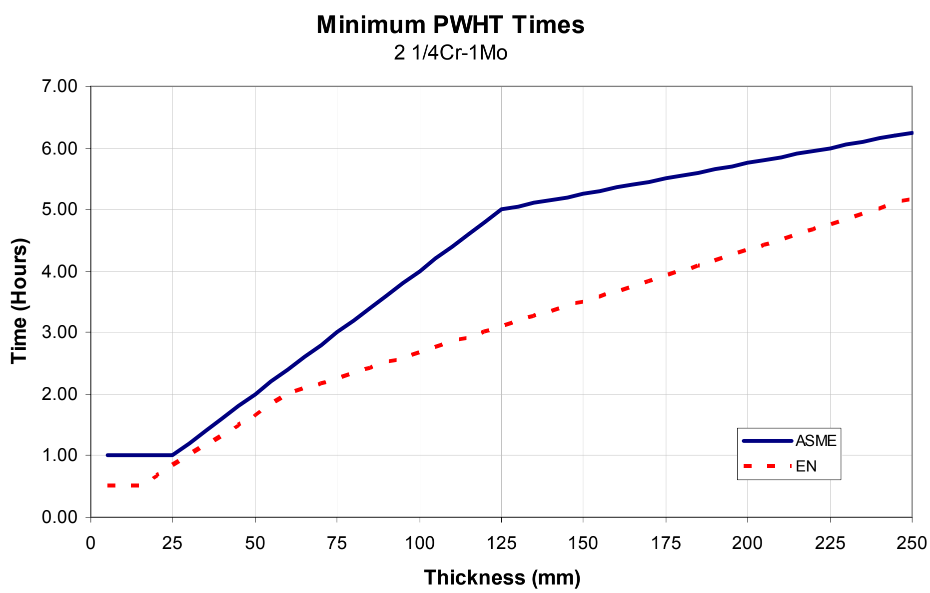


Figure 8 – Minimum PWHT Times for 2 1/4Cr-1Mo

3.6 NDE/Inspection Requirements

A very significant number of existing vessels have been constructed in compliance with the ASME codes. As such, most post construction inspection and maintenance programs are based on the NBIC NB-23 and API 510 codes. These are post construction codes. NB-23 is the basis for the National Board's commissioning of Authorized Inspectors who are involved in new construction and post construction issues. API 510 is the basis for the American Petroleum Institute's certification of Inspectors who are involved in post construction issues. For both codes there is a requirement for a through knowledge of the ASME Code, Addenda and Code Cases. The duties of Inspectors are listed in the ASME Codes (e.g., AG-303 and Part AI of Division 2).

By introducing vessels which are constructed in compliance with the EN code into existing plants, owners will incur costs relating to Inspector Training and Certification. In addition, there will be added costs due to the complexity of maintenance which will in many cases require separate weld and NDE procedures in order to comply with different construction codes.

4 COST STRUCTURE BREAKDOWN

One of the major factors to consider when choosing the most appropriate pressure vessel code to use is the overall cost impact associated with that decision. Of course, one of the major contributors of that impact will be the initial vessel purchase costs.

The cost breakdown structure indicated in Figures 9 through 15 provides the typical percentage breakdown of the major cost factors associated with new pressure vessel purchases sorted by common vessel groups. In Figures 9 – 15, shop costs includes welding, testing, NDE, handing equipment, etc., and material costs include plate and forgings, forming, heat treatment, etc.

In the following charts, various items are depicted as percentages of the overall selling price. These are defined as follows:

- **Material cost** is the cost of the all material for fabrication
- **Labor cost** is the cost for the craft labor and non craft labor.
- **Shop overhead cost** is the cost for the equipment (welding machines, rollers, grinding equipment, the crane, fork-lift. etc.), the consumables, the utilities, the inspectors, NDE, and the shop foremen
- **Office overhead cost** is the cost to run the office accounts payable, the designers, the engineers, the administration, and the equipment used by these people.
- **The Profit** is an average percentage of industry profit margins for that particular industry.

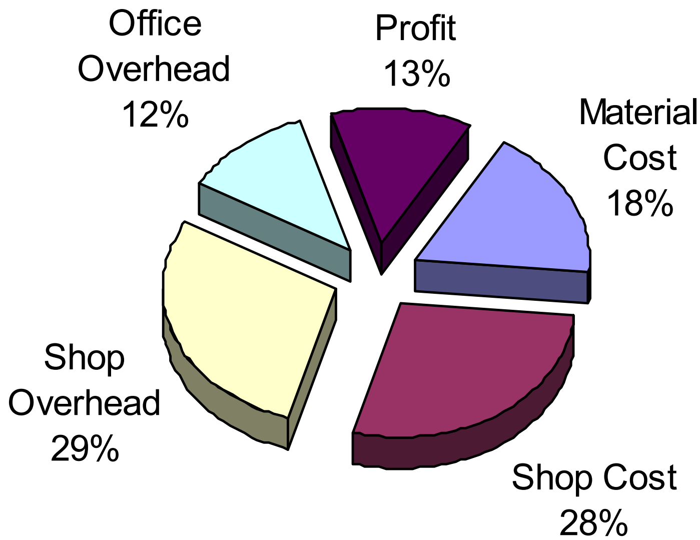


Figure 9 – Overall Cost Drivers for a Medium Carbon Steel Vertical Drum

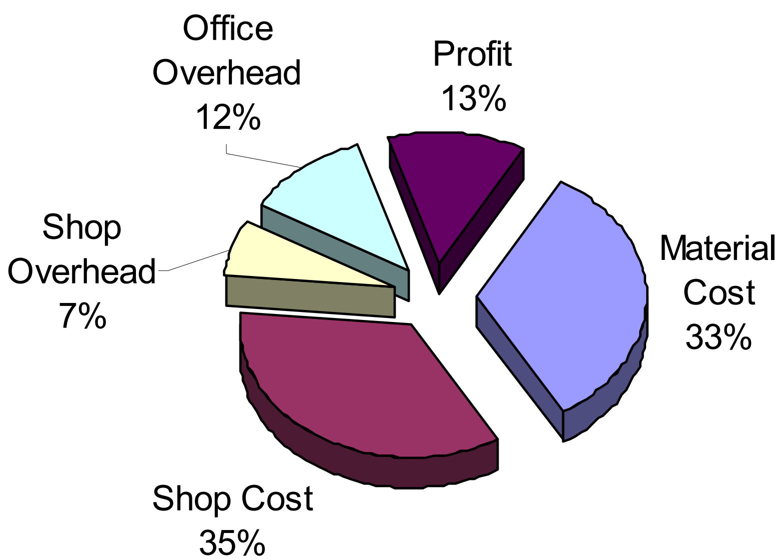


Figure 10 – Overall Cost Drivers for a Large Carbon Steel Vertical Drum

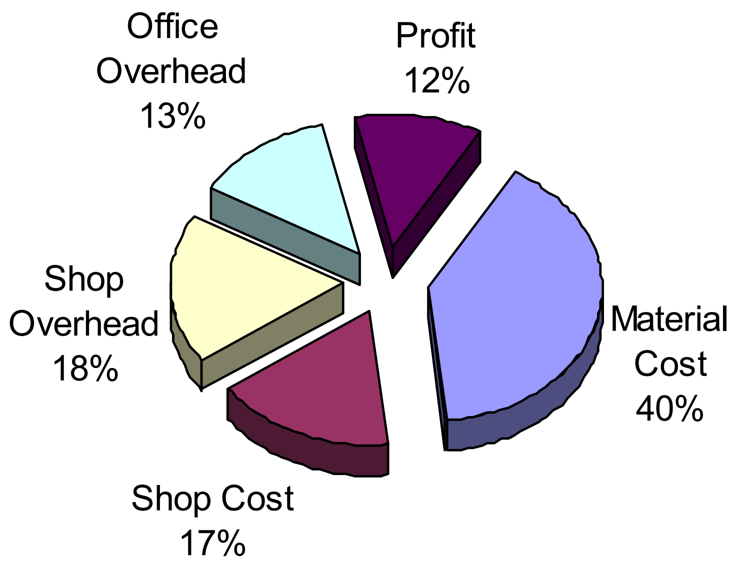


Figure 11 – Overall Cost Drivers for a High Pressure Carbon Steel Horizontal Drum

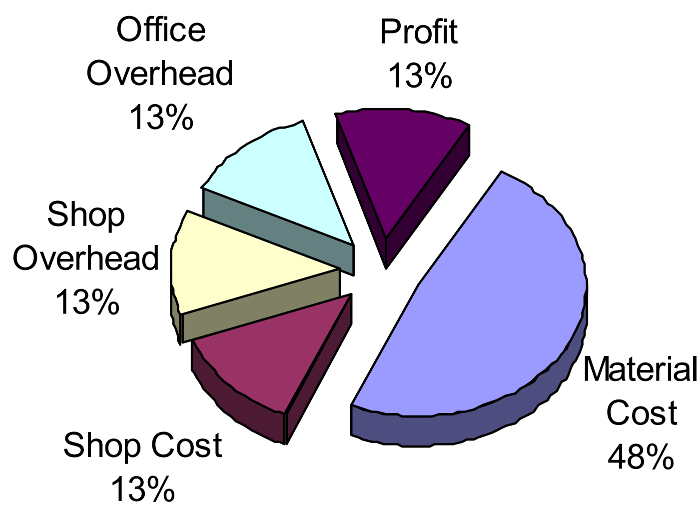


Figure 12 – Overall Cost Drivers for a Trayed Carbon Steel Column with Stainless Steel Cladding

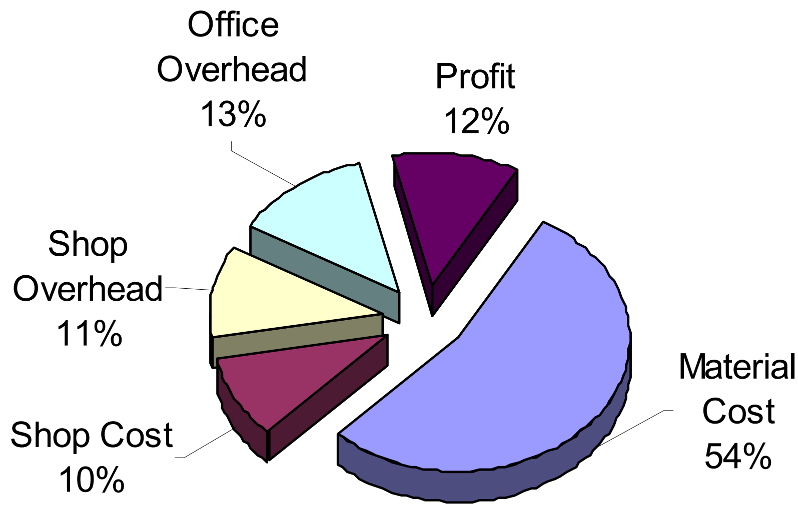


Figure 13 – Overall Cost Drivers for a 1 1/4 Cr-1/2 Mo Steel Reactor

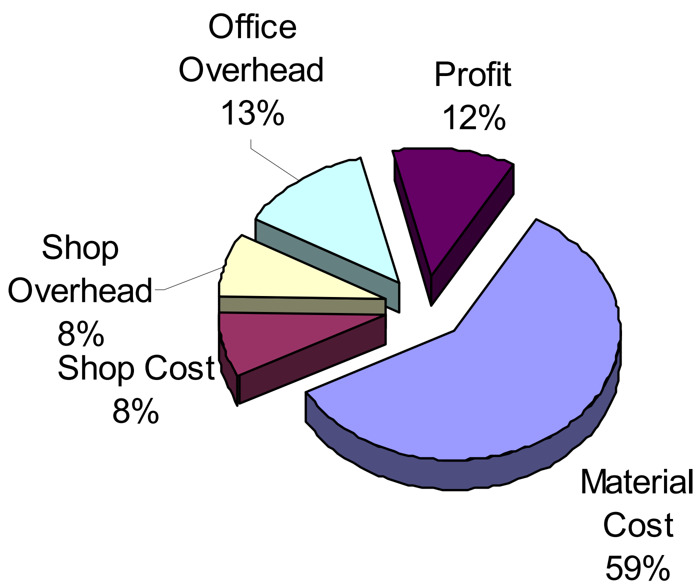


Figure 14 – Overall Cost Drivers for 2¼C Cr - 1Mo and 2 ¼Cr - 1Mo - ¼V Steel Reactors

As can been seen from the graphs, as the material is upgraded from carbon steel to chrome moly, or stainless steel the material portion of the cost continues to increases. Thus, there can be a very dramatic economic advantage if the material quantity can be optimized and reduced.

One can also see that the other factors associated with the cost such as shop, engineering, and profit cost do not vary much between the materials that are used. In addition, from the results of the code

comparison and industry research survey it did not appear that the non-material costs provided any significant change on the overall vessel cost between the different pressure vessel codes.

Therefore, the primary cost differentiator for new vessel purchase costs is going to be the cost associated with the material quantity of the vessel. Since this cost will vary among the different pressure vessels types and will vary with the selection of the pressure vessels codes, it is critical that the costs are reviewed on an entire site or project basis so that a true reflection of the overall purchase costs are used to make the proper decision on the code to be used for the new vessel purchases.

To emphasize the importance of proper consideration of the vessel codes on a project, the following chart portrays the cost of the vessel types on a typical E&C type clean fuel project. One can see that the primary costs are going to be in the high temperature reactors and large towers. Since the amount of material in these items is the primary cost drivers, it is essential that the quantity be optimized.

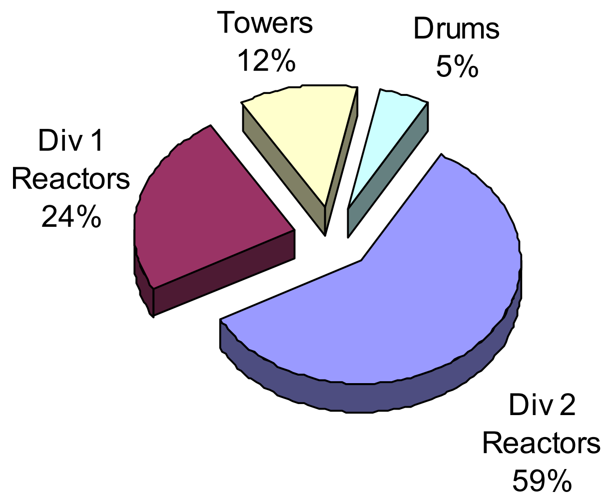


Figure 15 – Typical Cost Associated With Vessel Types On Clean Fuel Projects

5 SURVEY ANALYSIS

A survey was conducted on five key factors associated with pressure vessel codes, ASME Section VIII, Division 1, Division 2, PD5500, and EN 13445. These key factors included quantities used, cost differences, schedule differences, preferences, and future use. To help ensure that a broad range of results were available for comparison, the survey was submitted to owner/users, material suppliers, and fabricators from all over the world and results were obtained from 8 different countries, including 3 EC members.

The results of the surveys are included in the following sections and they are organized by the five key factors discussed above. In general, it appears that the ASME Section VIII pressure vessel code is the most common code currently in use. It provides a good economic advantage especially for the complex/costly vessels and it seems to be the current and future preference by the survey participants.

The following chart provides the general summary of the annual quantity percentage of vessels provided to each of the requested pressure vessel codes. This is an average value of all the completed surveys that were obtained.

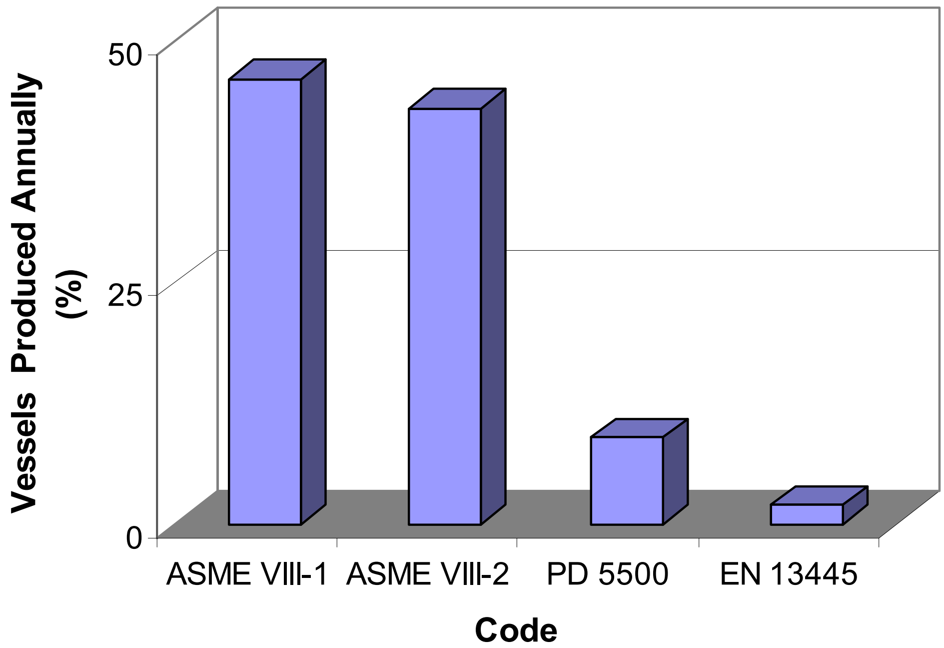


Figure 16 – Quantities of Vessels Produced Annually

The following charts provide the summary of the anticipated cost percentage differences between each of the requested pressure vessel codes for each of the major vessel types provided in the survey. These are medium size carbon steel, large size carbon steel, high pressure carbon steel, trayed column, 1 ¼ Cr - ½ Mo reactor and 2 ¼ Cr - 1 Mo- V reactor. These are average values of all the completed surveys that were obtained.

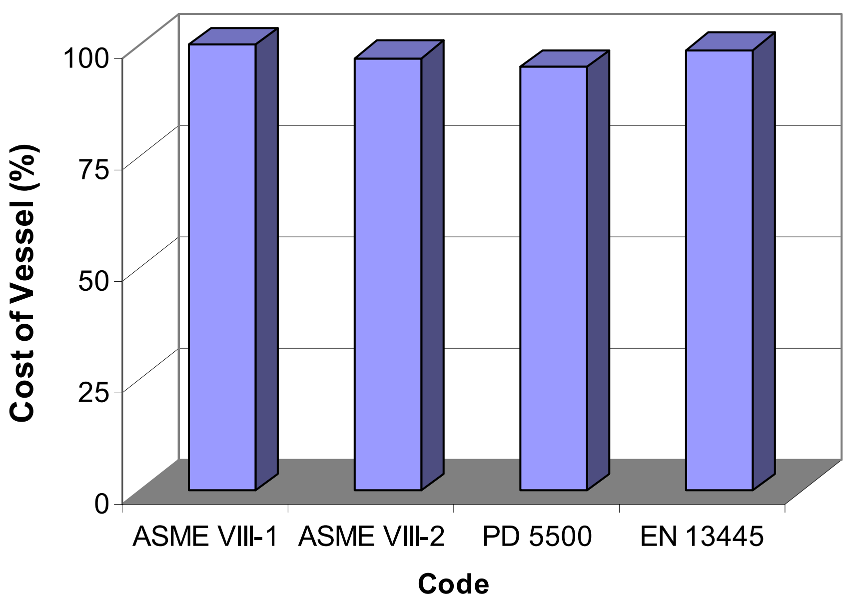


Figure 17 – Medium Size Carbon Steel Vertical Drum
(8'-0" Dia. x 20'-0" T/T x 50 psig @ 400°F with a 10'-0" skirt)

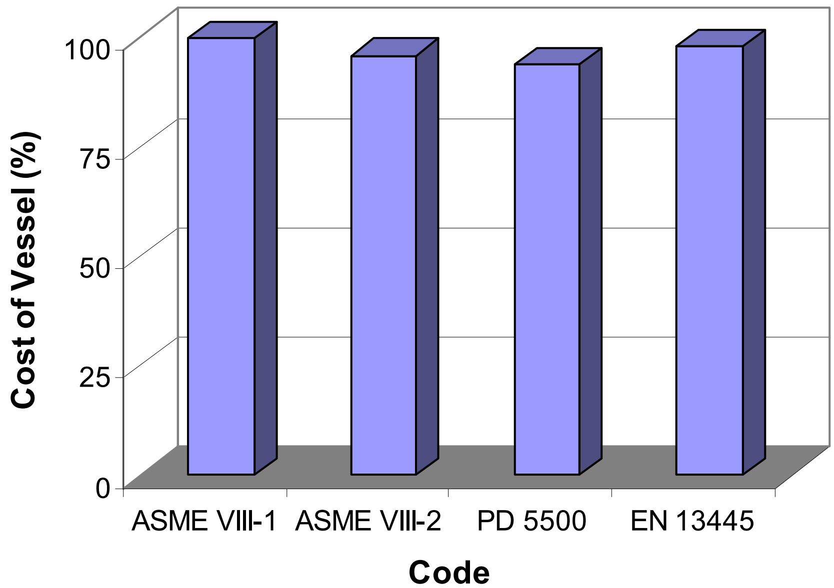


Figure 18 – Large Carbon Steel Vertical Drum
(25'-0" Dia. x 80'-0" T/T x 400 psig @ 500°F with a 10'-0" skirt)

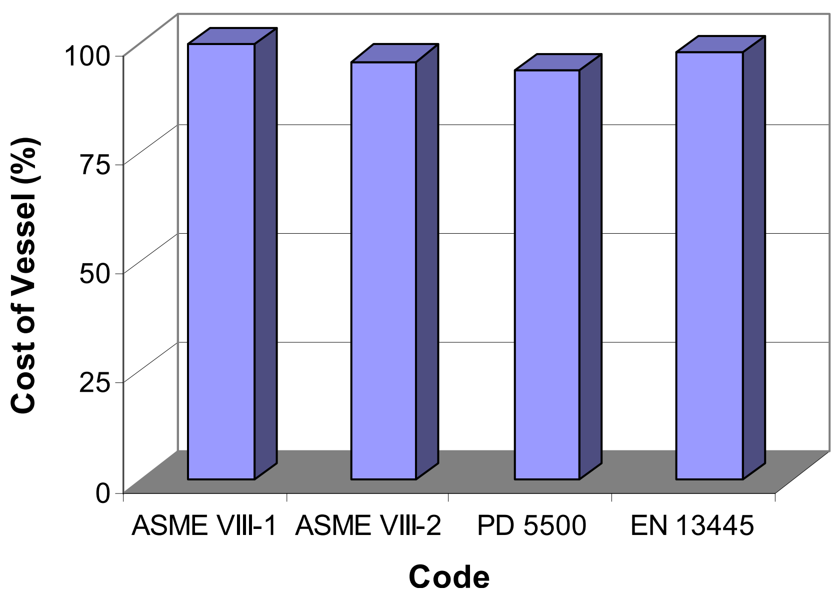


Figure 19 – High Pressure Carbon Steel Horizontal Drum
(8'-0" Dia. x 30'-0" T/T x 1500 psig @ 400°F with saddles)

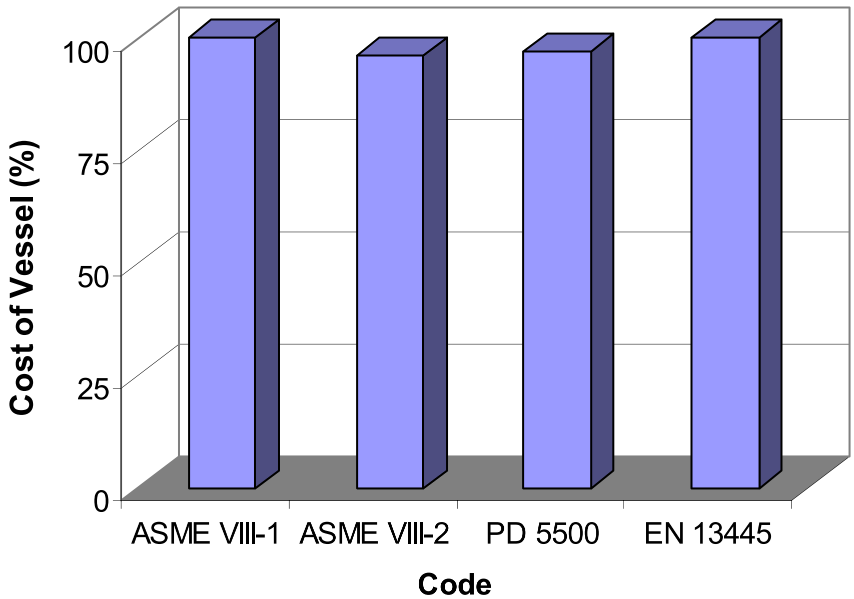


Figure 20 – Trayed Column Carbon Steel with SS Clad
(10'-0" Dia. x 150'-0" T/T x 100 psig @ 600°F with a 20'-0" skirt)

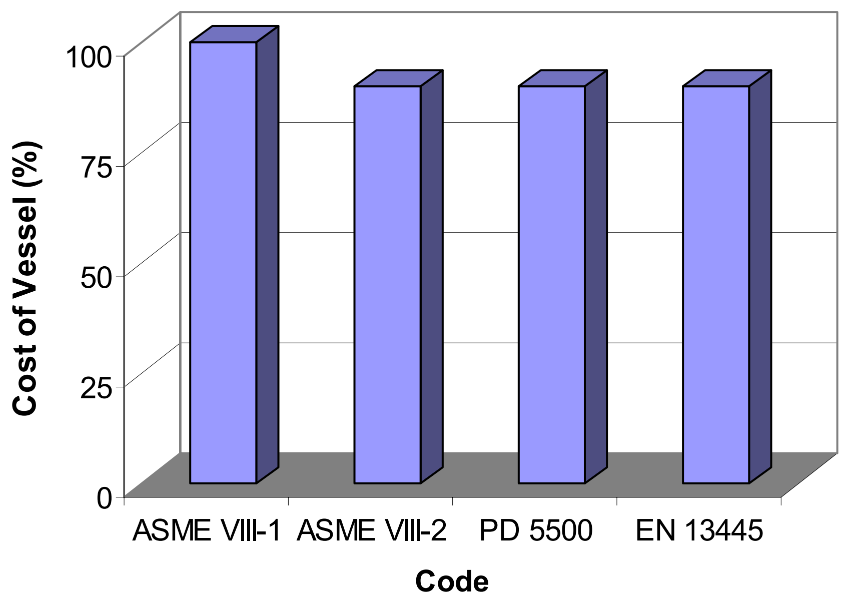


Figure 21 – Reactor 1 ¼ Cr – ½ Mo
(10'-0" Dia. x 40'-0" T/T x 1000 psig @ 800°F with 347 single pass overlay and a 15'-0" skirt)

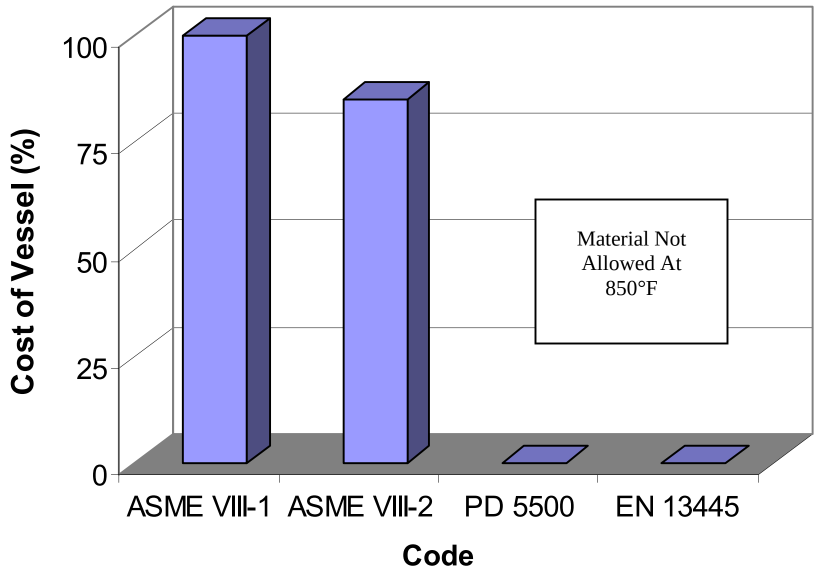


Figure 22 – Reactor 2 ¼ Cr – 1 Mo - V
(10’-0” Dia. x 60’-0” T/T x 2000 psig @ 850°F with 347 single pass overlay and a 15’-0” skirt)

The following chart provides the summary of the projected schedule duration differences between each of the codes. These are averaged based on all the survey results obtained.

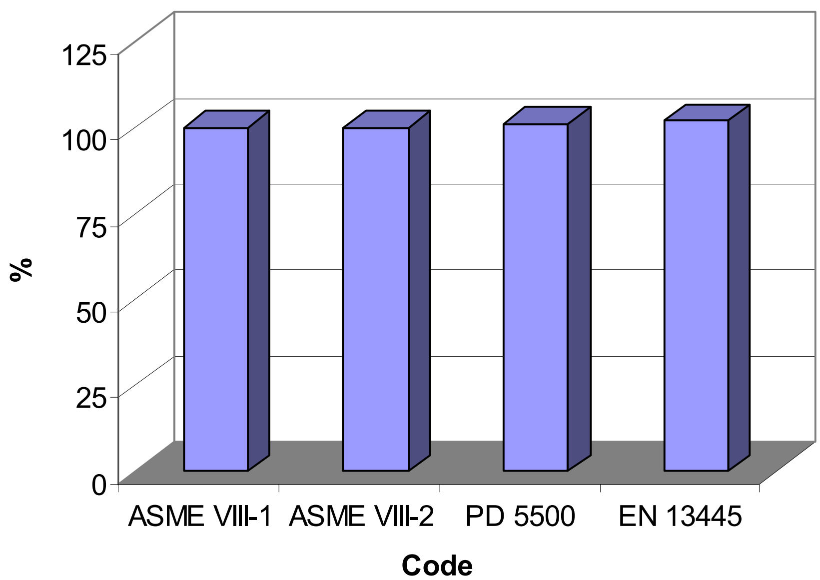


Figure 23 – Schedule Duration

The following chart provides the preference rank based on the expressed preferences of the survey respondents. Without exception, the respondents selected the ASME Section VIII, code as their first preference. This is based on the frequency of use, overall economic advantages, and familiarity with the ASME Codes internationally.

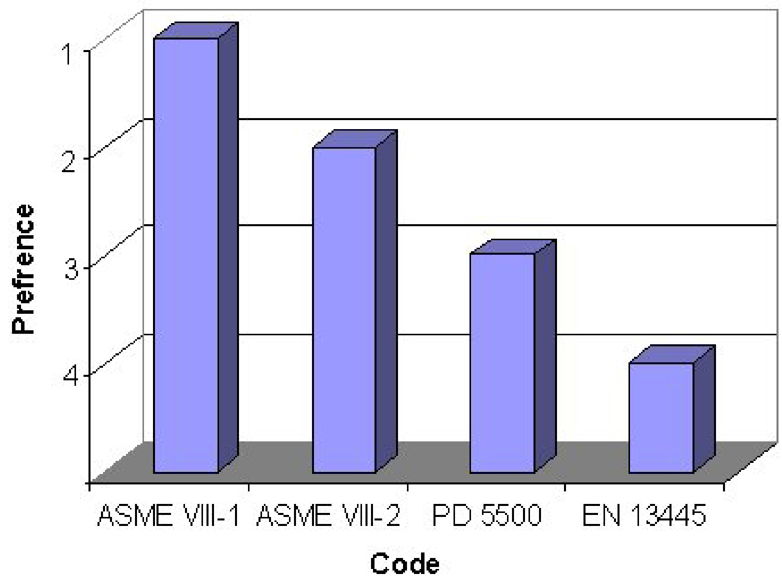


Figure 24 – Preference Rank

The following chart provides the anticipated future use of each of the vessel codes. These charts are averaged based on the anticipated yearly totals for the next five years.

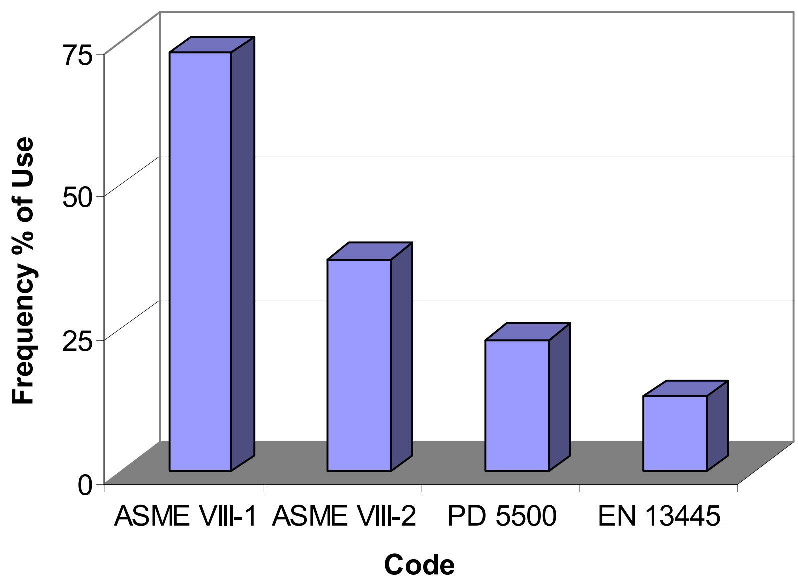


Figure 25 – Future Frequency

In general the survey confirmed that throughout the global industry there is a strong preference to use the ASME codes for pressure vessel design and manufacturing. Even though the PD5500 or EN 13445 may have a few specific areas or cases where there is a small economic advantage, when considering the overall aspects of the entire organization, plant, or project cost, the ASME code seems to provide a better overall advantage.

Overall, the primary cost advantage lies with the highest allowable stress that can be used. Of course this is limited to designs which are governed by pressure loads and not wind or seismic loads. The perception of the industry survey participants is that ASME is addressing the areas where they have slightly lower allowable stresses and that ASME will continue to be competitive in every type of Vessel and they will continue be the most common global construction code for the industry.

6 CONCLUSION

As has been discussed, EN 13445 does provide the designer the opportunity to design and construct thinner and less expensive vessels in very specific cases. The previous study cited several of these cases and concluded that there were economic reasons to construct every type of vessel based on that code as a general rule. As has been discussed here, there were a number of issues in the previous study which bring those conclusions into question.

The ASME codes have a number of advantageous features which make them highly competitive. As noted in the economic analysis, the majority of pressure vessel expense in plant construction is concentrated in a small number of high pressure thick walled vessels. ASME code cases allow the use of Division 3 design rules in the design of these vessels resulting in thinner walls and lighter weights. A typical example is Code Case 2390 for composite reinforced pressure vessels for compressed hydrocarbon gas services with external FRP wrapping that very significantly reduces the cost and weight of a range of CNG vessels. Although Division 3 is generally intended for pressure vessels above 10 ksi, it may also be used for vessels at lower design pressures. There are certain code cases (such as CC 2390-1 for Composite Reinforced Pressure Vessels) that include additional requirements for vessels of new and novel designs that can be constructed to Division 3 rules, which results in thinner walls and lighter weights.

The presence of rules in the ASME Section VIII, Division 1 allowing stresses based on time dependant properties permits design at temperatures in which the materials are in the creep zone, thus allowing higher operating temperatures. Higher allowable stresses in forged materials used extensively in thick wall vessels results in thinner walls with resulting reduced weights. All of these factors combine to provide significant economic savings in constructing high end vessels such as reactors in compliance with the ASME codes.

Since economics tend to favor the use of the ASME codes in the high end vessels and the added complexity of project administration, design, and inspection to multiple codes will tend to offset the savings from construction of the towers, tanks and other vessels which comprise the minority of the overall pressure vessel budget on the typical project. When coupled with the added life cycle ownership costs that a plant constructed to multiple codes will impose on the plant owners, economic analysis of combined costs will tend to favor the ASME codes.

A review of the EN standard has shown several important and innovative features. The ASME is in the process of rewriting Section VIII, Division 2, which will make a range of Division 2 vessels even more competitive with the EN standard. This rewrite is an opportunity to incorporate the latest advances in pressure vessel design, as well as new and innovative features that will enable the ASME Code to remain the preeminent pressure vessel standard.

ACKNOWLEDGMENTS

Kenneth Kirkpatrick, Engineer – Assistance with preparing document.

John Kehre, Engineer – Assistance with preparing document.

REFERENCES

[1] "Comparative Study on Pressure Equipment Standards", June 2004, Final Report, European Commission, Enterprise Directorate—General.

[2] ASME, 2004, ASME Boiler and Pressure Vessel Code, American Society of Mechanical Engineers, New York.

[3] European Committee for Standardization, CEN EN 13445:2002, Unfired Pressure Vessels, European Standard.

[4] Preiss, R., and Zeman, J., 2004, "Comparative Study EN 13445/ASME Section VII Div. 1 and 2", ASME PVP conference, San Diego, PVP-Vol. 480, p. 285, PVP20045.

[5] API RP 934, Specification for Heavy Wall Pressure Vessels of 2 1/4Cr-1Mo and 3Cr-1/2Mo Steels in High Temperature Hydrogen Service, API.

[6] Code Case 2514, "Use of SA-182 F22V, SA-336 F22V, SA-541 22V, SA-542 D Cl.4a and SA-832 22V to 850 deg. F", ASME.

[7] Code Case 2390, "Composite Reinforced Pressure Vessels", ASME.

[8] BS PD5500, "British Code for Unfired Fusion welded Pressure Vessels".

[9] BS EN 13445 Unfired Pressure Vessels

[10] API 510 – Pressure Vessel Inspector Certification

[11] NBIC-23 – National Board Inspection Code, Manual for Boiler and Pressure Vessel Inspectors

INTENTIONALLY LEFT BLANK

PART II - PVP2006-ICPVT11-93059: Design Fatigue Life Comparison of ASME Section VIII and EN 13445 Vessels with Welded Joints

Prepared by:

Arturs Kalnins
Lehigh University
Bethlehem, PA

Guido G. Karcher
Consulting Engineer
Little Egg Harbor, NJ

Mahendra Rana
Praxair, Inc.
Tonawanda, NY

ABSTRACT

In a recent study conducted by the European Commission, design fatigue life of welded vessels allowed by the ASME Boiler and Pressure Vessel Code was compared with that of the European Standard EN 13445. The allowable number cycles of the ASME Code was reported to be much larger than that of EN 13445, and, therefore, the ASME Code was regarded as unconservative for welded regions. This paper investigates the reason for the reported discrepancy between the two design codes. It is found that, when calculating the allowable cycles reported in the study using the ASME Code as a basis, no fatigue strength reduction factor on stress was used, which is contrary to the ASME Code design rules for welded joints. This paper recalculates the allowable cycles according to ASME Code rules and concludes that they are comparable with those of EN 13445.

NOMENCLATURE

FS = fatigue strength

FSRF = fatigue strength reduction factor

WRC = Welding Research Council

NDE = non-destructive examination

RT = radiographic (x-ray) testing

UT = ultrasonic testing

MT = magnetic particle testing

PT = dye (liquid) penetrant testing

VT = visual examination testing

1 INTRODUCTION

The European Commission conducted a comparative study [1] (henceforth identified as the EC Study) of economic and non-economic issues of the ASME Boiler and Pressure Vessel Code [2] (henceforth identified as ASME Code) and the European Standard EN 13445 [3] for unfired pressure vessels (henceforth identified as EN 13445). The full report of the EC Study was not made available to the authors of this paper, but a summary was published on the Internet [4] and its main results were presented in a panel session at the ASME/JSME 2004-PVP conference [5]. Supplemental information was provided in a private communication. As far as it is known from the summary [4], the EC Study addressed conceptual designs. No indication was given that the vessels had been actually constructed and tested.

The EC Study considered nine examples. For two of the examples, fatigue analysis was required, and in both, the locations for potential fatigue failure were in welded regions. The reported results are summarized in Table 1.

Table 1 - Allowable Design Cycles Reported in EC Study

	EN 13445	ASME Code
Example 3	33,576	more than 5×10^6
Example 4 Batch	13,100	$200\text{x}10^6$
Example 4 Stirrer	212,000[1]	more than 10^{11}

As seen from Table 1, for both examples the maximum allowable design cycles of the ASME Code were reported in the EC Study [1] to be far greater than those of EN 13445.

Contrary to the conclusions in the EC Study, the reported results in the ASME Code column of Table 1 do not represent the number of cycles allowed by the ASME Code. They were obtained without applying any factors on the cycled stress range, which are needed to account for the reduction in fatigue strength due to the welds or other discontinuities. ASME Code's Section III and Section VIII-Division 2 require that such factors be used (reference AD-412.1, and Appendix 5, 5-111 to 5-122, in Section VIII-Division 2). As shown in this paper, when the ASME cycles are recalculated with appropriate factors, the maximum allowable design cycles of the ASME Code are comparable with those of EN 13445.

2 ASSESSING DESIGN LIFE FOR WELDED JOINTS

Two types of S-N curves are available for fatigue analysis of welded joints. One is based on test specimens that include weld details and the other is based on smooth-bar test specimens. When applied to locations that are affected by welds, EN 13445 uses the former and the ASME Code uses the latter.

In order to account for the reduction of fatigue strength at a location affected by a weld, ASME Code's Sections III and VIII-Division 2 apply a *fatigue strength reduction factor* (FSRF) to the stress range that is being evaluated. The product is then entered on an ASME Code's design fatigue curve to obtain the maximum allowable design cycles. This paper will focus on the rules in Section VIII, Division 2. It is noted that Section III provides similar rules.

---

[1] This number was not listed in the summary of the EC Study [4] but was received in a private supplemental communication on the EC Study.

3 FSRF'S IN SECTION VIII DIVISION 2

Section VIII-Division 2, Appendix 5, 5-112, specifies FSRF=4 for certain types of fillet welds and Article D-4, AD-412.1, specifies FSRF=2 on the membrane stress range and FSRF=2.5 on the bending stress range for Type No. 2 joints, leaving it up to the analyst to determine appropriate FSRFs for other cases. Many publications outside the codebooks have provided FSRFs for specific cases from which the designer can draw. The publication selected for this paper is Bulletin No. 432 of the Welding Research Council (WRC) [6]. The Bulletin consists of two articles, one by Carl Jaske [7] and the other by John Hechmer and Elmer Kuhn [8].

In the first article of the WRC Bulletin No. 432 [7], Jaske extends the definition of FSRFs in Div. 2, Appendix 4, 4-112(o), to one that is applicable to welded joints. The definition in equation (1) involves fatigue strength (FS), which is defined as the value of the applied cyclic stress that produces fatigue failure at a certain number of cycles.

$$FSRF = \frac{FS - without - weld}{FS - with - weld} \tag{1}$$

"FS-without-weld" in the numerator refers to the design fatigue S-N curves used in Sections III and Section VIII-Division 2. This definition distinguishes an FSRF for welds from that applicable to a smooth location at a notch, for which the local geometry is determinable, and to which notch sensitivity provides the connection with the theoretical stress concentration factor. For a location affected by a weld, the local geometry is typically not determinable, the theoretical stress concentration factor is unknown, and the kind of FSRFs given in WRC Bulletin No. 432 [6] are needed.

The main point of Hechmer and Kuhn's article [8] of Bulletin No. 432 is that the reduction in fatigue strength of a welded joint depends mainly on the weld type and weld quality, and that the degree of examination (NDE) of the weld region provides adequate assessment of the quality. While other parameters that affect the reduction in fatigue strength are also discussed in Bulletin No. 432, the weld type and quality are identified as essential in estimating the maximum allowable design life of a welded joint.

According to Hechmer and Kuhn [8], a full-penetration weld in as-welded condition deserves an FSRF of 1.2. The other extreme is a fillet weld, for which FSRF=4. For other weld types and quality levels, Reference [8] provides FSRFs that are listed in Tables 1 and 2, which are copied from page 32 of the WRC Bulletin No. 432 [8] and shown in Figure 1 at the end of this paper.

As an illustration of how it works, consider the full penetration weld in the vessel tested recently by the Paulin Research Group and reported by Chris Hinnant [9]. The part of the vessel that is of interest here consisted of a cylindrical shell with a welded-on flat head, like the joint identified by a circle in Figure 2. Attachment welds were full penetration with cover fillets on ID and OD of the shell. The welds had received full RT and VT, and failure occurred at 30,500 cycles. According to Tables 1 and 2 of [8] shown in Figure 1, the Quality Level is 4 and FSRF=2 for this weld. Using ASME Code's design fatigue curve for carbon steel, the allowable cycles are 1550, which gives a factor of almost 20 with respect to the failure cycles. Of course, this one test does not "validate" the FSRFs in [8] – it is assumed that Jaske, Hechmer and Kuhn have done that – but it does provide one datum point for the magnitude of the factor on the allowable cycles that the ASME Code can provide with respect to failure. This is not surprising as the ASME design fatigue curve is a "design curve" and not a failure curve.

4 EXAMPLE 3 OF EC STUDY

Example 3 refers to an autoclave vessel for which fatigue assessment is mandatory. A sketch of the vessel is shown in Figure 2. The weld subjected to fatigue analysis in the EC Study [1] was the corner weld between the shell and the flat end, marked by a circle in Figure 2. The summary of the EC Study gives the maximum allowable cycles in Table 1 but does not identify the weld that was subjected to fatigue analysis. Dr. Reinhard Preiss of TÜV Austria kindly provided additional information in private communication.

4.1 Fatigue Analysis in the EC Study

Dr. Preiss reported some details of the fatigue analysis that was conducted in the EC Study [1]. Detail 2.1(c) in Figure 3 shows the weld type. The root of the one-sided weld was considered as the critical location. It was assumed that the inside of the weld could be visually inspected and that Testing Group 1 or 2 was applied, which requires, as the minimum, 100% RT or UT. With these assumptions, weld class FAT 63 was used. The pressure range of 0.26 MPa (38 psi) was cycled inside the shell, and the calculated stress range was reported as 246 MPa (35.7 ksi). Using the FAT 63 S-N line, the maximum number of allowable design cycles according to EN 13445 was reported on page 7 in the summary of the EC Study [4] as 33,576.

For the calculation of the maximum number of allowable design cycles of the ASME Code, the EC Study [1] used the same stress range of 246 MPa. No FSRF was applied to the stress range. According to the design fatigue S-N curve of Division 2 for stainless steel, Appendix 5, Fig. 5-110.2.2M, Curve C, more than 5 million cycles were reported on page 20 of the summary of the EC Study [4]. As it will be shown in the following section, this result does not represent the number of cycles allowed by the ASME Code if correctly evaluated.

4.2 Fatigue Analysis by ASME Code

The vessel being compared with EN 13445 is a Section VIII-Division 1 vessel. Division 1 does not provide rules for fatigue analysis but UG-22, requires “consideration” of cyclic loadings. If fatigue analysis is required, U-2(g) may be invoked, which permits using methods external to Division 1, but they must lead to designs that are “as safe as those provided by the rules of Division 1”. For the fatigue analysis of this paper, Section VIII-Division 2 is chosen as the source and the FSRFs are taken from WRC Bulletin No. 432 [6].

The welded joint of Example 3 belongs to Category C and is classified in Table UW-12 of Division 1 as Type (7) joint. It is applicable to the corner joints of Fig. UW-13.2 (see Figure 4), and the particular joint of Example 3 is shown in sketch (e-1). Neither RT/UT (as per UW-11) nor MT/PT (as per UHA-34) is required for this joint. It is assumed that no NDE will be applied to the root of the one-sided weld. For such a situation, Tables 1 and 2 of Bulletin No. 432 shown in Figure 1 justify Quality Level 7 and an FSRF between 3 and 4. The designer can decide which FSRF to choose based on the other parameters that affect the fatigue strength of welded joints, some of which are given in WRC Bulletin No. 432 [6].

Regarding the stress calculation, the EC Study [1] indicates greater shell and flat end thicknesses for the Division 1 design than those for the EN 13445 design. These greater thicknesses were not used for the fatigue analysis of this paper because doing so would not offer a fair comparison between the two design codes. The objective in this paper is to compare the design life for exactly the same design. Therefore, the same structural stress range that was calculated for the EN 13445 design in the EC Study, 246 MPa (35.7 ksi), will be used.

Using FSRF=4, the alternating stress amplitude is

$$S_a = 4 \times 246 / 2 = 492 MPa \qquad (2)$$

According to the design fatigue S-N curve of Division 2 for stainless steel, Appendix 5, Fig. 5-110.2.1M, the number of allowable cycles according to the ASME Code is 6,450. The corresponding number of allowable cycles for FSRF=3 is 20,000.

As already stated, it is the designer's responsibility to choose an FSRF (in this case, from 3 to 4) that will lead to a safe design for the application. Appendix V of Hechmer and Kuhn [8] provides guidelines for that choice, which depend on the quality of the welding process that will be applied to the joint. For example, a welding process that has been used successfully on many welds to which full NDE has been applied should merit an FSRF=3 even if the weld receives no NDE. Since nothing is known about the weld process that will be used for the corner weld of Example 3, FSRF=4 and the 6,450 maximum allowable cycles are selected.

Since these cycles assumed a weld with no NDE, a fair comparison with EN 13445 would be provided by the weld class FAT 40, and not FAT 63, because the latter assumed 100% RT and VT of the weld root region. This would give 8,600 allowable cycles according to EN 13445. Figure 5 shows the EN 13445 (FAT 40) and ASME Code's FSRF=4 weld design fatigue S-N curves. (The latter is obtained by lowering the Div. 2 design fatigue curve by the factor of 4 on stress.)

These results show that, as already stated in the Introduction, the maximum allowable design cycles of the ASME Code are comparable with those of EN 13445 and not near the "more than 5 million" cycles reported in the EC Study [1].

5 EXAMPLE 4 OF EC STUDY

The geometry of the vessel is shown in Figure 6. This is a stirring vessel for which fatigue assessment is mandatory. Two cyclic actions are involved, batch and stirring operation.

5.1 Batch Operation

Few details are given in the summary of the EC Study [4] for this case. It is reported only that the allowable number of batch cycles according to EN 13445 is 13,100 and that those according to the ASME Code are 200 million. A possible procedure that led to these results can be reconstructed for the purposes of this paper as follows.

The critical location is selected at the welds of one of the large nozzles, identified by the circles in Figure 6. Assuming Detail No. 3.2 of Table P.3 in Annex P of Clause 18 in EN 13445, full penetration, as-welded condition, and Testing Group 3 for this weld, the weld class is FAT 63. An approximate stress range of 337 MPa can then be back calculated from the 13,100 allowable cycles. This range can be used to estimate the allowable cycles according to the ASME Code.

Since Table P.3 indicates fatigue failure in the shell adjacent to the weld toe, VT is assumed of the weld region. Tables 1 and 2 of Bulletin No. 432 justify FSRF=2.5. According to the design fatigue S-N curve of Division 2 for stainless steel, Appendix 5, Fig. 5-110.2.1M, the allowable cycles are estimated at 12,500. Figure 7 shows the EN 13445 (FAT 63) and ASME (FSRF=2.5) weld design fatigue S-N curves for this weld type and NDE, and the allowable cycles.

Reader is cautioned that the calculated stress range, weld type, and NDE are based on the limited data that were made available to the authors and may not be what were used in the EC Study. While the estimated number of ASME cycles may not be accurate because of the assumptions stated above, they do confirm that correct use of the ASME Code does not support the claim that it would allow 200 million cycles for the assumed design conditions.

5.2 Stirrer Operation

Again, few details are given in the summary of the EC Study [4] also for this case, except that the fluctuating load components rotate about the stirrer axis and are assumed to act in the most unfavorable way. Design for an infinite number of cycles is required.

Dr. Reinhard Preiss provided more information in a private communication. The critical detail is the weld between the mounting flange reinforcement ring and the upper dished head (Figure 8). That weld is identified by Detail No. 5.3 of Table P.5 in Annex P of Clause 18 in EN 13445, which is rated at the weld class of FAT 71. The calculated stress range is reported as 150 MPa.

With this information, the maximum number of allowable design cycles according to EN 13445 is 212,000. Dr. Preiss reported that the same stress range of 150 MPa was also used to calculate the allowable cycles of the ASME Code. Since that stress range is below the last entry of the design fatigue S-N curve of Division 2 for stainless steel, Appendix 5, Fig. 5-110.2.2M, Curve C, 100 billion cycles were reported in the EC Study [4] as allowable by the ASME Code. Again, this number does not represent the allowable cycles of the ASME Code.

Assuming VT for this weld, Tables 1 and 2 of Bulletin No. 432 (Figure 1) justify FSRFs from 2.0 to 2.5. According to the design fatigue S-N curve of Division 2 for stainless steel, Appendix 5, Fig. 5-110.2.2M, Curve C, the allowable cycles are estimated at 2,700,000 for FSRF=2 and 1,100,000 for FSRF=2.5. Figure 9 shows the EN 13445 (FAT 71) and ASME weld design fatigue S-N curves for both FSRF=2 and 2.5. As seen from Figure 9, the ASME curves do allow more cycles in the high cycle regime than the EN 13445 curve. However, the differences are within the normal scatter of the test data in this cycle regime and depend on someone's judgment on where to draw the curves. The important point is that for both FSRF values the ASME cycles are far from the 100 billion cycles reported in the EC Study [4].

6 DISCUSSION

A summary of corrected allowable cycles for Examples 3 and 4 is given in Table 2.

Table 2 - Allowable Design Cycles for Example 3 and 4

	EN 13445	**ASME Code**
Example 3 RT+VT	33,576	N/A
Example 3 no NDE	8,600	6,450
Example 4 Batch	13,100	12,500[2]
Example 4 Stirrer	212,000	1,100,000

The entry for the ASME Code cycles in the first row was marked as "not applicable" (N/A) because it was regarded that 100% RT and VT of the root of the weld was not possible. It is questionable how such NDE could be carried out to satisfy the EN 13445 Code requirements considering the multiple-exposure RT that would be required and the limited visual access for VT of such a corner weld.

[2] Because of the limited data made available to the authors, this number was obtained by assuming a critical location, FAT class, and NDE that may not be those used in the EC Study.

7 CONCLUSIONS

1. The large number of cycles attributed in the EC Study [1] to the ASME Code rules was obtained without applying fatigue strength reduction factors on the stresses that are cycled.
2. Fatigue strength reduction factors must be used in the application of Section VIII-Division 2 fatigue rules to welded joints.
3. Recalculation of the maximum allowable design cycles using ASME Code rules with appropriate FSRFs shows that they are comparable with those of EN 13445 and far less than those reported in the EC Study.
4. When the design fatigue life obtained from two design codes is not the same, it is impossible to judge which is "conservative" and which is "unconservative", as it was done in the EC Study, without comparing specific test data and agreeing on a margin that design fatigue life should have with respect to test life.
5. When such comparative studies are carried out and widely published, all of the details, data, and design assumptions used should be made available to the engineering community so that it can be readily reviewed and evaluated. Making many such details unavailable to the public has resulted in a less open discussion of important safety issues for cyclic design and evaluation of pressure equipment than it could have been had it been made available to the public.

ACKNOWLEDGMENTS

The authors express their appreciation to John Koehr, Director of ASME Standards Technology LLC, and Gerry Eisenberg, Director of ASME Codes and Standards Pressure Technology, for technical and editorial support in writing the paper. The authors also thank Dr. Reinhard Preiss of TÜV Austria for providing additional information on the fatigue analysis that was performed in the EC Study.

REFERENCES

[1] "Comparative Study on Pressure Equipment Standards", June 2004, Final Report, European Commission, Enterprise Directorate—General.

[2] ASME, 2004, ASME Boiler and Pressure Vessel Code, American Society of Mechanical Engineers, New York.

[3] European Committee for Standardization, CEN EN 13445:2002, Unfired Pressure Vessels, European Standard.

[4] Summary of "Comparative Study on Pressure Equipment Standards", July 2004, European Commission, DG Enterprise, Internet address: http://ped.eurodyn.com/standards.

[5] Preiss, R., and Zeman, J., 2004, "Comparative Study EN 13445/ASME Section VII Div. 1 and 2", ASME PVP conference, San Diego, PVP-Vol. 480, p. 285, PVP2004-2705.

[6] Jaske, C. E., Hechmer, J. L., and Kuhn, E. J., 1998, "Fatigue Strength Reduction and Stress Concentration Factors for Welds in Pressure Vessels and Piping", WRC Bulletin 432, June 1998, pp. 1-55.

[7] Jaske, C. E., 1998, "Report No. 1: Interpretative Review Weld Fatigue-Strength-Reduction and Stress-Concentration Factors", WRC Bulletin 432, June 1998, pp. 1-26.

[8] Hechmer, J. L., and Kuhn, E. J., 1998, "Report No. 2: Fatigue-Strength-Reduction Factors Based on NDE", WRC Bulletin 432, June 1998, pp. 27-55.

[9] Hinnant, C. 2006, " Fatigue Life Estimates and Testing of a Welded Flat Head Pressure Vessel", 2006PVP/ICPVT-11, paper 93967.

Table 1—Weld Surface Fatigue-Strength-Reduction Factors

Weld Condition (surface condition)	*Quality Levels**						
	1	*2*	*3*	*4*	*5*	*6*	*7*
Full penetration							
Machined	1.0	1.2–1.5	1.5	2.0	2–2.5	2.5–3.0	3.0–4.0
As-welded	1.2	1.3–1.6	1.7	2.0	2–2.5	2.5–3.0	3.0–4.0
Partial penetration							
Final surface							
Machined	n.a.	1.2–1.5	1.5	2.0	2–2.5	2.5–3.0	3.0–4.0
As-welded	n.a.	1.3–1.6	1.7	2.0	2–2.5	2.5–3.0	3.0–4.0
Root	n.a.	n.a.	n.a.	n.a.	n.a.	n.a.	3.0–4.0
Fillet							
Toe–machined	n.a.	n.a.	1.5	n.a.	2–2.5	2.5–3.0	3.0–4.0
Toe–as-welded	n.a.	n.a.	1.7	n.a.	2–2.5	2.5–3.0	3.0–4.0
Root	n.a.	n.a.	n.a.	n.a.	n.a.	n.a.	3.0–4.0

Table 2—Criteria for Weld FSRF

FSRF	*Quality Level*	*Definition*
1.0	1	Machined or ground weld that receives a full volumetric, and a surface that receives MT/PT and a VT (visual).
1.0–1.2	1	As-welded weld that receives a full volumetric, and a surface that receives MT/PT and VT.
1.2–1.5	2	Machined or ground weld that receives a partial volumetric, and a surface that receives MT/PT and VT.
1.3–1.6	2	As-welded weld that receives a partial volumetric, and a surface that receives MT/PT and VT.
1.5	3	Machined or ground weld surface that receives MT/PT and VT, but the weld receives no volumetric.
1.7	3	As-welded weld surface that receives MT/PT and VT, but the weld receives no volumetric.
2.0	4	Weld has received a partial or full volumetric and the surface has received VT, but no MT/PT.
2.0–2.5	5	VT only of the surface; no volumetric nor MT/PT examination.
2.5–3.0	6	Volumetric only.
3.0–4.0	7	Weld backsides that are non-definable and/or receive no inspection.

Figure 1 - Weld Quality Levels and FSRFs from WRC Bulletin No. 432, [8]

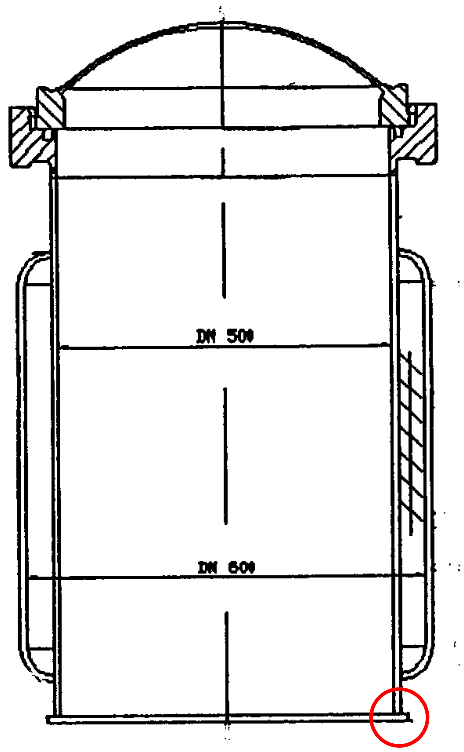


Figure 2 - Autoclave vessel of Example 3 in the EC Study [1]

EN 13445-3:2002 (E)
Issue 9 (2004-02)

Table P.2 — Shell to head or tubesheet

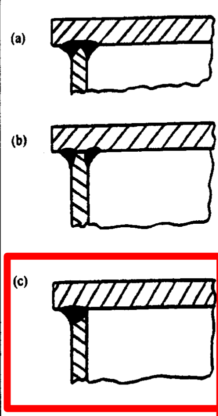
Detail No	Joint type	For principal stresses acting essentially normal to the weld			
		Sketch of detail	Comments	Class	
				Testing group 1 or 2	Testing group 3
2.1	Welded-on head	(a) (b) (c) 	Head plate must have adequate through-thickness properties to resist lamellar tearing.		
			Full penetration welds made from both sides: - as-welded; - weld toes dressed (see 18.10.2.2).	71 80	63 63
			Partial penetration welds made from both sides: - refers to fatigue cracking in shell from weld toe - refers to fatigue cracking in weld, based on stress range on weld throat	63 32	63 32
			Full penetration welds made from one side without back-up weld: - if the inside weld can be visually inspected and is proved free from weld overlap and root concavity. - if the inside cannot be visually inspected.	63 40	40 40

Figure 3 - EN 13445 weld classes for Example 3, [3]

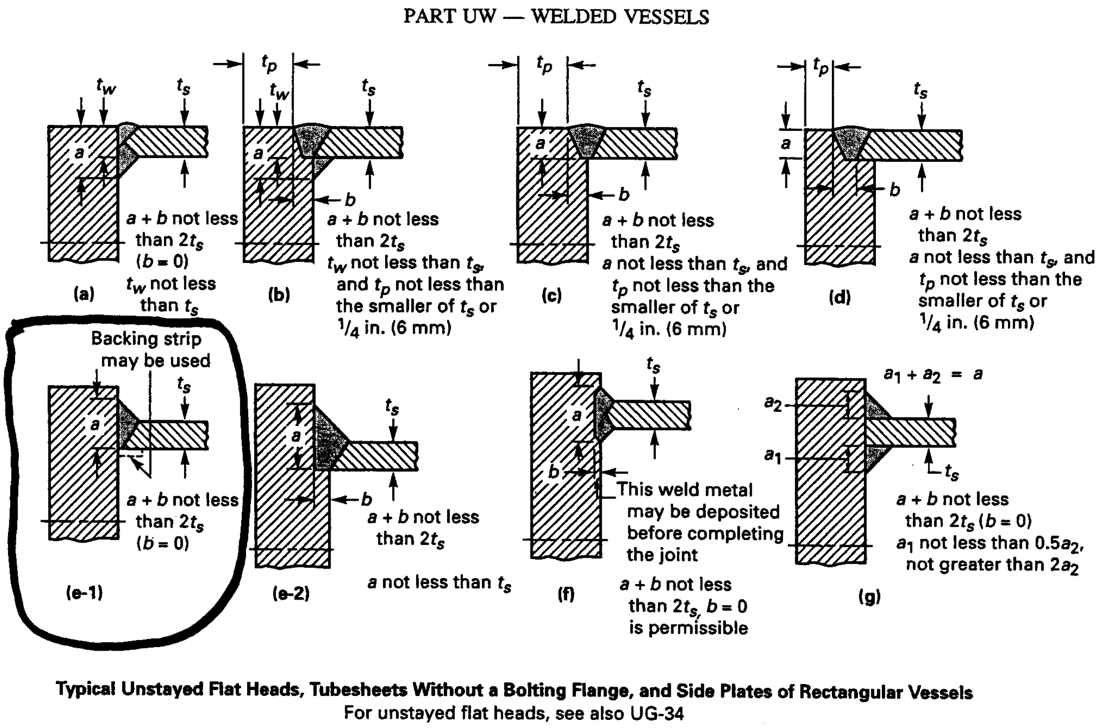


Figure 4 - Section VIII-Division 1 corner joint for Example 3, [2]

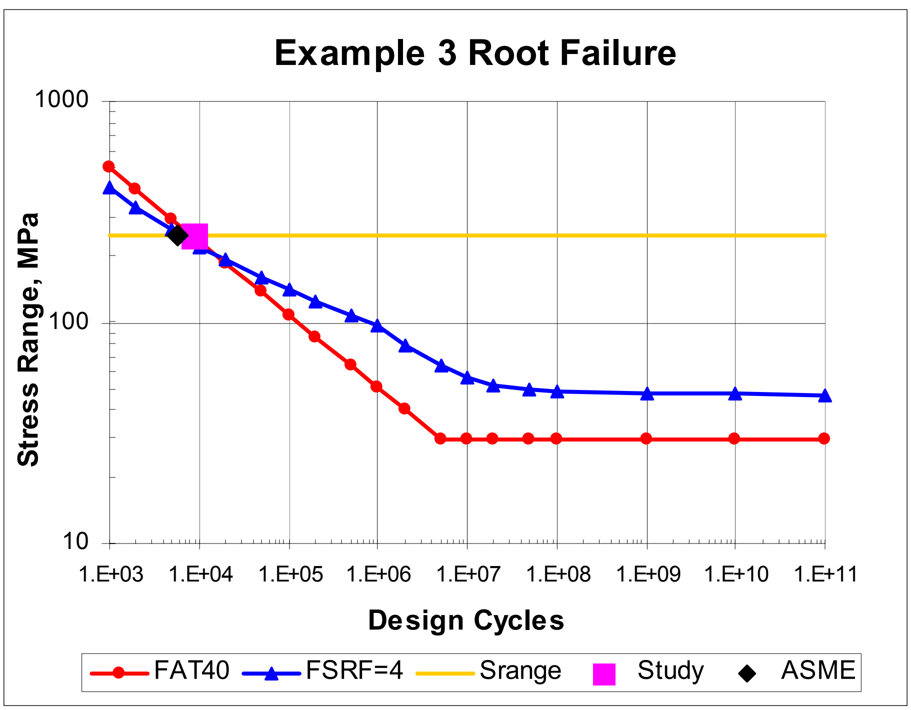


Figure 5 - Weld design fatigue S-N curves and allowable cycles

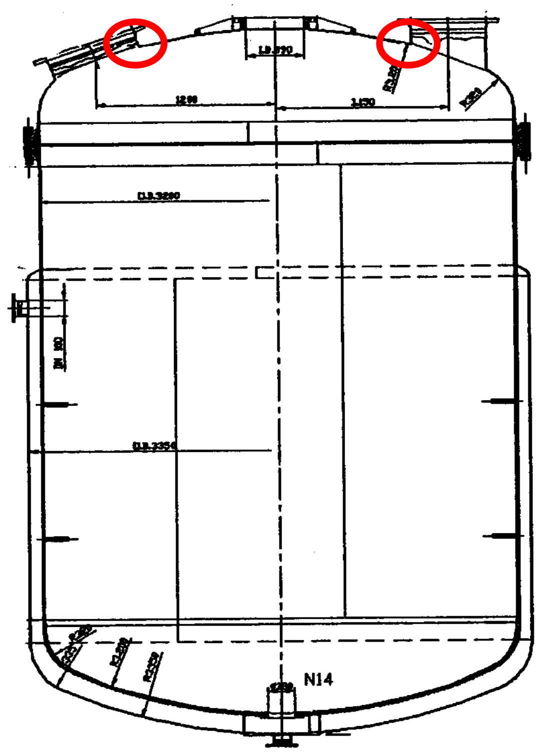


Figure 6 - Example 4 in the EC Study for batch operation [1]

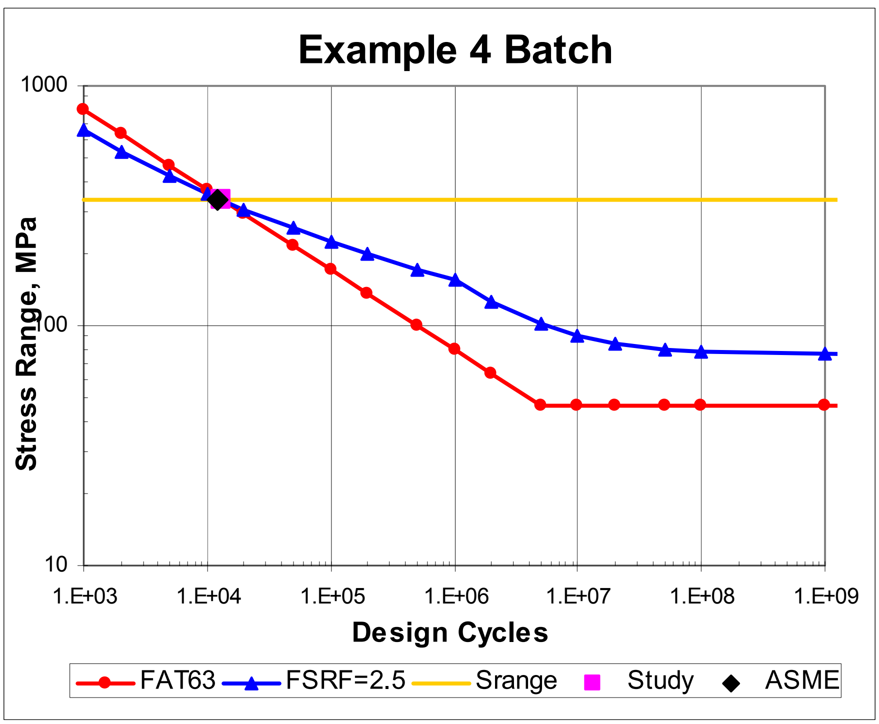


Figure 7 - Weld design fatigue S-N curves and allowable cycles

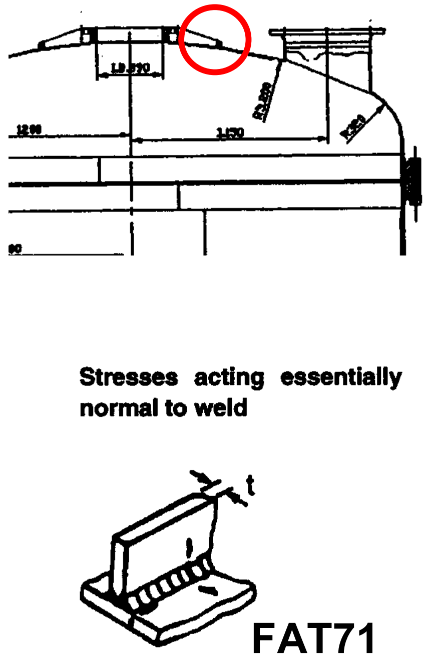


Figure 8 - Example 4 in the EC Study for stirrer operation and its weld type [1]

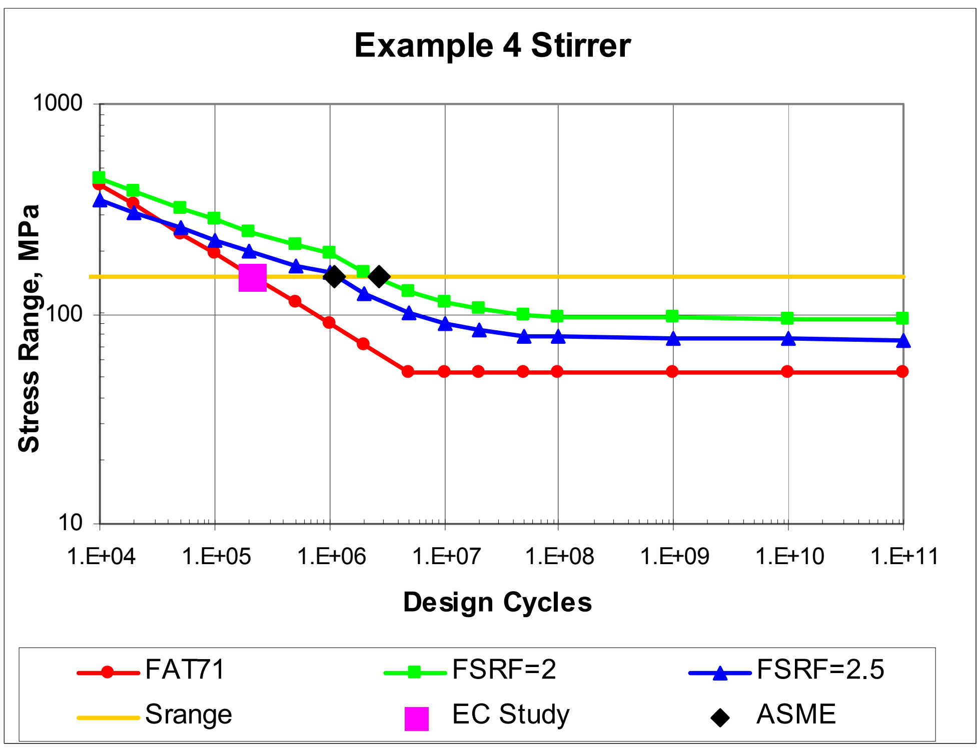


Figure 9 - Weld design fatigue S-N curves and allowable cycles