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RECOMMENDATIONS FOR DESIGN OF VESSELS FOR ELEVATED TEMPERATURE SERVICE

Vincent A. Carucci
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These Bulletins contain final Reports from projects sponsored by the Welding Research Council, important papers presented before engineering societies and other reports of current interest.

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FOREWORD

The PVRC Elevated Temperature Design (ETD) Committee was formed in 1969 with a charter to identify research needs and to support the ASME Section III Elevated Temperature Design Committee activities in the development of design rules for nuclear components. The Committee has since expanded its role to address the needs of other industries, such as Refining, Petrochemical, and Fossil Power.

ASME Section VIII Division 1 has had design allowable stresses in the creep range for many years. ASME is also developing new code rules for Section VIII, Division 2 at elevated temperature. The goal of the PVRC Committee on Elevated Temperature Design is to support those activities.

The ETD Committee is organized into four subcommittees and this project was completed for the Subcommittee on Petrochemical and Balance-of Plant Criteria, chaired by J. F. Cervenka, under PVRC Project 96-03.

The title of this project is "Design Considerations for ASME Section VIII, Division 1 Pressure Vessels Operating at Elevated Temperature" by V. A. Carucci , R. C. Chao and D. J. Stelling of Carmagen Engineering Inc. The objective of the project was to develop additional guidelines for the design of Section VIII Division 1 pressure vessels when the vessels are designed for operation at temperatures where the long term creep properties govern design.

The ASME Boiler and Pressure Vessel Code, Section VIII, Division 1 has a long and successful service history, including elevated temperature service, however experience has shown that in service cracking problems can develop in pressure vessels at elevated temperature. The experience indicates that the cracking has occurred at locations of high local discontinuity stresses, high local thermal gradients, and weld joints. The purpose of this project is to determine the current industry experience with specified practices and guidelines and screen out those design practices that have had problems from those that have been successful. The goal is to develop design rules, guidelines and mechanical design details that are recommended to Subcommittee VIII of the ASME Boiler and Pressure Vessel Code for Section VIII Division 1 pressure vessels operating in the creep regime. The focus is not on complex analytical methods, but practical design guidelines.

The report is general in scope, however the focus is on the carbon and low alloy steels. It is anticipated that many of the recommendations made will become standard practice in the Refining, Petrochemical and Fossil Power industries.

William J. Koves
Chair, Committee on Elevated Temperature Design

CONTENTS

Recommendations for Design of Vessels for Elevated Temperature Service

Vincent A. Carucci, Raymond C. Chao, Douglas J. Stelling,*

1.0 Introduction

The ASME Code Section VIII, Division 1 [1] permits pressure vessels to be designed and operated at temperatures that are in the creep regime of the materials of construction. However, the Code does not directly address some design issues that become relevant for pressure vessels that operate at high temperatures. For example:

- Div. 1 does not explicitly recognize that some "Code Acceptable" details (e.g., pad-reinforced nozzles) in our opinion should not be used at high temperatures.
- Div. 1 currently does not provide definitive details for when the metal temperature and wall thickness of the pressure vessel increase (e.g., specific support skirt-to-shell design details that are used for vertical pressure vessels).
- Div. 1 does recognize that some loadings may be short-term by providing higher allowable stresses for wind and earthquake loadings. Otherwise, Div. 1 lacks specific guidance on time-at-temperature issues.

Table 1 illustrates several typical high-temperature process plant applications with typical design conditions and materials of construction.

Experience has shown that some pressure vessels operating at temperatures in the creep range of the materials have experienced incidents of cracking after some period of operation. Such cracking has most often occurred at locations that exhibit stress concentrations and/or local thermal gradients, and weldments (in the heat affected zones). Typical locations where cracking has occurred are the following:

- Nozzle reinforcing pad attachment welds.
- Nozzle-to-shell junctions, both where there is connected piping and at manways.
- Support attachment points (e.g., skirt-to-shell attachments and lug-to-shell attachments).
- Intermediate head attachment welds to the shell.
- Other welded attachments to the vessel.
- Main circumferential and longitudinal welds away from any obvious large stress concentrations or thermal gradients.

Paragraph UG-22 of Section VIII Div. 1 requires that all loadings be considered. However, Section VIII Div. 1 provides varying levels of details of how to account for some loads. Div. 1 does not address "all" loads possible. Although Div. 1 does not require an official certified Design Specification, design requirements are to be provided in a purchase order or some other document. However, many pressure vessels have operated for long periods at temperatures in the creep range without experiencing any such failures. Obviously, based on this experience, there are ways to design, fabricate, inspect, and operate Division 1 pressure vessels such that they can achieve long-term operating reliability without cracking while operating in the creep range.

Because of some poor experience that has occurred, some companies have developed their own design requirements to improve the reliability of Division 1 pressure vessels that operate at temperatures in the creep range. These requirements are generally more restrictive than those published in Division 1 rules, and therefore may have to be justified to corporate management when the increased costs associated with their use are identified. On the other hand, companies that are less knowledgeable make no allowance for these high temperatures in their pressure vessel designs and operate pressure vessels that may be less reliable than they should be.

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Table 1
Typical Process Plant Elevated Temperature Processes

Process	Materials	Typical Design Temperature, °F	Typical Design Pressure, psig
Reforming	Low Cr	1000-1025	100-200
Dehydrogenation	Low Cr Type 304	1000 1200	60 60
Thermal Hydro-Dealkylation	Incoloy 800H for Heat Exchanger Hot End	1450	500
Styrene	Type 304	1200	Vacuum
FCCU Reactor	Low Cr	1025	50
Hydrocracking	2 ¼ to 3 Cr	850	3000
Delayed Coking	C-Mo, Low Cr	850 – 950	80 - 140

The purpose of this report is to present proposed rules, guidelines and mechanical design details that are suitable for Division 1 pressure vessels that operate at temperatures in the creep range of the material. The ultimate goal is for the ASME to consider these for inclusion in Section VIII, Division 1 as a combination of both mandatory rules and a new non-mandatory appendix that provide guidelines for the design of such pressure vessels. In addition, areas for future study are suggested which should result in additional guidelines to achieve improved pressure vessel reliability.

The ASME Code encompasses a broad range of acceptable material chemistries. However where materials and materials-rated requirements are discussed in this report, the focus is on carbon steel, C- ½ Mo, 1 Cr- ½ Mo, 1 ¼ Cr- ½ Mo, and 2 ¼ Cr-1 Mo. These materials are used in a broad variety of high-temperature applications and appear to have experienced the most in-service problems. Other topics discussed in this report are applicable to other material types as well.

This report does not attempt to provide a complete literature review of this broad subject nor purport to provide detailed analytical solutions to the problem. Much work has already been done and is continuing in this area. In addition, the Division 1 "Design by Rules" approach is not amenable to detailed analytical treatment and there is no intent to complicate this. This report does provide the following:

- Summarizes typical design considerations and details that may be appropriate for high-temperature applications.
- Proposes when such "special" considerations should be applied.
- Provides a simple approach that may be used when users wish to consider an "extended" nominal design basis as part of the initial vessel design.
- Proposes consideration be given to multiple design conditions including short-time variations in temperature, pressure, or both, above the normal design temperature and design pressure in setting the design conditions for pressure vessels.

To avoid potential confusion or misunderstanding regarding information and recommendations presented later in this report, the following terms are defined here.

Elevated Temperature: Temperatures above which the allowable stress is controlled by the time-dependent properties of materials. See creep range.

Creep Range: Temperatures above which creep and stress-rupture strength of materials govern the selection of allowable stresses.

Design Property: The design property is used to establish the allowable stress. It is a time-dependent property. The design property does not imply the retirement or replacement life of the vessel.

2.0 User Experience and Citations From Published Literature

Sufficient published data and anecdotal information exist to show that pressure vessels and piping systems operating at temperatures in the creep range can be prone to premature failure. This section highlights some information contained in several published documents.

2.1 API/MPC User Survey

Extensive work was done under the sponsorship of API and MPC [2] and resulted in compilation of user experience with high temperature pressure vessels. This work concentrated only on catalytic reformer reactors fabricated of 1 Cr- ½ Mo and 1 ¼ Cr- ½ Mo: materials; however, much of the related experience is also applicable to carbon steel and higher alloy materials that operate at temperatures in the creep range. The operating conditions of the reactors in this survey are in the range 900-1000°F at 150-540 psia. The 80 reactors that were part of this survey had an average service life of 17 years (ranging between 3 and 35 years). Of these 80 reactors:

- 44 experienced some degree of cracking at welds in the pressure containing shell. Most of the cracking was in the coarse-grained heat affected zone, and creep cavitation was observed often.
- 30 had cracks at nozzle or manway attachment welds that required either weld repair or replacement of the forgings. Most of these cases involved integrally reinforced, set-in type nozzles without reinforcing pads. Only minor cracking was observed at a few nozzles that were of butt-welded design.
- 18 had cracks at main longitudinal weld seams.
- 11 had cracks at circumferential (girth) or shell-to-head welds.
- At least 17 reactors had cracks at welds attaching external support bands or plate lugs to the shell. The attachments were made with heavy fillet welds to the shell, and on the ID and OD of the skirt or platform supports. Cracking of these welds was caused by poor weld profile or weld quality, thermal gradients leading to high local thermal stresses, and stress concentrations.

The data clearly show that stress concentrations and high local thermal stresses are major factors that contribute to crack initiation in these high-temperature applications. Specific materials, chemistry control, heat treatment and welding-related factors clearly also contribute to the problems experienced, especially at locations that are remote from obvious stress concentrations and thermal gradients.

Another potential factor that was identified as a cause for cracking is notch sensitivity in the weld HAZ. Cr-Mo steels operating in the 850 - 1050°F temperature range have experienced intergranular failure due to low ductility in the presence of multiaxial stresses, similar to PWHT cracking. Research has suggested that this occurs when hardening within the grains prevents stress relaxation and grain boundary sliding must occur. This can cause cracking if the grain boundary microstructure is unfavorable for deformation. While similar to PWHT cracking, this notch sensitivity appears at lower (i.e., operating) temperature and over a longer period of time.

HAZ cracking will most likely occur in structures that have been given an improper PWHT, or in those where PWHT was not done. Stress relaxation continues at the operating temperature, and is worsened due to the addition of operating stresses to the weld residual stresses already present.

2.2 Other Published Experience and Citations

Cervenka's paper [3] did not cite any specific examples of user experience. However, it referred to many cases of poor design and maintenance practices on vessels in elevated temperature service and referred to good operating experience in many other applications. It also clearly refers to mechanical design details that are acceptable for low-temperature applications but are not reliable when vessels are operated at temperatures in the creep range.

Antalffy et. al. [4] and Baxter [5] focus on one component in a particular high-temperature application, i.e., skirt support attachments to vertical coke drums. Coke drums are an essential component in the delayed coking process. These drums are used to separate the coke from lighter hydrocarbons. A coke drum is a vertical cylindrical shell with an elliptical top head and a conical bottom. They range from 60 ft to 100 ft in height and 15 ft to 30 ft in diameter. Skirts attached near the shell-to-cone weld area are used to support the drums. Most coke drums are made of C- ½ Mo or Cr - Mo steel and are internally clad with stainless steel.

The coke drums are operated in pairs with only one drum filled at a time. While one drum is filled, the other is emptied and then prepared for the next fill cycle. Measured from fill start to fill start, a coking cycle typically lasts between 24 and 48 hours. During the cycle, the process temperature varies from 200°F to 900°F. However, the drum metal temperature operates at a maximum of about 875°F.

The 875°F maximum metal temperature is below that listed in the table included in Section 4.0 for Cr - Mo steels. However, the higher localized cyclic stresses make coke drum supports good examples of where optimum low stress intensification design details are needed even though the metal temperature is below that indicated in this table. After several years of operation in this environment, bulges and cracks tend to start at weld seams. The cracks start at stress concentration points, and the skirt-to-shell attachment is one of those typically affected.

Antalffy [4] describes the results of finite element stress analyses done on four different skirt-to-shell attachment details. These analyses clearly showed that the particular attachment detail used affects the predicted fatigue life at the attachment weld (ranging from approximately 500 to almost 11,000 cycles (See Figures 1 through 5). The paper also illustrates a "hot-box" detail (Figure 6) used to minimize the local thermal gradients at the skirt-to-shell attachment. The stress analyses done were for specific drum geometries and operating cycles. The quantitative results will vary with changes in these variables, especially the heatup and cool-down rates. However, the basic trends among the four designs will be similar and clearly show the importance of these details.

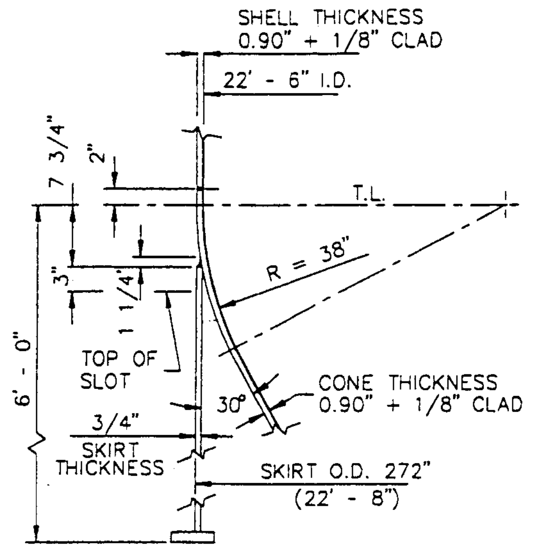

Fig. 1—Conventional Coke Drum Skirt Attachment

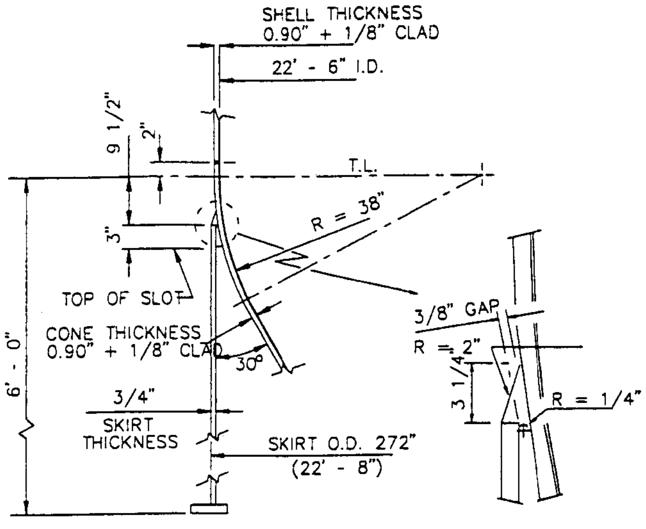

Fig. 3—Modified Coke Drum Skirt Attachment

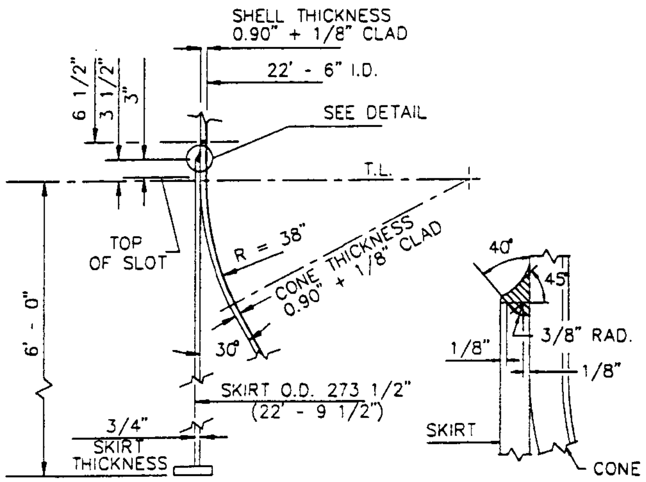

Fig. 2—Coke Drum Skirt Attachment to Outside of the Shell

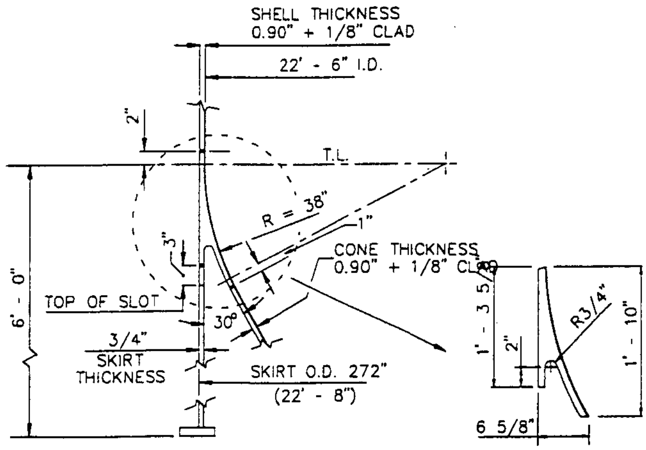

Fig. 4—Special Integral Arrangement for Coke Drum Skirt Attachment

While not related to a pressure vessel, the failure experience with a high temperature piping system reported by Buchheim et. al. [6] also merits some discussion. Here, a long-seam welded, 1 ¼ Cr- ½ Mo pipe failed after about 100,000 hours of operation at a nominal hoop stress of 6 ksi with an operating temperature in the range 970-1000°F. The pipe was NPS 36 and was ½ inch thick. Based on a detailed finite element stress analysis, it was concluded that peaking at the weld seam caused high localized bending stresses in the HAZ that resulted in an 18 inch long, through-wall crack. The peaking at the weld seam was reported to be noticeable, but was within the fabrication tolerances of ASTM A 691 [7]. Although this failure experience was in a pipe, the same concepts may be of potential concern in pressure vessels operating at temperatures in the creep range.

McLaughlin et. al. [8] reported cracking experience with a 40-year-old, carbon steel vessel with a maximum service temperature of 970°F. The cracks were at the shell-to-head weld in an area that was 1.75 inches thick with a design membrane stress of 2.6 ksi.

Most in-service pressure vessel failures initiate at welds or their associated heat affected zones. This is especially true in high-temperature applications where the weld metal properties may be affected by the high temperature and depart from those of the base material more than in lower temperature applications. Tu et. al. [9] reported on analytical work done to address several aspects of weld design in high-temperature applications, namely: weld location, weld preparation geometry, and weld design allowances. Finite element analyses and a damage mechanics approach were used in this work. The following conclusions were drawn in this work:

- The stresses in and near welds that are near geometrical discontinuities (e.g., nozzle-to-shell junctions) are increased from the stress in the surrounding area by the combination of the geometric discontinuity and the material discontinuity due to the high temperature and do not relax away. Therefore, welds should be located as far as possible away from structural discontinuities.

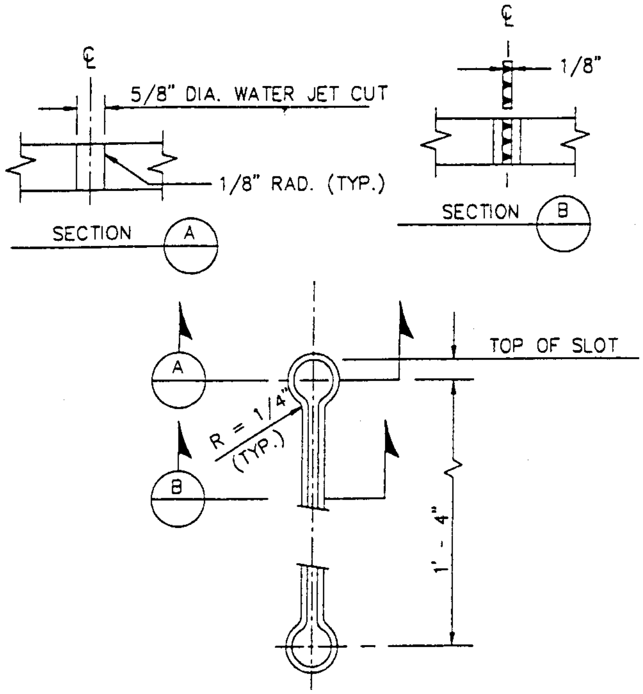

Fig. 5—Skirt Slot Details for Coke Drums

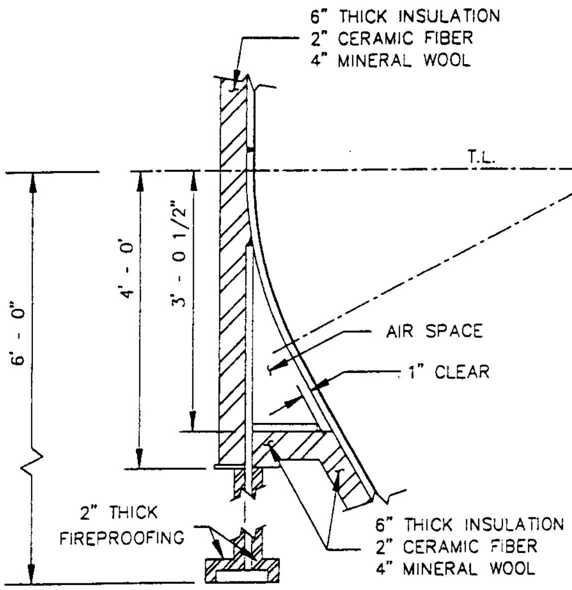

Fig. 6—Skirt Hot Box Details for Coke Drums

- Based on preliminary results, the double V-shaped weld preparation is more likely to have a high local stress intensification (when compared with a single V-shaped groove), and is thus more likely to experience creep damage that may not easily be detected by non-destructive testing. The analyses also concluded that it is preferable for the weld angle to be no larger than 45° and the root gap should be smaller than 0.25t. Considering these results, the double V-shaped weld preparation should not be given a high design priority for pressure vessels that operate in the creep range.
- Material and stress discontinuities at a weld result in stress concentrations that do not rapidly fade away with distance from the weld. Therefore, they must be treated as primary stresses. The estimated creep lifetime of weld metal is also less than that of the base material. If the pressure vessel design is based on the properties of the base material, a stress reduction factor must be used to ensure that the operating stress level in the weld metal is low enough to provide adequate design lifetime without creep damage. Section III, Division 1, Subsection NH. [21] uses a stress reduction factor approach based on uniaxial stress. However, the Subsection NH stress reduction factor was found to be relatively conservative when compared to that calculated using a procedure that considers multiaxial stresses.
- If the pressure vessel design is based on the strength of the base material (which it always is), then a weld stress reduction factor may be required in determining the required component thicknesses. This reduction factor accounts for the fact that the welds can have shorter predicted creep life than the base material in elevated temperature applications. This approach is taken in Subsection NH, but Tu's work shows that Subsection NH may be overly conservative.

2.3 PVRC Experience Survey

A user experience survey was conducted as part of the data collection effort for this report. The goals of this survey were twofold:

- Add to the number of problems that could be documented with actual design and operating data and bring these into the public domain.
- Determine what material selection, design, fabrication, and inspection requirements knowledgeable users typically apply to pressure vessels operating at temperatures in the creep range of the materials.

The respondents included two end users, one licensor, one contractor, one from a state regulatory authority, a consultant, and one from a power company. No new explicit details on failure experience were reported; however, some information was obtained from discussions with these individuals and other information provided. Several items noted are as follows:

- One end user uses 200,000 hr. creep rupture data to develop allowable stresses for reactors in high-temperature service. The only material used for their applications is 2 ¼ Cr-1 Mo.
- Greater attention must be paid to stress concentrations, local thermal gradients, and cyclic conditions for high-temperature applications. Thus, several mechanical design details that are acceptable per Division 1 should not be used.

- Material selection, material and fabrication quality control, and inspection become more important in high-temperature applications.
- Usually, more attention is paid to "heavy wall" pressure vessels (more than about 2 inches thick), and especially in high-temperature applications. Limitations are placed on the use of particular mechanical design details, and more stringent material quality control and inspection acceptance criteria are applied.
- One respondent suggested that a "risk-based" assessment approach be taken for pressure vessels that operate at temperatures in the creep range of the material. With this approach, a "probabilistic risk model" would be developed that considers service/environmental conditions, failure phenomena, damage mechanism, material characterization and endurance. What, if anything, special is done in a particular case would depend on the results of the risk assessment.

3.0 Comparison of Allowable Stress Bases

Several design codes were reviewed to identify the allowable stresses used for design temperatures that are in the creep range. The results follow.

3.1 ASME Code Section I [10] and Section VIII, Division 1

At temperatures where creep and stress rupture strength govern the allowable stresses, the maximum allowable tensile stress value is the lesser of the following:

- 100% of a conservatively estimated average stress to produce a minimum creep rate of 0.01% in 1000 hr.
- 67% of the average stress to rupture at the end of 100,000 hr.
- 80% of the minimum stress to rupture at the end of 100,000 hr.

In addition, it is noted that the stress values for high temperature are based, whenever possible, on representative properties of materials under laboratory conditions and no consideration has been given for the effects of corrosive environment, abnormal temperatures, or other design conditions.

The Div. 1 Design Criteria is inferred in UG-23(c) considering both sustained and transient loadings and their related allowable stresses, although no allowable stresses or criteria are provided for transient loadings. The manufacturer is responsible for making sure these items and all others are appropriately considered.

3.2 ASME Code Section VIII, Division 2

The main body of Section VIII, Division 2 [11] of the ASME Code does not currently permit the design of pressure vessels for temperatures that are in the creep range of the material. However, there is a long-standing Code Case (1489-2) [22] that permits the design of Division 2 pressure vessels provided the following criteria are met:

- The allowable stress tables of Division 2 may be extended to higher temperatures by using the Division 1 stress values. This also applies for materials approved for Division 2 application by a Code Case.
- The vessel or part is exempted from a fatigue analysis by the provisions of Para. AD 160.1 of Division 2, and such exemption is made a part of the User's Design Specification.

Note that consideration is being given to develop design rules to permit extending Division 2 into the creep range.

3.3 ASME B31.3 [12], Process Plant Piping

The basic allowable stress for materials other than bolting, cast iron, malleable iron, and structural grade materials is determined using the same basis as in the ASME Code Section VIII, Division 1 noted above. However, stresses due to sustained loads are permitted to exceed the nominal design allowable stresses for limited time durations.

3.4 API RP 530 [13], Calculation of Heater Tube Thickness

The minimum required thickness of a heater tube is calculated using both an elastic basis and a stress-to-rupture basis. The larger of the two calculated thicknesses is used for design. The allowable stress used for the latter case is 100% of the minimum stress to rupture in the required design life of the heater tube. The nominal tube design life is 100,000 hours, although allowable stresses for 20,000 through 200,000 hours may be determined using the Larson-Miller parameter plots included in API 530.

3.5 British Standard BS 5500 [14]

The temperatures at which time dependent properties must be considered in BS 5500 are given in Table 2.

Table 2—Temperature When Time-Dependent Properties are Considered in BS5500

Material Group	Nominal Chemistry	Temperature Limit, °F
0	C, C-Mn	750
M2	C-Mo	750
M5	3 ½ Ni	750
M6	9 Ni	750
M4	Low Alloy Mn, Cr, Mo, V	875
M7	1 Cr-½ Mo	875
M8	½ Cr -½ Mo- ¼V	915
M12	1 ½ Cr-1 Mo–V	915
M9	2 ¼ Cr-1 Mo (YS ≤ 50.75 ksi)	860
M10	5 Cr-½ Mo	860
M11	9 Cr-1 Mo	860
M9	2 ¼ Cr-1 Mo (YS > 50.75 ksi)	660
Austenitic SS	Types 321 & 347	1000
Austenitic SS	Types 304 & 316	1040

Below the creep range of the material, the allowable tensile stress is the lesser of 2/3 of the material's specified minimum yield strength or the material's room temperature specified minimum tensile strength divided by 2.35. This may be greater than that in Section VIII, Division 1, which uses 2/3 of yield strength and 1/3.5 of tensile strength at temperature, or Div. 2 which uses 2/3 of yield strength and 1/3 of tensile stress.

In the creep range of the material, the allowable tensile stress is equal to 77% of the mean stress to rupture in the specified design lifetime of the vessel (e.g., 100,000 hr.). Note that design stresses for some materials are based on performing a fitness-for-service review at 2/3 of the agreed upon design life. The tables in BS 5500 show allowable stress values for 100,000 hr , 150,000 hr., 200,000 hr., and 250,000 hr. nominal design lives; however, notes to the tables indicate that the allowable stresses are based on extended extrapolation of creep-rupture data (i.e., from shorter times and higher stresses).

Below the creep range, BS 5500 Appendix A can be used for design-by-analysis.

3.6 Dutch Stoomwesen Code

For pressure vessels at temperatures in the creep range, the maximum allowable tensile stress is the least of the following:

- The average stress to produce 1% creep in 100,000 hr.
- 67% of the average stress to rupture in 100,000 hr.
- 83% of the minimum stress to rupture in 100,000 hr.

For furnace tubes the allowable stress is the least of the following:

- 100% of the yield strength at the design temperature plus 15°C (27°F).
- The average stress to rupture in 100,000 hr. at the design temperature plus 15°C (27°F).
- 1.25 times the minimum stress to rupture in 100,000 hr. at the design temperature plus 15°C (27°F).

The Stoomwessen Code considers that the material is in the creep range when the average value of the stress to rupture in 100,000 hrs. is less than the yield strength at temperature. This is essentially the same temperature at which time-dependent properties govern under ASME rules for most materials.

The Stoomwessen Code also has a Division 2-like design criteria that can be used for the assessment of stresses when they are not covered by simple formulae. These criteria are also limited to temperatures below the creep regime.

4.0 Temperature at Which Creep Should be Considered in ASME Section VIII Div. 1

It is recommended that Division 1 clearly identify the temperature(s) above which special design considerations and details should be applied for high-temperature applications. As noted above, BS 5500 identifies an appropriate temperature for each generic material chemistry above which material creep must be considered. It is proposed that a similar approach be used for Division 1 (with some adjustments to the temperature break points) considering the most common materials used for pressure vessel construction.

For metal temperatures at and below these values (Table 3), the base case Division 1 design rules and mechanical design details may be applied (supplemented by any specific user-specified requirements). For operation above these values, special design considerations and details suitable for high-temperature operation should be considered. However, the requirements of Sections 6.0, 7.0, and 8.0 of this report should also be considered at temperatures below those shown in the table for heavy wall pressure vessels (shells greater than ~2 in. thick), regions of high local stress concentration, and severe cyclic service applications. Good engineering judgement and experience must be used to help decide when more stringent requirements should be used. Work done by others) [15, 16] investigated the effects of discontinuity stresses at elevated temperatures and should also be referred to when appropriate.

5.0 Proposed Design Basis For Multiple Design Conditions Including Short-Time Excursions

In some cases, more than one design condition must be considered in the design of pressure vessels. At present, the Code requires that the most severe coincident design pressure and design temperature that occurs in normal operation be taken as the design condition. For designs where time-dependent properties such as creep and creep-rupture govern the design, this basis is extremely conservative, as the most severe design condition may occur for only a very short total time duration during the nominal design life of the vessel.

The ASME B31.3 Code for process piping permits an increase in the allowable stress of up to 33% for short-term, high-pressure or high-temperature excursions that last for less than 10 hours per occurrence and no more than 100 hours cumulative time per year. ASME B31.3 permits an increase in allowable stress of up to 20% for such excursions if they last for less than 50 hours per occurrence and no more than 500 hours per year. This allowance is made independent of the temperature (i.e., either below or above the creep range of the material). Below the creep range, this would mean that the stress could approach 90% of the minimum specified yield strength (S_y) for ferritic materials and equal to (S_y) for austenitic materials.

This "short-term excursion" design approach has never been allowed in the ASME Boiler and Pressure Vessel Code, Section VIII. However, such a short-time allowable stress criteria, especially in the creep range, would certainly be welcome to handle short-term, high-temperature conditions. Technically, it is certainly difficult to rationalize the difference in design approach between pressure vessels and piping systems.

One question that should be addressed is what happens to a pressure vessel or vessel component that is operating in the creep regime at a stress level equal to the long-term allowable and then the temperature or the pressure is increased for a short but limited period of time, say 10 hours. We know that the material's creep rate would go up, but since the time is so short, the additional creep strain would be very small. If the time for the temperature or pressure increase is limited to 10 hours/occurrence and 100 hours/year, this would amount to 1000 hours in a 10-year nominal design life, 2000 hours in a 20-year nominal design life, and 3000 hours in a 30-year nominal design life.

One way to consider multiple design conditions that are time-dependent for a pressure vessel is by applying a use-fraction rule, like Miner's rule, such

Table 3—Section VIII Div. 1 Time-Dependent Temperature Consideration

Material Chemistry	Metal Temperature, °F
CS	650
C-½ Mo	800
1 Cr- ½ Mo	850
1 ¼ Cr-½ Mo, 2 ¼ Cr-1 Mo,	850
3 Cr-1 Mo, 5 Cr-1 Mo	800
9 Cr-1 Mo	850
Type 304	1000
Type 316, 321	1050
Type 347	1050

as that used in the ASME Section III, Division 1, Subsection NH. In ASME B31.3, similar rules are given for allowable variations in elevated temperature service. Appendix V of ASME B31.3 uses a Linear Life Fraction Rule to provide a method for evaluation of variations above the nominal design conditions when the nominal design temperature is such that the material creep-rupture properties control the design. It is proposed that consideration be given to adapting this procedure to pressure vessels.

Appendix 1 provides a procedure for the design of pressure vessels or components with multiple design conditions that may be considered for inclusion as a non-mandatory appendix to the ASME Code Section VIII, Division 1.

Here again, the user must specify the alternative design conditions to be considered, and these must be shown on Form U-1.

6.0 Creep-Fatigue Considerations

Creep-fatigue damage mechanisms can be relevant failure modes in high-temperature applications and many damage evaluation methods have been developed. However based on a review done by Viswanathan [17], none of these damage rules have proven to be superior to the others, and the best predictions agree with actual test results only within a factor of 2-3.

A simplified method for creep-fatigue damage evaluations has been proposed by Manjoine [18]. This method requires materials data which include standard tensile, creep-rupture and fatigue data over the relevant design temperature ranges, and special creep-fatigue data. However, even the application of this method is too complicated within the context of the Division 1 "design by rules" approach.

Further complicating creep-fatigue considerations is that high-temperature oxidation or other material factors can also play a significant role in causing crack initiation. In some cases, these factors dominate over the creep-fatigue damage mechanism. Since most pressure vessels are subject to steady loading conditions with only a relatively few cyclic operations, it would be prudent to provide a set of rules in Division 1 within which creep-fatigue damage mechanism would not be of concern and thus creep-fatigue evaluation may be exempt.

For the purpose of this report, and pending further development of complete rules within ASME for the evaluation of cyclic operation of pressure vessels operating in creep regime, it is proposed that an approach similar to that of Code Case 1489-1 of Division 2 be used for Division 1 vessels. Namely, the vessel or vessel components may be exempt from performing a creep-fatigue analysis if the following provisions are met:

- The total number of normal heat-up and cool-down cycles shall not exceed 100.
- The maximum rate of heat-up and cool-down shall not exceed 100°F, as measured by inlet fluid temperature.
- The requirements of Section 7.0 of this report on mechanical design details shall be met.
- The requirements of Section 8.0 of this report on weld geometries and weld inspection shall be met.
- No bimetallic welds are permitted in pressure containing parts or in welds attaching non-pressure containing parts to the pressure containing shell.

The user may also consider applicable service experience with comparable equipment operating under similar conditions to determine whether or not a creep-fatigue analysis shall be performed.

The user should also be cautioned that in high-temperature operating environments, material oxidation and other material factors also play a significant role in the material's susceptibility to cracking. These conditions must be evaluated on a case-by-case basis and appropriate safeguards included in the vessel's material selection, design, and operation.

7.0 Preferred Mechanical Design Details

Based on actual operating experience, it is clear that certain pressure vessel mechanical design details contained in Section VIII, Division 1 are more prone to damage than others when operated at temperatures in the creep range. Such damage usually manifests itself as cracks that initiate in the welds or the associated HAZ, at stress concentration points, or locations of high local thermal stresses.

An attempt could be made to use detailed stress analysis in each case to determine appropriate design temperature/time limits and geometries. However, such a detailed analysis approach is contrary to the "design by rules" philosophy used in Division 1. Furthermore, some Division 1 mechanical design details are simply inappropriate for applications that operate at temperatures in the creep range.

Therefore, it is proposed that guidelines for preferred mechanical design details in elevated temperature applications be incorporated into Division 1. These guidelines could initially be contained in a non-mandatory appendix of Division 1 until sufficient user experience and feedback is obtained to warrant them being made mandatory (with any appropriate modifications).

The authors do not claim credit for developing these preferred mechanical design details and guidelines (although we do have experience in dealing with the subject and specifying and/or designing such details). Rather, these are design concepts that have been in use by knowledgeable end-users, contractors, and pressure vessel manufacturers for many years. However, identifying them in the ASME Code as "preferred" (and later mandatory) will serve to unify the design approach used and lend further legitimacy to any increased costs that may be incurred through their use.

The preferred mechanical design details are divided into categories in the following sections and referenced to the relevant ASME Code paragraphs where appropriate.

7.1 Nozzles and Other Openings

Nozzles and other openings are stress concentration points in general and also can be areas that have high local thermal stresses. Many of the vessel cracks experienced have occurred at nozzles. Fig. UW-16.1 of Division 1 contains many nozzle details that the ASME Code accepts for Division 1 pressure vessels; however, users already restrict the use of many of these details. For example, many users already prohibit the use of all the designs that use partial-penetration welds, whatever the vessel design temperature, and some users have additional use restrictions on other nozzle design details. The prohibition of partial penetration nozzle attachment welds is even more important for high-temperature applications to minimize stress concentrations and local thermal stresses. Several other nozzle details also tend to have either higher stress concentrations or local thermal gradients as well, though they are attached by full penetration welds.

To minimize the potential for cracking at nozzles and other openings in creep range applications, the nozzle welding details should be limited to Fig. UW-16.1 (f-1), (f-2), (f-3), (f-4) for nozzles over NPS4. Fig. UW-16-1 (a), (b), (c), (d), (e), may be used for nozzles less than or equal to NPS 4, and (y-1) or (z-1) may be used for nozzles less than or equal to NPS 2. These acceptable details are illustrated in Figure 7. Note that in the case of Figs. UW-16.1 (a) and (c), the backing strip must be removed and the surface ground to a smooth contour to reduce local stress concentrations.

Additional consideration should also be given in situations where a relatively cold fluid stream is being injected into a hot vessel, or where a hot stream is being introduced into a vessel that has an internal lining (and thus has a "cold" shell). In either case, excessively high local thermal stresses may be introduced at the nozzle-to-shell junction if the nozzle details do not consider this. Figure 8 illustrates conceptual design details that may be used in each case. In all cases, the nozzle attachment to the vessel is made oversized, is then reduced in diameter to match the size of the connecting pipe, and an internal "thermal" sleeve is installed. In this way, the local thermal gradients occur in the thinner component and resulting thermal stresses are minimized.

For the hot/cold nozzle transitions, the specific design details used must be developed for each case. For example in the case of the internally-lined vessel:

- The nozzle size should be selected such the nozzle metal temperature at the shell attachment point is approximately equal to the shell metal temperature.
- The nozzle material at the shell attachment point should have the same chemistry as the shell material (typically carbon steel), and the reducer material may be a higher alloy that is suitable for the high temperature and matches that of the connecting pipe (e.g., a chrome alloy or austenitic stainless steel).
- The diameter transition should typically be made using a standard reducer rather than a pipe cap to provide increased structural flexibility and reduce the sharpness of the thermal gradient.
- For relatively small diameter nozzles ($\leq$ NPS 2) and depending on the materials and temperatures involved, it may also be possible to have the nozzle welded directly to the vessel shell with an internal extension through the vessel lining and not have a diameter transition.

7.2 Vessel Supports

Vessel support attachment points are areas of relatively high loads and can be subject to significant local thermal gradients and stresses in creep range applications. The problem becomes more significant in cyclic services (e.g., delayed coking drums) where fatigue stresses commonly cause fatigue cracking. The local thermal stress problem is also significant in cases where the vessel shell is more than about 2 in. thick, even in non-cyclic services, because there is a significant thermal gradient through the thickness of the shell.

All vessel supports (skirt, lug, or column) should be attached to the vessel shell or head using full penetration welds. Two methods for achieving this for skirt-supported vessels are illustrated in Figures 9 and 10. In the case of Figure 9, the bottom head, skirt, and bottom shell course are each butt-welded to a forged ring section. In Figure 10, the bottom head is first built up with weld metal where the skirt is to be attached, and the skirt is then butt-welded to the weld buildup. In both design concepts, clear access is provided to both make and inspect the butt welds, thus increasing the potential for achieving good overall weld quality.

Skirt-supported vessels in high-temperature applications will have a relatively high thermal gradient at the skirt-to-shell intersection. Therefore, a "hot-box" design detail (illustrated in Figure 11) should be used to minimize the local thermal gradient and resulting stress. It is especially important to minimize this thermal gradient in the case of "heavy wall" pressure vessels (i.e., wall thickness greater than approximately 2 in.).

For vessels in cyclic service (e.g., delayed coking drums), additional design refinements may be required to reduce the thermal fatigue stresses further and increase the long-term reliability of the vessel. Several design concepts that may be considered are

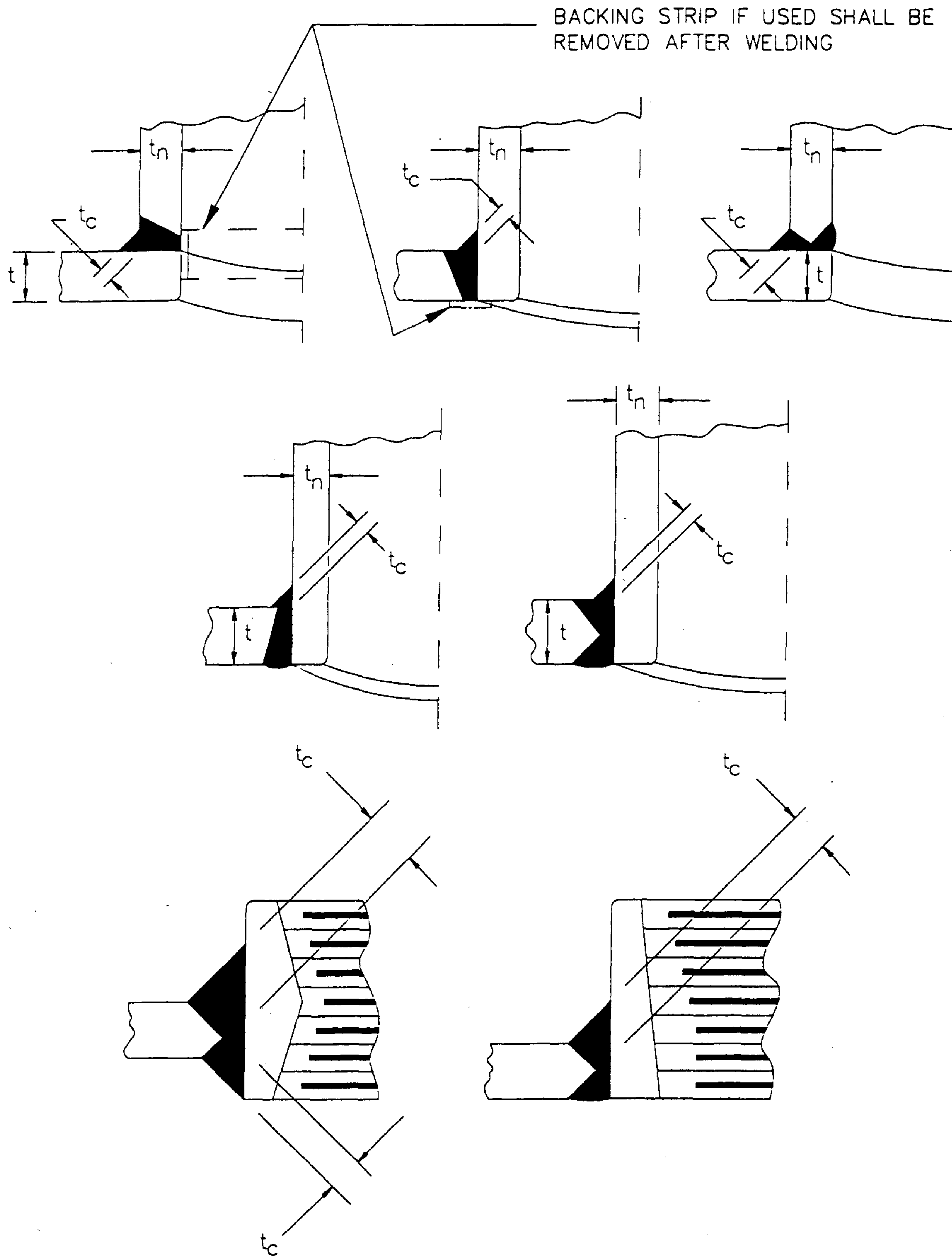

Fig. 7—Acceptable Nozzle Attachment Details*
*See Figure UW 16.1 of Reference 1

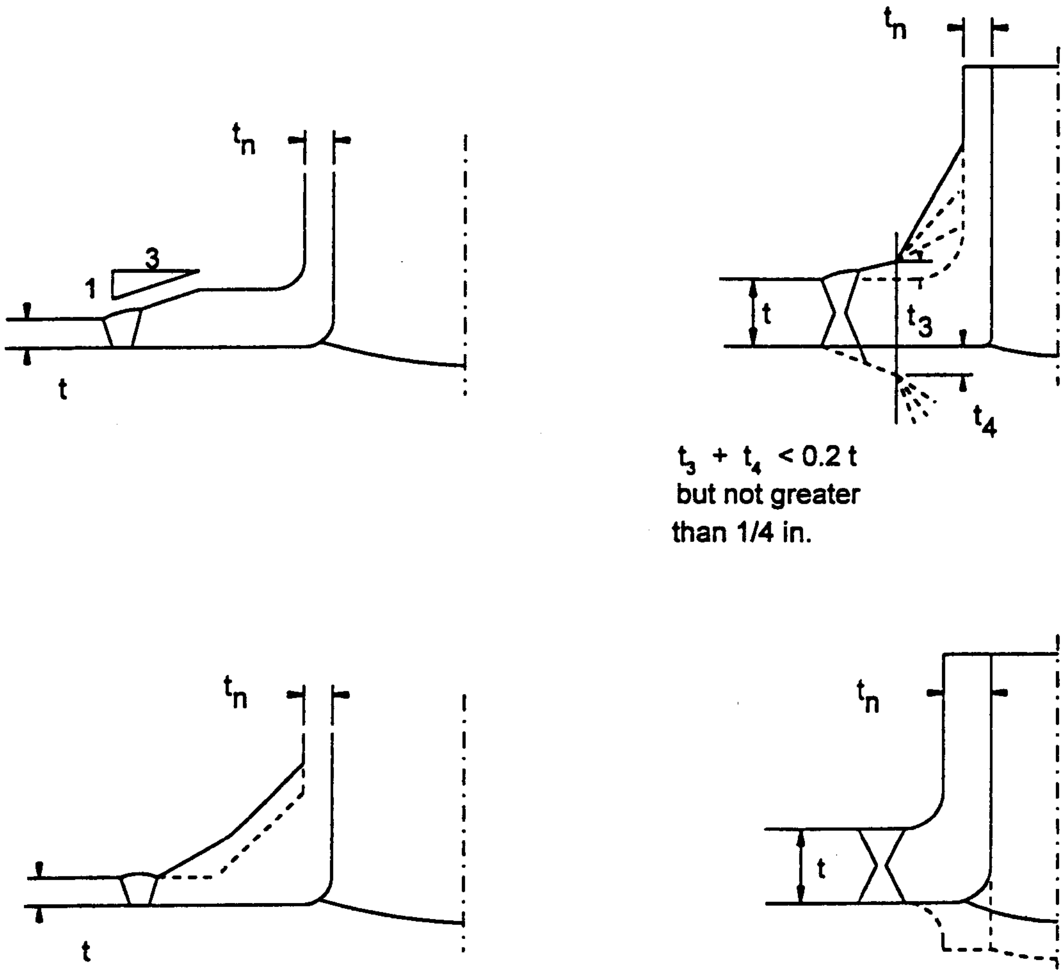

Fig. 7 (Cont'd)— Acceptable Nozzle Attachment Details*
*See Figure UW 16.1 of Reference 1

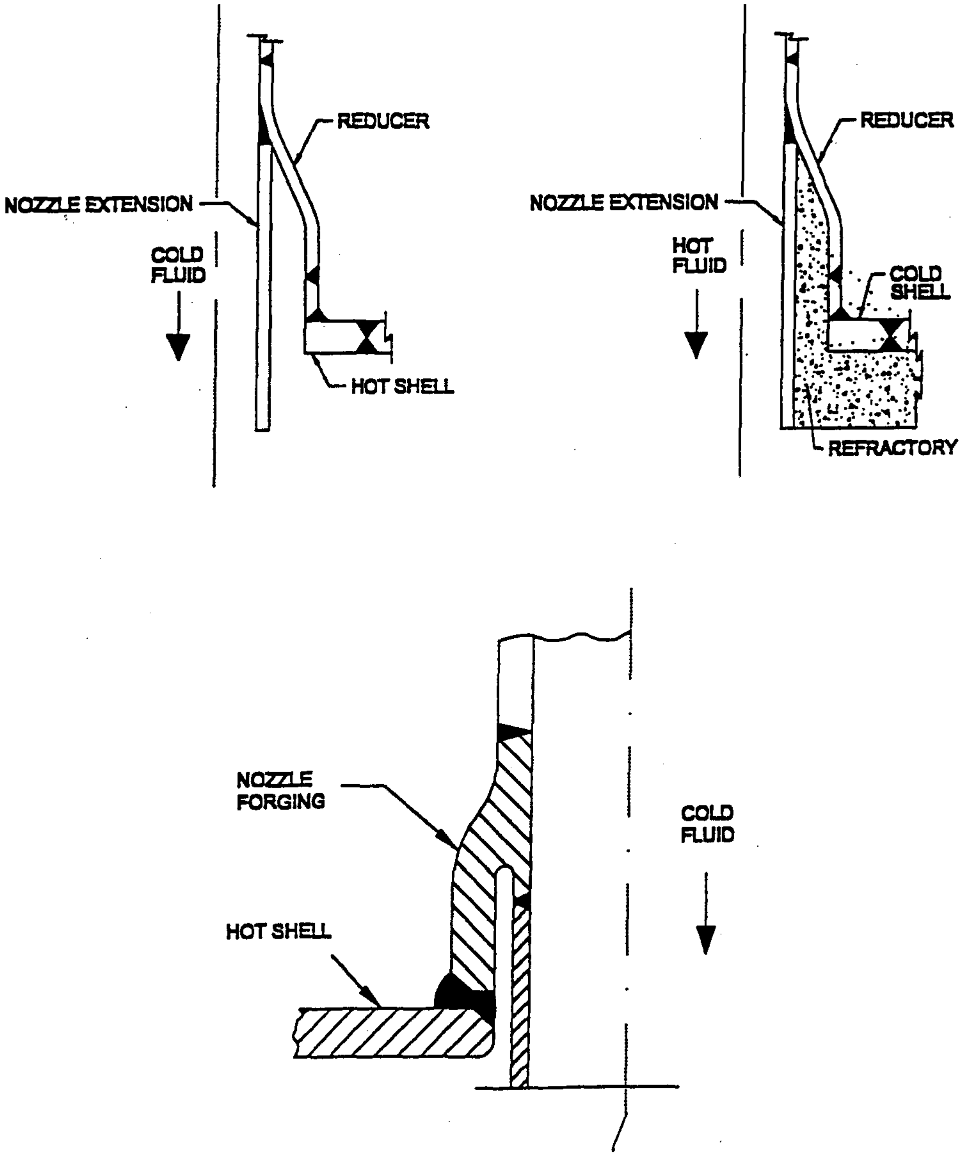

Fig. 8—Nozzle Details at Hot/Cold Temperature Transitions

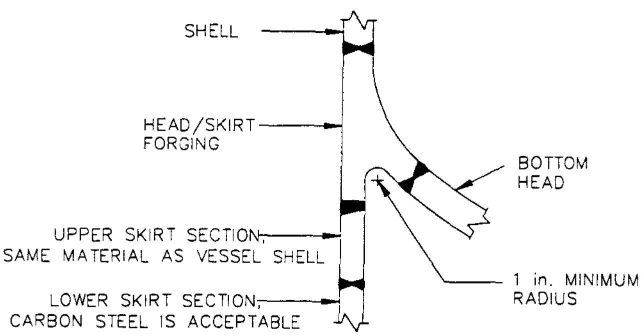

Fig. 9—Forged Ring at Skirt-to-Shell Attachment

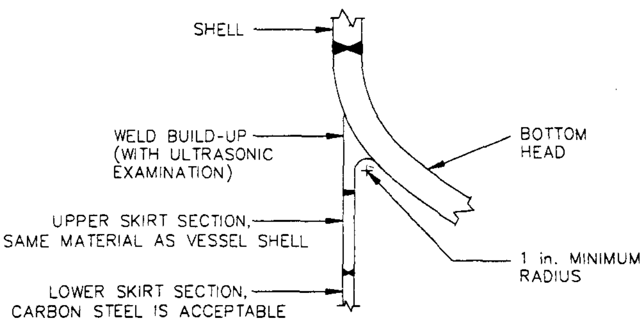

Fig. 10—Weld Buildup at Skirt-to-Shell Attachment

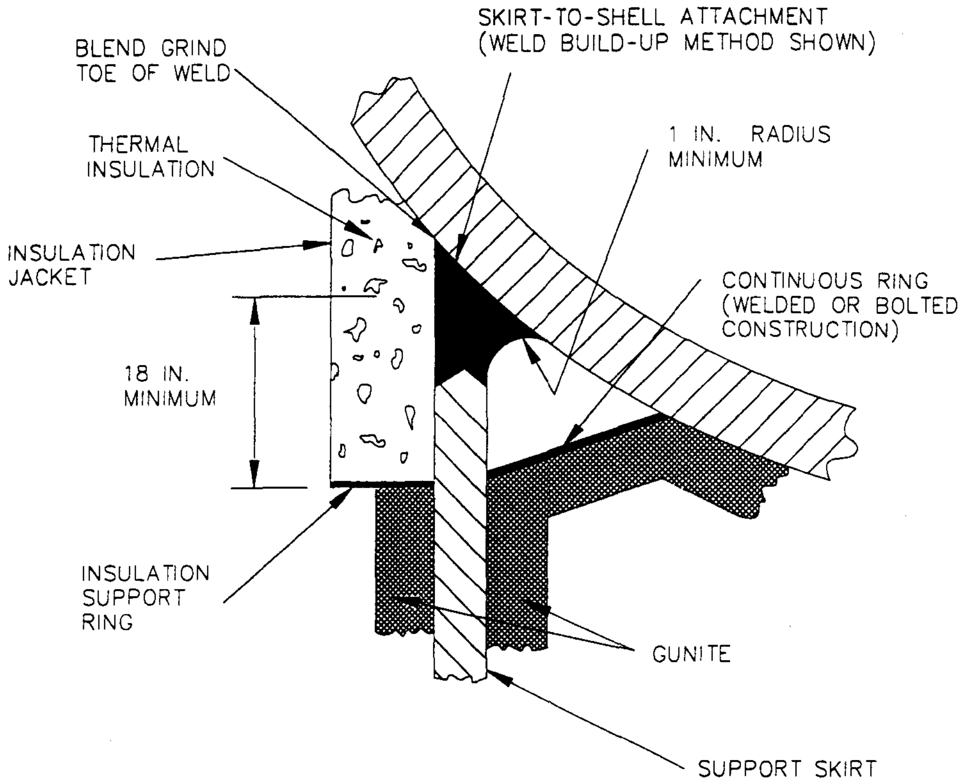

Fig. 11—"Hot-Box" At Skirt-to-Shell Junction

illustrated in Figures 3 through 6. [4] The slot detail at the upper portion of the skirt shown in Figure 5 increases the flexibility of the skirt in the junction region and reduces the local stresses. The optimum number, size, and location of the slots must be determined by detailed stress analysis. The use of the butt-welded construction shown in Figure 6 reduces the stress intensification at the skirt-to-shell junction. Here again, use of a hot-box design as illustrated in Figures 6 and 11, in addition to these design concepts, is especially advantageous.

7.3 Intermediate Heads

All of the intermediate head attachment details shown in Fig. UW-13.1 of Division 1 are prone to cracking in creep range applications. This is true even for the detail most commonly used, UW-13.1(f) [Figure 12], which is prone to cracking at the fillet weld. Therefore, the use of a butt-welded alternative using a forged ring, as illustrated in Figure 13, should be considered in high-temperature applications because it eliminates weld details that are more prone to cracking.

7.4 Main Vessel Shell and Head Seams

Table UW-12 in Division 1 identifies permissible weld joint types that may be used for weld joint categories A, B, C, and D as defined in Fig. UW-3. Joint Types (4) and (5) should not be used in high-temperature applications. These joint types use fillet welds, which are more prone to cracking in high-temperature applications. If Joint Type (2) is used, the backing strip should be removed to avoid the local stress concentration. Here again, many users already require the use of full penetration welds [i.e., Types (1), (2), or (3)] regardless of service, and require that backing strips always be removed.

Fig. UW-13.1 illustrates head attachment details. The only details that should be used for external heads in high-temperature applications are the butt-welded geometries shown in Fig. UW-13.1(d).

7.5 Attachment of Pressure Parts to Flat Plates at a Corner Joint

Fig. UW-13.2 in Division 1 illustrates acceptable details for attaching a flat plate to a pressure part to form a corner joint. Details (g) and (l) of Fig. UW-13.2 that use only fillet welds for the attachment should not be used in high-temperature applications, and only butt-welded attachments should be used.

7.6 Other Non-Pressure Attachments to the Vessel Shell and Heads

Typical non-pressure attachments to vessel shells and heads include vacuum stiffening rings, insulation support rings, internal tray and catalyst bed supports, ladder clips, and local reinforcement plates. The following guidelines should be used in high-temperature applications.

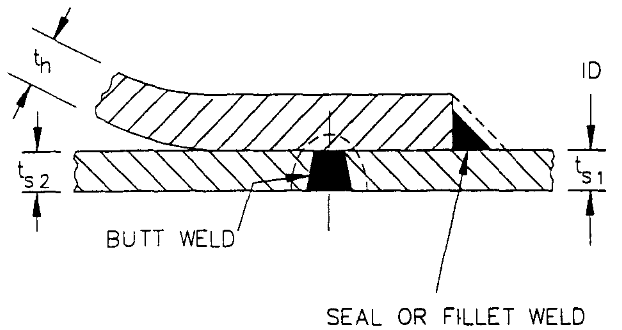

Fig. 12—Commonly Used Intermediate Head Attachment* (Not Suitable in the Creep Range)
*See Figure UW 13.1 of Reference 1

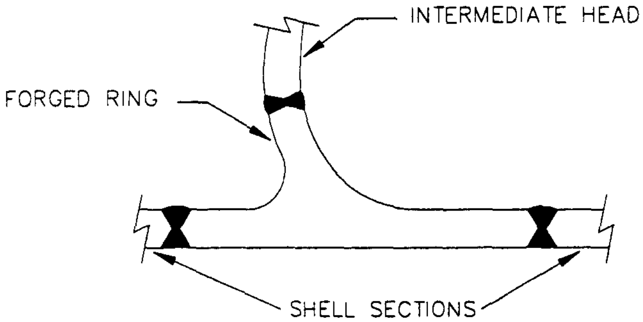

Fig. 13—Intermediate Head Attachment for High-Temperature Applications

- All items that must be attached using a corner joint (e.g., a vacuum stiffening ring) should be attached with continuous welds from both sides to reduce local stress concentrations and thermal gradients where the welds stop and restart. Therefore, the weld attachment details shown in Fig. UG-30 for attaching stiffening rings should not be used.
- In cases where the attachment supports significant loads (e.g., a reactor catalyst bed support ring) and is attached using a corner joint, the attachment to the shell should be made with a full-penetration weld. This further reduces the local stress concentration and thermal gradient.
- There will be occasions where pipe must be supported from a vessel shell and significant piping loads will be transmitted to the shell. Here again, attachments to the shell should be made using full-penetration welds. Depending on the actual loads, temperatures, and stresses involved, it may be necessary to use a thickened, butt-welded shell insert plate to achieve acceptable local stresses in the vessel shell. Use of a reinforcing plate that is fillet-welded to the shell should generally be avoided since local thermal gradients and the resulting stresses could cause cracking at the fillet welds. If a reinforcing plate is used, the fillet welds should have an elongated geometry and be ground to a smooth contour to reduce local thermal gradients and stresses, and a detailed analysis made to confirm the design.

8.0 Other Considerations

Besides the preferred mechanical design details discussed in Section 8.0, other issues relevant to pressure vessels operating at temperatures in the creep regime must also be addressed. In particular, materials, fabrication, inspection and heat treatment are known to play a significant role in the long-term reliability of pressure vessels operating at elevated temperatures. Therefore, it is prudent that additional requirements in these areas supplementing those in Division 1 are implemented. These additional requirements are discussed in the following paragraphs.

8.1 Materials

Among the materials commonly used for pressure vessels in elevated temperature services, 1 ¼ Cr- ½ Mo and 2 ¼ Cr-1 Mo steels have been studied extensively due to their susceptibility to cracking in elevated temperature applications. A multi-task experimental study was recently completed by Lundin et. al. [19] to develop recommendations for the application of 1 ¼ Cr- ½ Mo steel at temperatures in excess of 825°F. For many years, special material requirements have already been specified by users to enhance the reliability of equipment fabricated from Cr-Mo steels in elevated temperature service. These requirements are summarized below.

- For applications of 1 ¼ Cr- ½ Mo at 825°F and higher, Lundin et. al. recommended that users specify:
  - Class 1 (60/35 ksi tensile, yield strength) materials in preference to Class 2 (75/45 ksi tensile, yield strength) materials to reduce problems during fabrication, repair and service. It may be noted that there is no difference between Division 1 allowable stresses for Class 1 and Class 2 at temperatures above 900°F.
  - Carbon content in the range of 0.10-0.13% to achieve satisfactory material properties with minimum fabrication problems.
  - The composition of the material in terms of a newly defined MPC factor to assess the sensitivity to low ductility heat affected zone (HAZ) cracking.
- For 2 ¼ Cr-1 Mo steels, user specifications have typically included the following requirements to minimize temper embrittlement:
  - The limits on residual tramp elements shall be:

$$P \leq 0.012\%$$

$$Sn \leq 0.018\%$$

$$As \le 0.004\%$$

$$Sb \le 0.004\%$$

$$S \le 0.007\%$$

– For each heat of plate and forging:

$$J = (Si + Mn)(P + Sn) \times 10^4 \le 120$$

For each heat or lot of electrode:

$$X = (10P + 5Sb + 4Sn + As)/100 \le 15$$

– A sample from each heat shall be subjected to a step-cooling procedure and meet specified Charpy V-notch toughness requirements. The step-cooling procedure is as follows:

Heat to 1100°F at a rate of 100°F per hour and hold for 1 hour
Cool to 1000°F at a rate of 10°F per hour and hold for 15 hours
Cool to 975°F at a rate of 10°F per hour and hold for 24 hours
Cool to 925°F at a rate of 10°F per hour and hold for 60 hours
Cool to 875°F at a rate of 5°F per hour and hold for 100 hours
Cool to 600°F at a rate of 50°F per hour
Air cool from 600°F to ambient.

To be acceptable, the sum of the initial 40 ft-lb transition temperature plus three times the shift in the Charpy V-notch transition temperature caused by the step cooling procedure shall not exceed 100°F.

The above requirements have also been specified by some users for 1 ¼ Cr- ½ Mo material.

- All internal and external attachments, such as brackets and insulation clips, welded directly to the vessel shall be fabricated of the same material specification and grade as the vessel material.
- Weld metal chemistry shall be within the limits specified for the wrought material (except that it may be lower in carbon content.) Verification of the chemistry of production welds by weld metal sampling is required.
- The strength of the weld deposit shall match the strength of the base metal.

8.2 Fabrication

Division 1 provides general fabrication requirements covering many issues that are necessary to achieve long-term reliability of pressure vessels. In services operating in the creep range, however, poor fabrication practice has been identified as being one of the major contributors to the cracking of equipment of Cr-Mo steels [2]. In particular, poor weld details which result in high stress concentrations and local thermal stresses must be avoided. The recommendations listed below are intended to minimize these detrimental effects at welded joints.

- All welds, including attachment welds between pressure and non-pressure parts, shall be full penetration welds.
- All fillet welds shall be ground to a smooth, concave contour and the toes blended smoothly into the adjacent base metal. The grinding wheel shall be oriented perpendicular to the line of the weld. No notches or undercuts that create stress concentrations are permissible.
- Welded joints shall not be located at geometric discontinuities (e.g., thickness tapers shall be in the plates or forgings rather than in the welds).
- The maximum offset of joint alignment at circumferential and longitudinal butt welded joints shall not exceed 1/16 in. Peaking at longitudinally welded joints is not permitted.

The users may also consult Reference 19 which presents additional fabrication guidelines for successful fabrication of pressure vessels of 1 ¼ Cr- ½ Mo steel for use at elevated temperatures into the creep temperature regime.

8.3 Base Material and Weld Inspection Requirements

Division 1 contains a base level of inspection requirements for materials and welded joints used in pressure vessels. For example, Para. UW-11 requires 100% radiography of all butt-welded joints greater than 1- ½ in. thick (or thinner based on the particular materials used). Otherwise, spot radiography (or even no radiography) of butt-welded joints is permitted. Some additional requirements for magnetic particle, dye penetrant, and ultrasonic inspection are specified, but these are minimal. Requirements for the particular inspection methods used and suitable acceptance criteria for any indications found are also specified. The current Division inspection requirements are not sufficient for pressure vessels used in high temperature services.

Any failure in a high-temperature pressure vessel will most likely initiate at a weld. The likelihood that such a failure will occur increases if the welds contain defects since creep crack growth and creep-fatigue interaction effects are greater at such local discontinuities. Therefore, every effort should be made to mitigate injurious defects in the welds. By the same token, the base material used for high-temperature pressure vessels should also be of a generally higher quality, especially if it is relatively thick, to further reduce the likelihood of experiencing an in-service failure.

Some users, contractors, and vessel manufacturers already specify more than the basic Division 1 inspection requirements for all pressure vessels. They also specify still more stringent requirements for "heavy-wall" pressure vessels, regardless of design temperature. Therefore, it is recommended that similar requirements be specified for pressure vessels operating in the creep range. The following inspection requirements should be included in the overall quality assurance program for pressure vessels designed to operate at temperatures in the creep regime to enhance their reliability:

- The base materials shall be inspected as follows:
  - Plates (and plate like forgings such as forged shell rings) shall be ultrasonically examined per SA-435/SA-435M. In addition, all areas within two plate thickness from the location of a major load bearing attachment weld shall be straight-beam examined. The acceptance standard shall be in accordance with the ASME Code Section VIII, Division 1, Appendix 12.
  - All forgings (except plate-like forgings and standard flanges that meet ASME codes) shall be ultrasonically examined per SA-388/SA-388M.
  - Plate and forged ring edges, bevels, and the edges of nozzle cutout areas shall be magnetic particle examined.
- Prior to final PWHT, the welds shall be inspected as follows:
  - All welded joints shall be visually examined. No undercuts or other surface defects that create stress concentrations are permitted.
  - All butt welded joints in the shell and heads including those joining integrally reinforced nozzles to the shell or heads shall be 100 percent radiographically or ultrasonically examined in accordance with the requirements of Division 1, Par. UW-51 and Par. UW-53, respectively.
  - All groove welds, such as those between the shell and inserted nozzle neck, where radiographic examination is not practicable, shall be ultrasonically examined in accordance with Division 1, Par. UW-53. The root and final weld pass shall be examined by the magnetic particle method, or by the liquid penetrant method for non-magnetic materials.
  - All weld buildups that are used for the attachment of vessel skirt or other attachments to the shell or heads shall be ultrasonically examined prior to welding the attachment.
  - Magnetic particle examination (D-C prod only) shall also be performed as follows:
    + All fillet welds, including those which are a part of other welds, such as groove welds.
    + Repaired areas and areas where temporary attachments have been removed after grinding.
    + For double butt welded joints, the back-gouged surfaces and both sides of the completed weld.
    + Weld joint surfaces that will be covered by weld overlay.
    + Weld buildup on shell or heads.

  For non-magnetic materials, the liquid penetrant method shall be used.
- After the final PWHT, all main seam welds (both inside and outside surfaces) and all attachment welds shall be magnetic particle examined. The liquid penetrant method shall be used for non-magnetic materials.
- After hydrostatic test, all weld metal surfaces including welds connecting permanent attachments to the pressure shell shall be 100% visually examined.

8.4 Heat Treatment

The requirements for postweld heat treatment (PWHT) of welded pressure vessels are covered in Division 1. However, for pressure vessels fabricated from Cr-Mo steels, higher PWHT temperatures have been generally recognized as being necessary to improve the heat affected zone ductility. The following summarizes the heat treatment requirements which have been specified by many users for pressure vessels fabricated from these materials:

- PWHT shall be required for all pressure vessels fabricated from Cr-Mo steels.
- PWHT shall be performed at a minimum temperature of 1325 - 1375°F. The hold time shall be in accordance with Table UCS-56 of Division 1 except that the minimum hold time shall be one hour.
- Alternative PWHT at lower temperatures but for longer periods of holding time as permitted in Table UCS-56 of Division 1 shall not be used.
- A minimum preheat temperature of 300°F shall be maintained continuously during the entire welding operation.
- The hardness of the base metal, weld metal and the heat affected zone of all completed welds (after PWHT) shall not exceed 225 HB for 1 CR - ½ Mo or 1 ¼ Cr - ½ Mo materials and 235 HB for 2 ¼ Cr-1 Mo material.

9.0 Future Work

In gathering the information used in this paper and developing the recommendations contained herein, several additional items were noted that are worth additional consideration as a part of any future effort in this area. These are summarized below.

- Define appropriate design criteria and limitations for the use of specific weld details in high-temperature applications (e.g., particular butt-weld bevel details). Work that has been reported implies that a double V-shaped weld preparation should be avoided in vessels that operate in the creep range. However, this weld preparation is commonly used in heavy wall pressure vessels. The analytical work that was reported for single V-shaped and double V-shaped weld preparations should be extended to other butt-weld bevel details (e.g., J-groove, narrow gap, etc.).

- When the work to extend Division 2 into higher temperature applications is complete, determine what additional guidelines and requirements can be incorporated from that into Division 1. Included in this is to define appropriate butt-weld strength reduction factors.
- Develop simplified code rules for enveloping cyclic operating conditions that are acceptable without the need for further detailed analysis. These might take the form of rules to keep creep-fatigue damage and ratcheting to acceptable levels provided the cyclic conditions and component geometrics are within specific boundaries.
- Develop additional material selection and fabrication requirements that are applicable to austenitic stainless steels.
- Develop design requirements to be included in ASME B31.1 [20] and ASME B31.3 for piping systems that operate at temperatures in the creep range of the material. Several requirements developed for Division 1 pressure vessels are directly applicable to piping systems as well. Items such as creep-fatigue interaction, elastic follow-up, etc. will have more relevance to piping systems than for pressure vessels and would require additional work.
- Develop a model technical specification that may be used for the purchase of Division 1 vessels operating at elevated temperature.
- In the longer term, develop a risk-based assessment methodology that may be used for assessing the design requirements for pressure vessels that operate at temperatures in the creep range. Some users may wish to use a procedure such as this to help determine the need to use special design requirements for such pressure vessels and/or to employ different levels of additional requirements.

10.0 Summary

The ASME Code Section VIII, Division 1 allows pressure vessels to be designed and operated at temperatures that are in the creep regime of the materials of construction. The Code does not directly address design issues that become relevant for pressure vessels that operate at such high temperatures. Therefore, knowledgeable users typically impose additional design restrictions based on their experience, and thus incur some increased pressure vessel fabrication costs. Unfortunately, less knowledgeable users and those that cannot justify the increased initial capital cost merely follow Division 1 design rules and have an increased risk of premature vessel failure. Including design guidance, and later mandatory requirements, within Division 1 for higher temperature pressure vessels will help make the increased costs easier to justify and ultimately lead to generally more reliable pressure vessels.

The current exclusion of additional design requirements from Division 1 for pressure vessels that operate at temperatures in the creep range of the material is non-conservative. Conversely, Division 1 is overly conservative in the manner in which it handles multiple operating cases that might include short-term elevated temperature excursions. Based on current rules, short excursions must be treated as a long term design case and the vessel designed accordingly. This results in vessels that are thicker, and thus more expensive, than they need to be.

This paper contains proposals for consideration by ASME that address both issues. Restrictions on the use of particular Section VIII Division 1 design details are proposed. In addition, two alternative allowable stress bases are suggested. The first stress basis permits the user to employ lower than the normal allowable stresses. These lower allowable stresses are based on time-dependent material properties longer than the 100,000 hours currently used in Division 1. The second stress basis permits the user to consider multiple design conditions and short-term temperature excursions in a manner similar to that used in ASME B31.3, and thus not unnecessarily increase vessel component thicknesses for such short occurrences. In all cases, these proposals permit continued use of the "design-by-rules" philosophy of Division 1 and the Division 1 allowable stress tables, and can be adapted to future changes in Division 1. Figure 14 provides a flow chart to assist in applying recommendations contained in this paper.

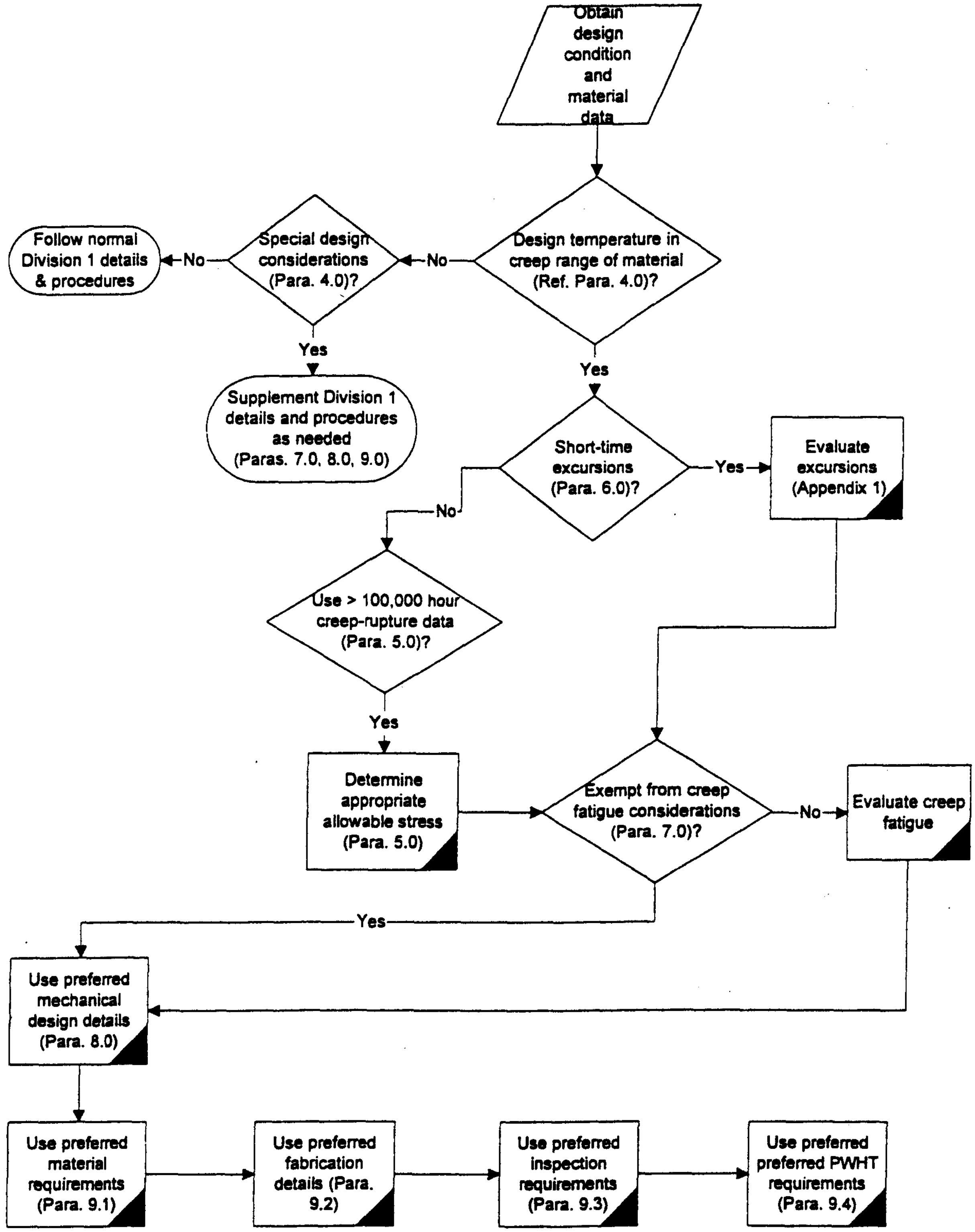

Fig. 14—Vessel Design Flow Chart

11.0 Acknowledgment

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12.0 References

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Appendix 1
Design Considerations for Pressure Vessels Operating in the Creep Regime with Multiple Design Conditions

Application

This appendix covers application of a Linear Life Fraction Rule, which provides a method for evaluating multiple design conditions, including short-term variations in temperature or pressure, or both, above the nominal design temperature and design pressure of the vessel or vessel component. This method only applies when the nominal design temperature is such that the allowable stress is based on the time-dependent creep-rupture properties of the material. This appendix is required only when specified by the owner.

Life Fraction analysis only addresses the gross strength of the component and does not consider local stress or strain effects. It is the responsibility of the designer to provide construction details suitable for elevated temperature design.

Definitions

The following terms are defined for the purposes of this appendix:

Nominal Design Pressure
: The maximum long-term pressure for which the component is being designed.

Maximum Allowable Working Pressure
: The maximum allowable pressure for continuous operation for the component at the nominal design temperature.

Nominal Design Temperature
: The maximum long-term temperature for which the component is being designed. This temperature must be high enough such that time-dependent, creep-rupture properties of the material govern the allowable stress.

Operating Condition
: Any condition of pressure and temperature under which the "design conditions" are not exceeded.

Excursion Condition
: Any condition under which the pressure, temperature, or both exceed the design conditions.

Service Condition
: Any operating condition or excursion condition. All service conditions must be at temperatures such that the allowable stress is based on the time-dependent, creep-rupture properties of the material.

Duration
: (a) The extent of any service condition, hours/year.
: (b) The cumulative extent of all repetitions of a given service condition during the service life, hours.

Service Life
: The life assigned to the vessel for design purposes, years

Design Basis

The Life Fraction analysis shall be done in accordance with one of the following design basis options as selected by the owner:

A. All service conditions in the creep range and their duration are included.
B. To simplify the analysis, less severe service conditions need not be individually evaluated if their durations are included with the duration of a more severe service condition.

Design Criteria

The following design criteria shall also be followed.

1. All existing temperature limits in the Code for the materials of construction shall be met.
2. Only carbon steels, low and intermediate alloy steels, and austenitic stainless steels are included.
3. This procedure shall only be used if the vessel is considered to be in non-cyclic service. Creep-fatigue interaction effects shall be considered if the exemption criteria stated in Para. 7.0 of this report are not satisfied.

Procedure

The cumulative effect of all service conditions during the service life of the vessel shall be determined by the Linear Life Fraction Rule in accordance with the procedure below. Calculations shall be done for each section of the vessel that has a different design pressure and/or design temperature.

Calculations for Each Service Condition, i

The following steps shall be repeated for each service condition considered:

1. Pressure-based Equivalent Stress
   Compute a pressure-based equivalent stress, $S_{pi,}$ using Eq. 1.
   $$S_{pi} = S_{dt}P_{oi}/P_d) \qquad (1)$$
   Where:
   S_{pi} = Pressure-based equivalent stress for service condition i, psi
   S_{dt} = Allowable stress at the nominal design temperature, psi
   P_{oi} = Operating pressure during service condition i, psig
   P_d = Maximum allowable working pressure at the nominal design temperature, psig
2. Compute the maximum longitudinal stress, S_{Li}, during service condition i.
3. The equivalent stress, S_i, for use in Step 4 is the larger of S_{pi} or S_{Li}.
4. Effective Design Temperature
   From the allowable stress table in Section II Part D, determine the effective design temperature, T_{Ei}, corresponding to the equivalent stress, S_i. Use linear interpolation if necessary.
5. Larson-Miller Parameter
   Compute the Larson-Miller Parameter, LMP_i, using Eq. 2.
   $$LMP_i = (C + 5)(T_{Ei} + 460) \qquad (2)$$
   Where:
   C = 20 (carbon, low, and intermediate alloy steels)
   C = 15 (austenitic stainless steels)
   T_{Ei} = Effective design temperature at service condition i, °F
6. Rupture Life
   Compute the allowable rupture life, $t_{ri,}$ using Eq. 3.
   $$t_{ri} = 10^a \qquad (3)$$
   Where:
   $a = [LMP_i/(T_{oi} + 460)] - C$
   t_{ri} = Allowable rupture life associated with a given service condition i, hr.
   T_{oi} = Operating temperature during service condition i, °F

Cumulative Creep-Rupture Usage Factor

The cumulative usage factor, u, is the summation of individual usage factor $u_i = t_i/t_{ri}$ for all service conditions considered. Compute this using Eq. 4.

$$U = \sum t_i/t_{ri} \qquad (4)$$

Where:

i = As a subscript: 1 for the prevalent operating condition; i = 2,3, etc. for each of the other service conditions considered.
t_i = Duration associated with any service condition, i.

If $u \leq 1.0$, the usage is acceptable for all service conditions considered. If $u > 1.0$, then the usage factor is unacceptable and the designer shall increase the design conditions until the usage factor is acceptable. This could involve increasing the Nominal Design Temperature or reducing the number and/or severity of the excursions until the usage factor is acceptable. If neither of these options are feasible, then the actual thickness of the part must be increased, thereby increasing its MAWP and reducing the cumulative usage factor.

The following examples illustrate the application of this procedure. Note that these examples were simplified by neglecting the longitudinal stress, S_{Li}. This is based on the tacit assumption that S_{Li} will be less than S_{Li} in most cases. However, S_{pi} must be calculated in all actual applications of this procedure since situations will occur where S_{Li} governs (e.g., the lower sections of tall vessels exposed to high wind or seismic loads).

Example 1

Material:	SA-387-Gr22-1		
Nominal Design Pressure, P_d, psig:	425		
Nominal Design Temperature, T_d, °F:	1050		
Code Allowable Stress, S_{dt}, psi:	5700		
Service Life, n, years:	25		
Operating Conditions:	(1)	(2)	(3)
Temperature, T_{oi}, °F:	1000	1200	1025
Pressure, P_{oi}, psig:	400	125	450
Duration, t_i/n, hrs:	8000	200	400
Pressure-Based Equivalent Stress, $S_{pi} = S_{dt}\,(P_{oi}/P_d)$, psi:	5365	1676	6035
Effective Design Temperature, T_{Ei}, °F:	1059	1186	1043
$LMP_i = (C + 5)\,(T_{Ei} + 460)$	37971	41154	37568
$a = [LMP_i / (T_{oi} + 460)] - C$	6.007	4.792	5.298
Allowable Rupture Life, $t_{ri} = 10^a$, hrs	1016839	61918	198685
$u_i = t_i/t_{ri}$	0.197	0.081	0.050
$u = \Sigma t_i/t_{ri}$	0.197	0.277	0.328

In this example, the cumulative usage factor is well below 1.0, indicating that the design is too conservative. The designer may reduce the nominal design temperature, thereby increasing the allowable stress (and reducing the component thickness) to achieve a more cost-effective design.

Example 2

Material:	SA-387-Gr 22-1		
Nominal Design Pressure, P_d, psig:	425		
Nominal Design Temperature, T_d, °F:	1025		
Code Allowable Stress, $S_{dt,}$, psi:	6850		
Service Life, n, years:	25		
Operating Conditions:	(1)	(2)	(3)
Temperature, T_{oi}, °F	1000	1200	1025
Pressure, P_{oi}, psig	400	125	450
Duration, t_i, hrs	8000	200	400
Pressure-Based Equivalent Stress $S_{pi} = S_{dt}\,(P_{oi}/P_d)$, psi	6447	2015	7253
Effective Design Temperature, T_{Ei}, °F	1030	1169	1016
$LMP_i = (C + 5)\,(T_{Ei} + 460)$	37259	40732	36906
$a = [LMP_i / (T_{oi} + 460)] - C$	5.520	4.537	4.853
Allowable Rupture Life, $t_{ri} = 10^a$, hrs	330773	34444	71209
$u_i = t_i/t_{ri}$	0.605	0.145	0.140
$u = \Sigma t_i/t_{ri}$	0.605	0.750	0.890

In this example, the service conditions are the same as those in Example 1, but the nominal design temperature is reduced by 25°F. This results in a 20% increase in the basic allowable stress and hence the component would be 20% thinner than that required for the original case. The cumulative usage factor in this case is still below 1.0, which indicates that the component is satisfactory for the service conditions.

Example 3

Material:	SA-387-Gr 22-1		
Nominal Design Pressure P_d, psig:	425		
Nominal Design Temperature T_d, °F:	1000		
Code Allowable Stress S_{dt}, psi:	8000		
Service Life, n, years:	25		
Operating Conditions:	(1)	(2)	(3)
Temperature, T_{oi}, °F	975	1200	1025
Pressure, P_{oi}, psig	400	125	450
Duration, t_i/n, hrs./year	8000	200	400
Pressure-Based Equivalent Stress, $S_{pi} = S_{dt}(P_{oi}/P_d)$, psi	7529	2353	8471
Effective Design Temperature, T_{Ei}, °F	1010	1152	992
$LMP_i = (C + 5)(T_{Ei} + 460)$	36756	40309	36290
$a = [LMP_i / (T_{oi} + 460)] - C$	5.614	4.282	4.438
Allowable Rupture Life, $t_{ri} = 10^a$, hrs	410928	19161	27394
$u_i = t_i/t_{ri}$	0.487	0.261	0.365
$u = \Sigma\, t_i/t_{ri}$	0.487	0.748	1.113

In this example, the long-term operating temperature is 25°F less than those in Examples 1 and 2. However, a 25°F reduction in the nominal design temperature from that in Example 2 results in a cumulative usage factor greater than 1.0; therefore, the design is not acceptable. The designer must increase either the nominal design temperature or the component thickness and repeat the calculations to ensure that the cumulative usage factor does not exceed 1.

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