Assessment of high temperature bolted flange joints gasketed with DeltaV-Seal™ 800HT for pilot plant by applying EN 1591-1 code (Part 1)

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PIPEOTECH

Contents

1. INTRODUCTION

For an established process system, choosing temperature and pressure (T/P) ratings of equipment is a relatively straightforward exercise. However, choosing maximum operating pressure and temperature combination for major pieces of equipment in a pilot plant can be much more challenging. This has become more and more common during the last decades as many of the new 'green' process designs are pushing the T/P conditions to even higher levels mainly for improved process chemistry, energy input/output balance and equipment utilization efficiencies.

Choosing the correct combination, or design point, is one of the key decision processes in any pilot-plant project, meriting significant analysis and thought. Choosing a design point that is too low can render a plant useless, but even in a pilot plant, it may not be in the best interest of the project merely to guess high. Further, choosing a point that is inappropriately high can increase the cost and delivery schedule of the pilot plant by a significant fraction without providing any meaningful benefit. An inappropriately high point may also hamper operation of the full plant by unnecessarily limiting the types of closures, equipment, materials, valves, and instrumentation that can be used, resulting in increased downtime or a reduction in the ability of the plant to deliver the desired measurements, products, and data in a timely fashion.

A breakpoint analysis for the proposed materials of construction should be carried out as a preliminary design exercise, to establish an appropriate design point, also refer to Section 6. By identifying natural limits for T/P for the required materials of construction, natural breakpoints for the design and T/P combinations for various pieces of equipment can be determined. It goes without saying that crossing a natural breakpoint can come with a significant cost, schedule, or operability penalty and should not be taken lightly. This analysis should obviously include both time-independent and time-dependant materials properties and how these impact on integrity and reliability of materials and equipment throughout the entire pilot plant process system.

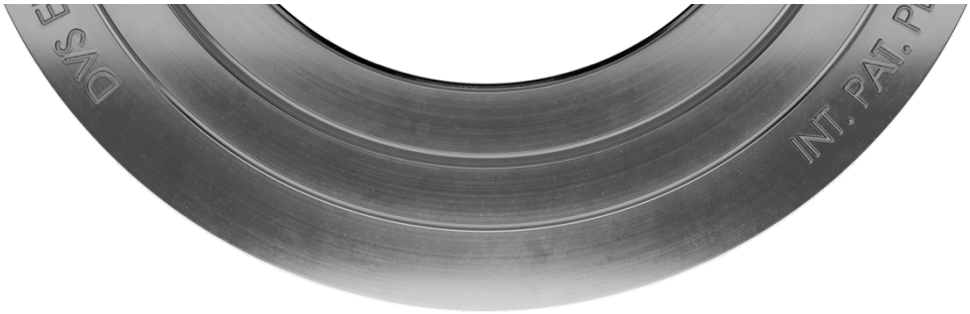
In the present investigation, Pipeotech has applied the EN 1591-1 code using DeltaV-Seal™ 800HT on a pilot plant case where the piping design point was decided upfront. The task has specifically focussed on proving mechanical/material structural stability and leak tightness class for bolted flange

joints (BFJ) by applying the optimization algorithms of the code on all system components (pipe, flange, gasket, and lubricant). This has been done simultaneously for the assembly, testing, and service load cases (LC) on BFJ's gasketed with Pipeotech's new high temperature gasket DeltaV-Seal™ 800HT for one selected size/pressure class (3"/#300) where the internal process operating temperatures were as high as maximum 850°C. Since these calculation temperatures are in the creep range, pressure vessel and piping code requirements for creep design had to be considered.

It should be noted that the EN 1591-1 model does not take account of creep of flanges and bolts and assumes that these materials have been selected to avoid excessive creep. However, creep and relaxation of the gasket are included in the mechanical model and the gasket creep under compression is approximated by the creep factor PQR which is the ratio of the residual and the original gasket surface pressure at load conditions at a defined flange stiffness. Also included in this model is the additional deflection (ΔeGc) of the gasket due to creep and this phenomenon is in the current revision (2013) directly handled by the treatment of gasket deflection including the viscous contribution (in previous revision this was handled via the PQR – value).

As will be demonstrated in this paper, proving leak tightness and mechanical/structural stability of gasketed BFJ can be quite challenging and standard equipment lifetimes postulated by piping/vessel design codes should not be taken for given for BFJ's operating in the creep range, especially not so for pilot plants. Here it is worth mentioning that the ASME VIII, Division 1, App.2 gasket calculation code is not recommended since this code does not consider thermal expansion of the BFJ components which can render an otherwise mechanically proven BFJ susceptible to either leakage or bolt overload at high temperatures in cases where there are dissimilar materials included in the BFJ.

The current Part 1 paper is outlining the general principles for creep design of BFJ and focusses on a set of specific plant design parameter values. In a subsequent paper, Pipeotech will consider a broader range of design parameter values including BFJ pressure classes and sizes.



2. DESIGN CONDITIONS

2.1 Flange materials

The following general design conditions apply to the pilot plant case:

- Flange material was alloy 800H (ASTM B564 UNS N08810)
- Flange design according to B16.5 WNRF Cl.300 with a design temperature of 880°C at 0.5 bar(g)
- Size range varies from NPS ½" to 6" on the piping
- Piping system design code was ASME B31.3 2020 with the piping systems categorized as Normal Fluid Service & Elevated Temperature Fluid Service

2.2 Process chemistry and operating conditions

The piping systems were carrying different gas mixtures. Typical composition and operating temperature were given as:

- Gas mixtures containing (wt.%): H_2 0.05%, CO 0.5%, CO_2 61.4%, H_2O 38.1%
- Operating pressure: 0.2 bar(g)
- Operating temperature: 850°C

Other details of the process chemistry were not known to Pipeotech at the outset of this work. However, the stated gas mixture chemistry/components are typically occurring in steam reforming of natural gas (steam methane reforming (SMR)). In hydrogen production processes, the produced syngas is normally undergoing the water-gas shift reaction in a separate reactor (WGS) producing H_2 (blue hydrogen after purification) and CO_2 for capture and storage in accordance with the following red-ox reaction:

$$CO + H_2O \Leftrightarrow CO_2 + H_2$$

At low to moderate pressures, the equilibrium state of the WGS reaction is approximately described by the relation:

$(Y_{CO2} \times Y_{H2})/(Y_{CO} \times Y_{H2O}) = K_{eq}(T) = 0.0247 *\exp[4020/T(K)]$ where

- T(K) is the WGS reactor temperature
- K_{eq} is the reaction equilibrium constant
- Y_i is the mole fraction of species in the reactor contents at equilibrium

At the selected design point (850°C operation), the $K_{eq}(T)$ – value is equal to approximately 0.9 which means that the gas mixture composition should be at close to equilibrium since the known equilibrium temperature for the WGS reaction is 823°C. However, the stated gas mixture composition gives a $K_{eq}(T)$ –

value equal to approx. 0.16 which means that the stated gas mixture composition is quite far from equilibrium and therefore will likely change with time until an equilibrium composition is achieved.

Operational/chemistry changes during service are obviously a fundamental aspect of a pilot plant design process and associated impact on exposed materials should be considered.

2.3 Design code

Piping design code was stated as ASME B31.3 (7) with piping systems as Normal Fluid Service & Elevated Temperature Fluid Service.

2.4 Materials selection

Pipeotech has reviewed the process and operational design conditions and assessed the effect on materials selection for the different components. This work is described in the following sections.

2.4.1 Flanges

As mentioned, the flange material was selected as alloy 800H forgings (ASTM B564 UNS N08810) which is an austenitic nickel-chromium-iron alloy characterized by:

- High creep strength
- Very good resistance to oxidation
- Good resistance to combustion gases
- Very good resistance to carburization
- Good microstructural stability at high temperatures

As will be discussed in subsequent section of the paper, the single most important mechanical/material damage mechanism at 850°C is creep (not considering chemical/environmental interactions with exposed materials). This is applicable to any metallic material and every material has specific creep curves (creep strain/creep rupture as a function of time, applied stress and temperature) that need to be considered and the associated design lifetime be addressed for all components and included in the system mechanical stability and leak tightness calculations (EN 1591-1) including definition of necessary safety factors and limit state conditions, see below.

These high temperature creep damage factors should also form the basis for inspection and maintenance planning for the pilot plant.

2.3.1.1 Creep curves for alloy 800H (UNS N08810)

An extensive creep investigation of alloy 800H was published by the Japan Atomic Energy Research Institute during 1998 (3) at temperatures ranging from 700°C to 900°C. True creep strain (%) was measured at constant stress at 850°C and the results are plotted below in Figure 1 for the following applied stresses: (TTF = Time to Failure, i.e., creep rupture)

- 38 MPa (TTF = 4144.5 hours)
- 50 MPa (TTF = 645.8 hours)
- 65 MPa (TTF = 117.8 hours)

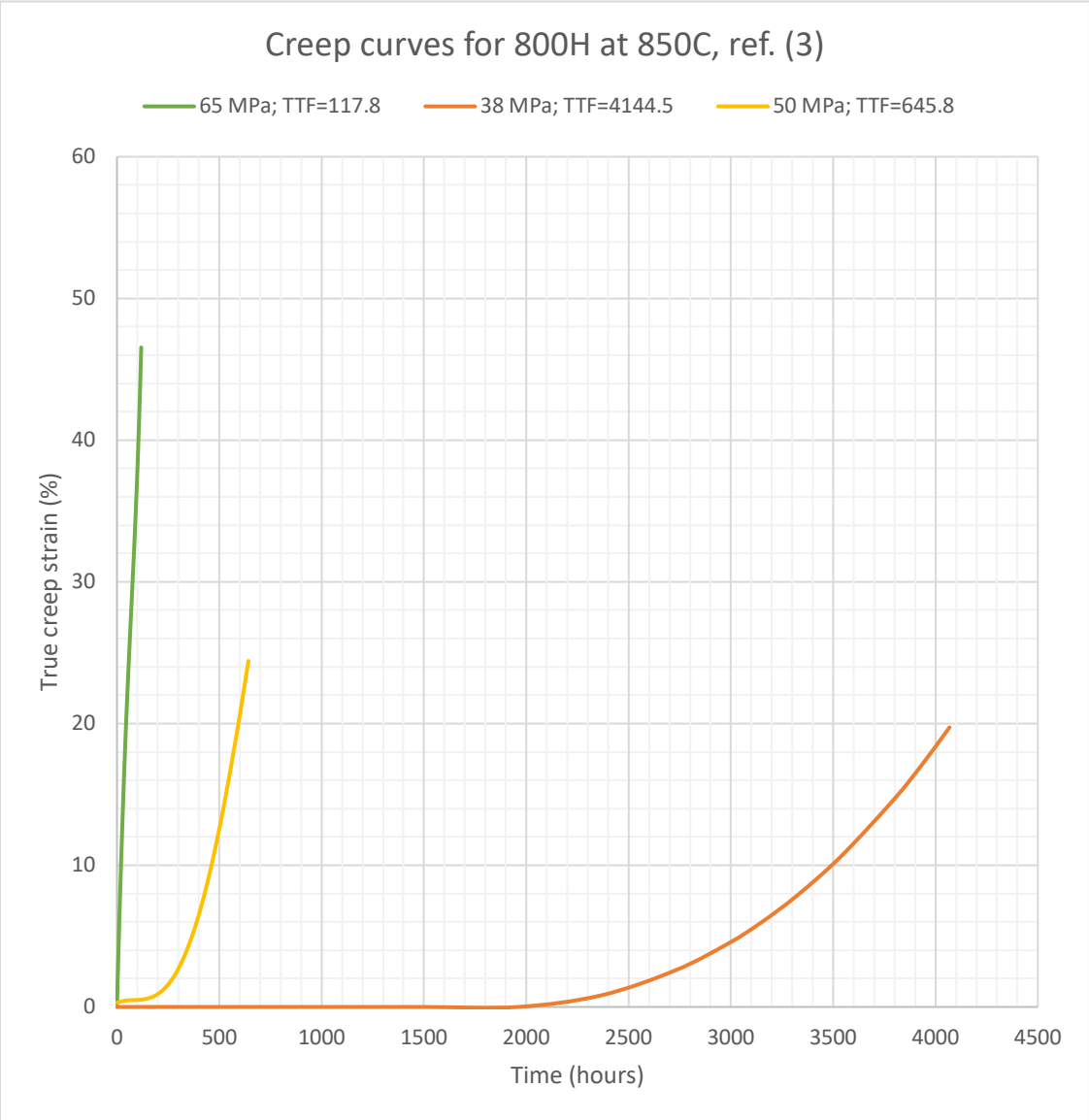


FIGURE 1.

As can be expected, the lower the applied stress the longer TTF. For example, the result of this investigation shows that an applied constant stress of 38 MPa gave a TTF-value of 4144.5 hours with a total true creep strain of 20% at the point of failure. The same graph shows that a defined limit state condition of 10% total creep strain is reached after 3500 hours.

The same set of test data can be expressed as time to reach a defined limit state. In Figure 2 below the time to reach a limit state of 1% true creep strain has been plotted against applied stress and shows this limit state is reached after 100000 hours if the applied stress is 30 MPa.

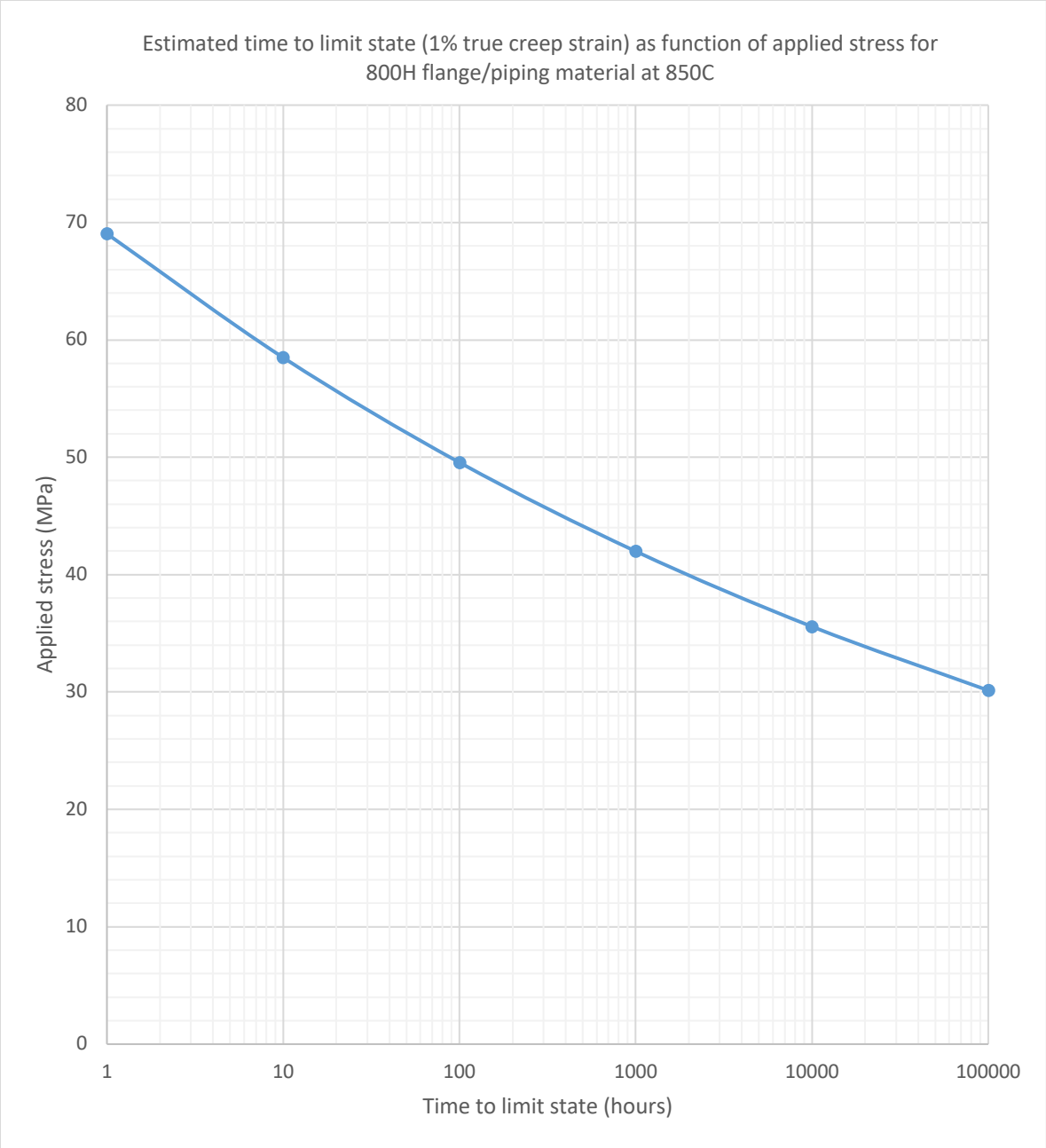


FIGURE 2.

These creep data will be utilized for definition of limit states and applied safety factors for the flanges required for the EN 1591-1 modelling with the main task to derive the minimum required stress to be applied to the BFJ which provides a reasonable duration before the limit state condition is reached, see subsequent paragraphs of this paper.

2.4.2 Piping materials

At the outset of this investigation, the pipework material was not given. Pipeotech therefore decided to apply the same material grade for the pipes as has been applied for the flanges, namely alloy 800H. This is motivated by the need to minimize as much as possible dissimilar material issues like differences in thermal expansion which

should be minimized at these high temperatures.

The corresponding seamless/welded pipe material is standardized in ASTM B407 (UNS N08810).

Therefore, the same creep data as given in Section 2.3.1.1 is also applicable to the piping part of the BFJ's and was used for definition of limit states and applied safety factors for the pipes required for the EN 1591-1 modelling, see subsequent paragraphs of this paper.

2.4.3 Bolting materials

Generally, 850°C is a very high temperature when it comes to designing with threaded fasteners, preloaded assemblies, etc. For example, the ASME code (ASME II Part D) only allows the following three bolting materials to be used at a maximum temperature of 899°C:

- SB-572 UNS N06002
- SB 408 UNS N08800
- SB-572 UNS R30556 (HAYNES® 556® Alloy)

with the following allowable design stresses at 899°C respectively:

- 8.14 MPa
- 6.70 MPa
- 12.3 MPa

It should be noted that all tabulated data from codes and standards only serve as a guide to provide some basic understanding in how to deal with a complex design problem like the current one. Complex in the sense that in this case time dependent properties must be considered to avoid either excessive creep deformation or failure from stress rupture as described in the previous section (limit state design) for the flange and piping material and this section for the bolting material. In addition, the environment may play a significant role because of potential corrosion concerns at this service temperature (850°C).

In the current pilot plant investigation, Pipeotech decided to apply R30556 bolting material to provide maximum possible mechanical strength and stability to the BFJ's. Furthermore, an extensive set of high temperature creep data is available from Haynes web site for their 556 alloy for bolting.

Two sets of data were considered:

1. Limit state defined as 0.5% creep strain
2. Limit state defined as 1% creep strain

Note that in the following all stated creep strains are assumed to be permanent.

For the 1st set of data, the applied stress was plotted against operation temperature at constant TTF; 10 hours, 1000 hours, and 1000 hours respectively, hence the three curves shown in Figure 3 below.

From this set of data, the maximum allowable stress at an operation temperature of 850°C could be derived with the results indicated on the graph:

- Allowable stress = 70.1 MPa to obtain 10 hours to reach a limit state of 0.5% creep strain

- Allowable stress = 49.0 MPa to obtain 100 hours to reach a limit state of 0.5% creep strain
- Allowable stress = 35.4 MPa to obtain 1000 hours to reach a limit state of 0.5% creep strain

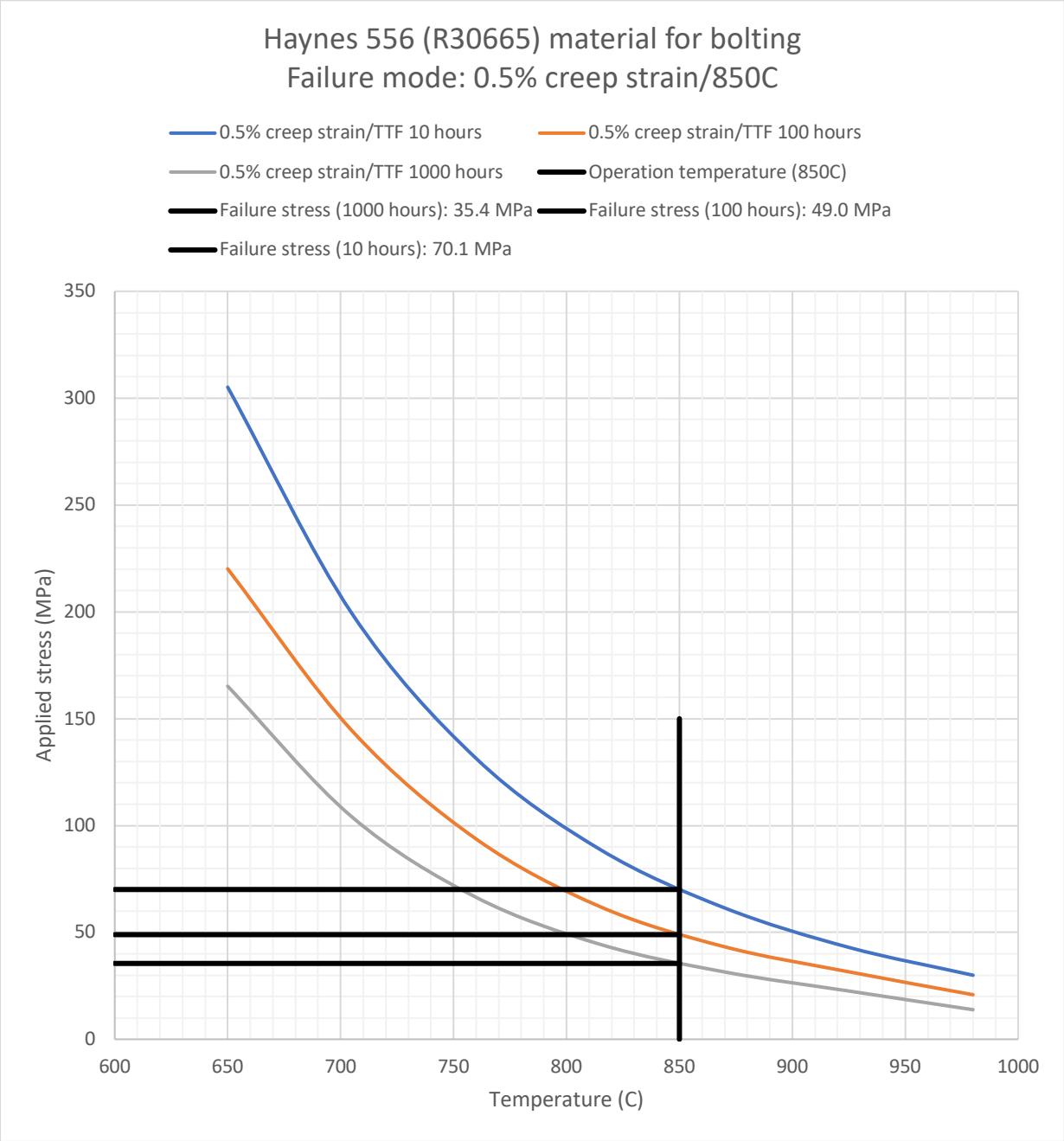


FIGURE 3

Likewise, for the 2nd set of data, the applied stress was plotted against operation temperature at constant TTF; 10 hours, 1000 hours, and 1000 hours respectively, hence the three curves shown in Figure 4 below. However, increasing the strain at limit state to 1% provided for a fourth set of TTF at 10000 hours.

From this set of data, the maximum allowable stress at an operation temperature of 850°C could be derived with the results indicated on the graph in Figure 4 below:

- Allowable stress = 80.4 MPa to obtain 10 hours to reach a limit state of 1% creep strain
- Allowable stress = 59.1 MPa to obtain 100 hours to reach a limit state of 1% creep strain
- Allowable stress = 43.6 MPa to obtain 1000 hours to reach a limit state of 1% creep strain
- Allowable stress = 32.3 MPa to obtain 10000 hours to reach a limit state of 1% creep strain

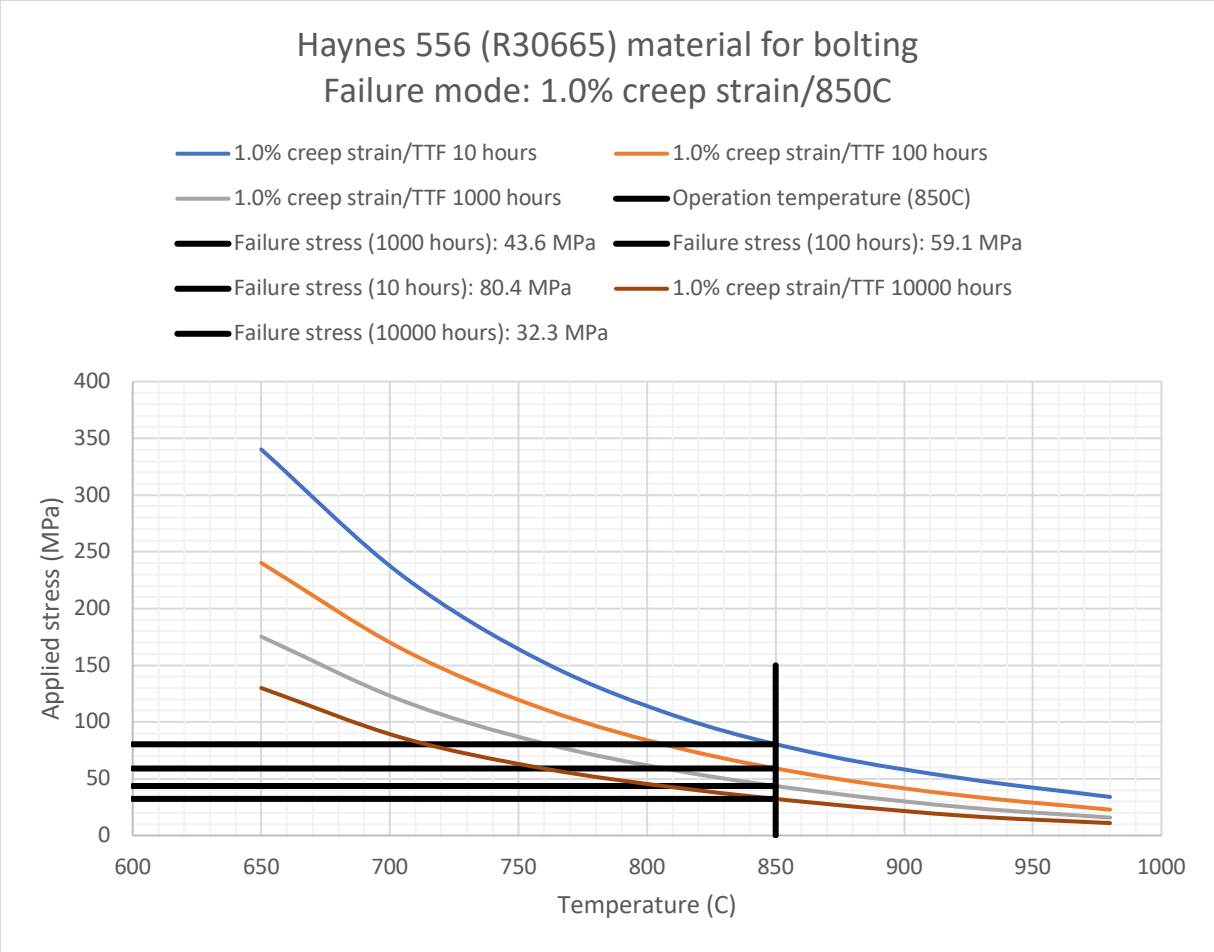


FIGURE 4

The same set of test data can be expressed as time to reach a defined limit state. In Figure 5 below the time to reach a limit state of 1% true creep strain has been plotted against applied stress and shows this limit state is reached after 100000 hours if the applied stress is 24 MPa.

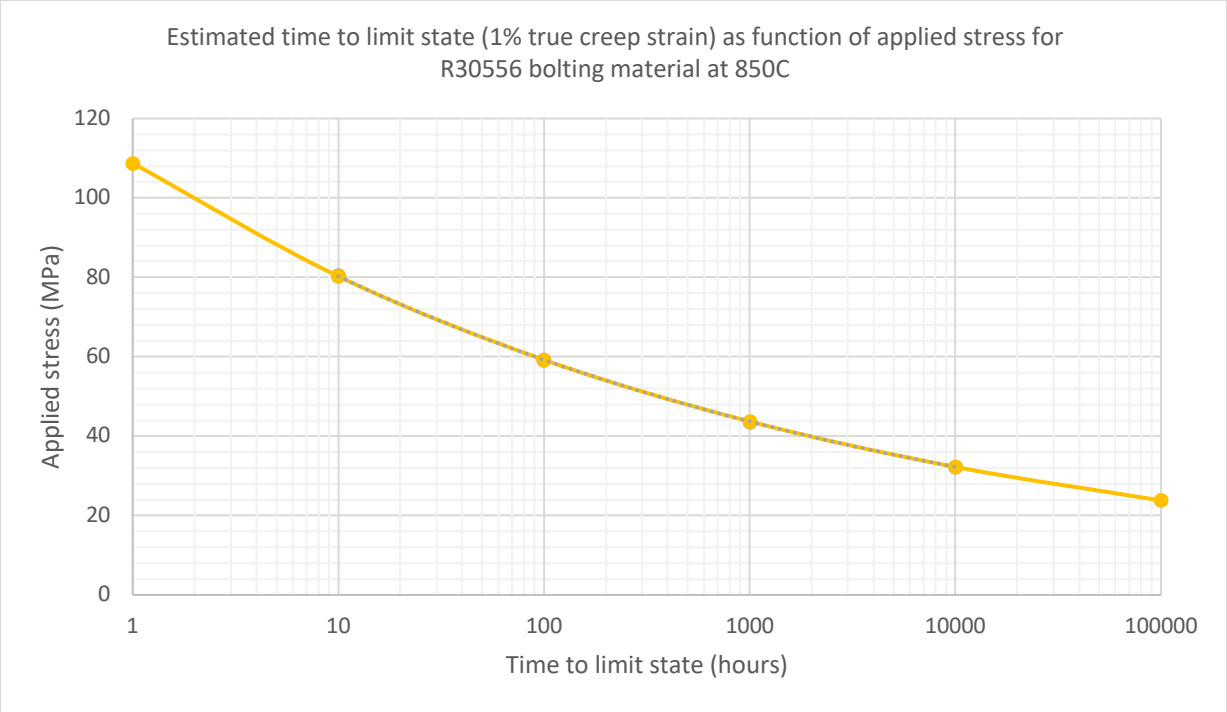


FIGURE 5

These creep data will be utilized for definition of limit states and applied safety factors for the bolts required for the EN 1591-1 modelling, see subsequent paragraphs of this paper.

2.4.4 Code requirements for allowable stresses and creep

The ASME codes for pressure vessels (BPVC VIII Div.1/II Part D) and piping (B31.3) provide the same allowable (tensile) stresses. For temperatures exceeding 815°C creep is considered by assuming a 100000-hour service life and limiting the allowable stress such that creep deformation (strain) does not exceed 1%. For N08810 pressure forgings and seamless pipes/tubes the maximum allowable stress (S_{max}) is equal to 9.03 MPa at 850°C. For R30556 bolting the S_{max} value is stated as 18.2 MPa at 850°C.

The stress values for high temperatures are based, whenever possible, on representative uniaxial properties of the materials obtained under standard ASTM testing conditions or equivalent. The stress values are based on basic properties of the materials and no consideration is given for corrosive environment, for abnormal temperature and stress conditions, or for other design considerations.

Unfortunately, the ASME codes do not provide actual creep-rupture equations (time to limit state as function of applied stress) and hence to obtain these it is required to go to the source references (manufacturers, R&D institutes, etc.) where the actual creep testing was performed by laboratory testing and results provided as open-source literature. The same applies to most of the ASTM high temperature material standards which normally do not contain information regarding time to

limit state (1% creep strain and/creep rupture) as function of applied stress.

In contrast, such equations are available in the EN pressure vessel code EN 13445-3 and associated EN/DIN material standards, e.g., EN 10028-7 Flat products made of steel for pressure purposes, Part 7: Stainless steels where strength for rupture data are provided for 10000, 30000, 50000, 100000, 150000, 200000 and 250000 hours.

2.4.5 Summary of allowable stresses/creep data for the pilot plant

The codified maximum allowable stress (S_{max}) and the obtained applied stresses from laboratory testing ($S_{applied}$) for the two BFJ materials applicable to the pilot plant are summarized in Table 1 below for operation temperature 850°C

BFJ component	Material	S_{max} (MPa)	Code/standard	Applied stress ($S_{applied}$) at TTF (MPa) 100000 hours/1% creep strain	Source	$S_{max}/S_{applied}$
Flange	N08810	9.03	Ref. (4)	30	Ref. (3)	0.30
Pipe/tube						
Bolting	R30556	18.2		24	Ref. (11)	0.76

TABLE 1

The summary shows that the laboratory testing stress data are reasonably close to the codified stress for the bolting material (0.76) but is quite far off for the flange/pipe/tube material (0.30). The differences are due to the conservatisms built into codes and standards.

The ratio $S_{max}/S_{applied}$ can be considered as equivalent to a safety factor (SF) and e.g., if an allowable stress of 24 MPa had been applied to the bolting, this would have meant a reduction of the SF from 1.0 to 0.76.

As will be shown in the subsequent section of this paper, the maximum allowable stresses (S_{max}) are extremely small and posed a large challenge to provide sufficient loading compression to the BFJ to achieve required leak tightness and stability based on a standard pilot plant lifetime requirement for materials and equipment based on the EN 1591-1/ASME II, Part D codes. Even the $S_{applied}$ values were too low and hence a 100000-hour pilot plant material/equipment lifetime turned out not to be achievable.

During the EN 1591-1 modelling work, ref. to Section 4, it became apparent that the bolting material was the critical component and more specifically the strong effect of the coefficient of thermal expansion (CTE). The principal effect of the CTE can be summarized as two cases:

1. Case 1: $[CTE]_{bolt} < [CTE]_{flange}$
2. Case 2: $[CTE]_{bolt} > [CTE]_{flange}$

Case 1 may cause overloading of the bolt whereas Case 2 may cause leakage

of the BFJ. The selected materials have the following CTE-values at 800°C in accordance with published datasheets:

- $[CTE]_{flange}$ = 17.89e-6 m/m/°C
- $[CTE]_{bolt}$ = 16.7e-6 m/m/°C

and hence there is a big risk for bolt overloading (Case 1) in this case during service at 800°C which was also encountered during the EN 1591-1 modelling. At these high temperatures, even the smallest CTE-difference between bolt and flange may cause design problems with associated risk for leakages of gasketed BFJ's.

In the current case, there were principally three ways to reach a design solution:

- Reduce the safety factor (SF) which – in a creep regime – is equivalent to reduce the plant design lifetime from the standard 100000-hour lifetime
- Reduce the service temperature of the BFJ components
- Combination of the above

As shown in section 4, and after extensive iteration efforts, a design solution was achieved with the following assumptions:

- Set SF = 0.6 for the bolting material
- Reduce the service temperature of BFJ components from the internal process temperature of 850°C to 797°C by doing temperature distribution simulations, ref. to Section 3

As can be seen in Figure 6 below a SF=0.6 for the bolting material corresponds to a maximum allowable stress of 48.22 MPa for the R30556 bolting material and a design lifetime of approximately 0.75 years if 1% permanent creep strain is acceptable and a BFJ average temperature of 797°C is achievable. Note that the curves in Figure 6 are based on the same data analysis as was performed under Section 2.3, now applying data at 797°C (not shown here, the graph for 0.5% creep strain included for reference).

The EN 1591-1 calculations show, ref. to Section 4, that these design conditions will provide a stable BFJ with a leakage class of $L_{0.0001}$ for at least 1.14 years.

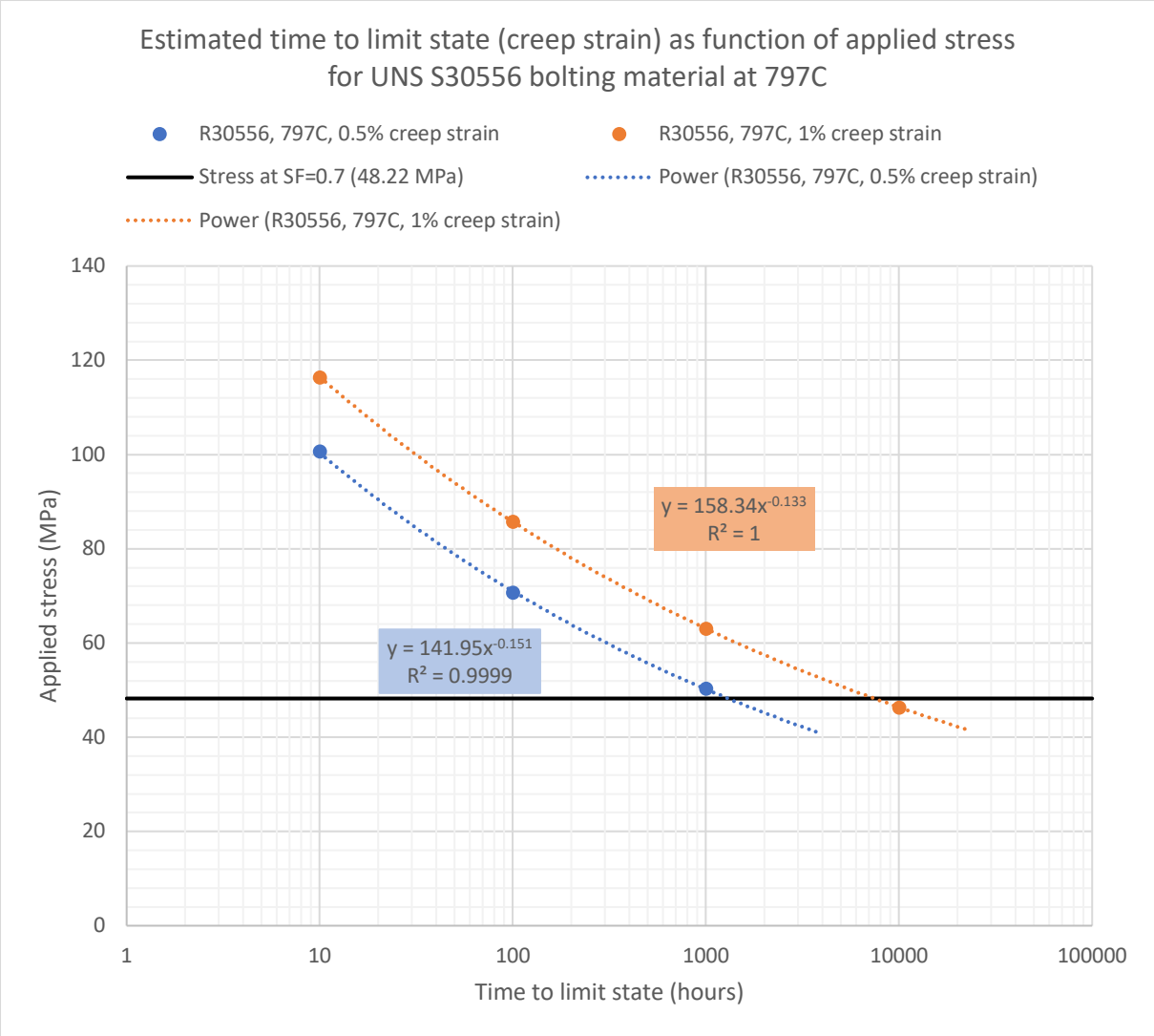

Estimated time to limit state (creep strain) as function of applied stress for UNS S30556 bolting material at 797C
R30556, 797C, 0.5% creep strain
R30556, 797C, 1% creep strain
Stress at SF=0.7 (48.22 MPa)
Power (R30556, 797C, 0.5% creep strain)
Power (R30556, 797C, 1% creep strain)
y = 158.34x^-0.133
R² = 1
y = 141.95x^-0.151
R² = 0.9999
Applied stress (MPa)
0
20
40
60
80
100
120
140
1
10
100
1000
10000
100000
Time to limit state (hours)

FIGURE 6.

3. TEMPERATURE DISTRIBUTION

3.1 Introduction

As shown in the previous sections of this paper, the derived allowable stresses are extremely low and would render it practically impossible to demonstrate mechanical stability of the BFJ with a high tightness class under the stated design conditions. Hence, the gasket industry's challenge to provide suitable gasket products for these environments is understandable. Here, it has been assumed that these high temperature piping systems are completely insulated for heat conservation and personal safety including the flanges. Hence, a quite even temperature distribution between all the BFJ components would have to be assumed and hence the same temperature would have to assumed for the flange ring and the fasteners as stated in paragraph 303.1.1 of (7).

However, the stated operation temperature of 850C is the temperature of the process environment contained by the piping/equipment and is not necessarily the actual temperature of the materials exposed to creep conditions if one considers a case of non-insulated flanges with the rest of the pipe work insulated. This is also in accordance with paragraph 301.3.2 of (7):

For fluid temperatures 65°C (150°F) and above, unless a lower average wall temperature is determined by test or heat transfer calculation, the temperature for uninsulated components shall be no less than the following values:

(1) valves, pipe, lapped ends, welding fittings, and other components having wall thickness comparable to that of the pipe — 95% of the fluid temperature

(2) flanges (except lap joint) including those on fittings and valves — 90% of the fluid temperature

(3) lap joint flanges — 85% of the fluid temperature

(4) bolting — 80% of the fluid temperature

Accordingly, the following temperatures could be considered for non-insulated flanges:

- Flanges: 765°C
- Fasteners: 680°C

To investigate this, Finite Element Method (FEM) simulations were performed of the temperature

distribution of uninsulated flanges to get an indicative evaluation of actual material temperatures in comparison with the code requirements.

3.2 Temperature distribution of uninsulated flanges

The FEM of the temperature distribution of uninsulated flanges was performed with the following boundary conditions (BC):

- Internal temperature: 850°C
- BFJ size/pressure class: 3”/#600
- Fasteners made of alloy UNS R30556 which has CTE = 16.85e-6 at 850°C
- Flanges made of alloy UNS N08810 which has CTE = 18.00e-6 at 850°C
- DeltaV-Seal™ gasket made of alloy UNS N08811/N08810
- Thermal conductivity [W/m/K] of fasteners (UNS R30556): 26 W/m/K at 850°C (estimation)
- Thermal conductivity [W/m/K] of flange/pipe/gasket (UNS N08810): 25.5 W/m/K (estimation)
- Natural convection on non-insulated external surfaces: 5 W/m^2

3.3 Results of FEA simulation of temperature distribution

The results of the FEM simulations are shown below in:

- Figure 7 (temperature distribution in flange)
- Figure 8 (temperature distribution in fastener)
- Figure 9 (temperature distribution in stud)

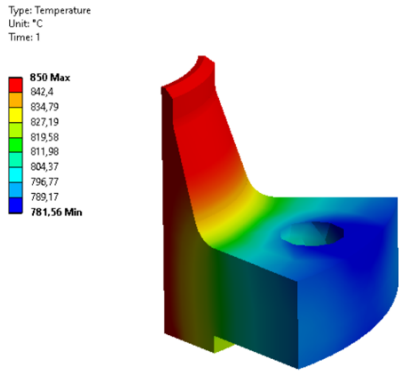


FIGURE 7 (FLANGE)

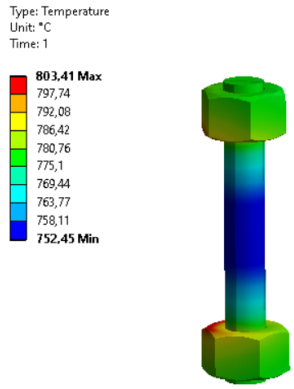


FIGURE 8 (FASTENER)

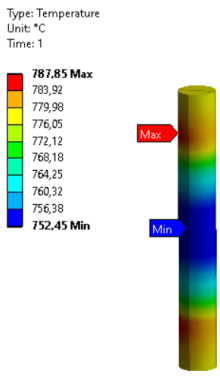


FIGURE 9 (STUD)

The results are summarized in Table 2 below:

Item #	Component	Minimum Temperature (°C)	Maximum Temperature (°C)	Component Average (°C)	BFJ Average (1 & 3) (°C)
1	Flange	782	850	816	797
2	Stud	752	788	770	
3	Fastener	752	803	778	

TABLE 2

3.3 Summary

The min/max plots and Table 2 show that the average temperature of the 3"/#600 BFJ is approximately 797°C which is significantly higher than the code requirements for non-insulated flanges, especially in the case of the fasteners. Hence, to assure a conservative assessment, this temperature was considered for the bolt, flange and pipe materials in the strength and stability calculations presented in Section 4.

4. EN 1591-1 MODELLING

4.1 Introduction

As stated before, the EN 1591-1 model iteration provides checking/verification of the BFJ stability and leak tightness simultaneously for all load cases, i.e., assembly, testing and operation applying the specified T/P conditions for each of the load cases. The results are stated as two parameters:

- minimum required compression force
- maximum allowable compression force

under the checked condition that all load ratios are equal to or less than one (1). The results also show which of the BFJ components is critical and hence makes it possible to change input parameters so that solutions can be achieved.

Pipeotech's recently completed EN 13555 testing program for DeltaV-Seal™ 800HT has provided a comprehensive set of gasket data which have been implemented in the EN 1591-1 software data base, refer to the gasket datasheet published elsewhere. Especially the results of leakage testing (loading/unloading) and creep/relaxation testing at 880°C have proven to be important for this modelling. The obvious advantage to apply this gasket to this BFJ material combination is that this gasket is manufactured of the same material grade as the flange, namely UNS N08810/N08811.

The obtained leakage curves for DeltaV-Seal™ 800HT at 880°C are shown in Figure 10 below.

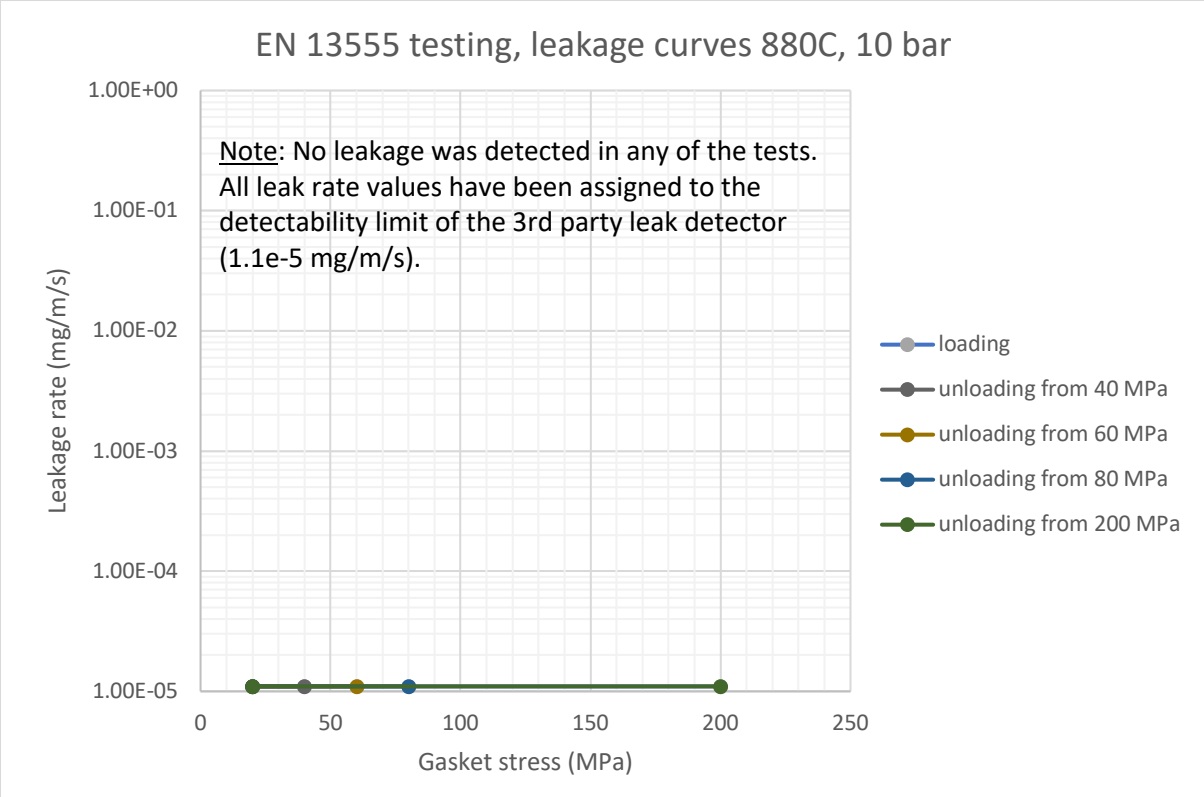


FIGURE 10

Maximum applied force during loading was 71.5 kN corresponding to a gasket stress of 200 MPa and minimum applied force during unloading was 7.2 kN corresponding to a gasket stress of 20 MPa during these tests. As noted on the graph, there was no leakage detected in any of these 880°C tests which were performed with DN40 PN40 gaskets.

4.2 Input data for the pilot plant

An extensive effort was required to derive an acceptable set of input data, refer to previous sections of this report. This was done by iteration of several parameters such as required leak tightness class (L), gasket seating stress, thermal expansion coefficients (TEC), allowable stresses and external forces and moments acting on the BFJ resulting from the high process temperatures.

For the testing LC the test pressure was considered in accordance with paragraph 345.4.2 of (7) since the design temperature (850°C) is greater than the test temperature (20°C), and hence the minimum test pressure was calculated using the formula:

$P_T = 1.5 * P * S_T / S$ where

- P = internal design gage pressure (0.5 barg)
- P_T = minimum test gage pressure
- S = allowable stress at component design temperature for the prevalent pipe material (15 MPa)

- S_T = allowable stress at test temperature for the prevalent pipe material (115 MPa)

Hence, the test pressure was considered as 5.75 barg.

External forces and moments acting on the BFJ allowing for high temperature effects on the piping systems were assumed as follows:

- Axial force = 3500 N
- Bending moment = 450 Nm

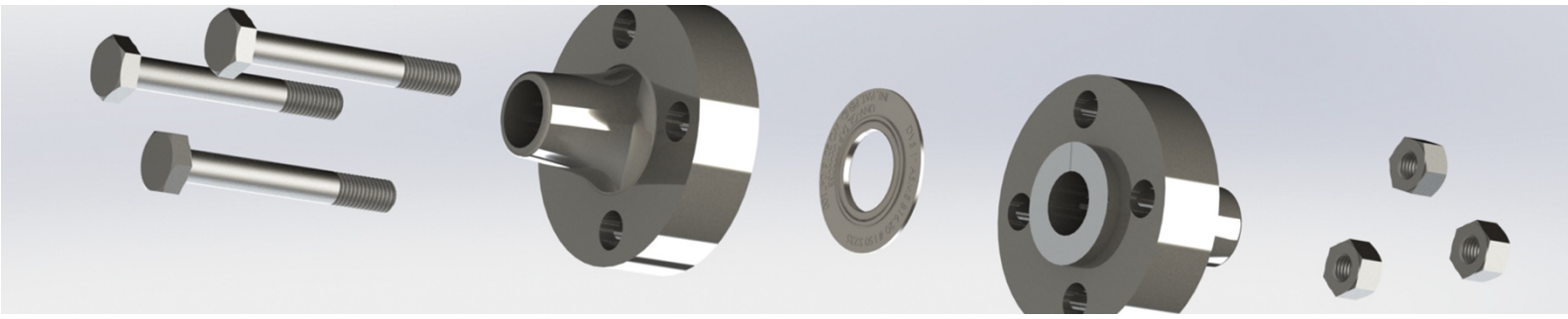
4.3 Results of EN 1591-1 modelling

Selected results of the EN 1591-1 modelling process of a 3”/#300 BFJ are shown in Appendix 1. The calculation results can be summarized as follows:

- Total required assembly force during bolt-up is 31.4 kN
- Required torque is 11.7 Nm which is applicable for the assumed lubricant (Molykote™ P-37 Paste containing zirconium dioxide solids)

In summary, the results show that it is possible to prove mechanical stability and a leak tightness class of $L_{0.0001}$ for a 3”/#300 BFJ gasketed with DeltaV-Seal™ 800HT operating in an insulated piping system with non-insulated flanges where the internal normal operating process temperature is 850°C under the assumed design conditions as stated in this paper.

It should be noted that the presented case is based on the ASME II, Part D standard allowable stresses. However, the assumed creep lifetime of the ASME standards which normally assume a limit state condition of 1% permanent creep strain after 100000 hours (11.4 years) of plant exposure has been reduced to 0.75 years as discussed in Section 2. As stated in Section 1 it would probably be reasonable to assume a shorter design lifetime for a typical pilot plant operating at such high temperatures where the plant operation experience is often limited.



5. CONCLUDING REMARKS

This paper has outlined the general basis and principals for application of critical creep design parameters for design of gasketed BFJ's operating under high temperatures. In this Part 1 paper a set of specific design parameters has been applied and the results of mechanical stability calculations and tightness class (L) determinations are presented for a 3"/#300 BFJ. In a subsequent paper a wider range of pilot plant design parameters will be presented.

The paper demonstrates a BFJ design approach that can be applied to any high temperature pilot plant development project. This approach should be applied in the early stages of the project development (FEED, pre-FEED) to obtain maximum gain in terms of optimization of the pilot plant design.

Pipeotech's new gasket DeltaV-Seal™ 800HT is fundamental to achieve BFJ stability combined with a high tightness class in these challenging high temperature process environments.

6. REFERENCES

The following references have been applied in this work:

1. EN 1591-1, December 2013, Flanges, and their joints – Design rules for gasketed circular flange connections – Part 1: Calculations
2. EN13555:2021 Flanges and their joints – Gasket parameters and test procedures relevant to the design rules for gasketed circular flange connection
3. Jaeri – Tech 98-010, Alloy 800H, 1998, Japan Atomic Energy Research Institute
4. ASME BPV 2010 II, Part D, Properties (Metric), Materials
5. ASTM B564, Rev.2018, Standard for Specification for Nickel Alloy Forgings
6. ASME B16.5-2020, Pipe Flanges and Flanged Fittings, NPS ½ Through NPS 24, Metric/Inch Standard
7. ASME B31.3-2020, Process Piping, ASME Code for Pressure Piping, B31
8. High-Temperature Characteristics of Stainless Steels, A Designer's Handbook Series No. 9004, NiDi (Nickel Development Institute)
9. Behaviour of flange joints in steam generator under thermal loads, N.R Nelson, Department of Mechanical Engineering, Indian Institute of Information Technology Design and Manufacturing, E35, Web of Conferences 309, 01082 (2021), https://doi.org/10.1051/e3sconf/2021130901082
10. WRC Bulletin 510, 2nd Edition, May 2022, Analysis of the Effects of Temperature on Bolted Joints, ISSN 2372-1057
11. Haynes International – Haynes 556 alloy (UNS R30556)
12. Haynes International – Hastelloy X alloy (UNS N06002)
13. ASME B36.19 – 2022 Welded and Seamless Wrought Stainless-Steel Pipe
14. EN 13445-3: 2021, Unfired pressure vessels – Part 3: Design (Chapter 19 Creep design)

APPENDIX

Selected screenshots from EN 1591-1 calculations using TÜV-certified software from Lauterbach Verfahrenstechnik:

- Table 11 (LC's)
- Table 12 P/T conditions
- Table 13 (flange material data)
- Table 14 (pipe and bolt material data)
- Table 15 (calculated gasket results)
- Table 16 (gasket stresses)
- Table 17 (checked load ratios and specified total assembly force and nominal bolt torque during bolt-up)

Load cases		Assembly-Test-Service		
Accuracy of iteration		0.01	-	
Number of times that the joint is re-made	N_R	1		
Temperature under assembling conditions	T_0	20	°C	

Table 11.

Pressure / Temperature

		Service		Test	
Are all temperatures the same?		☑		☑	
Pressure	P	0.02	MPa	0.575	MPa
Temperature	T	797	°C	20	°C

Table 12

Material data

Flanges

		1st Flange			2nd Flange	
Material		N08810-SB-564--Class:Annealed-Size:			N08810-SB-564--Class:Annealed-Size:	
Assembly						
Modulus of elasticity	E_{F0}	196300	N/mm²	$\tilde{E}_{F0}$	196300	N/mm²
Nominal design strength	$R_{e,F0}$	115	N/mm²	$\tilde{R}_{e,F0}$	115	N/mm²
Safety factor	S_{F0}	1		$\tilde{S}_{F0}$	1	
Allowable stress	f_{F0}	115	N/mm²	$\tilde{f}_{F0}$	115	N/mm²
Wall thickness allowance	c_{1F}	0	mm	$\tilde{c}_{1F}$	0	mm
Corrosion allowance	c_{2F}	0	mm	$\tilde{c}_{2F}$	0	mm
Service						
Modulus of elasticity	E_{FI}	145000	N/mm²	$\tilde{E}_{FI}$	145000	N/mm²
Nominal design strength	$R_{e,FI}$	15.4	N/mm²	$\tilde{R}_{e,FI}$	15.4	N/mm²
Safety factor service	S_{FI}	1		$\tilde{S}_{FI}$	1	
Thermal expansion coefficient	α_{FI}	17.89	1e-6/K	$\tilde{\alpha}_{FI}$	17.89	1e-6/K

Table 13.

Shells

		1st Flange			2nd Flange	
Material		N08810-SB-407-Class:Annealed			N08810-SB-407-Class:Annealed	
Assembly						
Nominal design strength	$R_{e,S0}$	115	N/mm²	$\tilde{R}_{e,S0}$	115	N/mm²
Safety factor	S_{S0}	1		$\tilde{S}_{S0}$	1	
Allowable stress	f_{S0}	115	N/mm²	$\tilde{f}_{S0}$	115	N/mm²
Nominal design stress equivalent cylinder	f_{E0}	115	N/mm²	$\tilde{f}_{E0}$	115	N/mm²
Wall thickness allowance	c_{1S}	0	mm	$\tilde{c}_{1S}$	0	mm
Corrosion allowance	c_{2S}	0	mm	$\tilde{c}_{2S}$	0	mm
Service						
Nominal design strength	$R_{e,SI}$	15.4	N/mm²	$\tilde{R}_{e,SI}$	15.4	N/mm²
Safety factor service	S_{SI}	1		$\tilde{S}_{SI}$	1	

Bolts

Material		R30556-SB-572-Class:Annealed	
Assembly			
Minimum rupture elongation	A_{min}	35 ?	%
Modulus of elasticity	E_{B0}	215000	N/mm²
Nominal design strength	$R_{e,B0}$	172	N/mm²
Safety factor	S_{B0}	1	
Allowable stress	f_{B0}	172	N/mm²
Thermal expansion coefficient	α_{B0}	14.7	1e-6/K
Service			
Modulus of elasticity	E_{BI}	148000	N/mm²
Nominal design strength	$R_{e,BI}$	28.93	N/mm²
Safety factor service	S_{BI}	0.6	

Table 14.

Inside diameter	d_{Gi}	88	mm
Outside diameter	d_{Ga}	117	mm
Nominal temperature (test)	$T_{G,Nom,T}$	20	°C
Nominal temperature	$T_{G,Nom}$	880	°C
Conversion of leakage rate to other conditions		☐	
Leakage rate	L	0.0001	mg/(s·m
Leakage pressure	P_G	10	bar
Required minimum design seating stress for tightness class *L* Assembly	$Q_{min(L)}$	20	N/mm²
Gasket seating stress before unloading	Q_A	40	N/mm²
Average effective compressive stress of gasket	Q_{G0}	56.78	N/mm²
Modulus of elasticity for unloading / reloading	E_{G0}	66524	N/mm²
Average modulus of elasticity for unloading / reloading	E_{Gm}	66524	N/mm²
Coefficient of friction between gasket and flange contact facing	μ_G	0.3	
Thermal expansion coefficient	α_{Gl}	15	1e-6/K
Initial theoretical uncompressed thickness of gasket	e_{Gt}	3.8	mm
Pressed gasket thickness under assembling conditions at compressive stress Q_{G0}	$e_{G(QG0)}$	3.8	mm
Number of gaskets	n_G	1	

Gasket stress

Gasket seating stress before unloading	Q_A	40	N/mm²
Required gasket stress *Subsequent condition*	$Q_{(S)min,L}$	20	N/mm²
Maximum gasket seating stress, safe and without damaging at T_0	Q_{smax}	1150	N/mm²

Table 15.

Tightness

Required gasket force F_{G0req} 30613 N

		Assembly	Test	Service	
Actual gasket stress	Q_{Gl}	56.78	22.12	123.2	N/mm²
Required gasket stress	$Q_{(S)min,L}$	20	20	20	N/mm²
Desired gasket stress	Q_A	40 N/mm²			
Gasket stress sufficient?		✓	✓	✓	

Gasket force assembling sufficient $F_{G0req} < F_{G0} \Leftrightarrow 30613\ N < 31816\ N$ ✓

Table 16.

Load ratios

		Assembly	Test	Service	
1st Flange					
- *Flange*	Φ_F	0.05373	0.05597	0.9992	-
(hub overloaded)		✓	✓	✓	
(ring overloaded)		✓	✓	✓	
Maximum flange rotation	Θ_F	0.007871	0.009888	0.02725	°
2nd Flange					
- *Flange*	$\tilde{\Phi}_F$	0.05373	0.05597	0.9992	-
(hub overloaded)		✓	✓	✓	
(ring overloaded)		✓	✓	✓	
Maximum flange rotation	$\tilde{\Theta}_F$	0.007871	0.009888	0.02725	°
Flange joint					
- *Bolt*	Φ_B	0.1515	0.1098	0.9956	-
- *Gasket*	Φ_G	0.05646	0.01923	0.1071	-
Allowable flange rotation / inclination	Θ_{all}				1 °
Condition allowable flange rotation / inclination	$Max(\Theta_i, \tilde{\Theta}_i) \leq \Theta_{all} \Leftrightarrow 0.02725° \leq 1°$				✓
Strength condition fulfilled		✓	✓	✓	

Nominal forces / nominal moments

Specified bolt force *assembly*	$F_{B0,spec}$	34100	N
Nominal bolt torque	$M_{t,nom}$	11.67	Nm
Nominal bolt force	$F_{B0\ nom}$	34100	N

Table 17.

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