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<th>詳細内容</th>
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<td>Title</td>
<td>Buckling Assessment Procedure for Large Diameter Vessel with Local Thin Area Subjected to Combined Pressure and External Moment</td>
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Buckling Assessment Procedure for Large Diameter Vessel with Local Thin Area Subjected to Combined Pressure and External Moment

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Abstract

The newly-developed p-M diagram provides a means for readily evaluating the collapse and/or buckling load of pressure equipment with external flaws simultaneously subjected to internal pressure, p and external bending moment, M due to earthquake, etc. In this paper, some FEAs for large diameter vessels with an external flaw were conducted under (1) pure external bending moment, and (2) subjected simultaneously to both internal pressure and external bending moment, in order to determine the plastic collapse load by applying the twice-elastic slope (TES) as recommended by ASME and to determine the buckling load. The p-M line adopted in the Ibaraki FFS rule based on the measured yield stress indicates that the safety margin for the TES loads at the LTA is about 1.2-1.8 and 1.5-2.0 for the buckling loads. The Ibaraki FFS rule that prevents buckling by applying Donnell's equation of FS=2 can assure adequate levels of integrity and safety.

1. Introduction

There are numerous cases in which the external surfaces of vessels such as towers, piping and storage tanks are partially corroded under their insulation. Therefore, assessment of the local thin area (LTA) should be undertaken for both tension and compression bending due to earthquake, etc.

It is important to know the limit loads for the safe operation of the vessels. Chattopadhyay et al. (2006) proposed several definition of the term 'limit load'. In this paper, the plastic initiation load (p_pl.init or M_pl.init) is characterized by the point of deviation from the elastic slope of the load-strain curve (ASME Div1, 2004). These loads will generally give a lower boundary to the plastic limit load determined by other methods. As shown in Figure 1, plastic collapse load (p_TES or M_TES) indicates a load where appreciable plastic deformation occurs (almost equal to general yielding and not necessarily the physical collapse load of the vessel), determined by applying twice-elastic slope (TES) as recommended by ASME Div2 (2004), whose strain is the maximum principal strain. Plastic instability load (p_max or M_max) is characterized by the zero slope of the load-strain curve, which means the maximum load or break point. Buckling load (M_Buckle) is a load where precipitous drop-off occurs, which is important for large diameter-to-thickness ratio (D/t) vessels (see Figure 1).

1

1
Figure 2 shows the process for a cylinder proceeding from plastic deformation to plastic instability (or break) load. A specific sequence of events (plastic initiation and plastic instability or breakage) will generally occur in response to the applied load. Where the ratio of diameter-to-thickness is large, buckling may occur at any stage of these events.

The procedures described in the Ibaraki FFS rule (2006) based on the p-M (internal pressure ratio and external bending moment ratio) method (Konosu, 2006) were used to predict the plastic initiation condition and the collapse condition for cylinders with an LTA subjected to combined internal pressure and bending moment. Numerous experimental tests and FEA were conducted on cylinders with several sizes of external flaws to verify the procedures of the Ibaraki FFS rule (Konosu, 2007). In this paper, attention was focused on the procedure for assessing the buckling of large D/t vessels with a local thin area.

Nomenclature

\( a \): Flaw depth

\[
A_b^0 = \frac{2\left[1 - (1 - \tau)^3\right]}{2\left[1 - (1 - \tau)^3\right] \cos \psi_b + \left[1 - y\tau\right]^{3/2} - 1}\sin \theta
\]  : Coefficient of bending stress in reference stress

\( 2c_L \): Longitudinal flaw length

\( 2c_\theta \): Circumferential flaw length

\( E \): Young’s modulus

\( FS \): Design factor of buckling stress for seismic loading

\( k \): Increasing factor of allowable stresses for combination of design load and earthquake loading or wind loading

\( LTA \): Local thin area

\( M \): External bending moment

\[
M_r = \frac{|M|}{M^L} : \text{Parameter in } p - M \text{ diagram approach}
\]

\[
M_s = \frac{1}{1 - \frac{a}{t} + \frac{a}{t} \frac{1}{M_F(a)}} : \text{Bulging factor for a part-through flaw}
\]

\[
M_c^L = \frac{\pi R_b^3 \left(1 - (1 - \tau)^3\right)}{4A_b^0} \sigma_f : \text{Plastic collapse moment for an external surface flaw under pure external moment}
\]
$M_{\text{cutoff}}$: Cut off value to prevent buckling
$M_{\text{buckle}}$: Moment where a precipitous drop-off occurs
$M_{\text{max}}$: Plastic instability moment determined as the plastic instability limit load
$M_{\text{pl.init}}$: Moment where the moment versus strain curve deviates from linearity
$M_{\text{TES}}$: Collapse moment determined under the twice-elastic slope method
$p$: Internal pressure

$p_r = \frac{p}{p^*}$: Parameter in $p-M$ diagram approach

$p^L = \frac{2\tau(2-\tau)}{4-6\tau+3\tau^2} - \frac{1-\alpha}{M_s} \sigma_f$: Plastic collapse internal pressure for an external surface flaw under pure pressure

$p_{\text{pl.init}}$: Pressure where the pressure versus strain curve deviates from linearity
$p_{\text{TES}}$: Collapse pressure determined under the twice-elastic slope method
$p_{\text{max}}$: Plastic instability pressure determined as the plastic instability limit load

$R_o = D_o / 2$: Outer radius of cylinder
$R_i = D_i / 2$: Inner radius of cylinder

$T_{\text{ES}}$: Twice-elastic slope

$t$: Wall thickness

$t^* = \left[1 - \frac{\cos \theta \sin \theta + \theta}{\pi} \cdot \frac{y(2 - 3y\tau)}{2 - 3\tau}\right] \cdot t$: Effective wall thickness

$t_c$: Nominal thickness minus maximum metal loss at LTA

$y = \frac{a}{t}$: Relative measured flaw depth

$\alpha = \frac{\pi a c_L}{t(\pi a_L + 4t)}$: Relative rectangular flaw depth

$\lambda = \frac{1.818 c_L}{\sqrt{R_o t}}$: Shell parameter for a through-wall flaw

$\lambda_a = \frac{1.428 c_L}{\sqrt{R_o a}}$: Shell parameter for a part-through flaw

$2\theta \approx \frac{\pi c_{\theta}}{(2 - y\tau)R_o}$: Angle (rad.) of an external circumferential surface flaw

$\sigma_{ys}^{\text{mean}}$: Measured yield stress

$\sigma_{ys}^{\text{min}}$: Specified minimum yield stress

$\sigma_{ys}^{\text{min}}$: Specified minimum yield stress at design temperature
\[ \tau = \frac{t}{R_o} : \text{Ratio of thickness to outer radius of a cylinder} \]

\[ \psi_b = \frac{y \theta}{2} - \frac{y \tau}{2} : \text{Angle of compressed region with no axial stress action} \]

2. IBARAKI FFS rule assessment procedure

The Ibaraki FFS rule assessment procedure based on p-M method is a simple evaluation procedure, which can be readily used to assess a vessel with an LTA simultaneously subjected to internal pressure and bending moment due to earthquake, etc (Konosu, 2006).

Figure 3 indicates the p-M diagram (structural integrity assessment diagram) in the Ibaraki FFS rule, which is currently being revised. The flow stress, \( \sigma_f \) is considered as the specified minimum yield stress of the material with a safety factor of 1.5 in order to evaluate on the conservative side. The cut-off value, \( M_r^{\text{cutoff}} \), may be determined by the following equations (1) to (4) (Miller, 1995; Donnell, 1943). The Ibaraki FFS rule adopts Donnell’s equation in accordance with the Japanese High Pressure Gas Safety Law (2006; KHK, 1997). ASME Sec. VII Div.2-2007 (ASME Div.2, 2007) and API 579-1/ASME FFS-1 (API/ASME, 2007) adopt Miller’s equation. In Miller’s equation, the allowable stresses for combinations of design loads and earthquake loading or wind loading are increased by a factor of \( k = 1.2 \).

\[ M_r^{\text{cutoff}} = \frac{466}{FS \cdot (331 + \frac{D_o}{t^*})} \frac{\sigma_{ys}}{\sigma_f} \quad \text{for} \quad 100 \leq \frac{D_o}{t^*} < 600 \quad (\text{Miller}) \quad (1) \]

\[ FS = \frac{1}{k} \left( 2.407 - 0.741 \left( \frac{F_{sc}}{\sigma_{ys}} \right) \right) = \frac{1}{k} \left( \frac{1}{331 + \frac{D_o}{t^*}} + \frac{451.41 + 2.41 \frac{D_o}{t^*}}{k} \right) \]

\[ \frac{1.667}{k} \leq FS \leq \frac{2.0}{k} \]

\[ F_{sc} = \frac{466\sigma_{ys}}{1.0 \left( 331 + \frac{D_o}{t^*} \right)} \]

\[ M_r^{\text{cutoff}} = \frac{1.081 \sigma_{ys}}{FS} \frac{\sigma_f}{\sigma_f} \quad \text{for} \quad \frac{D_o}{t^*} < 100 \quad \text{and} \quad \gamma \geq 0.11 \quad (\text{Miller}) \quad (2) \]

\[ M_r^{\text{cutoff}} = \frac{1.4 - 2.9 \gamma}{FS} \frac{\sigma_{ys}}{\sigma_f} \quad \text{for} \quad \frac{D_o}{t^*} < 100 \quad \text{and} \quad \gamma < 0.11 \quad (\text{Miller}) \quad (3) \]

\[ FS = \frac{1.667}{k} \]
\[ k = \begin{cases} 1.0 & \text{: Design load} \\ 1.2 & \text{: Combinations of design loads and earthquake loading or wind loading} \end{cases} \]

\[ \gamma = \frac{\sigma_{ys} D_0}{E t^*} \]

\[ M'_{\text{cutoff}} = \frac{1.2 E t^*}{FS \cdot D_0 (1 + 0.004 \frac{E}{\sigma_{ys}}) \sigma_f} \quad \text{for } \frac{D_0}{t^*} = 600 - 3000 \quad \text{(Donnell)} \quad (4) \]

\[ FS = 2.0 \]

As the internal pressure is expected to suppress the buckling behavior under low-pressure conditions (Fung, 1967), holding the buckling moment of \( p=0 \) irrelevant to the internal pressure will result in conservative estimates.

3. Material and the geometric parameters of the cylinders with LTA

The material used in the FEA analysis was typical Japanese carbon steel (STPG370) as shown in Table 1. True stress-strain relation, as shown in Figure 4, was experimentally derived on the assumption that the specimen volume is constant up to the maximum load point. Shown currently in the above-mentioned Figure is the dotted curve adopted in FEA. Table 2 shows the geometric parameters of the cylinders.

4. FEA model

The buckling loads or plastic instability loads of vessels with an external local thin area are calculated by FEA. The applied loads are pressure and moment. The moment is loaded in the direction in which compression stress occurs at the local thin area. Material nonlinearity is considered as well as geometrical nonlinearity. The FEA software, ABAQUS, is used. Three-dimensional shell elements (S4R) are applied. The FEA models were analyzed by using the experimentally obtained true stress-strain relation shown in Figure 4. Young’s Modulus, yield stress and Poisson’s ratio are 203GPa, 326MPa and 0.3, respectively. FEA models adopted are 1/4 models from the symmetry. The thick flat head is set to one end of the cylinder. The moment is loaded at the center of the flat head. Pressure is loaded to the inside of the cylinder and the flat head.

The boundary conditions of the FEA model are the symmetry condition. And vertical direction (2-direction) at the center of the flat head is restrained. The typical configuration of a vessel with LTA and finite element mesh for the model of V_XI are shown in Figure 5.

Load cases are two cases as follows:

MC: LTA in compression under bending moment loading
P+MC: After applying initial pressure that is almost 60% of \( p^i \), the same moment as MC is loaded.
5. FEA results

Table 3 summarizes the FEA results of plastic initiation loads, TES loads and plastic instability or buckling loads. The load-strain curves are shown in Figures 6 to 8. These strains are logarithmic axial strain at the center of the inside surface of the LTA. But as only V_XIP+MC can not extract TES at the inside surfaces as shown in Figure 9, then the TES was extracted at the outside surface of the LTA as shown in Figure 8(b). In Figure 9 the axial strain at center of LTA changes the direction from minus to plus between 24800 and 25900kNm of the applied moment. The reason is suddenly buckling inward at the center of LTA (point A) and buckling outward at point B of LTA as shown in Figure 10. Therefore the axial strain of inside surface at center of LTA on V_XIP+MC changes from compression to tension as shown in Figure 9.

Refer to Figure 8(a), as V_XIMC (D/t=300) occurred suddenly buckling without plastic behavior, then post buckling analysis by the modified Riks method prepared in ABAQUS was conducted to verify the accuracy of the buckling load obtained by the general analysis method. The results are shown in Figure 11. As buckling loads which are obtained from Figures 8(a) and 11(a) are the same, it is considered that the general analysis method can accurately determine the buckling load. Figure 12 shows the buckling deformation around the LTA at the last increment of V_XIMC by the modified Riks method. From the figure, the buckling occurs inward at the center of LTA, the same as V_XIP+MC and buckling outward at point B in Figure 10 does not occur because initial internal pressure is not loaded. For the case of V_XIMC shown in Figure 11, the buckling load and the subsequent maximum load are almost the same. Although the subsequent maximum load may become larger than the buckling load depending sizes of a cylinder and an LTA in some cases, the deformation at the subsequent maximum load is considered to be very large (140mm) as shown in Figure 11(b). Therefore, the evaluation at the buckling load or the plastic instability load (limit load) which can be predicted by the general analysis method is useful and conservative.

The analysis results by the modified Riks method for V_XMC are shown in Figure 13. The maximum plastic instability and TES loads obtained by the modified Riks method are almost the same as those obtained by the general analysis method as shown in Table 3. Therefore the general analysis method can accurately determine the limit load (plastic instability load). In Figure 13, the relation of moment and strain at center of LTA of model V_XMC depicts a phenomenon of the plastic instability like in Figure 1(a). As shown in Figure 14, the plastic instability (or buckling) behavior. The deformation around the LTA at the last increment of V_XMC by the modified Riks method is shown in Figure 14. The deformation occurs outward in the circumferential direction at the LTA and over the substantial area around the LTA.

From Table 3, the effects of pressure are as follows:

(1) Effect of pressure on TES load: where initial internal pressure is maintained, TES load due to moment is a little smaller (2-3%) than that due to pure moment, and the TES loads of them are virtually almost same.
Effect of pressure on limit load or buckling load: where initial internal pressure is maintained, limit load or buckling load due to moment is larger (12-37%) than that due to pure moment. It can be seen internal pressure tends to suppress buckling behavior.

6. Comparison of FEA results with conventional buckling equations for vessel without LTA
There are no buckling equations applicable to the vessel with a LTA. Therefore, the effective wall thickness, \( t^* \), for the vessel with a LTA is proposed to use instead of wall thickness based on approximation which simply takes into account only the effect of section modulus reduction as follows (Konosu, 2006):

\[
t^* = \left[ 1 - \frac{\cos \theta \sin \theta + \theta}{\pi} \cdot \frac{y(2 - 3y)}{2 - 3\tau} \right] \cdot t
\]  

(5)

Figure 15 shows \( M_{\text{pl.init}} \), \( M_{\text{TES}} \), and \( M_{\text{max}} \) of model V_X, XI and XII due to pure moment. These moments are changed to stresses by following equation.

\[
\sigma_c = \frac{4M}{\pi R_o^3 \left( 1 - \left( \frac{t^*}{R_o} \right)^4 \right)}
\]  

(6)

From Figure 15 incorporated FEA results (V_I, V_III) for Do/t*<150 (Konosu, 2007), the effects of Do/t* on limit stress are as follows:

Where 300>Do/t*, it is considered that buckling (or plastic instability) stress matches well Miller’s equation of design factor FS=1. Where Do/t*>325, buckling (or plastic instability) stresses tend to be lower than Miller’s equation of design factor FS=1. For TES stresses (loads), Miller’s equation with FS given by equations (1) to (3) is more suitable than Donnell’s equation of FS=2. For plastic initiation stresses for Do/t*>160, however, Donnell’s equation of FS=2 is preferable to Miller’s equation with FS because of conservatism. For the range of Do/t*<100, plastic initiation stresses can be predicted by (yield stress)/1.5 (Konosu, 2007), which is preferable to the value by (yield stress)/1.285 adopted by ASME Sec.Ⅷ Div.2-2007 (ASME, 2007) and API 579-1/ASME FFS-1 (API/ASME, 2007), where 1.285=1.667/(1.2x1.081). Therefore, Ibaraki FFS rule adopting (yield stress)/1.5 for Do/t*<160 and Donnell’s equation of FS=2 for Do/t*>160 is considered to be preferable as a code.

As API 579-1/ASME FFS-1(API/ASME, 2007) adopts \( t_c \) (nominal thickness minus maximum metal loss at LTA) instead of \( t^* \) in Equations (1) to (3), the curve given by Miller’s equation in Figure 15 tends to shift toward the left side of Do/t* in accordance with the relation between \( t_c \) and \( t^* \).

7. Evaluation as per p-M diagram
Plastic initiation, TES and max loads on the p-M diagram where flow stress is assumed to be measured yield stress (\( \sigma_f = \sigma_{ys}^{mean} \)) are shown in Figure 16. When D/t* is over 200, TES and
the buckling load or plastic instability load ($M_{\text{max}}$) lie inside the p-M line especially under pure moment condition. Therefore $M_{\text{r,cutoff}}$ is defined to prevent buckling in the Ibaraki FFS rule. Donnell’s equation of $FS=2$ indicates that the safety margin for the TES loads at LTA is about 1.2-1.8 and about 1.5-2.0 for the buckling load or plastic instability load ($M_{\text{max}}$) as shown in Table 4. And as the specified minimum yield stress is used in the Ibaraki FFS rule, actually the safety margin of Donnell’s equation of $FS=2$ increases to 1.7-2.4 for the TES loads and 2.0-2.7 for the buckling load or plastic instability load ($M_{\text{max}}$).

8. Conclusions

(1) The general analysis method by FEA in accordance with material and geometrical nonlinearity can determine accurately the buckling load in comparison with post buckling analysis by the modified Riks method.

(2) The effects of pressure are as follows:

   ① Where initial internal pressure is maintained, TES load due to moment is a little smaller (2-3%) than that due to pure moment, and both of the TES loads are virtually almost the same.

   ② Where initial internal pressure is maintained, buckling load (or plastic instability) load due to moment is larger (12-37%) than that due to pure moment. Therefore, internal pressure tends to suppress buckling (or plastic instability) behavior.

(3) Comparisons to buckling equations are as follows:

   ① Where $300>\text{Do}/t^*$, it is considered that buckling stress (or plastic instability) matches well Miller’s equation of design factor $FS=1$. Buckling stresses of large diameter vessels of $\text{Do}/t^*>325$ tend to be the lower than Miller’s equation of design factor $FS=1$.

   ② For TES stress (load), Miller’s equation taking account of the design factor ($FS$) given by equations (1) to (3) is more suitable than Donnell’s equation of $FS=2$.

   ③ For plastic initiation stresses for $\text{Do}/t^*<160$, however, Donnell’s equation of $FS=2$ is preferable to Miller’s equation with $FS$ because of conservatism. For the range of $\text{Do}/t^*<100$, plastic initiation stresses can be predicted by $(\text{yield stress})/1.5$ (Konosu, 2007), which is preferable to the value by $(\text{yield stress})/1.285$ adopted by ASME Sec.Ⅷ Div.2-2007 (ASME, 2007) and API 579-1/ASME FFS-1 (API/ASME, 2007), where $1.285=1.667/(1.2x1.081)$. Therefore, Ibaraki FFS rule adopting $(\text{yield stress})/1.5$ for $\text{Do}/t^*<160$ and Donnell’s equation of $FS=2$ for $\text{Do}/t^*>160$ is considered to be preferable as a code.

   ④ Donnell’s equation of $FS=2$ based on $\sigma_{\text{mean}}^{\text{ys}}$ indicates that the safety margin for the TES loads at LTA is about 1.2-1.8 and about 1.5-2.0 for the buckling loads (plastic instability loads).

(5) And as the specified minimum yield stress is used in the Ibaraki FFS rule, actually the safety margin of Donnell’s equation of $FS=2$ increases to 1.7-2.4 for the TES loads and
2.0-2.7 for the buckling loads (plastic instability loads). The Ibaraki FFS rule adopted Donnell’s equation of FS=2 can assure adequate levels of integrity and safety.

Acknowledgements

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Fung,Y.C., Sechler, E.E., 1957, Buckling of Thin-Walled Circular Cylinders under Axial Compression and Internal Pressure, Journal of aeronautical sciences, May, p.351·356
Konosu, S., et al., 2007, Plastic collapse load for vessel with external flaw simultaneously subjected to internal pressure and external bending moment -Experimental and FEA results, Proceedings of ASME PVP2007·26410.
Figure 1 Definition of various limit loads

Figure 2 Plastic deformation process of a cylinder with a local thin area
Fig. 3 p-M diagram in Ibaraki FFS rule

\[ p_r = \frac{p}{p^L} \]

\[ M_r = \frac{|M|}{M_r^L} \]

If the flow meets the following equation,

\[ \left( \frac{c_r}{t} \right) < \frac{\pi}{4} \left( \frac{\pi}{2 \arccos \left( \frac{\sin \theta}{2} \right)} - 1 \right) \]

\[ \frac{p}{\sigma_f} = \min \left( \frac{p^L}{\sigma_f}, \frac{2\alpha}{\pi} \left( \arcsin \left( \frac{y \sin \theta}{2} \right) - y \theta \right) \left( 1 - \frac{|M|}{M_r^L} \right) \right) \]

Here,

\[ p^L = \frac{2\alpha(2-\alpha)}{4 - 6\alpha + 3\alpha^2} M_s \sigma_f, \quad M_r^L = \frac{\mu R_t^3}{4 A_b} \left( 1 - (1 - \tau)^4 \right) \]

\[ \sigma_f = \min \left\{ \alpha_{\sigma_{\mu}}, \alpha_{\sigma_{\sigma_{\mu}}} \right\} \]

\[ M_{r,\text{conf}} = \frac{S'}{\sigma_f} \]

\[ S' = \frac{1.2E_t \tau^*}{FS \cdot D_r (1 + 0.004 \frac{E}{\sigma_{\sigma_{\mu}}})} \]

\[ FS = 2.0 \]

\[ \tau^* = \left( 1 - \frac{\cos \theta \sin \theta + \theta \sin \theta}{\pi} \right) \cdot t \]
Fig. 4 True stress - true strain curve used in FEA

(a) Whole configuration and LTA  (b) Finite element mesh of whole model

(c) Finite Element mesh around LTA

Fig. 5 Typical configuration of vessel with LTA (V_XI) and finite element mesh
Fig. 6 Strain history at LTA and TES for V_XII (General analysis method)

Fig. 7 Strain history at LTA and TES for V_X (General analysis method)

Fig. 8 Strain history at LTA and TES for V_XI (General analysis method)
Fig. 9 Strain history of inside surface at LTA for V_ XIP+MC (General analysis method)

Fig. 10 Deformation and axial strain of inside surface at LTA for V_ XIP+MC (General analysis method)

Fig. 11 Strain and displacement histories for V_XIMC (Modified Riks method)
Fig. 12 Deformation around LTA at last increment for V_XIMC (Modified Riks method)

Fig. 13 Strain and displacement history for V_XMC (Modified Riks method)

(a) Strain history at center of LTA  (b) Radial displacement history at center of LTA

Fig. 14 Deformation around LTA at last increment for V_XMC (Modified Riks method)
Equations for assessing buckling and FEA results

Fig. 15 Influence of Do/t* for vessel with LTA on buckling stress
Fig. 16 Comparison of TES and buckling or plastic instability load (M_max) at LTA determined by FEA versus p-M line based on $\sigma_f = \sigma_{ys}^{mean}$
Table 1 Specified chemical composition and mechanical properties of test materials

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Average data of 3 pieces of specimen

Table 2 Configuration of vessel and flaw

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<td>300 (308)</td>
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Table 3 Results of FEM analysis

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</thead>
<tbody>
<tr>
<td></td>
<td>V_XII&lt;sub&gt;M&lt;/sub&gt;</td>
<td>V_X&lt;sub&gt;M&lt;/sub&gt;</td>
</tr>
<tr>
<td>Do/t (Do/t*)</td>
<td>200 (207)</td>
<td>300 (308)</td>
</tr>
<tr>
<td>Do mm</td>
<td>2600</td>
<td>3900</td>
</tr>
<tr>
<td>t mm</td>
<td>13</td>
<td>13</td>
</tr>
<tr>
<td>a mm</td>
<td>6.5</td>
<td>6.5</td>
</tr>
<tr>
<td>2cθ mm</td>
<td>350</td>
<td>400</td>
</tr>
<tr>
<td>2cL mm</td>
<td>250</td>
<td>300</td>
</tr>
<tr>
<td>p MPa</td>
<td>0.9</td>
<td>0.6</td>
</tr>
<tr>
<td>pL MPa</td>
<td>1.54</td>
<td>1.01</td>
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<td>ML kN.m</td>
<td>21570</td>
<td>49100</td>
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<td>Analysis Method</td>
<td>General</td>
<td>General</td>
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<tr>
<td>M_pl.init kN.m</td>
<td>12400</td>
<td>10100</td>
</tr>
<tr>
<td>M_TES kN.m</td>
<td>14120</td>
<td>13660</td>
</tr>
<tr>
<td>M_max kN.m</td>
<td>16880</td>
<td>19010</td>
</tr>
<tr>
<td>General: Static general analysis</td>
<td>Riks: Modified Riks method</td>
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Table 4 Safety margin of Donnell’s equation of FS=2 for TES and buckling load or plastic instability load (M_max)

<table>
<thead>
<tr>
<th>Do/t*</th>
<th>(\sigma_{ys}^{\text{mean}}) BASE</th>
<th>(\sigma_{ys}^{\text{min}}) BASE</th>
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</thead>
<tbody>
<tr>
<td></td>
<td>V_XII&lt;sub&gt;M&lt;/sub&gt;</td>
<td>V_X&lt;sub&gt;M&lt;/sub&gt;</td>
</tr>
<tr>
<td></td>
<td>V_XII&lt;sub&gt;M&lt;/sub&gt;</td>
<td>V_X&lt;sub&gt;M&lt;/sub&gt;</td>
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<tr>
<td>207</td>
<td>308</td>
<td>325</td>
</tr>
<tr>
<td>(M_TES/ML)/(M&lt;sub&gt;rcutoff&lt;/sub&gt;(Donnell:FS=2))</td>
<td>1.26</td>
<td>1.80</td>
</tr>
<tr>
<td>(M_max/ML)/(M&lt;sub&gt;rcutoff&lt;/sub&gt;(Donnell:FS=2))</td>
<td>1.52</td>
<td>2.03</td>
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