A Collapse Stress Analysis of a Heat Exchanger Subjected to External Pressure in a Nuclear Power Plant

Jae-Do Kwon*, Yong-Sun Lee, Choon-Yeol Lee, Seung-Wan Woo
Department of Mechanical Engineering, Yeungnam University

The collapse pressure of tubes is determined experimentally by Tschoepe and Maison for various materials with different geometries. The results are compared with those obtained by ASME Codes UG-31 and UG-28. A collapse pressure is the pressure required for the incipient yielding stress of the tubes with and without ovality. This collapse pressure is compared with the experimental results by Tschoepe and Maison. The present investigation is towards finding the collapse pressure required to bring the entire wall of tubes into a state of plastic flow for the pipes, with ovality and without ovality. This collapse pressure is compared with the collapse pressure obtained through experiments in the present investigation. The experimental results are compared with the pressure obtained by FEM (finite element methods). The FEM results are then compared with results obtained through an approximate plastic analysis of the strain hardening material, SA312-TP304 stainless steel. The structural integrity evaluation is performed for the heat exchanger used in an actual nuclear power plant by using various methods described in this paper. The results obtained by the various analyses and the FEM are discussed. Consequently, the paper is oriented towards an actual design purpose of a heat exchanger in an industrial environment, rather than for the purpose of an academic research project investigation.

Key Words : Collapse Pressure, Heat Exchanger, Elasticity, Plasticity, Perfect Plastic, Strain Hardening, Ovality, Structural Integrity

1. Introduction

In recognition of the limitation of the retracted figure UG-31 of the ASME Boiler and Pressure Vessel Code, Section Ⅷ, Division 1 and in the interest of confirming the adequacy of external pressure charts used for the design of tubes, the subcommittee on shells of the Pressure Vessel Research Committee sponsored an experimental program in fiscal years 1977-1978. The committee developed rules in the form of charts which could be applied to steel vessels at a pressure less than 3.4 MPa and for temperatures not exceeding 371℃. The allowable external pressure was fixed at one-fifth of the collapse pressure (Greene Jr.,). The original committee on the strength of vessels under external pressure was chartered in the early 1930's to review existing empirical formulas for vessels subjected to external pressures. The second major effort of the experimental program for pressure vessels subjected to external pressures in the last 50 years is described in ISSN 0043-2326 (Tschoepe and Maison, 1983). ISSN 0043-2326 is unique in that it takes into account dimensional parameters such as diameter to length (D/L), diameter to thickness (D/t), material variability and initial imperfections due to permissible tolerances in fabrication (ovality) which were ignored in the original consideration. The results obtained through the experimental program are compared with those obtained by the UG-31 and UG-28 of

* Corresponding Author,
E-mail : jdkwon@ynucc.yeungnam.ac.kr
TEL : +82-53-810-2462 ; **FAX** : +82-53-813-3703
Department of Mechanical Engineering, Yeungnam University, 214-1 Dae-dong, Kyungsan, Kyungbuk 712-749, Korea. (Manuscript **Received** January 14, 2000 ; **Revised** August 10, 2000)

the ASME Code and the results obtained by the analysis of elastic instability. The analysis performed in ISSN 0043-2326 is considered as the pressure that is required for an incipient yielding to be the collapse pressure, assuming that there are perfect plastic materials and the results of the analysis are compared with those of the experiments(Tschoepe and Maison, 1983).

The paper describes the collapse pressure as the pressure required to make a cross section of tubes result in complete plasticity. The collapse pressure is approximately determined, including the ovality of the specimen used in ISSN 0043-2326, using the "thin or thick tubes" theory. The analysis of the collapse pressure is performed, assuming that the materials are perfect plastic and strain hardening materials that don't include ovality. The collapse pressure obtained by the FEM is compared with the pressure obtained by analysis and experimentation. The structural integrity of a heat exchanger in a nuclear power plant is portrayed by the scenario established in this paper. Consequently, the present paper is oriented towards a structural design in an industry, rather than strictly for academic purposes.

2. Collapse Pressure of a Heat Exchanger

2.1 Analysis

The collapse pressure which is defined as a pressure required to incipient yielding is obtained using elastic instability including ovality(Timoshenko and Gere, 1961) and these results are compared with the experimental results in ISSN 0043-2326. The elastic instability is given as follows in Timoshenko and Gere, Thoery of elastic stability, 1961 :

$$\sigma_\theta = \frac{P_y}{2t/D_0} + \left(\frac{6P_y}{2t/D_0}\right)\left(\frac{u/t}{1-P_y/P_e}\right) \qquad (1)$$

where $P_e=[2E/(1-\nu^2)](t/D_0)^3$, t : the thickness of a tube, D_0 : the outside diameter, E ; Young's modulus, u : the ovality(difference between the maximum and minimum diameter) and $\sigma_\theta=\sigma_y$.

From Eq. (1) using $\sigma_\theta=\sigma_y$, the pressure P_y can be found. The smaller value of the two roots in Eq. (1) is chosen. If the ovality $u=0$, then Eq. (1) can be written as,

$$\left(P-2\sigma_y\frac{t}{D_0}\right)(P-P_e)=0 \qquad (2)$$

which is the pressure required for incipient yielding based on a thin cylinder. The results obtained by Eq. (1) are compared with the experimental values in ISSN 0043-2326 and those results obtained by ASME Codes, UG-31 and UG-28. Those results are tabulated in ISSN 0043-2326.

The present paper is intended to obtain the pressure required to bring about the entire wall of tubes into a state of perfect plastic flow for pipes with ovality and without ovality. When ovality is included in the analysis, the result is nothing but the approximation. If Tresca's yielding criteria is employed, maximum pressure to bring plastic flow into the entire cross section without ovality is given as,

$$P_{ult}=\sigma_y \ln\frac{r_0}{r_i} \text{(Timoshenko, 1956)} \qquad (3)$$

The tangential stress corresponding to P_{ult} is given as,

$$\sigma_\theta=\sigma_y\left(1+\ln\frac{r_0}{r_i}\right) \qquad (4)$$

If ovality is included in the analysis, the collapse pressure is approximately obtained using,

$$P^*_{ult}=P_{ult}(P_y/P) \qquad (5)$$

Equation (5) is approximately obtained by assuming that the ratio of collapse pressure required to bring incipient yielding with ovality (P_y) to that without ovality(P) would be identical to that of the collapse pressure required to bring complete plastic flow into the cross section with ovality and without ovality. P_{ult} is defined as the ultimate pressure to cause the plastic flow through the entire section without ovality and P^*_{ult} is the corresponding pressure with ovality.

It is noted that Eqs. (1) and (2) are associated with the solution of a thin tube while some of the specimens utilized in the experiments in ISSN 0043-2326 are thick wall tubes($D_0/2t<10$). Therefore, the approximate collapse pressure requiring the entire cross section of pipes with ovality deforming to plastic flow, is found using the thick wall tube theory. Lame's solution of a thick wall cylinder subjected to an external pressure is obtained using the elastic solution.

The external pressure corresponding to σ_θ at $r = r_0 (\nu \sim 1/2)$ is

$$P_0 = \frac{2(t/D_0)(1-t/D_0)\sigma_\theta}{1-2t/D_0(1-t/D_0)} \leq 2(t/D_0)\frac{\sigma_\theta}{1-t/D_0} \tag{6}$$

where $t/D_0 \ll 1$ is assumed. The pressure corresponding to $\sigma_\theta = \sigma_y\{1+\ln(r_0/r_i)\}$ for the thick wall cylinder is

$$P_0 = 2(t/D_0)\sigma_y \frac{1+\ln(r_0/r_i)}{1-(t/D_0)}$$

The numerator is the pressure corresponding to $\sigma_\theta = \sigma_y\{1+\ln(r_0/r_i)\}$ for a thin cylinder. Therefore, the correction factor from the thin cylinder solution to that of the thick wall cylinder is $1/(1-t/D_0)$ and

$$\begin{aligned} P_0/2(t/D_0) &= \sigma_y\{1+\ln(r_0/r_i)\}/(1-t/D_0) \\ &= \sigma_y\{1+\ln(r_0/r_i)\}(4t/D_0)/(1-r_i^2/r_0^2) \end{aligned} \tag{7}$$

The right hand side of Eq. (7) is the corrected tangential stress of the thick wall cylinder where ovality is zero. Equation (7) is substituted into the left hand side of Eq. (1),

$$\sigma_y\{1+\ln(r_0/r_i)\}\frac{4(t/D_0)}{1-R^2} = \frac{P_{ult}^{**}}{2t/D_0} + \frac{6P_{ult}^{**}}{2t/D_0}\left(\frac{u/t}{1-P_{ult}^{**}/P_e}\right) \tag{8}$$

Equations (1) and (8) are solved with respect to P_y and P_{ult}^{**} using various t/D_0 and u/t given in ISSN 0043-2326 for various materials. It is noted that Eqs. (1) and (8) are obtained with the assumption that all materials are perfect plastic materials and these assumptions are conservative for actual design purposes.

2.2 Finite elements and strain hardening plastic analyses

Finite element analysis is performed finding a collapse pressure, for both the perfect plastic and the strain hardening materials. The analysis includes the effect of L/D on collapse pressure conditions. Collapse pressure using the FEM is determined by the pressure where the pressure gradient with respect to radial displacement varies abruptly within the inside surface. This can be found internally in personal computer (PC). The collapse pressure is found for both perfect plastic and strain hardening materials with ovalities and without ovalities and for various L/D. The FEM analyses for strain hardening materials requires the Ramberg-Osgood equation. The experimental data of SA-304 at room temperature (Structural Alloy Hand Book, 1974) is shown in Fig. 1. The comparison of the digitized experimental stress-strain curve with the result of curve fitting is shown in Fig. 2. The Ramberg-Osgood equation is obtained using the results of the curve fitting, and is noted as follows:

$$\frac{\varepsilon}{\varepsilon_y} = \left(\frac{\sigma}{\sigma_y}\right) + 7.1113 \times 10^{-4}\left(\frac{\sigma}{\sigma_y}\right)^{10} \tag{9}$$

where $\varepsilon_y = \sigma_y/E$, $E = 193.1$ GPa, $\sigma_y = 172.4$ MPa.

Equation (9) can be represented in an alternative form as,

$$\varepsilon - \frac{\sigma}{E} = B\sigma^n \tag{10}$$

where $B = 2.823558 \times 10^{-51}$, $n = 1/m = 10$. The left hand side of Eq. (10) is the plastic strain.

If a material is power law material, the stress components of the cylindrical material subjected to an external pressure can be found in Boyle and Spence, Stress analysis for creep, 1983.

$$\sigma_r = \frac{2P_0}{1-R^{2m}}\left\{\left(\frac{r_i}{r}\right)^{2m} - 1\right\}$$

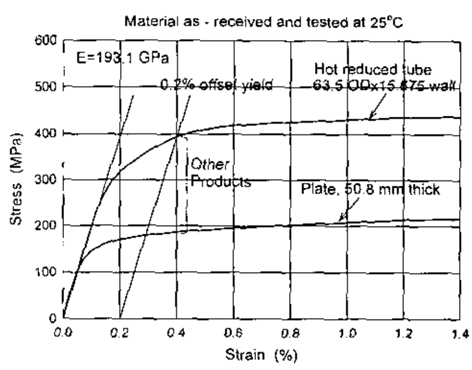

Fig. 1 Stress-strain response of various type 304 stainless products

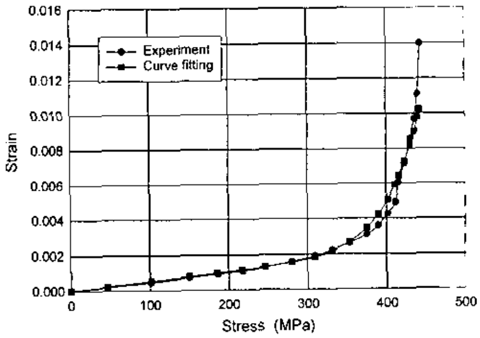

Fig. 2 Comparison of stress-strain curve obtained by experimental(Fig. 1) and curve fitting

$$\sigma_\theta=\frac{2P_0}{1-R^{2m}}\left\{(1-2m)\left(\frac{r_i}{r}\right)^{2m}-1\right\}$$

$$\sigma_z=\frac{1}{2}(\sigma_r+\sigma_\theta)$$

$$\bar{\sigma}=\frac{\sqrt{3}}{2}\left|\sigma_r-\sigma_\theta\right|=\frac{\sqrt{3}}{2}\,\frac{2mP_0}{1-R^{2m}}\left(\frac{r_i}{r}\right)^{2m}$$

$$\varepsilon_r=\frac{3}{4}B(\bar{\sigma})^{n-1}(\sigma_r-\sigma_\theta)=-\varepsilon_\theta$$

$$=\frac{3}{4}B\left(\frac{2m}{1-R^{2m}}\right)^n\left(\frac{\sqrt{3}}{2}\right)^{n-1}\left(\frac{r_i}{r}\right)^2P_0^n \quad (11)$$

As seen in Fig. 1, strain variation with respect to stress variation is very sensitive when the strain exceeds 0.06% and behaves almost as a perfect plastic material. Therefore, effective stress required to bring strain arbitrarily $\varepsilon=0.008$ is found using the equation,

$$E\varepsilon-\bar{\sigma}-EB\bar{\sigma}^n=0 \quad (12)$$

Equation (12) is solved by using the Newton-Raphson numerical method and $\bar{\sigma}$ in Eq. (11). The effective stress $\bar{\sigma}$ corresponding to $\varepsilon=0.008$ is found to be $\bar{\sigma}=465.8$ MPa while $\bar{\sigma}$ is found to be $\bar{\sigma}=427.5$ MPa using Eq. (9). This error is caused by $\sigma_y=172.4$ MPa in Eq. (9) and $\sigma_y=189.6$ MPa in Eq. (10). The yielding stress 189.6 MPa is the yielding stress of SA312-TP304 at 37.8℃ of the header which is used to evaluate structural integrity. The 34.5 MPa error may be tolerated through a curve fitting process. The collapse pressure at $r=r_i$ is found to be $P_0=182.2$ MPa.

Alternatively, the collapse pressure can be approximately obtained using the effective strain,

$$\bar{\varepsilon}=\frac{\sqrt{2}}{3}\left[(\varepsilon_r+\varepsilon_r'-\varepsilon_\theta-\varepsilon_\theta')^2+(\varepsilon_\theta+\varepsilon_\theta')^2+(\varepsilon_r+\varepsilon_r')^2\right]^{1/2} \quad (13)$$

where $\varepsilon_z+\varepsilon_z'=0$ under the plane strain. ε_{ij}' are elastic strains and ε_{ij} are plastic strains. The effective strain in Eq. (13) is bounded by

$$\frac{\sqrt{2}}{3}\{(\varepsilon_r-\varepsilon_\theta)+(\varepsilon_r'-\varepsilon_\theta')\}<\bar{\varepsilon}<\sqrt{\frac{2}{3}}\{(\varepsilon_r-\varepsilon_\theta)+(\varepsilon_r'-\varepsilon_\theta')\} \quad (14)$$

The plastic strain and elastic strain can be found using Eq. (11) and the solution of elasticity. The radial elastic displacement of a cylinder (ovality=0) subjected to an external pressure is given as,

$$u=\frac{1}{2G}\left\{(1-2\nu)\frac{-P_0}{1-R^2}r-\frac{P_0}{1-R^2}\,\frac{r_0^2}{r}\right\}$$

$$\varepsilon_r'=\frac{1}{2G}\,\frac{-P_0}{1-R^2}\left\{(1-2\nu)-\frac{r_i^2}{r^2}\right\} \quad (15)$$

Approximately $\nu\sim 1/2$, then

$$\varepsilon_r'\sim\frac{1}{2G}\,\frac{P_0}{1-R^2}\left(\frac{r_i}{r}\right)^2$$

$$\varepsilon_r'-\varepsilon_\theta'=\frac{2P_0}{2G}\,\frac{1}{1-R^2}\left(\frac{r_i}{r}\right)^2 \quad (16)$$

Plastic strain at $r=r_i$ is

$$\varepsilon_r-\varepsilon_\theta=2\left(\frac{3}{4}\right)B\left\{\frac{2m}{1-R^{2m}}\right\}^n\left(\frac{\sqrt{3}}{2}\right)^{n-1}P_0 \quad (17)$$

The first term of Eq. (13) is

$$\frac{\sqrt{2}}{3}\{(\varepsilon_r-\varepsilon_\theta)+(\varepsilon_r'-\varepsilon_\theta')\}$$

$$=\frac{\sqrt{2}}{3}\left[2\left(\frac{3}{4}\right)B\left(\frac{\sqrt{3}}{2}\right)^{n-1}\left(\frac{2m}{1-R^{2m}}\right)\right]P_0^n+\frac{2P_0}{2G}\,\frac{1}{1-R^2} \quad (18)$$

where $n=10$ and $m=1/n=0.1$.

The pressure required for a total lower bound of an effective strain, with $\varepsilon_L=0.008$ at $r=r_i$ can be obtained using $R=r_i/r_0=16.9926/24.13$ and Eq. (18). The pressure is found to be 186.0 MPa.

Similarly the pressure requiring the third term of the upper bound of an effective strain, $\overline{\varepsilon_u}=0.008$ in Eq. (14) is found to be $P_0=171.2$ MPa. The actual pressure to bring the effective strain in Eq. (14), $\bar{\varepsilon}=0.008$ is bounded between

$$171.2<P_0<186.0 \text{ MPa}$$

The average value of the two bounded pressures is approximately $P_0=178.6$MPa.

2.3 An example of a heat exchanger collapse pressure calculation

Collapse pressure is found using the actual heat exchanger dimensions and pressure in a nuclear power plant. For example, two components of the heat exchanger are illustrated in this section. The dimensions and operational pressure of the header is given as follows. The schematic model of the header is shown in Fig. 3.

◯ Header

○material : SA312-TP304 ○ovality, $u=0$

○test pressure : 21.4 MPa

○yielding stress : $\sigma_y=189.6$ MPa, ultimate stress : $\sigma_u=503.3$ MPa

allowable stress : $\sigma_a=126.2$ MPa, Young's modulus : 192.4 GPa

2.3.1 Collapse pressure of the heat exchanger header

The collapse pressure is obtained using ASME Fig. UG-31. If the allowable stress, 126.2 MPa and $t/D_0=7.1374/49.53=0.15$ are known, then

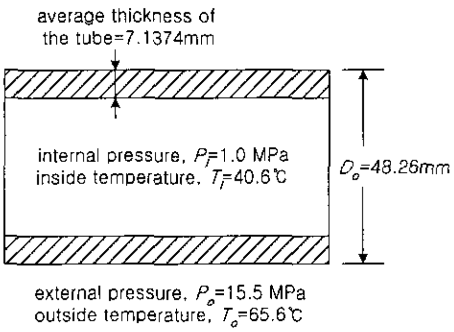

Fig. 3 Dimensions and operational pressure of the header

from Fig. UG-31, the collapse pressure can be calculated as $P_d=17.9$ MPa. The pressure required for incipient yielding can be found from Eq. (2) and the value of the header, $2t/D_0=0.296$. The pressure for the incipient yielding is found to be $P_d=56.1$ MPa. Using a thick wall Lame's solution, the tangential stress, σ_θ is maximum at $r=r_i$,

$$\sigma_\theta=\frac{P_0-P_i}{1-R^2}+\frac{P_iR^2-P_0}{1-R^2}\quad\left(R=\frac{r_i}{r_0}\right)\quad(19)$$

$= -58.5$ MPa (operating pressure, $P_0=15.5$ and $P_i=1.0$ MPa)

$= -67.6$ MPa (testing pressure, $P_t=21.4$ MPa)

Using ASME Code UG-28, the allowable pressure for the header is found to be $P_a=22.6$ MPa. Using Eq. (3), the ultimate pressure to bring the plastic deformation of the entire cross section is found to be $P_{ult}=189.6\times\ln(r_0/r_i)=66.5$ MPa. The safety factors, the ratio of allowable pressure to the applied pressure are summarized in Table 1. The applied pressure is $15.5-1.0=14.5$ MPa for the normal operating condition and $21.4-1.0=20.4$ MPa for the testing pressure.

In Table 1, safety factors associated with UG-31 and UG-28 show minimum value as expected since the codes are made so that the code limit pressure is very conservative. The safety factor of yielding stress (*mark) in Table 1 is large compared with that of the allowable stress (**mark) in Table 1. This is caused by the calculation of the pressure required for the incipient yielding is performed using a thin cylinder formula even though the ratio of mean radius of the header to the average thickness is approximately 2.9. The hoop stress obtained by the thin cylinder formula

Table 1 *Summary of the safety factors of the header obtained using various methods*

UG-31	UG-28	Yielding stress*	Allowable stress**	Eq. (3)***
17.9/14.5=1.24	22.6/14.5=1.55	56.1/14.5=3.87	126.2/58.5=2.16	66.5/14.5=4.59
17.9/20.4=0.88	22.6/20.4=1.10	56.1/20.4=2.75	126.2/67.6=1.87	66.5/20.4=3.26

* : Ratio of pressure required for incipient yielding to applied pressure

** : Ratio of allowable stress to the tangential stress caused by applied pressures 14.5 MPa and 20.4 MPa, respectively

*** : Ratio of Eq. (3) to applied pressure results into a lower value than that obtained by the thick wall cylinder on the inside surface. Therefore, the safety factor associated with *mark in Table 1 is less reliable than that of **mark in Table 1. The safety factor associated with Eq. (3) (***mark) in Table 1 shows maximum value even though a strain hardening effect of the pipe is neglected and this value can be considered as a limiting value of the header collapse pressure. When the ratios of safety factors associated with UG-31 or UG-28 to those of Eq. (3) (***mark) in Table 1 are considered, those ratios are exceeded 3.0. Therefore, the code of UG-31 or UG-28 implies the safety factors more than 3.0.

3. Results and Discussion

The comparisons of experimental collapse pressure with the results obtained by ASME Codes, UG-31 and UG-28 are made in ISSN 0043-2326. ISSN 0043-2326 includes the ratio of the experimental collapse pressure(P^*) to P_y obtained by Eq. (1). The results show that P^*/UG-31 ~ Avg. 6.96 and P^*/UG-28 ~ Avg. 6.0. The ratio of P^*/P_y ranges from a minimum of 1. 28 to a maximum of 2. 42 depending on the tested materials. The collapse pressure obtained by ASME Codes, UG-31 and UG-28 exceeds the safety factor 3 and the codes are very conservative. The tubes would not collapse at the incipient of yielding. The collapse pressure measured by the experiments are compared with that obtained by using Eq. (8) and P^{**}_{ult}. The pressure P^{**}_{ult} is found at room temperature using the geometries of the tubes and the mechanical properties given in ISSN 0043-2326. The ratios P^*/P^{**}_{ult} show a minimum of 1.07 to a maximum of 1.72 which are closer than those ratios obtained by experiment (Tschoepe and Maison, 1983). The deviation between the experimental data and P^*/P^{**}_{ult} or P^*/P_y may be caused by the strain hardening effect of the tested specimens. The values of P^*/P_y and P^*/P^{**}_{ult} of SA-192 (the material tested in ISSN 0043-2326) are given in Table 2. The "*" in the 9th and 12th

Table 2 Failure pressure prediction including plasticity (SA 192)

Steel(SA 192) : σ_y=239.9 MPa, E=189.6 GPa, at 21.1℃ (Experiment)												
T (℃)	σ_y (MPa)	P^*	P_{ult}	P	P_y	$P^*_{ult}=P_{ult}\times P_y/P$	P^*/P^*_{ult}	P^*/P_y	P_e	P^{**}_{ult}	P^*/P^{**}_{ult}	Specimen ID#
21.1	239.9	20.7	20.8	20.2	11.8	12.3	1.722	1.78	30.9	12.7	1.668	S1
21.1	239.9	22.4	20.6	19.7	14.2	14.9	1.501	1.57	28.7	15.2	1.476	S2
21.1	239.9	22.4	20.5	19.7	12.8	13.4	1.671	1.74	28.7	13.7	1.638	S3
21.1	239.9	25.6	24.6	23.5	15.8	16.5	1.551	1.62	49.0	17.2	1.492	S4
21.1	239.9	25.5	24.5	23.0	16.3	17.7	1.445	1.53	46.1	18.1	1.412	S5
21.1	239.9	25.6	24.5	23.0	17.4	18.4	1.392	1.48	46.1	18.9	1.360	S6
21.1	239.9	56.5	29.3	27.4	15.0	16.0	3.527	3.78*	77.2	17.4	3.25*	S14
21.1	239.9	57.9	51.9	46.5	36.6	40.8	1.421	1.58*	380.3	48.8	1.187*	S16
21.1	239.9	58.6	51.7	46.5	33.7	37.4	1.567	1.74*	380.3	45.0	1.303*	S17

*Thick Wall Cylinder

P^* : Experimental collapse pressure in ISSN 0043-2326

P_{ult} : Collapse pressure required to the entire cross section deforms to the rigid plastic flow

P : Collapse pressure, $P=\sigma_y(2t/D_0)$

P_y : Eq. (1)

$P^*_{ult}=P_{ult}\ P_y/P$: Approximate solution of collapse pressure required to the entire cross section deforms to the rigid plastic flow with ovality

P^*/P^*_{ult} : Safety factor P^*/P_y : Safety factor

P_e : Eq. (1) P^{**}_{ult} : Eq. (8)

P^*/P^{**}_{ult} : Safety factor

Table 3 Results of finite element analysis for collapse pressure(MPa)(inside surface)

	Boundary conditions	L/D	Perfectly plastic collapse press.	Strain hardening collapse press.	Remarks
Material SA312-304 Header	Radial=0 Axial=0 complete circle	1.0	113.8	262.0	ovality=0
		2.5	80.7	193.1	
		5.0	76.5	186.2	
		10.0	76.5	186.2	182.2 [see p. 6th, P_0=182.2 MPa] 178.6 [Avg. of Eq. (14), see p. 7th 171.2<P_0<186.0 MPa]
		50.0	76.5	186.2	
	Free end	10	60.0	144.8	no length effect
	Axial=0	10	76.5	180.6	
			(66.5 : Table 1)		
SA192 specimen I.D. S14	ovality u/t 0.111	10	21.1	-	Table 2, P^{**}_{ult}=17.4 Experiment 56.5

columns of Table 2 indicate a thick wall cylinder ($t/D_0>0.05$) and the other tubes indicate a thin cylinder ($t/D_0<0.05$). The difference between the P^*/P_y and the P^*/P^{**}_{ult} of Table 2 is not significant for the thin tubes while those values show a significant difference for the thick wall cylinder. This implies that the thick wall cylinder is more affected by plastic deformation.

The results of the collapse pressure obtained by the FEM analysis are shown in Table 3. The collapse pressure of SA312-TP304 is found by the FEM analysis assuming that the material behaves as both a perfect plastic material and a strain hardening material without ovality. Table 3 includes the effect of L/D on the collapse pressure. ISSN 0043-2326 found that the collapse pressure does not affect it if $L/D \geqq 10$. However, it is found through FEM analysis that if $L/D \leqq 5$, then the collapse pressure does not have an affect on L/D while the collapse pressure increases with a decrease of L/D for both the perfectly plastic and the strain hardening materials. This statement is true when ovalities are not considered. The collapse pressure that includes the strain hardening effect is approximately 2.4 times larger than that of the perfect plastic material for $L/D \leqq 10$. It is noted that the results shown in Table 3 are obtained by using the dimensions of the header described in Section 2.3. The collapse pressure, including the strain hardening effect for the case of zero ovality, is compared with that of the approximate calculations by plastic analysis and the results are shown in Table 3. The error between the FEM result and the analysis is found to be approximately 4% which can be acceptable margin of error. The collapse pressure of SA-192 which is used in the experiment (Tschoepe and Maison, 1983) is compared with the FEM result and the approximate plastic analysis in Table 3. The collapse pressure in the experiment is 56.5 MPa while the FEM analysis and plastic analysis (Eq. (8)) show approximately 21.1 MPa and 17.4 MPa, respectively. These calculations do not agree whatsoever, and the reason is not clear. The ovality of specimen, identity number (I.D) used in ISSN 0043-2326 is S14 (u/t=0.11 and t/D_0=0.058) which is significantly larger than those of other specimens such as u/t=0.041, t/D_0=0.097 (S16) or u/t=0.058, t/D_0=0.097 (S17) as noted in ISSN 0043-2326. The pressure of the incipient yield for the specimen S14 is 27.4 MPa while those of specimens, S16 and S17 are 46.5 MPa when σ_y=239.9 MPa, as shown in Table 2.

The comparison of collapse pressure of the perfect plastic and strain hardening material of

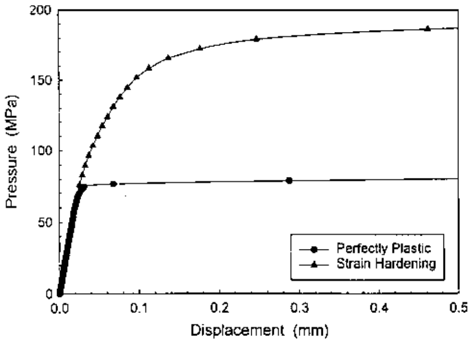

Fig. 4 Collapse pressure of SA312-TP304 (CCW-Header)-Axial and radial direction Fixed, $P_e = 76.5$MPa(perfectly plastic) $P_c = 186.2$MPa(strain hardening), L/D=10 -ovality, u=0

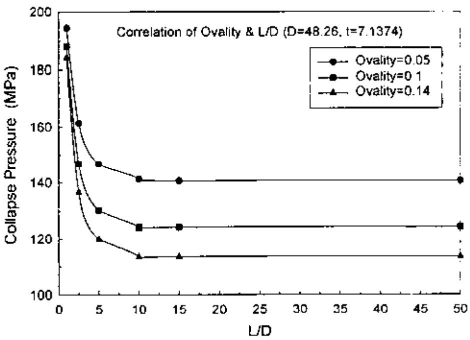

Fig. 5 Collapse pressure of SA312-TP304 (Header) for various L/D -Axial and radial direction Fixed, including ovality and strain hardening

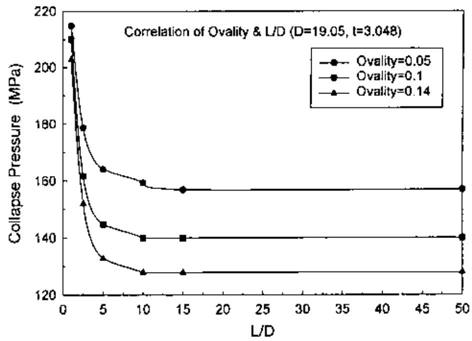

Fig. 6 Collapse pressure of SA213-Grade 316(Cooling tube) for various L/D-Axial and radial direction Fixed, including ovality and strain hardening

SA312-TP304(header) is shown as $L/D=10$ in Fig. 4. The pressure of the perfect plastic material is 76.5 MPa while that of the strain hardening material(Eq. (9)) is 186.2 MPa. Therefore, the collapse pressure is affected significantly by the strain hardening effect. Figures 5 and 6 show the collapse pressure variation with respect to different ovalities. Collapse pressure decreases with an increase in ovality. Figure 5 shows the effect of the collapse pressure on L/D with various ovalities. The pressure is not affected by L/D when the L/D exceeds 10 for strain hardening materials with ovality while it is not affected if $L/D \leq 5$ for perfect plastic materials without ovality.

The collapse pressure which is not affected by L/D requires a larger L/D with ovality than L/D without ovality. Figure 6 is the collapse pressure variation with respect to L/D for various ovalities. The t/D in Fig. 5 is smaller than that in Fig. 6. The collapse pressure in Fig. 6 is larger than that in Fig. 5, which was expected. The collapse pressure of the ovality, $u/t=0.05$ is approximately 141.3 MPa and that of the ovality, $u/t=0.1$ is approximately 124.1 MPa as noted in Fig. 5. The collapse pressure is decreased approximately 13% when $u/t=0.05$ increases to 0.1. The collapse pressure of specimens, S16 and S17 in Table 2 are 57.9 MPa and 60.0 MPa, respectively, which are larger than that of S14(56.5 MPa) as seen in Table 2. These results are consistent with those shown in Figs. 5 and 6. The ovalities of S16 ($u/t=0.041$) and S17($u/t=0.058$) are almost half of the S14 ovality, 0.11. Therefore, it can be conjectured that the collapse pressure of S14 should be at least 13% less than the collapse pressures of S16 and S17, which should be less than $57.9 \times 0.87 = 50.3$ MPa. Therefore, the collapse pressure of S14, 56.5 MPa, may be considered as an experimental error.

4. Summary and Conclusion

The paper reviews collapse pressure(P^*) which was measured in various experiments (Tschoepe and Maison, 1983). The ratios of P^*/UG-31, P^*/UG-28 and P^*/P_y obtained by ISSN 0043-2326 are compared with those obtained by plasticity analysis, including the effect of strain hardening. Collapse pressure of the header and cooling tube used in actual nuclear power plants are obtained by various types of analysis. The scenario to evaluate collapse pressure is established when applied pressure exceeds ASME Code criterions, *UG*-31 and *UG*-28. Consequently, the present paper serves as a guide to evaluate the structural integrity and practicality of a heat exchanger in industries, rather than simply an academic investigation. Through the analyses and calculations of tubes subjected to an external pressure, the following conclusions can be made:

(1) Though the review of ISSN 0043-2326, it can be concluded that:

(a) Even through there are limitations and arguments to use ASME UG-31 charts, UG-31 is safe, and there are plenty of margins for calculating collapse pressures. No conclusions or deductions can be made whether UG-31 is more conservative than UG-28 criterion.

(b) The ratios of experimental measurements to UG-31 or UG-28 exceed safety factor 3, and all tubes do not collapse at the pressure of the incipient of yielding.

(2) Through the FEM and the collapse plastic analyses, it can be concluded that:

(a) The safety factor obtained by the collapse pressure required to bring about plastic deformation along the entire tube cross section using the thick wall cylinder theory is the smallest value. The largest safety factor is the one obtained by the pressure required at incipient yielding using the thin cylinder theory. This conclusion is made without including the strain hardening effect.

(b) The collapse pressure obtained by assuming a perfect plastic material is approximately 42% of the pressure, including the strain hardening effect when $L/D \leqq 5$. The error between the pressure obtained by the FEM and the approximate plastic analysis, including strain hardening effect of SA312-TP304 is 4%. Therefore, the strain hardening plastic analysis employed in the paper can be used for collapse pressure calculations.

(c) The effect of L/D on the collapse pressure is decreased if $L/D \geqq 5$ for either the strain hardening or perfect plastic material, with an zero ovality. The effect of L/D on the collapse pressure with ovality of tubes is diminished if $L/D \geqq 10$ which is in agreement with that in ISSN 0043-2326.

(d) The header and cooling tubes used in the present calculations satisfy both ASME Codes UG-31, UG-28 and the collapse pressure of the heat exchanger is less than the results obtained by various analyses.

Acknowledgement

The results presented here were obtained in the course of a research project supported by a grant from the Korea Electrical Engineering and Science Research Institute, Grant #98-051.

References

Boyle, J. T. and Spence, J., 1983, *Stress Analysis for Creep*, Butterworths, London.

Greene Jr., A. M., History of the ASME Boiler Code, referred from ISSN 0043-2326.

Structural Alloys Hand Book, 1974, Stainless 304, p. 61.

Timoshenko, S. P., 1956, *Strength of Materials, Part* II, D. Van Nostrand Co, New York.

Timoshenko, S. P. and Gere, J. M., 1961, *Theory of Elastic Stability*, Mcgraw Hill, New York.

Tschoepe, E. and Maison, J. R., 1983, The External Pressure Collapse Tests of Tubes, Welding Research Council Bulletin, Bulletin 284, ISSN 0043-2326.