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Seal Plate에 용접되어 있는 Nuclear Class 2 & 3 배관의 국부응력 해석기법 단순화 방안 개발

Development of the Simplified Local Stress Analysis Methodology for the Nuclear Class 2&3 Piping welded to the Seal Plate

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제 출 문

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본 보고서를 "Seal Plate에 용접되어 있는 Nuclear Class 2 & 3 배관의 국부응력 해석기법 단순화 방안 개발"에 대한 기술 보고서로 제출합니다.

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요 약 문

배관의 내부나 외부에는 러그, 버팀대, 보강재 혹은 기타 부착물이 용접되어 있거나 보울트로 체결되는 경우가 많다. 이때 배관의 반경 방향으로의 열팽창, 내력으로 인한 팽창, 축방향 인장 하중으로 인한 원주 방향으로의 수축 등이 이들 부착물에 의해 구속 받게 되고 그 결과 배관계의 부착물 주위에는 불연속 국부 응력이 발생하게 된다. ASME Section III NB-3651.3, NC-3645, ND-3645에서는 Class 1, 2, 3 배관계에 부착물이 있을 경우 설계시 국부응력에 의한 배관 계의 영향을 평가하도록 요구하고 있다.

본 보고서에서는 Seal Plate에 용접되어 있는 Class 2 & 3 배관의 국부응력을 산출하는 단순화 방안을 개발하였고, 이로부터 구한 이론해와 범용 구조해석 프로그램인 ANSYS Ver. 5.1을 사용한 수치 해석 결과를 비교하여 이론해의 타당성을 검증하였다. 이론해와 수치 해석 결과는 잘 일치 하였으며, 전체적으로 이론해가 보수적인 결과를 보여 주었다 이 보수성은 실제 설계시 안전 계수로써 고려될 수 있을 것으로 판단되며, 본 보고서에서 개발된 Seal Plate에 용접되어 있는 Class 2 & 3 배관의 국부응력 해석 단순화 방안은 Seal Pate 설계와 Class 2&3 배관의 국부응력 해석에 효율적으로 이용되리라 기대된다.

ABSTRACT

Lugs, brackets, stiffeners and other attachments may be welded, bolted and studded to the outside or inside of piping and the local stresses arise because of the radial thermal expansion of the piping, the dilatation of the piping due to its internal pressure, the circumferential contraction of the pipe as a results of an axial tensile force, etc., constrained by those. So the evaluation of the local stress for the piping constrained by the attachment in accordance with the ASME Section III, NB-3651.3, NC-3645 and ND-3645 are required for the Class 1, 2, and 3 piping.

In this report, the formula for the local stress analysis for the piping welded to the seal plate was developed and the results from the theoretical analysis were compared with the results analyzed by the ANSYS. The results from the theoretical analysis agree well to the results analyzed by the ANSYS with a conservatism. The conservatism in the theoretical analysis can be considered as a safety factor in the design stage. So, the formula developed in this report can be used very effectively for the design of the seal plate and the local stress analysis of the nuclear class 2&3 piping welded to the seal plate

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1 Introduction

Incompatibility of deformation at the joint due to a discontinuity of the membrane action which occurs at all points of external restraint or at the junction of cylindrical shell and attachment possessing different stiffness characteristics produces bending moments and shearing forces, and these bending moments and shearing forces cause the discontinuity stress at the joint.

In the piping system, lugs, brackets, stiffeners and other attachments may be welded, bolted and studded to the outside or inside of the piping and local stresses arise because of the radial thermal expansion of the piping, the dilatation of the piping due to its internal pressure, the circumferential contraction of the pipe as a results of an axial tensile force, etc, constrained by those. So the evaluation of the local stress for the piping constrained by the attachment in accordance with the ASME Section III, NB-3651.3, NC-3645 and ND-3645 are required for the Class 1, 2 and 3 piping.

The evaluation of the piping due to the local stresses can be done with the finite element method but a lot of time and effort for the modelling of the piping with the attachment are needed. So the efficient methodology are required to evaluate the local stress. In this report, the methodology for the evaluation of the local stress of the Nuclear 2 and 3 piping welded to the circular plate which is embedded in the Reactor Building and Service Building as a seal plate shown in figure 1 using a plate and shell theory was proposed and verified by the results of ANSYS.

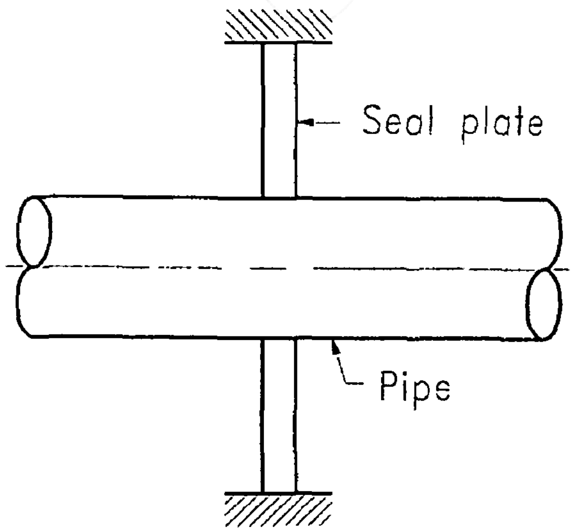

Figure 1 Typical seal plate

2 Formulation of the problem

The assumptions for the formulation are as follows.

1) The plate and shell theory follow the linear elasticity
2) The temperature in the circular plate does not vary over the thickness of the circular plate.
3) The cylinder is infinitely long and the temperature in the cylinder is taken to be symmetrical about axis and independent of the axial coordinate.

2.1 Stress due to thermal expansion and the pressure

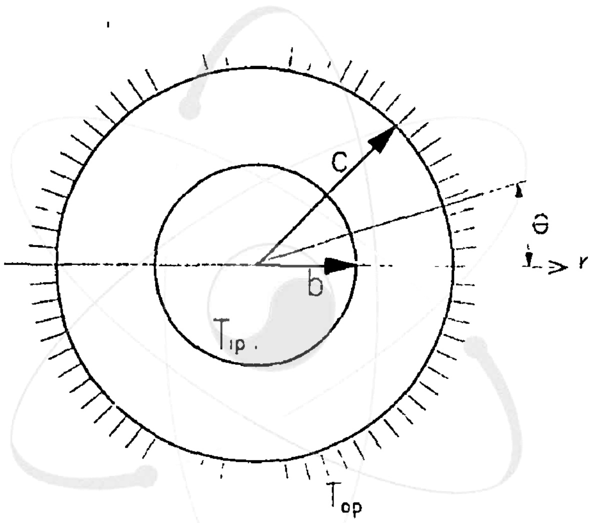

Figure 2. Circular plate with a concentric hole

2.1.1 The displacement of the circular plate in the radial direction due to thermal expansion

2.1.1.1 The displacement equation of the circular plate in the radial direction due to thermal expansion

If ε_r represents the actual radial strain, $\varepsilon_r - \alpha T$ represents the part due to the stress, so the stress-strain relations are

$$\varepsilon_r - \alpha_p T = \frac{1}{E_p}(\sigma_r - \nu\sigma_\theta) \qquad (1)$$

$$\varepsilon_\theta - \alpha_p T = \frac{1}{E_p}(\sigma_\theta - \nu\sigma_r)$$

where

E_p : Young's modulus of the circular plate
α_p : Thermal expansion coefficient of the circular plate
ν : Poisson's ratio

Solving (1) for σ_r and σ_θ

$$\sigma_r = \frac{E_p}{1-\nu^2}\{\varepsilon_r + \nu\varepsilon_\theta - (1+\nu)\alpha_p T\} \tag{2}$$

$$\sigma_\theta = \frac{E_p}{1-\nu^2}\{\varepsilon_\theta + \nu\varepsilon_r - (1+\nu)\alpha_p T\}$$

The equation of equilibrium is

$$\frac{d\sigma_r}{dr} + \frac{\sigma_r - \sigma_\theta}{r} = 0 \tag{3}$$

The unit elongations for the plane-stress are

$$\varepsilon_r = \frac{du_p}{dr} \tag{4}$$

$$\varepsilon_\theta = \frac{u_p}{r}$$

Substituting eqs. (2) and (4) into eq. (3), we obtain

$$\frac{d}{dr}\left[\frac{1}{r}\frac{d(ru_p)}{dr}\right] = (1+\nu)\alpha_p\frac{dT}{dr} \tag{5}$$

Integration of eq. (5) yields

$$u_p = (1+\nu)\alpha_p\frac{1}{r}\int_b^r Tr\,dr + C_1 r + \frac{C_2}{r} \tag{6}$$

$$\sigma_{rp} = -\alpha_p E_p\frac{1}{r^2}\int_a^r Tr\,dr + \frac{E_p}{1-\nu^2}\left[C_1(1+\nu) - C_2(1-\nu)\frac{1}{r^2}\right]$$

2.1.1.2 Temperature distribution for the circular plate

If only conduction is considered, the temperature distribution for the circular plate is

$$T=\frac{T_{op}-T_{ip}}{\ln\frac{c}{b}}\ln r+T_{op}-\frac{T_{op}-T_{ip}}{\ln\frac{c}{b}}\ln c \tag{7}$$

In reality, there will be heat loss due to radiation. If it is considered, the temperature gradient will be steeper. Rather than to solve the nonlinear differential equation, one way to approximate the temperature distribution is to fit $T(r)=a_1 + a_2r + a_3r^2$ to match $T(b)=T_{ip}$, $T(c)=T_{op}$ and $\frac{d}{dr}(T(c))=0$.

Then,

$$T=T_{op}+(T_{ip}-T_{op})\left(\frac{c-r}{c-b}\right)^2 \tag{8}$$

2.1.1.3 The boundary condition and the displacement in radial direction for the circular plate

If the circular plate at the outer diameter is fixed and the circular plate at the inner diameter is free, the boundary conditions are expressed as follows

$$\begin{aligned} u_p\,|_{r=c}&=0 \\ \sigma_r\,|_{r=b}&=0 \end{aligned} \tag{9}$$

Applying the boundary condition (9) into eq. (6), the eq (6) is

$$u_p=(1+\nu)\alpha_p\frac{1}{r}\int_b^r Trdr+C_1r+\frac{C_2}{r} \tag{10}$$

where

$$C_1=-\frac{(1-\nu^2)\alpha_p}{(1-\nu)c^2+(1+\nu)b^2}\int_b^c Trdr$$

$$C_2=-\frac{(1+\nu^2)b^2\alpha_p}{(1-\nu)c^2+(1+\nu)b^2}\int_b^c Trdr$$

$$\int_b^c Trdr=C_3\left(\frac{c^2}{2}-\frac{b^2}{2}\right)+C_4\left\{c^2\left(\frac{c^2}{2}-\frac{b^2}{2}\right)+2c\left(\frac{c^3}{3}-\frac{b^3}{3}\right)+\left(\frac{c^4}{4}-\frac{b^4}{4}\right)\right\}$$

$$C_3=T_{op}$$

$$C_4=\frac{T_{ip}-T_{op}}{(c-b)^2}$$

2.1.2 The displacement of the long cylinder in the radial direction due to thermal expansion

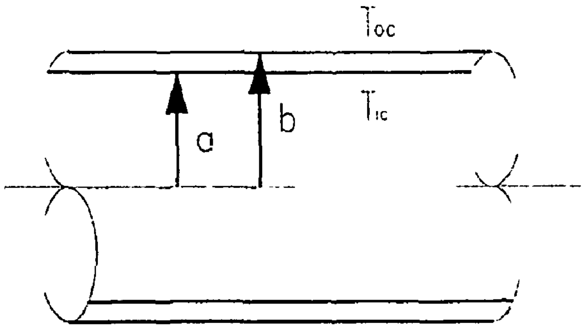

Figure 3 Long cylinder

2.1.2.1 The displacement equation of the long cylinder in the radial direction due to thermal expansion

The stress-strain relations are

$$\varepsilon_r - \alpha_c T = \frac{1}{E_c}[\sigma_r - \nu(\sigma_\theta + \sigma_z)]$$
$$\varepsilon_\theta - \alpha_c T = \frac{1}{E_c}[\sigma_\theta - \nu(\sigma_r + \sigma_z)] \qquad (11)$$
$$\varepsilon_z - \alpha_c T = \frac{1}{E_c}[\sigma_z - \nu(\sigma_r + \sigma_\theta)]$$

where

E_c : Young's modulus of the cylinder
α_c : Thermal expansion coefficient of the cylinder

The unit elongation in the three perpendicular directions are

$$\varepsilon_r = \frac{du_p}{dr}$$
$$\varepsilon_\theta = \frac{u_p}{r} \qquad (12)$$
$$\varepsilon_z = \frac{dw}{dz}$$

Using the symbol for the unit increase in volume, we obtain

$$\Delta = \varepsilon_r + \varepsilon_\theta + \varepsilon_z = \frac{1-2\nu}{E_c}(\sigma_r + \sigma_\theta + \sigma_z) + 3\alpha T \qquad (13)$$

From eq. (12) and (13), we find

$$\begin{aligned} \sigma_r &= \frac{E_c}{1+\nu}(\varepsilon_r + \frac{\nu}{1-2\nu}\Delta) - \alpha_c E_c T \\ \sigma_\theta &= \frac{E_c}{1+\nu}(\varepsilon_\theta + \frac{\nu}{1-2\nu}\Delta) - \alpha_c E_c T \\ \sigma_z &= \frac{E_c}{1+\nu}(\varepsilon_z + \frac{\nu}{1-2\nu}\Delta) - \alpha_c E_c T \end{aligned} \qquad (14)$$

The equation of equilibrium is

$$\frac{d\sigma_r}{dr} + \frac{\sigma_r - \sigma_\theta}{r} = 0 \qquad (15)$$

Substituting eqs. (12) and (14) into eq. (15), we obtain

$$\frac{d}{dr}[\frac{1}{r}\frac{d(ru_c)}{dr}] = \frac{(1+\nu)}{(1-\nu)}\alpha_c \frac{dT}{dr} \qquad (16)$$

Integration of eq. (16) yields

$$\begin{aligned} u_c &= \frac{1+\nu}{1-\nu}\alpha_c \frac{1}{r}\int_a^r Trdr + C_5 r + \frac{C_6}{r} \\ \sigma_{rc} &= -\frac{\alpha_c E_c}{1-\nu}\frac{1}{r^2}\int_a^r Trdr + \frac{E_c}{1+\nu}(\frac{C_5}{1-2\nu} - \frac{C_6}{r^2} + \frac{\nu}{1-2\nu}\varepsilon_z) \end{aligned} \qquad (17)$$

2.1.2.2 The boundary condition and the displacement in radial direction for the long cylinder

If the inner and outer surfaces are free, the boundary conditions are expressed as follows

$$\begin{aligned} \sigma_r |_{r=a} &= 0 \\ \sigma_r |_{r=b} &= 0 \end{aligned} \qquad (18)$$

Applying the boundary condition (18) to eq. (17), equation (17) is

$$u_c = \frac{1+\nu}{1-\nu}\alpha_c \frac{1}{r}\int_a^r Trdr + C_5 r + \frac{C_6}{r} \tag{19}$$

where

$$C_5 = \frac{(1-2\nu)(1+\nu)}{1-\nu}\frac{\alpha_c}{b^2-a^2}\int_a^b Trdr - \nu\varepsilon_z$$

$$C_6 = \frac{(1+\nu)}{1-\nu}a^2\frac{\alpha_c}{b^2-a^2}\int_a^b Trdr$$

$$\int_a^b Trdr = C_7\left(\frac{b^2}{2}\ln b - \frac{a^2}{2}\ln a - \frac{b^2}{4} + \frac{a^2}{4}\right) + C_8\left(\frac{b^2}{2} - \frac{a^2}{2}\right)$$

$$C_7 = \frac{T_{oc} - T_{ic}}{\ln\frac{b}{a}}$$

$$C_8 = T_{oc} - \frac{T_{oc} - T_{ic}}{\ln\frac{b}{a}}\ln b$$

2.1.3 The total displacement of the piping in the radial direction due to thermal expansion and internal pressure.

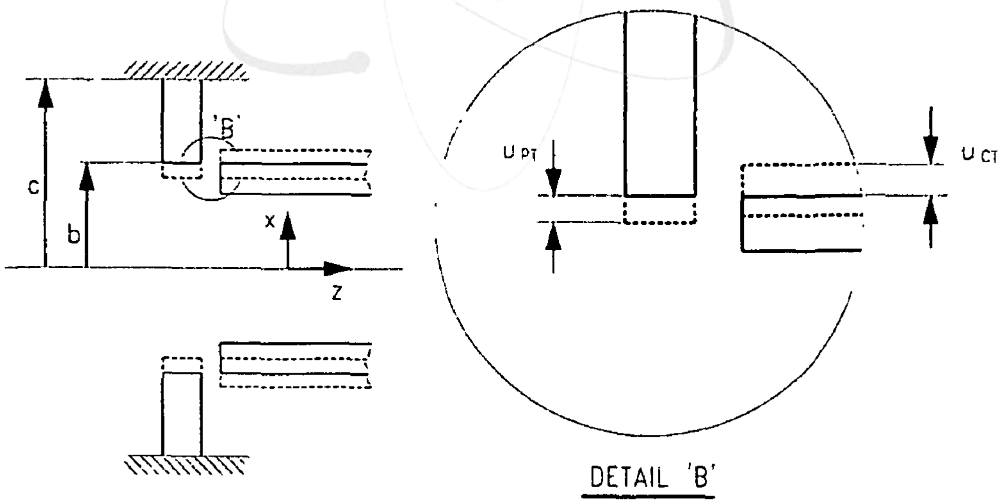

Figure 4. Free body diagram at the junction between circular Plate and long cylinder

The displacement of the circular plate at the inner diameter due to thermal expansion is

$$u_{PT} = C_1^* b + \frac{C_2^*}{b} \tag{20}$$

where

$$C_1^* = \frac{(1+\nu)^2 b^2 \alpha_p}{(1-\nu)c^2 + (1+\nu)b^2} \int_b^c Trdr$$

$$C_2^* = \frac{(1-\nu)^2 \alpha_p}{(1-\nu)c^2 + (1+\nu)b^2} \int_b^c Trdr$$

$$\int_b^c Trdr = C_3(\frac{c^2}{2} - \frac{b^2}{2}) + C_4 \{ c^2(\frac{c^2}{2} - \frac{b^2}{2}) + 2c(\frac{c^3}{3} - \frac{b^3}{3}) + (\frac{c^4}{4} - \frac{b^4}{4}) \}$$

$$C_3 = T_{op}$$

$$C_4 = \frac{T_{ip} - T_{op}}{(c-b)^2}$$

The displacement of the long cylinder at the outer surface due to thermal expansion is

$$u_{CT} = \frac{1+\nu}{1-\nu} \alpha_c \frac{1}{b} \int_a^b Trdr + C_5^* b + \frac{C_6^*}{b} \tag{21}$$

where

$$C_5^* = \frac{(1-2\nu)(1+\nu)}{1-\nu} \frac{\alpha_c}{b^2 - a^2} \int_a^b Trdr - \nu\varepsilon_z$$

$$C_6^* = \frac{(1+\nu)}{1-\nu} a^2 \frac{\alpha_c}{b^2 - a^2} \int_a^b Trdr$$

$$\int_a^b Trdr = C_7(\frac{b^2}{2} \ln b - \frac{a^2}{2} \ln a - \frac{b^2}{4} + \frac{a^2}{4}) + C_8(\frac{b^2}{2} - \frac{a^2}{2})$$

$$C_7 = \frac{T_{op} - T_{ip}}{\ln \frac{b}{a}}$$

$$C_8 = T_{oc} - \frac{T_{oc} - T_{ic}}{\ln \frac{b}{a}} \ln b$$

Therefore, the total displacement of the piping in the radial direction due to thermal expansion is

$$\Delta u_T = u_{CT} - u_{PT} \tag{22}$$

The total displacement in radius, with thermal and pressure effects[3] combined, is

$$\Delta u = \Delta u_T + \frac{Pb^2}{E_c t_c}(1 - \frac{\nu}{2}) \tag{23}$$

2-1-4. Radial constraint force due to Δu

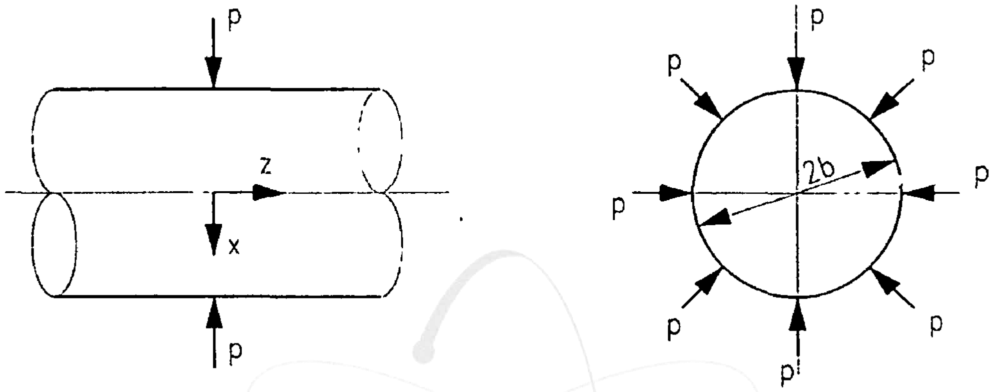

Figure 5. Axisymmetrically loaded cylindrical shell.

The radial constraint force per unit circumferential length is equal to

$$P = 8\beta^3 D \Delta u \tag{24}$$

where

$$\beta^4 = \frac{3(1-\nu^2)}{a^2 t_c^2}$$

$$D = \frac{E_c t_c^3}{12(1-\nu^2)}$$

The corresponding radial shear force and the meridional bending moment as a result of P are equal to

$$Q_{PT} = \frac{P}{2} \tag{25}$$

$$M_z = \frac{P}{4\beta} \tag{26}$$

2-1-5. Unconstrained stress and discontinuity stress.

2-1-5-1. Unconstrained stress

The unconstrained stresses due to internal pressure are as follows

. Longitudinal stress

$$\sigma_z = \frac{Pb}{2t} \tag{27}$$

. Circumferential Stress

$$\sigma_\theta = \nu \frac{Pb}{2t} \tag{28}$$

2-1-5-2. Discontinuity stress.

The discontinuity stress in the pipe caused by Δu are as follows.

. Longitudinal stress

$$\sigma_z = \pm \frac{6M_z}{t_c^2} \tag{29}$$

. Circumferential stress

$$\sigma_\theta = \frac{E_c \Delta u}{b} + \nu \frac{6M_z}{t_c^2} \tag{30}$$

2-1-5-3. Combination of the unconstrained stress and the discontinuity stress

The unconstrained stresses and the discontinuity stresses are to be combined.

. Circumferential stress

$$\sigma_\theta = \nu \frac{Pb}{2t_c} + \frac{E_c \Delta u}{b} + \nu \frac{6M_z}{t_c^2} \tag{31}$$

Longitudinal stress

$$\sigma_z = \frac{Pb}{2t_c} + \frac{6M_z}{t_c} \tag{32}$$

2-2. Stress due to piping loads

2-2-1. Grouping of loading

To simplify the analysis, the piping loads will be grouped into one axial force, one transverse bending moment, and one transverse shear force. The grouping is not unlike the one in ASME Code.

Axial force = F_z

Transverse shear force = $Q_{PL} = \sqrt{F_x^2 + F_y^2}$

Transverse bending moment = $M_\theta = \sqrt{M_x^2 + M_y^2 + M_z^2}$

2-2-2. Unconstrained stresses

. Longitudinal stress

$$\sigma_z = \frac{F_z}{A} + \frac{M_\theta}{Z} \tag{33}$$

where

A : metal area of cross section
Z : section modulus of the pipe

. Circumferential stress

$$\sigma_\theta = \nu \frac{F_z}{A} \tag{34}$$

2-2-3. Discontinuity stresses

The discontinuity stress due to the transverse shear force Q_{PL}, both longitudinal and circumferential is estimated by using the following empirical formula [1]

$$\sigma_{max} = K^{*} \frac{Q_{PL}}{t_c^2} \ln\left(\frac{b}{t_c}\right) \qquad (35)$$

where

$$K^{*} = 0.03 - 0.00017(\theta^{o} - 90^{o})$$

2-2-4. Combination of the unstrained stresses and Discontinuity stresses due to the piping loads.

. Longitudinal stress

$$\sigma_z = \frac{F_z}{A} + \frac{M_\theta}{Z} + K^{*} \frac{Q_{PL}}{t_c^2} \ln\left(\frac{b}{t_c}\right) \qquad (36)$$

. Circumferential stress

$$\sigma_\theta = \nu \frac{F_z}{A} + K^{*} \frac{Q_{PL}}{t_c^2} \ln\left(\frac{b}{t_c}\right) \qquad (37)$$

2-3. Total stress due to pressure, thermal expansion and piping loads

The total stresses are obtained by combining the stresses due to pressure, thermal expansion and piping loads as follows:

For longitudinal stress

$$\sigma_z = \frac{Pb}{2t_c} + \left| \frac{6M_z}{t_c^2} \right| + \frac{F_z}{A} + \frac{M_\theta}{Z} + K^{*} \frac{Q_{PL}}{t_c^2} \ln\left(\frac{b}{t_c}\right)$$

For circumferential stress

$$\sigma_\theta = \nu \frac{Pb}{2t_c} + E\frac{\Delta u}{b} + \left| \frac{6\nu M}{t_c^2} \right| + \frac{\nu F_z}{A} + K^{*} \frac{Q_{PL}}{t_c^2} \ln\left(\frac{b}{t_c}\right)$$

2-4 Maximum stress and stress intensity

For longitudinal stress

$$\sigma_z = \frac{Pb}{2t_c} + \left| \frac{6M_z}{t_c^2} \right| + \frac{F_z}{A} + \frac{M_\theta}{Z} + K \cdot \frac{Q_{PL}}{t_c^2} \ln\left(\frac{b}{t_c}\right)$$

For circumferential stress

$$\sigma_\theta = \nu \frac{Pb}{2t_c} + E\frac{\Delta u}{b} + \left| \frac{6\nu M}{t_c^2} \right| + \frac{\nu F_z}{A} + K \cdot \frac{Q_{PL}}{t_c^2} \ln\left(\frac{b}{t_c}\right)$$

For radial stress

$$\sigma_r = P$$

The stresses which are calculated in the above in longitudinal, circumferential and radial direction are principal stresses because there is no shear stress. So, the maximum stress and the stress intensity can be determined using the following expression.

maximum stress = σ_z or σ_θ

stress intensity = maximum stress + P

3. Results and discussion

The local stresses for the 6″ pipe with various thickness welded to the seal plate with O.D. =10.75″, thickness = 1″ in longitudinal and circumferential direction were calculated by the method proposed in this report. And the local stress analysis for the same models were performed by ANSYS Version 5.1 in order to compare with the results of the theoretical analysis. Since the structure is axisymmetric while some of loadings such as weight and mechanical loads are non-axisymmetric, the analyses utilize the capability of ANSYS axisymmetric structure with non-axisymmetric loading in finite element analysis. For stress analysis the PLANE25 4 node axisymmetric harmonic structural solid element is used in the finite element model. For thermal analysis, temperature distribution are assumed to be axisymmetric and the same finite element model is used except that element type is changed to PLANE55 2-D thermal element The finite element model for the stress analysis is shown in figure 5.

The material properties and dimensions are shown in the Table 1 and Table 2.

The assumptions for the analysis are as follows.

. The temperature of the fluid inside the pipe = 400^{o} F
. The temperature of the outside circular plate = 100^{o} F
. The internal pressure of the pipe = 450 psi

Figure 6 shows the results of the stress intensities obtained from the finite element analysis results by the ANSYS and theoretical analysis results. The theoretical analysis results agree well to the finite element analysis results with about 20% of conservation for the stress intensities. So, if conservatism in the theoretical analysis is considered as a safety factor in the design stage, the methodology proposed in this report can be used for the evaluation of the local stress.

Table 1. Material Properties and Dimension

	PIPING	SEAL PLATE
E	$29. \times 10^{6}$ psi	$29. \times 10^{6}$ psi
ν	0.3	0.3
α	6.5×10^{-6} in/in/ °F	6.5×10^{-6} in/in/ °F
k	1.95×10^{-4} Btu/sec. in. °F	1.95×10^{-4} Btu/sec. in. °F
O.D.	6.625″	10.75″
I.D.		6.625″
t	Refer to Table 3	1″

Table 2. Thickness of the Piping

Model No.	t
1	0.109″
2	0 134″
3	0 219″
4	0.28″
5	0 432″
6	0 562″
7	0.718″
8	0.864″
9	1.0″
10	1.125″

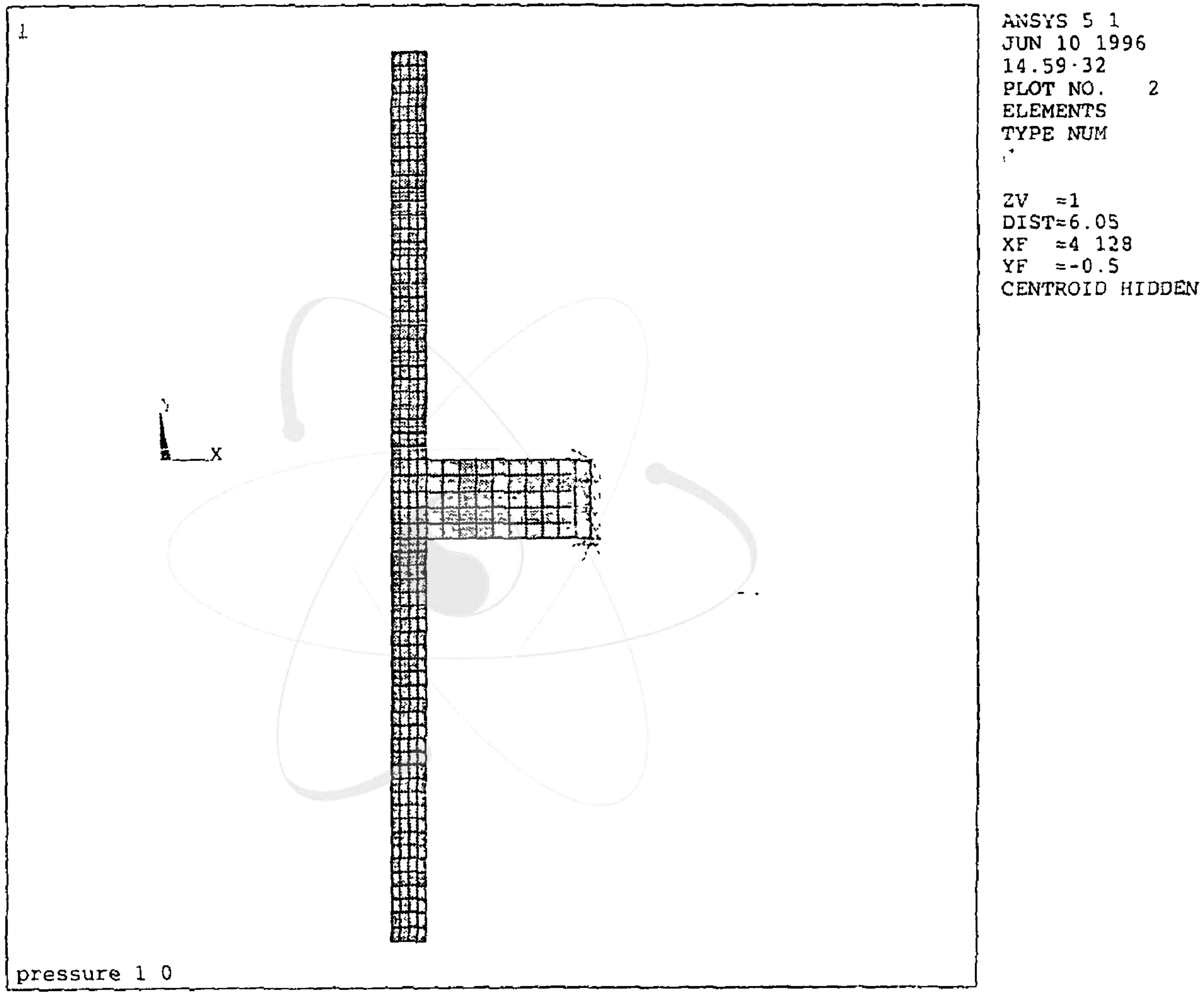

Figure 6 Finite element analysis model

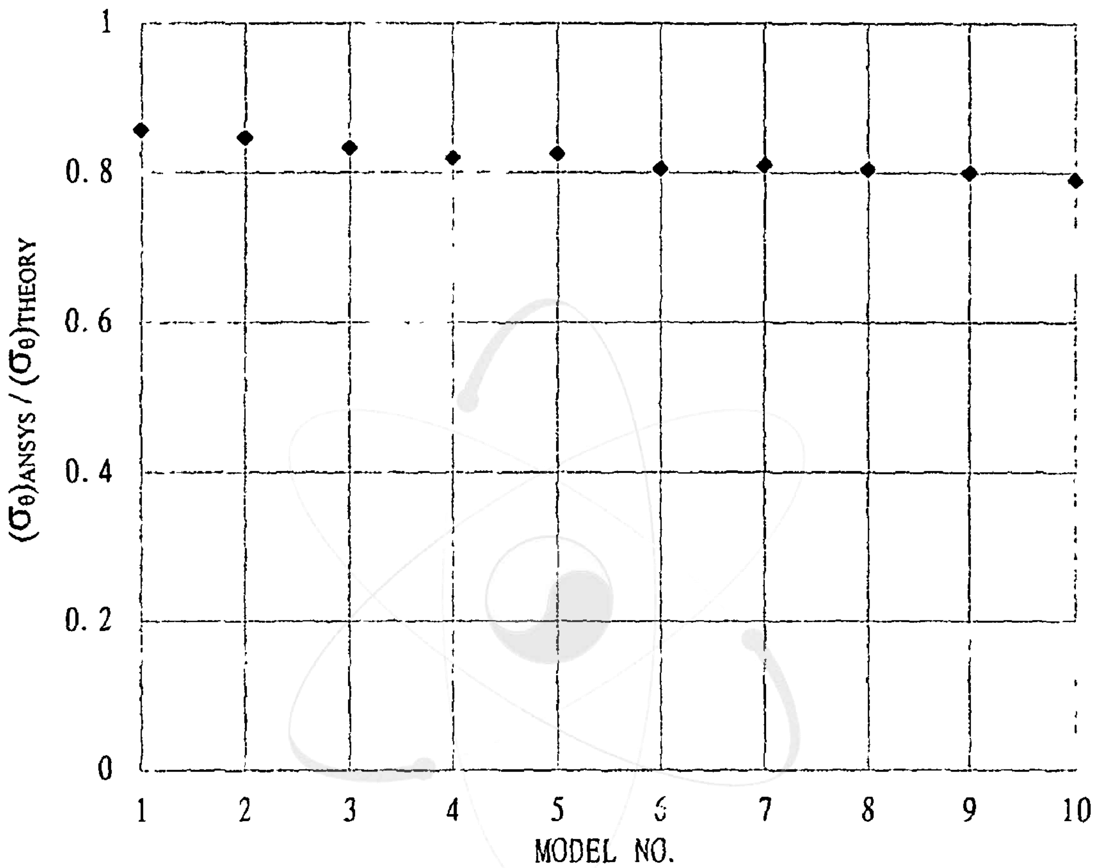

Figure 7 Comparison of the stress intensities at the junction between the piping and seal plate

4. Conclusion

The formula for the local stress analysis for the Nuclear class 2 & 3 piping welded to the Seal Plate was developed and the results from the theoretical analysis were compared with the results analyzed by ANSYS.

The results from the theoretical analysis agree well to the results analyzed by the ANSYS with about 20% conservatism. The conservatism in the theoretical analysis can be considered as a safety factor in the design stage. So, the formula which was developed in this report can be used very effectively for the design of the seal plate and the local stress analysis of the nuclear class 2&3 piping welded to the seal plate.

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서 지 정 보 양 식

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제목/부제	Seal Plate에 용접되어있는 Nuclear Class 2&3 배관의 국부응력 해석기법 단순화 방안 개발
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초록 (15-20줄내외)

배관계에서는 불연속 응력의 일종으로 러그, 버팀대, 보강재 그리고 그 이외의 다른 부착물이 배관의 내부나 외부에 용접되어 있거나 보울트로 체결되어 있어 배관의 반경 방향으로의 열팽창, 내부 압력으로 인한 배관의 원주방향으로의 수축 등으로 국부 응력이 존재하게 되어 ASME Section III NB-3651.3, NC-3645, ND-3645에서는 Class 1, 2 ,3 배관계에 부착물이 있는 경우 국부 응력에의한 배관계의 영향을 평가하도록 요구하고 있다

본 보고서에서는 Seal Plate에 용접되어 있는 Nuclear Class 2&3 배관의 국부 응력 해석을 위한 이론식을 수식화하고 범용 구조 해석 프로그램인 ANSYS Ver 5.1의 해석 결과와 비교하여 이론해로 구한 해석결과가 ANSYS 해석결과와 비교하여 볼 때 다소 보수적이지만 잘 일치하였다 이론해에 의한 보수성은 실제 설계시 안전 계수로써 고려할수 있으리라 사료된다 그러므로 본 보고서에서 개발된 Seal Plate에 용접되어 있는 Nuclear Class 2 & 3 배관의 국부응력 해석 기법 단순화 방안은 Seal Plate 설계와 Nuclear Class 2 & 3 배관의 국부응력해석에 매우 효율적으로 이용되리라 사료된다.

주제명키워드 (10단어내외)	열팽창, 내부압력, 국부응력, 응력해석

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Researcher and Department	Jung, Seung Ha (Power Reactor Mechanical Engineering Department)

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Abstract (15-20 Lines)

Lugs, brackets, stiffners and other attachments may be welded, bolted and studded to the outside or inside of piping and the local stress arise because of the radial thermal expansion of the piping, the dilatation of the piping due to its internal pressure, the circumferential contraction of the pipe as a results of an axial tensile force etc., constrained by those. So, the evaluation of the local stress for the piping constrained by the attachments in accordance with the ASME Section III NB-3651.3, NC-3645 and ND-3645 are required for the nuclear class 1 , 2 and 3 piping.

In this report, the formula for the local stress analysis for the piping welded to the seal plate was developed and the results from the theoretical analysis were compared with the results analyzed by the ANSYS. The results from the theoretical analysis agree well to the results analyzed by the ANSYS with a conservatism. The conservatism in the theoretical analysis can be considered as a safety factor in the design stage. So, the fomula which was developed in this report can be used very effectively for the design of the seal plate and the local stress analysis of the nuclear class 2 & 3 piping welded to the seal plate.

Subject Keywords (About 10 words)	Thermal Expansion, Internal Pressure, Local Stress, Stress Analysis