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ANALYSIS ON INTENSITY OF SINGULAR STRESS FOR BONDED PIPE IN COMPARISON WITH BONDED PLATE

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Abstract: In our previous research, the intensity of singular stress at the end of interface for bonded plate was discussed under arbitrary material combinations. Also, it was found that the bonded strength of butt joint can be evaluated in terms of the singular stress in good accuracy. In this study, the intensity of singular stress for bonded pipe is newly discussed in comparison with the one of bonded plate. The finite element method is applied to calculate the intensity of singular stress with varying the material combination systematically. This method focuses on the result of first node, which locates on the end of the interface. Since few studies are available for bonded pipe, in this study, first, the singular stress field at the end of the interface of the bonded pipe is investigated under several boundary conditions. Next, the effect of the material combination on the intensity of singular stress is discussed. This investigation may contribute to a better understanding of the debonding strength and initial interfacial cracks of bonded pipe.

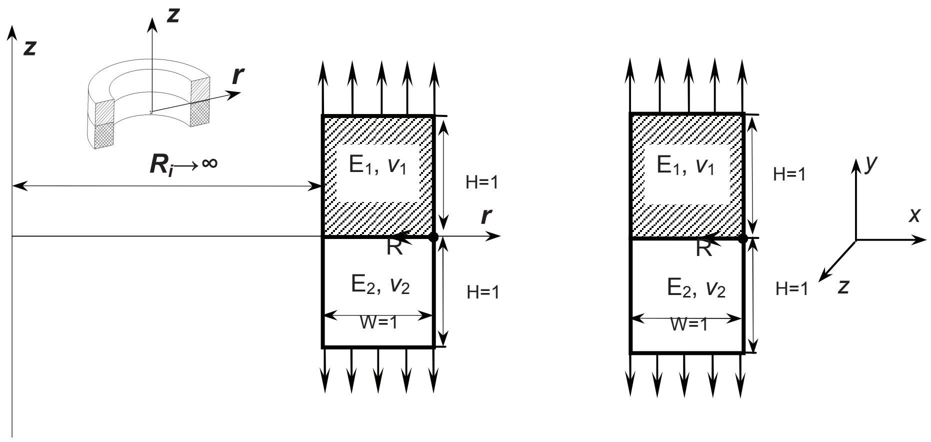
Keywords: adhesion; fracture mechanics; stress intensity factor; interface; bonded pipe

1 INTRODUCTION

The adhesive pipe joints have been widely used in offshore, space and aviation engineering recent years since it has number of advantages over the traditional pipe joint, such as no welding residual stress, lightweight, lower costs, easy to process and corrosion-resistant. With the rapid growth in the use of adhesive pipe joint, many research works have been done to establish the evaluation criteria of this kind of pipe joint [1-7].

However, the improper selection of material combination will cause stress singularity at the end of interface, which may result in the failure of the joint. Thus a rational selection of material combination is crucial to the strength of the adhesive pipe joint. Noda et al. have investigated the intensity of stress singularity for arbitrary material combination in a boned strip [8]. So far only few researches have considered the stress intensity of adhesive pipe joint, and no result of arbitrary material combination has been obtained.

In order to obtain the stress intensity near the corner interface, a basic result is necessary. Teranishi and Nisitani proposed a highly accurate numerical method named the zero element method to determine the stress intensity factor of a homogenous plate [9]. Anyway, this method cannot be used directly into the problem of adhesive pipe since there are non singular terms in stress components. In this research, the stress intensity will be evaluated by using an extended method proposed by Oda et al. FEM is also employed in this paper.



(a) Bonded pipe (b) Plate

Fig. 1 Structure of bonded pipe and reference problem

There are many kinds of adhesive pipe joints; the most commonly used joining methods for pipes are adhesive-bonded socket joints, tubular lap joints, heat-activated coupling joints, and flanged joints. In this research, the basic adhesive bonded pipe shown in Fig.1 is studied. Figure 1(a) shows the structure of pipe joint, and Fig. 1(b) shows the structure of the bonded strip under plain strain condition, which is the reference problem of this research. This research focuses on the intensity of singular stress of arbitrary material combination in a bonded pipe. For the sake of universality, the inner radius of the pipe is chosen as infinite. In this study we assume R_i=10^5.

2 PROBLEM DESCRIPTIONS

The plain strain problem shown in Fig. 1(b) is used as the reference problem, in which the stress near the end of interface can be expressed as

$$\sigma_i^{PLT} = \frac{K_{\sigma_i}^{PLT}}{R^{1-\lambda}} \left(i = x, y, z\right), \tau_{xy}^{PLT} = \frac{K_{\tau_{xy}}^{PLT}}{R^{1-\lambda}} \tag{1}$$

Here R is the distance from the end of interface. This problem has been analyzed by Chen-Nisitani and Noda et al., and the intensity of singular stress was accurately calculated by using body force method (shown in Fig. 2) [10].

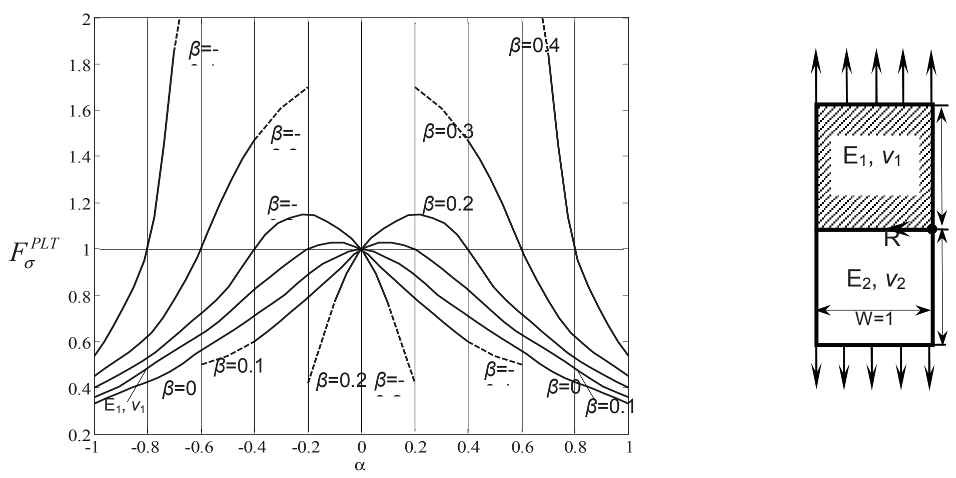


Fig. 2 F_σ for a boned strip in Fig. 1(b)

F_σ is the dimensionless of intensity of singular stress defined by

$$F_\sigma^{PLT} = \frac{K_\sigma^{PLT}}{\sigma_y^\infty (2W)^{1-\lambda}}, \tag{2}$$

While the stress in the unknown problem shown in Fig. 1(a) is expressed as:

$$\sigma_j^{PIPE} = \frac{K_{\sigma_j}^{PIPE}}{R^{1-\lambda}} + \tilde{\sigma}_j^{PIPE} \ (j = r, z, \theta), \ \tau_{rz}^{PIPE} = \frac{K_{\tau_{xy}}^{PIPE}}{R^{1-\lambda}} + \tilde{\tau}_{rz}^{PIPE} \tag{3}$$

This research focuses on the intensity of singular stress of arbitrary material combination in a bonded pipe. To obtain the intensity of singular stress, the zero element method is used. However, according to this method, the stress ratio $\sigma_{ij}^{PIPE}/\sigma_{ij}^{PLT}$ should be consistent with each other and independent of element size when FEM is employed. Table 1 is the results for plate and butt joint. The ratios of all stress components show very good consistent with each other. However the non singular terms in equation (3) can lead uncertainty for the ratio of bonded pipe and plate. Table 2 shows the results of this ratio if we directly apply the zero element method to our new problem. As we can see in Table 1, the ratios of $\sigma_{z0,FEM}$ and $\sigma_{\theta 0,FEM}$ are quite different from that of $\sigma_{r0,FEM}$ and $\tau_{rz,FEM}$.

Table 1 Ratio of $\sigma_{ij0,FEM}^{PLT} / \sigma_{ij0,FEM}^{BJ}$ $(E_1 = 1000, \nu_1 = 0.25548042, E_2 = 113.79748, \nu_2 = 0.20656946)$

	$\sigma_{r0,FEM}^{PIPE}/\sigma_{x0,FEM}^{PLT}$		$\sigma_{z0,FEM}^{PIPE}/\sigma_{y0,FEM}^{PLT}$		$\sigma_{\theta 0,FEM}^{PIPE}/\sigma_{z0,FEM}^{PLT}$		$\tau_{rz,FEM}^{PIPE}/\tau_{xy,FEM}^{PLT}$	
Material	Mat. 1	Mat. 2	Mat. 1	Mat. 2	Mat. 1	Mat. 2	Mat. 1	Mat. 2
$e_{min} = 2^{-13}$	0.5253	0.5254	0.5254	0.5254	0.5254	0.5253	0.5253	0.5254
$e_{min} = 2^{-17}$	0.5250	0.5253	0.5252	0.5252	0.5252	0.5252	0.5252	0.5253

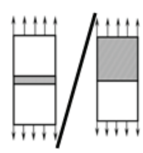
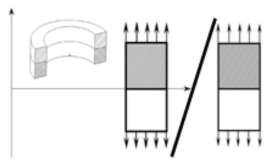


Table 2 Ratio of $\sigma_{ij0,FEM}^{PIPE} / \sigma_{ij0,FEM}^{PLT}$ $(E_1 = 1000, \nu_1 = 0.25548042, E_2 = 113.79748, \nu_2 = 0.20656946)$

	$\sigma_{r0,FEM}^{PIPE}/\sigma_{x0,FEM}^{PLT}$		$\sigma_{z0,FEM}^{PIPE}/\sigma_{y0,FEM}^{PLT}$		$\sigma_{\theta 0,FEM}^{PIPE}/\sigma_{z0,FEM}^{PLT}$		$\tau_{rz,FEM}^{PIPE}/\tau_{xy,FEM}^{PLT}$	
Material	Mat. 1	Mat. 2	Mat. 1	Mat. 2	Mat. 1	Mat. 2	Mat. 1	Mat. 2
$e_{min} = 2^{-13}$	1.0207	1.0207	***1.0148***	***1.0135***	***-0.12834***	***0.88466***	1.0206	1.0207
$e_{min} = 2^{-17}$	1.0204	1.0204	***1.0163***	***1.0154***	***0.22911***	***0.92672***	1.0204	1.0203



Therefore the zero element method can not be employed directly. It is necessary to eliminate the affect of these non singular terms in Eq.(3) for the use of zero element method. Then next chapter will mainly focus on how to make the zero element method suitable for the new unknown problem shown in Fig. 1(a).

3 NUMERICAL METHOD FOR THE ANALYSIS OF THE STRESS INTENSITY FOR BONDED PIPE

At the end of interface, the second terms in equation (3) have the expressions as

$\left(\tilde{\sigma}_{r0}^{PIPE}\right)^{\text{mat1}}, \left(\tilde{\sigma}_{z0}^{PIPE}\right)^{\text{mat1}}, \left(\tilde{\sigma}_{\theta 0}^{PIPE}\right)^{\text{mat1}}, \left(\tilde{\tau}_{rz}^{PIPE}\right)^{\text{mat1}}$ in material 1;

$\left(\tilde{\sigma}_{r0}^{PIPE}\right)^{\text{mat2}}, \left(\tilde{\sigma}_{z0}^{PIPE}\right)^{\text{mat2}}, \left(\tilde{\sigma}_{\theta 0}^{PIPE}\right)^{\text{mat2}}, \left(\tilde{\tau}_{rz}^{PIPE}\right)^{\text{mat2}}$ in material 2.

These 8 stress components should meet the boundary conditions of bonded interface and free edge of the outer surface. And the compatibility of deformation should also be satisfied. As a result, these components should conform to the following equations.

$$\left(\tilde{\sigma}_{r0}^{PIPE}\right)^{\text{mat1}} = \left(\tilde{\sigma}_{r0}^{PIPE}\right)^{\text{mat2}} = \left(\tilde{\tau}_{rz}^{PIPE}\right)^{\text{mat1}} = \left(\tilde{\tau}_{rz}^{PIPE}\right)^{\text{mat2}} = 0 \tag{4}$$

$$\left(\tilde{\sigma}_{z0}^{PIPE}\right)^{\text{mat1}} = \left(\tilde{\sigma}_{z0}^{PIPE}\right)^{\text{mat2}} = \tilde{\sigma}_{z0}^{PIPE} \tag{5}$$

$$\left(\tilde{\varepsilon}_{\theta 0}^{PIPE}\right)^{\text{mat1}} = \left(\tilde{\varepsilon}_{\theta 0}^{PIPE}\right)^{\text{mat2}} = \tilde{\varepsilon}_{\theta 0}^{PIPE} \tag{6}$$

$$\left(\tilde{\varepsilon}_{r0}^{PIPE}\right)^{\text{mat1}} = \left(\tilde{\varepsilon}_{r0}^{PIPE}\right)^{\text{mat2}} = \tilde{\varepsilon}_{r0}^{PIPE} \tag{7}$$

Transpose Eq. (6), and substitute Eq. (4), (5) into it gives

$$\left(\tilde{\varepsilon}_{\theta 0}^{PIPE}\right)^{\text{mat1}} - \left(\tilde{\varepsilon}_{\theta 0}^{PIPE}\right)^{\text{mat2}} = \frac{1}{E_1}\left[\left(\tilde{\sigma}_{\theta 0}^{PIPE}\right)^{\text{mat1}} - \nu_1\left(\tilde{\sigma}_{z0}^{PIPE}\right)^{\text{mat1}}\right] - \frac{1}{E_2}\left[\left(\tilde{\sigma}_{\theta 0}^{PIPE}\right)^{\text{mat2}} - \nu_2\left(\tilde{\sigma}_{z0}^{PIPE}\right)^{\text{mat2}}\right] = 0$$

Thus

$$\left(\frac{\nu_1}{E_1} - \frac{\nu_2}{E_2}\right)\tilde{\sigma}_{z0}^{PIPE} = \frac{\left(\tilde{\varepsilon}_{\theta 0}^{PIPE}\right)^{\text{mat1}}}{E_1} - \frac{\left(\tilde{\varepsilon}_{\theta 0}^{PIPE}\right)^{\text{mat2}}}{E_2} \tag{8}$$

Similarly, for Eq. (7), there is

$$\frac{\left(\tilde{\sigma}_{\theta 0}^{PIPE}\right)^{\text{mat1}}}{\left(\tilde{\sigma}_{\theta 0}^{PIPE}\right)^{\text{mat2}}} = \frac{1+\nu_2}{1+\nu_1}\cdot\frac{E_1}{E_2} \tag{9}$$

From Eq. (8) and Eq. (9) we can obtain

$$\frac{\left(\tilde{\sigma}_{\theta 0}^{PIPE}\right)^{\text{mat1}}}{\tilde{\sigma}_{z0}^{PIPE}} = -\frac{\nu_1 - \frac{E_1}{E_2}\nu_2}{\frac{\nu_1 - \nu_2}{1+\nu_2}} \tag{10}$$

and

$$\frac{\left(\tilde{\sigma}_{\theta 0}^{PIPE}\right)^{\text{mat2}}}{\tilde{\sigma}_{z0}^{PIPE}} = -\frac{\nu_2 - \frac{E_2}{E_1}\nu_1}{\frac{\nu_2 - \nu_1}{1+\nu_2}} \tag{11}$$

For axis symmetric problem under cylindrical coordinate system, there is

$$\begin{cases} \varepsilon_r = \dfrac{\partial u_r}{\partial r} \\ \varepsilon_\theta = \dfrac{u_r}{r} \\ \gamma_{rz} = \dfrac{\partial u_r}{\partial z} + \dfrac{\partial u_z}{\partial r} \end{cases} \tag{12}$$

Recall Eq.(6) we can obtain:

$$\left(\tilde{\varepsilon}_{\theta 0}^{PIPE}\right)^{\text{mat1}} = \left(\tilde{\varepsilon}_{\theta 0}^{PIPE}\right)^{\text{mat2}} = \tilde{\varepsilon}_{\theta 0}^{PIPE} = \varepsilon_\theta = \frac{u_r}{r} = \frac{1}{E_1}\left\{\left(\tilde{\sigma}_{\theta 0}^{PIPE}\right)^{\text{mat1}} - \nu_1\left[\left(\tilde{\sigma}_{r0}^{PIPE}\right)^{\text{mat1}} + \left(\tilde{\sigma}_{z0}^{PIPE}\right)^{\text{mat1}}\right]\right\} = -\frac{(1+\nu_1)\nu_1 E_2 - (1+\nu_2)\nu_2 E_1}{(\nu_1 - \nu_2)E_1 E_2}\tilde{\sigma}_{z0}^{PIPE}$$

Thus

$$\left(\tilde{\sigma}_{z0}^{PIPE}\right)^{\text{mat1}} = \left(\tilde{\sigma}_{z0}^{PIPE}\right)^{\text{mat2}} = \tilde{\sigma}_{z0}^{PIPE} = -\frac{(\nu_1 - \nu_2)E_1 E_2}{(1+\nu_1)\nu_1 E_2 - (1+\nu_2)\nu_2 E_1}\frac{u_r}{r} = -\frac{(\nu_1 - \nu_2)E_1 E_2}{(1+\nu_1)\nu_1 E_2 - (1+\nu_2)\nu_2 E_1}\frac{u_{r0}^{PIPE}}{R_i + W} \tag{13}$$

Substituting Eq. (13) into Eq. (10), (11) gives

$$\left(\tilde{\sigma}_{\theta 0}^{PIPE}\right)^{\text{mat1}} = \frac{(1+\nu_2)(\nu_1 E_2 - \nu_2 E_1)E_1}{(1+\nu_1)\nu_1 E_2 - (1+\nu_2)\nu_2 E_1}\frac{u_{r0}^{PIPE}}{R_i + W} \tag{14}$$

$$\left(\tilde{\sigma}_{\theta 0}^{PIPE}\right)^{\text{mat2}} = \frac{(1+\nu_1)(\nu_1 E_2 - \nu_2 E_1)E_2}{(1+\nu_1)\nu_1 E_2 - (1+\nu_2)\nu_2 E_1}\frac{u_{r0}^{PIPE}}{R_i + W} \tag{15}$$

And recall Eq. (4)

$$\left(\tilde{\sigma}_{r0}^{PIPE}\right)^{\mathrm{mat1}} = \left(\tilde{\sigma}_{r0}^{PIPE}\right)^{\mathrm{mat2}} = \left(\tilde{\tau}_{rz}^{PIPE}\right)^{\mathrm{mat1}} = \left(\tilde{\tau}_{rz}^{PIPE}\right)^{\mathrm{mat2}} = 0 \tag{16}$$

So far, all the eight non singular terms in Eq. (3) have been solved and can be eliminated from the singular stress calculated by FEM, so that the first element method can be applied to this new unknown problem.

4 NUMERICAL RESULTS AND DISCUSSION

Figure 3 shows one of the FEM model for the bonded pipe. There are two models in this research with the minimum element sizes 2^{-13} and 2^{-17} respectively.

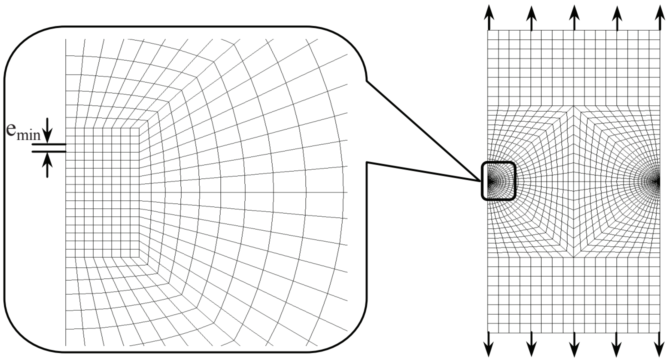


Fig. 3 FEM model

Since the non singular terms have been expressed as shown in Eq.(13)-(16), the ratio $\sigma_{ij}^{PIPE}/\sigma_{ij}^{PLT}$ should be independent of element size when using FEM.

Next we will introduce the results in Eq.(13)-(16) to eliminate the non singular terms. When $e_{min}=2^{-13}$, the displacement of the first node locating on the end of interface (outer surface) is

$$u_r = -73.7971190230$$

And outer radius R_i+W=100001, thus

$$\varepsilon_\theta = \frac{u_r}{r} = \frac{-73.7971190230}{100001} = -7.3797\times 10^{-4}$$

Submit ε_θ into Eqs.(13)-(15), we can get a perfect result of $\sigma_{ij}^{PIPE}/\sigma_{ij}^{PLT}$, which have at least 4 significant digits (See Table 3).

Table 3 Ratio $\sigma_{ij}^{PIPE}/\sigma_{ij}^{PLT}$ excluding non singular terms when $e_{min}=2^{-13}$

$e_{min}=2^{-13}$	$\sigma_{r0,FEM}^{PIPE}/\sigma_{x0,FEM}^{PLT}$		$\dfrac{\sigma_{\theta 0,FEM}^{PIPE}-\tilde{\sigma}_{\theta 0,FEM}^{PIPE}}{\sigma_{z0,FEM}^{PLT}}$		$\dfrac{\sigma_{z0,FEM}^{PIPE}-\tilde{\sigma}_{z0,FEM}^{PIPE}}{\sigma_{y0,FEM}^{PLT}}$		$\tau_{rz0,FEM}^{PIPE}/\tau_{xy0,FEM}^{PLT}$	
Material	Mat. 1	Mat. 2	Mat. 1	Mat. 2	Mat. 1	Mat. 2	Mat. 1	Mat. 2
Average	1.0207		1.0207		1.0207		1.0207	
Separate	1.0207	1.0207	1.0207	1.0207	1.0207	1.0207	1.0206	1.0207

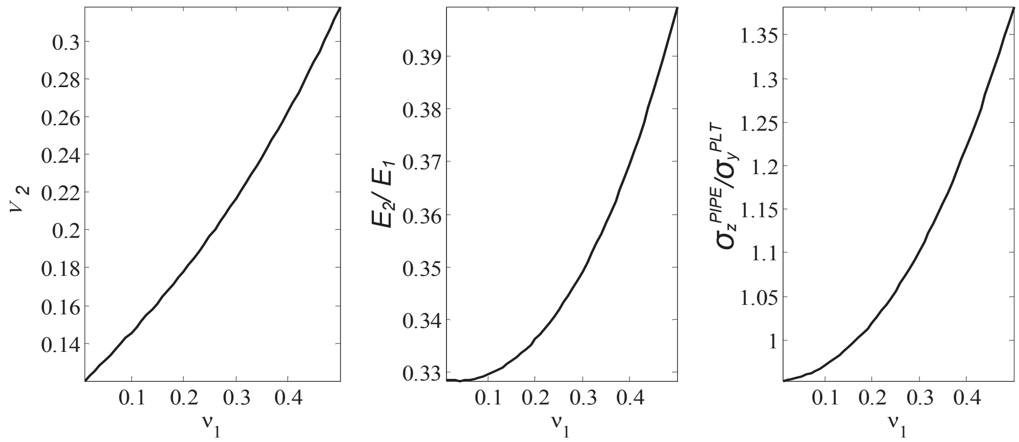
Table 4 Ratio $\sigma_{ij}^{PIPE}/\sigma_{ij}^{PLT}$ excluding non singular terms when $e_{min}=2^{-17}$

$e_{min}=2^{-17}$	$\sigma_{r0,FEM}^{PIPE}\big/\sigma_{x0,FEM}^{PLT}$		$\dfrac{\sigma_{\theta 0,FEM}^{PIPE}-\tilde{\sigma}_{\theta 0,FEM}^{PIPE}}{\sigma_{z0,FEM}^{PLT}}$		$\dfrac{\sigma_{z0,FEM}^{PIPE}-\tilde{\sigma}_{z0,FEM}^{PIPE}}{\sigma_{y0,FEM}^{PLT}}$		$\tau_{rz0,FEM}^{PIPE}\big/\tau_{xy0,FEM}^{PLT}$	
Material	Mat. 1	Mat. 2	Mat. 1	Mat. 2	Mat. 1	Mat. 2	Mat. 1	Mat. 2
Average	1.0204		1.0204		1.0203		1.0203	
Separate	1.0204	1.0204	1.0204	1.0204	1.0204	1.0204	1.0204	1.0204

The result when $e_{min}=2^{-17}$ (see Table 4) also shows the same good consistency as shown in Table 1. It is also found that the ratios are almost independent of element size; the accuracy is up to 4 decimal places. Therefore the zero element method is available for the problem in this research. And the intensity of singular stress in bonded pipe F_{σ}^{PIPE} can be expressed as the multiple of $\sigma_{ij,FEM}^{PLT}$, which has already been solved accurately.

$$F_{\sigma}^{PIPE}=\frac{\sigma_{ij,FEM}^{PIPE}}{\sigma_{ij,FEM}^{PLT}}F_{\sigma}^{PLT}$$

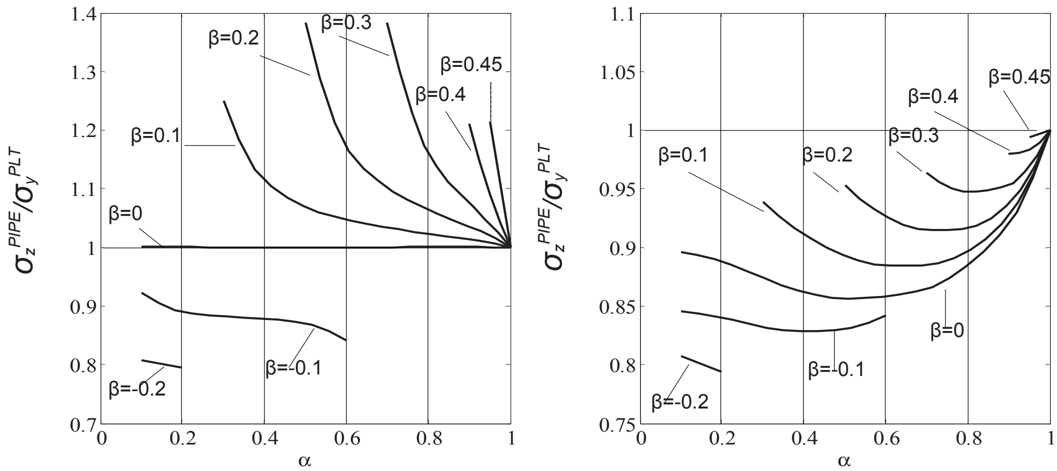
So in this research we mainly consider the ratio $\sigma_{z0,FEM}^{PIPE}\big/\sigma_{y0,FEM}^{PLT}$ when *α, β* are fixed. For plane strain problems, the Dunder's parameter *α* and *β* can fully control the intensity of singular stress near the end of interface, however, for the bonded pipe (axis symmetric problem), the intensity of singular stress can't be totally dominated by these parameters. Figure 4 is the result when $\alpha=0.5$, $\beta=0.2$. Fig. 4 (a) shows thatv_2 varies from 0.1177 to 0.3182 whilev_1 varies from 0 to 0.5; Fig. 4 (b) shows that E_2/E_1 varies from 0.3284 to 0.3994 while v_1 varies from 0 to 0.5. Fig. 4 (c) shows that $\sigma_z^{PIPE}/\sigma_y^{PLT}$ varies from 0.9532 to 1.3796 with the variation range of 42.64%.



(a) (b) (c)

Fig. 4 $\sigma_z^{PIPE}/\sigma_y^{PLT}$ varies when *α* and *β* are fixed as (0.5, 0.2)

Therefore in this study only the maximum and minimum value of $\sigma_{z0,FEM}^{PIPE}\big/\sigma_{y0,FEM}^{PLT}$ are considered. Figure 5 shows this result. In this research, only the results for $\alpha\geq 0$ in *α-β* space have been investigated since switching material 1 and 2 (mat. 1↔mat. 2) will only reverse the sings of *α* and *β* ((*α, β*)↔(-*α, -β*)). So it is unnecessary to draw full map about α and *β*. When $\alpha<0$, the result is the same as that when *α* is positive.



(a) Maximum value of $\sigma_z^{PIPE}/\sigma_y^{PLT}$

(b) Minimum value of $\sigma_z^{PIPE}/\sigma_y^{PLT}$

Fig. 5 Maximum and minimum value of $\sigma_z^{PIPE}/\sigma_y^{PLT}$

5 CONCLUSIONS

In this study the intensity of singular stress for bonded pipe is newly discussed in comparison with the one of bonded plate. First, the non singular terms in stress components of bonded pipe were derived and eliminated so that the proportional method can be applied. This method focuses on the result of first node, which locates on the end of the interface. Then, finite element method is applied to calculate the intensity of singular stress with varying the material combination systematically. And finally the following conclusions can be obtained.

1) The numerical results showed very good consistency among all the ratios of stress components, verifying the rightness of the previous derivation of non singular terms.
2) It is found that the minimum value of stress ratio between axis symmetric problem and plane strain problem is always less than 1, while the maximum value varies from 0.8 to nearly 1.4. The stress ratio for all material combinations converge to 1 if the material combination can reach α=1 or α=-1.
3) Notably, the maximum value keeps constant to 1 when β=0. This result can provide a basic understanding of the intensity of singular stress near the end of interface on a bonded pipe.
4) This investigation may contribute to a better understanding of the debonding strength and provide a better choice of material combination for bonded pipe.

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