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Intensity of singular stress field over the entire bond line thickness range useful for evaluating the adhesive strength for plate and cylinder butt joints

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1. Introduction

Adhesive joints are widely used in numerous industrial sectors, such as automobile, shipbuilding and aeronautics [\[1](#page-15-0)–3]. Compared with the other traditional joints, adhesive joints have advantages of light weight, low cost and easy to process. However, different material properties cause singular stress at the interface end, which may lead to debonding failure in structures $[4-12]$ $[4-12]$. The bonded strength can be expressed in terms of the intensity of the singular stress field (ISSF). The ISSF K_{σ} and the normalized ISSF F_{σ} can be determined from the interface stress as shown in Eq. [\(1\)](#page-0-4) [\[13,14\]](#page-15-2) by using the local polar coordinate (r, θ) indicated in [Fig. 1\(](#page-1-0)a), (b).

$$
\text{ISSF } K_{\sigma} = \lim_{r \to 0} [r^{1-\lambda} \times \sigma_{\theta}(r)|_{\theta = \pi/2}], \text{ Normalized ISSF } F_{\sigma} = \frac{K_{\sigma}}{\sigma W^{1-\lambda}} \tag{1}
$$

Since the singular index $\lambda \neq 0.5$ different from the singular index for cracks $\lambda = 0.5$, the term ISSF (=Intensity of Singular Stress Field) is used instead of SIF (=Stress Intensity Factor) usually used for cracks generally.

[Fig. 2\(](#page-2-0)a) shows the adhesive joint strength for plate butt joint expressed as the critical remote tensile stress *σc* [\[15\].](#page-15-3) It is known that the debonding stress increases with decreasing the adhesive thickness [15–[19\]](#page-15-3). In Ref. [\[19\]](#page-16-0), the effect of joint component mechanical properties and adhesive layer thickness on stress concentration was discussed for a bonded cylindrical specimen. In Ref. [\[7](#page-15-4)–9] the ISSF is

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considered under arbitrary material combinations for $h/W = 0.1$ and 0.001. Our previous studies have indicated that the normalized ISSF decreases with decreasing the bondline thickness as shown in [Fig. 2](#page-2-0)(b) under tension [\[7\]](#page-15-4) and under bending [\[8,9\]](#page-15-5). From the critical remote tensile stress shown in [Fig. 2](#page-2-0)(a), (b), the critical ISSF can be calculated when the debonding occurs. As shown in [Fig. 3,](#page-2-1) the debonding strength can be expressed as a constant value of ISSF [\[12,20\].](#page-15-6)

From the above discussion, it is seen that the solution for ISSF shown in [Fig. 2\(](#page-2-0)b) is quite useful for evaluating the adhesive strength. For large adhesive thickness h, the normalized ISSF F_{σ} becomes constant as shown in [Fig. 2\(](#page-2-0)b), and therefore can be estimated easily for any material combination (see [Appendix A](#page-12-0) [\[14\]](#page-15-7)). However, for small bondline thickness h, which is necessary for evaluating normal adhesive layers, the normalized ISSF F_{σ} decreases with decreasing h and does not become constant. In this paper, therefore, the ISSF vs. h relation will be clarified mainly focusing on the small adhesive thickness. As a threedimensional fundamental solution, the cylindrical butt joint in [Fig. 1\(](#page-1-0)c) is also considered to be compared with plate butt joint. The aim of this paper is to provide the solutions of ISSFs useful for evaluating the adhesive joint strength. In this study, arbitrary material combinations will be considered for the future use of adhesive joint in wide engineering fields.

Nomenclature

1.1. Mesh-independent technique to evaluate the ISSF of plate butt joint

In this section, a mesh-independent technique will be explained for the readers to understand how to obtain accurate ISSFs although similar methods have been used in [\[9,12,20\]](#page-15-8). In the first place, a plate butt joint as shown in [Fig. 1\(](#page-1-0)a) is considered. When the bondline thickness *h* is significantly less than the adhesive width $W(h/W \rightarrow 0)$, the solution may be regarded as the bonded semi-infinite plate as shown in [Fig. 1](#page-1-0)(b). It is known that the interface stress $\sigma_{ii}(ij = rr, r\theta, \theta\theta)$ at the edge can be expressed in the form $\sigma_{ii} \propto 1/r^{1-\lambda}$ when α ($\alpha - 2\beta$) > 0. Notations α and β denote Dundurs' parameters [\[21\]](#page-16-1) expressed by Poisson's ratio ν and shear modulus G , and notation λ denotes the singular index at the interface expressed as the root of the following equations [\[22,23\]](#page-16-2).

$$
\left[\sin^2\left(\frac{\pi}{2}\lambda\right) - \lambda^2\right]^2 \beta^2 + 2\lambda^2 \left[\sin^2\left(\frac{\pi}{2}\lambda\right) - \lambda^2\right] \alpha\beta + \lambda^2 \left[\lambda^2 - 1\right] \alpha^2 + \frac{\sin^2\left(\lambda\pi\right)}{4} = 0\tag{2}
$$

$$
\alpha = \frac{G_1(\kappa_2 + 1) - G_2(\kappa_1 + 1)}{G_1(\kappa_2 + 1) + G_2(\kappa_1 + 1)}, \ \beta = \frac{G_1(\kappa_2 - 1) - G_2(\kappa_1 - 1)}{G_1(\kappa_2 + 1) + G_2(\kappa_1 + 1)}\tag{3}
$$

*σ*_γ[∞], *σ*_{*γ*}[∞] Uniform applied stress

$$
\kappa_j = \begin{cases} \frac{3 - \nu_j}{1 + \nu_j} \text{ (plane stress)} & (j = 1, 2) \\ 3 - 4\nu_j \text{ (plane strain)} \end{cases}
$$
(4)

The ISSF K_{σ} at the adhesive dissimilar joint end is defined from the real interface stress σ_y^{real} as shown in Eq. [\(5\)](#page-1-1).

$$
\text{ISSF } K_{\sigma} = \lim_{r \to 0} [r^{1-\lambda} \sigma_{y}^{real}(r)], \text{ but } \text{ISSF } K_{\sigma} \neq \lim_{r \to 0} [r^{1-\lambda} \sigma_{y}^{FEM}(r)] \tag{5}
$$

However, the ISSF cannot be easily determined by FEM since real interface singular stress σ_y^{real} is different from the FEM stress σ_y^{FEM} , which is largely depending on the mesh size. In the previous papers [\[8,9,12\],](#page-15-5) therefore, the FEM stress ratio $\sigma_y^{FEM}/\sigma_{y(\text{Re}f)}^{FEM}$ was considered by using a reference problem which has been solved very accurately in the previous study. It should be noted that the FEM stress ratio of the unknown reference problems is independent of the mesh size if the same FEM mesh is applied. This is because the FEM errors of two problems are nearly the same. As the reference solution, a simply bonded plate can be used since the ISSF has been analysed very accurately by using the body force method [\[14\]](#page-15-7) (see [Appendix A](#page-12-0)). Since the FEM stress ratio and the reference solution are very accurate, the new results also can be obtained very accurately.

In this study, the ISSF of a simply bonded plate will be used as the reference problem, as is shown in Eq. [\(6\).](#page-1-2)

(a) Plate butt joint (b) Semi-infinite butt joint (c) Cylinder butt joint Fig. 1. Adhesive butt joints ([Fig. 1\(](#page-1-0)b) is equivalent to [Fig. 1\(](#page-1-0)a) when h/W≤0.01 in [Fig. 1\(](#page-1-0)a)).

(a) Critical remote tensile stress σ_c [15] (b) Normalized ISSF F_{σ} under constant σ [12]

Fig. 2. Relationship between critical remote tensile stress *σc*, normalized ISSF *Fσ* and bondline thickness *h*.

$$
\frac{K_{\sigma}}{K_{\sigma(\text{Ref.})}} = \lim_{r \to 0} \frac{[r^{1-\lambda} \sigma_y^{real}]}{[r^{1-\lambda} \sigma_{y(\text{Ref.})}^{real}]} = \lim_{r \to 0} \frac{\sigma_y^{FEM}}{\sigma_{y(\text{Ref.})}^{FEM}}
$$
(6)

Here, the subscript (Ref.) denotes the ISSF or the stress for reference problem.

The finite element analysis is carried out for the butt joints by using the commercial software ANSYS 16.2. [Fig. 4](#page-3-0)(a), (b) shows the FEM mesh for the butt joint for $h/W=0.001$ and the bonded plate for $h/$ $W=1$. Because of symmetry, quarter models are considered for analysis. The finite-element mesh consists of two-dimensional four-node element named PLANE42 and finer subdivisions are used around the interface end. As shown in [Fig. 4\(](#page-3-0)b), the same mesh division pattern is used to eliminate FEM error. The total number of elements have to be larger if the adhesive layer is thin since the interaction between the two interface ends becomes larger. Therefore, the total number of elements 196,794 is necessary for $h/W = 0.001$, but the total number of elements 2560 is enough for $h/W=1$. [Table 1](#page-3-1) shows an example of stress ratio for the butt joint over the bonded plate by using the mesh in [Fig. 4](#page-3-0) with different minimum mesh sizes e_{min} . In [Table 1](#page-3-1) it should be noted that $\sigma_y^{\text{mat1}} = \sigma_y^{\text{mat2}}$, but $\sigma_x^{\text{mat1}} \neq \sigma_x^{\text{mat2}}$, $\sigma_z^{\text{mat1}} \neq \sigma_z^{\text{mat2}}$ at the interface. Here $\sigma_{ij}^{\text{mat1}}$ denotes the stress for matrial 1 and $\sigma_{ij}^{\rm mat2}$ denotes the stress for matrial $2.$

As shown in [Table 1,](#page-3-1) however, all the stress components ratios σ_{ij}^P / σ_{ij}^{REF} are continuous across the interface and coincide with each other. The results are independent of the element size when the mesh

independent technique is employed by using the same FEM mesh pattern.

2. Effect of bond line thickness on the ISSF for plate butt joint

In the previous papers $[7-9,20]$ $[7-9,20]$ for the plate butt joint as shown in [Fig. 1](#page-1-0)(a), the normalized ISSF for bonded plate F^P_σ was defined in Eq. [\(7\)](#page-2-2).

$$
\text{Normalized ISSF } F_{\sigma}^{P} = K_{\sigma}^{P} / \sigma W^{1-\lambda}.\tag{7}
$$

In Eq. [\(7\),](#page-2-2) the ISSF for bonded plate K^P_σ is normalized in terms of the remote tensile stress *σ* and the plate width *W* . This is because the ISSF is controlled by the width *W* . Namely, if *W* becomes larger, the ISSF becomes larger.

However, if the bondline thickness *h* is small, the width *W* does not affect the ISSF K_{σ}^P anymore. Consider a small adhesive thickness joint as shown in [Fig. 1](#page-1-0)(b), which has two singular points at the two interface ends. If *h* becomes smaller, the interaction between two interface ends becomes larger. Therefore, the ISSF is controlled by *h* instead of *W* , and therefore the ISSF K_{σ}^{P} should be normalized by *h* instead of *W*. In other words, for small *h*, the butt joint in [Fig. 1](#page-1-0)(a) can be regarded as the bonded semi-infinite plate as shown in [Fig. 1](#page-1-0)(b). In this case, the ISSF K_{σ}^{P} in [Fig. 1](#page-1-0)(b) should be normalized in terms of the remote tensile stress *σ* and the adhesive thickness *h* as shown in Eq. [\(8\).](#page-2-3)

$$
\text{Normalized ISSF } F_{\sigma}^{P*} = K_{\sigma}^{P} / \sigma h^{1-\lambda}.\tag{8}
$$

[Table 2](#page-4-0) shows F^P_{σ} values and normalized value of $F^P_{\sigma}/F^P_{\sigma}$ $|_{h/W\to\infty}$. [Fig. 5](#page-4-1) shows F_{σ}^{P} vs. h/W relation for several material combinations. As shown in [Table 2](#page-4-0) and [Fig. 5](#page-4-1), when $h/W \geq 1$, the normalized ISSFs F^P_σ are always the same. This is due to Saint'-Venant's Principle stating that the effects of two different but statically equivalent loads are the same at sufficiently large distances from load, that is, $h/W \geq 1$. As shown in [Table 2,](#page-4-0) the normalized ISSF F_{σ}^P has the same value in the range $h/W \geq 1$ since the thickness effect can be negligible.

[Table 3](#page-5-0) shows F_{σ}^{P*} values and [Fig. 6](#page-5-1) shows F_{σ}^{P*} vs. h/W relation. It is seen that when the bondline thickness is small, the F_{σ}^{P*} value always becomes constant. The plate butt joint in [Fig. 1](#page-1-0)(a) can be regarded as a bonded semi-infinite plate in [Fig. 1](#page-1-0)(b) when the relative bondline thickness $h/W \leq 0.01$. From [Figs. 5 and 6,](#page-4-1) it is found that $F_{\sigma}^{P*} = K_{\sigma}^{P} / \sigma h^{1-\lambda}$ is more suitable for small h since F_{σ}^{P*} is insensitive to *h*/*W* compared to $F^P_{\sigma} = K^P_{\sigma}/\sigma W^{1-\lambda}$. As shown in [Table 3,](#page-5-0) the normalized ISSF F_{σ}^{P*} has almost the same value in the range $h/W \le 0.01$ within 0.6% deviation and in the range $h/W \leq 0.1$ within 10% deviation since the width effect is smaller.

Fig. 3. Relationship between K_{∞} and h [12]. [Table 4](#page-6-0) and [Fig. 7](#page-6-1) show the normalized ISSFs F_{σ}^{P*} under arbitrary

(b) Detail around the interface end

Fig. 4. FEM mesh for the plate butt joint $h/W = 0.001$ and simply bonded plate $h/W = 1$.

Table 1 *Mesh-independent FEM stress ratio σ^{<i>i*}₀, *FEM σ*^{*i*}_{*8*}*iFEM* when *E*₁ = 1000, *ν*₁ = 0.23, *E*₂ = 105.06, *ν*₂ = 0.32,h/*W* = 0.001.

$\sigma_{x0, FEM}^{P}/\sigma_{x0, FEM}^{REF}$			$\sigma_{y0, FEM}^{P}/\sigma_{y0, FEM}^{REF}$		$\sigma_{\rm ZO, FEM}^{P}/\sigma_{\rm ZO, FEM}^{REF}$		$\tau_{xy, FEM}^{P}/\tau_{xy, FEM}^{REF}$	
Material	Mat.1	Mat.2	Mat.1	Mat.2	Mat.1	Mat.2	Mat.1	Mat.2
$e_{min} = 2.5^{-15}$ $e_{min} = 2.5^{-18}$	0.3604 0.3604	0.3603 0.3604	0.3604 0.3604		0.3604 0.3604	0.3604 0.3604	0.3603 0.3603	

material combinations useful for h/W≤0.01 and within 10% error for h/ W
subseteq 0.1. Since the solution for bonded plate $h/W \ge 1.0$ is indicated in [Appendix A,](#page-12-0) the accurate results can be obtained by the interpolation in the range for $0.01 \leq h/W \leq 1.0$ under arbitrary material combination.

3. Mesh-independent technique to evaluate the ISSF of cylindrical butt joint

In this section, the mesh-independent technique will be explained for the readers to understand how to obtain accurate ISSFs for cylindrical butt joints although the similar method was used to analyze bonded cylinder and bonded pipe in [\[24\].](#page-16-3) The ISSF of a semi-infinite butt joint K^P_σ has been analyzed in the previous section. To obtain the ISSF of cylindrical butt joint K^C_σ , the new results of K^P_σ can be used as the reference solution. [Table 5](#page-7-0) shows an example of stress ratio for the cylindrical butt joint in [Fig. 1](#page-1-0)(c) over the semi-infinite butt joint in [Fig. 1](#page-1-0)(a). Different ftom [Table 1,](#page-3-1) the ratios of stress components are not always consistent with each other even though the same FE mesh is applied. It should be noted that the value of $\sigma_{\theta 0, FEM}^C / \sigma_{z 0, FEM}^P$ is quite different from other stress ratios. Therefore, we have to consider the mesh-independent technique for axi-symmetric problems in some special aspects.

The difference between [Tables 1](#page-3-1) and [5](#page-7-0) can be explained in the following way. For the plane strain problem as shown in Fig. $1(a)$, the strain in the z-direction is zero. While for the axi-symmetric problem as shown in [Fig. 1](#page-1-0)(c), the strain in the θ direction on the outer cylinder surface can be expressed as [\[25\]:](#page-16-4)

$$
\varepsilon_{\theta} = \frac{u_r}{(W/2)},
$$

which can lead to non-zero stresses [\[24,26\].](#page-16-3) Then the stress of the unknown problem shown in [Fig. 1\(](#page-1-0)c) is expressed as:

 F_{σ}^{P} and $F_{\sigma}^{P}/F_{\sigma}^{P}$ $\vert h/\sqrt{W} \rangle$ of butt joint with varying the bondline thickness.

$$
\sigma_j^C = \frac{K_{oj}^C}{R^{1-\lambda}} + \tilde{\sigma}_{j0}^C = \hat{\sigma}_j^C + \tilde{\sigma}_{j0}^C (j = r, z, \theta),
$$

$$
\tau_{rz}^C = \frac{K_{z_{vy}}^C}{R^{1-\lambda}} + \tilde{\tau}_{rz}^C = \hat{\tau}_{rz}^C + \tilde{\tau}_{rz}^C.
$$
 (9)

where R is the local distance from the axisymmetric interface end.

In Eq. [\(9\),](#page-3-2) the first terms $\hat{\sigma}_{j}^{C}$ and $\hat{\tau}_{rz}^{C}$ denote singular stress and the ̂ second terms $\tilde{\sigma}^C_{j0}$ and $\tilde{\tau}^C_{rz0}$ denote non-singular stress [\[26](#page-16-5)–28] as

$$
(\tilde{\sigma}_{r0}^C)^{\text{matl}}, \quad (\tilde{\sigma}_{z0}^C)^{\text{matl}}, \quad (\tilde{\sigma}_{\theta0}^C)^{\text{matl}}, \quad (\tilde{\tau}_{rz0}^C)^{\text{matl}} \text{ in material 1;}
$$

 $(\tilde{\sigma}_{r0}^C)^{\text{mat2}}, \quad (\tilde{\sigma}_{z0}^C)^{\text{mat2}}, \quad (\tilde{\sigma}_{\theta0}^C)^{\text{mat2}}, \quad (\tilde{\tau}_{rz0}^C)^{\text{mat2}} \text{ in material 2.}$

These eight stress components should satisfy the boundary conditions for bonded interface and free edge of the outer surface as well as the compatibility condition. As a result, they are reduced to the following equations.

Fig. 5. F_{σ}^{P} is constant when $h/W \ge 1.0$.

$$
(\widetilde{\sigma}_{r0}^C)^{\text{matl}} = (\widetilde{\sigma}_{r0}^C)^{\text{matl}} = (\widetilde{\tau}_{rz0}^C)^{\text{matl}} = (\widetilde{\tau}_{rz0}^C)^{\text{matl}} = 0
$$
\n(10)

$$
(\tilde{\sigma}_{z0}^C)^{\text{matl}} = (\tilde{\sigma}_{z0}^C)^{\text{matl}} = \tilde{\sigma}_{z0}^C \tag{11}
$$

$$
\left(\tilde{\varepsilon}_{\theta 0}^C\right)^{\text{matl}} = \left(\tilde{\varepsilon}_{\theta 0}^C\right)^{\text{matl}} = \tilde{\varepsilon}_{\theta 0}^C \tag{12}
$$

$$
\left(\tilde{\varepsilon}_{r0}^C\right)^{\text{matl}} = \left(\tilde{\varepsilon}_{r0}^C\right)^{\text{matl}} = \tilde{\varepsilon}_{r0}^C \tag{13}
$$

By substituting Eqs. (10) , (11) into Eq. (12) , we have

$$
\begin{aligned} (\widetilde{\varepsilon}_{\theta 0}^C)^{\mathrm{matl}} - (\widetilde{\varepsilon}_{\theta 0}^C)^{\mathrm{matl}} &= \frac{1}{E_1} [(\widetilde{\sigma}_{\theta 0}^C)^{\mathrm{matl}} - \nu_1 (\widetilde{\sigma}_{\theta 0}^C)^{\mathrm{matl}}] \\ &- \frac{1}{E_2} [(\widetilde{\sigma}_{\theta 0}^C)^{\mathrm{matl}} - \nu_2 (\widetilde{\sigma}_{\theta 0}^C)^{\mathrm{matl}}] = 0. \end{aligned}
$$

Thus

$$
\left(\frac{\nu_1}{E_1} - \frac{\nu_2}{E_2}\right) \widetilde{\sigma}_z^C = \frac{(\widetilde{\sigma}_{\theta 0}^C)^{\text{matl}}}{E_1} - \frac{(\widetilde{\sigma}_{\theta 0}^C)^{\text{matl}}}{E_2}
$$
\n(14)

Similarly, for Eq. [\(13\)](#page-4-4), we have

$$
\begin{aligned} (\tilde{\varepsilon}_{r0}^C)^{\text{matl}} - (\tilde{\varepsilon}_{r0}^C)^{\text{matl}} &= -\frac{\nu_1}{E_1} \big[(\tilde{\sigma}_{\theta 0}^C)^{\text{matl}} + (\tilde{\sigma}_{z0}^C)^{\text{matl}} \big] \\ &+ \frac{\nu_2}{E_2} \big[(\tilde{\sigma}_{\theta 0}^C)^{\text{matl}} + (\tilde{\sigma}_{z0}^C)^{\text{matl}} \big] = 0. \end{aligned}
$$

Substitute Eq. [\(14\)](#page-4-5) into the above equation, we have

$$
\frac{(\widetilde{\sigma}_{\theta0}^C)^{\text{mat1}}}{(\widetilde{\sigma}_{\theta0}^C)^{\text{mat2}}} = \frac{1 + \nu_2}{1 + \nu_1} \cdot \frac{E_1}{E_2}.
$$
\n(15)

From Eqs. [\(14\)](#page-4-5) and [\(15\)](#page-4-6) we can obtain

$$
\frac{\left(\widetilde{\sigma}_{\theta 0}^C\right)^{\text{mat1}}}{\widetilde{\sigma}_{z0}^C} = -\frac{\nu_1 - \frac{E_1}{E_2}\nu_2}{\frac{\nu_1 - \nu_2}{1 + \nu_2}}.\tag{16}
$$

And

 F_{σ}^{P*} and $F_{\sigma}^{P*}/F_{\sigma}^{P*}$ lh/_{W→0} of butt joint with varying adhesive thickness.

$$
\frac{(\widetilde{\sigma}_{\theta 0}^C)^{\text{mat2}}}{\widetilde{\sigma}_{z0}^C} = -\frac{\nu_2 - \frac{E_2}{E_1}\nu_1}{\frac{\nu_2 - \nu_1}{1 + \nu_2}}.\tag{17}
$$

For axis symmetric problem under cylindrical coordinate system, there is

$$
\begin{cases}\n\varepsilon_r = \frac{\partial u_r}{\partial r} \\
\varepsilon_\theta = \frac{u_r}{r} \\
\gamma_{rz} = \frac{\partial u_r}{\partial z} + \frac{\partial u_z}{\partial r}\n\end{cases}
$$
\n(18)

Recall Eq. [\(12\)](#page-4-3) we can obtain:

$$
(\tilde{\varepsilon}_{\theta 0}^C)^{\text{matl}} = (\tilde{\varepsilon}_{\theta 0}^C)^{\text{matl}} = \tilde{\varepsilon}_{\theta 0}^C = \varepsilon_{\theta} = \frac{u_r}{r}
$$

$$
= \frac{1}{E_1} [(\tilde{\sigma}_{\theta 0}^C)^{\text{matl}} - \nu_1 \{ (\tilde{\sigma}_{r0}^C)^{\text{matl}} + (\tilde{\sigma}_{z0}^C)^{\text{matl}} \}]
$$

$$
= -\frac{(1 + \nu_1)\nu_1 E_2 - (1 + \nu_2)\nu_2 E_1}{(\nu_1 - \nu_2) E_1 E_2} \tilde{\sigma}_{z0}^C
$$

Thus

$$
(\tilde{\sigma}_{z0}^C)^{\text{mat1}} = (\tilde{\sigma}_{z0}^C)^{\text{mat2}} = \tilde{\sigma}_{z0}^C = -\frac{(\nu_1 - \nu_2)E_1E_2}{(1 + \nu_1)\nu_1E_2 - (1 + \nu_2)\nu_2E_1} \frac{u_r}{r}
$$

$$
= -\frac{(\nu_1 - \nu_2)E_1E_2}{(1 + \nu_1)\nu_1E_2 - (1 + \nu_2)\nu_2E_1} \frac{u_r^C}{(W/2)}
$$
(19)

Substituting Eq. [\(19\)](#page-5-2) into Eqs. [\(16\), \(17\)](#page-4-7) gives

$$
\left(\tilde{\sigma}_{\theta 0}^C\right)^{\text{matl}} = \frac{(1+\nu_2)(\nu_1 E_2 - \nu_2 E_1)E_1}{(1+\nu_1)\nu_1 E_2 - (1+\nu_2)\nu_2 E_1(W/2)}\tag{20}
$$

$$
\left(\tilde{\sigma}_{\theta0}^C\right)^{\text{mat2}} = \frac{(1+\nu_1)(\nu_1 E_2 - \nu_2 E_1) E_2}{(1+\nu_1)\nu_1 E_2 - (1+\nu_2)\nu_2 E_1(W/2)} \tag{21}
$$

And recall Eq. [\(10\)](#page-4-2)

$$
(\tilde{\sigma}_{r0}^C)^{\text{mat1}} = (\tilde{\sigma}_{r0}^C)^{\text{mat2}} = (\tilde{\tau}_{rz0}^C)^{\text{mat1}} = (\tilde{\tau}_{rz0}^C)^{\text{mat2}} = 0
$$
\n(22)

The validity of Eqs. (19)–[\(22\)](#page-5-2) to express non-singular stress components will be discussed in [Tables 6](#page-7-1)–8. By using the material combination shown in [Tables 5,](#page-7-0) [6](#page-7-1) shows the radial displacement at the interface end, u_{r0}^C , and the non-singular stresses which are obtained from Eqs. (19)–[\(21\)](#page-5-2) and [\(22\).](#page-5-3) Here, displacement u_{r0}^C is independent of the element size. [Table 7](#page-7-2) shows the singular stresses by subtracting the nonsingular stresses in [Table 6](#page-7-1) from the stresses at the interface end. [Table 8](#page-7-3) shows the ratios of the singular stresses at the interface end of the cylindrical butt joint to those of the semi-infinite butt joint. It is found that the ratio 0.9937 is independent of the element size e_{min} . Since the ratio is also independent of the stress components, the validity of (19)–[\(22\)](#page-5-2) has been confirmed. From the comparison between [Tables](#page-7-0)

[5](#page-7-0) and [8,](#page-7-3) it is seen that $\sigma_{r0, FEM}^C$ and $\tau_{rz, FEM}^C$ do not have the non-singular stresses because $\widetilde{\sigma}_{r0, FEM}^{C} = \widetilde{\tau}_{rz0, FEM}^{C} = 0$. The correct ratio of the ISSF can be calculated from $\sigma_{r0, FEM}^{C}$ and $\tau_{rz, FEM}^{C}$ easily since the subtraction process is not necessary.

4. Effect of bondline thickness on the ISSF for cylindrical butt joint

For plane stress and plane strain problems, Dundurs' parameters (α, β) β) fully control the solution and results [\[21\].](#page-16-1) Under fixed (α , β), therefore, the ISSFs are always the same for plane problems. However, since the cylindrical butt joint is axi-symmetric, (α, β) cannot totally control the ISSFs. [Fig. 8](#page-7-4) shows an example when $(α, β) = (0.8, 0.3)$. Fig. 8(a) and (b) show the possible material combinations under $(\alpha, \beta) = (0.8, 0.3)$. Here, v_2 and E_2/E_1 are calculated by varying v_1 from 0 to 0.5. It can be seen that v_2 changes from 0.183 to 0.250, and E_2/E_1 changes from 0.107 to 0.139. [Fig. 8\(](#page-7-4)c) shows $K_{\sigma}^C/K_{\sigma}^P = [\sigma_{z0, FEM}^C - \tilde{\sigma}_{z0, FEM}^C]/\sigma_{y0, FEM}^P$ and $\sigma_{z0, FEM}^{C}/\sigma_{y0, FEM}^{P}$ calculated by varying v_1 from 0 to 0.5. It is seen that $K_{\sigma}^C/K_{\sigma}^P$ changes from 0.998 to 1.081, and $\sigma_{z0, FEM}^C/\sigma_{y0, FEM}^P$ changes from 0.998 to 1.032. Different from plane problems, $K_{\sigma}^C/K_{\sigma}^P$ and $\sigma_{z0,FEM}^{C}/\sigma_{y0,FEM}^{P}$ are not constants under fixed (α, β). Therefore, in this study the maximum and minimum values will be focused to evaluate the strength of cylindrical butt joint.

For several material combinations, [Table 9](#page-8-0) shows normalized ISSF

Fig. 7. Normalized ISSF F_{σ}^{P*} of a semi-infinite butt joint in [Fig. 1\(](#page-1-0)b) which is useful for $h/W \le 0.01$ in [Fig. 1\(](#page-1-0)a).

 F_{σ}^C defined in Eq. [\(23\).](#page-6-2) And [Fig. 9](#page-8-1) shows F_{σ}^C vs. h/W relation.

$$
\text{Normarized ISSF } F_{\sigma}^C = \frac{K_{\sigma}^C}{\sigma W^{1-\lambda}}.\tag{23}
$$

As shown in [Fig. 9](#page-8-1) when adhesive thickness h is large, the normalized ISSF F_{σ}^C always becomes constant. In [Table 9](#page-8-0), the normalized ISSF F_{σ}^C has the same value in the range $h/W \ge 1$ since the thickness effect can be negligible.

[Table 10](#page-8-2) shows normalized ISSF F_{σ}^{C*} values defined in Eq. [\(24\)](#page-6-3). And [Fig. 10](#page-9-0) shows F_{σ}^{C*} vs. h/W relation.

$$
\text{Normalized ISSF } F_{\sigma}^{C*} = \frac{K_{\sigma}^{P}}{\sigma h^{1-\lambda}}.\tag{24}
$$

It is seen that when the bondline thickness is small, the F_{σ}^{C*} value always becomes constant. From [Figs. 9 and 10,](#page-8-1) it is found that $F_{\sigma}^{C*} = K_{\sigma}^{C}/\sigma h^{1-\lambda}$ is suitable for evaluating the adhesive strength when the bondline thickness is small, because F_c^{C*} is more insensitive to small h/W than $F_{\sigma}^C = K_{\sigma}^C / \sigma W^{1-\lambda}$. As shown in [Table 10,](#page-8-2) the normalized ISSF F_{σ}^{C*} has almost the same value in the range $h/W \le 0.01$ within 0.3% deviation and in the range $h/W \le 0.1$ within 4% deviation since the width effect is smaller.

[Figs. 11 and 12](#page-9-1) show the maximum values of $K_{\sigma}^{C}/K_{\sigma}^{P}$ and the $\sigma_{z0, FEM}^{C}/\sigma_{y0, FEM}^{P}$ by varying α from -0.2 to 1.0 when $\beta = 0.2$ and $\beta = 0.3$. Those values were calculated in a similar way as shown in [Fig. 8.](#page-7-4) For the bad pair *α*(*α* – 2*β*) > 0, the solid line indicates the ISSF ratio K^C_σ/K^F_σ and the broken line indicates the stress ratio $\sigma_{z0, FEM}^C / \sigma_{y0, FEM}^P$. For $\alpha(\alpha - 2\beta)$ > 0, the singular stress appears at the interface end, and therefore $K^C_{\sigma}/K^P_{\sigma}$ may be useful for evaluating the debonding strength [7–[9,12,20\].](#page-15-4) For the good pair α (α – 2 β) < 0, the solid line indicates the stress ratio $(\sigma_{z_0, FEM}^C / \sigma_{y_0, FEM}^P)_{max}$. In this case, the singular stress does not appear at the interface end.

It is found that the ISSF ratio $(K_{\sigma}^C)_{\text{max}}/K_{\sigma}^P \rightarrow \infty$ as $\alpha \rightarrow 2\beta$. However, it should be noted that the singular stress field disappears since the singular index $\lambda \to 1$ as $\alpha \to 2\beta$. Therefore, the stress ratio $(\sigma_{z0, FEM}^C / \sigma_{y0, FEM}^P)_{max}$ may be useful than the ISSF ratio $K_{\sigma}^C / K_{\sigma}^P$ around $\alpha = 2\beta$.

[Figs. 13 and 14](#page-9-2) and [Tables 11 and 12](#page-10-0) show the maximum and minimum values of $K_{\sigma}^C/K_{\sigma}^P$ and $\sigma_{z_0, FEM}^C/\sigma_{y_0, FEM}^P$ calculated by varying (α , β). As mentioned above, $K^C_{\sigma}/K^P_{\sigma}$ is useful for predicting the debonding strength for bad pairs $\alpha(\alpha-2\beta) > 0$, this is because the stress singularity occurs at the interface end when $\alpha(\alpha-2\beta) > 0$. On the other hand, *σ*^{*C*}_{*z*0,*FEM*}/*σ*_{*z*0,*FEM*} may be useful for predicting the debonding strength for good pairs $\alpha(\alpha-2\beta) \leq 0$. However, when $\alpha \cong 2\beta$, it is not known whether

Table 6

Non-singular stresses of cylindrical butt joint.

Table 7

Singular stresses of cylindrical butt joint.

$e_{\rm min}$	$\sigma_{r0, FEM}^{C}$ - $\widetilde{\sigma}_{r0, FEM}^{C}$		$\sigma_{\rm z0, FEM}^{C}$ - $\widetilde{\sigma}_{\rm z0, FEM}^{C}$		$\sigma^C_{\theta 0, FEM}$ – $\widetilde{\sigma}^C_{\theta 0, FEM}$		$\tau_{\text{\tiny{PZO}},\text{\tiny{FEM}}}^{C}$ - $\widetilde{\tau}_{\text{\tiny{PZO}},\text{\tiny{FEM}}}^{C}$	
	Mat.1	Mat.2	Mat.1	Mat.2	Mat.1	Mat.2	Mat.1	Mat.2
2.5^{-15} 2.5^{-18}	-1.5377 -2.3816	0.9911 .5356	4.1917 6.4919		0.6104 0.9454	1.3238 2.0503	0.2144 0.3323	

Table 8

The ratios of singular stresses at the interface e of the cylindrical butt joint and the semi-infinite butt joint.

Fig. 8. (a) v_2 , (b) E_2/E_1 , (c) K_c^C/K_σ^P and $\sigma_{z0, FEM}^C/\sigma_{y0, FEM}^P$ values depending on v_1 under fixed $(\alpha, \beta) = (0.8, 0.3)$.

 $K_{\sigma}^C/K_{\sigma}^P$ or $\sigma_{z0, FEM}^C/\sigma_{y0, FEM}^P$ is suitable for predicting the strength because $(K_{\sigma}^C)_{\text{max}}/K_{\sigma}^P$ goes to infinity when $\alpha \to 2\beta$. [Figs. 13 and 14](#page-9-2) and [Tables 11](#page-10-0) [and 12](#page-10-0) are useful for $h/W \le 0.01$ in [Fig. 1\(](#page-1-0)c). Since the solution for $h/$ $W\geq 1.0$ in [Fig. 1](#page-1-0)(c) was shown in the [Appendix B,](#page-13-0) the accurate results can be obtained by the interpolation also in the range for $0.01 \leq h$ / W≤1.0.

[Fig. 15](#page-10-1) shows the variations of the parameters in the α - β space for

the materials combinations among metal, ceramics, resin, and glass [\[29\]](#page-16-6). Although $(K_{\sigma}^C)_{\text{max}}/K_{\sigma}^P$ in [Fig. 13](#page-9-2) goes to infinity around the equal pair condition, $(K_{\sigma}^C)_{\text{max}}/K_{\sigma}^P$ is less than 1.5 for most of the bad pair region α (α – 2kβ) \geq 0, k = 1.0–0.61(β ² – 0.25) as indicated in [Fig. 15.](#page-10-1)

$$
\frac{(K_{\sigma}^{C})_{\text{max}}}{K_{\sigma}^{P}} \le 1.5 \text{ when } \alpha(\alpha - 2k\beta) \ge 0, \ k = 1.0 - 0.61(\beta^{2} - 0.25) \tag{23}
$$

In the previous studies [\[24\],](#page-16-3) the authors obtained $k = 1.35 - 0.78$ for the bonded cylinder and $k = 1.3 - 0.6\beta$ for the bonded pipe with the infinite inner radius. As shown in [Fig. 15](#page-10-1), the butt joint ISSF ratio satisfies less than 1.5 in the wide range of the bonded cylinder and the bonded pipe. This is because the butt joint has the small 3D effect on the ISSF in comparison with the bonded cylinder and the bonded pipe.

[Fig. 15](#page-10-1) also shows that almost all $(α, β)$ of engineering materials are distributed in $0 \leq |\beta| \leq 0.3$ [\[24\]](#page-16-3), therefore, the stress ratio *σ*_{*ζ*0,*FEM*/*σ*_{*γ*0,*FEM*} can be discussed in this range. It should be noted that the} stress ratio $(\sigma_{z0, FEM}^C/\sigma_{y0, FEM}^P)_{\max}$ is always finite in this range. Comparing [Fig. 13](#page-9-2) with [Fig. 14,](#page-9-3) it is found that the value of $\sigma_{z_0, FEM}^C / \sigma_{y_0, FEM}^P$ varies depending on (α, β) but the value of $(\sigma_{z_0, FEM}^C / \sigma_{y_0, FEM}^P)_{max}$ is in the small range for most of good pairs satisfying $\alpha(\alpha - 2\beta) < 0$ and $0 ≤ |β| ≤ 0.3$. Also, the difference between $(σ_{z0, FEM}^C/σ_{y0, FEM}^P)_{max}$ and $(\sigma_{z0, FEM}^C / \sigma_{y0, FEM}^P)_{min}$ is small in this region. The value range and the maximum and minimum value difference can be expressed in Eq. [\(24\)](#page-6-3).

Table 9 F_{σ}^C and $F_{\sigma}^C/F_{\sigma}^C|_{h/W\to\infty}$ of cylindrical butt joint by varying the bondline thickness.

Fig. 9. F_{σ}^C is constant when $h/W \ge 1.0$.

Table 10 F_{σ}^{C*} and $F_{\sigma}^{C*}/F_{\sigma}^{C*}|_{h/W\to 0}$ of cylindrical butt joint with varying the bondline thickness.

$$
0.971 \le \left(\frac{\sigma_{z0, FEM}^C}{\sigma_{y0, FEM}^P}\right)_{\text{max}} \le 1.143, \left(\frac{\frac{\sigma_{z0, FEM}^C}{\sigma_{y0, FEM}^P}\right)_{\text{max}} - \left(\frac{\sigma_{z0, FEM}^C}{\sigma_{y0, FEM}^P}\right)_{\text{min}}}{\left(\frac{\sigma_{z0, FEM}^C}{\sigma_{y0, FEM}^P}\right)_{\text{max}}} \le 0.1, \left(\frac{\sigma_{z0, FEM}^C}{\sigma_{y0, FEM}^P}\right)_{\text{max}} + \left(\frac{\sigma_{z0, FEM}^C}{\sigma_{y0, FEM}^P}\right)_{\text{min}} \le 0.1, \tag{24}
$$

The difference between $(\sigma_{\text{CO,FEM}}^C / \sigma_{\text{y0,FEM}}^P)_{\text{max}}$ and $(\sigma_{\text{CO,FEM}}^C / \sigma_{\text{y0,FEM}}^P)_{\text{min}}$
is less than 10% in Eq. [\(24\),](#page-6-3) and therefore, Dundurs' parameters can

almost control the results and be useful for axisymmetric bonded structures. For two-dimensional problems, Dundurs' parameters are most useful since they control the results completely (no difference).

Since $(K_{\sigma}^C)_{\text{max}}/K_{\sigma}^P$ goes to infinity when $\alpha \to 2\beta$, it is not clear whether $K_{\sigma}^C/K_{\sigma}^P$ or $\sigma_{z0, FEM}^C/\sigma_{y0, FEM}^P$ is suitable for predicting the strength at present.

Useful parameter is unknown near the equal pair
\n
$$
\alpha (\alpha - 2k\beta) < 0
$$
, $k = 1.0 - 0.61(\beta^2 - 0.25)$ and $\alpha (\alpha - 2\beta)$
\n ≥ 0 in Fig. 15\n(25)

5. Experimental evaluation of debonding strength of cylindrical butt joint and plate butt joint

The debonding strength of the cylindrical butt joints was studied experimentally by several researchers [\[30\].](#page-16-7) [Fig. 16](#page-11-0) shows the schematic illustration of the specimens. In this experiment of Naito et al. [\[30\]](#page-16-7), the adherent is aluminum alloy 5052-H34 (Young's modulus E_1 = 69.6 GPa, Poisson's ratio $v_1 = 0.33$) and the adhesive is polyimide (E_2) = 3.77 GPa, ν_2 = 0.342). [Table 13\(](#page-11-1)a), (b) show Dundurs' parameters (α , β) and singular index λ . The length of the adherent *l* is 38.1 mm and the adhesive thickness t is varied from 0.2 mm to 0.6 mm. ([Table 14\)](#page-11-2).

[Fig. 17\(](#page-12-1)a) shows the tensile strength σ_c which increases with increasing the adhesive thickness. In the experiment, the fracture was initiated at the axisymmetric interface end between the adhesive and the adherent. [Fig. 17\(](#page-12-1)b) shows the dimensionless of ISSFs for the cylindrical butt joint $F_{\sigma}^C = K_{\sigma}^C / (\sigma_{\sigma}^{\infty} W^{1-\lambda})$ and $F_{\sigma}^{C*} = K_{\sigma}^C / (\sigma_{\sigma}^{\infty} h^{1-\lambda})$ ob-tained by the method shown in [Section 4](#page-6-4). In [Fig. 17](#page-12-1)(b) F_{σ}^C and F_{σ}^{C*} increase with increasing the adhesive thickness. However, F_{σ}^{C*} is insensitive of h/W and almost constant within 2%. It is seen that F_c^{C*} can be used conveniently to evaluate the adhesive strength. [Fig. 17\(](#page-12-1)c) shows the critical ISSF at $\sigma_z^{\infty} = \sigma_c$, $K_{\sigma c} = K_{\sigma}^C|_{\sigma_z^{\infty} = \sigma_c}$. The $K_{\sigma c}$ values are almost constant independent of the adhesive thickness. It can be confirmed that the ISSF is useful for evaluating the debonding strength.

Similarly, the debonding strength of the plate butt joints [\[15\]](#page-15-3) was considered again by using the present results. [Fig. 18\(](#page-12-2)a), (d) shows the tensile strength σ_c . σ_c increases with increasing the adhesive thickness. In Suzuki's experiment [\[15\]](#page-15-3), it was observed that the fracture is initiated from the interface end between the adhesive and the adherent. [Fig. 18](#page-12-2) (b),(e) shows the ISSFs for the cylindrical butt joint $F_{\sigma}^{P} = K_{\sigma}^{P}/(\sigma_{z}^{\infty}W^{1-\lambda})$ and $F_{\sigma}^{P*} = K_{\sigma}^{P}/(\sigma_{z}^{\infty}h^{1-\lambda})$ obtained by the method shown in [Section 4.](#page-6-4) In [Fig. 18](#page-12-2)(c),(f) F^P_σ and F^{P*}_σ increase with increasing the bondline thickness. However, F_{σ}^{P*} is also insensitive of h/W and almost constant with 2%. It is seen that F_{σ}^{P*} can be used conveniently to evaluate the adhesive strength. [Fig. 18\(](#page-12-2)c),(f) shows the critical ISSF at

Fig. 10. F_{σ}^{C*} is constant when $h/W \le 0.01$.

Fig. 11. Maximum values of $K_{\sigma}^C/K_{\sigma}^P$ and $\sigma_{z_0, FEM}^C/\sigma_{y_0, FEM}^P$ when $\beta = 0.2$.

Fig. 12. Maximum values of $K_{\sigma}^C/K_{\sigma}^P$ and $\sigma_{z_0, FEM}^C/\sigma_{y_0, FEM}^P$ when $\beta = 0.3$.

Fig. 13. Maximum value of $K_{\sigma}^C/K_{\sigma}^P$ and $\sigma_{z_0, FEM}^C/\sigma_{y_0, FEM}^P$ which is useful for h/W ≤0.01 in [Fig. 1](#page-1-0)(c).

Fig. 14. Minimum value of $K_{\sigma}^{C}/K_{\sigma}^{P}$ and $\sigma_{z_0, FEM}^{C}/\sigma_{y_0, FEM}^{P}$ which is useful for h/W ≤0.01 in [Fig. 1](#page-1-0)(c).

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Table 11

Maximum and minimum values of $K_{\sigma}^{C}/K_{\sigma}^{P}$ which is useful for $h/W \leq 0.01$ in [Fig. 1\(](#page-1-0)c).

β	-0.4	-0.3	-0.2	-0.1	$\boldsymbol{0}$	0.1	0.2	0.3	0.4
α									
-1	1.220	1.102	0.951	0.696	0.615				
	0.977	0.945	0.838	0.697	0.636				
-0.9	1.294	1.141	0.991	0.738	0.652				
	0.986	0.949	0.845	0.703	0.646				
-0.8		1.187	1.044	0.819	0.720				
		0.956	0.855	0.722	0.670				
-0.7		1.260	1.121	0.906	0.779				
		0.978	0.875	0.748	0.709				
-0.6			1.258	0.988	0.829	0.650			
			0.889		0.737	0.684			
-0.5			1.364	0.771		0.687			
				1.043	0.887				
			0.902	0.791	0.758	0.704			
-0.4				1.108	0.919	0.708			
				0.811	0.776	0.721			
-0.3				1.153	0.938	0.736			
				0.834	0.796	0.736			
-0.2					0.952	0.779	0.688		
					0.825	0.749	0.658		
-0.1					0.962	0.795	0.698		
					0.861	0.763	0.683		
0			0.987	0.989		0.803	0.710		
			0.961	0.895		0.775	0.698		
0.1			0.987	0.990	0.991				
			0.972	0.914	0.924				
0.2			0.987	0.991	0.992				
			0.981	0.938	0.942				
0.3				0.992	0.993	1.153			
				0.951	0.954	0.971			
0.4				0.992	0.994	1.052			
				0.960	0.965	0.972			
0.5				0.993	0.994	1.022	1.228		
				0.966	0.973	0.977	0.988		
0.6				0.994	0.995	1.010	1.108		
				0.970	0.980	0.982	0.987		
0.7					0.994	1.003	1.056	1.205	
					0.985	0.986	0.989	0.994	
0.8					0.995	1.000	1.029	1.079	
					0.987	0.990	0.992	0.995	
0.9					0.996	1.000	1.008	1.018	1.091
					0.989	0.995	0.996	0.997	0.999
1					0.996	0.996	0.997	0.998	1.000
					0.991	0.996	0.997	0.998	1.000

Upper: maximum value, lower: minimum value.

 $\sigma_z^{\infty} = \sigma_c$, $K_{\sigma c} = K_{\sigma}^P \big|_{\sigma_z^{\infty} = \sigma_c}$. The $K_{\sigma c}$ values are almost constant independent of the bondline thickness. It can be confirmed that the ISSF is useful for evaluating the debonding strength.

6. Conclusions

In this study, the ISSF variations were clarified over the entire thickness range of plate and cylinder butt joints. An effective meshindependent technique was applied to obtaining the ISSFs under arbitrary material combinations. A reference solution was used to eliminate FEM error since the solutions are available for simple bonded plate solved by the body force method. Then, the following conclusions can be summarized.

(1) For the plate butt joints, the ISSF $F_{\sigma}^{P*} = K_{\sigma}^{P}/\sigma h^{1-\lambda}$ normalized by the bondline thickness *h* becomes constant with decreasing the bondline thickness when $h/W \leq 0.01$. In this case, the adhesive joint can be regarded as a bonded semi-infinite plate. If the adhesive layer is thin, F_{σ}^{P*} is more suitable because the variation is smaller than the variation of $F_{\sigma}^{P} = K_{\sigma}^{P}/\sigma W^{1-\lambda}$. To improve the interface strength, thin adhesive layers are desirable. For a certain value*β*, it is found that *F_c*^{*P∗*} decreases with increasingα. Since the solution for *h*/*W*≥1.0 in [Fig. 1](#page-1-0)(a) was shown in the [Appendix A,](#page-12-0) the accurate results can

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Maximum and minimum values of $\sigma_{z_0, FEM}^C / \sigma_{y_0, FEM}^P$ which is useful for $h/W \leq 0.01$ in $Fig. 1(c)$ $Fig. 1(c)$.

Upper: maximum value, lower: minimum value.

Fig. 15. Dundurs' parameters for the several engineering materials and the range of $(α, β)$ satisfies ISSF ratio < 1.5 [\[24\].](#page-16-3)

be obtained by the interpolation also in the range for 0.01≤h/ W≤1.0.

(2) For the cylindrical butt joint, the circumferential strain at the interface end, ε_{r0}^C , is not influenced by the stress singularity because ε_{r0}^C is obtained from the radial displacement u_{r0}^C and the cylinder radius. It was found that the non-singular stresses caused by the $\varepsilon_{r(c)}^C$

Fig. 16. Schematic illustration of cylindrical butt joint.

are contained in the FEM stresses at the interface end. The accurate method was therefore used for calculating the ISSF from the ratio of the stress obtained by subtracting the non-singular stress to the stress of the semi-infinite butt joint adopted as the reference solution. The stress-free boundary condition causes the nonsingular stresses $\tilde{\sigma}_{r0, FEM}^{C} = \tilde{\tau}_{rz0, FEM}^{C} = 0$. The ISSF can be calculated easily without subtraction process of the non-singular stresses when the radial stress $\sigma_{r0, FEM}^{C}$ or the shear stress $\tau_{rz, FEM}^{C}$ is used.

(3) For a certain material combination, the ISSF F_{σ}^{C*} normalized by adhesive thickness h becomes constant with decreasing the bondline thickness when $h/W \leq 0.01$. Thin adhesive layer can be used to improve the interface strength of the cylindrical butt joint. Since the ISSFs of the cylindrical butt joint cannot be totally dominated by the Dundurs' parameter α and β , the maximum and minimum values of the $K_{\sigma}^C/K_{\sigma}^P$ and the $\sigma_{z_0, FEM}^C/\sigma_{y_0, FEM}^P$ were shown in the charts

Table 14

Results of plate butt joint [\[15\].](#page-15-3)

(a) Dundurs' parameters (α, β) and order of singular index λ									
Materials	Adherend		Adhesive		Dundurs' parameter		Singular index		
	E_1 [GPa]	v_1	E_2 [GPa]	v_2	α				
S35C/Epoxy resin A S35C/Epoxy resin B	210 210	0.30 0.30	3.14 2.16	0.37 0.38	0.969 0.978	0.199 0.188	0.685 0.674		

(b) Tensile strength σ_c , F_c^C , F_c^{C*} , $K_{\sigma c}$ for the specimen in [Fig. 16](#page-11-0) with $l = 38.1$ mm, $t = 0.2$ ~0.6 mm, $W = 12.7$ mm

and tables for various (α , β). The value $K_{\sigma}^{C}/K_{\sigma}^{P}$ may be useful for predicting the debonding strength under the bad pairs α (α - 2β) > 0. On the other side, the $\sigma_{z0, FEM}^C / \sigma_{y0, FEM}^P$ may be more important for predicting the debonding strength under the good pairs α (α - 2 β) \leq 0. Since the solution for $h/W \geq 1.0$ in [Fig. 1\(](#page-1-0)c) was shown in the [Appendix B](#page-13-0), the accurate results can be obtained by the interpolation also in the range for 0.01≤h/W≤1.0.

Appendix A. ISSF for the bonded plate

Fig. A1 shows the ISSF F_{σ}^P for the bonded plate calculated by varying Dundurs' parameter (α, β) [\[14\].](#page-15-7) Then, the non-dimensional function of ϵ has been already clarified by Carpenter and Byers [\[31\].](#page-16-8) The bonded plate in [Fig. A1](#page-13-1) can be regarded as a plate butt joint with a very thick adhesive layer for $h/W \ge 1.0$. The F_σ^p values are obtained by the body force method under the bad pair condition of $\alpha(\alpha-2\beta) > 0$ [\[14\]](#page-15-7) and obtained by FEM under the good pair condition of $\alpha(\alpha - 2\beta)$ < 0 [7–[9,20\]](#page-15-4). Since the solution for thin adhesive layer*h*/*W* \leq 0.01 is indicated in [Table 4](#page-6-0) and [Fig. 7](#page-6-1) under aribitrary material combination, the accurate results can be obtained by the interpolation also in the range for $0.01 \le h/W \le 1.0$. ([Table A1\)](#page-13-2).

Fig. A1. ISSF for the bonded plate useful for $h/W \ge 1.0$ in [Fig. 1](#page-1-0)(a).

Appendix B. ISSF for the bonded cylinder in comparison with the bonded plate

In the previous study [\[24\],](#page-16-3) the ISSF of bonded cylinder was compared with the ISSF of bonded plate under arbitrary material combination. The bonded cylinder can be regarded as a cylindrical butt joint with a very thick adhesive layer for $h/W \ge 1.0$ in [Fig. 1](#page-1-0)(c). [Table B1](#page-13-3) and [Fig. B1](#page-14-0) show the maximum values and the minimum values of K_G^C/K_G^P and $\sigma_{\text{SO,FEM}}^C/\sigma_{\text{yO,FEM}}^P$ calculated by varying(α , β). The solid lines indicate K_G^C/K_G^P under

Table B1 Maximum and minimum values of $K_{\sigma}^C / K_{\sigma}^P$ useful for $h/W > 1$ in [Fig. 1\(](#page-1-0)c).

		β										
		-0.45	0.4	0.3	-0.2	-0.1	0.0	0.1	0.2	0.3	0.4	0.45
α	-1.0	0.995	0.981	0.937	0.898	0.866	0.839					
	-0.9		1.146	0.996	0.935	0.892	0.859					
			0.992	0.944	0.899	0.863	0.834					
	-0.8			1.089	0.977	0.919	0.879					
				0.957	0.906	0.865	0.832					
	-0.7			1.321	1.032	0.948	0.899					
				0.976	0.918	0.870	0.833					
	-0.6				1.121	0.981	0.918	0.802				
					0.936	0.88	0.837					
	-0.5				1.346	1.022	0.937	0.827				
					0.962	0.895	0.843	0.804				
	-0.4					1.084	0.955	0.845				
						0.916	0.854	0.808				

(continued on next page)

Table B1 (continued)

(b) Minimum values

Fig. B1. $K_{\sigma}^C / K_{\sigma}^P$ and $\sigma_{z0, FEM}^C / \sigma_{y0, FEM}^P$ in (α, β) map useful for $h/W \ge 1.0$ in [Fig. 1.](#page-1-0)

Table B2

 $\alpha(\alpha-2\beta)$ and $\sigma_{z0,FEM}^C/\sigma_{y0,FEM}^P$ under $\alpha(\alpha-2\beta)$ < 0. The dashed lines indicate $\sigma_{z0,FEM}^C/\sigma_{y0,FEM}^P$ with $\alpha(\alpha-2\beta)$ > 0. The circle marks indicate $\sigma_{z0,FEM}^C/\sigma_{y0,FEM}^P$ for $\alpha(\alpha-2\beta) = 0$. All $K_{\sigma}^C/K_{\sigma}^P$ values are distributed between $(K_{\sigma}^C)_{\max}/K_{\sigma}^P$ and $(K_{\sigma}^C)_{\min}/K_{\sigma}^P$. Because $(K_{\sigma}^C)_{\max}/K_{\sigma}^P$ goes to ∞ when *α* → 2β, the solid lines are very important for predicting the debonding strength except for the bad pair condition near *α* \cong 2β. Because there are only 10% differences between $(K_{\sigma}^C)_{max}/K_{\sigma}^P$ and $(K_{\sigma}^C)_{min}/K_{\sigma}^P$ except for the bad pair condition near $\alpha \approx 2\beta$, $K_{\sigma}^C/K_{\sigma}^P$ and $\sigma_{z0, FEM}^C/\sigma_{y0, FEM}^P$ can be almost controlled by (*α*, β). Since the solution for thin adhesive layer $h/W \le 0.01$ is indicated in [Tables 11,](#page-10-0) [12,](#page-10-2) [Figs. 13 and 14](#page-9-2) under aribitrary material combination, the accurate results can be obtained by the interpolation also in the range for $0.01 \le h/W \le 1.0$. [\(Table B2\)](#page-15-10).

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