Zhang et al. Applied Adhesion Science 2013, 1:9
http://www.appliedadhesionscience.com/content/1/1/9

Applied Adhesion Science
a SpringerOpen Journal

RESEARCH Open Access

# Strength analysis of adhesive joints of riser pipes in deep sea environment loadings

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**Abstract**

Nowadays, adhesive joints are widely used in riser pipes, which are subjected to many kinds of loadings in deep sea, such as external pressure, internal pressure, tension, torsion, bending, and also a combination of these loadings. Adhesive joints of riser pipes are the most dangerous parts in term of strength, as singular stress fields exist at the end of the interface between the adhesive and the adherends, so it is very important to evaluate the strength of adhesive bonded joints for riser pipes in deep sea environment loadings. In this research, the strength of adhesive joints of riser pipes is studied under external pressure, internal pressure, tension, torsion, bending loadings, and it is found that singular stress fields exist around the end of the interface. The riser pipe under external pressure, internal pressure and tension loading is more prone to break than under bending and torque loading.

**Keywords:** Adhesive bonded joint; Singular stress field; Riser pipe; Sea environment

## Background

Due to low manufacturing costs, low stress concentration and ease of maintenance, adhesive joints are most frequently used in numerous industrial sectors such as automobile, shipbuilding, aeronautical, etc., replacing or supplementing traditional joining technologies, such as welding or riveting. Adhesive bonded joints are also widely used in riser pipes because of their light weight, high strength, and high corrosion resistance [1]. With the wide use of adhesive joints, many research works have been done to evaluate their strength including experimental and analytical methods [2-8].

The mismatch of different materials properties may cause stress singularity at the end of the interface between different materials, which leads to failure of the bonding part in structures, so it is very important to analyze the stress singularity field for evaluating the strength of adhesively bonded joints. Many researchers did some valuable work to analyze the stress singularity field at the end of interface between the adhesive and the adherends, such as Koguchi et al. [9], Kilic et al. [10], Van Tooren et al. [11], and Goglio and Rossetto [12], Tilscher [13]. Zhang [14] proposed one easy method to evaluate the effect of adhesive thickness and length on the strength of adhesive joints. Moreover, Zou and Taheri [15] analyzed stress distributions of adhesive bonded sandwich pipe joints subjected to torsional loadings. Da Silva et al. made a detailed and comprehensive comparion between studies about adhesively bonded joints [16,17].

The most commonly used joining methods for pipes are adhesively-bonded socket joints, tubular lap joints, heat-activated coupling joints, and flanged joints. In this research, the adhesively bonded socket joint as shown in Figure 1 was studied. Adhesive joints of riser pipes in deep sea are subjected to many kinds of loadings, such as external pressure, internal pressure, tension, torsion, bending, and also combinations of these loadings. As singular stress fields exist around the end of interface between the adhesive and adherends, the riser pipe is more prone to break near the end of interface of the joint, so it is very important to evaluate the strength of adhesive joints for riser pipes in the sea environment. In this research, the strength of adhesively bonded joints of riser pipes was studied under out pressure, inner pressure, tension, torsion, bending loadings, and the strength of adhesive bonded joints under each loading was compared based on the intensity of singular stresses.

## Methods

### *Theoretical analysis*

At the end of the interface, as shown in Figure 2, it is known that the interface stress $\sigma_{ij}(ij = rr, \theta\theta, r\theta)$ goes to infinity at the edge of the joint and has a singularity of $\sigma_{ij} \propto 1/r^{1-\lambda}$ when $a(a\text{-}2\beta) > 0$. $K_\sigma$ is the parameter used to evaluate the intensity of singular stresses, where $K_\sigma = \lim_{r\to 0}\left(r^{1-\lambda} \times \sigma_\theta\right)$. Besides, when $\theta = \pi/2$, the singularity of the stress $\lambda$ at the joint interface can be expressed by the following equation [18-20].

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

Where

$$a = \frac{G_1\,(k_2+1)-G_2\,(K_1+1)}{G_1\,(k_2+1)-G_2\,(K_1+1)} \qquad \beta = \frac{G_1\,(k_2-1)-G_2\,(K_1-1)}{G_1\,(k_2+1)-G_2\,(K_1+1)}$$

$$k_j = \begin{cases} \dfrac{3-\nu_j}{1+\nu_j}\,(plane\ stress) \\ 3-4\nu_j\,(plane\ strain) \end{cases}, k_j = (j = 1, 2) \qquad (2)$$

Here $r$, $\theta$ are the polar coordinates around the interface edge, $a$, $\beta$ are Dunders' parameters which are expressed by Possion's ratio $\nu$ and shear modulus $G$.

For the 3D model, as shown in Figure 3, there are also several interface ends, and in this paper, the singular stress fields for the interface end in the 3D adhesive joint model was studied under several kinds of deep sea loadings.

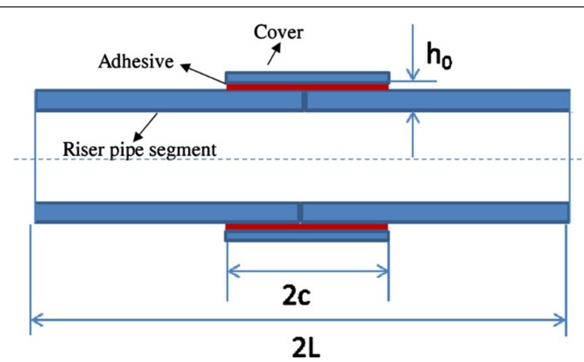


**Figure 1 Adhesively bonded socket joint of riser pipe.**

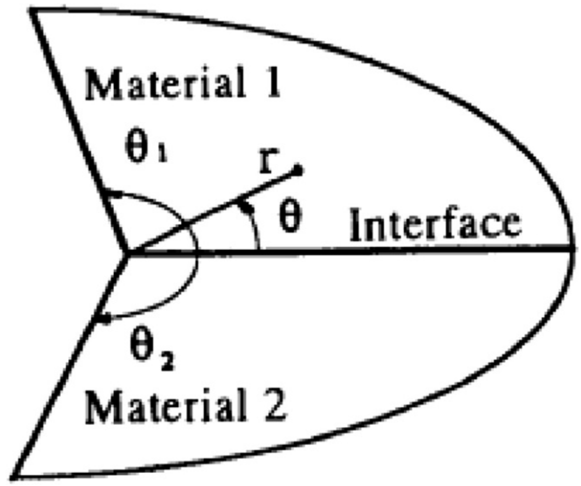


**Figure 2** Model of the end of the interface.

### Numerical model

In this paper, the scaling model with the following dimensions was used: adhesive thickness $h_0 = 6$ mm, cover thickness $h_1$ =10 mm, total length 2 L = 200 mm and coupling length 2c = 25 mm. The outer diameter of the adherend was 140 mm and the inner diameter was 110 mm. The adherend steel elastic properties were E = 70,000 MPa, $\nu = 0.33$. The adhesive material elastic properties were E = 3500 MPa, $\nu = 0.30$, which means that the existence of singular stress fields condition $\alpha(\alpha\text{-}2\beta) > 0$ is satisfied at the edge of interface between adhesive and adherend.

The commercial software ABAQUS was used to perform the analysis. The finite element method model was constructed using 3-D solid elements (Figure 3). Two mesh densities were used to conduct the analysis. A coarse mesh with the mesh density of 32 rows of elements circumferentially 2 rows (radially) · 30 rows (axially) was used to model the pipe region, while the mesh density was doubled to model the joint region. Moreover, the mesh of the joint region was graded along the axial direction of the pipe, finer toward the free edges of the joint.

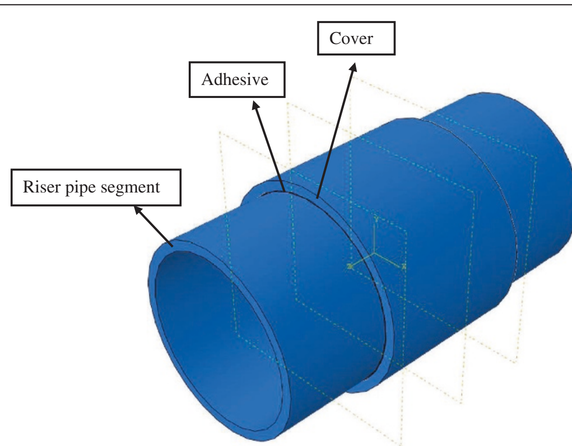


**Figure 3** 3D FEM model of the adhesive bonded joint of a riser pipe.

## Results and discussion

### External pressure loading case

In this paper, a riser pipe under a depth of 1500 meters water was considered, so the out pressure $P_{op}$ = $\rho gH$ =14.7 MPa was applied the adhesive joint. The boundary conditions for this case were : ur = 0, at x = 0; ux = 0, x = 0; and us = 0, x = 0. According to the theroretical analysis, a singular stress field exists around the end of the interface between the adhesive and the adherends. Here, one example of the stress distribution at the interface of the joint subjected to the external pressure is shown in Figure 4. It is found that stresses at the end of interface tend to infinity, which indicates that a singular stress field exists around the end of the interface.

### Internal pressure loading case

Generally, the internal pressure of a riser pipe is higher than the external pressure to prevent buckling of the pipe, and in this research, an internal pressure $P_{ip}$ = 15.0 MPa was chosen. The boundary conditions for this case were: $u_r = 0$, at $x = 0$; $u_x = 0$, $x = 0$; and $u_s = 0$, $x = 0$.

Figure 5 shows the stress distribution at the interface between the adhesive and the adherends of the joint subjected to internal pressure. The stress values at the end of the interface are related to element sizes, but Figure 5 indicates that the stresses around the end of the interface tend to infinity with the element sizes in this paper, which means that a singular stress field exists around the end of the interface. From the comparion with the stress distribution under external pressure, it can be concluded that stresses increase more quickly under internal pressure than external pressure at the end of the interface, which means that the riser pipe under internal pressure is more dangerous than under external pressure if the same increase of stresses happens.

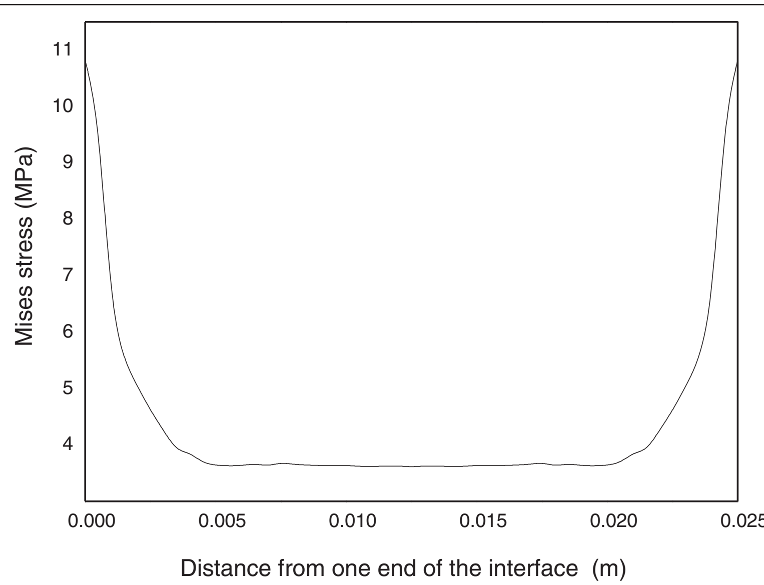


**Figure 4 Stress distribution at the interface of the adhesive joint of a riser piper under external pressure in deep water.**

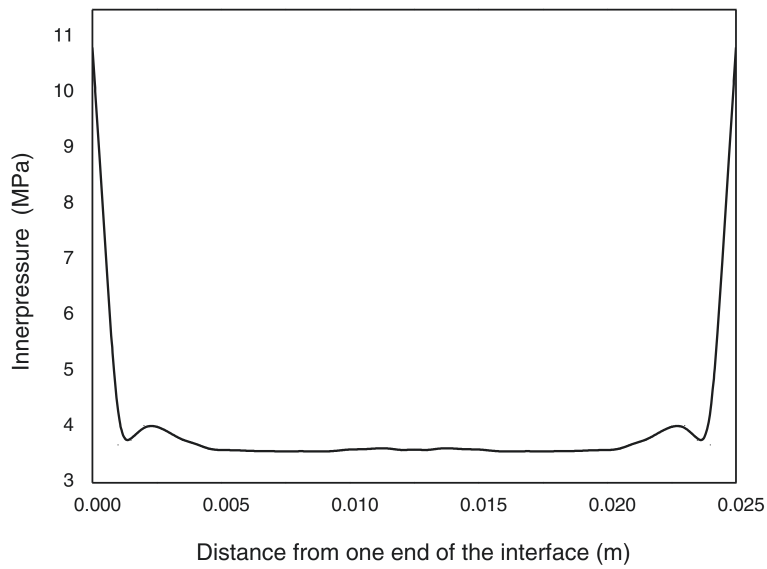


**Figure 5** **Stress distribution at the interface of the adhesive joint of a riser piper under internal pressure in deep water.**

### Tension loading case

The riser pipe in the deep sea also suffers tension loading. The tension loading $T = 20.0$ MPa was considered in this paper, and the boundary conditions are the same as those used for out pressure and inner pressure, which are ur = 0, at x = 0; ux = 0, x = 0; and us = 0, x = 0. Figure 6 shows the stress distribution at the interface between the adhesive and the adherends. Stress values at the end of the interface are related to the elment sizes, but it can be found that stresses at the end of the interface tend to infinity

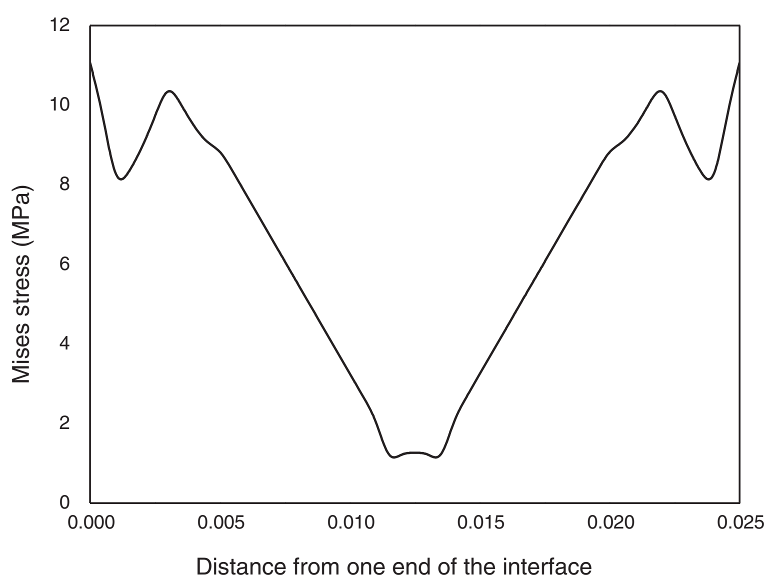


**Figure 6** **Stress distribution at the interface of the adhesive joint of a riser pipe under tension in deep water.**

with the element sizes in this paper, which indicates that a singular stress field exists at the end of the interface.

### Bending loading case

The bending loading was applied by a load $P = 200$ N prependicular to the pipe axis, and the left end of the riser pipe was fixed as: $u_r = 0$, $\theta r = 0$,at $x = 0$;$ux = 0$, $\theta x = 0$, at $x = 0$; and $u_s = 0$, $\theta_s = 0$ at $x = 0$ Figure 7 shows the stress distribution at the interface. It indicates that stress distribution is unsymmetric, and stresses decrease from on end of the interface, which is because moments along the pipe are not symmetric. However, the stress distribution indicates that both ends of the interface have singular stress fields.

### Torque loading case

A torque loading was applied by a load $P = 200\ N$ which was tangent to the circumference of the riser pipe section, so the torque is equal to $200 \times 140 = 28000$ M mm. Figure 8 shows the stress distribution at the interface, and it indicates that the stress also trends to infinity.

### Comparison of stress intensity fields between the cases of external pressure, internal pressure, tension, bending and torque loading

Stress values at the end of the interface are related to the element sizes, and models under different loading cases have the same mesh sizes in this paper. From the comparison, it can be said that different loading cases have different gradients of stress lines. The gradient of stress lines indicates the index of the singular stresses singularity at the end of the interface. If the loading values increase, the stress values at the end of the interface will also increase, without changing the gradient of the stress lines. So the gradients of stress lines are only related to the loading conditions if the mesh sizes do not change.

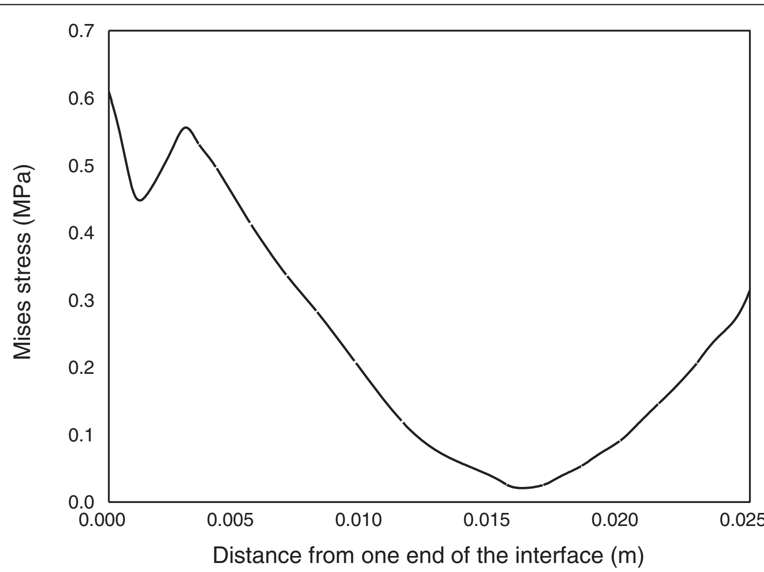


**Figure 7** **Stress distribution at the interface of the adhesive joint of a riser piper under bending in deep water.**

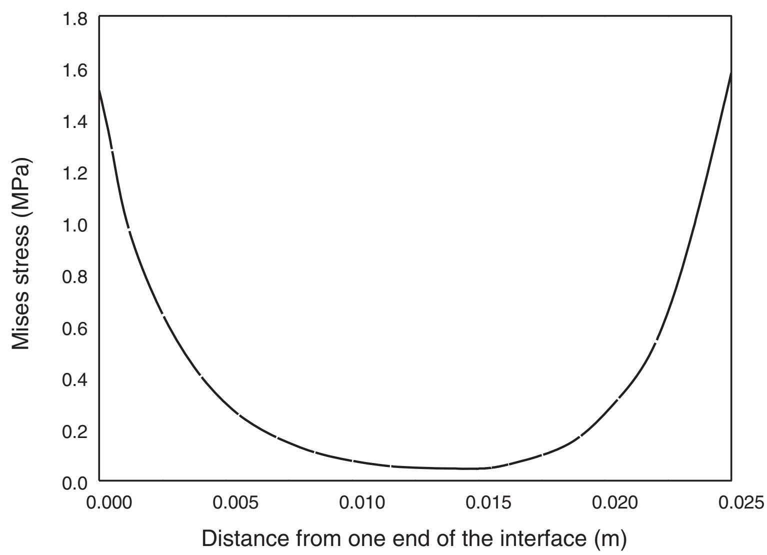


**Figure 8** **Stress distribution at the interface of the adhesive joint of a riser piper under torque.**

Figure 9 shows that the gradients of stress lines for external pressure, internal pressure and tension loading cases are larger than that for bending and torque loading cases, which means that stresses go to infinity more quickly for cases of external pressure, internal pressure and tension compared to cases of bending and torque. Therefore, failure happens more easily around the end of the interface, which means that the riser pipes under external pressure, internal pressure and tension loading are more dangerous than under bending and torque loading in deep sea environment.

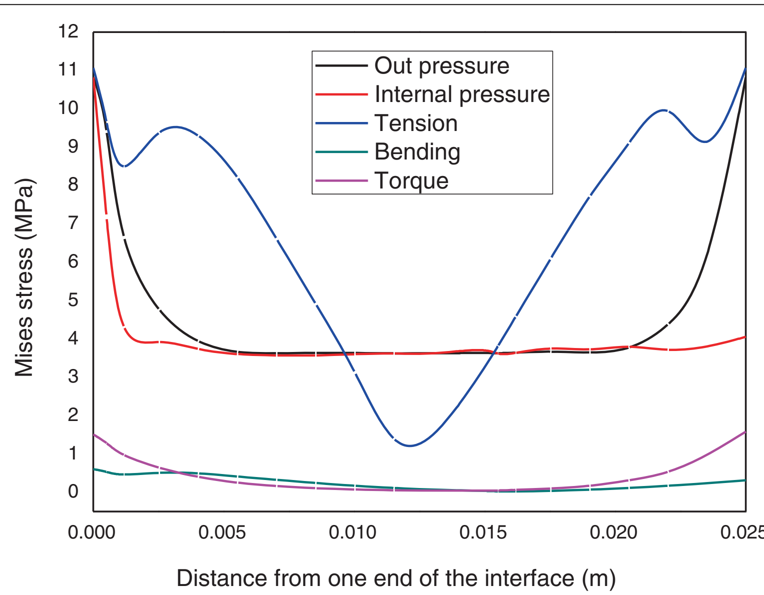


**Figure 9** **Comparison of stress distributions at the interface of the adhesive joint of a riser piper under different loadings.**

## Conclusions

In this paper, a 3D adhesive joint model was constructed using FEM, and the singular stress field around the end of the interface between the adhesive and the adherend was analyzed for cases of external pressure, internal pressure, tension, bending and torque loadings. Singular stress fields exist for all the loading cases, and the stress singularity is larger for cases of external pressure, internal pressure and tension loading compared to the cases of bending and torque loading, which means that the riser pipe under external pressure, internal pressure and tension loading is more prone to break than under bending and torque loading in deep sea environment.

### Competing interests

The authors declare that they have no competing interests.

### Authors' contributions

YZ carried out the FEM analysis and drafted the manuscript. TYQ participated in the FEM analysis. NAN carried out the theoretical analysis. MLD participated in its design and coordination. All authors read and approved the final manuscript.

### Acknowledgements

The authors appreciate the reviewer's constructive comments and detailed feedback for improving the manuscript. This research was supported by Science Foundation of China University of Petroleum, Beijing (No. 2462013YJRC44), and the National Basic Research Program of China (973 Program) Grant No. 2011CB013702.

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Received: 2 September 2013 Accepted: 30 October 2013
Published: 30 December 2013

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doi:10.1186/2196-4351-1-9
**Cite this article as:** Zhang *et al.*: **Strength analysis of adhesive joints of riser pipes in deep sea environment loadings.** *Applied Adhesion Science* 2013 **1**:9.