The Strength of Joints Combining Adhesives with Bolts*

(Cases where Adherends are Pipe Flanges and Circular Flanges)

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This paper deals with the strength of joints combining adhesives with bolts. The stress distribution in the adhesives and a variation of axial bolt force were analyzed using an axisymmetrical theory of elasticity when an external load was applied to the combination joint, in which two pipe flanges or two circular flanges were clamped by nuts and bolts with an initial clamping force after being joined by adhesives. In addition, a method to estimate the strength of the combination joint is proposed. Experiments were performed and analytical results are consistent with the experimental results concerning the variation of axial bolt force and the strength of combination joints. It is seen that the strength of the combination joint is greater than that of adhesive joints, and that it increases with a decrease in the bolt pitch diameter. Furthermore, it is seen that the sealing performance of the combination joint is improved when the adherends are pipe flanges.

Key Words: Elasticity, Stress Analysis, Adhesion, Bolt, Combination Joint, Pipe Flange, Circular Flange, Strength

1. Introduction

Recently, adhesive joints have been used in mechanical structures as the performance of bonding materials has advanced. But adhesive joints have not been used in the principal parts of structures because of large deviations in the strength of the adhesive. On the other hand, mechanical joints such as bolted, welded and riveted types, which have given satisfactory results, have merits and demerits in their characteristics. Thus, combination joints⁽¹⁾, which develop the merits of mechanical joints, have been used.

In the same way, a joint which combines an adhesive with bolts (or rivets) has appeared^{(1),(2)}. The first object of the joint is to increase the strength because of the unreliability of the strength of adhesions, thus, joints must be designed from a fail-safe standpoint. The second object is to make two joining elements distribute an external load more efficiently by combining the merits of adhesive joints and bolted joints. Although combination joints have been used in practical structures⁽³⁾, there has been little

research^{(4)~(7)} carried out on the fundamental characteristics of the combination joints.

In this paper, the characteristic strength of the combination joint is examined in order to extend the use of combination joints, which the authors^{(6),(7)} have investigated. These are joints in which two pipe flanges or two circular flanges are clamped by nuts and bolts with an initial clamping force after being joined with an adhesive. In the analysis, replacing adherends (pipe flanges or circular flanges) and an adhesive with finite hollow cylinders or finite solid cylinders, respectively, the stress distributions in the adhesive and the load factor (the ratio of an increment of axial bolt force to an external load) are analyzed as a three-body contact problem using an axisymmetrical theory of elasticity⁽⁸⁾. Then, a method to estimate the fracture strength of the combination joint is proposed using the stress distribution in the adhesive. In the experiments, the load factor was measured with strain gauges. The fracture strength of the combination joints was also measured. The analytical results are compared with the experimental results.

2. Method of Analysis

2.1 The case where adherends are pipe flanges

2.1.1 Stress distribution in joint Figure 1 shows a combination joint of an adhesive with bolts, in which two pipe flanges are clamped by N nuts and

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bolts with an initial clamping force F_f after being joined with an adhesive. The axial tensile load W is produced, due to an application of internal pressure P , and an increment of axial bolt force F_t is produced; then the total compression force at the adhesive layer becomes $F_f - F_c$ because the force F_c is eliminated from the adhesive layer. It is necessary to know the stress distribution in the adhesive layer in order to evaluate the joint strength. Next, the adherends and the adhesive layer are replaced with finite hollow cylinders [I] and [II], respectively, as shown in Fig. 2. The properties of the finite hollow cylinder [I] are as follows. Its inner diameter is $2a$, the outer diameter is $2b$, the height is $2h_1$, Young's modulus is E_1 and Poisson's ratio is ν_1 . The diameter of the bearing surface is denoted by $2c$ and the bolt pitch diameter by $2d$. For the finite solid cylinder [II], they are denoted by $2a$, $2h_2$, E_2 , and ν_2 , respectively. Figure 2(a) shows the case where the force $F_b(=(F_f + F_t) \times N)$ acts on the bearing surfaces ($d - c \leq r \leq d + c$) as a uniform stress p_1 and Fig. 2(b) shows the case where an external load W acts on the surfaces ($a \leq r \leq e$, $z_1 = +h_1$) as a uniform stress p_2 .

In the analysis of the cases of Fig. 2(a) and (b), the boundary conditions are expressed as follows, developing the stress distributions into series of Bessel functions⁽⁸⁾ $C_0(\gamma_s r)$, where $C_0(\gamma_s r) = J_0(\gamma_s r) - J_1(\gamma_s b)Y_0(\gamma_s r)/Y_1(\gamma_s b)$, $J_n(\gamma_s r)$ is the first kind of Bessel function of the order n and $Y_n(\gamma_s r)$ is the second kind of Bessel function of the order n . The finite hollow cylinders [I] (adherends)

Fig. 1 A joint combining adhesives with bolts (the case of pipe flanges)

$$\left. \begin{aligned} r=a: \sigma_r^I &= -P, \\ \tau_{rz}^I &= 0 \\ r=b: \sigma_r^I &= \tau_{rz}^I = 0 \\ z=h_1: \sigma_z^I &= a_0 + \sum_{s=1}^{\infty} a_s C_0(\gamma_s r), \\ \tau_{rz}^I &= 0 \end{aligned} \right\} \quad (1)$$

The finite hollow cylinders [II] (adhesives)

$$\left. \begin{aligned} r=a: \sigma_r^{II} &= -P, \\ \tau_{rz}^{II} &= 0 \\ r=b: \sigma_r^{II} &= \tau_{rz}^{II} = 0 \end{aligned} \right\} \quad (2)$$

At the interface between the finite hollow cylinders [I] and [II]

$$\left. \begin{aligned} (\sigma_z^I)_{z_1=-h_1} &= (\sigma_z^{II})_{z_2=h_2} \\ (\tau_{rz}^I)_{z_1=-h_1} &= (\tau_{rz}^{II})_{z_2=h_2} \\ (u_z^I)_{z_1=-h_1} &= (u_z^{II})_{z_2=h_2} \\ \left(\frac{\partial w^I}{\partial r} \right)_{z_1=h_1} &= \left(\frac{\partial w^{II}}{\partial r} \right)_{z_2=h_2} \end{aligned} \right\}, \quad (3)$$

where u is the displacement in the r directions, w is the displacement in the z directions and superscripts I and II indicate the finite hollow cylinders [I] and [II], respectively. The finite hollow cylinders under the boundary conditions (1)-(3) are analyzed based on the method demonstrated in Ref. (8).

2.1.2 The load factor When an external load W is applied to a combination joint which is joined

(a) The case of initial clamping

(b) The case where internal pressure acts

Fig. 2 Model for analysis (the case of pipe flanges)

with an initial clamping force F_f , an increment F_t in axial bolt force is produced. From the viewpoint of bolt strength, it is necessary to know the load factor $\phi^{(9),(10)}$ (the ratio of F_t to W). As shown in Fig. 2(a), when the force $F_b (= F_f + F_t)$ acts, the compressive spring constant K'_{cg} is defined by the equation $K'_{cg} = F_b / 2\varepsilon_1$ using the mean displacement ε_1 at the bearing surfaces. The tensile spring constant K_{pt} is defined by the equation $K_{pt} = W / 2\varepsilon_2$ using the mean displacement ε_2 at the bearing surfaces produced by the external load W . Using the spring constants mentioned above, the load factor $\phi^{(10)}$ is obtained by Eq. (4).

$$\phi = \frac{F_t}{W/N} = \frac{K_t}{K_t + K_c} \left(\frac{K'_{cg}}{K_{pt}} \right) \quad (4)$$

where $K_t^{(10)}$ is the spring constant for a bolt-nut system and $K_c^{(10)}$ is the compressive spring constant for a hollow cylinder around a bolt hole. The notation N is the number of nuts and bolts. The values of K'_{cg} and K_{pt} are obtained by analyzing the cases of Fig. 2(a) and (b). The compressive force F_c , which is eliminated from the adhesive layer, is obtained by the equation $F_c = (1 - \phi)W$.

2.1.3 Analysis of bending moment produced in bolts With an application of a tensile load W , a bending moment M_b is produced in bolts in addition to an increment F_t of axial bolt force because the bearing surface is inclined. In the same method as in Ref. (11), the bending moment M_b is obtained and the bolt strength is evaluated by the maximum stress produced in the bolts. The maximum stress is given by the following equation,

$$\sigma_{\max} = \frac{F_f}{A} + \left(\frac{\phi}{A} + \frac{M_b}{Z} \right) \left(\frac{W}{N} \right) \quad (5)$$

where A is the sectional area and Z is the modulus of the section.

2.1.4 Estimation of the joint strength It is assumed that the stress at the adhesive which reaches the fracture stress is the joint strength. The strength is estimated by using the stress which is obtained by superposing the stress of Fig. 2(a) on the stress of Fig. 2(b), that is, when the sum of the stress σ_b due to the load W_b and the compressive stress σ_t due to the initial clamping force F_f and the increment of axial bolt force F_t becomes equal to the fracture stress σ_a , which is obtained by using the fracture load W_a . Then, the strength of combination joint W_b is given by Eq. (6),

$$\frac{W_b}{A_a} = \frac{W_a}{A_a} + N \cdot \frac{F_f + F_t}{A_a} \quad (6)$$

where N is bolt number and A_a is the interface area and F_t is given by Eq. (7).

$$F_t = \frac{W_b \phi}{N} \quad (7)$$

Thus, the load W_b when the joint ruptures is given by the following equation.

$$W_b = \frac{W_a + NF_f}{1 - \phi} \quad (8)$$

2.2 The case where adherends are circular flanges

2.2.1 Stress distribution in joint Figure 3 shows a combination joint of an adhesive with bolts, in which two circular flanges are clamped by nuts and bolts with an initial clamping force F_f after being joined with the adhesive. The adherends and the adhesive layer are replaced with finite solid cylinders, respectively, and the stress distribution is analyzed based on the above method 2.1.1 and the author's investigation⁽¹²⁾.

3. Experimental Method

Experiments were carried out in order to measure the load factor and the strength of joints combining an adhesive with bolts. Figure 4 shows pipe flanges and circular flanges used in the experiments. After these flanges were joined by epoxide adhesive [Scotch-weld 1838, Sumitomo 3M Co., Ltd.], they were clamped by bolts of M16. A combination joint which consisted of the pipe flanges shown in Fig. 4(a) was applied to an internal pressure by an oil-hydraulic pump. The magnitude of the internal pressure was measured with a pressure transducer, and the axial bolt force was measured with strain gauges glued to the shank of bolts. The outputs were recorded by an X-Y recorder through a dynamic amplifier. In addition, tensile tests were also carried out using a material testing machine in the case where a tension $W (= P\pi a^2)$ due to an internal pressure acts on the joints. Figure 4(b)

Fig. 3 A joint combining adhesives with bolts (the case of circular flanges)

shows the specimens of pipe flanges and circular flanges manufactured from S45C(JIS). Two kinds of flanges with the bolt pitch diameter $2d$ were manufactured in order to examine the effect of the bolt pitch diameter. Figure 4(c) shows the bolts used in the experiments. Two strain gauges were glued to the opposite side of the shank.

4. Analytical Results and Comparisons with Experimental Results

4.1 The case where adherends are the pipe flanges

4.1.1 Stress distribution at the interface of adhesive layer Figure 5 shows the analytical results of the stress distributions at the interface of the adhesive layer⁽⁸⁾. In the numerical calculations, the

(a) Pipe flange (for an internal pressure test)

(b) Pipe flange and circular flange (for a tensile test)

(c) Bolt (for the tensile test)

Fig. 4 Dimensions of flanges and bolts used in experiments

dimensions of the hollow cylinder [I], are determined as the same as those of the specimen used in the experiments, and the thickness $2h_2$ of the adhesive layer is taken as 0.06 mm from the measurement results. Young's modulus and Poisson's ratio of the adhesive layer are 3.6 GPa and 0.38, respectively. The bolt pitch diameter $2d$ is set at 55 and 75 mm and the web diameter at $2e=35$ mm. The number N_s of terms in the series is determined as 80. The ordinate is the ratio of the maximum principal stress σ_1 at the interface of the adhesive layer ($z_2=+h_2$) to the mean stress σ_{zm} and the abscissa is the ratio of the distance r from the center to the inner radius a of pipe flanges.

Figure 5(a) shows the analytical results in the case of initial clamping state. The value σ_1/σ_{zm} near the inner circumference ($r/a=1.1$) increases with a decrease in the value $2d$. In the case of $d/a=5.0$, the nondimensional maximum principal stress near the inner circumference becomes negative, so that no effect of the initial clamping force is expected. Figure

(a) In the case of Fig. 2(a)

(b) In the case of Fig. 2(b)

Fig. 5 Analytical results of principal stress distribution at the interface ($z_2=h_2$) (the case of pipe flanges)

5(b) shows the analytical results in the case of Fig. 2(b), where the solid line shows the case where a tensile load is applied to the joint shown in Fig. 4(b) and the dotted line shows the case where an internal pressure is applied. The value σ_1/σ_{zm} increases near the inner circumference. The stress at the interface between the adherends and the adhesive is estimated by superposing the stress shown in Fig. 5(a) on the stress in Fig. 5(b). From the results, it is predicted that the strength of the combination joints increases with a decrease in the bolt pitch diameter $2d$.

4.1.2 Variation in axial bolt force Figure 6(a) shows a variation in the axial bolt force indicated by the solid line when an internal pressure is applied to a combination joint shown in Fig. 4(a). The ordinate is an axial bolt force $F_f + F_t$ and the abscissa, an internal pressure P . The initial clamping force F_f is 20 kN, and the bolt pitch diameter is 105 mm. The dotted line shows an experimental result in the case where a flat-face gasket (thickness 10 mm) manufactured from aluminum is used instead of an adhesive. In the case where the gasket is used, separation of the contact surface occurs with increase in the internal pressure, so that F_t plots a curved line. Then, fluid leakage occurs at the internal pressure of about

(a) The case of internal pressure test

(b) The case of tensile test

Fig. 6 Comparisons of numerical results with experimental ones with respect to variation of axial bolt force

21 MPa. On the other hand, in the case of the combination joint, F_t increases slightly and linearly with an increase of the internal pressure. However, the fracture does not occur at the adhesive layer and leakage does not occur until an internal pressure is applied to the limit value of 49 MPa. Thus, the combination joint is useful under a static load.

Figure 6(b) shows a variation in the axial bolt force under a tensile load. The ordinate is an axial bolt force $F_f + F_t$ and the abscissa is an external load per bolt W/N . The initial clamping force F_f is 10 kN and 20 kN. The increment F_t in the axial bolt force increases linearly until the adhesive layer ruptures. After the rupture of the adhesive layer, the axial bolt force $F_f + F_t$ becomes the external load W/N . The inclination of the linear line is independent of the initial clamping force F_f and is held constant. It is preferable for the bolt design to restrain the increment in the bolt axial force caused by the separation of the contact surfaces. The load factor indicates the inclination of the straight line. In the case of $2d=55$ mm, the load factor ϕ is 0.057 5; in the case of $2d=75$ mm, $\phi=0.011$ 4. The experimental results are 0.031 5 and 0.008 6, respectively. In Fig. 6(b), the dotted lines are the numerical results in the case where $2d$ is 55 mm and F_f is 10 kN and 20 kN. Both results are in fairly good agreement.

4.1.3 Maximum stress produced in bolt Figure 7 shows experimental and analytical results of a maximum stress taking account of a bending moment produced in bolts of a combination joints shown in Fig. 4(a) under an application of internal pressure P . Both results are in fairly good agreement. The long and short dashed lines in Fig. 7 represent the stress F_t/A obtained from the load factor. Because no separation of the contact surface occurs, the value of the bending stress is smaller than that of the joint with a gasket⁽¹³⁾.

4.1.4 Strength of combination joint Table 1 shows comparisons between the strength of combination joints W_b and the strength of adhesive butt joints W_a . Each experiment was performed five times using

Fig. 7 Comparisons in the case where bending moment is taken into consideration

the specimens shown in Fig. 4(b), and the mean values are shown in Table 1. The strength W_a of the adhesive butt joint with no bolt holes, of which the dimensions are the same as in Fig. 4(b), was 109.1

Table 1 Comparisons with respect to joint strength (the case of pipe flanges)

		(kN)					
		Combination Joint					
2d (mm)		55			75		
F _t		5	10	20	5	10	20
EXP	W _a	103.6			108.5		
	W _b	129.5	140.8	159.1	122.8	126.3	126.7
NUM	W _b	113.9	118.6	128.0	108.2	108.0	107.5
	W _a	109.6			109.6		
	W _b	125.6	132.6	146.8	114.4	118.2	125.8

(a) The case where compression acts on the bearing surface

(b) The case where tension acts

Fig. 8 Analytical results of principal stress distribution at the interface ($z_2 = h_2$) (the case of circular flanges)

kN. The strength of the combination joint W_b is larger than that of the adhesive joints W_a , and the value W_b increases with a decrease of the bolt pitch diameter $2d$. The strength of adhesive butt joints was 103.6 kN in the case of $2d = 55$ mm and 108.5 kN in the case of $2d = 75$ mm. From Fig. 5 and Eq. (8), the values of the rupture load in combination joint $\bar{W}_b$ are shown in Table 1. The analytical values are a little smaller than the experimental values.

It was observed from the ruptured surfaces that the adhesive layer was ruptured simultaneously. In addition, taking into consideration that an increment F_t of axial bolt force becomes W/N after an adhesive layer of a combination joint, we tried to estimate the strength of the combination joints as follows. First, using a hollow cylinder specimen⁽⁶⁾, the fracture stress of the adhesive layer is obtained as $\sigma_a = 60.3$ MPa. Next, it is assumed that the area in the adhesive layer, in which the stress shown in Fig. 5(b) is above the mean stress, is ruptured simultaneously. The radius r/a of the area which is ruptured simultaneously is obtained as 3.4 ($r = 25.2$ mm) from Fig. 5(b). Then, the strength of the adhesive joint W_a is obtained as $W_a = \pi r^2 \sigma_a = 109.6$ kN. Furthermore, using the value W_a , the strength of the combination joint is estimated. Taking account of the stress distribution shown in Fig. 5 and the load factor shown in Table 1, the strength of the combination joint $\bar{W}_b$ is estimated by Eq. (8) and the value $\bar{W}_b$ is shown in Table 1. These results are fairly consistent with the experimental results.

In the case of the adhesive joint, the joint is broken after the adhesive layer is ruptured. But in the case of the combination joint, the shape of the joint is maintained after the adhesive layer is ruptured.

4.2 The case where adherends are circular flanges

4.2.1 Stress distribution at interface of adhesive layer Figure 8 shows the analytical results of the stress distributions at the interface of the adhesive layer⁽¹²⁾. In the numerical calculations, the thickness $2h_2$ of the adhesive layer was taken as 0.06 mm from the measuring results. Young's modulus and Poisson's ratio of the adhesive layer were 3.6 GPa and 0.38, respectively. The bolt pitch diameter $2d$ was set at 55 and 70 mm, the diameter $2c$ of the bearing surface at 16.3 mm, and the web diameter $2e$ at 35 mm. The number of terms in series was determined as 80. The ordinate is the ratio of the maximum principal stress σ_1 at the interface of the adhesive layer ($z_2 = +h_2$) to the mean stress σ_{zm} , and the abscissa is the ratio of the distance r from the center to the outer radius a of circular flanges.

Figure 8(a) is the case of initial clamping. It is

Table 2 Comparisons with respect to joint strength
(the case of circular flanges)

		(kN)			
		Combination Joint			
2d (mm)		55		70	
F _f		10	20	10	20
EXP	W _a	115.8		115.9	
	W _b	139.0	152.0	127.0	130.8
NUM	W _b	125.6	131.8	116.4	116.7
	W _a	108.7		108.7	
	W _b	129.5	142.8	118.4	126.4

seen that the value σ_1/σ_{zm} at the center increases with a decrease in the value $2d$. In the case of $d/a=0.78$, the stress at the center becomes extremely small. Figure 8(b) is the case where an external load is applied. It is seen that the value σ_1/σ_{zm} at the center increases. The stress at the interface is estimated by superposing the stress shown in Fig. 8(a) on the stress in Fig. 8(b). From the results, it is predicted that the strength of the combination joints increases with a decrease of the bolt pitch diameter $2d$.

4.2.2 Strength of combination joint Table 2 shows comparisons concerning the joint strength. The specimens shown in Fig. 4(b) were used, and the experimental and analytical values were obtained in the same method as in 4.1.4. The analytical results are slightly smaller than the experimental results. However, they are in fairly good agreement. In this case, it is seen that the strength of the combination joint increases. The joint's shape was maintained after the adhesive layer was ruptured.

5. Conclusions

This paper describes the strength of a combination joint consisting of an adhesive with bolts. In the case where adherends are pipe flanges and circular flanges, stress distributions at the interface of the adhesive layer, the load factor and the joint strength were examined. The following results are obtained.

(1) The stress distribution in combination joints was analyzed using an axisymmetrical theory of elasticity.

(2) According to (1), the load factor, the maximum stress and the joint strength were analyzed. In numerical calculations, the effects of the bolt pitch diameter on the load factor and the joint strength were clarified.

(3) Experiments were performed with respect to the load factor, the maximum stress in bolts and the joint strength. It was shown that the analytical results were fairly consistent with the experimental ones.

(4) It was seen that the strength of combination joints was greater than that of adhesive joints, and that the strength of the combination joints increased with a decrease in the bolt pitch diameter. In the combination joint, the joint's shape was maintained after the adhesive layer was ruptured. In addition, in the case of pipe flanges, the sealing performance was improved.

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