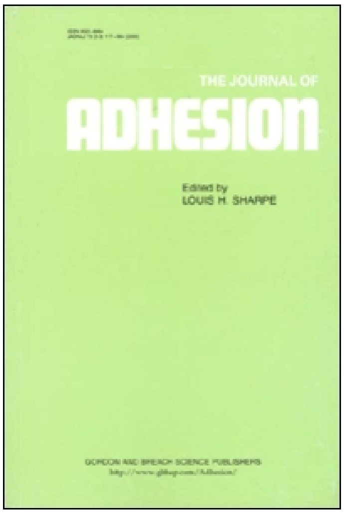

The Journal of Adhesion
Publication details, including instructions for authors and subscription information:
http://www.informaworld.com/smpp/title~content=t713453635

The Strength of Joints Combining an Adhesive with a Bolt

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To cite this Article Sawa, Toshiyuki and Kobayashi, Takashi(1988) 'The Strength of Joints Combining an Adhesive with a Bolt', The Journal of Adhesion, 25: 4, 269 — 280
To link to this Article: DOI: 10.1080/00218468808071267
URL: http://dx.doi.org/10.1080/00218468808071267

J. Adhesion, 1988, Vol. 25, pp. 269–280
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Printed in the United Kingdom

The Strength of Joints Combining an Adhesive with a Bolt

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(Received April 10, 1987; in final form October 27, 1987)

In this paper, discussion is made on the strength of joints combining adhesives with bolts. The stress distributions in adhesives and the variation in bolt axial force are analyzed when an external load is applied to the combination joint. The joint consists of two hollow cylinders clamped by a bolt and a nut with an initial clamping force after they are joined by an adhesive. The analytical result is almost consistent with the experimental result, therefore the strength of the combination joints can be estimated. The strength of the combination joint is greater than that of the adhesive joint. In addition the variation in bolt axial force in the combination joint is less that that in the bolted joint. The characteristics and the availability of combination joints are made clear.

KEY WORDS Elasticity; strength; adhesion; bolt; combination joint; hollow cylinder.

1. INTRODUCTION

Recently, adhesive joints have been used in mechanical structures as the performance of bonding materials has advanced. But the 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, 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 trial which combines the adhesive joint with bolts (or rivets) has appeared.[2] The first object of these joints is to increase the strength, because of the unreliability of the strength of adhesions, and also to design joints safely (from a fail-safe viewpoint). 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[4,5] carried out on the fundamental characteristics of the combination joints.

In this paper, discussion is made on the strength of the combination joint. This is a joint in which two hollow cylinders are clamped by a bolt and a nut with an initial clamping force after being joined with an adhesive. Theoretically, stress distributions in the adhesive layer and the variations in bolt axial force are analyzed by a three-dimensional theory of elasticity.[6] For verification, experiments are made, and agreement between the theoretical and the experimental results are discussed.

2. METHOD OF ANALYSIS

2.1 Stress distribution in joint

Figure 1 shows a combination joint of an adhesive with a bolt, in which two hollow cylinders are clamped by a bolt and a nut with an initial clamping force F_f after being joined with an adhesive. With an application of the tensile load W, the bolt axial force becomes $F_f + F_t$ and 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 stresses in the adhesive layer in order to evaluate the strength of the joint. So the adherends and the adhesive layer are replaced with finite hollow cylinders [I] and [II], respectively, as shown in Figure 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. For the finite hollow cylinder [II], they are denoted by $2a$, $2b$, $2h_2$, E_2 and ν_2 respectively.

Figure 2(a) shows a case where the force F_b $(= F_f + F_t)$ acts on the

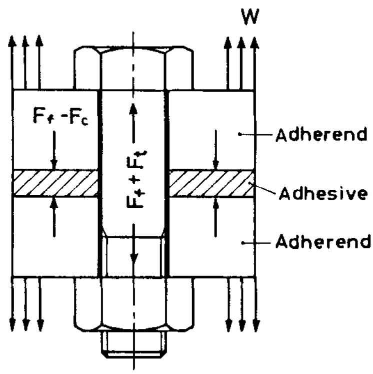

FIGURE 1 Combination joint of adhesive with bolt (the case where adherends are hollow cylinders).

bearing surface as a uniform stress P_1 and (b) shows an external load W applied to the surfaces ($c \leqq r \leqq b$, $z_1 = \pm h_1$) as a uniform stress P_2. F_f is an initial clamping force and F_t is an increment in bolt axial force.

In order to analyze the cases of Figure 2(a) and (b), boundary conditions are expressed as follows using the Bessel series.

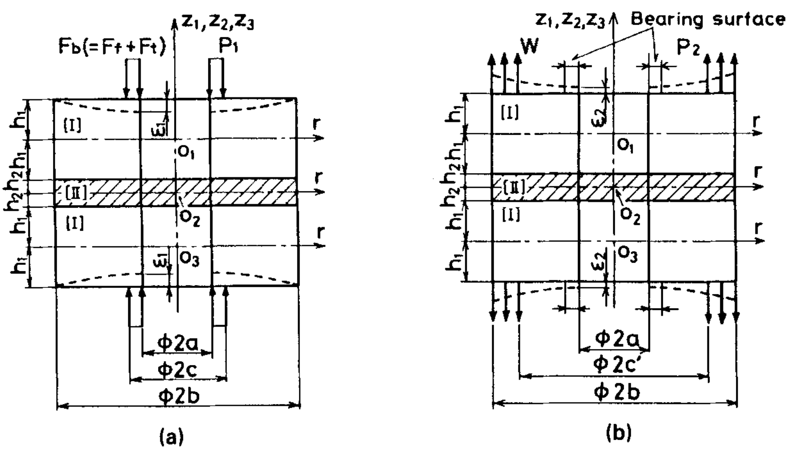

FIGURE 2 Model for analysis (a) the case where clamping force acts on the bearing surfaces (b) the case where an external load is applied.

The finite hollow cylinders [I] (adherends)

$$\begin{aligned} r = a&: \ \sigma_r^{\mathrm{I}} = \tau_{rz}^{\mathrm{I}} = 0 \\ r = b&: \ \sigma_r^{\mathrm{I}} = \tau_{rz}^{\mathrm{I}} = 0 \\ z_1 = h_1&: \ \sigma_z^{\mathrm{I}} = a_0 + \sum_{s=1}^{\infty} a_s C_0(\gamma_s r) \\ &\quad \tau_{rz}^{\mathrm{I}} = 0 \end{aligned} \tag{1}$$

The finite hollow cylinder [II] (adhesive)

$$\begin{aligned} r = a&: \ \sigma_r^{\mathrm{II}} = \tau_{rz}^{\mathrm{II}} = 0 \\ r = b&: \ \sigma_r^{\mathrm{II}} = \tau_{rz}^{\mathrm{II}} = 0 \end{aligned} \tag{2}$$

The boundary of the finite hollow cylinders [I] and [II]

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

where u is the displacement in the r direction, w is the displacement in the z direction and superscripts I and II indicate the finite hollow cylinders [I] and [II]. The finite hollow cylinders under the boundary conditions (1)–(3) are analyzed based on the method demonstrated in the Reference 6.

2.2 Increment in bolt axial force

The force ratio, which is an important factor to estimate the strength of bolts, will now be examined. As shown in Figure 1, when external load W is applied to a combination joint with an initial clamping force F_f, an increment F_t in bolt axial force is produced. The ratio of F_t to W, (force ratio[7,8]), is analyzed. As shown in Figure 2(a), when the force $F_b\,(=F_f + F_t)$ acts, the compressive spring constant K_c is defined by the equation $K_c = F_b/2\varepsilon_1$ 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 external load W. The problem to obtain the force ratio is statically indeterminate. Thus, using the spring constants mentioned above, the force ratio Φ is expressed by Eq. (4)[7] using the condition that the elongation in a bolt is equal to the elongation between the bearing surfaces of the adherends.

$$\Phi = \frac{F_t}{W} = \frac{K_t}{K_t + K_c}\left(\frac{K_c}{K_{pt}}\right) \tag{4}$$

where K_t is the spring constant for a bolt-nut system.[9] The values of K_c and K_{pt} are obtained by analyzing the cases of Figure 2(a) and (b). The compressive force F_c, which is eliminated from the adhesive layer, is obtained by the equation $F_c = (1 - \Phi)W$.

3. EXPERIMENTAL METHOD

Experiments were done to measure force ratio and strength of combination joints of an adhesive with a bolt. Figure 3(a) shows the hollow cylinder used in the experiments. The inner diameter $2a$ was held constant at 15 mm and the ratio b/a of the outer radius to the inner one was varied with three values. Figure 3(b) shows the bolt used in the experiments. The bolt was manufactured by turning and its size was M12 (JIS). Two strain gages were glued to the opposite side of the shank. The cylinders were bonded by an epoxide

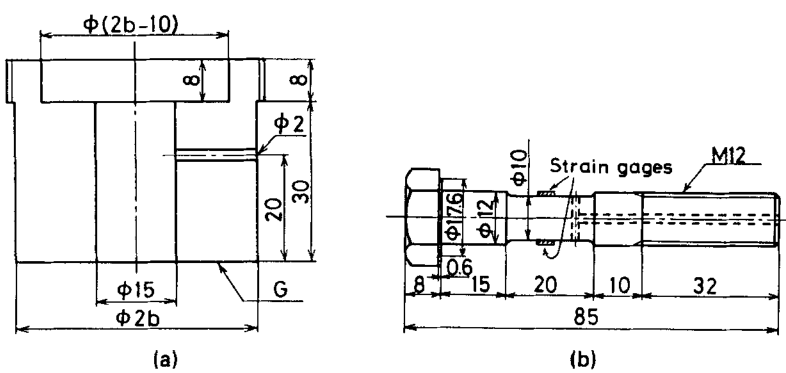

FIGURE 3 Specimens and bolt used in experiments (a) hollow cylinder specimen (b) bolt.

adhesive [SUMITOMO 3M Co. Ltd. in Japan, Scotch-Weld 1838]. Tension tests were made using a material testing machine [Instron type, SHIMADZU, DCS-25T]. The load was measured by a load cell and the strains in the adherend and the variation in bolt axial force were measured by strain gages. Both output signals were recorded by an X–Y recorder using dynamic amplifiers.

4. COMPARISONS OF ANALYTICAL RESULTS WITH EXPERIMENTAL RESULTS

4.1 Stress distribution at boundary of adhesive layer

Figure 4 shows analytical results of the stress distributions at the boundary of the adhesive layer. This corresponds to the case indicated in Figure 2. The dimensions of the finite hollow cylinders [I] conform to those of the specimens used in the experiment. Thus, a comparison can be made between the analytical and the experimental results. The inner diameter $2a$, the height $2h_1$, Young's modulus E_1 and Poisson's ratio ν_1 are taken as 15 mm, 30 mm, 206 GPa and 0.3, respectively. The outer diameter $2b$ is varied as 30, 45 and 60 mm. The thickness $2h_2$ of the adhesive layer is taken

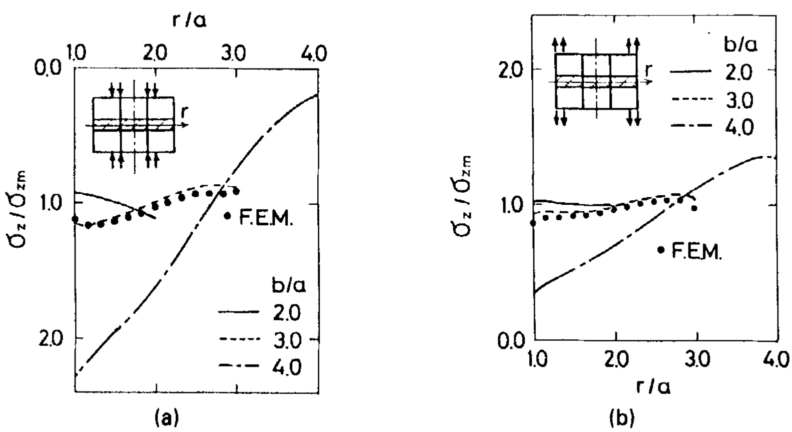

FIGURE 4 Stress distribution at the boundary ($z_2 = \pm h_2$) of adhesive layer ($h_1/a = 2.0$), (a) analytical result for the case of Figure 2(a), (b) analytical result for the case of Figure 2(b).

as 0.06 mm from the measuring results. Young's modulus and Poisson's ratio of the adhesive layer are 3.6 GPa and 0.38 respectively. The diameter of the bearing surface is taken as 17.6 mm and the width $(b-c')$ of the position of load application is 5 mm. In the analyses, 50 terms are taken in the Bessel series. The ordinate is the ratio of the normal stress σ_z at the boundary of the adhesive layer $(z_2 = \pm h_2)$ to the mean normal stress σ_{zm} and the abscissa is the ratio of the distance r from the center to the inner radius a of the hollow cylinder. As shown in Figure 4, at smaller ratios of b/a (2.0, 3.0) the ratio σ_z/σ_{zm} tends not to change much. But with increase in the ratio b/a to 4.0, there are marked decreases in the compressive stress and marked increases in the tensile stress.

As a result, comparisons are made where the ratio b/a is small, and it is supposed that ruptures are caused by smaller external loads with an enlargement in the ratio b/a. For verification, analyses using the finite element mehtod (F.E.M.) were also carried out. The results for the case of $b/a = 3.0$ are shown in Figure 4. The results by elasticity are in fairly good agreement with the results by F.E.M. The comparisons between the analytical and the experimental results, (Figure 2), are presented in Figure 5. The strain distributions have a similar tendency to the stress distributions

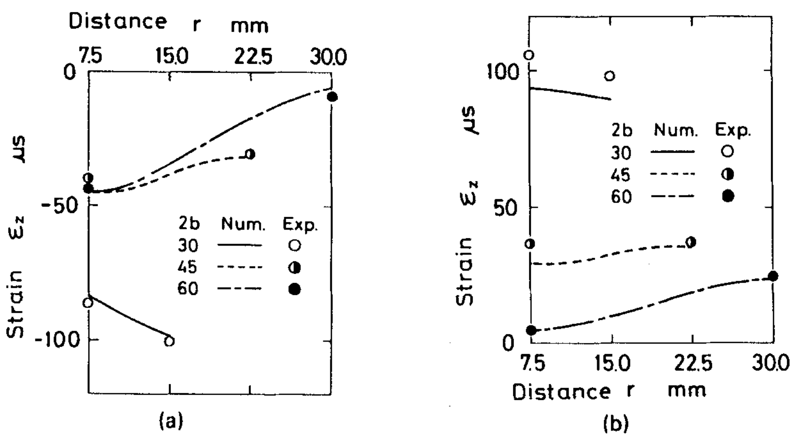

FIGURE 5 Comparisons of analytical and experimental results with respect to the strain distribution near adhesive layer, (a) the case where clamping force acts on the bearing surfaces ($F_f = 10$ kN), (b) the case where external load is applied ($W = 10$ kN).

shown in Figure 4. The measured results at the inner and the outer surfaces of the hollow cylinder are indicated in Figure 5. They are in fairly good agreement with the analytical results.

4.2 Increment in bolt axial force

Figure 6 shows the variation in the bolt axial force between the combination joint and the bolted joint without an adhesive. The

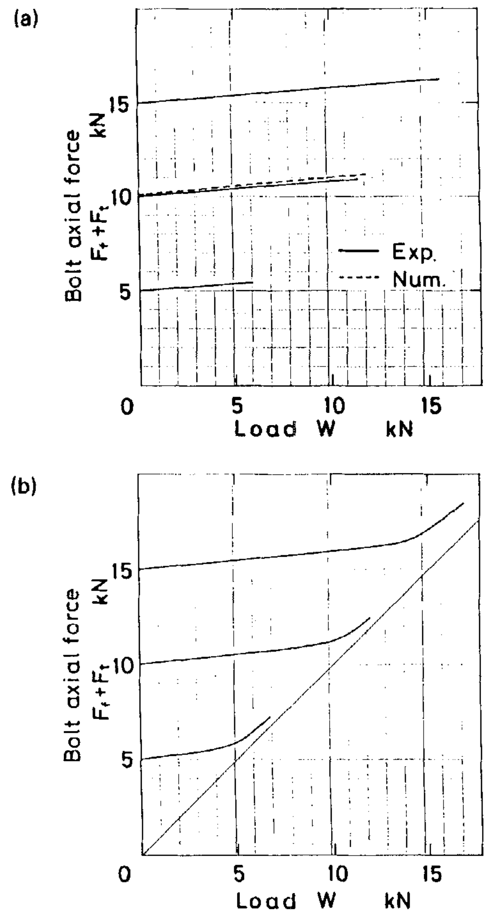

FIGURE 6 Variation of bolt axial force, (a) the case of a combination joint, (b) measured example of a bolted joint.

TABLE I
Comparisons of analytical and experimental results with respect to the force ratios

2b	30	45	60
$\phi_{\text{Num.}}$	0.091	0.020	0.004
$\phi_{\text{Exp.}}$	0.08	0.02	0.00

ordinate is a bolt axial force $F_f + F_t$ and the abscissa is an external load W. The initial clamping force F_f is varied as 5, 10 and 15 kN. In the case of the bolted joint without an adhesive, an increment F_t in the bolt axial force increases linearly with the increase of the external load W. When a separation of the contact surfaces is caused, F_t increases non-linearly. When the load W continues to increase, the contact surfaces separate completely. At this time, the bolt axial force $F_f + F_t$ equals the external load W. On the contrary, for the combination joint, the increment F_t in the bolt axial force increases linearly until a rupture of the adhesive layer occurs. The tangent of the linear line is independent of the initial clamping force F_f. It is preferable for the strength design of bolts to restrain the increment in the bolt axial force caused by the separation of the contact surfaces. The force ratio indicates the inclination of the straight line. In the case of $2b = 30$ mm, the spring constants K_c and K_{pt} are obtained as 933.5 and 2233 N/μm respectively as shown in Figure 2. The spring constant K_t is obtained as 260.6 N/μm utilizing Reference 9. Using these spring constants, the force ratio Φ is obtained as 0.091 from Eq. (4). The analytical value of the force ratio is shown in Figure 6(a) where the initial clamping force F_f is 10 kN. Table I shows comparisons between numerical and experimental results with respect to the force ratio. Both results are in fairly good agreement.

4.3 Strength of combination joint

Table II shows comparisons between the strength of combination joints and adhesive butt joints. The outer diameter $2b$ of the hollow cylinders is 30 mm and the thickness of adhesive layers is approximately 0.06 mm. In the case of adhesive joints, the average value of the breaking load W_a, which is obtained from three measurements,

TABLE II
Comparisons of the strength of joints (the case of $2b = 30$ mm)

(kN)

n	W_a	W_b
1	29.1	42.7
2	27.7	47.2
3	32.1	42.7
Mean	29.6	44.2

a: Adhesive
b: Combination

is 29.6 kN. It is 44.2 kN in the case of combination joints. The breaking load for combination joints is greater by 4.6 kN than the value added from the initial clamping force F_f to W_a. One reason is that part of the external load W is distributed to the bolt as shown in Figure 7. In this figure, the ordinate is the bolt axial force $F_f + F_t$ and the abscissa is the external load W.

In the case of the bolted joint, where separation of the contact surfaces occurs, F_t increases greatly as indicated by the dotted line. When the contact surfaces are completely separated by enlarging the external load W, the value of $F_f + F_t$ equals the value of W. On the contrary, F_t increases linearly in the case of a combination joint. When the adhesive layer is ruptured by an application of load

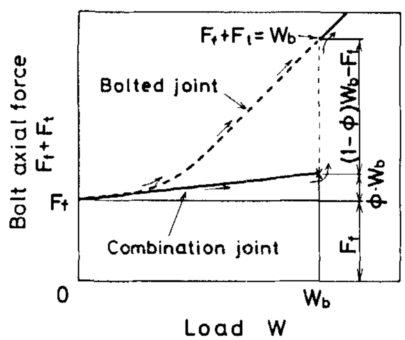

FIGURE 7 Estimation of the strength of a combination joint.

$W = W_b$, F_t suddenly increases and it equals the value of F_t of the bolted joint. It is thought that when the value of the force $F_c - F_f = (1 - \Phi)W - F_f$ which is transmitted to the adhesive layer reaches the value W_a, which is the breaking load of the adhesive joint, the joint is ruptured in the adhesive layer. Thus, the breaking load W_b is given by the following equation,

$$W_b = \frac{W_a + F_f}{1 - \Phi} \tag{5}$$

where the outer diameter $2b$ is 30 mm, the breaking load of the adhesive joint is determined as 29.6 kN, shown in Table II, and the force ratio is 0.091, shown in Table I. Using these values, the breaking load W_b is estimated as 43.6 kN. This result is in a fairly good agreement with the result shown in Table II. The breaking stress σ_a is obtained by the equation $\sigma_a = W_a/A$, (A = area of adhesion), because the deviation of the stress distribution is small in the case of $2b = 30$ mm as shown in Figure 4.

Next, using the value σ_a, the strength of the combination joint, of which the outer diameter $2b$ is 45 mm, is estimated. Taking account of the stress distribution shown in Figure 4 and the force ratio shown in Table I, the strength of the combination joint is estimated as 82.9 kN. The strength obtained by experiments is 75.5 kN. It is possible to estimate the strength of the combination joint using Eq. (5). According to Eq. (5), W_b becomes greater with an enlargement in F_f. If F_f is enlarged to such an extent that the adhesive layer is not ruptured, it is effective in improving the strength of the combination joint. In the case of the adhesive joint, the shape is lost after the rupture of the adhesive layer. But in the case of the combination joint, the shape is maintained after the adhesive layer is ruptured. Thus, the combination joint satisfies the first object, which is the fail-safe viewpoint. In the case of $2b = 45$ mm, which has a larger adhesive area, a bolt reaches its yield near holes drilled in the shank. However, the shape of the joint is maintained. It is considered that the combination joint is safe for a greater external load, since bolts, which have enough strength to bear the breaking load, are used and the initial clamping force is satisfactory. It is necessary to select the size and strength of bolts so that the shape is maintained after the adhesive layer is ruptured.

5. CONCLUSIONS

This paper describes the strength characteristics of a combination joint consisting of an adhesive with bolts. In the case where adherends are hollow cylinders, stress distributions at the boundary of the adhesive layer, the force ratio and the strength of the joints are exmained. The following results are obtained:

(1) The stress distribution in combination joints is analyzed using a three-dimensional theory of elasticity.

(2) According to (1), the stress distribution at the boundary of the adhesive layer is analyzed where an initial clamping force and an external load are applied separately. The effects of the ratio of the outer diameter to the inner one are made clear.

(3) To verify the results of (1), the strains near the adhesive layer are measured and compared with the analytical results. The agreement is thought to be fairly good. In addition, analyses using F.E.M. are made and the stress distribution near the adhesive layer almost agrees with the result of (2).

(4) The methods to estimate the strength of the combination joint using the stress distribution at the boundary of the adhesive layer are demonstrated. The strength of the combination joint is greater than that of an adhesive joint. In the combination joint, the joint's shape is maintained after the adhesive layer is ruptured. In addition, availability of combination joints is indicated.

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