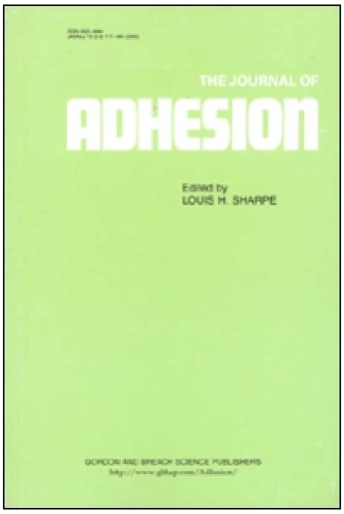

The Journal of Adhesion
Publication details, including instructions for authors and subscription information:
http://www.informaworld.com/smpp/title~content=t713453635

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To cite this Article Lee, Dai Gil , Kim, Ki Soo and Im, Yong-Taek(1991) 'An Experimental Study of Fatigue Strength for Adhesively Bonded Tubular Single Lap Joints', The Journal of Adhesion, 35: 1, 39 — 53
To link to this Article: DOI: 10.1080/00218469108030434
URL: http://dx.doi.org/10.1080/00218469108030434

J. Adhesion, 1991, Vol. 35, pp. 39–53
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Printed in the United Kingdom

An Experimental Study of Fatigue Strength for Adhesively Bonded Tubular Single Lap Joints

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(Received September 28, 1990; in final form March 25, 1991)

In this paper, manufacturing technology of the tubular single lap adhesive joint was studied to obtain reliable and optimal joint quality. In addition, a surface preparation method and a bonding process for the joint were devised. The effect of the adhesive thickness and the adherend roughness on the fatigue strength of the joint was experimentally investigated. From experiments, it has been found that the fatigue strength of the joint increased as the adhesive thickness decreased and the optimal arithmetic surface roughness of the adherends was about 2 μm.

KEY WORDS Manufacturing technology; adherend surface roughness; surface preparation; bonding process; adhesive thickness; torsional strength.

INTRODUCTION

Adhesives have been widely used as a means of joining materials. Recently, the science and technology of adhesion and adhesives have progressed significantly and major advances have been made.[1] The reason for the wide use of adhesives is that adhesives possess properties that enable them to adhere readily to other materials and to have an adequate strength so that they are capable of transmitting the applied loads or forces from one substrate to the other.

With the wide applications of fiber-reinforced composite materials in aircraft, spaceships, even robot arms and machine tools to exploit their high specific stiffness, high strength and high damping properties,[2,3,4] the design and manufacture of joints have become a very important research area because the structural efficiency of a composite structure is established, with very few exceptions, by its joints and not by its basic structures.[5] The design of joints has a special significance because the joints are often the weakest areas in the composite structures.[6]

There are two kinds of joints: mechanical and bonded. Mechanical joints are made by fastening the substrates with bolts or rivets. An adhesive interlayer between the adherends is used in bonded joints. Bonded joints have several advantages over mechanical joints. They distribute the load over a larger area than in the mechanical joints. They require no holes and add very little weight to the structure. However, they are difficult to disassemble, and their strength is affected by adherend geometry, quality of surface treatment, service temperature, humidity and other environmental conditions.

There are several types of tubular lap joints, such as single lap joint, double lap joint, stepped lap joint, and scarf joint. In these joints, the single tubular lap joint is most popular, due to its ease of manufacture and its relatively low cost. However, the single tubular lap joint leads to high stresses and rapidly changing stress gradients in the end region of the adhesive layer.

Static studies, by analytical and finite element methods, for the single tubular lap joint have been made more extensively by many researchers than for other configurations. Alwar and Nagaraja[7] used the finite element method to obtain the stress in a tubular joint subjected to torsion. The time dependent properties of the adhesive were taken into account in the finite element solution. Adams and Peppiatt[8] refined the solution of Volkersen[9] and arrived at a closed form solution for the shear stress, $\tau_{r\theta}$, in an adhesively bonded tubular lap joint subjected to a torque (Fig. 1). They also analyzed adhesively bonded tubular lap joints which were subjected to axial and torsional loads, using the finite element method, when the adhesive had a fillet. Chon[10] analyzed the adhesive tubular lap joint whose adherends were composite materials in a closed form by a method similar to Adams'. Hipol[11] analyzed the tubular lap joint comprised of a steel tube adhesively bonded to a composite tube subjected to torsion. He used the finite element method to characterize the stress concentration associated with the boundary layer effect in the end region of the adhesive layer. Graves and Adams[12] used the finite element method to calculate the stresses in a bonded joint whose adherends were orthotropic composite material subjected to torsion. He obtained the stresses in the adherends using ply by ply analysis. Also, he obtained the stresses in the adherends with

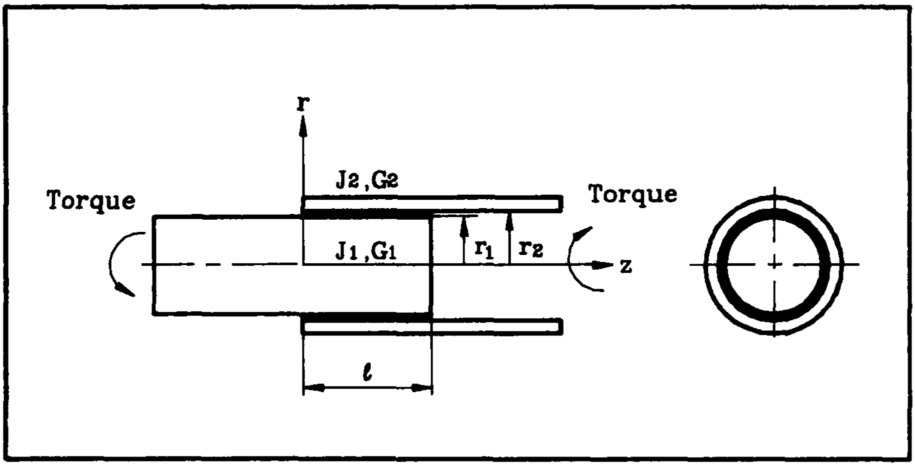

FIGURE 1 Adhesive bonded tubular single lap joint subjected to a torque.

smeared laminate properties. Hart-Smith[13] analyzed several adhesively bonded joints such as double lap, single lap, scarf, and stepped lap joint and created computer software for analysis of stress distribution in the various joints. Imanaka *et al.*[14] investigated the effect of filler addition on the fatigue strength of an adhesively bonded butt joint under cyclic load. They also investigated the effect of the adhesive thickness on the fatigue strength.

While the static theoretical investigations of the adhesive tubular lap joint are extensive, the experimental verifications of the static and dynamic performances of the adhesive joint are rare. Since many adhesive joints are subjected to alternating stresses, it is necessary to design the adhesive joint based on the dynamic fatigue strength. The importance of dynamic fatigue is that under fluctuating loads joints will fail at stress levels much lower than under monotonic loading. Also, for a given alternating stress amplitude, the dynamic fatigue time will be much shorter than the static fatigue time which was obtained when a constant stress of the same magnitude has been applied.[1]

The parameters which affect the static and dynamic fatigue strength of adhesive joints are many, such as surface roughness of adherends, adhesive thickness, geometric shapes of adherends, curing and environmental conditions (pressure, temperature, and humidity).

In this paper, the effect of the adhesive thickness and the adherend roughness on the fatigue strength of adhesively bonded tubular single lap joints has been investigated by an experimental method. Carbon steel which has 0.45% carbon content (S45C) was chosen as the adherend material in order to control accurately the surface roughness and adhesive thickness.

PROPERTIES OF THE ADHESIVE USED IN THE JOINT

The adhesive material used in tests was an epoxy resin (IPCO 9923) manufactured by Imperial Polychemicals Corporation of U.S.A. The epoxy was prepared using a one-to-one mix ratio between resin and hardener as recommended for thin film, bonding and repairing applications. It was a ductile epoxy which was toughened by adding rubber. Its pot life was 75 minutes at 25°C and its maximum use temperature was 120°C. Figure 2 shows the tensile test results for the epoxy adhesive. Table I shows the mechanical properties of the adhesive and Table II shows the experimental conditions for determining lap shear strength (ASTM D-1002-72).

TEST SPECIMENS

The size and the shape of test specimens were chosen according to the theory of Adams and Peppiatt.[8] Their theory for the tubular single lap joint, without an adhesive fillet, which was subjected to torsion (Fig. 1) is reproduced here for reference. The shear stress $\tau_{r\theta}$ in the adhesive was given as follows:

$$\tau_{r\theta} = \frac{T\alpha}{2\pi a^2}\left[\left(\frac{1-\psi(1-\cosh\alpha\ell)}{\sinh\alpha\ell}\right)\cosh\alpha z - \psi\sinh\alpha z\right], \qquad (1)$$

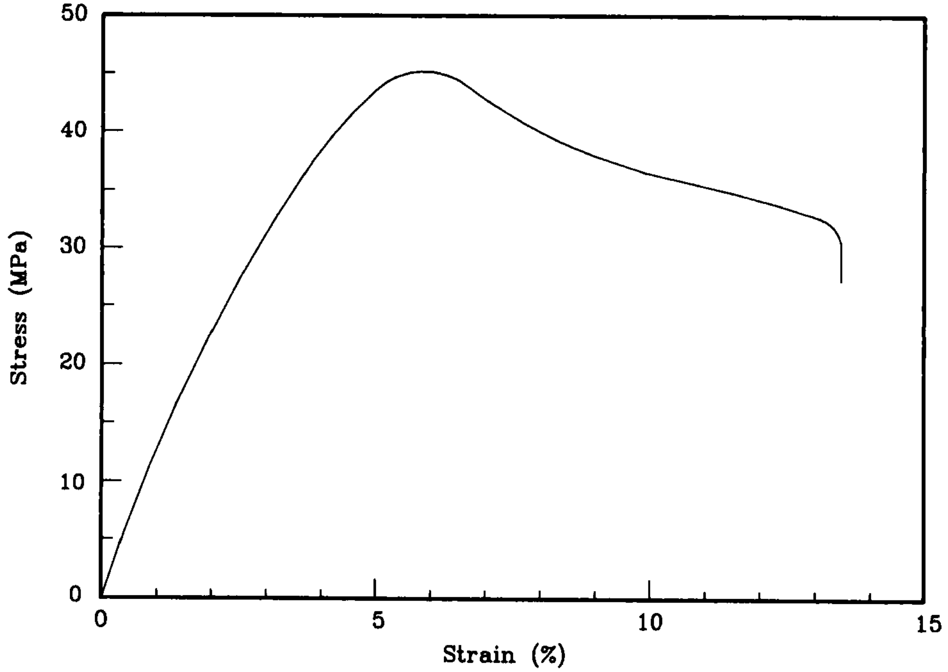

FIGURE 2 Tensile stress-strain behavior of the epoxy resin (IPCO 9923) used in the adhesive joint.

TABLE I
Mechanical properties of the epoxy adhesive (IPCO 9923)

Lap shear strength (MPa)	13.7 (ASTM D-1002-72)
Tensile modulus (GPa)	1.3
Poisson's ratio	0.41
Shear modulus (GPa)	0.46
Tensile strength (MPa)	45
Cure condition	100 hours at 25°C, 1 atmosphere

TABLE II
Experimental conditions for determining lap shear strength (ASTM D-1002-72)

Cure condition	100 hours at 25°C, 1 atmosphere
Adherend material	S45C steel
Adherend size (L × W × T)	97.7 × 25.4 × 1.6 mm
Adherend surface roughness	1.5 μm (arithmetic)
Adherend surface treatment	10 minutes in 10% nitric acid + 90% ethanol solution
Bonding area (L × W)	12.7 × 25.4 mm
Adhesive thickness	0.1 mm

$$\psi=\frac{G_2J_2r_1}{r_2G_1J_1+r_1G_2J_2}, \tag{2}$$

$$\delta=\frac{2\pi a^2\,r_1G_a}{G_1J_1\eta}, \tag{3}$$

$$\alpha=\left(\frac{\delta}{\psi}\right)^{1/2}, \tag{4}$$

$$a=\frac{r_1+r_2}{2}, \tag{5}$$

$$\eta=r_2-r_1, \tag{6}$$

where J_1 and J_2 are the sectional polar moments of inertia of the adherends, G_1 and G_2 are the shear moduli of the adherends, G_a is the shear modulus of the adhesive, r_1 and r_2 are the inner and outer radii of the adhesive, respectively, and ℓ is the adhesive length of the joint. Here, subscripts 1 and 2 represent inner and outer adherends, respectively.

According to Eq. (1), the shear stress at $z=0$ is larger than the shear stress at $z=\ell$ when ψ is larger than 0.5. The minimum shear stress occurs at the intermediate value of z. If we express this value of z as z_{min}, it is expressed as

$$z_{\min}=\frac{1}{2\alpha}\,\ell n\,\frac{1-\psi+\psi\cosh\alpha\ell+\psi\sinh\alpha\ell}{1-\psi+\psi\cosh\alpha\ell-\psi\sinh\alpha\ell}. \tag{7}$$

The shear stress at $z_{\min}$ is usually much smaller than the shear stresses at both ends ($z=0$, $z=\ell$). This is a kind of stress concentration phenomenon. Owing to this stress concentration, the torque capability of the adhesive joint does not increase indefinitely with the increase in the joint length, but levels off quickly.

In the present experiments, the joint length 15mm was used as depicted in Figure 3 and Figure 4, because the torque capability saturates quickly when the joint length was larger than this value when the adhesive thickness was 0.25 mm (10 mil).

It can be found out from Eq. (1) that as the adhesive thickness changes, the static torque capability also changes. Figure 5 shows the calculated maximum torque capability $T_{\max}$ and the calculated average shear stress τ_a in the adhesive with respect to adhesive thickness. In this calculation, τ_a was defined by the following equation

$$\tau_a=\frac{T_{\max}}{2\pi a^2\ell}. \tag{8}$$

It can be seen in Figure 5 that $T_{\max}$ and τ_a decrease about 70% and 60%, respectively, as the adhesive thickness decreases from 1mm to 0.1 mm. This result was also observed by Hipol through FEM simulations.[11] In addition, $T_{\max}$ becomes saturated quickly with the adhesive thickness if the adhesive thickness is larger than 0.5 mm. Therefore, in the present experiment, the fatigue test was performed with the adhesive thickness smaller than 1.8 mm.

Since the accurate machining and control of surface roughness of steel adherends were easier than those of polymeric composite materials, S45C steel was chosen as the adherend material. In order to control the surface roughness of the adherends,

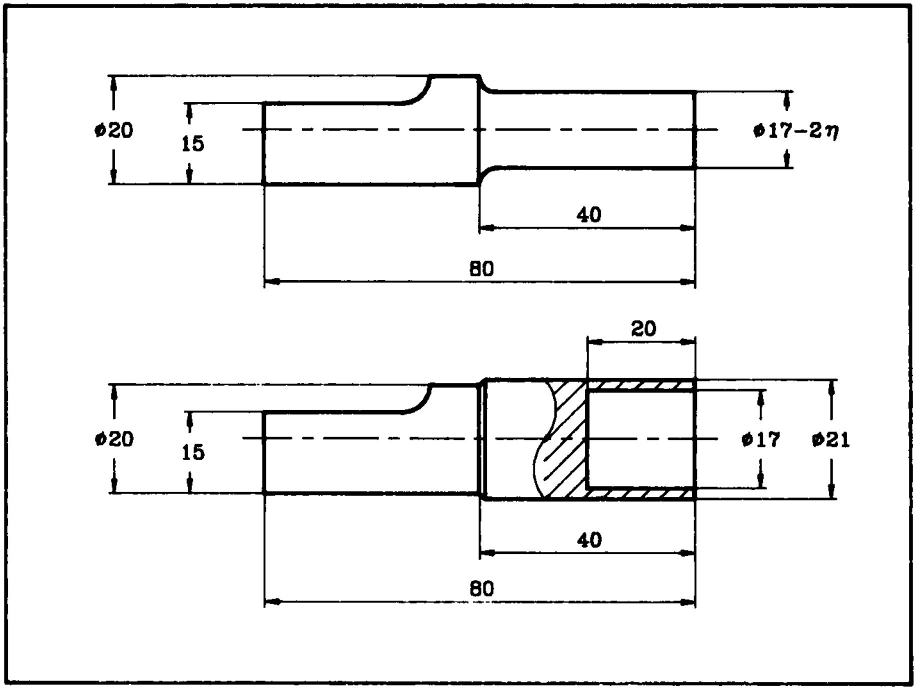

FIGURE 3 Dimensions of the adherends of the tubular single lap joint.

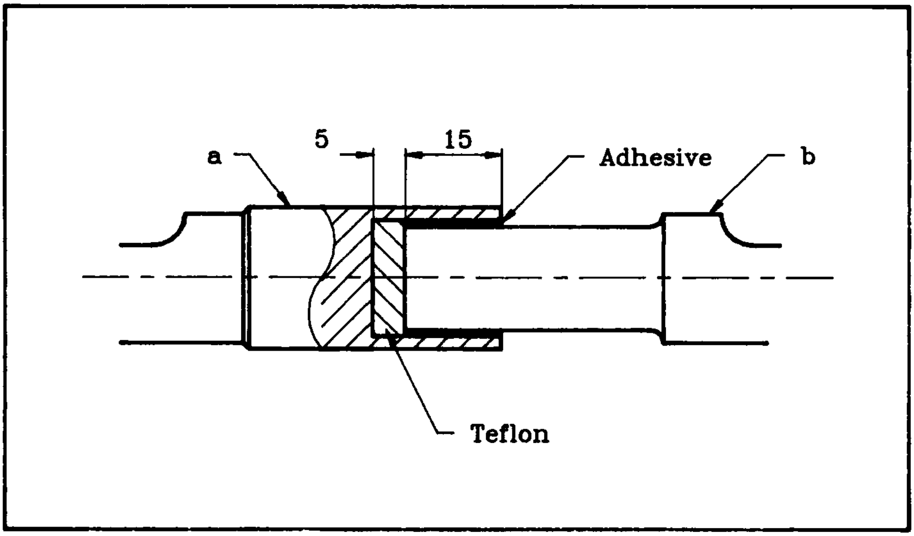

FIGURE 4 Adhesively bonded tubular single lap joint used in the fatigue experiments.

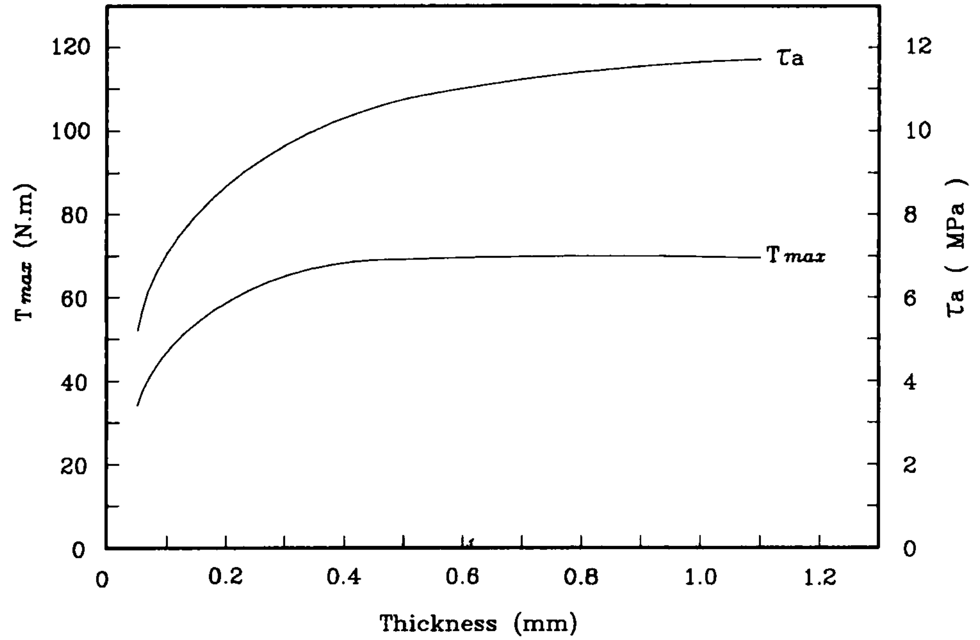

FIGURE 5 Effects of the adhesive thickness on the maximum static torque transmission capability (T_{max}) and the average shear stress (τ_a) in the adhesive when T_{max} is applied to the joint in Fig. 4 (T_{max} and τ_a are calculated from Eqn. 1).

the outer surface of the inner adherend was turned by a lathe and the inner surface of the outer adherend was bored out. Since close control of the surface roughness of the adherend in the bonding operation is indispensable, the surface roughness of the adherend was measured by a surface roughness tester after machining. The surface roughness tester used was SZ-3A type made by Kosaka in Japan. Figure 6a shows the surface roughness of the adherend when machined by a lathe. The surface roughness in this figure was produced when the cutting speed, feed rate and depth of cut of machining were 100 m/min, 0.1 mm/rev, and 0.2 mm, respectively. The arithmetic surface roughness of the machined surface was approximated by the following equation:[15]

$$R_a = \frac{0.032\, f^2}{r_a}, \tag{9}$$

where f is the feed rate (mm/rev) and r_a is the nose radius of the cutting tool. However, there are many sharp points in the machined surface and these sharp points might work as crack initiators in the adhesive under cyclic loading. Therefore, it was concluded that the machined surfaces should not be used even though average arithmetic surface roughness was governed by the Eq. (9). Figure 6b shows the surface roughness of the same adherend when abraded with 180 mesh abrasive paper after machining in a lathe. The sharp points of the adherend were greatly reduced by abrasion, as seen in this figure.

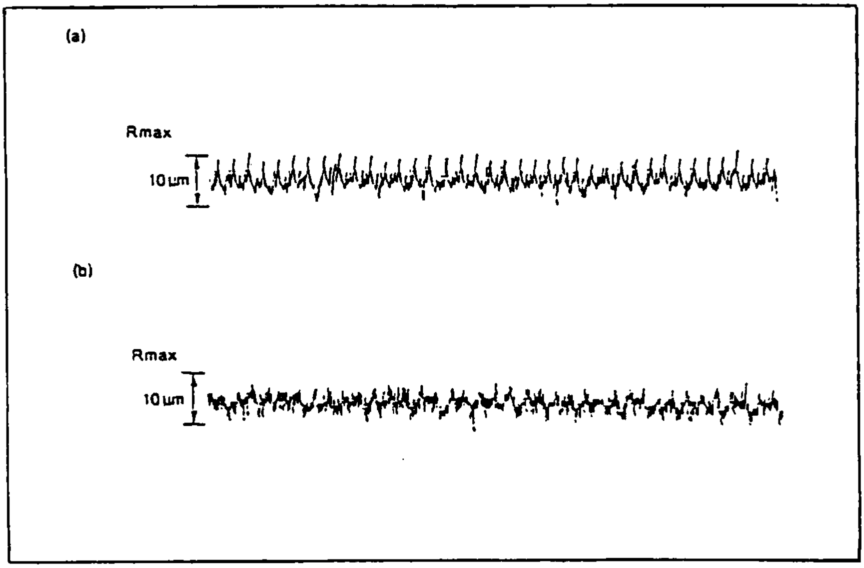

FIGURE 6 Surface roughnesses of the adherends: a) when turned with lathe and b) when abraded by an abrasive paper (mesh number 180) after machining in a lathe.

In order to check the relationship between the surface roughness and the mesh number of the abrasive paper, the adherend was abraded by several abrasive papers with different mesh numbers from 60 to 1200 in a dry condition. In this test, it was found that the abraded surface roughness was dependent on the adherend material, the mesh number of the abrasive paper, abrasive speed, abrasion pressure and the surface roughness of the original adherend. Even though it was not easy to get quantitative results, we found that the surface roughness decreased as the abrasion speed and the mesh number increased. Also, the required abrasion time was dependent on the surface roughness of the original adherend. Figure 7 shows the surface roughness of a S45C adherend which was 20 mm in diameter and was abraded by turning in a lathe. In this test, the abrasion speed was fixed at 4 m/min. The original arithmetic surface roughness was 1 to 1.3 μm and the pressing force on the abrasive paper, which was bonded on a flat steel plate, was 300 N. The surface roughness was measured in the axial direction. The deviation of the surface roughness was rather large when the mesh number of the abrasive paper was less than 400. This was attributed to the nonuniform abrasion pressure and the clogging of the abrasive paper. The roughness of the abraded adherend varied little when the mesh number was larger than 400.

Since the test specimen "a" in Figure 4 was a hollow tube and "b" was a solid rod, a 5mm length gap was introduced by inserting a Teflon slice. In order to prevent the adhesive from penetrating the gap, the Teflon slice was made with close tolerance.

The bonding operation was performed in a room where the relative humidity was kept less than 40% by dehumidification. The mixed adhesive (resin and hardener) was stored for 10 minutes in a vacuum chamber to eliminate air bubbles. The adhe-

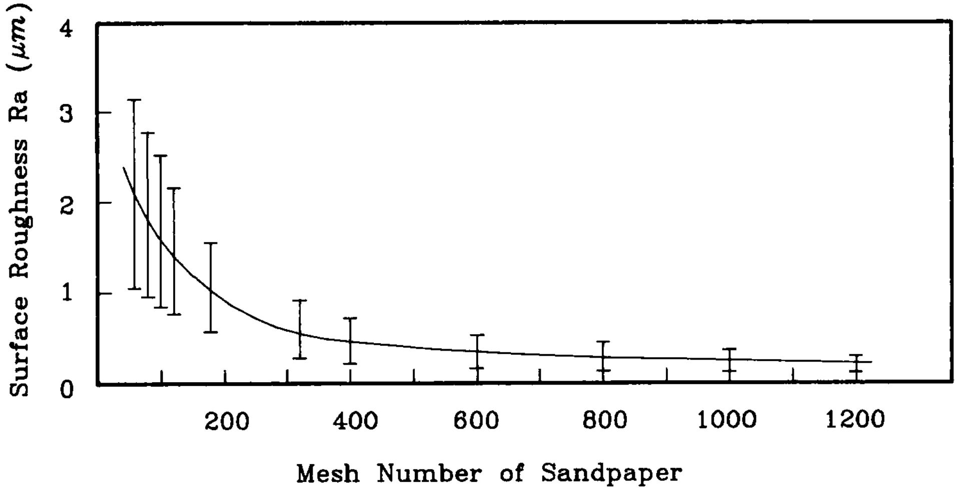

FIGURE 7 Relationship between the surface roughnesss and the mesh number of the abrasive paper (Adherend diameter 20 mm, Abrasion speed 4.0 m/min).

sive material was then carefully poured into an injector so as not to introduce additional air bubbles. The adhesive in the injector was slowly injected onto the bottom surface of the outer adherend. After this, the inner adherend was pushed to the bottom surface of the outer adherend slowly. Since the concentric bonding of the adhesive tubular lap joint was indispensable for the reliable joint quality, the joint after wetting with the adhesive was clamped using a V-block as shown in Figure 8.

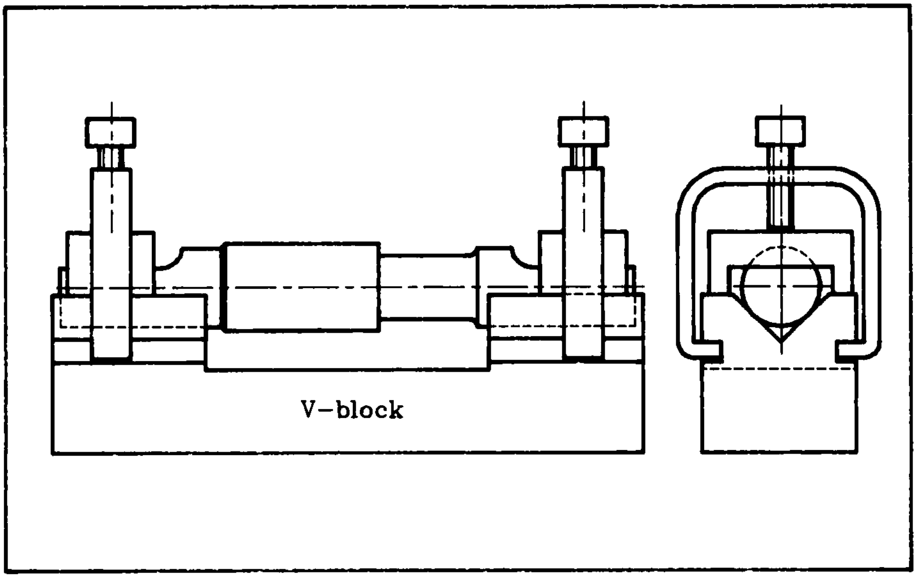

FIGURE 8 V-block used in the adhesive bonding to remove eccentricity between adherends.

The joint clamped in the V-block was cured in an autoclave, to suppress the size of the bubbles which might still remain, under an air pressure of 0.7 MPa. Since the epoxy resin used in this study had minimum viscosity at around 80°C, the joint was cured for 18 hours at 80°C. Then, the joint was postcured for 6 days at 30°C. In order to eliminate the effect of the adhesive fillet on the stress concentration in the adhesive, the outside adhesive fillet (Fig. 4) was cut by a razor before testing.

EXPERIMENTS

The fatigue tester used in this work was a Shimadzu TB-10 made by Shimadzu Corporation in Japan. It had a dynamic capacity of ±50 N • m for both bending and torsion applications. The bending and torsional moments of the tester were produced by rotating the eccentric mass, and the frequency of the cyclic stress was fixed at 2,000 rpm. If the frequency of the cyclic stress is high, there is a chance of a temperature rise in the adhesive. In experiments, however, since the adherends were made of steel and had larger masses than the adhesive, the heat generated in the adhesive could be dissipated easily through the steel adherends. Therefore, the problem of the high frequency of the cyclic stress was not taken into consideration in this study.

The fatigue test results were expressed by the average fatigue strength S_a which is defined by the following equation:

$$S_a = \frac{T_{amp}}{2\pi a^2 \ell}, \tag{10}$$

where T_{amp} is the amplitude of the cyclic torque, and all fatigue experiments were done under the stress ratio R $(\sigma_{min}/\sigma_{max}) = -1$.

Since the dynamic characteristics of the adhesive material were dependent upon the environmental temperature, the fatigue tester was enclosed in an acrylic chamber and the inside temperature was maintained at 30 ± 1°C by a temperature controller. Also, the relative humidity inside the chamber was maintained at less than 60%.

EXPERIMENTAL RESULTS AND DISCUSSIONS

Figure 9 shows a fatigue test specimen which is mounted on the clamping device of the fatigue tester. Figure 10 depicts the fatigue test results (*i.e.* S-N curve) of the tubular single lap joint whose adherends were abraded with an abrasive paper whose mesh number was 100. The adhesive thickness was 1mm. The average endurance limit of the torsional joint in this case was about 3.5 MPa. Since the lap shear strength of the adhesive was 13.7 MPa (ASTM D-1002-72), the average endurance limit is less than 35% of the lap shear strength.

In order to investigate the dependence of the fatigue strength on the adherend surface roughness, the fatigue tests were carried out on tubular single lap joints with different adherend surface roughnesses. Figure 11 shows the relationship between

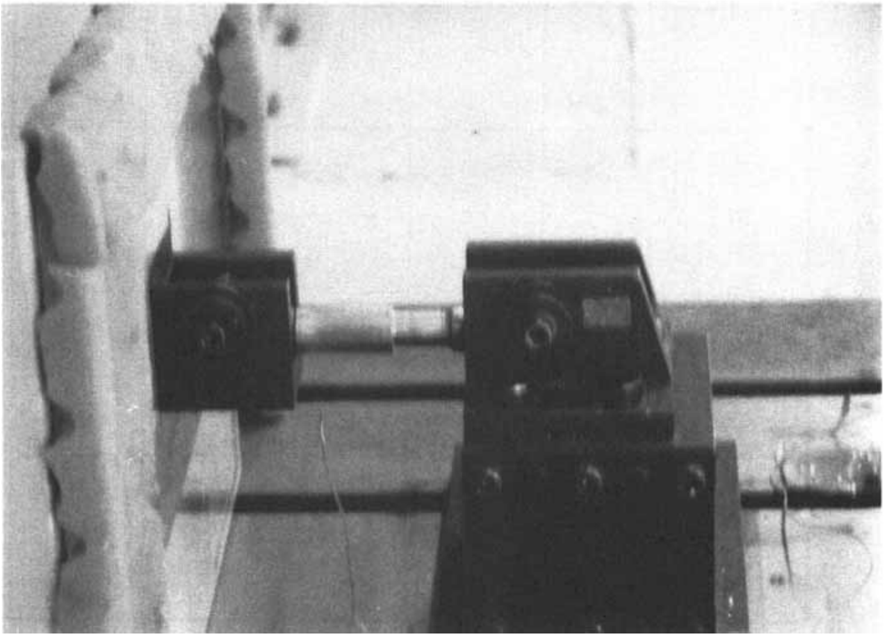

FIGURE 9 Photograph of the fatigue test specimen which is mounted on the clamping device of the fatigue tester.

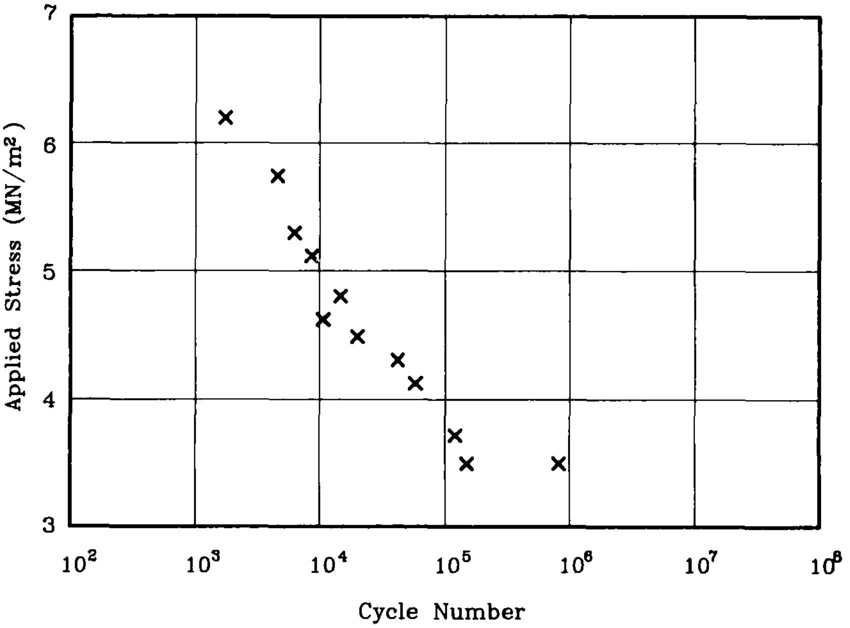

FIGURE 10 S-N curve of the adhesively bonded tubular single lap joint (the fatigue strength is the average shear stress in the adhesive and the adhesive thickness used in experiments was 1 mm).

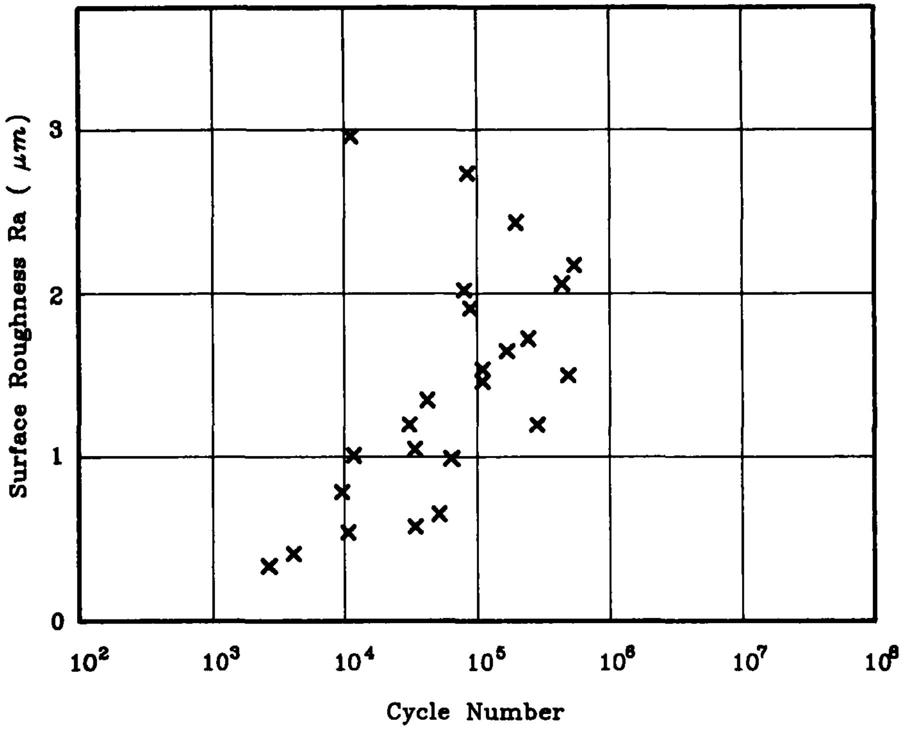

FIGURE 11 Effect of the surface roughness on the fatigue strength of the adhesively bonded tubular single lap joint (Adhesive thickness = 1 mm, Shear stress amplitude = 4.0 MPa).

the fatigue life and the adherend surface roughness when the adhesive thickness was 1 mm and the amplitude of the cyclic load was 4.0 MPa. The surface roughnesses of the adherends in this test were measured by the surface roughness tester after abrading the adherend surface by abrasive papers with different meshes. It was found out from this figure that the fatigue life decreased quickly when the arithmetic surface roughness was less than 1 μm. The fatigue life changed slowly when the surface roughness was in the range of 1.5–2.5 μm. If the surface roughness was larger than 2.5 μm, the fatigue life decreased quickly also. The failure of the adhesive usually occurred near the outer surface of the inner adherend. To look at the failure mechanism of the adhesive, the outer adherends of the several failed adhesive joints after fatigue testing were machined carefully in a lathe. Figure 12 shows the failed inner surfaces of the adhesives which were made flat after having been removed from the tubular adhesive joint. The surface of the failed adhesive taken from the adherends with 0.6 μm arithmetic surface roughness was very smooth and there was no adhesive left on the outer surface of the inner adherend. However, another failed adhesive surface from the adherends with 2.0 μm arithmetic surface roughness was not smooth and some part of the adhesive had peeled off and remained adhered to the outer surface of the inner adherend. When the arithmetic surface roughness was larger than 2.5 μm, it was found that the adhesive had usually

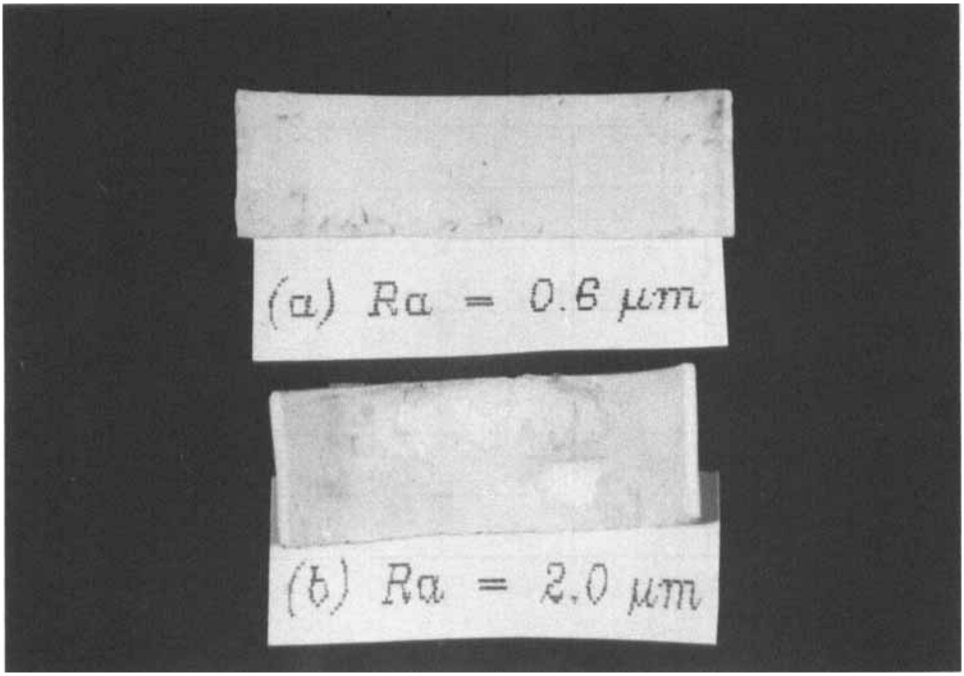

FIGURE 12 Surface fractographies of the failed adhesives in the fatigue tests. (a) arithmetic surface roughness of the adherend: 0.6 μm (shear stress amplitude: 4.0 MPa, failure cycle: 5.1×10^4). (b) arithmetic surface roughness of the adherend: 2.0 μm (shear stress amplitude: 4.0 MPa, failure cycle: 5.7×10^5).

several broken lines. This may suggest that the failure was propagated from the rough inner surface to the inside adhesive when the surface roughness of the adherend was larger than 2.5 μm. Therefore, the optimal arithmetic surface roughness of the adhesive tubular joint was around 2.0 μm.

Figure 13 shows the relationship between the adhesive thickness and the fatigue life. In this test, the arithmetic surface roughnesses of the testpieces varied from 1.5 to 2.5 μm, and the average amplitude of the cyclic load was 4.0 MPa. The endurance limit increased as the adhesive thickness decreased. This phenomenon was contrary to the static strength of the tubular single lap joint. However, it was found that the adhesive bonding operation without introducing eccentricity was very difficult if the adhesive thickness was less than 0.15 mm. Therefore, an adhesive thickness of 0.15 mm may be the optimal value for the tubular single lap joint subjected to cyclic torque. The fatigue limit of the adhesive joint with adhesive thickness of 0.15 mm was larger than 4.0 MPa which was more than 50% of the average shear stress, τ_a of the adhesive.

The decrease in the fatigue life with the increase in the adhesive thickness can be attributed to defects such as voids, dust and cracks which can be accumulated easily in the thicker adhesive.

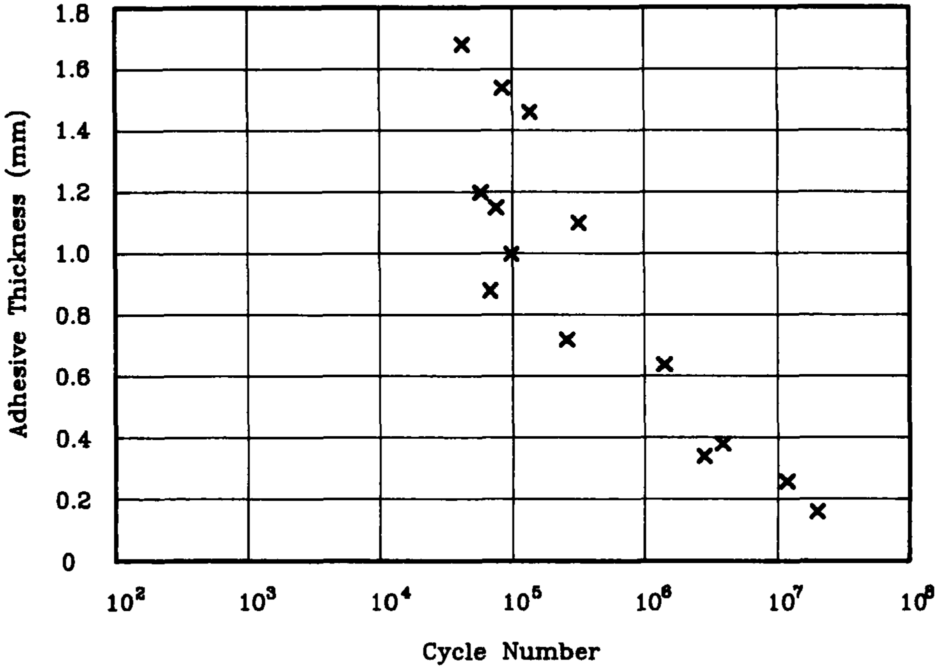

FIGURE 13 Fatigue life-adhesive thickness behavior of the adhesively bonded tubular single lap joint (Shear stress amplitude = 4.0 MPa, 1.5 μm ≤ R_a ≤ 2.5 μm).

CONCLUSIONS

In this research, the manufacturing technology of the adhesive tubular single lap joint was studied. The influences of adhesive thickness and surface roughness of the adherends on the torsional fatigue strength of tubular single lap joint were experimentally investigated. From the experimental observations, the following conclusions may be made:

(1) The optimal arithmetic surface roughness of the adherend was around 2 μm and the fatigue strength did not change much when the surface roughness varied from 1.5 to 2.5 μm.

(2) The torsional fatigue strength increased as the adhesive thickness decreased although this phenomenon was contrary to the static case. However, the optimal adhesive thickness was around 0.15 mm because the concentric bonding of the adhesive joint is impractical if the adhesive thickness is less than 0.15 mm.

(3) A machined surface should be abraded by abrasive papers in order for it to be suitable for use as the adherend surface. The optimal mesh number of abrasive paper for the adhesive tubular lap joint subjected to dynamic torque was around 100.

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