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THE EFFECT OF ANGLE ADHESIVE WITH BONDED IN Z TYPE MATERIALS

ABSTRACT

The usage of adhesives as connecting on method is increasing rapidly in today world. Many of research, development and engineering have been made to find the most important parameters for a success adhesion. In this study, stress analysis of Z type bonded sheet that connected with various adhesives has been investigated. The adhesive thickness was constant (0.20 mm) b lap joint length and lap joint angles (θ=15°, 30°, 45°) were varied for analysis. This study has deal with the effect of overlap angle on predicting of the damage load of adhesively bonded joints via a linear FEM(Finite Element Method). All of the analysis were carried out with Ansys (10.0) computer software with a generally purposed finite element application. Experimental results were compared with numerical results and were found quite reasonable.

Keywords: Adhesive, Stress Analysis, Interface, Finite Element Method (FEM), Finite Element Calculations

YAPIŞTIRICI İLE BİRLEŞTİRİLMİŞ Z TİPİ BAĞLANTILARDA AÇININ ETKİSİ

ÖZET

Günümüzde birleştirme yöntemi olarak yapıştırıcıların kullanımı hızlı bir şekilde artmaktadır. Başarılı bir yapıştırmada en önemli parametreleri bulmak için birçok araştırma, geliştirme ve mühendislik çalışmaları yapılmıştır. Bu çalışmada Z şeklinde bükülmüş ve değişik yapıştırıcılarla yapıştırılmış çelik sacların gerilme analizi yapılmıştır. Çalışmada yapıştırıcı kalınlığı (0.20 mm), b bindirme mesafesi sabit alınarak ve bindirme açıları(θ=15°, 30°, 45°) değiştirilerek analiz gerçekleştirilmiştir. Çalışmada lineer Sonlu Elemanlar Metodu(SEM)'nun yardımıyla yapıştırıcı bağlantının hasar yüküne açının etkisi kullanılarak araştırılmıştır. Bütün analizler genel sonlu elemanlar yazılımı olan Ansys (10.0) ile gerçekleştirilmiştir. Analiz sonuçları ile deneysel sonuçlar karşılaştırılmış, sonuçların oldukça iyi bir uyum gösterdikleri görülmüştür.

Anahtar Kelimeler: Yapıştırıcı, Gerilme Analizi, Arayüz, Sonlu Elemanlar Metodu(SEM), Sonlu Eleman Hesaplamaları

1. INTRODUCTION (GİRİŞ)

Adhesive joints have been used in mechanical structures, the automobile and aerospace industries, electric devices, and so on. Due to the many advantages offered by this method of joining, such as stress concentration reduction, the possibility to assemble dissimilar and/or thin materials, and protection against corrosion etc. Some studies have been carried out on the stress distribution of adhesive joints under static loadings such as tensile loads, bending moments and cleavage loads [1].

Adhesive bonding offers many advantages over classical fastening techniques such as welding, riveting and mechanical fastening. The substantial reduction in weight that can be achieved using adhesive bonding is an important advantage, especially for lightweight structures. However, the most common and most important factor influencing the long-term behaviour of unprotected adhesively-bonded metal joints is the presence of high humidity or liquid water [2].

When loaded in the tensile mode of adhesively bonded joints, they developed a linear stress pattern along the bonded overlap. Peak stresses, which may be several times the average failure stresses, are produced at the ends of the lap because of two factors: the differential strain induced between the adhesive and the adherents by the load, and the bending of the joint due to an eccentricity that results from the presence of the overlap. As the failure of a simple lap joint is determined by the maximum stresses at the ends of the overlap, joint modifications that produce a more uniform stress distribution yield stronger joints.

Many ideas have been suggested to reduce the high stresses that occur at the ends of the overlap. These ideas can be grouped into two general categories: material modification and geometrical modification. Material modification includes changing the material properties or fracture characteristics of adhesive, for example, by rubber toughening. Geometrical modifications involve altering the shape of the adherend and/or adhesive. Among these methods are pre-formed adherends, taper, fillets, rounding, adherend shape optimization, etc.[3].

Higuchi et al. have reported on the stress propagation of adhesive butt joints of T-shaped adherends subjected to impact tensile loads. In addition, it has been found that the characteristics of adhesive butt joints under impact loadings are different from those under static loadings. In practice, it is necessary to know the stress propagation and the stress distribution of adhesive joints subjected to impact bending moments from a reliable design standpoint, and to know the difference in the characteristics of adhesive butt joints under impact and static loadings [4].

A method for making the shear stress uniform along the bond length was presented by Cherry and Harrison [5]. This method was based on simple static equilibrium conditions. The tensile strains on both adherents were set equal to each other at each point by modifying the adherent thickness. It was assumed that the displacements through the thickness of the adhesive were negligible, the adhesive layer was thin enough so that the edge effects could be ignored, the bond length was much greater than the adherent thickness, and that the plane faces remained parallel to each other. Furthermore, peel stresses were not considered in this model. The ideal adherent profile for making the shear stress uniform was found to be a symmetric taper of the adherent along the bond line. It was also found that in addition to being a function of the adherent thickness, the shear stress was also a function of the Young’s modulus of the adherents.

Borgmeier and Devries also studied the effect of the modification of the lap joint geometry by tapering the adherents. A fracture mechanics approach was used for predicting adhesive joint failure to facilitate its application to practical joint configurations. In these studies, two groups of samples were tested: unmodified, and modified with tapered adherents. They reported that tapering of the adherents reduced the rate at which shear stress increased as the bond termini were approached. This, in principle, results in a more uniform distribution of the shear stresses over the overlap region of the joint [6].

In their stress analysis of single lap joint using FEM, Baylor and Sancaktar [7] showed that if the mesh density along the transverse direction of the overlap was greater than 3 elements per mm, then the variation in maximum principal stress and von-mises stress with mesh density would be effectively removed. It was also shown that for an adhesive thickness of 0.2 mm, 25 elements per mm in the peel direction would result in the uncoupling of these stresses with mesh density. Therefore, the FEM used in this work was designed with these two mesh densities as constraints on design.

The effects of loading rate, fiber sizing, test temperature and global strain level on the adhesion strength between carbon fibers and a thermosetting epoxy (Epon 815) are studied using the single fiber fragmentation test procedure. Analytical methodology describing the viscoelastic behavior observed is also presented. The possibility of rate-temperature-interphase thickness superposition for the interfacial strength function is illustrated based on the analytical models discussed. Experimental data are discussed using Weibull statistics and also presented in the form of percent relative frequency histograms for the fiber fragments in a collective fashion. The use of histograms allows for interpretation of the skewness in the data population [8].

2. RESEARCH SIGNIFICANCE (ÇALIŞMANIN ÖNEMİ)

In this study, the mechanical behaviours of bonded Z ties steel using two adhesives with different properties under a tensile load was analyzed. Experimentally results are compared with numerically results (FEM). In order to assess the performance of the adhesives (E type adhesive and W type adhesive) in this work, tensile experiments on the joints with different angle lap joint were carried out. The FEM calculations were performed in elastic deformation and it was assumed that the strain rate of the adhesive was small. The effects of angle adherends and the geometry of Z shaped adherends stresses at the interfaces were examined. Furthermore, the characteristic of adhesive joints of Z shaped joints subjected to tensile loads were examined by FEM.

After the stress analysis in the Z shaped joints was performed via non-linear finite element method by considering stress behaviours of adhesives and adherend (steel; $Fe_{49}Cr_{15}Mo_{14}C_{18}B_3Er_1$), experimental results were compared with the FEM results obtained by Temiz [9].

3. FINITE ELEMENT CALCULATIONS (SONLU ELEMAN HESAPLAMLARI)

Fig. 1 shows a model for calculations of a Z shaped adhesive joint. Figure 1 shows a model for FEM calculations of a Z shaped adhesive joint. Coordinate system (x,y) of specimens is used as shown in figure 1. Supports are inserted into edges of the adherends shown in Figure 1 to attach object to the specimen. Tensile load is applied in the x direction shown in Figure 1. The adherends thickness by t, adhesives thickness by n, adherend angle by θ, section of overlap angle by a, section of overlap non-angle by b, shown in Figure 1. The geometrical parameters and material properties used in the FEM analysis are given in Table 1 and Table 2, respectively.

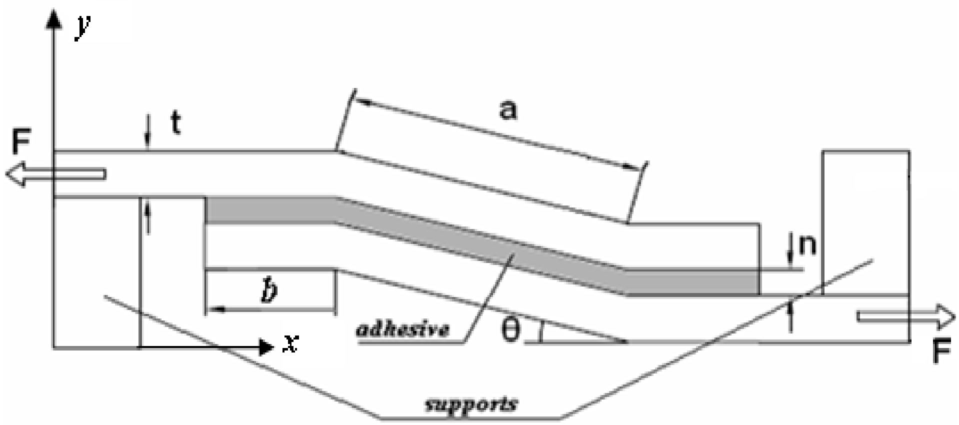


Figure 1. Geometry and tensile load (F) of specimen
(t: adherent thickness, a: overlap length I, b: overlap length II, n: adhesive thickness, θ: overlap angle)
(Şekil 1. Numunelerin çekme yükü ve geometrisi
(t: yapıştırıcı kalınlığı, a: bindirme mesafesi I, b: bindirme mesafesi II, n: yapıştırıcı kalınlığı, θ: bindirme açısı))

Figure 2 shows an example of mesh divisions. Also, the stress analysis of the Z ties adherend was carried the von-Mises yield criterion was used to calculate the equivalent stress (σ_{eqv}) distributions in the adhesives and adherends. In the analysis of the Z ties steel adhesive joints, 2D non-linear FEM was carried out.

Table 1. Geometrical parameters of the specimens used in experimental and numerical studies (all dimensions in mm)
(Tablo 1. Deneysel ve numerik çalışmalarda kullanılan numunelerin geometrik parametreleri(bütün ölçüler mm'dir))

Adh.Thick. (t)	Overlap Leng (a)	Overlap Length (b)	Adhes. Thickness(n)	Overlap Angle (θ)
5	30	13	0.20	15°
5	30	13	0.20	30°
5	30	13	0.20	45°

Table 2. The Mechanical properties for the adherends and the adhesives used in study
(Tablo 2. Çalışmada kullanılan yapıştırıcıların ve malzemelerin mekanik özellikleri)

Steel($Fe_{49}Cr_{15}Mo_{14}C_{18}B_3Er_1$)		E Adhesive	W Adhesive
E_x (GPa)	210	1.68055	1.92454
ν	0.32	0.28	0.30

E:Young's modulus; ν:Poisson's ratio

In this study, Ansys finite element package was utilized to evaluate the stresses. The Ansys code version 10.0 and two dimensional volume elements, Plane 82 and plane 2, were employed for the joints. The mesh density can affect the strain predictions in the adhesive layer. The mesh density remained 1 elements/mm. In adhesive geometries the mesh in the adherends was denser than adherends. However, further dimension changes cause only little effect when a specific size of finite element is reached.

A smaller element size will generally give a higher strain. For this reason, the size of the elements in the mesh was reduced until a stable maximum strain value had been achieved.

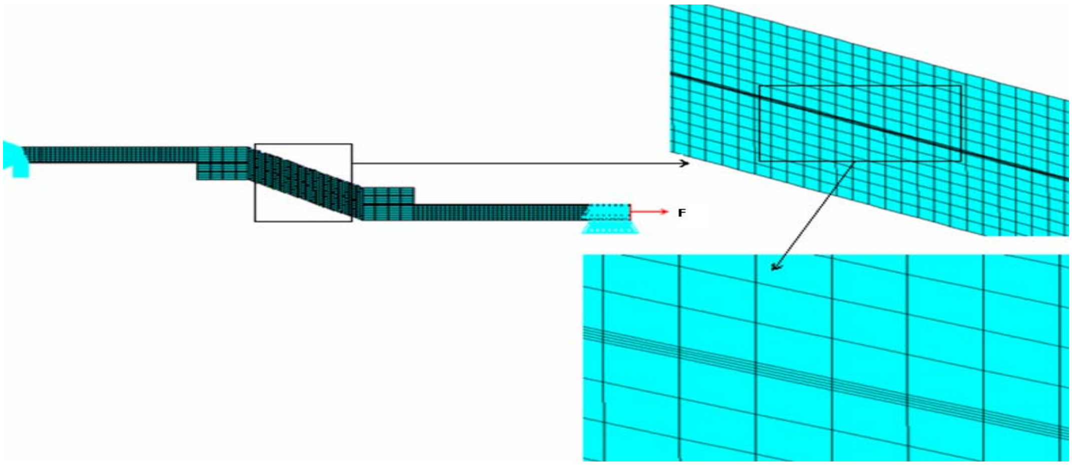


Figure 2. Finite element meshes of steel plates
(Şekil 2. Çelik levhanın sonlu eleman meshlenmesi)

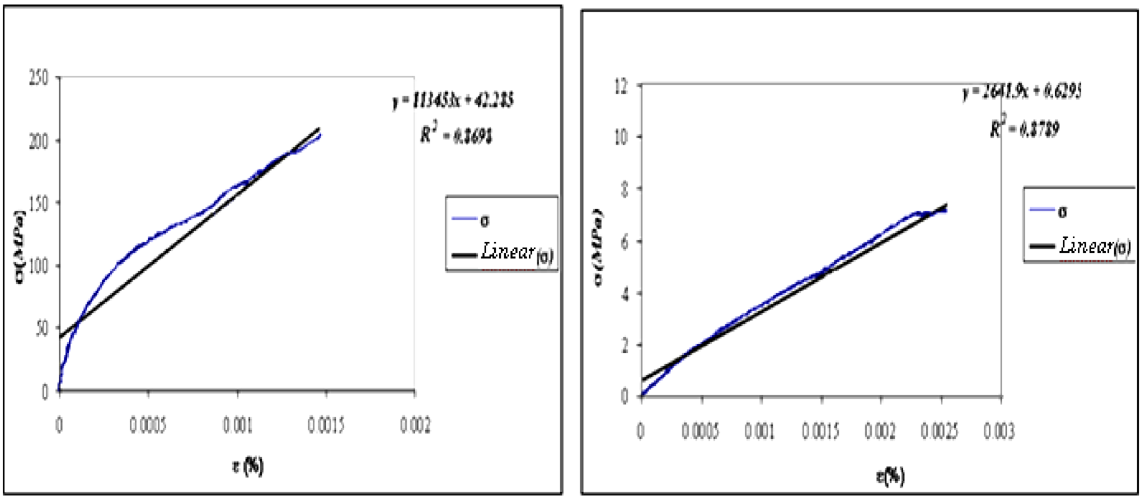
Consequently, 5 elements through the adhesive thickness were used in the models, as shown in Fig. 2, and the number of elements was varied for each overlap length. In the joints of adherend with adhesives, the nominal bondline thickness considered in all cases was 0.20 mm. The adhesive layer was divided into five meshes of 5 mm thickness in the y (thickness) direction after the effect and accuracy of the mesh divisions on the stress wave propagations and stress distributions were examined. When the minimum thickness of element was chosen (t=5 mm), it was confirmed that a difference in the calculated results of the interface stress distributions was very small.

4. EXPERIMENTAL METHOD (DENEYSEL METOD)

Figure 1 shows the dimensions of the specimens used. The specimens were made of steel($Fe_{49}Cr_{15}Mo_{14}C_{18}B_3Er_1$), and they were joined by an E and W adhesives of which Young's modulus was recpectively 1680.55 and 1924.54 MPa and Poisson's ratio was respectively 0.28 and 0.30. The surface impurities were removed using aseton, the interfaces of the specimens were joined by the adhesive, and the joint was cured at room temperature for 24 hours [10].

The stress-strain (σ-ε) behaviours of the adhesives was determined from bulk dumb-bell (dog bone) specimens tested under the conditions specified. Three specimens were tested to failure at a crosshead speed of 1 mm/min. The other experimental details are described in Ref. [11]. Typical tensile stress-strain curves for the two adhesives are shown in Fig. 3a and 3b, while the geometrical parameters and materials properties used in the FEM are given Table 2-3, respectively.

The tests were performed using Instron 1114 machine at room temperature (23°C) and 50% relative humidity. During tensile testing, the crosshead speed was maintained at 1 mm/min, and a 5 kN load was used. Three or four specimens were tested for each experimental condition analyzed, and the average values were shown in table 3. Also, the stress analysis of joint was carried the von-Mises yield criterion was used to calculate the equivalent stress (σ_{eqv}) distributions in the adhesives and adherends.



(a) (b)

Figure 3. Tensile stress-strain behaviours of adhesives:
a) E Type Adhesive; b) W Type Adhesive
(Şekil 3. Yapıştırıcıların gerilme-şekil değiştirme davranışları:
a) E tipi yapıştırıcı, b) W tipi yapıştırıcı)

Table 3. Experimental loads and damage loads
(Tablo 3. Deneysel yükler ve hasar yükleri)

Adherends angle (θ)	E Type Adhesive			W Type Adhesive		
	F_E	F_{FEM}	F_R	F_E	F_{FEM}	F_R
15°	135.323	142.740	1.0548	185.640	191.880	1.0336
30°	134.986	145.158	1.0753	186.420	190.320	1.0209
45°	131.349	144.066	1.0968	182.113	188.448	1.0347

F_E (N): Experimental damage load of adhesives; F_{FEM} (N): Damage load predicted from FEM adhesives; $F_R = F_E / F_{FEM}$ (Experimental load/ Finite Element Analysis load)(N).

5. FEM RESULTS WITH COMPARISON EXPERIMENTAL RESULTS (DENEYSEL SONUÇLARIYLA FEM SONUÇLARININ KARŞILAŞTIRILMASI)

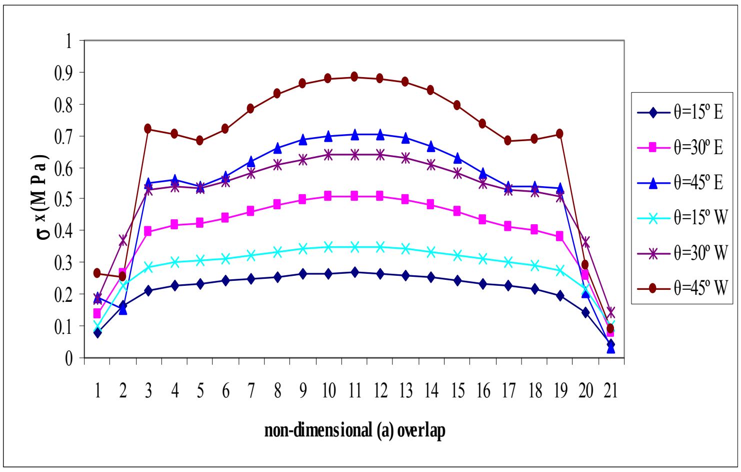
In the FEM calculations, the dimensions and the material constant used are the same as those used in the strain response measurements(Figure 1 and Table 2).

The solution in finite element considering non-linear material behaviour is reached by dividing the total load in steps to track the equilibrium paths and iterating to a converged solution at each load increment. In this study, the number of load steps for each joint type changed due to changing predicted damage loads.

The results predicted from FEM and obtained experimentally are shown in Table 3. When the FEM results are compared with the experimental results, the results found are compatible with FEM results. For this reason, in addition to other parameters such as the dependence on strain and the lack of yield criterions of adhesives, it can be said that the residual thermal stresses occurred due to the applied pressure during curing process at elevated temperature need to be taken into consideration so as to simulate accurately the mechanical behaviours of adhesively bonded joints. But, in practice, the magnitude of these stresses is difficult to predict. Therefore, more detailed investigation which comprises the mechanical and thermal properties of adhesives at different temperatures needs to be performed in order to explain the effect of curing pressure on the strength of adhesively bonded joints.

In order to predict the damage load, the stress (σ) of adhesives given in Table 1 was used and the adhesives was assumed to fail when the von-Mises equivalent stress (σ_{eqv}) calculated at any point of adhesive layer reaches the stress (σ) of the adhesives.

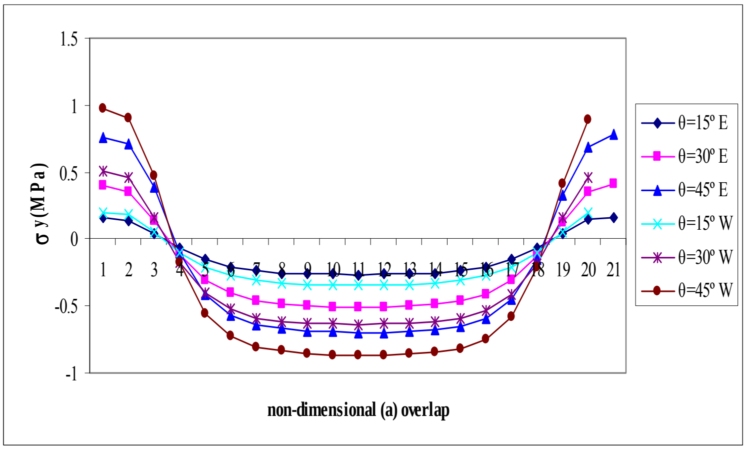
The present FEM analysis results have shown that the most critical points are along the adherend-adhesive interfaces and the maximum peel (σ_y) and shear (τ_{xy}) stresses are located between the centerline and the adherend-adhesive interfaces and at the opposite corner ends of overlap. For this reason, the bondline on the adhesive side was taken into consideration for the stress analysis (see Figure 5 and 7) and all of the stress (σ_x , σ_y, τ_{xy}, and σ_{eqv}) distributions were normalized (Figure 4, 5, 6 ve 7).



E: E type adhesive W: W type adhesive

Figure 4. σ_x normal stress distributions along the overlap length on the adhesive (adherend thickness=5 mm).

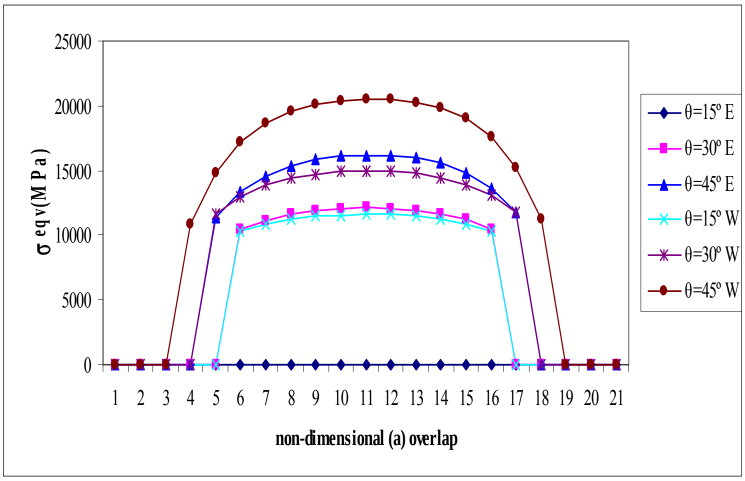
(Şekil 4. Yapıştırıcıda bindirme mesafesi boyunca σ_x gerilme dağılımı(adherend kalınlığı: 5 mm)).



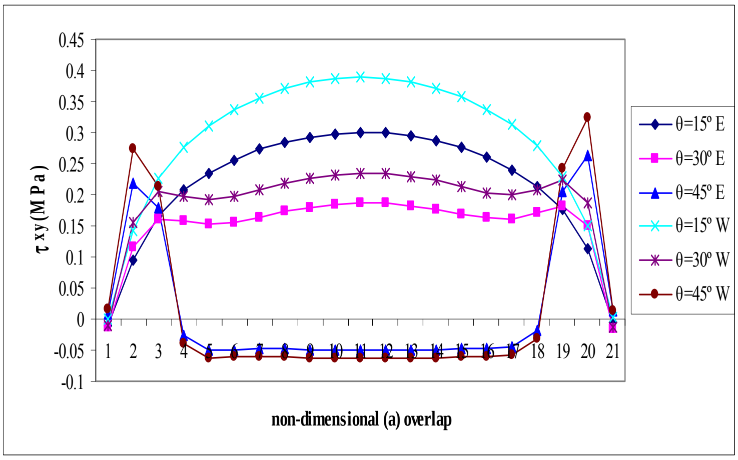
E: E type adhesive W: W type adhesive

Figure 5. σ_y normal stress distributions along the overlap length on the adhesive (adherend thickness=5 mm).

(Şekil 5. Yapıştırıcıda bindirme mesafesi boyunca σ_y gerilme dağılımı(adherend kalınlığı: 5 mm)).



E: E type adhesive W: W type adhesive
Figure 6. σ_{eqv} von-Mises equivalent stress distributions along the overlap length on the adhesive (adherend thickness=5 mm).
(Şekil 6. Yapıştırıcıda bindirme mesafesi boyunca σ_{eqv} von-Mises gerilme dağılımı(adherend kalınlığı: 5 mm)).



E: E type adhesive W: W type adhesive
Figure 7. τ_{xy} shear stress distributions along the overlap length on the adhesive with bonded Z type of steel(adherend thickness=5 mm).
(Şekil 7. Yapıştırıcıda bindirme mesafesi boyunca τ_{xy} kayma gerilme dağılımı(adherend kalınlığı: 5 mm)).

6. CONLUSIONS AND DISCUSSION (SONUÇLAR VE TARTIŞMA)

The maximum stress is described in this article. The stress components and the results of FEM calculations for the Z shaped joint are described in Figure 4, 5, 6 and 7 Figures shows the stress propagations at the positions of the interfaces adhesives. In this case, the stresses were examined up to increase and decrease. In this study, a stress means the stress an element. In addition, the interface stress shows at the interface of the adhesive. It is observed that stresses becomes maximum at the interfaces. From the results, it can be concluded that the stress of Z shaped joints becomes maximum at the position interfaces. Figure 4, 5, 6 and 7 shows the stress components σ_x figure 4 is the highest, while the normal stress component of which the direction is the same as the direction of tensile loadings is substantial when an tensile load is applied to the adhesive joints. The stress (σ_x) is minimum at the overlap a non-dimension in the boundary. However, as described before, the stress indicates at edges.

From the results, the maximum stresses in the special case shown in Figure 4, 5, 6 and 7 become highest at the overlap a length of the specimens. Figure 7 shows the distribution of shear stress τ_{xy} at the interfaces when the elapsed. It is found that the stress is substantial in this special case. Theoretically, the stress must be zero at the boundary. However, as has been described before, the stress occurs at point of the specimens along the edge. Thus, the stress is not zero at a overlap length boundary. In addition, it is also emphasized that the stress distribution in this special case (Figure 4) is different from that shown in Figure 5 (Z shaped). The effect of the overlap angle on the stress distribution at the interfaces is examined by FEM calculations (see Figure 4). It was found that the position where the highest value in centre of overlap length increased. It is found that the highest value of stresses σ_x increases along the overlap length of the adhesive as the value of θ increases.

Furthermore, the effect of overlap angle on stresses propagation is examined. In the FEM calculations, the overlap angles are changed and the calculations were done under the same conditions. In addition, it is also observed that stress distribution in the adhesive joint. Stress distribution repeated the results on the obtained stresses are different from the stress state of joints subjected to tensile loads. In this study, Z shaped adhesive butt joint subjected to stresses was calculated by FEM. In the FEM calculations employed is Ansys (version: 10.0).

In order to predict the ultimate strength given in Table 2, the adhesive was used. Therefore, the equivalent stress (σ_{eqv}), normal stresses and shear stresses were calculated using the von-Mises yield stress. A solution in FEM considering non-linear material behaviour is reached by dividing the total load in steps to track the equilibrium paths and iterating to a converged solution at each load increment. Hence, each load step was applied for all joint types. The loads in joints are non-linear. Consequently, exposes the adhesives both shear (τ_{xy}) and peel stress (σ_y). The peel stress (σ_y) at the free ends of the overlap is very important in this region. Consequently, when all of the specimens tested are examined, during tensile test, it can be stated that the damage in adhesives according to adherends and overlap length angle (θ) occurs (see Table 3.).

It is an important point to be considered that the increase in overlap length angle causes an increase in the damage load occurred, when Table 3. is examined. Also, the damage occurs within the adhesives and is partly cohesive and adhesive, but very close to the steel adhesive interface. Finally, it can be concluded that interfacial bond damage occurs in the joints.

Figure 7 indicates that more shear stress are transferred from the end to the centre of the overlap with increasing the adherend overlap angle (θ), due to the reduced the elastic deformations on the adhesives. Therefore, the effect of shear stresses on the failure and strength of the adhesively bonded joints increases. Similarly, it is evident that more equivalent stress is transferred from the end to the centre of the overlap with increasing the adherend overlap angle, as seen from Figure 6.

As observed for the normal and shear stresses along the bondline on adherends (Figures 4,5,6 and 7) σ_x, σ_y, and τ_{xy} shear stress distribution are higher for the joints with W type adhesive.

Similarly, when the von-Mises equivalent stresses are examined together it can clearly be stated undertakes elastic deformations on the adhesives. This situation provides the important increase in the performance of the joint with W type adhesive.

Consequently, a fairly good agreement is observed between the FEM results and experimental results.

6. CONCLUSIONS (SONUÇLAR)

Adhesives are used in many fastening applications as an alternative bonding method. Nevertheless, the designers have not enough trustworthy data yet due to changing adhesive strength. This study has deal with the effect of overlap angle on predicting of the damage load of adhesively bonded joints via a linear FEM. The results obtained are as follows;

- It is clear from figures between 4, 5 and 6 that σ_x, σ_y, and σ_{eqv} stresses were reduced at a overlap point. The σ_y stresses were increased for the same conditions.
- With the use of both adhesives, for θ=15°, θ=30° and θ=45°, the σ_y stresses were decreased at an overlap point. As for b overlap point, for b=13, when angle was increased from 15° to 30° the σ_y stresses were increased and when angle was increased from 30° to 45° the σ_y stresses were decreased.
- With the use of both adhesives, for θ=15°, θ=30° and θ=45°, the τ_{xy} stresses were decreased at an overlap points.
- For both adhesives, the σ_{eqv} stresses were decreased at a overlap point as can be in Figure 6. With the use of both adhesives, for b=13 mm and b=25 mm and at a overlap point, the σ_{eqv} stresses were increased and at "b" overlap point the σ_{eqv} stresses were decreased. With the use of both adhesives, for θ=15°, θ=30° and θ=45°, the σ_{eqv} stresses were increased at "a" overlap point. As for b overlap point when angle was increased from 15° to 30° the σ_{eqv} stresses were increased and when angle was increased from 30° to 45° the σ_{eqv} stresses were decreased.
- As can be seen in figures above σ_x, σ_y, and τ_{xy} stresses of the W type adhesive were higher than those of E type adhesive. It is because of that, the elasticity module of the W type adhesive is higher than those of E type adhesive.
- The σ_{eqv} stresses of W type were lower than those of E type. For both adhesives, geometrical exchange has considerable effects on maximum stresses, dependent upon the load.

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