THE CHOICE OF THEORY FOR ADHESIVE LAYER EFFORT

Jan Godzimirski, Marek Rośkowicz, Sławomir Tkaczuk

Military University of Technology, Department of Mechatronics
Kaliskiego Street 2, 00-908 Warszawa, Poland
tel.:+48 22 6839575, fax: +48 22 6839318
e-mail: jan.godzimirski@wat.edu.pl

Abstract

In the paper, the problems of adhesive layer effort was considered. The experimental tests of adhesives bonds were conducted in order to determine breaking loads. A technique a finite element mesh of adhesive layer subjected to shearing was proposed. Next the numerical calculations were conducted in order to determine distribution of stresses in adhesive layers at breaking loads. The components of stress tensor were used for the analytical calculations of reduced stresses according to seven hypotheses. It was found out that the maximal principal positive stresses hypothesis, the maximum principal strain hypothesis and the strain energy hypothesis make possible to estimate the adhesive layer effort with similar accuracy. Shape and the dimensions of the steel (S355J2G3) single lap specimen, the compression curves for Araldite AW 136H, the maximal principal stress values in the most loaded element of the adhesive layer depending on the elements' dimensions, modelling versions of the tree-layer adhesive layer's edges, distribution of the maximal principal stresses along edge section, comparison of the maximal principal stress distribution along adhesive layer's length, the stress-strain curve for Belzona 1111 adhesive and for Epidian57/Z1 adhesive, distribution of maximum principal stresses in adhesive layer of single lap bond subjected to shearing, are presented in the paper.

Keywords: *adhesive bonds, adhesive layers stresses, adhesive layer effort*

1. Introduction

Adhesive bonding is a typical method of joining applied in numerous branches of industry [1].There is complex state of stresses in adhesive layers of mostly adhesive joints: lap joints, strapped joints, trunnion-sleeve joints, strap-sleeve joints as well as adhesive bonds subjected to peeling [2-6]. Therefore, calculating of adhesive strength demand joints to determine an adhesive layer effort, according to some theory. Carried out an appropriate research prove that commonly used Huber's hypothesis (von Misses hypothesis) for metallic elements [7] is not useful for estimate adhesive layer effort. The hypothesis of the maximal principal stress seems to be right for the assessment of the adhesive layer effort [8, 9].

The experimental tests of adhesive bonds and numerical calculations were conducted in order to verify usefulness this hypothesis and to compare its accuracy with another hypotheses. The strength of various adhesive bonds was determined, the numerical tests were conducted for these bonds and the stresses were determined in the adhesives layers. The components of stress tensor in the most loaded adhesive layers elements were stated. The components of stress tensor were used for calculations of reduced stresses according to diverse hypotheses. It was assumed to accept this hypothesis as best for determination of the adhesive layer effort for which arithmetic average deviation of reduced stresses from their average value will be the lest (for all varied adhesive joints bonded the same adhesive).

2. Experimental test

Adhesive bonding with use two-epoxy adhesive - Belzona 1111 and Epidian57/Z1 - were tested. The adherends were made of steel S355J2G3 or aluminum alloy 2024T4 sheets. Glued surface of all specimens used in the test were prepared by sand blasting and washed with extraction naphtha and drying for 10 minutes at temperature 50ºC. The identical adhesive layer thickness was obtained by engrossed in adhesive two thin threads. Shown in Fig. 1 and Fig. 2 specimens were used for strength determination of various adhesive bonds.

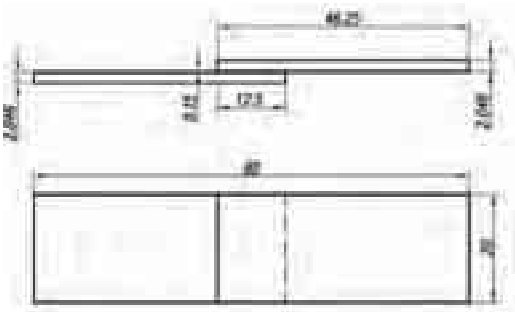


Fig. 1. Shape and the dimensions of the steel (S355J2G3) single lap specimen (dimensions in mm)

The specimens were bonded in devices under generated by pressure springs. Epidian57/Z1 adhesive was cured for 1 hour at temperature 60ºC and Belzona 1111 adhesive for 72 hours at ambient temperature. The specimens thickness was measured in order to determine adhesive layer thickness – about 0.15 mm. Made specimens were subjected to shearing in the ZD-10 testing machine or were twisted in shown in Fig. 3 device (only sing lap specimens). The values of twisting moment were measured by means of Bahco I20-DM-30 torque spanner. In the strength tests, six specimens were prepared for every measuring point. The test results were elaborated statically: Student-Fisher's method was applied to calculate the confidence interval for the level α= 0.95. The results of strength tests for the given specimens are shown in Tab. 1.

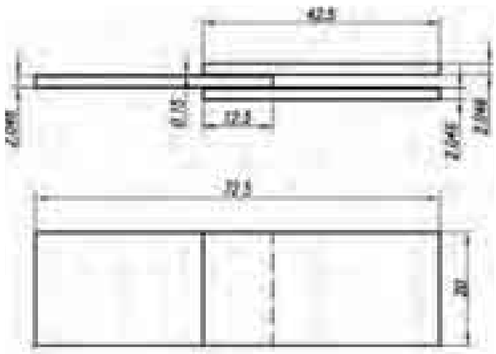


Fig. 2. Functional diagram and the dimension of the steel (S355J2G3) double lap specimen (dimensions in mm)

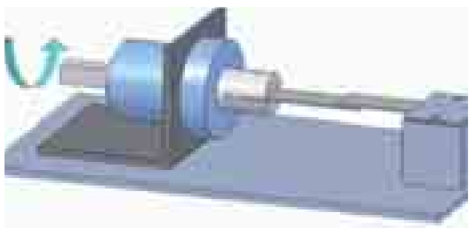


Fig. 3. Scheme of the instrument for single lap specimens twisting

3. Selection of numerical model of adhesive layer of joint subjected to shearing

Finite element modelling has been successfully used to model adhesive joints of various configurations and enables a detailed analysis of the stress and strain state. However, the nature and geometry of adhesive joints presents some unique challenges to finite element modelling. Theoretical stress singularities are present in many adhesive joints configuration, especially in joints with sharp edge adherends. Stress concentration tends to exist at the ends of overlaps and refined meshes are required in regions of high stress gradient [2].

Tab. 1. Breaking loads of tested specimens

Type of adhesive	Kind of specimens and load manner	Breaking loads
Belzona 1111	Steel, single lap, subjected to shearing	4.36 ± 0.45 kN
	Steel, single lap, twisted	8.14 ± 0.72 Nm
	Steel, double lap, subjected to shearing	13.85 ± 0.29 kN
	Aluminum alloy (2024T4), single lap, subjected to shearing	2.945 ± 0.173 kN
Epidian57/Z1	Steel, single lap, subjected to shearing	5.78 ± 0.42 kN
	Steel, single lap, twisted	9.08 ± 0.45 Nm
	Steel, double lap, subjected to shearing	13.70 ± 0.50 kN
	Aluminum alloy, single lap, subjected to shearing	3.22 ± 0.15 kN

The numerical calculations were made to verify whether the calculated stress value in adhesive layer is dependent on the density of mesh (the size of elements, which modelled the adhesive layer) and whether models of adhesive layer used allow for convergent solution. The numerical tests were conducted for adhesively bonded single lap joint. The specimens were modelled as made from the 2024T4 aluminium alloy bonded with Araldite AW136 adhesive and HY996 hardener. The value of failure load was adopted at 7000 N. This value was confirmed by experimental tests. The measurements of the tested joint are shown in Tab. 2.

Tab. 2. The geometric data of the simple single lap sample model

thickness of adhesive layer [mm]	adhered thickness [mm]	single lap sample length [mm]	width of sample [mm]	length of lap [mm]
0.1	2	100	25	12.5

The mechanical properties of the adhesive, declared in the calculations, were determined based on the compression curves ($\sigma = \sigma(\varepsilon)$) of the bulk adhesive which was experimentally set with cylindrical specimens (Fig. 4).

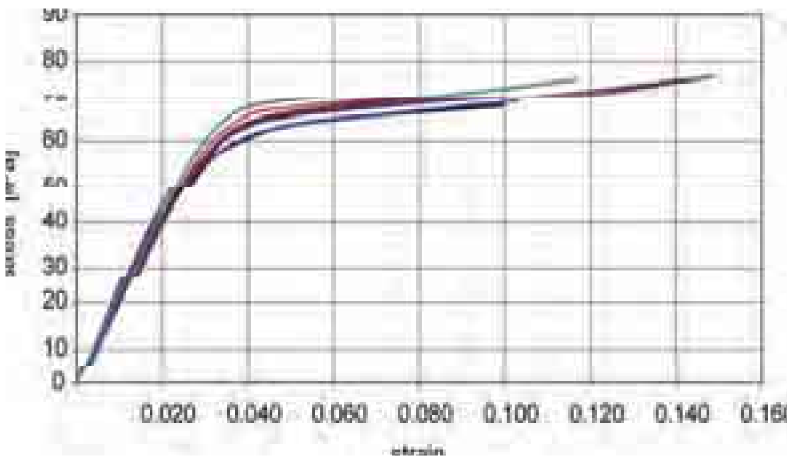


Fig. 4. The compression curves for Araldite AW 136H

Because of its regular shape and its manner of load, the simple single lap specimen was modelled by plate rectangular elements. The adhesive layer was modelled by 3 or 6 layers of the finite elements. The geometrical dimensions of the adhesive layer's elements were being changed in order to achieve larger density of mesh. Four three-layer mesh models and four six-layer ones were created. The geometrical dimensions of the elements for numerical tests models are shown in Tab. 3.

Tab. 3. The geometrical dimensions of the quadrangular (rectangular) elements used for the adhesive layer modelling

Number of adhesive's element layers	name of element	Geometrical dimensions of the adhesive's element	
		width „b" [mm]	height „h" [mm]
3 layers	05x033	0.05	0.033
	025x033	0.025	0.033
	0125x033	0.0125	0.033
	00625x033	0.00625	0.033
6 layers	05x0166	0.05	0.0166
	025x0166	0.025	0.0166
	0125x0166	0.0125	0.0166
	00625x0166	0.00625	0.0166

The numerical label of a width and a height of the elements were included in their markings. The height of the elements was 0.033 mm in the three-layer adhesive layer model and 0.0166 mm in the six-layer one. The maximal principal stress values in the most loaded element of the adhesive layer as function of depending on the elements' dimensions is presented in Fig. 5.

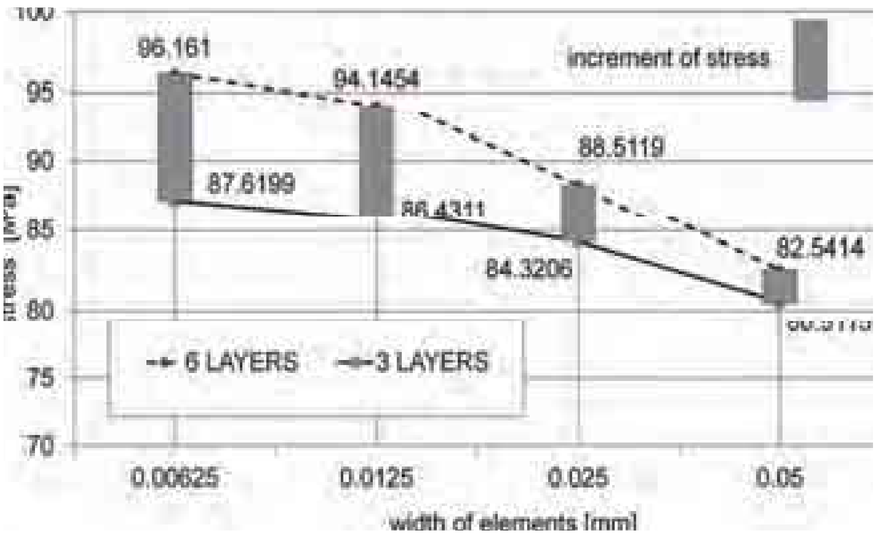
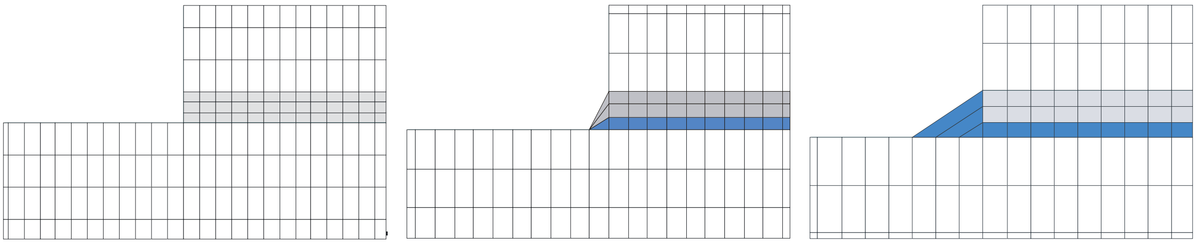


Fig. 5. The maximal principal stress values in the most loaded element of the adhesive layer depending on the elements' dimensions

The continuous increase of stresses along with the reduction of the elements' dimensions was found in both: the three-layer and the six-layer models of the adhesive. In search of such method of adhesive layer modelling which would allow to obtain convergent results of numerical calculation, essential property of the adhesive bonds was taken into account. Namely, the edge of the adhesive layer is not perpendicular to the adherent surface even after careful removal of the adhesive flash. Therefore, the comparative investigations were carried out. During them, a different geometry of the adhesive layer's edge was considered. This geometry results from the existence of a small (equal in dimension to the thickness of the adhesive layer) adhesive flash, which occurs on the boundary between the adhesive layer and adherents. The numerical calculations were made for two model versions of the adhesive layers with "flash" (Fig. 6).



a) b) c)

Fig. 6. Modelling versions of the tree-layer adhesive layer's edges with simple single lap bonds: a) traditional (without "flash"), b) 1st version, c) 2nd version (with "flash")

The exemplary results of the numerical calculations made for the adhesive layer modelled according to 1st version are shown in Fig. 7.

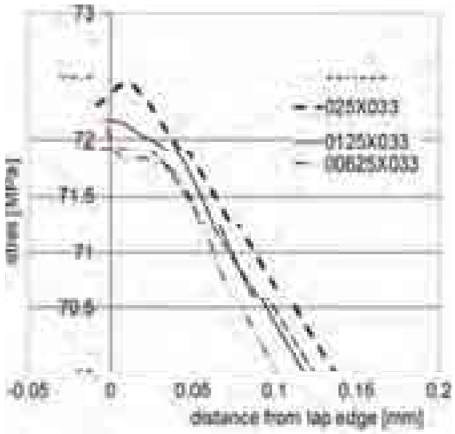


Fig. 7. Distribution of the maximal principal stresses along edge section of the lower adhesive layer for 1st version of the "flash" modelling (adhesive layer modelled by the 3 elements layers)

Based on the analysis of the numerical calculation results, it was concluded that the modelling of the adhesive layer "flash", especially according to the 1st version, caused the lack of considerable increase of stresses in the edge elements of the adhesive layers along with the thickening of mesh. Also, it was found that taking into account the adhesive "flash" in the adhesive layer model has significant effect on the stress distribution in the adhesive layer, causes uniformity of the stress distribution in all layers of the adhesive layer's elements (Fig. 8). This allows modeling the adhesive layer by one layer of elements.

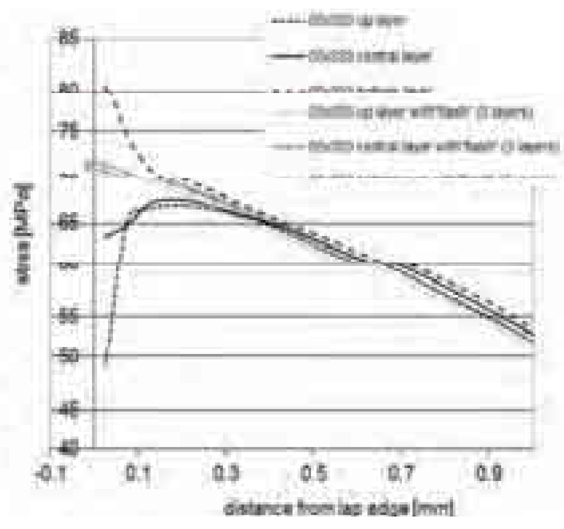


Fig. 8. Comparison of the maximal principal stress distribution along adhesive layer's length for three layers of

elements both with and without "flash"

The numerical investigations conducted lead to the following conclusions:

– small (equal in dimension to the thickness of the adhesive layer) "flashes" occur at the edges of overlaps in the joint subjected to shearing cause considerable reduction of the stress concentrations in adhesive layers as well as the uniformity of stresses along the thickness of the adhesive layer,
– the uniformity of stresses along the thickness of the adhesive layer allows modeling the adhesive layer with one element layer, thus simplifying in great extent the numerical modelling of the adhesive joint.

4. Numerical investigations

MSC Nastran for Windows program was used for the numerical investigation of experimentally tested adhesive joints. The actual thickness of specimen's sheets overlaps lengths, thickness of the adhesive layer and the spacing of the testing machine handles were used in numerical calculations. The computational numerical models were formulated of the basis of actual dimensions of the tested specimens. The adhesive layers were modelled by one layer of hexahedron finite elements [10]. The geometrical dimensions of adhesive layers elements were equal to 0.5x0.5x0.15 mm. The adherends were modelled by four layers of elements (Fig. 9).

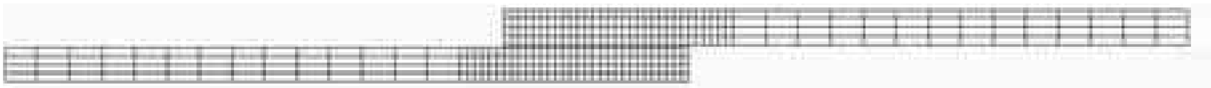


Fig. 9. Mesh of single lap specimen

The tested models of bonded joints were loaded with average forces according to experimental tests in respective groups. The stretched single lap specimens were extra loaded with a displacement resulting from the fact that they were clamped in rigid holder of testing machine.

Due to symmetry of double lap, specimens only half-bonded joints were modelled (Fig. 10) and loaded by half-breaking load.

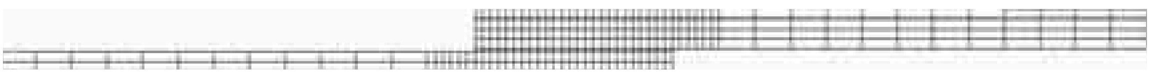


Fig. 10. Mesh of double lap specimen

The calculations were conducted for non-linear characteristics of adhesive. The $\sigma = \sigma(\varepsilon)$ curves of adhesive layers were adopted based on the set of compression curves (Fig. 11, 12). The linear property of adherends were assumed (steel – E = 210 GPa, ν= 0.3; aluminum alloy – E = 72 GPa, ν= 0.3).

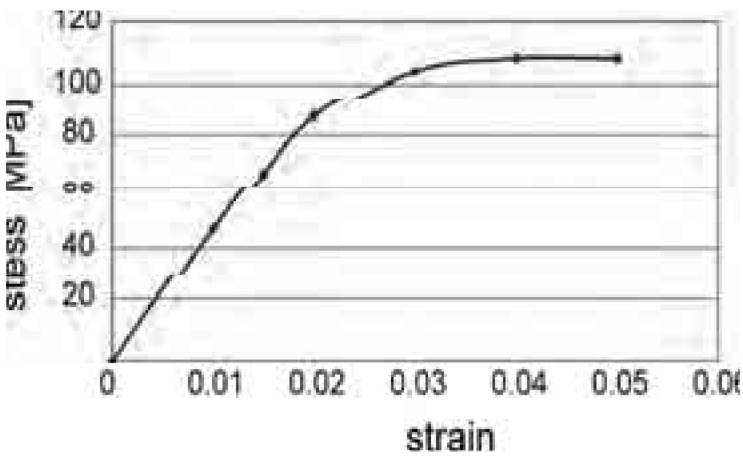


Fig. 11. The stress-strain curve for Belzona 1111 adhesive (compression curve) [11]

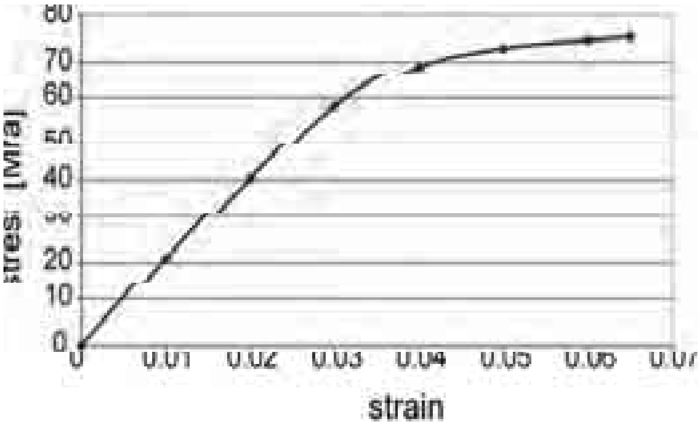


Fig. 12. The stress-strain curve for Epidian57/Z1 adhesive (compression curve) [11]

The contour plot of stresses in adhesive layers are presented in Fig. 13-15.
Calculated components of stress tensor and principal stresses are presented in Tab. 4 and 5.

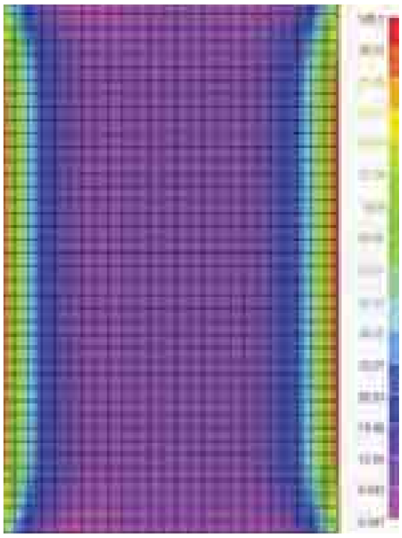


Fig. 13. Distribution of maximum principal stresses in adhesive layer of single lap bond subjected to shearing

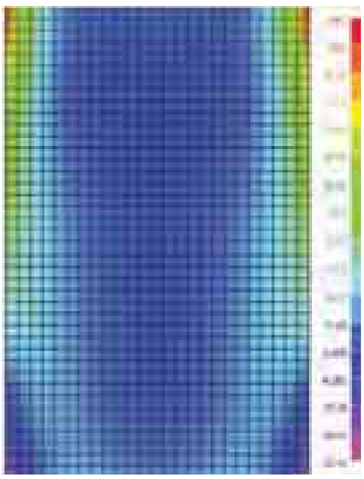


Fig. 14. Distribution of maximum principal stresses in adhesive layer of twisted single lap bond

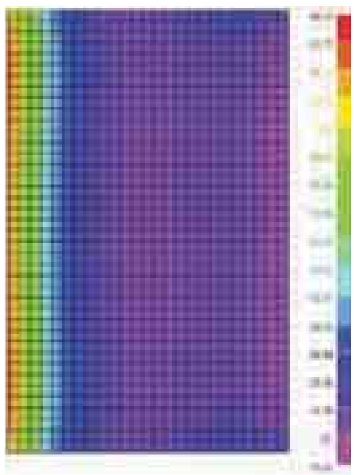


Fig. 15. Distribution of maximum principal stresses in adhesive layer of double lap bond subjected to shearing

Tab. 4. Components of stress tensor and principal stresses in the most loaded element for Belzona 1111 adhesive layer at breaking loads

Component of stress tensor	Single lap, steel, sheared	Single lap steel, twisted	Double lap, steel, sheared	Single lap, aluminum alloy, sheared
σ_x	38.29	44.80	28.47	36.98
σ_y	63.45	80.51	46.37	54.75
σ_z	35.19	42.73	25.07	31.22
τ_{xy}	43.03	26.68	50.24	41.88
τ_{yz}	0.28	14.74	9.32	0.21
τ_{zx}	0.01	0.36	0.14	0.01
σ_1	95.72	97.94	89.27	88.67
σ_2	6.05	27.17	-14.50	3.05
σ_3	35.19	42.93	25.13	31.21

Tab. 5. Components of stress tensor and principal stresses in the most loaded element of made of Epidian57/Z1 adhesive layer at breaking loads

Component of stress tensor	Single lap, steel, sheared	Single lap steel, twisted	Double lap, steel, sheared	Single lap, aluminum alloy, sheared
σ_x	30.66	33.71	17.25	27.32
σ_y	53.49	61.51	29.45	44.48
σ_z	29.24	32.90	15.92	24.74
τ_{xy}	34.56	16.31	37.20	31.28
τ_{yz}	0.20	13.83	0.22	0.17
τ_{zx}	0.5	0.18	1.61	0
σ_1	78.48	73.07	62.64	68.33
σ_2	5.68	21.97	-14.32	3.46
σ_3	29.25	33.06	15.92	24.74

5. Reduced stresses calculation

It may be assumed that if a hypothesis is corresponding for evaluation of adhesive layer effort the reduced stress calculated according this hypothesis should be near one another for specified adhesive and specified bonding theology regardless of joint sort (shape and loading means).

The calculations were made for seven hypothesis [12]:

1. Huber's (von Misses's):

$$\sigma_z = \sqrt{\sigma_x^2 + \sigma_y^2 + \sigma_z^2 - \sigma_x\sigma_y - \sigma_y\sigma_z - \sigma_z\sigma_x + 3\left(\tau_{xy}^2 + \tau_{yz}^2 + \tau_{zx}^2\right)}, \tag{1}$$

2. maximum shearing stress:

$$\sigma_z = \max\left(|\sigma_1 - \sigma_2|, |\sigma_2 - \sigma_3|, |\sigma_3 - \sigma_1|\right), \tag{2}$$

3. maximum positive principal stress:

$$\sigma_z = \max\left(|\sigma_1|, |\sigma_2|, |\sigma_3|\right), \tag{3}$$

4. maximum principal strain:

$$\sigma_z = \max\left[\sigma_1 - \nu(\sigma_2 + \sigma_3), \sigma_2 - \nu(\sigma_1 + \sigma_3), \sigma_3 - \nu(\sigma_1 + \sigma_2)\right], \tag{4}$$

5. strain energy:

$$\sigma_z = \sqrt{\sigma_1^2 + \sigma_2^2 + \sigma_3^2 - 2\nu(\sigma_1\sigma_2 + \sigma_2\sigma_3 + \sigma_3\sigma_1)}, \tag{5}$$

6. Drucker-Prager's:

$$\sigma_z = \frac{1+\alpha}{2\alpha}(\sigma_1+\sigma_2+\sigma_3) + \frac{\alpha-1}{2\alpha}\sqrt{\frac{1}{2}[(\sigma_1-\sigma_2)^2+(\sigma_2-\sigma_3)^2+(\sigma_3-\sigma_1)^2]}, \tag{6}$$

where: $\alpha = \frac{K_d}{R_{L,T}}$,

K_d – compression strength,
$R_{L,T}$ – tensile strength,

7. true strain:

$$\sigma_z = \sqrt{\sigma_1^2+\sigma_2^2+\sigma_3^2 - 2\nu\left(\frac{2-\nu}{1+2\nu^2}\right)(\sigma_1\sigma_2+\sigma_2\sigma_3+\sigma_3\sigma_1)}. \tag{7}$$

Calculated for analyzed hypothesizes and two tested adhesives mean percent deviation of maximal reduced stresses from their mean value are present in Tab. 6.

Tab. 6. Mean percent deviation of maximal reduced stresses from their mean value

Kind of hypothesis	Belzona 1111	Epidian 57/Z1
	mean per cent deviation	
Huber's (von Misses)	9.68%	10.99%
maximum shearing stress	10.59%	12.71%
maximum positive principal stres	4.23%	7.28%
maximum principal strain	3.00%	6.92%
strain energy	4.67%	5.89%
Drucker-Prager's (α= - 0,6)	17.58%	30.04%
Drucker-Prager's (α= - 1)	5.65%	11.40%
true strain	8.26%	10.12%

5. Conclusions

Adhesive layer of lap joints subjected to shearing may be modelled for numerical calculation by one layer of finite elements.

The experimental tests, numerical investigate and analytical calculations conducted led conclusion that from examined hypothesizes the maximal principal positive stresses hypothesis, the maximum principal strain hypothesis and the strain energy hypothesis make possible to estimate adhesive layer effort with similar accuracy.

Cohesive failure was characteristic for bonded joints with Belzona 1111 adhesive specimens. For this adhesive the mean percent deviation of maximal principal positive stresses from their mean value didn't exceed 3%.

The scientific thesis funded from Narodowe Centrum Badań i Rozwoju funds in years 2011-2012 as a research project.

Reference

[1] Cagle, Ch. V., *Kleje i klejenie*, WN-T, Warszawa 1977.
[2] Adams, R. D., *Structural Adhesive Joint in Enginery*. Elsevier Applied Science Publishers. London and New York 1984.
[3] Benson, N. K., *Influence of Stress Distribution on the Strength of Bonded Joints*. Adhesion. Fundamentals and Practice, Elsevier, Amsterdam 1970.
[4] Gardon, J. L., *Peel Adhesion. A Theoretical Analysis*, J. Appl. Polymer Sci. 7, 2, pp. 643-666, 1963.

[5] Kuczmaszewski, J., *Fundamentals of Metal-Metal Adhesive Joint Design*, Oddział PAN w Lublinie 2006.
[6] Godzimirski, J., Rośkowicz, M., Tkaczuk, S., *Wytrzymałość połączeń klejowych*, (Strength of Adhesive Bonds), WAT, Warszawa 2010.
[7] Jakubowicz, A., Orłoś, Z., *Wytrzymałość materiałów*, (Mechanics of Materials), WN-T, Warszawa 1978.
[8] Kubissa, J., *Problemy wyznaczania nośności klejonych połączeń metali. (Problems of Strength Determination of Adhesive Bonds of Metals),* Inżynieria i Budownictwo, 453-454, 8-9, pp. 169-171, 1982.
[9] Godzimirski, J., *Określanie naprężeń w spoinach klejowych metodą elementów skończonych. (Strength Determination in Adhesive Layers by Using Finite-element Method),* Biuletyn WAT, 399, 11, 77-81, 1985.
[10] Godzimirski, J., Tkaczuk, S., *Numerical Calculations of Adhesives Joint Subjected to Shearing,* Journal of Theoretical and Applied Mechanics, 45, 2, pp. 311-324, 2007.
[11] Godzimirski, J., Komorek, A., Rośkowicz, M., Tkaczuk, S., Smal, T., *Tworzywa adhezyjne – zastosowanie w naprawch sprzętu technicznego. (Adhesive Plastics – Application at Technical Equipment Repair,* WN-T, Warszawa 2010.
[12] Rusiński, E., Czmochowski, J., Smolnicki, T., *Zaawansowana metoda elementów skończonych w konstrukcjach nośnych (Advanced Finite Element Method for Supporting Structure),* Oficyna Wydawnicza Politechniki Wrocławskiej, Wrocław 2000.