PROBLEMATIC ASPECTS OF NUMERICAL COMPUTATION OF THE HYBRID JOINTS

Problematyka obliczeń numerycznych połączeń hybrydowych

Marek ROŚKOWICZ ORCID 0000-0003-0501-0622
Jarosław GĄSIOR ORCID 0000-0003-3205-0486
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A b s t r a c t: This paper presents the results of numerical calculations to compare the stress fields of hybrid (mechanical-adhesive) joints. ANSYS ver.19 was used as a computing tool with its solver based on finite element method (FEA – Finite Element Analysis). Based on the results obtained, a comparative analysis of the stresses in the adhesive joints for adhesives with different Young's modulus including different mounting schemes of mechanical fasteners and pressures caused by the assembly of the mechanical fasteners was performed. For modelling purposes of hybrid joints it is assumed that calculations will be made for single-lap joints where specimens with AW 2024T3 material parameters were joined. The adhesive layer was modelled by two layers of elements, adding i.a. pressure to the adherend caused by a preliminary tension of mechanical fasteners. In order to reduce the negative peeling effect at the ends of the adhesive layer, the case of mounting mechanical fasteners closer to the joint edge was also considered – at a distance of one diameter of the fitting instead of two. It has been found that using adhesive with lower value of Young's modulus can increase the load capacity of the joints due to possibility of absorbing higher loading by the mechanical fasteners. Moving the mechanical fasteners closer to the lap edge will result in a positive reduction of stress responsible for peeling of adhesive layer, as well as the pressure caused by the assembly of the bolts.
K e y w o r d s: hybrid joints, hybrid joint modelling, numerical computation of hybrid joints stress distribution in adhesive layer

S t r e s z c z e n i e: W pracy zaprezentowano rezultaty obliczeń numerycznych, których celem było porównanie pól naprężeń występujących w połączeniach hybrydowych (mechaniczno-klejowych). Jako narzędzie obliczeniowe wykorzystano aplikację Ansys v.19 opartą na metodzie elementów skończonych – MES (ang. FEA – Finite Element Analysis). Na podstawie otrzymanych wyników wykonano analizę porównawczą naprężeń występujących w spoinach połączeń adhezyjnych dla klejów o różnym module Younga, dla różnej geometrii montażu łączników oraz dla przypadku z uwzględnieniem nacisków wywołanych montażem łączników mechanicznych. W modelowaniu połączeń hybrydowych przyjęto, że obliczenia zostaną wykonane dla połączeń jednozakładkowych, w których łączone były elementy o parametrach materiału AW 2024T3. Spoinę połączenia klejowego modelowano jedną warstwą elementów, obciążając ją dodatkowo m.in. naciskami wywołanymi napięciem wstępnym łączników mechanicznych. W celu zmniejszenia negatywnego zjawiska oddzierania występującego na końcach zakładki połączenia adhezyjnego przeanalizowano również przypadek montażu łączników mechanicznych bliżej krawędzi połączenia – w odległości równej jednej średnicy łącznika zamiast dwóch. Stwierdzono, że zastosowanie kleju o mniejszym module może zwiększyć nośność połączeń ze względu na większe wytężenie łączników mechanicznych. Przesunięcie łączników mechanicznych bliżej krawędzi zakładki wpływa pozytywnie na obniżenie naprężeń wywołujących zjawisko oddzierania spoiny klejowej, podobnie jak naciski wywołane montażem śrub.
S ł o w a k l u c z o w e: połączenia hybrydowe, modelowanie połączeń hybrydowych, rozkład naprężeń w spoinie połączenia adhezyjnego

Introduction

Hybrid joints as a combination of adhesive and mechanical joints are an interesting alternative to joining components made of both metal and composite materials. Especially when connecting components made of polymer composite materials that are more susceptible to bearing stresses than metals, the use of adhesive in addition to mechanical joints allows increasing load capacity joints without damaging composite material as a result of exceeding the allowable pressures. And as aerospace manufacturers are increasingly interested in polymeric composites for airframe construction [19], the problem of effective composite assembly is now up to date.

In addition to the technical problems [4], the assembling of aviation constructions also has formal problems due to limited confidence in the use of adhesive joints in the main structures. There is also a technical barrier for the non-destructive testing of adhesive joints in used machinery effectively (e.g. aircraft) [7]. Therefore, the guidelines defined by the aviation authorities favour the use of hybrid joints instead of just adhesives [10].

Studies on hybrid joints show that they have improved strength characteristics compared to mechanical and adhesive joints in both static and durable area [3, 6, 9].

As the authors of [1, 4, 5, 16, 17] indicate, one of the reason for their increased fatigue life is the lower stress peaks that occur in the vicinity of the assembling holes for mechanical fasteners.

The load capacity and fatigue life of hybrid joints depend on the strength and geometric parameters of the adhesive layer (including the susceptibility of the adhesives to deformation), but also on the arrangement of the mechanical fasteners in the joint. As the experimental interaction between mechanical and adhesive interface parameters in a hybrid joints is burdensome, modern numerical solving tools can be used to optimize the load capacity of the hybrid joints.

The purpose of the calculations was to compare the stress distributions of the hybrid joint for adhesives with different Young's modulus and to evaluate the stress changes in the adhesive layer when the mechanical fasteners were moved closer to the edge of the lap. The calculation also takes into account the effect of the pre-loading caused by the assembly of the mechanical fasteners on the load capacity of hybrid joint. Also, the differences in stress values were assessed if a default tetragonal mesh and a more advanced hexagonal mesh are used to build the numerical model.

Numerical model of joint

The numerical calculations were performed for a single-lap hybrid joint model, made up of two elements with dimensions of 100 x 25 x 2 mm. The length of the lap was 50 mm and the thickness of the adhesive 0.1 mm. The adhesive was modelled with two layers of finite elements, while the adherends are modelled with five layers of elements. The diameter of the holes for mechanical fasteners is 3 mm. Two fasteners were modelled by placing them over a distance measured from the lap edge to the hole's axis and equal to one mandrel diameter of the fastener (1d) or two diameters (2d). The fastener model had two solutions. In the first solution, it was a single body fitting made up of a 3 mm diameter shank and 5.4 mm diameter heads and 2 mm high (Fig. 1a), while in the second solution it was made up of two parts: the head with the shank and the ring, which being able to move in relation to the shaft of the shank (Fig. 1b). This allows simulating continuous load occurring in the mechanical (bolt) joint after its assembly. Figure 2 shows an example of the geometric model of a hybrid joint created in the ANSYS Workbench Design Modeller with fasteners mounted at 2d distance from the edge of the lap (2d variant).

In addition, in order to assess the influence of the finite elements mesh created in modelling joint on the results obtained, the calculations were performed for the mesh tetra type (tetrahedral) and the hexa (hexahedral) type of mesh. Figure 3 shows the models covered with tetra mesh (a) and hexa mesh (b) prepared for numerical analysis. The tetra mesh is the default grid of finite elements offered to the user by ANSYS Meshing. The main advantage of this type of grid is the relatively small amount of work in the design of the model and the shorter

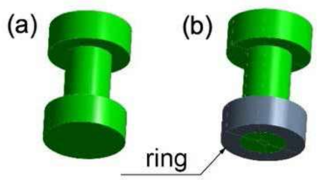

Fig. 1. Geometrical models of mechanical fasteners

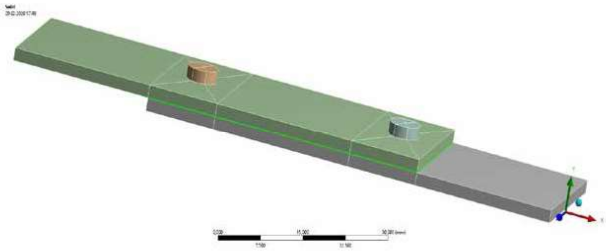

Fig. 2. Example of a geometric model of a hybrid joint in a 2d variant

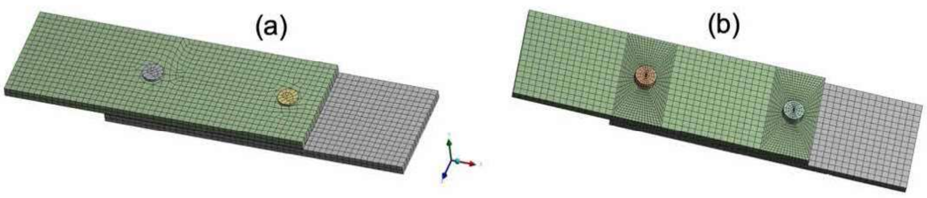

Fig. 3. Models built of Tetra mesh (a) and Hexa mesh (b)

computation time. Therefore, this solution is attractive to users and is often used by them [14].

The use of a second type of mesh – hexa – is more time consuming, but more regular mesh design has a positive effect on the quality of the results obtained. Table 1 summarizes the data for the 2d variant specimen to reflect the effect of mesh type on the number of elements and nodes present in a joint based on the same geometric model.

The following solutions are used to define the material properties of the hybrid joint elements. The joined adherends have been given the non-linear properties of aluminium alloy series 2024T3, adhesive presents the non-linear properties of Epidian 57/Z1 and Raychem S1125, mechanical fasteners have properties of steel. Table 2 shows the material parameters of the main elements used in the joint construction.

The initial load conditions of model are based on the experimental test conditions of the joints installed in the fatigue testing machine performed by authors own [17]. Hence, the following conditions are defined in the joint model (Figure 4):

a) all degrees of freedom (D_x, D_y, D_z, R_x, R_y, R_y, R_z = 0) was arranged at one end of the joint to reproduce the joint conditions in the machine's handle – Fig. 4 – A,
b) for the free end of the joint, the load was defined as a force acting on the cross-section surface of the joint and degrees of freedom were blocked according to the diagram: D_z=0, R_x, R_y, R_z = 0 – Fig. 4 – B,
c) The free end of the joint has been moved by a value of the thickness of the sample (2 mm) in a direction perpendicular to the direction of the force applied, thereby reproducing the axial conditions of mounting joint in the holder of the machinery during experimental tests – Fig. 4 – C,
d) The pressure generated by the mechanical fasteners after its assembly has been reproduced by the surface load of the head and the ring of the mechanical fastener – Fig. 4 – D ÷ G. The head and ring loads have opposite directions so the resultant load is reset. The load values are based on the measured torque appeared during assembly of the mechanical fasteners – approximately 2 Nm. When taking into account the torque value and the external diameter of the thread, it has been estimated that the nut tightening force is 3,3 kN.

Table 1. Compare the number of entities and the compute nodes of the connection model for Tetra and Hexa meshes

Sample model	Number of elements – Tetra grid	Number of cells – Hexa grid	Number of Nodes – Tetra grid	Number of Nodes – Hexa grid
2d	9971	28792	41963	140946

Table 2. The parameters used to define the material properties of each main element in the hybrid joint

Adherends AW 2024 T3	***σ[MPa]***	0	330,0	348,5	366,0	411,0	469,0	507,0	540,0	540,0	-	-
	ε	0	0,005	0,010	0,020	0,040	0,080	0,120	0,160	0,192	-	-
Adhesive Epidian 57/Z1	***σ [MPa]***	0	20,830	40,344	58,82	69,913	72,951	74,690	75,453	-	-	-
	ε	0	0,010	0,020	0,030	0,040	0,050	0,00	0,650	-	-	-
Adhesive Raychem 1125	***σ [MPa]***	0	0,50	0,75	1,00	1,20	1,35	1.,50	1,70	1,85	2,00	2,20
	ε	0	0,001	0,002	0,003	0,004	0,005	0,006	0,007	0,008	0,009	0,01
Fasteners Steel	***E [GPa]***	200										
	υ	0,3										

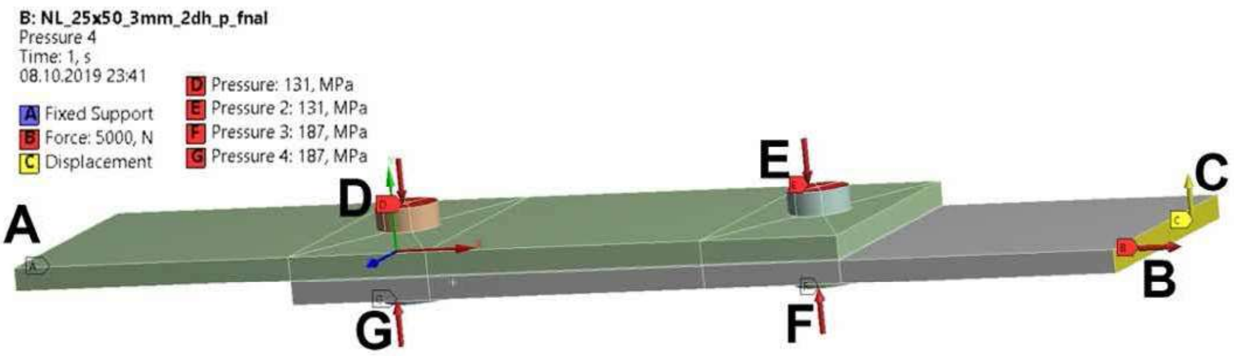

Fig. 4. Boundary conditions set in the sample calculation process

Table 3. Types of contact elements defined in the connection

Combined elements (CE)	Area of CE – adhesive layer	Heads of mechanical fasteners – CE	Tie rod spindles – internal hole surfaces	Mechanical coupler rings – mating surfaces	Mechanical coupler rings – mandrel surfaces
Contact type	Bonded	Frictional f=0,2	Frictional f=0,02	Frictional f=0,2	Frictional f=0,02

Since the hybrid joint is made up of several bodies, it was necessary to define the contact elements between the adjacent planes of the specimen elements. The types of contact elements used in the joint are shown in Table 3.

The calculation was performed in a non-linear range.

Calculation results

Experimental studies show that the destruction of a hybrid joints is a two-stage process, firstly the adhesive joint is damaged then the mechanical joint [5, 6, 9, 18, 19]. Therefore, the changes in the adhesive layer of the hybrid joint were analysed in detail. The adhesive layer was evaluated by analysing the stress distribution fields. The main purpose of the studies was to assess the changes in the adhesive joint after assembling the fasteners closer to the lap edge (so-called 1d variant instead of the 2d variant) and after using a more deformable adhesive material (Raychem S1125 instead of Epidian 57/Z1). In addition, changes in the adhesive stress fields as a function of the type of used finite element mesh were also tested during the calculation taking into account the pressures from the assembly of mechanical fasteners. The results of the calculations are presented below, taking into account the objectives of the studies adopted, including:

A) The type of mesh

The comparative calculations used a finite elements mesh of tetra to build the computing model, which is used by ANSYS as the default. Selecting and using this type of grid for calculations does not require a lot of work, but at the same time in complex geometry areas can lead to the formation of singular points. Because there are holes in the samples the usage of tetra mesh causes specific grid deformation, it was also decided to use hexa-type mesh. The mesh of hexa elements is much more demanding in terms of designer skills and time consuming, but the quality of the calculation site is higher. In the present case, the use of the hexa mesh also allowed to concentrate its elements in the vicinity of the holes for mechanical fasteners (Figure 5). Hence, there are more elements and nodes in this type of mesh which implicates increase quality of results.

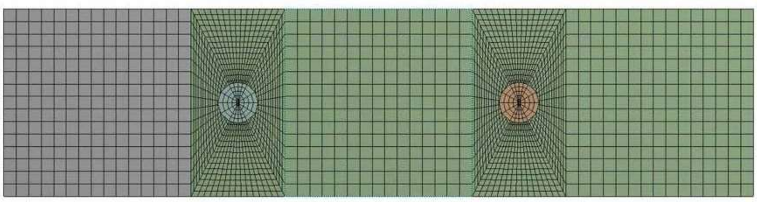

Fig. 5. Hexa mesh view with visible concentration of computing elements near holes

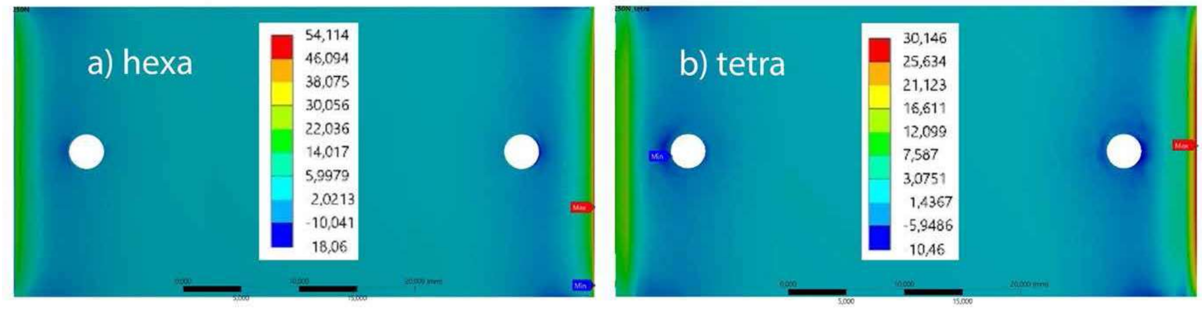

Fig. 6. Compare the distribution of normal stresses perpendicular to the adhesive layer

Using the models covered with two types of mesh, calculations were made according to the above conditions, assuming a force value equal to the joint load capacity (for hybrid joints with Epidian 57/Z1 adhesive, the mean load capacity value determined in the experimental tests was 8250 N) [17]. The calculations were performed by assigning the properties of Epidian 57/Z1 to the adhesive joint. Examples of normal stress distributions perpendicular to the adhesive layer (which are a measure of the magnitude of the peeling effect in the adhesive) are shown in Figure 6. For a model with a hexa mesh, the maximum stress values are almost twice as large and have a value close to the stress values of butt joints, bonding with Epidian 57/Z1 [7].

Figure 7 shows maps of the distribution of the maximum principal stresses in the adhesive joint for models built on a mesh of tetra and hexa. For a model with a hexa mesh, the maximum principal stresses, which are a measure of adhesive cohesion, are lower at the ends of the lap and are closer to the actual values of adhesive decohesion. Therefore, it was decided to use models based on the hexa mesh for further calculations.

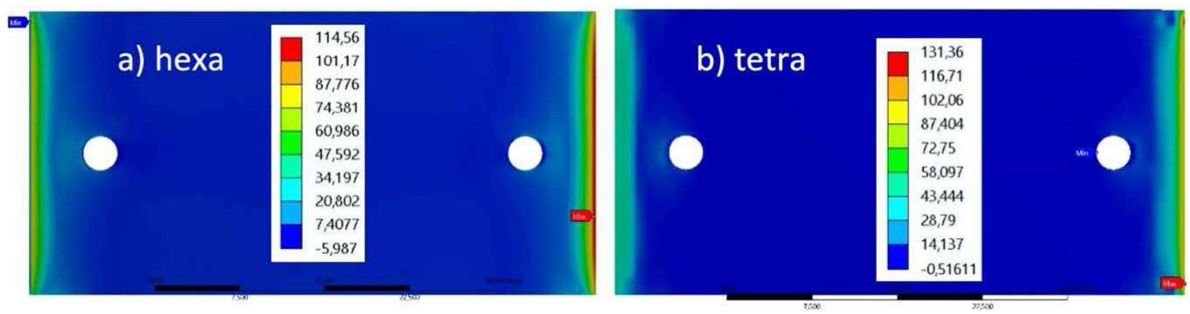

Fig. 7. Distribution of the maximum principal stresses present in the adhesive joint

B) Influence of assembly pressures

The numerical calculations for hybrid joints often exclude the effects of high torque mechanical fasteners on the adhesion layer [12, 15]. Usually the fasteners are modelled, without the pre-tension in the fasteners resulting from the way they are assembled is not taken into account (e.g. in the case of bolts, this is the torque which causes the bolt spindle to stretch and compress the fasteners on a samples surface equal to the surface area of the heads). A comparison calculation was performed of a model constructed with hexa mesh, reproducing the pressures that result from the nut and head on the combined elements – boundary condition d), Section: Numerical model of joint. The calculations were made for Epidian 57/Z1 adhesive, loading the model with force 8250N. The distributions of the normal, shear and maximum principal stresses are shown in Figures 8, 9 and 11.

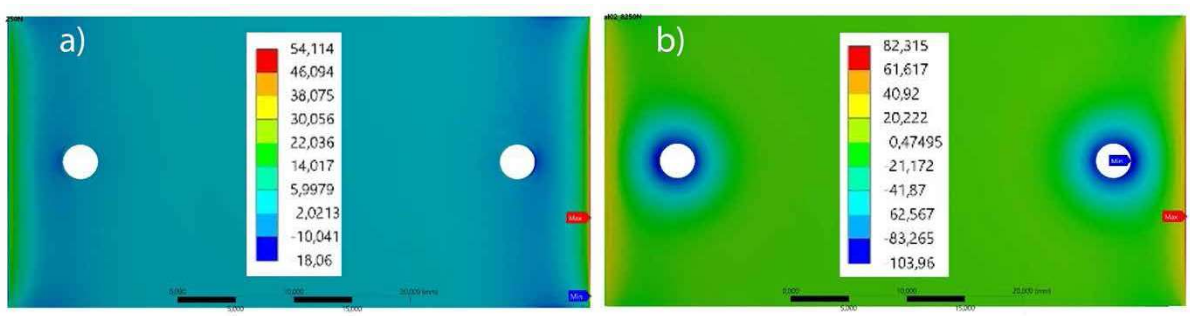

Fig. 8. Comparison of the distribution of normal stresses perpendicular to the adhesive joint
(a) without mounting pressures from the bolts, (b) taking into account the mounting pressures from the bolts

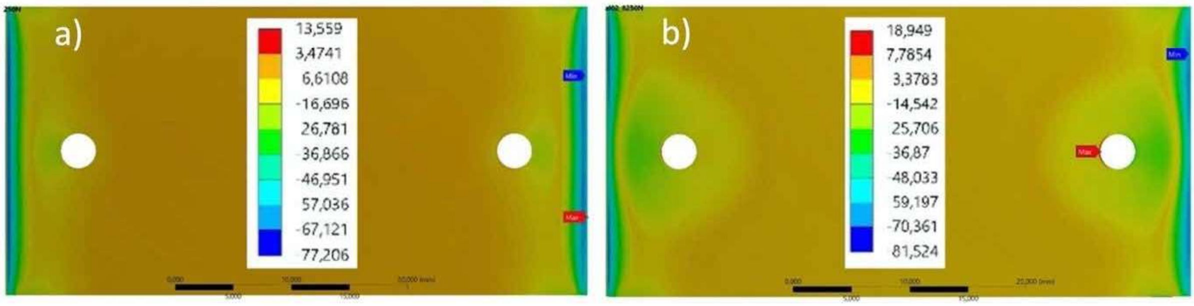

Fig. 9. Comparison of the shear stress distribution in the adhesive joint
(a) without the mounting pressure from the bolts, (b) taking into account the mounting pressure from the bolts

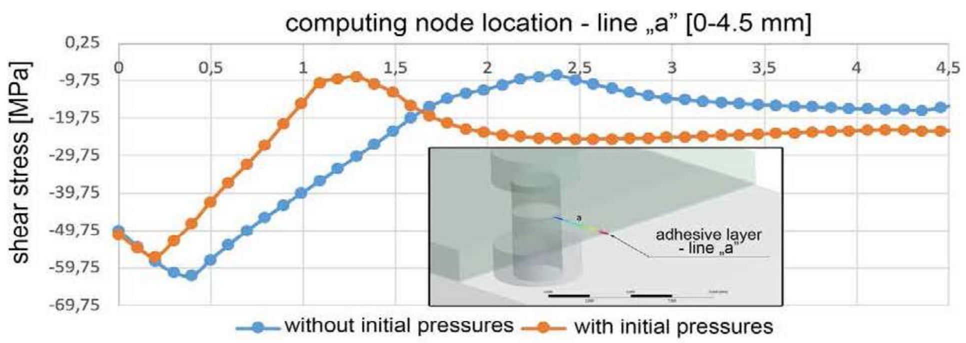

Fig. 10. Comparison of the distribution of the shear stresses on the line "a" adhesive joint
(a) without the initial (assembly) pressures, (b) taking into account the initial (assembly) pressures

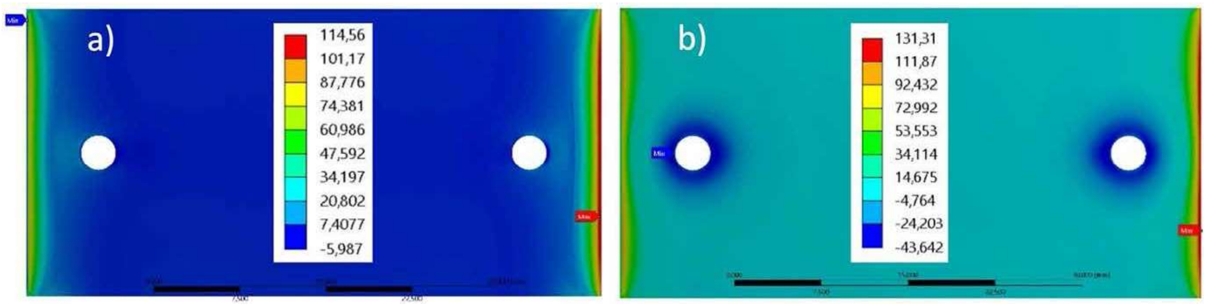

Fig.11. Comparison of the distribution of the maximum principal stresses in the adhesive joint
(a) without pressure from the fasteners, (b) taking into account the pressure from the fasteners

Analysis of the resulting distributions shows a significant effect of assembly pressures on changes not only in the distribution of normal stresses perpendicular to the adhesive in the vicinity of the holes for the fasteners – values increased (Figure 8), but also in the increase of stress values at the ends of the lap. In the area of shear stresses there is a clear zone of their variation that radiates from the holes – Figure 9. In addition, the shear stress values at the lap ends are lower as well as the difference between the stress at the lap ends and the hole area – Figure 10.

The distribution of the maximum principal stresses in the lap also changes due to assembly pressures – Fig. 11. The concentration zone at the ends of the lap is narrower and further narrowed at the middle end of the lap due to assembly pressures.

C) Assembling of mechanical fasteners – 1d and 2d option

Experimental studies [6, 12, 17] show that in the two-stage destruction of a hybrid joint, the adhesive joint is damaged firstly, then the mechanical joint is destroyed. As the results of the calculations presented above show, the positive effect of mechanical fasteners on the stress distribution in the adhesive joint is limited close to a circle with radius of approximately 1,5 diameter. According to the assembling conditions guidelines for mechanical joints, the minimum installation distance from the edge of

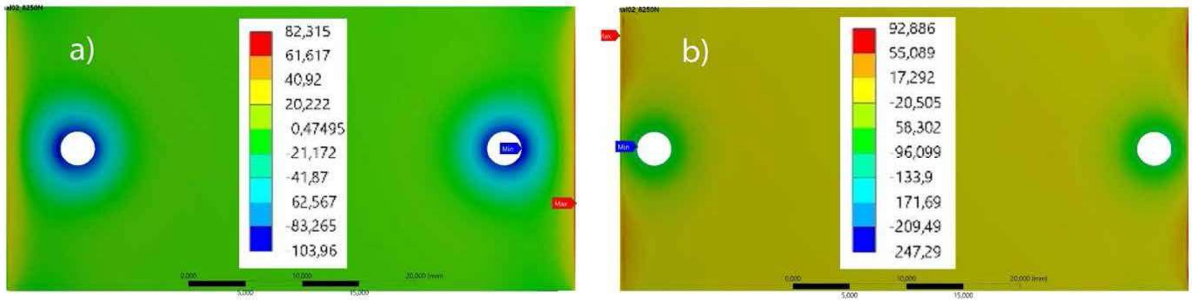

Fig. 12. Comparison of the distribution of normal stresses perpendicular to the adhesive joint
a) variant 2d, b) variant 1d

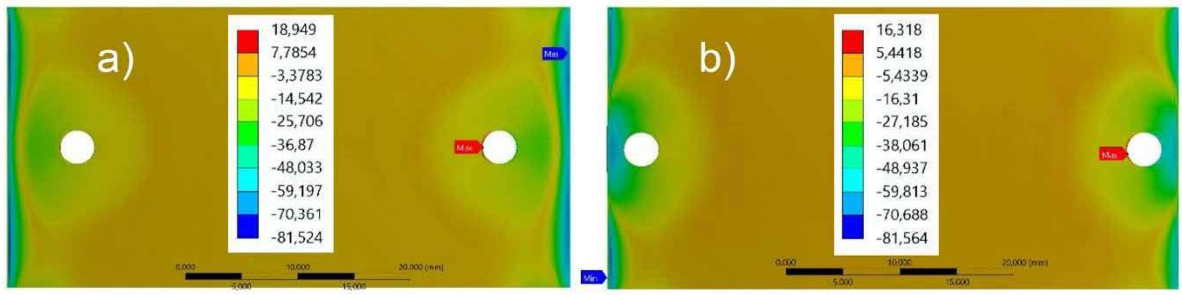

Fig. 13. Comparison of the shear stress distribution of the adhesive joint
a) variant2d, (b) variant 1d

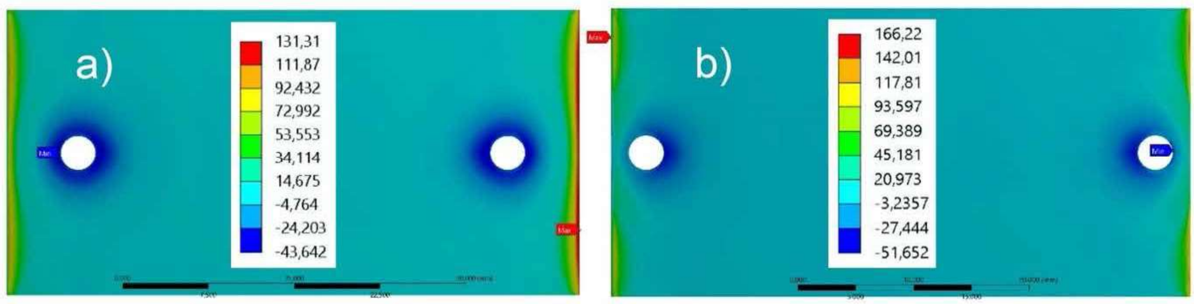

Fig. 14. Comparison of the maximum principal stress distribution of the adhesive joint
a) variant 2d, b) variant 1d

the lap should be 2d (2d variant) and this diagram was used in the calculations above. However, considering the field size where the fasteners have a positive effect on the adhesive joint, it was decided to perform calculations for a scheme where the mechanical fastener will be mounted closer to the edge of the lap at a distance of one diameter (1d variant) from the edge of the lap. It was assumed that the presence of fasteners closer to the edge will reduce the negative stress peaks at the ends of the lap and the mechanical properties of the hybrid joints, e.g. its load capacity, will be improved. The computations were made for model with Epidian 57/Z1 adhesive, loading the joint with a force of 8250 N. The distributions of the normal, shear and maximum principal stresses are shown in Figures 12, 13 and 14.

When analysing the results obtained, it was found that moving the fasteners closer to the lap edge separates the stress concentration zone into two parts and creates a much smaller area around the holes. Therefore, there is no continuous stress concentration zone at the ends of the lap as in the 2d variant, but two smaller zones symmetrically aligned with the hole's axis. The magnitude of the stress concentration field in variant 1d is then lower compared to the 2d variant. The own experimental studies [17] show a significant improvement in the load capacity of hybrid joints when the modified installation scheme 1d variant is used – Figure 15.

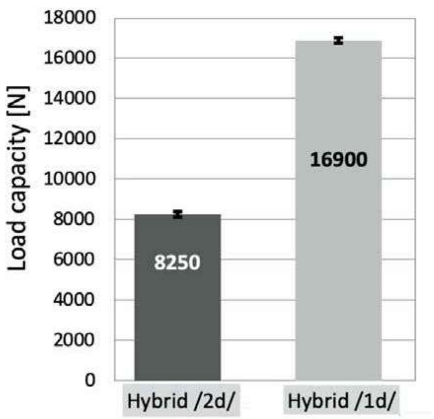

Fig. 15. Comparison of joint load capacities for the variant 2d and 1d [17]

D) Type of adhesive material

The proper use of the mechanical parameters of the hybrid joints is primarily based on the optimal distribution of the total load between the adhesive and mechanical joints, in order to different types of joints, i.e. the adhesive and mechanical joints as combination forming the hybrid joint, present properly load sharing to create homogenous stress distribution. Using a simplified hybrid joint model (Figure 16) as the authors of the publication [9, 15], it can be assumed that the use of a more deformable adhesive (i.a. lower value of Ea_d) should result in a higher absorbing and load sharing by the mechanical joint and as a consequence higher load capacity the hybrid joint

The research presented in [1, 2] shows that the adhesive parameters have a significant effect on the level of load absorption by mechanical fasteners. In order to achieve the condition that at least 10% of the load in the hybrid joint is transferred to mechanical fasteners, the authors recommend the use of deformable adhesive materials. In own calculations, the hybrid joint was modelled with using two adhesives, of which Epidian 57/Z1 was a highly rigid, while Raychem S1125 was a flexible one. The distribution of the shear and maximum principal stresses for the 5000 N load in variant 1d with initial pressures is shown in Figures 17 and 18.

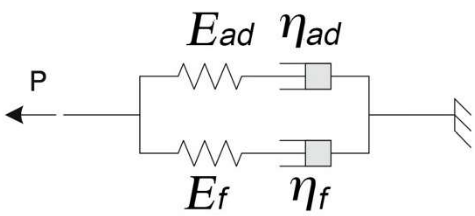

Fig.16. Simplified hybrid joint model (E_{ad} E_f - elastic modules of adhesive and mechanical fasteners, η_{ad}, η_f - *viscosity coefficients* of adhesive and mechanical fasteners [9]

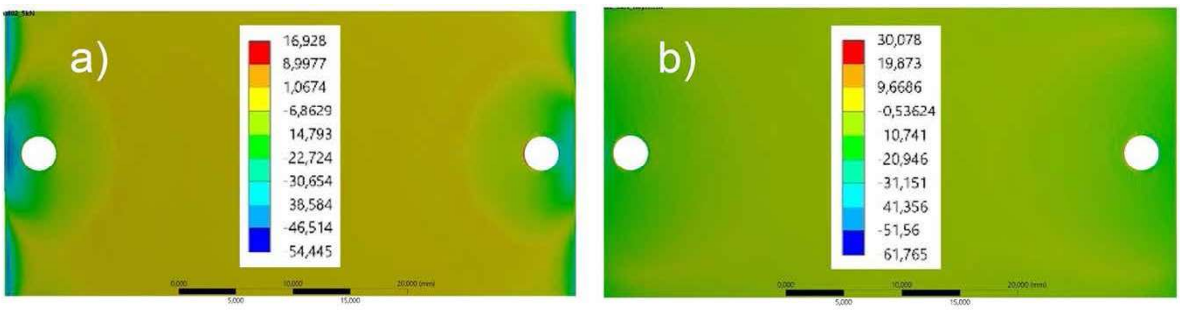

Fig. 17. Comparison of the shear stress distribution of the adhesive joint
a) Epidian 57/Z1, variant 1d, (b) Raychem S1125 variant 1d

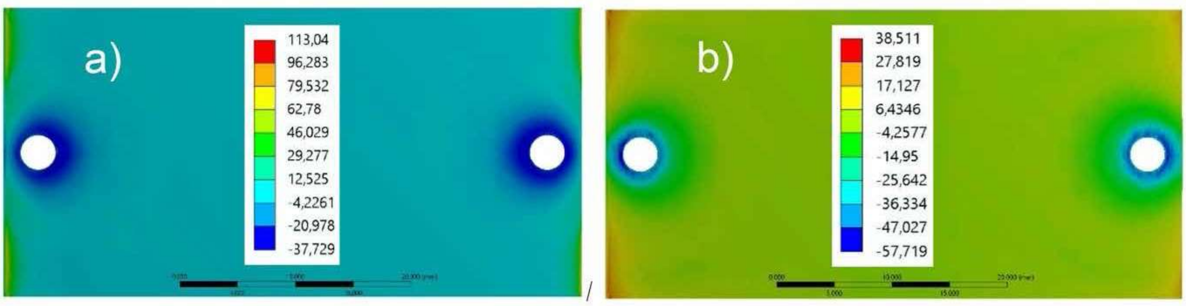

Fig. 18. Comparison of the maximum principal stress distribution of the adhesive joint
a) Epidian 57/Z1, variant 1d, b) Raychem S1125, variant 1d

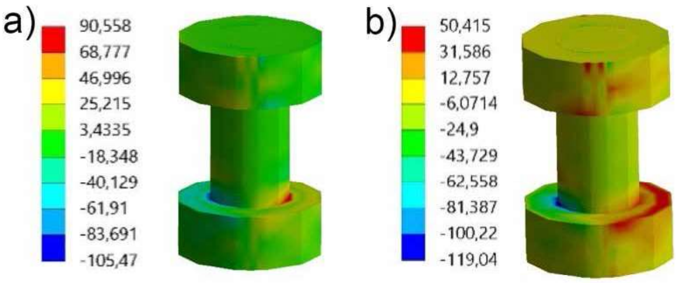

Fig. 19. Shear stress distribution in mechanical fasteners in the hybrid joint with (a) adhesive Epidian 57/Z1, (b) adhesive Raychem S1125

For the model with Raychem S1125 adhesive properties, the variation in stress distribution in the adhesive layer is lower. For hybrid joints, the smaller differences between the stress values at the ends and at the centre of the lap mean i.a. their increased of fatigue life [13]. In addition, the shear stress values in the mechanical fasteners in the model with Raychem S1125 adhesive are almost twice higher, which means that the fasteners are more effort – Figure 19.

Conclusions

On the basis of the computations made, it was found that:

- On the one hand, the density of the mesh elements and their regular shape around the holes (hexagonal mesh) result in a significant increase of the calculation task and increase of the time taken to prepare the model and perform the calculations by about 5 times compared to the model with the tetragonal default grid. On the other hand, the results obtained, e.g. the stress distribution in the hybrid joint are more similar to the stress values estimated from experimental studies.
- The pre-load of the hybrid joint including adhesive is the result of the assembly process of the mechanical fasteners. And since the calculated stress status of the adhesive joint is the basis for estimating the fatigue life of the hybrid joint, the assembly pressures should be taken into account in defining the boundary conditions of this type of joint (if a model with a limited number of mechanical fasteners is considered, the so-called higher level model in the form of a sample or a smaller part of the structure).
- The results of the computations obtained for the two assembly variants of the fasteners (1d and 2d) do not clearly indicate that there is a significant reduction in stress at the lap ends, but when the fasteners are moved closer to the lap edge, it separates the stress accumulation zone into two parts reducing the negative stress peaks. Additionally, the areas with significantly less uneven stress distribution are created around the holes, what positively affects the load capacity and durability of hybrid joints.
- The results of the computations made also indicate that the use of adhesive material more susceptible to deformation is a positive factor in solving the stress compensation problem on lap length. For this material, the process of "passing" the load from the adhesive to the mechanical fasteners is more efficient and results in a higher absorbing as well as load sharing by the mechanical joint. The consequence is higher load capacity the hybrid joint.

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Marek Rośkowicz, DSc.
The Institute of Aviation Technology, Military University of Technology in Warsaw
ul. gen. Sylwestra Kaliskiego 2, 00-908 Warsaw, Poland
email: marek.roskowicz@wat.edu.pl

Jarosław Gąsior, M.Sc., Eng.
The Military Centre for Standardization, Quality and Codification
ul. Nowowiejska 28A, 00-909 Warsaw, Poland
email: j.gasior@ron.mil.pl