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A R C H I V E S O F M E T A L L U R G Y A N D M A T E R I A L S

Volume 60 2015 Issue 4

DOI: 10.1515/amm-2015-0442

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THE INFLUENCE OF GEOMETRICAL PARAMETERS IN SOCKET - PIN CONNECTIONS ON THE VALUE OF OPENING FORCE

WPŁYW PARAMETRÓW GEOMETRYCZNYCH W POŁĄCZENIACH TYPU GNIAZDO - TRZPIEŃ NA WARTOŚĆ SIŁY OTWIERAJĄCEJ

The paper presents an analysis of the influence of a number of technological aspects of both the socket and the pin on the value of the force required for joint disconnection. A number of numerical simulations were made in Abaqus program to examine effects of such parameters as: presence of an interference fit, use of spherical latches, application of different rigidity of the pin by making cuts with variable width and length, use of different angles of inclination of the working part of the connection. Models of different simple joints presented in this work, can also operate in large structures forming panels of aircraft structures. For this purpose one of the analyzed geometry of the connection was applied to create a 3-D panel model of the structural element in CAD – SolidWorks program. All analysed models with different geometries were subjected to simulation of opening process. The corresponding critical forces were estimated for the beginning of the failure process. The detailed discussion of all model parameters was included to specify their influence on the whole disconnection of joints. It should be noted that aerospace structures work under complex loading states and further numerical studies are required to extend the presented results.

Keywords: socket-pin joints, deformation processes, FEA (finite element analysis)

W pracy przedstawiono analizę wpływu kilku zagadnień technologii wykonania zarówno gniazda jak i trzpienia na wartość siły potrzebnej do rozłączenia połączenia. Przeprowadzono szereg symulacji numerycznych w programie Abaqus, badając wpływ takich parametrów jak: występowanie wcisku, użycie sferycznych zatrzasków, zastosowanie różnych sztywności w trzpieniu poprzez wykonanie rozcięć o zmiennej szerokości i długości,

zastosowanie różnych kątów pochylenia części roboczej gniazda. Prezentowane w pracy modele pojedynczych połączeń mogą także funkcjonować w większych strukturach tworzących panele konstrukcji lotniczych. W tym celu, dla jednego z analizowanych rozwiązań geometrii wykonano model CAD panelu w programie SolidWorks, który następnie poddano symulacji otwierania. Należy jednak zwrócić uwagę, że konstrukcje lotnicze pracują w złożonych stanach naprężeń i tym samym konieczne są dalsze badania numeryczne prezentowanych rozwiązań.

1. Introduction

Permanent innovations in aircrafts technologies require application of:

- multiphase engineering materials with specially designed composition to satisfy exploitation requirements of critical parts of airplanes,
- continuous searching for novel and more effective joining technologies to decrease weight of airplanes and in parallel to increase their security.

The novel materials are different types of composites obtained as mixtures of various phases and further subjected to specific technological process in the manufacturing, e.g. [1, 2]. The materials engineering allows for designing of almost arbitrary internal structure of the composites, particularly important for industrial demands. A very important example are functionally graded materials, possessing gradation of the material properties, (e.g. [3-11]).

Composites built up as a sequence of layers are other types of gradation of material properties. Introduction of layers between material components (e.g. [12-19]) or joined adherends (e.g. [20-25]) and as TBC covering of turbine blades (e.g.[26-32]), etc. are other examples of improvements in composites or joining technologies.

Nano-composites (e.g. [33]) or adhesives with different nano-particles or carbon nano-tubes [34, 35] are good applications of particle reinforced composites in modern aerospace joints technology.

Generally, novel solutions in joining technologies of structural parts for the aerospace can be performed by:

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1. application of new composite materials for adherends and using single joining technique as: riveting, spot welding, clinching or adhesive bonding (e.g. [25, 36])
2. combination of two simple joining techniques, e.g.: riveted-bonding (e.g. [37, 38]), spot welded-bonding (e.g. [39, 40], clinched-bonding (e.g. [41-49]), etc.
3. creation of the new joint geometry in the form of the new geometry lock, e.g. [50]

Modelling of connections using mechanical fasteners (rivets, screws), or hybrid adhesive joints is a complex problem. This results from necessity of a dense mesh to represent mechanical fasteners, a large number of contact pairs causing that the problem becomes highly nonlinear, necessity of using damage and fracture criteria (e.g. [51-58]) for both mechanical fasteners and adhesive.

The aim of this work is to propose other possibility of a new joining technology by creation of the novel geometry lock (point 3).

In the joining of aircraft structures many techniques are used (e.g. [25, 36-38, 14-58]). The oldest of them is riveting [51, 52], which despite its advantages requires a violation of construction material continuity in the form of holes for rivets. Each hole provides the stress concentration, and can be a source of cracks initiation. Another method used for over 50 years is adhesive bonding [55]. An advantage of bonding is the uniform stress distribution in the assembled parts as well as tightness ensuring. To adhesive bonding disadvantages we can include i.e. joints aging and the need for careful surface preparation. To combine the advantages and disadvantages of above mentioned connections, hybrid joints are used such as: riveted - adhesive, spot weld - adhesive (e.g. [39,40, 56, 58]), clinched - adhesive [41 -50, 52] and others.

A new alternative to the currently used connections like mechanical, adhesive and hybrid may be socket - pin connection type. These are purely mechanical connections strongly dependent on the shape. They have different names such as: Interlock, Snaplock, Snapfit Gridlock. The principle of operation of these connections lies in the fact, that between the socket and the pin occurs a suitable interference fit or formed clip, which carries the load. The advantages of the structure in which this type of connection is used is very fast assembly, consisting in pressing joined parts together. Application of the socket - pin connection results in elimination of human error occurrence and reduction of production costs, because an individual connection is made by CNC machine tools. At present, these connections are at the stage of research and development as evidenced by numerous articles and reported patents in recent years. Solutions, both for aluminum and composite materials, are under consideration.

The paper proposed 5 new geometries of the socket-pin connections.

2. The state of art in designing of the socket-pin connections

The socket – pin connections consist of only two elements which are integral parts of larger structures. This type of connection is referred to as purely mechanical. By making subdivision, there are three kinds of this connections:

1. with simple geometry, i.e. without a formed clip,
2. with complicated geometry forming the specified shape of the clips
3. hybrid joints including above concepts a) and b).

In the first type connection sockets and pins have relatively simple shape and the load is carried mainly by friction. The level of force value depends on appropriately designed geometry of the both parts. In case of the second type joints, the connections consist of the sockets having notch and adequately shaped pin. The load is carried by shear stresses curried by the clip. In the third type connection there is both a clip and relatively large interference fit between the socket and pin. This type of hybrid joints has the highest strength in comparison to previous solutions due to the existence of clip and lack of clearance.

All of the above selected connections are widely used to attach baseboard, cell phone housing, fixing car upholstery etc. This type of application relates mainly to joining plastic elements, however there is no wide application with respect to metals and composites. The reason for such a large spread of the socket – pin connections is the fact, that the plastic materials have a relatively low Young's modulus (e.g. ABS 1,4 – 3,1 GPa) and a tensile strength in the range of several tens of MPa (e.g. ABS 40 MPa). Closing of socket – pin connections is done on the basis of their mutual pressure without the use of tools. This causes a deflection of beam with a clip or deformation of socket arms. Due to the high Young's modulus for metal (eg. Aluminum 69 – 70 GPa) the deflection of the beam or the socket arms is very difficult and at low displacement plastic strains can occur. This problem can be solved with using relatively thin-walled beams or socket arms in range of 0,5 – 1mm as well as clips with similar width. However, such small dimensions lead to technological problems, especially when there is a lot of socket – pin connections in the whole structure. Therefore both numerical controlled machines and tools with special shapes should be applied.

The Grid – Lock connections type, e.g. [50], is ideal for many commercial applications such as space and arms industry. They can replace the "honeycomb" structure e.g. flight stabilizer used in the F-15E fighter which is subjected to the extreme and complex loads. However, this type of connections uses an adhesive layer which may be damaged and the identification of this phenomenon is difficult.

A major problem that can occur in the application of the socket – pin connections is the amount of clearance affecting the closing force. If the clearance value is lower than the threshold designed, it can result in damage of connection or connecting elements. Therefore, the authors in paper [54] proposed a solution that in the misfit range of ±0,3 mm allows to keep a constant value of the closing force.

Brock and Wright [55] created a tool for the design of the snap – fit connections. It allows, with known dimensions, to determine such parameters as: opening and closing force and plastic stresses. Rusli et. all. [56] pointed out that the snap – fit connections should be compressed, but with a small force of several tens of Newton's.

Another problem, that occurs when using the socket – pin joints, is disconnection. The authors in papers [57, 58] propose the use of a shape memory material for joints manufacturing. In this case the pin model consists of two parts: the outer – for load carrying and the internal – used to trigger opening. The inner element is made of shape memory material in the form of a simple beam, which - when heated to 90°C - returns to its original shape to form an obtuse angle. Then the damage of the external part occurs and the connection is open.

In paper [59] the problem of the connection disassembling has also been considered. In this case shape memory material was also used, but for the entire model of the pin with the clip. To release the connection one had to use a temperature in the range 130°C – 150°C at which the deflection of beam occurred.

The conclusion resulting from the above literature analysis is that the problem of the socket–pin connections designing and further behavior under loading in the structural elements is very complex and requires further intensive study, particularly in relation to metals and composites. This paper will contribute to solution of this problem.

3. Innovative clips geometries and finite element models creation

The paper presents 5 types of socket – pin mechanical models (Fig. 1):

1. using interference fit between the socket and pin (Fig. 1a),
2. with locking clips (Fig. 1b),
3. with a different width of a cut in the pin (Fig. 1c),
4. with a different length of cut in the pin (Fig. 1d)
5. with different angles of inclination of joined parts (Fig. 1e)

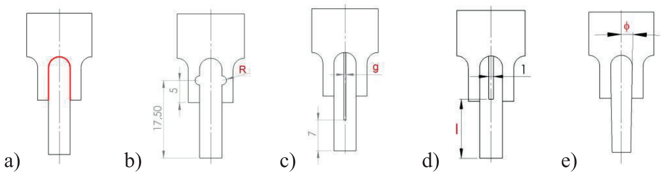

Fig. 1. Proposed types of the socket-pin connections

The dimensions of cross-section for the main model are shown in Fig. 2. The thickness of all connections was assumed equal to 5mm.

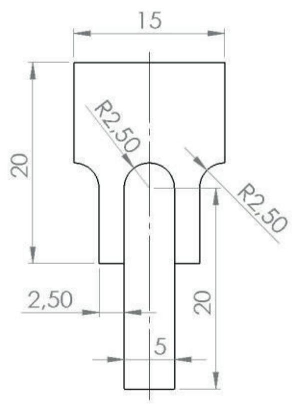

Fig. 2. The dimensions of the cross section of the connection

It should be noted, that the connection assembly was made by sliding (Fig. 3). In this way we do not introduce to the joint any initial state of deformation. The connection is free of stress before beginning of the loading process. This assumption is important, because in the paper we will discuss the opening process to assess the level of failure force.

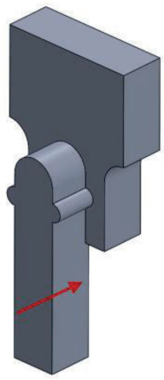

Fig. 3. The method of assembly by sliding.

After assembling the analysed models were subjected to the uniaxial quasi-static deformation process, Fig. 4. In order to perform this process, two reference points RP1 and RP2 were made, which were connected by "coupling" constraints with fixing and loading surfaces. The point RP1 a displacement vector was imposed, whereas the point RP2 was fixed, i.e. all degrees of freedom were removed.

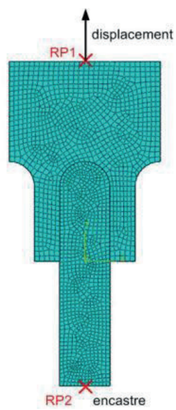

Fig. 4. The mesh of finite elements

The numerical finite element (FE) analysis was done with application of the commercial code ABAQUS. Figure 4 presents also the FE mesh of both parts of the joint. For creation of the FE numerical models we applied standard the FE type C3D8R. The number of elements in different models was included in the interval from 18 000 to 21 000. The important step in the creation of the FE models was inclusion of:

- the interference fit along contacting surfaces of the sockets and pins. In the FE contact modeling we created along the surfaces contact pairs in the "clearance" overlap and further we selected "initial clearance" as "uniform value across the slave surface". It should be noted that the simulations taking into account in the FE model the interference fit, consisted of two steps: interference creation and opening.

- the dynamic friction along contacting surfaces due to their movement. In all considered models a friction coefficient was equal to 0.3.

In the numerical simulations of the opening process of the socket – pin connections we used the aluminium alloy (i.e. an elastic – perfectly plastic material) as the base material for manufacturing the joints. The following material parameters were taken into account: E = 70 GPa, ν= 0,3 and σ_y = 300 MPa.

4. Results of numerical simulations

All the results presented below for simple models are in the form of force – displacement graphs. The values were recorded for RP1 point which was loaded by displacement. In all considered models 3 or 5 various versions with different values of the characteristic parameter were analysed numerically.

4.1. Models with interference fit between the socket and pin (Fig. 1a)

A total number of 5 versions of the model with interference fit were considered for the following values of the interference fit: from 0.01 mm to 0,05 mm with the interval 0.01 mm, Table 1. During the whole opening process there was no plastic strains in any analysed case.

The maximum value of Mises stress occurred for model 1_5 with value 235.8 MPa, much below σ_y . Table 1 shows the values of maximum forces necessary to open connections, whereas Fig. 5 presents the force-displacement diagrams.

TABLE 1
Maximum forces for the model with interference fit

Model number	1_1	1_2	1_3	1_4	1_5
	Value of interference fit [mm]				
	0.01	0.02	0.03	0.04	0.05
Maxiumum force [N]	82.46	173.65	258.2	336.57	432.91

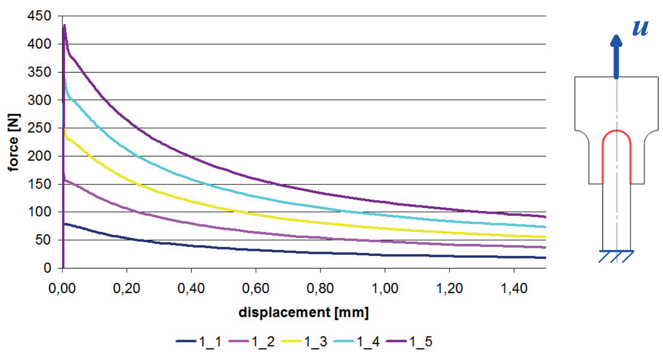

Fig. 5. Models with interference fit

4.2. Models with the locking clip (Fig. 1b)

Contrary to the previous connection, the model (Fig. 1b) has much more complicated geometry. This type of models have clips on the lateral surfaces of the pin in the form of half – cylinders with a radius of 1mm, 0.5mm and 0.25mm. The socket of the connection has grooves with corresponding shape of the clip. We assumed in calculation that there is no interference fit between joined parts. Below, in Table 2, the maxiumum forces necessary to open connection were listed. Figure 6 presents the corresponding force-displacements diagram for 1 mm thickness of the joint.

TABLE 2
Maximum forces

model number	2_1	2_2	2_3
	clip radius 2R [mm]		
	1	0,5	0,25
Maximum force [N]	1194,16	1896,09	1175,38

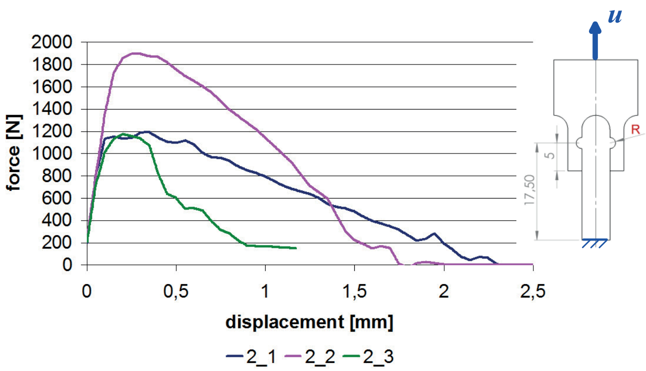

Fig. 6. Models with clips

In all of the considered versions of the models one can observe places with the level of the Huber – von Mises stress exceeding the yield stress σ_y. The disconnection of the joints requires providing a relatively high energy. Therefore, the force to opening is very high, i.e. of the level from 1000 to 2000 N per mm of the connection thickness. The maximum of the force corresponds to the second geometry of the joint. It should be noted that this type connections are relatively difficult to implement and may require tools with special shapes.

4.3. Models with a different width of the cut in the pin (Fig. 1c)

In these models we designed cuts in the pin with different widths. Additionally, we introduced the interference fit with value of 0.02 mm to strengthen of the connection. The cut width “g” in the various models was equal to: 1mm, 0.5mm and 0.25 mm. By increasing the cut width the pin rigidity is reduced which leads to decreasing of the opening force. Change of the cut width from 1mm to 0,25 mm results in the force increase by 20%. Furthermore, we can obtain 29-fold decrease in opening force using a model with cut as compared to the model 1_2.

TABLE 3
Maxiumum opening forces for the model presented in Fig. 1c

Model number	3_1	3_2	3_3
	Cut width „g" [mm]		
	1	0,5	0,25
Maximum force [N]	5,09	5,76	6,1

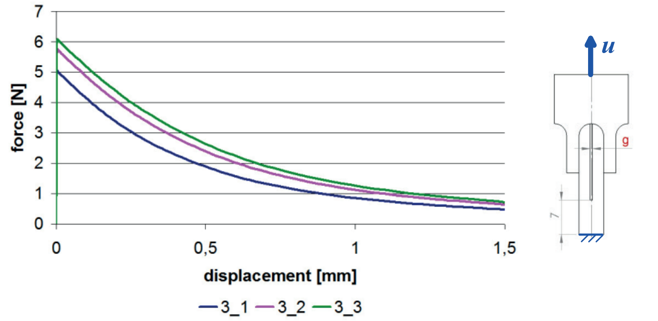

Fig. 7. Models with cut width „g"

4.4. Models with different cut length of the pin (Fig. 1d)

This model analyses the influence of the cut length of the pin on the overall behavior of the connection. Similarly to the previous model we assume the interference fit with the value of 0,02 mm. The cut width in each case was 1mm. Calculations were carried out for 5 versions of the model with cut length as in Table 4. During the whole deformation there was no visible plastic strains and the maximum Huber – von Mises stresses were in range from 14.5 MPa (model 4_1) to 105 MPa (model 4_5). By using different cut lengths the opening force can be very effectively controlled. Change of the cut length "d" with 13 mm causes about 32-fold increase in the force required to disconnection.

TABLE 4
Maximum opening forces for the model presented in Fig. 1d

Model number	4_1	4_2	4_3	4_4	4_5
	Cut lenght „d" [mm]				
	15,5	9,5	7,5	4,5	2,5
Maximum force [N]	5,09	16,29	32,46	76,22	160,88

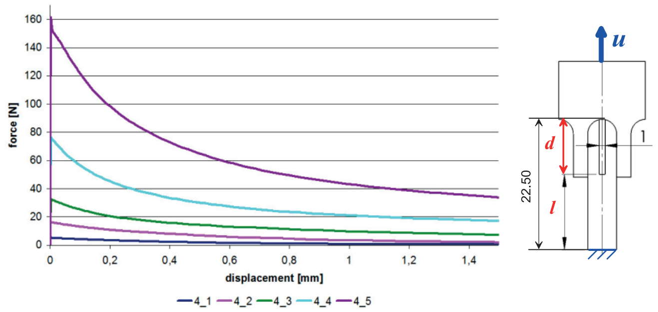

Fig. 8. Models with different cut length „d"

4.5. Models with different angles of inclination of joined parts (Fig. 1e)

In these simulations interference fit has not been applied, nominal dimensions for parts in contact were the same for the pin and the socket. The yield stress σ_y was not reached in any of the considered cases. Manufacturing of this connection type requires the use of high precision and maintaining close tolerances which may result in the fact that implementation of the process becomes expensive. By slightly change of the inclination angle, i.e. from 10 to 0,250, one can obtain more than a 4-fold decrease of the opening force. Table 5 shows the maximum force values for the considered models.

TABLE 5
Maximum opening forces for the model presented in Fig. 1e

Model number	5_1	5_2	5_3	5_4
	Inclination angle ϕ			
	10	0,750	0,50	0,250
Maximum force [N]	219,85	164,49	103,59	50,44

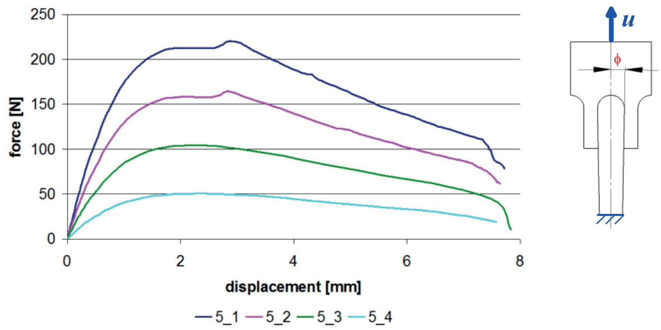

Fig. 9. Models with different inclination angle of the joined parts

4.6. 3-dimensional model of the fuselage panel

The application of the simple socket – pin model with different angles of inclination of joined parts (Fig. 1e, section 4.4) was implemented for construction of the 3-D fuselage panel. The panel consists of two halves, i.e. top and bottom parts (Fig. 10). For joining both parts in this structure we applied the model 5_2 with the inclination angle equal to 0.750. For such value of the inclination angle we did not observe any local stresses level above the yielding point, i.e. σ_y.

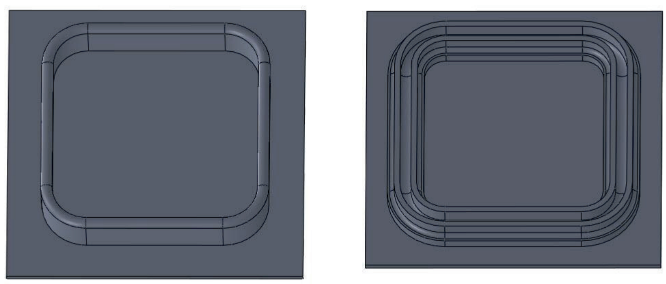

Fig. 10. Parts of assembly for fuselage panel

Figure 11 shows a vertical cross section through the fuselage panel with its basic dimensions. The material model remained the same as for the simple connection 5_2. For building of the numerical model we applied C3D8R finite elements – totally 59 000.

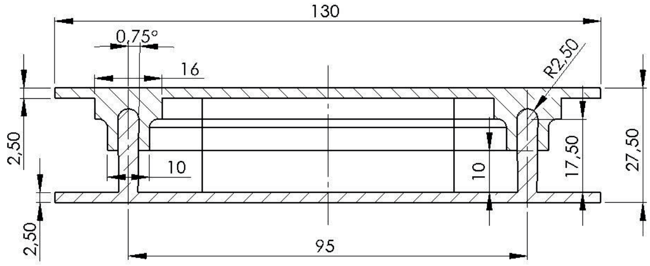

Fig. 11. Cross section of closed panel

Figure 12 presents the boundary conditions which allows for analysis of the opening process of the panel.

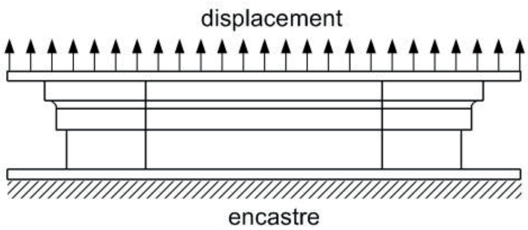

Fig. 12. Boundary conditions

It should be noted that in the simple clip model 5_2, the pin surfaces which did not participate in the contact could easily deform. However, we observed quite different situation in the case of fuselage panel, where the circuit formed by mechanical connection was closed. This results in appearance of additional stresses that lead to creation of the plastic deformations in the socket (Fig. 13).

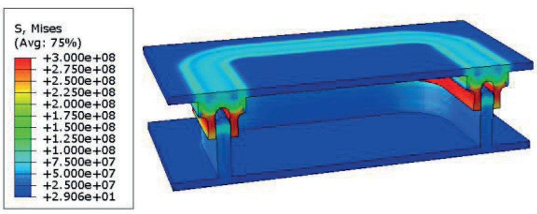

Fig. 13. Huber – von Mises stress distributions during connection opening

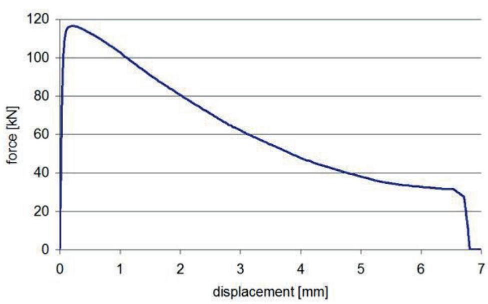

Fig. 14. Force – displacement graph for fuselage panel

Figure 14 shows the force – displacement graph for the whole opening process of the structural panel. The maximum force was equal to 116 kN.

5. Conclusions

This work presents a review of the literature of the currently considered mechanical connections with application of the different joining techniques. The particular aim of the paper was discussion of the socket – pin mechanical connection models. Their biggest advantage is quick assembly without use of any additional tools. Assembly of the joints were done by pressing the two halves of the structure. The obtained numerical results presented in this work for simple models, proved the possibility of the opening force adjustment in a very large range. The lowest value of the carrying force, in range of 1 N/mm, was obtained for models with the longitudinal cut in which variation of the width dimension negligible affects the results. By reducing the cut length, we can increase the opening force to about 32 N/mm. The significant influence has also the interference fit between the socket and pin. In this case, using the interference – fit with value equal to 0.05 mm the increase of the opening force up to 86 N/mm was obtained. The important parameter is also the angle of inclination of the contacting surfaces of the upper and bottom part of the lock, i.e. for the inclination angle of 10 the carrying force was equal to 43 N/mm. However, the highest values were obtained for connections with clips. In this case, the maximum force for simple model was 234 N/mm (model 2_2).

Considering the results for the fuselage panel, it should be noted that there is no direct translation of results from the simple model with the same cross – section of both pin and socket. The reason for this is a completely closed circuit formed by the mechanical connection in the fuselage panel which works much more efficiently in comparison to the simple joint. Presented mechanical models, having a lot of technological parameters, require further research especially in complex load states. Moreover, different degradation and cracking models (e.g. [60 – 73]) should be incorporated in the analysis to get more detailed mechanical response of the presented joints. The work will be continued.

Acknowledgement:

Financial support of the National Centre for Research and Development (Poland) - Project "Block Structures - Mechanical joining innovations to replace conventional fasteners in aerostructures", contract No INNOLOT/I/5/NCBR/2013 is gratefully acknowledged.

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Received: 15 September 2014.