Stress Analysis in Adhesive Cylindrical Assemblies made by Hybride Materials

OVIDIU NEMES[1]*, FREDERIC LACHAUD[2], ABDELKADER MOJTABI[3], MARIAN BORZAN[4], STEFAN GRIGORAS [5]

[1] Technical University of Cluj Napoca, Faculty of Materials Science and Engineering, Environmental Engineering Department , 103-105 Muncii Blvd., 400641, Cluj-Napoca, Romania

[2] Ecole Nationale Superieure d'Ingenieurs de Constructions Aeronautiques, Departement Genie Mecanique, 1 place Émile Blouin, 31056 Toulouse cedex 5, France

[3] IMFT, Universite Paul Sabatier Toulouse III, UFR-MIG, Departement de Mecanique, 118 route de Narbonne, 31062, Toulouse Cedex 4, France

[4] Technical University of Cluj Napoca, Faculty of Machine Building, TCM Department, 103-105 Muncii Blvd., 400641, Cluj-Napoca, Romania

[1] Gh. Asachi Technical University, TCM Department, 71A Mangeron Blvd. 700050, Iasi, Romania

This work presents a theoretical model of calculation of cylindrical assemblies joined with adhesive, based on an energy method. After the determination of the cinematically acceptable field of stresses, according to the applied load, a variational calculus on the expression of elastic potential energy leads to the complete expression of the stress field in the whole assembly. A first parametric analysis (geometrical and physical parameters) is carried out on an assembly of tubes and makes it possible to deduce the optimal length and the thickness of the adhesive. The model is validated by comparison with a finite elements model. For the assembly, the total force-displacement behavior is well defined. Thus the analytical model makes it possible to determine the rigidity of the assembly and to obtain a simple formulation very rapidly which gives the total behavior of the assembly.

Keywords: adhesive joints; glass fibers; modeling; finite element analysis, stress transfer

Adhesive bonding is a powerful technique of assembly which makes it increasingly possible to replace or to supplement the traditional methods (welding, riveting or bolting). However, the optimization of this type of assembly requires determination of the stresses in the adhesive and the substrates. Stresses are strongly influenced by the geometrical and physical parameters of the assembly [13, 14].

The mechanical performance of an adhesive-bonded joint is related to the distribution of the stresses in the adhesive film. Consequently it is essential to know this distribution which, because of its complexity, makes prediction of fracture difficult.

The first studies were developed on plane assemblies with simple covering loaded in traction. The work of Volkersen [1] developed in 1944 led to a false evaluation of the maximum level of stress because the effects of inflection of the supports were not taken into account.

Compared to the number of recent scientific publications concerning plane joints, with simple or double lap [2-4], there are only a few theoretical publications studying the mechanical behaviour of adhesive-bonded joints, with symmetry of revolution, subjected to traction [5, 6].

The first theoretical studies concerning cylindrical adhesive-bonded joints were carried out by Lubkin and Reissner [7] and Volkersen [8]. They assumed that the supports did not become deformed.

The influence of the pressures exerted inside and outside the tubes was considered [9].

It was investigated the action of moments and axisymmetric forces, thus introducing inflection into the tubes [10]. In this work the orthoradial stresses were not taken into account.

The most recent work, concerning the type of assembly considered here, was performed [6]. A first stress field based on the equilibrium equations and the conditions of continuity of the stresses at the interfaces using an equation of compatibility was built. Then it was calculated the potential energy associated with this field and, using the theorem of minimal complementary energy, a system of differential equations whose solutions determined the optimal stress field was obtained.

However, although the field of the stresses obtained verifies some of the compatibility equations well, it does not verify them all.

The first numerical example treated in the present paper shows the zones of stresses which appear at the ends of the joints. The later ones also provide an outline of the consequence of varying some geometrical and physical parameters on the distribution and intensity of shear stresses in the adhesive.

All the techniques developed, based on the resolution of the associated differential equations, encounter a difficulty which has not yet been overcome: how to take into account the boundary conditions at the ends of the joint. Since damage declares itself in these zones, it is important to model the edge effects.

A technique based on the minimization of the potential energy was used [11, 12]. The first stage consists in building a statically acceptable stress field, i.e. verifying the boundary conditions and the equilibrium equations. The second stage consists in calculating the potential energy generated by such a stress field. In the third stage, the second theorem of potential energy is used to minimize this in order to determine the stress distributions.

The analytical approaches which relate to cylindrical bonded interfaces are applicable for estimated calculations, therefore for preparatory projects, but there is no doubt that, under complex loading, digital simulation is a stage that cannot be circumvented if we want to optimize an adhesive assembly.

* *email address: ovidiu.nemes@sim.utcluj.ro.*

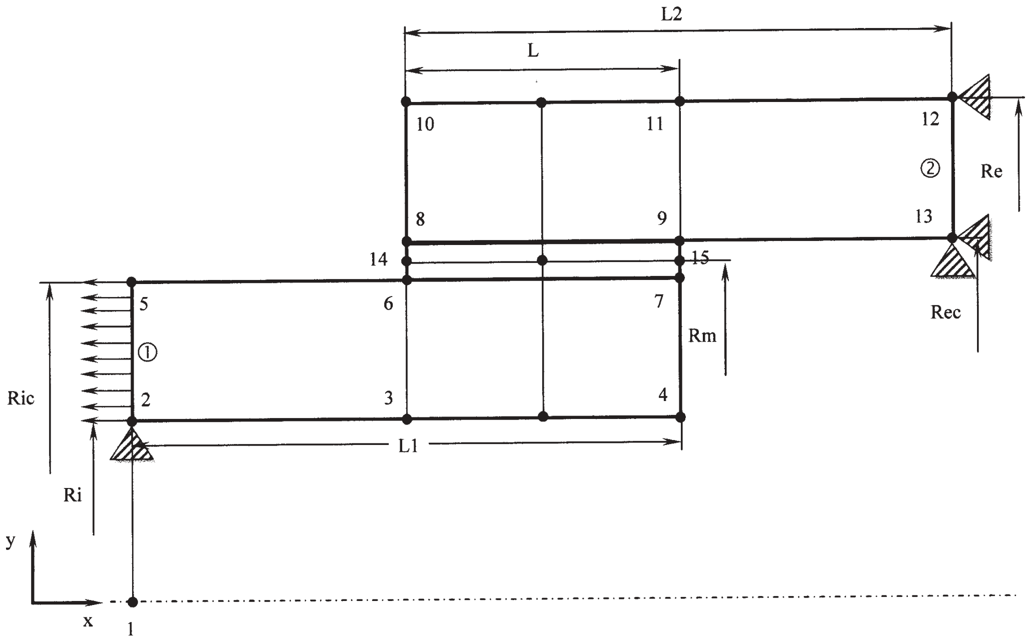


Fig.1. CAD diagram of a cylindrical adhesive bonded joint

Numerical modeling by finite elements

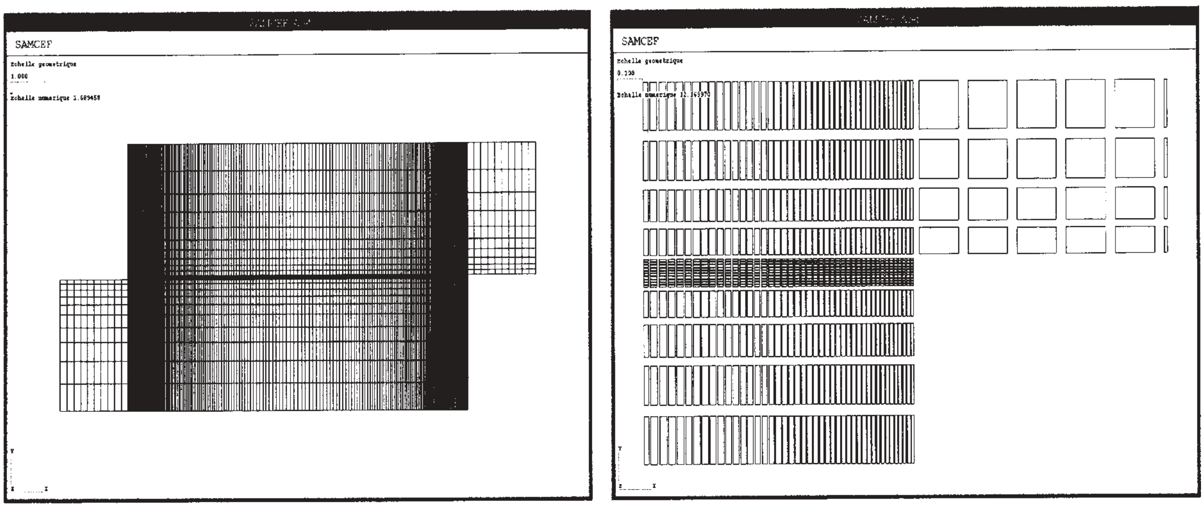
Meshing and boundary conditions

The objective of this study is to compare our analytical models of the adhesive-bonded joints with models made of finite elements.

For the numerical analysis by finite elements of the adhesive-bonded joints, we used the computer code SAMCEF from SAMTECH®.

The diagram of the C.A.D., basis of the finite element model, is presented in figure 1. The diagram also describes the boundary conditions and the loading applied.

The cylindrical assembly is modeled by 2D quadrangles of degree 2 finite elements with the axisymmetric assumption ($x \to z$, $y \to r$, $z \to \to\theta$). The displacements along x and y in face 2, of the external tube and those along y in face 1 of the internal tube are blocked. The load is applied as a pressure on face 1 (fig. 1).



a) b)

Fig 2. Numerical modeling of a cylindrical bonded joint with quadrangles elements: a) assembly; b) detail

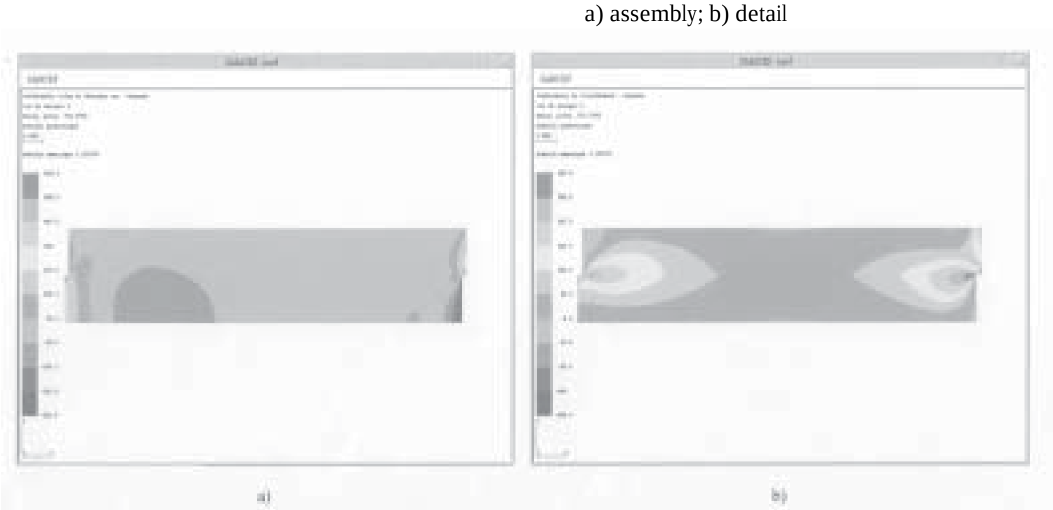


Fig. 3. Stress distribution in cylindrical bonded joint modeled with quadrangles elements for f = 100 MPa: a) orthoradial stress ($\sigma_{\theta\theta}$); b) shear stress (τ_{xy})

Table 1
ANALYZED ASSEMBLY CONFIGURATION

Tube 1	Adhesive	Tube 2	r_i [mm]	r_{ic} [mm]	r_{ec} [mm]	r_e [mm]	L [mm]	f [MPa]
Aluminium AU 2024 T3 E = 75000 MPa G = 28846 MPa υ = 0.3	**Araldite** Redux 312 E_c = 2500 MPa G_c = 1000 MPa υ_c = 0.35	**Glass/Epoxy +/- 45°** E_x = 14470 MPa E_y = 14470 MPa G_{xy} = 12140 MPa υ = 0.508	10	12	12.2	13.2	50	100

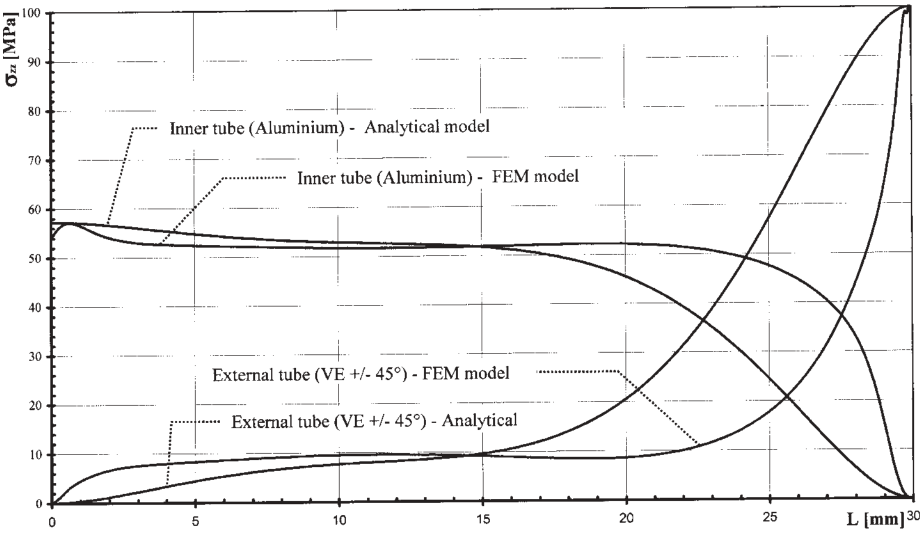
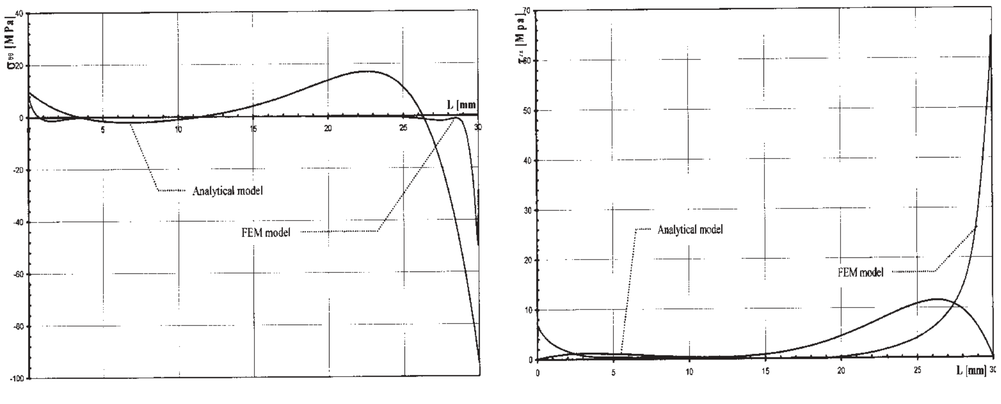


Fig. 4. Load transfer in an AU 2024 T3-AV 119-VE ±45° assembly



a) b)

Fig. 5. Stress distributions in the adhesive layer of an cylindrical assembly AU 2024 T3-AV 119-VE ±45° for f = 100 MPa: a) The orthoradial stress ($\sigma_{\theta\theta}$); b) The shear stress ((τ_{rz}).

Figure 2 shows an example of the grid used in this study where all the finite elements are quadrangles. We imposed ten finite elements according to the adhesive.

Figure 3 shows the stress distributions in the assembly in the form of cartography.

Stress distributions

Load transfer

To compare the analytical model with the finite elements model [15] we determine the load transfer in the middle of the bonded substrates (fig. 4) in a cylindrical metal-composite assembly with the characteristics given in table 1.

We can note some differences in the variation of the load given by the ideal model in the case of a metal-composite assembly, (fig. 4), while it still has a similar evolution.

The point of equivalence in stress in the two substrates is shifted (according to the length of the join) from approximately 15% in the finite element analysis. It should be noted that the position of this point varies according to the characteristics of the substrates: it is centered compared to the length of the join for substrates of equivalent total rigidities, and shifts on both sides as a function of the ratio of the rigidities of the bonded substrates.

The ideal analytical model has the same aspect as the finite element model (FEM).

Stress analysis in the adhesive

The stresses in the adhesive make it possible to predict the failure of the adhesive-bonded joint. Their distribution is thus of primary importance on the prediction of this force.

Figure 5 presents the distribution of the orthoradial and shear stresses, according to the covering length. In this case, the stresses given by the ideal model are similar to those given by finite elements. The greatest differences (30 %) are seen on the maximum amplitudes where the ideal model underestimates these values: the edge effect due to a local inflection of the substrates is not taken into account.

Conclusions

Adhesive joining is a simple method of assembly. Its interest lies in the fact that it minimizes the machining of the parts to be assembled. The performance of the adhesive bonded joints depends on the performance of the adhesive. The latest generations of adhesives, delivered in the form of film, make it possible to minimize the number of operations to make the join and greatly increase the mechanical resistance. However, the designing engineer must have at his disposal methods and/or reliable computer codes for pre-dimensioning with known margins.

The objective of our study was to entirely develop analytical models for dimensioning adhesive-bonded joints. To this end we placed ourselves in the case of a cylindrical assembly.

The basis of our analytical model was the analysis of the stresses applied to an elementary volume of the assembly under consideration, observing the boundary conditions, the geometry and materials of the assembly. The application of an energy method made it possible to obtain the solution of the problem in stress in any point of the structure. The behavior law enabled us to obtain the deformations then, by integration, the displacements. The problem in stress, deformation and displacements was thus entirely defined.

The model validation is presented by comparison with finite elements models. For the assembly, the total force-displacement behavior is well defined. Thus the analytical model makes it possible to determine the rigidity of the assembly and to obtain a simple formulation very quickly, which gives the total behaviour of the assembly.

The comparison was also carried out on the distribution of stress in the substrates and the adhesive joint. We showed that the transfer of force by joining was well determined by the model. The distribution of stress in the adhesive remained very close to the solution given by finite elements.

The analytical model underestimated the stresses in the adhesive leading to an over-estimate of the forces at rupture. However, this model is reliable and allows fast analysis of this type of assembly.

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