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Analytical solution to calculate the stress distribution in pin-and-collar samples bonded with anaerobic adhesives (following ISO 10123 standard)

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ABSTRACT

In this work, we analyse cylindrical joints bonded with anaerobic adhesives, applying the principles found in a paper of Nemeş et al. [Int J Adhes Adhes 2006; 26(6) :474]. Nemeş' paper gives an analytical solution for the different stresses that appear on the three elements of the cylindrical assembly (two tubes and the adhesive) over the whole joining surface.

A detailed study of this paper allowed us to develop a new mathematical model to be applied to a pin-and-collar specimen, in particular to the standard system, which appears in ISO 10123.

From the mechanical and geometrical properties of the components and the axial loading applied on the system, it has been possible to obtain the intensity and distribution of stresses in the assembly graphically, using the mathematical program MathCAD. Consequently, it is possible to calculate the so far unknown value of maximum shear stress.

So knowing the shear stress, the model allows (i) to predict the distribution of stresses in the adhesive bond and (ii) to carry out a parametric study of the bond; that is to say, it allows to evaluate the influence of geometrical parameters and the influence of the selected adhesive in the stress distribution within the bond.

It is, therefore, a methodology, which will make possible to calculate, quickly and simply, the distribution of stresses and the maximum shear strength in the adhesive. Moreover, it makes unnecessary to carry out numerous mechanical tests.

1. Introduction

Adhesive bonds must be specifically designed to obtain the maximum possible effectiveness. However, the complexity involved in the adherend–adhesive–adherend system is tremendous, while useful information about these unions is rather limited. Most of the tests applied by manufacturers are valid, although merely for comparative ends [1], but their use is limited for engineers in charge of design.

When an adhesive joint is designed, the idea is that the distribution of stress and strain be as uniform as possible throughout the bond. However, real distribution of stresses is not only determined by the loading level of the pieces that are joined together, but also by other factors such as geometry of the pieces, their stiffness and the stiffness of the adhesive [2,3].

Analytical solutions are applied to predict the mechanical behaviour of the adhesive bond, allowing prediction of the adhesive bond strength using experimental information about the types of material used. Numerous works have been made covering simple overlapping unions [4–7] and tubular joints [8–10].

This analytical solution is further complicated when a non-linear system is considered. These systems have their origin in the rotation of the bond due to the application of eccentric loadings, with complicated geometries, or when using different types of materials [11–13]. The use of the finite element method is highly applied in the simulation of simple overlapping joints, although the process becomes more complicated for complex geometries [8,14–17], or in the case of three-dimensional models [18,19].

Application of certain simplifications facilitates obtaining results regarding the influence of geometric and material parameters on the strength of the bond, taking into consideration the limitations of the model that involve the adopted simplifications.

Experimental results allow validating the results obtained with algebraic solutions. This approach has a reduced economic cost, given that neither an important number of experimental tests is necessary, nor are long computer calculations necessary, as would be the case with finite element methods, although the latter do allow obtaining more precise results.

Some attempts have been made in previous works to determine maximum stress in pin-and-collar joints with L-603

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and L-638 anaerobic adhesives (supplied by Henkel Adhesivos and Tecnologías, Spain) using the Volkersen model [20], and experimental trials according to ISO-10123 [21] and ASTM E229 [22]. Although the shape of distributions appeared to be adequate, the maximum stresses obtained were too low [23]. The Goland and Reissner model [8,24] was also tested, although results did not adapt to the expected [25].

A new way of approaching the study has been to use the model developed by Nemeș and Lachaud [26] to predict the stress distribution in cylindrical joints. This model is based on a variational method [27] applied to the energy potential in the bond, and it can be used to predict the field of stresses in the bond or the influence of the material and geometric parameters on the stress field.

2. Modified Nemeș–Lachaud model for the pin-and-collar specimen

The following stages have been followed to adapt the Nemeș and Lachaud model according to the new geometry:

1. Construction of an acceptable static field,
2. Calculation of potential energy associated to the stress field,
3. Minimization of this energy by the variational method,
4. Resolution of the differential equation that is obtained,
5. Validation of the mathematical model that is obtained.

In relation to the mechanical properties of the material with which the pin (1) and collar (2) are manufactured, these will be the same and isotropic, which implies that the Young's modulus (E), shear modulus (G) and Poisson's ratio (υ) will be the same. Since this is a block pin instead of a tube, the internal pipe radius (r) is zero. The different parameters that are used are outlined in Fig. 1. The Young's modulus, shear modulus and Poisson's ratio of the adhesive (C) will be E_C, G_C and υ_C, respectively.

In Fig. 2, the differences between the models pertaining to the two concentric tubes and the pin-and-collar new geometry can be observed. The adopted hypothesis has been:

- Radial stress in the three elements is $\sigma_{rr}^{(i)} = 0$;
- Rotation symmetry implies that the shear stress are $\tau_{r\theta} = \tau_{z\theta} = 0$;
- Direct longitudinal stresses in the adhesive (C) are neglected, $\sigma_{zz}{}^{(C)} = 0$;
- The axial stress will only be a function of the axial variable, z.

The stress field for a differential element of each element section will be:

For the pin (1):

$$\tau_{rz}^{(1)}(r,z) = \frac{(-r^2)}{2\cdot r}\cdot\frac{d\sigma_{zz}^{(1)}}{dz}$$

$$\sigma_{\theta\theta}^{(1)}(r,z) = \frac{(-r^2)}{2}\cdot\frac{d^2\sigma_{zz}^{(1)}}{dz^2}$$

For the adhesive (C):

$$\tau_{rz}^{(C)}(r,z) = \frac{(-r_{ic}^2)}{2\cdot r}\cdot\frac{d\sigma_{zz}^{(1)}}{dz}$$

$$\sigma_{\theta\theta}^{(C)}(r,z) = \frac{(-r_{ic}^2)}{2}\cdot\frac{d^2\sigma_{zz}^{(1)}}{dz^2}$$

For the collar (2):

$$\tau_{yz}^{(2)}(r,z) = \frac{(r_e^2 - r^2)\cdot(r_{ic}^2)}{2\cdot r\cdot(r_{ec}^2 - r_e^2)}\cdot\frac{d\sigma_{zz}^{(1)}}{dz}$$

$$\sigma_{\theta\theta}^{(C)}(r,z) = \frac{(r_e^2 - r^2)\cdot(r_{ic}^2)}{2\cdot(r_{ec}^2 - r_e^2)}\cdot\frac{d^2\sigma_{zz}^{(1)}}{dz^2}$$

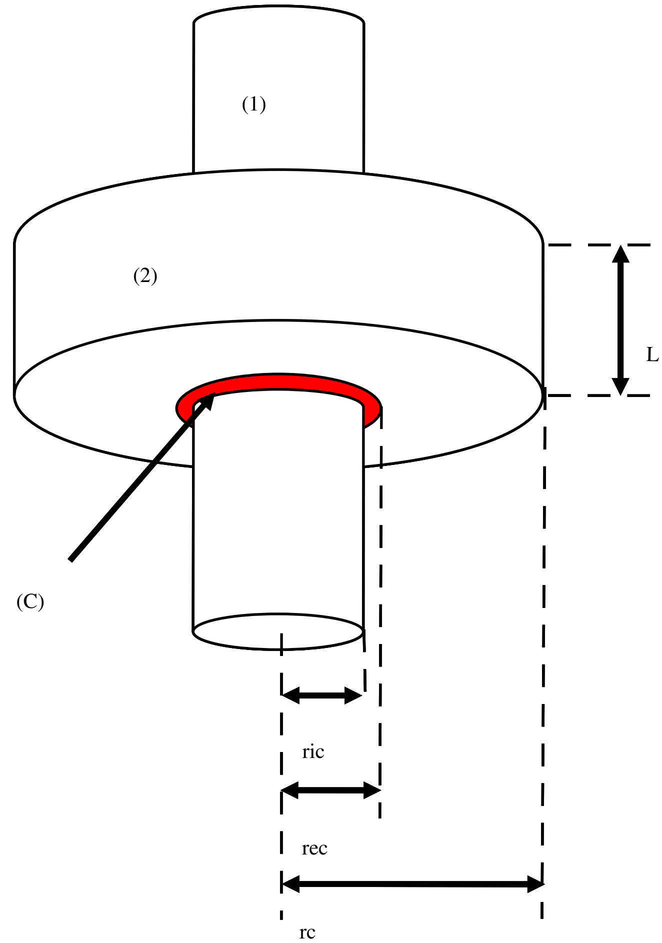

Fig. 1. Geometric parameters of the pin-and-collar sample.

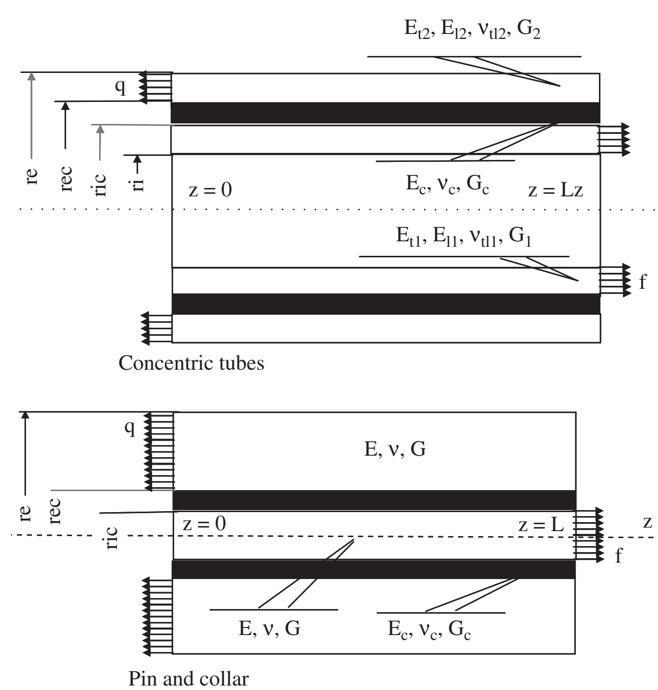

Fig. 2. Concentric tubes [26] and pin-and-collar models.