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A parametric study of the elastic stress distribution in a pin-loaded tube with multiple pinning

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ABSTRACT

This paper describes a numerical study of the stress distribution in a pin-loaded tube, loaded in tension, where more than one pin is used. The effect of changing the inter-pin distances, pin diameters and pin orientation (i.e. in-line or crossed) have been analysed and discussed. The study shows that there are significant through thickness effects as a result of out-of-plane deformations coupled with an unequal load distribution between each of the pins. Various configurations of pin positions are compared, and it is demonstrated that the joint design can be optimised by crossing, changing the diameter or moving particular pins. It is shown that constructing a joint employing three pins has advantages over a two pin arrangement and that the peak circumferential stress around a pin hole can be reduced by approximately 42% compared with using only a single pin: especially significant when fatigue loadings are considered.

1. Introduction

The use of pins to load a tube presents a straightforward means of forming a structural connection. Such connections have been used in a number of applications including civil, navel and aerospace; with notable examples found in aircraft engine subframes. In this particular case it is imperative that such connections are correctly designed not only on a strength basis, but also to avoid issues of fatigue crack initiation and propagation.

The pin-loaded lug can be considered as a two-dimensional equivalent of the pin-loaded tubular joint. The early two-dimensional finite element work of Grant et al. [1] considered the stresses around a pin-loaded lug which was loaded in tension along the axis of the lug. A later publication by Grant and Flipo [2] took a more detailed look at this connection using three-dimensional finite element models where the 'edge effects' of the pin-lug contact were described at the edges of the bore. There are also numerous other publications, experimental and numerical, for example by Frocht and Hill [3] and Strozzi et al. [4], that consider the design of the pin-loaded lug.

The pin-loaded tube has only received a limited amount of attention with the numerical work of Grant and Smart [5] representing the first paper published on this geometry; but only for a joint with a single pin. Later the pin-loaded tube, again having only one pin, was studied in 2000 by Ahmad and Stanley [6] using a photo elastic method which provided results in terms of stress indices. A further publication by these authors [7] went on to consider inline and crossed pinning; however no more than two pins were used. In these publications the geometric variation was limited, and the load-share between the pins may be judged as being unrealistic; they did not take into account the elastic effects of the shank to which the pins were attached. However, all models in these papers considered a tension loading along the axis of the tube.

For the sake of completeness, it is worth mentioning the experimental and numerical works of Grant and Smart where an analysis of a fatigue damaged, singly pinned, pin-loaded tube was considered [8,9]. In these two publications the manner in which such a joint fails was considered and correlations were made between experimental crack growth rate and J-integral analyses.

The purpose of this paper is to describe the elastic stress distribution in a pin-loaded tube with multiple pinning loaded in tension; a design which may be considered as a realistic engineering solution and a three-pin approach can be seen in some aircraft subframe structures. The effect of the variation of distance between each pin, the diameter of a pin and the pins' orientation to one another (i.e. in-line or crossed) have been studied; which should provide an insight for the would-be designer who wishes optimise

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of this type of connection. The paper published in 1997 [5], previously mentioned, reported the results of a parametric study of the elastic stress distribution in a singly pin-loaded tube: this work has been used as a basis for this new study. The results have consequently been compared with the findings presented in this article in order to unify the two studies, where the current analyses have been made with up to three pins. The fit of the pin in a hole, or level of pin interference, was investigated in this study along with the effect of changing the tube wall thickness, and the size of the pin/pin hole. It was evident that the use of a clearance fit should be avoided and that there were benefits in using a degree of interference fitting especially if cyclic loads were present; although this might not always be practical. That said, the zero clearance case was a realistic 'minimum' level of interference recommended.

It has been previously shown that there was a significant difference in the stress values and distribution between the inner and outer surface of the tube. The result will therefore be reported at the inner and at the outer surface separately, and in a similar manner to that of the singly-pinned analyses. In addition, as the position of the maximum circumferential stress around the pin hole is an indicator as to the likely site of crack initiation, the position of this maximum stress will be noted for each pin hole.

2. Modelling techniques

The multiple pin-loaded models used in this work consisted of a tube, a shank and the transversely running connecting pins. The analyses were defined using Abaqus CAE and run using the ABAQUS solver [10]. The same code, albeit an earlier version, was used in the previous study of a tube loaded with a single pin [5]. As the geometry of the tube is doubly symmetric, only a quarter of the tube has been modelled.

The quarter tube (Fig. 1(a)) is composed of C3D20R elements (continuum three-dimensional 20-noded reduced integration bricks) and meshed with a sweep method. Each of the pin holes within the tube contains the equivalent of 64 elements around a bore (i.e. 32 elements on the half bore modelled) and 6 elements through the thickness (Fig. 1(b)).

The pins have been simulated by a rigid surface as opposed to a meshed pin, which was used in Ref. [5]. To justify this choice, the rigid surface model has been chosen after first considering results produced from preliminary models using a meshed pin; only minor differences were shown. This technique has the benefit of simplifying the model and also reduces computing time for each analysis. This rigid surface is equivalent to a meshed pin with an infinite Young modulus. Preliminary investigations also involved changing element types from linear to quadratic continuum elements with full and reduced integration for the tube, pins and shank. Additionally, mesh convergence studies were made paying particular attention to the region of the tube surrounding a bore. Furthermore, principal stress extraction allowed overall comparison with the photo-elastic models of Ahmad and Stanley's work [6]. Although their experimental setup, geometry and loading, was slightly different from the work presented in this paper it gave confidence in the simulations.

For this type of structural connection there will be edge effects which give rise to very localised increases in stress down the pin holes in the tube [2]. Whist the meshes used in this work have not been optimised to capture fine edge effects that maybe present they *have* been designed to present a realistic simulation of the load sharing, the position of the maximum circumferential stress and the pin on which may be deemed to be 'critical'. It may be worth noting that when comparing a 2D and 3D (edge optimised model) in a pin-loaded lug the error associated with the max circumferential stress was 2% for a clearance fit used, 5% for a snug fit and 15% for the interference fitting pin [2]. Although these values are significant when fluctuating loads are considered the differences are likely to be significantly less for the 3D models used in this work.

The contact between each pin and the tube is a node-to-surface interaction. The tangential behaviour of the contact is set as frictionless while the normal behaviour is set as hard and calculated by the augmented Lagrange method. This type of contact does not allow the *slave surface* to penetrate the *master surface*. In our case, the pin is defined as the master surface and the tube is the slave surface. By convention, the master surface is always the hardest one, so, as the pin is modelled by a rigid surface it was set as the master. (It should be noted that the type of pins used in these types of applications are usually case hardened and may be treated as incompressible.)

The interaction between each pin has been modelled by way of a shank which was given elastic properties. This shank is a full cylinder (1/4 model) concentrically fitted inside the tube as a clearance so that the shank diameter is kept constant. It is worth noting that some initial thoughts were given towards modelling the shank with spring elements where the stiffness would be adjusted to meet the properties of the shank. Using rigid surfaces for pins would present this as a reasonable way forward; however, with the numerous parametric changes required in this study, and with the additional complexity of using crossed pins, a meshed shank was used.

The analysis is composed of an initial step in which the boundary conditions are created and a second step during which the load is applied to the pin. As the pins are linked to one another by a solid shank, the pin-loading was created by way of a negative pressure applied to the end cross-sectional face of the shank so creating force along its longitudinal axis. The corresponding end cross-sectional face of the tube was fixed.

A comprehensive script file has been written to facilitate the creation of the large number of geometric definitions required.

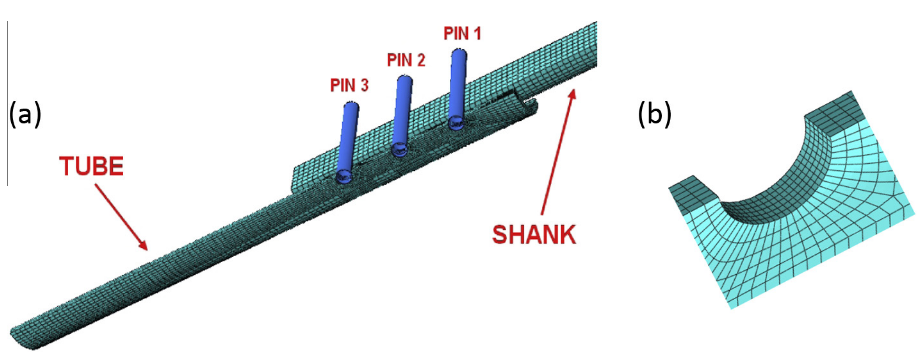

Fig. 1. (a) Finite element of typical pin-loaded tube model with (b) details of meshing around a hole.