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Simplified finite element analysis of bolted T-stub connection components

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ABSTRACT

In order to predict the T-stub behaviour, simplified theoretical models are provided by Eurocode 3 that allow to evaluate the T-stub stiffness and strength. Conversely, there are no codified rules to predict the plastic deformation capacity. Therefore, the prediction of the T-stub ductility is still an open issue in the connection modelling. Even though theoretical models are very important in order to recognise the parameters that govern the stiffness, resistance and ductility of bolted T-stubs, they cannot always be applied with confidence, because of the simplifying assumptions usually made for gaining closed form solutions. Therefore, the information coming from simplified theoretical models needs to be integrated with that obtained either by experimental results or by means of finite element simulations.

For this reason, in this paper, a simple simplified FEM model of a bolted T-stub with only one bolt row has been developed using SAP2000 computer program aiming to show how even a widespread commercial software can be used to estimate the plastic deformation capacity of bolted joints' components.

The accuracy of the FEM model has been verified by means of a comparison with available experimental results. In particular, all the specimens that were tested at the Material and Structure Laboratory of Salerno University in 2001 have been modelled and the results obtained are presented and discussed.

1. Introduction

The modelling of bolt rows of beam-to-column connections is currently carried out by means of equivalent T-stubs. Therefore, the knowledge of whole force versus displacement curve of axially loaded bolted T-stubs is of primary importance for the analysis of the moment-rotation response of bolted connections, in particular by means of the component approach [1,2].

The prediction of the force versus displacement curve of axially loaded bolted T-stubs can be performed by means of analytical models or finite element models which can be either 2D or 3D models.

Analytical models have been developed with the primary aim of predicting the initial stiffness and the plastic resistance, leading to codified rules. This is the case of EN 1993-1-8 [3] which provides design rules for the evaluation of the initial stiffness and the full plastic resistance by means of formulations based on elastic [4,5] and pure plastic theories [6,7]. However, the codified approach neglects the influence of strain-hardening and geometric non-linear effects. In addition, concerning the component ductility, only same qualitative principles are provided [8].

To fill this gap some theoretical approaches have been proposed. The first attempt of an analytical prediction of the plastic deformation capacity of bolted T-stubs was carried out by Piluso et al. [9,10] by assuming that the bending moment diagram along the T-stub flanges is characterised by a point of zero moment coincident with that occurring when a perfectly plastic behaviour is assumed and by computing the resulting plastic hinge rotations of the T-stub flange by integrating the curvature diagram resulting from the bending moment diagram. Finally, the plastic displacements are computed from the knowledge of the rotation of the plastic hinges developed by the T-stub flanges and the corresponding kinematic mechanism. However, for sake of simplicity the model disregarded compatibility requirements between bolt and flange deformation. The ultimate condition was identified as the occurrence of the ultimate natural strain in the extreme fibres of the T-stub flanges.

The main merit of the above model is that, it has demonstrated by means of a closed form solution that the plastic deformation capacity of a bolted T-stub failing according to type-1 mechanism of EN 1993-1-8, i.e. flange yielding, quadratically increases with the distance between the bolt axis and the T-stub stem and is inversely proportional to the flange thickness [1,9,10].

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Successively, starting from the above studies, a highly simplified formulation has been proposed by other researchers [11]. Moreover, non-linear model requiring an incremental solution technique was presented by Swanson and Leon [12,13] who provided also a wide experimental analysis [14] while an incremental non-linear analytical model has been recently proposed by Lemonis and Gantes [15] accounting also for the compatibility requirement between the bolts and the flange deformation. A simple beam representation for the flange and an axial spring for the bolt are used. Also contact phenomena are accounted for by means of the proposed incremental procedure.

The accuracy and the reliability in the prediction of the force versus displacement curve of bolted T-stubs can be improved by adopting numerical models based on FEM structural analysis. Such models can be either 2D or 3D models. The most accurate results can be obtained by means of 3D FEM models accounting for both geometrical and mechanical non-linearities, contact and friction phenomena, bolt preloading effects and residual stresses. Such FEM models are typically based on the use of solid elements which can be either 8-node or 20-node brick elements and 15-node wedge elements [16–20]. These 3D FEM models generally allow to accurately predict the ultimate resistance. The initial stiffness can be also evaluated with high accuracy, provided that bolt preloading is properly modelled. In fact, bolt preloading significantly affects the initial stiffness of bolted T-stubs [21] while has only a minor and generally negligible influence on the ultimate resistance. Conversely, the ultimate plastic deformation capacity is always very difficult to be predicted because the failure mode can be governed either by the T-stub flange or by the bolts. In particular, the highest uncertainties occur in the prediction of the bolt failure. In fact, three potential failure mechanisms can developed: (1) bolt shank failure under combined action of axial force and secondary bending; (2) stripping of the bolt threads (3) stripping of the nut threads.

However, to the best of authors' knowledge, even in the case of sophisticated 3D FEM models, failure modes (2) and (3) cannot be predicted because bolt and nut threads typically are not modelled. In addition, even if such a modelling was carried out, significant uncertainties would remain because the actual bolt material properties cannot be measured, being unavoidably referred to a coupon specimen extracted from a bolt different from those adopted for the bolt T-stub specimen. Further discussion on the modelling of bolts in tension can be found in a recent work where different finite element models are examined [22].

It has to be considered that sophisticated 3D FEM models are still not applicable for practical design and even parametrical analyses with such models are highly cumbersome. For this reason, the development of accurate 2D FEM models maintains a significant research interest. In fact, even though 2D FEM models are not able to rigorously account for the influence of the interaction between the bolt head/nut and the T-stub flange because of the actual 3D behaviour, they can be applied by properly exploiting the concept of effective width which constitutes the basic simplification commonly required even in the application of analytical models. In particular, the interest in 2D FEM models is due to their ability to provide a good compromise between the computational effort required by highly accurate 3D models and the coarseness of analytical models, requiring many simplifying assumptions to lead to closed form solutions.

A 2D FEM model adopting geometrical and mechanical properties consistent with the prying model of Eurocode 3 has been proposed by Girão Coelho et al. [23] and implemented by means of Lusas software [24]. The flange material is modelled according to a multilinear true stress-natural strain model as suggested in [10] and the bolt behaviour is modelled according to [12]. The FEM model is based on the use of beam elements belonging to the Kirchoff beam group, neglecting shear deformability. As already pointed out by other researchers, the computed values of the strength and deformation capacity are not always predicted accurately because of the model sensibility to strain hardening parameters and bolt ductility.

Starting from the background mentioned above, in this paper a new 2D FEM model is proposed and implemented by means of SAP2000 structural analysis software [25]. The model is conceived to account for both geometrical and mechanical non-linearities and contact phenomena. In addition, also the influence of the bolt head and nut is considered. Contact phenomena between the bolt shank and the flange hole are also included in the model. In particular, mechanical non-linearities are modelled by means of the plastic hinge fibre model available in SAP2000 [25]. The numerical results obtained by means of the proposed 2D FEM model are compared with the experimental test results obtained in a previous research programme [26]. The role of the uncertainties due to the modelling of the bolt material properties is also investigated by means of a lower bound and upper bound approach.

The above approach accounts that experimental data is stochastic by nature and is always subject to some variation. According to [27], calibration of input data has been avoided, because it means an unjustified modification of the input data applied to a numerical model in order to shift the numerical results closer to experimental results.

2. The proposed simplified fem model

The finite element model proposed has been implemented in SAP2000 software. Reference is made to a single bolt row whose behaviour is described by an equivalent bolted T-stub having an effective width computed according to Eurocode 3 [3]. The whole T-stub behaviour under axial loading is characterised by two symmetry axes. Therefore, the finite element model is referred to one fourth of the whole T-stub (Fig. 1).

The vertical symmetry axis (VSA) is accounted for by means of a horizontal double pendulum restraining the beam element modelling the T-stub flange. However, the restrained point A of the T-stub flange is located at a distance from the vertical symmetry axis equal to $0{,}5t_w + 0{,}8r$, being t_w the thickness of the T-stub web and r the root radius of the flange-to-web connection. Such distance accounts for the almost rigid behaviour of the flange-to-web connection zone and is evaluated consistently with Eurocode 3 model. The horizontal symmetry axis (HSO) is taken

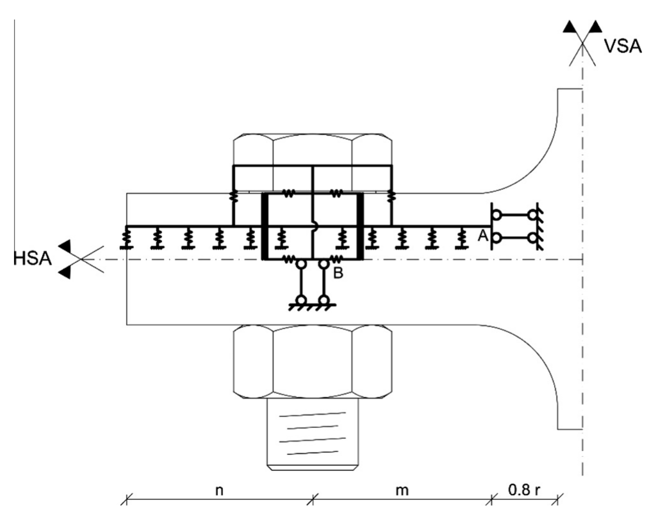

Fig. 1. T-stub model.