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Experimental characterization of the bending fatigue strength of threaded fasteners

Henrik Wentzel *, Xiyue Huang

Department of Solid Mechanics, Royal Institute of Technology (KTH), SE-100 44 Stockholm, Sweden

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ABSTRACT

The fatigue strength in bending of pre-stressed steel bolts is investigated and compared to the fatigue strength in axial tension. The strength is measured in terms of maximum engineering stress amplitude, neglecting any stress concentration in the threads. The experimental results reveal that the fatigue limit is 76% higher in bending than in axial tension. A finite element model is used to compute the stress state in the threaded region for both axial tension and bending. It allows fitting a volume based weakest link model to the experimentally observed failure probabilities. Based on the good fit of the weakest link model it is argued that randomly distributed defects in the highly stressed thread root determine the fatigue strength.

1. Introduction

Bolted joints are widely used for assembly of engineering structures thanks to their ease of assembly and disassembly and their low cost. Fatigue failures of bolted joints are often costly and occasionally fatal. This motivates the characterization of the fatigue strength of bolts.

The testing of the axial fatigue strength of bolt and nut assemblies is the subject of international standardization, for example [1]. Fatigue failures occur primarily in the bolt, the stresses in the nut are generally much lower. The fatigue strength of the bolt is expressed in terms of the engineering stress amplitude, e.g. force amplitude over cross sectional area, thus ignoring the stress concentrations in the thread root and in the transition from bolt shank to bolt head. The fatigue strength in terms of engineering stress can be compared to loads obtained through handbook calculations or finite element simulations using the nominal geometry of the bolt, e.g. ignoring the threads. One such handbook is [2].

In applications many bolts suffer fatigue failure due to bending loads. This is simply because the joint can support large dynamic axial forces without generating dynamic stress in the bolt, while this capacity is much lower in bending [3]. Hobbs et al. performed fatigue experiments on bolts using eccentric loading [4]. However, in the eccentric test setup the alternating axial stress is an order of magnitude larger than the bending stress.

There is no standard test for the bending fatigue of threaded fasteners and there is virtually no data on the bending fatigue strength publically available. This is probably because it is not trivial to construct a testing device that generates a bending load with high tensile pre-load without using two hydraulic cylinders.

Tests on smooth or notched laboratory specimens typically show higher fatigue strength in bending than in axial tension [5]. This can be explained by statistical effects, or by considering the stress gradient [6]. Threaded fasteners have high stress gradients both in axial tension and in bending due to the geometry of the thread. It may be meaningful to consider statistical effects based on the weakest link theory, such as the stressed volume approach [7] or the stressed area approach [8].

In this paper is presented an experimental setup for bending testing of threaded fasteners. Experiments in both axial tension and bending have been carried out on industrial high strength steel bolts, and the results are reported here. A finite element model of the bolt and nut assembly is used to resolve the stress state in the thread root. The stress state is thereafter used to fit a statistical model based on the stressed volume approach to the experimental data.

2. Test object

The tested bolts are ISO-metric M14 flange bolts with coarse thread, strength class 10.9 and length 140 mm, heat treated after rolling of the threads. All bolts are from the same manufacturer and the same batch. The bolts are used together with ISO-metric

* Corresponding author. Tel.: +46 730469876.
E-mail address: hwentzel@kth.se (H. Wentzel).

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flange nuts of strength class 10. Both nuts and bolt are ordered in the regular production flow of a vehicle manufacturer. In Fig. 1 is shown a photographic image of the bolt and nut assembly. Marks are clearly visible on the ridges of the threads, as seen in the close-up image in Fig. 2, most certainly due to handling of specimen. However, the bottom of the threads appears undamaged.

Three bolts were equipped with strain gauges to enable strain measurements at the positions 7 mm, 48 mm, and 73 mm from the bolt head. The strain gauges were positioned on the unthreaded shank of the bolt and aligned with its axis, one on each side of the bolt so as to permit measurements of both the axial strain and the bending strain at the different positions. Thus in total 6 strain gauges were employed on each instrumented specimen. In the bending test the bolts were carefully positioned so that the strain gauges aligned perpendicular to the pivot line so as to measure the maximum bending strain.

The effective cross-sectional area of the threaded part of the bolt is $A_{eff} = 115\ \text{mm}^2$ [3], corresponding to an effective radius of $r_{eff} = 6.05$ mm. The effective radius is used to calculate the engineering stress in the threaded part of the bolt. On the non-threaded bolt shank the radius is 7.00 mm.

3. Test equipment and procedures

Testing was conducted at the Laboratory of the Department of Solid Mechanics in two different servo-hydraulic test machines.

3.1. Axial testing

Procedures and equipment for axial fatigue testing of bolts are well described in the standard [1]. The bolts were tested in conformity with the standard.

The bolts were inserted through cups that were mounted in the test machine. The machine was a servo-hydraulic MTS with a capacity of 160 kN with a control unit from Instron, M8520+. The load was measured with the machine's internal load sensor and the displacement was measured with the machine's internal LVDT. To minimize the effect of any misalignment, spherical washers were used under both the nut- and the bolt head.

After assembly the bolts were given a pre-tension of 73 kN, corresponding to approximately 635 MPa. Note that the yield strength of the bolt material is at least 900 MPa. Thereafter a time-varying sinusoidal load of amplitude F_a was added until fatigue failure occurred.

The engineering amplitude stress is

$$\sigma_e = \frac{F_a}{A_{eff}}. \tag{1}$$

During the testing the stress vary between $635 - \sigma_e$ to $635 + \sigma_e$. The load ratio, often referred to as the R-value, $R = \max(\sigma)/\min(\sigma)$, depends on the stress amplitude.

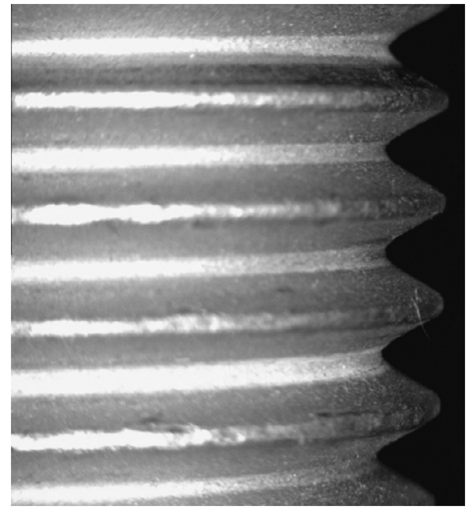

Fig. 2. Close-up photograph of the threads.

In [9] a distinction is made between the number of cycles to crack nucleation and the number of cycles to failure. No such distinction is made here, but failure is defined to have occurred when the displacement exceeds 5 mm. At this displacement the bolt is broken into two pieces that can be separated by hand.

3.2. Bending testing

There is no ISO-standard for bending testing of bolts and a novel test setup has been developed in this work. The setup consists of two rigid parallel plates that are joined with a hinge. A test machine is used to actuate the hinge, thereby changing the angle between the plates. The test specimen (bolt and nut) is positioned through drilled holes in both plates and pre-tension is obtained by applying torque on the nut.

When the hinge is actuated the bolt head and nut head are subjected to forced relative displacement (mainly rotation) and in this way bending stresses are induced in the bolt. In this setup the hinge is realized with a hardened steel wedge resting in a groove, allowing the wedge to pivot around the line of contact. The test setup is depicted in Fig. 3 where the most important measures are indicated.

It should be noted that the test setup induces both bending and axial stress in the bolt. By positioning the centre of the bolt close to the line of contact the axial stress amplitude is minimized. However, the test was difficult to control when the bolt was directly over the line of contact, so a distance of $L_B = 1.25$ mm was maintained. In this way the machine could work in tension through the complete load cycle which proved stable. Shallow slots were machined in the plates to allow for accurate positioning of the bolt and nut.

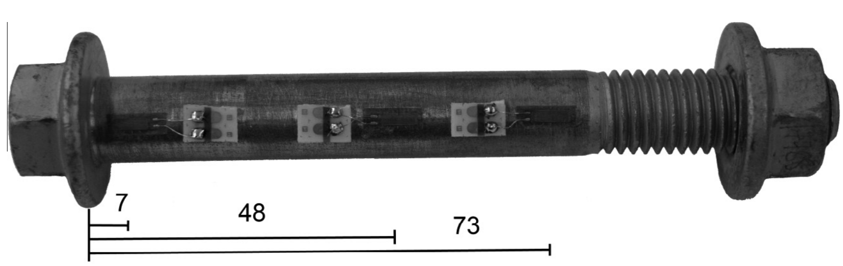

Fig. 1. An instrumented specimen of bolt and nut with positions of the strain gauges indicated.