

Nut Resilience and Vibration Fatigue Limit of Bolted Joints

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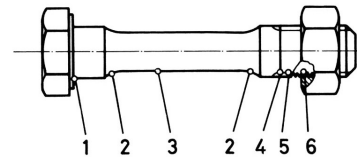
ABSTRACT

An increasing number of materials with a lower specific density are being used as part of the drive towards lightweight construction. Plenty of tests have been carried out on light metal bolts. However, the only bare essentials are known about the effect of nut resilience on the vibration fatigue limit. As bolts are subject to both static and cyclical stress during their lifetime, their load-bearing capacity under cyclical stress is hugely important.

The general measures available to improve the vibration fatigue limit will be presented. The effect of varying the modulus of elasticity of internal threads on the vibration fatigue limit of standard high-strength bolts will be shown.

INTRODUCTION

With a share of 65% - 70%, bolted joints are the form of detachable joint that can most commonly be used in mechanical engineering and vehicle construction /1/. As this mechanical element is found in large quantities and is frequently used in both the industrial and the private sector, its use is often unproblematic. While the load-bearing capacity is often higher under static stress than with smooth components, the cyclical load-bearing capacity is significantly reduced here as a result of the high stress concentration. The durability of a bolted joint under vibration stress can thus be reduced in some cases to less than 10% of its potential durability /2/. The reason for this reduction in load-bearing capacity is the notches connected in series, which induce extremely high stresses in the notch root in some places. **Fig. 1** gives an overview of the notches on a bolt with the corresponding stress concentration factors k_t .



Pos.	1	2	3	4	5	6
k_t	3 - 5	~ 1,1	1	2	~ 2	5 - 8

Fig. 1: Bolted joint with notched areas and stress concentration k_t /e.g. 2/

The bolted joint component system consists of the screw bolt as the central component, the nut or a basic body with an internal thread and the stressed parts. However, nuts and stressed parts should under no circumstances be regarded as passive elements. Indeed, these can materially affect the stress and the load capacity of the joint /3/. The vibration fatigue limit can be improved by implementing a range of different measures. The superelevations in stress along the screw axis are shown schematically in **Fig. 2**. The targeted application of internal compressive stresses through rolling after heat treatment can be used to improve the local component strength in the area of the highest stress concentration (**Fig. 2a**).

The maximum stress in the first load-bearing thread can be reduced by standardising the bolt force on all the internal threads (**Fig. 2b**). This can be done using appropriate design or material-based measures, such as using tensile nuts or nuts with higher resiliencies.

Apart from improving the vibration fatigue limit, a safely applied assembly preload force and the residual clamping force still present after or during a given operating period are important for the functionality of the system /2, 5 - 7/. If the residual clamping force under operating conditions is so severely reduced that there is lifting of the stressed parts on one side, eccentrically stressed bolted joints are subject to hyperbolically increasing additional bolt forces that can induce a fatigue fracture if they exceed a critical level (**Fig. 2c**). The residual clamping force is primarily determined by

the level of surface pressure on the bearing surfaces of bolts and nuts and by relaxation procedures in the stressed system /3/.

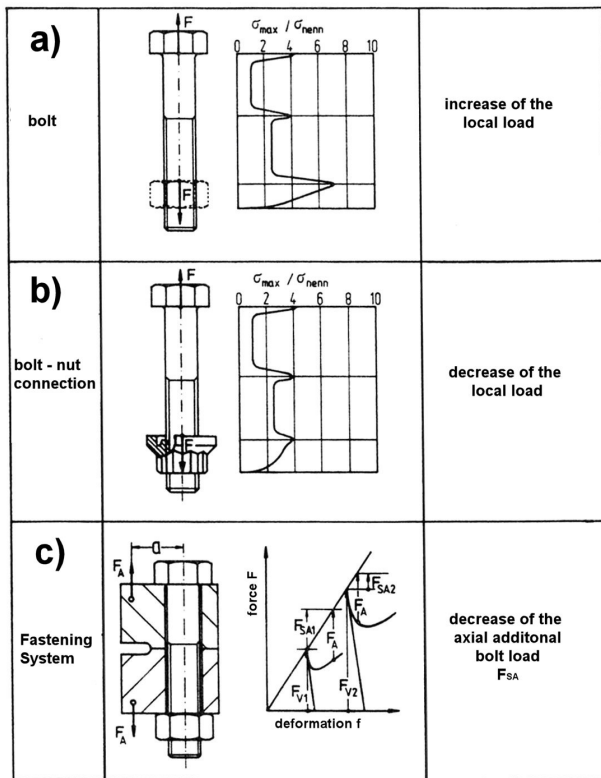


Fig. 2: Influence on the cyclic strength of bolted joints /4/.

MAIN SECTION

To improve the vibration fatigue limit of bolt/nut joints, an attempt must accordingly be made to reduce the peak stresses at the places exposed to the greatest stress.

The method for standardising the bolt force described in **Fig. 2b** using several paired threads can be implemented through design or material-based measures. Design measures include, for example, using tensile nuts, changing the thread tolerances, differences in the angles of the flanks of the thread and differences in the thread pitch. According to /2/, tensile nuts are used only rarely, as the result is rather limited and the costs are relatively high. An alteration of the thread dimensions within the standardised tolerances produces rather limited results. Higher tolerances such as 6g/6H should be given precedence over narrower tolerances such as 4h/4H, as these increase the elastic bending resiliencies /2, 8/. Differences in the angles of the flanks of the thread alter the introduction of the force. In what is referred to as the endurance limit range, even differences in the flank angle of 5° failed to produce any significant increase /2, 9/. Differences in the thread pitch trigger an introduction of force, starting from the side facing away from the bearing surface of the nut. As the preload force increases, the neighbouring threads are

gradually involved in the transfer of force. This produces a more uniform distribution of load in the paired threads /10, 11/.

Standardisation of the distribution of thread load and an improvement of the vibration fatigue limit are thus effected by changing how the force is introduced or by increasing the resilience. The following formula generally applies to the resilience δ in the linear elastic range:

$$\delta = \frac{l}{E \cdot A}.$$

It is often not possible to change the geometric parameters l and A ; or in larger components they do not produce any significant increase in the vibration fatigue limit. Changing the modulus of elasticity (i.e. material-based measure) increases the resilience and can therefore help improve the vibration fatigue limit. Textbooks /1, 2, 12, 14/ therefore generally state that using nuts with a lower modulus of elasticity than steel (e.g. grey cast iron, titanium, aluminium, magnesium) increases the vibration fatigue limit. However, there is no information about any increase in load-bearing capacity if these various materials are used.

Comparative investigations on the subject of the effect of the modulus of elasticity of the nuts on the vibration fatigue limit of bolted joints tend to be few and far between. /15/ for example features investigations into the vibration fatigue limit characteristics of larger bolts. These included tests with constant minimum stress ($\sigma_u = 10 \text{ N/mm}^2$) on modified metric bolts of dimensions M25 x 2 with nuts made of material 34CrNiMo6, GGG60 and a titanium alloy RT18. Up to a limit of $3 \cdot 10^6$ stress cycles, the tests, not verified statistically, revealed a significant increase in the vibration fatigue limit of the bolted joint from 85 N/mm² to up to 155 N/mm² as the modulus of elasticity fell. An article by Wiegand from as early as 1939 /13/ referred to the positive effect of nuts with a low modulus of elasticity on the vibration fatigue limit of steel bolts. In this instance, steel bolts of dimensions M22 x 1.5 with an mean stress of around 150 N/mm² were paired with steel nuts and with nuts made of “cast iron” and an “aluminium alloy”. A 23% increase in the vibration fatigue limit was noted as a result of changing from the steel nut to the nuts with a lower modulus of elasticity.

The characteristics of continuous stress in respect of bolts of the M8 and M10 sizes standard in car manufacturing in property classes 8.8 and 10.9 were examined in dedicated tests. High-strength bolts are usually paired with steel nuts. For this purely academic case, vibration tests were carried out on M10 x 90 – 10.9 ISO 4017 bolts rolled after heat treatment with standard hexagon nuts made of ISO 4033 steel of property class 10 and the aluminium alloy Al9. In a second series of tests, bolts rolled before heat treatment of dimensions