Theoretical analysis of preloaded bolted joints subjected to cyclic loading

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Abstract In this paper the performance of preloaded bolted joints is analysed. The prior torque tightening creates compression in the clamped members and tension in the bolt. So, when an external tensile load is applied to the joint a large amount of it is taken up by the members, thereby relaxing their compression, at the expense of only a slight increase in bolt tension. In this way preloading the joint has the effect of changing the bolt fatigue regime from a low mean load with high alternating load to a high mean load with a low alternating load. This is beneficial for its fatigue life.

Keywords fatigue; preloading torque; bolted joint

Introduction

When dealing with aircraft structures the design philosophies [1] have changed over recent years. Fatigue life, durability and damage tolerance have replaced static strength and safe life as the key concepts of design. Furthermore, the number of structural failures in service has been reduced by improved materials, sophisticated design analysis and thorough testing programmes. Yet the main concern engineers have about aircraft structures is the detrimental ageing factors, such as metal fatigue.

Aircraft structures are constructed by inserting mechanical fasteners through holes in the material. This use of fasteners transforms the series of material shapes into a structure by the formation of joints. One of the most common methods of joining aircraft structures is by using nuts and bolts, which allows easy assembly and disassembly without the need for specialist tools.

Bolts require a hole to be made in the material and the presence of this hole will produce a stress concentration around that area, thus reducing the fatigue life of the structure. For this reason, joints are often blamed for causing weak points within the structure and for increasing the weight of the overall assembly, making the performance of the joint critical to weight reduction and to keeping the strength constant.

There are a number of factors that influence the performance of a bolted joint within the structure. An important one is the level of preload, or torque, applied to the bolt, which compresses the joined elements and increases the friction between them, thus facilitating the load transmission from one to another. This has been studied by several authors [2–4], who have detailed the performance of bolted joints.

Although preloading the bolts can facilitate load transmission between the joined elements and alleviate the stress concentration in the surrounding area, it raises the question of fatigue life of the bolts themselves, which support tension from the beginning, and quite often they suffer from an alternating load in service.

This paper analyses the behaviour of bolted joints under cyclic loads, and shows how their fatigue life can be improved by applying preloading torques.

Bolt preloading

Torque tightening of a bolted joint places the bolt in tension and the clamped members in compression. Effectively the resulting force exerted on the nut stretches the bolt and clamps the members of the joint together. In this way the bolt and the surrounding region of the members of the joint behave as a compound bar (see Fig. 1).

In fact, tightening the nut is equivalent to reducing the active length of the bolt by the following amount:

$$\delta = np \tag{1}$$

where n is the number of turns or part of a turn and p the pitch of the thread.

The shortening of the bolt's active length is taken up by the combined compression of the members of the joint and the extension of the bolt itself. So, the compression of the members of the joint, δ_m, and the tensile extension of the bolt, δ_b, will be such that:

$$\delta_m + \delta_b = \delta \tag{2}$$

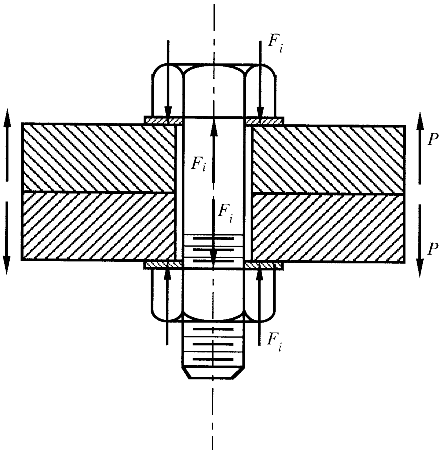

Fig. 1 *Forces acting upon a preloaded bolted connection. Before the external load,* P*, is applied, bolt tension and compression of the clamped members are of the same value.*

Also, the resultant compression load on the members of the joint and the tensile load on the bolt initially, i.e. when there is not yet any external load, are of the same value, F_i, since they are action and reaction forces. Accordingly,

$$\delta_\mathrm{m} = \frac{F_\mathrm{i} l}{E_\mathrm{m} A_\mathrm{m}} = \frac{F_\mathrm{i}}{K_\mathrm{m}} \tag{3}$$

and

$$\delta_\mathrm{b} = \frac{F_\mathrm{i} l}{E_\mathrm{b} A_\mathrm{b}} = \frac{F_\mathrm{i}}{K_\mathrm{b}} \tag{4}$$

where l is the bolt length and E_m, E_b, A_m and A_b are the respective modulus of elasticity and sections of the compressed region of the members and of the bolt. Then K_m and K_b are their respective stiffness values.

As a consequence, the force derived from the torque to turn the nut of the bolt can be related to the shortening of the bolt as follows:

$$\frac{F_\mathrm{i}}{K_\mathrm{m}} + \frac{F_\mathrm{i}}{K_\mathrm{b}} = \delta \tag{5}$$

Member stiffness can usually be obtained accurately only by finite element modeling or experimentation. If the material is of adequate thickness, then a pressure cone of compressed material in the shape of a frustum of a cone will be present and the average area should be considered. Otherwise, if the material thickness is small, the compressed area can be assumed constant, in which case the compressive zone in the member forms a cylinder. If this is three times the bolt diameter, d, the stiffness of the member can be assumed to be:

$$K_\mathrm{m} \approx \frac{2\pi d^2 E_\mathrm{m}}{l} \tag{6}$$

while the bolt stiffness is given by:

$$K_\mathrm{b} = \frac{\pi d^2 E_\mathrm{b}}{4l} \tag{7}$$

All this can be represented and better understood graphically – see Fig. 2. From such a diagram, developed by Shigley [5]:

$$K_\mathrm{m} = \mathrm{tg}\alpha \quad K_\mathrm{b} = \mathrm{tg}\beta \tag{8}$$

If the two materials have a similar modulus of elasticity, then, according to equations 6 and 7:

$$K_\mathrm{m} \approx 8K_\mathrm{b} \tag{9}$$

Fig. 2 shows the effect of preload F_i on the deflection of the joint before any external load is applied. The preload causes a compressive deformation, δ_m, in the joint members, inversely proportional to their stiffness, K_m, while the bolt is loaded in tension and deforms by δ_b, inversely proportional to its stiffness, K_b. In the process

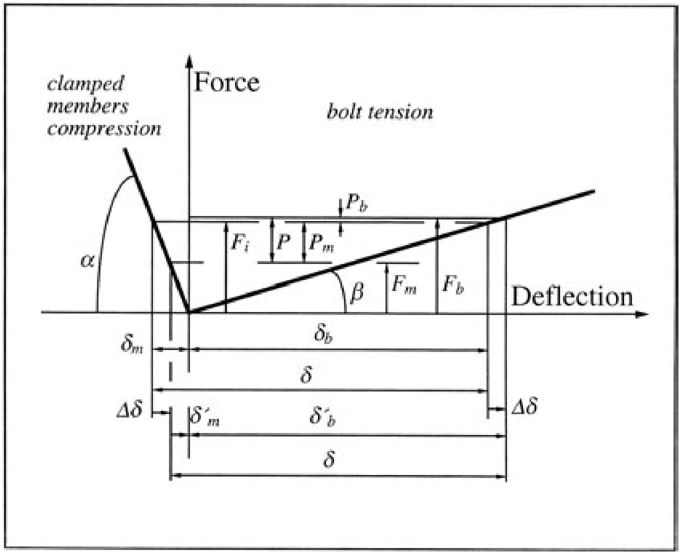

Fig. 2 *Force-deflection characteristics of a bolted joint before any external load is applied and once a load,* P, *has been applied.*

the sum $\delta_m + \delta_b$ equals the reduction, δ, of the bolt's active length corresponding to its torque tightening.

External loading of a preloaded joint

Now, if an external axial load, P, is applied to try to pull apart the members of the joint (see Fig. 1), it will tend to uncompress those members. Therefore, the load will be shared by the joint members and by the securing bolt in such a way that it will represent a relaxation force, P_m, for the compression of the clamped members and a further tension, P_b, for the bolt. This means that:

$$P_m + P_b = P \tag{10}$$

whereas the difference between the resultant bolt tension, $F_b = F_i + P_b$, and the resultant compressive force on members, $F_m = F_i - P_m$, is the external load:

$$F_b - F_m = P \tag{11}$$

as represented in Fig. 2.

Also, the new deflections of both the clamped members and the bolt, δ'_m and δ'_b, still must satisfy the condition of equation 2:

$$\delta'_m + \delta'_b = \delta \tag{12}$$

which means that the deflection introduced by the external load, P, in both the joint members (reducing their shortening) and the joining bolt (increasing its extension) is the same, $\Delta\delta$. Fig. 2 can be used to show that the portion, P_b, of load P taken by the bolt, that is the tension added to the bolt by the external load, is:

$$P_b = \frac{P}{r+1} \tag{13}$$

where r is the ratio of member stiffness to bolt stiffness, K_m/K_b. And so the resultant bolt load, once the external load is applied, is:

$$F_b = F_i + P_b = F_i + \frac{P}{r+1} \tag{14}$$

On the other hand, the portion, P_m, of external load P assumed by the clamped members will be:

$$P_m = \frac{rP}{r+1} \tag{15}$$

and the resultant compression in members:

$$F_m = F_i - P_m = F_i - \frac{rP}{r+1} \tag{16}$$

Fluctuating loads

According to what has been described above, it is of interest to analyse how a joining bolt behaves in a fatigue context, when the joint (see Fig. 1) has to transmit an external tensile load varying from 0 to P.

The bolt itself will be loaded in different ways, depending on whether it is preloaded or not. If no preload is applied, then the external load goes directly on the bolt and it will be loaded with a tensile load varying from 0 to P, which would represent a mean load $P/2$ and an alternating load of the same value.

Otherwise, if the bolt is preloaded by torque tightening with a force, F_i, then, when the external load is null, the bolt is tensioned by a force F_i, and when the external load is P, the bolt has a tensile load given by equation 14. In this case the mean load would be $F_i + \frac{P}{2(r+1)}$, which is of the order of the preload, whereas the alternating load would be $\frac{P}{2(r+1)}$, which is much lower than if there were no preload.

In parallel, the clamped members, when the force resulting from the tightening torque is F_i, if the external load is null are subjected to a compression F_i, and when the external load is P their compression is released down to the value given by equation 16. This would represent a mean compression load $F_i - \frac{rP}{2(r+1)}$, with a fluctuation $\frac{rP}{2(r+1)}$.

TABLE 1 *Effects of preloading on bolt tension and member compression*

Preload (F_i) (kN)	External load (P) (kN)	Resultant bolt tension (F_b) (kN)	Resultant member Compression (F_m) (kN)
0	1.8	1.8	1.8
1.7	1.8	1.9	0.1
1.7	0	1.7	1.7

As an example, in the case of a joint carrying an external tensile load varying from 0 kN to 1.8 kN, without any preload, the bolt would suffer a 0.9 kN mean load and an alternating load of equal value. Similarly, the clamped members would be subjected to a mean compressive load of 0.9 kN with an alternating load of equal value. However, if the bolt is torqued with a preload of 1.7 kN and the ratio of member stiffness to bolt stiffness is $K_m/K_b = 8$, as given by equation 9, the bolt will take only 0.2 kN from the external load (see equation 13), the rest of the external load, 1.6 kN, being assumed by the clamped members (see equation 15). This would represent a 1.8 kN mean load for the bolt with an alternating load of only 0.1 kN, while the clamped members would have a 0.9 kN mean compression with an alternating load of 0.8 kN. Table 1 summarises all these results.

Three points may be observed:

(1) Clamped members remain in compression despite an external tension load of the order of preload and even greater.
(2) The proportion of external load taken by the bolt can be only as low as 10% of the total load.
(3) If clamped members were more flexible, as would occur using a gasket seal, the ratio K_m/K_b would be much smaller and so the resultant bolt load would be much greater.

Fig. 3 represents the bolt tension for the two cases, and shows how significantly its fatigue regime changes when preload is introduced. Similarly, Fig. 4 represents the member compression in the two cases, and shows that its fatigue regime also changes when preload is introduced, although not as much as the bolt's.

Fatigue life

It is well known that, in relation to fatigue, the magnitude of the alternating load is more important than the absolute load. In fact, the fatigue life of a structural member supporting a fluctuating load depends mainly on the load variation, as shown by the classic empirical *S–N* diagram (Fig. 5), in which *N* is the number of load repetitions the member can endure when supporting an alternating stress, *S*. In this diagram the successive curves represent the fatigue life of any structural element under a fluctuating load, for different values of the mean stress. It clearly sows that with low mean stress the curve shifts rightwards, indicating a longer life.

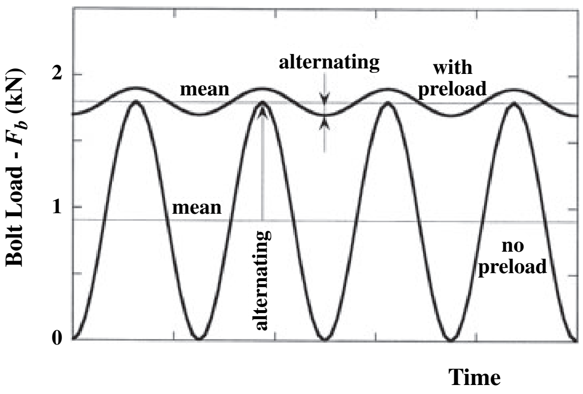

Fig. 3 *Fatigue regimes of the bolt in the two cases. With no preload: low mean load and high alternating load. With preload: high mean load and low alternating load.*

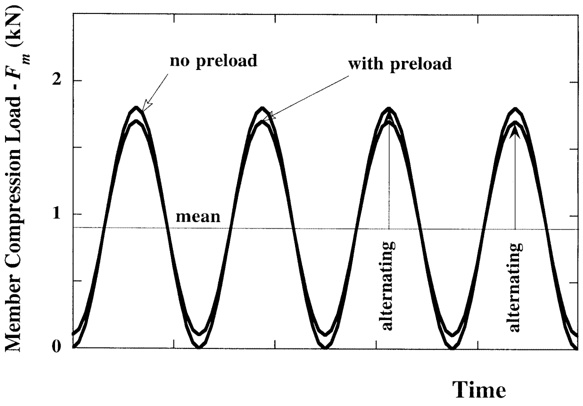

Fig. 4 *Fatigue regimes of the clamped members in the two cases. Although the mean compression remains the same when a tightening torque is applied, the alternating load diminishes a little.*

Preloading bolted joints changes the fatigue regime of the bolts and improves their fatigue performance by reducing their alternating load, even when their mean load is increased. Effectively, the change introduced by preloading the joint, shown in Fig. 3, results in shifting from a curve corresponding to a low average stress to another curve with a higher average stress, as shown in the *S–N* fatigue diagram of Fig. 5. However, as the alternating stress is also very much diminished when preloading is applied, the final result of the two effects is a substantial increase in the fatigue life of the bolt. As a consequence, preloaded bolts can prolong their life under fatigue very significantly.

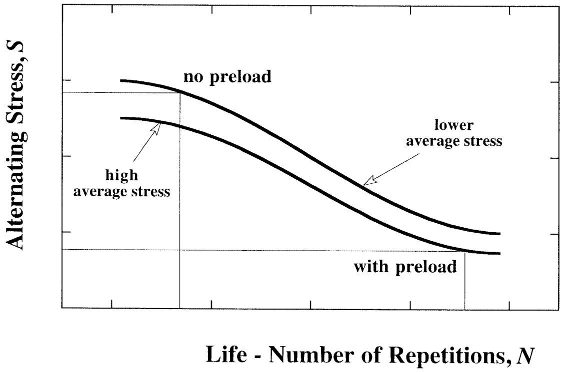

Fig. 5 *Classic* S–N *diagram of bolt fatigue performance. Although preloading raises the average stress, as it also causes a remarkable reduction of the alternating stress it brings about a real improvement since it prolongs bolt life significantly.*

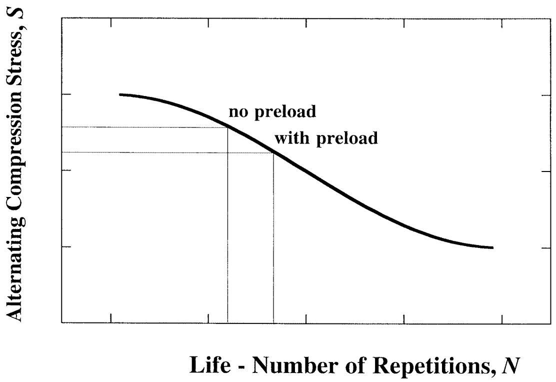

Fig. 6 S–N *diagram representing member fatigue performance. Preloading does not change the average compressive stress, but diminishes the alternating stress, which causes a prolongation of the fatigue life of the clamped members.*

As for the clamped members, they also experience an improvement in their fatigue life. Effectively, the change in their fatigue regime, as shown in Fig. 4, diminishes the alternating stress they are under, with no change in their average stress. This, according to the *S–N* diagram of Fig. 6, means at least a small prolongation of their fatigue life.

Conclusions

It has been shown how preloading of bolted joints by torque tightening makes them work under a more favourable fatigue regime, with a lower alternating tensile load.

Accordingly, as alternating load is the prime cause of fatigue failure, preload is very beneficial.

With torque tightening, clamped members also have better resistance to external load, as their compression is also reduced, due to initial preloading.

Furthermore, preloading the joint can ensure a friction grip between clamped members, thus better transmitting shear load while diminishing stress concentrations in the compressed area.

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