Failure of Fastening Screws and Their Preventive Methods 5th report

Factors related to fatigue strength of bolts and conventional methods

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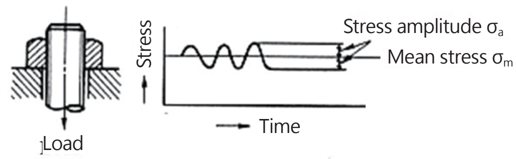
Introduction

The author has referred in the previous reports that a surprisingly large number of bolts are used in a widely variety of machines and equipment, such as electrical equipment, machine tools, construction machinery, rolling-stock, steel towers, bridges, transportation equipment, etc. Therefore, it might be thought that bolts cause, in reality, the largest number of failures among mechanical parts. As far as bolts failures, about 90% of failure cases are caused by fatigue and followed by delayed fracture (5%), SCC (3%), and static fracture (2%) including corrosion. In addition, as environmental failure is limited to high-tensile bolts which are very sensitive to a corrosive environment, 3rd and 4th reports had introduced about environmental failures of bolts which is called "delayed fracture" and "stress corrosion crack".

This report will explain about the factors related to fatigue strength of bolts and introduce the conventional methods for fatigue strength improvement of bolts.

Table 5.1 Tensile fatigue limit of steel bolt

Nominal diameter [mm]	6	8	12	20	30	42	48
Fatigue limit σ_w [kgf/mm^2]	6	6	5	4	3	3	3



Note: More strictly, the fatigue limit is indicated in terms of $\sigma_a+\sigma_m$ by definition

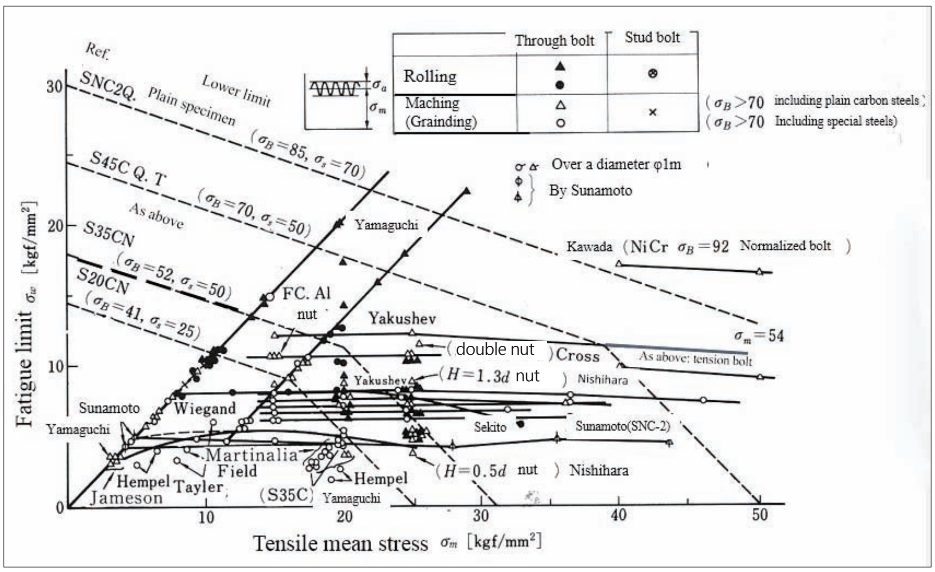


Fig.5.1 Effect of mean stress on fatigue limit of bolts

The effect of mean stress on the tensile fatigue limit of bolt is shown in ***Fig.5.1***[4]. So far as ***Fig.5.1*** is concerned, the mean stress below about 40kgf/mm^2 has little effect on the fatigue limit. However, some work reports that the mean stress has little effect on bolts with low tensile strength but shows its effect on bolts whose tensile strength has been increased by heat treatment[5]. According to an experiment conducted by the authors, the fatigue limit of bolts with a tensile strength σ_B=110 kgf/mm^2 (quenched and tempered structure) is decreased by about 20% when the mean stress σ_m is increased from 18 to 56 kgf/mm^2[2(6)].

Figure 5.2 shows the relation between the tensile strength and the tensile fatigue limit of bolts[4]. The tensile fatigue limit varies considerably but the tensile fatigue limit increases, although only slightly, with increasing tensile strength. These variations in the fatigue limit of bolts are attributable to the transmission of force due to contact between the bolt threads and the nut threads.

Figure 5.3 shows the size effect on the tensile fatigue limit of bolts[4]. The tensile fatigue limit decreases with increasing nominal bolt diameter, but the tensile fatigue limit varies considerably (see ***Figs.5.1*** and ***5.2***, respectively). From ***Fig. 5.3***, the tensile fatigue limit of a bolt 3 inches in nominal diameter will be only 2.5~3.0 kgf/mm^2 if a conservative value is desired. This value is very low compared with the tensile strength. The shape

Fatigue strength of conventional bolts

Only a little work has been directed towards bolt fatigue. The tensile fatigue limit of representative steel bolts are shown in ***Table 5.1***[1]. According to the definition of fatigue limit, the fatigue limit means a limiting stress below which an infinite number of stress cycles can be applied. In the case of partially tensile pulsating fatigue, the fatigue limit should be expressed in terms of the sum of stress amplitude σ_a and mean stress σ_m. In the case of bolts, however, the effect of mean stress is comparatively small, as described later (see ***Fig. 5.1***) and the mean stress is not always constant. Accordingly, a comparison in terms of stress amplitude σ_a only will facilitate understanding. In this paper, therefore, the fatigue limit is expressed in terms of stress amplitude only unless otherwise specified. From ***Table 5.1***, the tensile fatigue limit of steel bolts of normal diameters is 5~6kgf/mm^2 but decreases with increasing nominal diameter. This decrease is called the "size effect", which is particularly significant in the case of fatigue failure. However, as the size effect for fatigue failure of steel structures is only 10~15% in terms of decrease in fatigue limit[2], it is considered that the decrease in fatigue limit of a bolt is very remarkable. The probable reason is localized loading between the bolt threads and the nut threads due to low machining accuracy, which is one of the factors governing the fatigue strength of bolts as described later (ref. 7th report)[3].

factor (stress concentration factor) α of a bolt is about 4[7]. The notch factor β is expressed as the fatigue limit of a plain specimen, σwo/fatigue limit of a notched specimen,σw. Normally, the stress concentration factor α of a notched specimen is nearly equal to β when this factor is about 2. If α is greater than 2, α becomes greater than β. In the case of bolts, however, the notch factor β becomes 8~10 and β becomes greater than α even if the fatigue limit of a plain specimen with a tensile stress σ_B=100 kgf/mm^2 is assumed to be σwo~σ_B/2=50 kgf/mm^2, because the fatigue limit, σw, of a bolt is 5~6kgf/mm^2. This indicates that the fatigue limit of bolts is far lower than that of conventional notched specimens. Conversely, improvement of the fatigue limit is far more difficult for bolts than for normal notched specimens. Therefore, the designers should not calculate the value based on maximum stress on the notched bottom for anti-fatigue design.

Measures taken so far for the improvement of fatigue strength of fastening screws and their effects

3.1 Cause of low fatigue strength of bolts

There are several causes of low fatigue strength of bolts. The first cause is uneven load sharing[8] among the threads of a bolt (see *Table 5.2* and *Fig. 5.4*). In the case of a bolt with eight threads for engagement with the nut, about one-third of the total load (which is taken as 100%) is applied to the first thread, as seen from *Table 5.2*. The loads applied to the second and ensuing threads decrease sharply. The ratio of the load applied to the fourth and ensuing threads in particular less than 10%. The same also applies to the bolts with six or ten threads. It is estimated that this type of load is applied to nearly half the height of each thread in a concentrated manner.

Table 5.2 Percentage load distribution to screw threads (Total:100%)

No. of thread	P_1	P_2	P_3	P_4	P_5	P_6	P_7	P_8	P_9	P_{10}
6	33.7	22.9	15.8	11.4	8.7	7.5				
8	33.3	22.3	15.0	10.2	7.0	5.0	3.9	3.3		
10	33.1	22.2	14.9	10.0	6.7	4.6	3.1	2.3	1.6	1.5

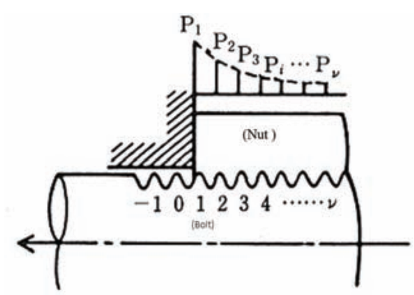


Figure 5.4 Load distribution to screw threads

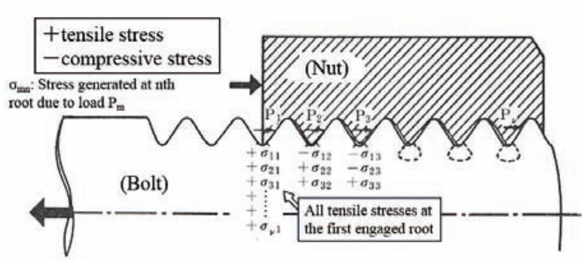


Fig. 5.5 Stresses generated at the roots of bolt threads

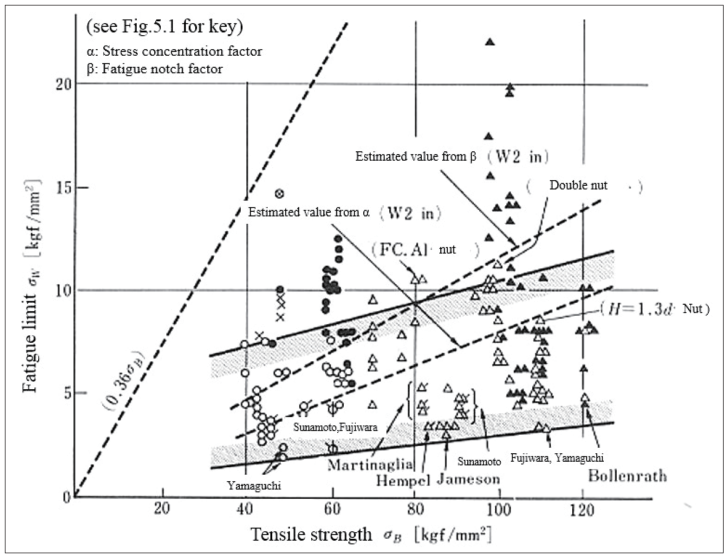


Fig.5.2 Relation between tensile strength and tensile fatigue

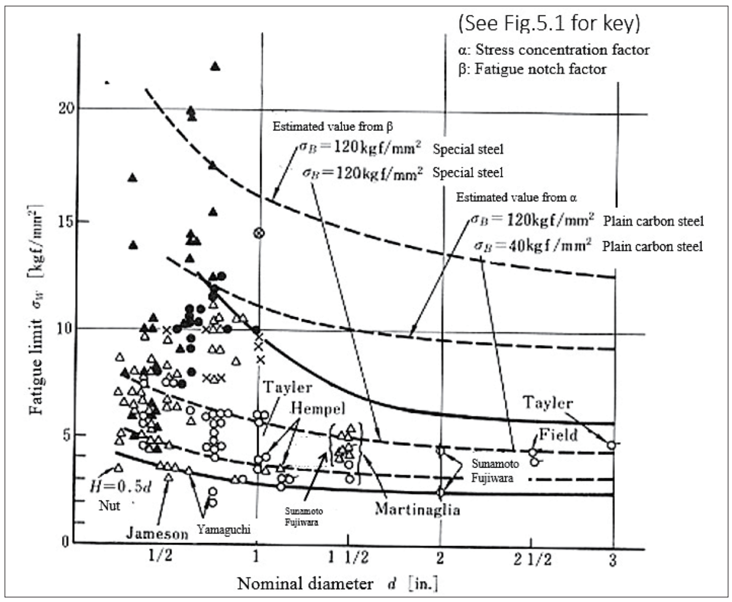


Fig. 5.3 Size effect on fatigue limit of bolts

On the basis of the load-sharing condition shown in *Fig.5.4*, the stress developed at each thread root is explained by reference to *Fig.5.5*. The loads to be shared by the threads are represented by P_1, P_2, P_3...P_ν, beginning with the first thread in engagement with the nut thread. The stresses developed at the roots of the first thread, second thread...and νth thread by the load P_1 are represented by +σ11, -σ12, -σ13,...-σ1ν, respectively. Similarly, the stress developed at the root of the first , second νth thread by the load P_1 are represented by $+\sigma_{21}$, $+\sigma_{22}$, $-\sigma_{23}$,... $-\sigma_{2\nu}$, respectively. In general, the stress developed at the root of the nth thread by the load Pm is σmn. Here, '+' indicates the tensile stress, while '-' indicates the compressive stress. Accordingly, the stress induced when m≧n is a tensile stress, but a compressive stress is created when m<n. The stress developed in a single bolt is the sum of the stress developed at the roots of all the threads. The stress induced at the roots of the second and ensuing threads under the load-sharing condition as shown in *Fig.5.5* are decreased as the compressive and tensile stresses cancel each other to some extent. However, only the tensile stresses are induced at the root of the first thread at the end section of nut. It is therefore apparent that the total stress developed at this root becomes the largest. The

above description is substantiated by the results of experiments conducted by Seike et al[9] on the stress concentration in bolts (see ***Fig.5.6***). In other words, the stress concentration factor α at the root of the first bolt thread at the end section of the nut is 4.5, which is close to the value (α=3.86) obtained by the photoelastic test[8]. It is therefore, easily understood that bolt failure mostly occurs at the root of the first bolt thread at the end of engagement with nut. In general, bolt failure occurs at three points, i.e. the end face of the nut, incomplete threads, and the underhead fillet. The probabilities that failure occurs at these points are said to be 65, 25, and 10%[7], respectively. So far as the fatigue failure of clamp bolts is concerned, nearly 100% of failures occur at the root of the first bolt thread at the end section of the nut unless special measures are taken for the improvement of fatigue strength. As is apparent from the examples of failure of fastening screws described in Second report, all failures, except one example (***Fig.2.20***), occurred at the root of the first bolt thread at the end section of the nut.

Uneven load sharing has been cited as a cause of low fatigue strength of bolts. As the bolt is a kind of notched material and the external force is transmitted through contact between the bolt threads and the nut threads, a high stress concentration factor and localized loading can also be cited as causes of low fatigue strength of bolts. These causes are explained in more detail in later.

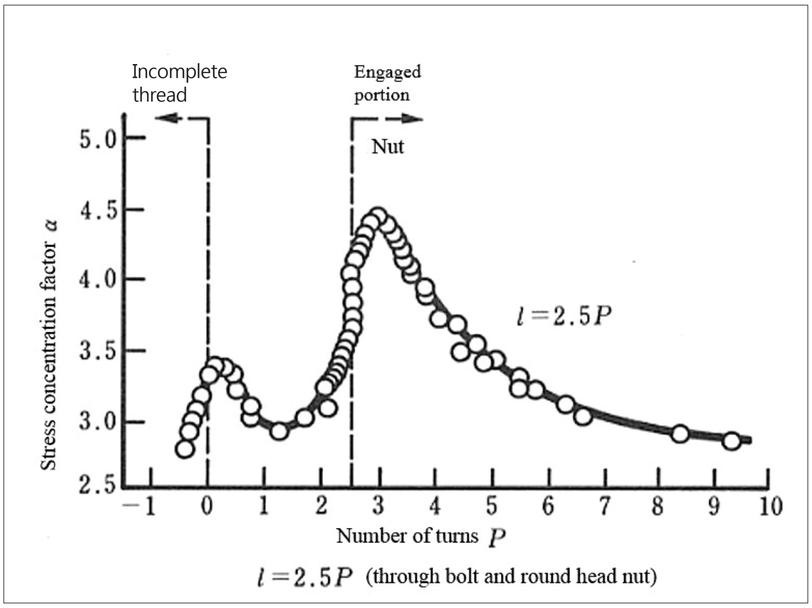


Fig. 5.6 Mutual interference of stress concentration at root of bolt

The measures taken so far for improvement of the fatigue strength of fastening screws are roughly divided into measures for nuts and measures for bolts. The measures are described below.

Measures for nuts

Figure 5.7 shows the transmission of force from the bolt to the nut[7]. In general, complicated stress concentration can be quantitatively determined by likening it to the flow of water. ***Figure 5.7(a)*** shows a conventional type of fastening with a bolt and nut. From ***Table 5.2*** and ***Fig.s 5.2~5.7(a)***, it is seen that the stress is concentrated at the boundary between the bolt and the nut. To improve the fatigue strength of a bolt, therefore, the stress concentration at the nut end should be decreased. One such measure is to change the flow of stress by providing a weir at the nut side, an example of which is shown in ***Fig. 5.7(b)***. That is, the flow of stress to the end of the nut (the end of the bolt threads) is changed by providing an annular

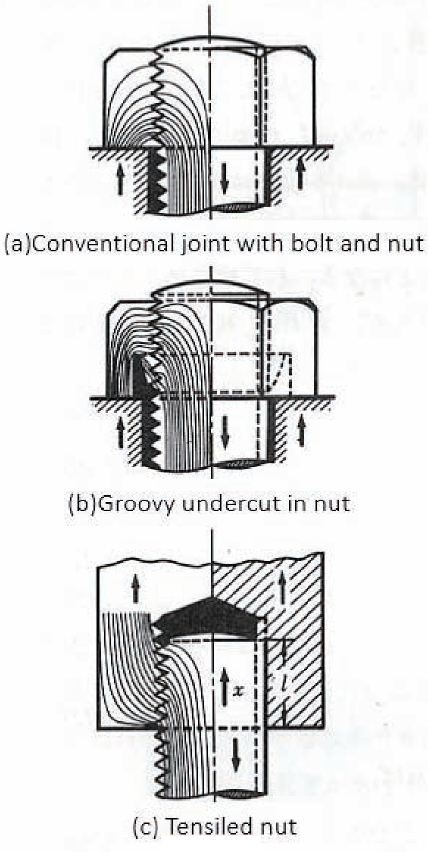


Fig. 5.7 Stress flow from bolt to nut

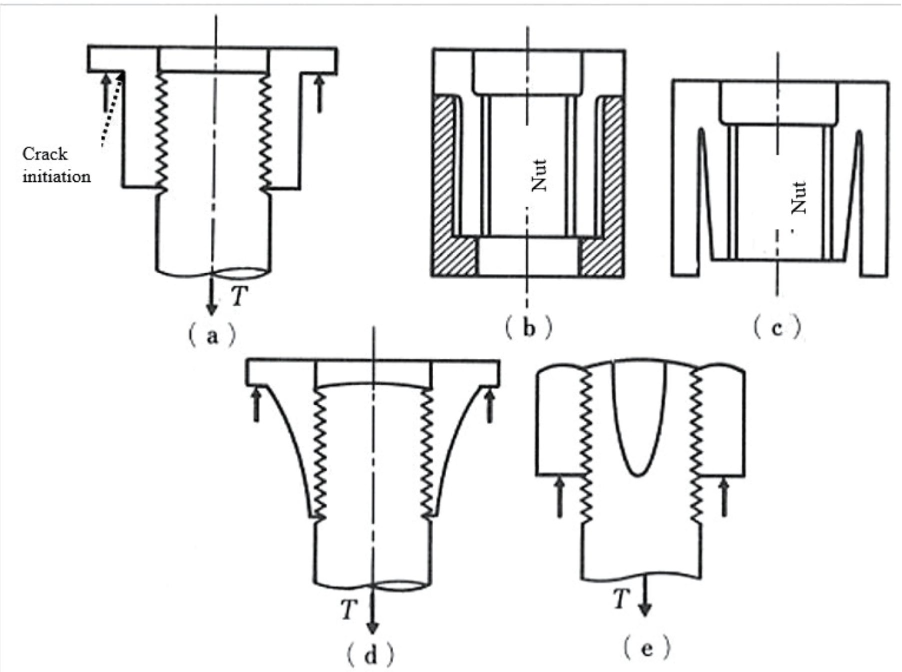


Fig. 5.8 Conventional methods for improving the fatigue strength for nut

groove at the nut end. The most desirable method is to pull the nut as shown in ***Fig.5.7(c)***. The smooth flow of force from the bolt to the nut can be ensured by pulling the nut. The bolt is elongated but the nut is compressed under a tensile force. As the 'strain' is concentrated at the nut end due to the difference in deformation between the bolt and nut, the fatigue strength of the bolt is decreased. If the strain concentration is decreased, the fatigue strength of the bolt will be considerably improved. It seems that the idea of 'nut pulling' is based on this idea.

Figure 5.8 shows examples of the measures proposed for nuts so far[7,10]. Strictly speaking, the example shown in ***Fig.5.8(e)*** is not a measure for nuts. However, the measure can be included in the measures for nuts as it is based on the same idea as other measures. It will be easily understood that the measures shown in ***Fig.5.8(a),(b),(c)*** and ***(d)*** are based on the same measures. Although the measure shown in ***Fig.5.8(e)*** is a measure for bolts, the measure aims at ensuring equal elongation of the bolt at the point where it engages with the nut.

However, the measures described above have not been put into practice for the following reasons. With the measure shown in ***Fig. 5.8(a)***, for example, a fatigue test had been conducted but the fatigue strength could not be improved as expected. Depending on the individual case, fatigue failure due to shear occurred from the flange-shaped root of the nut. The improvement in fatigue strength by this method is only 1-2kgf/mm^2. One reason is as follows. Except in special cases, fatigue failure initiates at the bolt. If a measure is taken for the nut, therefore, the fatigue strength cannot be improved as expected. As there are several factors, other than uneven load sharing, which govern the fatigue strength of bolts, the measures proposed for nuts are effective for only one of the factors. Moreover, the bolts shown in ***Fig.5.8(b)*** and ***(c)*** are so complicated in shape that they are unsuitable for mass production and their application becomes somewhat difficult.

Measures for bolts

There are only a few examples of measures taken for improvement of the fatigue strength of bolts. One such example is shown in ***Fig.5.9***[7]. The diameter of the body of the bolt is smaller than the nominal diameter. This type of bolt is called a bolt with reduced shank. If the diameter of the body is smaller than the nominal diameter over the entire bolt length, the bolt may be set out of center with respect to the axis of the bolt hole. In this condition, a bending load may be applied to the bolt in addition to a tensile load. If a bending load is applied to the bolt, the fatigue strength of the bolt is decreased[1]. And therefore, it becomes necessary to prevent this decrease in strength. The reason why the measure shown in ***Fig.5.9(d)*** is more desirable than that shown in ***Fig. 5.9(c)*** is that the stress concentration at the incomplete thread and the underhead fillet can be decreased.

The effect of the bolt with a reduced shank is described below. Normally, bolts are used in a tightened condition. To use this type of bolt under the most favorable condition from the standpoint of fatigue strength, it is necessary to understand the relation between

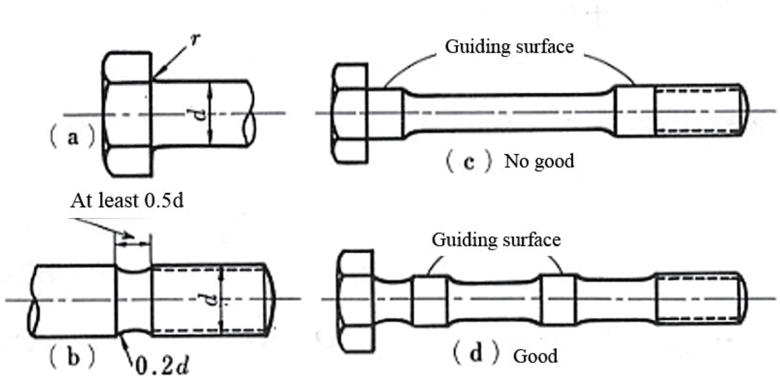


Fig. 5.9 Conventional methods for improving
The fatigue strength of bolts [(d) better than (c)]

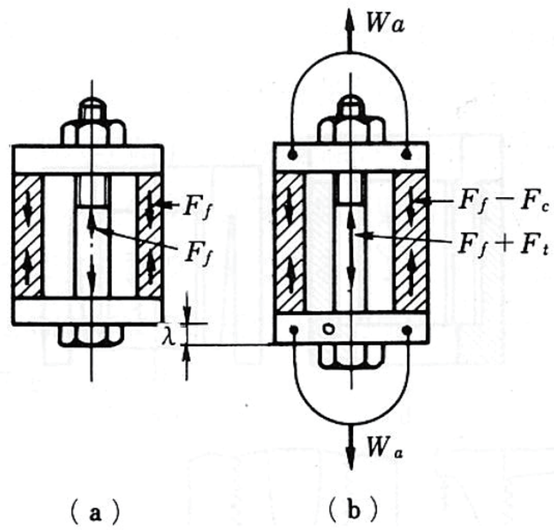


Fig. 5.10 Balancing relation between external force
and internal forces applied to the screw-fastened member
(Wa external force; Ff, Ft, Fc, internal forces)

the external force applied to fastening screw and the internal forces. The internal force means the load to be shared by the bolt and the fastened part, depending on the applied external force, but does not mean the stress.

Figure 5.10 shows the external force applied to the bolt, nut and fastened part and the internal forces[1]. ***Figure 5.10(a)*** shows the condition in which the tensile force F_f induced in the bolt shank is balanced with the compressed force F_f induced in the fastened part when the bolt is tightened. Assume that an external force W_a is applied to the screw fastening. In this case, a tensile internal force Ft is applied to the bolt shank and compressive force F_c is lost from the fastened part as shown in ***Fig.5.10(b)***. Under this condition, the fastening length is increased by λ. Let K_t be the tensile spring constant of the fastening screw (load per unit elongation) and Kc the compressive spring constant of the fastened part (load per unit contraction). In this case, K_t and K_c are expressed as shown below:

$$F_t = K_t \cdot \lambda \qquad (5.1)$$

$$F_c = K_c \cdot \lambda \qquad (5.2)$$

From the balance of forces

$$W_a = (F_f + F_t) - (F_f - F_c) = (F_t + F_c) \qquad (5.3)$$

Substituting equations (5.1) and (5.2) into equation (5.3):

$$W_a=(K_t+K_c)\cdot\lambda$$

$$\therefore \lambda=\frac{1}{K_t+K_c}W_a \qquad (5.4)$$

Substituting equation (5.4) into equations (5.1) and (5.2):

$$F_t=\frac{K_t}{K_t+K_c}W_a \qquad F_c=\frac{K_c}{K_t+K_c}W_a \qquad (5.5)$$

The ratio of the tensile force Ft added to the bolt by the external force W_a to the external force W_a applied to the fastening screw is expressed in terms of the internal force coefficient φ of the bolt as shown below:

$$\varphi=F_t/W_a=\frac{K_t}{K_t+K_c} \qquad (5.6)$$

F_t and F_c in equation (5.5) can be expressed as shown by using φ:

$$\left.\begin{aligned} F_t&=\varphi\cdot W_a \\ F_c&=(1-\varphi)\cdot W_a \end{aligned}\right\} \qquad (5.7)$$

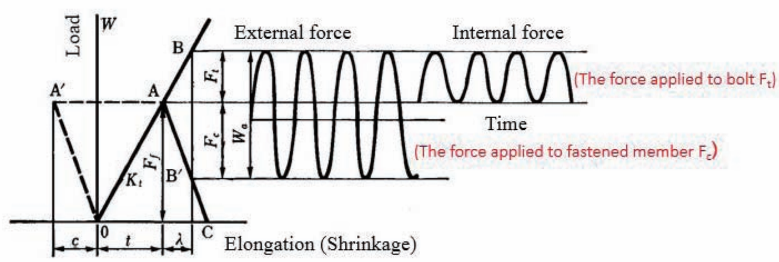
The relation shown above is illustrated in ***Fig.5.11.*** The relation between the force applied to the bolt and the fastened part and the elongation of the bolt (contraction of the fastened part) in the case where an external force is applied to the fastening screw is shown in ***Fig. 5.11*** with load as ordinate and elongation of bolt (contraction of fastened part) as abscissa[1,11,12]. This diagram is always introduced when the force applied to the bolt, particularly the fatigue strength of the bolt, is discussed. As is not easy to understand this diagram in spite of its simplicity, a brief explanation is given below.

The condition in which a tensile force Ff is being applied to the bolt is indicated by the point A. The relation between the force applied to the bolt and the elongation of the bolt is indicated by the line OAB. However, the line AB'C indicates the relation between the force applied to the fastened part and the contraction of this part. When an external force Wa is applied during the application of a tensile force F_f to the bolt, the elongation of the bolt, λ by the force W_a is equal to the elongation of the fastened part, δ, so far as the fastening between the bolt and the part is tight. Accordingly, both the bolt and the fastened part are elongated until the length of the line BB' becomes equal to the external force W_a.

The forces to be shared by the bolt and the fastened part when the external force Wa is applied can be easily explained by the following from the start of fastening. The force to be shared by the bolt, F_t, and the force to be shared by the fastened part, F_c, are as shown by the stress wave form in ***Fig.5.11.*** If the external force W_a is increased until $F_c=F_f$, the force by which the bolt is fastening the part becomes zero, and therefore the screw fastening becomes loose. If the screw fastening becomes loose, the whole external force W_a is applied to the bolt. To prevent loosening of the screw fastening by the application of the external force W_a, the following equation must be satisfied:

$$F_f\geq K_c\cdot W_a/(K_t+K_c) \qquad (5.8)$$

Conversely, the screw fastening becomes loose if the



K_f Spring constant of bolt, W_a External force less than tightening force
K_c Spring constant of fastened member, F_f Tightening force to bolt

Internal force becomes about one-fifth of external force W_a

$$F_t=\frac{K_t}{K_t+K_c}W_a \quad \text{(Internal force due to external varying force } W_a\text{)}$$

Maximum force applied to bolt

$$F_{tmax}=F_f+\frac{K_t}{K_t+K_c}W_a$$

Fig. 5.11 Relation between the force and shrinkage in the bolt and fastened member

external force W_a expressed by the following equation is applied to the bolt being tightened by the force Ff:

$$W_a\geq\left(1+\frac{K_t}{K_c}\right)\cdot F_f \qquad (5.9)$$

When the bolt is tightened to such an extent that it is not loosened by the external force W_a, the amplitude of the load to be borne by the bolt decreases with decreasing spring constant K_t of the bolt so far as the external force Wa and the compressive spring constant of the fastened part, K_c, are constant. In other words, the force Ft to be borne by the bolt which is more likely to elongate within the elastic region is smaller with respect to the same external force W_a. For example, K_t for a long or narrow bolt is smaller than a short or thick bolt.

In the case of a fastening screw to which an external force W_a with constant

☆In the case of effectively fastened bolt :
External force is applied into
not only "bolt" but also "fastened member"

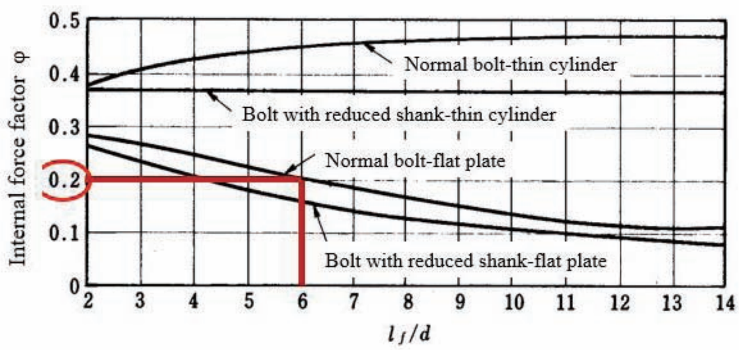
★In the case of loosened bolt :
All of external force is applied to only bolt ⇒
Instantaneously broken

∴Anti-loosening screw becomes important against fatigue

Anti-loosening screws: e.g. "Anti-loosening nut (ref. ninth report)

Fig. 5.13 Importance of the not-loosening fastened screw

amplitude is applied, the fluctuating load to be borne by the bolt decreases with decreasing spring constant of the bolt and increasing spring constant of the fastened part. This is very advantageous from the standpoint of fatigue.



(l_f : Tightening length · d : nominal diameter)

Ex.) in the case of φ=0.2

Considering $F^t = \varphi W_a$ · F_t = 6kgf/mm² (fatigue limit of conventional bolt), this bolt can be endurable to external force, W_a = 6 / 0.2 = 30kgf/mm². That is, the conventional bolt actually shows the fatigue limit of 30kgf/mm², if the fastened screw does not appear loosening.

Fig. 5.12 Quick calculation diagram of internal force factor for a typical fastened screw (for Kt = Kc)

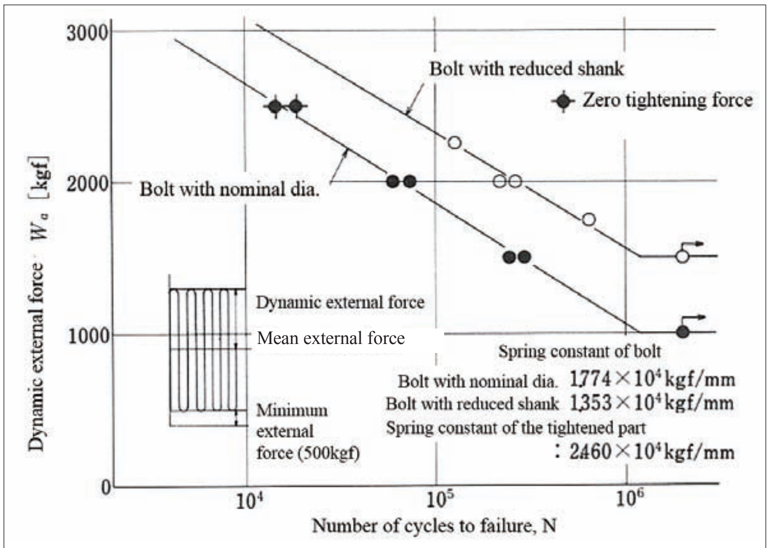


Fig. 5.14 S-N curve for fastening screws
(comparison between a bolt with reduced shank and a bolt with nominal dia.

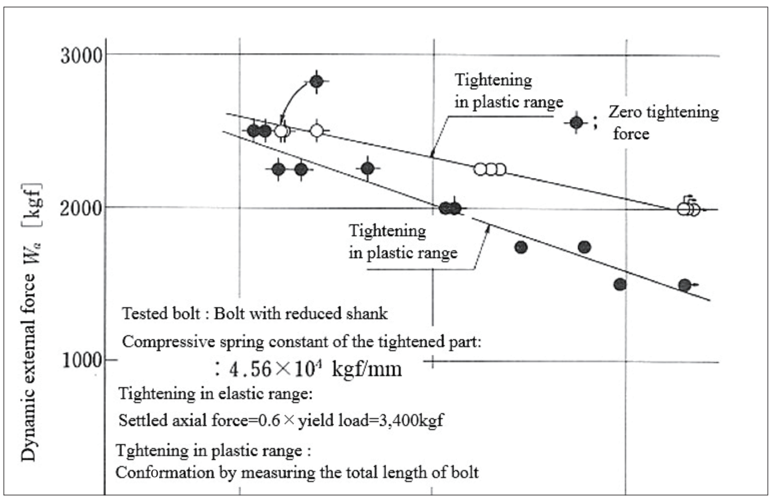


Fig. 5.15 S-N curves for fastening screws (for fastening in the plastic range)

However, the maximum stress F_{tmax} applied to the bolt becomes below:

$$F_{tmax}=F_f+\frac{K_t}{K_t+K_c}Wa \qquad (5.10)$$

As the fatigue strength of bolt is mainly effected by the amplitude of stress, it is very important to prevent loosening of the screw fastening by the application of the external force W_a (see ***Fig.5.13***[13]).

Figure 5.12 shows the relation between the internal force coefficient φ and lf/d (fastening length/outside diameter of thread)[14]. When conventional bolts are used, the internal force coefficient φ is 0.1-0.3. As described previously[1], the fatigue limit of the bolt itself is 5-6kgf/mm². Tolerable stress variations corresponding to variations in external force are 15-60kgf/mm² according to the equation (5.5). Accordingly, there may be cases where the fatigue limit of a fastening screw becomes higher than that of a welded structure.

On the basis of the results described above, this type of bolt is very effective as it does not receive the whole external force. Accordingly, the bolt has been used as a connecting rod bolt in engine casings and the like.

The fatigue strength of the bolt is hardly affected by the mean stress (see ***Fig.5.3***) and the bolt does not become loose (the external force W_a does not satisfy equation (5.9) at all times). However, the bolt cannot be used where variations of external force W_a are directly applied to the bolt (for example, in a piston rod screw and hanging bolt).

Fatigue strength of a fastening screw

Figure 5.14 shows the S-N curve of a fastening screw with dynamic external force W_a as the ordinate and the number of cycles as the abscissa(15). The frequency is 1,800 cycles/min and the spring constants of the bolt with reduced shank and the bolt with nominal diameter body are 1.353×104 and 1.774×104 kgf/mm, respectively. The compressive spring constant of the fastened part is calculated as 2.460×104 kgf/mm. The internal force coefficients φ of the bolt with reduced shank and the bolt with nominal diameter body are calculated as 0.355 and 0.419, respectively, from equation ***(5.5)***. As shown in ***Fig.5.15***, the fatigue strength of the part fastened with the bolt with reduced shank (with smaller internal force coefficient φ) is larger than that of the part fastened with the bolt with nominal diameter body. All failures occurred at the bolt thread at the end section of the nut. For these reasons, extensive studies must have been made on the prevention of looseness to improve the fatigue strength of the bolt.

The fatigue strength in the case of fastening with a bolt with reduced shank in the elastic range is shown in ***Fig.5.15*** compared with that in the case of fastening with the same bolt in the plastic range[15]. The test was conducted by the same method as that shown in ***Fig.5.15***, except for the different type of load cell used. Accordingly, the compressive spring constant of the fastened part is calculated as 4.56×104 kgf/mm. In the case of fastening in the elastic range, the axial tension is set at '0.6×yield load=3,400kgf'. In the case of fastening in the plastic range, the spring constant was checked by measuring the overall length of the bolt. Fastening in the plastic range improves the fatigue strength by 43% (at 2×106 cycles), i.e. from 1,400 kgf to 2,000 kgf in terms of the dynamic external force. The probable reasons are that the compressive residual stress is developed by cold working of the thread root of the bolt and the loads to be shared by the bolt threads as shown in ***Table 5.2*** and ***Fig.5.4*** are changed. Yet another reason is that localized loading due to very small errors in pitch processing between the bolt and nut is decreased by plastic deformation of the bolt threads (see 6th report).

The results of a study on the effect of prestressing which was conducted with the bolt made of SNCM630 (root diameter:φ25mm) are described below[16]. In this study, the fatigue strength of the bolt was examined by applying a mean stress σm of 18 kgf/mm^2 instead of the bolt tightening force. As a result, the fatigue limit of 6.0 kgf/mm^2 was increased to 9.0 kgf/mm2 by the application of a prestress of 43 kgf/mm^2. In other words, the fatigue limit of the bolt is increased by 50% by prestressing (see ***Fig.5.16***). The difference in results shown in ***Fig.5.15*** and ***5.16*** is attributed to the difference in the test methods employed. In the former method, the bolt was tightened under a load higher than the yield point and the fatigue test of the fastened screw was conducted under this condition. In the later method, however, the fatigue test was conducted by applying the specified mean stress after static yielding of the bolt in tension and load removal. Although the mean stress applied to the bolt in the former method differs from that in the latter method, both methods are based on what is nearly the same concept. G.H.Yunker pointed out that the fatigue strength was improved by tightening the bolt under a load higher than the yield point[16]. He attributed this improvement to the effect of compressive residual stress induced by cold working of bolt threads. The authors are of the opinion that the uniform load sharing among bolt threads and the decrease in localized one-side loading are also responsible for the improvement for the following reason. The effect of residual stress on fatigue strength is frequently likened to the effect of mean stress. Compared with the simple fatigue strength of a plain specimen, the fatigue strength of a bolt is little affected by the mean stress. Accordingly, an increase in the fatigue limit of a bolt by 50% due to the effect of compressive residual stress only is considered questionable.

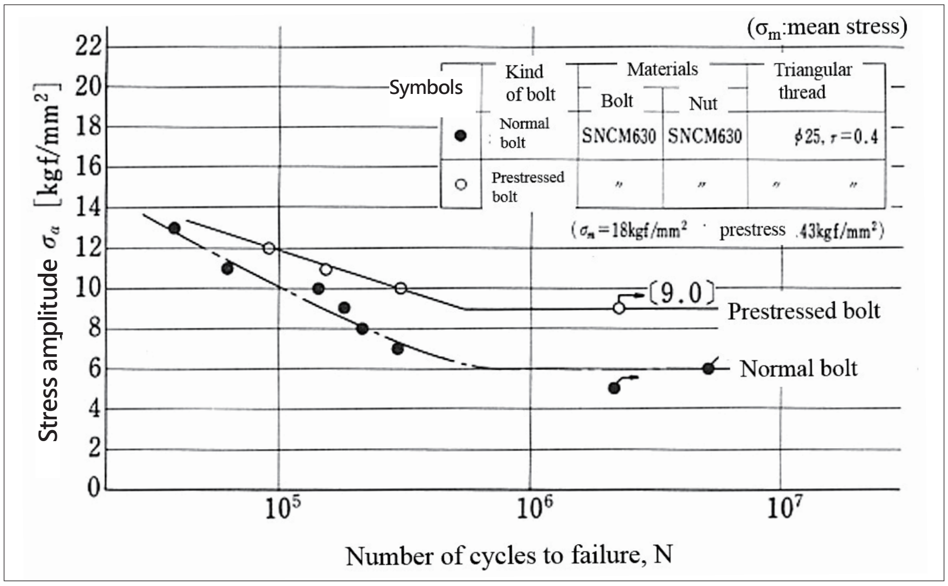


Fig. 5.16 S-N curves of prestressed bolt

Conclusions

The main results described above in this report are concluded below:

Size effect of bolt; the fatigue limit of steel bolts of normal diameter is 5~6kgf/mm^2 but remarkably decrease with increasing nominal diameter. In addition, though the stress concentration factor α is larger than the notch factor β in the case of normal fatigue, the notch factor β becomes greater than the stress concentration factor α in the case of bolts.

Effect of mean stress: the fatigue strength of bolts is hardly affected by mean stress.

Effect of tensile strength of materials: the fatigue limit of bolts does not necessarily increase with increasing strength of materials.

Cause of low fatigue strength of bolts: one of the main reasons is due to uneven load sharing among the threads of a bolt. That is, about one-third of the total load is applied to the first thread and decreases remarkably for the latter threads.

The fatigue strength of the part fastened with the bolt with reduced shank is larger than that of the part fastened with the bolt with nominal diameter body.

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