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Loosening of Bolted Joints by Small Plastic Deformations

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Loosening of Bolted Joints by Small Plastic Deformations

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Formulas are derived for several cases of loosening of bolted joints caused by stresses in excess of elastic limits and accompanying permanent deformations. It is shown that loosening occurs during the initial application of an external force, but that subsequent repetitions of the same force cause no further loosening. External impact loads, which require a definite energy absorption of the joint, result in loosening upon both the initial load application and later repetitions. The effects of initial tightening, gaskets, and spring washers are investigated. Loosening under moderate loads may be completely avoided if the initial bolt tension does not exceed a certain maximum value. Loosening is unavoidable under extremely high loads, but the amount of loosening can be decreased by use of higher initial tensions. Illustrative experimental data are reported.

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Loosening of Bolted Joints by Small Plastic Deformations

O. A. PRINGLE

An important requirement for the satisfactory performance of a bolted joint is the retention of a suitable clamping force. Decrease or loss of clamping force may result in such detrimental effects as slip, misalignment, slack, leakage, and reduced fatigue strength. Unfortunately, loosening can occur in a variety of ways, among which are rotation of the nut; crushing of surface irregularities; extrusion of dirt, paint, plating, or scale; creep; and plastic deformation.

This paper will examine in detail several cases in which loosening is caused by plastic elongation of the bolt. Such plastic deformation will occur whenever the stress in the bolt exceeds the elastic limit. Obviously this can be prevented by keeping peak stresses below the elastic limit at all times, however such a solution is not always practical or economical. For efficient use of bolts, clamping forces should be as high as possible, and it is common practice to tighten bolts initially to stresses near or even above the elastic limit. With an initial tightening stress near the elastic limit, peak stress due to imposition of an external load may easily cause this limit to be exceeded. Furthermore, it may be anticipated that requirements for bolted-joint applications of the future will be increasingly severe, necessitating minimum factors of safety while demanding maximum reliability. In view of this, a need is indicated for methods by which the amount of loosening may be predicted quantitatively, which in turn should permit rational determination of the best means of mitigating loosening in a particular application.

General discussions of bolted joint design considerations have been presented by Stewart (1)[1] and Almen (2). Some of the relations among the various forces and deflections in bolted joints were explained by Dolan (3). Studies of loosening due to nut rotation have been made by Goodier and Sweeney (4), and by Sauer, Lemmon, and Lynn (5). Roberts (6) investigated the effects of loosening on leakage. One of the most serious effects of loosening, loss of fatigue strength, has been treated by Steward (1), Almen (2), Dolan (3), Boomsma (7), and Erker (8).

[1] Underlined numbers in parentheses designate References at the end of the paper.

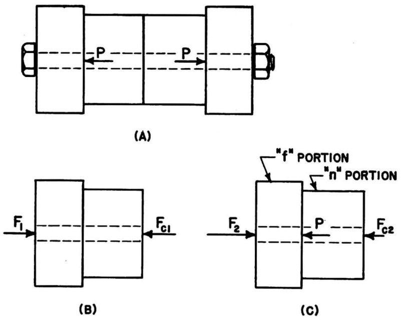

Fig. 1 Bolted joint loaded in tension

LOOSENING BY EXTERNAL FORCE

A bolted joint loaded in tension may be visualized as shown in Fig.1(a). The bolt is tightened initially to tension F_1, creating clamping force F_{c1} at the mating surfaces. When the external force P is applied, the bolt tension increases from F_1 to F_2 while the clamping force decreases from F_{c1} to F_{c2}. The free-body diagram is shown in Fig.1(c), and the equilibrium equation is $F_2 = P + F_{c2}$.

As the bolt load increases from F_1 to F_2, the bolt elongates an amount ΔX_b. At the same time, the compressive force on the f-portion of the clamped parts increases from F_1 to F_2, resulting in a contraction of ΔX_{cf}. Also, the compressive force on the n-portion decreases from F_{c1} to F_{c2}, hence it elongates an amount ΔX_{cn}. The total springback of the clamped parts is the difference between ΔX_{cn} and ΔX_{cf}. As long as the mating surfaces do not separate, the bolt elongation must equal the clamped parts springback, thus the necessary deflection relation is $\Delta X_b = \Delta X_{cn} - \Delta X_{cf}$.

For a particular case, the foregoing relationships may be shown conveniently on a force versus deflection diagram as in Fig.2. It is assumed that the elastic portion of the bolt force-deflection curve is a straight line of slope k_{be} and the plastic portion is a straight line of slope k_{bp}. The plastic straight line is an acceptable approximation as it need be fitted to the actual curve over only the small range to be included in the loosening calculations. It is also assumed that the force versus deflection

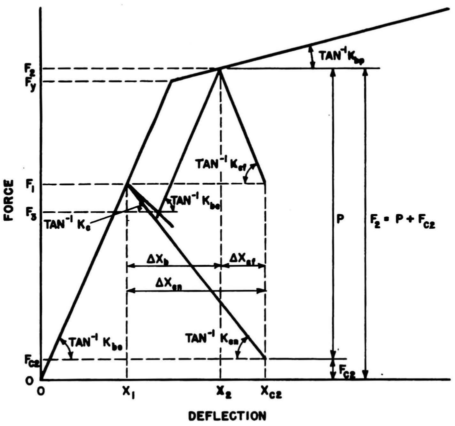

Fig. 2 Force-deflection diagram for bolted joint with initial tension above the elastic limit

curve for the total clamped parts is a straight line of slope k_c, the curve for the f-portion is a straight line of slope k_{cf}, and the curve for the n-portion is a straight line of slope k_{cn}. For the particular case shown, the initial tightening load F_1 is less than the elastic-limit load F_y, the maximum bolt load F_2 is greater than F_y, and the clamping force F_{c2} is greater than zero so the mating surfaces do not separate.

The bolt is tightened initially to point F_1, X_1. As the external force P is applied, the bolt load increases to F_2, and the corresponding bolt elongation ΔX_b is indicated. The load on the n-portion decreases from F_{c1} to F_{c2}, and its elongation ΔX_{cn} is the X projection of its line between F_{c1} and F_{c2}. Meanwhile, the load on the f-portion has increased from F_1 to F_2, and the contraction ΔS_{cf} of this portion is the X projection of its line between F_2 and F_1 as shown. The construction satisfies the previously mentioned equilibrium equation and deflection relation, therefore the maximum bolt load F_2 is correctly located for a given value of P. If P is removed, the bolt springs back from point F_2, X_2 along an elastic recovery line of slope k_{be}. The final state of the clamped parts is along a line of slope k_c drawn from the initial state at F_1, X_1. For equilibrium, the final bolt tension F_3 must equal the final clamping force F_{c3}, therefore these forces are found at the intersection of the above lines. As a result of the first application of external force, loosening has occurred, and the bolt tension has decreased from F_1 to F_3.

Formulas for the case illustrated in Fig.2 will be derived. After expressing the deflections in terms of forces and slopes, the deflection relation becomes

$$\frac{F_2 - F_y}{k_{bp}} + \frac{F_y - F_1}{k_{be}} = \frac{F_1 - F_{c2}}{k_{cn}} - \frac{F_2 - F_1}{k_{cf}}$$

Substituting F_{c2} from the equilibrium equation and solving for F_2 results in

$$F_2 = \left(\frac{1 + \frac{k_c}{k_{be}}}{1 + \frac{k_c}{k_{bp}}}\right) F_1 + \left(\frac{\frac{k_c}{k_{cn}}}{1 + \frac{k_c}{k_{bp}}}\right) P + \left(\frac{1 - \frac{k_{bp}}{k_{be}}}{1 + \frac{k_{bp}}{k_c}}\right) F_y \qquad (1)$$

The equations of the bolt recovery line and clamped parts force-deflection line are

$$F_2 - F = k_{be}(X_2 - X)$$

$$F_1 - F = -k_c(X_1 - X)$$

Solving for F_3 at the intersection of these lines, and expressing deflections in terms of forces and slopes, one obtains

$$F_3 = F_1 - \left(\frac{\frac{k_{be}}{k_{bp}} - 1}{\frac{k_{be}}{k_c} + 1}\right)(F_2 - F_y) \qquad (2)$$

Equations (1) and (2) pertain to the initial application of the external force. As a result of work-hardening, the elastic limit of the bolt is assumed to have then increased in magnitude to F_2. Upon a second application of external force, F_y in the equations should be made equal to the previously calculated F_2, and F_1 should be made equal to the previously calculated F_3. It will then be found that F_2 equals F_y. Since F_2 does not exceed F_y during the second load application, no plastic deformation of the bolt occurs, and consequently there is no loosening. Similarly, additional repetitions of the same external force will cause no further loosening.

Equations (1) and (2) do not apply if $F_2 < F_y$. For this particular case an expression for F_2 may be obtained by simply substituting $k_{bp} = k_{be}$ in equation (1), which in effect replaces the plastic portion of the bolt force-deflection curve by a continuation of the elastic portion. Then

$$F_2 = F_1 + \left(\frac{\frac{k_c}{k_{cn}}}{1 + \frac{k_c}{k_{be}}}\right) P \qquad (3)$$

Since F_2 does not exceed F_y in this case, there is no loosening, and $F_3 = F_1$.

For a given external force P, there is a maximum initial tension F_1 which should not be ex-

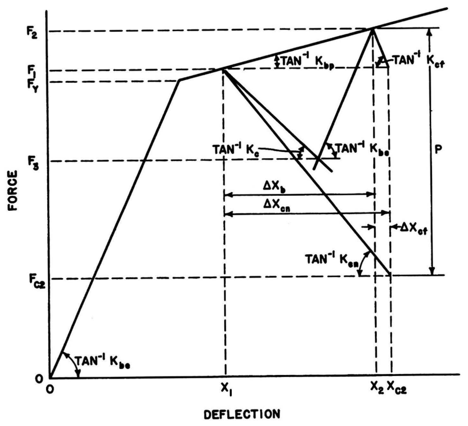

Fig. 3 Force-deflection diagram for bolted joint with initial tension above the elastic limit

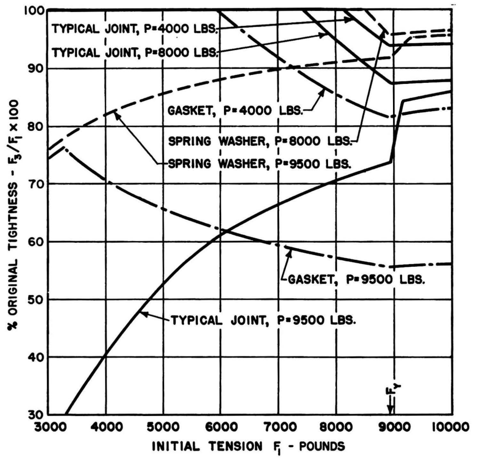

Fig. 4 Effects of initial tension, gaskets, and spring washers on loosening due to external force

ceeded during tightening if loosening is to be avoided. This maximum initial tension is reached when F_2 from equation (3) equals F_y, thus

$$F_{1\,max} = F_y - \left(\frac{\frac{k_c}{k_{cn}}}{1 + \frac{k_c}{k_{be}}}\right) P \qquad (4)$$

Fig.3 illustrates the force-deflection relationships for the case in which the initial tension exceeds the elastic limit. The formula for F_2 may be obtained by simply replacing k_{be} in equation (3) by k_{bp}; i.e., by changing the slope from that of the elastic portion to that of the plastic portion. The final bolt tension after loosening for the case of Fig.3 is found from the previously given equations of the bolt recovery line and the clamped parts force-deflection line. Substituting for deflections the appropriate equivalents in terms of forces and slopes and solving simultaneously, one obtains

$$F_3 = \left(\frac{1 + \frac{k_c}{k_{bp}}}{1 + \frac{k_c}{k_{be}}}\right) F_1 - \left(\frac{\frac{k_{be}}{k_{bp}} - 1}{\frac{k_{be}}{k_c} + 1}\right) F_2 \qquad (5)$$

Equations (1), (3) and (4) assume that $F_{c2} > 0$, so that the mating surfaces do not separate. To check this assumption, F_{c2} should be determined from the equilibrium equation. A negative value of F_{c2} is physically impossible and signifies that actually $F_{c2} = 0$ and $F_2 = P$. F_3 may then be obtained from equation (2) if $F_1 < F_y$ or equation (5) if $F_1 > F_y$.

If desired, the foregoing formulas may be modified to allow for a reduction in bolt elastic limit due to combined tensile and torsional shear stresses. Evidence seems to indicate, however, that most of the torque on the bolt due to tightening is relieved by radial rubbing of the thread surfaces due to transverse contractions and expansions.

The use of the foregoing equations will be illustrated by several sample calculations, the results of which are plotted in Fig.4. For the curves labeled "typical joint" the values assumed were $k_{be} = 1.35 \times 10^6$ lb./in., $k_{bp} = 0.225 \times 10^6$ lb/in., $k_{cf} = \infty$ (assuming negligible deflection of the f-portion), $k_{cn} = k_c = 6.0 \times 10^6$ lb/in., and $F_y = 8920$ lb. These values approximate a 3/8 in. - 16 UNC x 3 in. heat-treated steel bolt clamping rigid steel parts. As shown in Fig.4, for P = 4000 lb there is no loss of tightness if $F_1 \leq 8190$ lb. However, if $F_1 > 8190$ lb, loosening occurs. For the larger external load of 8000 lb loosening begins at a lower value of F_1 and is greater in amount. For the extremely large external force of 9500 lb loosening occurs at all values of F_1, however the fraction of initial tightness retained increases with larger values of F_1. Discontinuities in the curves occur at the elastic limit and also at points where the mating surfaces cease to separate. The curves show that favorable decreases in loosening may be realized at initial tensions above F_y. It may, however, be inadvisable to take advantage of this effect as the maximum bolt load F_2 may become excessive, resulting in insufficient factor of safety on ultimate strength. Also, excessive plastic flow in the threads may result in binding of the nut. In general, the curves show that if loosening is to be completely avoided for small or moderate values of external load, the initial tension must not exceed the maximum value as calculated by equation (4). On the other hand, loosening is unavoidable at extremely high values of external load, but the amount of loosening can be decreased by tightening to higher values of initial tension.

For the curves in Fig.4 labeled "gasket" the value of k_{cn} was reduced to 0.5×10^6 lb/in. to approximate the inclusion of a soft gasket between the mating surfaces of the joint. Values of other variables were unchanged. In general, the gasket increased loosening, with the exception that loosening decreased for the combination of the ultra-high external load and low values of initial tension.

For the curves in Fig.4 labeled "spring washer" the values assumed were $k_{cf} = 0.5 \times 10^6$ lb/in., $k_{cn} = 6.0 \times 10^6$ lb/in., and $k_c = 0.462 \times 10^6$ lb/in. These values approximate the addition to the typical joint of a spring washer under the bolt head or nut. Values of the other variables were unchanged. The location of the spring washer is such that it becomes in effect the f-portion of Fig.1(c), hence a low value of k_{cf} was used. It can be seen that the spring washer results in substantially decreased loosening in all instances.

It should be noted that some bolted joints may be subjected after fabrication to a test load much higher than the external load expected in service. Possible loosening due to the test load should not be overlooked. Also, it is obvious that lock-nuts or similar devices cannot be effective against the type of loosening under discussion.

LOOSENING BY EXTERNAL IMPACT

An external impact load, such as that caused by the collision of two moving bodies, will require the bolted joint to absorb a certain amount of energy. Formulas for loosening under impact load will be derived by equating the external impact energy W to the strain energy absorbed by the joint. The first formula to be derived will be for the case illustrated in Fig.2. The strain energy absorbed or released by each component of the joint will be equal to the area under the applicable portion of its force-deflection curve. For Fig.2 this results in

$$W = \left(\frac{F_y + F_1}{2}\right)(X_y - X_1) + \left(\frac{F_y + F_2}{2}\right)(X_2 - X_y) + \left(\frac{F_2 + F_1}{2}\right)(X_{c2} - X_2) - \left(\frac{F_1 + F_{c2}}{2}\right)(X_{c2} - X_1)$$

Substituting $F_{c2} = F_1 - k_{cn}(X_{c2} - X_1)$, replacing the deflections by their equivalents in terms of forces and slopes, and solving for F_2 results in

$$
\begin{aligned}
F_2 = {} & \left[\frac{\frac{k_{cn}}{k_{bp}}\left(1 - \frac{k_{bp}}{k_{be}}\right)}{1 + \frac{k_{cn}}{k_{cf}} + \frac{k_{cn}}{k_{bp}}}\right] F_y + \left[\frac{1 + \frac{k_{cn}}{k_{cf}} + \frac{k_{cn}}{k_{be}}}{1 + \frac{k_{cn}}{k_{cf}} + \frac{k_{cn}}{k_{bp}}}\right] F_1 \\
& + \Bigg\{ \left[\frac{\left(\frac{k_{cn}}{k_{bp}}\right)^2 \left(1 - \frac{k_{bp}}{k_{be}}\right)^2}{\left(1 + \frac{k_{cn}}{k_{cf}} + \frac{k_{cn}}{k_{bp}}\right)^2} + \frac{\left(1 - \frac{k_{bp}}{k_{be}}\right)\left(1 + \frac{k_{cn}}{k_{be}} - \frac{k_{cn}}{k_{bp}}\right)}{\left(1 + \frac{k_{bp}}{k_{cf}}\right)\left(1 + \frac{k_{cn}}{k_{cf}} + \frac{k_{cn}}{k_{bp}}\right)}\right] F_y^2 \\
& + \left[\frac{2\left(\frac{k_{cn}}{k_{bp}}\right)\left(1 - \frac{k_{bp}}{k_{be}}\right)\left(1 + \frac{k_{cn}}{k_{cf}} + \frac{k_{cn}}{k_{be}}\right)}{\left(1 + \frac{k_{cn}}{k_{cf}} + \frac{k_{cn}}{k_{bp}}\right)^2} - \frac{2\left(1 - \frac{k_{bp}}{k_{be}}\right)\left(1 + \frac{k_{cn}}{k_{cf}} + \frac{k_{cn}}{k_{be}}\right)}{\left(1 + \frac{k_{bp}}{k_{cf}}\right)\left(1 + \frac{k_{cn}}{k_{cf}} + \frac{k_{cn}}{k_{bp}}\right)}\right] F_1 F_y \\
& + \left[\frac{\left(1 + \frac{k_{cn}}{k_{cf}} + \frac{k_{cn}}{k_{be}}\right)^2}{\left(1 + \frac{k_{cn}}{k_{cf}} + \frac{k_{cn}}{k_{bp}}\right)^2} - \frac{\frac{k_{bp}}{k_{cf}}\left(1 + \frac{k_{cf}}{k_{be}}\right)\left(1 + \frac{k_{cn}}{k_{cf}} + \frac{k_{cn}}{k_{be}}\right)}{\left(1 + \frac{k_{bp}}{k_{cf}}\right)\left(1 + \frac{k_{cn}}{k_{cf}} + \frac{k_{cn}}{k_{bp}}\right)}\right] F_1^2 + \frac{2W k_{bp}}{\left(1 + \frac{k_{bp}}{k_{cf}}\right)\left(1 + \frac{k_{cn}}{k_{cf}} + \frac{k_{cn}}{k_{bp}}\right)} \Bigg\}^{1/2}
\end{aligned}
\qquad (6)
$$

F_3, the bolt tension after loosening, may be found by substituting F_2 from equation (6) into equation (2).

Equations (6) and (2) pertain to the first application of impact energy W. Upon a second application, F_y in the equations should be made equal to the previously calculated F_2, and F_1 should be made equal to the previously calculated F_3. It will then be found that F_2 again exceeds F_y, and further loosening occurs.. Similarly, each additional repetition of the impact load causes additional loosening, although in decreasing amounts. Thus there is continuous loosening under repeated impact, with the possibility of complete loss of tightness after many cycles.

Equation (6) does not apply if $F_2 < F_y$. For this case, an expression for F_2 may be obtained by substituting $k_{bp} = k_{be}$ in equation (6); that is, by replacing the plastic portion of the bolt curve by a continuation of the elastic portion. Then

$$F_2 = F_1 + \sqrt{\frac{2W k_{be}}{\left(1 + \frac{k_{be}}{k_{cf}}\right)\left(1 + \frac{k_{cn}}{k_{cf}} + \frac{k_{cn}}{k_{be}}\right)}} \qquad (7)$$

Since $F_2 < F_y$ in this case, there is no loosening, and $F_3 = F_1$.

For a given impact energy W, there is a maximum initial tension which should not be exceeded during tightening if loosening is to be avoided. This maximum initial tension is reached when F_2 from equation (7) equals F_y, thus

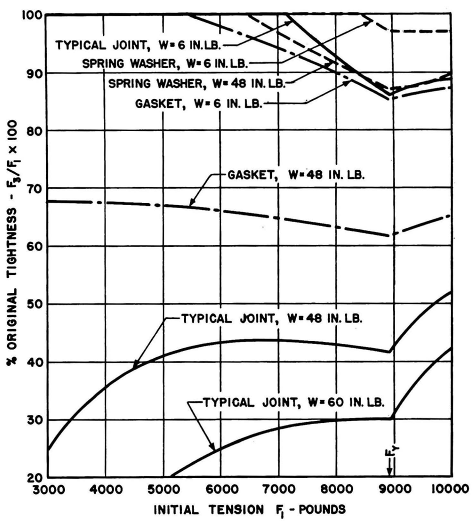

Fig. 5 Effects of initial tension, gaskets, and spring washers on loosening by one application of impact load

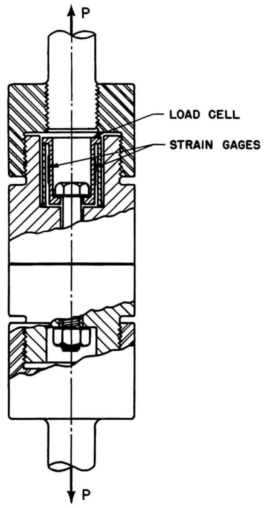

Fig. 6 Static testing fixture

$$F_{1\,max} = F_y - \sqrt{\frac{2Wk_{be}}{\left(1 + \frac{k_{be}}{k_{cf}}\right)\left(1 + \frac{k_{cn}}{k_{cf}} + \frac{k_{cn}}{k_{be}}\right)}} \qquad (8)$$

If $F_1 > F_y$, as in the case in Fig.3, the appropriate formula for F_2 may be obtained by replacing k_{be} with k_{bp} in equation (7); i.e., by changing the slope of the bolt curve from that of the elastic portion to that of the plastic portion. Then F_3, the bolt tension after loosening, is found from equation (5).

Equations (6) through (8) assume that $F_{c2} > 0$, so that the mating surfaces do not separate. To check this assumption, F_{c2} should be calculated. Starting with $F_{c2} = F_1 - k_{cn}(X_{c2} - X_1)$ and replacing the deflections by equivalents in terms of forces and slopes, one obtains

$$F_{c2} = \left(1 + \frac{k_{cn}}{k_{cf}} + \frac{k_{cn}}{k_{be}}\right)F_1 + \frac{k_{cn}}{k_{bp}}\left(1 - \frac{k_{bp}}{k_{be}}\right)F_y - \frac{k_{cn}}{k_{cf}}\left(1 + \frac{k_{cf}}{k_{bp}}\right)F_2 \qquad (9)$$

Equation (9) applies if $F_1 < F_y$ and $F_2 > F_y$, and should be used in conjunction with equation (6). If $F_2 < F_y$, the corresponding formula may be derived by replacing k_{bp} in equation (9) by k_{be}, to obtain

$$F_{c2} = \left(1 + \frac{k_{cn}}{k_{cf}} + \frac{k_{cn}}{k_{be}}\right)F_1 - \frac{k_{cn}}{k_{cf}}\left(1 + \frac{k_{cf}}{k_{be}}\right)F_2 \qquad (10)$$

Equation (10) should be used with equation (7). If $F_1 > F_y$, the corresponding formula may be obtained by replacing k_{be} in equation (10) by k_{bp}.

A negative value of F_{c2} obtained from the above equations is physically impossible and signifies that actually $F_{c2} = 0$. If so, equations (6) through (8) do not apply. For the case in which $F_{c2} = 0$, $F_1 < F_y$, and $F_2 > F_y$, the appropriate formula may be obtained in a manner duplicating the derivation of equation (6), with the exception that F_{c2} is substituted as zero. The result is

$$F_2 = \sqrt{\left[\frac{\frac{k_{bp}}{k_{cn}}\left(1 + \frac{k_{cn}}{k_{cf}} + \frac{k_{cn}}{k_{be}}\right)}{1 + \frac{k_{bp}}{k_{cf}}}\right]F_1^2 + \left[\frac{1 - \frac{k_{bp}}{k_{be}}}{1 + \frac{k_{bp}}{k_{cf}}}\right]F_y^2 + \frac{2Wk_{bp}}{1 + \frac{k_{bp}}{k_{cf}}}} \qquad (11)$$

After obtaining F_2 from equation (11), F_3 is found from equation (2).

For the case in which $F_{c2} = 0$, $F_1 < F_y$, and $F_2 < F_y$, let $k_{bp} = k_{be}$ in equation (11) to obtain

$$F_2 = \sqrt{\left[\frac{\frac{k_{be}}{k_{cn}}\left(1 + \frac{k_{cn}}{k_{cf}} + \frac{k_{cn}}{k_{be}}\right)}{1 + \frac{k_{be}}{k_{cf}}}\right]F_1^2 + \frac{2Wk_{be}}{1 + \frac{k_{be}}{k_{cf}}}} \qquad (12)$$

Since $F_2 < F_y$ in this case, there is no loosening, and $F_3 = F_1$.

For a given impact energy, the maximum initial tension permissible without subsequent loosening is reached when F_2 from equation (12) equals F_y. This leads to

$$F_{1\,max} = \sqrt{\left[\frac{1 + \frac{k_{be}}{k_{cf}}}{\frac{k_{be}}{k_{cn}}\left(1 + \frac{k_{cn}}{k_{cf}} + \frac{k_{cn}}{k_{be}}\right)}\right]F_y^2 - \frac{2Wk_{cn}}{1 + \frac{k_{cn}}{k_{cf}} + \frac{k_{cn}}{k_{be}}}} \qquad (13)$$

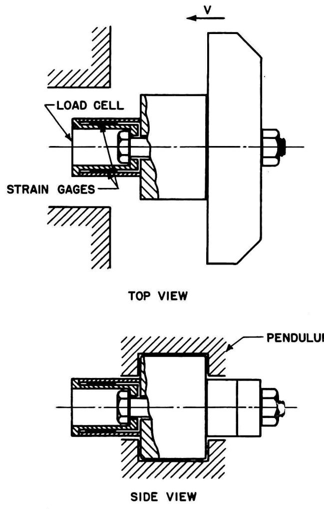

Fig. 7 Impact testing fixture

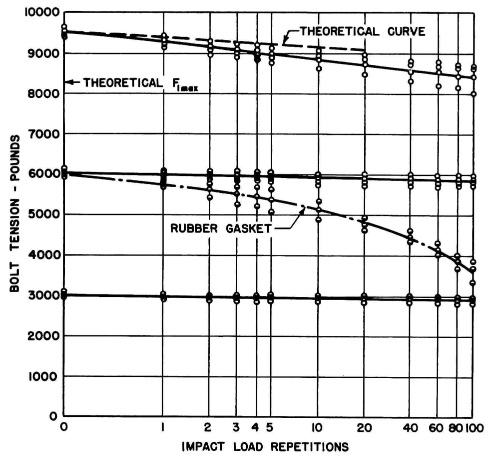

Fig. 9 Loosening caused by repeated impact with 48 in. lb pendulum energy

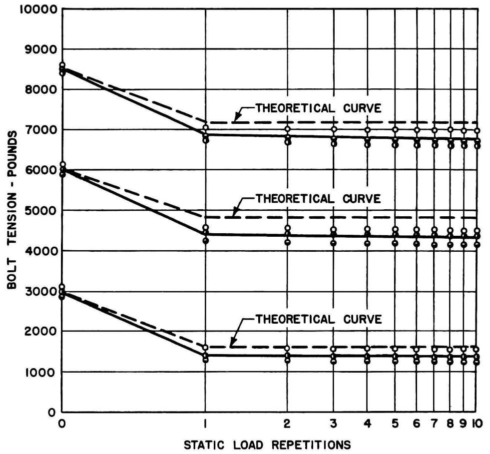

Fig. 8 Loosening caused by repetitions of an external force of 9250 lb

If $F_{c2} = 0$ and $F_1 > F_y$, the appropriate formula for F_2 may be obtained by replacing k_{be} in equation (12) with k_{bp}, thus changing the slope of the bolt curve from that of the elastic portion to that of the plastic portion. F_3 is then found from equation (5).

It should be noted that the energy method used in deriving the foregoing formulas does not provide the complete picture of the transient motions and wave propagation effects which are involved. Also, in high-speed impact the increase in elastic limit with strain rate should be taken into consideration.

The use of the impact equations will be illustrated by several numerical examples, the results of which are plotted in Fig.5. Values of k_{be}, k_{bp}, F_y, k_{cf}, k_{cn}, and k_c were those used in Fig.4. The curves show that if loosening is to be completely avoided for moderate values of impact energy, the initial tension must not exceed the maximum value obtained from equation (8) or (13). Loosening is unavoidable at extremely high values of impact energy, but the amount of loosening can be decreased by tightening to higher values of initial tension. Gaskets appear to increase loosening somewhat under moderate impact loads, and to markedly reduce loosening under extremely large impact loads. In all instances loosening is substantially reduced by the addition of a spring washer. The effect is particularly remarkable at high impact loads; the 48 in. lb. curve shows that not only is loosening drastically reduced, but for initial tensions below 6480 lb. there is no loosening at all.

EXPERIMENTAL RESULTS

The bolts tested were 3/8 in.-16 UNC-2A x 3 in. hexagon-head cap screws of 1038 steel heat treated to SAE Grade 5, used with heavy hexagon steel nuts. The static testing fixture, shown in Fig.6, was used to apply an external tensile force to the joint with the aid of a hydraulic tensile testing machine. The impact testing fixture, shown in Fig.7, was used to impose an external impact load on the joint by means of a pendulum-type impact machine. The impact fixture was supported in a slot in the pendulum in a manner similar to standard tensile impact specimens. The bolt tension was measured with a load cell consisting of two concentric sleeves to transfer

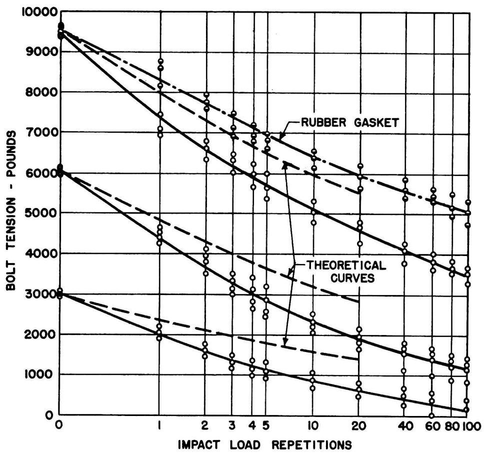

Fig. 10 Loosening caused by repeated impact with 192 in. lb pendulum energy

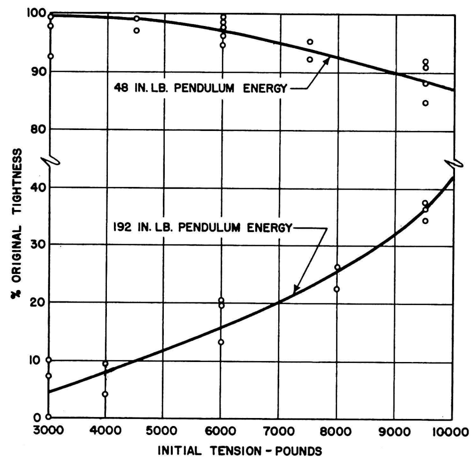

Fig. 11 Effect of initial tension on loosening after 100 impact load repetitions

the load from the bolt head to the fixture block. Four resistance-wire strain gages were cemented to the inner sleeve in an arrangement which cancelled signals due to temperature effects or bending stresses. Instruments used were a strain-gage amplifier with chopper and null-balance dial, and a cathode-ray oscilloscope. Bolt tensions before and after loosening were measured by the null balance dial using the square wave formed by the chopper on the oscilloscope screen.

Values of k_{be}, k_{bp}, F_y, k_{cf}, k_{cn}, and k_c were determined for the bolts and fixtures and these values were used to compute theoretical curves for comparison with the experimental curves.

Representative results are shown in Figs.8 through 11. Fig.8 shows loosening due to repetitions of a large external force. It is seen that the actual loosening was somewhat greater than that calculated. Most of this discrepancy is probably due to loosening from other causes, such as contact deformation of surface irregularities or extrusion of scale. It was also found that an extremely small amount of loosening occurred on each successive load repetition after the first. This additional loosening may be chargeable to small nut rotation per cycle or to continued extrusion of scale. Loosening by nut rotation, if any, would be extremely small in amount due to the limited number of load cycles.

Figs.9 and 10 show loosening due to repeated impact. At the moderate load of 48 in-lb there is substantially no loosening for initial tensions of 3000 and 6000 lb, while there is definite loosening for an initial tension of 9500 lb. Under the very large load of 192 in-lb, substantial loosening occurred at all values of initial tension. Curves are included showing the effect of a 1/16-in. rubber gasket. No theoretical curves were calculated for the rubber gasket, as some creep of the rubber was present and this effect was not included in the formulas. However, the general effect of the gasket was as predicted in the discussion of Fig.5. As in the case of the static tests, the actual loosening was in general greater than the calculated loosening, the difference again being due to loosening from other causes.

Fig.11 presents the impact data in the manner of Fig.5, showing the effect of initial tension on loosening. As expected, loosening was increased by use of higher initial tensions at moderate loads, but was decreased by higher initial tensions under very large loads.

CONCLUSIONS

The designer must determine the amount of loosening which is admissible in a particular application. The toleration of some loosening by plastic deformation will permit the greatest external load for a given size bolt or the smallest bolt for a given load. The amount of such loosening can then be minimized by specifying high tightening torques. Use of spring washers or, depending on conditions, addition or removal of gaskets may bring the loosening to within acceptable limits.

If, however, the application demands the complete absence of loosening, the initial tension must be kept below the calculated maximum value by a margin of safety. The maximum initial tension for no loosening increases as the external load is decreased, hence if high clamping forces are desired the load must be small for a given size bolt or a larger bolt must be used for a given load.

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