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**REPRINT**

# New Criteria for Self-Loosening of Fasteners Under Vibration

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A UNIQUE TEST method and apparatus now make it possible to reproduce conditions of vibration that are not only certain to loosen bolted joints but which also closely simulate actual conditions. It can be shown that properly preloaded fasteners loosen as a result of rotation as soon as relative motion occurs between the mating threads and between the bearing surfaces of the fastener and the clamped material.

In addition to fatigue failure, self-loosening is the most frequent cause of failure of dynamically loaded, bolted joints. This may be the reason for the voluminous literature on the operation and effectiveness of self-locking elements designed to prevent self-loosening of threaded fasteners, with resultant severe damage to machine and vehicle assemblies. Most of the existing literature, however, gives the impression that there have been many approaches tried without finding the correct solution to this problem. To be sure, much test data have been obtained in both the United States and Europe in an attempt to explain these action-reaction phenomena. Unfortunately, up to the present time, there does not seem to have been sufficient exploration to develop any truly practical answers.

Goodier and Sweeney (1),\* as well as Sauer, Lemmon, and Lynn (2), have tested only axial dynamically loaded, bolted connections. In spite of their failure to obtain a

complete self-loosening of threaded fasteners, they have offered an explanation for partial loosening, which does contain the basic element of the mechanism of self-loosening of bolted connections. That is, bolted connections are loosened by the relative motion between the thread flanks and other contact surfaces of the clamped and clamping parts.

In 1966, Paland (3) precisely formulated and affirmed the prior theory and also presented exact proof. Nevertheless, neither could he produce total self-loosening (that is, the total loss of clamping force), even by extreme axial impact loading.

In 1966 the author approached the problem from another direction. My paper, "Investigations of the Mechanism of Self-Loosening and Optimal Locking of Bolted Connections" (4-6), presented data from tests mainly carried out on transversely loaded joints. There was no difficulty in completely rotating loose nonlocking fasteners as well as some so-called self-locking fasteners.

New tests, which affirmed that transverse vibration generates the most severe conditions for self-loosening, led to the design of a new testing machine, which is supposed to give practical answers to the phenomena of self-loosening as well as quantitative data on the locking performance of self-locking elements. After a general discussion here of the mechanism of self-loosening, this new machine and its operation will be described. The various possibilities of recording and evaluation will be shown and discussed, with

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\*Numbers in parentheses designate References at end of paper.

## ABSTRACT

The theory of self-loosening of preloaded bolted connections when subjected to vibration is discussed. The significance of self-loosening as a cause of failure is explained, and design guidance to avoid self-loosening is given. The test methods are described and discussed in connection with

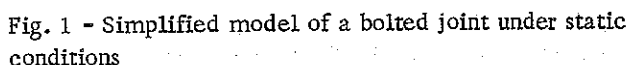
a newly designed testing machine that yields quantitative data for evaluating locking properties. These methods can be applied to all types of locking elements. Finally, a simplified method for broad scale testing and inspection is proposed.

It is not the intention of this author to compare the performance of self-locking elements, not even of those that have been used in these test series for the purpose of evaluation and trial of new test methods and machinery. To give final answers on the self-locking performance of various elements, much more test data must be obtained on a statistical basis after agreement on suitable test methods has been reached.

The theory of the mechanism of self-loosening is based on the well-known law of physics that defines the effects of friction on two interacting solid bodies. As soon as the friction force between two solid bodies is overcome by an external force working in one direction, an additional movement in any other direction can be caused by the action of forces that can be essentially smaller than the friction force.

$$Q = L \tan(-\varphi + \rho) + L \tan \rho \quad (1)$$

This effect can be demonstrated by examples from daily life: A cork can be more easily removed from a bottle if



Similar to a load lying on a fixed slope, self-locking exists in a bolted connection as long as no relative motion arises between the thread flanks and the contact surfaces of clamping and clamped parts. The following off-torque is needed for loosening the bolt or the nut:

The bolt could loosen by itself (without the application of external off-torque) only if  $T_{off}$  were to become either

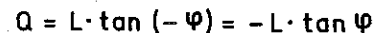
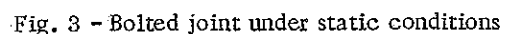


Fig. 2 - Simplified model of a bolted joint under vibration



a zero or negative value. Even under ideally lubricated conditions, the thread friction angle will not be less than 6 deg and the helix angle will not exceed 3° 20' on sizes down to M5 diameter (equivalent to 3/16 in.). Therefore, the sum between the square brackets in Eq. 2 can never become smaller than, or equal to, zero; that is, the product of the equation can become zero only if the preload drops to zero. This case, which exists only if no relative motion occurs in the threads or at the bearing area interfaces, has been very widely generalized.

However, if relative motion occurs between the threaded surfaces and/or other contact surfaces of the clamped and clamping parts because of an external force, the direction of which is either tangential or radial, the bolted connection will become free of friction in a circumferential direction (Fig. 4). This means that the preload acting on the thread, which is a slope, creates a force in a circumferential direction and results in the rotational loosening of the bolt or the nut. The maximum value for total elimination of the circumferential friction force resisting the existing internal off-torque is then equal to

$$T_{\text{off}} = F_v \frac{d_2}{2} \tan -\varphi = -F_v \frac{d_2}{2} \tan \varphi \quad (3)$$

The question now is: How do these relative motions acting between the thread flanks and other contact surfaces of the clamped and clamping parts occur?

For axially loaded joints, Goodier and Sweeney have already pointed out that pulsating tension of a clamped bolted connection creates radial sliding motions between the thread flanks of the bolt and nut or at the interface of the clamped bearing surfaces. The reasons for this are the contraction of the bolt according to Poisson's ratio and the dilation of the nut walls caused by axial tension. Paland proves this rule arithmetically and by measuring the tangential strain on the surface of the nut. He comes to the conclusion that a loaded nut widens elastically in a radial direction at the

area near the bearing surfaces and contracts in the upper part. For the widening area, for nuts of M10 diameter (approximately 3/8 in.), he specifies values for the relative motion as

$$S/F = (7 \cdots 8) \cdot 10^{-6} \text{ mm/kgf} \quad (4)$$

These very small amounts of radial displacement by expansion of the nut would explain why Paland (3), in spite of heavy impact loading in an axial direction of the bolt, still needed a small external off-torque to turn the nut completely loose. Goodier and Sweeney (1) have succeeded at the end of 500 cycles in producing angles of  $5.5 \times 10^{-3}$  revolutions, approximately equal to 2 deg, when loading a 3/4 in. nut-bolt connection on a tensile testing machine. Nevertheless, this loosening had already caused a preload drop of 27%.

Sauer, Lemmon, and Lynn (6) loaded a 5/16 in. nut-bolt connection in a fatigue testing machine with a working load equal to 80% of the preload. They recorded a maximum loosening angle of 6 deg after 25,000 cycles when using cadmium plated nuts and bolts. Their loosening curves show that the rotating process had come to a stop. The recorded rotation loosening angle of 6 deg nevertheless had caused a decrease of the clamped length of 24  $\mu\text{m}$ ; this is equal to a decrease of 700 kgf (1550 lb), when related to a clamping length of  $2d_1$ . Using a Grade 8.8 bolt (equal to Grade 5) this reduced the preload by 42%.

For dynamically transverse loaded joints, the relative motion between the thread flanks and the contact surface of the bearing areas can occur in magnitudes up to the maximum allowance of the thread. These large effects appear when transverse loadings, which have to be transmitted by grip friction, exceed the friction force between the clamped parts,  $\mu F$ , the friction force being delivered by the clamping force  $F$ . The resultant transverse slippage between the clamped parts forces the bolt to assume a pendulum movement, which leads to relative motion in the thread hole and thus between the thread flanks.

If the amplitude of such transverse slippage of the bolt is large enough, slippage of the nut or bolt head bearing surface will finally occur and make the joint totally free of friction in a circumferential direction. It can be easily realized too that, contrary to the conditions of axial loadings, relative motion between the flanks will occur in all parts of the nut thread when the joint slips under transverse force. Thus, the internal off-torque force becomes sufficient to turn the bolt or nut completely loose as soon as the friction is eliminated in the bearing area as well as in the thread area. Such transverse slippages are more common in practice than is usually accepted. Experience shows that these joints most frequently fail by self-loosening.

An example of self-loosening is the fastening of the crown wheel to the body of a differential gear in an automobile. The driveshaft transmits the moment to the crown wheel by a bevel wheel and from there by grip friction to the body

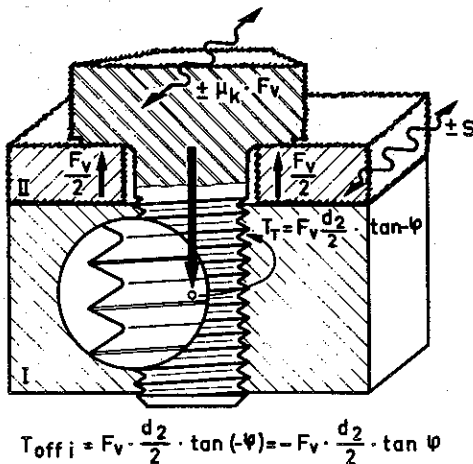


Fig. 4 - Bolted joint under transverse vibration

of the differential gear. Quite often the transverse slippage between the crown wheel and body occurs around the area of these two teeth, which are actually in contact, because the adjacent bolt connection is more heavily loaded than the more distant ones. Finally, in the limits of the elastic deformation on the crown wheel, a transverse slippage takes place and forces the bolt to move as a pendulum in the tapped hole of the body.

During the rotation of the crown wheel, one bolt after the other enters the area of the two teeth in contact; thus one bolt after the other rotates loose. Also, in cases where the joint is axially or coaxially loaded, transverse slippage can occur because such loadings can cause elastic deformation of the clamped parts (for example, a bearing cover).

Proof for the validity of the whole theory of the loosening mechanism of transversally loaded joints can be easily determined. Schoellhammer, of the Laboratories of Daimler-Benz AG., Stuttgart, has determined the external off-torque of several fasteners, including self-locking types under dynamic transverse loading. In these tests the frequency of transverse impact was increased to the point that the external off-torque decreased, and the fastener finally rotated loose without the necessity of any external off-torque. For these tests he used a vibration machine, with the fastener to be tested clamping a plate against a block.

Two air actuated impact hammers induced the relative motion between the plate and the block, and by variation of the air pressure the induced impulses could be varied. The bolts tested were M10 Grade 8.8 (equal to SAE Grade 5), with and without self-locking elements and also with prevailing torque type self-locking nuts. The bolts and nuts were tightened with a torque wrench, but the actual preload was not determined. When a certain impulse with increasing frequency was reached, the external off-torque for loosening

became zero. The nuts under test rotated completely loose by themselves.

A similar principle was used in the earlier work of this author. It consisted of two plates clamped by the fastener to be tested, which in turn was transversely loaded dynamically by using a 6 ton fatigue testing machine. Transverse loads were measured by a tensile load cell, and the amplitude of the relative movement between the two clamped plates (caused by the transverse loading) was recorded by a displacement pickup. Both sets of data were plotted by a high speed recorder. When tightened, both bolt torque and preload were measured by load cells. Thus, the bolt could be tightened to the exact preload.

The properly tightened joint was vibrated and then, during the vibration process, the bolt was loosened and the necessary external off-torque was measured. With increasing induced transverse force, the external off-torque decreased and, as Fig. 5 shows, the curve passed through  $T_{\text{off}} = 0$  for a certain amount of induced vibration. Further increase of induced vibration resulted in negative torque values, indicating clearly that there is an internal off-torque developed by the wedge effect of the axially loaded thread. The proof for the elimination of self-locking was given when an internal off-torque of  $-0.58 \text{ mkgf}$  ( $50.4 \text{ in.-lb}$ ) was recorded. This measured internal off-torque was approximately equal to the one obtained by a calculation for the case of a bolt completely free of friction:

$$T_{\text{off}} = F \frac{d}{V} \frac{2}{2} \tan(-\varphi) \quad (5)$$

The same proof could be obtained by using a vibration device in which a joint is clamped by one single bolt in the center of a torsional vibration.

#### TO PREVENT SELF-LOOSENING BY MEANS OF DESIGN

According to previous research work, it has to be assumed that total self-loosening does not occur in joints that are dynamically loaded only in an axial direction. This may be at least valid in all cases where the joint is properly tightened. But it has been proven that at least partial rotation of the nut can occur when the ratio between working load and preload is high, and particularly if the load in the thread is largely diminished by the compression amplitude of an alternating working load.

The partial loosening by rotation leads to a further loss of preload, and thus to a stop of the rotating process, because the internal off-torque is proportional to the preload, which itself is reduced by the rotation process. On the other hand, the loss of the preload increases the danger of fatigue failure because the total amount of alternating working load (axial) is felt by the bolt.

On the other hand, total rotational loosening can occur in transversely loaded connections as soon as the clamping

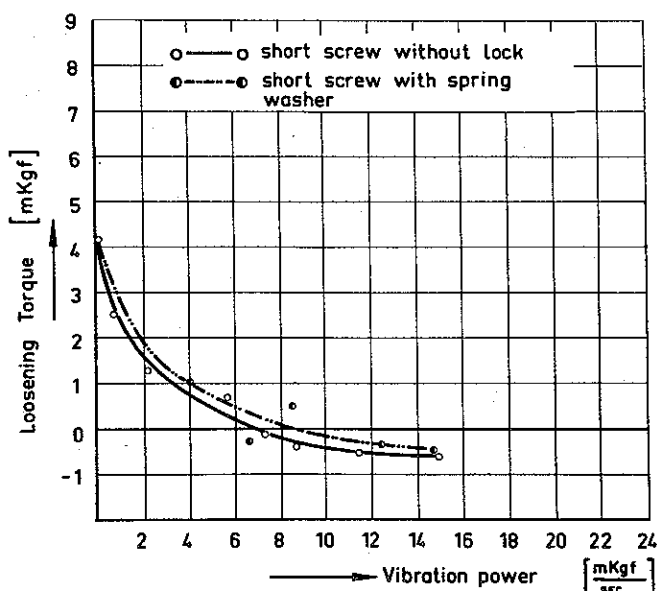


Fig. 5 - Loosening torque for preloaded fasteners under transverse vibration

load can no longer maintain sufficient grip friction so that transverse slippage between the clamped parts occurs. Transverse slippage, as a matter of fact, is quite common because it is caused not only by transverse loads or transverse components of a load in any direction, but also by coaxial loads that deform the clamped part elastically (that is, by all joints that are loaded eccentrically, such as bearing caps). The proposed concept of the mechanism of self-loosening or self-rotating of fasteners completes the picture of bolt failures under vibration by introducing the "self-rotating process." Many cases of fatigue failures under obviously very low preload, which could not be explained by plastic deformations or even by insufficient tightening, were probably caused by partial self-rotating.

In Fig. 6 an attempt is made to illustrate the failure mechanism of dynamically loaded bolted connections (7). This chart has been made up to resemble a road map with road numbers. There is a total of 12 different lines, which lead either to fatigue failure of the bolt (infrequently the clamped part or the nut) or to self-loosening of the fastener. This illustration shows that the basic rule for avoiding bolt failure, either by fatigue breakage or by self-loosening, is to supply

and to maintain sufficient clamping load. Since this paper deals with the "self-loosening" branch of the failure mechanism, the design rule for avoiding self-loosening or loss of preload by partial self-rotating is as follows: Design in such a way that no relative movement occurs between the contact bearing surfaces normal to the axis of the bolt or between the thread flanks of the fastener components.

This requires that a sufficient residual clamping force remains when the scatter of preload values after tightening and the embedding phenomena (Setzen), due to plastic deformations of the fastener or the clamped parts, is taken into consideration. The residual clamping load should never fall short of the value  $F_T/\mu$  (transverse load/friction coefficient) to make sure that no transverse slippage can occur. Modern rules for calculation of bolted connections (8 and 9) are therefore based on the necessary residual clamping force, the estimated degree of embedding (Setzen), and the scatter of the preload obtained by tightening, as represented in the following equation:

$$F_{\max} = A [F_R + F_W + \Delta F_V] \quad (6)$$

where:

$$\Delta F_V = z \frac{C_S C_P}{C_S + C_P} \quad (7)$$

Tabulated values for the different factors are given in Ref. 8.

As it was previously shown, using the example of the crown wheel, there exist cases where these demands can be met only if the joint fasteners are uneconomically overdimensioned. There is no doubt that many dynamically loaded bolted connections are beyond an exact calculation; therefore the relative movements cannot always be excluded by means of the joint design only. In these cases, use is recommended of fasteners with self-locking elements that tend to resist the internal off-torque which occurs when inherent self-locking is lost.

#### PROPOSED NEW TEST METHODS AND MACHINERY

Since relative movements between clamped parts and fasteners cannot be avoided by means of the joint design only, and self-locking fasteners have to be used for preventing "rotating loose," a criterion has to be found for the effectiveness of such self-locking fasteners. Several test methods have been tried in the past and are still in use (10-19). Most of them are more or less qualitative methods, which result in "rotating loose" or "not rotating loose" under certain specified conditions (some of them in preloaded condition; some in an unpreloaded condition). It is obvious that the latter ones are practical only for prevailing torque types of locking fasteners. Other specifications give values

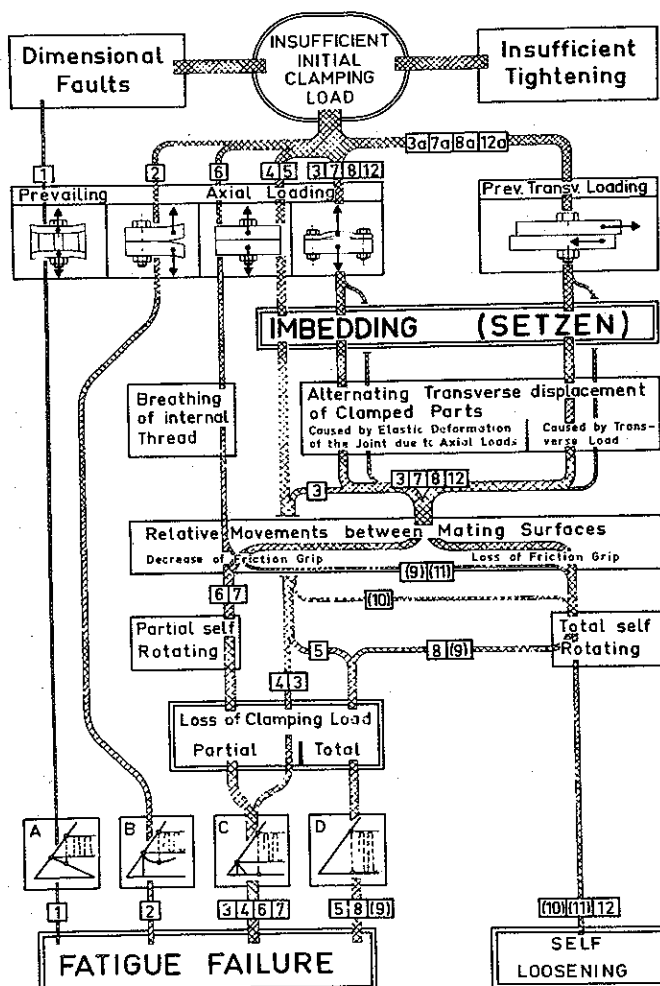


Fig. 6 - Graphical illustration of failure mechanism of dynamically loaded bolted joints

for installation torque, breakaway torque, and prevailing torque.

With this paper an entirely different approach is proposed. Since transversely loaded joints tend more to self-loosening, the test procedure suggested imitates these actual conditions. The first attempts were made with a device consisting of two parts clamped together by the specimen, with load cells and a displacement pickup to record transverse load, preload, and displacement.

The product, transverse force times displacement, was called vibration energy. Its maximum values were significantly different for the various locking elements.

This test method yielded results that could not be reproduced, since the machine used for the tests was a resonant type of fatigue machine. This type generates a force that is a function of the machine frequency. When starting a test, the transverse force increased with increasing frequency as a function of time until the loosening process of the tested fastener was initiated. Depending upon the starting speed of the machine, a different maximum force (and therefore maximum energy) was necessary for rotation loosening of a nut or bolt. Therefore, a new vibration machine was designed for these tests.

The prototype of the new machine was built by the Corporate Machine Building Dept. of the SPS Co. This machine generates a transverse sliding motion between two clamped parts by means of an adjustable eccentric. The resulting transverse force is independent of frequency and starting speed of the machine. It is for a given machine stiffness (spring constant of the load transmitting parts), strictly a function of the eccentric adjustment.

Figs. 7 and 8 show the heart of the machine. The tested bolt clamps the U-shaped top part onto the bed, an integral

part of the machine frame. The clamping force is measured with a compression load cell through which a bushing with internal threads is placed for testing bolts. When testing nuts, this bushing may hold a threaded stud, or a special bushing with external threads may be used.

The U-shaped top part is displaced parallel to the bed. The force is transmitted by a load cell mounted on flexure plates, a connecting rod being attached to the load cell by crosswise flexure plates and by an adjustable eccentric. The U-shaped top part is separated from the bed by flat strips of needle bearings, to avoid galling.

The relative movement between bed and top part is measured with special linear differential transformers. The tightening and loosening angles were measured with a linear potentiometer, which was attached to the test specimen by a flexible shaft. Exchangeable threaded bushings and inserts in the U-shaped top part make the mechanism usable for testing bolts from 1/4 in. (or M6) up to 5/8 in. (or M16) thread sizes. This range covers the majority of the common locking elements.

The machine is driven by a 3.6 kW (5 hp) a-c motor, which is located in the machine housing. A V-belt drive and different pulleys allow testing frequencies of 1500, 3000, 4500 cpm.

Fig. 9 shows three different applications for the machine. In the upper picture the U-shaped top part is connected to the load cell with a bolted plate, to eliminate all play. The arrangement is used for tests with pure transverse loading. This plate is replaced in the middle picture by an U-shaped sheet metal specimen that transmits the transverse forces when clamped with the U-shaped top part to the bed. With this arrangement the fatigue strength of bolted sheet metal joints can be evaluated (see also Fig. 21). This is of special interest if the fastener damages the surface of the clamped parts, for example, when using locking fasteners with serrated bearing surface. In the bottom picture the connecting plate is replaced by a connecting rod. It is thereby possible to rotate the bed so that it forms any

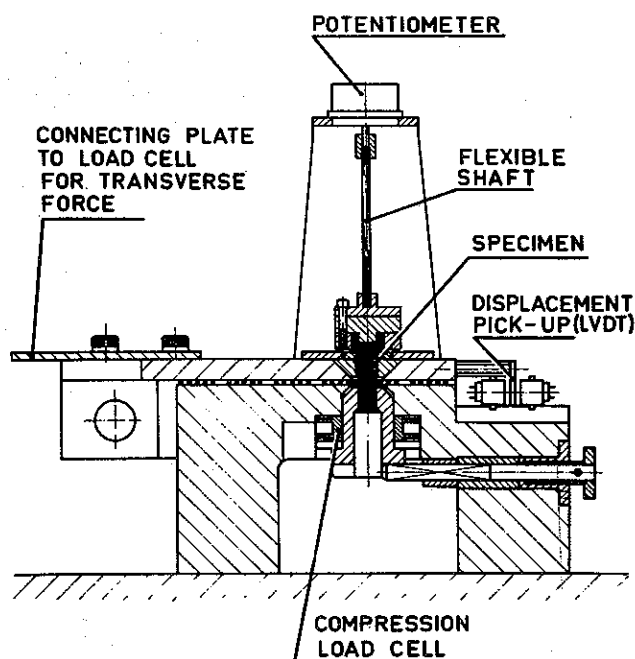


Fig. 7 - Central part of the vibration machine

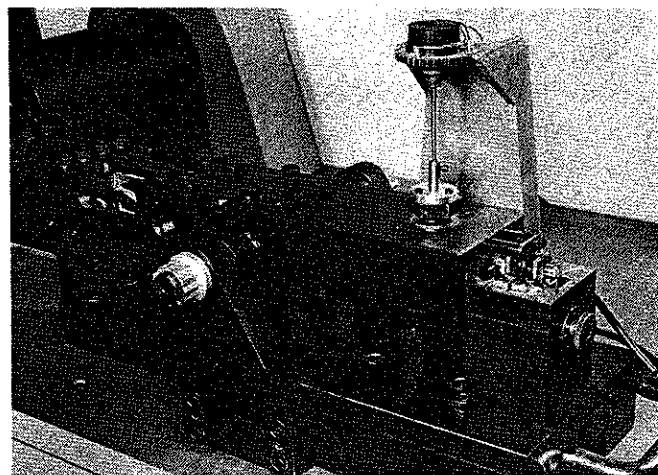


Fig. 8 - Photograph of central part of the vibration machine

angle between 90 and 180 deg with the frame of the machine. The center of rotation coincides with one bearing of the second connecting rod. This arrangement allows loading the test specimens by any combination of transverse and axial forces.

The recording equipment used for the tests described above is shown in Fig. 10. Transverse force  $F_T$  and displacement "d" are plotted on a Visicorder as a function of time. The bolt preload is recorded on a third channel. By using a two channel oscilloscope, the transverse force  $F_T$  is recorded as a function of displacement "d" in X-Y hookup on one channel and the preload on the second channel. Running the tests with a special low speed drive (10 cpm), the same information was plotted on an X-Y<sub>1</sub>-Y<sub>2</sub> recorder.

The electrical values of transverse force  $F_T$  and displacement "d" were multiplied by a specially built electronic multiplication circuit. The result  $F_T \cdot d$  is plotted as a function of preload on an X-Y<sub>1</sub>-Y<sub>2</sub> recorder. The tightening and loosening angle measured with a linear potentiometer

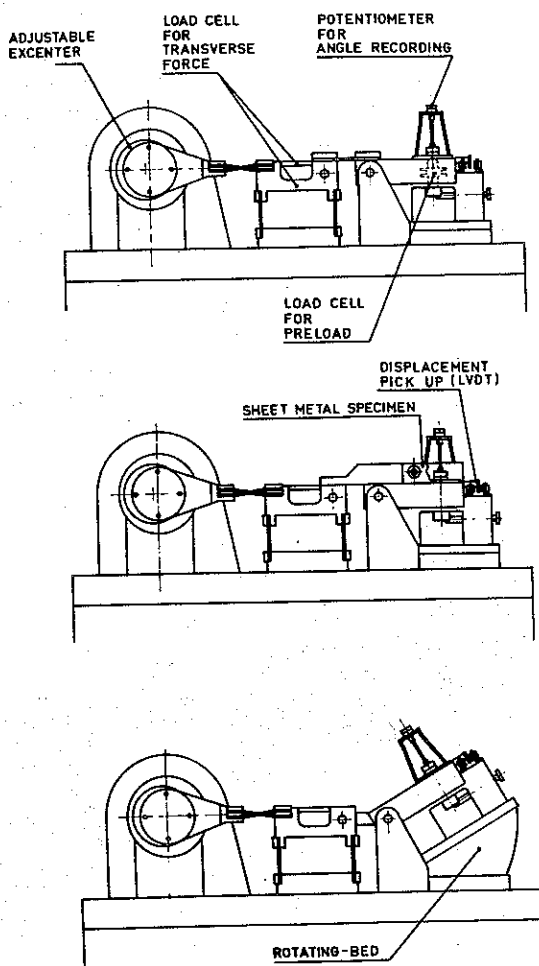


Fig. 9 - Three different possibilities of application of the vibration machine

is recorded as a function of preload on the second channel of the X-Y<sub>1</sub>-Y<sub>2</sub> recorder.

To check out the machine and to evaluate the possibilities and limitations of the various testing, measuring, and recording methods, the test specimens were restricted to nonlocking screws of different thread tolerance (with and without spring washers) and to two types of free spinning self-locking screws with serrated bearing surfaces. This restriction has been made because of lack of time and because these types of screws are used in great numbers for sheet metal constructions, particularly in the automotive industry. The described test methods and machinery can be used for testing all kinds of self-locking nuts and also self-locking screws of the prevailing torque and adhesive type. A comprehensive study with specimens of all groups of self-locking fasteners is in preparation. Geometric shapes

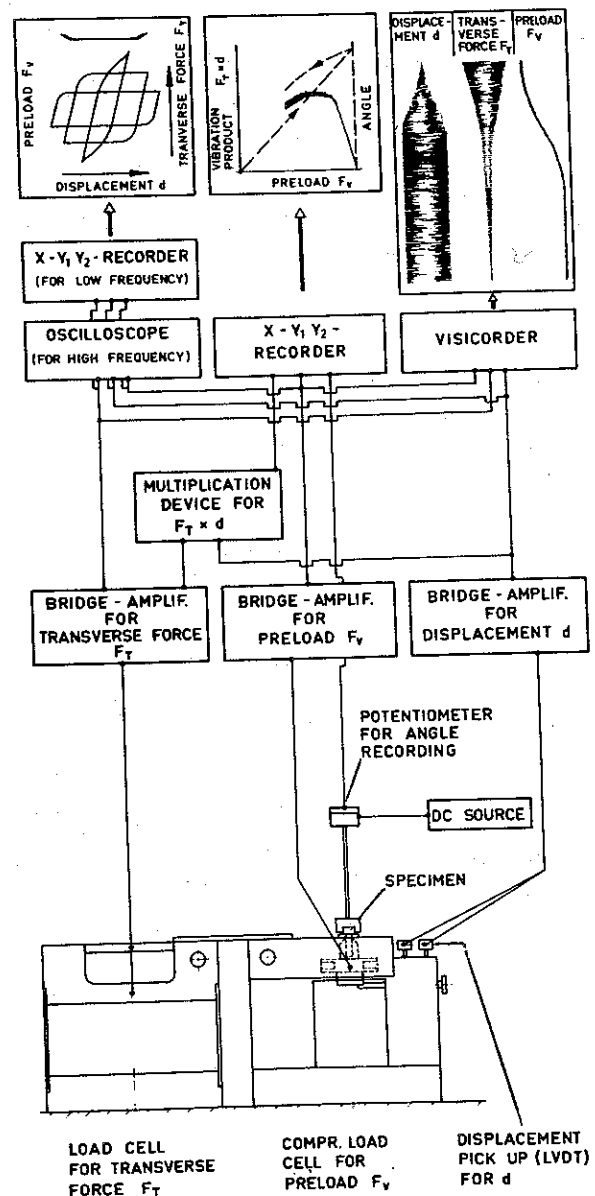


Fig. 10 - Plan of instrumentation and recording

and dimensions of the test specimens are given in Table 1 and the mechanical properties in Table 2. Screws of the size M10 × 1.5 and 3/8-16 have almost equal nominal thread diameter, pitch diameter, and pitch so that their test results can be compared.

All fasteners were oiled before testing and were used with plain washers under the head, having a Rockwell hardness of 77R<sub>B</sub> (HB<sub>30</sub> = 137 kgf/mm<sup>2</sup>). The free spinning

types of locking fasteners were additionally tested on washers of higher hardness ( $R_c = 39 - 41$ ).

During all tests the specimens were originally preloaded to 2500 kgf = 5500 lb. This relates to 75% of the proof load of screws with a strength level of ISO Class 8.8 (similar to Grade 5). All tests except those run with a special low speed drive (10 cpm) were run with a frequency of 3000 cpm.

X-Y PLOT OF TRANSVERSE FORCE AND DISPLACEMENT

Table 1 - Dimensional Specification of Tested Specimens

| No.                                   | Specimen                                                                                                                                                                               | Dimensional Specification                                                                                                                                                      |
|---------------------------------------|----------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------|--------------------------------------------------------------------------------------------------------------------------------------------------------------------------------|
| <b>A Nonlocking Screws</b>            |                                                                                                                                                                                        |                                                                                                                                                                                |
| A <sub>1</sub>                        | Hex head cap screws,<br>DIN 933, M10 × 1.5 × 30                                                                                                                                        | Pitch: 1.5 mm<br>Thread tol: Bolt 6g<br>Nut 6H<br>Nominal max. thread<br>allowance: 0.344 mm<br>0.01355 in.                                                                    |
| A <sub>2</sub>                        | Socket head cap screws,<br>1960 Series,<br>3/8 -16 × 1-1/4 in.                                                                                                                         | Pitch: 1.586 mm<br>Thread tol: Bolt 2A<br>Nut 2B<br>Nominal max. thread<br>allowance: 0.290 mm<br>0.01142 in.                                                                  |
| A <sub>3</sub>                        | Socket head cap screws,<br>1960 Series,<br>3/8 -16 × 1-1/4 in.                                                                                                                         | Pitch: 1.586 mm<br>Thread tol: Bolt 3A<br>Nut 3B<br>Nominal max. thread<br>allowance: 0.193 mm<br>0.00759 in.                                                                  |
| <b>B Spring Washers</b>               |                                                                                                                                                                                        |                                                                                                                                                                                |
| B <sub>1</sub>                        | Hex head cap screws,<br>DIN 933 (A <sub>1</sub> ) plus<br>spring washer DIN 128                                                                                                        |                                                                                            |
| <b>C Free Spinning Locking Screws</b> |                                                                                                                                                                                        |                                                                                                                                                                                |
| C <sub>1</sub>                        | Flange head, hex cap<br>screws, M10 × 1.5 × 30,<br>with triangular teeth,<br>groove and bearing plat-<br>form near the shank                                                           |  Thread tol: Bolt 6g<br>Nut 6H<br>Nominal max. thread<br>allowance: 0.344 mm<br>0.01355 in. |
| C <sub>2</sub>                        | Flange head hex cap<br>screws, M10 × 1.5 × 30,<br>with long radial teeth<br>and circumferential<br>bearing ring, edges of<br>teeth and bearing ring<br>forming one continuous<br>curve |  Thread tol: Bolt 6g<br>Nut 6H<br>Nominal max. thread<br>allowance: 0.344 mm<br>0.01355 in. |

(HYSTERESIS CURVES) - The schematic drawing in Fig. 11 explains the relationship of force to deflection and to displacement in a slipping, bolted connection subjected to dynamic transverse forces. The dotted line represents the force/displacement diagram of two plates, which are clamped together by a screw with narrow thread tolerance. It will be assumed that the head of the screw cannot slip on the plate. This screw is bent to the left by the top plate at point  $A_1$

(left dead center). When the top plate moves to the right, the screw bends to the right until (owing to the moment in the thread) the slipping in the thread begins at point  $B_1$ ; the force/deflection line is less steep up to point  $D_1$ .

At  $D_1$  the allowance in the thread has been used up by slipping and the screw thread is supported by the opposite side of the internal thread. From there on to point  $E_1$  (right dead center) the screw is bent more. The same process repeats during the other half of the cycle when the screw is bent back to the left dead center. The included area represents the friction energy absorbed by the joint, which is being transformed into heat. The area is very small, since only the thread slippage creates friction energy; the bending of the screw is pure elastic deflection.

The curve, indicated by a dash-dot line, represents the force/displacement diagram of a bolted joint for which the bolt head too cannot slip on its bearing surface. The thread tolerance, however, is so wide that the external screw thread

is contained by the internal thread exactly at the left and right dead center. This means that the slipping in the thread starts at point  $B_2$  and ends at point  $E_2$ . The enclosed area is larger, owing to the longer slippage displacement in the threads; therefore the absorbed friction energy is larger too.

The solid line in the diagram represents a joint with a fastener that can slip under the head. This is the case when the friction grip  $F_V^\mu$  is overcome by the transverse force

$F_T$ . Starting from the left dead center at point  $A_3$ , the screw is bent to the right by the transverse force, which does not yet break through the friction grip of the head and the top plate. At point  $B_3$ , slipping in the threads starts.

Before thread slipping can proceed to  $D'_1$  (for narrow thread tolerance) or to  $D'_2$  (for wide thread tolerances), slipping under the screw head starts at point  $C_3$  and ends at point  $E_3$ , the right dead center.

Drawing lines through  $D'_1$  and  $D'_2$  parallel to the load/deflection line of the load transmitting machine part ( $C_m = \tan \phi$ ), we obtain the distances  $W_1$  and  $W_2$  as the intersections of these parallel lines with line  $C_3E_3$ . They represent the part where slipping in the thread as well as under the head occurs. Fastener rotation can take place only on these distances because only there is the screw free of friction.

Table 2 - Mechanical Properties of Specimens

| No.            | Proportional        | UTS,                | HV <sub>0,3</sub>      |               | Remarks                                                              |
|----------------|---------------------|---------------------|------------------------|---------------|----------------------------------------------------------------------|
|                | Limit,              |                     | (Diamond pyr 300 load) |               |                                                                      |
|                | kgf/mm <sup>2</sup> | kgf/mm <sup>2</sup> | kgf/mm <sup>2</sup>    |               |                                                                      |
|                | psi                 | psi                 | Core Hardness          | Case Hardness |                                                                      |
| A <sub>1</sub> | 68                  | 82                  | 270                    |               |                                                                      |
|                | 96,600              | 116,600             |                        |               |                                                                      |
| A <sub>2</sub> | 110                 | 117                 | 360                    |               |                                                                      |
|                | 156,400             | 166,400             |                        |               |                                                                      |
| A <sub>3</sub> | 110                 | 117                 | 360                    |               |                                                                      |
|                | 156,400             | 166,400             |                        |               |                                                                      |
| B <sub>1</sub> | 68                  | 82                  | 270                    | ---           | Mech. properties of<br>screw same as A <sub>1</sub>                  |
|                | 96,600              | 116,600             |                        |               |                                                                      |
| C <sub>1</sub> | 84                  | 94                  | 275                    | 405           | Case hardness mea-<br>sured 0.05 mm<br>(0.00197 in.) from<br>surface |
|                | 119,500             | 133,700             |                        |               |                                                                      |
| C <sub>2</sub> | 85                  | 98                  | 260                    | 520           |                                                                      |
|                | 120,900             | 139,400             |                        |               |                                                                      |

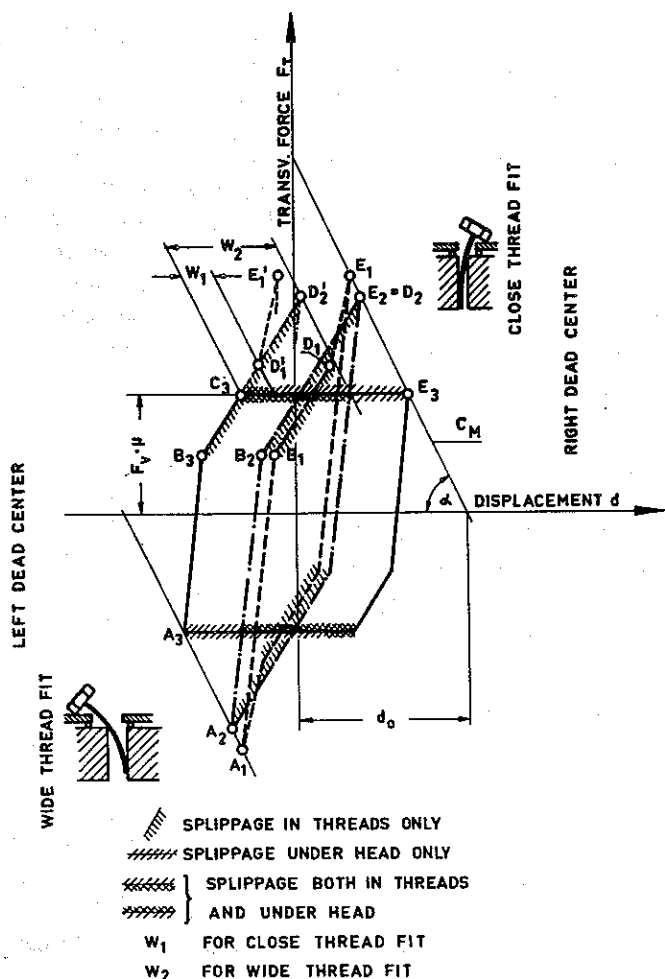


Fig. 11 - Schematic description of the self-loosening process under transverse vibration

It can be seen that the distance  $W_1$  for threads with close tolerance is smaller than the distance  $W_2$  for threads with wide tolerance. Therefore, screws with close thread fit must have a higher resistance to self-rotating loosening than screws with wide thread fit. These considerations explain that the prevailing torque type of locking screw works by reducing or eliminating the thread allowance by various means. The self-forming screws must be added to this category for the same reason.

According to Fig. 11, a bolted connection that slips under the bolt head will absorb the most energy; the hysteresis loop encloses a considerable larger area. Where the clamped parts are not separated by flat strips of needle bearings and where they are clamped together directly, the interaction of force, deflection, and displacement is basically the same. Merely the friction energy between the clamped parts ( $F_V \mu 2d_{\max}$ ) has to be generated additionally every cycle. When the transverse force  $F_T$  is large enough to force a relative displacement of the clamped plates, the same process as that described in Fig. 11 takes place. The use of strips of needle bearings between the clamped plates merely prevents galling; it does not favor either group of locking elements.

The diagrams recorded during tests have basically the same shape as the schematic drawing of Fig. 11. Fig. 12 shows the transverse force-displacement diagram of a non-locking screw  $A_1$  and of a free spinning locking screw  $C_2$ , recorded at a machine speed of 10 cpm. The diagram is presented for 100, 80, 60, 20, and 0% of the original bolt preload, which is also shown on the same plot as a function of displacement (upper part of chart). It can be seen that

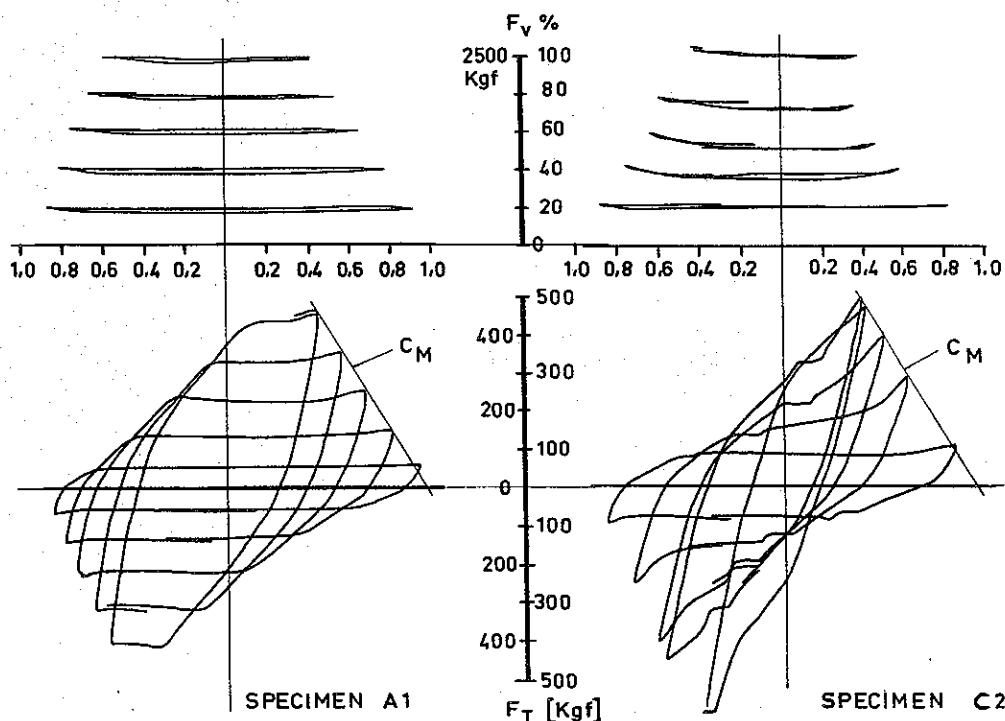


Fig. 12 - Preload  $F_V$  and transverse force  $F_T$  versus displacement "d" (hysteresis curve) for nonlocking screw  $A_1$  and free spinning locking screw  $C_2$ ; frequency 10 cpm

the nonlocking screw  $A_1$  starts to slip under the head in the fully preloaded condition; however, the screw  $C_2$  with serrated head bearing surface does not start to slip until the preload is decreased to 40% of the original preload. The tangent curves in Fig. 12 represent the load/deflection curve (spring constant  $C_M$ ) of the load transmitting parts of the machine.

Fig. 13 shows the complete loosening process of a nonlocking screw  $A_1$  recorded with an X-Y recorder at a test frequency of 10 cpm. The recorder was turned off at a remaining preload of 10% of original preload, but the screw had completely turned loose after 200 cycles.

Fig. 14 shows the same process for the same eccentric adjustment (zero load amplitude  $d_0 = \pm 0.8$  mm), but recorded on an oscilloscope during the 3000 cpm test frequency. Both diagrams, recorded at 10 cpm and at 3000 cpm, are equal when the different scales are taken into consideration; even the number of cycles causing complete loosening is approximately 200 for both speeds. This result gained in several tests shows that the self-loosening of screws is independent of frequency. It simply depends upon the occurrence of relative motions and on the length of such motions during one cycle. If the forces causing these relative motions are inertia forces, they are a function of the square of the frequency. In this case the frequency indirectly influences the loosening process.

This test method yielded interesting results when record-

ing the angle of rotation during loosening in addition to the force/displacement hysteresis curve. It becomes apparent that the loosening process of a nonlocking fastener starts with the very first cycle. The recorded angle of rotation loosening was 9 deg after 25 cycles; it was constant, since 0.36 deg rotation occurred per cycle. The recording of preload and angle of rotation as a function of displacement during 10 cpm tests gave an answer to the question of what causes the relative high loss of preload of all free spinning, self-locking fasteners at the beginning of the vibration process. This phenomenon was especially encountered at test series described later.

Fig. 15 shows the first 50 cycles at 10 cpm frequency for free spinning locking screws  $C_1$  and  $C_2$  (the first five cycles are recorded, then every fifth; screws seated on a soft washer). The screws lose 25% and 14.4% respectively of their original preload during the test. Recording the angle of rotation showed that the high loss of preload at the beginning was not only the result of the "digging in" of the teeth but also of a rotation of 1.3 deg for  $C_1$  and 0.6 deg for  $C_2$ . After 30 cycles, the rotating process stops and further loss of preload is due to brinelling of the teeth into the surface.

The loss of preload after 50 cycles is the same when the screws are seated on hardened washers; however, it is caused to a larger degree by rotation. That this process is dependent largely on the shape of the teeth becomes obvious by the following: The screws  $C_2$  having long rectangular teeth lost

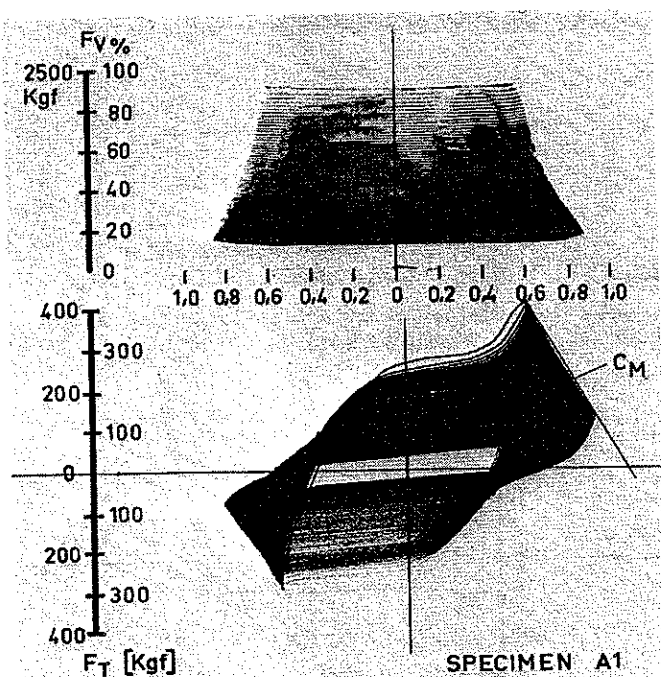


Fig. 13 - Preload  $F_V$  and transverse force  $F_T$  versus displacement "d", showing a continuous self-loosening process of a nonlocking screw  $A_1$ ; frequency 10 cpm

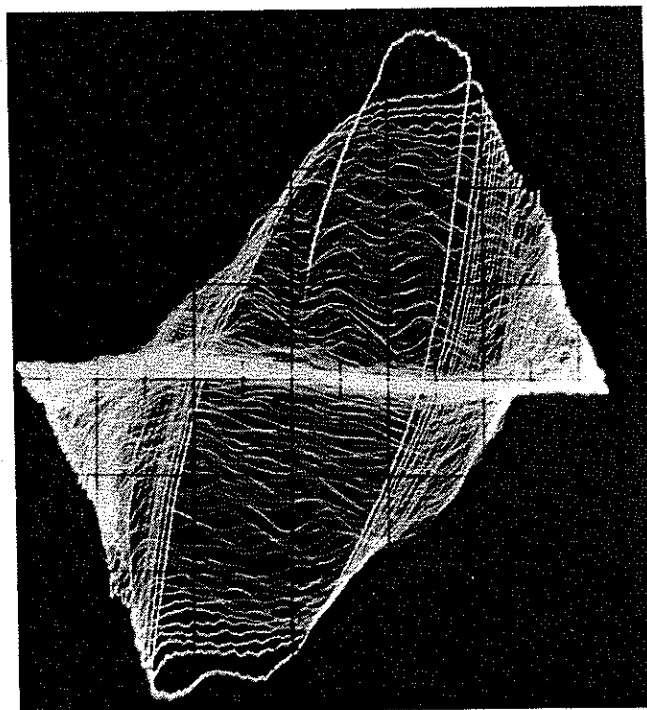


Fig. 14 - Transverse force  $F_T$  versus displacement "d", showing a continuous self-loosening process of a nonlocking screw  $A_1$ ; frequency 3000 cpm

## SELF-LOOSENING OF FASTENERS

only half as much preload after 50 cycles under equal conditions and turned only half the angle as the screw  $C_1$ , which has triangular high teeth.

Fig. 15 ( $C_2$ ) also shows clearly the two bends in the hysteresis curve which characterize the limits of thread slippage. The hysteresis curve recorded by the X-Y recorder has the same shape as the one drawn in Fig. 11, which shows a screw that does not slip under the head. This kind of test is primarily suited for evaluating rotating and brineling processes. It can be used especially well at slow testing speeds so that X-Y recorders may be employed. It is not so well suited for large test series at high testing frequencies when a statistical evaluation is desired, since in that case the hysteresis curves will have to be photographed from an oscilloscope. Fig. 16 shows maximum output of the vibration machine and actual hysteresis curves.

**VIBRATION PRODUCT** - A fastener connection that slips under alternating transverse force absorbs exactly that amount of energy represented by the included area of the hysteresis diagram. Assuming the extreme case of an absolutely stiff screw, all absorbed energy would be in form of friction and would be for 1/4 cycle:

$$E_F = d \cdot F_T \quad (8)$$

At the other extreme, pure bending of the screw (no friction), 1/4 cycle, would absorb

$$E_B = \frac{d}{2} F_T \quad (9)$$

In the second case the energy for one full cycle would be zero. The largest possible energy per 1/4 cycle for a given

eccentric adjustment would be the following rectangle, expressed by

$$\left( \frac{F_T}{2} \right) \left( \frac{d}{2} \right) = \frac{C_M}{4} d_0^2 \quad (10)$$

The limiting values of displacement and transverse force, the exceeding of which causes the preloaded screws to rotate loose, will be used for the rating of locking abilities of various locking elements. Both values are of equal importance, since a slipping joint will be kept from total separation if large displacements in connection with small forces, as well as large transverse forces in connection with relatively small displacements, are absorbed. It is therefore proposed to introduce the product of transverse force times displacement  $F_T d$  as a criterion for the locking ability of a locking element subjected to transverse displacements. The product will be called "vibration product" V.

The vibration product V was obtained in two ways: First, displacement "d" and transverse force  $F_T$  were recorded on a Visicorder together with the bolt preload. Values for "d" and  $F_T$  were then read from the Visicorder chart and multiplied. As a second method, the product was obtained by electrically multiplying the signals of the two amplifiers of displacement and transverse force. The multiplication circuit was built from commercially available components.

By multiplying "d" and  $F_T$  electrically it was possible to record the vibration product as a function of the (decreasing) preload on an X-Y recorder. Simultaneously, the tightening and rotation loosening angle was registered as a function of preload on the second channel of the X-Y re-

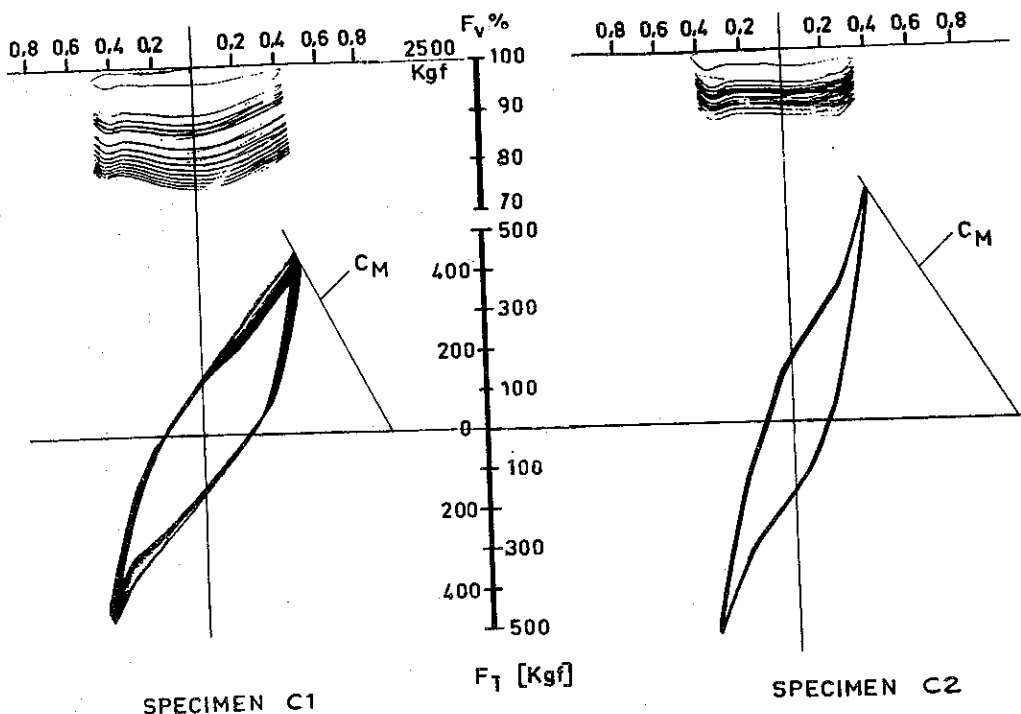


Fig. 15 - Preload  $F_V$  and transverse force  $F_T$  versus displacement "d", showing the first 50 cycles of vibration for free spinning self-locking screws  $C_1$  and  $C_2$ ; frequency 10 cpm

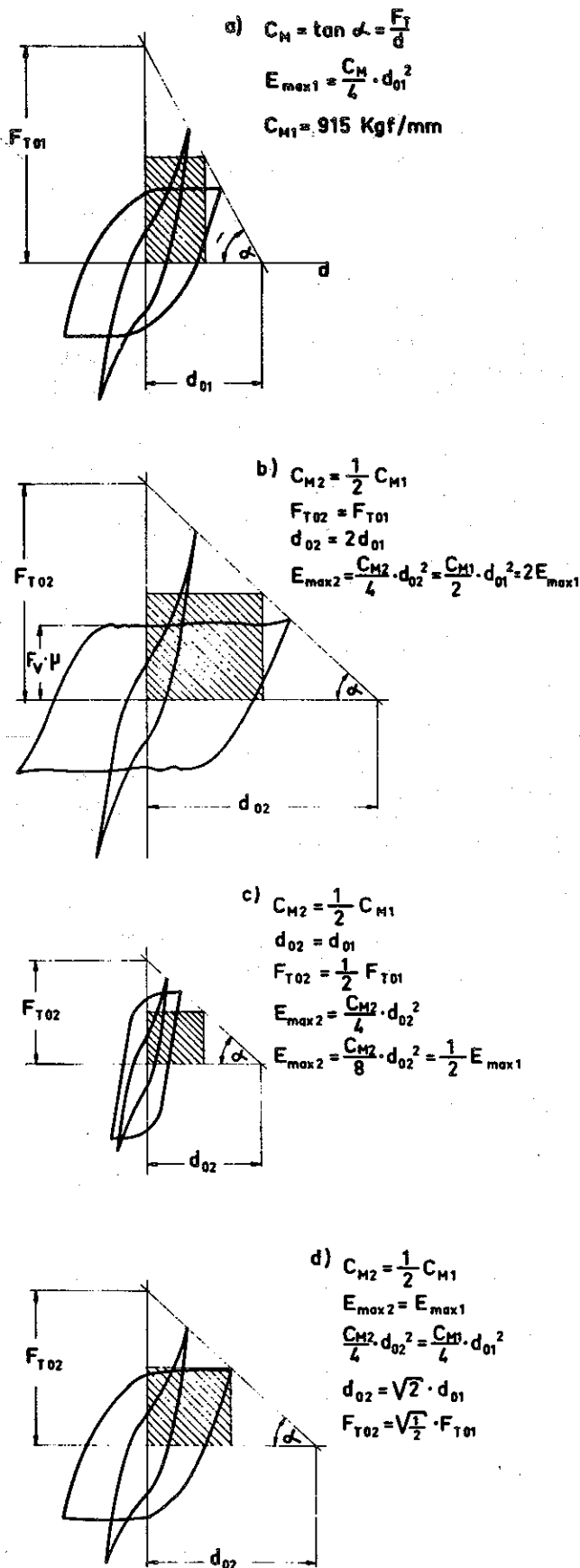


Fig. 16 - Maximum output of the vibration machine and actual hysteresis curves

corder. Recording the angle was done in the following way: First the screw was tightened to 500 kgf (1100 lb) preload (snug tight); then the screw head was connected with the angle recording potentiometer and the screw subsequently tightened to 100% preload.

Fig. 17 shows recordings obtained, as described above, of the vibration product and the tightening and rotation loosening angle as functions of the preload. The machine settings were as follows: eccentric adjustment to produce  $E_{\max} = 210 \text{ mm kgf}$ , frequency = 3000 cpm, no. of cycles = 3000 (provided total loosening did not occur). Fig. 17 shows also the performance of screw  $C_1$  after one tightening, tested at 3000 cpm frequency and an energy level of  $E_{\max} = 210$

mm kgf. When seated on a soft washer, the screw retains 50% of its preload after 3000 cycles, whereas the vibration product hardly changes. The angle  $A = 6$  deg, representing the rotation loosening, and the angle B corresponds to the amount of embedding during vibration. These can be determined when the line of the tightening angle of a plain unserrated screw of the same dimensions is added to the recording in Fig. 17. If this is done, one sees that the screw loses 12% preload by partial rotation loosening and 39% by embedding, a total of 0.08 mm (0.003 in.), which corresponds to the angle B. The total angle of tightening is 84 deg; this is 34 deg (C) more than the tightening angle 50 deg (D) of an unserrated screw of equal dimensions. The difference of 34 deg represents the embedding and the elastic deformation of the flange during tightening (0.15 mm = 0.006 in.). The same picture shows the loosening process of the same screw  $C_1$  under the same conditions except that it is seated on a hard washer. The screw rotates approximately 60 deg and loses all its preload.

Finally, Fig. 17 gives the performance of a screw  $C_2$ , which is of the same basic type. Loss of preload and vibration product are the same when seated on a soft washer; however, the tightening angle is only 40 deg compared with 84 deg for  $C_1$ ; the reason being a less deep embedding and a smaller elastic deformation of the flange. The rotation loosening angle of 6 deg is approximately the same. A significant difference shows up when the screw is seated on hard washers. In this case the screw  $C_2$  rotates loose only 6 deg after 3000 cycles, the shape of the angle diagram indicating that the loosening process has come to rest after 6 deg.

Many more tests were run in the same manner with the fasteners reported in Table 1. They were evaluated at different energy levels (zero load amplitude adjustments or eccentric positions), recording preload, angle of rotation, and vibration product, the latter being the electronically computed product of displacement and transverse force. Only a few of them are reported in Fig. 17, to describe the test procedure.

This mode of testing gives a quick basic knowledge of the performance of a locking element, a disadvantage being

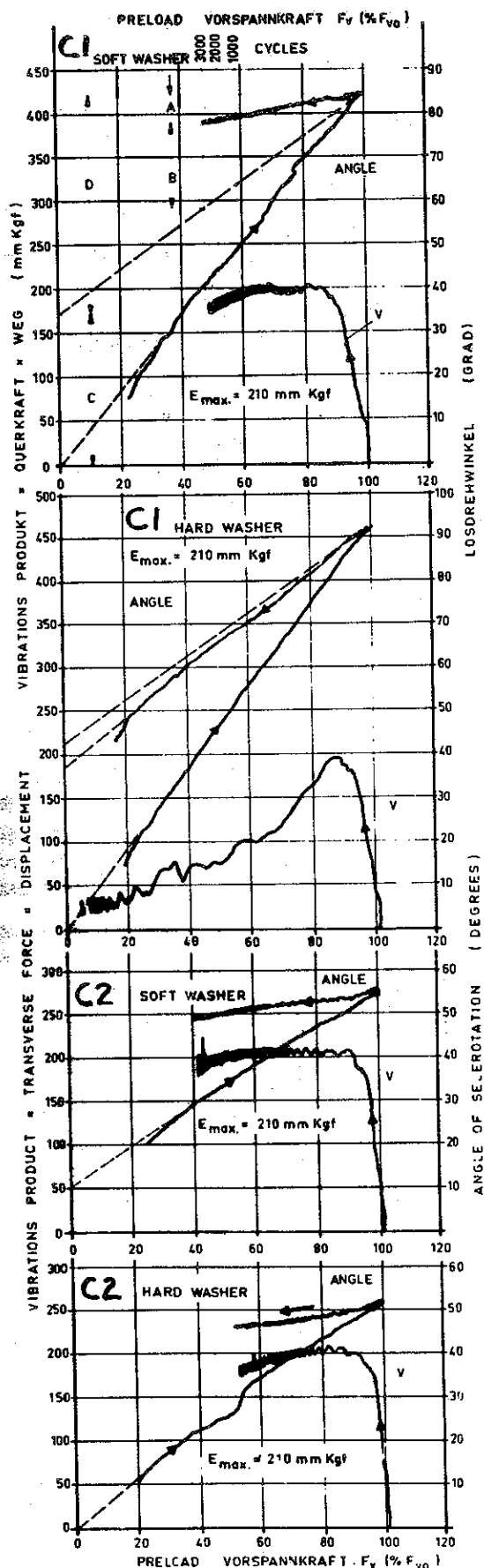


Fig. 17 - Vibration product (transverse force  $\times$  displacement) and loosening angle versus preload

that the number of cycles is not recorded. This can be avoided by lifting the pen of the X-Y recorder after fixed numbers of cycles, thereby creating interruptions in the diagram to provide reference to the number of cycles. The procedure can be recommended only for laboratories well outfitted with electronic equipment. The testing requires three bridge amplifiers, one X-Y<sub>1</sub>-Y<sub>2</sub> recorder, one electronic multiplication circuit, and one linear potentiometer to record angles.

A similar setup of electronic recording devices is needed if the vibration product, preload, and angle of rotation are plotted versus the number of cycles. When running the tests described here, it was not yet possible to feed the electronically multiplied values of transverse force and displacement directly into the Visicorder. They had to be recorded separately and then multiplied manually. When processed this way, the test result curves can be plotted in a way similar to S-N curves (Figs. 18-19). The difference, however, is that one does not obtain a clear datum point (such as fatigue failure) for a given test level whereas with increasing cycles a more or less steep declining vibration product curve can be plotted.

Either the curve goes through zero, which means that the screw vibrates loose completely, or the curve reaches asymptotically a final vibration product value, which means that the screw has vibrated loose and the loosening process has ended. The curves of low testing levels run flatter than the ones of high testing levels, and therefore they cross. After all curves are plotted in one diagram, a tangent curve can be drawn (V-N curve). This curve, being comparable to an S-N curve, represents for each number of cycles the maximum vibration product that the subject locking element can take when subjected to transverse vibrations.

In the V-N curves (Figs. 18-19), every datum point is the average of three tests. The recorded V-N curves confirm the theory of the influence of thread fit on nonlocking screws: Screws with national coarse thread 3/8 in. ( $A_2$  and  $A_3$ ) having a tighter thread fit than the ISO metric screws M10 withstand a higher vibration product at high number of cycles than do the metric ones. There is even a difference between screw  $A_2$  (tolerance 3A/3B) and screw  $A_3$  (tolerance 2A/2B); the V-N curve of  $A_3$  runs out with a higher vibration product than that for  $A_2$ . In the area of short cycles the V-N curves of metric hex head screws run with higher vibration product values, probably due to their larger bearing area under the head as compared with the bearing area of socket head cap screws  $A_2$  and  $A_3$ . The V-N curves for  $A_1$  and  $B_1$  are approximately identical, the smooth spring-washer ( $B_1$ ) having no ability to lock.

The V-N curves of free spinning screws  $C_1$  and  $C_2$  are practically horizontal; this means that these screws withstand the same high vibration product over a large number of cycles. Merely by seating the screw  $C_1$  on a hard washer,

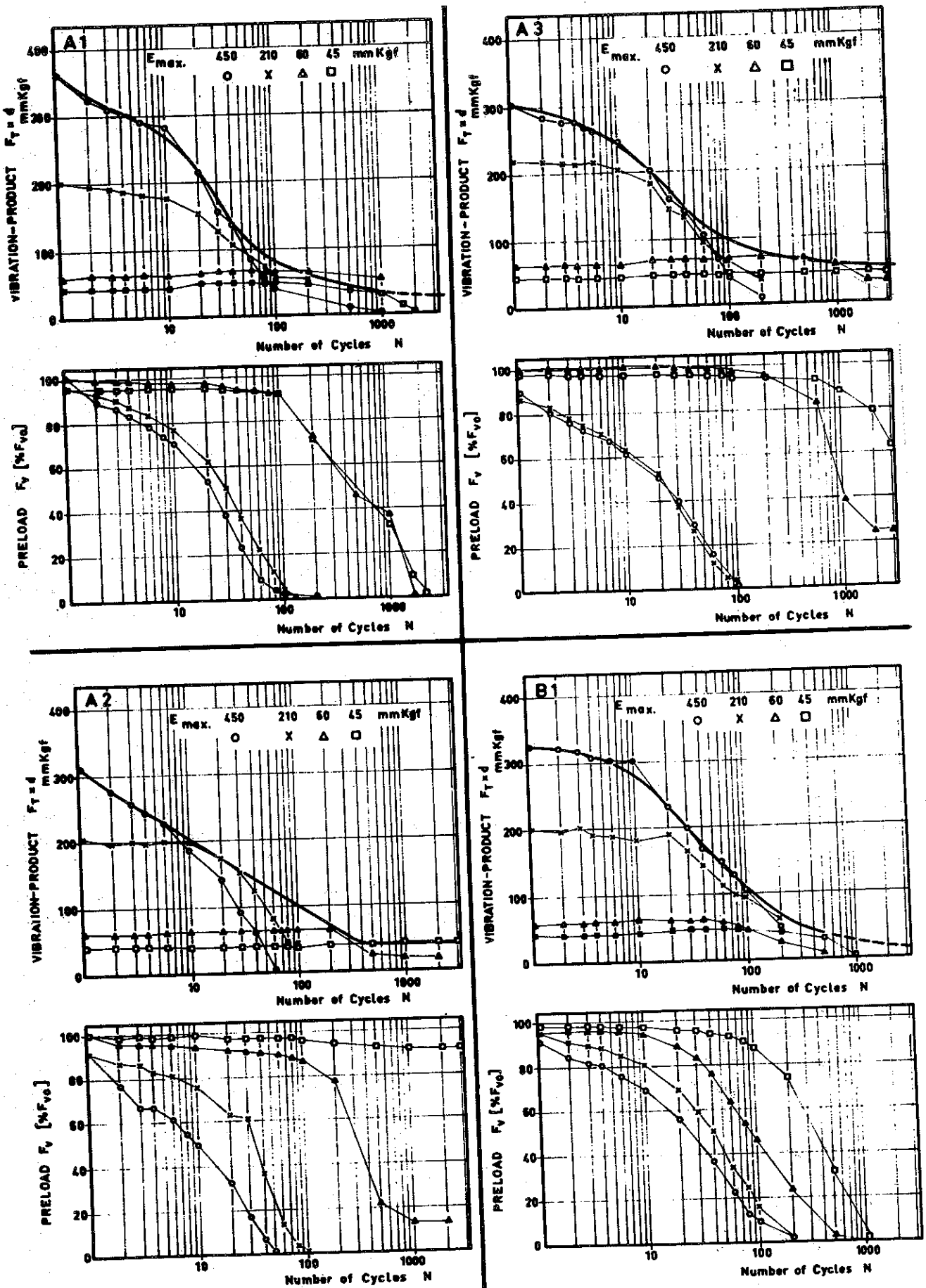


Fig. 18 - Vibration product and preload versus number of cycles (V-N and  $F_v$ -N curves) for specimens  $A_1$ ,  $A_2$ ,  $A_3$ ,  $B_1$

the performance becomes inferior; however, the screw  $C_2$  performs as well on hard as on soft washers.

The disadvantage of the testing procedure just described is that the V-N curves contain no information on the loss of preload. To describe the performance of a locking element completely in addition to V-N curves, so called  $F_V$ -N curves must be recorded, as has been done in Figs. 18 to 19.

**E-F-N CURVES** - The  $V$ - $F_V$  curves (Fig. 17) as well as the V-N curves (Figs. 18-19) show that the maximum vibration product is always close to the highest possible vibration energy that the machine is able to put out at the corresponding eccentric adjustment (zero load amplitude "d"). The fastener then either withstands this vibration product or rotates loose in relatively short time. It therefore appears to be advantageous to use the energy level

$$\left(\frac{C_M}{4}\right) d_0^2 \quad (11)$$

as a test level.

Beginning with the machine adjusted to produce a specific energy level corresponding to the desired testing level, one

plots as datum points those numbers of cycles at which the fastener retains 80, 50, and 20% of preload during its process of turning loose. After this is done for several testing levels, one connects all points of equal percentage preload and obtains E-N curves with the three preloads as parameter.

Figs. 20 and 21 show test results that were processed in this manner. The  $E$ - $F_V$ -N curves or abbreviated E-F-N curves are not so precise as the earlier described V-N and  $F_V$ -N curves. However, they have the advantage of containing the preload as parameter. If the designer wants to select a locking element that must retain high preloads during application, he will choose an element that has an 80% E-F-N curve at high energy values. If he is only trying to keep the joint from total separation and prevent loss of the fastener, he can select an element with a high 20% E-F-N curve.

Besides the advantage just described, this testing method requires only one bridge amplifier as the total recording instrumentation. It is mandatory that either the manufacturer of the machine or the user determine conclusively the spring constant  $C_M$  of all load transmitting parts of the machine. After this is done, the eccentric can be adjusted to produce the zero load amplitude  $d_0$ , which yields the de-

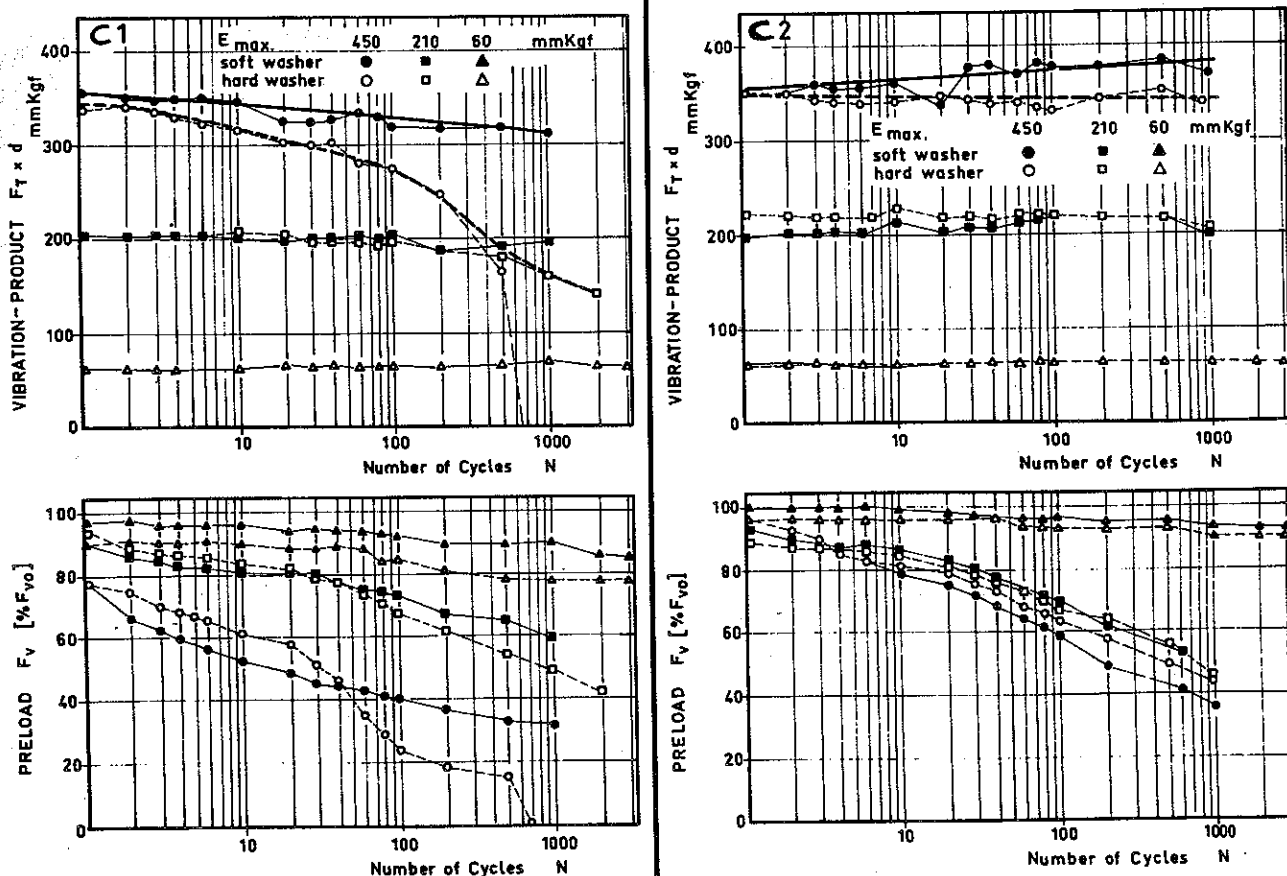


Fig. 19 - Vibration product and preload versus number of cycles (V-N and  $F_V$ -N curves) for specimens  $C_1$ ,  $C_2$

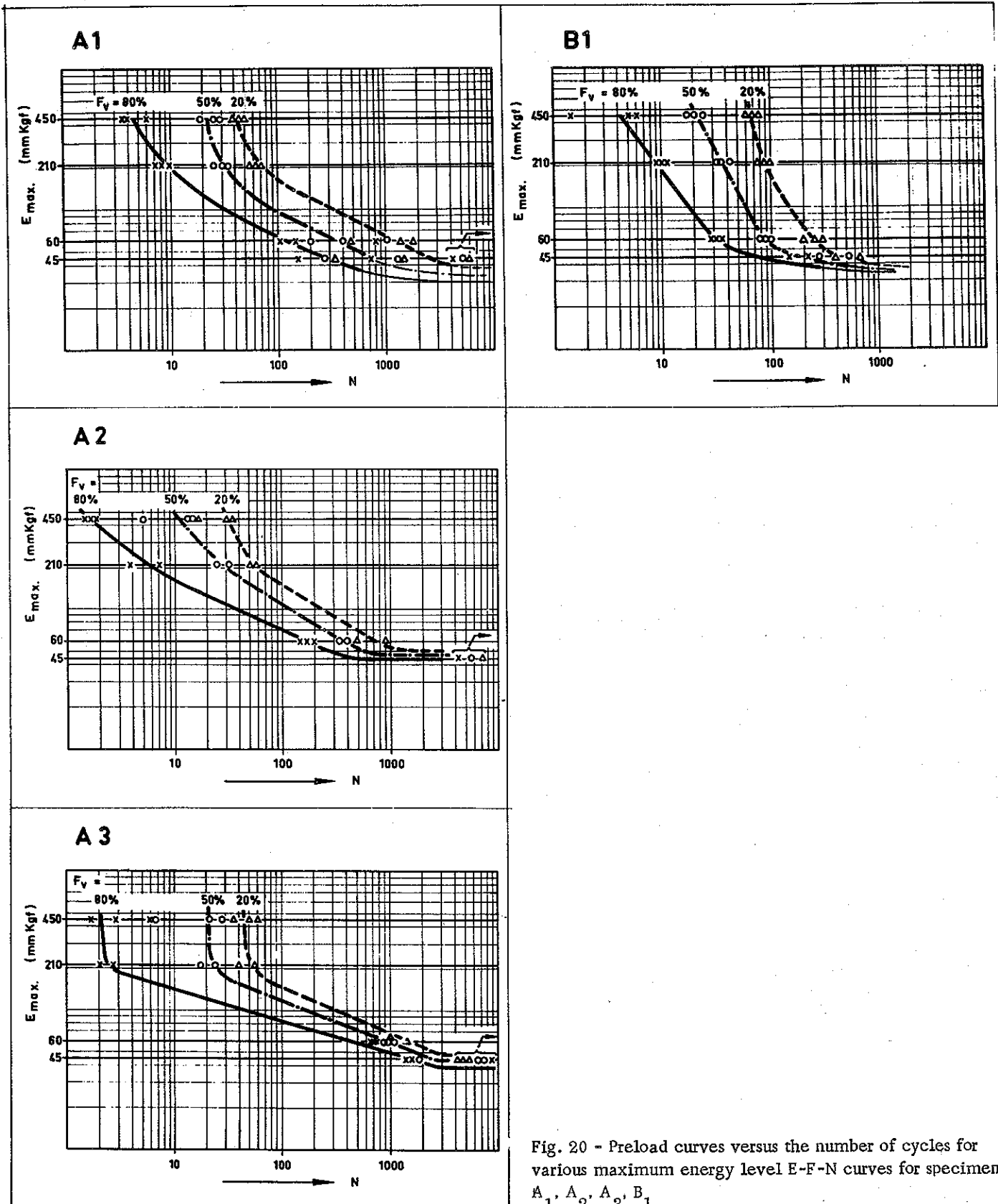


Fig. 20 - Preload curves versus the number of cycles for various maximum energy level E-F-N curves for specimens  $A_1, A_2, A_3, B_1$

sired energy level according to the equation

$$E_{\max} = \left( \frac{C_M}{4} \right) d_0^2 \quad (12)$$

As presented in Fig. 16, results gained from different machines must be comparable, even if the spring constants of the load transmitting parts are in the ratio of 1:2. Fig. 16 (d) shows that transverse vibration tests, run at equal energy levels,  $E_1(\max) = E_2(\max)$ , and on machines with spring constants in the ratio of 1:2 resulted in comparable force/deflection/displacement processes. If, as reference for the test levels, the transverse force or the zero load amplitude (eccentric adjustment) were used in the case of two machines with 1:2 spring constants, the results would be very different force/deflection/displacement processes; see Figs. 16(b) and (c).

The procedure to generate E-F-N curves is as follows:

1. Adjust the zero load amplitude with the eccentric of the vibration machine to obtain the first desired energy level:

$$d_0 = 2\sqrt{\frac{E_{\max}}{C_M}} \quad (13)$$

2. Preload the specimen to 75% of proof load; read preload off bridge amplifier.
3. During vibration process, read numbers of cycles corresponding to 80, 50, and 20% of preload  $F_V$  as indicated on bridge amplifier.
4. Repeat process at lower energy level until the preload does not drop below 80% after a predetermined number of cycles. Run several tests at each energy level, similar to fatigue testing, so that a statistical evaluation of the results is possible.

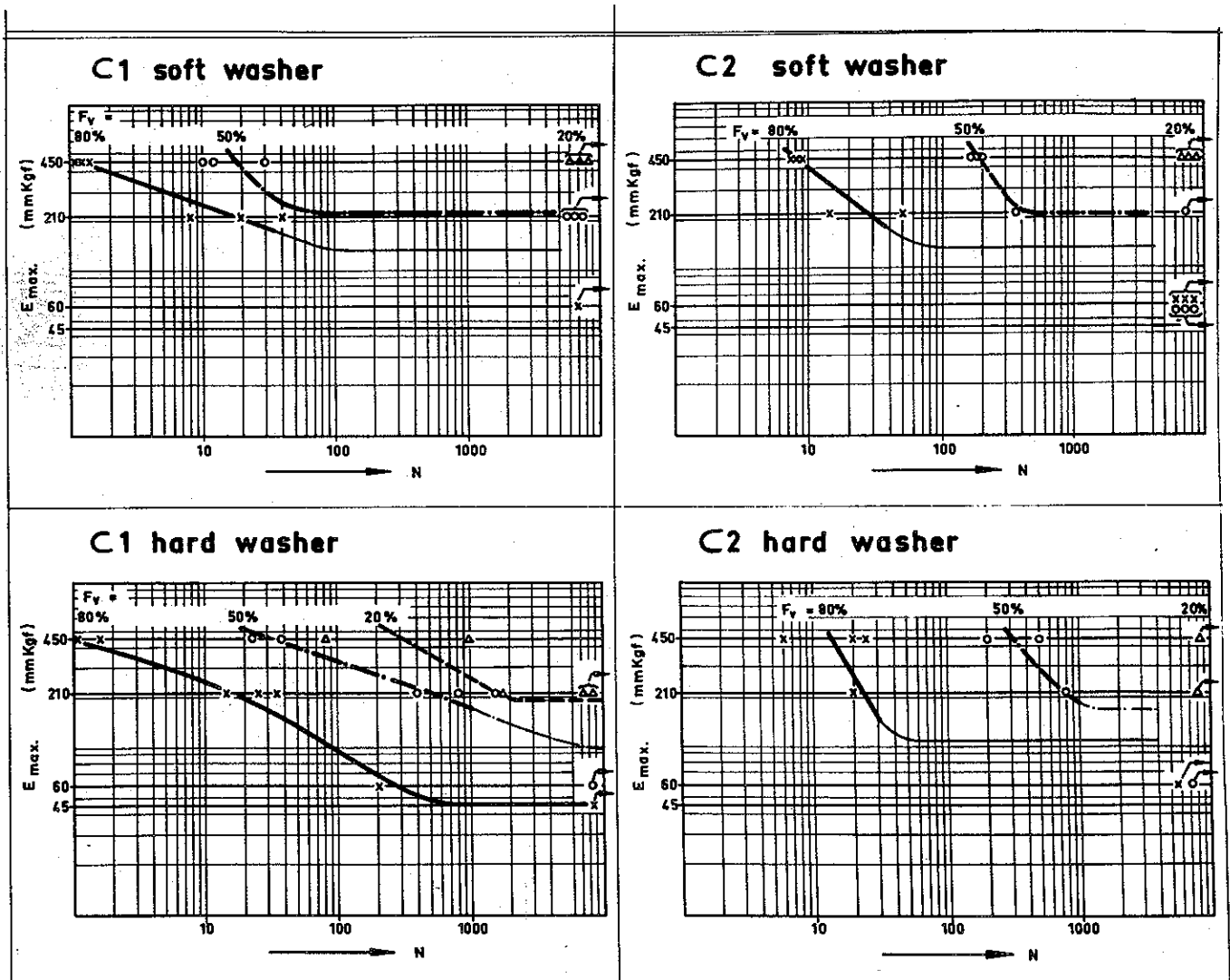


Fig. 21 - Preload curves versus the number of cycles for various maximum energy level E-F-N curves for specimens  $C_1$ ,  $C_2$  on hard and soft washers

**SURFACE INTEGRITY** - The value of self-locking elements with serrated head bearing surface is often diminished by the damage they cause to the surface of the clamped parts, despite their excellent locking performance. They cut notches in the surface of the clamped parts, which reduce the fatigue strength of sheet metal designs as well as of other machine components. The new vibration machine can be adapted to evaluate this influence, too, as is shown in Fig. 9. Fig. 22 shows the test setup for a sheet metal specimen.

The specimens used for the tests described here were made from 1 mm (0.04 in.) cold rolled body sheet, bent into a U-shaped channel. The sheet metal specimens were tested with two types of screws with serrated bearing surface ( $C_1$  and  $C_2$ ). For comparison, tests with nonlocking screws were run. The fasteners were tightened twice to 100% pre-

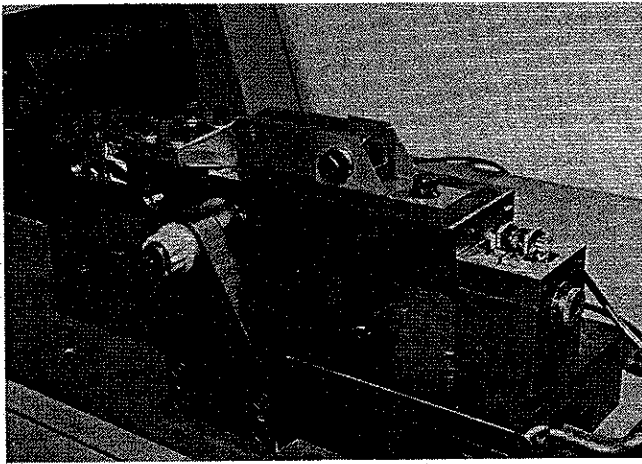


Fig. 22 - Testing device for clamped sheet metal specimens

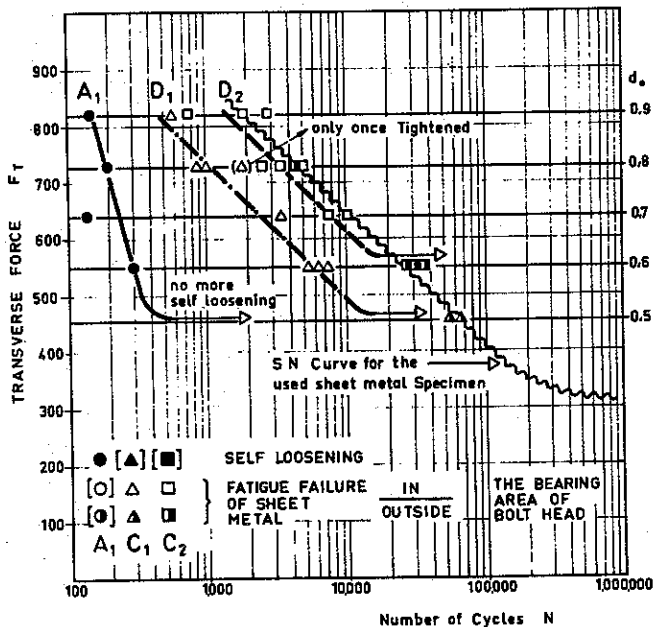


Fig. 23 - Transverse force versus number of cycles,  $F_T$ -N curves for clamped sheet metal specimens

load, unscrewed, cleaned, and seated for a third time to 100% preload prior to vibration testing. After taking out the flat strips of needle bearings, the tests could be run like standard pushpull fatigue tests.

The results are shown in Fig. 23. Some typical fatigue failures caused by screw  $C_1$  are shown in Fig. 24, and those caused by screw  $C_2$  in Fig. 25. These tests make clear the difference in the shape of the serration of the two screws.  $C_1$  cuts an annular notch, triangular in cross sections, as can be seen in the washer left over from other tests (Fig. 24). The fatigue failure occurs along that annular notch; as a result, a washerlike piece is cut out by the screw. The screw  $C_2$  has long radial teeth, the edges of which run out continuously into a smooth outer bearing surface. The damage done to the seating surface is minute, and therefore fatigue failures occur away from the fastened area, as is shown in Fig. 25. For this reason, the  $F_T$ -N curve of screw  $C_2$  is close to the S-N curve of the undamaged sheet specimen.

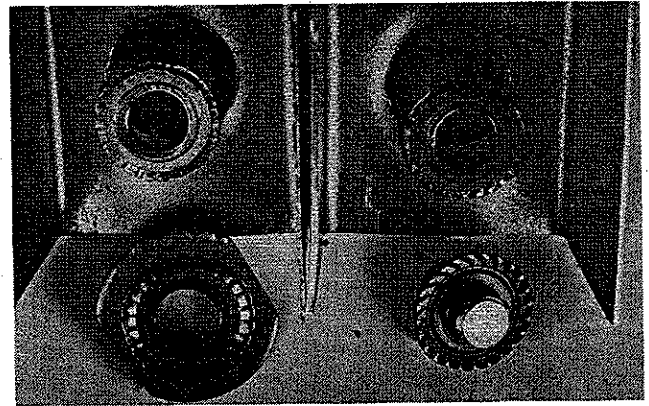


Fig. 24 - Fatigue failure of sheet metal specimens clamped with free spinning self-locking screws  $C_1$ ; shape of teeth and removed washer after vibration

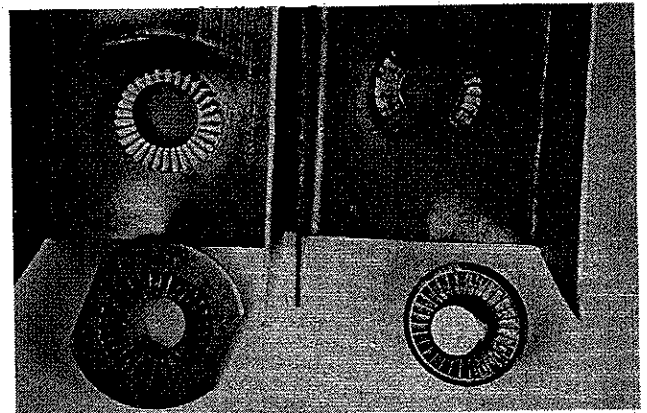


Fig. 25 - Fatigue failure of sheet metal specimens clamped with free spinning self-locking screws  $C_2$ ; shape of teeth and removed washer after vibration

## CONCLUSION AND FUTURE TEST PROGRAM

Self-loosening characteristics of fasteners can be achieved with a new vibration machine that generates relative motion in the clamped parts perpendicular to the axis of the fastener. Several values characteristic of the process can be measured during the test. These include preload, transverse force, displacement, and angle of rotation. The described machine is suited for further research work to study the mechanism of self-loosening. It must therefore be equipped with a series of electronic recording instruments.

The machine is also suited for large scale testing and inspection testing. For this kind of work, only one bridge amplifier is needed to record the preload. When so equipped, E-F-N curves can be recorded for large scale and inspection tests. Thereby a quantitative testing of locking performance of all kinds of locking elements becomes possible. The fasteners are tested in a preloaded condition. Test results of different locking types of fasteners such as those having prevailing torque, and free spinning torque, and adhesive type screws can be compared. The spring constant of the machine must be known. Results obtained from machines with different spring constants must be comparable.

The following details must be resolved preparatory to the specification of a standard test method:

1. Influence of screw length.
2. Performance of nuts tested on special externally threaded inserts or studs.
3. Vibration endurance limit; that is, after how many cycles has the self-rotation of a fastener or the loss of preload come to rest.
4. Influence of hardness of clamped parts; specification of standard test washers.
5. Influence of thread tolerance.
6. Influence of thread pitch.
7. Reusability.

Of special interest is the determination of that number of cycles after which no more self-loosening occurs. The exact shape, particularly the horizontal part, of all  $F_V$ -N, V-N, E-F-N, and  $F_T$ -N curves presented in this paper is

not yet known because only a limited number of tests have been run so far. It can only be estimated that the "vibration endurance limit" is smaller than the fatigue strength of materials and structures.

Furthermore, tests should be run to establish the influence of superimposed forces other than transverse forces. First, screening tests using angles of 15, 30, 45, and 60 deg showed that under an equal external force, screws loosened faster when tested under smaller angles. It can be assumed that the pure transverse force applied at an angle of 180 deg represents the most severe conditions. The influence of lubrication, tolerances, and surface finish should be investigated. The question is whether there is a correlation between static tests (breakaway torque and so on) and vibration

tests. The answer to this will probably be found by statistical methods.

When enough tests have been run to make it possible to give statistically evaluated data concerning the locking performance of locking elements (E-F-N curves, endurance limits for  $F_V = 80\%$ , 50%, 20%), then a true analysis of the value of locking elements will be possible. In addition to the then measurable value of locking performance the following factors must be taken into consideration: reusability (a), surface integrity and fatigue strength of clamped parts (b), fatigue strength of clamping parts (c), compatibility with clamped material and hardness (d), installation cost (e), and price (f). Factors (a) and (b) can also be evaluated with the described vibration machine and the proposed methods. When a catalogue with E-F-N curves is established by a large number of tests, short term tests with specific energy levels can be standardized as inspection tests, the minimum number of cycles for  $F_V = 80\%$ , 50%, or 20% have to be specified.

It was not intended to give at this stage of study an evaluation of the locking performance of the tested specimens. The results, therefore, shall only be considered as a means for the demonstration of the operation of the vibration machine and for the discussion of several possible test methods. A value analysis covering all the above mentioned factors will show which locking element is the optimum one for the actual application.

## SUMMARY

Preloaded screws (or nuts) rotate loose, as soon as a relative motion in the thread takes place. This motion cancels the friction grip and originates an inner off-torque proportional to the thread pitch and to the preload. The inner off-torque rotates the screw loose if the friction under nut or bolt head bearing surface is canceled by relative motions.

A dynamically loaded joint fails in most cases either by fatigue or by rotation loosening of the fastener. Even the fatigue failure is often initiated by partial loosening. If it is impossible to avoid relative movements by design, self-locking screws must be used. This means either that relative movements in threads must be diminished or avoided so that the formation of the inner off-torque will be thereby prevented, or that the formation of inner off-torque must be blocked up in the head or nut bearing surface.

Dynamic transverse forces are more dangerous than dynamic axial forces. Axial forces cause relative movements through expansion of the nut thread; transverse forces cause relative movements through rocking action of the screw in the internal thread (or rocking motions of the nut on the external thread). The relative motions caused by transverse forces are larger.

No existing dynamic test method measures the locking performance of locking elements. A new vibration machine is described which generates transverse forces and displacements in preloaded joints as well as combinations of trans-

verse and axial forces. Displacement, transverse forces, preload, and angle of rotation can be measured. Additionally, the fatigue strength of bolted sheet metal joints can be evaluated, which is a means of rating the damage that a locking element causes to the surface of the clamped parts.

Various test methods using different recording instruments are described and discussed in relation to nonlocking and self-locking screws of sizes M10 and 3/8 in. NC. The influence of thread fit on the loosening resistance of nonlocking screws, the influence of the shape of the serration of free spinning screws on locking performance, the influence of frequency on the self-loosening process, and fatigue strength of clamped sheet specimen were studied.

Finally, a simplified method for large scale testing and inspection testing is proposed for which only one bridge amplifier is needed with the vibration machine. This method produces energy/number of cycles curves (similar to S-N curves) with 20, 50, and 80% preloads as parameter (E-F-N curves). This test method could be developed into a standard test method after solving some detail questions. It would yield measurable values for the locking performance of nuts and bolts under dynamic loading. It could become the basis of a real value analysis for the use of locking elements.

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#### SYMBOLS AND ABBREVIATIONS

|       |                                                                                                                                                                                       |                 |                                                                                                                                                                                                          |
|-------|---------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------|-----------------|----------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------|
| $d$   | = Displacement of clamped components under transverse force, mm or in.                                                                                                                | $C_P$           | = Spring constant of clamped plates or components, kgf/mm or lb/in.                                                                                                                                      |
| $d_0$ | = Displacement of U-shaped top part of vibration machine in unloaded condition (zero load amplitude), mm or in.                                                                       | $C_S$           | = Spring constant of screw, kgf/mm or lb/in.                                                                                                                                                             |
| $d_1$ | = Major diameter of thread, mm or in.                                                                                                                                                 | $D_H$           | = Friction diameter of a bolt head or nut, mm or in.                                                                                                                                                     |
| $d_2$ | = Pitch diameter of thread, mm or in.                                                                                                                                                 | $E_B$           | = Energy for elastic deflection of a bolt in a transversely loaded joint during 1/4 cycle, mm kgf or in.-lb                                                                                              |
| $s$   | = Dilation of nut under axial load, mm                                                                                                                                                | $E_F$           | = Energy absorbed by friction in a sliding joint during 1/4 cycle, mm kgf or in.-lb                                                                                                                      |
| $z$   | = Amount of embedding (Brineling, "Setzen," plastic deformations) during dynamic dynamic loading of a joint, mm or in.                                                                | $E_{max}$       | = Maximum output of vibration machine for a certain eccentric adjustment, mm kgf or in.-lb                                                                                                               |
| $A$   | = Tightening or torque factor, which considers the preload deviations caused by differences of friction coefficients and applied torque in the design equation for bolted connections | $F$             | = Axial force in a fastener, kgf or lb                                                                                                                                                                   |
| $C_M$ | = Spring constant of load transmitting parts of vibration machine, Fig. 16 (a); kgf/mm or lb/in.                                                                                      | $F_{max}$       | = Maximum axial force in a prestressed fastener under working load, kgf or lb                                                                                                                            |
|       |                                                                                                                                                                                       | $F_R$           | = Residual clamping force necessary for maintaining grip friction of clamped components, kgf or lb                                                                                                       |
|       |                                                                                                                                                                                       | $F_T$           | = Transverse force in a bolted joint as well as in the vibration machine, kgf or lb                                                                                                                      |
|       |                                                                                                                                                                                       | $F_{T_0}$       | = Transverse force in vibration machine with eccentric adjustment $d_0$ , kgf or lb                                                                                                                      |
|       |                                                                                                                                                                                       | $F_V$           | = Preload in a fastener, kgf or lb                                                                                                                                                                       |
|       |                                                                                                                                                                                       | $F_W$           | = Working load in a bolted joint, kgf or lb                                                                                                                                                              |
|       |                                                                                                                                                                                       | $\Delta F_V$    | = Loss of preload in a bolted joint by embedding, kgf or lb                                                                                                                                              |
|       |                                                                                                                                                                                       | $L$             | = Weight, kgf or lb                                                                                                                                                                                      |
|       |                                                                                                                                                                                       | $Q$             | = Transverse force for moving loads resting on a horizontal plane and a slope, kgf or lb                                                                                                                 |
|       |                                                                                                                                                                                       | $T_{off}$       | = Off-torque for a threaded fastener under static conditions, m kgf or in.-lb                                                                                                                            |
|       |                                                                                                                                                                                       | $V$             | = Vibration product equals displacement "d" times transverse force $F_T$ in a transversely vibrated joint, a significant factor for the locking performance of a self-locking fastener, mm kgf or in.-lb |
|       |                                                                                                                                                                                       | $\alpha$        | = Angle of load deflection curve of load transmitting machine parts, deg                                                                                                                                 |
|       |                                                                                                                                                                                       | $\varphi$       | = Slope angle and helix angle, deg                                                                                                                                                                       |
|       |                                                                                                                                                                                       | $\mu$           | = Coefficient of friction in threads, $\mu = \tan \rho$                                                                                                                                                  |
|       |                                                                                                                                                                                       | $\mu_H$         | = Coefficient of friction in threads, under head or nut                                                                                                                                                  |
|       |                                                                                                                                                                                       | E-F-N curves    | = Curves of constant preload for certain vibration energy levels versus number of cycles                                                                                                                 |
|       |                                                                                                                                                                                       | $F_T$ -N curves | = Transverse force versus number of cycles, significant for surface integrity of clamped components                                                                                                      |
|       |                                                                                                                                                                                       | $F_V$ -N curves | = Decreasing preload under transverse vibration versus number of cycles                                                                                                                                  |
|       |                                                                                                                                                                                       | V-N curves      | = Vibration product versus number of cycles                                                                                                                                                              |

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