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The Strength of Large Bolts Subjected to Cyclic Loading

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The strength of a bolted assembly which carries a fluctuating load is determined by two distinct factors: (a) the magnitude of the cyclic load that can safely be carried by the bolts, and (b) the design of both the bolts and bolted parts, which determines the proportion of the externally applied load transmitted to the bolts. Accordingly, the paper is divided into two sections.

In the first section various problems of design as well as the effect of pre-stressing of the bolts are considered, while the second part is devoted to fatigue tests which have been carried out on 3-inch and 1-inch bolts of material, method of manufacture and design similar to those commonly used in heavy marine oil engines.

These tests indicate that the fatigue strength of large bolts having lathe-cut threads is liable to vary over a wide range and the strength under cyclic loading may be reduced to only 16 per cent of the load required to cause complete failure under static loading. This variation is attributed largely to slight inaccuracies in thread production. It is concluded that a small reduction in diameter of the shank of the bolt has several advantages but such a practice does not affect the fatigue strength of the bolt.

On the whole the 1-inch bolts were found to be capable of carrying a slightly higher fluctuating stress than those of 3-inch diameter, but insufficient data are yet available to draw any definite conclusions on the effect of size or pitch of the threads.

INTRODUCTION

The failure of marine engine parts in service can almost invariably be attributed to fatigue of the metal due to repeated application of stress, the effect of which is greatly intensified where corrosive conditions are present. Bolts, particularly those used in the running gear of main and auxiliary reciprocating machinery, are no exception and when a fracture occurs in a large bolt, such as those used in direct coupled oil engines, evidence of a spreading crack commencing at some point of stress concentration—whether in the screwed section, at a surface tool mark, at a fillet, or at some fault in the material—is invariably revealed. Fortunately such failures are rare in modern engines but nevertheless they are not unknown, as members of this Institute will be aware.

In recent years some attention has been given to the investigation of the strength of bolts of the type used in the aircraft engineering and automobile industries but knowledge of the strength of large bolts under cyclic loading has been entirely lacking. The British Shipbuilding Research Association therefore decided to institute a systematic series of fatigue tests on bolts of the type used in marine engines to investigate the factors influencing the strength when subjected to repeated stresses. Such an investigation is necessarily a long term project but some data have been obtained on the fatigue strength of both 3-inch and 1-inch bolts which also throw some light on the causes of failure. The results of these experiments are discussed in the second part of this paper.

The strength of bolts used to secure the parts of a dynamically loaded assembly depend to a large extent on factors other than the design of the bolts themselves; accordingly it was considered desirable to devote the first section of the paper to

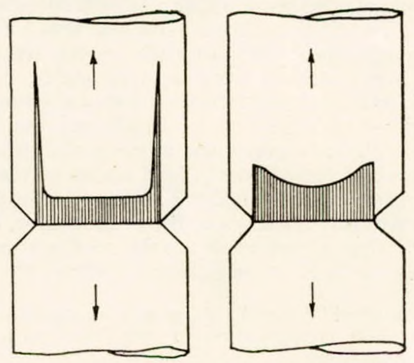


FIG. 1—*Effect of notch on stress distribution across tension member*

a general discussion of the problem. It was also thought that some introductory remarks on the fatigue of metals would not be out of place in a paper of this nature since the conventional text-book methods of stress computation, with which all engineers are familiar, takes no account of the effect of fluctuating stresses or the inherent stress raisers caused by minute defects in the material or discontinuities introduced by necessity in the design. Fig. 1 is included by way of an example to illustrate the great increase in stress set up at the base of a notch* in a member loaded in tension. The ratio between the theoretical maximum stress at the notch and the nominal stress across the section, obtained by conventional methods, is known as the stress-concentration factor, which may be derived in many cases by theoretical stress analysis or experimental methods such as the use of photo-elastic models. This stress-concentration factor increases with the sharpness of the notch.

It is very seldom that machine parts fail by being stressed beyond the ultimate strength yet it is on this basis, with the use of a suitable factor of safety, that most design is carried out, mainly because of the lack of fatigue data relating to full size components. When a part is tested to destruction under static load an appreciable volume of the metal yields before fracture occurs but one of the notable features of a fatigue fracture is that cracking takes place with very little deformation, even though the material may exhibit a high degree of ductility under static loading. Such a crack may form after relatively few load applications if the stress is sufficiently high but, on the other hand, cracking may be delayed until many thousands or even millions of loading cycles have been applied. The reason for this is that at a point where there is some localized weakness in the material or a localized increase in stress, plastic slipping occurs within the crystalline grains; this may develop into a spreading crack after many repetitions of stress if the capacity of the material for further strain hardening, which arises from such plastic deformation, is locally exhausted.

The rate of propagation of the crack depends on the stress gradient in the vicinity of the notch and the magnitude of the peak stress of the cycle. It was thought until recently that once a crack started it would spread across the section, progressively weakening it, until fracture occurred suddenly but the results of investigations† carried out on mild steel suggest that it may be possible for large components to continue to operate satisfactorily under controlled stress conditions even with a fatigue crack present. In service, however, particularly in the case of marine propelling machinery, load variations do not occur in an orderly and predictable manner and it may happen that stresses sufficiently high to extend the crack occur infrequently when for some reason the part is subjected to higher loads than usual. The cumulative effect of such overloads may therefore only lead to ultimate failure of a large component after years in service.

In studies of the fatigue of metals the usual method of investigation is by determining the life of a number of carefully prepared identical specimens under reversed stresses of different magnitudes. The first specimen is usually tested at a relatively high stress range and subsequent test pieces are subjected to successively lower stresses until a stress range is reached at which a specimen remains unbroken after an arbitrary number of stress cycles which for steel is usually ten millions. When stress range is plotted against the number of cycles to fracture on semi-logarithmic co-ordinates, S-N curves of the type shown later in this paper are obtained. The range of stress at which the curve flattens out is known as the fatigue limit, below which it is assumed that a test piece would have an indefinite life. In some cases the S-N curve may not become absolutely horizontal, particularly when testing steel components containing stress raisers, but after about 10 million cycles the slope becomes so small that the range of stress for this endurance is likely to be only very little more than the safe range of stress for an infinite number of stress cycles.

The value of the fatigue limit obtained from a laboratory test also depends on the type of loading. A testing machine may be designed to apply direct axial, flexural or torsional stresses so that any value for the fatigue limit must be qualified by the type of loading applied.

The type of stress induced may be further classified as one of the following: —

(a) A "fluctuating" stress which varies between a maximum and minimum of the same sign (as in axially loaded bolts).
(b) A "repeated" stress where the minimum value of the loading cycle is zero.
(c) An "alternating" stress varying between a maximum and a minimum of opposite sign, or
(d) As a special case of (c) a "reversed" stress varying between a maximum and minimum of equal magnitude and opposite sign.

It will be seen that for a constant stress range the only difference between the various loading systems given above is the mean value of the cyclic stress applied. The work of early investigators indicated that the limiting stress range‡ decreased as the value of the mean stress of the cycle increased. Consequently, in order to allow the safe range of stress to be estimated for any mean stress different from that for which test data may be available, various relations between mean stress and the the limiting stress range have been put forward from time to time. It is now well established, however, that the relation between mean stress and limiting stress range varies both with the material under test and the type of applied stressing and, as far as engineering steels are concerned, the effect of mean stress is much less than is often assumed.

This point is illustrated by the curve shown in Fig. 2 in which the limiting stress ranges for a number of mean stress values are plotted for the steel used in the investigation of the strength of bolts reported in this paper. These results were obtained from ½-inch diameter unnotched specimens tested under direct axial loading. It will be noted that the drop in fatigue strength with increase of mean stress, although quite definite, is surprisingly small even though at the high mean stress values the maximum stress of the loading cycle approaches

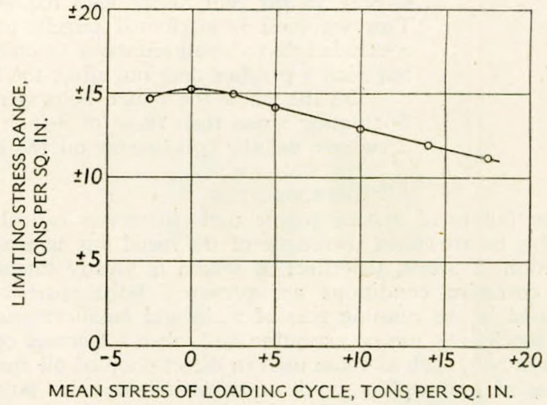


FIG. 2—*Effect of mean stress of loading cycle on limiting stress range of unnotched specimens (mild steel, 28-32 tons per sq. in. U.T.S., as used for test bolts)*

* The term "notched" is used to denote a specimen or component containing a stress concentration of any type, such as a screw thread, groove, hole, etc.

† Phillips, C. E. and Heywood, R. B. 1951. Proc.I.Mech.E., Vol. 165, p. 113. "The Size Effect in Fatigue of Plain and Notched Steel Specimens Loaded Under Reversed Direct Stress".

‡ The term "limiting stress range" is used throughout this paper to denote the maximum stress range that can be applied for an indefinite number of cycles without failure, where the mean stress of the loading cycle is not zero.

the ultimate strength of the material. It may therefore be taken that over the normal range of working stresses the effect of mean stress of the loading cycle is not of very great significance in the case of mild steel.

It has been found from laboratory tests that the fatigue limit of polished unnotched steel specimens is roughly proportional to the ultimate tensile strength of the material but the safe range of stress is considerably reduced if the specimen is notched. Unfortunately, the degree to which the strength is reduced by a given notch varies considerably for different qualities of steel and therefore the effect of introducing a stress raiser cannot be predicted for a particular steel unless actual tests are carried out. The higher strength alloy steels in general are more notch sensitive than low carbon steels, i.e. they suffer a greater reduction in strength for a given notch, and therefore in many applications there may be no advantage in the use of high tensile steels. Indeed, there are cases on record where the substitution of such a steel in place of an ordinary mild steel has resulted in a reduction in strength under cyclic loading.

In general the reduction in strength resulting from the presence of a notch has been found to be less than would be expected from application of the theoretical stress concentration factor mentioned above. The ratio between the actual fatigue limits of an unnotched specimen and one with a notch is therefore termed the effective stress concentration factor or strength reduction factor, the difference between this and the theoretical stress concentration factor being more marked in the case of ductile materials.

In addition to the variation of the strength reduction factor of a given notch for different materials it is found to depend also on the size of the specimen, at least in test pieces of relatively small size. There is a tendency for the fatigue strength of geometrically similar notched specimens to be reduced as the size is increased but there are indications that above a certain size the strength reduction due to a given notch remains constant for components made from similar material. The problem is further complicated by the fact that the properties of a given steel may differ for forgings of different sizes even though the ingots may be from the same cast; as yet, however, little experimental data are available regarding the effect of size on the fatigue strength of parts greater than about two inches in diameter.

In addition to the points mentioned above on the aspects of fatigue that have a direct bearing on the strength of dynamically loaded bolts, various other factors including the forging, heat treatment and finishing processes used may influence the problem. Only a very brief outline has been attempted and for detailed information reference should be made to a most comprehensive review* of the many papers published on the fatigue of metals prepared by the staff of the Batelle Memorial Institute in the United States.

From the foregoing it will be apparent that, although there is available a large mass of data on the fatigue of metals, very little of it can be applied directly by the designer of heavy marine engines because, in the first place, very few tests have been carried out on the type of material generally used in such engines but mainly because so little information is available on the strength of large specimens. It is evident that before designs can be more widely based on the resistance of materials to fatigue stresses rather than static stresses much more experimental work on full scale components, such as that described in this paper, will be necessary.

I. STRESSES IN ENGINE BOLTS

Design Stress

In practice the estimation of the actual working stresses in such parts as engine running-gear bolts is very difficult, if not impossible, because of various indeterminate factors. Theoretically, fluctuating stresses in the connecting rod bolts of certain oil engines such as the four-stroke cycle single-acting type should be quite small; that in fact this is not so is demonstrated by failures that have occurred in such engines. It may be concluded, therefore, that in service the running gear of direct-coupled engines must be subjected to inertia stresses of considerable magnitude which probably arise from acceleration of the propeller and shafting when the vessel is pitching in a seaway, particularly when in the light condition.

In double-acting and opposed-piston engines the load transmitted, excluding such extraneous factors, can be calculated with reasonable accuracy from the cylinder pressure and inertia of the reciprocating parts. In such cases it is assumed that the whole of the load is carried by the bolts and they are made of such a size that the corresponding stress, based on the cross-sectional area at the root of the threads, does not exceed a nominal value, usually of the order of 5,000 to 8,000lb. per sq. in., depending on the properties of the material used. Since in marine propelling machinery the first essential is reliability such low nominal stresses are fully justified; moreover long experience has shown that by the use of such a purely arbitrary "design" stress satisfactory results are usually obtained in service.

From further consideration of the problem, however, it must be concluded that the design stress is a purely hypothetical figure and bears little relationship to the actual stresses set-up in the bolts securing an assembly subjected to cyclic loading. Both the tightening stress and the design of the components play an important part in determining the actual operating stresses, as will be shown in the following section.

Fluctuating Tensile Stress

If bolts are fitted which are initially unstressed, as soon as a tensile load is applied to the assembly the faces of the two halves will open owing to the elastic extension of the bolts; the load in the bolts will then be equal to the externally applied load. Large bolts of the type under consideration are normally hardened-up using a spanner and heavy hammer, however, stressing the bolts in tension and the bolted parts in compression. Assuming for the moment that the bolted parts are absolutely rigid, the bolts being the only elastic components of the assembly, it follows that the stress in the bolts could

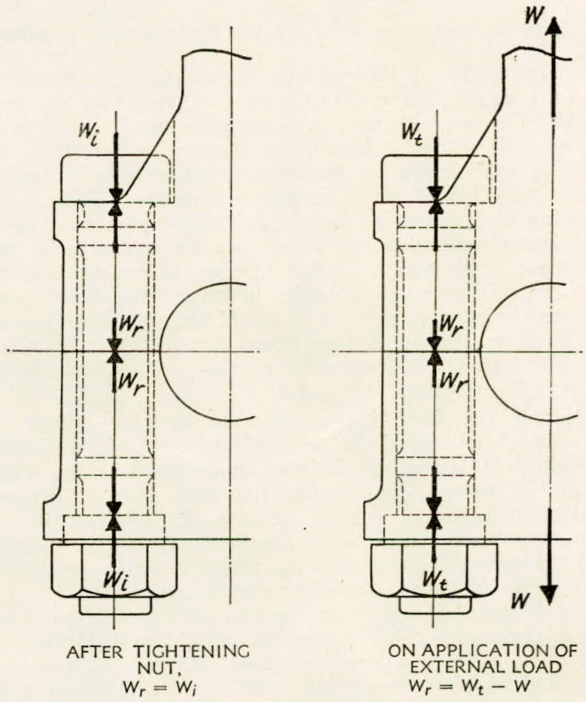


FIG. 3—*Forces acting on bearing assembly*

* 1941. "Prevention of the Failure of Metals Under Repeated Stress". John Wiley and Sons, New York.

only be increased by opening of the joint faces, between which there is a force W_r, equal to the initial tightening load in the bolt (see Fig. 3). Under such conditions the stress in the bolt would remain constant upon the application of an external load W unless this load was of sufficient magnitude to separate the bolted parts, i.e. greater than the pre-load W_i set up on tightening. In these circumstances (the pre-load being greater than the applied load) the fluctuating working load would cause no variation in stress in the bolts. Consequently, there would be no tendency to failure by fatigue, since the fatigue endurance of the metal depends on the stress fluctuation.

In practice, of course, the bolted parts cannot be made perfectly rigid and the bolts are subjected to the initial tightening stress plus a fluctuating component, the value of which depends upon the relative stiffness of the bolts and the bolted parts.

It can be shown that when the working load is less than that required to cause parting-line separation the total load in the bolt at any point in the loading cycle can be expressed approximately in the form

$$W_t = W_i + CW, \quad \text{.........................(1)}$$

where W_t = total load in bolt,
W_i = initial tightening load,
and W = instantaneous value of applied working load.

The value of the coefficient C is given by $\dfrac{k_a}{k_a + k_b}$, where k_a and k_b represent the elastic constants, or the deformation per unit load, of the bolted parts and the bolt respectively. If the ratio $\dfrac{k_b}{k_a}$ is represented by K the above expression for C may be rewritten in the form $\dfrac{1}{1+K}$ and the total load in the bolt is therefore given by

$$W_t = W_i + \frac{W}{1+K}. \quad \text{.........................(2)}$$

It will be noted that the second part of the expression represents the fluctuating component of load which is superimposed on the steady load W_i, this additional load being only a fraction of the externally applied load W.

Unfortunately the value of the fraction $\dfrac{1}{1+K}$ cannot be calculated readily in the majority of cases since the bolted parts are of irregular shape. The elastic constant for the bolt k_b can be estimated with reasonable accuracy but in order to apply this analysis to design it will be necessary to determine values of K experimentally for typical bearing assemblies and other dynamically loaded parts in marine engines. This matter is receiving some attention by the British Shipbuilding Research Association and when sufficient data are available it will be possible to estimate the actual stresses in engine bolts more precisely. It appears probable, however, that for forged or cast steel bearings of normal design K has a value of 4 or more. If this is so the fluctuating component of stress in the bolt is reduced to at least *one-fifth* of the applied load providing the bolt is adequately tightened.

From the foregoing it is apparent that effective pre-stressing of bolts is of the greatest importance in reducing the possibility of failure by fatigue, and to provide an adequate margin of safety, by allowing for some loss of tension by bedding together of mating surfaces in service, the initial tightening load should be at least twice the applied working load. Measurement of this tightening load in the bolt presents some difficulty, however, and it appears that the only reliable method is to tighten the bolt until the required extension is obtained. Such refinements would probably be hardly practicable in the majority of cases, but since there is little likelihood of overstressing large engine bolts, hardening-up as far as possible should provide an adequate safeguard.

In this connexion it is of interest to note that in the report* of a sub-committee set up by the Institute in 1945 it is concluded that slackening of the nuts of the connecting-rod bolts of auxiliary Diesel engines is one of the main causes of failure. In this report various details for preventing loss of pre-stress are considered.

From the point of view of design it is evident from expression (2) that the value of K should be made as large as possible to reduce the cyclic load in the bolt. This may be accomplished by designing the bolts for maximum elasticity, i.e. the length should be as great as possible and the cross-sectional area of the shank should be reduced to a minimum. The use of bolts having a reduced shank diameter also has other advantages which will be discussed later.

An increase in stiffness of the bolted parts has a similar effect in increasing the value of the ratio of the elastic constants K. The bolted members of a dynamically loaded assembly should therefore be made as rigid as possible.

Bending Stresses

In the derivation of the expression given in the preceding section, one of the assumptions made is that the stress applied to the bolt is purely tensile. It is doubtful, however, whether this condition ever occurs in practice even in components where the load is supposed to be axial and it may be taken that all bolts are subjected to superimposed bending loads of an unknown magnitude. Static bending stresses may be caused by slight misalignment of the faces of abutting joints or unsymmetrical elastic deformation of the bolted parts. In addition to these factors, the line of application of the external load does not usually pass through the bolt axis and consequently in an assembly such as a bottom end bearing the fluctuating stress at one side of the bolt will be considerably greater than the mean value.

It follows that bending stresses can be minimized by suitable design of the bolted members and accuracy of production. The effect of bending on the fatigue strength of a bolt is also mitigated by the use of long bolts having a reduced shank diameter but it is evident from the foregoing discussion that when considering the strength of bolts subjected to cyclic loading the design of the bolted parts is of great importance. This is a case where the strength of one part may be influenced to a large extent by other parts of the assembly.

Stress Concentrations

It has been found from tests and actual failures in service that three positions in a bolt carrying fluctuating stresses are particularly vulnerable to fatigue failure. These occur at the points of stress concentration indicated in Fig. 4, namely (1) the junction of the shank with the underside of the head; (2) the junction of the screwed section with the shank; and (3) at the first thread engaging with the nut.

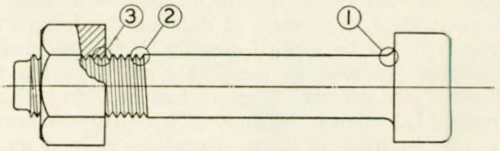


FIG. 4—*Points of stress concentration*

At these points the mean stress across the section is magnified by an amount depending upon the abruptness of the change in cross sectional area. The smaller the radius at the root of the thread or the smaller the fillet the greater is the concentration of stress and hence the greater the reduction in strength under cyclic loading. It is for this reason that an adequate fillet radius should be provided under the head; this question is discussed further in the consideration of the results of the fatigue tests given in section II.

* 1945-6. "The Failure of Auxiliary Diesel Engine Connecting Rod Bolts, A Survey of Cause and Prevention". Trans.I.Mar.E., Vol. 57, p. 85.

The distribution of stress across the screwed section probably approximates to the form shown diagrammatically in Fig. 1, but the stress concentration produced by a number of adjacent grooves, as in a screw thread, is less than that at the root of a single notch of similar shape and depth. At the end of the thread in a full diameter bolt there is no stress relieving groove and consequently the concentration of stress is greater at this point than at any other position along the screwed section outside the nut. Reduction of the diameter of the bolt below the screwed section to less than that at the root of the threads, with the provision of an adequate fillet radius, reduces the stress concentration at this point.

The main point of weakness in dynamically loaded bolts, however, occurs at the first one or two threads in the nut (position (3) in Fig. 4). This is due to the fact that in bolts and nuts having uniform pitch, extension of the bolt and compression of the nut occurs when loaded, so transmitting a large proportion of the load through the first active turn of the thread as illustrated diagrammatically in Fig. 5. Thus, in addition to the concentration of stress at the root of the thread due to the notch effect, the position is made worse at this point by the concentration of load transmitted through the nut.

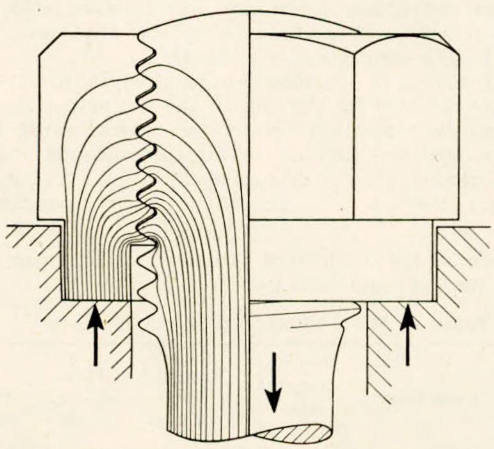


FIG. 5—*Diagram to illustrate distribution of load along a nut*

Various methods of improving the distribution of load along the nut are possible. Of these the use of threads on the bolt and in the nut of slightly different pitch or the use of tapered threads in the nut are hardly practicable in marine engine practice owing to production difficulties. A simple method of improving the load distribution is to use a more ductile material for the nut. Thus, the old practice of using a wrought iron nut in conjunction with a steel bolt had something to commend it although it was not from the point of view of relief of stress concentration that this procedure was adopted.

In theory uniform distribution of load along the nut can be obtained by tapering the outside of the nut and taper boring the end of the bolt to the depth of thread engagement. A type of nut known as the "tapered-lip nut" which approximates to the theoretical shape has been used in various forms to some extent. Applications of this principle are illustrated in Fig. 6. Design (a) has been used in aero-engine practice, that illustrated in Fig. 6(b) was used in German marine engines for securing balance weights to the crank webs and at (c) is shown the design of nut finally used to overcome troubles which developed in the connexion of the piston rods to the crossheads in the double-acting oil engines used in certain German warships.

The effect of modifications in nut design has been investi-

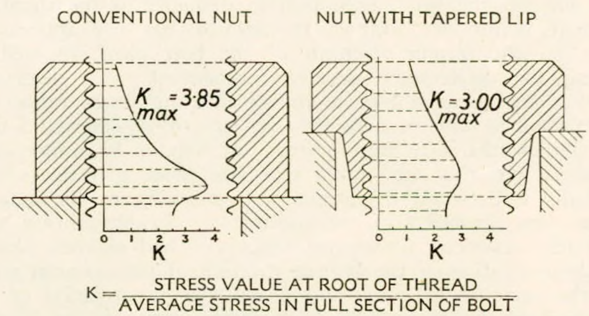
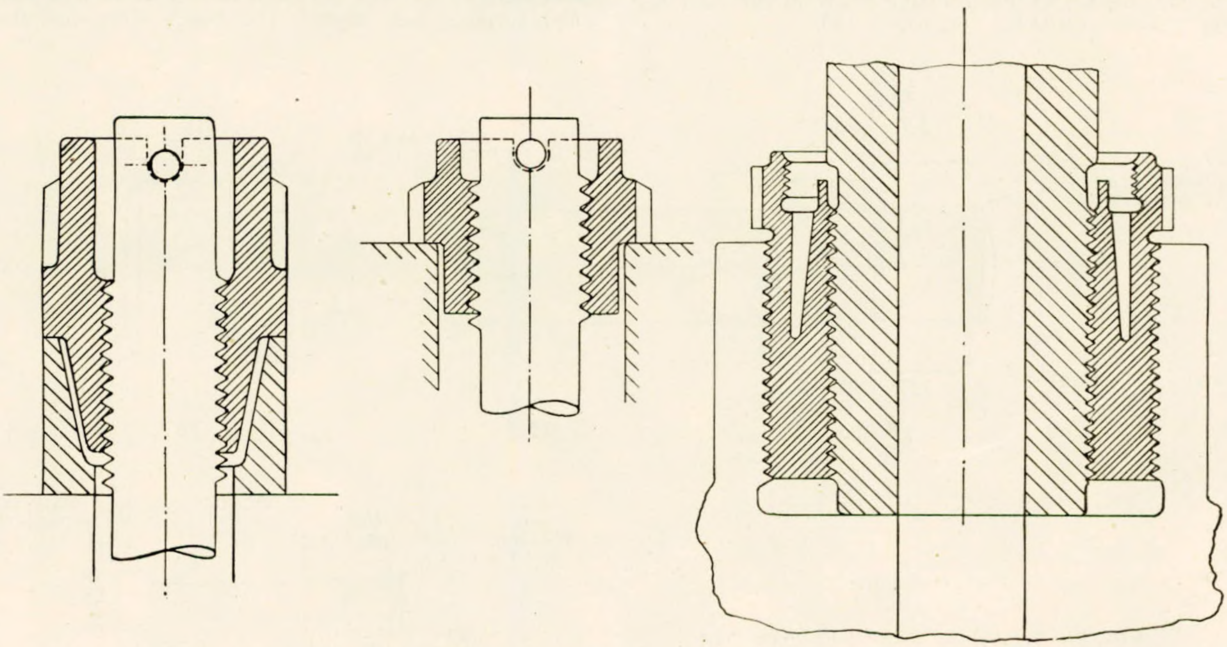


FIG. 7—*Comparison of stress values at the roots of bolt threads with nuts of conventional and "tapered lip" types*



(a) (b) (c)

FIG. 6—*Modified nut forms for improving load distribution along the threads*

gated by Hetényi* by means of photo-elastic models and the diagrams shown in Fig. 7 give an indication of the stress relief that may be expected by using a tapered-lip nut.

The distribution of load along nuts is, however, influenced to such a large extent in practice by the accuracy of thread profile and pitch that extremely small pitch errors would entirely mask the effect of design modifications. In mild steel bolts and nuts some local yielding generally occurs which tends to bring about an improved load distribution. This effect is less pronounced in steels of a higher yield point; consequently greater accuracy of pitch and thread form is desirable in bolts of higher tensile steels and modified nut designs would probably be more effective in such cases.

On looking through the remarks in this section it might be inferred that the number of variables are such that the difficulties involved in an estimation of the operating stresses in dynamically loaded bolts are insurmountable. It is hoped, however, that as more actual stress measurements obtained under running conditions become available it will be possible to fix average values for those quantities which are at present unknown.

II. FATIGUE TESTS

As pointed out in the introduction the primary object of these tests was to determine the fatigue strength of bolts of a type similar to those normally used in slow-running marine Diesel engines. Additional tests were also carried out to ascertain whether the small reduction in diameter of the shank of the bolt, commonly adopted in practice, has any appreciable effect on the fatigue strength of the bolt itself, as well as having the advantages already enumerated. To determine whether any great difference in the limiting stress range can be expected in bolts of different sizes, specimens screwed 3-inch and 1-inch diameter were tested. It should be understood, however, that this was not a true size effect investigation as normally understood in fatigue testing owing to the lack of geometrical similarity of the specimens. Supplementary tests on ½-inch diameter unnotched specimens had as their objects the determination of the fatigue strength of the material without the presence of stress-raisers and the investigation of the influence of the magnitude of the mean stress of the loading cycle on the limiting stress range.

* Hetényi, M. 1943. "A Photo-Elastic Study of Bolt and Nut Fastenings". Trans.A.S.M.E., Vol., 65, p. A93.

Material and Manufacture of Specimens

The steel used for the test bolts and fatigue specimens was similar to that normally specified for the running-gear bolts of heavy oil engines, namely, an acid open-hearth steel having an ultimate tensile strength of 28-32 tons per sq. in. and containing not more than 0·05 per cent sulphur and phosphorus. To ensure maximum uniformity of material all the steel used was obtained from the same cast and had the following analysis: —

Carbon, per cent	Manganese, per cent	Silicon, per cent
0·16	1·0	0·18

It will be noted that the manganese content is rather higher than in normal mild steel. This is necessary to give a high impact value which is normally required in material to be used for bolts. In this connexion it is of interest to note that there does not appear to be any direct relationship between the impact value, as determined by the Izod test, and the notched fatigue strength of a material.

Steel having a specified tensile strength of 21-25 tons per sq. in. was used for the nuts of the test bolts. As material of this tensile strength is below the normal range for acid steel, basic steel was supplied having the following analysis: —

Carbon, per cent	Manganese, per cent	Silicon, per cent
0·115	0·51	0·18

Details of the mechanical properties of the materials used for both the nuts and bolts are given in Table I.

TABLE I—MECHANICAL PROPERTIES OF MATERIAL

Bloom size, inch	Used for	Tensile strength, tons per sq. in.	Elongation, per cent	Izod values, ft. lb.	Brinell Hardness No.*
$6\frac{1}{2}\times6\frac{1}{2}$	3-inch bolts	31·6 to 32·4	35	66 to 84	137 to 143
$6\frac{1}{2}\times6\frac{1}{2}$	3-inch nuts	25·6	40	—	—
$5\frac{1}{2}\times5\frac{1}{2}$	1-inch bolts	30·8	38	97 to 108	143 to 163
5×5	1-inch nuts	27·2	40	—	—

* Hardness values were obtained from each forging.

All the steel used for the test pieces was supplied by the steelmakers in the form of rolled blooms of various sizes from which forgings were made. The 3-inch bolts were forged in

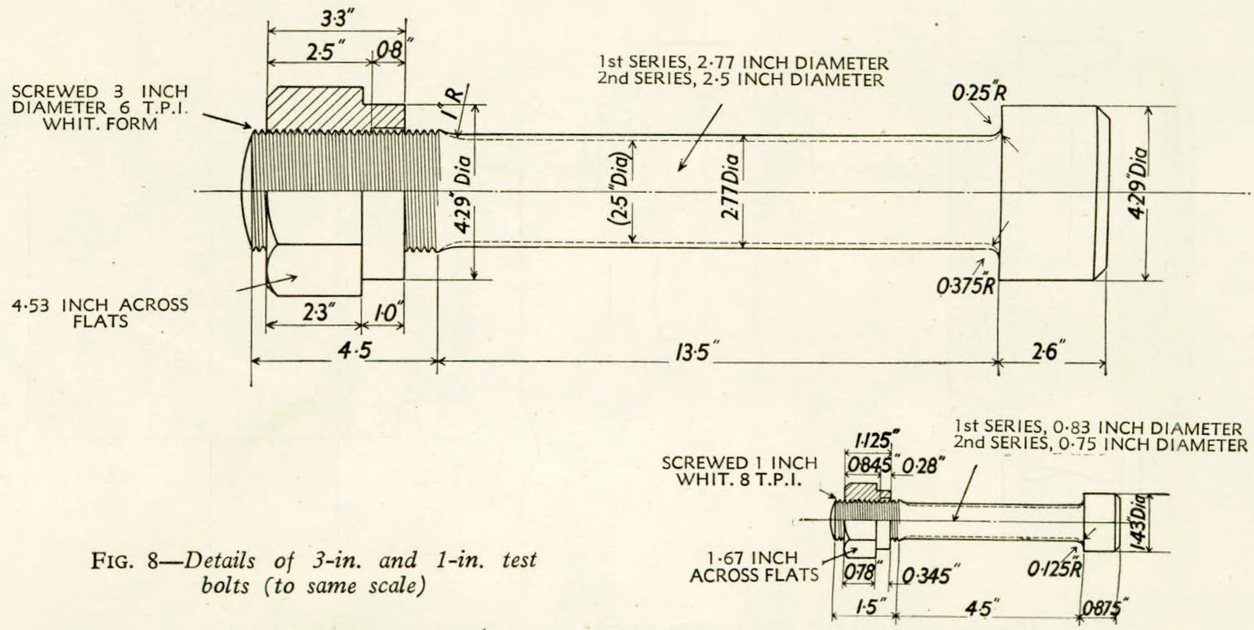


FIG. 8—*Details of 3-in. and 1-in. test bolts (to same scale)*

pairs, head to head, as is the normal practice and this procedure was also adopted for the 1-inch bolts. The $\frac{1}{2}$-inch unnotched fatigue specimens were machined from lengths of $1\frac{1}{4}$-inch diameter bar forged down from $5\frac{1}{2}$-inch blooms. All forgings were given a normalizing and tempering heat treatment which was varied in accordance with the size.

When testing screwed components in fatigue it is usual to specify ground threads of the greatest accuracy to reduce as far as possible scatter of the test points but, as the object of this investigation was to obtain results which can be applied directly to marine engine design problems, it was decided not to depart from normal production methods. The bolts were given a smooth turned finish and care was taken to make the fillet under the head as smooth as possible; the threads of both the bolts and nuts were lathe-cut, using a single point tool, and finished with a ground thread chaser.

Details of the test bolts are given in Fig. 8, from which it will be noted that the nuts were similar to those commonly used when side-locking set screws are fitted. The pitch of the threads adopted was in accordance with usual practice.

Test Procedure

Two series of tests were carried out on each of the two sizes of bolt specimens. In the first series the body of the bolt was made just less than the thread-root diameter while in the second series the shank was turned down to 90 per cent of the root diameter, i.e. 80 per cent of the thread core area.

In all the fatigue tests the minimum stress* of the loading cycle was maintained constant at 3 tons per sq. in. The choice of this value presented some difficulty since it was known that the limiting stress range would decrease with increasing values of mean stress. On the other hand, in practice, a bolt which has a high initial pre-stress is not subjected to great cyclical stress fluctuations, as pointed out in an earlier section. To simulate as far as possible the most adverse conditions likely to occur in service, the minimum value of the fatigue loading cycle was therefore chosen to represent the lowest probable prestress obtaining under normal operating conditions.

* The stresses quoted in this paper refer to the mean stress calculated on the area at the root of the threads.

The machine used for the testing of the 3-inch bolts was a Losenhausen hydraulic pulsator. This machine was developed in Germany primarily for the fatigue testing of actual machine components and has been used to a large extent in the testing of welded structures. The cyclic load range of the machine is 100 tons, which may be applied within the limits of 105 tons in tension or compression. A diagrammatic arrangement of the machine is shown in Fig. 9 from which it will be seen that in construction it is similar to that of a normal vertical hydraulic tensile-testing machine, the upper clamping head A being the loading member. The lower end of the test piece is held in an adjustable crosshead B which is attached to the framework of the machine. A fluctuating tensile load may be applied to the test piece through the top piston C by means of the variable stroke pulsator pump E, while a constant compressive load may be applied by the lower piston D. Thus, when the desired loading lies wholly within the tensile range only the top cylinder is used but if part or all of the loading cycle is required within the compressive range the mean stress of the loading cycle applied to the specimen is depressed by the superimposition of a compressive load by the bottom piston. The frequency of loading is governed by the speed of the pulsator pump; in these tests the speed of operation was 266 cycles per minute.

The fluctuating load limits are indicated by two pressure gauges G and H which are connected in turn to the cylinder of the testing machine through a rotary valve operated by the shaft of the pulsator pump when the pressure is at a maximum or minimum respectively. When set to the required value the maximum value of the loading cycle is automatically controlled by pump J while the function of pump L is to maintain a constant pressure in the accumulator cylinder K which is connected to the lower cylinder of the machine.

In assembling a test bolt in the machine the greatest care was taken to ensure axial loading as far as possible. Spherical seating plates were used to hold the specimens in the grips of

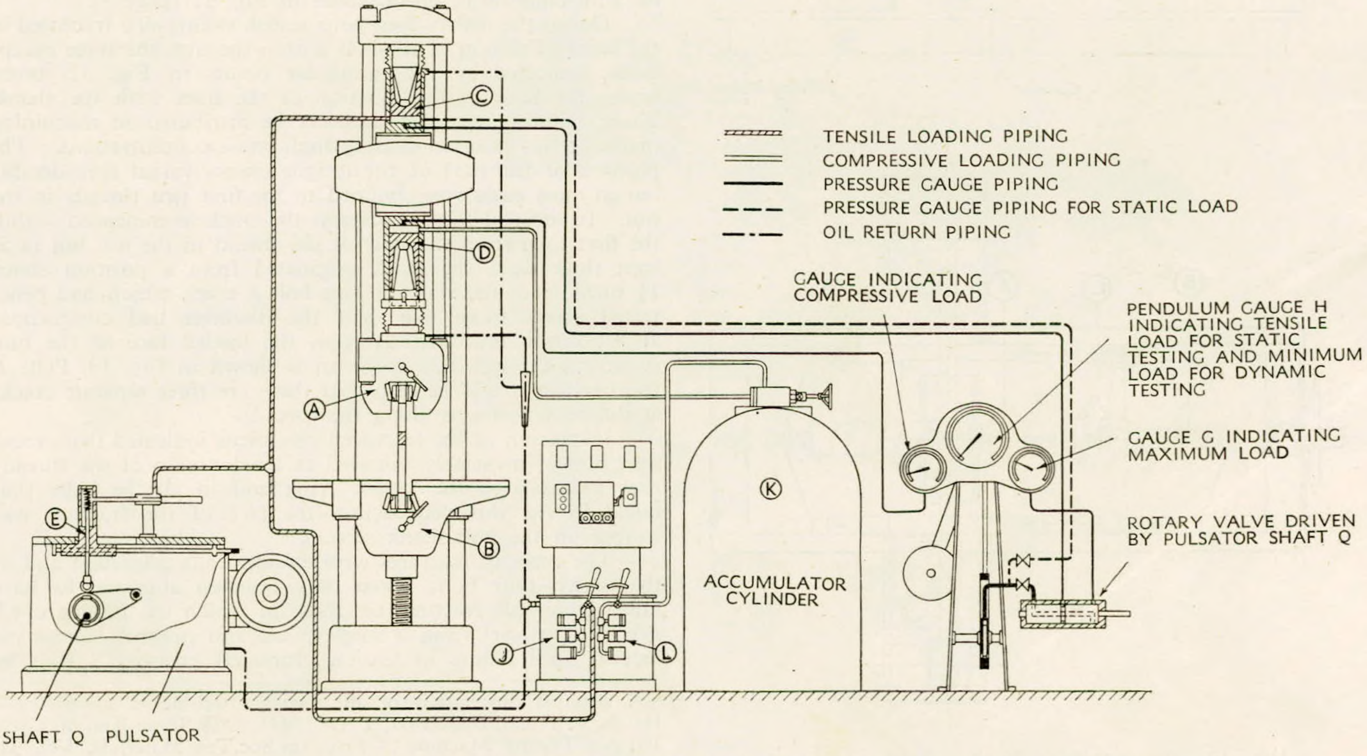


FIG. 9—*Diagrammatic arrangement of the Losenhausen fatigue testing machine*

the machine and, by careful alignment, bending stresses in the shanks of the bolts, as shown by strain gauge tests, were reduced to less than 5 per cent of the maximum direct stress.

After assembly of the specimen in the machine and setting of the loading cycle the machine was operated continuously until fracture of the bolt occurred. The method of testing adopted was the normal "loading-down" procedure in which a number of similar specimens are tested, the first one being subjected to a relatively high fluctuating stress range and subsequent test pieces being tested at successively lower stresses. A total of twenty-four bolts were tested, fourteen in the first and ten in the second series.

It will be appreciated that to run a test to 10 million cycles would mean continuous operation of the machine for nearly a month; it was therefore decided not to attempt to obtain such an endurance after results had been obtained for a life of 3 million cycles, in view of the variation in results indicated by the tests completed.

The machine used for the testing of the 1-inch bolts was an Avery-Schenck horizontal fatigue testing machine having a load range of 20 tons, within the limits of 20 tons in tension or compression. This machine is of the dual mass resonance type, one mass being formed by the oscillating system and the other by the machine base. Fig. 10 shows a general arrangement of the machine. One end of the specimen A is attached to the frame of the machine by means of an adjusting screw B, an elastic loop dynamometer C being interposed to measure the loads. The other end of the test piece is attached to the oscillator beam D which is excited by an out-of-balance rotating mass driven by a variable speed D.C. motor. Variation of the motor speed alters the amplitude of vibration and consequently the magnitude of the cyclic stress applied to the test piece, the frequency of loading normally being between 2,600 and 2,900 per minute. An initial static load can be imposed by springs E which are anchored to the main frame and adjustable by means of a straining screw F.

Fifteen 1-inch bolts were tested in all, five of these having a reduced shank diameter. Owing to the relatively high speed of the machine used it was convenient to carry out extended endurance tests without unduly prolonging the running time Two specimens were subjected to 50 million loading cycles without failure.

In addition to the fatigue tests on bolt specimens, unnotched fatigue specimens were tested, as already mentioned, to determine the fatigue limit of the material without any stress raising discontinuities and the effect of mean stress of the loading cycle. Some sixty specimens of the type shown in Fig. 11 were tested in order to obtain a series of S-N curves for loading cycles having various mean stress values ranging

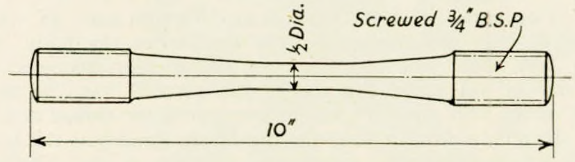


FIG. 11—*Details of small scale unnotched specimens*

from 2·5 tons per sq. in. in compression to 17·5 tons per sq. in. in tension. The machine used was a Haigh fatigue-testing machine designed to apply a range of load of up to 6 tons at a frequency of 3,000 cycles per minute. As this machine is fairly widely known details of construction and operation are not considered necessary*.

Results

The results of the tests on 3-inch bolts are shown plotted on semi-logarithmic co-ordinates in Fig. 12 (page 9).

Out of the twenty-four bolts tested, twenty-one fractured in the screwed section of the bolt within the nut, the three exceptions, indicated by the triangular points in Fig. 12, broke under the head at the junction of the fillet with the shank. These failures can most probably be attributed to machining marks at this position causing high stress concentrations. The position of the start of the fatigue cracks varied considerably but in most cases was confined to the first two threads in the nut. In most of these specimens the cracks commenced within the first turn from the start of the thread in the nut but in at least three cases the crack originated from a position about 1½ turns from the start; in one bolt a crack which had penetrated across more than half the diameter had commenced at a position four threads from the loaded face of the nut. A section through this specimen is shown in Fig. 13, Plate 1, from which it will be seen that there are three separate cracks at different positions along the thread.

Inspection of the fractured specimens indicated that cracks were almost invariably initiated at the junction of the thread-root radius with the loaded flank and in all the bolts that failed in the threaded section the face of the fracture was convex on the bolt-shank section.

The types of fractures were by no means consistent and of the twenty-four bolts tested only fourteen appeared to have failed by simple fracture, i.e. those in which the fatigue crack appeared to start from a single point and progress across the section until failure in tension ultimately occurred. It is by

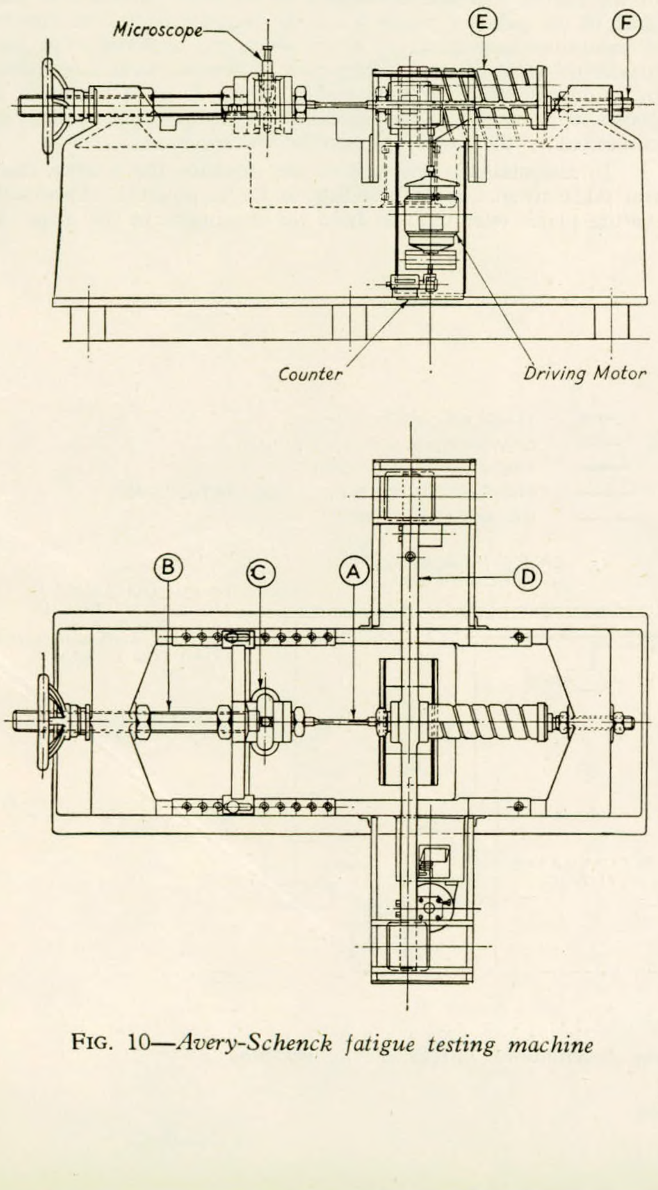
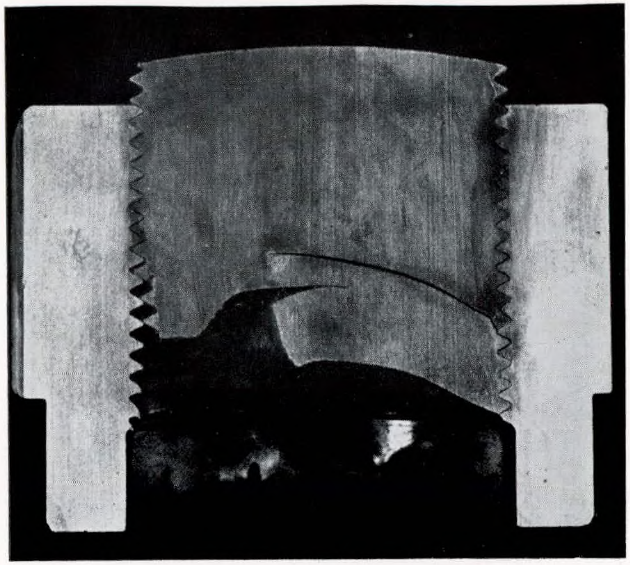
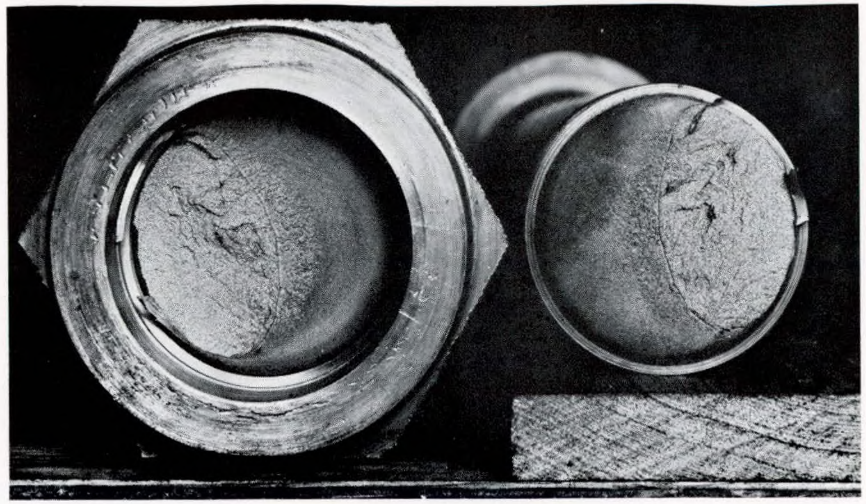


FIG. 10—*Avery-Schenck fatigue testing machine*

* A detailed description of the Haigh machine is given in:—Haigh, B. P. and Robertson, T. S. 1931. "A Seven-Ton 50 Cycle Fatigue Testing Machine". Proc.Am.Soc.Test.Materials, Vol. 31, p. 221.





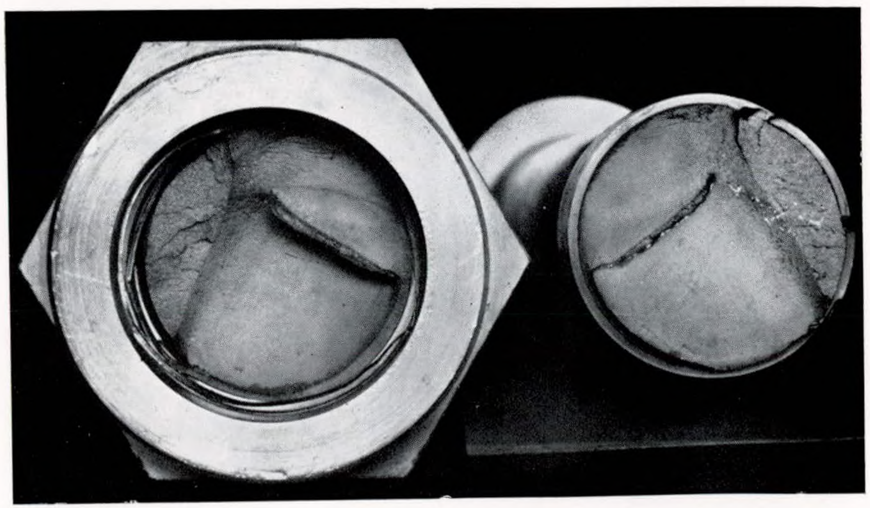
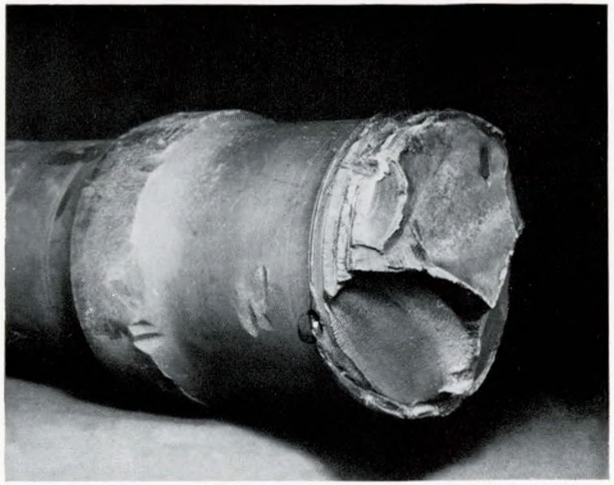


FIG. 13 *(top left)—Section through 3-in. specimen showing compound fracture (complete failure after 1,890,000 cycles, 3·0 to 10·3 tons per sq. in.)*

FIG. 14 *(top right)—Example of simple fatigue fracture of 3-in. bolt (failure after 196,000 cycles, 3·0 to 12·8 tons per sq. in.)*

FIG. 15 *(bottom left)—Example of compound fatigue fracture of 3-in. bolt (complete failure after 1,225,000 cycles, 3·0 to 11·0 tons per sq. in.)*



FIGS. 16(a) *(above)* and (b) *(right)—Compound fatigue fracture of 4-in. diameter bottom-end bolt*

FIG. 19—*Example of fatigue fracture at fillet under head of bolt (failure after 79,000 cycles, 3·0 to 15·0 tons per sq. in.)*

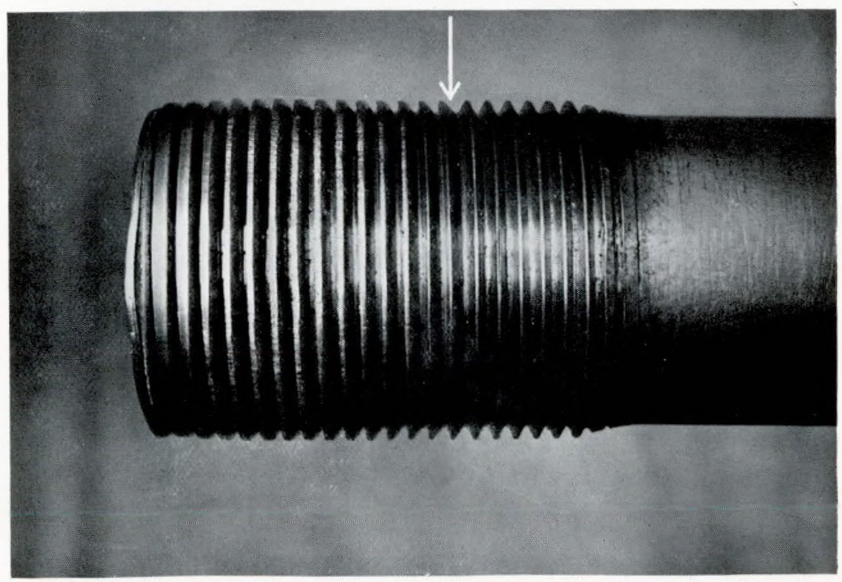
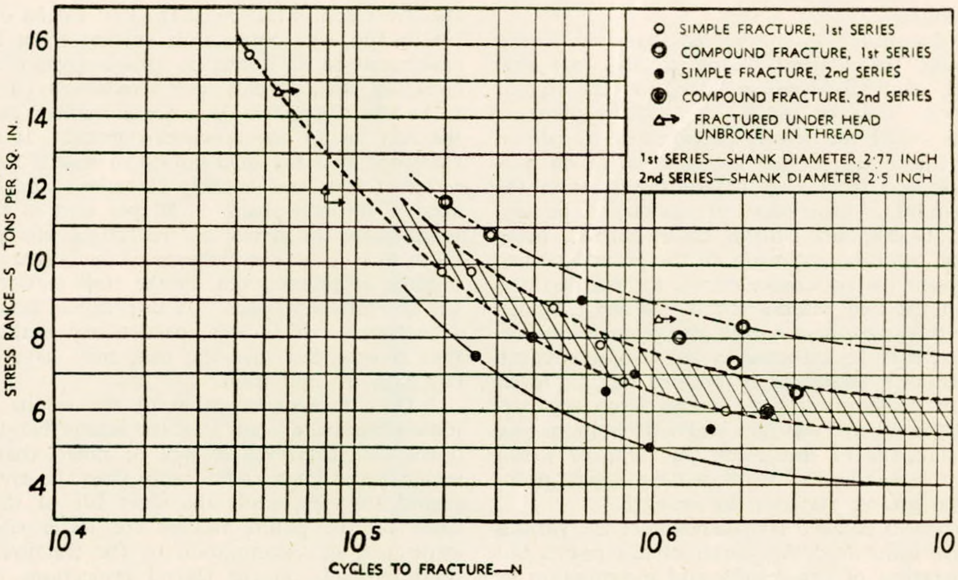


FIG. 20—*Threads of 3-in. tensile test specimen after failure. (Arrow indicates position of bottom of nut before testing)*



(MINIMUM STRESS OF LOADING CYCLE, 3 TONS PER SQ. IN. THROUGHOUT)

FIG. 12—*S-N diagram for 3-in. bolts*

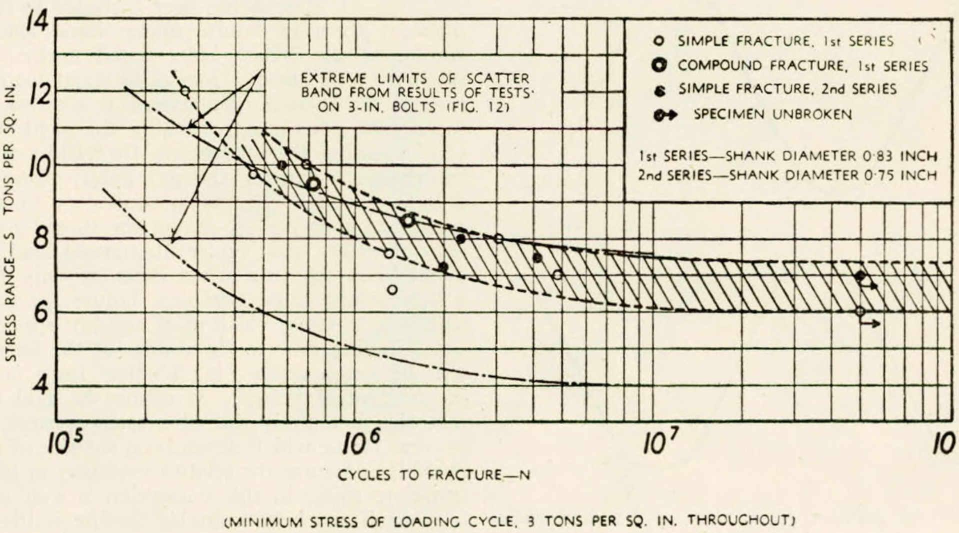
no means certain that all these were simple fractures, however, since the broken ends of the bolts were not removed from the nuts except in two cases where the nuts were cut off for examination of the threads. A typical example of a simple fatigue fracture is shown in Fig. 14, Plate 1.

In seven of the bolts compound fractures, i.e. those where fatigue cracks had commenced from more than one point, were evident on visual examination. Some of these specimens revealed the fact that where multiple cracks occurred the crack nearest the top or unloaded face of the nut was able to penetrate almost across the section without complete failure, thus, in some cases transferring the whole of the load to little more than a single turn of the thread. Even then thread failure did not occur and final fracture was delayed until another crack had commenced nearer the loaded face of the nut and had penetrated to sufficient depth to allow the intervening material to fail in shear.

The most outstanding example of this phenomenon is illustrated by the nut shown in section in Fig. 13, Plate 1. In this particular specimen complete failure did not take place until a third crack had spread across more than half the section. Another example of a compound fatigue fracture is illustrated in Fig. 15, Plate 1, which shows the fracture faces. It is of interest to note that failures of this type have been known to occur in service. An example is illustrated in Fig. 16, Plate 2, which shows a fractured bottom end bolt of 4-inch diameter which failed after being in service for fifteen months.

From the results of the 1-inch bolt tests shown in Fig. 17 it will be noted that the majority of the bolts of this size failed by simple fracture, only two specimens exhibiting the type of multiple cracking which occurred in several of the large bolts. As in the 3-inch bolts the crack generally commenced within the first turn of the thread engaged in the nut.

The results of the tests on the ½-inch unnotched fatigue specimens are given in Fig. 2, from which it will be seen that the fatigue limit of the material under reversed axial stress is ± 15·4 tons per sq. in. Even when the mean stress of the loading cycle is increased to 17·5 tons per sq. in. the limiting stress range is only reduced to ± 11·2 tons per sq. in.; under these conditions the fluctuating stress has a maximum value of 28·7 tons per sq. in.



(MINIMUM STRESS OF LOADING CYCLE, 3 TONS PER SQ. IN. THROUGHOUT)

FIG. 17—*S-N diagram for 1-in. bolts*

Consideration of Results

In these series of tests it was anticipated that some variation in results would be obtained owing to the fact that ordinary production methods were employed in the manufacture of the test bolts; only by the use of carefully prepared laboratory specimens could the results be expected to fall on or close to a single line when plotted in the form of an S-N diagram. As will be seen from Fig. 12 the results from the 3-inch bolt tests exhibit a great deal of scatter. The two extreme curves in this diagram (shown chain-dotted), being the boundary lines of the zone enclosing all the plotted points, may be said to represent the endurance curves for the best and poorest specimens in the two batches tested. When extended to an endurance of 10^7 cycles these curves give some indication of the differences that may be expected in the fatigue strength of large bolts in practice. It should be pointed out, however, that this variation in the limiting stress range, from approximately 4 to 7·6 tons per sq. in., was obtained with bolts having a reasonably high standard of finish for this type of work; any deterioration in the standard of workmanship would probably result in an even greater variation in strength.

Space does not permit detailed consideration of the various factors that may have influenced the scatter of test points but from careful consideration of the results and examination of the broken test pieces it was concluded that the wide variation in strength could be attributed largely to small errors in pitch and form of the nut and bolt threads. Two bolts, picked at random, were measured to obtain some indication of the accuracy of thread cutting and, although the cumulative pitch error did not exceed one thousandth of an inch over a length of four inches, periodic errors in pitch amounting to almost 0·0015-inch between adjacent threads were found in some cases. Considerable variation in thread-flank angles was also revealed as well as irregular profiles. Typical profiles enlarged to ten times actual size are reproduced in Fig. 18.

On the whole the internal threads in the nuts were found to be more accurate than the external threads but slight errors in pitch of the order of 0·001-inch per inch were revealed. Even such a small error is sufficient to alter the load distribution along the threads in engagement and cause differences in stress concentration.

It will be noted that for the most part the plotted points from the second series of tests in Fig. 12 lie towards the lower endurance curve but the opposite tendency is apparent in the results from the 1-inch tests shown in Fig. 17. As mentioned earlier, the only way in which a reduction in shank diameter can affect the fatigue strength of the bolt is by relieving the stress concentration at the first thread on the bolt (position (2) in Fig. 4). Since it is evident from the results that the concentration of stress is much greater at the first thread engaging with the nut such a reduction in diameter is unlikely to be effective when there are a number of free threads below the nut, as in the specimens tested. It may be concluded, therefore, that the distribution of results from each of the two series of tests was entirely fortuitous and that a reduction in area of the bolt shank to 80 per cent of the thread root area is ineffective in increasing the fatigue strength. This does not mean to say that the diameter of the bolt should not be reduced, indeed, as pointed out in the first section such a procedure has several advantages. It is possible, however, that a greater reduction in shank area, particularly if the turned down portion is extended into the nut, may have some strengthening effect on the bolt itself.

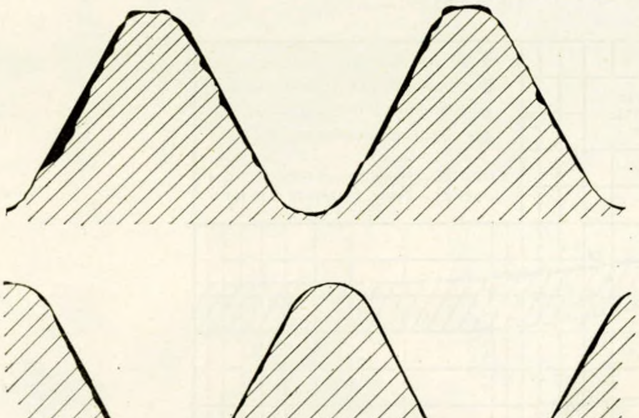


FIG. 18—*Typical thread profiles from 3-in. bolts and nuts (external thread above, internal thread below; magnification ×10). Variations from true profile shown in black*

On closer examination of the results of the 3-inch bolt tests it became evident that the scatter band could be narrowed down considerably. It will be noted that all the specimens which had a greater endurance than the average failed by compound fracture, while the short life of those test bolts indicated by the points nearest the lower endurance curve was attributed, on examination of the fracture faces, to excessive bending loads on the thread projections. Both these effects arise from inaccuracies in thread cutting and therefore, by disregarding the points mentioned, a narrower band (enclosed by the dotted curves in Fig. 12) was obtained which may be said to represent an endurance band for bolts having reasonably well formed threads. It will be seen that this band encloses most of the results where simple fracture occurred and the difference in limiting stress range between the upper and lower curves is reduced to 1 ton per sq. in. (from approximately 5·4 to 6·4 tons per sq. in.).

The fact that three of the 3-inch bolts failed at the fillet under the head indicated that the stress concentration at this point is almost as great as that at the first thread in the nut. Two of these specimens were from the second series of bolts where the fillet radius to shank diameter ratio was 6/40; this suggests that even a fillet radius of $\frac{1}{8}$ of the shank diameter is rather small when allowance is made for the irregularities in surface finish that invariably occur, especially when the bolt diameter is reduced, so increasing the stress at this section. The fractured faces of one of these bolts are shown in Fig. 19, Plate 2, in which the two zones of the fracture can be seen particularly clearly; that part to the left, when looking on the bolt head, is the relatively smooth surface of the fatigue crack, which has evidently spread from left to right, while the remaining section, which failed suddenly, has a rough texture typical of a tensile break.

Fig. 20, Plate 2, has been included to illustrate the entirely different form of failure under static load. This specimen, similar to the 3-inch bolts tested under cyclic loading, was subjected to a steadily increasing axial load. Plastic deformation of the threads commenced at a stress of 15 tons per sq. in., a value closely approaching the yield strength of the bolt. On increasing the load above the yield point, rapid failure of the threads occurred, the nut finally being stripped from the bolt at a load of 152 tons, equivalent to a stress of approximately 25 tons per sq. in. on the thread root area. It will be seen, therefore, that under fluctuating load conditions the bolt could break under a stress range of only 16 per cent of that which would cause complete failure, or 25 per cent of that sufficient to cause yielding, if steadily applied.

Turning now to the results of the tests on 1-inch bolts it will be noted from Fig. 17 that there is much less scatter of the plotted points. It cannot be said with any certainty that this is entirely due to greater accuracy of thread cutting as some factor which depends on the size of the specimen might possibly influence the relative variation in life. One point that comes to mind in this connexion is that once a crack forms in a small specimen complete failure is likely to follow fairly rapidly and thus varying rates of crack propagation have little influence on the results.

It will be noted from the endurance band shown shaded in Fig. 17 that, on the whole, the 1-inch bolts had a slightly greater limiting stress range than the 3-inch specimens. This tendency becomes more marked at higher stress ranges but for the most part the results obtained from the 1-inch bolts fall within the scatter band of those from the 3-inch specimens. The divergence of the curves with increase in stress range is of importance because in service it is unlikely that fatigue failures arise from a great number of load applications at a stress slightly greater than the fatigue limit but rather from relatively few applications of very high stress which may occur when for some reason the component is subjected to heavy overloading.

No definite conclusions can be drawn from a comparison of the results obtained from the two sizes of test bolts owing to the lack of geometrical similarity. Also, it must not be overlooked that a difference in fatigue strength could possibly be attributed to the variation in properties of the material in forgings of different sizes. It would appear from the limited data yet available, however, that any difference in the safe stress to which bolts over a considerable range of size could be subjected would be likely to be overshadowed by the variation in strength of large bolts, unless more accurate methods of thread cutting are employed.

ACKNOWLEDGEMENTS

The author's thanks are due to the Council and Director of Research of the British Shipbuilding Research Association for permission to publish this paper.

Discussion

MR. T. W. BUNYAN, B.Sc. (Member), who opened the discussion, said it was an interesting reflexion that several hundred years of the art or practice of engineering had invariably utilized the bolted connexion which, however, still constituted a problem. In fact, on reading Mr. Taylor's paper one found one was left with large gaps in one's knowledge of this apparently simple device. Even the most up-to-date statistics on failures in running gear, shafting, rudders and deck machinery of ships would feature the failure of bolted connexions in a major rôle. If the problem was still a real one with ships' machinery, where stresses were kept down to nominal dimensions and where a ½-inch bolt was invariably used when a ⅛-inch bolt would do the job—a concession to that all-embracing factor known as the personal element—the problem of the bolted connexion which faced the designers of aircraft engines must have been much more acute. This, indeed, could be judged by the amount of experimental data available on the fatigue strength of comparatively small high tensile bolts sponsored by the industry internationally, to which Mr. Taylor made reference in his paper.

He did not wish to comment on the results of the fatigue tests included in the paper. Reference would be made to these matters later by his colleague, Dr. Attia, who had been associated with some of the work done for the British Shipbuilding Research Association on the stresses in large bolts measured under running conditions. There were, however, certain points in the paper to which he would like to refer.

The research, as far as it had gone, had been valuable and worth while, but he was sorry that the author had been unable to include results on bolts with much greater reduction in shank diameter. In no case had failure of a specimen occurred in the shank. He had no doubt that this most important matter would be included in the programme.

He underlined the author's comments on page 235 of the paper regarding the use of high tensile steel, which had been responsible for some serious failures due to the fact that its greater notch sensitivity had not been fully appreciated. There were naturally many advantages which could be taken of the higher yield point of high tensile material, and, indeed, it would be impracticable to design an aircraft engine without resort to high tensile steels. But it must and did involve a greater appreciation of stress concentration effects, and the design of stressed parts must be arranged accordingly. Conjointly, it would be most instructive if the future programme could also include some tests with loose-fitting nuts. The fit of the nuts of bottom-end bolts in an aircraft engine was surprisingly slack.

The author dealt with fluctuating tensile stresses on pages 235 and 236. The importance of adequate tightening could not be overemphasized. Most of the failures of bolts, particularly large bolts, were due to undertightening rather than overtightening.

He agreed with the author that with bolts three inches in diameter and over there appeared to be no practical means of assessing adequate tightening, except, perhaps, by measuring the overall extension of the bolt, which was sometimes most inconvenient. Reliance on the experience and integrity of the fitters or engineers on the job was a most uncertain method, particularly if there were insufficient room to get a good swing at the bolts with a heavy hammer, as appeared to be the case with most of the more important bolts in an engine.

The importance of the relative elasticity of the bolted assembly and the bolt could not be emphasized enough. In these days, with welded entablatures and seatings and higher engine revolutions, one heard more and more frequently of transverse and vertical vibration of engines. This made it essential satisfactorily to fit holding-down bolts and chocks. It would be appreciated that the holding-down bolt, which had probably the smallest length-diameter ratio of any bolt on a ship, would have a very small elastic strain when fully tightened up. It was therefore essential that the tank top, chock and bedplate flange should have no spring in themselves and should be solid when bolted up. It was assumed, of course, that the bolting faces of the chocks were solid around the bolt and that the chocks were carefully bedded to engine bedplate and seating. If this were not the case, it was almost impossible to get the bolted assembly solid.

The author had recalled what was likely to happen if the elasticity of the bolted assembly was of the same order as the bolt itself; namely, that the bolt was subjected to dynamic load. With badly fitted holding-down bolts, therefore, it was problematical whether the bolt failed before it slackened off. It frequently happened that the landings for the bolt heads or nuts were not spot faced. This almost invariably ensured that the bolt started off with a high bending stress at the head or in the way of the first thread, which was bad enough. But it also ensured a condition in which local yielding would take place under dynamic load, permitting the full dynamic stress to be applied to the bolt.

Studs were a particular feature on their own—but then no one would fit a stud where it was at all possible to fit a bolt. This should apply in particular to the bolted assemblies of castings, such as brackets, and so on, on columns and the like.

Bottom-end bolts were a much simpler problem, but naturally they had been more in prominence because of the very spectacular damage which could result on failure of one bottom-end bolt. No doubt it would be some time before the design of large bottom-end bolts was really satisfactory, but further tests by the B.S.R.A. would certainly lead, in the future, to this knowledge being available.

Another most interesting consideration was the coupling bolt. It was sometimes met with that the criterion of a good fit was that the bolt would just be driven home, using the heaviest device that could be swung, slid or otherwise manipulated in the confines of the shaft tunnel. It was surprising what a little experience could accomplish in this direction. The nut was then applied and slogged up, and everyone was satisfied that a good job had been done. The tightening strain of the bolt under these conditions was probably insignificant, as it was problematical how much penetration was obtainable with so tight a bolt. It was not surprising, therefore, that in the presence of transverse vibration or malalignment of the line shafting, failure of the bolts was a common occurrence.

DR. H. H. ATTIA, B.Eng., M.Sc. (Member) said he would like to compliment the author on presenting a difficult problem in such a simple way.

In the summary it was stated that the strength of a bolted assembly which carried a fluctuating load was determined by the magnitude of the cyclic load that could safely be carried by the bolts.

There must be innumerable magnitudes of cyclic loads that could safely be carried by a bolt, provided it received the appropriate tightening in each case and the number of cycles did not exceed certain limits. It was suggested that the number of applications of a cyclic load was at best as important as its magnitude in determining the "safety line" and, without reference to it, this statement was meaningless.

It was known that even dynamic stresses above the fatigue limit might be harmless and sometimes beneficial if applied less than certain numbers of cycles. For any magnitude of cyclic stress, there was probably a limiting number of cycles below which the fatigue strength of a material might be improved and above which it might be decreased.

With regard to "the theoretical stress concentration factor", it would be appreciated that this "elastic" factor depended only on the geometry of the notch and was independent of the material. Theoretically, it was applicable for the same notch in any elastic material whether it was plaster or tungsten—regardless of the intrinsic properties of the material itself. It was, however, generally uneconomical to use in design, as it was for metals almost always higher than "the actual or effective stress concentration factor" which represented the ratio:—the fatigue limit without stress concentration over the fatigue limit with the stress concentration, for a particular material, size and notch. This latter factor depended on the properties of the material among other factors and generally differed from the theoretical factor according to the "notch sensitivity" of the material, which was somehow related to its ductility.

The author mentioned in the second paragraph on page 002 that "fatigue cracking takes place with very little deformation, even though the material may exhibit a high degree of ductility under static loading".

He could assure the author that considerable plastic deformation did actually take place by slip in the crystalline structure of the metal—depending on its ductility—before fatigue cracking occurred. The strain hardening of the metal under cyclic loading was a result of this plastic deformation which eventually led to the initiation of a fatigue crack. It must not be assumed that because there was no appreciable change in the external dimensions of a fatigued component there was no deformation. A good example of how deceptive external deformations of metals could be, was given by the results of the work of Smith and Wood (published by the Royal Society) who had shown that, although a residual external elongation was usually observed after yield of a specimen under tension, residual contraction actually occurred in the lattices of the grains of the metal. Such contraction was known to contribute to the increased hardness and tensile strength usually resulting from the plastic deformation and yield of a metal.

It was only when the "submicroscopic cracks"—formed by the process of slip within the grains—crossed crystal boundaries and joined one another that the conventional fatigue crack was actually initiated. This usually happened when the capacity for further deformation by slip of the highly stressed grains of the metal was locally exhausted. During this strain hardening process the strength of the metal was locally increased and at the same time its "notch sensitivity", and it was due to the ascendancy of one over the other that local weakening or strengthening of the material resulted under fatigue loading which might or might not arrest the propagation of an initiated crack.

In the third paragraph on page 234, the author stated that "it may be possible for large components to continue to operate satisfactorily under controlled stress conditions, even with a fatigue crack present". This was rather a dangerous general conclusion to be drawn from the limited evidence available. He suggested that the possibility of this happening in practice with a visible crack was rather rare under marine service conditions and that in any such rare case there must have existed some special combination of factors peculiar to the case which were mainly responsible for the lack of crack propagation. It was probable that crack propagation might depend on many other factors relating to the crack itself, in addition to the properties of the material achieved at the end of the crack, stress gradient and magnitude of the peak stress of the cycle, which were mentioned by the author.

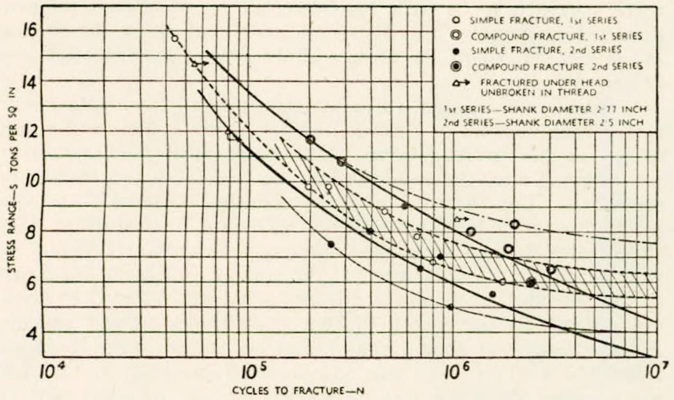
It was also most probable that fatigue cracking was generally initiated by a few cycles of excessive cyclic stresses rather than by millions of cycles of moderate ones—in which case it would appear that the usefulness of the value of "the fatigue limit" by itself was rather limited. The results of fatigue tests, complete with S-N curves, could only be made full use of, when statistical analyses of investigations of service loadings—including cyclic overstressing of loaded components—could be made to indicate possible stress ranges under service conditions and the numbers of cycles for each stress range occurring within a given period. When such information was available, it was possible that the life of each component might be accurately predicted from the S-N curves and the designers might then be able to decide the appropriate scantlings of a component by the use of a "*real* factor of safety" and not a fictitious misnomer as at present used.

This brought to his mind the story of Nevil Shute and his aeronautical fatigue scientist who rightly refused to fly in an aircraft in which some component or another had already passed its safe operational fatigue life. The day might come when this becomes a reality in marine engineering.

This method of statistical investigation might also be the answer to the author's conclusion on page 235 that "From further consideration of the problem, however, it must be concluded that the design stress is a purely hypothetical figure and bears little relationship to the actual stresses set up in the bolts securing an assembly subjected to cyclic loading".

He did not agree, however, with the author's statement in the last paragraph of page 236 that "there is little likelihood of overstressing large engine bolts" and his recommendations for hardening up as far as possible to provide an adequate safeguard. Results of tightening tests at which he was present had indicated that unnecessarily high initial stresses might be inadvertently induced in bolts by the normal method of hammering up the spanner. During two consecutive tightening operations by the same method and gang, the same degrees of nut rotation produced initial stresses in the bolts varying by as much as 600 per cent. It was therefore evident that no reliance could be placed on the method of assessing tightening stresses in a bolt by measuring the angular rotation of the nut. These stresses could only be accurately assessed by measuring the extension of the bolt and once a predetermined tightening stress—depending on the external load and the properties of the assembly—had been induced, the main issue and necessary precaution then became the maintenance of this tightness.

Referring now to Fig. 12, and admitting the considerable scatter in the fatigue test results of the 3-inch diameter bolts, it seemed to him that the S-N curves in this figure had been drawn rather too optimistically. They did not appear to pass through the majority of the relative points in each series as normally accepted. To do so, these curves would become much steeper than those drawn in that figure, and, bearing in mind the fact that the maximum number of cycles run in these tests was only three million, it would appear that the fatigue limit—on a ten million cycle basis—should be much lower than that suggested by these curves. He did not see how the chain dotted curves in Fig. 12 could be assumed to represent the endurance curves for the best and poorest series of specimens, and it was suggested that any conclusions drawn from such an assumption could only be misleading. It would appear from redrawing these curves (Fig. 21) that the fatigue



(MINIMUM STRESS OF LOADING CYCLE, 3 TONS PER SQ IN. THROUGHOUT)

FIG. 21

limit for the 3-inch diameter bolts—on a ten million cycle basis—could only be between 3 and 4 tons per sq. in. rather than between 4 and 7·6 tons per sq. in. as the author claims.

In connexion with the fracture of the three 3-inch diameter bolts which failed at the fillets under the head, he suggested that this fracture was due to some other defect at the nucleus of the fracture such as an undercut, a material defect, or a crack already initiated at that point, rather than due to the stress concentration at this fillet $\left(\text{with a ratio of } \frac{R}{D} = \frac{6}{40}\right)$. There was no doubt that the most severe stress raisers in a bolt were the deep sharp notches formed by the threads and he could not see how the stress concentration of such a comparatively moderate fillet could exceed that at the sharp notch of the bottom of the threads, particularly the first two threads engaging in the nut.

With regard to the mechanical properties of the material of the bolts mentioned in Table I, it was noted that the tensile strength for the 1-inch diameter bolts was *lower* than that for the 3-inch diameter bolts, yet the hardness of the former was *higher* than that of the latter. In his experience, he had found that smaller forgings produced under identical manufacturing conditions (reduction and heat treatment, etc.) from the same cast of steel as large ones, almost invariably gave higher tensile strengths than the larger ones. Furthermore, it was generally found that the tensile strength and hardness of steels were usually related, viz., the higher the tensile strength of the steel the higher its hardness, yet in this case the opposite was reported. Could the author shed some light on this interesting matter?

Finally, from consideration of the results of these tests, he could not help reaching the conclusion that the strength of dynamically loaded bolts was determined not only by the man with the wrench, the designer, the tightening and external loads and the material of the bolts, etc., but largely by the accuracy and care of the thread cutting, machining and finishing operations.

MR. A. F. C. BROWN described photo-elastic tests on screw threads which had been carried out by the National Physical Laboratory. He explained that the results had been embodied in a paper by Mr. V. M. Hickson and himself which had been offered to the Institution of Mechanical Engineers, and it was with the agreement of the Institution that he communicated some of these results.

The work was concerned with stresses at the root radii in screw threads, measured by the "frozen stress" photo-elastic method. The main object of the work was to check some theoretical results arrived at by Dr. Sopwith and given in another paper to the Institution of Mechanical Engineers.* In this paper, Dr. Sopwith described a special form of tension nut which was mentioned in Mr. Taylor's paper. Dr. Sopwith's nut was peculiar in that not only was it tapered, but the bolt was hollowed out as well in sympathy. At the same time as the test on the tension nut, the opportunity was taken of examining the stresses in an ordinary bolt and nut which had already been done by Hetényi, as quoted by Mr. Taylor.

A distinction had already been clearly drawn that evening between stress concentration and load concentration. Stress concentration was a geometrical effect brought about by the shape of the notch, whereas load concentration was a different effect which was illustrated in Fig. 5 of Mr. Taylor's paper. The results he was about to show were influenced by both these factors and referred to purely elastic conditions.

The first slide showed the form of test specimen used, the material being Fosterite. It consisted of a 1½-inch Whitworth stud with a tension nut at one end and an ordinary nut at the other.

The frozen stress technique was as follows. A comparatively small load was applied to the specimen which was then heated in an air oven to about 90 deg. C. After time had been allowed for the heat to penetrate, the model was cooled slowly with the load still on. After the load was removed, the model retained the strains which had been imposed when it was hot and it could in fact be cut into slices for examination in a photoelastic polariscope. From measurements on longitudinal slices through the roots of the threads, it had been possible to determine the stresses in the various threads.

The next slide showed the results for the ordinary stud and nut. The tensile stress in the stud rose to a high value in the thread nearest to the bearing face of the nut, the maximum value being twelve times the stress in the full body of the stud. Such a high value would not occur in a steel stud owing to the stress peak being relieved by local yielding. The tensile stress on the nut, on the other hand, was quite low. This was because the tensile stress set up as a result of the bending of the threads was counteracted by the general compression in the nut. In the stud, on the other hand, the tensile stresses were augmented by general tension in the stud.

Another slide showed the corresponding pair of curves for the tension stud and nut. The peak in the curve for the stud stresses had been largely removed, showing that the load had been more uniformly distributed along the engaged portion of the thread. It was also of interest that the stress in the nut became comparable with that in the stud.

As Mr. Taylor implied in his paper, a stud and nut of this kind were not very practical. The design was complex and required very accurate machining. Mr. Brown, however, considered the results to be valuable, not only because they

* Sopwith, D. G. 1948. "The Distribution of Load in Screw Threads". Proc.I.Mech.E., Vol. 159, p. 373.

proved the correctness of Dr. Sopwith's theory but because they demonstrated a means of greatly improving the load distribution in a screwed joint.

MR. E. W. CRANSTON, Wh.Sc. (Member of Council) said the author was to be congratulated on investigating a subject of great importance to designers and users of all types of reciprocating machinery, especially as there was very little information published in this country. However, certain engine-builders had made large-scale investigations similar to those described in the paper, and some of the results obtained had been published on the Continent.

These investigations showed that the designer of machinery, besides using certain rules derived from past experience, must pay particular attention to bolts, studs and nuts subject to fluctuating stresses and also to those subject to shock loading. By this he meant bolts holding piston closing covers, which were subject to hydraulic shock loading from the piston cooling of Diesel engines. In modern designs it was usual for important bolts used under these conditions, to find that the diameter of the body had been reduced wherever possible to approximately the diameter at the bottom of the threads, and also that a radius had been formed between the body and the head. For cases of shock loading, it was usual to make the bolts as long and as elastic as possible, increasing their length if necessary by inserting distance collars beneath the nuts. Two of the most important considerations were the choice of material and the type of thread to be used. The former was determined from the size of the bolt and the resulting stresses, whilst the latter was very often determined from other considerations apart from fatigue strength.

A certain amount of work had been carried out on the fatigue strength of bolts of different material and with different types of thread. It was evident from the paper that information of this sort was not generally known, and the following examples might be of interest.

The bolts tested were generally similar to those described in the author's tests, shown in Fig. 8, but with a body diameter of 31 mm. (1·22 inches), threaded portion 50 mm. (1·97 inches) long and unthreaded portion 280 mm. (11·02 inches) long. One set of bolts was turned from forged steel very similar to that used by the author, i.e. with analysis of carbon 0·18 per cent, manganese 0·91 per cent, silicon 0·19 per cent, phosphorous 0·017 per cent and sulphur 0·028 per cent, and ultimate tensile strength of 29·2 tons per sq. in.

The types of thread tested were Whitworth 1½-inch, Acme K No. 38 (38 mm. = 1·496-inch diameter), Standard International 39 mm. diameter by 4 mm. pitch, British Standard Pipe 1⅛-inch and Standard International Fine 36 mm. diameter by 2 mm. pitch. The lowest stress in the fatigue tests was always maintained at 15 kg. per sq. mm. (9·5 tons per sq. in.), a much higher figure than that given by the author. The loading cycles were 350 per minute. The fatigue limits for the various threaded bolts were in the order set out above; i.e. the Whitworth thread was the best at 4·8 tons per sq. in.; then Acme K No. 38 at 4·3 tons per sq. in.; S.I. 39 by 4 at 3·8 tons per sq. in.; 1⅛-inch B.S.P. and S.I.F. 36 by 2 at 3·2 tons per sq. in. Thus the original Whitworth thread showed up better than most other threads which could be used.

For similar bolts of chrome-nickel steel with a composition of carbon 0·28 per cent, silicon 0·15 per cent, manganese 0·50 per cent, chrome 0·80 per cent and nickel 3·40 per cent and ultimate tensile strength of 52 tons per sq. in., three types of thread were tested, with the result that the merits of Whitworth and Acme K threads were reversed. The fatigue limits were Acme K No. 38—6·9 tons per sq. in.; 1½-inch Whitworth—5·7 tons per sq. in.; and S.I. 39 by 4 mm.—5·4 tons per sq. in.

In studying the above results, it would be found—especially for the first-mentioned tests—that there was some relation between the fatigue limit and the radius at the root of the thread. The ratios of radius at root of thread to the core diameter of the thread were:—

1½-inch Whitworth thread ...	0·0178
Acme K No. 38	0·0161
1⅛-inch B.S.P.	0·00907
S.I. 39 × 4	0·00694
S.I.F. 36 × 2	0·00349

The paper pointed out that it was important to design the parts to be held together in such a way that there was no bending of the bolts, or at least that this was at a minimum. It was also important to choose a symmetrical arrangement of bolting, i.e. not one short and one long bolt for holding down a part such as a fuel injector body on a Diesel engine.

Not only must bolts be carefully designed but also studs and set-bolts, which were often subject to dynamic loading—for example, cylinder cover studs. It had been found that the best arrangement was to have collar studs, each with a portion below the collar turned to a diameter equal to or a little below the core diameter of the thread. Where collar studs could not be used, the next best arrangement was to fit studs which "bottomed" in the tapped holes. Short studs should not be used, and if necessary spacing collars should be placed under the nuts.

The design of the nut to go with dynamically loaded bolt or stud needed careful consideration, especially where the latter was highly stressed. Figs. 5, 6 and 7 gave guidance, but there appeared to be little full-scale testing of bolts with nuts designed to improve the distribution of load along the threads in contact. Perhaps the author could extend his investigations in this direction.

Apart from design, there was scope for great improvement in the manufacture of threaded components, as the author had found in his investigations. The body of a bolt, especially the radius under the head, should be free from tool marks or scratches. The thread should be accurately formed, and one method used for large bolts and studs might be of interest. The thread was turned or milled almost to size and finished by thread-rolling. This improved the surface finish on the flanks of the threads, i.e. the surface irregularity was reduced to about one-seventh to one-eighth of that of ordinary turned threads. Fatigue tests had shown that the stress range could be raised from 8·2 to 10·5 tons per sq. in. with breakage of the test pieces after about 10 million cycles, i.e. an improvement of about 28 per cent in fatigue strength, with threads which had been rolled.

During erection of an engine during manufacture and after overhauling, it was important that bolts and studs should be fitted and tightened up correctly. The faces under the heads of bolts and nuts should be square with the axis in every case, not only for running gear bolts but also for foundation bolts, which were also subject to dynamic loading and were often not treated with the care they deserved. This point had been covered by Mr. Bunyan.

The best way of tightening important bolts, including tie-rods, was to use a hydraulic stretching device which loaded the bolts and the structure to be held together by amounts previously determined. Other methods included:—

(*a*) the use of spanners with torque measuring attachments;

(*b*) tightening by turning the nut through a certain angle from the hand tight position;

(*c*) heating the bolt to a certain temperature and tightening the nut by hand; and

(*d*) measuring the extension of the bolt whilst tightening.

Some of these methods had definite disadvantages, but all should give improved results over the more usual tightening procedure which depended on the strength, training and experience of the fitter or erector.

MR. R. COOK, M.Sc. (Member of Council) said that as a colleague of the author he might, perhaps, be permitted to make a few remarks as an addendum to the paper.

He would like first of all to refer to the fatigue tests on the 3-inch bolts. It was clear from the test described by the author that the fatigue strength was dependent to a very large extent on the accuracy of the screw-cutting and that large variations in strength could occur with the normal method of cutting threads in a lathe with a single-point tool. Little is known regarding the relative merits of alternative forms of screw cutting, such as milling, grinding and rolling in these large sizes. A fresh series of tests had therefore been put in hand in which these factors were being investigated. Incidentally, it was obviously desirable that in the new series of tests the threads in the nuts should be of the highest possible degree of accuracy so as to eliminate one variable factor. As a preliminary, therefore, a number of specimen nuts were prepared, using various alternative methods of thread-cutting. When they were examined for accuracy, it was found that the nuts varied very considerably, according to the method employed.

Another direction in which there was a marked lack of authoritative data was connected with studs. A previous speaker had made claims regarding a certain type of stud, and other claims had been made from time to time for various other designs. As far as he was aware, however, there were no really reliable data in the form of results of fatigue tests as to their relative performance under dynamic loading. This was another matter which was now being actively investigated.

The author had referred in his paper to the report of a committee of the Institute in 1945 on the failure of auxiliary Diesel engine connecting rod bolts. This subject was not unrelated to the present paper, and perhaps a few remarks would not be out of place.

Three practices had been adopted at one time or another in an attempt to reduce the extent of such failures; namely, periodical heat treatment, renewal after a specified period of service, and periodical examination. It was now generally recognized that periodical heat treatment was not advisable, because in the first place it was very doubtful whether it did any good in improving resistance to fatigue or to shock loading. Secondly, incipient flaws were likely to be intensified. Thirdly, unless there was proper control, there was a good deal of risk of doing more harm than good. The latter consideration applied with special force to alloy steels, where there was also a risk of distortion.

The second practice was the one which was favoured by the committee of the Institute. It would appear that this was because a number of superintending engineers reported marked reductions in breakages following upon the introduction of such a policy. The author and himself had had occasion to look into this matter some three or four years ago, and they came to the conclusion—rather reluctantly—that they could not agree with this policy. There seemed to be strong grounds for doubting whether any particular period of replacement (four years was recommended) would be short enough to reduce the incidence of failures. The evidence did not seem very convincing, because obviously other precautions might well have been taken at the same time, and they obscured the issue. Thus, for example, any superintending engineer who was faced with a crop of bolt failures and decided upon a policy of bolt renewal would at the same time probably take exceptional steps to see that the bolts were thoroughly and adequately tightened up when they were renewed.

At the time, it had seemed to the author and himself that periodical inspection was the most logical of the three procedures. He was not quite sure whether the author would still agree with that. It would be interesting to hear his views. There was a proviso—that it should be arranged for at intervals of not longer than, say, a year; that inspection should be thorough, probably involving sending the bolts ashore for the purpose; and also that exceptional steps should be taken to see that the bolts were adequately tightened. Otherwise, one might do more harm than good.

When one realized that, as shown clearly by the author, the dynamic load going on to a properly tightened bolt was only a fraction of that applied to the bolted member, it was difficult to resist the conclusion that faulty assembly was responsible for the majority of bolt failures. It would seem that a policy aimed at ensuring proper assembly was the one which in the long run would prove most effective.

COM'R(E) J. I. T. GREEN, O.B.E., R.N. (Member) quoted from his own experience a case in which a Diesel engine was held together by long tie-bolts which failed very rapidly on service. The trouble was cured exactly as recommended in the first part of the paper, by rebedding the mating parts of the structure and applying a calculated tension to the bolts.

Concerning the second part of the paper, the frequency of application of the load on the 1-inch bolts was ten times that on the 3-inch bolts. He understood that speed of loading had some effect on results, particularly in notch brittleness tests. Could the author confirm that the use of different machines in the two cases would have no effect on the relative results?

MR. T. SCHUR, B.Sc., A.M.I.Mech.E., A.M.I.E.E., said that various methods of producing threads had been mentioned, among them grinding, cutting and rolling. He believed it was established that rolling was the best way to produce good threads for cyclic loading and that ground thread was actually less able to withstand cyclic loading than cut thread. He had always understood that this was due to the plastic deformation on the flanks which was produced by cutting or by over-rolling and was not produced by grinding. He would be glad to hear the author's views on this question. Perhaps fuller information might be revealed by further tests.

A great deal had been said about the importance of making the bolt as elastic as possible. Reduced shank diameter had been mentioned. This was clearly the best method of achieving the desired result. Where it was too expensive, however, a very stiff bolt could be made more elastic by making the thread portion longer than was often done. One very often found a bolt with a free thread portion of two or three threads only. The remainder had full shank diameter. Better results were obtained, he believed, if the free thread portion was as long and the shank as short as possible. Tests had shown that the free thread portion should be at least the length of one diameter, and this applied equally to bolts and studs.

Correspondence

MR. A. W. DAVIS, B.Sc. (Member) considered that the author had put forward a clear and concise argument for better design of bolts subjected to cyclic variations of stress, and for the need which existed in many cases for a higher quality of finish. The necessity for the avoidance of sharp changes in section, which embraced the need for good fillets, could never be over-emphasized.

Regarding the expression for the instantaneous load in a bolt uniting members in tension as given by equation (1), it was suggested that in the particular case of side connecting rods, k_b being large in proportion to k_a, the quantity CW was so small as not to be of serious consequence. On the other hand, trouble would most certainly arise if W_i were less than W—that was when separation occurred. Such a condition could be readily brought about by the explosion of excess oil in a cylinder consequent, say, upon a sticking fuel valve, the relief valve area being perhaps inadequate to avoid pressures arising to an extent that elongated the bolts in question. This could be simply verified with a spanner and a slogging hammer after such an occurrence and, if care were always taken to effect such hardening up after a heavy explosion, bolt failures in side rod members would be a rare occurrence.

MR. S. H. DUNLOP wrote that the author's paper was of a factual nature, supplying very interesting results, and his definite approach to the numerous problems did not permit of criticism, but merited several observations.

The author had gone a long way towards reducing this difficult problem to practical perspective; his work also afforded confirmation of various accepted principles. He showed conclusively the effect of slight inaccuracies in production, the value of prestressing and the necessity for an adequate fillet radius. It was well known that there was no direct relationship between impact strength and fatigue strength.

Specimen testing to provide a suitable material of indefinite life was still acceptable apparently, although the testing of the finished article might prove the accepted material unsuitable. Was the author implying in the concluding introductory sentence that tests of the commercially produced article should be applied before a definite material decision was made? This would be a very lengthy process and it was doubtful whether the tests would eliminate the human element factor arising in mass production.

Concerning the difference in performance between the 1-inch and 3-inch bolts, it was known that the fatigue strength of steels decreased about 4-16 per cent with increasing size of test piece. Although fatigue was responsible for most of the failures occurring in machine parts, it was still necessary to base design on the ultimate strength since this was the steelmaker's classification, and it was usually safe to take the fatigue strength as 40-60 per cent of the ultimate strength.

The prestressing of bolts was generally applied by Diesel engine manufacturers and the education of works' personnel in the use of such apparatus developed appreciation of its importance. The apparatus employed was supplied as standard equipment but its future intelligent application was doubtful and the author's conclusions that such refinements were hardly practicable led to consideration of its importance or otherwise.

The author's finding that prestressing reduced the effective fluctuation stresses to one-fifth was illuminating, but the conclusion drawn by the author that there should be as much prestressing as possible was scarcely acceptable. Hetényi, while showing that flexibility of structure resulted in bending fatigue failure, had drawn attention to the importance of *proper* preloading. In this connexion the author admitted the necessity for ductility of the nut, and of course, the whole mechanism of fatigue was one of strain hardening, the rupture of the atomic bonds leading to the formation of micro-cracks which grew into macro-cracks. It was important to bear in mind the effect of excessive prestressing on the life of other components.

The illustrated methods of improving the distribution of load along the nut were interesting; it appeared that thread accuracy was essential and employment of such methods would be limited to a few accepted problems. Commercially produced threads apparently produced a load distribution to meet normal requirements. Could the author give some indication as to when the use of commercial threads became unreliable, also when the new nut theory should be applied?

A rolled thread would maintain the prestress with less change than a cut thread because of better thread engagement, owing to improved surface finish. Was the author in favour of rolled threading despite the loss of accuracy entailed?

Higher strength alloy steels were becoming common in Diesel manufacture and it would be of interest to have the author's opinion of the conclusions expected after submitting such materials to similar fatigue tests. Would they follow those indicated in the present paper, with emphasis on the importance of body and thread finish?

The author's comments concerning the limitations of high tensile steels were somewhat controversial. Although the notch sensitivity increased with rising tensile strength, the actual endurance limit improved markedly. Accordingly, the statement that there might be no advantage in the use of high tensile steels called for qualification. The cases on record where high tensile steels had a lower fatigue strength than mild steel were altogether exceptional. The ratio of the fatigue limit to the ultimate tensile strength was usually 0·45 to 0·55 but it tended to fall to 0·4 for 100-ton steels.

W. Staedel (Mitt. der Material-pruf. Darmstadt, 1933, *14*, 22) had given the distribution of bolt failures as 65 per cent at the first thread in the bottom of the nut, 20 per cent at the last thread in the bolt and 15 per cent under the head. That a high stress concentration existed at the first thread in the nut had been shown photo-elastically by Hetényi.

Fatigue strength varied with the amount of forging, particularly owing to the elongation of non-metallic inclusions, causing marked directional differences in all mechanical properties. Non-metallic inclusions had a marked effect on the endurance limit, hard inclusions of silicates and aluminates being particularly detrimental, especially when near the surface.

It was evident from the author's results that calculations based on cylinder pressure and inertia of the reciprocating parts were unreliable, since provision must be made for cyclic loading. The variations disclosed by the author were so considerable as to indicate the advisability of generous allowances in design. The author's test pieces, finished with a ground-thread chaser, appeared to have been of a rather higher standard than the average production finish. His standard was particularly high in respect of pitch and form of threads, the pitch error being less than one thousandth in 4 inches, a small pitch error evidently having a great effect on stress concentration. Even with so high a standard of finish the great variation in the results was impressive, since the ordinary production finish would lead to wider variations. In practice it would seem advisable to reduce the cross sectional area of the bolt to much less than 80 per cent of that of the thread core and also to have a fillet radius of much more than one-eighth of the shank diameter.

The surface condition became increasingly important as the tensile strength was raised.

The author was to be congratulated on his successful application of tests corresponding more closely to actual service conditions than those adopted by previous investigators. It was to be hoped that he would follow up his valuable investigation with fatigue tests in torsion, preferably at ten million cycles; perhaps in subsequent tests he would consider the influential temperature fluctuations which occurred in practice.

MR. J. E. FIELD considered that the expressions (1) and (2) on page 236 were only really valid when the external load was applied at the outer ends of the bolted members, close to the bearing faces of the nut and bolt head. When the load was applied at the inner mating surfaces of the members, the total bolt load = W_i = initial tightening load (so long as no separation occurred). For points of application of the external load between these two, an expression

$$W_t = W_i + \frac{1}{L} C W$$

obtained, where 1 = distance between points of application of external load, L = distance between bearing faces of nut and bolt head.

Connecting rod assemblies of the type illustrated in Fig. 3 would, insofar as the direct tensile stresses were concerned, fall within this intermediate case. The external loads would be applied roughly at the underside of the bolt head (end of connecting rod) and at the inner surface of the lower half of the bearing assembly, through the crank pin. This would reduce C to about half the author's value, though it was difficult to say exactly where the effective points of load application were.

The author's expression always gave the greatest value for W_t and was safest for design purposes, particularly in view of the uncertainty as to W_i.

On page 237, the author recommended the use of a ductile material for the nut, in order to improve the load distribution. Ductility was hardly the point—the load distribution might be improved, insofar as nut material was concerned, by use of a material with either a low elastic modulus or an appropriately low yield point—the *amount* of yield required was very small. The author made this point himself on page 238, line 4, *et seq.*

It was not made clear on page 237 (column 1, line 19, *et seq.*) that the load transmission from nut to bolt was by cantilever bending of the threads and that the most serious stresses set up were the bending stresses near the roots of the bolt threads, as shown in Sopwith's analysis.*

It was, however, feasible to taper the *bolt* threads. This had been done successfully in the United States.†

A major factor in respect of the nut shown (page 237, line 14) was that it was "overhung", so that most of the nut was in tension, as well as the bolt, thus reducing the pitch differential caused by applying load to the joints (Figs. 6(a) and (b)).

MR. W. A. P. FISHER wrote that some tests‡ made by the writer and his associates were designed to demonstrate to the satisfaction of airframe designers that the deciding factor in the fatigue of bolts in tension was, as J. O. Almen put it some years ago, "the man with the wrench".

The need for controlled preloading of bolts in reciprocating parts was well known to engine designers. The effective application of the principle, however, presented difficulties. With small bolts, care must be taken not to approach the torsional strength of the bolt. With large bolts, on the other hand, the load to be imposed was so great, and available space frequently so limited, that the problem was to apply sufficient torque; and, in any case, with machine-cut threads the relation between torque and tensile load was apt to be somewhat variable. A recheck on tightness after a specified running period was always advisable. The elastic extension of a low tensile steel bolt stressed to, say, 15 tons per sq. in. was only 0·001 inch per inch length, and, the surfaces not being planished, the bedding-in of high spots must produce a measurable loss of tension. The provision of secure locking with, as nearly as possible, infinite choice of nut position and elimination of split pins and tab washers offered scope for ingenuity in the design office.

Most of the suggested design features were to be recommended even though the bolt were preloaded. The advantage of using a "tapered lip nut" was well established, also of reducing the shank diameter somewhat below the core diameter, when the first thread was engaged by the nut.

Fluctuating bending loads were of common occurrence, and were severe on the cross section immediately below the head. Hence, a generous head fillet radius, free from tool marks and preferably burnished, was all the more desirable.

The introduction of compressive surface stress in an axial plane at fillets was a well known and highly effective means for raising the fatigue strength. This occurred when threads were roll-formed or roll-finished. Did the author know of instances where these processes had been applied to large mild steel bolts?

When the author quoted the "limiting stress range", he evidently meant what most engineers would call the "limiting alternating stress" and not the full stress range from minimum to maximum. Conformity with normal practice would help to avoid ambiguity.

With reference to the tests made by the author, the occurrence of compound fractures indicated a high initial peak load at several threads' distance from the theoretical position shown in Fig. 5, since the first crack must have been that farthest from the bearing face. In Fig. 13, for example, the first crack started at the fifth engaged thread fillet. Spreading of the thread loading away from the first thread was obviously advantageous, provided no local peak of high pressure was produced. When using a low yield-point steel, accurate form and pitch, together with smooth surfaces (qualities obtained better by rolling than by machining), could give even contact, while better thread load distribution could be had through slight local yielding. It would be interesting to know what improvement in the lower limiting endurance curve in Fig. 12 would result from first tightening the bolt to preloading stress and then slackening off to the test condition.

DR. J. E. RICHARDS (Associate Member) wrote that the author drew attention to special types of nuts designed with the object of giving a better distribution of load and also to the fact that small errors in pitch might entirely mask the effects of these nuts. This might be illustrated by the magnitude of the relative displacement of the bottom and next thread of a one-inch diameter B.S.W. bolt which had been calculated by Sopwith* for a bolt load of 6,000lb. and found to be 0·00004 inches. It was doubtful whether an accuracy of this order could be achieved by any production machine tool and, in contrast to this, the allowable pitch error for the B.S.W. close fit thread was a mean of 0·0004 inches per thread assuming six threads were engaged and it was possible to have a pitch error of 0·0023 inches between adjacent threads and still be within the tolerances laid down. This threw considerable doubt on to the efficiency of these special nuts and consequently it would be interesting to know if their theoretical advantages had been proved by fatigue tests.

It had been suggested that a better load distribution would be achieved by using nuts having a lower elastic modulus than the bolts but it seemed probable that pitch errors were too great for this to give any appreciable relief. A more or less uniform distribution of load seemed to be achieved in practice by the yield stress of the material of the nut being appreciably lower than that of the bolt. In this way the large errors in pitch which must exist in practice might well make little difference to the fatigue strength. It appeared possible that the surface finish at the bottom of the thread of the bolt was a primary cause of the scatter of the results and that rolling the roots of the threads reduced this scatter and improved the fatigue strength.

The initial tightening of the bearing assembly was vitally important even for bolts with a very low residual stress and as this was so there might be some advantage in using smaller diameter bolts. For instance, the use of two $1\frac{1}{2}$-inch diameter bolts (which were about the largest size which could be tightened by a torque spanner) instead of one 3-inch bolt might be justified if the initial tightening of the smaller bolts could give a definite tightening stress of 12 tons per sq. in. and with the 3-inch bolts a stress of only 6 tons per sq. in. could be assured, even though the cross sectional area of the smaller bolts was only half that of the larger bolt. When this investigation had progressed further it might be possible to assess the relative efficiencies of various designs. At present the only published values of tightening stresses appeared to be those given by Labrow.§

MR. P. M. THRELFALL, M.Sc.Tech. (Graduate) thought that the distribution of an external load between the housing and the bolt of a previously tightened assembly might be illustrated by Fig. 22. Using the author's notation from page 236, the maximum external load which could be applied before abutment separation took place was given by:—

$$W = W_i \left(\frac{k_a + k_b}{k_b}\right)$$

* Sopwith, D. G. 1948. "The Distribution of Load in Screw Threads". Proc.I.Mech.E., Vol. 159, p. 373.

† Stoeckley and Macke. 1951. "The Effect of Taper on Screw Thread Load Distribution". A.S.M.E., No. 51-S-15.

‡ Fisher, W. A. P., Cross, R. H. and Norris, J. M. June 1952. "Pre-tensioning for Preventing Fatigue Failure in Bolts". "Aircraft Engineering", Vol. XXIV, No. 280.

§ Labrow, S. 1947. "Design of Flanged Joints". Proc.I.Mech.E., Vol. 156, p. 66.

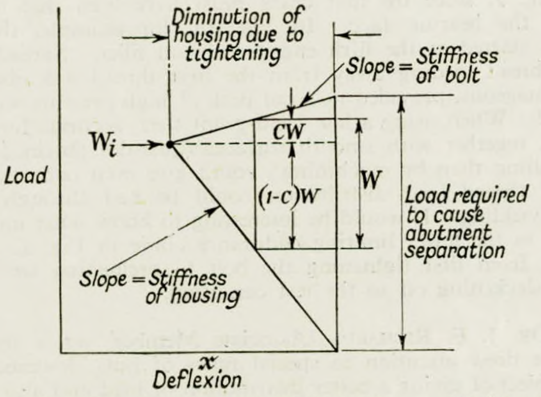


FIG. 22—*Showing the distribution of an applied load*

Additional load carried by the bolt $= CW = \left(\frac{k_a}{k_a + k_b}\right)W$.

Relief of compressive loading in the housing $= (1 - C)\,W$

$$= \left(\frac{k_a + k_b}{k_b}\right) W$$

When dealing with bolted connexions, two factors of safety must be observed, one determined by considering the maximum permissible working stress in the bolt, and the other determined by considering the minimum stress required in the bolt to avoid abutment separation. Failure to appreciate either of these factors would result in disaster.

The reduction in "load concentration" from 3·85 to 3·00 given by the "tapered-lip nut" shown in Fig. 7 was not very great. Had the author any comparable figures for the tapered thread and diferential pitch methods for obtaining a modified load distribution along the length of loaded thread? Whilst both these methods involved production difficulties, they should not be condemned on this score alone. The production difficulties should be considered in the light of the benefit to be gained.

In order to be able to make a more accurate assessment of the maximum load which large end bolts might be called upon to carry, it would be useful if the author could provide information regarding the magnitude of the acceleration of the propeller and shafting which arose due to the pitching of a vessel in heavy seas.

Author's Reply

The author wished first to thank all those who had taken part in the discussion, both verbally and in writing, for their valuable comments and expressions of appreciation. A large number of points calling for comment on the part of the author had been raised by contributors and he had attempted to deal with these in the order in which they appeared in the printed discussion.

The author's thanks were due to Mr. Bunyan for drawing attention to some of the important bolted connexions that were subjected to cyclic stresses. One did not, for example, tend to think of coupling bolts carrying fluctuating tensile stresses but it was clear that, if the shafting was not truly in alignment, which, of course, was difficult to ensure in practice, each bolt would be stressed in this way every time the shaft rotated.

Mr. Bunyan had suggested carrying out tests with loose-fitting nuts but it was the author's opinion that the fit of the nut was of secondary importance compared with the accuracy of pitch, profile and parallelism of the threads. Another point in this connexion was that, with very loose fitting threads, the bending stress at the thread roots would tend to be increased.

With regard to the reduction of the diameter of the shank of the bolt to much less than that used in the tests, he reminded Mr. Bunyan that such a practice could lead to undesirably high stresses being set up when tightening, unless some accurate method was adopted for measuring the initial pre-stress. With the present assembly procedure, a reduction in diameter to much below 80 per cent of the thread root area was not recommended, although it was desirable from the point of view of reducing the fluctuating load component. It would be seen from the results of the tests on unnotched specimens and the 3-inch bolts that to equalize the fatigue strength of the plain portion of the bolt and the section at the loaded face of the nut would require a reduction to about 20 per cent of the thread root area; this was hardly practicable. As pointed out in the paper, however, the extension of the reduced section of the shank into the nut might lead to an increase in fatigue strength owing to relief of the stress concentration at the first loaded thread.

Mr. Cranston had also touched on this question of the optimum reduction of diameter for bolts which carried shock loads. The author pointed out that in these circumstances the main factor to bear in mind was the capacity of the bolt to absorb strain energy. Since the energy stored in the bolt was proportional to the square of the stress and the volume of the material stressed, it followed that the energy absorbing capacity could be greatly increased by reduction of the shank diameter. Where one had shock loading, therefore, it would seem eminently desirable to make the plain portion of the bolt much smaller in diameter.

The author agreed with Dr. Attia that there was an innumerable number of stress ranges that could be applied for limited life; in the paper, the term "safe stress" indicated one that could be applied an indefinite number of times without failure. It was also agreed that plastic deformation took place within the crystalline structure of the metal under fatigue loading; indeed, this was the essential preliminary to the initiation of a crack, as pointed out in the paper. In making reference to the lack of deformation accompanying fatigue failure, it would be seen from the context that a comparison was being made between the phenomenological behaviour of components tested under static and fatigue loading.

On the question of crack propagation, the author wholeheartedly agreed that to leave a part in service when it was cracked was asking for trouble. What he had been anxious to point out, however, was that a cracked part might, in certain circumstances, continue to operate satisfactorily. Such circumstances undoubtedly did arise where the operator was unaware that a crack was present.

Dr. Attia had drawn attention to the lack of information about the actual stresses in engine running gear, shafting, etc., under operating conditions. It was evident that until more data were available on the frequency and magnitude of overloads, the designer was more or less in the dark. The experimental difficulties involved in the determination of operating stresses over a sufficient length of time to cover all service conditions were formidable but only by such work could the "real factor of safety" of the design be assessed.

In spite of Dr. Attia's remarks on the possibility of overstressing bolts by the normal hardening-up procedure, the author maintained that this was preferable to insufficient tightening and, in the absence of precise methods of measuring bolt extension, there was no alternative to hammering-up the nut as far as possible. The tightening stress obviously depended to a large extent on the human element, as well as the type of equipment used, but for large bolts of 4 inches in diameter or above the main hazard appeared to be the possibility of understressing rather than the reverse.

With regard to the modified S-N curves for the 3-inch bolts shown in Fig. 21, there appeared to be no evidence to suggest that the form of the curves would be different from those obtained from the results of the 1-inch specimens, which included points at 5×10^7 cycles. It was agreed that the scatter of the test results allowed considerable latitude in drawing the curves but in order to give the extreme limits in limiting stress range for all the bolts tested, the boundary curves should include all the test points; on this basis, following Dr. Attia's curves in Fig. 21 (shown as solid lines), the minimum limiting stress range for ten million cycles would be only about 2 tons per sq. in. These curves should be compared with those enclosing the shaded band in Fig. 12 and not the boundary curves. This point would probably be cleared up by the results of further tests which were in progress.

The failure of the 3-inch bolts by fracture at the fillet under the head was undoubtedly due to some local machining irregularity, probably coupled with a particularly favourable load distribution along the threads of the nut, which would tend to reduce the total concentration of stress at the roots of the first threads in the nut.

With regard to Dr. Attia's final remarks on the disparity of the figures given in Table I for the tensile strength and hardness of the test bolts, the author pointed out that the tensile strength values were determined from specimens cut from the blooms before forging. The fact that the 1-inch bolts had a greater forging reduction would account for the increase in hardness of some of the specimens, with a corresponding increase in strength. In this type of work one of the major difficulties was to ensure similar properties in specimens of different size.

The author thanked Mr. Brown for his remarks, which did not call for any comment on his part other than to congratulate the speaker on overcoming the experimental difficulties of the photo-elastic technique when applied to a screwed connexion, with such successful results.

The results of the comprehensive tests carried out on the Continent, quoted by Mr. Cranston, were of the greatest interest. It would be noted that the limiting stress range for the 1½-inch Whitworth bolts came within the range of values obtained for the 3-inch bolts given in the paper. It was also of interest to note that the increase in fatigue strength of the alloy steel bolts, which had a tensile strength 80 per cent greater than the mild steel, amounted to less than 20 per cent.

The specimens used in the tests mentioned by Mr. Cranston which had the greatest geometrical similarity to the 3-inch bolts were those screwed 1⅛-inch B.S.P. (thread root radius to core diameter ratio 0·009 compared with 0·0082 for 3-inch diameter 6 t.p.i.). These had a limiting stress range of 3·2 tons per sq. in. compared with a minimum value of about 4 tons per sq. in. for the large bolts (from Fig. 12). This would indicate a small size effect of an opposite nature to that usually observed but the lower strength of the smaller bolts might possibly be attributed to the greater value of the minimum stress of the loading cycle applied to the 1⅛ B.S.P. bolts.

The author was interested in Mr. Cranston's remarks regarding the merits of various stud designs. As mentioned by Mr. Cook, there did not seem to be available any test data on the relative fatigue strengths of studs of different types and this subject was being investigated by the B.S.R.A.

With regard to the testing of improved designs of nuts on a large scale, no work was in hand at the moment on this subject but it might receive attention at a later date. Investigations into the effect of nuts of different designs and materials on the fatigue strength of bolts had been made by Wiegand in Germany but these tests were on threads of only ⅜-inch diameter. For further details the reader was referred to a paper by S. M. Arnold,* which included a review of Wiegand's results.

The question of determining the tightening stress in a bolt from the angle through which the nut was turned from the hand tight position had been referred to by Dr. Attia; it could be concluded that this procedure was not satisfactory.

The author endorsed Mr. Cook's view that, discounting the possibility of faulty material, the majority of bolt failures could be attributed to incorrect assembly. If the design of the bolt and the bolted parts was such that stress concentrations were minimized and cyclic stresses reduced to a minimum, and precautions were taken to ensure adequate tightening, there should be no necessity to renew bolts unless for some reason they had been subjected to severe overloading. On the other hand, if exceptionally heavy loads were transmitted to the bolts at infrequent intervals, the policy of bolt renewal after an arbitrary time in service might have something to commend it. He still maintained, however, that detailed periodical inspection, coupled with the precautions mentioned above, was the most logical procedure to prevent the severe damage resulting from bolt failures.

In reply to Commander Green's query regarding the effect of the speed of the testing machines, the author said that experimental work by a number of investigators had shown that this factor was negligible up to speeds of 10,000 cycles per minute. When testing specimens at fairly high speeds and at high stresses, i.e. well above the limiting stress range, considerable heat was generated, however, particularly in the case of large test pieces. This had a modifying effect on the fatigue strength. The amount of heat generated depended on the volume of highly stressed material and, since in the bolt specimens this was quite small, owing to the high stress concentration, it was not thought that the difference in the speed of testing of the two sizes of bolts was of any significance.

With regard to Mr. Schur's comment on the relative fatigue strength of cut and ground threads, it was usually considered that the reason for the somewhat lower strength of components with ground surfaces was that this process left the surface layers of the metal in a state of tensile stress. Such a residual stress would be added to the working stress, thus increasing the maximum value. This explanation was not wholly satisfactory, however, since it had been shown that the effect of increasing the mean stress of the loading cycle was quite small. Processes such as rolling, which produced a compressive residual stress, undoubtedly led to an increase in fatigue strength for most materials but modification of the properties of the surface material might also have a considerable influence.

As pointed out by Mr. Davis, the fluctuating stresses in most bolts in service were well below the limiting stress range when the bolts were properly tightened. The suggestion that an excessive overload could lead to a permanent extension of the bolts, with consequent loss of pre-stress, appeared to be quite feasible and probably accounted for failures of the particular bolts mentioned.

In reply to Mr. Dunlop's first point regarding the testing of full size components, the author wished to indicate that the use of untried materials was not recommended unless full information was available on the fatigue resistance of test pieces in the form of the finished component. Data obtained from tests on small unnotched specimens were of little use to

* Arnold, S. M. 1943. "The Effect of Screw Threads on Fatigue". "Mechanical Engineering", Vol. 65.

the designer as the reduction in strength for a given stress concentration varied for different materials. Although it was agreed that the unnotched fatigue limit of steels usually fell within the limits of 40 to 60 per cent of the ultimate tensile strength, the limiting stress range of an actual component which had stress raising discontinuities, as shown by the tests reported in the paper, could be as low as, or even lower than, 12 per cent of the U.T.S. These points were well illustrated by the results quoted by Mr. Cranston, from which it would be seen that the limiting stress range of the alloy steel bolts was only 10 per cent of the U.T.S., while the value for similar bolts in mild steel was 13 per cent of the U.T.S.

The question of the degree of pre-stressing of bolts raised by Mr. Dunlop had been referred to in the author's reply to Dr. Attia.

With regard to Mr. Dunlop's comments on the accuracy of thread cutting in commercially produced bolts, the author pointed out that the most accurate method of production was not necessarily the most costly. Improved types of screw-cutting machines were coming into more general use in the marine engineering industry, which were capable of forming bolt threads in a fraction of the time required for lathe cutting. Inaccuracies in thread cutting did not necessarily lead to a reduction in fatigue strength; indeed, as shown by the results of some of the tests, a more favourable distribution of load than that obtained by perfect threads could arise. The main point was that, with badly formed threads, the fatigue strength could not be predicted with any accuracy and might have a very low value.

There were no experimental data available on the effect of modified nut designs of the type illustrated in Fig. 6, and since their use was likely to lead to some improvement in fatigue strength only when the most accurate production methods were employed, the author could not give any hard and fast rule as to when such nuts should be used. As stated in the paper he was of the opinion that mild steel tapered-lip nuts would be most effective when used in conjunction with high tensile steel bolts.

Rolled threads were probably more accurate than several other methods of production and the author agreed that their use would be desirable. It was doubtful, however, whether thread-rolling would be an economical proposition where relatively few bolts of each size were required.

Higher strength alloy steels would appear to have little advantage over mild steel for large bolts which were subjected to cyclic loading; the author was of the opinion that accuracy of production would assume even greater importance if such materials were used.

Mr. Dunlop also raised the question of the optimum diameter for the reduced section of the bolt, which the author had referred to in his reply to Mr. Bunyan. In reply to Mr. Dunlop's final remarks, the author said that the torsional fatigue strength of large specimens was being investigated by the B.S.R.A. and the effect of temperature on the fatigue strength of various materials was receiving attention.

The author thanked Mr. Field for his further refinement of the expression for the cyclic load in a pre-tightened bolt. He doubted, however, whether the fluctuating component of stress would in practice be reduced to half that obtained by substituting $\frac{k_a}{k_a + k_b}$ for the value of C and thought that bending stresses would probably have a greater effect than the point of application of the external load. As mentioned in the paper, it was hoped to obtain experimental values for C which could be used for design purposes; these would take account of such factors as mentioned by Mr. Field.

It was agreed that the nut material should have a low yield point rather than high ductility.

In reply to Mr. Field's third point regarding the importance of bending of the thread projections, the author added that the point at which the load was applied across the width of the thread projection, which was governed by the accuracy of the mating threads, determined the magnitude of the bending stresses at the thread root section. From the results of the fatigue tests on 3-inch bolts, it was found that the part played by bending stresses varied very considerably.

Although tapering of the bolt threads was sometimes adopted, as mentioned by Mr. Field, the author maintained that such a method of improving the load distribution along the nut was hardly likely to find wide acceptance in the marine engineering industry; the use of a low yield material for the nut would undoubtedly be more simple. Tapering of the threads could only be used when a most accurate method of thread cutting was employed and, while errors in pitch of the magnitude found in the test bolts were likely to occur, there would be little point in the introduction of such a modification.

The author was interested in Mr. Fisher's suggestion as to the possibility of improving the load distribution by first loading the bolt to a suitable pre-stress and then slackening off to the test condition (pre-stress 3 tons per sq. in.). The actual pre-stress applied in practice in typical assemblies was at present being investigated and when further information was available on this point he agreed that Mr. Fisher's suggestion would be worth trying.

In reply to Mr. Fisher's question regarding the application of thread rolling methods to large bolts, the author was not aware of the use of this method in the marine engineering industry. He understood, however, that a machine had recently been developed which was capable of forming threads by rolling up to a pitch of 4 t.p.i. and up to 4 inches diameter.

The term "limiting stress range" was used in the paper to indicate the full stress range from the constant minimum value (which was constant at 3 tons per sq. in. for the bolt tests) to the peak value of the loading cycle, except when referring to the results of the tests on the ½-inch diameter unnotched specimens. These results had been distinguished by the insertion of $\pm$ before the numerical values.

The author thanked Dr. Richards for his detailed comments on the accuracy of thread cutting that would be necessary to obtain the best results from nuts of the tapered-lip type. He agreed that such accuracy was hardly attainable by normal production methods but if such design modifications were coupled with the use of material having a relatively low yield point for the nut, he considered that the load distribution along the nut could be improved considerably. He was not aware of any fatigue tests on nuts of modified form other than those carried out by Wiegand (referred to in the reply to Mr. Cranston).

The author could not agree entirely with Dr. Richards's suggestion that variations in the surface finish at the roots of the threads could have had a considerable influence on the scatter of the results, although he was of the opinion that isolated machining marks could lead to a considerable reduction in fatigue strength. This conclusion was based on a careful examination of the fractured specimens, which revealed that there was no variation in the position of crack initiation, which would be expected if surface irregularities had had a significant effect.

The use of more than one bolt of a smaller diameter in place of a large bolt which might be difficult to tighten properly offered interesting possibilities. Considering further the example mentioned by Dr. Richards, it would appear that the substitution of two 1½-inch bolts in place of one of 3 inches diameter would not reduce the strength of the assembly appreciably, although the cross sectional area of the two bolts would be only half that of the one large bolt, owing to the reduction of the fluctuating load component arising from the increased elasticity of the two smaller bolts. This would only apply, of course, provided the pre-stress was of sufficient magnitude to prevent parting-line separation.

The diagram given by Mr. Threlfall (Fig. 22) clearly showed the distribution of the external load applied to a pre-tightened bolted assembly. It would be seen from the expression given by Mr. Threlfall for the maximum external load which could be applied before abutment separation that, if K

(the ratio k_b/k_a) had a value of 4 or more, this maximum external load would be only little more than the tightening load. It was for this reason that the author suggested that the tightening load should have a value of at least twice the applied working load.

The reduction in stress concentration arising from the use of tapered threads had been investigated by Hetényi and the results were given in the paper referred to in the footnote on p. 238. The photo-elastic model used by Hetényi had tapered threads equivalent to a tapering of the nut of 0·003 inch per inch of diameter and per inch length of the nut in a steel threaded connexion. This design reduced the stress concentration to 3·1, compared with 3·0 for the tapered-lip nut. The author was not aware of any tests with nuts and bolts of differential pitch.

With regard to Mr. Threlfall's last point, the question of operating stresses had been mentioned in the author's reply to Dr. Attia's contribution to the discussion.

INSTITUTE ACTIVITIES

Minutes of Proceedings of the Ordinary Meeting held at the Institute on Tuesday, 14th October 1952

An Ordinary Meeting was held at the Institute on Tuesday, 14th October 1952, at 5.30 p.m. Mr. S. Hogg (Vice-Chairman of Council) was in the Chair. A paper by Mr. B. Taylor, B.Sc.(Eng.), A.M.I.Mech.E. (Member), entitled "The Strength of Large Bolts Subjected to Cyclic Loading", was presented and discussed. Thirty-eight members and visitors were present and seven speakers took part in the discussion.

A vote of thanks to the author proposed by the Chairman was accorded with acclamation. The meeting ended at 7.40 p.m.

Local Sections

Cardiff

Annual General Meeting

The annual general meeting of the Cardiff and District Section was held on Friday, 24th October 1952. The following officers and committee members were elected for the ensuing year:

Committee: Messrs. J. E. Church (Chairman), J. H. Evans (Vice-Chairman), T. G. Thomas, F. R. Dale, G. K. Beard, S. Sedgwick, H. G. Wickett, H. J. Hobart, H. S. W. Jones, I. J. Thomas, R. H. Rees and J. M. Morton. Mr. W. Patton was re-elected as Honorary Secretary, Mr. W. Gracey as Honorary Assistant Secretary and Mr. G. Thomas as Treasurer.

Lecture on "Steam Generators"

The first lecture of the 1952-53 session was given at the South Wales Institute of Engineers, Park Place, Cardiff, on Monday, 10th November 1952, before a large and enthusiastic assembly. The lecturer was Mr. H. F. Reeman, M.I.Mech.E., A.I.E.E., and his subject was "Steam Generators". He described the various types of boilers that had been in use for the generation of steam, starting with the rotary boiler used by the Egyptians, progressing to the types in use in the nineteenth century, and ending with a description of the modern watertube boiler at present in service in both the British and American navies. Lantern slides were used to illustrate the lecture, which was completed by the showing of a sound film on the welding methods employed in boiler manufacture. The film illustrated the development of welding, beginning with the early form of pressure welding employed by the blacksmith in his hammer welding. The modern processes of gas flame fusion welding by hand were shown, as also was the method of electric fusion welding by machine. The film dealt particularly with the use of these processes in the fabrication of high pressure boiler tubes with wall thickness up to six inches. The flash welding of boiler tubes, and of studs in curved boiler tubes, by automatic machines was then demonstrated.

Afterwards the audience had ample time for asking questions, which the lecturer answered in a lucid and masterly manner. A vote of thanks to Mr. Reeman was proposed by Mr. G. Wickett (Member) and heartily endorsed, and Mr. D. Skae (Member) voiced the thanks of those present to the Chairman, Mr. J. E. Church (Member of Council) for the able manner in which he had conducted the meeting.

Fourth Annual Dinner

The fourth annual dinner of the Section was held at the Royal Hotel, Cardiff, on Thursday, 13th November 1952, there being 189 members and guests present. Mr. J. E. Church (Chairman of the Section) presided and the principal guests were Lord Howard de Walden (President of the Institute) and the Deputy Lord Mayor of Cardiff, Councillor H. E. Edmonds, J.P.

In proposing the toast of the City and Port of Cardiff, Mr. A. E. H. Brown, the chief docks manager, said that Wales and Cardiff had every right to be proud of the City and the way it had developed. The port and City of Cardiff were indivisible and if the port did not flourish, neither would the city. The post-war years had been difficult owing to the definite turn over to oil fuel, but there were still many markets open to coal, and both Canada and America wanted much more Welsh anthracite coal.

Mr. B. I. Llewellyn, Chairman of the Cardiff and Bristol Channel Incorporated Shipowners' Association, in toasting the shipping industry, expressed his concern over the position of the smaller shipping companies, who, he said, were being forced out of existence by the present high level of taxation and increasing costs.

Mr. R. G. M. Street, managing director of the South American Saint Line, in a stimulating response, spoke of the increase in the number of ships which had visited the port this year, at a time when shipping in other ports in the United Kingdom was decreasing.

Mr. T. G. Thomas (Member) had been responsible for the organization of the dinner and the large number of members and guests who attended was evidence of its growing popularity. Local members of the Institute fully appreciated the presence at the function of Lord Howard de Walden, Mr. Stewart Hogg (Vice-Chairman of Council), and Mr. J. Stuart Robinson, M.A. (Secretary), as their presence showed that the Institute was keenly interested in the welfare of the Local Sections.

Swansea

A meeting was held at the Central Library, Alexandra Road, Swansea, on Friday, 17th October 1952, at 7.0 p.m., when Mr. G. R. Goldsworthy gave an illustrated lecture on "The History of the Tube Industry". The Chair was taken by Mr. F. W. King (Member) and fifty-two members and visitors were present, including a number of students from Swansea Technical College who had recently commenced the training course for engineer officers in the Merchant Navy.

Captain(E) C. T. Phillips (Member) proposed a vote of thanks to the author for his entertaining and instructive lecture; this was seconded by Mr. G. T. MacDonald (Member) and warmly acclaimed. The meeting ended at 9.30 p.m.

Sydney

Lecture on "Phenolic Laminates, Bearings and General Applications"

The Sydney Local Section held a meeting at Science House, Gloucester Street, Sydney, on Friday, 31st October 1952, at 8 p.m. Fifty-one members and guests attended and Mr. H. A. Garnett, the Local Vice-President, was in the Chair. Mr. W. C. Steanes delivered a lecture, well illustrated by lantern slides, on "Phenolic Laminates, Bearings and General Applications". A spirited discussion followed, when the large number of questions which were asked were answered effectively by Mr. Steanes; Messrs. Williams, Waddell, Flaherty, Buls, Butcher, Munro, McLachlan, Gale, Pollock, Porteous, Franklin and Hutcheson contributed to the discussion.

A vote of thanks was proposed by Mr. D. N. Findlay, seconded by Mr. Buls, and carried by acclamation.

Annual Dinner

The annual dinner of the Section was held at the Carlton Hotel, Sydney, on Thursday, 20th November 1952. There was an attendance of seventy-five, comprising forty-one members and thirty-four guests. The official guests included: Mr. Justice A. V. Maxwell of the New South Wales Supreme Court; Captain(E) E. A. Good, engineer manager, Garden Island Dockyard; Mr. V. J. F. Brain, Past President of the Institution of Engineers, Australia; Mr. D. Lyon McLarty, Director of the State Dockyard, Newcastle; Mr. A. Denning, Director of the University of Technology of New South Wales; and Mr. W. C. Steanes, who presented a paper to the Section in October. Mr. H. A. Garnett (Local Vice-President) presided at the dinner.

After the Loyal Toast, the toast of the Institute of Marine Engineers was proposed by Mr. Justice A. V. Maxwell in a most interesting and thoughtful address; Mr. H. P. Weymouth (Member) replied in excellent style. The toast of "Our Guests" was proposed by Com'r(E) F. W. Purves, R.A.N. (Member); Mr. J. C. L. Wong, superintendent engineer of the Blue Funnel Line in Hong Kong, not only replied most ably on behalf of the visitors, but gave a very interesting talk on conditions in Hong Kong and in China generally.

The dinner, which had done a great deal to bring the local members together, was again a most successful function and had come to be one of the more important events in the year for the Section.

Junior Section

Barrow-in-Furness

On Friday, 28th November 1952, a Junior Section meeting was held at the Technical College, Barrow-in-Furness, when Mr. J. C. R. Mathieson presented a paper entitled "Gas Turbines". Mr. Sandham, principal of the college, was in the Chair and there were present about 110 members, students, and members of the Barrow Association of Engineers. The paper, which was illustrated by lantern slides, covered the development of the gas turbine from the simple jet engine down to present day practice and showed at all stages how this compared, for efficiency and basic and overall fuel consumptions, with similar powered Diesel and steam turbine plants.

In view of the marine engineering interests of his audience, Mr. Mathieson went to some trouble to show how the gas turbine could be applied to ship propulsion and to marine auxiliaries, while indicating that it would be some little time before gas turbines became a serious rival to conventional prime movers for ship work.

At the end of the paper, which lasted seventy-five minutes, a lively discussion was opened by Mr. George Wood (Member) and this covered most of the lecturer's material and a few "red herrings" as well. The discussion lasted an hour before a vote of thanks to Mr. Mathieson was moved by Mr. C. H. Verity (Member) and was passed with acclamation.

Falmouth

On Tuesday, 11th November 1952, at 7.30 p.m., Mr. R. R. Strachan (Member) presented his lecture, "Refrigeration at Sea", to a good audience of students and members at the Falmouth Technical Institute; the meeting had been arranged with the co-operation of the Cornwall Education Authority and the principal of the Institute, Captain Kelly, was in the Chair.

The lecture, with the accompanying lantern slides, was a particularly good choice for presentation in Falmouth, as much of the work in the port is connected with refrigerated vessels, and a strong interest in the subject was evinced in the lively discussion which followed.

A vote of thanks to the author was proposed by Mr. P. Ewing (Member) seconded by Mr. R. Pritchard (Member), and carried enthusiastically.

Hull

A junior lecture entitled "Gas Turbines" was given by Mr. J. C. Barr at the Hull Municipal Technical College on Thursday, 13th November 1952, at 7 p.m. The lecture room was packed to capacity with 170 members, visitors and students and a large number of students who wished to attend had to be turned away. Mr. G. H. M. Hutchinson, chairman of the Local Section, was in the Chair.

Mr. Barr's most interesting and lucid lecture was followed by a very lively discussion which had to end before many of those who still had questions to ask were able to do so. A vote of thanks to the author was proposed and seconded respectively by Messrs. Hemingway and Wickenden, students in the full-time day engineering degree course at the college.

After the lecture, Mr. F. C. M. Heath (Local Vice-President) presented to Mr. K. Aldred the prize awarded by the Institute (session 1951-52) for the best year's work in heat engines by a student of the college.

Southend-on-Sea

On the evening of Wednesday, 19th November 1952, at the Municipal College, Southend-on-Sea, Mr. J. Hodge gave a lecture entitled "Gas Turbines". The Vice-Principal, Mr. B. Thomsett, B.Sc., was in the Chair and the audience, which consisted of students from the college, numbered over one hundred.

The lecture was in two sections, the first dealing with the theory and types of gas turbine plant, while the latter half dealt with its application, giving special reference to marine work. The slides which were used to illustrate various salient points were very clear and helpful. The nature of the questions that were asked during question time gave proof that Mr. Hodge had given a very interesting lecture.

Mr. I. S. B. Wilson (Member), who represented the Council, spoke for a short time on the advantages to be gained by students becoming junior members of the Institute.

A vote of thanks to Mr. Hodge, proposed by the Vice-Principal, completed the evening.

Membership Elections

Elected 8th December 1952

MEMBERS

Maurice Joseph Mackie Birrell
George Clare
David Hunter Rae Falconer
William Robert Foley
Edward John Kenneth Gibbs
W. T. A. Jordan, C.B.E., Engr. Capt., R.N. (ret.)
Hunter Thomson McMichael
Robert McVie, B.Sc.
Philip Reginald Marrack, Lt.-Com'r(E), R.N.
John Herbert Martin

Robert John Ramsay
Henry Robb
John Wilson
William Simpson Wrangham

ASSOCIATE MEMBERS
Francis James Dayton, Sub. Lieut.(E), R.C.N.
Anthony Farrow
James George Macdonald

ASSOCIATES
Clarence Athol Bell
Ernest Brady
George Lionel Budd, C.E.R.A., R.N.
James Richard Corless, Lieut.(E), R.N.
Herbert John Dean
William Ralph Woodleigh Dyne
George Fairweather
Aloysius Leonard Stanislaus Fernando
Brian Hall, Lieut.(E), R.N.
Alexander Hendry
Hugh Lowry Irwin
Albert John Kingston
Alexander Mackenzie
Frank William Moir
William Singleton Morrow
Thomas Joachim Mulhearn
Kenneth Gordon Neaves
Edward Dering Neville
James Pendlebury
William Alfred Price
Charles John Probett
Imtiaz Muhammad Qureshi, Sub. Lieut.(E), R.P.N.
Alfred Emslie Riach
Jack Russell
William Schofield
James Dewar Smith
Raymond Towell, C.E.R.A., R.N.
Gordon Owen Webb

GRADUATES
John Robert Webber
Albert Ernest Walker

STUDENTS
Michael Treloar Best
Anthony Charles Harrison
Shankar Kotiedath Menon
Jonathan Richard Michaelis
Joseph Gordon Poole
Philip Simpson Pratt
Ian Alastair Ramsay
Ronald Ritson

PROBATIONER STUDENTS
David Marshal Anstiss
Philip John Ash
William Christopher Baldry
Michael Barber
Brian John Barnard
Derek Franklyn Betts
Peter John Bray
Terence Byrne
Christopher James Capus
Frank Clark
Gordon Greaves Common
Frederick Charles Cousins
Colin Francis Crandon
Terence D'Alton
Michael Ellis
David Edwin Fothergill
John Leslie Frost
Peter Edward Grant
Roger Harrison
Arthur Leslie Hawkins
Jackson Cumming Hiddleston
Brian Illingworth
Brian Jameson
Keith Francis Jones
Keith Hughes Jones
Brian John Kelsall
Peter Mills Livesey
Peter Mervyn John Llewellyn
Benjamin Gareth Llewelyn
Guy Grant Macpherson
Ronald McDonald
Alan Miles
Robert Charles Morgan
Neil Denton Nimmo
John Kenneth Pointon
Kevin Francis Ponsford
Michael David Retter
David Huw Rogers
Derek Ian Rowan
Richard John Scully
David Hugh Shallis
Richard John Shepheard
John Harold Smart
Robert Edward Smith
Robert Horton Streather
Heddwyn Thomas
Thomas Mortimore Warren
Clive Anthony Wiles
David John Williams

TRANSFER FROM ASSOCIATE TO MEMBER
Harry Burnside Cleworth
Frederick Edward Lawrence

TRANSFER FROM ASSOCIATE TO ASSOCIATE MEMBER
Albert Benjamin Richard Sexstone
James Frederick Stephen, B.A.

TRANSFER FROM GRADUATE TO ASSOCIATE MEMBER
Henry Nicol

TRANSFER FROM STUDENT TO ASSOCIATE MEMBER
Jeffery Francis Webb, Lieut.(E), R.N.

TRANSFER FROM STUDENT TO GRADUATE
Henry Lyth

TRANSFER FROM PROBATIONER STUDENT TO STUDENT
Rex Young

OBITUARY

BERT BATEMAN (Member 7271) was born in 1904. He served an apprenticeship with J. P. Knight and Son, Rochester, from 1920-25, and then spent short periods consecutively with Besant and Company, of Cuxton, Kent, International Combustion, Ltd., and Humphreys and Glasgow, Ltd. In 1927 he joined Stephenson Clarke, Ltd., as a junior engineer, with whom he remained until 1947, serving for some years as chief engineer. During the war the ship in which he was serving was hit by a bomb which burst in the stokehold; Mr. Bateman, who was chief engineer, entered the stokehold through scalding steam to shut off the valves. Later, he rescued five injured men from below. For the great courage and determination he showed on this occasion, he was awarded an M.B.E. and Lloyd's Medal. Mr. Bateman was elected an Associate Member of the Institute in 1933 and transferred to full membership in 1934. He died on 9th October 1952.

WILFRID BAYLISS (Member 4107) was born in Portsmouth in 1882. When his family moved to Pontypridd, in South Wales, he started an apprenticeship there with Brown, Lennox and Co., Ltd., and by the time he was twenty-five he had been appointed to a managerial position with John Abbott, Ltd., of Gateshead. In 1912 he became manager of Joseph Wright and Co., Ltd., of Tipton, and during the first World War he was commended by the Admiralty for introducing new types of mooring equipment which proved invaluable to the fleet. Later, Mr. Bayliss became managing director of the company and also a director of several chain and anchor manufacturing firms in the Cradley Heath area. He retired from business through ill health in 1928 but he continued to write articles for engineering and technical journals. He died on 6th October 1952. Mr. Bayliss was elected to membership of the Institute in 1920.

HENRY REGINALD FLOWER (Member 6428) was born in 1889. He served an apprenticeship at Portsmouth Dockyard from 1905-11; from 1912-14 he was employed on Spanish naval construction at Ferrol, Spain. He was Admiralty chargeman in the engineering department of Portsmouth Dockyard for the next two years and from 1916-23 he was Admiralty engineer overseer to Earle's Shipbuilding and Engineering Co., Ltd., and the Humber Graving Dock Co., Ltd., of Hull. Until 1926 he was mechanical engineer to the colonial government in Hongkong and then, for a further year, he was managing engineer (director) with the Radio Communication Company of Hongkong. From 1927-29 Mr. Flower was superintendent engineer of the Dorado Railway Co., Ltd. (Ropeway Branch) in Colombia, South America, and in November 1929 he was appointed mechanical engineer to the La Guaira and Caraças Railway Co., Ltd., Venezuela; he returned to England in 1935 but shortly afterwards went back to Colombia as mechanical engineer to the Frontino Gold Mines, for whom he planned and constructed a mechanical hopper which proved very successful in use. In April 1937, however, Mr. Flower was invited by the Venezuelan government to become the consulting naval engineer to the Escuela Naval, Maiquetia, Caraças. He accepted, and was given the rank of Capitan de Corbetta, Escuela Naval; he was responsible for the preparation of engineer officers and cadets attached to the school. In this capacity, he personally prepared several training manuals written in Spanish, dealing with seamanship and naval engineering. In 1938 he was appointed director of the school, responsible only to the Director of Marine and Minister for War.

When war broke out in 1939, Mr. Flower decided to return to England, eventually settling in Farnborough, where he at once offered his services and was appointed Incidents Officer for the district. When the war ended, he retired, although he continued to act as advisor to certain firms with whom he had maintained contact throughout his career. Ill health finally compelled him to seek a warmer climate and he died in the South of France on 8th June 1952.

Mr. Flower was a member of the American Society of Mechanical Engineers and was elected to membership of the Institute in 1930.

GEORGE HENRY LEWIS OWEN (Member 5128), who died at Gillingham, Kent, on the 19th November 1952, at the age of sixty-six years, served an engineering apprenticeship in H.M. Dockyard, Chatham, whence he entered the service of the British India Steam Navigation Company, rising in the course of thirty-five years from junior engineer to be the Company's youngest chief engineer, and ultimately commodore chief engineer at the time of his retirement in 1942. He was associated with the building and installation of the engines for the Company's first motor ship, the *Domala,* and sailed in her as chief engineer. He saw service over most of the globe, and during World War II, after completing his sea service with the B.I.S.N. Company, he was superintendent engineer for the Ministry of War Transport to the Government of India at Calcutta. Later, he held an appointment as engineer in charge of ground plant under the Air Ministry at Cranwell until shortly after the end of hostilities. He was elected a Member of the Institute in 1924, and after his retirement he took an active part in the work of the Institute, both as a member of the Education Group Committee and the Committee of the Guild of Benevolence. For several years he acted as a scrutineer at the annual general meetings. He was a governor of the Medway Technical College and was a prominent working member of the Gillingham Conservative Association. He is survived by his widow, they having lost their only daughter a few years ago.

NORMAN HENRY REED (Member 3203) died, aged sixty-two, on 22nd November 1952. He served his apprenticeship with Palmers Shipbuilding and Iron Company at Jarrow, where his uncle, the late J. W. Reed, was manager, and gained drawing office experience in their shipyard at Alloa. He then went to sea and spent a year in the Far East before joining the Donaldson Line as sixth engineer. As soon as the war broke out in 1914 he enlisted as a trooper in the 12th Royal Lancers and was commissioned early in 1915 in the South Lancashire Regiment; he served with them throughout the war, in the Battle of the Somme in 1916-17 and Salonika in 1918, until he was demobilized in November 1918. In 1919 he joined his father, T. A. Reed, in his business as consulting engineer in Cardiff, and remained with him as assistant until he set up his own business as consulting engineer and surveyor in London in 1931. Despite his age, Mr. Reed served voluntarily for about a year during the second World War as a gunner with the Royal Artillery (A.A.) and then was employed by the Ministry of War Transport until 1944, when his health broke down and his London house was bombed. His only son, a bomb aimer in the Royal Air Force, was killed in June 1944 while on operation over oil refineries at Budapest.

Mr. Reed was elected to membership of the Institute in 1916; he had been a Liveryman of the Worshipful Company of Shipwrights since 1931 and a member of the Institution of Naval Architects from 1927-42.

JOHN MARSHALL SCHOLEFIELD (Member 12628), aged 38, was killed in the Harrow train disaster on 8th October 1952. He was a London superintendent engineer of Furness Withy and Co., Ltd., and was travelling to Liverpool to attend s.s. *Brazilian Prince.* Mr. Scholefield served an apprenticeship from 1929-34 with John Dickinson and Sons, Ltd., of Sunderland, and then spent six-and-a-half years at sea in vessels owned by Rowland and Marwood, the Port Line, the Canadian Pacific S.S. Company, and Stephenson and Clarke, Ltd. Then from 1943-49 he served with the Prince Line, Ltd., joining as junior third and rising in 1946 to chief engineer. He obtained a First Class Steam M.o.T. Certificate with Motor endorsement. He was elected a Member in 1949.

ALBERT EDWARD YEADON (Member 6999) was born in 1901 and served an apprenticeship with the Manchester Ship Canal Company. He spent nine years at sea, from 1922-31, and obtained a First Class B.o.T. Steam Certificate with a motor endorsement. He returned to the service of the Manchester Ship Canal Company as a chief engineer in 1932. He remained with the company for a number of years but in 1940 he was appointed a section engineer with Sir Alexander Gibb and Partners and from 1948 he was employed in a civilian capacity by the B.A.O.R. Mr. Yeadon died in Cologne on 1st August 1952 after a short and painful illness. He had been a Member of the Institute since 1943.

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