Full-scale Fatigue Tests of Diesel Engine Elements

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The paper deals with a series of experiments undertaken by Burmeister and Wain and by various Danish testing laboratories on their behalf.

In the opening paragraphs a comparison is made between laboratory tests and service experience, experiments are mentioned with small rotary bend tests, with nitriding which seems to be able to treble the fatigue strength of a notched bar, with cast iron, as well as a series of experiments which indicated that the fatigue strength does not continue to fall when a notch is made sharper.

In addition, the paper deals with experiments with springs, the fatigue strength of which has been greatly increased by shot peening, and also experiments with the fashioning of crank throws and with piston rods, the strength of which was more than doubled by a series of alterations in design, material, machining and tightening. The paper also refers to experiments with piston rods shrunk into the crossheads by oil pressure. Finally, fatigue tests with a large welded construction, loaded in bending and tested as welded and after annealing, are dealt with.

INTRODUCTION

In this paper a series of experiments undertaken during the past fifteen years by the author's company, partly carried out by themselves and partly undertaken on their behalf by various Danish testing laboratories to determine the fatigue strength of Diesel engine parts to full scale or at least to a comparatively large scale, will be presented and commented upon. The details in question (springs, crankshafts, threaded connexions and welded units) are, however, elements in most engine plants and the problems in a broader sense are common for all machinery and building elements subjected to varying stresses.

On the subject of fatigue strength one is inclined to think in terms of the Wöhler curves with their breaks (or bends) at 5 or 10 million stress reversals and obtained on comparatively small specimens designed to investigate the fatigue strength of different materials or different heat or surface treatments under well defined conditions. That it is not always easy to see the correlation between such tests and experience from practice is of course because test conditions deviate essentially from actual conditions, which, moreover, might change with time.

Among the causes of such deviation are the difficulties in estimating exactly the magnitude of the varying forces which have acted on an element and which are influenced by accuracy in manufacture, deformation under stress, and service conditions such as engine load and gas pressure, engine speed and possible speed surges, temperature differences (for instance with load changes) and of upkeep and maintenance. Conditions may further be complicated in practice by the fact that the stress on many engine parts is altered in magnitude due to wear, for instance, of bearings or contact surfaces or to a permanent set (threads, springs, etc.). Corrosion might also produce a steady reduction in fatigue strength, at least up to a couple of hundred million or so stress reversals; a similar condition is encountered in the case of aluminium alloys and various other metals, even without corrosion.

ROTARY BENDING TESTS

Before dealing with the full scale tests of engine parts, brief mention can be made of a series of fatigue tests in rotary bending. The Schenck-type testing machine of the Royal Danish Navy Laboratory was mainly used. This gives the test bar a constant moment over its whole length. As a rule, this machine can take only test bars up to 7·5 mm. diameter ($\frac{5}{16}$ inch) and only for bending tests. These small bars probably give more favourable results than could be expected from large test pieces and large engine parts. For a comparison of the influence of materials, surface treatment and machining methods, these tests, judiciously used, have proved their worth and a considerable part of the test results published today originate from such small rotary tests. Furthermore, it must be borne in mind that the fatigue strength in rotary bending is considerably higher than in compression-tension and in torsion.

Fig. 1 shows the results of a number of small rotary bending tests on different materials. Material No. 6 is nitriding steel, heat treated, but not nitrided; its fatigue strength corresponds fairly well to other materials with the same ultimate strength. By the nitriding process (No. 7) the fatigue strength is increased in the case of a smooth bar by 40 per cent, which in itself is noteworthy. But the strangest thing is that fatigue stress for the two forms of notching is not lower than that of the nitrided plain specimens. In other words, for the sharply notched bar the fatigue strength is nearly trebled by the nitriding(1).

As is well known, nitriding involves the heating of the material to approximately 520 deg. C. (970 deg. F.) in ammonia gas in which it remains for a certain time. Nitrides

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	Analysis										Bending fatigue strength (10×10^6)			
							Tensile strength	Elongation L = 5D	Impact strength, Charpy	Bending strength	Smooth, polished bar (D = 7·5mm)	Bar with collar R = 3mm	Bar with notch	Very roughly turned
No.	C	Cr	Ni	Mo	Al		kg/mm^2	%	kgm/cm^2	kg/mm^2	kg/mm^2	kg/mm^2	kg/mm^2	kg/mm^2
1	0·20					Forged	51	28	10		±24	±23	±13	
2	0·45					Heat treated	73	22	6·7		±35	±34		
3	0·22	0·8		0·3		Heat treated	60	24	4·8		±25	±22		
4	0·37	1·1	0·2	0·2		Heat treated	90	18			±42	±35	±21	
5	0·13		5			Heat treated	79	22	10		±43	±39	±22	
6	0·3	1		0·3	1	Heat treated	80	20			±42	±39	±19	
7	0·3	1		0·3	1	Heat treated and nitrated					±59	±59	±59	
8	0·6	Spring-steel				Heat treated	140	7			±45			±30
9	0·6	Spring-steel				Heat treated and shot peened					±65			±58
10	3·05	Pearlitic cast iron					25			49	±10·5	±11	±11	
11	3·8	Good ordinary cast iron					20			40	±10	±9·5	±11	

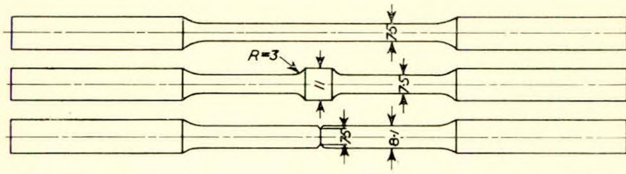

FIG. 1—*Experiments with small, rotating beam bend tests*

In the majority of these tests the sharpness of the notch was not checked microscopically, so final conclusions cannot be drawn as to the notch sensitivity of the materials (compare with Fig. 3)

are thereby formed in the surface of the bar and the favourable effect is probably due to these nitrides taking up a considerably larger volume than the amount of iron consumed in their formation, this resulting in powerful compression stresses in the surface of the material, i.e. at exactly those points where the greatest tensile stresses from the outside influence are concentrated. Thus, nitriding may be of benefit not only by forming a hard, wear-resistant surface, but also by increasing fatigue strength. Therefore, engine parts such as smaller crankshafts are sometimes nitrided over their entire surface.

Fractures presumably start just under the nitrided surface layer where the varying stresses are added to the tensile and shear stresses brought about by the nitriding; and as the effect of the notch does not extend very deeply into the material, the result consequently becomes the same in the notched as well as the smooth test bars.

SPRINGS

The same effect may, however, be attained by mechanical means. Materials Nos. 8 and 9 (Fig. 1) are spring steel. In the case of springs, it has sometimes been experienced that during the first months after the plant has been put into service a certain number of fractures occur; but when the broken springs have been replaced by new ones, these and the remaining springs will last for many years. In other words, an elimination of the inferior springs takes place. It is quite reasonable that such a hard material as spring steel is very sensitive to notches, and at the same time exposed to surface damage by winding and heating.

The most important factor about spring material, therefore, is probably not the analysis, but simply that the surface is faultless, that decarburization, scale and, above all, cracks, are avoided, and that hardness is adequate. In the case of larger springs, however, surface polishing is difficult; and magnetic examination of all the springs is not only an expensive and difficult job to be undertaken only in exceptional circumstances but also a purely negative measure, the best possible result of which can only be the discarding of all unsuitable springs.

Tests with shotpeening have therefore been undertaken. Spring steel, No. 8 (Fig. 1) was tested in rotary bending, in the smooth-turned condition, and also in the rough-turned condition; the turning tool was permitted intentionally to vibrate during the machining process so that cracks and fissures appeared in the surface. This was an attempt to duplicate the worst possible surface conditions, and it reduced the fatigue strength to two-thirds. The surfaces of a number of bars were then shot-peened, i.e. subjected to a bombardment by steel balls of 0·8 mm. (1/32 inch) diameter. This bombardment continued for fifteen minutes in a revolving drum so that the bars were bombarded from all sides(2). In this way it was seen that fatigue strength of the smooth-turned bars increased from ±45 to ±65 kg/mm.^{2} (64,000 to 93,000lb. per sq. in.) while that of the rough-turned bars increased from ±30 to ±58 kg/mm.^{2} (43,000 to 83,000lb. per sq. in.). In this way compression stresses in the surface were introduced by mechanical means. In the microphotograph (Fig. 2) can be seen a section of the surface of such a spring wire before and after shot peening. It can be seen that the line structure to a depth of 0·1 mm. (0·004 inch) is unmistakably affected by the blow on the surface, and incipient cracks become "closed by the compression" with the result that tension stresses can pass unimpeded. On examining the surface of a shot-peened spring broken during a test, a crust of a few tenths of a millimetre can usually be seen—almost like a case.

Next, a series of tests were carried out with actual springs (wire diameter 15 mm. = 19/32 inch, coil diameter 85mm. = 3⅜ inch) in a simple testing machine in the firm's research department. Results showed it to be possible to increase the calculated torsional stress in the spring for fracture at 10 million cycles from 0–40 kg per mm.^{2} (57,000lb. per sq. in.) or 20 ± 20 kg per mm.^{2} for ordinary spring wire to 0–60 kg per mm.^{2} (85,000lb. per sq. in.) or 30 ± 30 kg per mm.^{2} for correctly

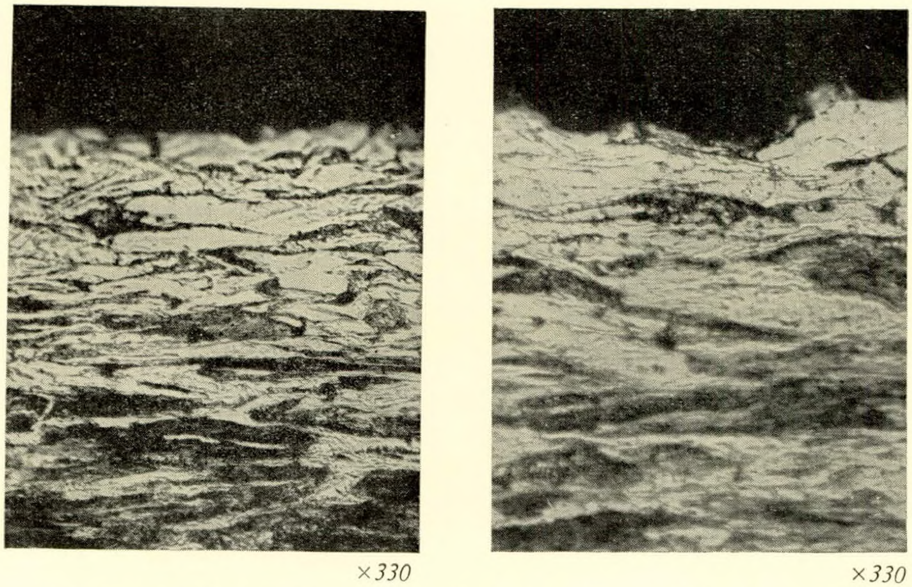

FIG. 2—*Microphotographs of spring wire before (left) and after shot-peening*

shotpeened springs. "Wet-peened" springs from another supplier were also tested. Their appearance was satisfactory but their fatigue strength was just the same as that of non-peened springs: 0–40 kg per mm.[2] (57,000lb. per sq. in.).

Springs of patent-hardened ("piano") wire look so well on the surface that one hardly has the heart to do anything to them; but tests show that shot peening gives an improvement very similar to that in the case of black springs.

Shot peening gave such good results that during the past four years all springs have been shot peened after heat treatment and this has resulted in a very noteworthy reduction in the number of spring fractures—although fractures due to extensive faults in the material cannot, of course, be prevented.

CAST IRON

Fig. 1 also shows the results of rotary bend tests with cast iron specimens.

Material No. 10 is pearlitic cast iron, while No. 11 is ordinary grey cast iron which, however, does not seem to be essentially inferior in fatigue strength.

Oddly enough, the notched bars appear to have a greater fatigue strength than the smooth polished bars. This phenomenon is not unusual for cast iron and is due to the fact that this material is in itself so abundant in graphite flakes, which form sharp notches; on these test bars, which have the same moment over their entire length, it is impossible to avoid having some of these notches in the surface of the smooth bar—hence the comparatively low fatigue strength of smooth bars (particularly when compared with the high static bending strength of cast iron). On the other hand there is little probability that such a notch will be located exactly in the bottom of the turned-in notch, and it also appears that several of the notched bars break outside the notch where the diameter is 8·1 mm., while the diameter is 7·5 mm. at the bottom of the notch; and it is according to the latter diameter that fatigue strength is calculated.

From this it does not follow, of course, that the strength of cast iron structures is not dependent on their shape. In practice it is nearly always the case that fatigue fractures in cast iron start from sharp corners, as with steel; generally speaking, the flux of forces is the same for both cast iron and steel and in actual structures fillets and sharp edges cover such great areas that it must be taken into account that somewhere on the rounding a notch from a graphite flake may add its effects to the stress concentration due to the form itself.

THE SHARPNESS OF THE NOTCH

The series of tests mentioned were primarily aimed at examining and comparing materials and surface treatments, but a series of tests was also made on the bend testing machine with varying notching alone. For these experiments steel with 0·20 per cent C was used. The actual fillet radius at the bottom of the notch was carefully measured by means of a thread-microscope, and in Fig. 3 the ascertained fatigue strength is plotted as a function of this fillet radius which is

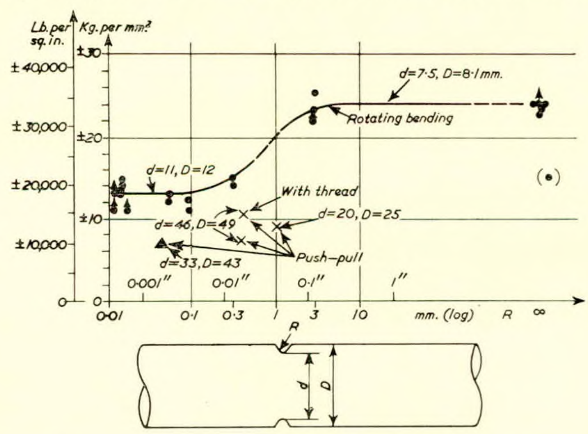

FIG. 3—*Fatigue strength at* 5×10^6 *cycles*

Each of the round marks indicates a single rod, the endurance limit of which is referred to 5×10^6 cycles by means of a Wöhler curve. The upper part of the curve corresponds to 7·5 mm. (0·3 inch) rods; if the corresponding tests were made with 11 mm. (0·43 inch) rods, this part of the curve would probably turn out somewhat lower, but the character of the curve would undoubtedly be unaltered. Each cross indicates the result of a whole Wöhler curve and the same applies to the mark ▲ which corresponds to the British experiments mentioned in the text

plotted as abscissa in logarithmic scale. The points marked with arrows pointing upwards correspond to bars which were not broken after the 5 million cycles which in this case was considered the limit. Thus there is a marked decrease in the fatigue strength when the fillet radius is reduced from 3 to

0·3 mm. and an additional smaller decrease when reducing from 0·3 to 0·1 mm.; but after this point it does not seem to impair the material if the notch is made sharper.

Of course there is some uncertainty in these determinations. It is possible that the turning tool to a varying degree may roll the surface at the bottom of the notch; but in reality the measurements indicate that the worst notch effect occurs with a 0·1 mm. notch and that an even sharper notch gives greater strength. Although this is not easily understandable, one is tempted to believe that there is something in this tendency; during and shortly after the war a number of mishaps with piston rods occurred on double-acting engines. How these mishaps were remedied will be discussed more fully later. To begin with the trouble was counteracted by magneto flux-testing the rods at intervals of approximately one year (Fig. 4). In this way *incipient cracks* were very often found, but usually they were so small that they disappeared on being lightly machined after which it was possible to permit the

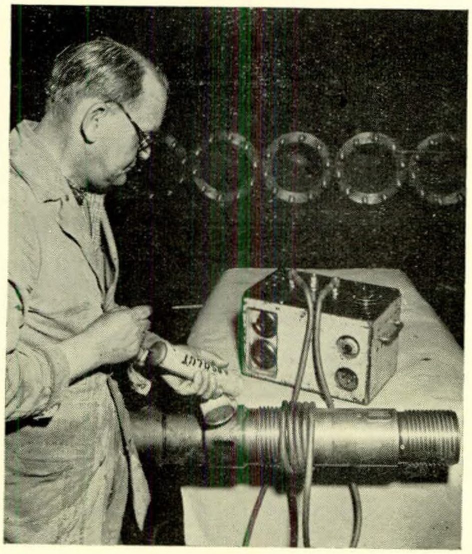

FIG. 4—*Magnetoflux testing of piston rod for double acting engine*

continued use of the rods. It never happened that a piston rod broke which was subjected to annual overhaul in this way. Consequently, the fractures must develop rather slowly, and this is in itself remarkable as an incipient crack must be regarded as a very sharp notch. An article in "Engineering"(3) discussed an interesting series of tests which showed that at the bottom of a sharp groove (fillet radius 0·05 mm. = 0·002 inch, diameter 33 mm. = 1·3 inch) at a stress of only ±3 kg per mm.^{2} (1·7ts. per sq. in.) quite small cracks, approximately 0·05 mm. (0·002 inch) deep, appeared as early as after approximately 10,000 cycles in tension-compression; at this low load the cracks did not, of course, result in fractures, and even with stress of approximately ±8 kg/mm.^{2} (5ts. per sq. in.) it took one million cycles from the appearance of the crack until the fracture took place. In practice one cannot expect that an incipient crack will not develop further; an established crack must always be considered to be a serious danger signal.

TENSION-COMPRESSION TESTS

Before passing on to the full-scale experiments, a couple of tension-compression tests which were carried out by the Laboratory for Building Technics at the Technical High School in Copenhagen deserve mention. Each cross (Fig. 3) is the result of a Wöhler curve determined by four or five bars. It will be seen that the tension-compression test bar d = 46, D = 49 mm. (= ab. 2 inches), only stands ±7·3 kg per mm.^{2} (10,400lb. per sq. in.) or half of that of the 11 mm. ($\frac{7}{16}$ inch) notched bending bar with the same fillet radius. This is partly due to the fact that the tension-compression strength is usually about 70 per cent of the bending strength and partly to the 0·4 mm. (0·016 inch) fillet naturally being relatively sharper for a 46 mm. bar than for an 11 mm. bar, and probably due to the "size effect" too. Various points of view regarding the size effect are brought forward from time to time—some maintain that it only applies up to 12 mm. ($\frac{1}{2}$ inch), others that it is demonstrable for considerably larger diameters, at least in connexion with notch effect.

A series of notches above one another relieve each other so that the strength of a thread with the same diameter (d = 46) and fillet radius 0·4 mm. as the single notch was ±11 kg per mm.^{2} (15,600lb. per sq. in.) against ±7·3 kg per mm.^{2} for the single notch; in this case the fractures occurred in the 1mm. (0·04 inch) fillet in the tool relief groove (d = 45·9 mm.) which is thus more dangerous than several 0·4 mm. (0·016 inch) radius fillets beside each other in the thread itself.

For comparison, the triangle (Fig. 3) shows the result of the determination of the fatigue strength which was made in connexion with the British investigations already mentioned by tension-compression tests with bars of similar size to those used in the experiments here described.

CRANKSHAFTS

On the pulsator machine of the Danish State Testing Laboratories, a series of tests were made with small crankshafts for truck Diesel engines (Fig. 5). This concerns crankshafts of the type which is supported for every second cylinder only

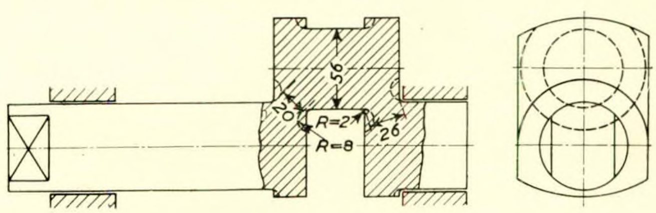

FIG. 5—*Crankshaft for truck engine 56 mm. (2 3/16 inch) diameter, tested for completely reversed bending endurance to full scale, first as the solid lines, then altered as shown by the dotted lines*

and the tests were made with a full-size throw, the adjacent main bearings, and a section of straight shaft which took the place of the second so that the bending stresses in the throw being examined were comparatively similar to those in the actual shaft, disregarding the fact, however, that the actual shaft is continuous. Interest was focused on the bending stresses and the throw was therefore tested only in the position corresponding to the top dead centre, and with stresses corresponding to tension and compression forces of equal magnitude in the connecting rod, which comes quite close to actual conditions, the acceleration force when the piston is in its bottom position being of the same magnitude as the difference between the ignition force and the acceleration force when the piston is in the top position. These small crankshafts were made of chrome-nickel steel, but as the test was made during the war it was not possible to obtain exactly the same material, for which reason 5 per cent nickel steel was used.

This type of crankshaft has a rather unfavourable stress distribution because crank pin journal and main bearing journal overlap each other without borings.

The shaft was first tested in its original form with 2 mm. (0·08 inch) fillet between webs and journal (Fig. 5) and then with 4 mm. (0·16 inch) fillets, which is the maximum attainable in practice on account of the surface pressure on the crank journal; with this an improvement of about 5 per cent only was obtained. The fillet radius was then increased to 8 mm. (0·32 inch) at the expense of the web. It is not evident beforehand that this result is an improvement because the web is so thin that there is a considerable reduction in the moment of resistance; but it proved that the strength of the shaft increased by 35 per cent, possibly due to the fact that the thickness of the web is reduced over such a length that it attains some flexibility and the stresses are better distributed. Most of the fully-forged and semi-built crankshafts produced by the author's firm are made according to this principle (Fig. 6), which was introduced nearly twenty years ago, based at that time on theoretical considerations.

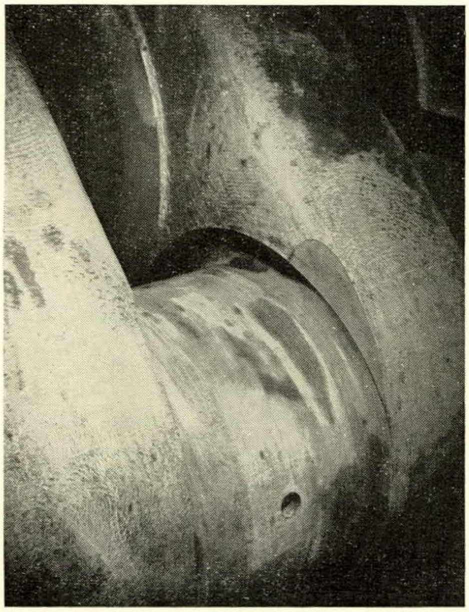

FIG. 6—*Crankshaft for a modern marine engine during roughturning*

By moving the fillet into the web it has been possible to increase the radius to 26 mm.

Tests were also carried out with turnings in the journal and with a rather large *eccentric hole drilled in the crankpin* so that there should be the least possible material near the critical area, with a fillet 3 mm.; also, with these two designs a 35 per cent improvement was obtained compared with the original design.

The deciding factor in connexion with this design seems to be that the stresses are prevented from concentrating around the transition between journal and web, from which place the fractures most often start.

For an ordinary forged crankshaft a 35 per cent improvement has thus been obtained by this method; but more can be obtained with cast crankshafts as the crank web can be made in such a way that the stresses are forced into the most remote part of the web; and by adopting quality cast iron of the Meehanite type, for example, the stresses can be reduced so much as to give the same margin of safety with a cast iron shaft as with normal shafts of mild steel; at the same time a considerable saving in cost, of course, is obtained. His company has had such a cast iron shaft for a somewhat bigger engine type in service for four years without the least inconvenience, and have cast a few additional shafts (Fig. 7) of the same type without altering bearing dimensions or cylinder distances. One disadvantage with cast iron is that the elastic modulus is somewhat lower than that of steel, so that the natural frequency of the crankshaft becomes lower (approximately 20 per cent) whereby critical numbers of revolutions are sometimes passed (which could be avoided by using steel shafts), while in other cases it is necessary to reduce the normal speed of the engine or to introduce vibration dampers in order to avoid extra stresses which are too high. In this connexion nodular iron is somewhat better.

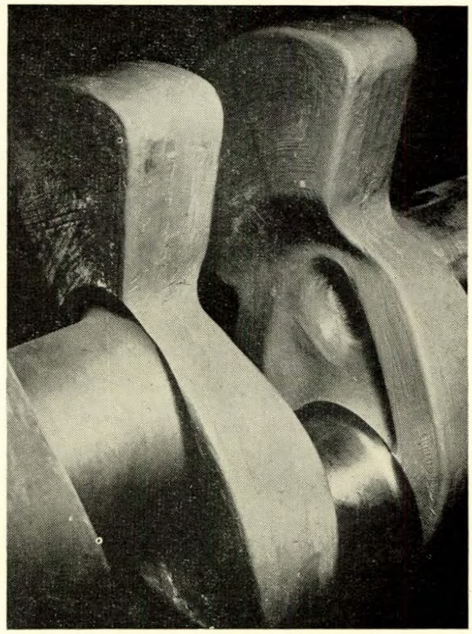

FIG. 7—"*Meehanite*" *crankshaft for auxiliary engine*

The area round the critical fillet rounding has been weakened so that the stresses are forced into the heavy ribs of the web

PISTON RODS

As mentioned before, piston rods for double-acting two-stroke engines in the years around the last war produced some difficulties, but these now have been overcome.

The crosshead connexion in question is shown in Fig. 8 and cracks have been encountered in both the upper and in the lower threads. The last mentioned location is noteworthy as the pretightening of the connexion results in the greater part of the varying load being transmitted directly from the rod via the top nut to the crosshead because the crosshead is more rigid than the portion of the rod traversing it. Even if the connexion is not properly tightened the stresses will only vary from zero to a tension corresponding to the gas force, whereas the stresses at the top end thread vary from tension to compression with a stress difference which is nearly double the value of that occurring in the lower thread with insufficient pretightening.

A series of tests with joints of this kind have been made on the pulsator machine of the Danish State Testing Laboratories, which can yield a maximum of ± 10 tons. It was therefore necessary to limit the rod diameter to 36 mm. (about $1\frac{1}{2}$ inches), the actual rods being 127 to 180 mm. (5 to 7 inches). The experiments were started by tightening the rod so that the *pretension* between the nuts would be 10 kg per mm.^{2} (14,200lb. per sq. in.) according to ordinary calculation methods; but it was noted that the rod broke in cross section A at a varying stress of $\pm 6{\cdot}7$ kg per mm.^{2} ($\pm$9,500lb. per sq. in.) measured in D, which corresponds to approximately ± 2 kg per mm.^{2} ($\pm$3,000lb. per sq. in.) only in A. By means of a tensometer, the elongation of the part of the rod traversing the crosshead B was checked, and it was seen that the pretension (in reality) was approximately 5 kg per mm.^{2} (7,000lb. per sq. in.). A series of tests with tightening and slackening showed (see Fig. 9) that a considerable setting takes place so that the tightening-up curve does not follow the calculated dotted line but has considerable curvature. By tightening and slackening several times, a considerable permanent set was also noticed, in such a way that stationary condition was not reached until after four to five successive tightenings. This has also been checked several times on actual rods; here the fault was somewhat less, but it was found necessary to tighten an angle

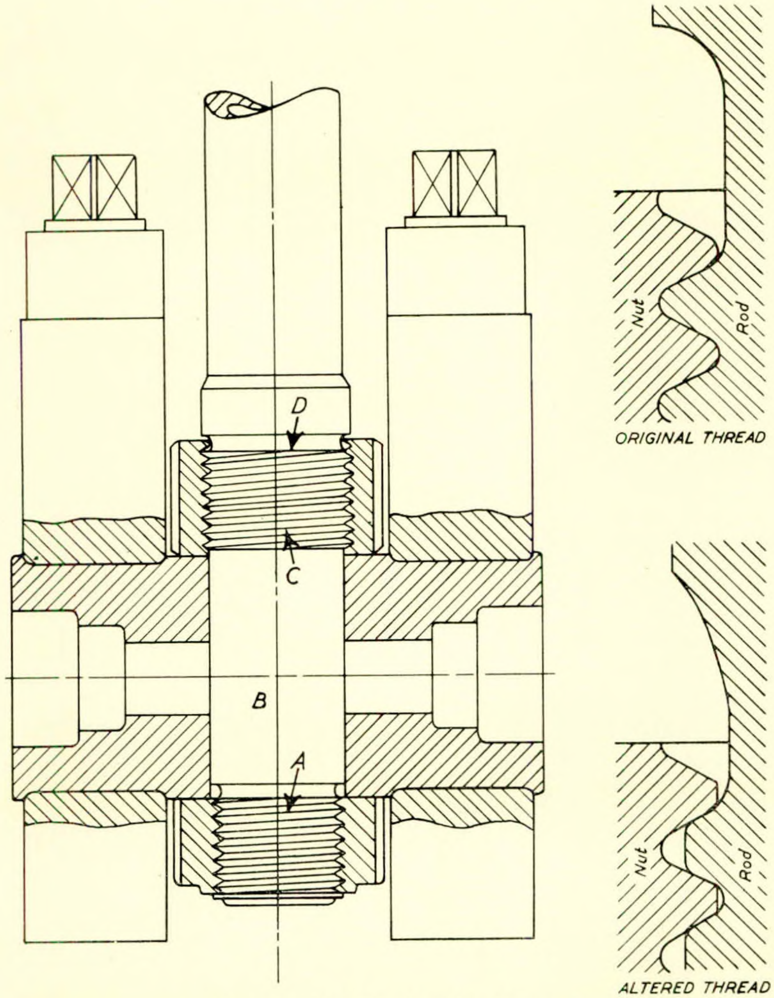

FIG. 8—*Crosshead and piston rod*

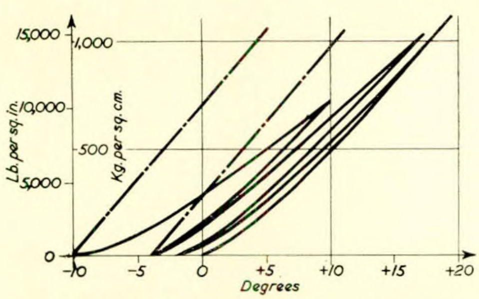

FIG. 9—*Pretension in the piston rod (point B, Fig. 8), calculated (dotted lines) and measured (solid lines)*

which is 20 to 40 per cent greater than that found by simple calculations. Instead of tightening and slacking four or five times, the rod may first be given a somewhat higher stress than that which is to be used in practice. The ensuing result is that piston rods or other vital threaded joints where a pretension is required which, theoretically, should equal for example a 13 deg. turn of the nut, are first pretightened, then tightened to 21 degrees, slackened, and again pretightened, after which they are tightened from the new point of pretightening to 18 degrees.

Also during running, wear and setting takes place, and it is therefore dangerous to place fixed tightening marks on nuts and crossheads, as the pretension thus becomes continuously less. The fact that many ships sailing during the war had difficult maintenance conditions has presumably been one of the causes of the many mishaps with piston rods which occurred in these very years.

As the test rod (Fig. 8) thus received the intended pretension, 10 kg per mm.^{2} (14,200lb. per sq. in.) it did not break until at ± 9 kg per mm.^{2} in D ($\pm 12{,}800$lb. per sq. in.), but the fracture still occurred in section A where the varying tensile stresses were hardly $\pm 2{\cdot}7$ kg per mm.^{2} (4,000lb. per sq. in.); there were no fractures at C or D where, as stated, the tension was ± 9 kg per mm.^{2} (12,800lb. per sq. in.). It was also seen that the fracture did not start at the bottom of the thread but at the transition between thread-rounding and flank, and this is no doubt due to the fact that to the simple tensile stresses in the cross section must be added the bending stresses in the thread, to which, of course, the notch effect must also be taken into consideration.

In a correctly machined thread those which are next to the contact surface of the lower nut will carry the greater part of the stress. An improvement may therefore be obtained by cutting the *lower nut* with a pitch a little higher than on the corresponding thread of the rod, thus causing all threads to bear more equally. As it is a question of a pitch increase of a magnitude of 0·03 mm. (0·0012 inch) per 1 inch, such work requires great accuracy. The same effect could, however, be achieved more simply by cutting both threads with the same pitch, but making one of them slightly conical. If the difference in pitch or the conicalness becomes too great, there is the risk, however, that the lowest individual thread may break off, and a difference in pitch in the opposite direction to that desired will worsen conditions correspondingly.

Tests were also made with serrated threads, but with no better results. A single rod of this type broke after approximately 46 million cycles; this experiment alone took nearly two months.

On the other hand a considerable improvement was attained by increasing the axial height of the individual threads, or—as is the present practice—*by reducing the radial height.* When the height of the individual threads on the rod is reduced (Fig. 9, lower right) the bending stress is also reduced, but at the same time it is correspondingly increased on the threads of the nut. This was, however, compensated by making the nut of special steel. In this way a fatigue strength of $\pm 10{\cdot}5$ kg per mm.^{2} (15,000 lb. per sq. in.) was obtained in D, and the fracture now occurred at the top D of the uppermost thread when the pretension was sufficient.

Later nuts were made flexible so that, to a lesser degree, the stress was placed on certain threads.

Tests were also made with *rolled threads.* This naturally called for some preparatory experiments to find a suitable degree of rolling, the correct flank angles on the roller, fillet radii, etc.; but the efforts produced as a result a fatigue strength of ± 15 kg per mm.^{2} (21,000lb. per sq. in.).

It should also be mentioned that even with the heavy deformation which takes place when a $\frac{1}{2}$in. thread is rolled direct from round-bar material, a greater fatigue strength has been measured than for rods with ordinary cut threads; threads from $\frac{1}{2}$ inch to $\frac{3}{4}$ inch for the author's company are therefore often made by rolling.

Within the same outer dimensions of piston rod and crosshead, the fatigue strength has thus been increased from $6{\cdot}7$ to 15 kg per mm.^{2} (9,500 to 21,000lb. per sq. in.) or more than doubled, without making the design materially more expensive; the special alloyed steel for the nuts costs somewhat more than unalloyed steel, nor is the rolling of the threads inexpensive, but the price increase on the whole design is nevertheless minimal compared with the increase in strength by 120 per cent. To obtain a similar improvement by increasing the diameter of the piston rod by 40 to 50 per cent would have required such great alterations of the engine design that it could not have been carried out in practice.

It will be noted that the progress thus made has only been possible by an improvement in *design* (lower threads and a more flexible nut), *material* (special steel nuts), *machining* (rolling of the threads and careful adherence to the pitch), and *maintenance* (correct tightening of the nuts). Actually one seldom gets to the bottom of a mishap in service without at least checking all four of these factors: design, material, machining, and service conditions. Usually it is possible to achieve improvements on several points.

To carry out the fatigue tests with high loads of ± 15 kg per mm.^{2} (21,000lb. per sq. in) it was necessary to increase *pretension* to over 20 kg per mm.^{2} (28,000lb. per sq. in.). That is to say, this is the simple tensile stress at B; to this comes the bending stress in the bottom of the threads, which is of the same order of magnitude, plus the notch effect which again nearly doubles the stress so that the yield point is greatly surpassed, the latter being about 26 kg per mm.^{2} (37,000lb. per sq. in.) for $0{\cdot}20$ steel. Actually, the tension is not much more than 26 kg per mm.^{2}, for as soon as the yield point is passed, the material in the surface yields so much that the tension spreads more deeply into the material or to other threads. But even though the yield point has been passed, the design can consequently stand a considerable varying stress.

If the thread joint is to be utilized to advantage, the pretension should not be too small; but of course there is also an *upper limit for the pretension,* and there is hardly any doubt that many small screws are tightened too hard. At any rate this risk is run with screws of 1in. diameter or less. The author's Company once made an experiment in which a number of experienced men from the workshop tightened a $1\frac{1}{4}$in. screw to the extent each considered adequate. The result was that the deformation measured on a cylindrical shaft varied corresponding to stresses from 25 kg per mm.^{2} up to approximately 40 kg per mm.^{2} (35,000 to 57,000lb. per sq. in.) (calculated as elastic deformation) so that, consequently, a very heavy yielding took place at the bottom of the threads where both bending stress and notch effect were added to the simple tension.

For smaller threads the risk of excessive tightening is far greater. Actually, most spanners are wrongly dimensioned. The torque for tightening is consumed almost entirely by friction, which increases with the cube of the diameter. Thus if a constant force is applied to the spanner, the latter should have a length approximately proportional to the cube of the diameter; in most standards the spanner length varies almost in proportion to the diameter. For a number of joints where it would be difficult to tighten according to degrees, the spanner length has been prescribed so that an adequate pretightening is obtained; for some vital bolts in smaller engines, torque spanners are used.

SHRUNK JOINTS

In the Laboratory for Building Technics at The Danish Technical High School some fatigue tests have been done with models of piston rods which were shrunk in the crosshead according to Svenska Kullagerfabriken's (SKF's) method with oil pressure (Fig. 10). It was hoped thereby to avoid the notch effect at the thread flank, which doubtless contributes to

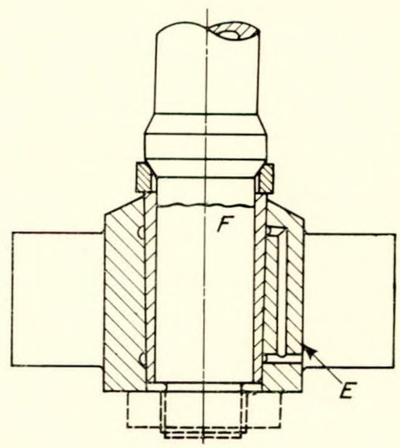

FIG. 10—*Shrunk joint crosshead-piston rod*

Pressure oil admission at E. For this design the fracture took place at F; for the final design the cone on top of the crosshead was made more acute angled. The inserted bush is necessary with a view to the vertical adjustment of the piston rod; its thickness and conicalness is shown exaggerated in the sketch

a reduction in the strength of the threaded joint. But it appeared that a similar stress concentration occurred at the shrinkage in spite of all efforts to make smooth transitions between the rod and its collar as well as in the form of a slender cone at the top of the crosshead. Also, the fatigue strength proved to be practically the same as for the best design of threads and nuts without rolling of the threads.

By rolling, the threads were improved considerably; a similar improvement could undoubtedly be attained by rolling the conical shrinkage surface. For the purpose in question there are, however, apart from the considerable cost involved in changing the design in existing plants, various practical drawbacks with the shrunk connexion. It has therefore been preferred to retain the threaded joint, since the fatigue strength is the same.

Preceding these tests was a long series of static tests to find the best arrangement of the oil grooves, the permissible oil pressure, a suitable compensation for the crosshead being out of round, etc. The coefficient of friction was found to be surprisingly uniform—0·16 to 0·17.

WELDED ELEMENTS

A final example concerns a full-scale test with a welded construction to investigate the merits of stress relief annealing. Judging from the Smith diagram, large welding stresses are required to reduce the fatigue strength materially, and somewhat contradictory statements on this point can be read. Those are based mainly on results with rather small welding test pieces. Some test pieces were therefore made consisting of 28 mm. (1⅛ inch) plate welded to 80 mm. (3⅛ inch) plate for a length of one metre (3ft. 3in.) (Fig. 11). This structure was exposed to varying one-sided bending by means of the pulsator machine at the Laboratory for Building Technics and the bending fatigue strength was found to be 0–10·3 kg per mm.^{2} (14,600lb. per sq. in.) or 5·15 $\pm$5·15 kg per mm.^{2} Thereupon others of the specimens were stress-relief annealed at 650 deg. C. (1,200 deg. F.), as a result of which the fatigue strength rose to 0–12·4 kg per mm.^{2} (17,600lb. per sq. in.) or 6·2 $\pm$ 6·2 kg per mm.^{2}

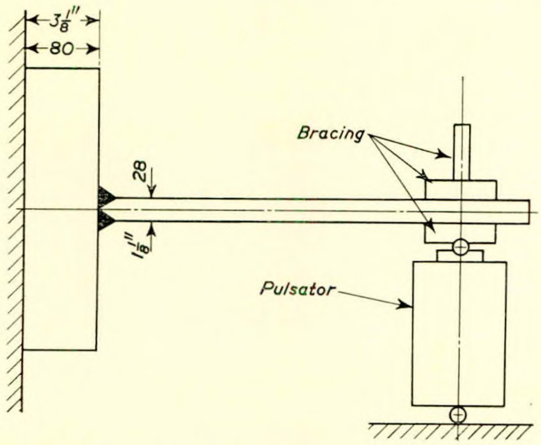

FIG. 11—*Welded construction tested to full scale at one-sided bending*

The weldings are one meter (3ft. 3in.) long (perpendicular to the paper)

For this test a set-up was made so that four plates could be tested simultaneously and the experiment was made with four blanks for each determination.

Steel of good weldability approximately 0·17 per cent C, 0·14 per cent Si, and 0·75 per cent Mn, was used for the test. The electrodes were 6 to 7 mm. thick (ab. ¼ inch), low hydrogen, but no preheating, no machining and no peening of the welds. All the fractures occurred in the body plates close to the welds where the greatest stress is increased by the notch effect. The hardness in this area after the welding was approximately 225 Vickers; it fell to 179 by the stress-annealing; at the same time the welding stresses in the outer run, and thereby in the transition zone, were of course considerably reduced.

Of late years the author's company have stress-annealed the most important welded structures for Diesel engines, and, as this test shows that there is a 20 per cent gain in fatigue strength and the cost is less than 20 per cent of the price of the material, stress annealing must at least in this case be said to be appropriate.

REFERENCES

(1) Archiv für das Eisenhüttenwesen, May 1952, p. 203.

(2) "The Iron Age", 29th March 1946, p. 40, and 5th April 1946, p. 66.

(3) Fenner, Owen and Philips. 25th May 1951. "Engineering", p. 637.

Discussion

Mr. A. G. Arnold (Member) said he had been asked to read a contribution from MR. F. G. VAN ASPEREN, Dipl.Ing.E.T.H. (Member).

Mr. van Asperen said that Mr. Wiene's paper contained many valuable remarks and conclusions which could be directly applied in practical cases.

With regard to the fracturing of springs, the statement about the elimination during the first few months of service of the weak numbers agreed with general experience. It was also agreed that shot peening increased fatigue strength considerably. The parallel drawn between the effect of closing the cracks by compression by either chemical nitriding or mechanical shot peening was, indeed, noteworthy.

It was not quite clear, however, that incipient and thus actually existing surface cracks were really rendered quite harmless by the bombardment of the steel balls. It seemed more likely that actually the peak stresses in the surface were being equalized.

With regard to the effect of sharpness of notches and the slowness of development of incipient cracks in the bottom of grooves, the explanation might perhaps be that extra stresses occurred during the mechanical machining of the grooves, causing the formation of incipient cracks. These stresses, however, did not remain but were equalized by the very formation of the cracks. The development of the cracks was probably caused later on, by tensions from the actual forces on the parts in service.

A very interesting item was the experience on the tightening of piston rods in crossheads. It might be helpful to draw the author's attention to similar problems occurring when tightening the nuts on long steel tie rods used for connecting the bed plate, columns and cylinder beams in large marine engines.

Previously the normal pre-tension was obtained by flame heating of the rod locally until the prescribed elongation was reached and the nut could be tightened some 140-150 degrees. Nowadays the normal practice was to prestress the rods hydraulically and then tighten the nut. This prestressing was done in two stages over the whole beam so that all nuts were tightened gradually. After the second tightening a control was made by applying the full pressure again and checking any undue clearance between nut and beam surface.

Experience showed, however, that special care should be taken that the bottom surface of the nut was not slanting. In that case, extra stresses might occur at one point of the thread, leading occasionally to incipient and slowly developing cracks in the rod. As a remarkable fact, it was observed at such a failure that the final broken surface did not show any rest fracture. It was most likely that the tension was gradually reduced by the developing crack. It would be interesting to learn the author's views on any similar cases that might have been experienced in either piston rods or tie-rods.

Finally, with reference to the tests on welded elements, it would be appreciated if the author would give his views as to whether welded elements should be used for moving parts, exposed as they were to mechanical and dynamic forces.

In case of failure the consequences would probably be more severe than in the case of stationary parts, such as bed plates or columns. It was therefore considered safer, in view of the need for elaborate control of the different welds, to manufacture moving parts, such as scavenge pump levers or piston cooling oil conduits of forged or cast steel, although a welded assembly was often tempting from the manufacturing point of view.

MR. ARNOLD then said it had been his pleasure to have worked with Mr. Wiene for a very long time and he knew something of the work that he had done in connexion with the development of the Diesel engine.

He considered Mr. Wiene's reference to springs on page 40 to be very interesting reading and was most appreciative of this interesting contribution. Some years ago, Mr. Wiene's company produced springs for fuel valves which consisted of a number of steel washers. These springs were a very great improvement over the ordinary coil spring, which his company had previously used in similar engines—in fact, they still had a considerable number of valves working with these springs in a most satisfactory manner, and he would like to know from Mr. Wiene whether there was any reason why his Company should have stopped using this type of spring, as they had been thinking of going back to it in a more general way. If, of course, this step would be retrograde, he would be very pleased if Mr. Wiene would let them have his views on this point.

Mr. Wiene had said much about piston rods and their failures, also about what had been done to reduce this most objectionable and serious feature on this type of engine. On page 6, Mr. Wiene referred to the nut being made very flexible but in Fig. 8 on the same page, where he showed a crosshead and piston rod, he did not show the flexibility of the nut. His company had a groove at the top nut for a very long time and thought this had contributed to a reduction in the number of failures. The tightening of the nuts to a predetermined amount to which Mr. Wiene referred was no doubt an excellent thing, and in his view had contributed to the lessening of the failures. They had adopted this method for a very long time, and had tightened the nuts by calculation, controlling the calculation by elongation of that part of the rod which was stretched, and in this way believed they had stressed the rod to 9·5 tons per sq. in. which was in excess of the inertia stresses. Mr. Wiene referred to a stress of 12·5 kg. per mm^2. He would be very pleased to have Mr. Wiene's considered opinion on this extremely important point, which he was sure would be very valuable to the industry as a whole.

Mr. Wiene referred to fitting the piston rod into the crosshead; he was afraid that hitherto this had not been considered of very great importance. He now thought with him that it was very important and he would like to have his views on just what type of fit he would recommend for fitting the piston rod into the crosshead of a two-stroke cycle double acting engine having a stroke of 1,400 mm.

He was not clear why Mr. Wiene should suggest that the piston rods in crossheads of single acting engines should be shrunk in as illustrated, and here again he would like Mr. Wiene, if possible, to be a little more explicit.

He would have been pleased if Mr. Wiene had made more reference to the possibility or otherwise of having the piston rods made of good 28/32 ton tensile steel. Mr. Wiene would be aware that there was a period when quite a number of eminent people considered that the tensile of the material used for piston rods should be increased considerably. They did not think this was right and consequently insisted upon material having a tensile of not more than 32 tons per sq. in.

In Mr. Wiene's illustration he did not show the hole through the centre of the rod, which allowed the piston cooling oil to return from the piston. Originally, there was one parallel hole of 70 mm. in diameter right through the rod, but latterly they had reduced the size of this hole over the length covered by the piston rod nuts, etc., and had had the hole in this portion of the rod reamered. He was afraid the lack of consideration given to the boring of these holes hitherto had contributed to a number of failures.

Also, they had adopted nuts made of nickel steel, the content of the nickel being 3½/5 per cent, and this had proved to be of great advantage. When this material had not been available, they had used steel to the specification of EN 19, which had also proved to be satisfactory. In an attempt to avoid the seizing of nuts on the rods, of which they had had a number, to get the differential of material, they tried wrought iron nuts, but this was not entirely successful, due no doubt to the inability to get wrought iron of sufficiently good quality and quantity.

It was clear that if the engine was allowed to run out of truth, the rod would be submitted to stresses for which it was not designed; consequently, it was essential that there should be a reasonable clearance between the guide bars and guide shoe. Generally, these clearances were maintained in their fleet to 12-15/1,000 inch. It was thought this amount might be reduced, consequently they arranged for a clearance on one guide only to be reduced to 8/1,000 inch; this experiment was made on a short coastal passage. It was not satisfactory, the white metal being adversely affected by this low clearance.

Another interesting point in Mr. Wiene's paper was that concerning stresses and the defects which had been revealed in some cases where fabricated bedplates were employed. They had had fabricated bedplates and had specified that they be annealed or stress relieved, and he was very pleased to have Mr. Wiene's confirmation of this very important point; it was not a unanimous view in the industry. Incidentally, they had never thought it was a progressive step to adopt fabricated steel bedplates and now that spheroidal graphite iron had been developed, they were seriously considering the adoption of this material. They had only had this type of bedplate since 1951, and he was bound to say that over this period they had not had any trouble; nor, of course, had they had the slightest trouble with a great number of bedplates, totalling over one hundred, made previously in ordinary cast iron.

Mr. L. F. Moore, B.Sc.(Eng.), said he proposed to confine his remarks mainly to "the sharpness of the notch" and "piston rods".

He would prefer to see the graph in Fig. 3 with ratios of fillet radius to root diameter as abscissæ, as this would indicate the geometrical similarity—or dissimilarity—between the specimens. Also it would be noticed that the threaded specimen with a root fillet of 0·4 mm. radius failed at the 1 mm. radius tool relief groove and, as it was possible that the thread itself would have failed at a higher stress than ± 11 kg./mm.^{2}, this point should be plotted on the abscissa corresponding to 1 mm. radius.

With reference to Mr. Wiene's remarks on size effect, the British Shipbuilding Research Association were certain that size effect was apparent with up to at least 3-in. diameter bolts (the fatigue strength decreasing with size), but it was possible that there was little decrease in strength above 4-in. diameter.

Perhaps Mr. Wiene would say whether the piston rod tests were carried out in a bending or a "push-pull" rig. He noticed that the actual pre-tension in the first test was 5 kg./mm.^{2}, whereas the alternating stress was 6·7 kg./mm.^{2} at section D. Possibly there was parting line separation at section C in this instance which would partly account for fracture of the thread at section A. This might also account for fracture at the same point in the second rod (pre-tension 10 kg./mm.^{2} and alternating stress of 9 kg./mm.^{2}), but this hypothesis would not suit the third case where the radial height of the thread was reduced. It would be interesting if Mr. Wiene could give some details of the accuracy of the threads, especially the pitch, in this connexion.

He agreed that a positive pitch error in the nut would increase the fatigue strength due to a better load distribution along the thread, but he thought that a negative pitch error had very little effect on the fatigue strength.

Mr. W. H. Falconer (Associate) said that the paper dealt with a subject that was often neglected by engineers. He knew of a case where a certain pump shaft fractured regularly and was replaced about every two years during the life of the ship. This might seem rather funny, but the consequences could have been serious.

He asked how Mr. Wiene had arrived at the modified thread form shown in Fig. 8. The nut did not appear to have been altered very much from those originally fitted. Photo-elastic stress analysis did provide a convenient method of investigating stress distribution and he wondered if it were considered for this particular problem.

The problem of the screw thread was essentially three dimensional but this particular problem could be fully investigated by the well-known "frozen stress" method. It was possible to work in three dimensions and therefore a stress pattern could be obtained for this type of connexion.

He agreed with Mr. Arnold's remarks concerning the method employed to detect defective rods but was of the opinion that this method could not be used by everyone. The operator must be experienced. An inexperienced operator might detect scratches and misinterpret them, and as a result otherwise serviceable piston rods would be taken out of service and scrapped.

With regard to "incipient cracks", did he understand that the threads were cut and that there was thus a reduction in the effective number of threads within the nut? If so, how much thread was cut away before the rods were eventually scrapped?

He agreed with Mr. Wiene that improvement had been made as a result of design, material, machining and maintenance. But he hoped Mr. Wiene would not now forget about the problem and consider it solved for this type of engine. The particular engines referred to in the paper had not been built in the last ten or twelve years, but there were many engines of a very similar design approaching an age at which the design referred to started to run into trouble. He hoped Mr. Wiene's company would continue their efforts to find a permanent solution. He agreed with Mr. Arnold that they were not yet out of the fog.

He appreciated that thread rolling produced a considerable increase in the fatigue strength of the rods and bolts. This was what one would expect; it was merely a controlled variation of shot blasting. Digressing slightly, he would like to know what effect thread rolling had on austenitic steels. He had heard that it had an adverse effect. Had Mr. Wiene done any research in this field, and if so, what was his explanation of this deviation?

Referring to the table on page 40, Mr. Wiene said that "Fractures presumably start just under the nitrided surface..., the result consequently becomes the same in the notched as well as the smooth test bars".

He wondered if Mr. Wiene would care to amplify his statement. It was observed that the result of nitriding a smooth bar only increased its fatigue strength by 40 per cent whereas nitriding a bar with a notch in it had increased the fatigue strength by 200 per cent.

MR. G. P. SMEDLEY, B.Eng., B.Met., said that Mr. Wiene had presented a number of exceedingly interesting examples of engine components which experienced fatigue loading in service. They indicated that there was a serious lack of reliable data relating to the notched fatigue strength of large sized forgings and castings. It must be admitted that theoretical notch sensitivity factors and the results of small scale fatigue tests offered very limited guidance in the assessment of the characteristics of engine parts of large size. Moreover, much of the data derived from the fatigue tests of actual components could not be applied generally as the variables were many in number.

There was some information, however, which was sufficiently comprehensive to permit reasonable analysis. This had led him to believe that the theoretical approach favouring stress concentration factors and ratios of notch dimensions to section sizes were misleading. So far these criteria had not proved reliable and he would like to show in one example how a simpler approach could lead to more practical information.

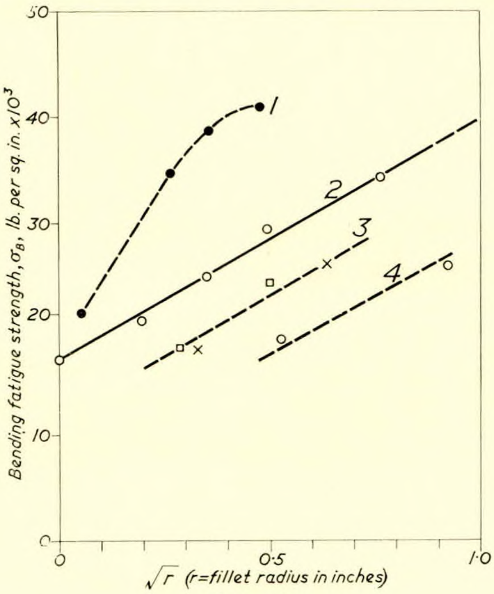

FIG. 12

(1)* C steel, U.T.S. 37·5 t.s.i., 0·375 inch ϕ
(2)* C steel, U.T.S. 37·5 t.s.i., 1·18 inch ϕ
(3)‡§ "46" C steel, 1·5 inch and 1·6 inch ϕ
(4)‡ "50" C steel, 5¼ inch ϕ

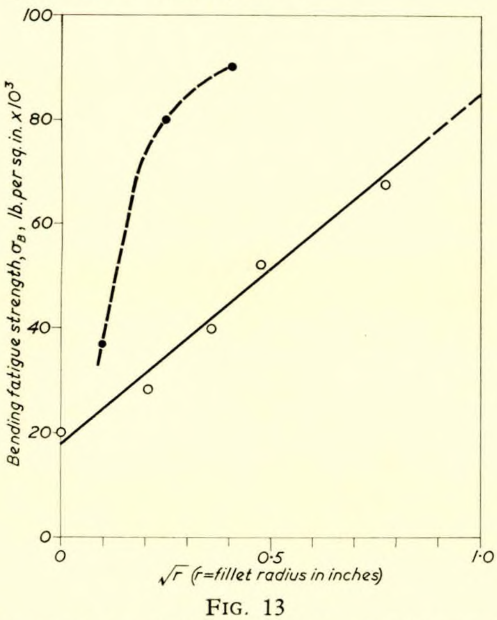

FIG. 13

Lehr and Mailander* Ni-Cr-W steel, U.T.S 77 t.s.i.
○ 1·18 inch ϕ specimens
● 0·375 inch ϕ specimens

The example comprised fillet radii between two sections of a shaft, having two different diameters. Lehr and Mailander* had carried out tests in reverse bending fatigue on 1·18in. diameter shafts having fillet radii and a ratio of larger to smaller diameter of about 2:1. Their results showed linearity on a simple σ_B, $\sqrt{r}$ plot for both the carbon steel and the Ni-Cr-W steel employed (Figs. 12 and 13).

Where σ_B = bending fatigue strength, lb. per sq. in.
r = fillet radius in inches.

The equations of the two plots were as follows:—

$$\sigma_B{}^1 = 15{,}600 + 24{,}400\ \sqrt{r} \qquad \text{.......................(1)}$$

For the C steel, U.T.S. = 37·5 T.S.I.

$$\sigma_B{}^2 = 18{,}000 + 67{,}000\ \sqrt{r} \qquad \text{.......................(2)}$$

For the Ni-Cr-W steel, U.T.S. = 77 T.S.I.

Three main conclusions could be drawn from these equations:—

(*a*) The fatigue strength could still be derived from the equations for the sharpest fillet, $r = o$.

(*b*) The alloy steel had a much greater notch sensitivity than the C steel, although its fatigue strength was always higher.

(*c*) The equation was limited to a maximum value when σ_B = unnotched fatigue strength (36,000lb. per sq. in. for the *c* steel; 81,000lb. per sq. in. for the alloy steel).

For 9¾in. diameter "25" carbon steel shafts tested in

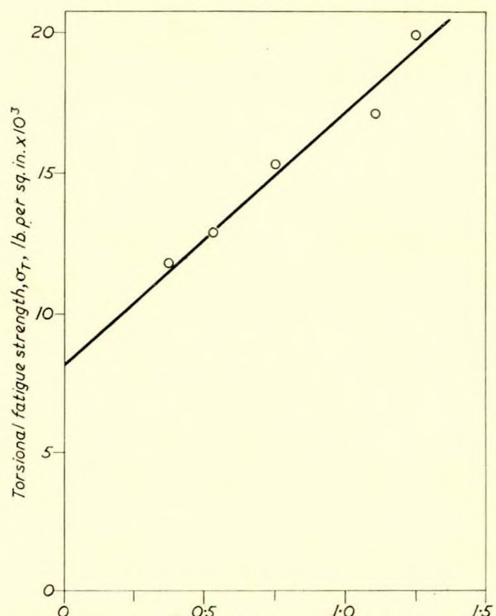

FIG. 14

Bunyan and Attia†
"25" carbon steel, 9¾-in. diameter shafts

reverse torsion, the results quoted by Bunyan and Attia† also displayed linearity on a similar plot (Fig. 14) and conformed to an equation:—

$$\sigma_T = 8{,}100 + 9{,}200\ \sqrt{r} \qquad \text{..........................(3)}$$

where σ_T was the torsional fatigue strength in lb. per sq. in.

It could be concluded therefore that the fatigue strength of a shaft of given diameter would be dependent on the size of

* Lehr, E. and Mailander, R. 1937-38. Arch. Eisenhuttenw, p. 563.

† Bunyan, T. W. and Attia, H. H. 1953. Trans.I.E.S.S., Paper No. 1170, p. 425.

fillet radius in accordance with the following expression:—

$$\sigma = y + c\ \sqrt{r} \qquad (4)$$

y and c are constants.

Size Effect

The limited results of Horger‡ and Peterson and Wahl§ for C steel shafts 1·5 inches, 1·6 inches and 5¼ inches in diameter tested in reverse bending, gave some indication of the size effect (Fig. 12). It would appear that the constant c (equation 4) was uninfluenced by the diameter of the shaft but that y decreased as the diameter increased. If this was the case the general expression including size effect could be written in the form:—

$$\sigma = a - b(fnD) + c\ \sqrt{r} \qquad (5)$$

a, b and c were constants for each particular grade of material and a particular type of loading (i.e. bending, torsion, etc.). The function of D was uncertain although it would appear to be a power approaching unity. A programme of systematic fatigue testing was required to establish its exact value, the value of b and their variation with material and loading condition. However, Lehr and Mailander also carried out tests on specimens of 0·375 inchϕ. The results showed a marked departure from relationship (4) (Figs. 12 and 13). He considered that surface effects had a serious influence on the fatigue strength of small specimens of both notched and unnotched types. For this reason small scale tests (D less than ⅜ inch) could not be employed usefully for predicting accurately the performance of large components.

The results of this analysis cast further doubts on the validity of the theory that under fatigue conditions the influence of a notch was a pure stress concentration effect. Equation (5) did not represent the conditions for other types of notches (e.g. grooves, screw threads, holes, etc.). Nevertheless the limited evidence, although somewhat conflicting, would indicate that if variables were treated separately and on a pure dimensional basis, more reliable relationships could be developed.

Mr. B. Todd, M.Eng., thanked Mr. Wiene for a most interesting paper which contributed much to knowledge of the fatigue strength of metallic parts.

There were two points on which he would like to comment, namely springs and piston rod failures.

Although he agreed with Mr. Wiene's statement on page 40 that the analysis was probably not the most important factor in springs but simply that the surface was faultless, he would like to qualify this in that the surface would depend to a large extent on the analysis.

Kenneford and Ellis* showed that in a series of spring steels, Si-Mn steel—a very widely used one—decarburized far more readily than any of the other steels tested.

Table I

Steel	Specification	Fatigue limit, tons per sq. in. Not decarburized	Decarburized to 5 thou.	Ratio
Si-Mn	S.T.A. 2b	± 58·5	± 32	0·55
Cr-V	S.T.A. 2c	± 55·0	± 36	0·64
Si-Cr	S.T.A. 2d	± 48·5	± 34·5	0·71
Cr-Mo	S.T.A. 12	± 49·5	± 39	0·79
Ni-Cr-Mo	En. 25	± 49·2	± 43·5	0·88

All heat treated to V.D.P. 550

Table I showed the results of fatigue tests on this series of steels based on an endurance limit of 5 million reversals. These results showed that although Si-Mn steel had the highest endurance limit where no decarburization occurred, in the decarburized condition it had the lowest. It must be borne in mind also that these steels were decarburized to the same extent, whereas in practice—given the same heat treatment—the Si-Mn steel would be more heavily decarburized than any of the others.

These results indicated that a spring material should be chosen in relation to the plant available to make the spring and that manufacturers who did not possess atmosphere controlled furnaces should not use easily decarburized steels.

It was interesting to read on page 43 that difficulties in piston rods had now been overcome, as he knew of numerous failures which had been revealed quite recently.

There was one very puzzling aspect of these failures to him as a metallurgist which he hoped an engineer could clarify for him, and that was the position of the failures.

In the lower thread the failures were usually in the first thread adjacent to the crosshead, as one would expect from the distribution of the load carried by the nut. However, in the upper thread, the failures were usually in position D, Fig. 8, and he had never seen one in position C, which would seem more likely than D due to the distribution of load on the nut. He realized that truncating the thread would give a better load distribution and reduce the stress concentration at C, but he did not think that with the pre-tension loads normally applied this would shift it from C to D.

It was interesting, therefore, to find that in the experiments conducted by Mr. Wiene fractures were only produced at D when, as stated on page 45, the pre-tension was sufficient. What was this pre-tension?

If these failures were caused purely by direct stresses, then failure might also be expected from the bore of the oil cooling hole outwards to the surface. The bore of this hole was usually quite rough and in his opinion contained more severe notches than the thread, which was carefully machined and rolled. Also thread rolling would cause an additional small residual tensile stress on the bore of this hole. Yet such failures had never been reported.

The appearance of the fracture surfaces, where fracture actually occurred, was typical of bending fatigue, i.e. a crack started at one side of the rod and proceeded across it until the final short time break which was on the opposite side. Direct stress might be expected to produce cracking all round the circumference and a final fracture towards the centre. Again, this had never been reported.

Was it possible that in addition to the direct stresses there was also an alternating bending stress on these rods? He had seen engines in which failures had occurred where locating pins between parts of the cylinder blocks had been badly deformed and almost sheared as though there had been relative movement between them; also semicircular markings on the crosshead face, as though the rod had been rocking to and fro.

He thought it would be worth while to carry out strain gauge tests on a rod in service to determine whether it was subject to any bending stress.

Mr. J. H. Milton (Member) said he would like to ask what might appear to be two very elementary questions.

First, it was stated that in the case of double acting engine piston rods a series of tests with slackening and tightening of the nuts showed that a considerable setting took place and that instead of tightening and slackening four or five times, the rod might first be given a somewhat higher stress than that which was to be used in practice. Was it to be assumed that this was to take up slight errors in machining?

Secondly, the author stated that during running, wear and setting took place—apparently more setting—and it was therefore dangerous to place fixed tightening marks on nuts and crossheads, as the pre-tension thus became continuously less. If this were the case, would there not be a limiting time after which it would be dangerous to run, as the pre-stress could become lower than the stress due to gas load, thus allowing surface separation to occur at the crosshead mating faces?

Mr. J. F. Chambers said that as one who had spent much time in the production of cast crankshafts, his interest was

‡ Horger, O. J. 1954. "Fatigue Characteristics of Large Shafts", "Fatigue", Publ. A.S.M.

§ Peterson, R. E. and Wahl, A. M. 1935. Trans.A.S.M.E., Vol. 57.

* Kenneford, A. S. and Ellis, G. C. 1950. Journal of the Iron and Steel Institute, Vol. 164, Part 3.

in Mr. Wiene's experiments on the influence of fillet radii and web form on crankshaft fatigue strength.

Pershaps he should say first, however, that his own tests confirmed his statement that surface finish had no appreciable influence on the fatigue strength of cast iron, and they had also had higher results on rough turned than on smooth bars.

In this country much work had been done by the Motor Industry Research Association on the influence of many features of design on the fatigue strength of cast crankshafts. Their results did not show the same strength increases as were shown by Mr. Wiene.

Taking the case of fillet radius, M.I.R.A. results indicated that an increase of 25 per cent in bending fatigue strength would be expected by an increase in radius from 2 mm. to 4 mm. (Mr. Wiene had reported practically no improvement), and that the strength increased proportionately with further increase in the fillet radius.

Taking the size of the shaft that was illustrated, a normally formed radius of 8 mm., if it could be accommodated, would on their work increase the strength by about 100 per cent over that of a 2 mm. radius. Increasing the radius to 8 mm. by undercutting into the web, however, would not be expected to give better strength than a normal 4 mm. radius. He was at a loss to understand the difference in these two series of experiments.

On shaft design, it was agreed that hollow bearings and crank pins increased the strength as compared with a solid shaft, and the optimum size of hole no doubt varied with the proportions of the shaft. The best result appeared to be obtained when the bore was approximately 50 per cent of the bearing diameter. Larger bores, recesses in the webs, and so on, did increase the flexibility of a shaft, but M.I.R.A. results showed practically no increase in the limiting fatigue strength by any of these variations. Such increased flexibility could probably be of value, however, in avoiding high stress under conditions allowing bending, say by worn bearings. But their results showed practically no increase from any of these forms of variation.

On surface treatment, Mr. Wiene mentioned the effect of shot-peening on steel springs and the increase in strength obtained as a result of the compression stress imposed on the surface layer. Again, quoting M.I.R.A., he would like to mention the effect of an imposed compression stress on the fillet of cast or forged crankshafts. M.I.R.A. cold rolled the fillet by the use of steel balls under heavy pressure, and obtained an improvement in strength of about 60 per cent under reversed bending, applying equally to forged and cast crankshafts.

Broadly speaking, M.I.R.A. results could be summarized by saying that attention to the size, accuracy and surface treatment of fillet radii appeared to be the only way to obtain significant improvement in the bending fatigue strength of cast crankshafts.

Mr. Wiene indicated that by using meehanite shafts of suitable design the same margin of safety could be maintained as with normal shafts of forged steel, and that his company had substituted the cast shafts without alteration in design. This was rather a sweeping statement.

Could Mr. Wiene say what was the nominal bending stress in the engines in question? If their experiments had been carried to the point of failure of the crankshafts, what did he regard as the maximum safe value of the nominal bending stress for specific types of cast iron?

Finally, he would express an opinion, which he believed was shared by Mr. Wiene, that suitably designed crankshafts of high duty iron could be of very great value to the builders of Diesel engines of quite a wide range of size.

Mr. B. K. Batten, M.Sc., said that in discussing the sharpness of notches the author remarked upon the slow development of some incipient cracks in piston rods which, as he rightly said, must be regarded as being very sharp notches. During some fatigue tests carried out at the research laboratory of Lloyd's Register of Shipping, it was noted that with sharp notches, whose natural stress concentration factor approached that of a crack, there were many shallow cracks formed round the notch which subsequently joined into a major crack, propagating rapidly to cause failure.

With large notches having a low stress concentration factor, however, only one or two cracks formed before failure.

This led them to offer a tentative theory regarding the mechanism of crack propagation. The presence of a sharp notch in a specimen gave rise to an area of stress concentration, and that might subsequently lead to the formation of a small crack having a higher stress concentration factor than the notch. This crack, however, was surrounded by work-hardened material which had a slightly higher ultimate tensile strength than it had in its unhardened state.

Furthermore, the nominal stress at the end of the crack had now been reduced by virtue of the depth of penetration of the crack towards the neutral axis of stress. Thus, at some stage during the propagation of the crack the resistance of the strain-hardened material surrounding it would be sufficient to stop the progress of the now less highly stressed crack extremity. Then the stress concentration effect of the original notch became again of primary significance.

With regard to crankshaft design, the removal of the pin and journal fillets into the web was still a matter of some controversy, though it would be agreed that greater flexibility of a web might be achieved.

One of the drawbacks of this design was that it seemed to offer greater opportunities for corrosion to occur which, of course, had the immediate effect of introducing surface cracks into the fillet. He would be interested to hear Mr. Wiene's comments on this.

Mr. Wiene had dealt very thoroughly with the problem of the tightness and design of bolts, with special reference to those for piston rods. The relation between the elasticity of the bolt and of the assembly was of some importance, and in the paper* given to the Institute by Mr. Bunyan it was suggested that the bolt should be at least four times more elastic than the bolted assembly. A vital point in any bolted assembly was to avoid introducing bending stresses in the bolt due to faulty fitting of abutting faces. This might be done by relieving the area in contact except in the immediate vicinity of the bolt.

The results of the tests on welding were interesting. His own experience of the fatigue strength of welded specimens was small, but preliminary tests had indicated that in the torsional fatigue of mild steel shafts the presence of a welding repair in the way of the notch could lower the torsional fatigue strength by as much as 25 per cent.

* Bunyan, T. W. 1955. "Practical Approach to Some Vibration and Machinery Problems in Ships". Trans.I.Mar.E., Vol. 67, p. 99.

Correspondence

Mr. R. W. Cromarty (Member of Council) wrote that Mr. Wiene's presentation of his paper and the subsequent discussion were listened to with great interest. It was felt that the following information about two-stroke double acting engine piston rod recent failures might be of some interest to members who lacked this type of experience and give them the background as to why tests such as Mr. Wiene presented under the heading "Piston Rods" were necessary.

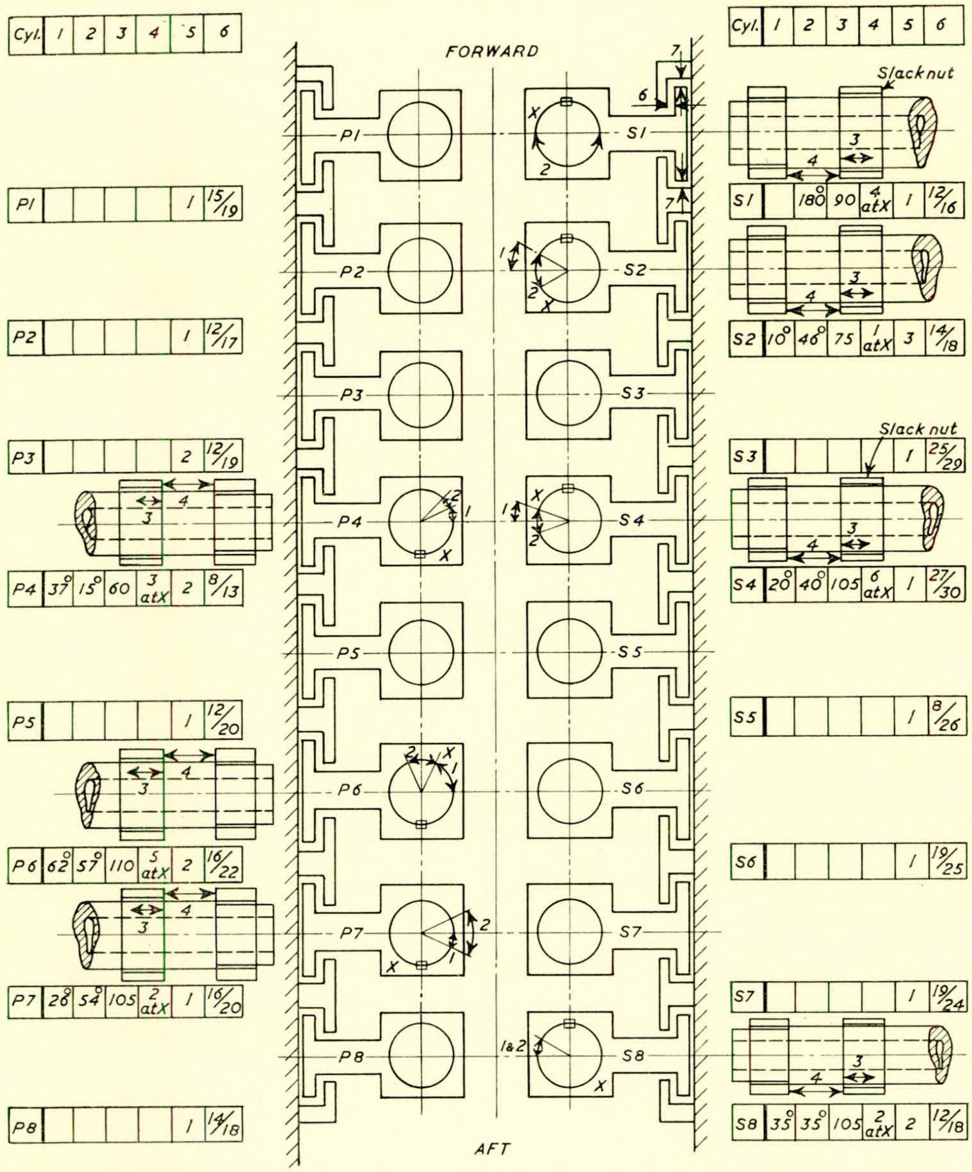

FIG. 15—*Twin screw double acting two-stroke engines: piston rod fractures and guide clearances*

(1) Angle from thwartships centre line
(2) Angle of fracture (rod has 65-mm. diameter hole)
(3) Height above top nut bearing face, in mm.
(4) Error in parallelness of nut bearing faces in 1/1,000 inch
(5) Error in parallelness of crosshead bearing faces in 1/1,000 inch
(6) Thwartships maximum and minimum guide clearances in 1/1,000 inch

Fig. 15 summarized the information. The ship had twin screws and the main engines were each of eight cylinders. The crossheads with their screwed piston rods were of the design shown in Fig. 8 in the paper, except that the nut bearing faces were relieved as shown in Fig. 16.

Seven piston rods were found fractured, all in way of the top nut. One of these rods (S.1) had been in service 23,823 hours and the other six 25,468 hours when, at the special survey last year, the fractures were discovered by the magnetic flux method. The S.1 (Starboard No. 1) rod, in addition to the fracture recorded in Fig. 15, had two other fractures directly under the centre of the top fracture, each approximately two inches long and at heights approximately 70 and 50 mm. above the top nut bearing face. Both the top nuts of S.1 and S.4 were found slack on the piston rod threads.

The original piston rods in this ship were made of an alloy steel, but were removed some four years ago and mild steel rods fitted.

It might not be appreciated from Mr. Wiene's paper that only the top nut was used for tightening (i.e. putting on pre-tension), because the fork of the connecting rod would not permit easy access for tightening the bottom nut. The method used with nuts that had been previously tightened was:—

(*a*) One man using a 2·0 metres long spanner tightened the top nut, using his full weight.
(*b*) The position of the top nut was then marked.
(*c*) Using a screw type tightener, the top nut was further tightened a calculated number of degrees, or to give a calculated elongation.

In considering the fractures, it would be noted that they occurred at different heights and different angular positions.

The different heights were probably due to slight errors in the threads, which were rolled, after being formed by grinding.

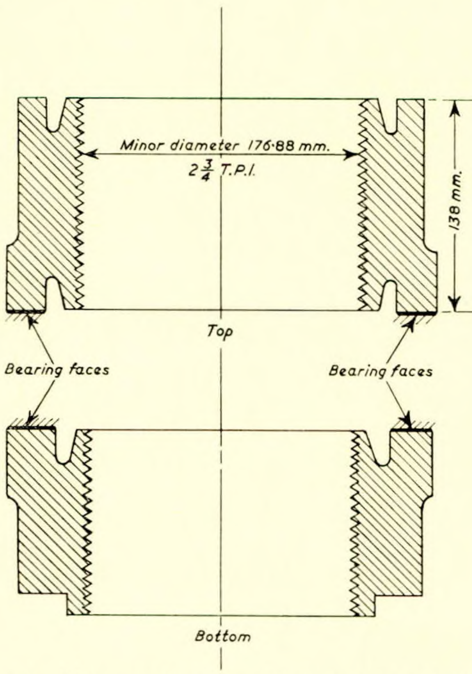

FIG. 16—*Piston rod nuts. Thread form similar to that shown in Fig. 8*

The different angular positions had been thought to be connected with the guide clearances which, by permitting crosshead movements, caused bending of the rods, resulting in the fractures appearing at various angular positions. A careful check of the guide clearances was made and these were summarized in Fig. 15. From a study of these there seemed to be no reason why they should be a major cause of the short life of these rods, and this was supported by Mr. Wiene's experiments, which produced similar fractures with only "push-pull" loading. It might at first appear that some of the clearances were excessive, but Mr. Arnold in his remarks struck a note of warning by relating his unfortunate experience when he reduced the clearance to less than 12/1,000 inch. Mr. Wiene did not give the length of his 36-mm. diameter test piece piston rod. The piston rods in the engines were about 16 feet long and about seven inches diameter with an approximate $2\frac{1}{2}$-in. diameter hole. Applying Euler's formula for long struts, the breaking, or buckling, load for these rods was some 530 tons. The gas load on the pistons alone was about one-fifth that of the breaking load. The margin for eccentric loading due to excessive guide clearances, inertia forces, temperature and pressure stresses, and possible stresses due to unhomogeneous material and faulty heat treatment, was not generous. However, it was considered that 20/1,000 inch would be a reasonable maximum working clearance, but Mr. Wiene's views would be appreciated.

Next, the nuts were placed on the rods at the position in which they would be when tightened up on the crosshead, and the distance between the bearing faces measured at four points 90 degrees apart. The maximum difference in this reading and its approximate position on the crosshead was shown in Fig. 15. At S.1, with a slack top nut, a higher reading was obtained, but whether this was original was not known and so the smaller reading was recorded. Again, a study of this data revealed nothing significant.

The distances between the nut bearing faces on the crossheads were similarly measured. The figures shown also represented within 2/1,000 inch the total depth that the bearing faces were found set in below the adjacent surfaces. The surfaces outside the bearing faces were also measured and it was found that they were quite parallel. Again, nothing significant was found.

Mr. Wiene would agree that these unexplained failures were a great anxiety to the shipowner, especially as the builders were also unable to find a solution. Replacing these rods after some 20,000 hours' service was a costly business and it would be most helpful if Mr. Wiene could give the fullest information of how he cured the failures that came his way in the immediate post-war years. Other speakers in the discussion stressed the point that piston and tie-rod failures were still prevalent.

The only crumb of comfort thrown to them by Mr. Wiene was that the life of such rods might be extended by cutting out the fractures when they were discovered at 12-month inspections, but this, of course, was no real satisfaction to anyone, nor were his suggested fancy threads, rolled threads, relieved nuts and the like.

A design similar to that shown in Fig. 8 was not a success in steam reciprocating engines. The standard type piston rod in these engines had a coned end fitted into the crosshead with a taper of about 1 in 5 and held in place by a nut. The stress on the area at the bottom of the thread on the rod rarely exceeded 7,000lb. per sq. in., so the stress at the top of the taper at the top of the crosshead was considerably less than that given by Mr. Wiene.

Did not the solution to the problem lie more in the direction indicated by Mr. Arnold, who had reduced the bore of the hole in these rods—that is, redesign the rod and its fastenings to reduce the stress? Especially as the above engines were run at 106 r.p.m., that is, at 80 per cent of their designed power, and would have to be run considerably below this to extend the life of these rods.

It was suggested that reducing the stress in working parts might also be the answer to crankshaft failures, of which there appeared to be an increasing number. Cutting fillets into the webs as shown in Fig. 5 thinned the web and looked wrong. Whilst in a laboratory results might be obtained which indicated some improvement in strength, there would be little doubt in the minds of those responsible for the reliable running of marine machinery in service that these fillets would eventually result in the web fracturing at this thinner section, especially as the design tendency during the years had been to thin the webs to shorten the engine length.

Would Mr. Wiene agree that the thinning of working parts in Diesel engines to reduce weight and overall dimensions had now reached beyond the stage of the extra "straw that broke the camel's back"?

This excellent paper with its frank admission of service failures was most refreshing after the negative attitude one encountered when service troubles were brought to the notice of the manufacturers.

MR. C. E. PHILLIPS considered that, though Mr. Wiene's paper contained no ideas which were entirely new, the collected investigations which had been carried out in Denmark by or on behalf of Burmeister and Wain were of great interest. The experiments on nitriding, for example, provided confirmation of earlier work in this field reported by Hengstenberg and Mailander* and by Sawert†.

The paper* reported tests on specimens with and without nitriding, and showed that nitrided specimens tested in rotating bending were completely insensitive to the presence of a 1-mm. deep V-groove with a very sharp root. The fractures were reported as occurring away from the notch. Sawert did not report on notched specimens but gave an interesting comparison between the effects of nitriding in direct stress, bending and torsional fatigue. In the latter two cases a 30 per cent improvement in fatigue strength was shown, but nitriding apparently had no effect on the direct stress fatigue limit. In view of

* Hengstenberg, O., and Mailander, R. 1930. "Fatigue Strength of Nitrided Steels". Krupp Mh, Vol. 2, p. 252.

† Sawert, W. 1943. Z.Ver.Deutch.Ing., Vol. 87, p. 609.

these findings the subject of nitriding and its effect on notch sensitivity would seem to be one which would well repay further study, since it was difficult to envisage the mechanism by which the stress concentration effect of a notch was completely suppressed by the introduction of surface compressive stresses, particularly as one would expect these to be accompanied by sub-surface tensile stresses.

At the Mechanical Engineering Research Laboratory they were also very interested in the fatigue tests carried out on the notched specimens of 0·2 per cent carbon steel, in which it was found that notches with a root radius of 0·1 mm. gave the lowest fatigue strength, and reducing the radius below this value resulted in an increase in strength. This particular problem had been investigated there in some detail. As the author stated, it was not at present completely understood why it should be so, but there appeared to be no doubt that a critical root radius existed for any given material. Decreasing the root radius below this value, i.e. increasing the sharpness of the notch, brought about an increase in fatigue strength. Work here had suggested that this behaviour was connected with the formation of non-propagating cracks at the roots of notches; it had been shown that with a root radius less than the critical value, the material at the notch root broke down at a very early stage in the fatigue life to form a crack, but that the crack, after extending for a certain distance into the material, did not appear to propagate if the alternating stress conditions remained unaltered. Rotating bending and direct stress fatigue tests carried out on specimens in which a fatigue crack had previously been formed had shown that the strength reduction factor due to a fatigue crack was no greater than that due to a mechanically formed notch. Under conditions of completely reversed stress (i.e. zero mean load) cracks or very sharp notches could have a less damaging effect as regards fatigue strength than notches which were less sharp at the root. It must be emphasized that the condition that the mean load was zero was very important; they were finding that the superimposition of mean tensile loads could bring about very different results.

The effect of truncation of threads found in the experiments with piston rods agreed to some extent with their own work on screw threads, though the extent of the apparent increase in strength reported in the paper was surprising in view of the amount of truncation. Rolled screw threads were, of course, stronger in fatigue than cut threads; with care, a doubling of the strength could be achieved.

Since they had never experienced a nut failure when using a normal sized nut, they were surprised that Mr. Wiene found the use of special steel nuts necessary to achieve the reported improvement in behaviour. It was also very difficult to account for the initial low fatigue strength of the piston rod threads.

Mr. P. Pluys wished to add a few remarks to Mr. Wiene's very interesting comments on the problem of piston rod failures of double acting Diesel engines.

He felt sure that the suggestions made in his paper would be welcomed by the many shipowners who unfortunately did not consider the question solved! It was probably right to say that most of the failures occurring in the bottom nut were due to a defective pretightening of the assembly. Even after several applications of a pre-tension 20 per cent higher than the final setting, it was interesting to observe that the real elongation of the piston rod was only about 30 per cent of the theoretical value calculated from the relative elasticities, the nut rotation angle and pitch of the thread. Moreover, if the remaining pre-tension was measured after a few months of service, it was found that it was still further reduced. This showed that it was imperative to increase as much as possible the elasticity of the rod relative to the crosshead and in this connexion much could be done by increasing the tightened length and decreasing slightly the rod cross area. It should also be advantageous if the horizontal boring through the crosshead pins and body could be avoided, so that the maximum rigidity could be achieved between the two nuts. In any case the only way to be sure of a given pretightening for such an assembly was to check the elongation which remained after a certain period of service and which should always be enough so as to avoid any slackening under firing pressures.

The top nut was, of course, in a worse condition because the full tensile and compressive stresses due to the firing load were applied to the first top threads. Moreover, it had been proved that bending stresses due to wear in the engine were superimposed and played a definite role in the failure process. Whatever could be done in the design of machining of a thread, he felt sure that the stress concentration would still be higher than in a well-designed conical seat with ample radius.

A proposal taking into consideration the preceding remarks

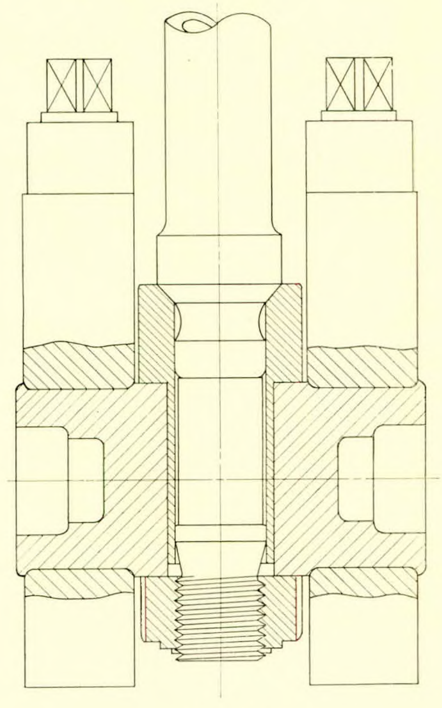

Fig. 17

was given, in Fig. 17, drawn to the same scale as Fig. 8. It introduced between the piston rod and the crosshead an intermediate block with a conical seat, made of harder steel. It should be mentioned that it could be applied to existing engines with a "double-nut" system without any machining to the crosshead. Such a solution had been tried on a ship in service over the past five months, and he would very much appreciate Mr. Wiene's comments on this design.

Mr. Bryan Taylor, B.Sc.(Eng.) (Member) considered that the work described in the paper was most valuable to engine designers, and the author and the company with which he was associated were to be complimented on releasing this information.

He wished first to comment on the results plotted in Fig. 3. In this diagram it was understood that the points indicated an endurance of 5×10^6 cycles, but it was not clear to him how the stress range corresponding to this endurance was obtained for each specimen tested. It appeared that some extrapolation of the results must have been adopted, and if so, this would seem to leave the accuracy of the plotted points open to some doubt.

He was also interested in the results plotted in Fig. 3 for threaded specimens, and assumed from the fatigue strengths obtained that these were test pieces of the type shown in Fig. 1, but with a thread cut in the central portion, and not specimens in the form of bolts and nuts. The author's confirmation or comments on this were sought.

Referring to the remarks concerning the failures of piston rods, which appeared in the section dealing with the sharpness of the notch, the writer was glad to have further confirmation that the spread of a fatigue crack could take place relatively slowly. He suggested that a possible explanation of this was that in large parts such as those in question, there was an appreciable stress gradient in the region of the point of maximum stress concentration. The progress of the crack through this surface layer must be rapid, but on reaching a zone of low stress the propagation of the crack would be retarded. Another point in this connexion was that a stress sufficient to cause further penetration of the crack might have arisen only infrequently and under exceptional conditions, thus allowing incipient fatigue cracks to be detected by examinations carried out at intervals in as much as one year.

The section dealing with piston rods was of the greatest interest and he was glad to note the emphasis placed by the author on the importance of an adequate prestress in screwed connexions. He agreed that the estimation of prestress by measuring the angle of nut rotation was liable to be misleading; in the case of assemblies of moderate size, however, he considered that this method could give reasonably consistent results, provided the point from which the angle of rotation was measured was determined by pulling-up the nut reasonably tightly by hand.

Again, referring to the modifications to piston rods it was noted that on reducing the depth of thread on the rod, alloy steel nuts were substituted to prevent fatigue cracks forming in the nuts. It would be of the greatest interest to learn whether the author actually observed failures in ordinary carbon steel nuts, as the writer failed to see the necessity of altering the material. It would seem that truncating the thread on the rod merely ensured that the bending stress on the thread roots due to the cantilever effect of the thread was not increased beyond that which should obtain in threads with accurately formed profiles.

In connexion with the tests made on specimens with rolled threads (or perhaps this should be termed after-rolled threads, since it was assumed that the rolling took place subsequent to normal thread cutting), he enquired whether the author could give some further information on the method of determining a suitable degree of rolling.

The increase in fatigue strength of 20 per cent obtained by stress relief annealing in the case of the welded specimens was a marked improvement. This might well underestimate the gain to be obtained by such treatment, however, as in most fabricated assemblies the plates were restrained in several directions. He would be glad to have the author's opinion on this matter.

MR. A. VANDEGHEN thought Mr. Wiene's very interesting paper gave important data about the problem of the piston rods of double acting two-stroke engines. These piston rods had certainly given trouble to all the builders of this type of engine.

In the B. and W. double acting coverless engines, of 59 cm. bore and 125/45 cm. stroke, which the writer's company had built for the Compagnie Maritime Belge, a few cases of piston rod cracks occurred in recent years, some at the lower thread, and most of them at the upper thread.

Like Mr. Wiene, he thought that the lower thread cracks were due to an insufficient pretightening. A series of measurements of the elongation of the rods were made at the C.M.B., and it was found, in accordance with Fig. 9 of the paper, that the real elongation, after first tightening and slackening, was only about 65 per cent of the elongation as theoretically calculated from the angle of rotation of the nut. Accordingly, his company recommended higher tightening angles than previously.

He thought that the problem of the top thread was more difficult because the upper part of this thread was running, without pretightening, under alternate stresses. Further, there were often at this place some bending stresses, maybe due to an asymmetrical distribution of the pressure around the crown of the main piston during the combustion period.

He believed that the best solution was to suppress the

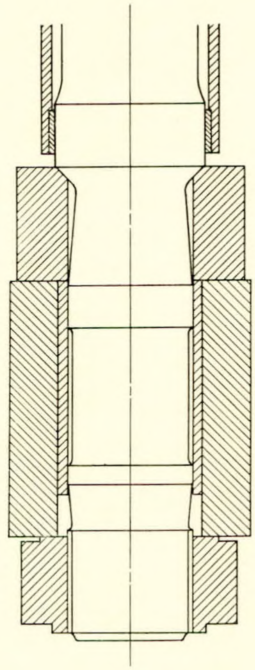

FIG. 18

upper thread. Consequently, his company had designed a rod with a lower thread only, and with an upper conical abutment, as shown in Fig. 18.

Another advantage of this design was that the length under tightening was about 30 per cent greater than previously, which decreased the loss of tightening by setting. This device could easily be adapted on existing crossheads. A rod of this type was now being tested in an engine of the C.M.B.

He agreed with Mr. Wiene that in big engines the small bolts were often overtightened. They used torquemeter spanners for all the small bolts that were important, and their operator's manuals contained data on the adequate torques.

Author's Reply

The author wished to thank everyone who had taken part in the discussion, and drew attention to the fact that the subject of the lecture was extremely wide and consequently the oral and written comments and questions were so numerous (covering more than fifty problems) that a thorough answer to them all would fill a complete book with a theoretical as well as a more practical consideration of the complete doctrine of fatigue problems. Because of this he would not endeavour to answer each individual question, but would go a little deeper into some of the more essential points.

Several questions dealt with the effect of nitriding. Fig. 19 showed approximately how it was imagined that stress was

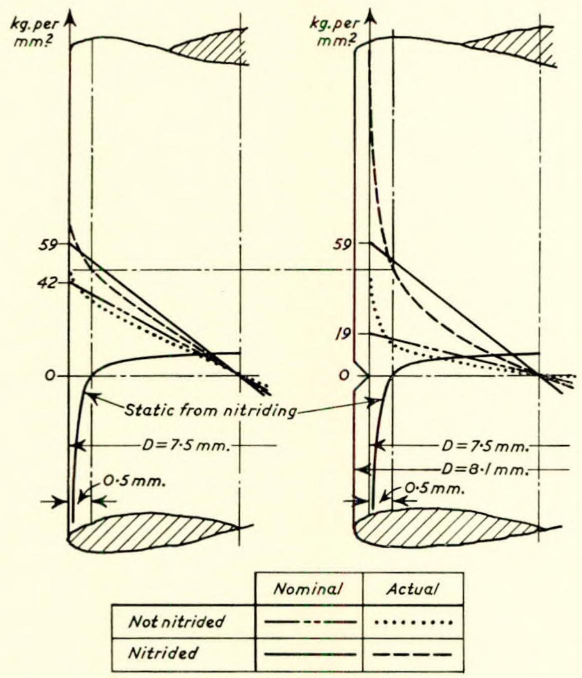

FIG. 19—*Suggested stress distribution at rotating bending tests with non-nitrided and nitrided material (as stated in Fig. 1, numbers 6 and 7); polished rod to the left and notched rod to the right. The crack starts when the stress in the non-nitrided part passes 48 kg/mm.², i.e. on non-nitrided rods from the surface, on nitrided rods 0·5 mm. under the surface.*

distributed on a smooth and a notched 7·5 mm. rotating bending-test piece both in the non-nitrided and the nitrided condition. In the smooth test, to the left, a nominal bending fatigue strength of 42 kg/mm.² was measured, calculated on plane stress distribution. Because of unavoidable minor faults in the surface the actual stress was somewhat higher, e.g. 48 kg/mm². Through nitriding, a very large static pre-compression was produced on the surface to a depth of approximately ½ mm. Therefore, the outer surface layer was able to stand a considerably higher tensile stress; but already at a nominal stress of 59 kg/mm.² on the surface the actual stress distribution would have the effect that the stress in the outer part of the non-nitrided core, 0·5 mm. under the surface, reached 48 kg/mm.², which was the real fatigue strength of the non-nitrided material.

The notching on the right of Fig. 19 had the effect that the nominal stress at the base of the notch dropped to 19 kg/mm.² in the non-nitrided sample, while the actual stress was still 48 kg/mm.², giving a notch factor of 2½. Nitriding added the large static pressure at the base of the notch, so that the material could stand far higher varying stresses; consequently, it again became the actual stress in the border layer between the nitrided and non-nitrided zone which determined the beginning of the crack, and when this stress reached 48 kg/mm.² the break commenced, which again gave a nominal stress of 59 kg/mm².

It appeared from this explanation that the useful effect of nitriding was only apparent if the varying stress on the surface were considerably higher than in the core of the material —either because the strain was due to bending or torsion or because of notching. No improvement had been obtained, according to Mr. W. Sawert's article referred to by Mr. Phillips on page 53, by nitriding a completely faultless test piece on a pure push-pull test. However, such cases did not occur in practice. It was of greater practical importance that this explanation showed that nitriding would only have a slight effect on machine parts which were so big that the nitrided layer was negligible compared with the dimensions.

As the effect of shot peening was analogous to the effect of nitriding, the above pointed to the fact that surface cracks in springs, which were comparatively small compared with the thickness of the peened layer, were rendered harmless by the peening—which answered one of Mr. van Asperen's questions.

The author agreed with Mr. Todd that the spring material was of some importance among other things because of decarburization, but he preferred to deal with this by saying that such precautions as were necessary for the material actually used must be taken to avoid decarburization.

Fig. 20 showed push-pull tests carried out at the Laboratory for Building Technique of the Danish Technical High School by a high frequency Amsler testing machine, rod diameter 8 mm., from which the importance of tensile- or compression *pre-stressing* appeared; this principle was the basis for the use of nitriding, shot peening and rolling against fatigue.

The Danish experiments, given in Fig. 3, were carried out in 1946 but were considered dubious until practical experience with piston rods in use confirmed the tendency, which had again been confirmed in the article quoted earlier[3] and by Mr. Phillips's and Mr. Taylor's written contributions to this discussion.

As far as the author understood Mr. van Asperen, he was of the opinion that cracks were formed during the machining and not observed during the inspection. In Fenner, Owen and Philips's experiment[3] it seemed, however, that the bottom of the notch had been controlled before or at the beginning of the fatigue test, as a more or less exact

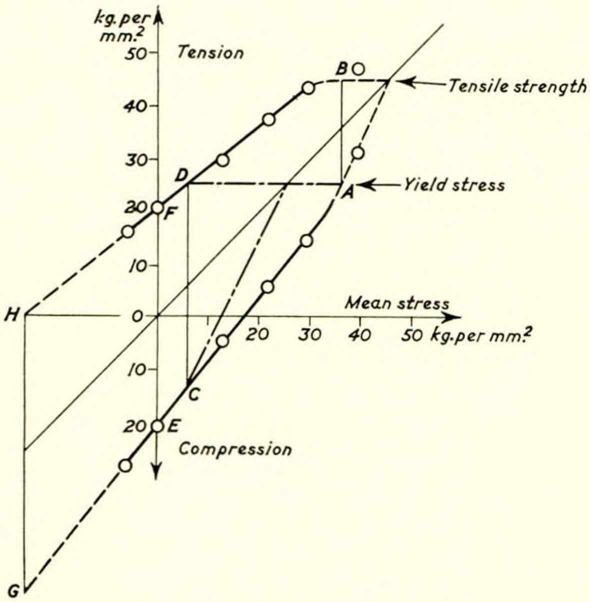

FIG. 20—*Smith-diagram indicating push-pull fatigue strength for 0·20 per cent C-steel with varying pre-stressing. Increased tensile pre-stressing reduces the varying stress which can be tolerated, but considerable stress can be tolerated far above the yield point and even varying stresses which momentarily reach the ultimate tensile strength can be tolerated more than ten million times. For practical reasons the experiment has not been continued as regards greater compression pre-stresses than 10 kg/mm.², but it is known from other experiments that the fatigue strength is constantly increasing with increasing pre-compression.*

moment could be stated for the appearance of the crack. Furthermore, one of their micro-photos showed that it was a question of a transcrystalline fracture and, therefore,. probably a real fatigue fracture. At any rate, the author had found minor cracks in piston rods which were not there before the piston rod was put into service and which apparently developed very slowly.

Mr. Batten's interesting observation, that a short crack in a sharp notch rather slowly elongated itself, whereas a crack in a large notch once formed more quickly continued half or the whole way round, might also be explained by the fact that under a sharp notch the stress decreased quickly so that the material, for instance, in 0·1 mm. depth, had been very little damaged and therefore needed many impulses before the crack continued to the depth—and some depth was necessary to make it continue in length if for some reason (a small slag inclusion, for example) it had been stopped momentarily at the very surface. Under a large notch the material in 0·1 mm. depth had nearly the same high stress as in the surface and when that was multiplied by the notch factor from the crack (which was much higher than from the large fillet) the stress was so high that the crack continued very quickly in depth and, therefore, also in length. Similar points of view had been stressed by the author at the Colloquium in Stockholm*. The author did not really understand why the nominal stress at the end of the crack should be reduced by virtue of the depth of the crack and he would expect the work-hardening effect to be the same with the large notch as with the sharp notch.

The author was thus in agreement with Mr. Taylor as to the importance of the stress gradient and also regarding the possibility that the cracks might appear in special circumstances, such as during starting, when the stresses were higher than during normal running.

It struck the author that there was some risk in employing Mr. Smedley's formulæ as they were purely empirical and based on a limited number of measurements; as remarked under (*c*), the curves gave a faulty result of extrapolation upwards, while the two points in Figs. 12 and 13 which were stated as radius nil, were poorly defined, as it was by no means unimportant whether this radius was 0·01 inch or 0·001 inch. Mr. Smedley himself also drew attention to the fact that the formulæ did not apply to the second experiment made by Lehr and Mailander or for other sorts of notches, threads or holes, etc. There was a better approach for fairly large radii through Bach's formulæ for curved beams. The author had had opportunity to test Timoshenko's formulæ† for large interior holes in thick cast iron by strain gauge measurements and actually found them completely confirmed. It had been found through many other experiments that Neuber's formulæ‡ also held good in many simple cases and for not too small fillets. The problem really started by complicated constructions or quite sharp notches for which no adequate theory was found.

Mr. Taylor was correct insofar as the experiments mentioned in Fig. 3—with thread—had been carried out with test pieces, as shown in the lower part of Fig. 1, but with a full thread instead of a single notch. However, the samples marked with a cross in Fig. 3 were taken as direct stress tests while those marked with a circle in Fig. 3 and all those in Fig. 1 originated from rotating bend tests.

With regard to crankshafts, the author was not able to explain fully the different results obtained by M.I.R.A., as referred to by Mr. Chambers, and himself. The increase in fatigue strength to nearly double by an increase in fillet radius from ⅛ inch to ½ inch found by M.I.R.A. seemed very high; it was nearly in accordance with the theoretic stress values but fatigue usually gave lower effects than were expected theoretically; the author admitted that his corresponding values were astonishingly low so that the truth might lie somewhere between the two results. As to the fillet cut into the web, the author thought that his result with a steel shaft was more reliable than M.I.R.A's with a cast shaft, as a big fillet cut into the web of a cast shaft brought the highest stresses into material originally cast near the centre of heavy parts which was not absolutely sound; therefore, the M.I.R.A. experiments produced poor results from this construction, which, theoretically and according to the author's experiment, should be satisfactory.

As to Mr. Cromarty's impression that thinning of the web (Fig. 5) looked wrong, one must remember that thinning the web in itself increases the stress but increasing the fillet radius decreases the stress and there was then a possibility of an optimum degree of cutting into the web. This point was first found in calculating according to the old Bach's formulæ for curved beams and it seemed to be confirmed by the experiment shown in Fig. 5.

The author did not remember to have seen corrosion at the big fillets in the webs and was at a loss to understand why they should be worse than small fillets in that respect, as Mr. Batten seemed to fear.

Th author was fortunate to be able to say that in marine engines built by his firm and its licensees crankshaft failure had been no problem at all. He assumed therefore that Mr. Cromarty's remarks about the strength of the crankshafts referred to cases where special circumstances pertained.

As regards the piston rod connexion, this had aroused the largest number of comments and enquiries, from which the author understood that fractures were still occurring at this point, he would add the following information.

The experiment was carried out with push-pull-loading (Mr. Moore) and the free length of the piston rod was 260 mm. (Mr. Cromarty).

A setting would take place during the first tightening of a new nut even if the threads had been accurately made

* International Union of Theoretical and Applied Mechanics, Colloquium on Fatigue, Stockholm, 25th-27th May 1955, Springer Verlag, 1956, p. 318.

† Timoshenko. 1934. "Theory of Elasticity", pp. 76-79.

‡ Neuber, H. 1937. Kerbspannungslehre. Berlin.

because the material at the bottom of the threads nearest to the crosshead was affected over the yielding point, but naturally this effect was increased occasionally because of unavoidable machining inaccuracies. There was hardly any danger, as Mr. Milton seemed to fear, that the pre-tension would fall unduly under the normal working of the engine as the fixing of the tightening angle must be done with sufficient safety to allow for ample time between overhauls of the piston rods.

When fracture occasionally took place in cross section A (Fig. 8) and never in cross section C—neither during the experiment nor in practice (as far as was known to the author, and in response to Mr. Todd's question)—this was probably due to the fact that the crack, as already mentioned, started at the point where the local bending stress from the thread joined the direct stress in the cross section of the rod. An upward pull in the rod would, in cross section A, increase these two stresses simultaneously, while in cross section C it would increase the direct tensile stress but decrease the bending stress in the thread, so that the actual variable stress near the base of the thread at C in points with tensile pre-tension was considerably smaller than near A.

In the higher situated threads of the top nut, conditions were extremely complicated, as a push on the rod had the effect that threads, which from the tightening-up were hardly touching, were now taking over a greater part of the loading. Through a subsequent pull these threads were completely relieved, while those placed farther down took over a greater part of the varying load; thus, in this way, none of the threads took a complete stress variation corresponding to the full variation between push and pull: this was probably why the top thread of the piston rod was at all able to stand the load.

This addition of the bending stress in the individual threads and the tension-compression on the cross section itself also explained why the cracks never started on the inside of the oil hole where there were—as remarked by Mr. Todd and Mr. Arnold—much coarser machining marks than on the outer side of the rod, but no extra bending from the thread and still less bending stresses from the crosshead shoe and misalignment, of which more will be said later.

At the same time it explained the observation made by both Mr. Cromarty (Fig. 15) and the author that fracture at the top thread within the upper half of the nut could occur at varying heights—extremely small variations in manufacture decided which thread would have the heaviest varying stress.

The intention with the modified shape of the thread (Fig. 8 at the right hand bottom corner) was just to reduce the effect of this bending on the rod as the point of attack of the force from the thread of the nut was moved closer to the thread base on the rod so that the stress should be lessened—this answered questions from Mr. Falconer and Mr. Taylor. Three-dimensional photo-elastic stress analysis was not in use in Denmark when these experiments were undertaken about ten years ago; it was therefore estimated that fatigue tests would lead to a reliable result more quickly, which at that time was important.

In a few cases the setting of the threads had been measured over ten to twelve years. An average setting was found of 1-deg. turn of the nut annually, corresponding with a reduction of stress of approximately 0·7 kg/mm.^{2} annually, assuming that the nuts were steadily tightened to fixed points on the crosshead. The author's firm used the same tightening method as given by Mr. Cromarty.

Naturally, it was correct as pointed out by Mr. Moore that a pre-tension of 5 kg/mm.^{2} was slightly too small for a varied stress of $\pm$ 6·7 kg/mm.^{2} but it was not discovered until the experiment was being carried out that the actual pre-tension was so much lower than had been calculated (Fig. 9). It was normal procedure to aim at a pre-tension which did at least equal the greatest tensile stress and the experiment proved this to be correct. Mr. Taylor's suggestion to fix the starting point for the angle measurement from a point fixed through pulling-up by hand was precisely what the author wanted—as against tightening to fixed marks on nuts and crossheads. As regards the fatigue strength, no inconveniences were discovered through an increase in the pre-tension to 20 kg/mm.^{2} (28,000lb. per sq. in.); that in practice one preferred a somewhat lower pre-stressing, e.g. 12 kg/mm.^{2}, which was almost double the tensile stress ever occurring, was due to a fear of seizing of the threads, a fault mentioned by Mr. Arnold and also observed by the author in a few cases. As a protection against such seizing, a material called "Molycote" had been used with good result.

The author found it a good idea, as mentioned by Mr. Arnold, to control the pre-tension through measurements on the outside of the rod. Unfortunately, it was rather difficult to calculate the exact stress obtained by means of this process as one did not know how many threads actually took part in the lengthening. This could be controlled by making a preliminary experiment in the workshop where the elongation inside the hole of the rod was measured, taking care to keep the measuring length well within the two threads. The measurements which the author mentioned as having been undertaken on real rods (cf. Fig. 9), were made according to this principle.

As regards strut buckling, several years ago the author had undertaken a number of calculations and found that the safety against strut buckling was much greater than the safety against fatigue. A minor inaccuracy in the erection would increase the risk of fracture both in fatigue and in strut buckling. As the restraint between the guides, as also mentioned by Mr. Arnold, could not be made more effective, an increase in the safety against strut buckling could only be achieved by means of an increase in the actual thickness of the rod above the top thread. Through this measure the rigidity of the rod would be increased so that the bending stresses at the top of the top thread from a possible misalignment would be increased, which would result in a diminishing of the rod's safety against fatigue crack; and as this safety was less than that against strut buckling, as already mentioned, it would be a most unsuccessful precaution.

As regards the bending stresses in the rods, these were measured on working engines just after the war by means of strain gauges. From the mass force originating from the weight of the one-sided crosshead shoe bending stresses arose of $\pm$ 1 kg/mm.^{2} (approximately 1,500lb. per sq. in.) when the clearance at the guide shoe was 0·3 mm. (12/1,000 inch). If the movements of the crosshead shoe were not restricted by the guide, this tension would increase to a little more than double and the same result was obtained if the play was over 0·7 mm. (28/1,000 inch). Through calculation it was possible to explain fully the tension-compression-stress measured in the rod section from combustion pressure and mass force and the measured transverse bending stresses; apart from the influence of the weight of the crosshead, there were lesser influences from the vibrations of the piston in the cylinder and from the crosshead's rotation round its horizontal longitudinal axis, vibrations which actually both got their impulses from the one-sided weight of the crosshead shoe. Consideration had been given to a suggestion to avoid these transverse bending stresses by trying to balance the crosshead shoe with a counterweight, but it was decided that it was unnecessary as the stresses were no greater than could be tolerated.

Moreover, by strain gauge measurements it was discovered that there were longitudinal bendings in the piston rod which were also roughly 1 kg/mm.^{2} when measured. These strains were probably due to misalignment (and partly also to the connecting rod changing its position longitudinally within its margins) and therefore varied from one plant to the next: and it was most probable that it was such stresses—which could equally well appear from transverse misalignments—which had caused the cases of piston rod fracture which had been mentioned by several of those contributing to this discussion. This corresponded with the fact that such fractures occurred most often in working plants after several years' service, so that irregularities might have arisen not only in the bearings and other moving surfaces but also in the construction of the frames, cylinder covers, cooling water jackets, etc. This agreed with Mr. Todd's observations of the displacements of the cylinder blocks in connexion with broken piston rods. But the author did not consider the marks of the nuts which were normally seen on the surfaces of crossheads which had been in use for several years as an indication of the fact that the piston had moved but as an unavoidable result of that small movement which was consequent on the bending of the crosshead during normal running. The wear thus caused gave a small contribution to the settlement already mentioned of approximately 1 degree annually which must be taken as a normal and harmless phenomenon so long as the tightening of the nut was altered accordingly.

No stresses were ascertained through these strain gauge measurements which could give any indication that the combustion pressure could be unevenly distributed across the top of the piston.

The author agreed with Mr. Falconer that it took some experience to judge an indication of magnetic flux in a thread. He himself got nothing good out of his first experiment in this direction and he had heard of a shipyard which had tested a set of piston rods without finding a single crack, with the result that one of the rods broke a few months after the examination. On the other hand, the fact that a crack was discovered was not always an indication that the piston rod in question was useless; if it was possible by means of grinding or by turning off one or two threads to remove the crack completely and to go well under the bottom of it, then it was, in the opinion of the author, quite permissible to re-use the piston rod but it must obviously be checked by magnetoflux after not more than one year.

As far as was understood from Mr. Cromarty's description (Fig. 15) it appeared that it was not a question of actual fractures but of small cracks starting in the seven piston rods concerned; experience had shown that the majority of such cracks did not continue through the cross-section of the rod but only resulted in a longer or shorter piece of a thread breaking off. It was probable that these rods could give service much longer if the cracks were ground away. Even taking into account that the stated guide clearances in some cases appeared rather large, it was difficult from Fig. 15 to see any connexion between the stated irregularities and the position of the cracks. Therefore it was more probable that the faults, as also pointed out by Mr. Cromarty, were due to small irregularities in the thread, and if no more important faults were discovered, as, for example, that the guide plane was not parallel with the cylinder's longitudinal axis, or that the crosshead pins were not at right angles with the cylinder axes, it was most probable that such minor faults would only result in the breaking off of shorter or longer pieces of thread.

The flexibility of the nuts mentioned was obtained by the author's firm in a manner similar to that shown by Mr. Cromarty (Fig. 16) and also described by Mr. Arnold.

The author's firm set the clearance between the piston rod and crosshead at 0·02-0·03 mm. (about 1/1,000 inch).

In the experiment the correctness of the nut threads was measured by means of an impression taken in dentist's wax; the deviations in pitch in the model rods and nuts was originally a maximum of 0·015 mm. per 1 inch, but after testing the most deformed thread settled up to 0·06 mm., corresponding to a 10-deg. nut revolution; it must be borne in mind that the loads were so high in the experiment, that they led to fracture after some million revolutions, corresponding to a much higher load than any rods were subjected to in practice.

As regards the material for these piston rods, the author agreed with Mr. Arnold that a good quality 28-32 tons tensile steel must be used rather than change to a steel with a higher tensile strength which gave simultaneously a lower elongation; and a certain setting, that is to say yielding, in fact a permanent elongation at the bottom of the threads was unavoidable.

The author agreed with Mr. Taylor and Mr. Phillips that normally nuts were not liable to crack, and he had not experienced cracks on piston rod nuts, but for reasons of space these piston rod nuts were far thinner than standard nuts, and because of this fact they were exposed to considerable bending stress and ring tensions; during the experiment it was measured that the exterior diameter of the nut varied considerably according to the loading. Therefore, the use of slightly more costly material for this proportionately small engine part was preferred than that the slightest risk should be taken of a serious breakdown, which could occur if a nut broke while the engine was going.

It was hardly advantageous to run the engines at a lower revolution number in order to prolong the piston rod life, as the mass forces which increased in accordance with the revolutions were advantageous in their effect on the piston rod as they counteracted the combustion pressure; that was to say, the piston rods were normally most heavily loaded at slow running.

During the experiment the author had varied the piercing of the crossheads, but it was impossible to measure any change in their rigidity.

When the shrinking of piston rods and crossheads was considered, the reason was that it was hoped to avoid thereby the bending stresses in the individual threads and the notch effect which could not be avoided with a thread. But the experiment showed that there was just as serious a notch effect from shrinking and, therefore, the author doubted the advantage of the method of construction suggested by Mr. Pluys (Fig. 17) and Mr. Vandeghen (Fig. 18). As far as was known to the author, several piston rods of this construction had broken and the author had seen starting cracks in the upper conical surface on rods of the same type. Moreover, there was the disadvantage in this construction that it was more difficult to adjust the height of these pistons. But the author fully agreed with the proposal to make the piece of the rod between the nuts as elastic as possible.

The author's company had for more than twenty years used hydraulic tightening of the long staybolts, by which method it was undoubtedly ensured (to a higher degree than by mechanical means) that the desired tightening was obtained, as mentioned by Mr. van Asperen. The uncertainty regarding these long elastic stays was undoubtedly even with pure mechanical tightening considerably less than on the rigid crosshead connexion because the set of the threads would be far less, expressed in a percentage of the whole elastic length. His company had in a few cases experienced fractures on these stays and found very small rest fractures. They ascribed these fractures to vibrations in the staybolts and had therefore during the last six years fitted bracing screws to prevent such vibrations.

As regards the experiment with the welded construction (Fig. 11), the author was not convinced that the most important effect of annealing was stress relief and, therefore, was not convinced that the improvement obtained in practice would be considerably bigger than the one obtained within the experiment, as stated by Mr. Taylor. The greatest improvement was most probably due to the softening of the local hardened zones, which usually appeared between the welding and the basic material, just there where the notching effect was at its highest. This experiment has been under discussion by the author in an article published by "The Welding Journal" probably appearing in April 1956*.

Mr. van Asperen's question as to whether it was justifiable to use welded elements for moving parts could hardly be answered in general. The importance of the experiment with welded structures must be seen in the fact that it gave the value of the strength figures obtainable in actual production of heavy construction, from which it could in each case be calculated if it were judicious to use welding of moving parts as well as stationary ones.

* See also footnote on page 57.

INSTITUTE ACTIVITIES

Minutes of Proceedings of the Ordinary Meeting Held at the Institute on Tuesday, 22nd November 1955

An Ordinary Meeting was held at the Institute on Tuesday, 22nd November 1955, at 5.30 p.m., when a paper entitled "Full-scale Fatigue Tests of Diesel Engine Elements", by P. E. Wiene, M.Ing.F., was presented and discussed. Mr. W. J. Ferguson (Chairman of Council) was in the Chair. Seventy-five members and visitors were present and eight speakers took part in the discussion.

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

Annual Conversazione

The Annual Coversazione was held at Grosvenor House on Friday, 2nd December 1955. The President, Mr. H. A. J. Silley, and Mrs. Silley received the 1,465 guests.

After dinner Sydney Jerome's Ballroom Orchestra played for dancing until 2 a.m., except for two interludes; there was a Floor Show at 10.30 p.m. when Wally Fryer presented the Champions of the Ballroom and the Vallie Trio entertained with "Romance in Song", and at midnight The Konyots, The Marlidor Trio and Les Bassi contributed a cabaret.

Section Meetings

British Columbia

A dinner meeting of the Section was held at Victoria in the Ward Room of the H.M.C.S. *Naden* on 10th February 1956. Thirty-nine members and eight visitors, including fourteen who had travelled from Vancouver, were present. An excellent dinner, carried through in true naval style, was presided over by Captain(E) C. I. Hinchcliffe, O.B.E., R.C.N.

After the dinner Mr. J. Brydon (Local Vice-President) gave a *résumé* of the proceedings which led up to this meeting and announced that the next meeting of the Section would be held in Vancouver on Friday night, 27th April 1956, and that this meeting would constitute the Annual General Meeting of the Section.

Following these remarks the meeting adjourned to the lounge where the speaker of the evening, Mr. R. D. Baver, B.Eng., B.A.Sc., post-graduate of the Massachusetts Institute of Technology, was introduced by Captain Hinchcliffe and gave an interesting lecture on "Radiography Applied to Welding Inspection of Ships' Hulls". A general discussion and question period followed, covering the field outlined by the speaker and other related applications of radiography, both gamma and X-ray.

Mr. W. Dey spoke in appreciation of the speaker and moved a vote of thanks on behalf of the members.

Kingston upon Hull and East Midlands

A meeting of the Kingston upon Hull and East Midlands Section was held on Thursday, 19th January 1956, at 7.30 p.m., at the Royal Station Hotel, Kingston upon Hull, when Mr. W. Lochridge presented his paper on "Trawling and Freezing at Sea (Fairtry)". Mr. H. F. Hesketh (Member) was in the Chair and seventy-one members and visitors attended. The author provided a great deal of information in describing this most interesting vessel and the new methods of trawling. The fact that the lecture was given in the chief fishing port of Great Britain ensured a vigorous discussion which was led by Messrs. A. Addy, F. T. Green, F. T. Morris and the Chairman. A vote of thanks to the lecturer was proposed by Mr. Addy and seconded by the Chairman.

Junior Lecture

On Thursday, 9th February 1956, a Junior Lecture was given at the College of Technology, Park Street, Kingston upon Hull, when Commander R. B. Cooper, M.B.E., B.Sc., A.R.S.C., read his paper entitled "Oil Fuel Burning". Approximately sixty marine and mechanical engineering students of the college attended, together with a fair number of senior members. An interesting discussion followed the lecture and a sincere vote of thanks to Commander Cooper was accorded.

After the meeting, Institute prizes were awarded by Mr. F. C. M. Heath (Vice-President) to H. R. English and D. M. Tree, the best first- and second-year students respectively in the Ordinary National Diploma course at the college organized in connexion with the Ministry of Transport Alternative Apprenticeship Scheme; and to N. W. Raine as the student taking the Higher National Certificate who accomplished the best year's work in heat engines.

Mr. G. H. M. Hutchinson (Chairman of the Section) was in the Chair and the Principal of the College, Mr. Emyln Jones, attended.

Scottish

December 1955 Meetings

Two general meetings of the Scottish Section were held during December. At the first meeting on the 14th, Mr. R. E. Zoller delivered a lecture on "The Selectable Superheat Boiler", illustrating the outstanding features with lantern slides. This lecture was very well received by the 105 members and guests present and fifteen took part in the discussion.

As the Chairman of the Section was abroad, the meeting was conducted by the Vice-Chairman, Captain(E) N. J. H. D'Arcy, R.N.(ret.). Appreciation of Mr. Zoller's lecture was heartily accorded in response to the vote of thanks proposed by Mr. R. Beattie.

The second meeting, on the 29th, was held jointly with the Institution of Engineers and Shipbuilders in Scotland. Mr. D. W. Low, O.B.E. (Chairman), presided and among those present was Professor A. M. Robb, President of the Institution of Engineers and Shipbuilders. One hundred and eighty members and guests attended to hear Mr. K. Maddocks deliver a lecture on "Nuclear Power for Commercial Vessels". The considerable interest aroused on Clydeside by this subject was reflected in the widely representative attendance, which covered all aspects of marine engineering and shipbuilding. In the time available for discussion, twelve speakers took part and added to the interest of the evening.

On the proposal of Mr. E. Norton (Member) a hearty vote of thanks was accorded to Mr. Maddocks.

January 1956 Meeting

A general meeting of the Scottish Section was held in Glasgow on 11th January 1956; Mr. D. W. Low, O.B.E.

The President, Mr. H. A. J. Silley, and Mrs. Silley

Mr. and Mrs. Silley, Mrs. Ferguson and Mr. W. J. Ferguson, M.Eng.
(Chairman of Council)

Annual Conversazione, 1955

Mr. and Mrs. F. J. Welch, Mr. Stewart Hogg, Mrs. J. S. Robinson, Mrs. Hogg, Mr. J. S. Robinson (Secretary) and Mrs. Villiers-Smith

Dr. S. F. Dorey, C.B.E., F.R.S., Mrs. D. Road, the late Mr. C. P. Harrison and Mrs. Harrison

Annual Conversazione, 1955

(Chairman), presided and forty-eight members and visitors were present.

At this meeting opportunity was taken to present Institute prizes to W. R. Nelson and W. D. Cherry, who were the best first- and second-year students taking the Ordinary National Diploma Course under the Alternative Training Scheme at Stow College of Engineering. The Institute's prize for Heat Engines was also presented to D. Mitchell, a student at Stow College taking the National Certificate Course. All three students were accorded hearty congratulations.

A most interesting lecture on "Control of the Accuracy of Marine Gears" was given by C. Timms, M.Eng. In this talk, Mr. Timms revealed the means adopted to test the accuracy of the components of a hobbing machine, also the measurement of the accuracy of the finished gear, and illustrated his various points with lantern slides.

The discussion was opened by Mr. A. W. Davis, B.Sc. (Member) and four others also took part. The appreciation of all for this highly instructive lecture was expressed by Mr. W. Young, whose proposal of thanks was enthusiastically accorded.

South Wales

A general meeting of the South Wales Section was held on 11th January 1956 at Cardiff Technical College. There was a very good attendance of members and students when Dr. A. Harvey, B.Sc., F.Inst.P. (Chairman), introduced the lecturer, Dr. J. E. Garside, M.Sc.Tech., pointing out how the Institute had provided many first class lecturers, but this was the first time a departmental head of another college had been chosen to give a lecture in Cardiff.

Dr. Garside used very few notes during his lecture and illustrated his points from time to time by lantern slides. His lecture covered the formation of metals, ferrous and non-ferrous metals, different types of each, and their applications, their failures and the cause of the failures. He then developed the theory of electronic action of different materials when in contact, or in fluids, and showed how pitting could be caused in steel plating.

Dr. Garside described the effect of mechanical stressing and fatigue and indicated the trend of development in materials, showing how aluminium was being recognized as a very versatile material.

A film on aluminium welding was shown after the lecture.

A vote of thanks to the lecturer was proposed and seconded by the students, and Mr. D. Skae (Vice-President) thanked Dr. Harvey for his continued support in allowing the Section to give these lectures.

Sydney

Annual General Meeting

The Annual General Meeting of the Sydney Section was held at Science House, Sydney, on Thursday, 22nd March 1956. Eng. Capt. G. I. D. Hutcheson, R.A.N.(ret.) (Local Vice-President) was in the Chair and eighty-two members and guests were present.

After the adoption of the Report and Balance Sheet, the Chairman announced the names of the office bearers for 1956 as follows:—

Chairman:	Eng. Capt. G. I. D. Hutcheson, R.A.N.(ret.)
Honorary Secretary:	N. A. Grieves
Honarary Treasurer:	J. R. Robertson
Committee:	E. L. Buls, W. G. C. Butcher, B. P. Fielden, D. N. Findlay, H. W. Lees and G. B. Williams

After the business meeting, a paper on "The Industrial Gas Turbine" was presented by D. C. Cooper (Associate Member); the paper was fully illustrated by lantern slides. Messrs. Findlay, Lees, Meredith and McLachlan contributed to the discussion. A vote of thanks to Mr. Cooper was proposed by Mr. Findlay and carried by acclamation.

West Midlands

January Meeting

A general meeting of the West Midlands Section was held at the Birmingham Exchange and Engineering Centre at 7.0 p.m. or Thursday, 12th January 1956. Mr. G. A. Plummer (Vice-Chairman) was in the Chair and the meeting was attended by forty-six members and guests. Mr. S. H. Griffiths delivered an illustrated lecture entitled "Recent Developments in Class I Welding".

Mr. Griffiths began by giving a brief description of the various methods adopted in the manipulation and weld preparation of steel plates before fabrication and discussed the relative merits of machine and hand welding and the bare wire, covered electrode and submerged arc processes. He continued by describing the non-destructive examination of welded joints, using either X-ray or gamma ray as a source of radiography. In concluding, the author referred to the difficulties encountered in the past when welding alloy steels and commented upon the success of the techniques recently developed for this purpose.

In the discussion which followed, fifteen members took part and the meeting closed at 9.15 p.m. with a vote of thanks to Mr. Griffiths, proposed by the Chairman.

February Meeting

At a general meeting of the Section held at the Birmingham Exchange and Engineering Centre, at 7.0 p.m. on Thursday, 9th February 1956, Mr. G. A. Plummer (Vice-Chairman) took the Chair; the meeting was attended by fifty-four members and guests.

Mr. R. Ecker presented a paper illustrated by slides entitled "The Design and Operation of High Temperature, High Pressure Steam Turbines".

He referred to the basic principles of steam turbine design and briefly outlined the major factors controlling the operating conditions and output. He then discussed the advance made in the development of turbines operating at high pressures and temperature and described some of the difficulties encountered, with particular reference to the limitations and subsequent revision of the steam tables. In conclusion he gave a brief description of some units of advanced design operating at 2,350lb. per sq. in. and 1,450 deg. F.

In the ensuing discussion six members took part and the meeting closed at 9.30 p.m. with a vote of thanks to Mr. Ecker, proposed by the Chairman.

Third Annual Dinner

The Third Annual Dinner of the West Midlands Section was held at the Imperial Hotel, Birmingham, on Thursday, 15th March 1956, and was attended by eighty members and guests.

The Chairman of the Section, Mr. H. E. Upton, O.B.E., was in the Chair; among notable guests were Mr. T. W. Longmuir (Vice-Chairman of Council), Rear-Admiral F. W. Billings (ret.), Capt. H. K. Hodgkin, R.N., and Mr. J. S. Robinson (Secretary of the Institute).

After an excellent dinner, the Chairman introduced Admiral Billings, who proposed the toast of "The Institute of Marine Engineers", and Mr. Longmuir responded. "The West Midlands Section" was proposed by Mr. G. A. Plummer and the Chairman responded.

The entertainment during the evening was provided by a number of local artistes.

The presence of the guests already mentioned contributed much to the success of the evening and officers and members of the Section wished to express their appreciation.

OBITUARY

CLEMENT PHILIP HARRISON

An appreciation by Mr. A. LOGAN, O.B.E. (Vice-President)

The recent death of Mr. Clement P. Harrison at the age of fifty-three means we have lost a Member who had our Institute's well-being at heart and one who was anxious to take his share of the work.

Elected as an Associate in 1935, Mr. Harrison transferred to Membership in 1941. He became a Member of Council in 1950, and it was during his four-year term of office, both in Council and on Committees, that his sense of sound judgement and integrity became known to his fellow office bearers.

Educated at Queen Mary's Grammar School, Walsall, Clement Harrison commenced his apprenticeship with the General Electric Company at their Witton Engineering Works in 1921, and at the time of his death he was Manager of the Marine Department of that same company—surely an outstanding tribute to his hard work and loyalty.

Our late colleague was closely associated with his company's development in the field of marine electrical equipment, and senior members will remember his paper, "Fuses and Circuit Breakers for Circuit Control and Protection in Marine Installations", published in the 1944 TRANSACTIONS, while junior members will recall the lecture on "The Use of Electricity at Sea" which he presented at meetings throughout the country during the years 1947/1950.

In recording our appreciation of the late Clement Harrison, we offer our sincere sympathy to his widow and thank her for the support she gave him in his service to the Institute.

SAMUEL THOMAS ACKLAND (Companion 14803) was born in 1895. He joined the Prince of Wales Dry Dock Co., Swansea, Ltd., in 1909 and after working through the various departments was appointed assistant secretary in 1937 and secretary two years later. In 1949 he was elected to the Board and held these two appointments until his resignation owing to ill health on 31st December 1955. He died on 27th January 1956. Mr. Ackland was elected a Companion of the Institute in 1954.

MAURICE BLUET (Member 13306) was born in 1900. After completing his initial studies at the Génie Maritime he was the engineer responsible for naval construction in Cherbourg in 1924-25. For the next sixteen years he was an engineer with the Compagnie de Construction Mécanique Procédés Sulzer, being promoted chief engineer in 1941, the position he occupied at the time of his death on 14th June 1955. He had been a Member of the Institute since 1951 and was a member of the Association Technique Maritime et Aeronautique.