To face page 51.

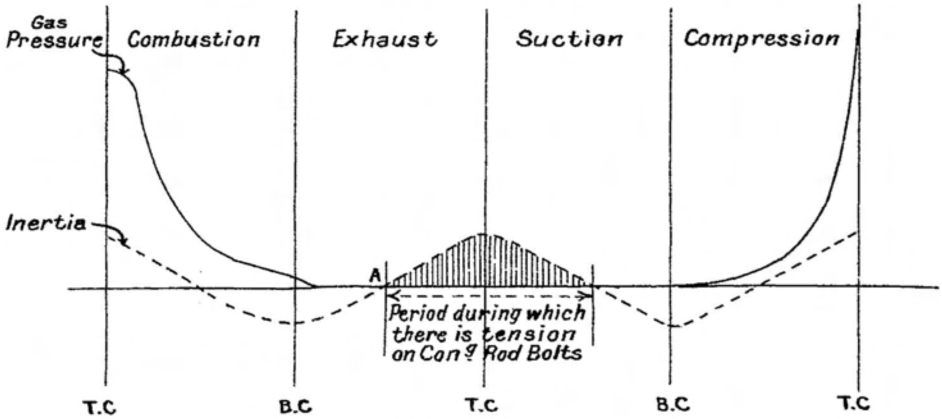

FIGURE 1.

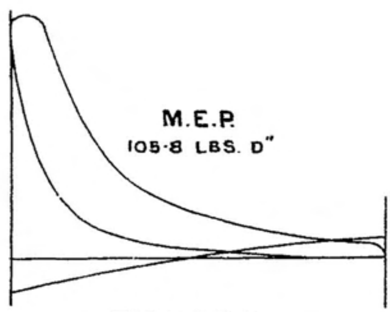

FIGURE 2.

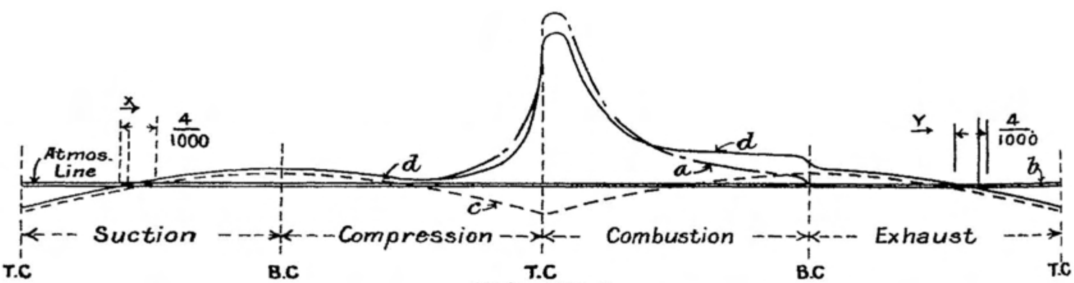

FIGURE 3.

CONNECTING ROD BOLTS–4 CYCLE DIESEL ENGINE.
NATURE AND DURATION OF STRESS.

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Malby & Sons. Lith.

CONNECTING ROD BOLTS FOR DIESEL ENGINES.

The Diesel Engine Users' Association have recently carried out an investigation as to the advisability of annealing or heat-treating the connecting rod bolts of Diesel engines at intervals with a view to assuring the continued reliability of these small, but highly important, links in the mechanism of the engine. A number of makers, users, and others were consulted and, as might, perhaps, be expected of such a highly controversial matter in the present position of the art, widely varying views were expressed, and no authoritative finding could be pronounced. The investigation, however, yielded other information respecting more obscure factors influencing the stresses on the bolts and other aspects of the problem of interest and value to the user, no less than the designer of Diesel engines.

A portion of the Association's summary of the investigations on this part of the work which appears likely to be of interest to naval readers is given below; this includes for completeness the illustration of the generally known effect of inertia forces for a four-cycle, single-acting, engine.

The mechancial forces acting upon the connecting rod bolts are the inertia forces originating in the piston, the centrifugal forces originating in the crank and in part in the connecting rod, and the forces set up by the securing of the bolts, in addition to stresses arising from piston and piston ring friction. These are the normal forces which are necessarily provided for as required in the design of the bolts at either end of the connecting rod. Stresses due to impact in the bearing caused by bearing clearances should also be taken into consideration. Abnormal stresses are those caused by excessive impact due to improper bearing clearances and to excessive piston friction, usually taking the form of slight, or more or less severe, piston seizures. The bolts are in tension during the latter half of the exhaust stroke, and during the first half of the suction stroke (*see* Fig. 1).

An actual indicator diagram illustrated in Fig. 2 has been developed and corrected for inertia forces in Fig. 3, from which it will be seen that there is a fluctuation of stress in the bolts at the points X and Y. An explanation of Fig. 3 is as follows :—

The cylinder dimensions are $16\frac{3}{8}$-in. diameter by $23\frac{5}{8}$-in., the mean effective pressure being 105·8 lbs. per sq. in. at 175 revolutions per minute. The crank pin dimensions are 9 ins. diameter by $9\frac{7}{16}$ ins. long, and the length of the connecting rod is 5·33 that of the crank. The weight of the piston is about 775 lbs. and the connecting rod 663 lbs., so that the total moving masses are about 1,218 lbs. The co-efficient of speed irregularity is $\frac{1}{70}$, and the mean velocity of the crank pin is approximately 16·8 ft. per second. The clearance of the bottom end was four thousandths, and these data have been utilised to construct the figure, where curve (*a*) represents actual gas pressure on the top of the piston, (*b*) is the curve representing the weight of the piston, (*c*) is the acceleration curve of the reciprocating and part of the rotating masses, and (*d*) shows the corrected curve of the pressure on the crank pin, indicating that at the points X and Y there is a fluctuation of stress.

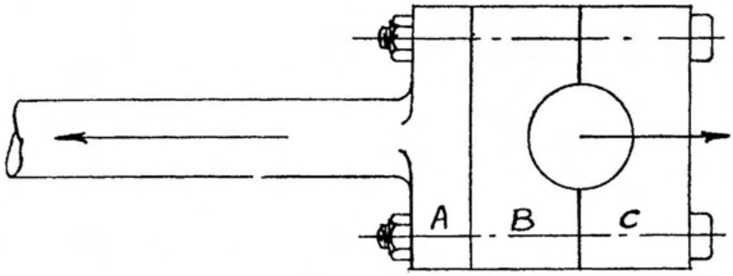
In view of the clearance on the bearings the load takes the form of an impact, and, therefore, the net force on the connecting rod bolt cannot be measured directly off the curve (*d*), but involves a set of calculations of a rather complicated nature. Curve (*d*) of course neglects the inertia of the crank webs, and any frictional losses which present.

Another factor which enters into the question of the determination of the maximum possible stress to which a connecting rod bolt may be submitted is the consideration of the torsional vibration of the crank shaft. It is essential to take into consideration the natural frequency of the system, consisting of crankshaft, fly wheels, &c., and to ensure that it does not coincide with the impulses originating from the variation in crank effort, because even a slight approach to the synchronising frequency would introduce considerable stress on the connecting rod bolt.

In the course of the consideration of the mechanical stresses occurring in the connecting rod bolts there was a sharp difference of opinion as to the effect of the initial bolt tension due to tightening the nut upon the total load upon the bolts. It was maintained on the one hand that, in the case of the bottom end connecting rod bolts, the initial stress is employed in holding the parts of the bottom end bearing together, and, until the inertia force exceeds the amount of the initial stress no additional load is put upon the bolts. On the other hand, the view was strongly held that the inertia stress is supplementary and additional to the initial stress upon the bolts. These latter views were developed in considerable detail, reference being made to the flat bottomed, or Marine type of connecting rod, and to the jaw pattern in which the bottom end consists of only two main portions, the difference in the nature of the problem being noted. The argument proceeded as follows :—

In considering the stresses produced in connecting rod bolts, the crank end of the rod and the bolts are assumed to be of the same material, thus simplifying the reasoning. It should, however, be borne in mind that all material is elastic, and that stress and strain cannot be separated. Just as soon as any material is subjected to any kind of stress, no matter how small, there is a corresponding strain or deformation.

When the bolts are tightened up (*see* sketch) they become strained in proportion. Also the resulting compressive force acting on the parts A, B, C, produces a relative strain in those parts, since in a body at rest one force cannot act alone, but must have an equal opposing force. This compressive force produces a corresponding pressure at the joints of AB and BC. Hence, when a pull, however small, is produced in the connecting rod, due to inertia force, &c., an equal opposing force (crank



effort) must act on C. These opposing forces, therefore, reduce the pressure at the joints of AB and BC, and proportionally relieve the compressive strain in the part B, which amount of strain is added to the initial strain in the bolts, and as stress and strain are inseparable, any increase in strain must necessarily result in increased stress.

The total pull due to inertia force, &c., cannot be added wholly as additional stress in the bolts, but such an amount must be added which is proportional to the relief of compressive strain in the part B. If the part B did not exist, and parts A and C only formed the end, the reasoning would be different, because no additional stress can be added to the bolts until the pressure at the joint AC, due to the initial stress in the bolts, has been overcome.

The reasoning given above was subjected to experimental demonstration carried out accurately with effective and suitable instruments as described and illustrated by diagrams (not reproduced).

The piston speeds developed by the Diesel engines on the books of the Association vary between 600 and 800 ft. per minute, the latter figure being the maximum for the majority of cases. At these piston speeds the stresses in the connecting rod bolts range from about 4,500 to 6,500 lbs. per sq. in. Taking the higher figure as a maximum and using a high tensile steel there appears to be ample safety (for medium piston speeds up to 800 ft. per minute) for general service conditions. It will be conceded that very severe abnormal occurrences are beyond the control or purview of the designer of a commercial engine. The fact that a broken connecting rod bolt is so rarely met with is a good indication that the stress limits employed by manufacturers generally are sufficient and ample.

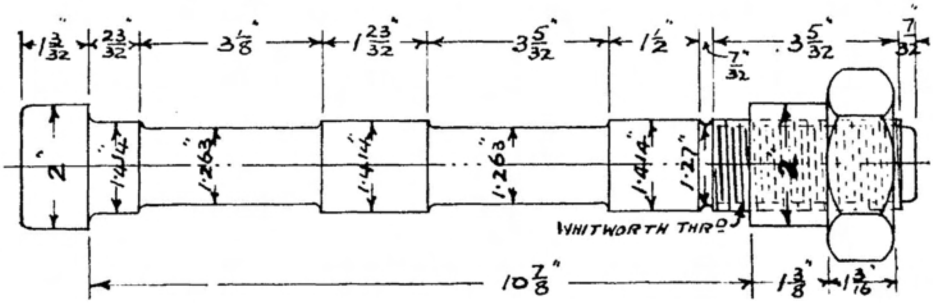
The design of the connecting-rod bolt has been a matter of most careful consideration on the part of manufacturers. In order to ensure that stretching, when it occurs, shall not take place in the thread of the bolt, the shank is turned down to a diameter smaller than that at the bottom of the thread, portions being left of full diameter for fitting purposes. A groove is turned in the bolt at the end of the screwed portion and, where change of sectional area is required, a fillet of large radius, according to the size of bolt, is provided. This design secures an ample stretching length and ensures that the bolt is not weakest at the screw thread, a defect which is serious in bolts having parallel shanks of diameter equal to the full diameter of the screw thread. The latter type of bolt will fail, if at all, at the screw thread immediately below the bottom of the nut in the majority of cases. The former type stretches and breaks in the shank. Among the numerous minor points touched upon during the investigation, that of the fastening of the bolts was not the least important. The Committee endorse the practice of completing the fastening with a castellated nut and split pin, an arrangement providing the most secure form of locking. They noted with interest and approval the statement of one engine manufacturer who expressed the opinion that there should be more than one castellation per face of the nut and more than one hole through the bolt to receive the split pin. He recommended as many castellations as the diameter permits, and the provision of several holes through the bolt. This practice will tend to prevent injudicious use of the spanner in tightening the nut, an error of judgment which received a good deal of comment on the part of several contributors to the discussion. A certain type of human nature is very prone to avoid the trouble of easing the face of a nut by forcing the material.

Some illuminating facts came to light in reference to this point. The energy exerted by the workman in tightening a nut by hand and finishing his work by the aid of a hammer is a quite unknown quantity. With small bolts an unduly heavy torsional stress very soon causes fracture, but with large bolts of $1\frac{1}{2}$-in. diameter and upwards the danger is more insidious because unsuspected in the generality of cases, while there is no visible result of the severe punishment inflicted on the material. Several tests were carried out upon connecting-rod bolts by members of the Association acting independently. The results obtained were similar. One of the tests is described, the particulars of the bolt being as follows :—

Diameter	45 mm.
Depth of head	29 mm.
Threads per inch	11
Depth of nut	95 mm.
Effective length under head to top of nut	410 mm.
Overall length of bolt	450 mm.
Elongation as found under average strain	·010 in.

Assuming a modulus of elasticity of 30,000,000 lbs. per sq. in., the tension on the bolt is of the order of 18,500 lbs. per sq. in., equivalent to a load of $8\frac{1}{4}$ tons.

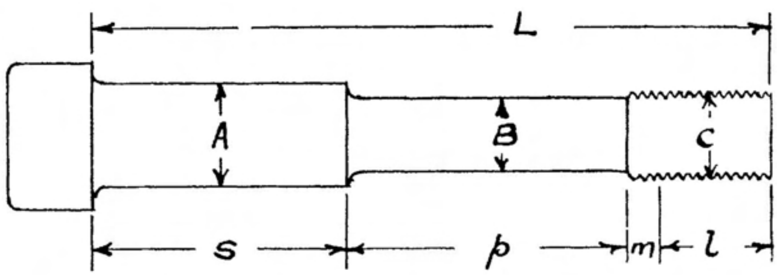
A second illustration, given in much fuller detail, indicates the effect of the use of excessive force, and will be found in the following notes. (The bolt under consideration had been heat treated after service : the modulus of elasticity had been carefully ascertained to be 25,000,000 lbs. per sq. in. after treatment.)



WORKING TENSION.

Strain when tightened up $=0\cdot008$ in. Strain when slogged up $=0\cdot01$ in.
Approximate tension in lbs. per sq. in. $=$ 18,400 (normal tightening).
Approximate tension in lbs. per sq. in. $=$ 23,000 (slogged up).

When the nut is tightened up, and assuming that all its threads have equal contact with those of the bolt, the tensile force is evenly distributed throughout the length of the nut, but the tensile stress produced in the bolt will vary from zero in a sectional plane in line with the free or outer end of the nut to a maximum in a sectional plane in line with the contact face of the nut; hence the strain in that portion of the bolt will relatively increase. The added stress of shear due to torsion when tightening up the nut may be as much as 17 per cent. of the tension on the bolt. (Hence the reason for the failure of a bolt at the face of the nut.) In dealing with the problem the various portions of the shank are re-grouped, as in the sketch below, an arrangement that does not affect the calculations for direct tensile stress.



L = total length of bolt from the contact face of head to outside face of nut in position before tightening up $= 13\frac{7}{16}$ ins.
$l = 2\frac{9}{16}$ ins. = total length of nut.
$m = \frac{7}{16}$ in. = length of screw not engaged with nut.
$p = 6\frac{1}{2}$ ins. = „ shank reduced in diameter.
$s = 3\frac{15}{16}$ = „ „ enlarged „
$A = 1\cdot57033$ sq. ins. = area of shank enlarged in diameter.
$B = 1\cdot25285$ sq. ins. = „ reduced „
$C = 1\cdot33756$ sq. ins. = effective area of screw.

$T = 0 \cdot 008$ in. $=$ total strain as measured $= U + V + W + X$.
$U =$ total strain in the nut length l.
$V =$,, screw length m.
$W =$,, shank length p.
$X =$,, ,, s.
$Y =$ total tensile strength acting on length ($m + p + s = 10\frac{7}{8}$ ins.).
$\frac{Y}{2} =$ average total tensile stress acting on length ($l = 2\frac{9}{16}$ ins.).
$E =$ Modulus of elasticity $= 25{,}000{,}000$.
$= \frac{\text{Unit stress}}{\text{Unit strain}}$

Unit strain in length $l = \frac{Y}{2\,CE} \therefore$ total strain $= \frac{Y}{2\,CE} \times l = U$

,, $m = \frac{Y}{CE} \therefore$,, $= \frac{Y}{CE} \times m = V$.

,, $p = \frac{Y}{BE} \therefore$,, $= \frac{Y}{BE} \times p = W$.

,, $s = \frac{Y}{AE} \therefore$,, $= \frac{Y}{AE} \times s = X$.

Now the total strain $= T = U + V + W + X = 0 \cdot 008$ in.

$$= \frac{Yl}{2\,CE} + \frac{Ym}{CE} + \frac{Yp}{BE} + \frac{Ys}{AE} = \frac{Y}{E}\left(\frac{l}{2\,C} + \frac{m}{C} + \frac{p}{B} + \frac{s}{A}\right)$$

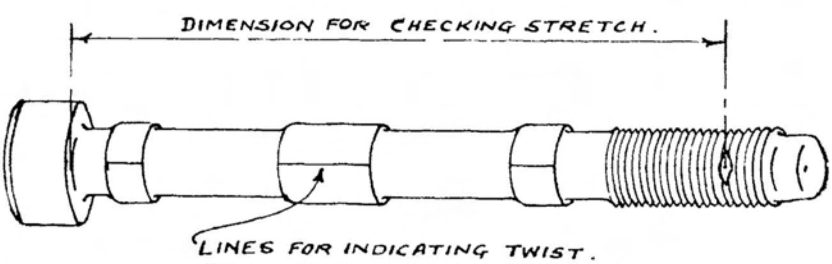
Hence $Y = 22{,}270$ lbs.

$\therefore$ unit tensile stress in area $A = \frac{22270}{1 \cdot 57033} = 14{,}812$ lbs. per sq. inch.

and ,, ,, $B = \frac{22270}{1 \cdot 25285} = 17{,}775$,, ,,

and ,, ,, $C = \frac{22270}{1 \cdot 33756} = 16{,}650$,, ,,

The Committee drew attention to the proposal of one of the principal Diesel engine manufacturers in England (Messrs. Mirrlees, Bickerton & Day, Ltd.) to so mark their connecting rod bolts that their condition with respect to twisting and excessive longitudinal strain can be readily determined when removed for examination from time to time.



On each bolt a definite length, preferably an even dimension, is to be marked by means of a centre dot on the thread, as near the end of the bolt as is practicable for an even dimension. The dimension to be taken from the underside of the head to the centre dot. To facilitate the locating of this dot a small portion of the thread to be filed down (*see* sketch). Measurement of the distance from the head to this dot should be made periodically, and will show if the bolt has been stretched. A longitudinal line should also be marked on the bolt, this line being preferably placed so as to pass through the centre dot. Inspection of this line will show if the bolt has been twisted. Whenever a bolt has been materially stretched or twisted, it should be rejected, and substituted by a new one. A stretch of 2 per cent. of the length or a twist $\frac{1}{8}$ in., measured on the surface of a bolt diameter of 1 in., would be considered as justifying a new bolt.