# THE EFFECT OF NON-CONTACT EXPLOSIONS ON WARSHIP MACHINERY DESIGN

by

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*This article is the reproduction of a paper read by Commander Bonny at a Meeting of the Institution of Naval Architects on March 18th, 1948, and is published with the Author's and Institution's permission. The original paper included two Appendices which have been omitted here ; Appendix I dealt with the calculation of the effect of a pressure pulse on an unrestrained flat plate, and its relation to the effect on items of machinery and equipment ; Appendix II gave some illustrations of the modifications required to existing machinery, details of which have already been supplied to the Fleet.*

Before the Second World War the problem of damage to machinery due to shock had not been one of importance ; the machinery was in general of a robust nature, as was essential for the conditions of naval service, and had withstood, without any outstanding troubles, the vibration met with at high speeds and also the shocks due to the firing of guns and the dropping of depth charges. Previous war experience had not revealed any weakness in this respect when ships were hit by shells or torpedoes, though the possibility of transmission of shock through the water had been considered in so far as it might affect condenser inlets and doors. Some damage from blast had also occurred when ship's guns were trained too close to the ship's structure.

Late in 1939, however, H.M.S. *Belfast* was mined in the Firth of Forth by a magnetic mine on the sea-bed, and the very extensive damage to machinery and hull structure caused by this mine exploding some 80 to 90 ft away from the hull focussed attention on this new type of attack.

This incident was followed by many more of a similar nature when the various types of non-contact mines employed by the enemy, as well as the explosion under water of near-miss bombs, caused a considerable number of ships to be laid up for long periods of repair. In all, over 100 ships suffered shock damage during the war—much of it of a serious nature.

Since, in general, warships are specially designed to withstand attack by torpedoes, mines, projectiles, bombs, and any other method of attack, this result was unexpected and formed a new and major problem for the naval designer at a critical period of the war.

An examination of the problem showed that in most cases the machinery and electrical equipment suffered greater damage than the hull itself, except where the ship suffered partial collapse of the structural girder. Of the machinery and equipment, by far the greatest part of the damage was caused by fracture of cast-iron castings, though considerable subsidiary damage and derangement was caused by the tripping of governors and defects in electrical breakers, lighting, and other electrical equipment—gyro compasses in particular suffering severe damage.

It is especially noteworthy that very little of this shock damage occurs from direct hits by either shell or bomb or from torpedo hits. In these cases the local devastation is severe, but once outside the direct radius of the explosion little damage other than by splinters is met.

Though the incidence of shock damage fell off after 1941-42, due to the various protective measures against magnetic and acoustic mines, the necessity for design against shock is still of great importance.

Since most of the design departments were affected by this problem, an inter-departmental committee, the Admiralty Shock Committee, was formed to investigate the problem both in theory and by experiment. This committee has carried out full-scale experiments on H.M.S. *Cameron*, a *Town* Class destroyer; on the submarine H.M.S. *Proteus*; on a target representing a section of a submarine; and on the destroyer H.M.S. *Ambuscade*, while further investigations into the effect on larger ships are being pursued.

Without going into the detail of these experiments, the following general picture of the effects of an underwater explosion will be of value in formulating principles for the design of machinery to withstand shock.

### Description of an Under-Water Explosion

Taking as an example a medium-size charge of 300 lb T.N.T. (or its equivalent of other explosive) in a container under water, then, on detonation, the solid explosive is changed almost instantaneously (about 1/20,000 sec) into an equal volume of gas at a very high temperature and at a pressure of the order of one million lb/sq. in. The water surrounding the charge possesses both a high density and a low compressibility and confines the gases in a bubble.

If the charge is fired in contact with a ship's hull, then the plating, which can only withstand a relatively low pressure in lb/sq. in., is blown in, and the pressure in the explosion is such that, together with the projectile effect of the splinters of plating, further inner plating is pierced unless elaborate systems of protection are fitted. The inrush of gases and splinters followed by water cause extensive local damage to any machinery in its path.

If, on the other hand, the charge is not in contact with a ship, then the bubble of gases expands rapidly outwards, the pressure falling (approximately adiabatically) as the sphere of gases increases in diameter.

The effect of the very rapid initial rise of pressure followed by the expansion of the bubble is to send through the water in all directions a pressure pulse travelling on a spherical front at the speed of sound in water (about 5,000 ft/sec) at a distance from the explosion but considerably higher close to the bubble owing to the high pressures involved.

This pressure pulse has a steep-fronted peak followed by a slower fall as the bubble expands, and as the front of the wave reaches each (spherical) layer of water so it compresses it and gives each particle of water in the layer an outwards velocity proportional to the pressure in the wave at that point. This particle then acts on a particle in the next layer and accelerates it while bringing itself nearly to rest, thus carrying on the wave front. Once the wave front has passed, the water reverts almost to normal, having only a small outward velocity except near the bubble.

Near the bubble, however, the velocity of the water is considerable, and if an explosion occurs close to a ship the pressure in the pulse may well be sufficient to rupture the hull, and the water between the bubble and the hull will be projected inwards, causing considerable damage.

In the general case the bubble continues to expand until the pressure of the gas equals the hydrostatic pressure, but, since at this point the water at the boundary has considerable outward velocity, the bubble carries on over-expanding for some time until the pressure is considerably below hydrostatic. The water then reverses its flow and the bubble contracts, but again overshoots the mark, the pressure rising to a second peak and sending out a secondary pressure pulse.

This oscillation of the gas bubble carries on until, consequent on the upward movement of the bubble due to its buoyancy, it breaks surface. These secondary pressure pulses are readily noticeable by an observer in a vessel close to an under-water explosion like hammer blows on the hull, and three or four are often noticed with fairly deep explosions.

While the pressures associated with the secondary pulses are considerably lower, they may be of importance if the charge is underneath a vessel, while in certain cases the maximum diameter of the bubble at the end of its initial expansion may be such that the bubble touches the skin of a ship. In this case the ensuing contraction of the bubble will take place against the ship's side, and the maximum pressure in the second pulse may be sufficient to rupture the plating even though the initial pressure pulse was insufficient to do so.

The pressure/time curve of the initial phase has been measured experimentally by several observers, and can be closely approximated to by an instantaneous rise of pressure to the maximum pressure $p_{max}$ followed by an exponential fall giving the pressure at any moment :

$$p_1 = p_{max} e^{-nt}$$

where $n$ is a function of the weight and composition of the charge and is given by

$$n = \frac{C}{W^{\frac{1}{3}}}$$

while $p_{max}$ is found to decrease approximately with the distance expressed in charge radii.

Hence

$$p_{max} = \frac{KW^{\frac{1}{3}}}{D}$$

where W = equivalent weight of T.N.T. ;

D = distance from centre of explosion ;

and from the experimental results the various constants can be evaluated, and, if great accuracy is required, allowances can be made for dissipation losses. The equivalent weights of T.N.T. for the various explosives used for under-water work can also be obtained experimentally ; non-detonating explosives, however, give very low values of maximum pressure owing to the time taken in burning.

## Effect at Surface of the Sea

When the spherical front of the pressure pulse reaches the surface it has to adapt itself to the atmospheric pressure that obtains there, and this can only be done by the reflection of the pulse as a negative pressure, *i.e.*, tensile pulse. This reflection of pressure of opposite sign occurs at free surfaces in all sound and stress wave problems.

As a result of this reflected pulse the pressure in the water just below the surface rapidly becomes negative, and when this tension exceeds a certain value (believed to be about 200 to 500 lb/sq. in.) the water breaks and the energy in the layer of water above carries the broken water upwards as a spray. This effect is greatest directly above the charge, falling off radially outwards due to the greater distance travelled by the pulse and to the decreasing value of the vertical component. As a result a dome of spray rises with a fairly definite radius and surrounded by a black ring. (Here it is believed breaking of the water is still occurring, but the energy is insufficient to throw the broken water clear of the surface, although sufficient to alter its power of reflection.) The maximum radius and height of this spray dome bear a definite relation to the size and depth of the charge, and from high-speed photographs it is possible to check the depth of the explosion from this data.

Finally, the bubble breaks through the surface and projects upwards the " plume " of water and gas ; this plume may vary considerably with the state of the bubble at the moment it breaks through, being most spectacular when the bubble is near a maximum pressure condition on breaking through.

With deeper explosions the dome is smaller, until finally no dome is formed, a shimmer with droplets of spray occurring on the surface as the pressure pulses arrive, while instead of a plume the gases come up in a mass of creamy water.

**Effect of Sea Surface and Bottom Reflections**

In the case of charges either on the bottom or close to the surface, the effect on a ship will be altered by the arrival of reflected waves at a short interval after the direct pulse. In the case of reflections from the bottom, the effect will be additive, and in the ideal case of a charge resting on a rigid sea-bed the pressure would be doubled ; however, as a rule, the increased effect is small, due to softness of the sea floor.

With a charge near the surface, a reflected tensile pulse will arrive at the ship before the direct pulse has died away and will tend to cut off the pressure in the pulse, though, since water can only withstand a limited tension, the reduction is not so large as would occur if this were not the case. In addition, with shallow explosions the bubble will break surface and vent before it has fully expanded, and the amount of energy imparted to the water will be less.

**Initial Effect on a Ship and subsequent Whipping**

When the main pressure pulse strikes the hull of a ship the shell plating, being relatively free to move, is given a very high acceleration and attains a velocity of some tens of feet per second in a thousandth of a second or less. The plating bends between the more rigid supporting frames, but these in turn gain velocity, the acceleration depending on the masses supported. As the pulse reaches the more distant parts so they are in turn accelerated and gain velocity, until after 20 to 30 thousandths of a second the whole ship has been set in motion. As the pressure dies away at any position, so the elasticity of the members will overcome the pressure and the part will tend to oscillate back to its position of rest at its natural frequency, generally with considerable damping.

If the movement of the plating is sufficient to cause plastic strain, the plates will be dished inwards, giving the typical " hungry look " ; severe dishing is usually accompanied by distortion of bulkheads, etc.

As the various sections of the ship are set in motion at different periods of time, the ship as a whole is deformed and, in addition to the local oscillations of plates and frames, larger items, such as bulkheads, will tend to vibrate, and, in turn, the whole ship will vibrate as a beam with two or more nodes. This latter is called whipping, and its magnitude varies considerably with the position of the explosion.

This whipping, though involving considerably lower accelerations than the initial shock on the arrival of the pulse, is associated with large displacements up to 1 or 2 ft, and is the phenomenon most noticeable to the human observer. It may also cause the throwing about of loose objects and damage to items on certain types of mountings. In addition, the interval between successive pulses is often of the same order as the period of whipping of the ship, so that with charges nearly under the ship a considerable increase in whipping may result from the secondary pulses.

**Waves of Stress and their Reflections**

In considering further the transmission of the pulse to parts inside the ship it is necessary to consider the theory of propagation of stress. Thus, if an impactive blow be given to one end of a bar of metal, a wave stress is propagated along the bar at a velocity equal to the velocity of sound in the metal, viz., $\sqrt{E/\rho}$, where E = Young's modulus and $\rho$ its density. This wave of stress is similar to the pressure pulse in the water except that in this case the motion is on a plane front instead of a spherical. Again each particle in being pushed by and pushing on its neighbours attains a momentary particle velocity proportional to the intensity of the stress.

When such a stress reaches a free end, then, as in the case of the free surface of the water, a wave must be reflected back along the bar of opposite sign (pull instead of push) to make the stress zero, and such stress will oscillate to and fro until it dies away. If, on the other hand, it strikes a rigidly fixed end, the reflected wave will be of the same sign and a force equal to twice the blow acts momentarily on the fixed surface. Similarly, with partially fixed ends and changes of section, varying reflected waves will be caused, and these waves can be superimposed to give the total stress at any instant at any position. Such waves occur even with normal rates of loading, but in view of their velocity (16,700 ft/sec in steel) the variations in strain caused by them are negligible. Even in the case of the shock wave experienced by light machinery items, the time of application of which is some 2 to 3 thousandths of a second, the effect of such wave propagations of stress does not show on record and can normally be neglected.

**Forces acting on internal Machinery and Fittings**

The pressure pulse having caused the plating and frames to deflect, these items will in turn exert forces on the various seatings, bulkheads, etc., and hence on the machines and fittings mounted on them. Depending on the items involved, the masses will be accelerated up to appreciable velocities, and as the pulse dies away they are brought to rest by their fastenings or internal stresses.

The resulting effects are many and varied. From the initial pressure pulse we get the very considerable fractures of main and auxiliary machinery and stretching of holding-down bolts, while from the later whipping cases have occurred of priming of boilers, throwing around of floorplates, ladders, and other loosely secured items, and the tripping of governors on dynamos, feed pumps, etc., besides the very considerable effects noticed on electrical machinery and equipment, gun mountings, rangefinders, and the small items such as wireless and radar and their aerials and arrays.

**Calculation of Effect on an Item**

In order to be able to design machinery to withstand shock, it is obviously necessary to get a fairly close approximation to the motion undergone by the body, so that a knowledge may be obtained of the accelerations to which a body may be subjected and their duration, as, in general, failure of a body is caused by relative motion ; thus, except for the most brittle materials, fracture is always preceded by strain which involves movement in the body.

However, as is the case with all calculations involving complicated bodies, such as ships' structures, there are too many factors about which insufficient data are as yet available to permit of a detailed calculation of shock effects.

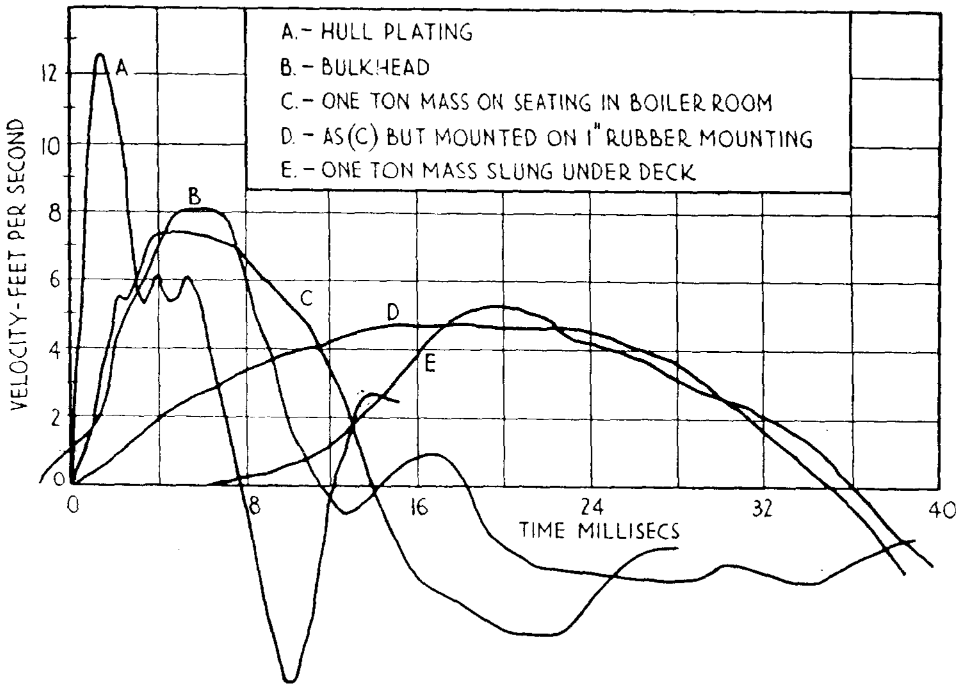


FIG. 1.—TYPICAL VELOCITY TIME CURVES

Such factors are :—

(i) The stiffness of the skin plating and its associated framing and the energy absorbed in dishing of the plating and distorting the frames.

(ii) The effect on such stiffness of added members, supports for machines, bulkheads, etc., and the resultant motion of the platforms and seating of machines, etc.

(iii) The increase in yield point and U.T.S. that occurs when metals, particularly low carbon steels, are subjected to rapid loading.

(iv) The effect of the reflections of the pressure pulse from the sea bottom and surface and diffraction of the pulse around corners.

(v) The reflection of the pressure pulse from the moving ship's plating and whether under certain circumstances this leads to cavitation in the water close to the plate—a matter the effect of which is most difficult to assess.

Nevertheless, a reasonably close approximation to the velocity that an item would attain, together with an appreciation of the acceleration, can be obtained from consideration of the behaviour of an unrestrained flat plate subjected to a pressure pulse.

Though, in fact, the ship's structure consists of relatively thin plates formed into panels by longitudinals and transverse frames, consideration of the various aspects of the problem shows that when considering fairly concentrated items of equipment the error is small if we take the item as equivalent to a plate of the total supported weight divided by the area contributing to its motion. This is particularly so since, for the weights per unit area commonly met with, the proportion of energy absorbed varies only slightly from a mean figure of approximately 0·5. Thus, provided a reasonable assumption of the area contributing to the motion can be made, the maximum velocity of an item can be obtained from a formula

$$\tfrac{1}{2}\, M\, V_{max}^2 = \text{constant} \times \frac{W \sin \theta}{D^2}$$

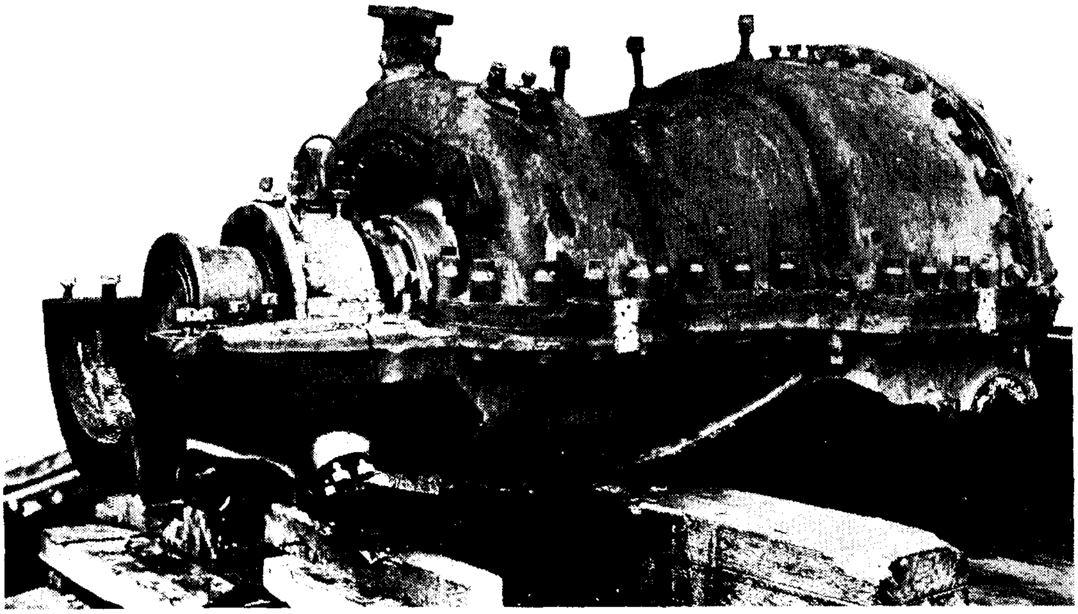

FIG. 2.—L.P. TURBINE OF A SLOOP

$$V_{max} = \frac{K}{\sqrt{M}} \frac{\sqrt{W \sin \theta}}{D}$$

Such an approximation, however, applies only up to the elastic limit of the structure under consideration, and above this the plastic deformation of the hull plating and frames will result in the absorption of energy that would otherwise contribute to the motion. The effect of plastic dishing is again, as yet, incalculable, and must be founded on practical experiment ; since the average ship will only stand a very few shocks of this magnitude this is both long and costly.

Sufficient data are, however, available to give an overall picture of the accelerating forces which items may have to withstand when subjected to a shock wave that just ruptures the hull. Note the greater the mass per unit area of hull the lower the acceleration, while position in the ship has a considerable bearing.

| | Accelerations | Duration in thousandths of a second |
|---|---|---|
| Shell plating ... ... ... ... | Several thousand $g$ | 1 or less |
| Items mounted directly on frames ... | 100-300 $g$ | 2 to 5 |
| Average auxiliary ... ... ... | 100-200 $g$ | 3 to 10 |
| Section of ship vibrating ... ... | 50-100 $g$ | 5 to 15 |
| Ship whipping as a whole ... ... | 2-5 $g$ | Several hundreds |

The motion of typical items such as the above can be seen from Fig. 1, showing velocity/time records as recorded by the seismographic type of velocity meter on actual trials.

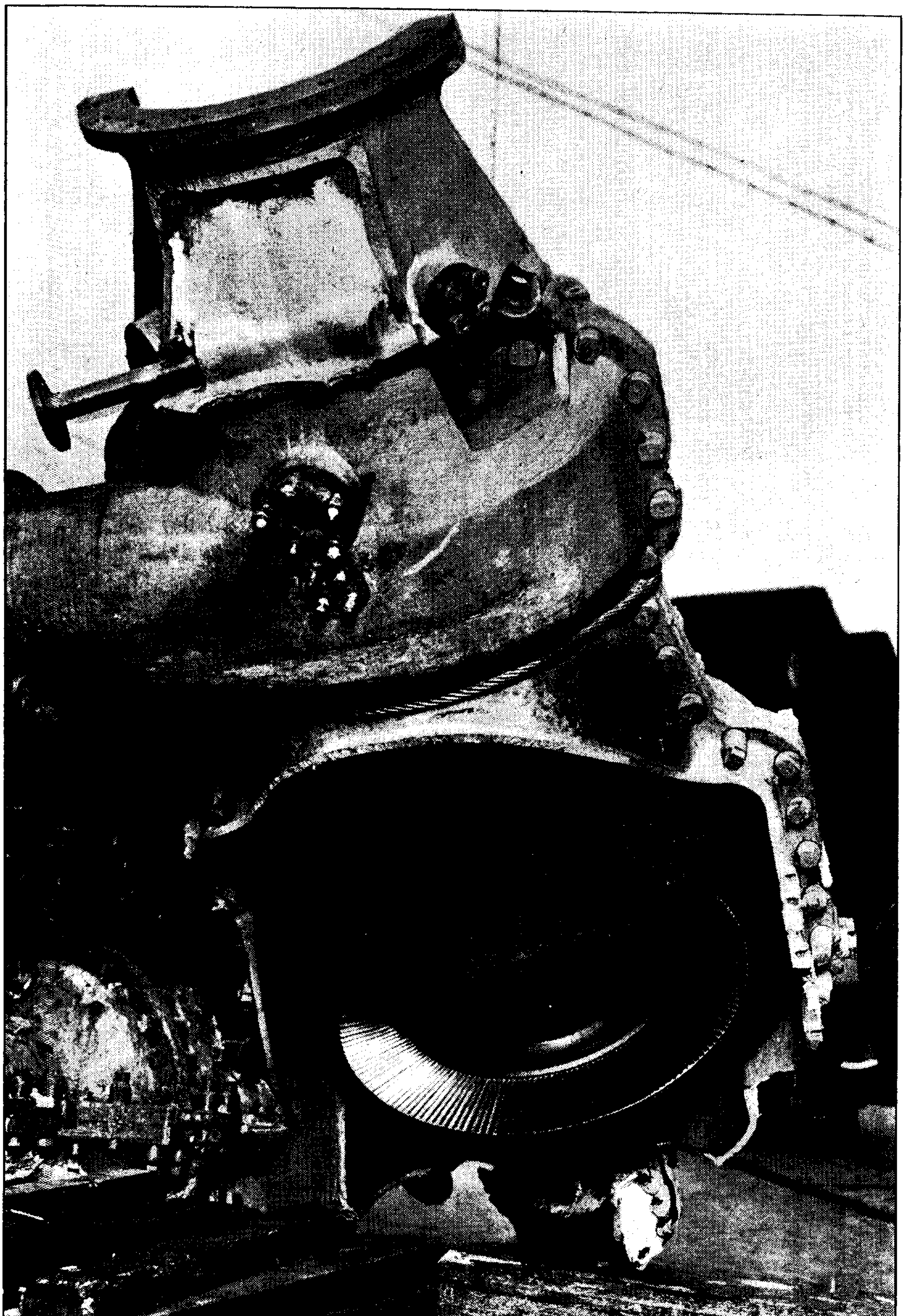

FIG. 3.—L.P. AND ASTERN TURBINE OF A SLOOP

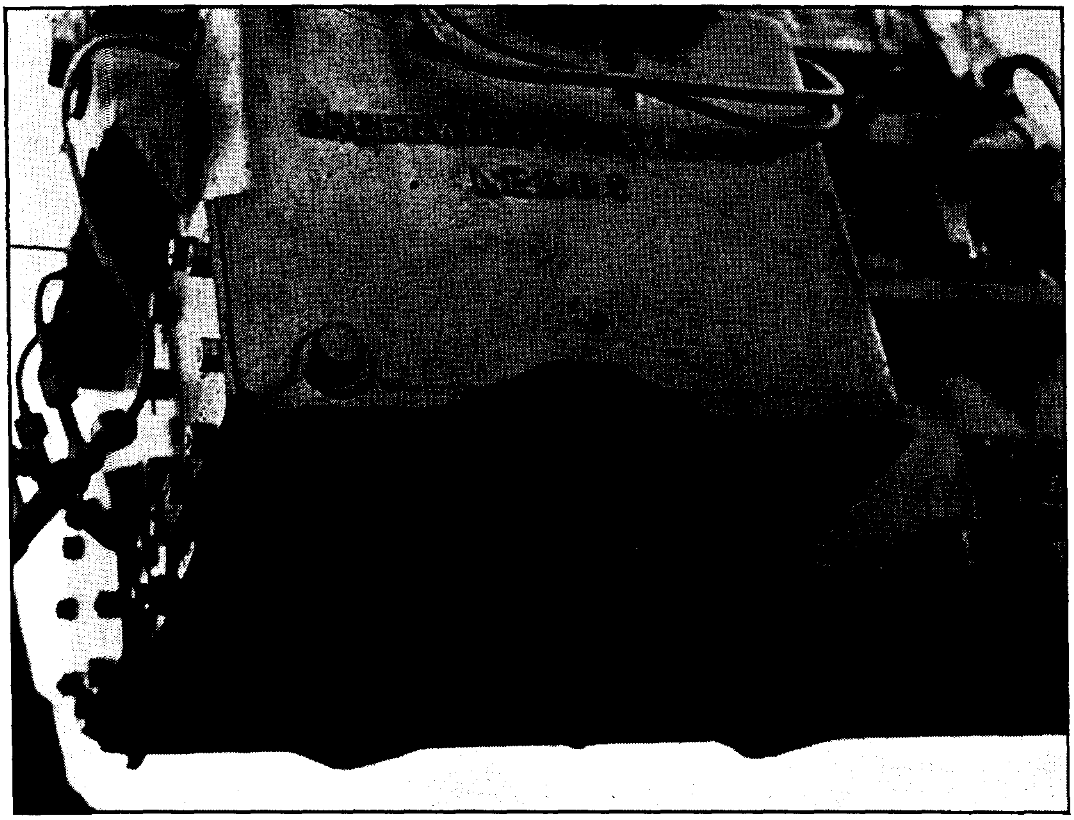

FIG. 4.—FRACTURED BASE OF RECIPROCATING GENERATOR

**Design of Machinery to Resist Shock**

Before going into the details of design it is necessary to view the picture as a whole. A warship is designed according to its class to stand a certain degree of punishment, and what is desired is that all the functions of a fighting ship can be carried out to the moment when the ship is so badly damaged that she can only limp to a refitting port.

It is essential that no one type of damage shall cripple her at an earlier stage than can be helped, nor should the refitting be unduly prolonged owing to the predominance of damage that cannot be readily repaired. Such damage is illustrated in Fig. 2, 3 and 4.

It was in this respect that shock damage was so serious, the replacement of large and intricate castings being a lengthy process and taking a considerably longer period than the hull repairs. In order to obtain a balance of qualities, it has been decided that items should be able to withstand shocks to the point at which uncontrollable flooding occurs in the compartment in which they are situated.

Since the ship is then in a bad way, no factors of safety are required, and in fact it is probably sound design for certain parts of an item to have yielded at this stage provided the machine will still just function and the yielded parts are easily replaceable.

Design figures for accelerations to be worked to are the next essential, and, as an example, the average auxiliary should be able to withstand

120 $g$ upwards,
60 $g$ downwards,
40-60 $g$ athwartships.

Certain other figures for special cases have been promulgated from time to

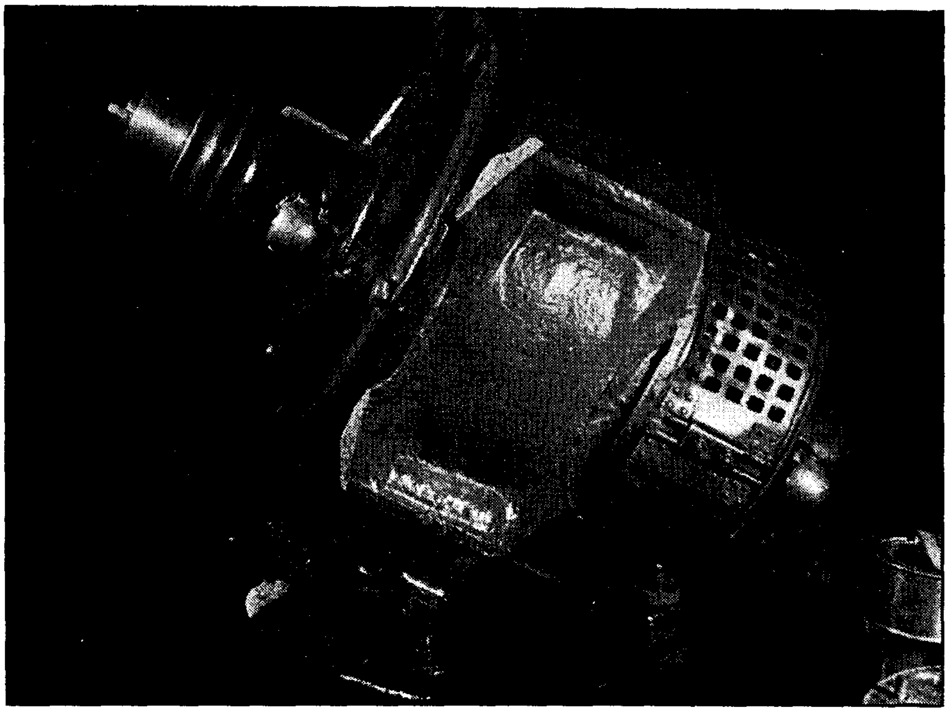

FIG. 5.—FRACTURED BASE OF ELECTRIC MOTOR

time, but it is to be hoped that a more comprehensive list of design figures can be promulgated to machinery designers when the experiments now in hand have been completed.

Next, it is desirable to define more closely the above figures of accelerating forces, etc. Thus an auxiliary machine designed to these figures should be capable of withstanding a force equal to, but not exceeding, 120 times its weight applied to its base for an indefinite period. Next, the base slows down and applies through the holding-down arrangements a retarding force of 60 times the weight of the machine indefinitely. A further requirement is that it can stand the base moving athwartships and applying a force of 40 to 60 times the weight of the machine.

Though, in fact, the accelerations may be of short duration, they may exceed those figures, and they are given as of indefinite period to simplify the design problem.

The stresses caused in the machine by these forces fall into three principal types :—

(i) Direct compressive (or tensile) stress due to the acceleration of a mass through a supporting column ; these seldom cause failure.

(ii) Bending stresses during acceleration due to the force not acting through the centre of gravity of a specific part of the machine—a common cause of failure.

(iii) Bending stresses during deceleration due to the holding-down arrangements acting on the feet of the machine and causing these to fracture or the holding-down bolts to stretch—a very common result (see Fig. 2, 3, 4 and 5).

If the materials used in construction are brittle, then fracture will occur, and it was evident from the first that cast iron was the chief sufferer from this type of damage, whereas ductile materials, which not only had a higher strength but could yield and distort slightly without necessarily putting the machine out

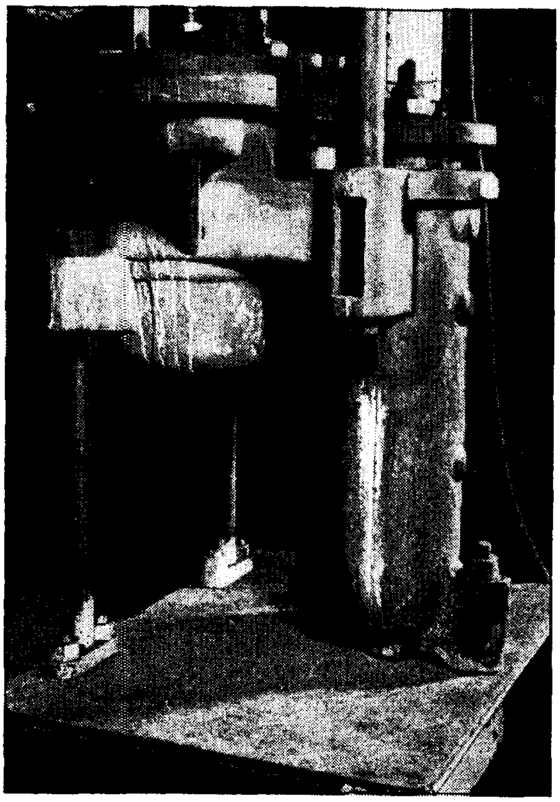

FIG. 6.—NORMAL TYPE OF PUMP BEFORE SHOCK

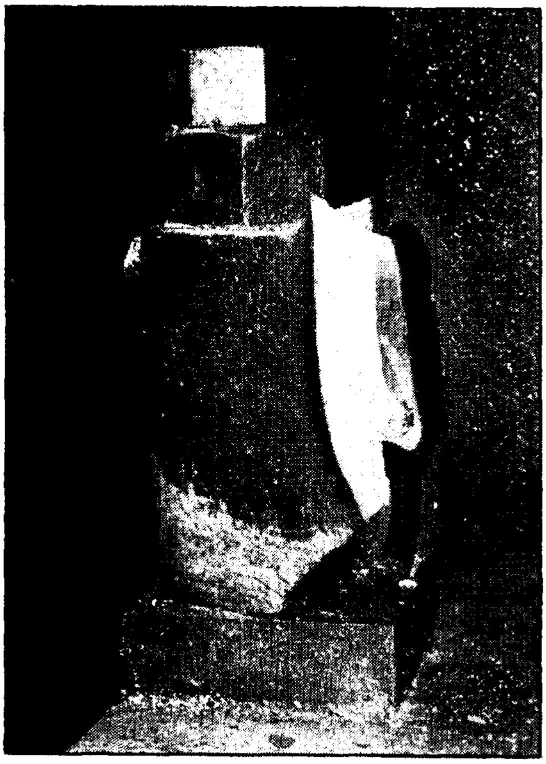

FIG. 7.—NORMAL TYPE OF PUMP AFTER SHOCK

of action, were seldom affected seriously except in cases of very poor design.

The prohibition of cast iron for all except a few items where its properties merit its special retention under due safeguards was the first obvious step, but could not be fully implemented owing to production difficulties in war-time. Cast aluminium is a similar source of weakness ; gunmetal is better, provided a reasonable ductility is assured.

In addition, then, to the avoidance of brittle materials, shock design requires :—

(i) The placing of the feet or supporting chocks and the design of the base so that the forces are transmitted directly to the main mass of the machine.

(ii) Design of the feet or bedplate and its holding-down arrangements so that the decelerating forces will not fracture or distort the main body of the machine.

Stretching of holding-down bolts or distortion of keeps is probably acceptable.

(iii) Suitable design of all overhanging masses and their supports to avoid excessive bending moments.

(iv) Dynamic balance of trip gears and similar mechanisms so that shock does not tend to trip turbo-generators, etc.

Design for (i) and (iii) is not, as a rule, difficult, while (ii) can be safeguarded against by making the holding-down bolts with adequate stretching lengths and considerably weaker than the foot ; similarly, keeps can be designed so that they will yield before the foot breaks. Thus, if 16 tons/sq. in. be allowed for a mild steel foot that should not yield, then the holding-

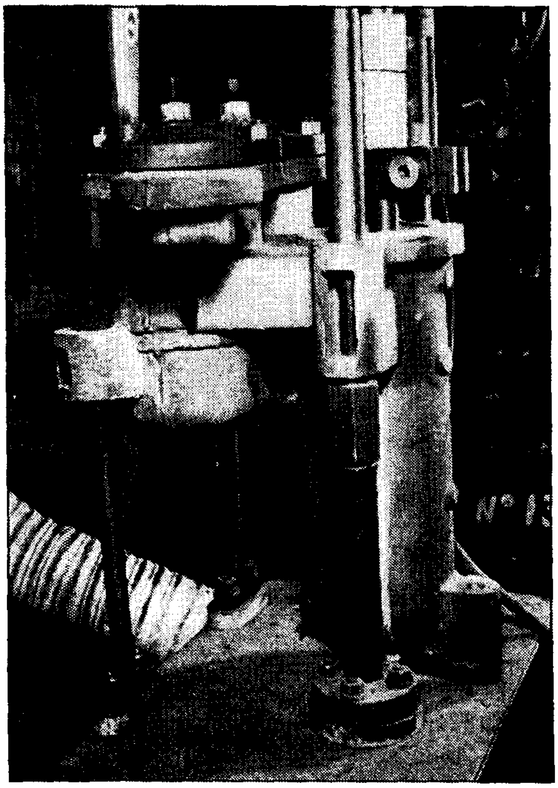

FIG. 8.—PUMP WITH ADDITIONAL SUPPORTS AFTER A MUCH HEAVIER SHOCK

down bolts or keeps should be stressed to 25 tons/sq. in. under these conditions to ensure yielding. Even higher stresses would not be amiss, for, owing to the short duration of the decelerating period, the actual elongation will be very small.

The effect of such design in minimizing damage is clearly shown in Fig. 6, 7 and 8 of an original and modified type of Weir's pump tested on a ship.

It is also good practice to mount auxiliary machinery on a fabricated bedplate or deep girder so that any slight movement of the base does not cause misalignment.

### Main Turbines

As can be seen by inspection of a turbine set and its mountings, the normal turbine is ill-suited to withstand shock forces, and with the cast-iron construction in general use at the beginning of the war many turbine units were wrecked by the magnetic mine and near-miss bombs. Typical damage is shown in Fig. 2, 3, 4 and 9, while Fig. 10 and 11 show diagrammatically the various types of damage that have occurred.

Since a large proportion of the damage was due to the decelerating forces acting on the feet, it is possible to make considerable improvement by fitting yielding keeps to the sliding feet and by adding resilient decelerating washers under the nuts of holding-down bolts, while damage to the gear-case joint was also minimized by reducing the number of bolts and fitting stretching lengths.

The general abolition of cast iron was the other obvious step, and, as far as possible, turbines were either fabricated or made from cast steel. Even so, it was doubtful if it would be possible to design the turbines to withstand the full shock effect in view of the distance of the feet from the centre of gravity, and the rigid resilient mounting (described later) is therefore being fitted under the feet.

With this mounting a severe shock will cause a temporary deflection of, say, $\frac{1}{4}$ in at the end of the turbine remote from the gear case and a lesser deflection at the gear-case end, but the flexible coupling should be capable of taking up this momentary misalignment and continue running at reduced speed should a permanent deflection be left.

### Athwartship Accelerations

In figures issued in 1941, a sideways acceleration of 5 *g* only was given, but higher figures have been worked to for electrical machinery, and it is clear from experimental data that considerable accelerations in athwartships direction (and also fore and aft for items mounted on bulkheads) may be met with from relatively shallow explosions, *e.g.*, near-miss bombs, particularly by items

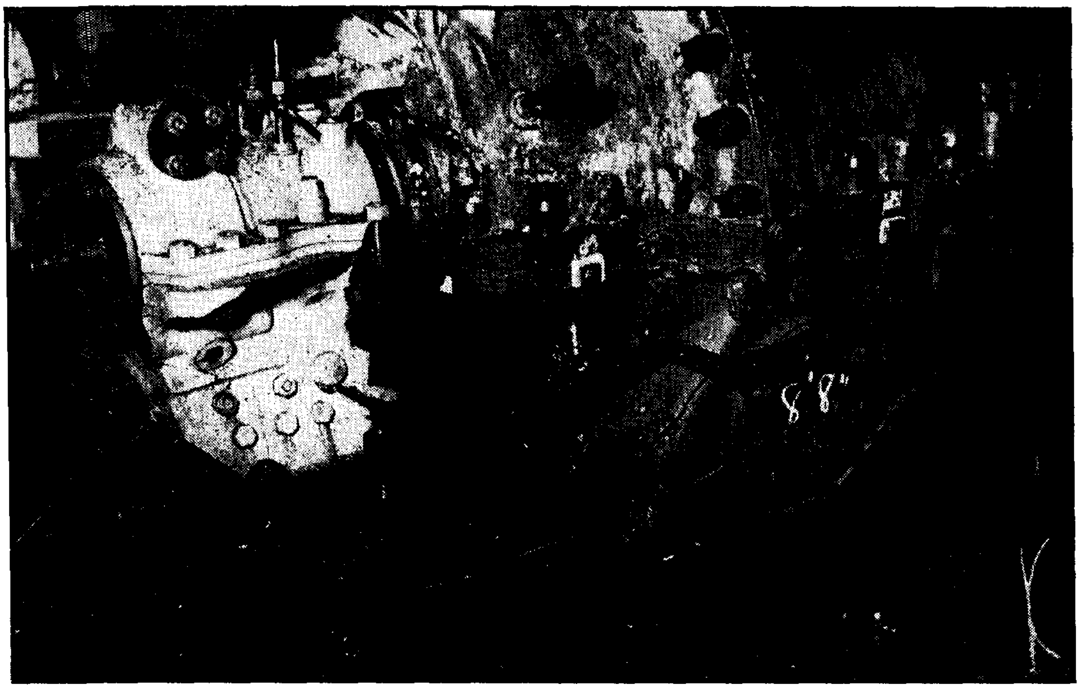

FIG. 9.—L.P. TURBINE OF A SLOOP

mounted on brackets on or near the side of the ship, and it is desired to work up to a figure of 60 $g$ where possible.

In a large proportion of machinery with a low centre of gravity relative to the holding-down arrangements, this sideways force does not raise many difficulties, it only being necessary to ensure that the shear stress in the holding-down bolts is not greatly in excess of the elastic limit—some plastic shearing being probably acceptable—and that under this stress the bedplate does not fracture at the bolt holes.

With the vertical type of machine fitted in many cases, where the centre of gravity is well above the feet, the sideways forces result in an overturning moment which may give excessive stresses in the holding-down bolts. This can best be overcome by lifting the holding-down flange to a position nearer the centre of gravity or widening the base, but where this is impracticable, it is probable that a slight reduction in $g$ value can be accepted, since the machines are generally self-contained and have considerable support at various levels from pipes, etc., while stretching of the holding-down bolts if not excessive, should not cripple the machine.

In this case the machine as a whole, but without the holding-down bolts, should be capable of withstanding the stated acceleration, the feet must be stronger than the holding-down bolts, and the bolts should be able to withstand the acceleration in shear without exceeding the yield point, whilst adequate stretching lengths must be provided.

**Modification to Existing Machinery**

One of the first requirements when this problem arose was to modify the existing machinery in ships to minimize the risk of damage from shock. Since major alterations could not be made, all that was generally possible were makeshift arrangements.

These consisted of the alteration of existing chocks and the addition of extra chocks—in many cases of wood—under the main mass, or the webs supporting

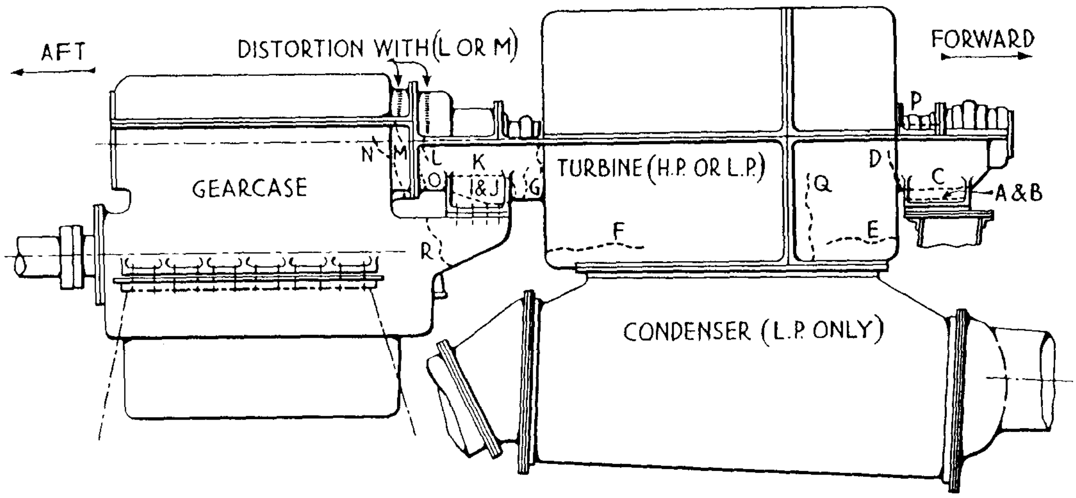


FIG. 10.—DIAGRAMMATIC ILLUSTRATION OF MAIN TURBINE SET, SHOWING FRACTURES

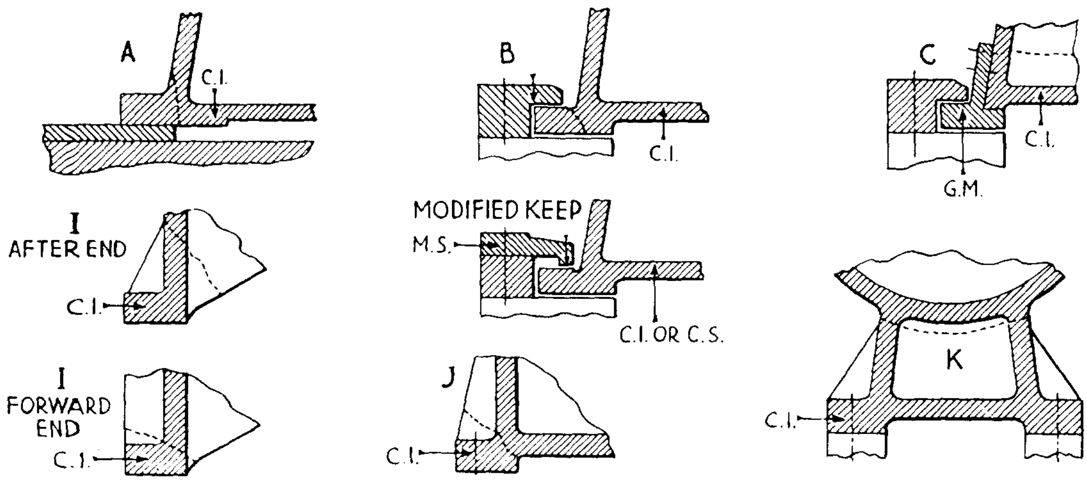


FIG. 11.—ILLUSTRATION OF FRACTURES ON FIG. 10

C.I. Cast iron  C.S. Cast steel  G.M. Gunmetal  M.S. Mild steel

it, so that the forces were transmitted more directly. Overhanging masses had separate supports arranged or were bracketed on to the main body (even this was not always sufficient), while the feet were protected by the provision of stretching lengths on the holding-down bolts, placing flexible washers under the nuts and alterations to keeps so that they would yield under the shock forces anticipated.

Some idea of the value of these modifications can be gathered from reports of twenty-nine ships fitted with yielding keeps to the turbine feet and subjected to shock. In eleven cases the keeps yielded and the turbines were not damaged ; in eight more half the keeps yielded sufficiently to save the turbines, and in the rest they afforded insufficient protection and the turbines were fractured.

## Resilient Mountings

Shock is not the only criterion of design, and it may not be reasonably possible to make a machine for a given purpose capable of withstanding the maximum probable shock forces.

In these cases, although the design has covered shock as far as possible, it may be necessary to find means of reducing the shock forces, and the simple and obvious solution is the insertion of some form of resilient (flexible) mounting between the machine and its seating. The machine and its mounting is then a simple spring system and is acted on by a disturbing force of a period which can be assessed from the experimental results and which is heavily damped after the first half-cycle. Taking any suitable approximation to the force, it is fairly simple to obtain the forced vibration of the spring system that will ensue. In such cases, where the period of application of the load is of the order of one half-cycle only, it is necessary to reduce the frequency of the item on its mounting to be one-half to one-third of the frequency of the applied force if a reduction in acceleration is to ensue.

Equally, the frequency of the machine on its mounting must not equal that of any other disturbing forces likely to be met with in the ship itself, such as the propeller or out-of-balance forces.

Since the disturbing force has a frequency of the order of 100 cycles per second and the propeller (assumed three-bladed) of about 15, it is seen that a frequency of between 25-35 cycles per second should be satisfactory.

In many cases it is not desirable to mount machinery on a simple resilient pad, since misalignment may result or the out-of-balance vibration may be excessive, and two alternative types of rigid/resilient mountings have been used for these cases.

(i) Shearing devices consisting of metal sleeves of form that fail in shear at a predetermined load. These are fitted in combination with resilient pads and ensure that the machine is rigid until under shock conditions the load exceeds a certain value ; the device then shears and the machine is thereafter resiliently mounted.

(ii) Corrugated plates of iron which crush uniformly under a given load. These again have resilient pads in parallel with them and also absorb energy in crushing.

The shearing devices did not prove highly satisfactory in service, as it was found that, owing to the variations in the forces at different parts of a machine, the devices did not all shear simultaneously and caused excessive bending moments and fractures at the unsheared supports.

The corrugated plates are considerably more satisfactory in this respect, since the variations in deflection of different parts of a given machine are smaller, but, nevertheless, relative deflection will occur both between the machine and its seating and possibly between parts of the machine itself, and consideration must be given to this before fitting such mountings.

With all these mountings the resilient pads are designed to bring the machine back approximately to its original position except for any permanent set due to excessive loading. Further, the resilient pads themselves may be liable to creep under permanent loading and cause misalignment.

The fitting of rigid/resilient mountings needs care if good alignment is to be maintained, and a high standard of accuracy in the plates and seatings is required.

### Choice of Materials for Resilient Pads

For this particular purpose a spring is required that will deflect, say, $\frac{1}{8}$ in to $\frac{1}{2}$ in under a load of, say, forty times the weight of the machine. The material must have a sufficiently long life and resistance to abrasion, oil and heat, and not be subject to creep under prolonged loads. A fairly high damping factor is also useful in preventing resonance conditions and also in absorbing part of the energy transmitted.

Since weight is an all-important factor, a simple rubber pad loaded in compression is in general the most suitable, the strain energy stored per unit volume being greater than in a spring involving bending stress. Steel springs are not satisfactory for this reason and are cumbersome. The use of a plain compression mounting also avoids any bottoming effects with a sudden sharp rise on the accelerations.

Many types involving rubber in shear have also been suggested, but this involves considerable reliance in the bonding of metal to rubber, which, in view of shipboard conditions, is undesirable unless any appreciable advantages are gained.

For heavy items a fairly hard rubber is desirable to keep areas reasonable, and types such as Dexine 118 or Walker's Questo, having a shore hardness of 80–85 deg are suitable. With these, the compression curves are fairly straight up to 30 to 40 per cent compression, when they gradually steepen. If noise suppression is also important, or when dealing with light and fragile parts, a much softer rubber is needed.

Nevertheless, deterioration of the rubber is a problem, and, though coated with oil-resisting varnish and fitted in oil shields, their life under adverse conditions is not good. The exercise of a little care in design would, however, obviate much of this—one cannot expect a rubber pad to last when placed inside the saveall of a lubricating or fuel oil pump, particularly when fitted with oil shields, which in this case act as oil baths for the rubber.

### Rubber Resilient Pads

In theory, all that is required is to obtain a mounting of a known frequency when loaded with a known mass, but the use of material such as rubber leads to further complications in that the load deflection curve is not straight, while other factors have a considerable influence on the deflection under load, namely :—

(i) Form factor—a plate of rubber with a large area under load and small free surface will be in effect stiffer than in a case where by cutting the plate into pieces the free surface is increased.

(ii) Effect of friction between surfaces of rubber and feet of item. The stiffness of a rubber mounting varies considerably between the extremes of well vaselined smooth surfaces and rough machined ones. A bonded rubber-to-metal joint would have the advantage of constancy but would lose the damping effect due to this friction.

(iii) Effect of dynamic loading on the frequency of rubber mountings. It is found that a rubber mounting when acted on by a pulsating load has a frequency that increases with the frequency of the load.

As a result of these conflicting factors, it is only possible to specify an approximate area/ton supported for these mountings, based on experimental results, and further investigations are in hand to obtain check figures.

### Decelerating Washers and Stretching Lengths in Holding-down Bolts

The placing of a simple rubber pad under the holding-down bolts (between suitable metal washers) was an obvious means of lessening the decelerating forces on feet and holding-down arrangements. This in many cases was all that was necessary to safeguard the machine since the feet in decelerating are usually the weakest part. The areas of rubber fitted are generally lower than that of the accelerating type of pad, as the maximum decelerating forces met with are usually lower than those in acceleration.

Since no weight is taken on the washers, questions of creep and vibration do not normally arise, but the washers should not be screwed down unduly tightly when fitted.

Another method of limiting the decelerating force on the feet is the use of stretching lengths in the holding-down bolts, and this can be fitted in conjunction with decelerating washers or without them where a more rigid arrangement is required.

The diameter of the stretching length should be sufficiently below the diameter at the bottom of the thread to ensure that notch effect does not cause fractures in the thread. The elongation of the stretching length will absorb a very considerable amount of energy before fracture occurs, besides making it simpler to design the feet so that fracture of the feet shall not occur. In order to ensure this, it is usual to allow a figure of 24-25 tons/sq. in. for the yield point of mild steel for bolts that are to yield, while allowing 16 tons/sq. in. for the feet that should not yield.

### Dynamic Yield Strength of Materials

It has for some time been known that the rate of application of the load affects the yield point of a metal quite considerably. Since this is an important factor in cases such as those under discussion, investigations have been carried out to find the yield point at various rates of loading, and the principal results to date have been published in various technical papers. Further tests are envisaged in which not only the rate of application, but the duration of the load, is varied.

For the moment, however, it seems best to leave this increase of permissible stress as a margin of safety in the design.

### Shock Testing Machinery

In view of the complex nature of the problem, it is clear that calculation alone cannot ensure a wholly satisfactory design, and some means of testing typical machines up to the design limit is necessary. Full-scale tests on ships are not satisfactory for this, since the vessels chosen, and their machinery, are normally obsolete, nor do more than one or two items get the full intensity of shock from a damaging shot.

The ship trials do, however, give us the necessary data on shock effects if we can find some means of reproducing these in a test shop.

Shock machines for testing electrical gear up to $\frac{1}{2}$ ton in weight have been installed at the Admiralty Experimental Laboratory. These are of the swinging hammer type, the intensity of shock being governed by the height the hammer drops.

This gives satisfactory results for the smaller items of gear and has enabled considerable investigation to be made into the movements of switchgear under shock, and very considerable improvements in design have resulted from its use.

For the heavier machinery items, however, the problem is somewhat more difficult, but endeavours are being made to design a testing machine suitable for this purpose on which typical machines could be tested out and the most economical shock resistant design worked out.

### Conclusion

Though there is much yet to learn concerning the forces brought into play by non-contact explosion, it is possible to make reasonable approximations to their magnitude and duration. The design of machinery to withstand such forces then requires only the application of straightforward principles of mechanics to obviate the major casualties that have occurred due to this cause.

The design problem is greatly simplified if brittle materials can be replaced by ductile ones, but where other considerations render this course undesirable much can still be done by avoiding the obvious causes of fracture.

Resilient mountings are a further step in the safeguarding of machinery, but in their turn involve upkeep problems, and the possibilities of improved design to lessen this must be borne in mind. Progress, owing to the difficult nature of trials, must, however, be slow, and it is essential to do what is possible now to improve design against this danger if serious damage in any future conflict is to be avoided.

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**Modifications to existing Machinery**

The method of protecting existing auxiliary machinery from shock damage is dealt with in C.A.F.O.2248/41 and its accompanying diagram 464/41.

The mountings described are designed to have a frequency of 30-50 c.p.s. in order that vibration troubles may be minimized while the mounting still fulfils its functions as an isolator of shock.

When fitting the mountings it is essential that adequate flexibility is allowed in pipe connections, etc. A valve chest secured to a bulkhead and connected to a pump by a short length of pipe should, for instance, be flexibly mounted by fitting resilient washers to the holding-down bolts.

The protection of the feet of main turbines is covered by C.A.F.Os. 211/41 and 575/41. C.A.F.O. 1151/41 refers to the fitting of decelerating pads.

A description of rigid resilient mountings for auxiliary machinery is given in C.A.F.O. 392/43. Mountings of this type are used for main engines and the larger auxiliaries of new construction ships when to make these machines sufficiently strong to be shock-resistant would involve an unacceptable increase in weight. The main disadvantage of this type of mounting is that renewal of the corrugated plates is required after a severe shock. In any case it will be necessary to inspect and in many cases to tighten the holding-down bolts of machinery after severe shocks although the hull may have suffered no apparent damage.

Experience, so far, indicates however that turbo-auxiliaries will continue to run without vibration even after the holding down bolts have been stretched as much as $\frac{3}{16}$ in. by an under-water explosion.—EDITOR.