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Boiler Corrosion and Boiler Feed Water Treatment

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The fundamental factors of the electro-chemical action which causes corrosion and the principal types of corrosion inside boiler equipment are described and illustrated. It is shown that the vital factors in boiler corrosion are the hydrogen ion concentration and the presence of dissolved oxygen. The physical facts concerning hydrogen ion concentration and the removal of dissolved oxygen are discussed and equipment for providing feed water free from oxygen is described and illustrated.

The paper deals with the factors affecting the electrode potential of iron and the conditions necessary to obtain immunity from corrosion, also the physical phenomena of the conditioning of boiler water to prevent the formation of scale and to provide the conditions which prevent corrosion.

Basic data are given and discussed concerning the substances and reactions involved in feed and boiler water conditioning and the physical phenomena of scale formation, solubilities of substances and the physico-chemical interchanges in water conditioning.

The basic facts of carbonate and phosphate conditioning, colloidal and coagulant treatment, oxygen scavenging, caustic cracking, etc., are dealt with together with the control of the alkalinity or hydrogen ion concentration of the boiler water.

THE CORRODIBILITY OF METALS

The disintegration of a metal, or corrosion, is a phenomenon which is experienced in widely different and varied forms but the cause of corrosion is fundamentally the same in all cases.

Metals which are liable to corrosion, such as steel and aluminium, are obtained from metallic oxides which are processed to give the desired metals, and the greater the energy necessary to produce the metal from the metallic ore, the greater is the tendency of the metal to return to its original state of a metallic oxide, due to corrosion. For example, gold does not corrode because it is already in a pure metallic state when mined. On the other hand, a large expenditure of energy is necessary in the manufacture of aluminium and magnesium, and these metals have a strong tendency to revert to their original states.

Corrosion is liable to take place whenever moisture or liquid is in contact with a metal, because the metal tends to revert to its original form of a metallic oxide under the influence of oxygen or into metallic salt compounds under the action of salts or acids.

The effect of oxygen on the surface of a metal is to cover the metal with an extremely thin coating of oxide, which does not necessarily lead to a destructive corrosive action but may, on the contrary, set up a state of passivity, thereby preventing further corrosive action.

The baser the metal, such as aluminium, for example, the more easily is such a metallic oxide coating formed and the more stable the coating is. For such a coating to give protection against destructive corrosion, it must be stable, must adhere firmly to the metal and must entirely cover the surface with a non-porous coating. These requirements are obtained only when the oxide coating is insoluble in the liquid in contact with the metal. The same desirable result may, however, be obtained with a coating which is self-repairing.

Metals such as aluminium and chromium have a strong tendency to form a protective oxide covering, whereas the oxide found on the surface of steel is generally porous and does not, therefore, ensure protection against further corrosion. The reddish brown coatings of rust which form on steel, and also the coatings of mill scale which form at high temperatures, not only do not prevent progressive corrosion, but, for the most part, even promote it.

The process which is usually considered to be the major cause of destructive corrosion, is the electro-chemical phenomenon which takes place when a metal is in contact with an electrolyte and two galvanic elements are present in the liquid, frequently on the surface of the same piece of metal.

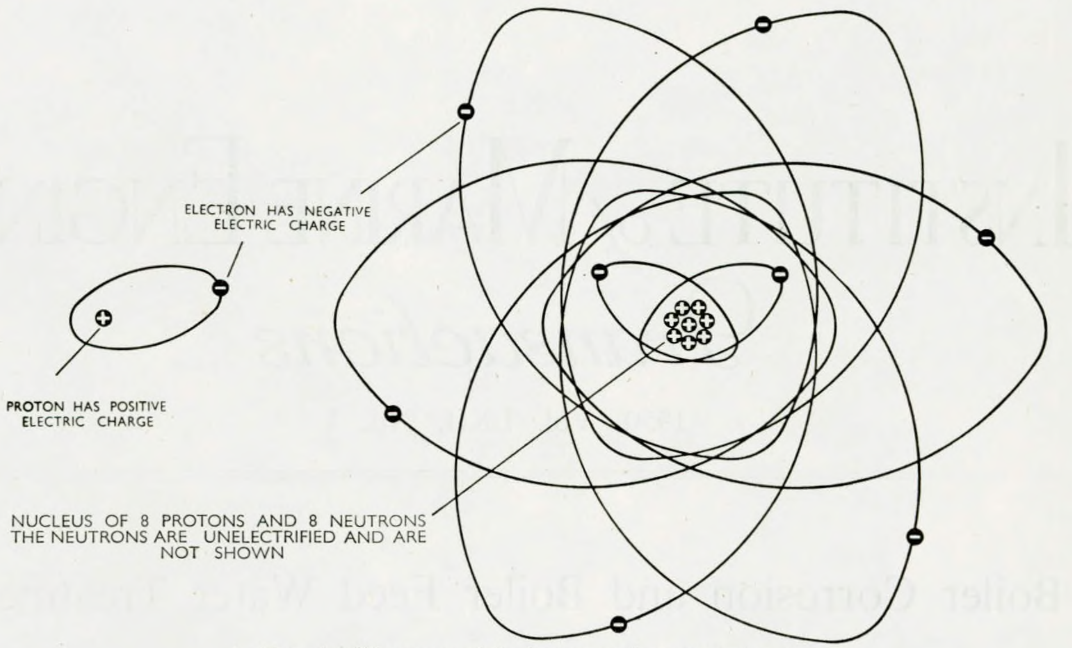


FIG. 1—*Diagrams of hydrogen and oxygen atoms*

The atomic number of an element is the number of protons in an atom of the element. The number of electrons in a normal atom is the same as the number of protons. The positive electric charge of each proton is equal in magnitude to the negative electric charge of each electron so that the resultant electric charge of a normal atom is zero.

The galvanic or local elements may be provided by differences in the chemical composition, segregations, inclusions or other causes which create variations in the surface structure of the metal. They may be provided by two or more dissimilar metals in contact with the same electrolyte. Local adhesions, such as oxide coatings, corrosion products, deposits of fungi, scale, coal, salt, particles of other metals, etc., may provide one of the elements. The electro-chemical action may be caused by different ventilation of parts of the metal surface, differences in concentration in the electrolyte, difference of temperature, currents, etc.

When two galvanic elements of different potential are present in an aqueous solution, due to any differences such as are mentioned above, there is a flow of electric current from

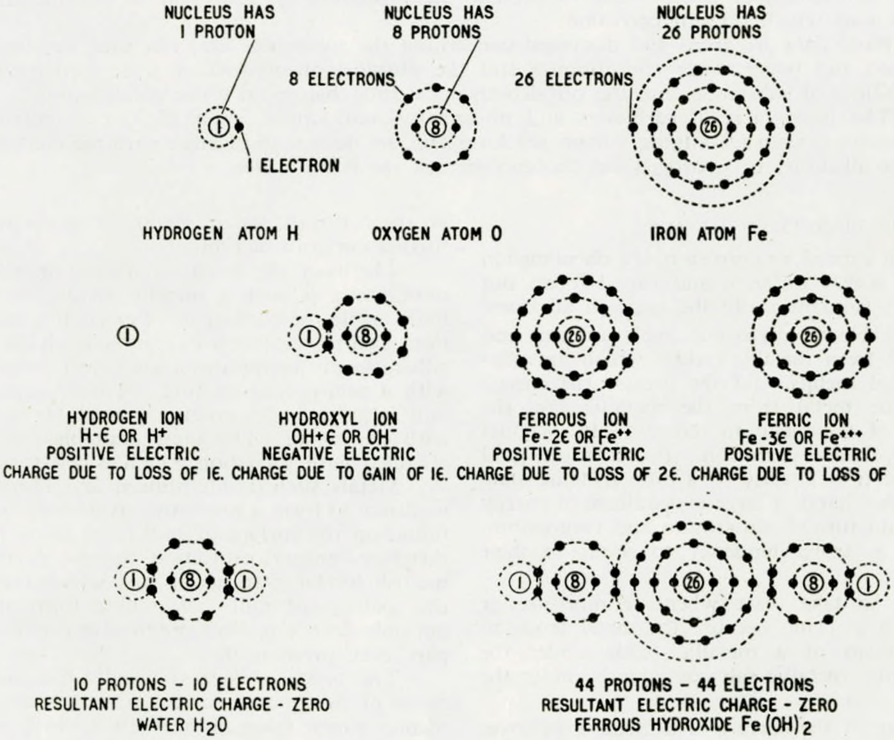


FIG. 2—*Some atomic and ionic groupings of hydrogen, oxygen and iron*

the base or anodic element to the noble or cathodic elements which may be present on the surface of the same piece of metal. Where the metal forms the anode of the couple, metal ions are dissolved into the liquid at the anode, where they combine with hydroxyl ions to form a metallic hydroxide, which is frequently insoluble, while hydrogen ions of the electrolyte migrate to the cathode, where they are discharged and released from the cathodic surface. Within the metal there is a flow of electrons from the anode to the cathode.

To understand this action it is necessary to have a picture of the structure of an atom and the electrical phenomena connected with it.

ATOMIC AND IONIC STRUCTURES AND ASSOCIATED ELECTRICAL PHENOMENA

An atom is composed of a number of protons and neutrons (an hydrogen atom is an exception in having no neutron) associated together in a close compact aggregation which is called the nucleus of the atom. Round this nucleus a number of electrons travel in orbits. An atom of any particular substance has a definite normal complement of electrons which is the same in number as the number of protons in the atom. Each proton has a positive electric charge and each electron has a negative electric charge, while the neutrons are unelectrified. When the normal complement of electrons is present, the electric charges of the protons and electrons balance each other and the atom has no resultant electric charge.

The smallest and simplest atom is the hydrogen atom which has one proton and one electron as shown diagrammatically on the left of Fig. 1. The oxygen atom which is shown on the right of Fig. 1, has eight protons in its nucleus, with eight electrons each travelling in its own orbit. The system is similar to the solar system with the nucleus as the sun and the electrons revolving round the nucleus like planets, the electrons being held in their orbits by the attraction between them and the protons, due to their opposite electric charges. According to the Bohr theory, the orbit of each electron is in a different plane, but they fall into groups which form what are known as space shells. The arrangement of the space shells of the electrons of atoms can be shown by conventional diagrams which show the electron orbits in the same plane. Fig. 2 gives such conventional diagrams for some of the atoms which are of interest in the subject under consideration. The space shells, when fully occupied, contain as a maximum, two electrons in the first space shell, eight electrons in the second, eighteen in the third and thirty-two in the fourth, which is as far as need be considered here. Atoms of each particular substance have a definite normal complement of electrons but an atom may, however, acquire one or more electrons, or it may lose one or more electrons, as compared with its normal complement.

When an atom gains an electron or electrons, it has a resultant electric charge which is negative; when it loses one or more of its normal number of electrons, its resultant electric charge is positive.

An atom or aggregation of atoms which has a greater or smaller number of electrons than the normal complement, is known as an ion and it possesses a negative or a positive electric charge respectively. Diagrams of an hydrogen ion, an hydroxyl ion, and ferrous and ferric ions, are shown in Fig. 2.

Ions or bodies with similar electric charges repel one another, while ions or bodies with opposite charges have an attraction for each other, so that, where ions or bodies are in contact with one another, there is a continual tendency towards electrical equilibrium by the interchange or sharing of electrons. This is illustrated in Fig. 2, by the combination of two hydroxyl ions, with negative charges, with a positively charged ferrous ion to form ferrous hydroxide which is neutral. When a compound of atoms is formed there is a similar sharing of electrons as illustrated in Fig. 2 by the combination of hydrogen and oxygen atoms to form water.

ELECTRO-CHEMICAL CORROSIVE ACTION WITH HYDROGEN EVOLUTION

Fig. 3 illustrates electro-chemical action causing progressive corrosion due to the evolution of hydrogen as a gas. The electrolyte shown is water in which there is always an extremely small quantity broken up into two kinds of ions having opposite electrical charges.

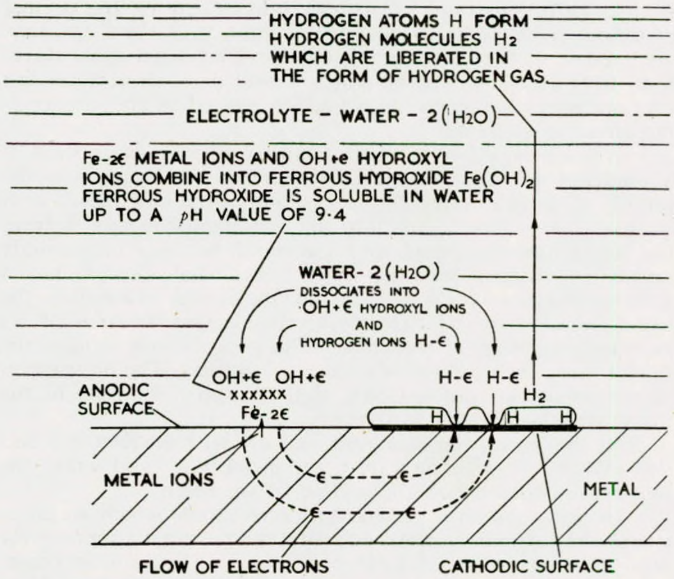


FIG. 3—*Diagram of electro-chemical action with hydrogen gas evolution causing corrosion of iron in an aqueous solution*

A polarizing layer of hydrogen is formed at the cathode, but a difference in potential exists between the liquid and the cathodic surface and hydrogen gas is liberated from the cathodic surface. In the absence of dissolved oxygen this can occur in liquids having *p*H values up to 9·4.

In the electro-chemical action, further dissociation occurs from the water H_2O into the two kinds of ions, hydroxyl ions $OH+e$ and hydrogen ions $H-e$, which are electrically charged ions with a negative and positive charge respectively. Each hydroxyl ion has one electron more than the normal hydroxyl compound and each hydrogen ion is without the electron possessed by a normal hydrogen atom. At the first opportunity each hydroxyl ion will part with its surplus electron and each hydrogen ion will acquire an electron. (Ions with their electrical charges are more usually written thus, H^+ and OH^-, the electrical charge being shown by the positive or negative sign.)

Due to differences, such as have been mentioned, there is a difference in potential between two parts of the surface of the metal in contact with the liquid and a current of electricity flows between the anodic surface and the cathodic surface. In the liquid the current of electricity is, by convention, said to flow from the anode to the cathode, while in the metal, the current is said to flow from the cathode to the anode.

The conduction of electricity through a liquid consists of the motion of both kinds of ions moving in opposite direction, the positive ions moving in the conventional direction of the current, while the negative ions move in the opposite direction. The conduction of electricity through the metal, however, consists entirely in the movement of electrons which pass through the system or nuclei of the metal from the anode to the cathode, which is the opposite direction to the conventional direction in which the electrical current is said to flow.

For the purpose of this paper, it is convenient, and makes for clarity, to consider only the movements of the ions and the electrons, and Fig. 3 illustrates the migrations of ions and electrons and the atomic and molecular changes which occur with electro-chemical action in an aqueous solution. The greater the difference in potential between the galvanic elements, the greater is the rate of movement of ions and electrons and the atomic and molecular changes.

A metal consists of metal ions arranged in a recurring geometrical pattern, called a lattice, and held together by a cloud of mobile electrons. Each metal ion, considered by itself, is short of one or more of the electrons which are the normal complement of an individual metal atom and these electrons are present in the surrounding cloud. The metal ions, therefore, have a positive charge which would drive them apart but they are held together by the attraction exerted by the surrounding cloud of electrons.

When a liquid is in contact with a metal surface there is a tendency for the passage of ions from the metal into the liquid. This is a reversible process, but with a base metal, such as iron in an aqueous solution, the dominant direction is from the metal into the liquid and the metal becomes increasingly negatively charged because, as it loses metal ions, it has a growing surplus of electrons, while the liquid adjacent to the metal becomes increasingly positively charged, because of an increasing shortage of electrons. This progressively reduces the loss of metal ions until equilibrium is reached. The destruction of the metal can proceed only if the surplus electrons in the metal are passed into the liquid.

In the liquid, the hydrogen ions are without electrons and will take surplus electrons from the metal when and where the ions make contact with the surface of the metal.

In doing so, they become hydrogen atoms which accumulate at the cathodic surface and alter its potential progressively until the movement of hydrogen ions to the cathode ceases unless the hydrogen atoms are removed from the surface of the metal by the formation of hydrogen gas bubbles. They may be removed by other means such as oxidization due to the presence of dissolved oxygen in the electrolyte.

The formation of hydrogen gas bubbles can occur if the acidic or hydrogen ion concentration at the cathodic areas gives a sufficient difference of potential compared to the metal potential at the anodic areas. With iron, this occurs freely with hydrogen ion concentrations corresponding to *p*H values up to about 5. As the *p*H value is increased above 5 the hydrogen potential at the cathodic surface falls until, for practical purposes, hydrogen evolution ceases with a *p*H value of 9·4. The mechanism of this action is discussed in detail later.

The escape of the molecular hydrogen from the cathodic surface permits a continuous flow of hydrogen ions to the cathode with a corresponding continuous removal of surplus electrons from the metal at the cathode. This allows a continuous passage of metal ions into the liquid at the anode so that the corrosive action at the anode is a progressive one. The greater the potential difference existing between anodic areas of the metal and the cathodic areas, the more vigorous is the corrosive action on the metal.

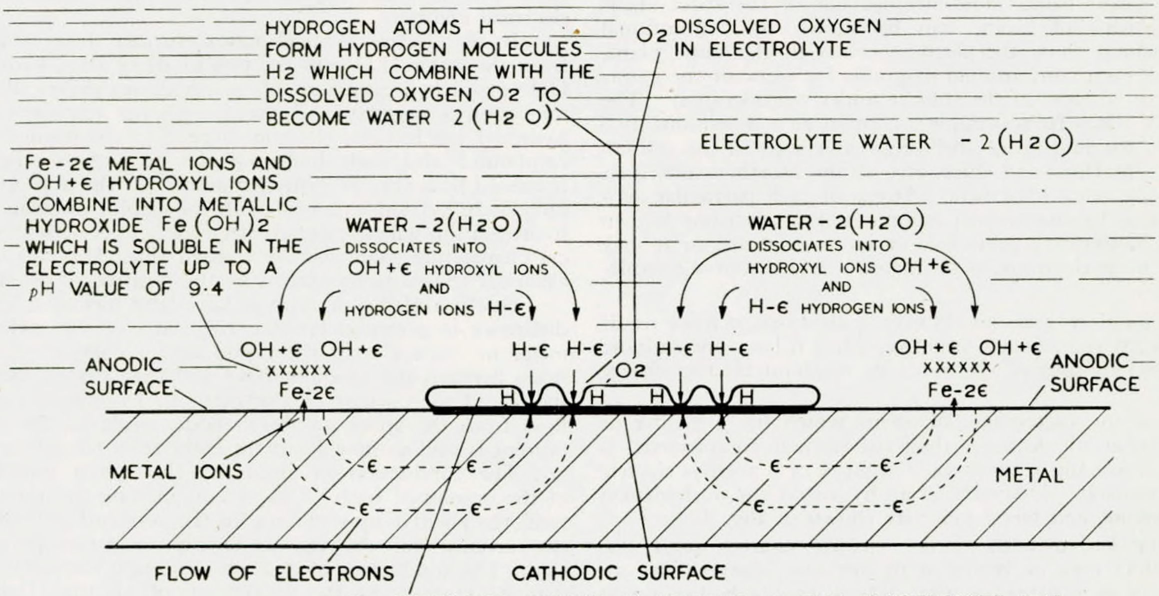
As shown in Fig. 3, as the metal ions $Fe-2e$ are dissolved into the liquid, they meet and combine with the hydroxyl ions, $OH+e$, which are migrating to the anodic surface, to form ferrous hydroxide $Fe(OH)_2$, the surplus electrons of the hydroxyl ions balancing the shortage of electrons in the metal ions to form a neutral compound. At the same time, the hydrogen ions, $H-e$, migrate to the cathodic surface, where they receive electrons from the metal to form atomic hydrogen on the cathodic surface.

In the usual aqueous solution, such as sea water or boiler water, the hydrogen atoms tend to form an insulating film on the cathodic surface which would stop the flow of the current of ions and electrons, an action which is called polarization.

If, however, the electrolyte is sufficiently acid, the hydrogen atoms will leave the cathodic surface as free hydrogen gas and the continuous release of hydrogen will permit the electro-chemical action to continue with progressive corrosion at the anode. With corrosion due to hydrogen evolution, the anodic points of attack are constantly changing, so that it is characterized by a general wastage of the surface of the metal.

ELECTRO-CHEMICAL CORROSIVE ACTION WITH OXYGEN REDUCTION

The corrosion which occurs with electro-chemical action can also be maintained by a supply of dissolved oxygen in the



WHEN DISSOLVED OXYGEN IS PRESENT IN THE ELECTROLYTE THE HYDROGEN WHICH TENDS TO FORM A POLARISING LAYER ON THE CATHODIC SURFACE IS REMOVED BY THE DISSOLVED OXYGEN COMBINING WITH THE HYDROGEN TO FORM WATER. IF THE ELECTROLYTE IS STRONGLY ACID THE HYDROGEN CAN ALSO ESCAPE IN THE FORM OF HYDROGEN GAS. WITH IRON THIS OCCURS WITH A *p*H VALUE UP TO ABOUT 5·0.

FIG. 4—*Diagram of electro-chemical action with oxygen reduction causing corrosion of iron in an electrolyte containing dissolved oxygen*

electrolyte. When dissolved oxygen is present, the atomic hydrogen H, which forms at the cathodic surface, coalesces into hydrogen molecules H_2 which combine with the dissolved oxygen O_2 to form water $2H_2O$, thereby removing the atomic hydrogen from the cathodic surface. This action is illustrated in Fig. 4 in which the general action is similar to that shown in Fig. 3 except that the removal of the hydrogen at the cathodic surface is effected by oxidization, the dissolved oxygen in the electrolyte combining with the hydrogen on the cathodic surface to form water, so that the hydrogen is removed from the surface of the cathode. The formation of ferrous hydroxide at the anodic surface removes the metal ions from the metal surface as the corrosive action. The continuous removal of the hydrogen from the cathodic surface by oxidization permits a continuous flow of hydrogen ions to the cathode with a corresponding continuous removal of surplus electrons from the metal at the cathode. This allows a continuous passage of metal ions into the liquid at the anode so that the corrosive action at the anode is a progressive one. The presence of dissolved oxygen in the liquid will remove hydrogen from cathodic surfaces in liquids at all practical *p*H values and it will be appreciated easily that the corrosive action associated therewith is directly proportional to the rate at which free oxygen is supplied to the liquid.

The removal of hydrogen from cathodic surfaces, with the corresponding corrosive action at the anodic surfaces, can be affected by hydrogen gas evolution and oxidization in action at the same time, provided that the liquid has a suitable acidity to permit hydrogen gas evolution.

It will be seen that corrosion depends upon the removal of hydrogen atoms from the cathodic surface areas either by hydrogen gas evolution, or by oxygen reduction with the formation of water. Fig. 3 illustrates the hydrogen evolution action which occurs when the liquid is acid in character and shows how the corrosive action is independent of the presence of dissolved oxygen. This type of corrosion usually wastes an area away in an even manner and may occur locally in a boiler if a pocket of water is sufficiently acidic. This may be caused by the presence of chlorides due to the ingress of sea water. When dissolved oxygen is present in the liquid, oxygen reduction will remove the atomic hydrogen from the cathodic areas as shown in Fig. 4. If the liquid is sufficiently acid and dissolved oxygen is also present, the hydrogen atoms will be removed from the cathodic areas by hydrogen gas evolution as well as oxygen reduction, so that the two actions can cause a very vigorous corrosive action of the metal at any anodic areas.

Electro-chemical action can be arrested by the polarization of the cathodic surface or by the stifling or blanketing of the anodic surface, either of which would have the effect of stopping the corrosion of the metal.

The presence of dissolved oxygen in the water depolarizes the cathode by oxidizing the insulating film of hydrogen as it forms. At the anodic surface, if the metallic hydroxide is formed close to the anodic surface, a protective oxide coating may be formed on the metal which will check the galvanic action, but if the metallic hydroxide is formed an appreciable distance away from the surface, or is porous or loose in its form, the action will continue.

As previously mentioned, oxygen may promote a protective oxide coating on a metal, thereby preventing destructive corrosion, but it may also cause the destructive corrosion which is usually experienced in feed systems and boilers when there is an appreciable supply of oxygen.

Protective layers of calcium salts, phosphates, etc., may be formed on boiler surfaces, etc., but they cannot be relied upon to form a non-porous protection. The acidic type of attack can be met by conditioning the boiler water to have the necessary alkaline character and consideration of the above then leads to the very important conclusion that progressive or destructive corrosion in a feed system or a boiler is then solely and entirely dependent on the presence of and a continuous supply of oxygen at the cathode of the local element.

This stresses the extreme importance of supplying a boiler with feed water which has been passed through a regenerative type condenser or a de-aerator to reduce its oxygen content to the lowest possible amount. For high pressure boilers, in par-

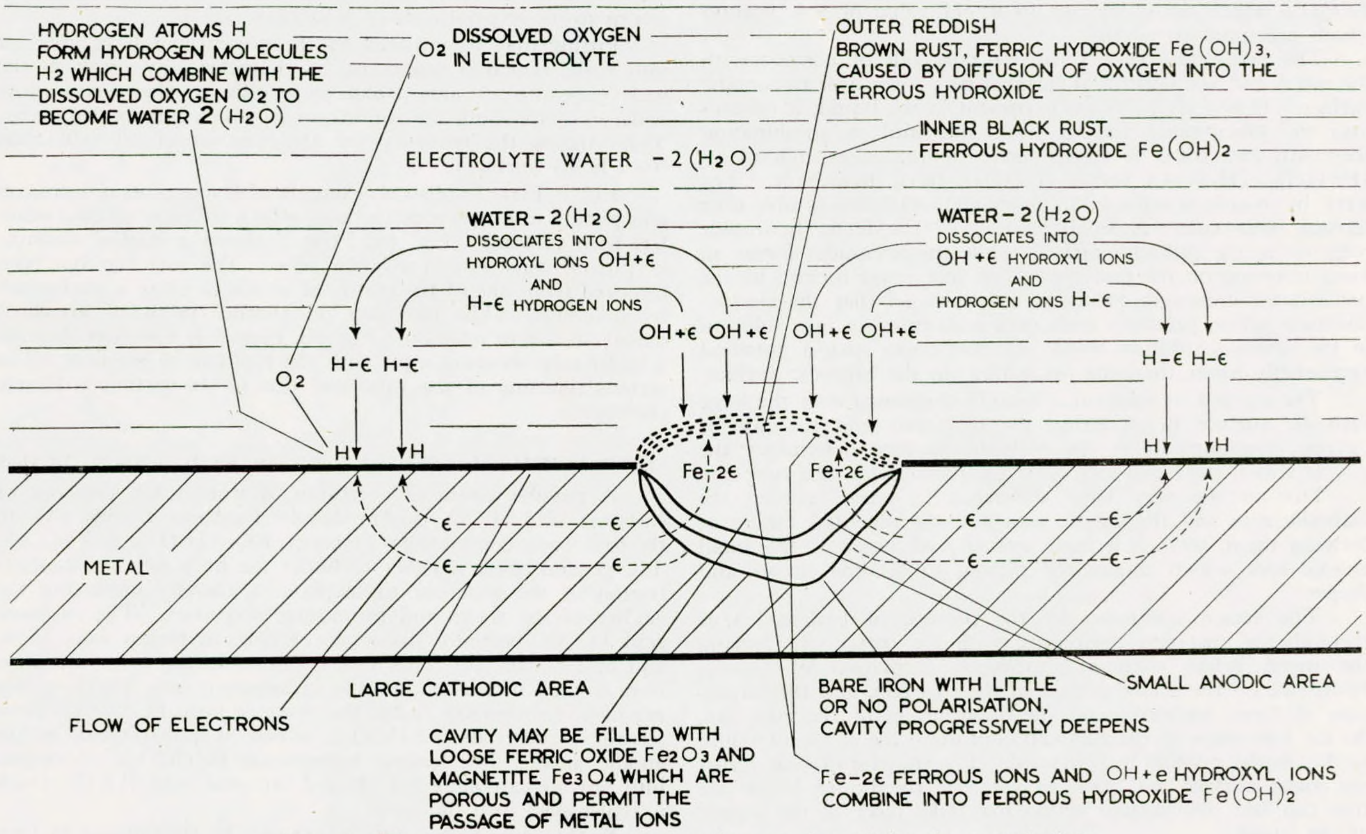


FIG. 6—*Diagram of electro-chemical action causing pitting in a steel plate*

ticular, de-aeration of the feed water is a paramount necessity.

Corrosion caused by electro-chemical activity may waste away an area of the surface, and an example of such corrosion is shown in Fig. 5, Plate 1. This type of corrosion usually occurs when steel is subjected to moisture in the atmosphere, when large portions of the surface may be attacked to a depth which depends upon the time of exposure. This type of corrosion may be called atmospheric corrosion and is caused by a continuous and ample supply of oxygen from the atmospheric air. Corrosion due to acidic attack, as illustrated in Fig. 3, wastes away the metal in a very similar way.

CORROSIVE PITTING

A much more serious type of corrosion may be experienced in boilers in service when the metal surface is immersed in liquid, particularly when standing, when electro-chemical action may cause pitting with a very serious thinning of the metal locally.

Serious pitting occurs mainly when a metal is immersed in a liquid and the action is illustrated in Fig. 6. It is a phenomenon intermediate between general corrosion, as described in relation to Figs. 3 and 4, and immunity, and may be due to incorrect water treatment which prevents general corrosion but is not adequate to give complete immunity and leaves small areas open to attack.

The electro-chemical action is the same as shown in Fig. 4, but, with pitting, the anodic surface is small in area and is surrounded by a very large cathodic surface which is covered with a protective thin invisible oxide film or other cathodic coating which may for example be mill scale or corrosion products. Although the dissolved oxygen content near the surface of the metal may be small, the action of the large cathodic surface is concentrated on the very small anodic surface, so that the intensity of corrosion at the anode may be high.

Due to electro-chemical action, metal ions $Fe-2e$ pass from the metal into the aqueous solution, where they combine with the hydroxyl ions $OH+e$ to form ferrous hydroxide $Fe(OH)_2$ which is soluble up to a *p*H value of 9·4, beyond which precipitation occurs.

The precipitate thus formed occurs out of contact with the metal, but, on settling, it forms a rust cover over the anodic surface. If dissolved oxygen is present in the liquid, it diffuses into the precipitated ferrous hydroxide and in combination therewith and water it forms ferric hydroxide as follows:— $4Fe(OH)_2+2H_2O+O_2$ forms $4Fe(OH)_3$ ferric hydroxide. The ferric hydroxide is colloidal in character, and is less soluble than ferrous hydroxide. A small portion of the ferric hydroxide migrates to the cathodic surface, while the remainder forms an outer covering on the rust cap. The rust cover formed by the precipitated corrosion products is porous so that the electro-chemical action proceeds with pure iron entering into solution in the aqueous solution below the rust cover with a potential appreciably lower than the oxide film on the cathodic surface.

The oxygen in solution is mainly consumed over the large cathodic surface in oxidizing the hydrogen and the rate of oxygen consumption at the cathodic surface determines the rate at which the metal ions leave the metal below the rust cap.

Due to the very large difference in areas between the cathodic area and the anodic area and the potential difference between them, there is a high intensity of attack at the small anodic area which causes the pitting to become deeper and deeper.

The action continues by the passage of hydroxyl ions through the rust cap combining with the metal ions leaving the metal below with a continuous formation of ferrous hydroxide on the under surface of the rust cap, and the formation of ferric hydroxide on the outer layers of the rust cap. As the formation of the rust cap continues, the access of liquid to the anodic pit will be restricted. The transfer of heat across the metal surface will tend to vaporize the liquid below the rust cap and intermittent action may take place in the supply of the liquid.

This action results in a partial drying of the products of corrosion, so that the loose brown rust sludge which forms initially and which contains more water than iron oxide, is de-hydrated and loses water, changing to ferric oxide Fe_2O_3 becoming darker and denser. As the drying action proceeds, this turns to granular black magnetite Fe_3O_4, which may be very stable, is a very good conductor and is strongly cathodic to iron. The rust cap may therefore be stratified with ferric hydroxide on the outside and layers of ferric hydroxide and ferrous hydroxide passing to ferric oxide and magnetite on the inside. This action can take place if the boiler is shut down and the rust sludge allowed to dry. The ferric oxides thus produced permit the passage of hydroxyl ions and the diffusion of oxygen, so that the electro-chemical action proceeds continuously. The anodic surface underneath the rust cap is protected from any stabilizing action of the outer liquid, and is maintained strongly anodic due to the dearth of oxygen below the rust cap. Pitting is therefore a very intensive and dangerous form of corrosion which is particularly likely to occur if a boiler is supplied with water incorrectly treated, or when a boiler is standing with unsuitable water conditions. Once started it cannot be stopped unless the rust cap is removed mechanically and protective influences allowed to cover the anodic surface with a protective film or coating. If the boiler is not completely filled with suitably de-aerated water or any portion has a water pocket, the surface of which is exposed to atmospheric air, the supply of oxygen made available can cause intensive pitting in a comparatively short time.

The tendency for the precipitated hydroxides of iron to lose water and to become more dense and stable is quite general, the rust passing through various stages and tending to be stratified from brown rust outside, changing to green or black below.

The iron ores which are the most commonly found in nature are, hæmatite, Fe_2O_3, magnetite Fe_3O_4, and ferrous carbonate, $FeCO_3$, which occurs as siderite, chalybite or spathic iron ore. It will be seen that the corrosive action described above and in the next section, causes the processed metal to return to its original state of a metallic oxide.

Pitting may be initiated by the presence of large areas of mill scale, which is magnetite in character, any break in the scale exposing an area which is anodic to the surrounding scale, so establishing the conditions in which pitting can occur. This stresses the necessity for the removal of all mill scale from boiler surfaces.

Fig. 7, Plate 1 shows an example of severe general corrosion and pitting in a feed pipe through which strongly aerated water has been flowing. Fig. 8a, Plate 1 shows a typical example of pitting corrosion in a boiler tube. The rust cap has been removed from one of the points of attack to show a pit formed by corrosion. The rust cap on another point of attack is clearly shown in position. Fig. 8b, Plate 1 is a section through a boiler tube showing a pit with the rust cap in position. The serious thinning of the tube wall due to the pitting is clearly shown.

ELECTRO-CHEMICAL CORROSIVE ACTION WITH CARBON DIOXIDE

A possible cause of corrosion of iron is the presence of carbonic acid H_2CO_3 due to dissolved carbon dioxide CO_2 in the feed water combining with water, CO_2+H_2O to give H_2CO_3. The general effect is either to buffer the hydrogen ion concentration of the water or to increase it, thereby increasing the acidity of the water and promoting corrosion. The carbonic acid H_2CO_3 partially dissociates giving hydrogen ions $H-e$, and bicarbonate ions HCO_3+e. The hydrogen ion concentration is thus increased and the bicarbonate ions HCO_3+e are available to combine with the ferrous ions $Fe-2e$ to form ferrous bicarbonate $Fe(HCO_3)_2$ which is precipitated at the anodic areas. The ferrous bicarbonate $Fe(HCO_3)_2$ dissociates into ferrous carbonate $FeCO_3$ and carbonic acid H_2CO_3 which is redissolved into the water.

With acidic water, this action can be continuous in proportion to the supply of carbon dioxide. If dissolved oxygen is present in the water, the ferrous carbonate is changed into ferric hydroxide which is converted into ferric oxide and water as follows:—$4FeCO_3+6H_2O+O_2$ forms $4Fe(OH)_3+4CO_2$. The carbon dioxide CO_2 is thereby regenerated thus enabling the process to be a continuous one so long as a supply of oxygen is available. If the ferric hydroxide $4Fe(OH)_3$ undergoes dehydration, ferric oxide $2Fe_2O_3$ is formed with the loss of water $6H_2O$.

Fig. 9 shows diagrammatically the electro-chemical corrosive action with dissolved carbon dioxide and dissolved oxygen in the water.

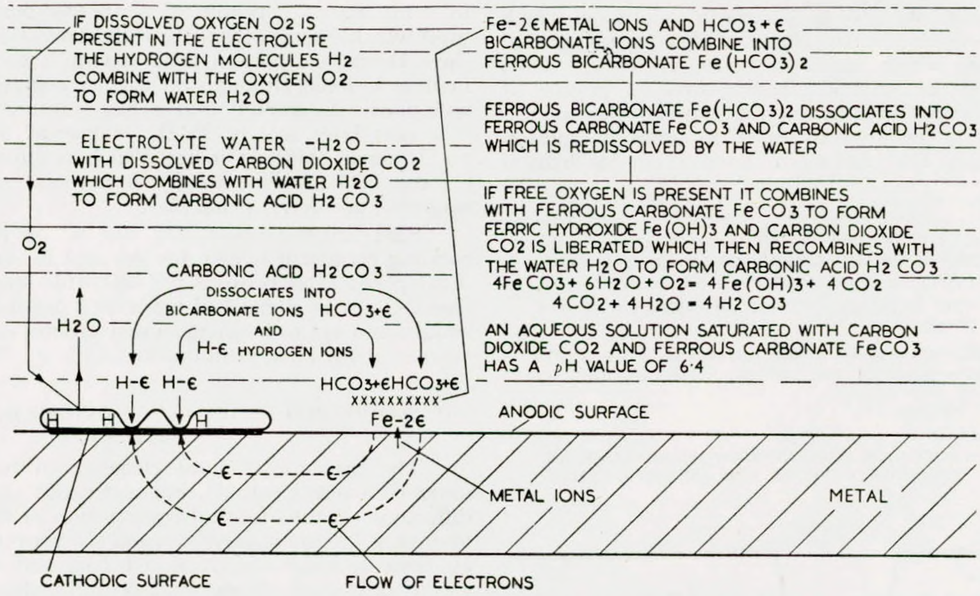


FIG. 9—*Diagram of electro-chemical action causing corrosion of iron in an electrolyte containing carbon dioxide,* CO_2

Hydrogen gas is evolved or liberated with *p*H values of 6·4 and under. If dissolved oxygen is present in the electrolyte it will also remove hydrogen from the cathodic surface by combining with the hydrogen to form water.

Carbon dioxide in water to saturation gives an acidity of *p*H 4·0, but this is varied by other solids when they are present. For example, when gypsum, limestone, and carbon dioxide occur together they provide a buffer action to maintain the water at *p*H 5·1. The precipitation of ferrous carbonate increases the *p*H value of the solution, and from some experimental tests, it would appear that with a continuous supply of CO_2 and no dissolved oxygen, the liquid would have a *p*H value stabilized at about 6·4 (6)*.

The presence of carbon dioxide in the water assists corrosion, principally by increasing the hydrogen ion concentration, but the amount left in feed water which has been subjected to efficient de-aeration is so small as to be negligible in its effect. Carbon dioxide may be absorbed by feed water in contact with the atmosphere, and it may also be formed in the water by the breaking up of carbonates or bicarbonates when heating occurs, but with the usual range of carbonate hardness in feed and boiler water and the very small amount of dissolved carbonic acid likely to be present, it need not generally be taken into consideration.

Carbon dioxide which escapes from the boiler water in the form of gas may pass through superheaters and turbines without causing any corrosion, but if it is absorbed into pockets of condensate which may form in any part of the boiler steam or feed systems, such pockets will have their acidity increased thereby, and will therefore be more liable to cause corrosion.

CAUSTIC CRACKING

Intercrystalline corrosion is another very dangerous form of corrosion, although it appears only under certain conditions. In this type of corrosion, the destruction occurs along the grain or crystal boundaries of the metal with an action similar to that described in pitting. Fig. 10a, Plate 2 shows a surface micrograph of an example of intercrystalline corrosion, while Fig. 10b, Plate 2 is a micrograph showing the penetration of the intercrystalline corrosion into the depth of the metal.

If a metal is subjected to repetitions of stress and corrosion, a progressive intercrystalline action proceeds which causes rupture at stresses which are much below the elastic limit. This is known as a corrosion fatigue failure and in the majority of cases the corrosive attack cannot be detected externally.

Caustic embrittlement or caustic cracking is a well-known form of intercrystalline corrosion, which has frequently been experienced in riveted constructions. Solid, forged or welded constructions are less subject to this type of corrosive attack.

The general nature of the start and development of caustic cracking is still the subject of a good deal of consideration. Caustic cracking may occur where there is a crack or fissure in which alkali can become concentrated as compared with the boiler water which is slightly alkaline. The dilute alkali renders the iron passive but any protective film will be dissolved as ferroate by the strong alkali. This occurs inside the crack with the result that electro-chemical action is set up and Fig. 11 illustrates a suggested explanation of the phenomenon (2). Due to deposition of sodium hydroxide in the base of the crack, there is a stronger concentration of sodium hydroxide at the base of the crack than on the surface, where it is more dilute in contact with the boiler water, so that an electro-chemical cell is formed between the concentrated alkali and the weaker alkali solution on the main surface of the metal. Hot concentrated sodium hydroxide dissolves iron readily as sodium ferroate Na_2FeO_2. The concentration in the crack is intensified if there is a leak of steam through the seam to the outside of the metal as may occur in a riveted construction. The electro-chemical action causes corrosion at the base of the crack which is strongly

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* The bracketed numerals refer to the bibliography at the end of the paper.

anodic, so that any tension applied to the metal due to the boiler pressure or other cause, gradually causes the crack to extend under the combined action of corrosion and stress. A crack may occur originally as a fatigue crack and may subsequently be developed by caustic cracking action.

It is also considered that adsorbed hydrogen may play a part because hydrogen which may become attached to the metal at the base of the crack may assume molecular form when it will expand in volume considerably and exercise a pressure which tends to increase the depth of the crack.

Sodium compounds are largely used in boiler water conditioning and the use of sodium hydroxide to increase the *p*H value is illustrated in the upper part of Fig. 11.

Caustic cracking, like pitting, occurs in conditions which are intermediate between the conditions which permit general corrosion and those which secure complete passivity. Some authorities consider that caustic cracking may be avoided if certain prescribed ratios are maintained between the sodium sulphate present and the total alkalinity in terms of sodium carbonate Na_2CO_3. The American Society of Mechanical Engineers advise that this ratio should be 1 to 1 for pressures up to 150lb. per sq. in., 2 to 1 up to 250lb. per sq. in., and 3 to 1 above that pressure. Sodium hydroxide depresses the solubility of sodium sulphate so that when sodium hydroxide becomes concentrated locally, sulphate is precipitated at the crack and affords protection. British Standard 1170-1947 advises that the sodium sulphate should be at least 2·5 times the caustic alkalinity in terms of NaOH.

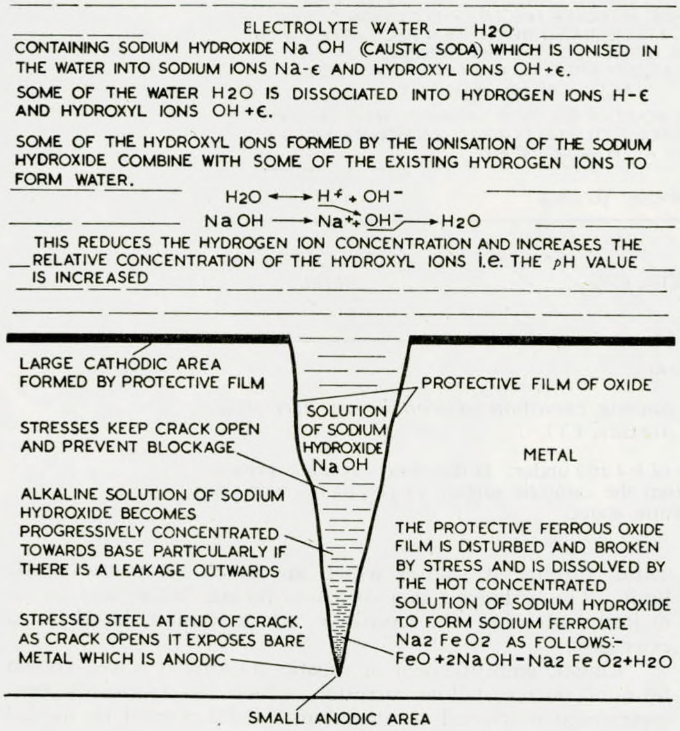


FIG. 11—*Diagram showing caustic corrosive cracking in a steel plate under stress*

CORROSION DUE TO HIGH TEMPERATURE OXIDIZATION

At high temperatures, iron is attacked by water vapour even in the absence of free oxygen. This so-called steam corrosion can occur at a temperature as low as 300 deg. C., but is negligible until the temperature exceeds 550 deg. C. (1,022 deg. F.). The main reaction which takes place between iron and steam at these temperatures is as follows:—

$$3Fe+4H_2O \text{ (Steam)} = Fe_3O_4+4H_2.$$

It will be seen that this reaction gives magnetite Fe_3O_4 which occurs as a hard black iron oxide scale on the surface of the metal. It occurs chiefly on the superheated steam side of boilers but the action can occur under a deposition of scale or corrosion products which limits the flow of liquid to the cavity under the scale or scab formation so that a local superheating action can occur. It is possible that this is one of the actions which occur under the scab over a pit.

MILL SCALE

When steel at high temperature is in contact with the air, a thick oxide scale is formed and the character of this scale to some extent depends upon the temperature of the steel. Steel which emerges from the rolls above 575 deg. C. will carry three layers, the bottom layer having a composition approximating to FeO, although on further cooling the ferrous oxide will partly decompose into a fine mixture of Fe and Fe_3O_4. The next layer will be Fe_3O_4, magnetite, and the outer layer Fe_2O_3, ferric oxide. On steel which is finished below 575 deg. C., the FeO layer is usually absent and the layers comprise magnetite and ferric oxide (2).

Steel finished above 575 deg. C. is readily descaled by pickling because it is easy for the acid to eat its way along the decomposed layer between the magnetite and the metallic iron. Descaling of steel finished below 575 deg. C. is a much slower process and apt to be sluggish and unsatisfactory because of the absence of the layer mentioned.

THE ELIMINATION OF DISSOLVED OXYGEN IN FEED AND BOILER SYSTEMS

The presence of local elements in boiler materials can never be avoided entirely, but experience shows that the usual differences in the chemical composition of the material are not important factors in corrosion and the various alloy steels which are used in boiler construction behave very much in the same way as mild steel. Forge scale or mill scale, corrosion products or oxide patches, are potent factors setting up the local elements necessary for destructive corrosion. Foreign deposits from the steam and feed circuit or from the feed water are also active sources of trouble. The corrosive action set up by any of these causes is greatly intensified if a strong salt solution is present which may occur in undrained or undrainable parts of a boiler when it is standing, the term "boiler" including all parts, such as, for example, the economizer and the superheater. When a boiler is shut down, care should be taken to avoid air getting into it because such air is greedily absorbed by the water in the boiler or in any undrainable pockets which are not dried out. During periods when a boiler is idle all parts likely to be attacked by corrosion should be completely drained or completely filled with de-aerated water having the necessary alkaline character.

These facts point to the necessity for the greatest possible cleanliness in the manufacture, operation and maintenance of boiler equipment. It will be appreciated that dissolved oxygen is the most important and the most dangerous factor in corrosion, being the real driving force in any progressive corrosive action. In the operation of boilers it is very important, therefore, that the amount of oxygen in the feed water should be reduced to the lowest practicable amount. The presence of gases and salts should be avoided as far as possible and a suitable alkalinity of the boiler feed water should be maintained so as to reduce electro-chemical action to a negligible amount.

Gases are soluble in liquids to an extent depending upon the gas, the liquid, the pressure and the temperature.

Atmospheric air contains nitrogen, oxygen, hydrogen, carbon dioxide, argon, etc., and each of these gases enters into any water which is in contact with the air to an extent which depends upon the solubility of the gas in water and the temperature of the water, and is proportional to the partial pressure of each gas. At atmospheric pressure and 32 deg. F., the absorption co-efficients, etc., of these gases are as given in

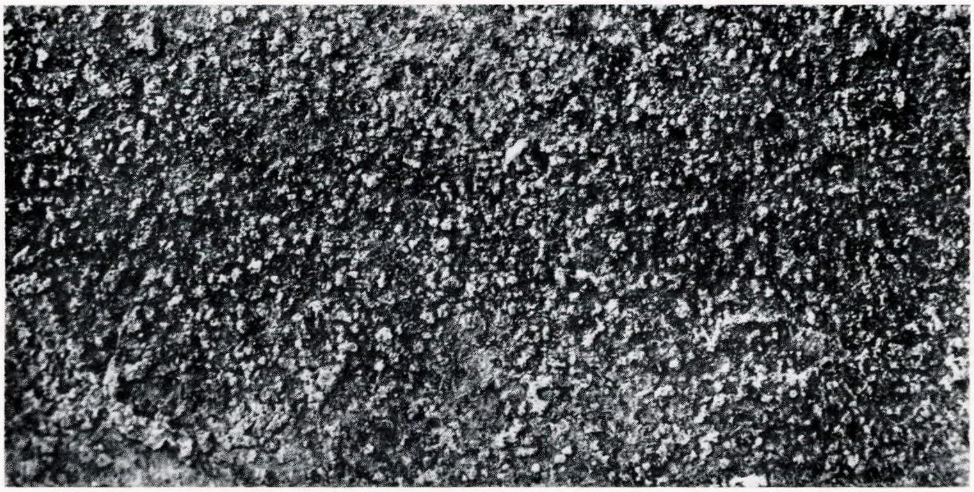


FIG. 5—*Atmospheric corrosion on a steel surface*

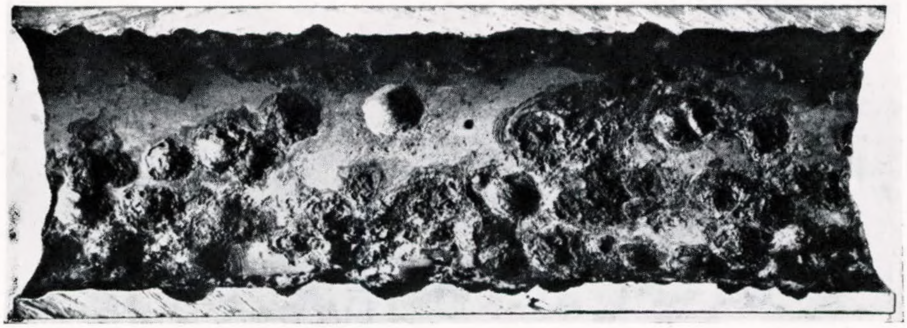


FIG. 7—*Corrosion and pitting in a steel feedpipe with aerated water*

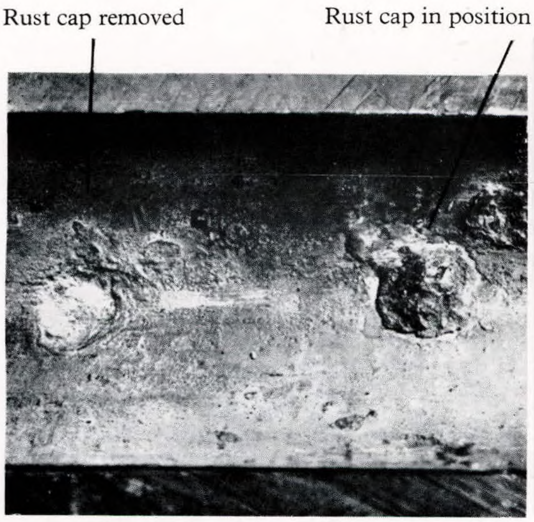


FIG. 8a—*Section of a boiler tube showing pitting*

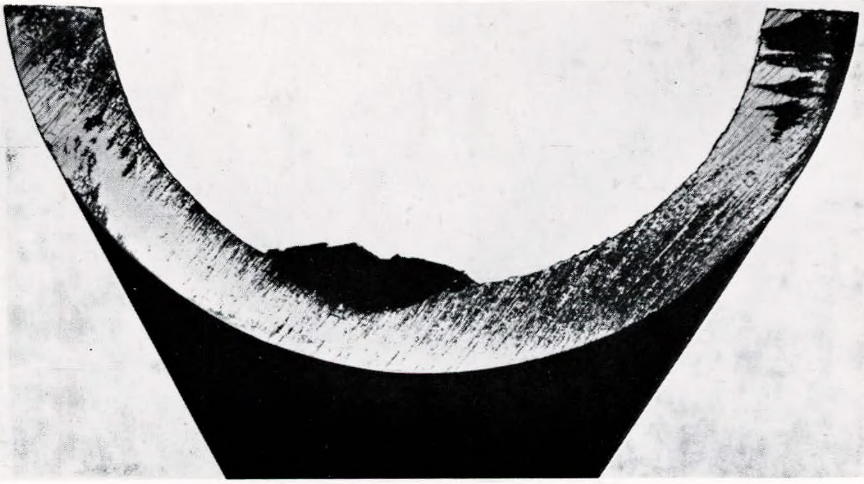


FIG. 8b—*Section of a boiler tube showing a pit with rust cap in position*

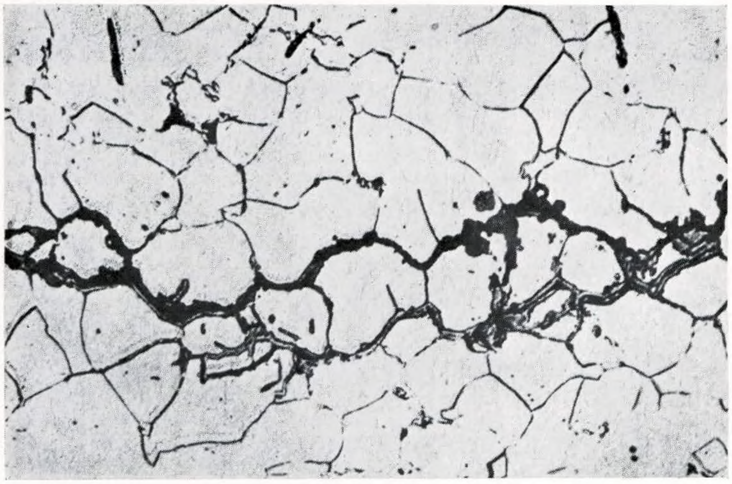


FIG. 10a—*Surface micrograph showing intercrystalline corrosion*

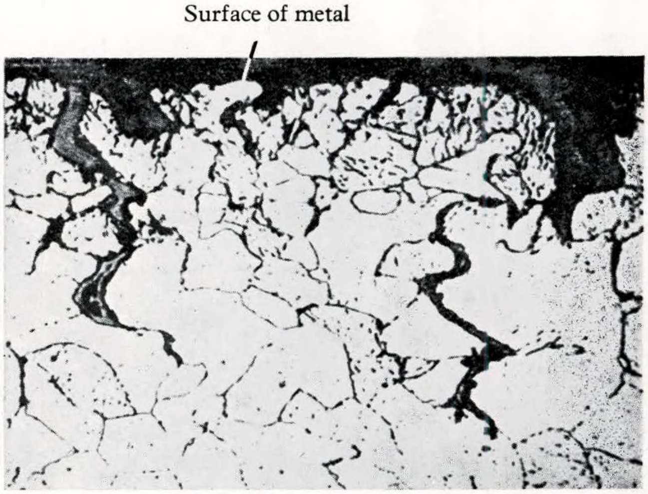


FIG. 10b—*Micrograph showing penetration of intercrystalline corrosion into the metal*

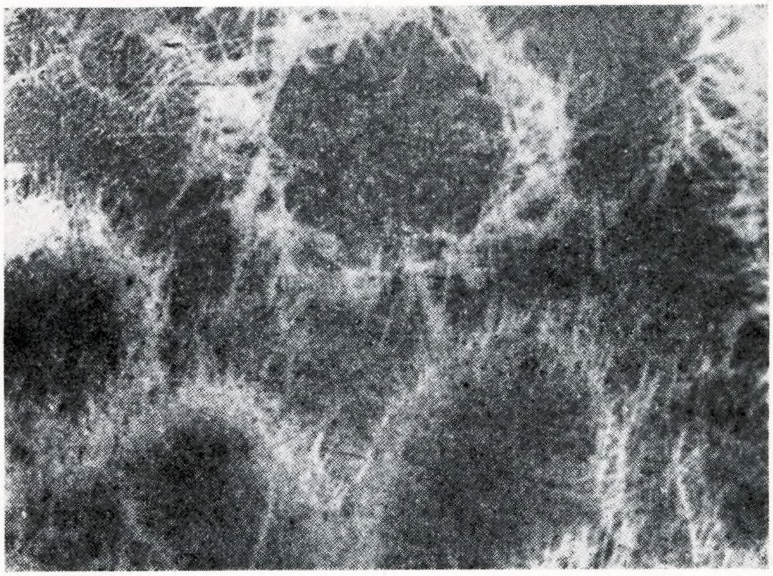


FIG. 25—*First stage of scale formation showing precipitation of rings of calcium sulphate crystals*

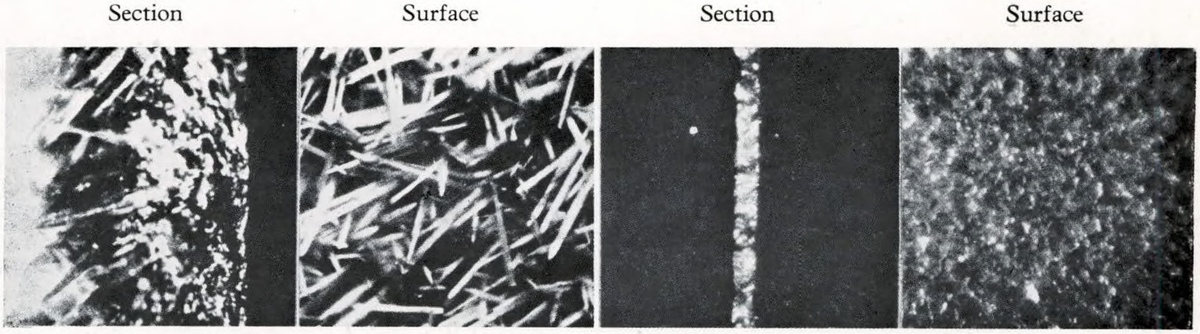


FIG. 26—*Cross-sectional and surface appearance of hemihydrate and anhydrite scales from an experimental boiler at stated pressures*

Table 1. The partial pressures are proportional to the volume percentages of the respective gases in the atmospheric air. By multiplying these partial pressures by the absorption coefficients, multiplied by 1,000, the volumes of the gases dissolved in one litre of water are obtained in millilitres of the gas measured at 32 deg. F. and a combined pressure of one atmosphere.

TABLE 1

Gas	Percentage by volume in the atmospheric air, per cent	Partial pressure of each gas, total 1 atmos.	Absorption coefficient	Amount of gas dissolved in one litre of water, ml.	Percentage of gas in the water, by volume, per cent
Nitrogen, N_2	78	0·78	0·0239	18·64	63·3
Oxygen, 0_2	21	0·21	0·0489	10·26	34·9
Carbon dioxide, CO_2	0·03	0·0003	1·713	0·51	1·8

It will be seen that the proportion of the gases in the water, as compared with the air, is modified by their respective solubilities. Fig. 12 shows the volume of oxygen which can be held by water at temperatures between 40 deg. F. and 212 deg. F. when fully saturated with the air at a pressure of 14·7lb. per sq. in. Air contains an average of about 0·03 per cent by volume of carbon dioxide and Fig. 12 also shows the solubility of carbon dioxide in pure water at atmospheric pressure and different temperatures. The solubility of carbon dioxide is very much greater than that of oxygen so that water can contain a great deal more than indicated above if the atmosphere contains more than stated or if it is absorbed from any other source.

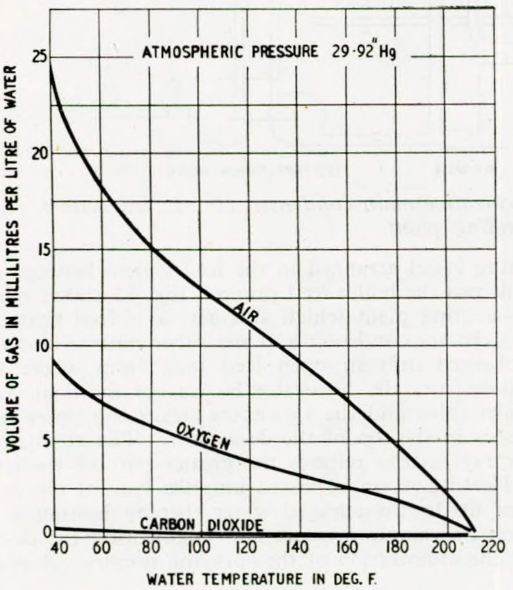


FIG. 12—*Volume of air, oxygen and carbon dioxide in water at different temperatures when saturated with air at atmospheric pressure*

The capacity of water for retaining a gas falls rapidly as the water temperature is increased because the water vapour pressure is increased and the partial pressure of the gas is reduced. This is the basic principle on which de-aeration is effected in regenerative steam condensers and mechanical de-aerators.

Water greedily absorbs gases up to the limit of solubility if exposed to contact with them, so de-aerated feed water should not be exposed to contact with atmospheric air after de-aeration has been effected.

Oxygen and carbon dioxide are the two most important gases to be considered in the de-aeration of feed water and it is fortunate that the treatment in a regenerative condenser or in a mechanical de-aerator which removes the oxygen is equally effective in removing the carbon dioxide. Nitrogen is a comparatively inert gas and can be neglected and fortunately the proportion of carbon dioxide absorbed from the atmosphere is small leaving oxygen as the really important gas to be dealt with.

The amount of oxygen and carbon dioxide present in water can be reduced by lowering the pressure on the surface of the water or by increasing the temperature.

For any given surface pressure on the water, as the water temperature is increased, the water vapour pressure is increased and the dissolved gas partial pressure is reduced until it reaches zero when the boiling point of the water is reached, at which point all the dissolved gas will have been driven out of the water. Stated briefly, the water must be subjected to boiling conditions.

Fig. 13 shows the effect of pressure and temperature on the solubility of oxygen and illustrates how the increase of the water vapour pressure to boiling point at any given temperature reduces the oxygen content to zero, because the partial pressure of the oxygen is reduced to zero. It should be borne in mind that a gradient is necessary to effect gas flow or heat transfer and it is therefore easier to avoid oxygen getting into water than to take it out after it has entered.

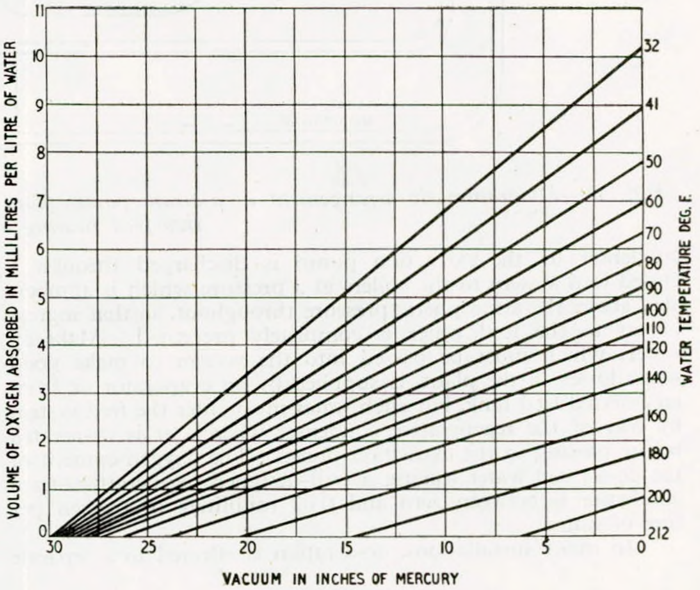


FIG. 13—*Volume of oxygen which can be contained in water at different temperatures and pressures*

Oxygen enters a steam and feed circuit at glands and other leakage points in the vacuum portion of the circuit, and in any make-up feed introduced from an external source contaminated by contact with air.

In a modern steam power plant, the steam condenser is invariably of the regenerative type, as shown in Fig. 14, with a closed feed system between the condenser and the boilers. The condenser is so designed that there is free access for the entering steam to pass to the base of the condenser so that the condensate formed in the condenser must pass through such steam and its temperature corresponds very closely to the temperature of the steam, so that the water cannot contain any oxygen or other non-condensable gas. The non-condensable gases are removed from a point on the condenser remote from the condensate level and suction outlet. The de-aerated water drawn from the

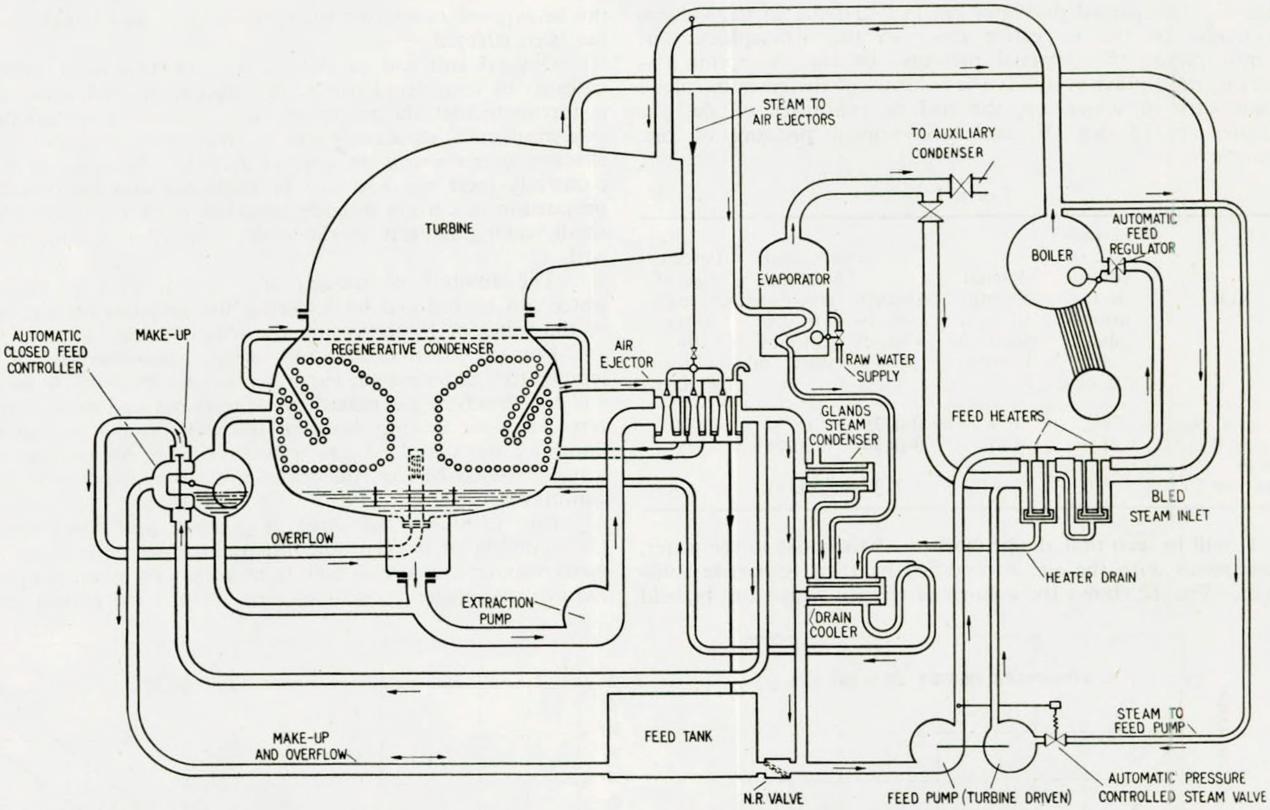


FIG. 14—*Diagrammatic arrangement of a steam power plant with regenerative main condenser, closed feed system, regenerative feed heating and evaporating plant*

condenser by the extraction pump is discharged through a closed feed system to the boilers at a pressure which is appreciably above the atmospheric pressure throughout, so that ingress of air to the feed water is completely prevented. Make-up water, which must be passed into the system to make good water losses in the plant, is supplied by an evaporator or from an external feed tank, but such water must enter the feed system by way of the regenerative condenser wherein it is de-aerated before passing to the extraction pump. The oxygen content of the de-aerated water leaving a well-designed regenerative steam condenser is between zero and 0·02 millilitres of oxygen per litre of water.

In many installations, de-aeration is effected in a separate de-aerating vessel arranged in the feed system between an open feed tank and the boiler feed pump. Fig. 15 shows one design of a de-aerating plant which also acts as a feed heater. Condensate from the condenser and any other returns of condensate are discharged into an open feed tank from where the feed water passes into the de-aerator by way of the float controlled water inlet valve and the air ejector steam condenser to spraying nozzles in the top of the de-aerator. The atomizing effect of the spray nozzles releases the greater part of the contained gases. Heating steam is passed into the base of the de-aerator and rises up to the sprayed water thereby heating it and so increasing its vapour pressure and augmenting the de-aerating effect of the atomization of the spraying nozzles. A portion of

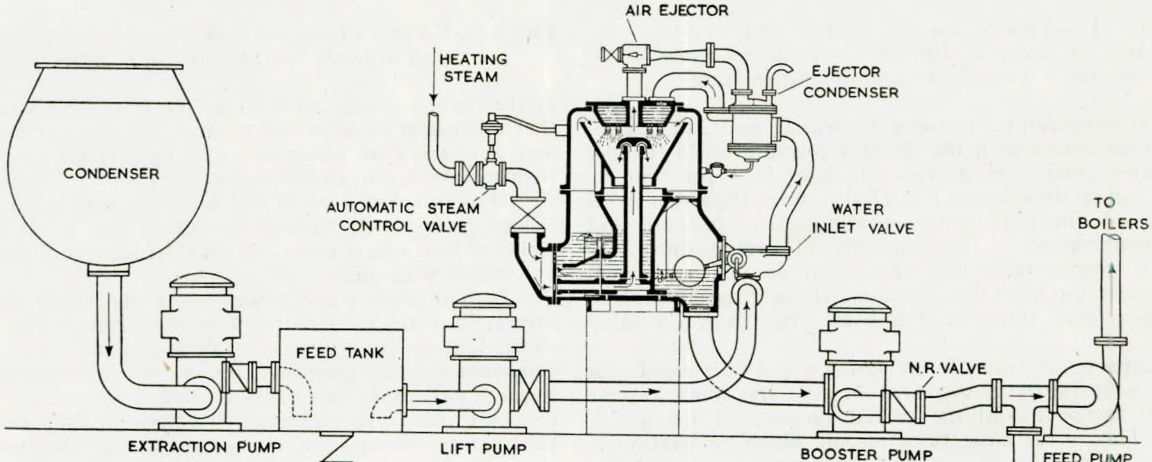


FIG. 15—*Diagrammatic arrangement of an open feed system with de-aerating plant*

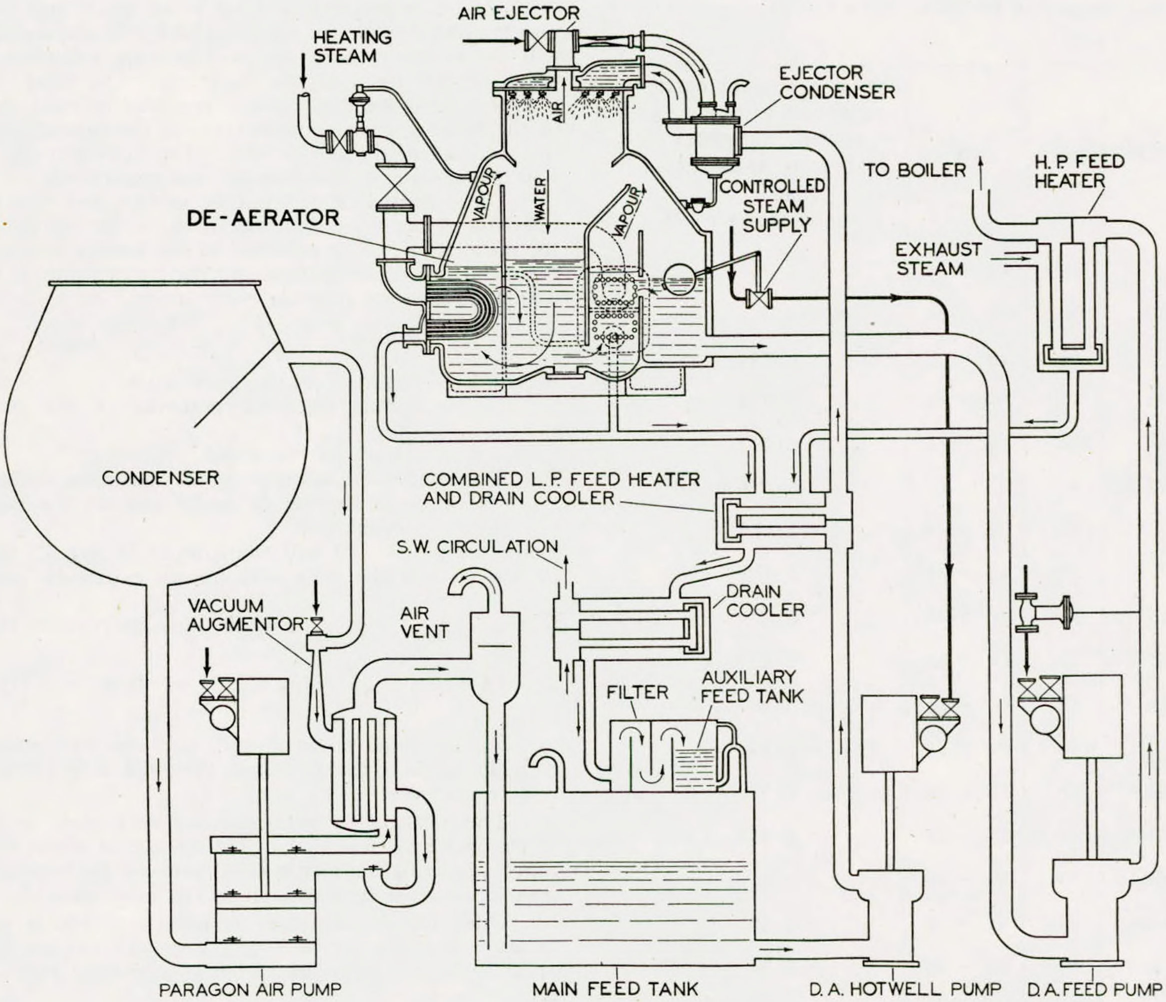


FIG. 16—*Marine feed system with de-aerator and direct acting pumps*

the heating steam is passed through a perforated plate submerged below the water level so that the water, as it passes through the base of the de-aerator, is combed by the incoming steam with the object of driving out any residual non-condensable gases left after the atomization and heating effected in the top of the de-aerator. The non-condensable gases are removed from the de-aerator by the air ejector. De-aerating plants give de-aeration between zero and 0·05 millilitres of oxygen per litre, depending upon the operating conditions for which they are designed.

Fig. 16 shows an arrangement of a de-aerator in a feed system with direct acting pumps for marine service in which the heating is effected by surface contact so that oily exhaust steam can be used and the resultant condensate cooled for filtration before it is passed into the feed system.

Regenerative steam condensers, closed feed systems and de-aerating plants have been very successful in removing dissolved oxygen and preventing corrosion in boilers, but they deal with only one phase of the problem although it is probably the most essential one. It is, however, also important to create the conditions which will reduce any possible electro-chemical action to a negligible amount.

POTENTIALS OF METALS

The electro-motive force causing the passage of the ions of a metal into the liquid in contact differs with different metals and is determined by their individual atomic structure and arrangement of electrons.

Table 2 gives the "normal electrode potential" of various metals in their salt solution of "normal" ionic activity measured relative to hydrogen as an arbitrary zero datum at the temperature of 25 deg. C. The potential increases with temperature and with the ionic concentration of the solution. However, the table is convenient for reference as indicating the relative values between the metals, the relativity remaining substantially similar for similar conditions of temperature and ionic concentration.

If two dissimilar metals are in an electrolyte, the voltage of the couple between the two metals is not the difference between the potentials of the two actual metals.

The metal which has the lower potential will become the anode and tend to dissolve anodically but the metal which has the higher potential will serve simply as a cathodic surface on which the positively charged ions in the electrolyte can make contact and take up surplus electrons from the metal. The metal of higher potential is said to be protected cathodically at the expense of the lower potential metal.

The list indicates, for the conditions specified, which metal will act as the anode relative to another metal. It also serves as a measure of the relative resistance of the various metals to acid attack.

It will be seen that zinc is a lower metal than iron or copper, which explains why zinc is frequently used to protect iron cathodically at the sacrifice of the zinc, while iron is used in a similar manner to protect copper.

The potential of a metal surface can be modified by many factors, such as "aeration", scale, and particularly by the

TABLE 2.—ELECTROMOTIVE FORCE SERIES OF ELEMENTS AT 25 DEG. C.

NOBLE END

ELEMENT	SYMBOL	ION	"NORMAL ELECTRODE POTENTIAL" POTENTIAL RELATIVE TO NORMAL HYDROGEN ELECTRODE AS ARBITRARY ZERO - POINT. POTENTIAL IN VOLTS
GOLD	Au	Au − 3ε	1·42
PLATINUM	Pt	Pt − 2ε	ABOUT 1·2
SILVER	Ag	Ag − ε	0·799
MERCURY	Hg	Hg_2 − 2ε	0·798
COPPER	Cu	Cu − 2ε	0·345
HYDROGEN	H	H − ε	0·000 DATUM
LEAD	Pb	Pb − 2ε	−0·126
TIN	Sn	Sn − 2ε	−0·136
NICKEL	Ni	Ni − 2ε	−0·250
COBALT	Co	Co − 2ε	−0·277
CADMIUM	Cd	Cd − 2ε	−0·402
IRON	Fe	Fe − 2ε	−0·441
CHROMIUM	Cr	Cr − 2ε	−0·71
ZINC	Zn	Zn − 2ε	−0·762
MANGANESE	Mn	Mn − 2ε	−1·05
ALUMINIUM	Al	Al − 3ε	−1·67
MAGNESIUM	Mg	Mg − 2ε	−2·34
SODIUM	Na	Na − ε	−2·712
POTASSIUM	K	K − ε	−2·922
LITHIUM	Li	Li − ε	−3·02

BASE END

The " normal " electrode potential of a metal is the potential difference between the metal and a solution containing one gram ion of the metal per litre of the solution, i.e. 1,000 grams of solution.
For example, the potential for iron given above is for 55·8 grams of iron ions per litre of the solution, where 55·8 is the atomic weight.
For hydrogen, the potential given above is for 1·008 grams of hydrogen ions per litre of solution.
The potential varies with ionic concentration and with temperature.

presence of the thin closely adherent protective oxide coating which may be formed and creates a cathodic surface. Any oxide film which will allow the passage of electrons will raise the potential of the metal surface to an extent depending upon the degree of porosity of the film.

For example, aluminium and chromium readily become covered with a thin tight oxide covering which raises the potential of the surface and increases the resistance to corrosion.

The thin protective oxide film referred to, which normally remains invisible, should not be confused with a rust or scale covering, although these, as they accumulate, may have some protective effect in diminishing the rate of corrosion.

In feed system and boiler corrosion problems, the datum from which corrosion takes place is the hydrogen ion concentration on cathodic surfaces.

Wherever a difference of potential exists, the surface which is at the lower potential will act as an anode and will dissolve into the liquid while the surface which is at the higher potential will act as the cathode and will provide a surface on which the hydrogen ions acquire electrons. The e.m.f. of the cell thus formed is the difference in potential between the potential of the hydrogen ion concentration at the cathodic surface and the potential of the anodic area. The hydrogen ion concentration of the water is therefore of vital importance.

The potential of any anodic surface will vary dependent upon the presence of elements in or on the surface other than the metal itself. The potential of the bare iron will vary with the metal ion concentration and the temperature in accordance with the following formula:—

$$E = E_0 + \frac{0{\cdot}059}{n} \times \frac{T}{273+25} \times \log_{10} C.$$

Where E is potential at the metal surface
E_0 is normal electrode potential of the metal at 25 deg. C.
n is valency of the metal
T is absolute temperature of the metal surface, deg. C.
C is concentration of metal ions in the liquid at the metal face.

For a value of $C = 1$ and temperature 25 deg. C. the potential of iron is $-0{\cdot}441$ volts and for an extremely low value of $C = \frac{1}{100{,}000{,}000{,}000{,}000}$ and the same temperature the potential is $-0{\cdot}441 + \left(\frac{0{\cdot}059}{2} \times \log 10^{-14}\right) = -0{\cdot}441 - 0{\cdot}413 = -0{\cdot}854$ volts.

For the same range of hydrogen ion concentration at the cathodic surface, the hydrogen potential is 0·000 and −0·826 volts respectively.

The potential of bare iron is always likely to be negative to the hydrogen potential in the absence of air in the solution. Since a potential difference exists between the hydrogen and the iron, dissolution or corrosion of the iron occurs.

Other factors, however, modify the anodic potential, such as the adsorption of hydrogen ions and oxygen ions on the metal surface and the deposition of corrosion products as described later.

HYDROGEN ION AND HYDROXYL ION CONCENTRATIONS (*p*H AND *p*OH VALUES)

The feed or boiler water contains hydrogen ions H^+ and hydroxyl ions OH^-, which have opposite electrical charges. The measure of the concentration of the hydrogen ions is usually referred to as the *p*H value and that of the hydroxyl ion concentration as the *p*OH value. The *p*H and *p*OH values are interdependent, so that it is customary to use the *p*H value only, since the *p*OH value is then known automatically, as will be seen from the following:—

(a) The concentration of hydrogen ions, and therefore hydroxyl ions, relative to the total number of molecules in the water has a definite value at any given temperature.

(b) The product of the concentrations of hydrogen ions and hydroxyl ions has the value of 10^{-14} if the water is at a temperature of 25 deg. C. This temperature is called the "relative" temperature.

Example:—

If the concentration of hydrogen ions is 10^{-4} at 25 deg. C. then the concentration of hydroxyl ions is 10^{-10}, because the product of the two concentrations must be 10^{-14}.

(c) If the hydrogen ion concentration and the hydroxyl ion concentration are equal, i.e., each is 10^{-7} in water at 25 deg. C., then the water is neutral. If there is a greater concentration of hydrogen ions than hydroxyl ions, i.e., *p*H is greater than 10^{-7}, the water is said to be acid; while, if the opposite is the case, the water is said to be alkaline.

Table 3 shows the range of concentration of hydrogen ions and hydroxyl ions and the corresponding *p*H values with water at 25 deg. C.

TABLE 3.—RELATIONSHIP BETWEEN CONCENTRATIONS OF HYDROXYL IONS AND HYDROGEN IONS IN WATER AT (25 DEG. C.) 77 DEG. F.

	CONCENTRATION OF HYDROXYL IONS IN SOLUTION P.P.M. BY WEIGHT	CONCENTRATION OF HYDROXYL IONS IN GRAM IONS PER LITRE OF SOLUTION		pOH VALUE	CONCENTRATION OF HYDROGEN IONS IN GRAM IONS PER LITRE OF SOLUTION		pH VALUE	CONCENTRATION OF HYDROGEN IONS IN SOLUTIONS P.P.M. BY WEIGHT	POTENTIAL OF HYDROGEN SATURATED PLATINUM POTENTIAL IN VOLTS
ACID RANGE	0·00000000017	10^{-14}	$\frac{N}{100,000,000,000,000}$	14	10^{0}	N "NORMAL"	0	1000	0·000
	0·0000000017	10^{-13}	$\frac{N}{10,000,000,000,000}$	13	10^{-1}	$\frac{N}{10}$	1	100	−0·059
	0·000000017	10^{-12}	$\frac{N}{1,000,000,000,000}$	12	10^{-2}	$\frac{N}{100}$	2	10	−0·118
	0·00000017	10^{-11}	$\frac{N}{100,000,000,000}$	11	10^{-3}	$\frac{N}{1,000}$	3	1	−0·177
	0·0000017	10^{-10}	$\frac{N}{10,000,000,000}$	10	10^{-4}	$\frac{N}{10,000}$	4	0·1	−0·236
	0·000017	10^{-9}	$\frac{N}{1,000,000,000}$	9	10^{-5}	$\frac{N}{100,000}$	5	0·01	−0·295
	0·00017	10^{-8}	$\frac{N}{100,000,000}$	8	10^{-6}	$\frac{N}{1,000,000}$	6	0·001	−0·354
NEUTRAL	0·0017	10^{-7}	$\frac{N}{10,000,000}$	7	10^{-7}	$\frac{N}{10,000,000}$	7	0·0001	−0·413
ALKALINE RANGE	0·017	10^{-6}	$\frac{N}{1,000,000}$	6	10^{-8}	$\frac{N}{100,000,000}$	8	0·00001	−0·472
	0·17	10^{-5}	$\frac{N}{100,000}$	5	10^{-9}	$\frac{N}{1,000,000,000}$	9	0·000001	−0·531
	1·7	10^{-4}	$\frac{N}{10,000}$	4	10^{-10}	$\frac{N}{10,000,000,000}$	10	0·0000001	−0·59
	17	10^{-3}	$\frac{N}{1,000}$	3	10^{-11}	$\frac{N}{100,000,000,000}$	11	0·00000001	−0·649
	170	10^{-2}	$\frac{N}{100}$	2	10^{-12}	$\frac{N}{1,000,000,000,000}$	12	0·000000001	−0·708
	1700	10^{-1}	$\frac{N}{10}$	1	10^{-13}	$\frac{N}{10,000,000,000,000}$	13	0·0000000001	−0·767
	17000	10^{0}	N "NORMAL"	0	10^{-14}	$\frac{N}{100,000,000,000,000}$	14	0·00000000001	−0·826

The "normal" or datum concentration of hydrogen ions is one gram ion per litre of solution, which corresponds to 1·008 grams of hydrogen ions per 1,000 grams of solution.

In chemically pure water, which has a pH of 7, the hydrogen ion concentration is reduced to one ten millionth part of a gram of hydrogen ions per litre of water.

For a pH value of 10 the water contains only one ten thousandth part of a millionth part of a gram of hydrogen ions per litre of water.

p.p.m. is " parts per million " by weight compared with pure water and is the weight of each substance in lb. per million lb. of pure water.

The pH value is the term commonly used to express the hydrogen ion concentration and is the logarithm of the reciprocal of the hydrogen ion concentration, e.g., a hydrogen ion concentration of 10^{-4} is referred to as a pH value of 4 and is understood to be measured at 25 deg. C. unless otherwise stated. Similarly, the hydroxyl ion concentration can be expressed as a pOH value.

It will be seen that water is alkaline in character when the pH value is greater than 7 (maximum up to 14) for a temperature of 25 deg. C.

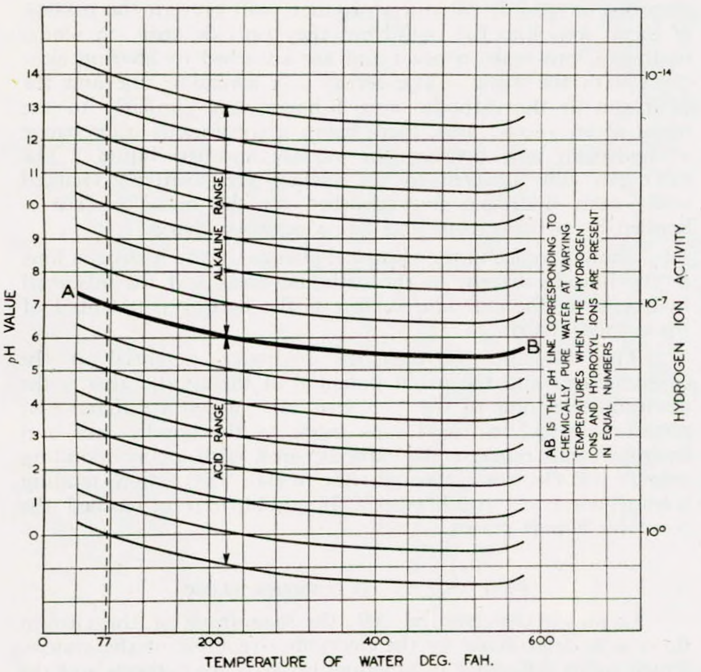


FIG. 17—*p*H *value of water in relation to water temperature*

When the temperature of the water is changed the concentrations of the hydrogen ions and hydroxyl ions change so that the pH value and the pOH value both fall as the water temperature is increased. This means that the concentration of hydrogen ions increases and any corrosive electro-chemical action is increased as the temperature is raised.

The curve AB in Fig. 17 shows how the pH value of pure water changes as the temperature is increased. The increase of hydrogen ion concentration with increase in temperature is also indicated for acid and alkaline aqueous solutions.

Whether a water shows an acid or an alkaline reaction depends entirely upon the proportion of its concentrations of hydrogen ions and hydroxyl ions relative to each other, and this proportion can be influenced by the addition of acids or alkalis.

Acids, such as hydrochloric, phosphoric, nitric, carbonic acid, etc., are chemicals, which, when added to water, increase its hydrogen ion concentration, i.e., lower its pH value.

Alkalis, such as caustic soda and trisodium phosphate are chemicals which reduce the hydrogen ion concentration of water, i.e., raise its pH value. Alkalis, such as caustic soda and trisodium phosphate are commonly used as additions to feed water or boiler water as necessary to maintain the water at the appropriate pH value offering the most favourable conditions to prevent corrosion. The importance of the pH value lies in the electric potential associated with it and the electric potential for different hydrogen ion concentrations is shown in Table 3. For a water with a given pH value, the effective potential of the cathodic surfaces, so far as corrosive activity is concerned, is determined by the potential associated with the pH value, i.e., the hydrogen ion concentration. This potential in relation to the potential of the iron will be discussed in detail later.

ELECTRO-CHEMICAL HALF CELLS

The metal and hydrogen potentials are illustrated in Fig. 18 in which circles have been used to indicate both ions and

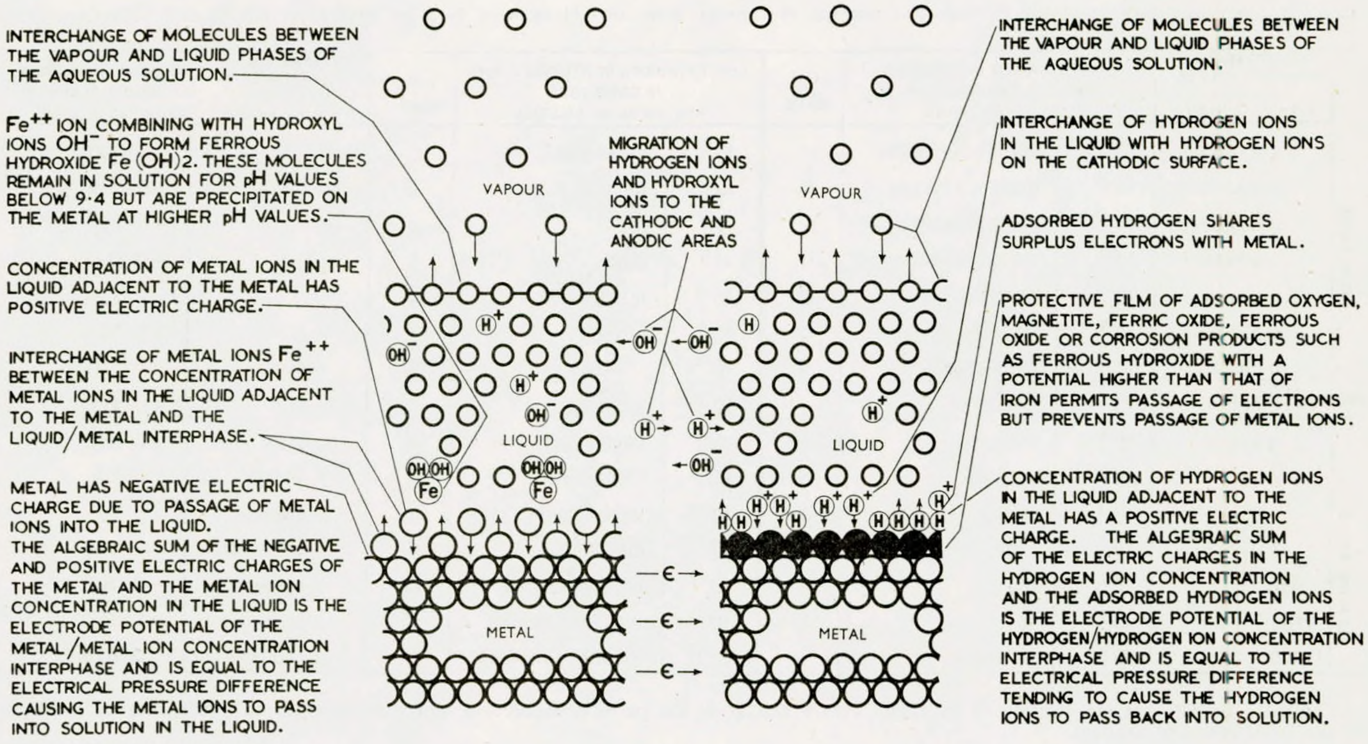


DIAGRAM OF CONDITIONS
AT ANODIC SURFACES

DIAGRAM OF CONDITIONS
AT CATHODIC SURFACES

FIG. 18—*Diagram of conditions at anodic* (left) *and cathodic* (right) *surfaces*

molecules. In the metal, the atoms are a compact aggregate relatively static with the electronic zones in close contact or slightly interpenetrating.

In the liquid, the ions and molecules of the liquid are slightly wider in spacing and are in random motion resulting in numerous collisions between the ions and molecules and the metal boundaries. In the vapour space, the molecules are much more widely spaced and at the liquid vapour surface, there is a continuous interchange of molecules between the liquid and the vapour. A very similar interchange takes place between the metal and what may be considered as the metal vapour adjacent to the metal surface.

An electro-chemical cell can be regarded as two half cells which can be considered separately (2). In both half cells, the action creating the potential is similar.

In the anodic half cell, the metal dissolves into solution by the metal ions passing into the liquid. This is a reversible action and at any given movement there is a number of free metal ions in the liquid adjacent to the metal face. The metal ions are positively charged and the metal is left with a negative electric charge. There is, therefore, an electrical pressure difference between the metal and the adjacent liquid which is called the solution pressure or the potential of the metal interphase. The interphase has a thickness of the order of a molecule.

Each metal or metal salt has a potential which depends upon the metal ion concentration at the interphase and the temperature. When anodic action occurs, the dissolution of the metal proceeds until the liquid adjacent to the metal is saturated with metal ions at which point there will be an equal number of metal ions entering and leaving the liquid and the dissolution of metal will cease unless other actions occur. For the dissolution of the metal to proceed, the metal ions must be removed or they must be neutralized by combination with negatively charged ions in the liquid.

This is illustrated in Fig. 18 by the combination of the positive ferrous ions with the negative hydroxyl ions to form ferrous hydroxide which is neutral. The corrosion product, ferrous hydroxide will be retained in solution in the liquid, or, if the liquid is saturated with ferrous hydroxide, it will be precipitated at the metal face and may tend to stifle the corrosive action.

Any metal areas which are cathodic due to the presence of adsorbed oxygen or corrosion products will prevent the passage of metal ions into the liquid but they provide areas on which hydrogen ions make contact and are adsorbed by sharing electrons with the metal. The action is a reversible one and the hydrogen at the cathodic area behaves very similarly to the metal at an anodic area, there being a continuous interchange of hydrogen ions between the surface and the liquid. The hydrogen ions adjacent to the surface are positively charged while each hydrogen ion adsorbed on the metal surface is bonded by two electrons and has a negative charge.

The difference in the electrical charges of the hydrogen ions in the liquid adjacent to the cathodic areas and the adsorbed hydrogen on the cathodic surface is the hydrogen potential of the cathodic surface.

The difference between the hydrogen potential at the cathodic area and the metal potential at the anodic area is the electromotive force of the complete cell. When a difference of potential exists, hydroxyl ions move to the anodic area and hydrogen ions move to the cathodic area, with a corresponding passage of electrons through the metal, the action tending towards electrical equilibrium. If equilibrium is reached the corrosive action ceases.

RESISTANCES TO CURRENT FLOW

In an electro-chemical cell, the magnitude of the current flowing is determined by the electromotive force of the system, which is the difference in potential between the cathode and the anode, and this is balanced by the resistances, which may be

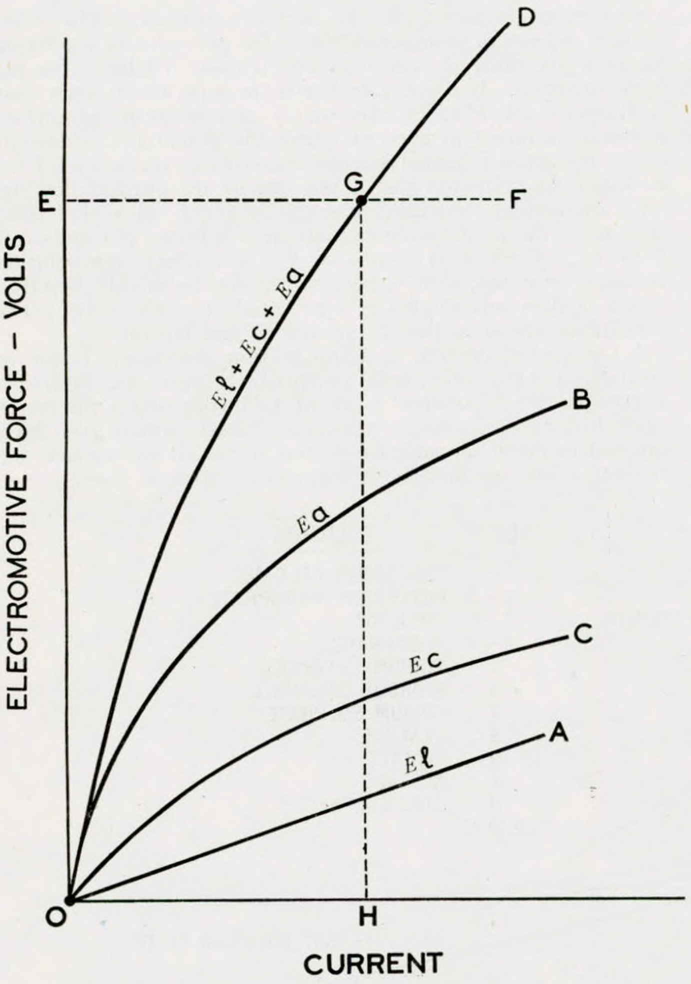


FIG. 19—*Diagram of resistances in an electro-chemical cell*
The corrosion rate is proportional to the current.

expressed as counter-electromotive forces, and can be divided as follows:—

$E = E_a + E_l + E_c + E_m$ where E is the electromotive force between the anode and the cathode.
E_a is the resistance to the passage of metal ions across the anode interphase.
E_l is the resistance to the passage of the hydroxyl and hydrogen ions through the liquid.
E_c is the resistance to the passage of hydrogen ions across the cathode interphase.
E_m is the resistance to the passage of electrons through the metal.

In most cases the resistance to the flow of electrons through the metal is so relatively small that it can be neglected. The effect of the factors mentioned is illustrated by Fig. 19 (8). Each individual resistance and the total resistance increase as the current flow is increased. Curve OA shows the resistance through the liquid while OB and OC show the resistances across the anodic and cathodic interphases respectively.

The summation of these resistances is shown by OD. If the electromotive force of the system is equal to EF then the intersection of OD with EF at G gives the rate of current OH which gives equilibrium between the driving force and the total resistance. The corrosion rate CR is directly proportional to the current flow so that OH also indicates the corrosion rate.

The interaction of these factors is perhaps more clearly shown by Fig. 20 where OA is the anodic-potential and OC the cathodic potential at no current flow. CD represents the resistance, i.e., potential difference, across the cathodic interphase, increasing as the current increases, while AE represents the increase of the resistance, i.e., potential difference, across the anodic interphase with respect to current flow. For a negligible resistance through the liquid, the current has a limiting value of OF corresponding to G, the intersection of curves AE and CD, but for a given value of liquid resistance, say HJ, the current or corrosion rate is reduced to OK.

It will be seen that the current, and therefore the corrosion rate, can be reduced by raising the potential of the anodic surface or lowering the potential of the cathodic surface. The potential difference or resistance across the cathodic interphase at the current flow of OK is equal to CL, while the potential difference across the anodic interphase is AM. If the anodic interphase potential is changed to A_1E_1, then the current would be reduced to OK_1 for the liquid resistance of J_1H_1.

The problem in boiler corrosion is to provide conditions which will reduce the current to a negligible amount. The cathodic potential is determined by the hydrogen ion concentration of the water and this must be maintained at a value which will induce stifling of anodic areas by the precipitation of small amounts of corrosion products, so as to raise any anodic potentials to the same value as the hydrogen potential, thereby eliminating any current flow which would maintain active corrosion.

CONDITIONS PRODUCING PASSIVITY

As explained, an electro-chemical cell can be regarded as two half cells coupled together by the main body of the liquid and each half cell can be considered separately. To prevent corrosion it is necessary to prevent the passage of the metal ions into the liquid, and this is effected if the whole of the metal surface has a potential equal to or electro-positive relative to the liquid hydrogen ion concentration potential. For this purpose, the liquid must provide conditions which will cause a negligible corrosive action to build up, between the metal and the liquid, an interphase which is equal to or electro-positive relative to the datum provided by the liquid. This

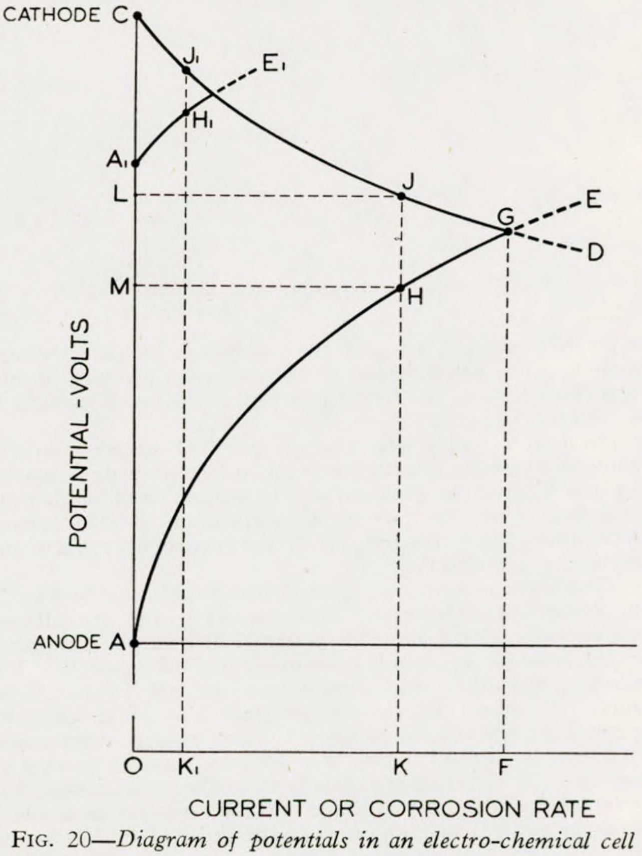


FIG. 20—*Diagram of potentials in an electro-chemical cell*

exists at cathodic areas, so the requirement is that any electro-chemical action which occurs must be small in magnitude and the interchanges at any anodic areas must form on the anodic surfaces an interphase which increases in potential relative to the liquid until the electro-chemical action is stopped. Electro-chemical action thereby creates an interphase which progressively becomes more positive in potential until the electro-chemical action is reduced to zero, i.e., it is self-eliminating.

This action is illustrated by Fig. 18. The surface of the metal adsorbs atoms of hydrogen and oxygen in close adhesion to the metal lattice so that an interphase is created between the metal and the liquid, and this interphase is at a higher potential than the bare metal. Hydrogen, oxygen and the corrosion products of iron are all higher in potential than iron and their adsorption or close precipitation on the surface of the iron raises the potential of the interphase. With a *p*H value of 9·4 the precipitation of $Fe(OH)_2$ required to raise the potential of the surface to equal the hydrogen potential and stop the passage of metal ions is extremely small. The relevant factors in the creation of passivity are the adsorption of hydrogen and oxygen on to the metal surface and the solubilities of the corrosion products, since precipitation of the corrosion products occurs when the solubility saturation point of any type of reaction in the solution is reached.

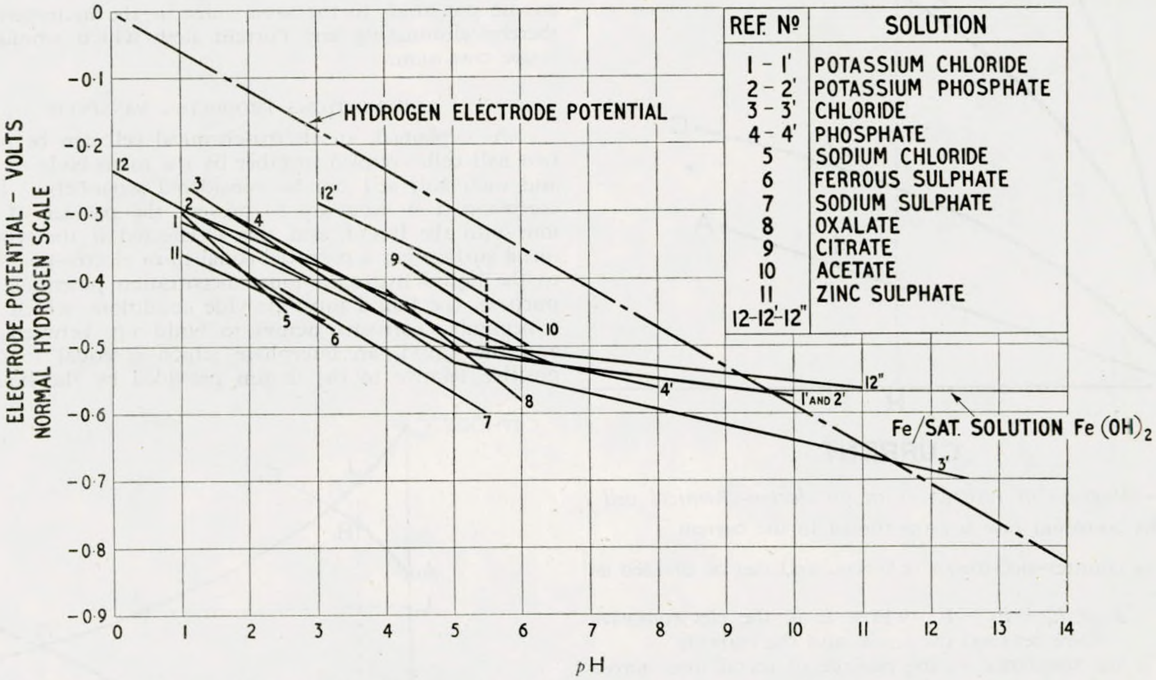


FIG. 21—*Experimental results showing the electrode potential of iron in relation to the hydrogen potential corresponding to* pH *value of various air free solutions*

Iron in a liquid which is air free will adsorb atoms of hydrogen which mesh with the metal surface forming a hydride film which raises the potential of the surface, until equilibrium is reached, when the current is stopped, or at such current which flows due to the removal of the corrosion products and hydrogen gas evolution.

The adsorption of hydrogen on iron is indicated by Fig. 21 which shows a collection of experimental results, by different experimenters, of the electrode potential of iron plotted against the *p*H value of the liquids used in air free solutions (9). The hydrogen potentials corresponding to the *p*H values of the liquids are shown by the chain-dotted line. The electrode potentials of the iron as measured in the various experiments, are shown by the full lines. For hydrogen gas to be evolved there must be a voltage gradient between the metal surface and the liquid, known as over-voltage, and the over-voltage at which hydrogen gas will just appear is of the order of 0·17 volts, the over-voltage increasing as the current increases. The over-voltage appears to be independent of the *p*H value of the liquid up to a *p*H value of 5, beyond which value it falls as the *p*H value increases. It is considered that the experiments show that hydrogen is adsorbed on the surface of iron until the surface potential is raised to a point where the potential corresponds to the hydrogen potential less the over-voltage necessary for the evolution of hydrogen gas at the rate of the current flowing. The gradient of the curves changes at a *p*H value of 5 until they cross the hydrogen potential curve between *p*H values of 9 to 10. When a *p*H value of 9·4 is reached, the solution becomes saturated with respect to ferrous hydroxide $Fe(OH)_2$ which is then precipitated at a potential of −0·57 volts. The conditions shown in Fig. 21 are for air free liquids.

Oxygen adsorption is a much more important factor in creating a high interphase potential, because the reversible oxygen electrode potential is about 1·23 volts above the reversible hydrogen electrode potential. Steel which has been exposed to moist air adsorbs oxygen atoms on the surface, the oxygen atoms meshing with the metal lattice to form a very thin invisible oxide film which is ferric oxide Fe_2O_3. This increases the potential of the oxide surface to an extent depending upon the completeness of the oxide film. With an electro-negative metal like iron the volume of the oxide film is liable to be less than that of the quantity of metal from which it is derived so that there is always the possibility of weak places or leaks in the oxide film through which Fe ions can pass into the water, and these weak places must be reinforced and repaired by the close deposition of corrosion products. These corrosion products can be ferrous carbonate $FeCO_3$, ferric hydroxide $Fe(OH)_3$, and ferrous hydroxide $Fe(OH)_2$. Precipitation of the corrosion products occurs when the saturation point is reached and this is determined by their solubility in the liquid. When iron goes into solution there are two kinds of ions, ferrous ions, $Fe-2e$, and ferric ions $Fe-3e$, but the number of ferric ions is extremely small as compared with the ferrous ions. The solubilities of the ferrous corrosive products are in the order of ferrous hydroxide, ferric hydroxide, and ferrous carbonate, and correspond to *p*H values of 9·4, 7·0 and 6·4 respectively (6). At *p*H values below these figures, the respective corrosion product will remain in solution but at higher

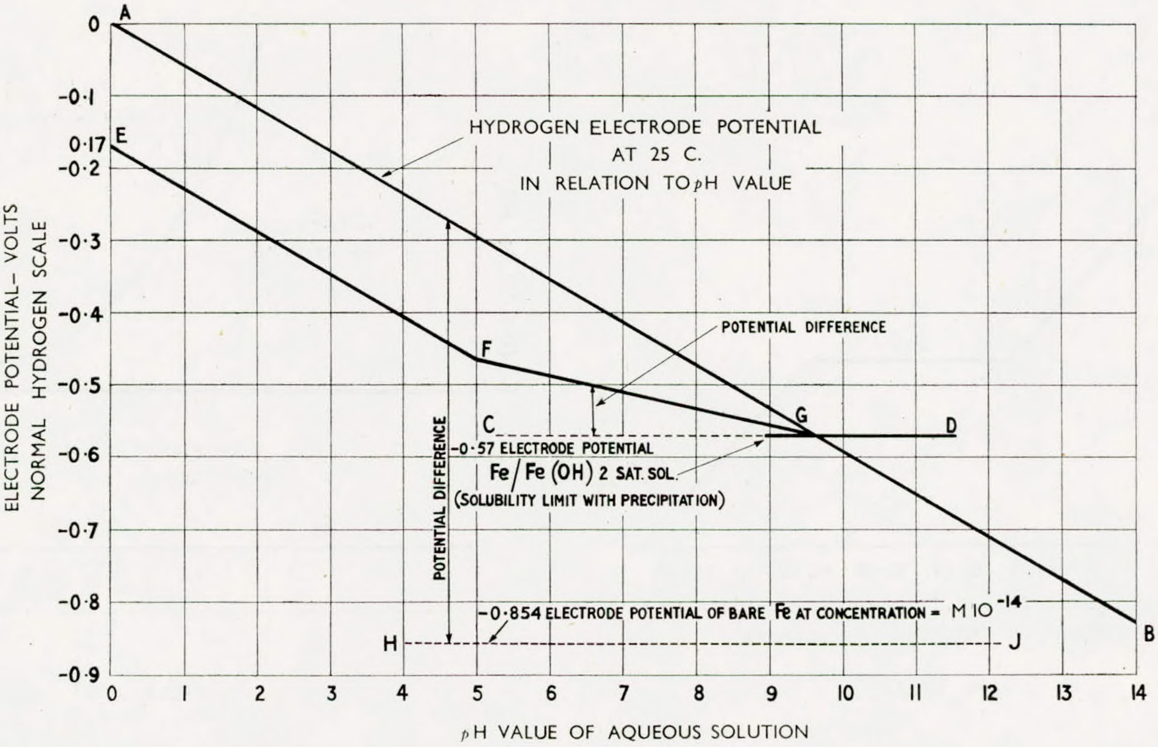


FIG. 22—*Diagram of possible potential differences causing corrosion of iron in aqueous solution of varying* pH *value*

*p*H values it will be precipitated. The closer the precipitation occurs to the surface of the metal the closer will the corrosion products adhere to or form a film on the surface of the metal.

Experiments with aerated liquids show that the electrode potential of iron in aerated liquids reaches a steady state value of −0·57 volts which is caused by the precipitation of ferrous hydroxide $Fe(OH)_2$ as the corrosion product, with a corresponding *p*H value of 9·4 for the liquid.

The potential conditions causing corrosion of iron may be illustrated as shown in Fig. 22. The cathodic potential will be that of the hydrogen ion concentration and for the different *p*H values is shown by the line AB.

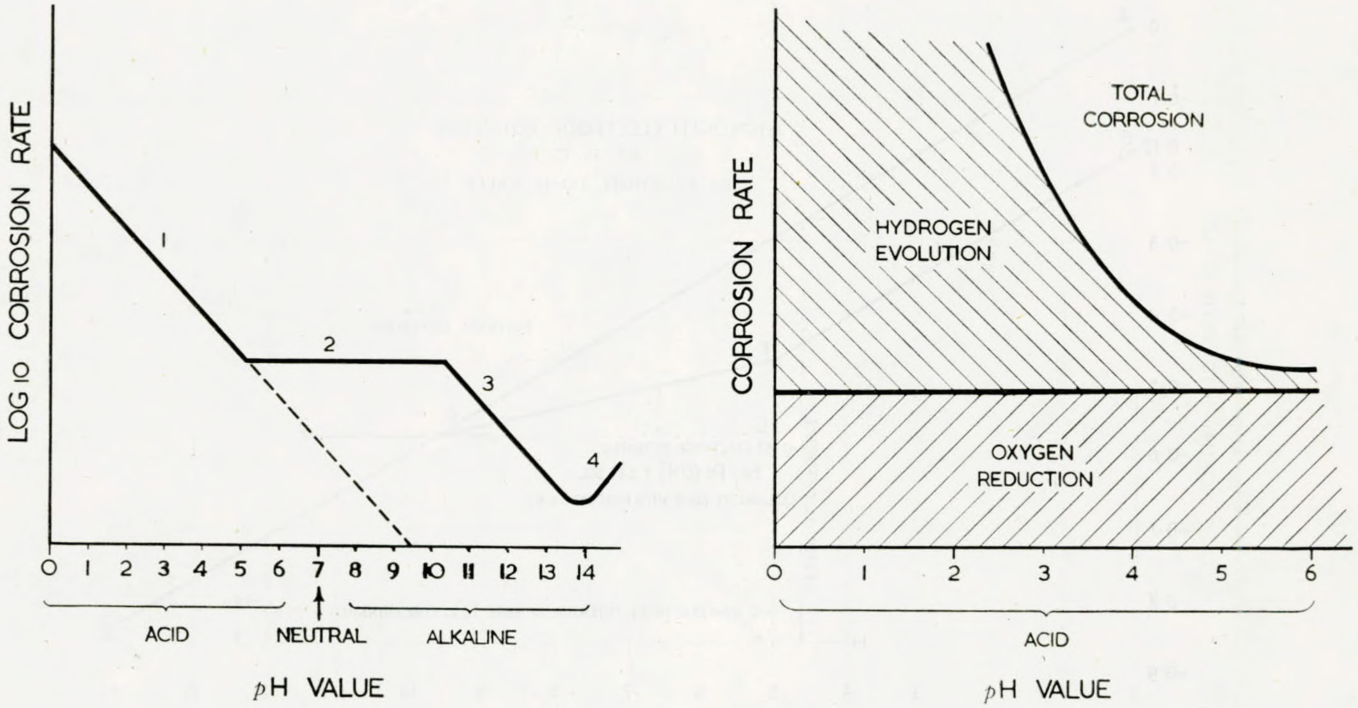
The potential of any anodic areas of the iron surface will be determined by various factors such as the adsorption of hydrogen and oxygen and the deposition of corrosion products. Bare iron with an extremely low metal ion concentration such as 10^{-14} has a potential of −0·854 volts, shown by the line HJ. Adsorption of hydrogen or oxygen on the metal face will raise the average value of the potential of the metal surface so that the potential of an anodic area may be anywhere between the lines AB and HJ. Due to hydrogen over-voltage, the cathodic potential is likely to be lower than AB. The line EFG represents the hydrogen gas over-voltage line at zero current and the electromotive force available to cause corrosion may be from EFG to any point below that line.

When corrosion occurs, ferrous hydroxide $Fe(OH)_2$ is formed and this is soluble in *p*H values up to 9·4; at this *p*H value and above the ferrous hydroxide is precipitated and if precipitated closely on the metal face it forms a protective film which has a potential of −0·57 volts, shown by CD. At *p*H values above 9·4, the formation of precipitated ferrous hydroxide should raise the potential of the metal face until it is equal to or higher than the hydrogen potential and so prevent corrosion taking place. If, at any place, the precipitation does not occur close to the metal surface but is loose, then pitting may occur under the loose skin of the corrosion products.

From experimental results, the rate of corrosion due to hydrogen gas evolution is proportional to the hydrogen ion concentration, and the rate of corrosion due to oxygen reduction is proportional to the amount of dissolved oxygen supplied. Fig. 23 shows how corrosive activity varies with the *p*H value of an aqueous solution (9). Since the hydrogen ion concentration follows a logarithmic law, the logarithm of the corrosion rate has been plotted in Fig. 23a giving a straight line for the corrosion rate caused by hydrogen evolution, this forming the first of the curves from which it will be seen that the corrosion rate due to hydrogen evolution falls until it is a negligible amount at a *p*H value of 9·4 and above. At all *p*H values below 9·4, hydrogen gas is evolved at the cathodic surface with the passage of ferrous ions into solution at the anodic surface where they combine with hydroxyl ions as shown in Fig. 4 to form ferric hydroxide or ferrous hydroxide. Ferric hydroxide is precipitated in small amount at a *p*H value of over 7·0 but it is colloidal in character and may migrate to the cathodic area so that its protective value is small.

Ferrous hydroxide passes into solution up to a *p*H value of 9·4 beyond which precipitation occurs. With a strongly alkaline solution, the precipitation occurs in close contact with the surface of the metal and forms a protective film thereby sealing up anodic areas. The higher the hydroxyl ion concentration the closer the precipitation occurs to the metal face.

If dissolved oxygen is present, the corrosion rate is proportional to the oxygen supplied and is indicated by line 2 in Fig. 23a. Due to the over-voltage effect it is considered that corrosion is primarily caused by dissolved oxygen at *p*H values above about 5 but at values below *p*H 5 corrosion can occur due to both hydrogen evolution and oxygen reduction occurring at the same time. As the *p*H value is increased above 9.4, the concentration of hydroxyl ions increases very rapidly so that the precipitation of ferrous hydroxide occurs in close optical contact with the metal, thereby preventing the passage of metal ions into the liquid.



a *b*

FIG. 23—*Corrosion rate for iron with* pH *values showing the effect of hydrogen evolution and oxygen reduction*

The third phase is shown by the line 3 where the strong precipitation of ferrous hydroxide stifles the corrosion even in the presence of dissolved oxygen, but a point is quickly reached at which the strong caustic action of the liquid dissolves any protective film and causes the formation of ferroate as shown by curve 4, where the corrosion rate increases again. This diagram shows clearly:—

(a) The necessity of a *p*H value of 9·4 as a minimum to avoid corrosion due to hydrogen gas evolution.
(b) The necessity of reducing the amount of dissolved oxygen to a minimum.
(c) The probable range in *p*H values in which passivity with immunity from corrosion may be obtained.

The use of too high a *p*H value carries with it the risk of caustic cracking and caustic corrosion and involves a high solid content in the boiler water or a large blow down. It also carries a risk of electro-chemical attack on any metals in the feed and boiler systems other than steel. The physical facts suggest that the *p*H value of the feed water should be maintained between 8·5 and 9·5, tending towards the latter value. The conditions for the boiler water are discussed later.

To secure passivity it will be seen that the liquid must be conditioned to create conditions which will cause the surface of the metal to attain a potential close to or greater than the hydrogen potential, and this is effected if the *p*H value is 9·4 or over, and all dissolved oxygen is removed.

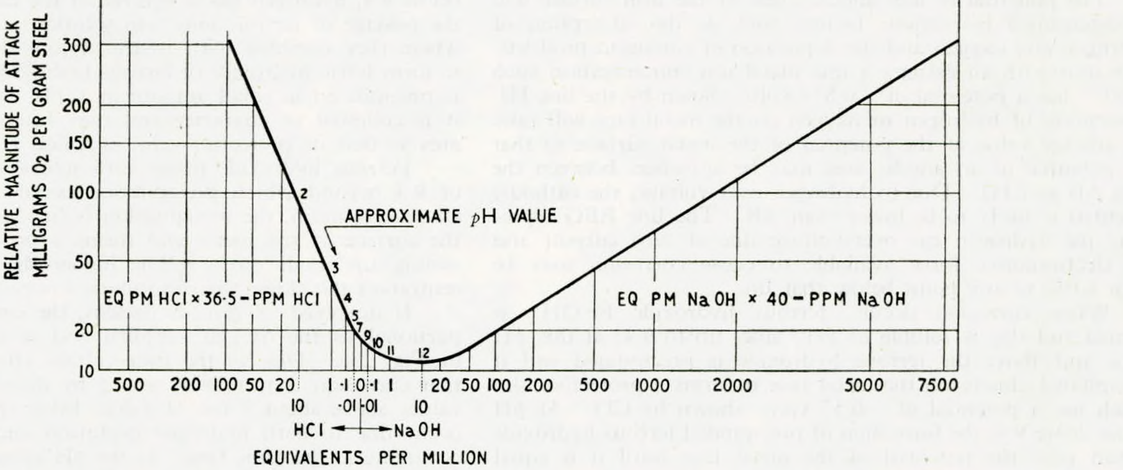


FIG. 24—*Relative attack on steel at 590 deg. F. by hydrochloric acid and sodium hydroxide*

Attack on powdered steel for 75 hours at 590 deg. F. and 1,418lb. per sq. in. gauge by water of varying degrees of acidity and alkalinity.

It should be appreciated that corrosion is an extremely complex problem in which there are many factors which are inter-dependent and vary over wide ranges. It is difficult, if not impossible to do more than give a very much over-simplified picture of the processes at the low pressures and temperatures which have been convenient for experimentation and the data given are all for atmospheric pressure and low temperatures.

Two sets of conditions have to be met in boiler operation. First, the standing or idle conditions where the data given apply, and the second, the running conditions, where the pressure and temperature are much higher and vary over wide ranges. In general, it would appear to be much easier to avoid corrosion in a boiler under running conditions than when standing, and this is confirmed by general experience.

There is very little data available on corrosion at high temperatures, but the importance of the matter will undoubtedly lead to investigations into corrosion processes at the pressures and temperatures of modern high pressure boilers. The potential of bare iron becomes more electro-negative as the temperature is increased so that any corrosion which may occur will probably be intensified. Further, the stress due to working pressures and differential temperatures increases the risk of corrosion. There is also the difficulty of obtaining and maintaining the required *p*H value at all parts of the plant at all times. It would appear that the *p*H value of the boiler water, for the operating conditions, should be such as will allow for the increase in hydrogen ion concentration at the operating temperature and should include some allowance to cover possible variations in control and ensure that not less than the desired *p*H value is maintained throughout the boiler at all times.

Fig. 24 shows some data on the corrosion of steel in water at a temperature of 590 deg. F., from which it will be seen that the magnitude of attack increases with increasing acidity or alkalinity, and is a minimum in the *p*H range of 11 to 12 (10). The test results on which Fig. 24 is based gave a minimum in corrosive attack at an alkalinity of 700 p.p.m. of NaOH, followed by a progressive increase in corrosion at higher alkalinities. There is therefore an increasing practice of operating high pressure boilers with an overall *p*H value of 11 to 12. This requires careful control over the feed and boiler water which must be conditioned and controlled within the desired limits.

BOILER WATER TREATMENT

The treatment of boiler feed water should have the following objects: —

(a) Prevention of scale formation in the feed and boiler systems.
(b) Control of the sludge formation and the total dissolved solids in the boiler.
(c) Avoidance of carry over with the boiler steam.
(d) Prevention of corrosion in the feed and boiler systems.
(e) Neutralization of any residual corrosive gases which may be left in the feed water.

It is extremely desirable that distilled water should be used in high-pressure boilers but there are many plants in which make-up feed is taken from a town supply or other convenient source. Water from such sources often contains undesirable solids from the earth's crust such as calcium and magnesium salts which are hardness producing substances. The earth's crust, the sea and the atmosphere are largely composed of eleven elements: oxygen, hydrogen, silicon, aluminium, iron, calcium, magnesium, sodium, potassium, carbon and chlorine. In marine plants, ingress of sea water into the feed systems may occur with consequent undesirable solid matter. A typical sample of sea water is as follows (3): —

	p.p.m.
Calcium bicarbonate, $Ca(HCO_3)_2$ ...	180
Calcium sulphate, $CaSO_4$	1,220
Magnesium sulphate, $MgSO_4$ ...	1,960
Magnesium chloride, $MgCl_2$ ...	3,300
Sodium chloride, NaCl	25,620
	32,280

("p.p.m." is "parts per million" by weight, compared with pure water and is the weight of each substance in pounds per million pounds of pure water.)

The total dissolved solids in sea water are about 32,000 parts per million, of which 25,600 are sodium chloride, the remainder being salts of calcium and magnesium. It is very important, therefore, to prevent the ingress of sea water into boilers, and on board ship it is now the almost invariable practice to use double distillation for make-up feed water to ensure the least possible sea contamination.

Where necessary, chosen solids are added to feed water to counteract any scale forming substances which may be in the water, and to increase its hydroxyl ion concentration to prevent corrosion.

Every possible step should be taken to reduce the presence of contaminating solids to a minimum so that the required amounts of corrective additions are also reduced to a minimum.

SUBSTANCES AND THEIR MOLECULAR GROUPINGS

To understand the corrective actions it is necessary to consider the principal molecular groupings which take place and those which are desirable. The main solids or metals which may be present in the water are shown in Table 4 below.

Hydrogen has been included in the list of metals because in many respects it behaves in the same way as a metallic substance.

When the above metals combine with hydroxide, OH, they form bases but when they combine with an acid radical they form salts. The calcium and magnesium salts are the chief cause of trouble in boilers.

Valence is the measure of the combining ratio of a substance or compound taking hydrogen as the datum of one. The number of atoms of hydrogen displaced by or combined with one atom of a given element is the measure of the valence of that element. Elements or compounds combine in the ratio of one to one for the same valence but inversely as their valence when their valences differ.

The valency is determined by the arrangement of the electrons in the outermost shell of an element, the shell tending to give up or acquire electrons in order to reach the most stable arrangement of electrons.

Metallic Elements

TABLE 4

Metal	Symbol	Valence or combining ratio	Relative atomic weight	Equivalent weight	Normal complement of electrons	Arrangement of electrons in shells	Ion
Calcium	Ca	2	40	20	20	2.8.8.2	Ca^{++}
Magnesium ...	Mg	2	24·3	12·15	12	2.8.2	Mg^{++}
Sodium	Na	1	23	23	11	2.8.1	Na^{+}
Potassium ...	K	1	39·1	39·1	19	2.8.8.1	K^{+}
Aluminium ...	Al	3	27	9	13	2.8.3	Al^{+++}
Iron	Fe	2	55·8	27·9	26	2.8.14.2	Fe^{++}
		3	55·8	18·6	26	2.8.14.2	Fe^{+++}
Barium	Ba	2	137·4	68·7	56	2.8.18.18.8.2	Ba^{++}
Hydrogen ...	H	1	1	1	1	1	H^{+}

The relative atomic weight is the weight of a given number of atoms of a substance relative to the weight of the same number of atoms of hydrogen. The atomic weights given are to the nearest convenient round figures.

The weight of 602,000,000,000,000,000,000,000, i.e., $6{\cdot}02 \times 10^{23}$ atoms of hydrogen is 1·008 grams while the weight of the same number of calcium atoms is 40 grams. 1lb. of hydrogen, therefore, contains the same number of atoms as 40lb. of calcium or 23lb. of sodium.

The weight of an ion can be taken as the same as the weight of the atom because a proton or neutron weighs 1,830 times the weight of an electron.

A chemical equation gives the ratios in which the atoms and molecules stated take part in the reaction given by the equation. The same ratios can be expressed in the respective atomic and molecular weights. If the weight of any substance taking part in the reaction covered by the equation is known, then the weights of the other substances taking part in the reaction are proportional to the respective atomic and molecular weights.

The equivalent weight is the atomic weight or molecular weight divided by the valence and is a convenient relation for enabling the weight of a substance to be expressed in terms of weight of another substance.

Other non-metallic elements which enter into this consideration are given below in Table 5.

Non-Metallic Elements

TABLE 5

Element	Symbol	Valence or combining ratio	Relative atomic weight	Equivalent weight	Normal complement of electrons	Arrangement of electrons in shells	Ion
Oxygen	O	2	16	8	8	2.6	O^{--}
Nitrogen ...	N	Variable (3)	14	4·7	7	2.5	N^{---}
Sulphur	S	Variable (2)	32·1	16	16	2.8.6	S^{--}
Chlorine	Cl	1	35·5	35·5	17	2.8.7	Cl^{-}
Phosphorus ...	P	3	31	10·3	15	2.8.5	P^{---}
Carbon	C	Variable (4)	12	3	6	2.4	$C^{\pm 4}$
Silicon	Si	Variable (4)	28·1	7	14	2.8.4	$Si^{\pm 4}$

The valence figure given in brackets is the common valence for the element.

Water

Water is the liquid mainly concerned with here and pure water is as follows: —

TABLE 6

Compound	Symbol	Combining ratio	Relative atomic weight	Equivalent weight
Water	H_2O	1	18	18

Acid Radicals

An acid radical is, in general, a molecular grouping which usually moves as a unit in any regrouping and the principal acid radicals to be considered are given in Table 7.

TABLE 7

Radical	Symbol	Valence or combining ratio	Relative molecular weight	Equivalent weight	Ion
Bicarbonate	HCO_3	1	61	61	$(HCO_3)^{-}$
Carbonate	CO_3	2	60	30	$(CO_3)^{--}$
Sulphate	SO_4	2	96·1	48	$(SO_4)^{--}$
Sulphite	SO_3	2	80·1	40	$(SO_3)^{--}$
Phosphate	PO_4	3	95	31·7	$(PO_4)^{---}$
Nitrate	NO_3	1	62	62	$(NO_3)^{-}$
Hydroxide	OH	1	17	17	$(OH)^{-}$
Chloride	Cl	1	35·5	35·5	Cl^{-}

It will be seen that the radicals are largely oxygen compounds. In their ionic form they have a very stable electronic configuration, as will be discussed later.

The molecular weight is the combined atomic weights of the atoms in the molecule of the given compound. For example the atomic weights of sulphur, S, and oxygen, O, are 32·1 and 16 respectively, so the molecular weight of sulphite, SO_3, is 32·1 plus $(3 \times 16) = 80{\cdot}1$.

Acids

The above acid radicals when they combine with hydrogen, H, form an acid, as follows (Table 8): —

TABLE 8

Acid	Symbol	Combining ratio	Relative molecular weight	Equivalent weight
Carbonic acid	H_2CO_3	2	62	31
Sulphuric acid ...	H_2SO_4	2	98·1	49
Sulphurous acid ...	H_2SO_3	2	82·1	41
Phosphoric acid ...	H_3PO_4	3	98	32·7
Hydrochloric acid ...	HCl	1	36·5	36·5

Carbonic acid H_2CO_3, when heated, breaks up into H_2O and CO_2.

Calcium Compounds

The compounds of the acid radicals with calcium and magnesium are the chief factors in scale formation, and are as given in Tables 9-12: —

TABLE 9

Compound	Symbol	Combining ratio	Relative molecular weight	Equivalent weight
Calcium bicarbonate	$Ca(HCO_3)_2$	2	162	81
Calcium carbonate	$CaCO_3$	2	100	50
Calcium hydroxide	$Ca(OH)_2$ (Slaked Lime)	2	74	37

The above are known as alkaline hardnesses (often called carbonate or temporary hardnesses). The following are all known as non-alkaline hardnesses (often called non-carbonate or permanent hardnesses).

TABLE 10

Compound	Symbol	Combining ratio	Relative molecular weight	Equivalent weight
Calcium sulphate	$CaSO_4$	2	136·1	68
Calcium silicate	$CaSiO_3$	2	116·1	58
Calcium nitrate	$Ca(NO_3)_2$	2	164	82
Calcium chloride	$CaCl_2$	2	111	55·5
Tricalcium phosphate	$Ca_3(PO_4)_2$	6	310	51·7

$Ca(HCO_3)_2$ is decomposed by heating when carbon dioxide CO_2 is driven off, and calcium carbonate $CaCO_3$ is precipitated, i.e., $Ca(HCO_3)_2 \rightarrow \underline{CaCO_3} + H_2O + \overline{CO_2}$.

(In a chemical formula, the total number of atoms of each particular element must be the same on each side of the equation. In the same way the total of the electric charges must balance out on each side of the equation.) A line under a symbol indicates a precipitate or solid, e.g., $\underline{CaCO_3}$. A line above a symbol indicates a gas, e.g., $\overline{CO_2}$.

$CaCO_3$ is largely insoluble and forms a soft sludge or soft scale. $CaSO_4$ is the chief constituent of hard scale.

Calcium silicate may be a constituent of the thin hard and non-conducting silicate scales which are troublesome to remove and may cause overheating.

Calcium hydroxide, commonly called slaked lime, is frequently used for boiler water conditioning for the removal of temporary hardnesses as follows: —

$$Ca(OH)_2 + Ca(HCO_3)_2 \rightarrow \underline{2CaCO_3} + 2H_2O$$

$$2Ca(OH)_2 + Mg(HCO_3)_2 \rightarrow \underline{2CaCO_3} + \underline{Mg(OH)_2} + 2H_2O.$$

Slaked lime also reacts with permanent magnesium hardnesses but produces an equivalent amount of temporary calcium hardnesses. This is known as the lime treatment and is frequently combined with the soda treatment mentioned later.

Tricalcium phosphate is a soft sludge which is formed in phosphate conditioning which will be described later.

Magnesium Compounds

TABLE 11

Compound	Symbol	Combining ratio	Relative molecular weight	Equivalent weight
Magnesium bicarbonate	$Mg(HCO_3)_2$	2	146·3	73·1
Magnesium carbonate	$MgCO_3$	2	84·3	42·1
Magnesium hydroxide	$Mg(OH)_2$	2	58·3	29·1

The above are known as alkaline hardnesses (often called carbonate or temporary hardnesses). The following are all known as non-alkaline hardnesses (often called non-carbonate or permanent hardnesses).

TABLE 12

Compound	Symbol	Combining ratio	Relative molecular weight	Equivalent weight
Magnesium sulphate	$MgSO_4$	2	120·4	60·2
Magnesium silicate	Mg_2SiO_4	4	140·7	35·2
Magnesium nitrate	$Mg(NO_3)_2$	2	148·3	74·1
Magnesium chloride	$MgCl_2$	2	95·3	47·6

$Mg(HCO_3)_2$ is decomposed by heating when CO_2 is driven off and magnesium carbonate $MgCO_3$ is formed, i.e., $Mg(HCO_3)_2 \rightarrow MgCO_3 + H_2O + \overline{CO_2}$.

$MgCO_3$ is relatively insoluble but it is decomposed into magnesium hydroxide and carbon dioxide as follows: —

$$MgCO_3 + H_2O \rightarrow \underline{Mg(OH)_2} + \overline{CO_2}.$$

$MgSO_4$ and $MgCl_2$ are soluble in boiler conditions. Magnesium chloride is decomposed to some extent under boiler conditions into magnesium hydroxide and hydrochloric acid as follows: —

$$MgCl_2 + 2H_2O \rightarrow \underline{Mg(OH)_2} + 2HCl.$$

Magnesium hydroxide is the commonest magnesium compound found in boiler scale and may form a hard scale but it is usually precipitated as a sludge with phosphate treatment.

Magnesium silicate forms a hard thin scale which is non-conducting, difficult to remove and may give rise to dangerous overheating.

As will be seen from the above, magnesium compounds produce acids which require neutralizing by alkaline additions.

Sodium Compounds

TABLE 13

Compound	Symbol	Combining ratio	Relative molecular weight	Equivalent weight
Sodium bicarbonate	$NaHCO_3$	1	84	84
Sodium carbonate	Na_2CO_3 (Soda ash)	2	106	53
Sodium hydroxide	NaOH (Caustic soda)	1	40	40
Sodium sulphate	Na_2SO_4	2	142·1	71
Sodium sulphite	Na_2SO_3	2	126·1	63
Sodium nitrate	$NaNO_3$	1	85	85
Sodium chloride	NaCl	1	58·5	58·5
Sodium aluminate	$Na_2Al_2O_4$	6	164	27·3

$2NaHCO_3$, when heated, breaks down into $Na_2CO_3 + H_2O + \overline{CO_2}$. Na_2CO_3, when subjected to the temperatures in high pressure boilers, breaks down and combines with water H_2O into caustic soda 2NaOH and $\overline{CO_2}$.

Sodium sulphite is used as an oxygen scavenger in the boiler after mechanical de-aeration.

A general characteristic of sodium compounds is their high solubility and, because of this, they are non scale forming.

Soda ash and caustic soda are both used for feed and boiler water conditioning because they decompose readily, they increase the alkalinity of the water, and they cause the precipitation of hardness forming substances as sludge. They are satisfactory for use up to boiler pressures of 250lb. per sq. in., but require careful consideration above that pressure because they may unduly increase the alkalinity in the boiler and the total dissolved solids present.

Sodium aluminate is a coagulant which is used to make a floc that settles easily and can be blown down.

Sodium Phosphates

TABLE 14

Compound	Symbol	Combining ratio	Relative molecular weight	Relative combining weight
Monosodium phosphate	NaH_2PO_4	3	120	40
Disodium phosphate	Na_2HPO_4	3	142	47·3
Trisodium phosphate	Na_3PO_4	3	164	54·7
Commercial trisodium phosphate ...	$Na_3PO_4 12H_2O$	3	380	127
Sodium metaphosphate	$NaPO_3$	1	102	102
Sodium hexametaphosphate	$(NaPO_3)6$	6	612	102

Sodium phosphates are widely used for water conditioning, particularly in high-pressure boilers, because they are very soluble and the phosphate radical is entirely stable at the higher operating pressures and combines with calcium and magnesium to form a sludge. The three types of phosphates mainly used in conditioning boiler water are sodium metaphosphate, disodium phosphate, and trisodium phosphate.

If sodium metaphosphate or disodium phosphate is added to boiler water containing caustic alkalinity it is converted into trisodium phosphate in the boiler.

Sodium metaphosphate is highly soluble and in a weak solution has a *p*H value of the order of 7. Disodium phosphate in a 1 per cent solution in water has a *p*H value of 9·3 and is therefore more alkaline, while trisodium phosphate in a solution of 1 per cent in water, has a *p*H value of 12·0.

Sodium metaphosphate has an advantage compared to the others in that it can be added to the feed water without causing deposition in the feed piping, but it is the more general practice to use phosphate conditioning direct into the boiler.

The different alkalinities of the phosphates mentioned enable them to be used to control alkalinity in the boiler, particularly when the feed water contains only small amounts of solid matter. Sodium metaphosphate and trisodium phosphate are those generally used when it is desired to control alkalinity by their use.

Phosphate conditioning has another advantage in that it tends to form an iron phosphate film on the boiler surfaces which is a protection against corrosion.

SCALE FORMATION AND OILY DEPOSITS

Scale formation is a complex process which is partly physical and partly chemical. When a bubble of steam is formed on a heating surface, the evaporation of the water into the bubble of steam causes a local concentration of the solids and the super saturation point of the solids may be reached when the precipitation of a ring of crystals occurs at the perimeter of the bubble on the heating surface. If the salts precipitated are highly soluble, when the bubble escapes from the heating surface, the subsequent flow of water across the surface where the bubble was formed causes the salts to re-enter into solution, but in the case of relatively insoluble salts, they may adhere to the heating surface and the repeated formation of circles of crystals builds up a scale formation.

Fig. 25, Plate 2 shows rings of calcium sulphate crystals precipitated (11) on a heating surface when water is evaporated and illustrates the beginning of scale formation. The scale formation thus formed can be augmented by the crystallization of solids in solution on to the solids precipitated by evaporation. The worst offender is calcium sulphate which crystallizes in long crystals which interlace and form a close adherent scale. This is shown in Fig. 26, Plate 2 (11) which shows the cross-sectional and surface appearance of hemihydrate and anhydrite scales produced experimentally under the following conditions: —

TABLE 15

Boiler pressure, lb. per sq. in. gauge	Duration of run, hours	Thickness of scale, inch	Crystal form of calcium sulphate in scale	Magnification
25	48	0·056	Hemihydrate	20
150	200	0·0032	Anhydrite	43

The three main substances which are precipitated in this manner are calcium sulphate, calcium silicate, and magnesium silicate. Calcium carbonate and magnesium hydroxide may be incorporated in the matrix of the scale in appreciable amounts. The presence of calcium carbonate in the scale reduces its hardness and the greater the proportion of calcium carbonate in the scale the softer the scale is until it becomes a sludge when the precipitate is formed chiefly of calcium carbonate.

The various other calcium and magnesium salts also play a part in the physico-chemical growth of scale, but the above are the main sources of trouble.

It will be appreciated that the metal of a boiler tube is maintained at a temperature intermediate between the gas temperature and the boiler water or steam temperature, and that, for safe working, the tube wall temperature must be reasonably close to the boiler water or steam temperature.

Scale formation, even of a thin character, offers a considerable resistance to heat flow and such resistance can be appreciably increased if it is combined with corrosive products or oil. Any coating of oil or scale on the water or steam side of a boiler tube will cause the tube metal temperature to rise because of the greater resistance to heat flow from the metal and such temperature rise may cause more vigorous scaling with cumulative effects resulting in rupture of the tube.

On the gas side, overheating of the tube wall causes the formation of forge scale which peels off and thereby thins the tube wall, thus contributing to the rupturing of the tube.

Defective cooling of the fire side of a tube is liable to cause serious bowing of the tube which may be such as to cause a permanent set every time the boiler goes on load. The repetition of this excessive stress every time the boiler is put into service and shut down is liable to give rise to transverse cracking of the tube wall which will be accelerated if any corrosive action is present. Scale formation may also give rise to "steam" corrosion.

Oil in the boiler water is a serious source of danger. A small amount of oil can be present without causing damage provided the boiler is free from scale forming substances but, if scale is formed, the presence of oil will promote the accumulation of scale and increase its resistance to the necessary flow of heat, thereby accelerating any destructive corrosion or overheating action which may take place.

In some cases, oil may combine with hardness producing substances and rust to form oily balls which may so seriously reduce the flow of water in a tube as to cause overheating of the tube, finally causing rupture. This danger is particularly present when a new boiler goes into service, or after a boiler has been opened up for some time, when rust may have formed inside. It is important, therefore, that the formation of scale should be avoided or minimized as far as it is practicable. To this end, the make-up feed should be as free from solid matter as possible, and any solid matter present should be treated chemically to make it non scaling. All possible steps should, of course, be taken to prevent oil entering a boiler.

HARDNESSES

The calcium and magnesium salts, which are the scale producing solids, are termed the hardness producing substances and a knowledge and measure of the various hardnesses in a water are necessary in considering the conditioning of the water. The hardnesses can be broadly divided into the following categories:—

(a) Total hardness.
(b) Alkaline or carbonate hardness (often called temporary hardness).
(c) Non-alkaline or non-carbonate hardness (often called permanent hardness).

The total hardness is the sum of the calcium and magnesium salts present and is usually expressed in terms of $CaCO_3$.

The alkaline hardness is the sum of all the bicarbonates, carbonates and hydroxides of the calcium and magnesium, while the non-alkaline hardness is the sum of all the remaining salts of calcium and magnesium, such as the chlorides, sulphates, silicates and nitrates.

TABLE 16

Degrees	Parts per million as $CaCO_3$
1 British	14·29 = 1 grain per Imperial gallon
1 American	17·16 = 1 grain per U.S. gallon
1 French	10·00
1 German	17·86

The bicarbonates are said to be "temporary hardnesses" because they are decomposed by heating or boiling into carbonates and carbon dioxide, while the magnesium carbonate is further changed into magnesium hydroxide and carbon dioxide. The calcium and magnesium salts which do not decompose in this manner are termed permanent hardnesses.

British, American, French and German degrees of hardness vary and Table 16 gives the relation between them in terms of $CaCO_3$.

Water can contain such a variety of solids in different proportions that it is convenient to be able to evaluate the scale producing solids in terms of a single solid or in degrees of hardness. This enables rough comparisons to be made between waters and permits a rough approximation to be made to the corrective treatment required. Modern boiler water treatment, however, is a much more exact science and a detailed analysis of the solid contents of feed water is necessary to enable the correct treatment to be applied, as is discussed later.

Scale formation is due to the precipitation of the hardness forming substances, the worst compounds being as stated previously. Scale can be avoided by conditioning the water so that no calcium sulphate, calcium silicate or magnesium silicate are formed at the evaporating surfaces, and any precipitates formed are sludge forming substances.

In the feed system and economizers, the main requirement to avoid scale formation is that supersaturation of calcium carbonate $CaCO_3$ or magnesium silicate Mg_2SiO_4 shall not occur.

FEED WATER ANALYSIS

When the compounds previously considered are present in solution in water they are in the form of the ions of their elements and radicals. In a chemical analysis, the quantities of the various ions are measured separately and the probable groupings of the ions into compounds are estimated on the basis of the combinations in which they would probably be precipitated from solution if the water is progressively removed. This estimate is based on the assumption that the ions are balanced to form various compounds in their increasing order of solubility as this will be the order in which they will be precipitated.

A water analysis may, therefore, be given in ionic form although the more usual and conventional way is to state the probable combinations. For water treatment purposes, the ionic analysis would be equally convenient since the knowledge required is the actual number of each ion present in the water and both forms of analysis give this information.

The quantities in a water analysis are usually given in grains per gallon or p.p.m. and, for convenience, they are often expressed in terms of a single compound such as $CaCO_3$. The equivalent weights given in the lists of elements, radicals and compounds enable the weight of any given substance to be obtained in terms of another substance by taking the ratio of their equivalent weights:—e.g., if a water contains 30 p.p.m. of magnesium sulphate, this is equivalent to $\frac{30 \times 50}{60{\cdot}2} = 25$ in terms of $CaCO_3$ where 50 and 60·2 are the equivalent weights of $CaCO_3$ and $MgSO_4$ respectively.

Table 17 gives a typical analysis of water from a London source.

TABLE 17

		p.p.m.	e.p.m.	p.p.m. as $CaCO_3$
Calcium carbonate	$CaCO_3$	197	3·94	197 Alkaline hardness
Calcium sulphate	$CaSO_4$	91	1·34	67 } Non-alkaline
Magnesium sulphate	$MgSO_4$	30	0·50	25·0 } hardness
Sodium nitrate	$NaNO_3$	34	0·40	20·0
Sodium chloride	$NaCl$	47	0·80	40·0
Silica	SiO_2	3	0·10	5·0
Iron oxide	Fe_2O_3	2	0·07	3·5
Free ammonia		Nil		
Albuminoid ammonia		Nil		
Total solids dried at 105 deg. C.		432		

It will be noted that the quantity of total solids as estimated by drying is greater than the sum of the constituents since many of the salts contain varying amounts of combined water of crystallization which is not completely removed by drying at 105 deg. C.

If the quantities are given in grains per gallon, multiply by 14·29 to obtain p.p.m.

To obtain the e.p.m. divide the p.p.m. by the equivalent weight of the compound. To obtain the p.p.m. as $CaCO_3$ multiply the e.p.m. by 50 which is the equivalent weight of $CaCO_3$.

The only alkaline hardness present in this water is the $CaCO_3$, which is 197 p.p.m. which is $\frac{197}{14{\cdot}29} = 13{\cdot}8$ degrees.

The non-alkaline hardness is the summation of the remaining calcium and magnesium salts in p.p.m. as $CaCO_3$ divided by 14·29, i.e., 67 + 25 = 92 ÷ 14·29 = 6·4 degrees.

The analysis could be equally usefully given in ionic form as follows:—

TABLE 18

		p.p.m.	e.p.m.	p.p.m. as $CaCO_3$
Calcium ions	Ca^{++}	105·6	5·28	264
Magnesium ions	Mg^{++}	6·0	0·50	25
Sodium ions	Na^+	27·7	1·20	60
Carbonate ions	CO_3^{--}	118	3·94	197
Sulphate ions	SO_4^{--}	88·2	1·84	92
Nitrate ions	NO_3^-	24·8	0·40	20
Chloride ions	Cl^-	28·5	0·80	40

For water conditioning, the calcium and magnesium ions require to be removed and their quantities together with the quantity of carbonate ions must be known to enable the required quantities of addition compounds to be estimated. The alkaline hardness is the amount of carbonate ions present, which equals 197 p.p.m., as $CaCO_3$. The non-alkaline hardnesses are the magnesium ions present plus the calcium ions not balanced with the carbonate ions (all expressed in terms of $CaCO_3$), i.e., 264 + 25 − 197 = 92 p.p.m. as $CaCO_3$. If the carbonate ions are in excess of the calcium ions the balance would be associated with magnesium ions and the hardness arrived at in the same way.

The water given above would be termed a hard water. The following analysis in Table 19 is of a Glasgow water which is a soft water.

TABLE 19

		p.p.m.	e.p.m.	p.p.m. as $CaCO_3$
Calcium carbonate	$CaCO_3$	5·6	0·112	5·6 Alkaline hardness
Calcium sulphate	$CaSO_4$	2·4	0·035	1·8 } Non-alkaline
Magnesium sulphate	$MgSO_4$	2·1	0·035	1·8 } hardnesses
Sodium chloride	$NaCl$	16·4	0·280	14·0
Silica	SiO_2	0·3	0·01	0·5
Iron oxide	Fe_2O_3	0·1	0·004	0·2
Total solids dried at 105 deg. C.		30		

Alkaline hardness 5·6 p.p.m. as $CaCO_3$ or 0·39 degrees.
Non-alkaline hardness 3·6 p.p.m. as $CaCO_3$ or 0·25 degrees.

The above analysis in the ionic form is shown in Table 20.

TABLE 20

		p.p.m.	e.p.m.	p.p.m. as $CaCO_3$
Calcium ions	Ca^{++}	2·95	0·148	7·4
Magnesium ions	Mg^{++}	0·42	0·035	1·7
Sodium ions	Na^+	6·45	0·280	14·0
Carbonate ions	CO_3^{--}	3·36	0·112	5·6
Sulphate ions	SO_4^{--}	3·38	0·070	3·5
Chloride ions	Cl^-	9·95	0·280	14·0

If the hardnesses are all expressed in terms of $CaCO_3$, simple calculations, as will be seen later, will enable the theoretical weights of the required conditioning compounds to be determined.

SOLUBILITIES

In the boiler, the precipitation of sludge instead of scale is required and in this connexion the temperature corresponding to the boiler pressure is an important factor. The temperature gradient in the water rises towards a heating surface but falls towards a cooling surface. As a general rule, therefore, substances whose solubilities increase with temperature, when concentrated beyond the saturation point, deposit as a sludge on heating surfaces but form a scale on cooling surfaces. Conversely, if a substance has a solubility which decreases with increase in temperature, when saturation point is reached, it forms scale on heating surfaces and a sludge on cooling surfaces.

Salts, bases and acids are all soluble in water to a greater or lesser extent but the factors which influence the solubility are complex and, in general, the degree of solubility of an element or compound can be determined only by experiment and is influenced greatly by the presence of other elements or compounds. In a boiler, the various solids are present in a comparatively dilute concentration, so far as the mass of the boiler water is concerned, but in hide outs and where the bubbles are formed at the evaporating surfaces, the concentration of solids reaches the limit of solubility and precipitation occurs. Both states have therefore to be considered.

The electronic configuration of a given compound and the weight of any given aggregation of molecules play a part in precipitation, but there is such a complexity of factors that, for the present, reliance has to be placed on experimental results.

It is interesting, however, to examine some of the factors. In their reactions, the atoms of elements tend to achieve the electronic arrangement of the nearest inert gas. An inert gas has the maximum possible number of electrons in each orbit and, after the first orbit, which has a maximum of two electrons, the outermost orbit always contains eight electrons. Such an arrangement is extremely stable and has practically no tendency to undergo a change. The electronic arrangements of the various inert gases are shown in Fig. 27.

The formation of compounds is due to the tendency of each atom to acquire a stable electronic arrangement with two electrons in the first orbit or eight electrons in the outermost orbit, which is the maximum the outermost orbit can contain. If an atom has three orbits only, the third orbit can contain up to eight electrons but if additional orbits are present, the third orbit can contain up to eighteen electrons, but the outermost orbit in all cases, from the second orbit to the sixth, can contain eight electrons only as a maximum.

The formation of compounds, therefore, tends to an arrangement in which each atom has the nearest approach possible to two electrons in a single orbit or, with multiple orbits, to eight electrons in its outermost orbit, and this may be

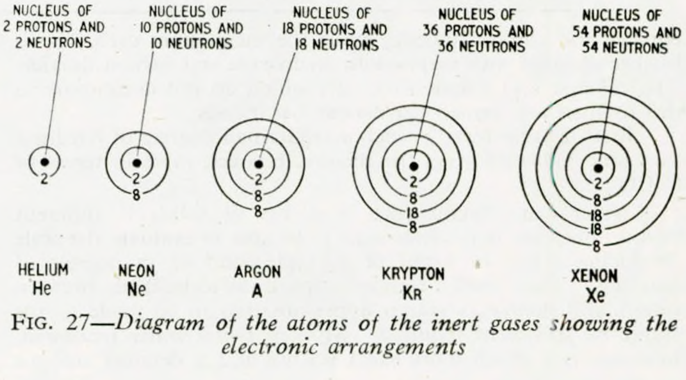
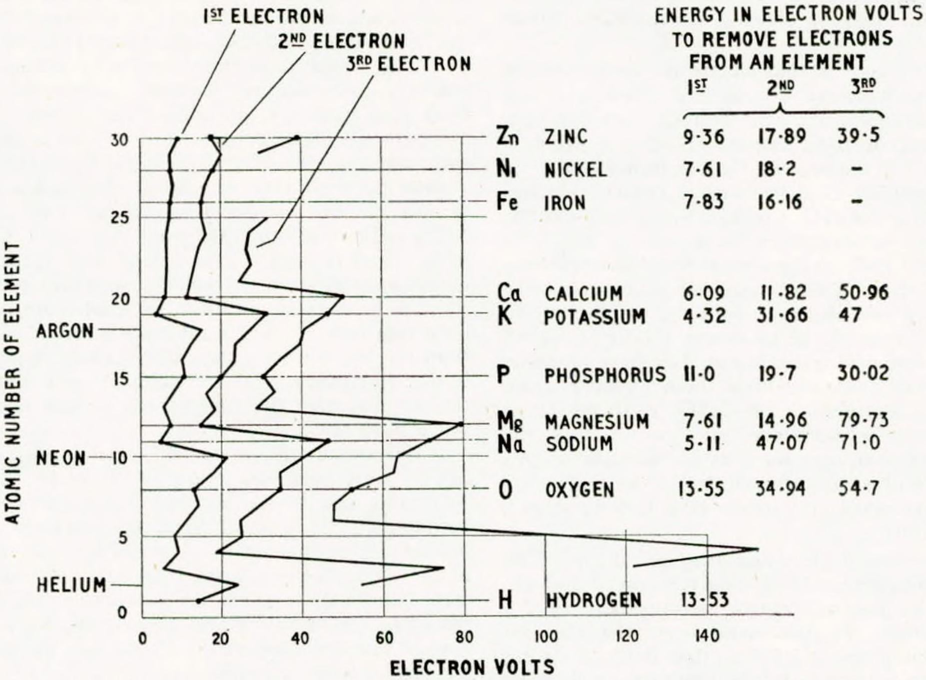


FIG. 27—*Diagram of the atoms of the inert gases showing the electronic arrangements*



ENERGY IN ELECTRON VOLTS TO REMOVE ELECTRONS FROM AN ELEMENT

	1ST	2ND	3RD
ZINC	9·36	17·89	39·5
NICKEL	7·61	18·2	-
IRON	7·83	16·16	-
CALCIUM	6·09	11·82	50·96
POTASSIUM	4·32	31·66	47
PHOSPHORUS	11·0	19·7	30·02
MAGNESIUM	7·61	14·96	79·73
SODIUM	5·11	47·07	71·0
OXYGEN	13·55	34·94	54·7
HYDROGEN	13·53		

FIG. 28—*Diagrams showing energy in electron volts (ionization potential) required to remove first, second or third electrons from an element*

obtained by losing or gaining an electron or electrons, as in ionization followed by ionic binding, or by sharing electrons as in what is termed co-valent binding, or by a combination of both forms.

The energy necessary to remove one electron from a particular atom is less than that required to remove two and so on, and these respective energies vary between the atoms depending upon how closely their electronic arrangement approaches the nearest stable configuration. Fig. 28 shows the energy necessary to remove electrons from the atoms of some of the elements concerned in this subject. Practically all chemical compounds contain an even total number of electrons and they tend to achieve the most stable arrangement. Ions, molecules or compounds in their movements in a gas or a liquid are continually coming into contact with each other, and when they approach one another sufficiently closely for interpenetration of their electronic shells to take place, the attractive or repulsive forces which exist or are set up may cause binding

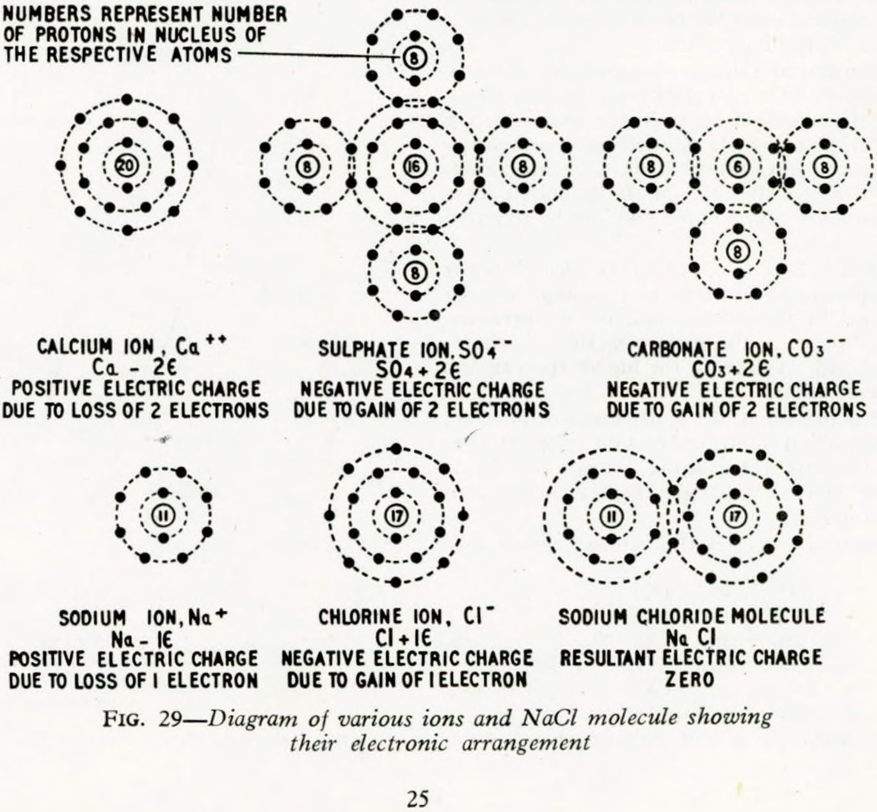


FIG. 29—*Diagram of various ions and NaCl molecule showing their electronic arrangement*

or dissociation depending upon the respective electronic configurations, and the gain, loss or sharing of electrons which takes place.

An ion may have a very stable electronic configuration but it will have a resultant electric charge and when two ions with opposite electric charges come into contact they are held together by the balancing of their electric charges in what is known as ionic binding. For example, the calcium ion $Ca-2e$ has an electronic arrangement of 2·8·8 with a positive electric charge, while the sulphate ion SO_4+2e has a negative electric charge and an electronic arrangement as shown in Fig. 29, in which the outer shell of each oxygen atom has eight electrons and the outer shell of the sulphur atom has eight electrons, each oxygen atom sharing two electrons with the sulphur atom. (The sulphate ion is an example of co-valent binding.) Both ions have a stable electronic arrangement but they have opposite electric charges which will hold and bind them together when they make contact. The compound of $CaSO_4$ is therefore an example of ionic binding between the calcium ion and the sulphate ion with the sulphate ion itself as an example of co-valent binding. Fig. 29 illustrates the electronic arrangements of various ions and compounds and shows how they tend to a stable electronic configuration.

A sodium atom has eleven electrons arranged 2·8·1. The outermost electron can be comparatively easily removed and the sodium ion which results has an electron arrangement of 2·8 which is extremely stable. In the same way, the electron arrangement of a chlorine atom is 2·8·7 so that it has a strong tendency to acquire another electron when it becomes a chlorine ion with 2·8·8 electrons, another very stable electron arrangement. Due to their extremely stable electronic arrangements with a small resultant electrical charge and their comparatively low weights, the sodium ion and chlorine ion are extremely soluble in water. The sodium and chlorine atoms in a sodium chlorine molecule share the two valency electrons as shown in Fig. 29, but the bonding is so slight that the ions separate very readily and are therefore extremely soluble. Due to their high solubility, chlorides remain in solution in the boiler water and are practically never precipitated at normal boiler water concentrations. The measure of the chlorides in the feed water and the boiler water can be used to determine whether sulphates are being precipitated in the boiler since the ratio of sulphates in the feed water to boiler water should be the same as the chloride ratio if no sulphates are being precipitated.

The various compounds of calcium, magnesium, sodium, and phosphate are composed of a co-valent ionic radical group bonded to an ionic metal or element by ionic binding, and it is this feature of ionic binding, which is used in boiler water conditioning to obtain the precipitation of the undesirable calcium and magnesium ions in the form of sludge and to leave in solution the ions which are very soluble and can be removed by blowing down.

When an acid, salt or a base is added to the feed or boiler water, the compound dissociates into ions to a greater or lesser degree, largely determined by the valency and the temperature. The more dilute the solution is, the greater is the percentage of the molecules ionized and, in general, the higher the valency, the less the degree of ionization.

When a compound is present in the boiler water or is added to the water, a large proportion of the compound splits up into ions which, in their movements in the liquid, reform and divide repeatedly, but for any given conditions there is a constant proportion divided into ions.

Some of the compounds met on this subject divide into ions as follows:—

$H_2O \rightarrow H^+ \text{ and } OH^-$
$NaOH \rightarrow Na^+ \text{ and } OH^-$
$CaCO_3 \rightarrow Ca^{++} \text{ and } CO_3^{--}$
$CaSO_4 \rightarrow Ca^{++} \text{ and } SO_4^{--}$
$Na_2CO_3 \rightarrow 2Na^+ \text{ and } CO_3^{--}$
$Na_3PO_4 \rightarrow 3Na^+ \text{ and } PO_4^{---}$

The radicals move about as a unit because they have a relatively stable electronic configuration and the loss or gain of electrons in the ionization is governed by the division into the most stable electronic arrangement for the respective ions.

The water therefore contains a mixture of ions such as calcium, magnesium, sodium, carbonate, sulphate, silicate, hydrogen, hydroxyl, phosphate, etc. These ions, in their movements, may make contact and combine, the attraction which they have for each other depending upon their valency and their atomic or molecular weight so that when there is a mixture of ions present in the boiler water any combinations which occur follow definite preferred lines and this fact is used in boiler conditioning. The preferential tendency for some ions to combine is used in boiler conditioning where, for example, sodium is used as a carrier for carbonate and phosphate conditioning because the sodium ion is much more soluble in water than the calcium and magnesium ions. Essentially the problem is the precipitation of the calcium and magnesium present in the water so that the compounds formed give a soft sludge and not a hard scale.

Carbonate and phosphate conditioning with sodium as the carrier, were proposed by Dr. R. E. Hall (12) in America many years ago, and he has recently suggested and tested the use of potassium (13), as an alternative to sodium, as a carrier for the phosphate ion owing to undesirable concentrations of sodium hydroxide causing caustic corrosion in hideouts which may exist due to defective boiler circulation or design. Both these elements, i.e., sodium and potassium, have a valency of one, have a very stable electronic structure in their ionic form and are very soluble in water.

If the boiler water contains ions of calcium, carbonate, sulphate, sodium and phosphate, the calcium and phosphate ions combine preferentially into tricalcium phosphate, while the sodium ions combine with the carbonate and sulphate ions to give sodium carbonate and sodium sulphate if sufficient concentration is allowed to occur, observing that the sodium ion is extremely soluble because its electronic configuration of 2·8 is extremely stable, and its weight is comparatively small. In the same way the chlorine ion is very stable with an electronic structure of 2·8·8 and this is the reason why sodium chloride is extremely soluble in water because the respective ions are very stable and have little tendency to bind with other ions.

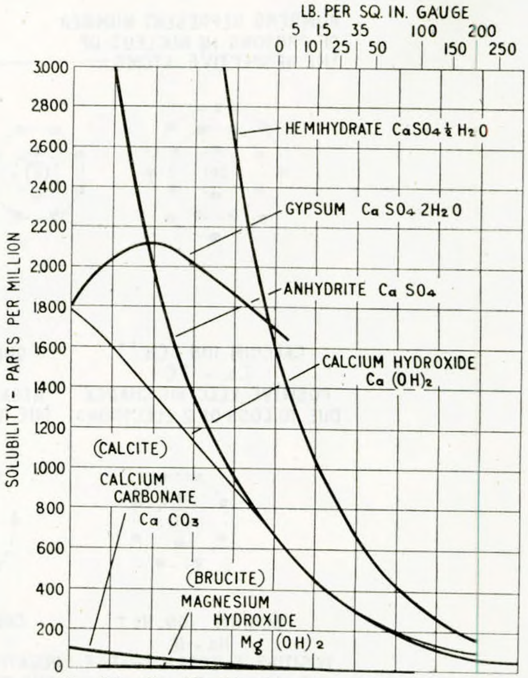


FIG. 30—*Solubility curves of scale-forming substances*

The conditioning salts which are commonly used at present, such as sodium carbonate, sodium hydroxide and trisodium phosphate, are therefore added in the desired proportions so as to effect the required interchanges with the sodium and calcium ions to give sodium carbonate and sodium sulphate, which are very soluble, and tricalcium phosphate which is insoluble and precipitates.

SOLUBILITY DATA OF SCALE FORMING SUBSTANCES AND SODIUM COMPOUNDS

Since calcium sulphate is the main hardness forming substance its solubility is of considerable interest. It can exist in three forms of solids:—gypsum, anhydrite and hemihydrate. Fig. 30 (11) shows its solubility with temperature, from which it will be seen that its solubility decreases rapidly as the temperature is increased. Fig. 30 (11) also shows the solubilities of calcium hydroxide and magnesium hydroxide.

The relative values of the data from which the curves are plotted at the higher temperatures can be more clearly seen from Table 21.

TABLE 21.—SOLUBILITY—PARTS PER MILLION (11)

Temperature, deg. C.	Temperature, deg. F.	Hemihydrate, p.p.m. $CaSO_4$	Anhydrite, p.p.m. $CaSO_4$	Calcite, p.p.m. $CaCO_3$	Calcium hydroxide, p.p.m. $Ca(OH)_2$	Magnesium hydroxide, p.p.m. $Mg(OH)_2$
100	212	1,645	650	14·8	658	4·5
150	302	530	222	7·3	225	2·2
200	392	165	76	4·8	99	0·8

The low solubility of $CaCO_3$ and $Mg(OH)_2$ is clearly shown by these figures.

The solubility of trisodium phosphate in water (16) is shown by Fig. 31. When the sodium ions are exchanged for calcium ions, the tricalcium phosphate which is formed is practically insoluble, its solubility being of the order of less than 5 p.p.m. at boiler temperatures.

It is probable that the precipitation is due to a strong tendency for combination between the molecules of tricalcium phosphate resulting in large aggregations of the molecules which

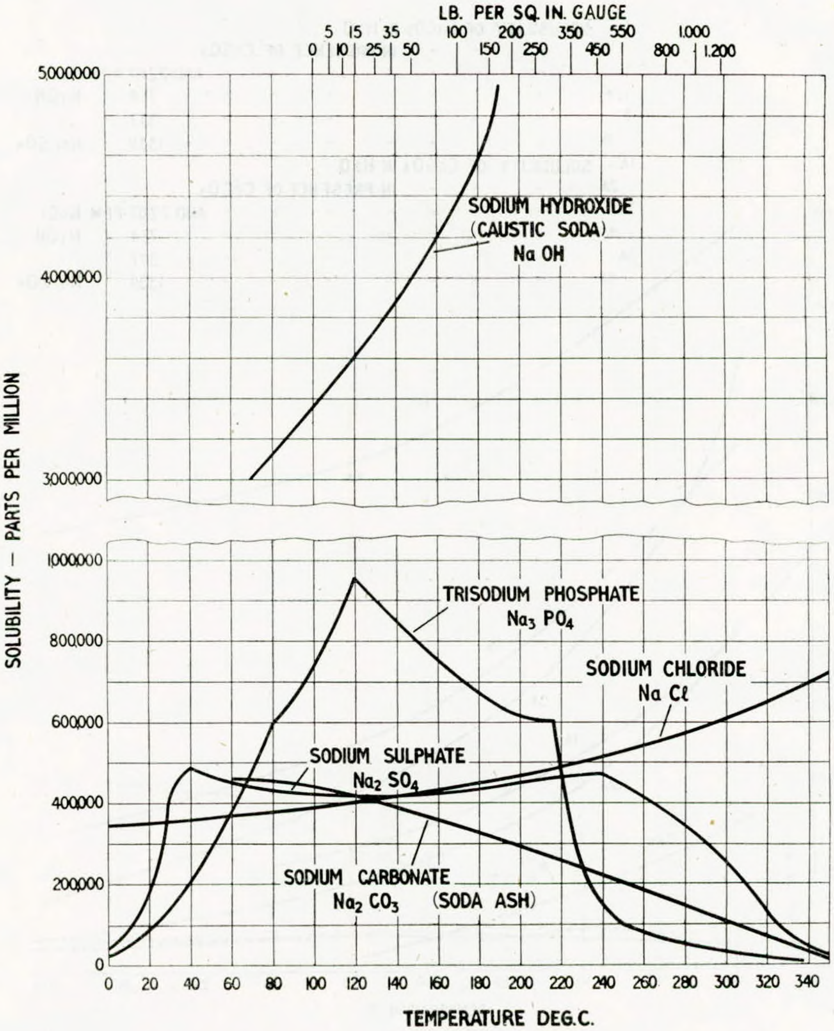


FIG. 31—*Solubility curves of sodium compounds in water*

resist any re-division and settle by reason of their weight. Such aggregations are moreover relatively self sufficient and do not bind themselves to the metal surfaces.

The high solubility of sodium sulphate (14) is shown in Fig. 31 which also shows the solubilities of sodium hydroxide, sodium carbonate, and sodium chloride. It will be seen that the sodium compounds (15) are soluble in water up to concentrations not likely to be reached in boilers in normal operation except in hide outs. Even at a temperature of 636 deg. F. (335 deg. C.) corresponding to a boiler pressure of 2,000lb. per sq. in., the solubility of trisodium phosphate is 8,000 p.p.m.

The curves given in Figs. 30 and 31 are in every case for a single compound in water. Boiler feed water, however, contains a variety of compounds, and the solubility data for any given compound is appreciably affected by the presence of other compounds.

Referring to Fig. 32, curve 1 shows the solubility of $CaCO_3$ in water while curve 1a shows the solubility of $CaSO_4$ in water, these curves being similar to those given in Fig. 30, for calcite and anhydrite respectively. If two salts such as $CaCO_3$ and $CaSO_4$ are together in solution in water their solubilities are interdependent and the respective solubilities of $CaCO_3$ and $CaSO_4$ in the presence of each other in water are shown by curves 2 and 2a in Fig. 32 (17). If, however, the two component system changes to a multi-component system by the addition of another salt or salts, the respective solubilities are still further modified. Solubility curves for a three-component system of $CaCO_3$ and $CaSO_4$ in water with various concentrations of NaCl, NaOH and Na_2SO_4 are shown by curves 3 to 6 and 3a to 6a in Fig. 32 (17). The addition of NaCl has little effect on the solubility of $CaCO_3$ but substantially increases the solubility of $CaSO_4$ as shown by curve 3a. NaOH, when added, as shown by curves 4 and 5, has little effect so far as $CaCO_3$ is concerned but, as shown by curves 4a and 5a considerably increases the solubility of $CaSO_4$. When Na_2SO_4 is added, the solubility of $CaCO_3$ is increased as shown by curve 6 and curve 6a shows that the solubility of $CaSO_4$ is reduced, probably because of the common SO_4 ion which is present. The curves, however, show that the general trend of the solubilities of the major calcium compounds in boiler water is similar to the simple systems and they illustrate the complex nature of the problem of controlling the concentrations of any given compounds in the boiler water.

One of the earliest methods of boiler water conditioning was based on the control of the relative amounts of carbonate to sulphate in the water with the object of ensuring the precipitation of calcium in a calcium carbonate sludge, while retaining sulphate in the solution in balance with sodium. If the two salts $CaCO_3$ and $CaSO_4$ are in solution together in water, their solubilities are interdependent, and if the CO_3/SO_4 ratio exceeds the ratio of their solubility product, calcium carbonate is precipitated. Sodium carbonate was therefore added to the water to control the CO_3/SO_4 ratio at the point where calcium carbonate would be precipitated in a soft sludge. Later work, however, has shown that sodium chloride, caustic soda and sodium sulphate, all reduce the equilibrium value of

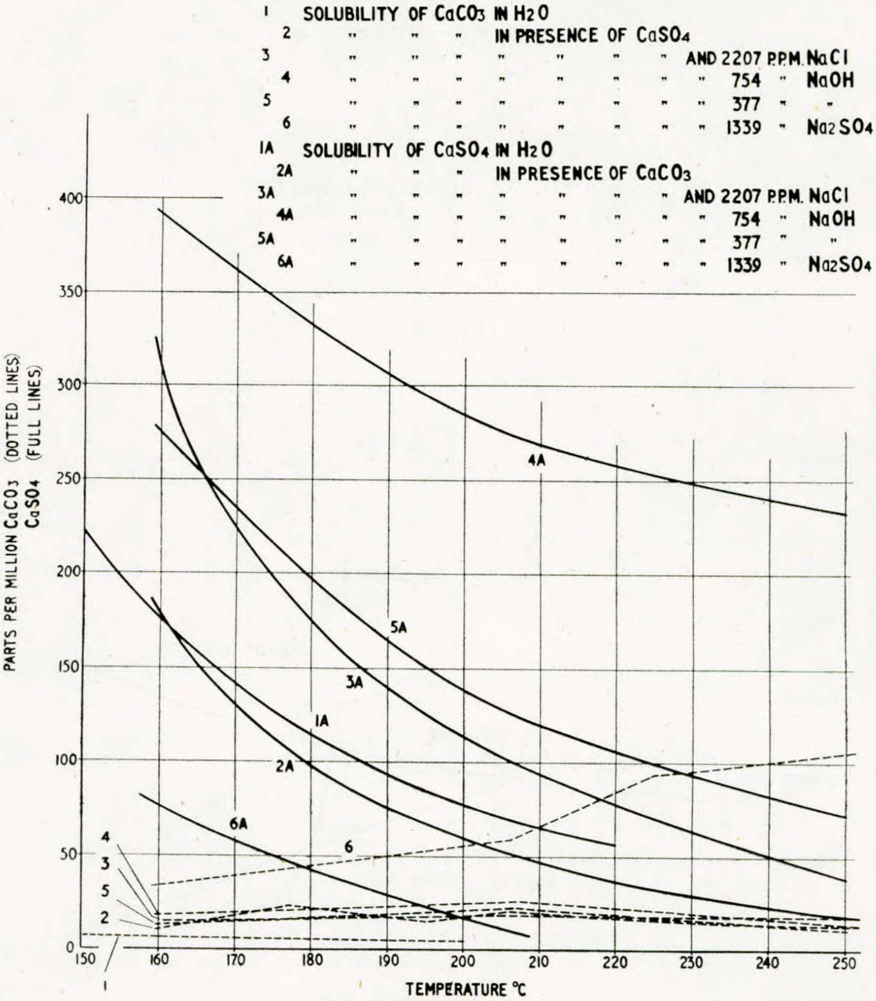


FIG. 32—*Diagram showing change in solubility of $CaCO_3$ and $CaSO_4$ in various solutions*

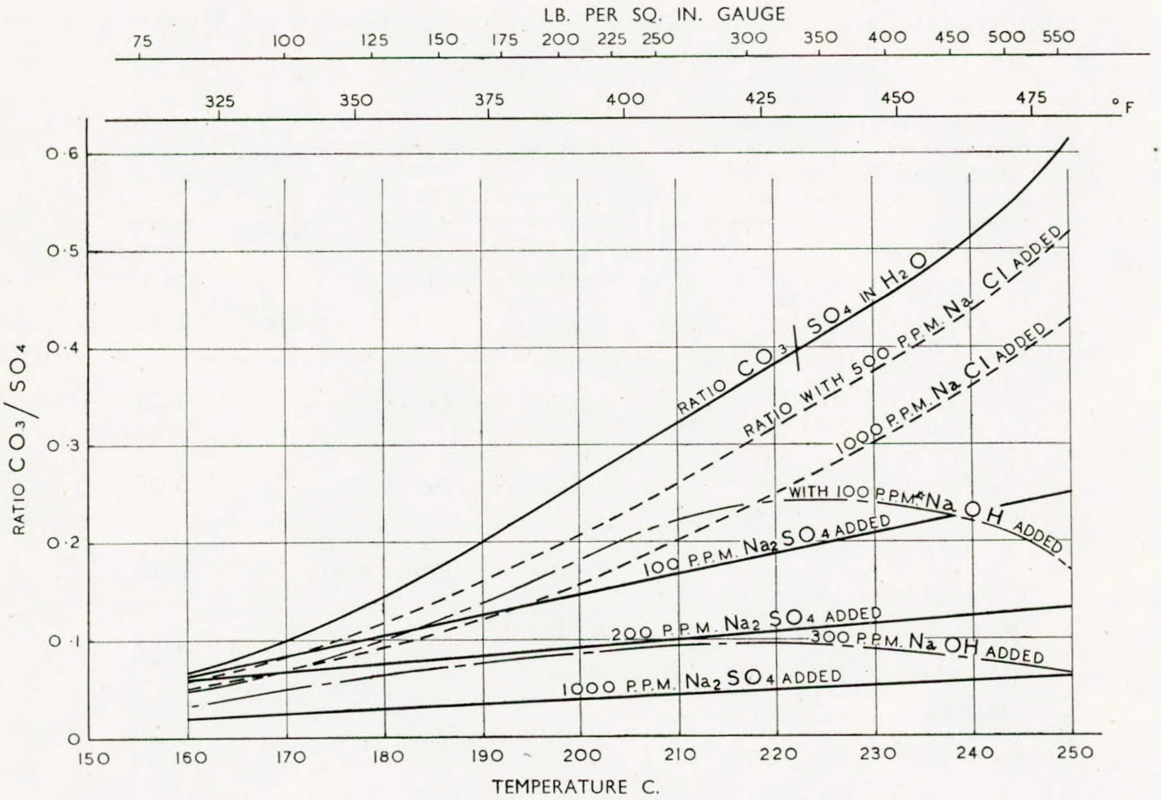


FIG. 33—*Ratio CO_3/SO_4 as varied by temperature and addition of different concentrations of sodium salts*

the CO_3/SO_4 ratio to an extent depending on the amount of salt added and the temperature, sodium sulphate having the most pronounced effect. The CO_3/SO_4 ratios corresponding to the solubility curves in Fig. 32 are given in Fig. 33. The curves show how the CO_3/SO_4 ratio varies with temperature and different concentrations of additional sodium salts, and illustrate the difficulty of controlling any two given compounds such as the carbonate/sulphate ratio to ensure that the precipitation of the desired compound is obtained.

Modern boiler water conditioning is therefore based on the control of alkalinity, the maintenance of zero hardness in the boiler water, and a reserve of phosphate in the boiler. Reagents may be supplied to promote the removal of scale forming matter as a flocculent sludge and oxygen scavengers may be supplied to remove residual dissolved oxygen.

LIME AND SODA CONDITIONING

Calcium hydroxide (slaked lime) is used to remove alkaline hardness and any magnesium compounds. Sodium carbonate (soda ash) is used to remove the other calcium compounds and the calcium compounds which are formed by the reactions between the calcium hydroxide and the magnesium compounds. The combination of these treatments is known as the lime and soda treatment and is commonly used to deal with natural waters before the water is supplied to the boilers. The reactions involved by the lime and soda treatment are shown in Table 22 which is largely self explanatory. The salts present are largely ionized in the water and the corrective salts added dissociate in the water into their respective ions. The modified mixture of ions present, leads to desired combinations between certain ions which result in compounds being formed which precipitate the calcium and magnesium in soft sludge compounds instead of hard scaling compounds, the remaining ions being so soluble that they remain in solution and are not troublesome from the scale forming aspect.

It will be seen from Table 22 that lime and soda conditioning results in the precipitation of calcium carbonate and magnesium hydroxide which are highly insoluble and form a light friable precipitate. Conditioning by the lime and soda method is usually applied to the feed water before it is delivered to the boiler, in which case the precipitate is light and friable. If the reactions occur in the boiler, the precipitate will form a thin friable porous scale which is easily removed.

The complete reactions with lime and soda conditioning give zero hardness in the water, i.e., the calcium and magnesium have been removed. If preliminary water treatment is not used, the alkaline hardnesses will be precipitated by the heating in the economizer and the boiler as a soft sludge or scale but the non-alkaline hardnesses will require to be treated by the addition of sodium carbonate to prevent scale.

The theoretical ratios between the corrective salts and the salts to be removed are shown in the Table and are obtained by inserting the respective molecular and atomic weights in the chemical formulæ as shown in the example in the table.

As previously shown, the calcium and magnesium hardnesses can be expressed in terms of a single compound such as $CaCO_3$ and this permits of a simple evaluation of the theoretical weights of lime and soda ash necessary to remove the calcium and magnesium.

Reference to Table 22 shows that $Ca(OH)_2$ is required to deal with the calcium alkaline hardness and the magnesium alkaline and non-alkaline hardnesses. The reactions with the magnesium non-alkaline hardnesses produce an equivalent amount of calcium non-alkaline hardness so that it can be taken that Na_2CO_3 is required to deal with the sum of the initial calcium and magnesium non-alkaline hardnesses.

The quantities of $Ca(OH)_2$ and Na_2CO_3 required to deal with a given water can therefore be calculated quite simply as follows:—

(a) Estimate the $Ca(OH)_2$ equivalent to the sum of the calcium alkaline hardnesses and the magnesium alkaline and non-alkaline hardnesses present in the water.

(b) Estimate the Na_2CO_3 equivalent to the sum of the

TABLE 22—LIME AND SODA TREATMENT FOR REMOVAL OF CALCIUM AND MAGNESIUM FROM FEED WATER

	A SCALE PRODUCING COMPOUND AND THE IONIZED FORM IN WHICH IT EXISTS IN SOLUTION IN THE MAKE UP FEED WATER	B SALT TO PROVIDE THE NECESSARY EXCHANGE OF IONS TO AVOID HARD SCALE FORMATION		RESULTANT SALT PRECIPITATING AS A THIN SOFT POROUS SCALE OR SLUDGE	SALT REMAINING IN SOLUTION IN THE BOILER WATER	WATER FORMED IN COMPLETING THE EXCHANGE	NUMBER OF LBS OF SALT B REQUIRED FOR EACH LB. OF SCALE PRODUCING COMPOUND A
	$Ca(HCO_3)_2$ IONIZED INTO $Ca^{++} + 2(HCO_3)^-$	+ $Ca(OH)_2$ IONIZED INTO $Ca^{++} + 2(OH)^-$	FORMS	$\underline{2CaCO_3}$		$+ 2H_2O$	0·45
	$CaSO_4$ IONIZED INTO $Ca^{++} + (SO_4)^{--}$	+ Na_2CO_3 IONIZED INTO $2Na^+ + (CO_3)^{--}$		$\underline{CaCO_3}$	$+ 2Na^+ + (SO_4)^{--}$		0·78
	$CaCl_2$ IONIZED INTO $Ca^{++} + 2Cl^-$	+ Na_2CO_3 IONIZED INTO $2Na^+ + (CO_3)^{--}$		$\underline{CaCO_3}$	$+ 2Na^+ + 2Cl^-$		0·96
	$Ca(NO_3)_2$ IONIZED INTO $Ca^{++} + 2(NO_3)^-$	+ Na_2CO_3 IONIZED INTO $2Na^+ + (CO_3)^{--}$		$\underline{CaCO_3}$	$+ 2Na^+ + 2(NO_3)^-$		0·65
	$Mg(HCO_3)_2$ IONIZED INTO $Mg^{++} + 2(HCO_3)^-$	+ $2Ca(OH)_2$ IONIZED INTO $2Ca^{++} + 4(OH)^-$		$\underline{Mg(OH)_2}$ + $\underline{2CaCO_3}$		$+ 2H_2O$	1·00
DOUBLE REACTION	$MgSO_4$ IONIZED INTO $Mg^{++} + (SO_4)^{--}$	+ $Ca(OH)_2$ IONIZED INTO $Ca^{++} + 2(OH)^-$		$\underline{Mg(OH)_2}$	$+ \boxed{Ca^{++} + (SO_4)^{--}}$		1 LB OF $MgSO_4$ REQUIRES 0·62 LB $Ca(OH)_2$
	$\boxed{Ca^{++} + (SO_4)^{--}}$	+ Na_2CO_3 IONIZED INTO $2Na^+ + (CO_3)^{--}$		$\underline{CaCO_3}$	$+ 2Na^+ + (SO_4)^{--}$		PLUS 0·86 LB Na_2CO_3
DOUBLE REACTION	$Mg(NO_3)_2$ IONIZED INTO $Mg^{++} + 2(NO_3)^-$	+ $Ca(OH)_2$ IONIZED INTO $Ca^{++} + 2(OH)^-$		$\underline{Mg(OH)_2}$	$+ \boxed{Ca^{++} + 2(NO_3)^-}$		1 LB OF $Mg(NO_3)_2$ REQUIRES 0·5 LB $Ca(OH)_2$
	$\boxed{Ca^{++} + 2(NO_3)^-}$	+ Na_2CO_3 IONIZED INTO $2Na^+ + (CO_3)^{--}$		$\underline{CaCO_3}$	$+ 2Na^+ + 2(NO_3)^-$		PLUS 0·72 LB Na_2CO_3
DOUBLE REACTION	$MgCl_2$ IONIZED INTO $Mg^{++} + 2Cl^-$	+ $Ca(OH)_2$ IONIZED INTO $Ca^{++} + 2(OH)^-$		$\underline{Mg(OH)_2}$	$+ \boxed{Ca^{++} + 2Cl^-}$		1 LB OF $MgCl_2$ REQUIRES 0·78 LB $Ca(OH)_2$
	$\boxed{Ca^{++} + 2Cl^-}$	+ Na_2CO_3 IONIZED INTO $2Na^+ + (CO_3)^-$		$\underline{CaCO_3}$	$+ 2Na^+ + 2Cl^-$		PLUS 1·11 LB Na_2CO_3

TYPICAL EXCHANGE GIVEN ON BASIS OF MOLECULAR AND ATOMIC WEIGHTS

$$\begin{array}{ccccccc} \overbrace{24\cdot3+2\times35\cdot5}^{MgCl_2} & + & \overbrace{40+2\times17}^{Ca(OH)_2} & - & \overbrace{24\cdot3+2\times17}^{Mg(OH)_2} & + & \overbrace{40+2\times35\cdot5}^{CaCl_2} \\ 95\cdot3 & + & 74 & & 58\cdot3 & + & 111 \\ & 169\cdot3 & & & & 169\cdot3 & \end{array}$$

$$Ca(OH)_2 \text{ REQUIRED PER LB } MgCl_2 - \frac{74}{95\cdot3} - 0\cdot78 \text{ LB}$$

$$\begin{array}{ccccccc} \overbrace{40+2\times35\cdot5}^{CaCl_2} & + & \overbrace{2\times23+60}^{Na_2CO_3} & - & \overbrace{40+60}^{CaCO_3} & + & \overbrace{2\times(23+35\cdot5)}^{2NaCl} \\ 111 & + & 106 & & 100 & + & 117 \\ & 217 & & & & 217 & \end{array}$$

$$Na_2CO_3 \text{ REQUIRED PER LB OF } CaCl_2 - \frac{106}{111} - 0\cdot96 \text{ LB}$$

$$Na_2CO_3 \text{ REQUIRED PER LB OF } MgCl_2 - \frac{106}{95\cdot3} - 1\cdot11 \text{ LB}$$

Lime—$Ca(OH)_2$ Soda ash—Na_2CO_3

calcium and magnesium non-alkaline hardnesses present in the water. (This allows for the reactions between $Ca(OH)_2$ and all the initial magnesium hardnesses.)

Taking the water analysis given in Table 17 the calcium alkaline hardness plus the total magnesium hardness equals 197 + 25 = 222 p.p.m. as $CaCO_3$. The equivalent weight of $Ca(OH)_2$ to match is $\frac{37}{50} \times 222 = 0\cdot74 \times 222 = 164$ p.p.m. $Ca(OH)_2$ where 37 and 50 are the equivalent weights of $Ca(OH)_2$ and $CaCO_3$ respectively.

The total calcium and magnesium non-alkaline hardnesses are 67 + 25 = 92 p.p.m. as $CaCO_3$. The equivalent weight of Na_2CO_3 to match is $\frac{53}{50} \times 92 = 1\cdot06 \times 92 = 98$ p.p.m. Na_2CO_3.

From the foregoing it will be seen that

(a) 0·74lb. of $Ca(OH)_2$ (0·82 for commercial $Ca(OH)_2$) is required to deal with each lb. of calcium alkaline hardness and magnesium alkaline and non-alkaline hardnesses present in the water.

(b) 1·06lb. of Na_2CO_3 are required to deal with each lb. of the calcium and magnesium non-alkaline hardnesses present in the water.

The above rules enable the quantities of $Ca(OH)_2$ (lime) and Na_2CO_3 (soda ash) required to deal with a given water to be calculated in a simple manner. The ionic analysis given for the same water provides the same information for the above calculations to determine the required quantities of $Ca(OH)_2$ and Na_2CO_3. The same weights can be obtained by using the atomic or molecular weights or by a detailed calculation for each compound as illustrated in Table 22.

In practice, greater quantities are required to obtain a complete chemical reaction of the salt to be removed and allowance must be made for the commercial composition of the corrective additions used. In some cases a comparatively large allowance must be made for the water of crystallization in commercial reagents.

Commercial $Ca(OH)_2$ has an equivalent weight of 41 so that the figure of 0·74 becomes 0·82. Na_2CO_3 in its commercial form can be taken as having an equivalent weight of 53.

Modern high-pressure boilers are usually supplied with distilled make-up water from which the hardnesses have been almost entirely removed in evaporators. The impurities to be dealt with are those carried over with the evaporator vapour and those due to condenser leakage. These are normally so low that phosphate conditioning only is necessary, but, if the make-up is not evaporated and contains appreciable impurities, it may be economical to use lime and soda conditioning for the make-up followed by phosphate conditioning in the boiler.

Sodium carbonate (soda ash) is not suitable for final conditioning at pressures above 250lb. per sq. in., it being difficult to maintain the CO_3^{--} formation, because it decomposes at high temperatures, and this led to the use of phosphates for final conditioning because the phosphate radical PO_4^{---} is entirely stable at all the higher operating pressures and is very satisfactory in producing a sludge precipitation instead of a scale. The decomposition of CO_3^{--}, with the soda ash treatment, into caustic soda and carbon dioxide, also tends to undue alkalinity in the boiler which is avoided by a phosphate treatment. Any treatment used must be adjusted as necessary to give the desired alkalinity or *p*H value in the boiler and soda ash may be used to control the alkalinity of the boiler water since it decomposes at high temperatures and produces sodium hydroxide.

CAUSTIC SODA CONDITIONING

Sodium hydroxide (caustic soda) is sometimes used instead of the lime and soda treatment and the principal reactions involved are shown in Table 23.

Caustic soda removes the calcium alkaline hardnesses and the magnesium alkaline and non-alkaline hardnesses by precipitating calcium carbonate and magnesium hydroxide as shown in Table 23. Caustic soda, however, does not deal with calcium non-alkaline hardnesses which require sodium carbonate to effect the reactions as shown in Table 22. If magnesium alkaline hardnesses are present the reactions with caustic soda produce sodium carbonate as shown in Table 23 and the sodium carbonate thus produced will react with the calcium non-alkaline hardnesses. If insufficient sodium carbonate is produced in these reactions, sodium carbonate in addition to caustic soda requires to be added. Due to this, it is not possible to provide a simple formula for the evaluation of the required

TABLE 23—CAUSTIC SODA TREATMENT FOR REMOVAL OF CALCIUM AND MAGNESIUM FROM FEED WATER

A SCALE PRODUCING COMPOUND AND THE IONIZED FORM IN WHICH IT EXISTS IN SOLUTION IN THE MAKE UP FEED WATER	B SALT TO PROVIDE THE NECESSARY EXCHANGE OF IONS TO AVOID HARD SCALE FORMATION		RESULTANT SALT PRECIPITATING AS A THIN SOFT POROUS SCALE OR SLUDGE	SALT REMAINING IN SOLUTION IN THE BOILER WATER	WATER FORMED IN COMPLETING THE EXCHANGE	NUMBER OF LB. OF SALT B REQUIRED FOR EACH LB OF SCALE PRODUCING COMPOUND A
$Ca(HCO_3)_2$ IONIZED INTO $Ca^{++} + 2(HCO_3)^{-}$ +	$2NaOH$ IONIZED INTO $2Na^{+} + 2(OH)^{-}$	FORMS	$\underline{CaCO_3}$	$+ 2Na^{+} + (CO_3)^{--} +$	$2H_2O$	0·49
$Mg(HCO_3)_2$ IONIZED INTO $Mg^{++} + 2(HCO_3)^{-}$ +	$4NaOH$ IONIZED INTO $4Na^{+} + 4(OH)^{-}$		$\underline{Mg(OH)_2}$	$+ 4Na^{+} + 2(CO_3)^{--} +$	$2H_2O$	1·09
$MgSO_4$ IONIZED INTO $Mg^{++} + (SO_4)^{--}$ +	$2NaOH$ IONIZED INTO $2Na^{+} + 2(OH)^{-}$		$\underline{Mg(OH)_2}$	$+ 2Na^{+} + (SO_4)^{--}$		0·66
$MgCl_2$ IONIZED INTO $Mg^{++} + 2Cl^{-}$ +	$2NaOH$ IONIZED INTO $2Na^{+} + 2(OH)^{-}$		$\underline{Mg(OH)_2}$	$+ 2Na^{+} + 2Cl^{-}$		0·84
Mg_2SiO_4 IONIZED INTO $2Mg^{++} + SiO_4^{----}$ +	$4NaOH$ IONIZED INTO $4Na^{+} + 4(OH)^{-}$		$2\,\underline{Mg(OH)_2}$	$+ 4Na^{+} + SiO_4^{----}$		1·14
$Mg(NO_3)_2$ IONIZED INTO $Mg^{++} + 2(NO_3)^{-}$ +	$2NaOH$ IONIZED INTO $2Na^{+} + 2(OH)^{-}$		$\underline{Mg(OH)_2}$	$+ 2Na^{+} + 2(NO_3)^{-}$		0·54
EXCHANGE GIVEN ON BASIS OF MOLECULAR AND ATOMIC WEIGHTS						
24·3 + 2 × 62 148·3	+ 2×23 + 2×17 + 80 228·3		24·3 + 2 ×17 58·3	+ 2× 23 + 2 × 62 + 170 228·3		
RATIO B/A − $80/148·3$ − 0·54						

Caustic soda—NaOH

caustic soda and the amount requires to be calculated in detail for each particular water analysis. The precipitates formed by the use of caustic soda are calcium carbonate and magnesium hydroxide which do not form a hard scale. In modern high-pressure boilers the principal use of caustic soda is to control the alkalinity or *p*H value of the boiler water.

Gloves and eyeshields should be worn when dealing with concentrated solutions of caustic soda, because it can cause serious injury to the skin and eyes if it touches them.

PHOSPHATE CONDITIONING

Various aspects of phosphate conditioning have been referred to in the section headed "Sodium phosphate compounds", which gives particulars of the various phosphates normally used. Phosphate conditioning is now the preferred treatment for high-pressure boilers, because, at boiler pressures above 250lb. per sq. in., the carbonate radical in sodium carbonate breaks up into sodium hydroxide and carbon dioxide, giving rise to undue alkalinity, whereas the PO_4^{---} ion is completely stable and very soluble at all the higher boiler pressures in use.

Sodium metaphosphate, or hexametaphosphate, disodium phosphate and trisodium phosphate are the phosphates most commonly used.

Sodium hexametaphosphate, monosodium phosphate and disodium phosphate are converted into trisodium phosphate in the boiler water in the presence of sodium hydroxide as shown in Table 24. This is due to their dissociation into their respective ions followed by a combination of the hydrogen and

TABLE 24—CONVERSION OF DISODIUM AND MONOSODIUM PHOSPHATE TO TRISODIUM PHOSPHATE BY ADDITION OF CAUSTIC SODA

A	B			B/A
$Na_2\ HPO_4$ IONISED INTO $2Na^+ + (HPO_4)^{--}$ +	$Na\ OH$ IONISED INTO $Na^+ + (OH)^-$	FORMS	$3Na^+ + (PO_4)^{---} + H_2O$	0·28
$Na\ H_2\ PO_4$ IONISED INTO $Na^+ + (H_2\ PO_4)^-$ +	$2Na\ OH$ IONISED INTO $2Na^+ + 2\,(OH)^-$		$3Na^+ + (PO_4)^{---} + 2H_2O$	0·66
EXCHANGE GIVEN ON BASIS OF MOLECULAR AND ATOMIC WEIGHTS				
23 + 97 120 200	+ 2×23 + 2×17 80		3 × 23 + 95 + 2 × 18 164 36 200	
RATIO $B/A = 80/120 = 0{\cdot}66$				

Disodium phosphate—Na_2HPO_4
Monosodium phosphate—$NaH_2\ PO_4$

TABLE 25—PHOSPHATE TREATMENT FOR REMOVAL OF CALCIUM AND MAGNESIUM AND TO AVOID CAUSTIC CONCENTRATION IN BOILER FEED WATER

A SCALE PRODUCING COMPOUND AND THE IONISED FORM IN WHICH IT EXISTS IN SOLUTION IN THE MAKE UP FEED WATER	B SALT TO PROVIDE THE NECESSARY EXCHANGE OF IONS TO AVOID HARD SCALE FORMATION		RESULTANT SALT PRECIPITATING AS A THIN SOFT POROUS SCALE OR SLUDGE	SALT REMAINING IN SOLUTION IN THE BOILER WATER	WATER FORMED IN COMPLETING THE EXCHANGE	NUMBER OF LBS OF SALT B REQUIRED FOR EACH LB OF SCALE PRODUCING COMPOUND A
$3Ca\ CO_3$ IONIZED INTO $3Ca^{++} + 3(CO_3)^{--}$ +	$2Na_3\ PO_4$ IONIZED INTO $6Na^+ + 2(PO_4)^{---}$	FORMS	$\underline{Ca_3\ (PO_4)_2} + 6Na^+ + 3(CO_3)^{--}$			1·09
$3Ca\ SO_4$ IONIZED INTO $3Ca^{++} + 3(SO_4)^{--}$ +	$2Na_3\ PO_4$ IONIZED INTO $6Na^+ + 2(PO_4)^{---}$		$\underline{Ca_3\ (PO_4)_2} + 6Na^+ + 3(SO_4)^{--}$			0·80
$3Ca\ Cl_2$ IONIZED INTO $3Ca^{++} + 6\,Cl^-$ +	$2Na_3\ PO_4$ IONIZED INTO $6Na^+ + 2(PO_4)^{---}$		$\underline{Ca_3\ (PO_4)_2} + 6Na^+ + 6Cl^-$			0·99
$3Mg\ (HCO_3)_2$ IONIZED INTO $3Mg^{++} + 6(HCO_3)^-$ +	$2Na_3\ PO_4$ IONIZED INTO $6Na^+ + 2(PO_4)^{---}$		$\underline{Mg_3\ (PO_4)_2} + 6Na^+ + 6OH^- + 6\,CO_2$			0·75
$3Mg\ SO_4$ IONIZED INTO $3Mg^{++} + 3(SO_4)^{--}$ +	$2Na_3\ PO_4$ IONIZED INTO $6Na^+ + 2(PO_4)^{---}$		$\underline{Mg_3\ (PO_4)_2} + 6Na^+ + 3\,(SO_4)^{--}$			0·91

Trisodium phosphate—Na_3PO_4

hydroxyl ions to form water, thereby leaving in solution the sodium ions and phosphate ions in the ratio of three to one.

If trisodium phosphate is used, it dissociates directly into the ratio of three sodium ions to one phosphate ion.

When calcium ions are present in the water, in their movements, they make contact with the phosphate ions and combine, due to their electronic forces, etc., in the ratio of three calcium ions to two phosphate ions.

The compound thus formed, $Ca_3(PO_4)_2$, tricalcium phosphate, is practically insoluble in the molecular structure of water and is precipitated as a soft sludge. If the caustic alkalinity of the boiler water is low, say below 100 p.p.m., the calcium may be removed as dicalcium phosphate, $Ca_2H_2(PO_4)_2$, or monocalcium phosphate $Ca(H_2PO_4)_2$, both of which are more soluble than tricalcium phosphate.

Table 25 gives the reactions which occur with sodium phosphate compounds and calcium and magnesium compounds in water. It will be seen that the sodium ions are left in solution in balance electrically with the carbonate, sulphate and chloride ions which formed part of the calcium salts and the calcium is precipitated with the phosphate.

With the removal of the calcium, any magnesium ions which may be present will combine with hydroxyl ions and will precipitate as $Mg(OH)_2$, which will be enmeshed in the tricalcium phosphate sludge and rendered harmless. The removal of the calcium ions also helps to prevent silicates forming scale by leaving the silicates in solution.

Sodium hexametaphosphate is commonly used in feed water because the change to trisodium phosphate takes place gradually and so avoids precipitation in the feed system. The rate of change can be considerably increased by heating or the addition of acid or alkali.

Trisodium phosphate is commonly injected direct into the boiler to avoid precipitation in the feed system or economizers. Phosphate treatment, it is claimed, has the advantage that it will dissolve any calcium deposits which may have been deposited.

An excess of phosphates is usually supplied to the boiler to maintain a reserve in solution in the boiler to ensure complete reactions and to deal with any increase in the ingress of scale forming solids.

A suitable reserve is 25 to 70 p.p.m. in terms of PO_4^{---}.

The use of sodium as the carrier for phosphate conditioning leaves the sodium in solution in the boiler. Dr. R. E. Hall, in America, has suggested that this may lead to strong sodium hydroxide concentration in hide outs and has recently proposed and tested the use of potassium as a carrier instead of sodium (13). He points out that potassium is very similar to sodium in its properties and has better solubility characteristics while avoiding the dangers of strong caustic concentrations.

The various sodium phosphate compounds are similar in their action in preventing scale formation but differ in their effect on the boiler water alkalinity.

The choice for use, therefore, depends upon the control of alkalinity and their effect increases in the order of sodium hexametaphosphate, disodium phosphate and trisodium phosphate. Consideration also requires to be given to the desirability of the avoidance of precipitation in feed piping and economizers.

COLLOIDAL AND COAGULANT TREATMENT

In some circumstances, not clearly understood, calcium carbonate, magnesium hydroxide and phosphate sludges may adhere to boiler surfaces, and to prevent this possibility, coagulant or colloidal treatment may be used.

Substances vary as to their solubility. In a true solution, the solute will consist of units which are not larger than single molecules, and the molecules are dispersed equally throughout the liquid. The molecules of some substances, however, or molecules of two substances present, may combine to form larger particles which settle in the liquid, due to the action of gravity, and are then said to be insoluble. In some cases, the insoluble solid particles are so small that they settle very slowly or remain suspended in the liquid indefinitely. The suspended particles consist of clusters of molecules or atoms and are known as colloids. A colloidal system consists of at least two substances one of which acts as a central connecting link to the other substance or substances in a symmetrical structure and the structure as a whole has a resultant electric charge due to the electronic configuration and this charge prevents them forming into greater clusters and keeps the colloidal particles dispersed in the liquid by the repelling action of their similar charges.

A nucleus of an atom has dimensions of the order of 10^{-12} cm., while the zone of the electrons surrounding the nucleus is of the order of 10^{-8} cm. Any particles having dimensions smaller than 10^{-7} cm., i.e., 10^{-8} cm., will exist in water in true solution. The average size of a colloidal particle is approximately 10^{-6} cm. The size or length of small particles is usually expressed in Angstrom units, a unit which is represented by the symbol A. 1A is 10^{-7} mm. (10^{-8} cm.) i.e., one ten millionth of a millimeter.

Particles up to 10A will exist in true solution in water. Particles of 10 to 1,000A (10^{-7} to 10^{-5} cm.) in diameter, are considered to be of colloidal size and will remain in suspension, while particles greater than 1,000A (10^{-5} cm.) in diameter will settle due to gravity. As a comparison for size, particles of about 1,500A are the smallest objects that can be seen by means of an ordinary microscope, while the largest known inorganic molecules have diameters of about 10A. A molecule of water is about 4A diameter.

A very important physical factor of colloids is the relatively extremely large surface area these particles present due to the extreme extent of the sub-division with occurs. For example, a cube of colloidal substance 1 cm. across its faces will have its surface area multiplied one million times when it is dispersed colloidally in water.

Colloids can be used to remove scale forming ions with opposite electric charges because attraction and bonding occur due to the opposite electric charges.

The neutralization of the electric charges which tend to keep the colloidal particles apart allows the colloidal particles to coagulate together and with the scale forming solids so that the agglomeration increases in weight until precipitation or a floc occurs. Precipitation takes place in the form of a loose soft sludge which will flow with the current of the liquid and can be blown down.

Substances such as starch, tannin, gels, casein, etc., are colloids in water and because of the large surface area provided by the very large number of very small particles and their similar electric charges, they are intimately dispersed throughout the water and become intimately but loosely bonded to any precipitating salts to form a flocculent mass which forms a floating easily removed sludge.

Substances such as sodium aluminate, ferric and ferrous sulphate, etc., in water give rise to the formation of their hydroxides which act in a similar way to colloids and promote coagulation with scale forming precipitates.

The presence of a colloid within the matrix of any scale has the effect of reducing the mechanical strength of the scale and facilitating its removal. If oil is present, the colloid will combine with the oil and assist in making it harmless or removing it.

Ferric and ferrous sulphate are used as coagulants in the external treatment of water and, in conjunction with lime and caustic soda, produce a floc of ferric hydroxide which assists in the removal of suspended matter.

Sodium aluminate $Na_2Al_2O_4$ is also used with the lime soda treatment as a coagulant. Sodium aluminate is a compound of sodium oxide Na_2O and aluminium oxide Al_2O_3. In solution in water, it can exist as a molecular dispersion or in a colloidal dispersion, and it is probable that the latter is the important phase. The colloidal dispersion contains particles of aluminium hydroxide, the surface layer of which is ionized and the system can be expressed as follows:—

$$\begin{array}{lll} Na^{+} & \dots & (OH)_2Al^{-} \\ 2Na^{+} & \dots & (OH)Al_2^{--} \end{array} \left[Al(OH)_3,\ aq. \right] AlO_3^{---} \quad \dots \quad 3Na^{+}$$

where the formula inside the bracket represents the nature of the solid phase in the colloid particle and the symbols outside indicate an electrical surface double layer (18).

The colloidal aluminium hydroxide dispersed in the water is negatively charged while magnesium hydroxide which may be present colloidally in the water is positively charged. Coagulation into a floc therefore occurs between the aluminium hydroxide and the magnesium hydroxide due to their opposite charges. There is no similar action between aluminium hydroxide and calcium carbonate precipitates, but such precipitates may be enmeshed in the matrix of the colloidal flocculent structure. Particles of other scale forming solids are carried by the floc and prevented from forming hard scale.

The action of sodium aluminate is improved by the addition of tannin which helps to form a coagulated floc of the finely divided precipitates so that the resultant aggregate is free flowing and can be blown down from the boiler easily.

Sodium aluminate together with magnesium hydroxide is also useful in precipitating silica from solution and avoiding hard silicate scaling. To ensure the removal of the silica it has been found to be important that magnesium is present in the water and if sufficient magnesium hydroxide is not available magnesium sulphate is sometimes added to the sodium aluminate tannin solution to prevent scaling due to silica.

OXYGEN SCAVENGING

It is now the invariable practice to supply a modern high-pressure boiler with de-aerated water but there is always the possibility that unusual circumstances may result in undesirable oxygen ingress. To meet this, and to supplement the mechanical de-aeration provided, sodium sulphite Na_2SO_3 is sometimes used to mop up any residual oxygen. Sodium sulphite should not be used to deal with any large quantities of oxygen, say above 0·2 ml/litre, because the amount required would unduly increase the total dissolved solids in the boiler.

In the water, Na_2SO_3 ionizes into $2Na^+$ and SO_3^{--} and combines with the dissolved oxygen in the water as follows:—

$$4Na^+ + 2SO_3^{--} + O_2 \rightarrow 4Na^+ + 2SO_4^{--}$$

If sodium sulphite is used, a slight excess of 20 to 30 p.p.m. should be carried in the boiler water.

CONDITIONING FOR CAUSTIC CRACKING

The conditions causing caustic cracking are still far from clearly understood, but it would appear to be due to very strong concentrations of sodium hydroxide, NaOH, in the base of a crack, or fissure, as illustrated in Fig. 11. Experiments in the U.S.A. appear to show that caustic cracking will not occur if the concentration of NaOH is less than 100,000 p.p.m. This is, of course, a higher concentration than is likely to occur as an overall value in boiler water but it can occur in cracks and hide outs.

Research has indicated that attack is inhibited if the surface is coated with sodium sulphate or sodium carbonate and on this basis boiler water is conditioned against caustic cracking by maintaining sufficient sodium sulphate and/or sodium carbonate to ensure their deposition when the concentration of sodium hydroxide reaches 100,000 p.p.m.

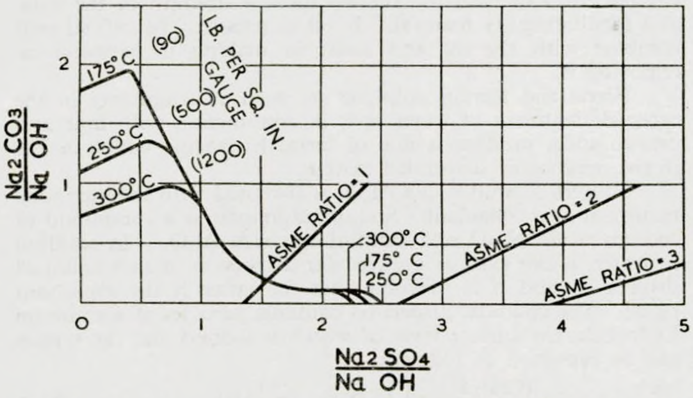


FIG. 34—*Curves of ratios $Na_2CO_3/NaOH$ and $Na_2SO_4/NaOH$ required in boiler water to prevent caustic cracking*

Sodium carbonate is liable to decomposition into sodium hydroxide while sodium sulphate is stable. Sodium sulphate is therefore the preferred inhibitor.

Straub in the U.S.A. established curves of ratios of $Na_2SO_4/NaOH$ and $Na_2CO_3/NaOH$ at different boiler temperatures which would ensure the deposition of Na_2SO_4 or Na_2CO_3 and these are given in Fig. 34 (19).

There is a lot of experience to show that if boiler water is conditioned so that the ratios are above the curves given, satisfactory protection will be afforded against caustic cracking. In 1936, the American Society of Mechanical Engineers laid down that the ratio of sodium sulphate Na_2SO_4 to the total alkalinity, in terms of sodium carbonate Na_2CO_3, should be not less than 1 to 1 up to boiler pressures of 150lb. per sq. in., 2 to 1 for boiler pressures of 150 to 250lb. per sq. in., and 3 to 1 for boiler pressures over 250lb. per sq. in., and these ratios are indicated in the diagram (19).

It has been noted that sodium chloride seems to assist in inhibiting caustic cracking. Sodium silicate appears to accelerate caustic cracking but the addition of a combination of iron and aluminium has been successful in counteracting the effects of sodium silicate.

Straub and Bradbury's investigations (20) into these aspects appear to show that for steam pressures up to 250lb. per sq. in. caustic cracking may be prevented by maintaining a sodium chloride ratio in the boiler water greater than 0·6 times the total alkalinity expressed as Na_2CO_3 along with a sodium sulphate content greater than 1·0 times the total alkalinity. For a steam pressure of 350lb. per sq. in. these sulphate or chloride ratios appear to be effective but may require to be somewhat larger. For steam pressures between 500 and 1,400lb. per sq. in. their results indicate that the presence of a soluble R_2O_3 content of greater than 0·6 times the SiO_2 content of the boiler water prevents caustic cracking. The R_2O_3 content may be a combination of iron and aluminium oxides and in the majority of the cases investigated aluminium as Al_2O_3 appeared to predominate (20).

Knowledge on this subject, however, is still far from complete and at present it is desirable to carry as high a sodium sulphate concentration as can be accepted within the total dissolved solids considered acceptable for a given boiler design. This will give the conditions least likely to allow caustic cracking to occur.

British Standard 1170-1947 recommends that the ratio of sodium sulphate Na_2SO_4 to the caustic alkalinity in terms of NaOH should have a minimum value of 2·5 at all times in all boilers with riveted or welded drums, which have not been stress relieved.

BOILER BLOW DOWN AND RECIRCULATION

The blow-down from a boiler is usually strongly alkaline and in land practice the blow down is usually effected continuously and is often passed through suitable equipment to use the heat in the blow-down for feed or other heating purposes, or it may be used in a water conditioning plant or a de-aerator, wherein the heat and the alkalinity may be utilized to increase the *p*H value of the feed water so as to prevent corrosion in the feed system and economizer.

In marine practice, it is almost the invariable practice in water tube boiler installations to provide make-up feed which has been doubly distilled to reduce the solid impurities to a minimum and so reduce the blow-down required as much as possible. Where continuous blow-down is practiced, the blow-down is sometimes led into the fresh water evaporator to conserve heat and water.

In land power stations, there is a growing practice of taking water from the boiler and discharging it through the economizer, back into the boiler. The quantity of water circulated in this way is about one to three per cent of the boiler output.

The alkalinity of such water is in the range of a *p*H value of 11 to 12 and this enables a lower *p*H value to be carried in the feed water before the mixing than would otherwise be the case to ensure that the economizers are protected against corrosion.

The total solids entering the boiler are thus reduced and the boiler can be operated with a lower total dissolved content or a smaller blow-down. Further, the continuous flow of such water provides continuous protection, particularly on light loads and where chemical conditioning is effected intermittently.

CONTROL OF BOILER WATER CONDITIONING AND ALKALINITY

It will be appreciated from the information already discussed that water treatment is a complex subject with a multiplicity of factors and that the water treatment adopted should be studied with a knowledge of the composition of the water used and any scale which may result. It is advisable, therefore, to consult the specialists in this field and follow their recommendations. For marine service, guidance has been provided by British Standard Specification 1170/1947 (3), which should be consulted for full details. It advises, broadly, that the boiler water should be conditioned as follows:—

(a) The alkalinity to phenolphthalein, in terms of $CaCO_3$, should be kept within the limits 125-250 p.p.m. with a recommended range of 175-225 p.p.m. for both water-tube and Scotch boilers. If the total dissolved solids exceed 2,000 p.p.m., the alkalinity should be increased 125 p.p.m. per 1,000 p.p.m. additional dissolved solids but should not exceed 1,000 p.p.m. to avoid corrosion of brass mountings.

(b) The caustic alkalinity, in terms of $CaCO_3$, should be kept within 100-200 p.p.m., with a recommended range of 125-175 p.p.m. for both water-tube and Scotch boilers. This corresponds to 80-160 p.p.m. and 100-140 p.p.m. in terms of sodium hydroxide NaOH. If the dissolved solids exceed 2,000 p.p.m. the caustic alkalinity should be increased by 100 p.p.m. (in terms of $CaCO_3$) or 80 p.p.m. (in terms of NaOH) per 1,000 p.p.m. additional dissolved solids. The caustic alkalinity should not exceed 850 p.p.m. (as $CaCO_3$) or 680 (as NaOH) to avoid corrosion of brass mountings.

(c) The chlorides in the boiler water should be kept below 120-480 p.p.m. in water-tube boilers, according to the design and rating of the boiler, and below 3,600 p.p.m. for Scotch boilers.

(d) The total dissolved solids should be kept below 500 to 2,000 p.p.m. for water-tube boilers, according to design and rating, as necessary to avoid priming and carry over, and below 15,000 p.p.m. in Scotch boilers.

(e) The hardness should be zero.

(f) A reserve of phosphate of the order of 25 to 70 p.p.m. should be carried in a water-tube or a Scotch boiler.

(g) A reserve of sulphite, as Na_2SO_3, of 20-30 p.p.m. should be carried in both types of boilers.

(h) The sulphate, measured as Na_2SO_4, should be 2·0 times the caustic alkalinity in terms of $CaCO_3$ or 2·5 times the caustic alkalinity in terms of NaOH, for both types of boilers.

(i) The dissolved oxygen content (of the feed water) should be between 0·02 and 0·05 ml/litre, the former figure being desirable for highly rated water-tube boilers and the latter for all other boilers.

In land power stations, a representative set of figures of American practice gives the total dissolved solids in the feed water as ranging from 0·2 to 14 p.p.m. with an average of 2·6 p.p.m. while the total dissolved solids in the boiler water ranged from 47 to 1,000 p.p.m. with an average of 360 p.p.m.

Chlorides pass through a boiler unchanged and the measure of the chlorides in the boiler water to the chlorides in the feed water gives a measure of the concentration of the solids in the water in the boiler. If no sulphate scale is forming, then the sulphate ratio will be the same as the chloride ratio but if it is less, sulphate scale is being formed.

The alkalinity is a measure of the total bicarbonates, carbonates and hydroxides in the water. The bicarbonates, carbonates and hydroxides may be in their ionic form or in suspension bonded with metallic ions. The alkalinity may be measured in terms of a single substance such as $CaCO_3$ or NaOH. 50 parts of $CaCO_3$ are equal to 40 parts of NaOH and the terms are interchangeable in that ratio when used as a measure of other substances. Alkalinity due to bicarbonates and carbonates is called carbonate alkalinity, while alkalinity due to hydroxides is known as caustic alkalinity. Hydroxides and bicarbonates cannot exist together because they combine to form carbonates.

Caustic alkalinity is a measure of the hydroxyl ions present in the water in balance with or bonded with metallic ions. Caustic alkalinity is therefore a measure of the OH present in the water above the number of hydroxyl ions corresponding to the neutral value of *p*H, which is 7 for the standard pressure and temperature, and is usually expressed in terms of $CaCO_3$.

To prevent corrosion it has been shown that it is desirable to maintain the *p*H value of the boiler water between 11 and 12 and, because of the concentration which occurs in the boiler, the feed water should be controlled between *p*H values of 8·5 and 9·5.

Corrective chemicals to prevent scale formation have their effect on the *p*H value of the boiler water, since any excess left in solution may alter the relative concentrations of hydrogen and hydroxyl ions. Substances are said to be alkaline in character if, when dissolved in water, they supply hydroxyl ions or, by the reactions which occur, they increase the number of hydroxyl ions present in the water. Conversely, substances which supply hydrogen ions when dissolved in water, or are responsible for reactions which result in an increase in the number of hydrogen ions, are said to be acidic. Acids added or formed in the water increase the hydrogen ion concentration while alkalis or salts with alkaline effects increase the hydroxyl ion concentration. For example, if sea water is allowed ingress to a boiler, the magnesium chloride, $MgCl_2$, combines with the water and hydrochloric acid HCl is produced together with a precipitation of magnesium hydroxide $Mg(OH)_2$ as follows:—

$MgCl_2 + 2H_2O$ forms $\underline{Mg(OH)_2} + 2HCl$, the hydrochloric acid being produced in the dissociated form of hydrogen ions H^+ and chlorine ions Cl^-, so that the hydrogen ion concentration is increased and therefore the possibilities of corrosion.

The measurement of the *p*H value of the boiler water is not a very convenient process because of the low hydrogen ion concentration at the *p*H value range usually worked. It is therefore much more convenient to measure the alkalinity of the boiler water and to work within a prescribed range.

The p.p.m. of hydrogen ions and hydroxyl ions corresponding to different *p*H values are given in Table 3. Pure water has a *p*H value of 7 with 0·0001 p.p.m. of hydrogen ions and 0·0017 p.p.m. of hydroxyl ions.

The addition of bicarbonates, carbonates and hydroxides will increase the number of hydroxyl ions. The bicarbonates will decompose with heating giving metallic ions, carbonate ions and carbon dioxide gas $\overline{CO_2}$. With correct conditioning, the calcium and magnesium ions will have been removed together with the balancing amount of carbonate ions and hydroxyl ions, and any carbonate and hydroxyl ions remaining will be in solution in balance with sodium ions. The hydroxyl ions increase the hydroxyl ion concentration while the carbonate ions react with hydrogen ions in the water to form carbon dioxide gas which also results in an increase in the hydroxyl ion concentration.

The substances used in boiler water conditioning which increase the *p*H value are sodium hydroxide NaOH, sodium carbonate Na_2CO_3, trisodium phosphate Na_3PO_4 in its anhydrous form, and commercial trisodium phosphate $Na_3PO_4 12H_2O$.

When sodium hydroxide NaOH is added to the water it dissociates into its ions, Na^+ and OH^- so that, theoretically, the OH^- concentration is directly increased and could be calculated from the respective relative weights of Na^+ and OH^-, 23 and 17.

When sodium carbonate Na_2CO_3 is added to the water, the dissociation gives two sodium ions $2Na^+$ for each carbonate ion $CO_3{}^{--}$. Some of the carbonate ions react with hydrogen ions H^+ in the water thus:—

$$CO_3{}^{--} + H^+ \text{ form } \overline{CO_2} + OH^-$$

The two surplus electrons in the radical $CO_3{}^{--}$ ion are used to supply the one short in the hydrogen ion and the other makes the combination of the hydrogen and oxygen atoms into an hydroxyl ion. An hydrogen ion is thus neutralized and an hydroxyl ion formed with the evolution of carbon dioxide as a gas so that the hydrogen ion concentration is reduced and the hydroxyl ion concentration is increased. The sodium ions remain in solution and the water, as a whole, remains electrically neutral.

When trisodium phosphate remains in solution in the boiler water it is in the ionic form of $3Na^+$ and $PO_4{}^{---}$.

The phosphate ion will combine with hydrogen ions, as they make contact, with reversible reactions as follows:—

$$(1)\quad PO_4{}^{---} + H^+ \rightleftarrows (HPO_4)^{--}$$

$$(2)\quad (HPO_4)^{--} + H^+ \rightleftarrows (H_2PO_4)^{-}$$

$$(3)\quad (H_2PO_4)^{-} + H^+ \rightleftarrows H_3PO_4$$

The phosphate ions, by the combining with hydrogen ions, cause an increase in the hydroxyl ion concentration.

In each case described above, the product of the numbers of hydrogen ions and hydroxyl ions in the water is maintained constant by the further dissociation of water molecules into hydrogen and hydroxyl ions until equilibrium is reached, the resultant effect being a reduction in the hydrogen ion concentration and an increase in the hydroxyl ion concentration.

The effect on the pH value by additions of varying amounts of the compounds mentioned is shown in Fig. 35 (21). The extent to which such compounds dissociate in water is variable depending upon temperature and other conditions but the diagram illustrates approximately the effect obtained. For example, if a water containing NaOH has a pH value of 11 the addition of 300 p.p.m. of NaOH will raise the pH value to 12. The p.p.m. of OH^- to give varying pH values is shown by the dotted line.

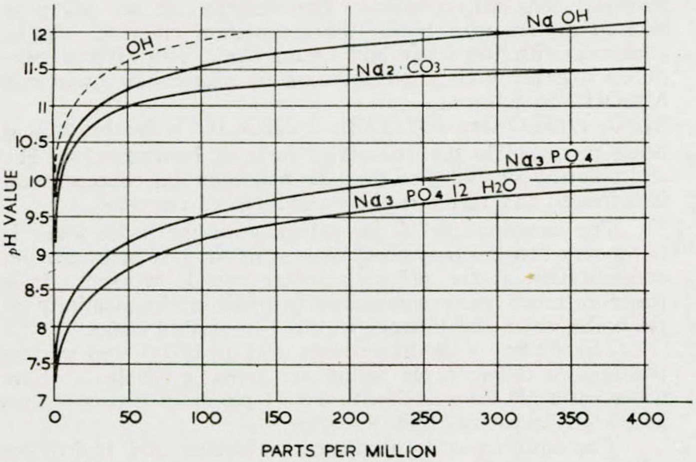


FIG. 35—*Effect on* p*H value by addition of varying amounts of chemicals*

The curves show clearly that the control necessary to maintain the boiler water within a given pH range such as 11 to 12 is not unduly sensitive. To obtain very high pH values requires very considerable additions of the suitable chemicals.

For practical convenience, boiler water control is usually carried out in terms of alkalinity but it should be appreciated that the object is to obtain the desired hydroxyl ion concentration necessary to prevent corrosion and maintain a self repairing action on the protective film over the steel surfaces.

The total dissolved solids must be kept below the maximum permissible for the design and rating of the boiler and the treatment should be that which will give the maximum pH value within the permissible total dissolved solids and the desired ratios of the other compounds appropriate to the boiler and the operating conditions. The presence of undesirable solids should therefore be kept as low as practicable, and the excesses of desirable compounds should be kept as close as possible to the minimum acceptable.

It will be seen from Fig. 35, that additions of sodium hydroxide have the most marked effect in increasing the pH value of the boiler water for a minimum increase in the total dissolved solids present after non scaling requirements have been met.

CONCLUSION

Boiler corrosion and feed and boiler water conditioning involve a multiplicity of complex factors, which are strange to the average engineer.

It is not intended that this paper should be a guide to the actual water treatment to be used and it should be emphasized that it is strongly advisable to consult specialists in this field and follow their advice. It is hoped, however, that it will help to a better understanding of the factors involved.

In dealing with chemical terms and equations there are different ways of expressing the same information and alternative ways have purposely been used in the paper by way of illustration.

ACKNOWLEDGMENTS

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Discussion

CAPTAIN(E) L. A. B. PEILE, R.N., said he felt sure he was speaking for many marine engineers when he said that he was usually wary of the reasoning behind chemistry, and in that frame of mind one tended to regard feed water treatment as a mystery. One was told what to do and one did it. The danger of that attitude was that circumstances regarding boiler operation might change and, still doing blindly what one had been told to do in different circumstances, one was, in fact, applying the wrong treatment. On the basis of the knowledge acquired from the author they should be able to recognize a change of circumstances and once again consult the specialist.

In the Navy one was faced with perhaps the most difficult conditions of boiler corrosion since many of the boilers spent much of their time idle. Corrosion troubles were readily generated under these conditions and became serious during subsequent operation.

Perhaps the most difficult thing to ensure was that the water in an idle boiler was in fact oxygen free. Any leakage on the suction side of the de-aerator extraction pump during the operation of filling the boiler, any oxygen present in the boiler being pumped up, or any leakage at the boiler mounting glands after the boiler had been completely filled would result in absorption of oxygen by the boiler water. He would like to emphasize Mr. Hillier's statement that water greedily absorbed oxygen. It was a very rapid process, a matter of seconds rather than hours. In this connexion, he should say that the Admiralty was not entirely satisfied that sodium sulphite was the complete oxygen scavenger. The affinity of oxygen for sodium sulphite might not be sufficiently marked in relation to the affinity for iron. They were not certain, so they had some experiments in hand in naval boilers in the idle condition.

He was glad that Mr. Hillier had pointed out that there was lack of knowledge about corrosion conditions in boilers operating at high pressures and temperatures. The trend was to suggest that with higher steam pressures the presence of oxygen might be even more serious as a factor in the corrosion problem, and that therefore it should be kept to a minimum. He could find no-one to give a reasoned statement as to what maximum oxygen content should be specified for feed water for high pressure boilers. He hoped the paper would stimulate research into boiler corrosion problems under advanced steam conditions.

Steam corrosion of iron was an alarming thought when considering boilers for operation at 1,050 deg. F. and above. The work that had been done in America by Rohrig, Van Duzer and Fellows (A.S.M.E Annual Meeting, New York, December 1943) suggested, however, that the rate of reduction of many steels suitable for superheater construction was acceptable at a temperature of 1,100 deg. F.

He anticipated that there might be some discussion among the metallurgists on the author's description of corrosion fatigue as an inter-crystalline failure. In Admiralty experience this phenomenon usually started as an inter-crystalline fracture, but continued in a transcrystalline manner to fail.

REAR-ADMIRAL(E) T. H. SIMPSON (Member) said, on the subject of pitting, that the naval boilers during the 1914-18 War and again in the first few years of the last war suffered to a very large extent from a form of pitting called scab pitting which was confined to the hottest part of the heating surface. There was no doubt that these scabs were formed initially as soft scabs and that the soft scabs insulated steel through the insulating layer, so that the steel underneath was overheated and magnetite was formed over the scab and quickly led to very severe pitting. On some occasions when the pits were cut out small salt crystals were found in the pits, and this led to the supposition that salt contamination was a predisposing cause, but in many cases this pitting occurred without any exceptional salt contamination.

In the years between the wars accounts of pitting of this description were rare, and it appeared that they were intimately associated with steaming at high forcing rates.

Until 1944 the only boiler water treatment which was in general use in the Navy was the addition of small quantities of lime to maintain the water slightly alkaline so that the *p*H value was 8·5, or something of that sort. This avoided general wastage, but it encouraged pitting. Pitting was very common, but the boilers were cleaned thoroughly at very regular intervals and the scabs were scraped off and on the whole large deep pits were successfully prevented from occurring except for the scab pitting mentioned above. The hard scabs could not be removed by normal cleaning methods.

He had been fortunate in being in the position of Fleet Engineer of the British Pacific Fleet in early 1944 when phosphate conditioning was first introduced in boiler water on a large scale. The results were quite incredible but many ships in the Fleet had already developed scab pitting, and although the phosphate treatment did not remove the scabs it appeared to arrest, or at least to diminish very greatly, the rate of pitting that went on underneath. All other corrosion problems completely disappeared, while the treatment made it possible to extend the intervals between boiler cleaning. It was a necessity in the Pacific campaign, when ships kept at sea for sixty days at a stretch.

Since he had left the Navy and joined an oil company he had become acquainted with the many difficulties of water treatment. A large proportion of the steam raised was used for process purposes and could not be recovered and returned as condensate to the boilers. A large quantity of make-up water was therefore required which was usually taken from the local river supply, often a very difficult supply and all too often it varied in the amount of its dissolved constituents. Generally speaking, a considerable quantity of low pressure steam was available or could be made available to evaporate the river water and return it to the boilers as condensate. The difficulty was a financial one. The evaporators to deal with large quantities of water cost twenty times the amount of the softening plants, and perhaps the author would consider whether it would not be possible to make them larger, say 100 tons an hour per unit, working to atmospheric conditions, at a price which might bear some reasonable relation to water softening plant.

Mr. R. J. Glinn said the paper had amplified information available in other papers, by dealing in more detail with atomic theories and the associated electro-chemical actions. This could readily be noted by comparing with, for example, the theoretical part contained in the recent Admiralty publication and, connected therewith, the paper given by Slater and Parr* in the present year (Nos. 4 and 5 of the bibliography).

It was suggested that perhaps the bibliography might be extended to include one or two other recently published works on corrosion. Two items he had in mind were Dr. Vernon's Cantor lectures and the Iron and Steel Institute's Special Report No. 41, which dealt with corrosion of steel by industrial waters and brought in a considerable amount of useful information on boiler corrosion.

Hamer's† paper indicated one of the early steps which emphasized the importance of marine water treatment and led to the issuing of the British Standard Specification on water treatment for marine boilers. He would like to emphasize that by carrying out the recommendations given in this publication—and he was very glad to see the author had mentioned these in his paper—had led to a considerable reduction of many of the troubles that occurred during the early part of the war and this Standard was now, he would suggest, the guide to avoid the corrosion troubles described by the author.

A lot of valuable information had been given in the author's paper, and he himself would keep it for reference purposes. He would ask the author, however, to consider, before it was finally published, whether certain cross-references could be given on each of the subjects dealt with.

The elimination of dissolved oxygen was now accepted practice. It had already been very fully dealt with and was essential to high pressure boilers. In connexion with land practice the removal of the final traces of oxygen under all conditions of working was much more essential with the higher pressure boilers than with lower pressure marine boilers, and this had been carried out with great success.

The use of phosphate was commented upon in various places in the paper, and he was thoroughly in agreement with its use, particularly to prevent hard scale formation which might occur owing to possible contamination of the feed water. Reference was made to this in the paper he had given on the subject of water in 1935 before the Institution of Mechanical Engineers‡, and the recommendations given at that time had been confirmed in practice and were still in line generally with those given by the author.

He was glad to note that the author referred to the growing practice in land power stations of re-circulating a small amount of boiler water back to the feed water. He had been advocating this method for a number of years in view of the benefits cited and of the results obtained at the Lancaster power station, which had been published in the technical journals. Other power stations were now adopting this method and some were at work. It was usual in these cases to employ a motor-driven pump, and he would ask whether the author could give any details of the type of pump he would recommend for reliable service under high pressure conditions.

Scale formation was referred to on p. 22, and the author was asked for his views on the use of X-ray examination of scales. It might be of interest to members to note that a British Standard Institution Committee was dealing with the examination of deposits encountered in the generation of steam. On account of the seriousness of silica scales on boiler tubes and turbine blades he would ask whether the author could amplify the reference to silica deposits in the paragraph on hardness. Although one must admit that these deposits had not been experienced so much in this country up to date, however, one heard from the United States of troubles on highly rated high-pressure boilers and on turbines and it was a point that would need to be watched in the future. If the de-mineralization process was used in place of evaporators for obtaining relatively pure make-up water—although the exchange processes had not been dealt with in the paper—any silica in the raw water was still present in the de-mineralized water, and in some cases this meant adding a de-silification process if one wished to avoid the possibility of silica scales. As this process would shortly be used in a number of power stations in place of evaporators, the author's views as to the relative merits of the two would be very pertinent.

Mr. M. S. Timms (Associate) said there were one or two points of chemical detail in the presentation of the theory of corrosion which might still be debatable. He did not propose, however. to raise them because he thought a discussion on purely chemical matters would probably be irrelevant on this occasion. However, as a practical man he might say that some busy engineers might be inclined to say to themselves, "On top of everything else this is too much!" It might therefore be permissible for him to comment that the essence of the paper could be condensed into a workable scheme of boiler water treatment and control which did not make undue demands upon the engineer's time.

Mr. Hillier had said that the prevention of boiler corrosion depended upon two factors: firstly, the reduction of the oxygen content of the boiler feed water to the lowest possible level; and secondly, the maintenance of an alkaline condition in the boiler water, the degree of alkalinity being controlled between fairly close limits. Too much emphasis could not be placed upon the need for controlling the boiler water alkalinity. Mr. Hillier had made clear that excessive alkalinity could result in an accelerated rate of corrosion and under these conditions the cure could be worse than the disease.

The second condition, the attainment of the desired degree of alkalinity, was in practice usually more easily achieved than the first condition, namely the elimination of oxygen, and there

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* Slater, I. E. and Parr, N. L. 1949. Proc.I.Mech.E., Vol. 160, p. 341, "Marine Boiler Deterioration".

† Hamer, P. and Stead, C. A. 1939-40. Trans.I.Mar.E., Vol. 51, p. 167, "Modern Boiler Feed Water Treatments and Suggestions for their Applications to Marine Boilers".

‡ Glinn, R. J. 1935. Proc.I.Mech.E., Vol. 129, p. 7, "Care of Modern Steam Generating Plant from the Water Side".

was no doubt that in practice, despite great efforts to maintain both the low oxygen content of the feed water and the desired degree of alkalinity, a considerable amount of corrosion was still being experienced in marine boilers steaming at 400 to 500lb. per sq. in. or thereabouts. This corrosion was affecting boilers of many different designs and was, he believed, causing a certain amount of anxiety to superintendent engineers in many companies. Investigations had revealed that in many of these cases the dissolved oxygen content of the feed water passing through the feed system was very low indeed under steady steaming conditions, but that there was a tendency for this to increase considerably under manœuvring conditions. Again it was customary—and, he understood, essential—to reduce the vacuum carried in turbine condensers when the main engines were at stand-by. This inevitably meant an increase in the dissolved oxygen content of the feed water unless the condensate temperature was allowed to rise correspondingly by reduction of the flow of cooling water circulating through the condenser, a practice which in his own experience it was not usual to follow. For instance, with the vacuum reduced to 20 inch of mercury and the condensate temperature at 90 deg. F., fairly typical conditions, the dissolved oxygen content might be as high as 1.5 ml. per litre, or about a hundred times the minimum content under steady steaming conditions. It was true that the volume of feed water passing to the boilers with main engines at stand-by was very much reduced, but it was a mistake, he thought, to assume that it was necessarily negligible. There might be turbine-driven feed pumps, electric generators running, and in many cases the condensate through these units passed ultimately back to the main condensers where the best de-aeration conditions were admittedly not being maintained. This state of affairs might continue for several days whilst a vessel was, for instance, riding in an exposed anchorage.

Finally, there was the water used for initial filling of boilers which in very many ships was necessarily cold and saturated with oxygen. Such water could give rise to very slight initial corrosion, particularly if the process of raising steam was delayed, and the boilers stood cold, perhaps for twenty-four hours, after filling, or the process of raising steam was prolonged, as was sometimes done in the interests of furnace brickwork. Once started in this way, the corrosion process, as Mr. Hillier had indicated, could continue whenever oxygen gained access to the boiler feed water. It was true that there were chemicals such as sodium sulphite which would remove oxygen from water under certain conditions, but it must be borne in mind that they were of limited value, for the continued addition of such materials to boiler feed water resulted in increasing the density of the dissolved salt content of the water and might introduce the need for blowing down of the boilers, a practice which was very much disliked by marine engineers for quite good reasons.

At this point he would like to digress for a moment to recommend the system of blowing down of boilers safely and easily over extended periods by the use of a micrometer type valve. As suggested by Mr. Hillier in his paper, blow down water could, under these conditions, be returned to the fresh water evaporators and heat and water so conserved. Such a system was quite commonly fitted in vessels built in the United States within the last five years or so. It struck him very forcibly by way of contrast with the difficulty of boiler corrosion which had been experienced with certain British built vessels that in his experience of some fifty or so vessels of American design and construction over the period of the last two or three years there had been very little if any report of boiler corrosion in these American ships. The question he would rather like, therefore, to put to Mr. Hillier, was this: despite the undoubted maintenance, while the ships were steaming steadily, of the conditions for the prevention of corrosion which had been described, some corrosion was nevertheless being experienced and appeared to be connected with the fact that the dissolved oxygen content of the feed water was not at all times as low as it might be. Was there, therefore, any room for improvement in the feed systems as commonly installed in British ships in order to obtain effective de-aeration of the boiler feed water at all times and irrespective of the condition of the main engine?

Mr. R. Cook, M.Sc. (Member of Council) said the paper was a long one and the uninitiated, after reading it, might well be excused for thinking there was very little that was not known about boiler corrosion and feed water treatment. The subject was, however, as the author had pointed out, very complex, and the truth was that there was much which was not known.

Take as a specific example, troubles which had been fairly widely experienced in recent years with their old friend, the Scotch boiler. Many marine engineers had complained of serious pitting corrosion in the last few years and they seemed to think that this trouble had coincided with the introduction of the modern mild steel tube in place of the old wrought iron tubes formerly fitted. So much was this the case that many superintendent engineers would certainly fit wrought iron tubes in preference to mild steel if they could be got, but unfortunately wrought iron tubes were no longer obtainable, either in this country or abroad. It appeared unlikely, from economic considerations, that the manufacture of wrought iron tubes would be resumed.

The experts said that the answer to this problem was adequate feed water treatment and reduction of oxygen to the utmost possible extent. Unfortunately, and he thought many superintendent engineers present would bear him out in this, that was a council of perfection in many ships employing Scotch boilers. The problem had been particularly serious in trawlers, for example, and somewhat of an *impasse* had been reached. Alternative solutions suggested to get over this trouble had been to subject the mild steel tube to various treatments, such as chromium diffusion, phosphating, and so on, designed to afford a protective coating to the mild steel tube. This line of attack was being pursued by the British Shipbuilding Research Association with a number of small model boilers which had been installed at the Chemical Research Laboratory.

The author had pointed out that most of the present data had been obtained at low pressures and temperatures, and it was hoped with these model boilers to be able to extend the knowledge of corrosion phenomena to at any rate about 250lb. per sq. in.

There was another point, and that was the comparative corrosion resistance of wrought iron and mild steel. It seemed to be generally accepted that under many conditions wrought iron had superior corrosion resistance properties as compared with mild steel, but although a good deal of work had been done in the past on this subject, it also appeared to be accepted that one really did not know the reasons for this superiority. If the reasons could be established beyond doubt, it would seem conceivable that mild steel might be modified so as to give improved corrosion resistance. Of course, the practical implications of this went far beyond the subject of the paper.

Mr. S. G. Christensen (Associate Member) said there was one point that appeared to be omitted—the possible effects of thermo-couples which might be set up in the boilers due to differences in temperatures. In some cases, corrosion had been seen going on where there had been normal boiler water conditions. Normally where there was a thermo-couple in a closed cycle—that was to say, if the flow of current took place in one part of the material—corrosion should not occur. If the cycle took place from the hot junction through the boiler water to the cold junction, there was, he thought, a chance of corrosion. This did not seem to have been mentioned, and he wondered whether anyone had any experience of or had done any investigations into this aspect.

To take up some of Mr. Cook's remarks on corrosion in Scotch boilers, he thought the answer to corrosion in boiler tubes in Scotch boilers was in many cases due to a simple cause; taking large numbers, it could be said that the majority of Scotch boilers were working with reciprocating main engines and quite a common arrangement was for the main engines to drive large capacity feed pumps which pumped the water up to a feed heater, which often acted as a de-aerator. On investigation it would be found that air leak-off cocks are usually fitted on the side of the air vessels by which the excess air could be drained off. From his experience over a number of years he found these cocks were not used; while at sea he was very much interested in cutting down corrosion and he found by using the air leak-off cocks he could bring up the feed temperature without rise of exhaust back pressure. If these air cocks were not used, the feed temperature fell and he thought this showed there was a good possibility of overloading the feed heater so far as its capacity to act as a de-aerator was concerned. That possibly was an answer for a lot of the corrosion found in Scotch boiler tubes today—the non-use of air leak-off cocks on the side of the air vessels.

MR. I. S. B. WILSON (Member) said he was pleased that the last two speakers had mentioned the Scotch boiler. There was an expression, "The poor are always with us", and he thought this was true of the Scotch boiler. The majority of steam plants associated with this type of boiler had the open feed system where the oxygen content of the feed water had not been guarded against. Therefore that side of corrosion in these boilers had not been counteracted in any way.

He would like to ask the author, who, after all, was applying most of his knowledge to the high evaporation rate boilers, whether, where no consideration was taken of the oxygen content, a feed water treatment such as was suggested by the knowledge given in the paper, would be of any advantage. With the welded construction of Scotch boilers coming to the fore, he thought this type of boiler would take on a new lease of life and the modern cargo vessels were going to see more of them.

He would like to ask whether, with the open feed system, cutting out the two buckets of soda ash, which was the normal feed treatment in that type of boiler, there would be some modified boiler treatment whereby improvements could be made. They had not got the closed feed system in all cases: he knew they could put it in, but he was speaking of vessels where expense was guarded against. Therefore, if they were to have an open feed system on ordinary cargo vessels with reciprocating engines some modified treatment would seem advisable, failing this they would have to come back to the two buckets of soda ash on Saturday mornings.

MR. S. J. MARLOW said it would seem to be suggested that water treatment was entirely a chemical operation to eliminate, so far as possible, the oxygen. He would like to know whether any results had been obtained with regard to electric treatment, which he believed was put forward during the past fifteen years.

Secondly, did the oxygen content increase with the use of injectors in place of the pumping method?

Iron tubes had been referred to as a material which would have a much greater life than commercial steel tubes, iron not being so susceptible to corrosive attacks.

He stated that he had known where lowmoor iron tubes had often been used, but, under the same conditions, these did not have the life of commercial steel tubes.

Correspondence

MR. C. A. STEAD wrote that this most comprehensive paper should prove extremely valuable as a general introduction to the scientific background of this subject, particularly for students, but whether the more theoretical aspects of the paper would be equally useful to the average practising marine engineer was another question and he thought that they should ask themselves what Mr. Hillier's theoretical considerations really meant to those who had to operate marine boilers and who, therefore, had to consider what type of water treatment should be adopted to prevent boiler corrosion and scale formation.

If one reduced to the simplest possible terms what the theoretical considerations explained by Mr. Hillier really meant, firstly the feed water should be slightly alkaline (or in other words its pH value should be on the alkaline side) to prevent attack on the ferrous metals of the feed system, secondly its oxygen content should be reduced to the lowest possible figure particularly for high-pressure boilers. To achieve this low oxygen content, some form of mechanical de-aeration of the feed water was essential and this could only be achieved by the adoption of such feed systems as those shown in Figs. 15 and 16, which were not usual in British practice. Any traces of oxygen remaining should then be dealt with by the addition of a small quantity of sodium sulphite sufficient to react with the traces of oxygen which might remain after de-aeration and to leave a small reserve of free sodium sulphite in the boiler water.

Turning to the boiler itself, the absence of oxygen in the feed water coupled with the small excess of free sodium sulphite in the boiler water should inhibit corrosion caused by oxygen (namely that due to so-called differential aeration). The presence of adequate alkalinity in the boiler water should further minimize corrosion due to this cause and also lessen the possibility of other forms of corrosion or pitting occurring. This boiler water alkalinity should contain a proportion of caustic alkalinity as this was the most effective alkali for supressing corrosion. A further small excess of alkali should be present in the boiler water either as carbonate or phosphate or both, in order completely to precipitate the scale forming salts from the boiler water, thus ensuring that it was always of zero hardness.

The soluble salts presented in the feed water and the alkalis which were added as feed water treatment would produce a quantity of dissolved salts in the boiler water. These dissolved salts had an upper limit for each class of boiler above which de-concentration by blowing down or washing out was therefore an essential part of water treatment if the dissolved salts should rise above the desired optimum figure.

The hardness which was precipitated by the use of alkalis appeared as a sludge which might, in some cases, adhere to boiler surfaces. For this reason, it was usually advisable physically to condition these with some organic material such as tannin or starch which produced a non-crystalline free flowing sludge which did not readily adhere to the heating surfaces.

The possibility of caustic embrittlement was minimized by maintaining a definite ratio of sodium sulphate to alkalinity in the boiler water. The sodium sulphate required might be present in the boiler water as a dissolved salt produced by the softening reactions or it might have to be added as a constituent of the treatment if this source was inadequate. Mr. Hillier had touched on the tests which were required in boiler water treatment but he had not emphasized the important point that these were an integral part of any chemical system of water treatment which could not be successful without a regular system of control testing which ensured that the correct conditions were maintained in the boiler water. Such tests should include a simple soap estimation of the hardness, an alkalinity test, a test for sodium sulphite if this re-agent was used, and preferably also in addition, a test for the dissolved solids present in the boiler water. In a few cases where full phosphate treatment was adopted, a determination of the phosphate content of the boiler was also necessary.

From the foregoing, he hoped it would be appreciated that the actual practical application of the theoretical principles so well explained by Mr. Hillier was an undertaking well within the capacity of marine engineers.

MR. T. H. TURNER (Member) wrote that he was very glad that Mr. Hillier referred to British Standard 1170 and it was fair to say that so far as water-tube boilers were concerned, marine engineers knew a lot more about boiler water control than they did in the previous decade. The pity was that they had not yet been able to help the trawlers' and tramps' Scotch boilers to the same extent and the difficulty of obtaining wrought iron tubes had increased the need for boiler water treatment in these Scotch marine boilers.

He believed that the principles laid down in B.S.1170 were capable of application to tugs and dredgers and other such craft, but at present the marine engineer took the line that he could not supply the supervision necessary for correct boiler water treatment in the small and frequently very old vessels.

Reference should also be made to B.S. 1328 : 1946 "Methods of Sampling Water used in Steam Generation" and also to B.S. 1427 : 1949 "Tests for Water used in Steam Generation", Group A Control Tests.

Author's Reply

Mr. Hillier would like strongly to support Captain Peile in the desirability of research into boiler corrosion problems under advanced steam conditions in view of the large amount of boiler equipment now in operation at high pressures. Although there is a very extensive literature on corrosion processes, there is almost a complete absence of data at the operating pressures and temperatures which are now in common use in boilers. For instance, it would be valuable to know the solubility of ferrous hydroxide in boiler water at different temperatures.

In reply to Admiral Simpson, in large evaporating plants the individual units are usually made as large as cleaning requirements will permit, and while this tended slightly to reduce the cost of the plants, the major part of the cost is due to the heating surface necessary to evaporate the water.

The total dissolved solids in the treated water from a softening plant are considerably greater than in water obtained from an evaporating plant so that evaporated water is always to be preferred, particularly for high pressure boilers or heavily loaded boilers, even though the first cost of the evaporating plant required may be higher.

Mr. Glinn's suggestion that the bibliography might be extended was given careful consideration but the literature on the subject was so extensive that the references were confined largely to those from which data given in the paper had been taken. He welcomed, however, the additional references given by Mr. Glinn.

He agreed with Mr. Glinn that de-aeration in land boiler installations had been very successful in preventing corrosion. There was not much difficulty in preventing corrosion in a boiler under steaming conditions given reasonable de-aeration and correct water treatment. Standby, port and idle conditions, however, required much more care and consideration than was usually given to them.

With regard to boiler recirculating pumps, a reciprocating pump had a high efficiency with a low running cost as compared with a centrifugal pump, but was more expensive unless the quantity of water handled was small.

For various reasons, the reciprocating pump was more flexible for the pumping pressures generally required, and would require less maintenance, particularly with high boiler pressures.

X-ray examination of scales was extremely useful in diagnosing the reactions and precipitations taking place in a boiler, and in giving guidance as to the water conditioning or corrections required to the water conditioning to get the best results in an individual case.

Serious cases of silica scale formation were, fortunately, not common, and he was unable to give any information outside of what could be obtained in various papers dealing with it. It was a subject which was being discussed and studied in the United States in connexion with some of the high pressure land boiler plants and was usually associated with a natural supply of water containing appreciable silica or with water treatment plant from which there was a silica carry over.

He had no doubt that the use of an evaporator to provide make-up water was much to be preferred to a demineralizator process because the residual dissolved solids present with the latter would be greater than present in make-up water obtained from an evaporator. This necessitated an increase in the total dissolved solids in the boiler or an increased boiler blow down. Further, it was difficult to maintain a consistently satisfactory performance with a demineralizator and ensure that no water escaped treatment at any time in passing through the apparatus.

Replying to Mr. Timms, almost every turbine steamer built in this country since 1925 had been fitted with a closed feed system and there had been very little serious trouble with corrosion although little or no water conditioning had been used in most of the ships.

He suggested that it would be very interesting if a superintendent engineer would arrange to run two or three ships without any boiler water conditioning and two or three ships with boiler water conditioning, and see how they compared with one another after several months' operation.

With the introduction of water treatment in recent years, there had been a number of cases of pitting and he suggested the pitting was mainly due to inadequate water treatment. It was essential that the alkalinity in the boiler should have an appreciable margin above the threshold of protection. In cold conditions, a pH value of 9·4 was the threshold of alkalinity to give protection. For hot conditions, when a boiler was under steam, it was probable that a pH value of 11 corresponded to the threshold of protection and that a higher value should be carried to provide an adequate margin to ensure protection.

Pitting occurred in conditions which prevented general corrosion but were not sufficiently alkaline to give complete immunity and in most cases the pitting would be eliminated if the alkalinity were raised appreciably. In general, it was probable that there was a tendency to carry too low an alkalinity, possibly due to an unjustified fear of the possible effects of a high alkalinity. With no boiler water conditioning, any small amounts of residual oxygen which entered the boiler might give rise to a slight general corrosion with negligible effects. Inadequate water treatment, however, could prevent general corrosion and protect 99·9 per cent of the surface, but unfortunately it created conditions in which the attack due to the same amounts of residual oxygen was concentrated on very small areas giving rise to pitting with its serious consequences which could be avoided if a higher and adequate alkalinity were carried.

The boiler designer usually stated the maximum total dissolved solids which could be accepted. If the sulphate ratio usually carried could be reduced, it would be possible to carry higher alkalinities without too high a total of dissolved solids. There seemed to be a case for an investigation as to whether the sulphate ratios commonly used were really necessary with solid, forged and welded constructions.

The question of standby, port and idle conditions was becoming more complicated with the large increase of various types of equipment in modern ships and was now receiving considerably more attention. There was an increasing practice of fitting special de-aerating equipment to deal with harbour conditions, and such equipment was also being arranged to ensure efficient de-aeration of the main feed under standby and manœuvring conditions.

It should, however, be appreciated that it was the total amount of free oxygen received by the boiler which mattered and that short periods of relatively poor de-aeration during manœuvring conditions were not of serious consequence if efficient de-aeration was obtained over long periods of steaming.

The problem of dealing with idle boilers and filling empty boilers was much more difficult and deserved much more consideration and attention than was usually given to it. More oxygen could get into or be put into a boiler during such periods than in months of steaming. Wherever possible, air should be prevented from getting into a standing boiler by carrying a steam pressure in the boiler well above that of the atmosphere.

When filling a boiler, it should be filled with de-aerated water while it was still under steam pressure and the boiler should subsequently be maintained at a pressure above that of the atmosphere either with steam, hydraulically, with de-aerated water if at all possible, or possibly by the use of an inert gas, such as nitrogen. This might require a complicated routine but it would be less costly and preferable to the maintenance which might otherwise be required.

An empty boiler should preferably be filled with de-aerated water at as high a temperature as could be made available and a steam pressure above atmosphere raised as quickly as circumstances permitted. A flow through the boiler should be established as soon as possible to remove any oxygen the water might have absorbed during the filling operation.

The use of much higher alkalinities in idle boilers and for filling empty boilers, say 1,000 p.p.m. of NaOH, i.e., a *p*H value of 12·5, merited careful consideration but would require arrangements for reducing the alkalinity when the boiler went on load and conserving the water removed from the boiler.

With regard to the question of blowing down boilers, a continuous blow down was to be preferred to an intermittent blow down and he considered that, with the wider adoption of boiler water conditioning for marine boilers, it would probably be only a matter of time before continuous blow down was seriously considered and the appropriate equipment provided for that purpose.

Mr. Cook raised the question of the difference in service obtained in Scotch boilers between wrought iron tubes and mild steel tubes.

It was difficult to see why there should be any difference so far as corrosion is concerned, but there was the possibility that the more homogeneous character of the wrought iron might allow the formation of a closer and better protective oxide film. Corrosion in Scotch boilers was, however, much more likely to be due to differences in the operation of the feed pumps and feed heaters, as mentioned by Mr. Christensen.

Engine driven feed pumps were usually provided with a reciprocating engine and had a swept volume much larger than required. Air cocks were fitted and opened to cushion the pumps so that the water was saturated with air and there was also a large amount of free air present in the water. A direct contact feed heater was usually provided and most of the air in the feed water would be liberated in the feed heater and discharged to the atmosphere or the condenser if the feed heater were properly vented. In many cases, however, the air discharge was not used or was partially closed and it was not surprising that serious corrosion was sometimes experienced.

There was an increasing practice of installing closed feed and de-aerating systems in Scotch boiler installations.

It was suggested that the research which Mr. Cook mentioned was in hand could profitably be extended to much higher pressures than mentioned.

The existence of differential temperatures, raised by Mr. Christensen, undoubtedly created differences in potential but it was doubtful if these were significant from the corrosion aspect given efficient de-aeration and correct water treatment.

The usual temperature differences present in a well-designed boiler did not give rise to any serious trouble but serious differential temperatures arising due to defective or erratic circulation could give rise to thermal-cracking which could be aggravated by corrosion.

In reply to Mr. Wilson, there was a growing practice of installing the closed feed system in Scotch boiler jobs; they were automatic in operation and it was a mistake to consider that they were expensive. It was just as important to look after de-aeration with Scotch boilers as with water tube boilers.

If an open feed system with reciprocating pumps was preferred, a de-aerator should be installed as shown in Figs. 15 or 16, and would ensure a de-aerated feed supply to the boilers. If the feed water contained a high dissolved oxygen content, there was no feed water conditioning which would prevent corrosion occurring.

The possibility of using electrical treatment, raised by Mr. Marlow, had been advocated from time to time but he knew of no successful application.

There was no reason why the use of an injector to feed the boiler should increase the oxygen content of the feed water.

In reply to Mr. Stead, the feed systems shown in Figs. 15 and 16 were both in service in British merchant ships.

The routine tests necessary for the control of boiler water conditioning had not been dealt with in the paper because they were fully covered by British Standard Specification 1170, which was included in the list of references.

He agreed that the testing routine and the practical application of the principles outlined in the paper were well within the capacity of marine engineers.

British Standard Specification 1170 had filled a need in the operation of marine boiler equipment, but it was suggested that there is room for clarification and simplification. There were a considerable number of ambiguities in the chemical references which made understanding of the problems more difficult for the average engineer than need be. He would also suggest that there should be a standardization of the reporting of the results of water analyses and that water analyses should be reported in either parts per million (p.p.m.) or equivalent parts per million (e.p.m.) and that the use of all other units should be discouraged. This would facilitate comparisons in performance and the units involved were easy to understand. They were both weight per weight units and the e.p.m. was obtained by dividing the p.p.m. by the equivalent weight of the substance concerned. All the results of the analyses required in connexion with water treatment could be expressed in e.p.m. He would also like to suggest that alkalinity should be reported in parts per million of OH.

With reference to Scotch boilers in trawlers and tramps, etc., mentioned by Mr. Turner, he considered that de-aeration of the feed water was the major step to be taken in overcoming the corrosion problems and that the equipment involved was comparatively simple to instal and operate.

Screwshaft Casualties—The Influence of Torsional Vibration and Propeller Immersion

S. ARCHER, B.Sc.*

The paper describes investigations carried out during the past two years by the research staff of Lloyd's Register of Shipping with a view to determining the principal factors underlying the many screwshaft failures in the "Liberty" class of vessel. Statistical inquiry is made into the circumstances of more than 100 cases of propeller losses at sea during the past three years, and the character of typical fractures is illustrated and discussed. The investigations have included the calculation and measurement of torsional vibration stresses, estimation of bending fatigue stresses, and consideration of the fatigue strength of the keyed propeller assembly in the light of published experimental data, and have led to a close study of propeller immersion as affected by deficiencies believed to exist in ballast capacity.

A "synthetic" racing test is described, as carried out on a moored "Liberty" ship heavily trimmed by the head for the purpose of measuring severe torsional vibration stresses occurring above the engine service speed.

The conclusion is reached that failures have primarily been the result of serious reduction in the fatigue strength of the shaft on account of severe overstressing in torsional and bending fatigue during periods of excessive racing occasioned by insufficient propeller immersion. In consequence the otherwise moderate torsional vibration stresses occurring near the maximum designed service speed of the engines have often been enabled to bring about early failure. The paper concludes with recommendations for the provision of increased water-ballast capacity in this class of oil-burning vessel and for the avoidance of continuous operation on the torsional vibration critical near the service speed.

The appendices include a theoretical treatment, believed to be novel in application, of the general case of torsional vibration of a damped two-mass system, with details of a convenient method of rapidly deriving propeller damping factors, together with a discussion of various precautions against the damaging effect on screwshafts or torsional and flexural vibration.

INTRODUCTION

Soon after the conclusion of the late war a pronounced increase in the screwshaft casualty rate of certain classes of 10,000-ton dead weight merchant vessels, built in America under the emergency programme, began to cause concern in maritime circles, at first mainly in America, but later also in Europe and elsewhere, as transfers of the U.S. surplus fleet to other flags began to take effect. The casualties took the form of fatigue cracks, or fractures, at the large end of the screwshaft cone.

GENERAL

The class of vessel chiefly affected has been the well-known "Liberty", or E.C.2-S-C.1 type, although the trouble has also been common in the Canadian "Victory" type and, to a lesser extent, in some of the oil-burning vessels of the "Canadian" type.

The approximate general specification of these vessels is as follows:—length 416 feet b.p., breadth 56·9 feet moulded, depth 37·33 feet moulded to upper deck, summer draught 27ft. 9in., gross tonnage about 7,160, dead weight tonnage 10,000.

Fig. 1 shows the general arrangement of the "Liberty" type vessel, including bunkering and ballasting capacities.

It may be stated that all these classes are generally similar as regards hull form and propelling machinery and differ in the former respect only slightly from the well-known U.S. built "Ocean" type, which, in turn, was based on the British "North Sands" design developed on the North-East coast, the prototype vessel being named S.S. *Empire Liberty.*

The principal differences between the various American-built vessels lie in the steam generating plant, in the superstructure, and extent to which welding is employed, and in the bunkering and ballasting arrangements

The propelling machinery of the U.S. built "Oceans" and "Libertys" and the Canadian "Victorys" consists of a triple-expansion steam engine having three cylinders, 24½-inch, 37-inch, and 70-inch diameter by 48-inch stroke, with a steam pressure of 220lb. per sq. in., and with superheat. The design was developed by The North-Eastern Marine Engineering Co., Ltd. The shafting sizes and arrangement, also the propellers, stern frames, and rudders are for all practical purposes identical (see Figs. 2 and 13). The rated service horse-power of the main engines is 2,500 i.h.p. at 76 r.p.m., and details of the 4-bladed, solid, manganese bronze propeller are indicated in Fig. 2. The rudder is of a special American design, known as the "Goldschmidt" type, whereas in the original "North Sands" class a conventional semi-balanced type was fitted.

The principal differences between the types are that whereas the "Ocean" vessels have coal-fired Scotch boilers of a total heating surface of 7,140 sq. ft., both the "Libertys" and the Canadian "Victorys" are oil-burning with water-tube boilers of approximately 10,000 sq. ft. heating surface. The "Canadians" have Scotch boilers, either oil- or coal-fired. Further, the "Libertys" and a few of the Canadian "Victorys" have deep tanks of a total capacity of 650 tons in No. 1 hold, in addition to the deep tank abaft the engine-room of capacity 760 tons. The latter is also fitted in the "Oceans", which have, however, a 500-ton cross-bunker forward of the boiler-room,

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*Senior Engineer Surveyor for Rcsearch, Lloyd's Register of Shipping

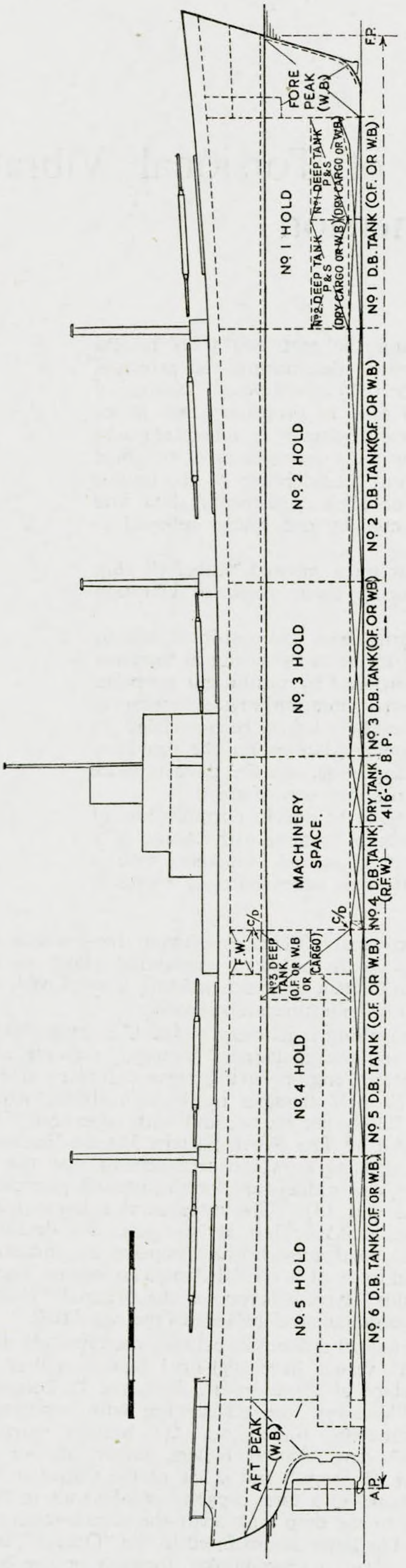


FIG. 1—*Ballast and bunker arrangements for "Liberty" vessels*

and a 600-ton 2nd deck bunker in way of the engine- and boiler-rooms. The presence of the cross-bunker, of course, cuts down the effective length of No. 3 hold, but on the other hand, the latter may be used as additional bunker space (955 tons) on long voyages.

STATISTICS

The construction of the largely welded dry cargo "Liberty" type of vessel was begun in the United States in January 1942, and altogether no less than 2,580 were built, of which 2,315 remained at the conclusion of hostilities, representing a total gross tonnage of nearly 17 million tons, or about 20 per cent of the world total for 1947 (13½ per cent in number) of all vessels greater than 500 tons gross.

Information at 1st December 1948, revealed that altogether a total of 583 "Liberty" screwshafts have been renewed, including during the past three years about 100 casualties at sea with resulting loss of propeller. The cost of these breakdowns in salvage and demurrage charges alone needs no emphasis.

Fig. 3 shows the monthly world totals of propeller losses at sea for the three years ending 31st December 1948, as derived from information published in Lloyd's List. The diagram also indicates the monthly proportion of "Liberty" casualties, and in the lower graph the number of non-"Liberty" failures. The figures in brackets on the "Liberty" graph give the estimated casualties per 1,000 "Liberty" vessels in service during each month and do not include "Liberty" tankers.

It will be seen that the heaviest casualty rate for "Libertys" occurred in the summer and autumn of 1947 and that the present rate is very considerably reduced.

The large drop occurring in early 1948 is most probably to some extent a reflexion of the recommendations of the American Bureau of Shipping, firstly, in May 1947, for more frequent examination of shafts (2 years instead of 3 years), and secondly, in January 1948, for a reduction in maximum engine speed to 66 r.p.m. loaded, and 70 r.p.m. light. On the other hand, it should be remarked that the smaller number of casualties at sea must inevitably have been influenced by the very

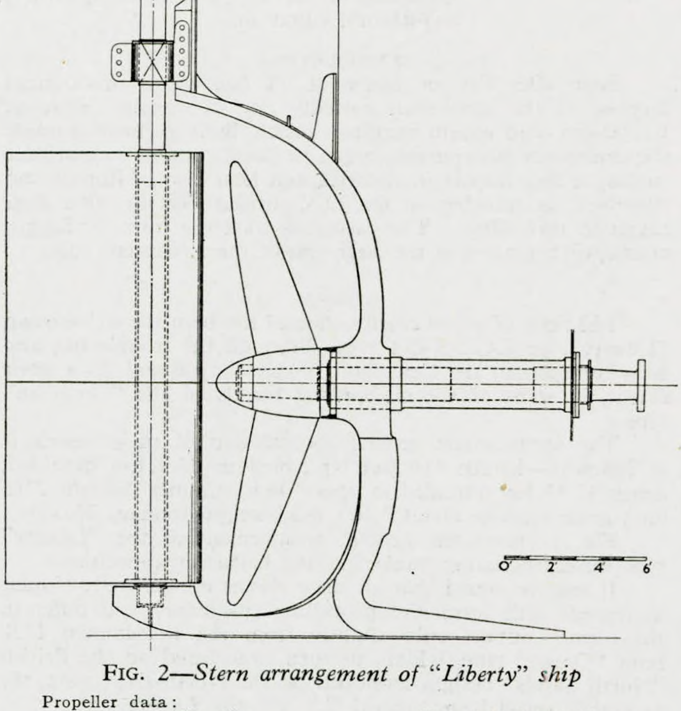
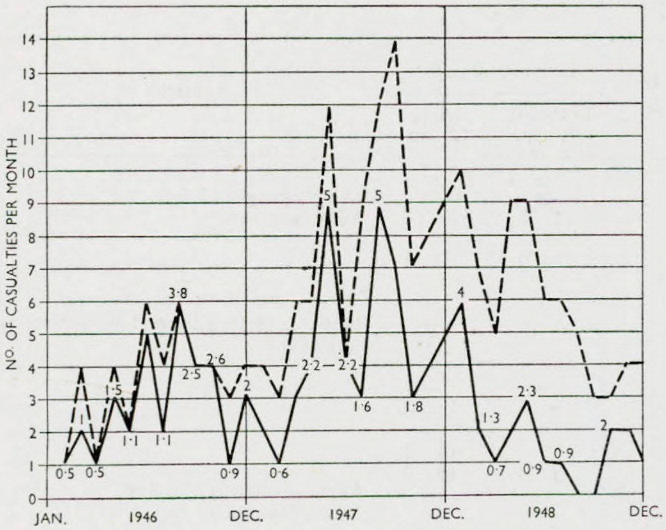


FIG. 2—*Stern arrangement of "Liberty" ship*

Propeller data:

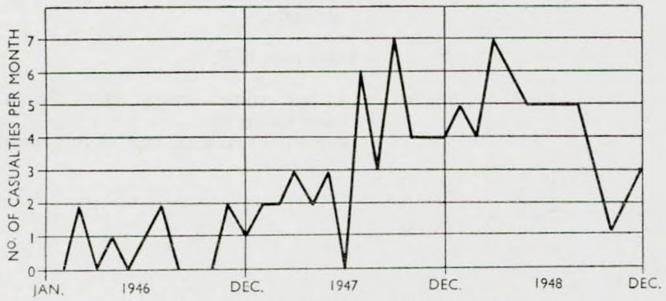
Diameter, 18ft. 6in.	No. of blades, 4 r.h.
Pitch varying (16ft. mean)	Material manganese bronze
Surface, 117 sq. ft.	Loss/diameter ratio 0·14



---- *Total casualties*

——— *"Liberty" casualties*

Figures on the curve give approximate casualties per 1,000 "Liberty" ships in service (based on United States statistics of number of "laid-up" vessels)



Casualties other than "Liberty"

FIG. 3—*Statistical survey of losses of propellers at sea over period January 1946 to December 1948 (based on Lloyd's list)*

appreciable reduction in numbers of "Liberty" vessels in service between the peak casualty period of July 1947, and the spring of 1948, say 20 to 25 per cent fewer vessels at sea. Accordingly, in the absence of parallel information on the rate of renewal of "Liberty" screwshafts from all causes, it would be premature to assume that all is now well. Meteorological sources state that North Atlantic weather in 1947 was not as a whole unusually stormy, although periods of uncommonly severe gales did occur in the early part of the year, particularly in early January.

Analysis of the statistics of non-"Liberty" propeller losses at sea shows that the ratio of casualties for steam-reciprocating vessels, compared with motor vessels, is approximately the same as that of their respective world totals in numbers, thus indicating that neither class of vessel is inherently more prone to screwshaft breakages at sea.

It may be of interest to observe that for the 12-monthly period ending 31st December 1947, of the total of 6,161 vessels classed with Lloyd's Register, of which more than 4,000 screwshafts were examined by the surveyors, only 13 were found broken and 74 condemned for cracks, representing respectively 0·21 per cent and 1·2 per cent of the total vessels classed, and the greatest number of renewals occurred in April and October.

Figs. 4(*a*) Atlantic, and (*b*) Pacific, show the geographical location of the individual "Liberty" vessels at the time of propeller loss, together with their approximate course, again based on Lloyd's List.

Table I is the key to Figs. 4(*a*) and (*b*) and lists the details of casualty voyages, also, where known, the loading condition at the time, with or without information as to the draughts when leaving the last port of call.

As would be expected from considerations of traffic density, the north Atlantic area includes by far the greatest number of casualties. Of the world total of 101 casualties, 55 occurred in the north Atlantic north of about latitude 30 deg. N., and of these no less than 52 were west-bound, and although information is lacking in the case of 3 vessels, at least 47 were on ballast voyages at the time. Of the "Liberty" world total casualties enquiries have revealed that no less than 81 occurred when in ballast, despite information being lacking in the case of 8 vessels.

It is perhaps of significance that the majority of breakdowns took place when on a course contrary to the prevailing winds, the principal casualty areas being the western Atlantic and U.S. eastern sea-board, the Channel approaches, the Azores, and mid-Pacific around Honolulu.

As regards the north Atlantic casualties the balance, or rather lack of balance, of trade, prevailing since the war, together with U.N.R.R.A. and Marshall Aid, has undoubtedly caused a predominantly easterly flow of cargoes, more particularly as carried in American bottoms. Furthermore, having regard to the relative costs of bunker oil in the United States, as compared with Europe (for example, about 33⅓ per cent greater in the U.K.), the majority of these vessels would bunker in the U.S. for a return voyage to Europe, the homeward leg of which would usually be made in ballast. Thus, those vessels returning from N.W. Europe and nearing the American seaboard, or those from Italy and S. France in the vicinity of the Azores, would at the time of casualty have steamed anything from 5,000 to 6,000 miles, which at 11 knots and 27 tons per day represents a total consumption of up to 540 tons of oil fuel. This is, of course, quite normal, but where, as is usual from considerations of contamination, oil fuel tanks are not replenished with water ballast and additional dry ballast is not carried, it will undoubtedly be found difficult in the "Liberty" type of vessel to strike a reasonable compromise between adequate propeller immersion and sufficient draught forward to give satisfactory steering qualities and minimize pounding damage.

Detailed discussion of the ballasting problem is included in a later section.

At this stage reference must be made to the almost complete immunity from screwshaft failures enjoyed by the coal-burning "Ocean" class, only one case of a condemned shaft having been reported, and this after a period of six years service. It may also be significant that this particular vessel had been converted to oil fuel seven months previously.

The "Ocean" class vessels were built in the U.S. to the order of the British Government over the period 1941-42. The total number built was 60, of which 18 became war casualties, and the remaining 42 have been operating for an average period of about 6½ years, mostly under British ownership, whereas the bulk of the "Liberty" casualties have occurred after a very much shorter period of service, in some cases only a few months.

THE CHARACTER OF SCREWSHAFT FAILURES

Fig. 5 shows details of the "Liberty" type propeller shaft tail end, from which it will be noted that the design follows normal practice and incorporates the modern sled-runner type keyway. A standard type of rubber-ring sealing arrangement is also indicated between the end of the bronze liner and the usual recess in the propeller boss. It should be noted that precisely similar keying and sealing arrangements were used on the "Ocean" class screwshafts.

In the Canadian "Victory" vessels, however, the semi-circular ended type of keyway was usually adopted, but it should be noted these shafts were by no means immune from failure as will be seen from Table II. A study of this table indicates that a common type of failure is characterized by cracking parallel to one or both faces of the keyway, more usually on the driving side, sometimes associated with cracks

TABLE I

Vessel No.	Port of departure	Destination	Ballast or loaded	Leaving draughts		Remarks
				Forward, ft. in.	Aft, ft. in.	
1	Marseilles	Hampton Roads	Troops	—	—	—
2	Colombo	Lourenco Marques	Ballast	—	—	Mean draught 12 feet
3	Bordeaux	U.S.A.	Ballast	11 7	19 11	2,000 tons sand
4	Casablanca	Hampton Roads	—	—	—	—
5	New York	Antofagasta	Ballast	6 0	17 0	—
6	Savona	Charleston	Ballast	—	—	—
7	Split	—	—	—	—	—
8	Havre	New York	Ballast	—	—	Between 14 ft. 6 in. and 16 feet mean draught
9	Trinidad	—	—	—	—	—
10	Port Alfred	New York	Ballast	—	—	—
11	Galveston	La Plata	Ballast	9 0	18 0	—
12	Trinidad	Buenos Aires	Ballast	—	—	—
13	New Orleans	Demerara	Ballast	8 7	15 3	—
14	Mobile	—	Ballast	7 2	16 0	—
15	London	Vancouver	Ballast	—	—	Mean draught 12 ft. 8 in.
16	Rotterdam	Philadelphia	Ballast	—	—	Mean draught 13 feet
17	Bordeaux	Montreal	Ballast	7 0	15 0	—
18	Shanghai	Vancouver	Ballast	—	17 6	—
19	Port Said	Naples	Ballast	—	—	—
20	Colombo	Vancouver	Ballast	8 6	16 3	1,170 tons O.F.
21	Leghorn	Charleston	Ballast	—	—	—
22	Glasgow	Vancouver	Ballast	—	—	720 tons O.F.
23	Savona	Hampton Roads	Ballast	—	—	—
24	Rouen	Hampton Roads	Ballast	—	—	Between 14 ft. 6 in. and 16 feet mean draught
25	Port Alfred	Paranam	Ballast	—	—	—
26	Hong Kong	San Francisco	Ballast	—	—	1,000 tons O.F.
27	Trinidad	Mackenzie	—	—	—	—
28	Singapore	Vancouver	Ballast	11 8	15 10	Loaded some cargo Port Swetn. and Colombo
29	Rouen	New York	Ballast	—	—	Between 14 ft. 6 in. and 16 feet mean draught
30	Durban	Buenos Aires	Ballast	11 2	16 0	—
31	Lourenco Marques	Buenos Aires	Ballast	12 6	15 6	—
32	Shanghai	W. C., N. America	Ballast	—	18 0	—
33	Civitavecchia	New Orleans	Ballast	—	—	—
34	Trinidad	Baltimore	Ballast	—	—	—
35	Ancona	Philadelphia	Ballast	—	—	—
36	Hamburg	Key West	Ballast	—	19 0	—
37	Hampton Roads	France or Alex.	Coal	—	—	—
38	Hong Kong	Honolulu	Ballast	—	—	500 tons O.F.
39	Antwerp	Savannah	Ballast	—	14 0	—
40	Durban	Montevideo	Ballast	8 10	15 10	—
41	New York	Antwerp	Part loaded	24 9	28 9	—
42	Venice	Hampton Roads	Ballast	—	—	—
43	Antwerp	Hampton Roads	Ballast	—	14 0	—
44	Liverpool	New York	Part loaded	9 0	17 9	1,166 tons cargo
45	Galveston	Mobile	Ballast	... 5 6	14 6	—
46	Rouen	Delaware Capes	Ballast	—	—	Between 14 ft. 6 in. and 16 feet mean draught
47	Rouen	Hampton Roads	Ballast	—	—	Between 14 ft. 6 in. and 16 feet mean draught
48	Yokohama	Balboa	Ballast	—	—	Bunkered Yokohama
49	Genoa	Hampton Roads	Ballast	—	—	—
50	Rouen	Demerara/U.S.A.	Ballast	—	—	Between 14 ft. 6 in. and 16 feet mean draught
51	Emden	Montreal	Ballast	—	13 0 16 0	—
52	Baltimore	La Guaira	Part loaded	18 10	25 4	—
53	Genoa	Hampton Roads	Ballast	13 3	17 0	—
54	St. Nazaire	New York	Ballast	7 0	14 0	—
55	Gibraltar	Hampton Roads	Ballast	—	—	—
56	Port de Bouc	Ceuta	Ballast	—	—	—
57	Cork	New York	Ballast	9 5	14 3	—
58	Lisbon	Baltimore	Ballast	—	—	—
59	Hampton Roads	New Orleans	Ballast	—	—	—
60	Lisbon	Hampton Roads	Ballast	—	—	—
61	Rouen	Hampton Roads	Ballast	—	—	Between 14 ft. 6 in. and 16 feet mean draught
62	Manila	U.S. Gulf Ports	Ballast	8 9	14 5	—
63	Yokohama	San Francisco	Ballast	—	—	—
64	Trieste	Baltimore	Ballast	—	—	—
65	Aarhus	Philadelphia	Ballast	—	17 9	—
66	Antwerp	New York	Ballast	—	17 8	—
67	New York	Baltimore	Ballast	6 0	13 0	—

TABLE I—*continued*

Vessel No.	Port of departure	Destination	Ballast or loaded	Leaving draughts		Remarks
				Forward, ft. in.	Aft, ft. in.	
68	Venice	United States	Ballast	—	—	—
69	Guam Honolulu	Balboa	Ballast	9 6	16 2	Bunkered Honolulu
70	Demerara	Trinidad	Part loaded	—	—	3,730 tons cargo 16 ft. 6¾ in. mean draught
71	Rouen	Hampton Roads	Ballast	—	—	Between 14 ft. 6 in. and 16 feet mean draught
72	Genoa	Hampton Roads	Ballast	7 10	13 0	—
73	Liverpool	Takoradi	Ballast	11 0	17 0	—
74	Rouen	New York	Ballast	—	—	Between 14 ft. 6 in. and 16 feet mean draught
75	Trieste	Ceuta	Ballast	—	—	—
76	Boston	Porto Alegre	General	—	—	—
77	Genoa	New Orleans	Ballast	6 2	17 3	—
78	Norrkoping	Charleston	Ballast	—	18 0	—
79	Avonmouth	Mobile	Ballast	6 4	17 6	—
80	Savona	Mobile	Ballast	—	—	—
81	Buenos Aires	Calcutta	Grain	27 8	29 2	—
82	Gibraltar	Hampton Roads	Ballast	—	—	—
83	Cherbourg	New York	Ballast	5 11	14 5	—
84	Helsinki	Key West	—	—	—	—
85	Marseilles	Hampton Roads	—	—	—	—
86	Rouen	Hampton Roads	Ballast	—	—	Between 14 ft. 6 in. and 16 feet mean draught
87	Tampa	Rotterdam	Loaded	26 8	27 11	—
88	Casablanca	U.S.A.	Ballast	—	—	—
89	Rio de Janeiro	Cuba	Ballast	—	—	—
90	Rouen	New York	Ballast	—	—	Between 14 ft. 6 in. and 16 feet mean draught
91	Calcutta	Camden, N. J.	Part loaded	18 7	21 6	—
92	Yokohama Honolulu	Balboa	Ballast	7 0	15 6	Bunkered Honolulu
93	Marseilles	New York	Ballast	17 8	20 10	—
94	Amsterdam	Hampton Roads	Ballast	10 3	15 0	—
95	Cork	New York	Ballast	8 0	15 2	—
96	Melbourne	Liverpool	Part loaded	22 7	25 5	—
97	Genoa	North of Hatteras	Ballast	—	—	—
98	Galveston	Fusan	Wheat	—	—	—
99	Abadan	New Caledonia	—	—	—	—
100	London	Portland, Or.	—	—	—	—
101	Duingun	Vancouver	Ballast	—	—	—

radiating from the forward end of the keyway, again usually, but not invariably, on the driving side. The parallel cracks are mostly situated about ¾ inch from the side of the keyway and frequently begin circumferentially for a short distance before turning parallel to the latter. They are sometimes accompanied by axial cracking along the root fillet (if any!) of the keyway (see ships O, T, and V, Table II), and the material between the parallel crack and the keyway is in some cases honeycombed with fine cracks. Fig. 6 (Plate 1) is a representative example of this type of failure (ship T), which, as shown in Table II, is almost invariably accompanied by a badly fitted propeller and, or alternatively key and not infrequently also by evidence of corrosive attack. In the latter case examination of the rubber sealing ring usually reveals the cause.

Where the propeller has been slack on the taper, it is also common to find a bronze deposit on the shaft surface, indicative of rubbing or fretting action between the surfaces. This class of failure bears strong symptoms of excessive bearing pressure on the sides of the keyway at the forward end, most probably due to heavy fluctuations of tangential loading, which, in the absence of frictional grip in the slack taper fit, have to be carried by the sides of the keyway alone. Where no proper fillet is provided at the root of the keyway, this type of cracking is facilitated.

As regards the cracks radiating from the forward end of the keyway these may be inclined at from 45 deg. to 60 deg. to the shaft axis and reach several inches in length. Cases also occur where circumferential cracking takes place in way of the sealing ring, particularly where the latter has permitted sea water leakage and, or alternatively where the shaft has been nicked circumferentially on account of careless machining of the liner end. It is of interest to note that micro-examination of some of the cracks described in Table II showed them usually to be trans-crystalline in character; further, the microstructure of the metal was fine-grained and uniform. In addition, mechanical tests showed the material to have a tensile strength of about 30 ton per sq. in. with a yield strength of 14 ton per sq. in., and a reversed bending endurance limit of ±13½ ton per sq. in. The possibility of defective material being responsible for the failures would thus appear to be ruled out, especially since a considerable number of different forges and steelworks were involved.

The failures so far described have referred to Canadian-built vessels, in most of which the round-ended type of keyway was used. Some illustrations of fractured "Liberty" type screwshafts will now be given. Fig. 7 (Plate 1), shows a typical torsional fatigue type of fracture, probably emanating from the forward end of the keyway. From the appearance and extent of the oyster-like ring markings reported it would seem that approximately one half of the section was cracked before final rupture took place. The vessel was two days out from Antwerp for New York in ballast with a leaving draught aft of 17ft. 8in. (see Table I and Fig. 4, vessel No. 66). Fig. 8 (Plate 2) is a beautiful example of the disastrously weakening effect of corrosion under the action of repeated stress accentuated by different kinds of stress-raisers. The fatigue cracks

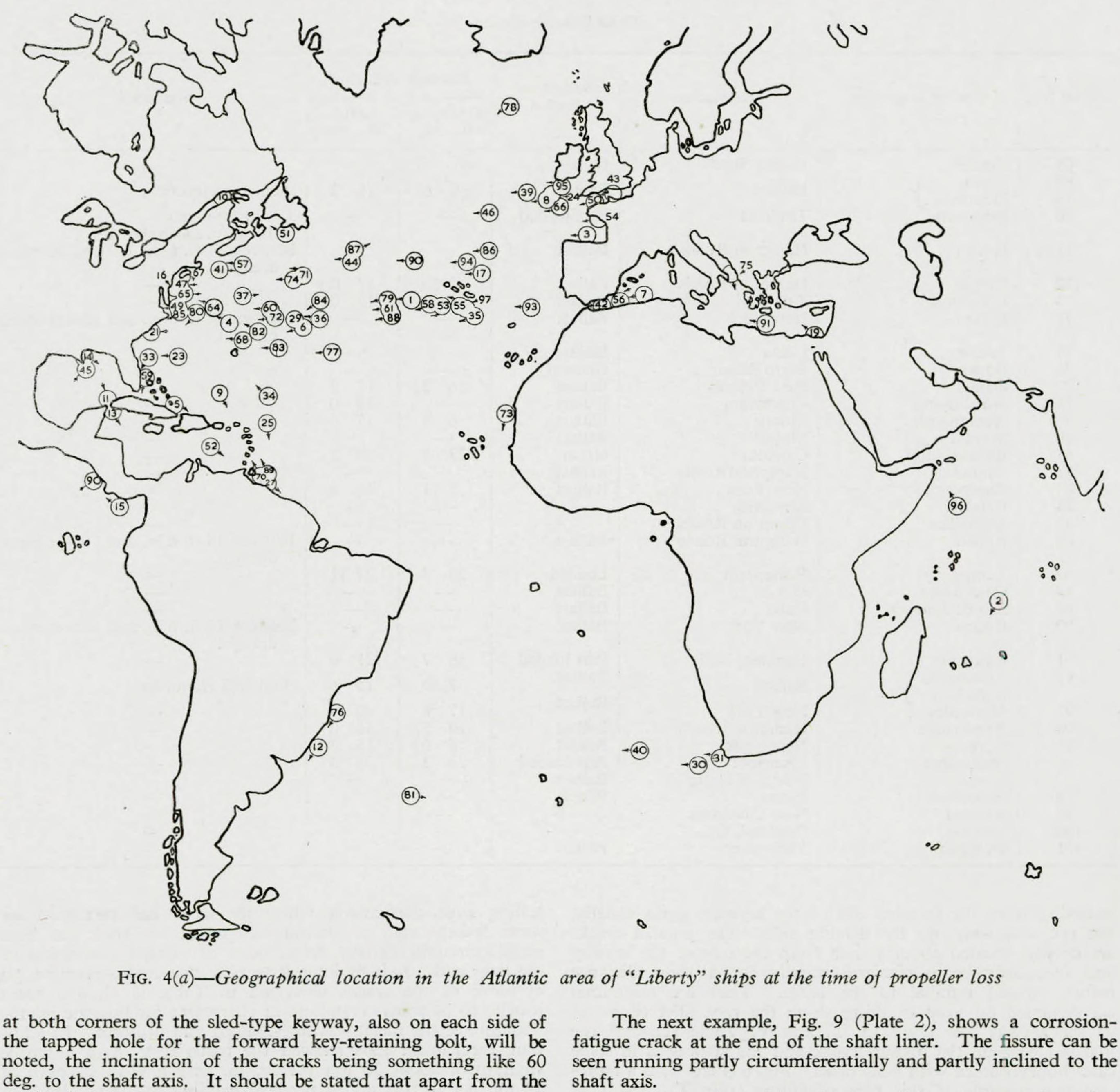


FIG. 4(*a*)—*Geographical location in the Atlantic area of "Liberty" ships at the time of propeller loss*

at both corners of the sled-type keyway, also on each side of the tapped hole for the forward key-retaining bolt, will be noted, the inclination of the cracks being something like 60 deg. to the shaft axis. It should be stated that apart from the cracks shown, the root of the keyway was cracked for its full length on the driving side and part length on the leading side. No fillet radius had been given to the keyway. The forward part of the root crack on both sides, when opened up, presented a zig-zag or saw-tooth appearance, clearly indicative of torsional fatigue stress, and this was confirmed by the appearance of a number of the characteristic corrosion-fatigue star cracks, again at about 60 deg. to shaft axis after hot etching the bottom of the keyway abreast the tapped hole. The shaft was also heavily corroded underneath the leaky rubber ring and was cracked circumferentially all round the shaft, except at the end of the keyway, to a depth of up to $\frac{11}{16}$ inch. Metallurgical examination of the shaft material, including mechanical tests, showed that the steel was of good quality, with a tensile strength of 30 ton per sq. in., 34 per cent elongation in 2 inch and over 30ft.-lb. Izod impact value. In this example it would appear from the orientation of the fatigue cracks that combined bending and torsional stresses were responsible for the failure.

The next example, Fig. 9 (Plate 2), shows a corrosion-fatigue crack at the end of the shaft liner. The fissure can be seen running partly circumferentially and partly inclined to the shaft axis.

Figs. 10 and 11 (Plate 3) show different views of another "Liberty" shaft which broke at sea, and here the familiar fatigue ring markings are clearly in evidence, indicating that the point of initiation of the crack was roughly at 90 deg. from the keyway, i.e., in line with one of the propeller blades. The fatigue crack, which extended over more than half the cross section, was only slightly helical, and suggests that combined bending and torsional stresses took part in bringing about failure. This vessel lost her propeller when two days out from Rouen for Hampton Roads. She was in ballast with a mean draught between 14ft. 6in. and 16 feet (see Table I and Fig 4, vessel No. 24).

Fig. 12, illustrates another type of "Liberty" fracture, the cracks luckily being detected at an early stage of their development and then only by the application of magna-flux methods, being invisible to the naked eye or to the glass, except when highly polished and under very strong oblique lighting. In this case, which is apparently typical of a fair number of

FIG. 7—*Typical example of screwshaft failure in "Liberty" type*

FIG. 6—*Typical example of cracked screwshaft in "Canadian Victory" type*

Plate 2

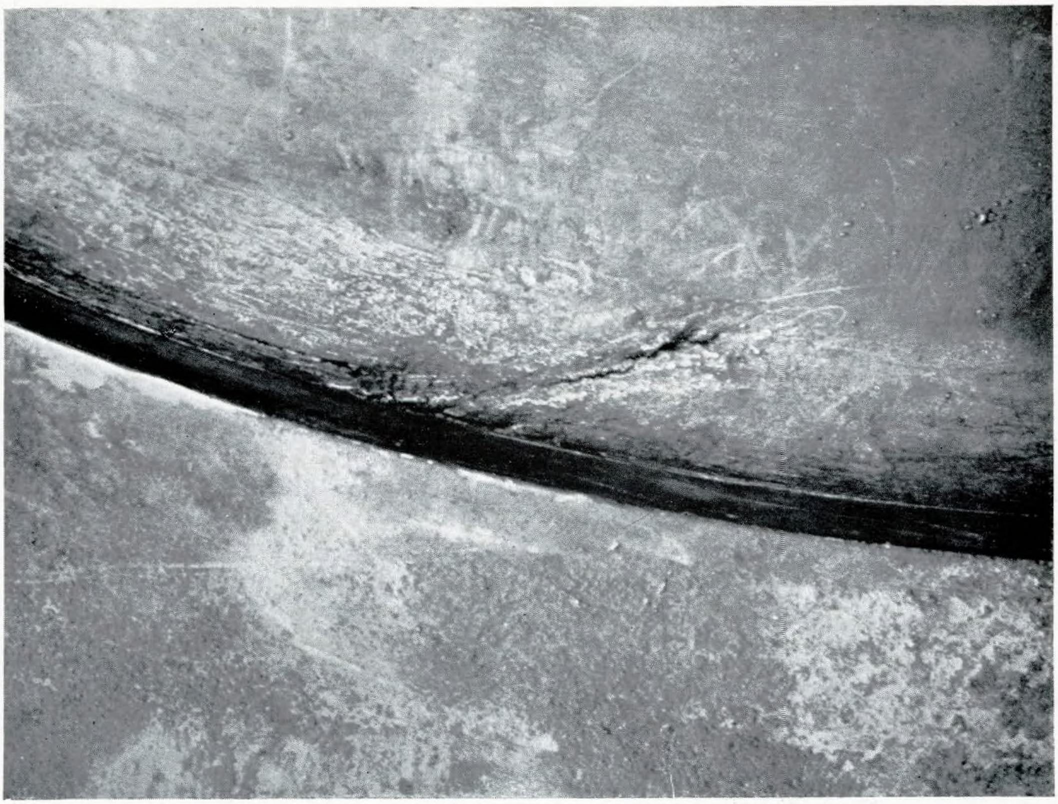


FIG. 9

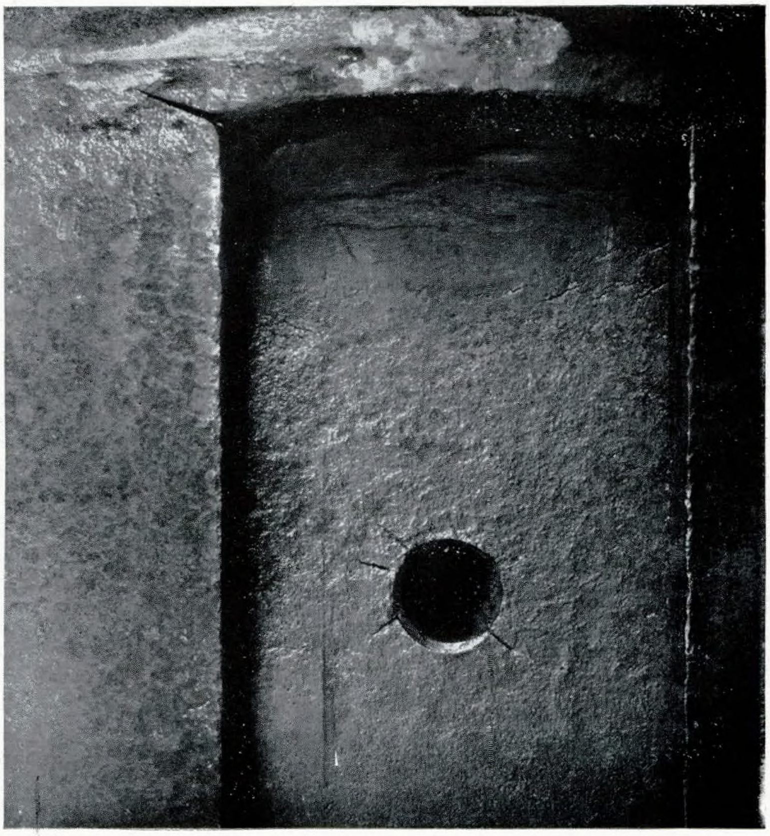


FIG. 8

Typical crack patterns in "Liberty" screwshafts

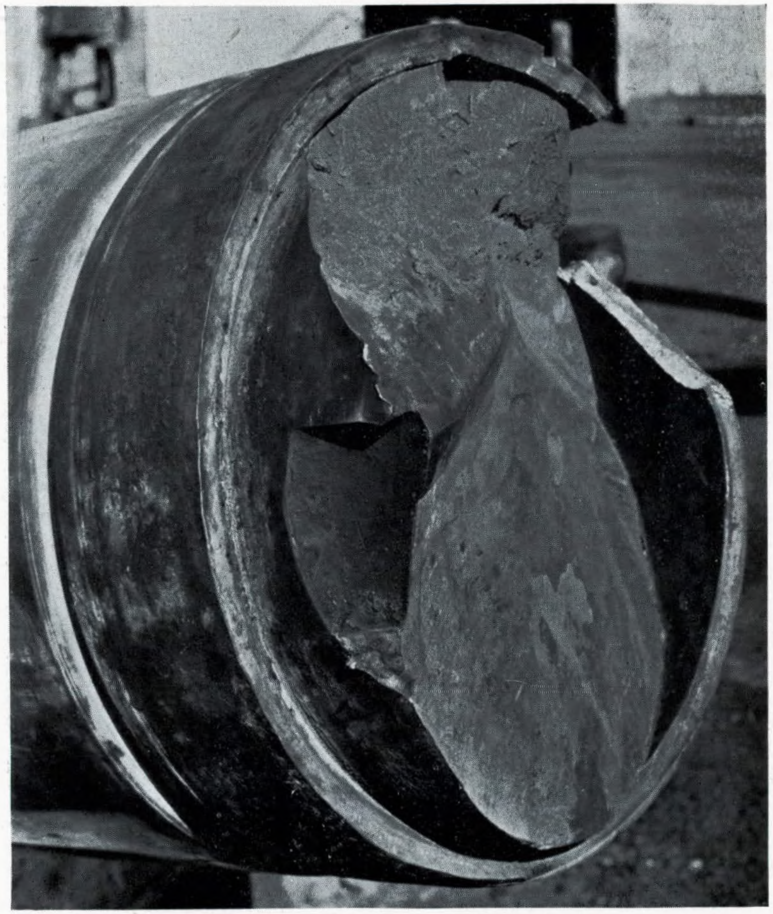


FIG. 10—*Fractured "Liberty" screwshaft (see also Fig. 11 for end view)*

FIG. 11—*Fractured "Liberty" screwshaft (see also Fig. 10 for side view)*

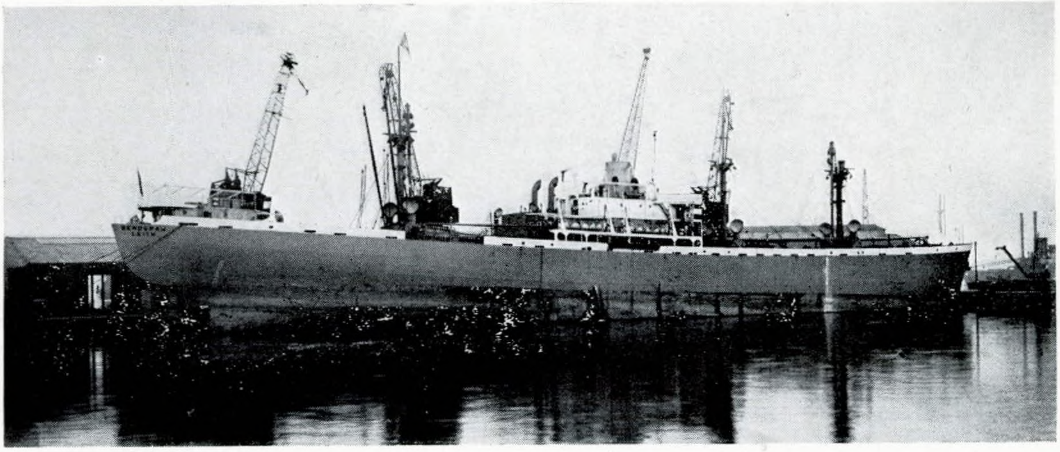


FIG. 18—*S.S.* Bendoran *in trimmed condition*

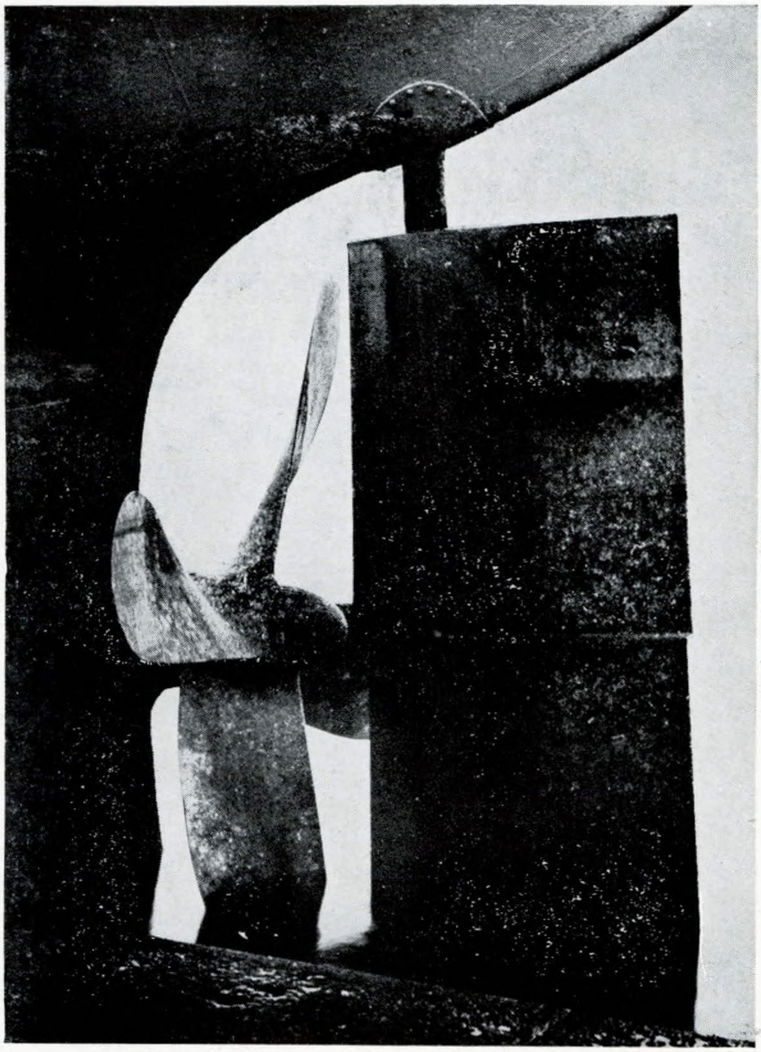


FIG. 19—*S.S.* Bendoran *in trimmed condition; close-up of stern*

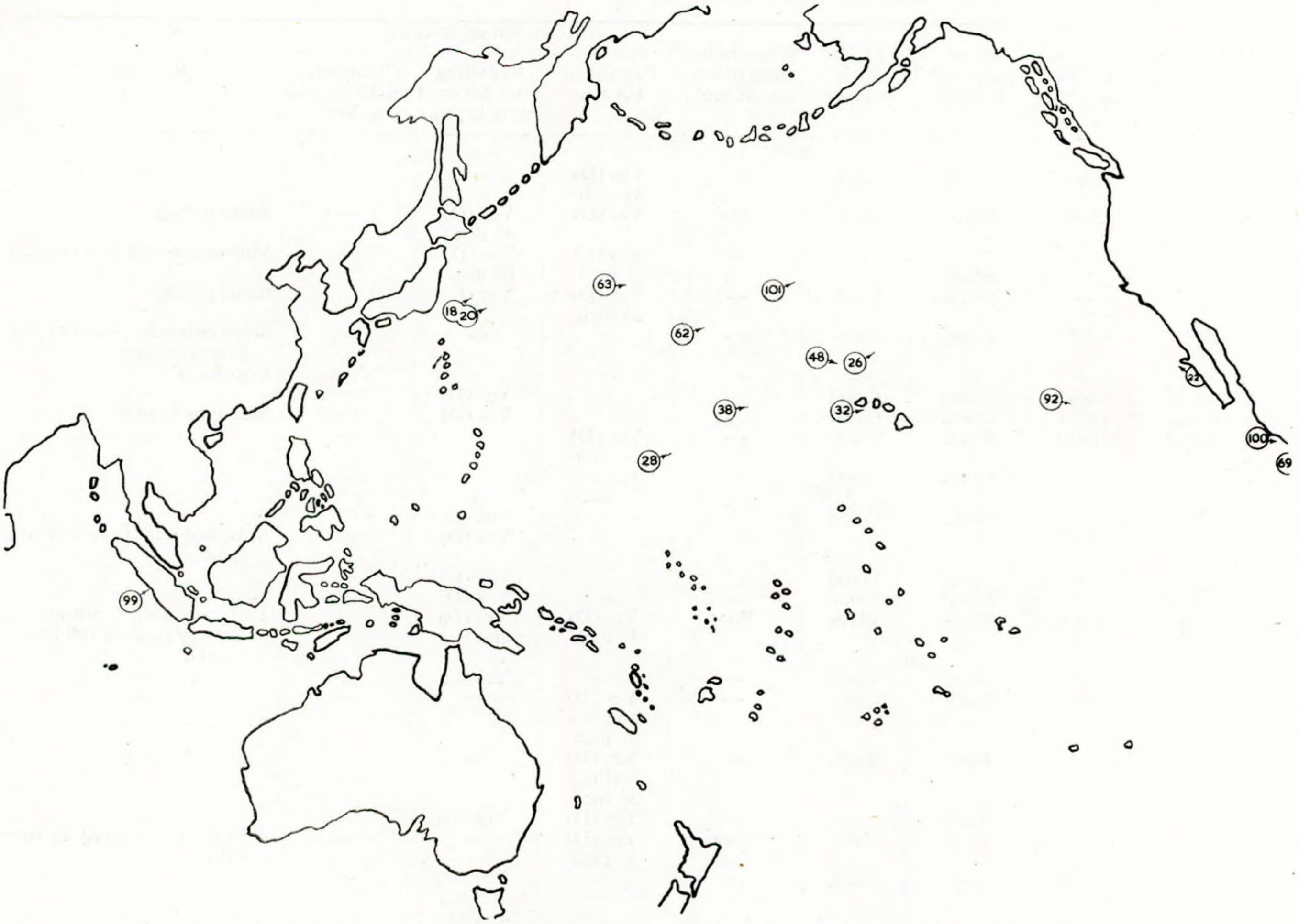


FIG. 4(*b*)—*Geographical location in the Pacific area of "Liberty" ships at time of propeller loss*

"Liberty" failures, there was evidence that the propeller was a slack fit on the taper at the forward end and the key was also a poor fit in the shaft. It will be noted that the cracks run along the root of the keyway (very small fillet radius was provided) for about 1 inch on both sides before extending through the sides of the keyway in a circumferential direction. The fracture probably corresponds to the early stages of that shown in Fig. 6 for the round-ended keyway in the Canadian "Victory" shaft. In the case of Fig. 12 the shaft was four years old and had been drawn for examination only six months previously. There was no indication of corrosive action.

It may be of interest to quote one more description of a "Liberty" type failure as actually reported by a surveyor in August 1948:—". . . on both sides of the keyway and at about

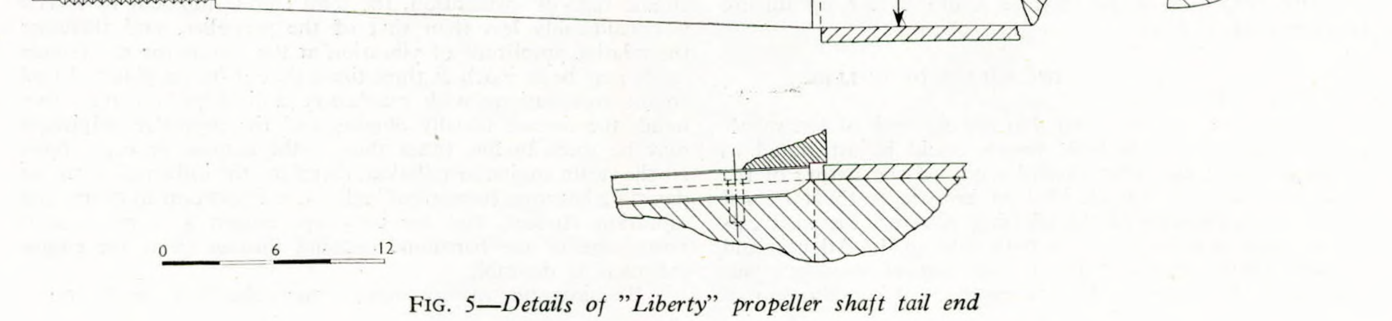


FIG. 5—*Details of "Liberty" propeller shaft tail end*

TABLE II—DETAILS OF TYPICAL SCREWSHAFT FAILURES IN CANADIAN "VICTORY" AND "CANADIAN" TYPE VESSELS

Ship	Condition of sealing ring	Compression of sealing ring	Fit of propeller on taper	Fit of key in keyway	Was a radius given to keyway at root?	Location of cracking			Remarks
						Parallel to keyway	Radiating from forward end of keyway	Circumferentially at end of liner	
A	Good	Slight	Good	Slack	—	Yes (D) 8½ inch	—	—	—
B	Good	Light only	Slack	Slack	No	Yes (D)	Yes (D) 45 deg.	—	Some pitting
C	Good	—	Slack	—	—	Yes (L) 4 inch	Yes (D) 60 deg.	—	Moisture oozed from cracks
D	—	—	Slack	Slack	—	Yes (D) 2½ inch	Yes (D)	—	Some pitting
E*	Bad	Slight	Poor	Poor	—	—	—	Yes	Some corrosion, shaft nicked at end of liner
F	Bad	—	—	—	—	—	—	Yes	Corrosion
G	Good	Good	Good	Good	—	—	Yes (D)	—	—
H	Good	Good	Good	Good	—	—	Yes (D)	—	Sled type keyway
I	Good	Good	Slack	Slack	—	Yes (D) 7 inch	—	—	—
J*	—	—	Good	Good	—	Yes (D) 7 inch	—	—	—
K	—	—	Good	Good	—	—	Yes (L)	—	—
L	Bad	—	—	—	—	—	Yes (D)	—	Scarphed joint in rubber ring failed
M	Good	—	Good	Good	—	—	Yes (D)	—	—
N	Good	—	Good	Good	—	—	Yes (D)	—	—
O	Bad	Light only	Slack	Slack	Poor	Yes (D) 3 inch	Yes (D) and (L)	—	Bad corrosion; bottom of keyway cracked full length (D)
P	Good	—	Good	Good	—	—	Yes (L)	—	—
Q	—	—	Slack	—	—	Yes (D) and (L) 9 inch	—	—	—
R	Good	Nil	Poor	Slack	—	Yes (D) and (L) 3½ inch	—	—	—
S	—	—	Slack	—	—	Yes (D)	Yes (D)	—	—
T	—	—	Slack	Poor	Poor	Yes (D) 8 inch	—	—	Pitted, also cracked in root fillet
U	—	Slight	—	Slack	—	—	Yes (D) and (L) 45 deg.	—	—
V*	—	—	Slack (nut slack also)	Slack 1/32 in.	—	Yes (D) 14 inch at root	—	—	After heavy weather in light condition

*="Canadian" type vessel, Scotch boilers, oil-fired, no deep tanks forward.

Notes.—(1) (D)=driving side of keyway when running ahead.
(L)=leading side of keyway when running ahead.
(2) All keyways of round-ended type, except where stated.

25 mm. from the forward end of the keyway, cracks had started and extended around the circumference of the shaft for about 45 mm., running aft at an angle of about 45 deg. Further, on the starboard side the crack extended downwards for the full depth of the keyway. There were also signs of a second crack on the starboard side at about 25 mm. aft of the main crack . . . There were signs of the key having borne heavily on the forward starboard side of the keyway . . ." This would seem to be a clear case of torsional fatigue loading, aggravated probably by a badly fitted key. That the cracks occurred on both sides of the keyway strongly suggests the action of an alternating tangential stress such as could be set up during heavy torsional oscillation.

INVESTIGATIONS INTO THE CAUSES OF FAILURE

(A) Torsional Vibration

As soon as it was suspected that the outbreak of screwshaft failures in American war-built vessels could be attributed to systematic causes, and after careful study of the nature of the reported fractures, it was decided to investigate the torsional vibration characteristics of the shafting system. Detailed calculations were therefore made on both sides of the Atlantic and agreement was reached that the 1-node natural frequency was of the order of 230 r.p.m. for the most usual combination of manganese bronze propeller and fully built crankshaft (a minority were fitted with cast-iron propellers) (see Table IV). It was thus apparent that the third order critical speed could be expected at about 76 r.p.m., i.e., coincident with the maximum service speed of the engines. Attempts were then made by various methods of calculation to estimate the maximum stress in the screwshaft resulting from operation on this critical at resonance. However, the results were found to vary widely, mainly on account of uncertainty as to the appropriate value of the engine damping factor. Reference to the normal elastic curve (Fig. 13(*b*)) will show that in the steam reciprocating engine type of installation, the total engine moment of inertia is considerably less than that of the propeller, and therefore the relative amplitude of vibration at the engine for the 1-node mode may be as much as three times that at the propeller. In oil engine installations with machinery amidships, on the other hand, the reverse usually obtains and the propeller amplitude may be three to five times that at the engine, or even more. In the steam engine installation, therefore, the influence of engine damping becomes correspondingly more important in estimating vibration stresses, and for the same reason a more accurate knowledge of the harmonic exciting torques from the engine cylinders is desirable.

Torsiograph measurements were therefore made on a

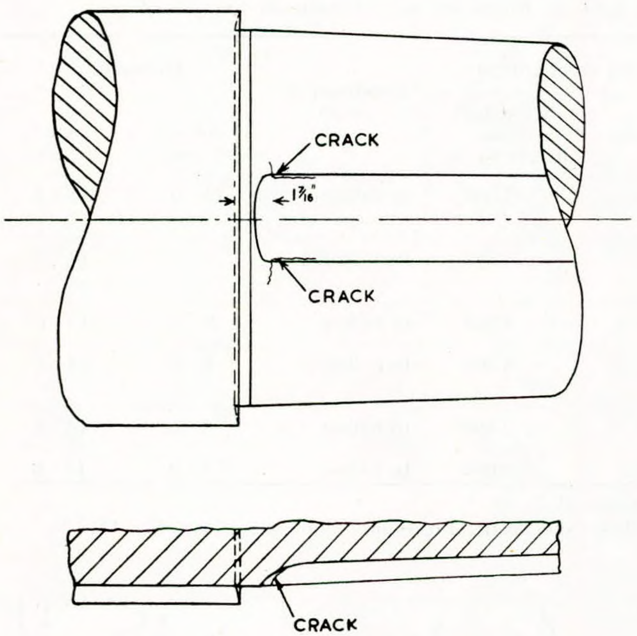


FIG. 12—*Cracked tail shaft, S.S.* Bendoran

number of "Liberty" vessels on both sides of the Atlantic and the recorded results of some of these tests are summarized in Table III. It will be seen that the calculated 1-node natural frequency has been confirmed and the measured stresses range from ± 2,850lb. per sq. in. to ± 4,400lb. per sq. in. It will also be noticed that vessel No. 5, S.S. *Clan Macbeth,* is of the "Ocean" type which, as previously mentioned, have identical engine-shafting-propeller systems to the "Libertys". The measured stress of ± 3,400lb. per sq. in. in this vessel compares with the average of ± 3,450lb. per sq. in. for the five "Libertys", thus confirming the similarity of the two types as regards torsional vibration stresses.

In the case of vessel No. 1 difficulty was experienced in forcing the engines above 77 r.p.m., and as a result some doubt remained as to whether the peak stress had in fact been reached.

This uncertainty was dispelled in vessel No. 2, when a maximum engine speed of 80 r.p.m. was attained, at which the measured stress had fallen to ± 2,600lb. per sq. in. In both these vessels the records were taken by means of a single Geiger torsiograph, belt driven from the intermediate shafting at several points. With this method of non-synchronous recording no account is taken of the fact that as a result of propeller damping there will, in general, be a phase difference between the records taken at any two positions along the shafting which is not exactly 0 deg. or 180 deg. (depending upon whether the two positions are on the same or opposite sides of the theoretical nodal position). Such a phase difference persists at resonance and therefore to derive the true amplitude of twist in the shafting it is necessary to use the measured vector difference of the two amplitudes. (See Appendix I.)

With these considerations in mind, therefore, the stresses recorded by Lloyd's Register in the case of the remaining three vessels in Table III, i.e., Nos. 3, 5, and 6, were all derived from amplitudes measured simultaneously by two phased Geiger instruments driven from two different points on the line shafting (see, for example, Figs. 13 and 14 for vessel No. 6). Further, as a check on the accuracy of this procedure, in the case of vessels Nos. 3 and 5 simultaneous electrical recordings

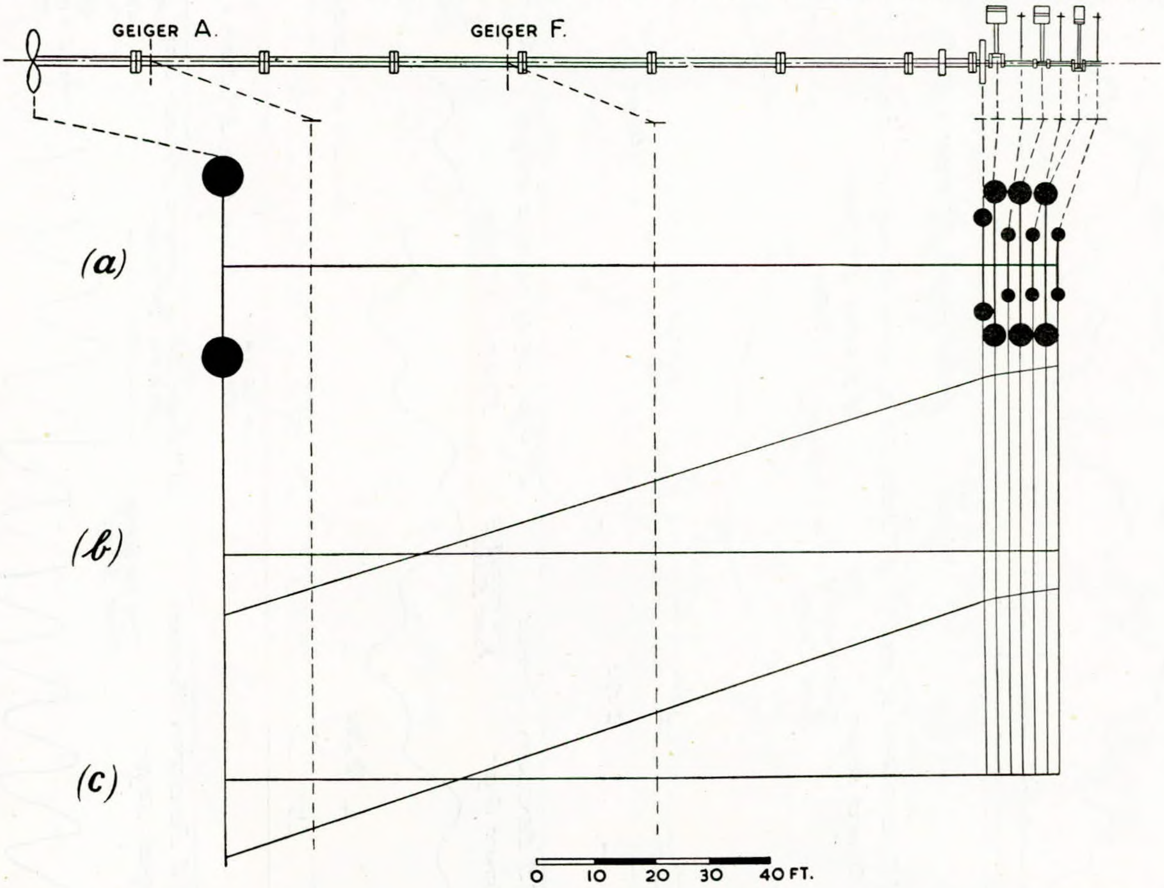


FIG. 13(*a*) (*b*) (*c*)—*Shafting arrangement and normal elastic curves for 1-node mode of torsional vibration, showing location of torsiographs for tests on S.S.* Bendoran

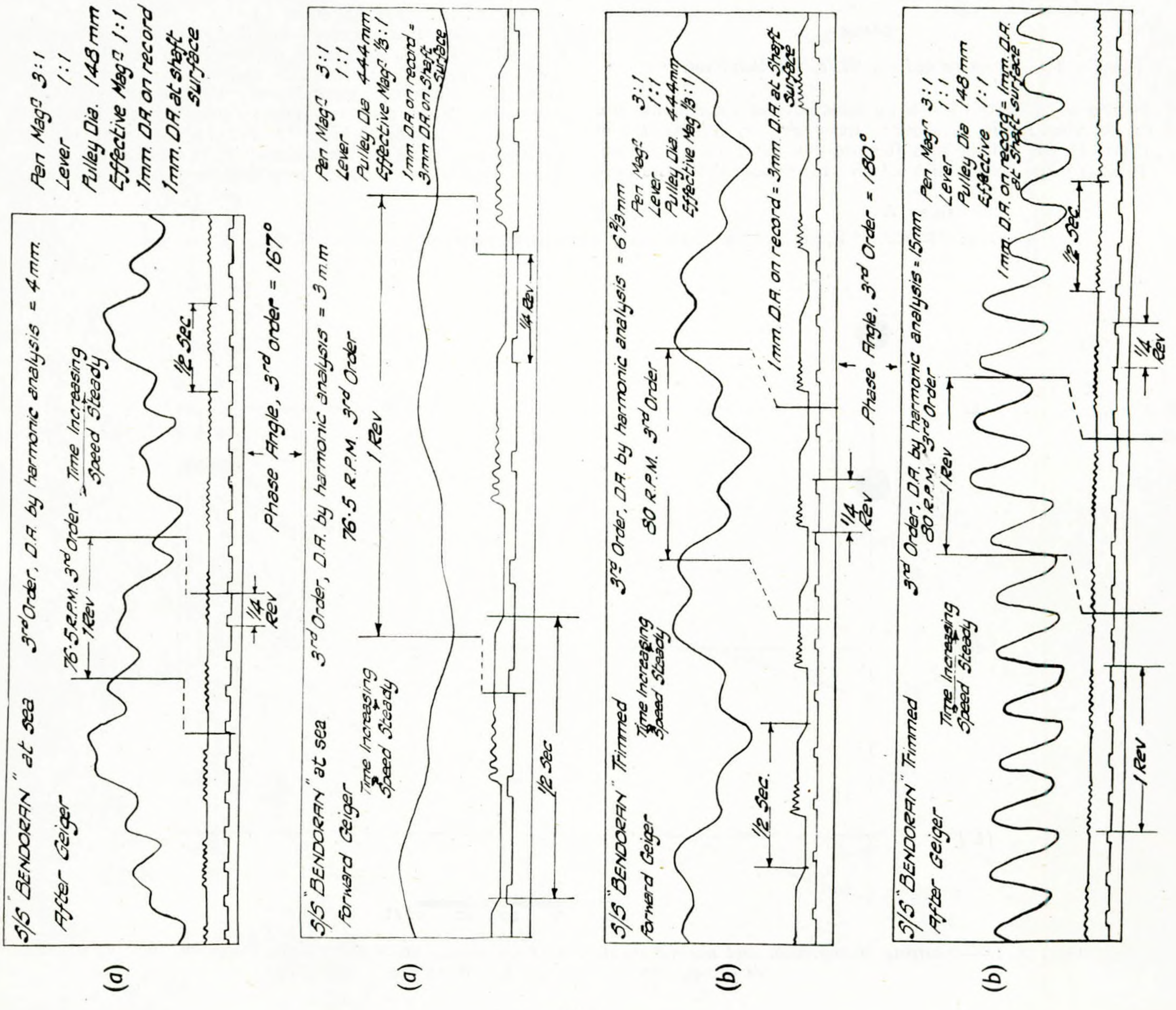
TABLE III—SUMMARY OF 1-NODE, THIRD ORDER STRESSES RECORDED BY TORSIOGRAPH

No.	Name of vessel	Type	Torsiographed by	3rd order critical		Condition of vessel	Draughts	
				r.p.m.	Screwshaft stress, lb. per sq. in.		Forward ft. in.	Aft ft. in.
1	S.S. *Samoa*	"Liberty"	Harland and Wolff	77	± 3,000	In ballast	13 0	15 6
2	S.S. *Benarty* ex *Ammla*	"Liberty"	Lloyd's	76	± 2,850	Part loaded	—	17 3
3	S.S. *Clan Macfadyen**	"Liberty"	Lloyd's	76·5	± 4,000	In ballast	9 0	13 6
4	S.S. *Ira Nelson Morris*†	"Liberty"	G.E.C. (for A.B.S.)	78·5	± 4,400	In ballast	8 0	14 6
5	S.S. *Clan Macbeth**	"Ocean"	Lloyd's	77	± 3,400	In ballast	8 6	13 6
6	S.S. *Bendoran*	"Liberty"	Lloyd's	76·5	± 3,000	In ballast	7 0	13 0

Note.—All stresses derived from Geiger torsiograph records except where stated.

* In these vessels Geiger stresses were confirmed by simultaneous recording from electrical resistance strain gauges (see Fig. 15).

† In this vessel stresses were recorded electrically.



were made by means of an a.c. bridge network of electrical resistance strain gauges secured to the shaft surface in the usual 45 deg. arrangement, the connexions being taken off through special slip rings and brush gear. The output from the bridge, after suitable amplification, was fed into an M.I.T.,* photographically recording oscillograph. Fig. 15 shows a typical example of M.I.T. film record for the 3rd order critical speed on the S.S. *Clan Macfadyen.*

For electrical recording of the relatively low frequencies of torsional oscillation characteristic of marine shafting installations, the a.c. bridge technique is strongly to be recommended if trouble-free operation and accurate results are required. The drawback to the ordinary d.c. excited bridge method is, of course, the difficulty of designing stable amplifiers for such low frequencies.

As a proof of the accuracy of both methods of recording employed, it can be stated that for vessels Nos. 3 and 5 the results showed agreement in stress within 4 per cent and 2 per cent respectively, between the two methods, 24-ordinate harmonic analysis being employed both for M.I.T. and Geiger records. The higher figure is quoted in Table III in each case.

During the trials on the S.S. *Clan Macbeth* on passage from London to Glasgow a south-westerly gale was encountered in the Channel and with the vessel drawing only 13ft. 6in. aft, severe racing occurred. The opportunity was therefore taken to record the torsional stresses in the shafting and the engine speed variations, using both the Geiger torsiograph and the M.I.T. electrical equipment. Fig. 16 shows curves of mean engine speed per revolution and mean transmission stress in screwshaft per revolution, as derived from strain gauge and timing traces. It is significant to observe that even with the h.p. engine expansion link shut well in and with an efficient governor set to cut off steam at about 85 r.p.m., speed surges still occurred between 50 and 112 r.p.m. This is a fundamental limitation of any non-anticipating type of governor, in of the torsional vibration strains in the intermediate shafting

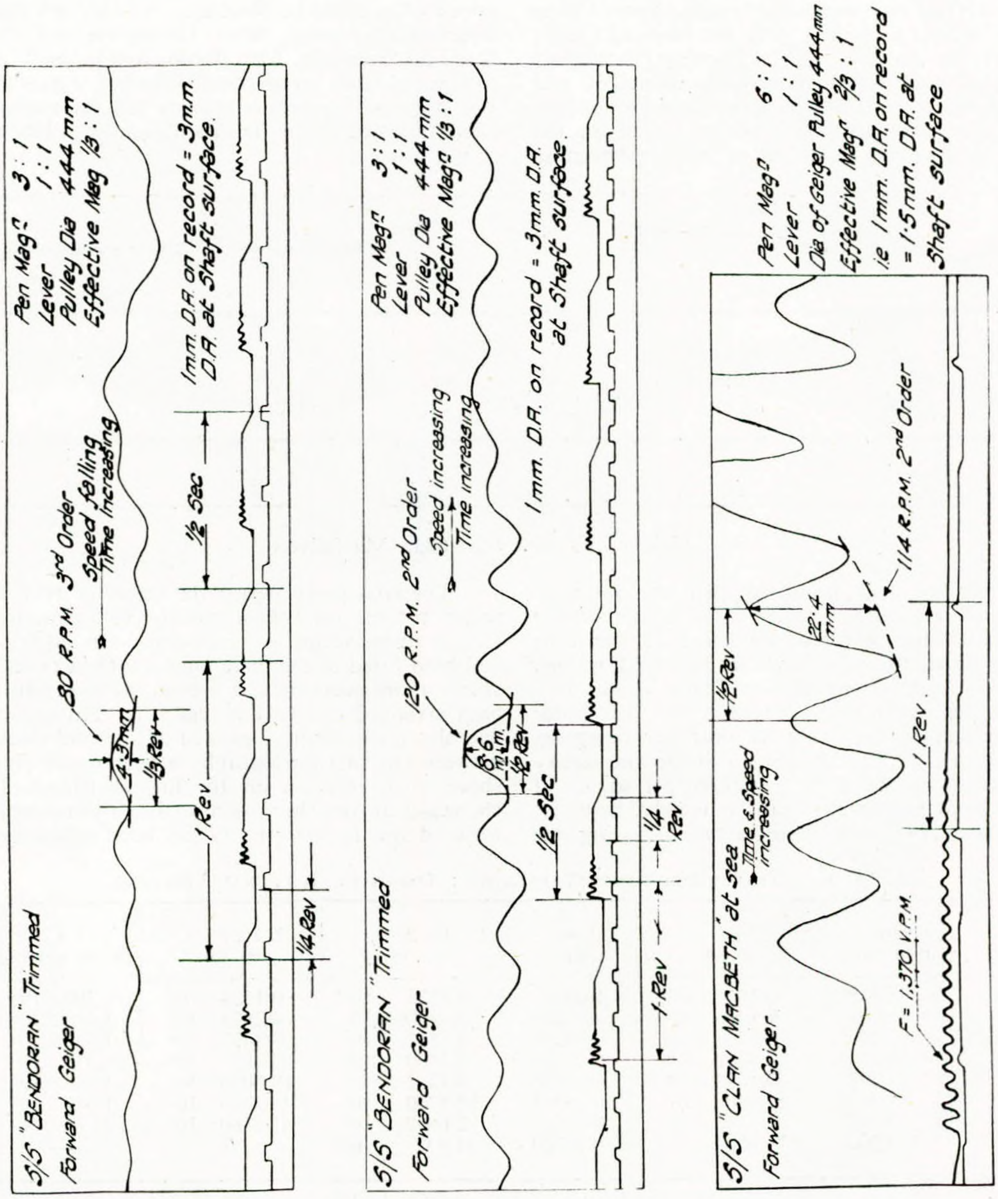


FIG. 14—*Geiger torsiograph records S.S.* Bendoran *and S.S.* Clan Macbeth

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* M.I.T. = Massachusetts Institute of Technology.

TABLE IV—NATURAL FREQUENCY TABULATION: ONE-NODE TORSIONAL VIBRATION

No. of mass	Mass	J/g, lb. inch sec.^{2}	J/g p^2, lb. in.	Θ radians	J/g p^2 Θ, in. lb.	ΣJ/g p^2 Θ, in. lb.	C, lb. in. per rad.	Twist, radians
1	Valve gear	245	$0{\cdot}143 \times 10^6$	1·0000	$0{\cdot}143 \times 10^6$	$0{\cdot}143 \times 10^6$	$1{,}390 \times 10^6$	0·0001
2	h.p. cylinder	14,705	$8{\cdot}5900 \times 10^6$	0·9999	$8{\cdot}590 \times 10^6$	$8{\cdot}733 \times 10^6$	$1{,}330 \times 10^6$	0·00656
3	Valve gear	245	$0{\cdot}1430 \times 10^6$	0·99334	$0{\cdot}142 \times 10^6$	$8{\cdot}875 \times 10^6$	$1{,}330 \times 10^6$	0·00667
4	m.p. cylinder	15,335	$8{\cdot}9500 \times 10^6$	0·98667	$8{\cdot}830 \times 10^6$	$17{\cdot}705 \times 10^6$	$1{,}337 \times 10^6$	0·01325
5	Valve gear	295	$0{\cdot}1722 \times 10^6$	0·97342	$0{\cdot}1678 \times 10^6$	$17{\cdot}8728 \times 10^6$	$1{,}108 \times 10^6$	0·01612
6	l.p. cylinder	18,343	$10{\cdot}7100 \times 10^6$	0·95730	$10{\cdot}2500 \times 10^6$	$28{\cdot}1228 \times 10^6$	$1{,}542 \times 10^6$	0·01821
7	Turning gear	3,557	$2{\cdot}0760 \times 10^6$	0·93909	$1{\cdot}948 \times 10^6$	$30{\cdot}0708 \times 10^6$	$23{\cdot}6 \times 10^6$	1·27300
8	Propeller	154,000	$90{\cdot}000 \times 10^6$	−0·33390	$-30{\cdot}07 \times 10^6$	0	—	—

$F=230{\cdot}6$ v.p.m. $p^2=0{\cdot}000584 \times 10^6$.

S.S. *Bendoran*

Built-up crankshaft. Manganese-bronze propeller, partially immersed.
(Draught aft approximately 13 feet)

so far as the expansive power of the steam remaining in the h.p. receiver is always available to accelerate the engines. Fortunately, the corresponding recorded variations of mean stress in the screwshaft per revolution only ranged from 150 to 1,300lb. per sq. in. Whilst recording with the forward Geiger torsiograph, however, the flank of the 2nd order, 1-node torsional vibration critical was picked up between about 100 and 114 r.p.m., at which latter speed it was approaching resonance (see Fig. 14(E)). The corresponding vibration stress in the screwshaft was calculated at ± 8,200lb. per sq. in. (see Appendix II). Previous calculations had indicated that the resonant speed for this critical, assuming the screw to be completely out of the water (and thus without the usual 25 per cent increase in moment of inertia to allow for entrained water), would be between 118 and 120 r.p.m. (see Table V and Fig. 13(*c*)). Further, it had been estimated that the 2nd order vibration stress at resonance, assuming no propeller damping, would be of the order of ± 15,000lb. per sq. in. in the screwshaft. The measured stress value of ± 8,200lb. per sq. in. at 114 r.p.m. thus appeared reasonable. Unfortunately, however, no speeds higher than 114 r.p.m. were recorded during the trial, and in consequence uncertainty remained as to the maximum value of the 2nd order resonant stress.

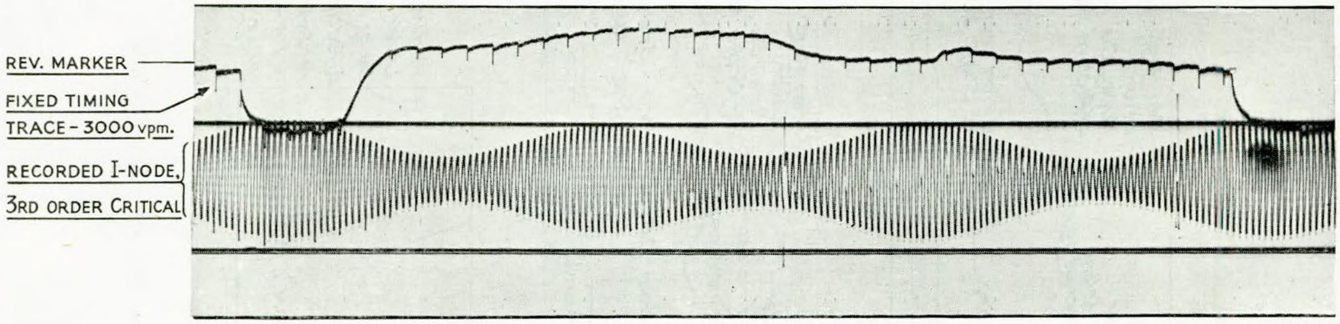

FIG. 15—*M.I.T. film, S.S.* Clan Macfadyen

Various alternative means were considered whereby this information could be obtained, and through the courtesy of the directors of Messrs. Wm. Thompson and Co., managers of Ben Line Steamers, Ltd., Leith, it was finally decided to carry out special trials on another "Liberty" ship, the S.S. *Bendoran.* The proposed procedure was to run the engine light with the vessel trimmed by the head sufficiently to bring the screw clear of the water.

The trial party joined the vessel in Hull and took torsiograph records on ballast passage to Leith, using two phased Geiger instruments as previously. In addition a tachometer had been fitted to the main engine and in order to derive engine speeds more accurately, a special 4-per-revolution contact had been arranged on the line shafting. An accurate timing trace was also provided by means of a ½-second clock. The distance between the two torsiographs was about 60 feet and Fig. 13(*a*) shows their location on the line shafting. The draughts of the vessel during the voyage to Leith were approximately 7 feet forward and 13 feet aft. It had been hoped that heavy weather

TABLE V—NATURAL FREQUENCY TABULATION: ONE-NODE TORSIONAL VIBRATION

No. of mass	Mass	J/g, lb. in. sec.^{2}	J/g p^2, lb. in.	Θ radians	J/g p^2 Θ, in. lb.	Σ J/g p^2 Θ, in. lb.	C, lb. in. per rad.	Twist, radians
1	Valve gear	245	$0{\cdot}1524 \times 10^6$	1·0000	$0{\cdot}1524 \times 10^6$	$0{\cdot}1524 \times 10^6$	$1{,}390 \times 10^6$	0·000109
2	h.p. cylinder	14,705	$9{\cdot}1400 \times 10^6$	0·99989	$9{\cdot}14000 \times 10^6$	$9{\cdot}2924 \times 10^6$	$1{,}330 \times 10^6$	0·006990
3	Valve gear	245	$0{\cdot}1524 \times 10^6$	0·99290	$0{\cdot}1513 \times 10^6$	$9{\cdot}4437 \times 10^6$	$1{,}330 \times 10^6$	0·007110
4	m.p. cylinder	15,335	$9{\cdot}5200 \times 10^6$	0·98580	$9{\cdot}3800 \times 10^6$	$18{\cdot}8237 \times 10^6$	$1{,}337 \times 10^6$	0·014100
5	Valve gear	295	$0{\cdot}1835 \times 10^6$	0·97170	$0{\cdot}1782 \times 10^6$	$19{\cdot}0019 \times 10^6$	$1{,}108 \times 10^6$	0·017150
6	l.p. cylinder	18,343	$11{\cdot}400 \times 10^6$	0·95455	$10{\cdot}8700 \times 10^6$	$29{\cdot}8719 \times 10^6$	$1{,}542 \times 10^6$	0·019380
7	Turning wheel	3,557	$2{\cdot}210 \times 10^6$	0·93517	$2{\cdot}0650 \times 10^6$	$31{\cdot}9369 \times 10^6$	$23{\cdot}6 \times 10^6$	1·35220
8	Propeller	123,200	$76{\cdot}600 \times 10^6$	−0·41700	$-31{\cdot}937 \times 10^6$	0	—	—

$F=238{\cdot}3$ v.p.m. $p^2=0{\cdot}000622 \times 10^6$.

S.S. *Bendoran.*

Propeller inertia excluding entrained water.

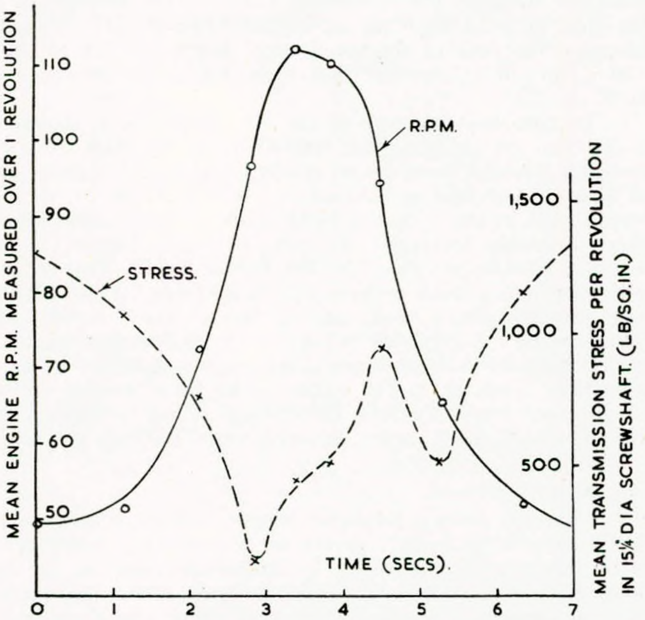

FIG. 16—*Variations of engine speed and mean transmission stress in screwshaft, S.S.* Clan Macbeth *during racing*

records might have been obtained on passage, but unfortunately there was fog with a flat calm sea. It was therefore necessary to fall back on the original proposal to trim the vessel on arrival at Leith. However, the 3rd order critical speed was recorded at sea, giving a stress of ± 3,000lb. per sq. in. in the screwshaft. Fig. 14(A) shows sections of the records at the resonant speed of 76-77 r.p.m. and the calculations are given in Appendix II. The 3rd order resonance curve is plotted in Fig. 17.

Preliminary calculations had indicated that with the vessel fully discharged a sufficient trim by the head could be obtained by emptying the after peak tank and filling the fore peak tank, the forward double bottom tanks, and Nos. 1 and 2 deep tanks, but in addition it would be necessary to flood No. 1 hold with about 600 tons.

Before proceeding with the test, careful consideration was given to the strength of the after bulkhead in No. 1 hold, also to the bending stresses in the hull girder, and to the stability of the vessel in the proposed heavily trimmed condition, bearing in mind the large free surface in No. 1 hold, and the difficulty of ensuring that all other tanks were either completely empty or pressed up. As a final check on the calculations an inclining experiment was carried out during the night before the tests with the vessel moored in Leith docks, the inclining moment being obtained by pumping 15 tons of oil fuel from one engine-room settling tank to the other. The results indicated that the stability in the trimmed condition would be adequate. Accordingly, the trimming operation was proceeded with, the final draughts being 18ft. 5in. forward and 3ft. aft. Under these conditions the propeller blade tips were immersed 1ft. 9in., which it was judged would provide negligible damping to the torsional vibrations.

Fig. 18 shows the vessel in her trimmed condition, and Fig. 19 gives a close-up view of the vessel's stern.

The purpose of the test was simply to measure the value of the 1-node, 2nd order stress in the screwshaft at resonance under comparable conditions to those obtaining during racing at sea. Accordingly, the h.p. engine expansion link was shut in to a minimum and the engine controlled on the hand throttle alone. Despite the short duration of the test, it was considered prudent to open the water service to the stern tubes. The operation of the engine controls during the race was entrusted to one of the Ben Line's senior chief engineers, in whose capable hands the engines were safely accelerated to a maximum speed of 140 r.p.m., as shown by the tachometer. It had been intended not to exceed 125 r.p.m., but owing to the negligible resistance load, accurate speed control was difficult. Fig. 20 shows the engine speed during the race plotted as mean r.p.m. per revolution against time in seconds as derived from the timing traces on the torsiograph records. On the diagram is drawn to the same scale, for direct comparison, a portion of the corresponding graph, Fig. 16, obtained from S.S. *Clan Macbeth* during racing at sea, from which it will be seen that the accelerations in the vicinity of the 2nd order critical, from say 100 r.p.m. upwards, are of the same order in the two cases, the speed rise in the S.S. *Bendoran* being, if anything, somewhat slower.

During the run-up, four 1-node critical speeds were clearly picked up, i.e., the 6th, 4th, 3rd, and 2nd orders, from which the 1-node natural frequency with the propeller substantially free was measured to be approximately 240 r.p.m. It will be noted from Fig. 20 that at 80 r.p.m., corresponding to the 3rd order critical speed, a pause was made for about four seconds during which the vibration built up to a steady value, but that the remaining criticals were passed through fairly rapidly. Fig. 14(B) shows phased sections of the records at the 3rd order resonant speed of 80 r.p.m. and the calculations for stress at resonance are given in Appendix II, from which it will be seen that the maximum value attained was about ± 8,000lb. per sq. in. in the screwshaft. It will be observed that owing to the virtual absence of propeller damping, the phase difference between points on opposite sides of the node is almost 180 deg., and the screwshaft stress is more than doubled, as compared with the measured value of this critical at sea, although the mean indicated pressure in the engine cylinders must have been very little more than the frictional mean pressure, or say, 10-12 per cent of that at full power. Clearly the major source of energy input must have been the 3rd order component of the inertia torques due to the reciprocating masses (see Appendix III). The same critical speed was also picked up on the run-down, but owing to the much greater rapidity with which it was passed through (see Fig. 20), the recorded stress amounted to only ± 4,900lb. per sq. in. as determined from a record taken on Geiger F alone (see Fig. 14(C)).

Fig. 14(D) is a section of the forward Geiger record showing the passage through the 2nd order critical during the run-up. Unfortunately, the after Geiger was hitting the stops during this period, but since the phase difference was 180 deg.,

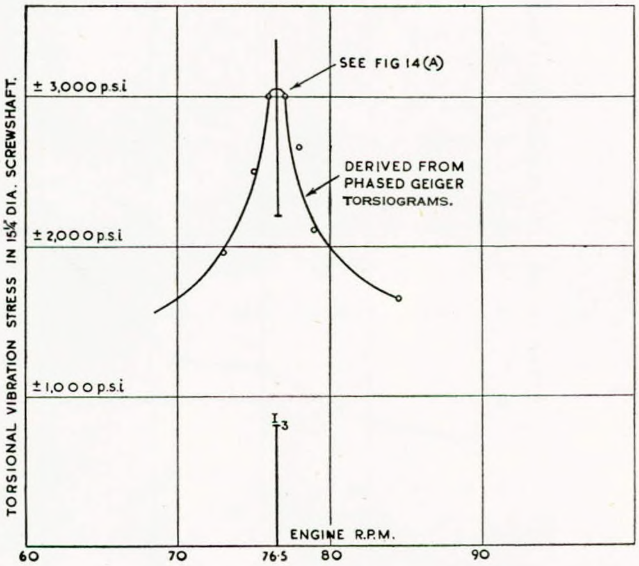

FIG. 17—*Resonance curve; 3rd order, 1-node, S.S.* Bendoran *at sea*

(Vessel's draughts: Forward 7ft., aft 13ft. approx.)

the stress can be readily calculated from the simple formula:—

$$f = \pm \frac{C_F \, \theta_F}{Z\left(1 - \dfrac{C_F}{J_2 p_c^2}\right)} \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots(1)$$

where f = vibration stress at resonance in 15¼ inch diameter screwshaft (± lb. per sq. in.).
Z = sectional modulus of shaft in torsion (inch^3).
θ_F = recorded amplitude of swing at forward Geiger (± radians).
C_F = calculated torsional stiffness of shafting between propeller and Geiger F (lb.-in. per radian twist).
J_2 = mass moment of inertia of dry propeller (lb.-in.-sec.^{2}).
p_c = phase velocity of vibration at resonance (radians per sec.).

Inserting the numerical values and the appropriate maximum value of θ_F from the record (see Appendix II), the maximum value of the 2nd order stress is found to be ± 9,850lb. per sq. in. in the 15¼-inch diameter screwshaft. In this case, owing to the acceleration of the engines and propeller, there was a mean torque at the critical of about 20 per cent of full load torque.

Resonance curves for the various critical speeds are plotted in Fig. 21, but for the 2nd order only the peak value is shown with, for comparison, the stress value of ± 8,200lb. per sq. in. at 114 r.p.m. as measured on the S.S. *Clan Macbeth* during racing at sea. It will be seen that these two points plot fairly well together, particularly bearing in mind that engine damping can vary widely in otherwise identical installations.

From the results of this test it is estimated that under racing conditions at sea the maximum value of the 2nd order vibration stress in the screwshaft may, on the average, be of the order of ± 12,500lb. per sq. in. (see Appendix III). Should, however, the peak of the speed surge happen to fall at about 120 r.p.m., it is possible that even higher stresses may be built up.

To complete the story of the S.S. *Bendoran* it should be added that on changing the screwshaft in dry-dock after the tests, the old shaft was found cracked under the magnetic test. as already described in relation to Fig. 12. Further, as luck would have it, when leaving Leith again in light ballast condition (draughts leaving: 6ft. 9in. forward, 14ft. 7in. aft; arriving Middlesbrough: 8ft. 2in. forward, 14ft. 5in. aft) the vessel ran into a south-easterly gale (wind force 8-9) with rough head seas and heavy swell, causing her to pound heavily with severe racing. Subsequent inquiry of the engineers revealed the interesting fact that the engines, although well linked-in, raced repeatedly from 40 to 140 r.p.m. under hand-throttle control (a governor has since been fitted), and it was estimated that the maximum speed surges occurred about 10 times per watch.

(*B*) *Bending Stresses*

That the known torsional fatigue stresses acting on the screwshafts of "Liberty" vessels are associated to some extent with bending fatigue stress is undeniable, but as to their numerical value, it is only possible to make somewhat speculative calculations at the present time as defined by more or less arbitrary assumptions. It is possible, however, that the obvious mechanical difficulties involved in their measurement by means of strain gauges will soon be overcome, in which case our knowledge of the actual stresses to which screwshafts are exposed in heavy weather will be greatly clarified.

Two calculations have been made to determine the possible

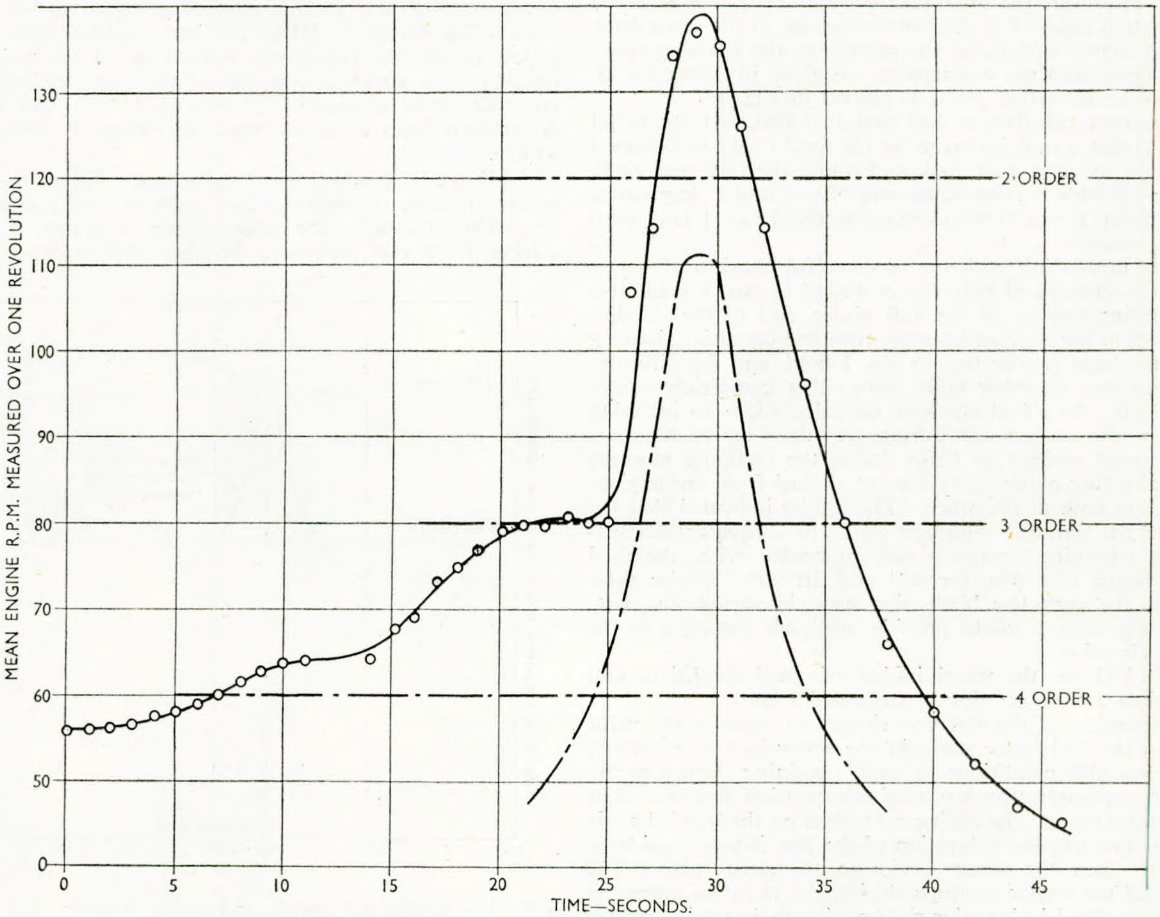

FIG. 20—*Engine speed-time curve for racing test, S.S.* Bendoran

– – – – *Derived from torsiograph records for S.S.* Clan Macbeth *during racing at sea*

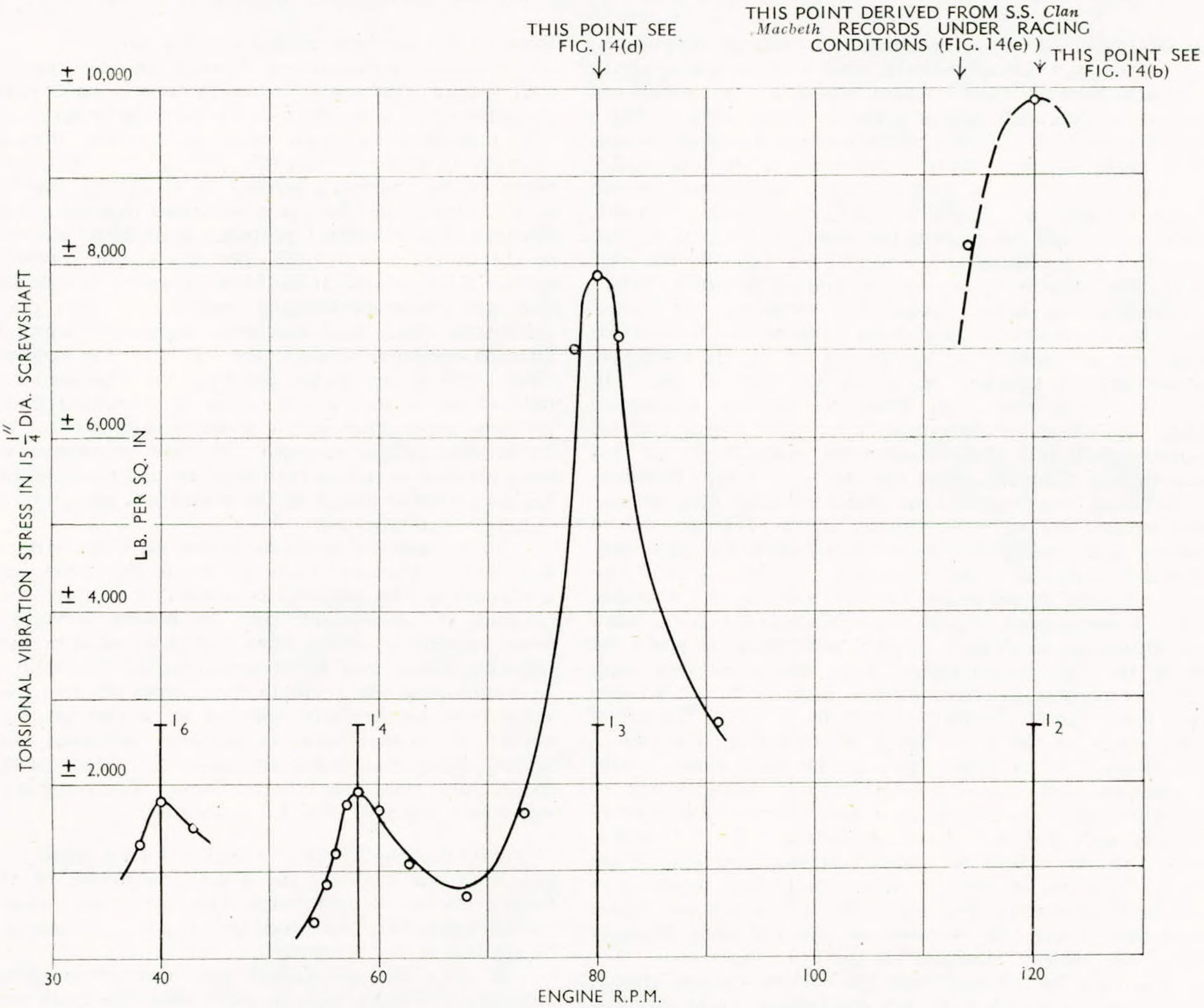

FIG. 21—*Resonance curves, S.S.* Bendoran *in trimmed condition*

(Vessel's draughts: Forward 18ft. 5in., aft 3ft.)

order of magnitude of bending fatigue stresses in the "Liberty" screwshaft, and details are indicated in Appendix IV (*a*) and (*b*).

Calculation (*a*) relates to the ballast condition in smooth water with a draught aft of, say, 14ft. 7in. and a mean draught of 11ft. 5in. (Condition VII, Table VI). In these circumstances, with two blades vertical the upper one will have about ⅔ of its length unloaded, and it is assumed that the reduction in unbalance caused by the inner ⅓ is negligible. The calculated stress value of ± 2,540lb. per sq. in., when added to the bending stress of ± 1,600lb. per sq. in. due to dead weight, gives a total variation of bending fatigue stress, at each of the four 90 deg. angular positions approximately defined by the blade centre lines (see Fig. 11), of ± 4,140lb. per sq. in.

Calculation (*b*) is admittedly more arbitrary and is intended to explore the conditions existing when the ballasted vessel is racing heavily to speeds of upwards of 120 r.p.m. in head seas, which have resulted in the vessel's speed being reduced to, say, 6 knots, probably quite a common condition in the North Atlantic in winter-time. The assumption is made that during re-immersion after a speed surge the revolutions have fallen to 100 r.p.m. by the time the lower blade in its vertical position has its centre of thrust at 80 per cent of tip radius from the shaft axis, also that the effective thrust under these conditions is about one half of that which would be exerted by each blade if the screw were normally submerged at the same slip. It will be noted that the resulting stress value obtained is about ± 8,000lb. per sq. in. It is realized, of course, that in practice the actual thrust would probably be reduced by cavitation effects and by the high degree of aeration of the surface water in the wake under the turbulent conditions envisaged. On the other hand, this reduction may be balanced to some extent by the impactive or blow effect as the blade enters the water at high velocity. However, the object is merely to attempt an approximation under certain specific conditions as an aid to bridging the gap between ignorance and fact.

In both calculations (*a*) and (*b*) use has been made of Troost's results for his series B.4.40, although his boss-diameter ratio was 16 per cent greater giving an 8 per cent smaller D.A.R. Also his blade thickness fraction of 0·045 compares with 0·0405 for the "Liberty" design. In other respects fair similarity exists.

(*C*) *The Strength of the Keyed Propeller Assembly in Fatigue*

In an attempt to assess the strength of the round-ended and sled-runner types of keyway under fatigue loading, it may be convenient at this point to summarize briefly the results of published experimental work.

Peterson* carried out rotating-beam endurance tests on a 36 ton per sq. in. tensile normalized medium carbon steel having an unnotched bending fatigue strength of ± 16½ ton per sq. in., and his results show nominal stress concentration factors of 1·61 and 1·32 for the round-ended and sled-runner type keyways respectively, i.e., a 22 per cent greater reversed bending fatigue strength in the case of the sled-runner type. The diameter of the specimens, however, was only 1 inch and the keyway section ¼ × ⅛ inch. Thus, the results could be

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* Peterson, R. E. 1932. Proc. Am. Soc. Test. Mat., Vol. 32, part 2, p. 413, "Fatigue of Shafts having Keyways."

appreciably affected by very small machining differences; further, it does not follow that the same ratio would apply in the 15¼-inch diameter shaft. Again, the tests were carried out in reversed bending only and it is by no means certain that a similar ratio would be obtained under reversed torsional stresses, or, as is probably more nearly correct for a ship's propeller shaft, under conditions of steady torsion with superimposed combined bending and torsional fatigue loading. Finally, Peterson's results did not include the effect of clamping stresses such as must be produced at the top of the taper by the edge of the propeller boss recess and by the end of the shaft liner.

As regards the latter Thum[*] has presented some comparative data from similar fatigue tests carried out in reversed bending on a soft steel of 27 ton per sq. in. tensile having an unnotched fatigue strength of ± 15½ ton per sq. in. His results agree fairly well with Peterson's for the unclamped condition, i.e., stress concentration factors of 1·7 and 1·25 for the round-ended and sled-runner types respectively, or 36½ per cent higher endurance limit for the latter type. However, when clamping was applied, the plain polished fatigue limit without keyway fell to ± 8¼ ton per sq. in. (53 per cent of unclamped), and the stress concentration factors for the round-ended and sled-runner types amounted to 2·33 and 2·59 respectively, i.e., the round-ended keyway was slightly stronger. However, Thum's work is probably subject to much the same sort of limitations as Peterson's when attempting to apply the results to the full-size propeller shaft, and until large-scale tests prove otherwise, it seems there is little to choose between the two types for ship's propeller shafts. The influence of clamping stress on the other hand, whether from the end of the shrunk-on liner or from the propeller boss recess, merits closer attention and it would seem prudent to endeavour, so far as possible, to avoid the double stress concentration effect of the keyway end and the clamping stresses. It is therefore suggested that the minimum distance between the end of the keyway and the end of the liner might be limited to say, ⅙ of the screwshaft diameter. For the "Liberty" design this would represent only 3 per cent increase in nominal shaft stress at the end of the keyway, allowing for the 1 : 12 taper.

It will have been noted from the various failures already described that in a number of cases the fatigue cracks, originating from the end of the keyway, commenced to propagate at first in a circumferential direction, suggesting at first sight the result of bending stresses. In this connexion, however, it may be well to quote some work carried out by Gough[†] in reversed torsion on 0·4-inch diameter "Armco" iron shafts having round-ended keyways 0·109-inch and 0·055-inch wide by 0·036-inch deep. He reported that for unclamped specimens in every case fracture commenced by a crack occurring at, and developing along, a junction of the bottom and one side of the keyway (root fillets 0·001 inch to 0·005 inch). When the section was reached at which the shaft diameter increased (i.e., where the keyway end met the fillet radius from the grip portion), the crack usually developed in a circumferential direction. Again, when the specimens were clamped with a keyed-on boss, the cracks started just inside the boss axially along both junctions of the bottom and sides of the keyway and finished up by a circumferential type of failure. The stress concentration factors for keyway alone and keyway clamped amounted to 1·14 and 1·5 respectively, i.e., about 24 per cent additional reduction in fatigue strength due to clamping. Similar experiments carried out on a 0·65 per cent carbon steel, without clamping and with or without keyways, gave fractures which in all cases were helicoidal, sometimes double helicoidal.

The above results tend to show that in a relatively soft mild steel, such as a propeller shaft forging, the presence of a circumferential crack does not necessarily imply that bending stresses are always responsible, although it is reasonably certain that where approximately 45 deg. helicoidal fractures occur, torsional stresses have played a major part.

Recent experiments by Dorey[‡] on the strength of mild steel marine shafting in reversed torsion have indicated that the presence of a round-ended keyway, without clamping, in a 10½ inch-diameter shaft may reduce the nominal fatigue strength to about ± 10,000lb. per sq. in. Assuming that the effect of the clamping stresses is to reduce this latter figure by a further 25 per cent, it is estimated that the nominal fatigue strength of a well-fitted propeller shaft keyed assembly may be of the order of ± 7,500lb. per sq. in. for reversed torsional stresses acting alone. It has been shown by Gough and Pollard[§] that when reversed bending stresses act upon plain polished small-scale mild steel specimens subjected simultaneously to reversed torsional stresses, the value of the applied torsional shear stress at the fatigue limit for the combined conditions is reduced below that which could be withstood under reversed torsional shear stress acting alone, a similar effect being found for bending fatigue strength. In these circumstances, it would seem prudent to reduce both bending and torsional fatigue loading on propeller shafts to the minimum, especially where these may act simultaneously.

Under steady conditions in smooth water, with some bending fatigue present, such as when the propeller is partly immersed in the ballasted condition, it is estimated from the evidence so far available that the fatigue strength of a well-fitted propeller assembly, expressed in terms of nominal reversed torsional stress, may be of the order of ± 6,000lb. per sq. in. It should always be borne in mind, however, that this estimated value may be seriously reduced in service on account of a variety of causes, such as extremes of stress concentration arising from machining errors, badly fitted parts, corrosive environment, and previous overstress. The latter is particularly important and will now be considered.

(*D*) *The General Effect of Overstress on Fatigue Strength*

Whatever may be the actual magnitude of the bending fatigue stresses it seems clear that under conditions of severe racing appreciable torsional and bending fatigue loading will be applied to the screwshaft.

It is a well-established fact that although mild overstressing in fatigue may actually raise the endurance limit of mild steel, the reverse applies when the combination of amount and numbers of cycles of previous overstress exceeds certain limiting values. For most materials it is possible to plot on the usual S-N diagram what is known as a "damage line", which expresses the limiting number of cycles of a given overstress which the material can withstand without failure occurring in less than 10^7 cycles when re-tested at the virgin fatigue limit. For many materials this damage line has a form similar to the S-N curve, but with the high stress end pulled down. Some cases have been reported where the damage line almost coincides with the S-N curve, indicating large capacity for absorbing punishment by overstress without damage. In other cases the damage line may be substantially horizontal from which it may be concluded that even the smallest amount of overstress applied for a very few cycles will produce damage. A good example of the latter type of behaviour has been reported by Russell and Welcker[||] for a 0·35 per cent annealed carbon steel having a reversed bending fatigue strength of ± 16 ton per sq. in., although it is stated that some improvement was obtained by suitable heat treatment.

A particularly important series of tests on the effect of overstress in reducing fatigue strength has been carried out by Müller-Stock[¶] and others. The material investigated was a low carbon steel (St. 37) having a reversed bending fatigue strength of ± 13·8 ton per sq. in. The material withstood

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[*] Thum, A. 1935. Trans.N.E.C.Inst. Eng. and Shipbuilders, Vol. 51, p. D.122 (discussion).

[†] Gough, H. J. 1924. Brit. Aero. Research Comm. Reports, Vol. 2, p. 488.

[‡] Dorey, S. F. 1948. Proc. I.Mech.E., Vol. 159 p. 399, "Large Scale Torsional Fatigue Testing of Marine Shafting."

[||] Gough, H. J. and Pollard H. V. 1935. Proc. I.Mech.E., Vol. 131, p. 3, "Strength of Metals under Combined Alternating Stresses."

[§] Russell, H. W. and Welcker, W. A. 1936. Proc. Am. Soc. Test. Materials, Vol. 36, part 2, p. 118.

[¶] Müller-Stock, H., Gerold, E. and Schulz, E. H. 1938. Archiv. f. Eisenhüttenwesen., Vol. 12, p. 141.

TABLE VI

Condition	"Liberty"									"Ocean"			Condition
	I	II	III	IV	V	VI	VII	VIII	IX	I	II	III	
Light weight (tons)	3,380	3,380	3,380	3,380	3,380	3,380	3,380	3,670	3,670	3,248	3,248	3,248	Light weight (tons)
Crew and stores	30	30	30	30	24	24	24	30	30	32	32	26	Stores
Fresh water	56	56	56	56	10	10	10	56	56	54	54	10	Fresh water (No. 4 Tw. Dk.)
No. 1 D.B. tank (O.F.)	132	132	132	132	—	—	—	144 (S.W.)	144 (S.W.)	9	9	—	Fresh water (No. 5 Tw. Dk.)
No. 2 D.B. tank (O.F.)	313	313	—	313	—	—	—	340 (S.W.)	340 (S.W.)	136	136	136	No. 1 D.B. tank (S.W.)
No. 3 D.B. tank (O.F.)	232	232	—	232	—	—	—	234	—	332	332	332	No. 2 D.B. tank (S.W.)
No. 4 D.B. tank (R.F.W.)	132	132	132	132	66	66	66	132	132	158	158	158	No. 3 D.B. tank (S.W.)
No. 5 D.B. tank (O.F.)	232	232	232	232	—	—	—	236	—	109	109	109	No. 4 D.B. tank (S.W.)
No. 6 D.B. tank (O.F.)	108	108	108	108	108	108	108	110	—	114	114	48	No. 6 D.B. tank (R.F.W.)
No. 3 deep tank (O.F.)	700	700	700	160	—	760 (S.W.)	760 (S.W.)	760 (S.W.)	760 (S.W.)	246	246	246	No. 7 D.B. tank (S.W.)
E.R. settling tanks (O.F.)	102	102	102	102	102	102	102	100	100	115	115	115	No. 8 D.B. tank (S.W.)
Fore peak tank (S.W.)	145	145	145	145	145	145	145	145	145	734	—	734	Deep tank abaft E.R. (S.W.)
After peak tank (S.W.)	155	155	155	155	155	155	155	155	155	124	124	124	Fore peak tank (S.W.)
No. 1 deep tank (S.W.)	—	—	—	—	—	—	—	—	—	166	166	166	After peak tank (S.W.)
No. 2 deep tank (S.W.)	—	420	—	—	—	—	420	420	420	955	955	—	No. 3 lower hold (bunkers)
No. 2 hold (dry ballast)	—	—	—	—	—	—	—	360	360	602	602	—	Second deck bunker (coal)
No. 3 hold (dry ballast)	—	—	—	—	—	—	—	615	615	500	500	280	Cross-bunker (coal)
No. 4 hold (dry ballast)	—	—	—	—	—	—	—	525	525	—	—	—	—
Ballast displacement (tons)	5,717	6,137	5,172	5,177	3,990	4,750	5,170	8,032	7,452	7,634	6,900	5,732	Ballast displacement (tons)
	ft. in.	ft. in.	ft. in.	ft. in.	ft. in.	ft. in.	ft. in.	ft. in.	ft. in.	ft. in.	ft. in.	ft. in.	
Mean draught in S.W.	12 5	13 5	11 5	11 5	9 1	10 7	11 5	16 10	15 8	16 0	14 8	12 6	Mean draught in S.W.
Trim by stern	8 10½	4 0½	11 5½	5 10	6 11	11 3½	6 4	8 10½	6 4½	7 9	4 1½	8 8	Trim by stern
Draught forward	7 11½	11 5	5 8	8 6	5 7½	4 11	8 3	12 4½	12 6	12 2	12 7	8 1½	Draught forward
Draught aft	16 10	15 5½	17 1½	14 4	12 6½	16 2½	14 7	21 3	18 10½	19 11	16 8½	16 9½	Draught aft

30,000 cycles of stress at an overstress of ± 20·3 ton per sq. in. It was found that when different batches of test pieces were all previously overstressed by 45 per cent at ± 20 ton per sq. in., but each batch for a different number of cycles, say, 5,000, 10,000, etc., then when re-tested to determine the respective batch fatigue limits, the new S-N curves were all substantially parallel to the virgin curve, but increasingly below it depending upon the numbers of cycles of previous overstress. Thus, at 25,000 cycles of overstress at ±-20 ton per sq. in. the fatigue limit had fallen as low as ± 10 ton per sq. in., or only 72 per cent of the virgin fatigue strength.

It therefore seems clear that where a shaft may be subjected periodically to cycles of comparatively heavy overstress, whether torsional, bending, or combined, the ability of the material to withstand continuously otherwise quite moderate fatigue stresses may well be seriously impaired.

(*E*) *The Ballasting Problem in the "Liberty" Ship*

From an examination of Fig. 4 and Table I there is strong evidence to suggest that the high percentage of screwshaft failures in the "Liberty", Canadian "Victory", and similar oil-burning types may be closely linked with the difficulty of ensuring adequate propeller immersion when in ballast.

Bearing in mind the almost complete immunity of the coal-burning "Ocean" vessels, it was decided to compare the two types by calculating the draughts under a number of different ballast conditions. The results of these calculations are summarized in Table VI and the following notes refer to the various conditions assumed therein:—

"LIBERTY" Type

Condition I.

Full bunkers including No. 3 deep tank.
Forward deep tanks empty.

Condition II.

As for Condition I, but showing effect of adding 420 tons water ballast in No. 2 deep tank forward.

Condition III (end of 6,000 mile round voyage).

As for Condition I, but 545 tons oil fuel burnt from double bottom tanks Nos. 2 and 3 and not replaced by water ballast.

Condition IV (end of 6,000 mile round voyage).

As for Condition I, but 540 tons oil fuel burnt from No. 3 deep tank.

Condition V.

Represents extreme condition at end of very long voyage (say after 60 days steaming at 11 knots). Only No. 6 double bottom and engine-room settling tanks left (210 tons = say 8 days steaming).
One half of reserve feed water used and reduced drinking water and stores.
No water ballast in any oil fuel tanks.

Condition VI.

As for Condition V, but showing effect of filling No. 3 deep tank with water ballast to improve propeller immersion.

Condition VII.

As for Condition VI, but showing effect of filling No. 2 deep tank with water ballast to improve immersion forward.

Condition VIII.

Special scheme of ballasting introduced by U.S.M.C. during war for North Atlantic convoys. Includes 1,500 tons dry ballast in Nos. 2, 3, and 4 holds and 680 tons oil fuel, mainly in double bottom.

Condition IX (end of 6,000 mile round voyage).

As for Condition VIII, but with double bottom fuel tanks Nos. 3, 5, and 6 burnt out, leaving 100 tons in settling tanks only (about 4 days steaming).

"OCEAN" Type

Condition I.

Full bunkers (2,057 tons), including No. 3 lower hold.
Full water ballast, including 734 tons in deep tank abaft engine-room.

Condition II.

As for Condition I, but without deep tank, in order to reduce compressive stress in deck plating in sagging condition.

Condition III (end of long voyage, say 44 days).

Fuel almost burnt out, only 300 tons left in cross-bunker (7½ days steaming).
All water ballast in, including deep tank; otherwise corresponds to "Liberty", Condition VI.
For reasonable immersion forward, however, the comparison should be with "Liberty", Condition VII.

For the "Liberty" type, comparing Conditions I and II, it is seen that with full bunkers the effect of filling No. 2 deep tank forward (420 tons S.W.) is to increase the draught forward by about 3ft. 6in., but reduce propeller immersion by 1ft. 4½in. Thus, in the case of Condition III, at the end of a short voyage of 6,000 miles with draughts 5ft. 8in. forward and 17ft. 1½in. aft, and assuming that for reasonable behaviour in head seas the draught forward should be at least 8ft., then the most that could be expected aft would be about 16ft., obtained by partial filling of forward deep tanks. This would leave 3ft. 9in. of the blade tips breaking surface, or nearly half the effective blade length.

If bunkers were consumed from No. 3 deep tank instead of Nos. 2 and 3 double bottoms, as in Condition IV, propeller immersion would suffer even more, with a draught aft of 14ft. 4in. only. This procedure, fortunately, is unlikely to be followed in practice.

Conditions V, VI, and VII are intended to represent the condition with fuel almost burnt out at the end of a very long voyage (Condition V), and to show the effects of filling water ballast, first in No. 3 deep tank to improve propeller immersion (Condition VI) and, secondly, of adding to this water ballast in No. 2 deep tank to correct draught forward to at least 8ft. (Condition VII). It is significant that under this latter condition a draught aft of only 14ft. 7in. can be expected, leaving 5ft. 2in. of the blade tips breaking surface.

For comparison with this latter condition a similar calculation has been made for the "Ocean" type, Condition III (end of long voyage), from which it will be seen that with about the same draught forward (8ft. 1½in.) more than 2ft. better immersion is obtained aft; in fact, with this class of coal-burner there seems little need to go much below 17ft. aft.

The comparison between the two types would clearly show even more to the advantage of the "Oceans", where, as is probably more common from considerations of the price and availability of steam coal, bunkering is carried out in U.K. ports, often before making a westward North Atlantic passage in ballast, instead of at the loading port as probably has been more usual practice with the "Libertys".

It is well known that the conditions of war service demanded deeper ballasting of tramp type steamers, partly on account of the routing of convoys in higher latitudes of the North Atlantic. As Sir Amos Ayre has stated*, in some cases this was provided in the form of additional dry ballast, but in later vessels of the British wartime programme permanent water

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* Ayre, A. L. 1945. Trans.I.N.A., Vol. 87, p. 8, " Merchant Shipbuilding during the War."

ballast tanks were added of approximately 1,000 tons capacity distributed abreast the tunnel, in the engine-room at the wings, and at each side—port and starboard—of the fore-hold bulkhead, thus ensuring the minimum loss of useful cargo space. In many cases, however, considerable dry ballast was carried in addition to this extra 1,000 tons.

That a similar problem existed with the "Libertys" is evident from the fact that during the war the U.S. Maritime Commission introduced a special scheme of ballasting in which 1,500 tons of dry ballast was carried in Nos. 2, 3, and 4 holds, although this was partly with a view to reducing hull stresses. Conditions VIII and IX, Table VI, show the disposition of the ballast under this scheme, the latter indicating the approximate condition after a North Atlantic return voyage with bunkers previously taken in North America sufficient for a 6,000-mile voyage (plus margin) only. It will be seen that by this means a favourable comparison with the "Ocean" class was obtained. Naturally, under peace-time conditions such a scheme would generally be impracticable, and it is understood few, if any, of these vessels carried this additional ballast over the period under review. The statistics included in Table I lend support to this statement.

One of the difficulties with the "Liberty" type appears to be that the capacity of the forward deep tanks cannot be fully employed to correct for the consumption of fuel forward and provide adequate trim for good steering and avoidance of pounding, without at the same time appreciably reducing the propeller immersion.

It may here be recalled that Sir Amos Ayre† has recommended for this class of vessel as suitable for peace-time service (except in the north-North Atlantic) a mean ballast draught of 16ft. 5½in. when fully bunkered (draught forward 12ft. 4in., aft 20ft. 7in.). This would approximate fairly well to "Liberty", Condition VIII, or "Ocean", Condition I of Table VI. H. Bocler‡ has also made tentative recommendations for minimum mean draughts in ballast based on mean depth allowing for side area of superstructure. From ballast draughts quoted in his paper for typical vessels of 400 to 450ft. in length it would seem that with a small quantity of fuel remaining and propeller 80 per cent immersed a mean draught of about 0·33 of mean depth would be representative of pre-war practice. On this basis for the "Liberty" vessel minimum arrival draughts of 16ft. aft and 9ft. 2in. forward (mean 12ft. 7in.) would seem appropriate, i.e., approximating to "Ocean", Condition III, but with less trim by the stern.

DISCUSSION OF AVAILABLE EVIDENCE

It is known that during the late war the "Liberty" and similar vessels were operated mainly in 9-knot convoys with engine speeds averaging about 60 r.p.m., that is, well below the 3rd order torsional critical at 76 r.p.m. This, together with the special precautions adopted in ballasting, probably protected the machinery against heavy racing. Since the war, however, most vessels have been operating with average engine speeds between 68 and 74 r.p.m., possibly with occasional short bursts of higher speeds. Furthermore, with the virtual abandonment of the war-time practice of carrying additional dry ballast, the arrival draughts of very many of these vessels have left a great deal to be desired (see Tables I and VI). The author has spoken to some masters and engineers, and it seems generally agreed that when in ballast these vessels can race very heavily. Furthermore, a good many of the engines were not originally fitted with governors and with, possibly in some cases, less experienced crews, speed control by hand throttle may not always have been too prompt.

At this stage it may be well to recall that in many instances (see Figs. 10 and 11) the appearance of the fracture has exhibited the well-known "oyster shell" or "tree ring" markings so characteristic of many fatigue failures. It is known that these markings are symptomatic of periods of different rates of propagation of the fatigue crack depending upon the intensity of overstress applied. These are the very conditions which characterize the service of a "Liberty" type screwshaft, i.e., occasional periods of heavy overstress during racing, followed by longer periods of continuous operation at moderate fatigue stresses in the vicinity of the service speed.

The latter stresses, i.e., from ± 3,000 to ± 4,000lb. per sq. in. in torsion, and say ± 4,000lb. per sq. in. in reversed bending, are not considered to constitute by themselves the prime cause of failure. Bearing in mind, however, the disastrously weakening effect of leaky rubber sealing rings and the possible stress concentrations caused by faulty fitting or machining of the keyed assembly, particularly when associated with the seriously damaging effects of previous heavy overstress during racing, it is considered that as a precautionary measure continuous operation in the vicinity of the 3rd order 1-node torsional critical, coincident with the designed maximum service speed of 76 r.p.m., should be avoided (see Appendix V).

This opinion is held despite the good record of the "Ocean" class, which have successfully withstood vibration stresses of equal magnitude at the same service speed, because in the "Liberty" vessels damage may well have been done already to the existing screwshafts by previous overstress during racing. It is important to note that such damage can exist without the actual initiation of cracks.

As regards the serious consequences of racing through the 2nd order critical at 120 r.p.m., with stresses of ± 10,000lb. per sq. in. and upwards in the screwshaft, it has been asked why more cases of shrinkage fit failures in the after L.P. crank webs have not been experienced (American sources quote only nine failures, of which four were associated with loss of propeller at sea), and reference has been made to the results reported by Forsyth* for a similar but smaller installation (without governor) in which the flank of the 2nd order critical was picked up at more than double the normal service speed of 70 r.p.m., giving rise to a stress of at least ± 6,000lb. per sq. in. in the screwshaft superimposed on estimated occasional maximum stresses due to acceleration of about 2,200lb. per sq. in. In this classic case it was stated that failure of the shrinkage fit in the after L.P. crank web occurred on six occasions within 4½ years. The trouble with this vessel was cured by increasing the shrinkage area of the after L.P. crank web and fitting a suitable governor.

As a matter of interest it has been thought well to make calculations on the relative strengths of the shrinkage fits of the after L.P. crank webs in the two installations, neglecting shear strength of the dowel pins (fitted in both installations). For this purpose a shrinkage allowance of 1/600 of the shaft diameter in the eye-hole has been assumed for each case and a value of 0·25 for the coefficient of friction. The latter figure may possibly be conservative, since R. Russell† has shown that for a small (and thus presumably dimensionally accurate) mild steel shrunk assembly with an allowance of only 1/1,500 of the diameter, the value of the coefficient of friction in torsion amounted to 0·29 and 0·9 when assembly was made with a rape oil film and perfectly dry, respectively.

Expressing the torsional strength of the shrunk web assembly at slip in terms of corresponding shear stress in the screwshafts of the two installations, and with the abovementioned assumptions (neglecting the shear strength of the dowel pins), the following results have been calculated, applying the usual Lamé formula for the radial grip pressure:—

Forsyth's vessel—14,000lb. per sq. in. in 12-inch diameter screwshaft.

"Liberty" vessel—23,500lb. per sq. in. in 15¾-inch diameter screwshaft.

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* Op.cit.

† Bocler, H. 1941-2. Trans.N.E.C.Inst.Eng. and Shipbuilders, Vol. 58, p. 241, " The Ballasting of Cargo Ships."

* Forsyth, G. H. 1943. Trans.I.Mar.E., Vol. 55, p. 101, " The importance of governing arrangements for Marine Installations with Special Reference to Torsional Vibration."

† Russell, R. 1933. Proc. I.Mech.E., Vol. 125, p. 493, " Factors affecting the Gripping Force, Shrink and Expansion Bits."

The principal factors affecting these results are the relatively large diameter screwshaft in the former vessel, and the fact that in the "Liberty" crankshaft the diameter of the eyehole in the web is increased by 1 inch over that of the journal and a correspondingly greater axial length provided.

From the foregoing it can be concluded that the almost complete immunity from crankshaft failures in the "Liberty" vessels by no means proves that their screwshafts have not been subjected to severe torsional stresses.

CONCLUSIONS

It is believed that the many failures in the screwshafts of vessels of the "Liberty" and similar types are primarily the result of insufficient propeller immersion in the ballast condition leading to excessive and frequent racing, which in turn has subjected the shaft to heavy vibration stresses, both torsional and bending. These stresses, taken in conjunction with existing stress raisers, have either been sufficient to cause early cracking with resulting rapid failure, or alternatively, have so weakened the shaft that fatigue failure has been brought about more slowly through the alternate action of overstress and the moderate vibration stresses occurring near the service speed. Contributory factors have been the effects of corrosion fatigue from leaky shaft seals, slack-fitting propellers and keys, and unnecessarily severe stress concentrations.

In the light of the foregoing it is difficult to avoid the conclusion that serious consideration should now be given to the provision of additional water ballast capacity in oil-burning ships of the "Liberty" and similar types. Where hull strength is adequate, a practical solution could be obtained by the fitting of additional tanks abreast the shaft tunnel in No. 5 hold and the resulting loss of cargo space could be offset to some extent by reducing the depth of the forward deep tanks, which, in any case, at the present time are largely ineffective in improving the vessel's immersion.

It is believed that where steps are taken to ensure propeller immersions comparable with those usual in the coal-burning "Ocean" class, together with precautions, as indicated, to avoid continuous operation at or near the existing 3rd order critical speed, these measures will confer adequate protection against further screwshaft failures in this class of vessel, particularly where attention has been paid to the many smaller, but nevertheless important, details, of which examples have been given in the paper.

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Lloyd's Shipping List.
Meteorological Office, M.O.1, Air Ministry.

APPENDIX I

THE CHARACTERISTICS OF THE 1-NODE MODE OF DAMPED TORSIONAL VIBRATION OF THE ENGINE-SHAFTING-PROPELLER SYSTEM WITH PARTICULAR REFERENCE TO THE INTERPRETATION OF SEISMIC TYPE TORSIOGRAPH RECORDS

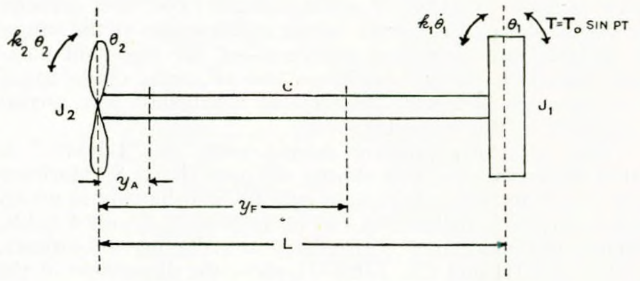

FIG. 22

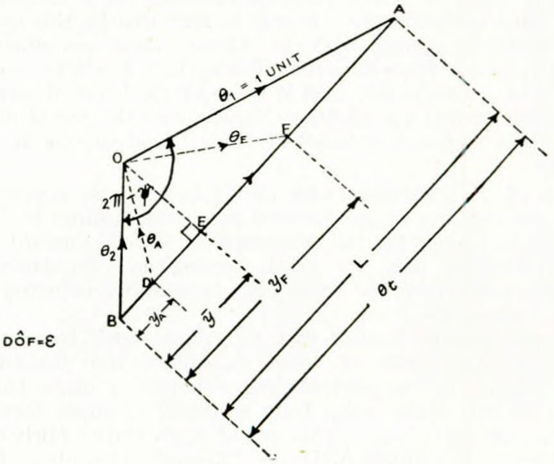

FIG. 23

Assumptions

(1) Engine may be idealized to a single concentrated equivalent mass.
(2) Moment of inertia of shafting is neglected, or allowed for.
(3) Harmonic exciting torque is applied at the engine only, is a major order and of constant amplitude.
(4) Propeller damping taken as proportional to vibration velocity.
(5) Engine damping may be expressed as an equivalent viscous torque proportional to vibration velocity.

Notation (See Figs. 22 and 23)

Subscripts 1 and 2 refer to engine and propeller respectively and c to undamped natural frequency.

Let J = mass moment of inertia (lb.-in.-sec.^{2})
C = torsional stiffness of shafting (lb.-in. per rad.)
θ = absolute amplitude of torsional vibration of masses (± rad.)
θ_t = vibratory twist in shafting (± rad.)
k = damping coefficient (lb.-in. per rad. per sec.)
T = harmonic exciting torque at engine of amplitude T_0 (± lb.-in.)
T_V = vibration torque in shafting (± lb.-in.)
p = phase velocity of vibration (rad. per sec.)
ψ = phase angle between θ_1 and θ_2 (rad.)
ϕ = phase angle between T_V and T (rad.)
t = time (sec.)
L = equivalent length of shafting of any convenient reference diameter corresponding to stiffness, C (inch)

Equations of motion:

$$J_1 \ddot{\theta}_1 + k_1 \dot{\theta}_1 + c\,(\theta_1 - \theta_2) = T_0 \sin p\,t \ldots\ldots\ldots\ldots\ldots (2)$$

$$J_2 \ddot{\theta}_2 + k_2 \dot{\theta}_2 + c\,(\theta_2 - \theta_1) = 0 \ldots\ldots\ldots\ldots\ldots\ldots (3)$$

The particular solutions of these equations can be shown to be given by

$$\theta_1 = \frac{g}{r^2 + s^2 p^2} \{[r\,b + p^2 (s\,f - r)] \sin p\,t - p\,[(b - p^2)\,s - r\,f]\, cos\, p\,t\} \quad \text{..............(4)}$$

and
$$|\theta_1|_{max} = \frac{g \sqrt{[(b - p^2)^2 + f^2 p^2]}}{\sqrt{[r^2 + s^2 p^2]}} \quad \text{............(5)}$$

also
$$\theta_2 = \frac{b\,g}{r^2 + s^2 p^2}(r \sin p\,t - s\,p \cos p\,t) \quad \text{.............(6)}$$

and
$$|\theta_2|_{max} = \frac{b\,g}{\sqrt{[r^2 + s^2 p^2]}} \quad \text{....................(7)}$$

where
$$r = p^2[p^2 - (a + b + e\,f)]$$
$$s = (b\,e + a\,f) - (e + f)\,p^2$$
$$a = c/J_1,\ b = c/J_2,\ e = k_1/J_1,\ f = k_2/J_2,\ g = T_0/J_1$$

Thus,
$$\begin{aligned} T_V &= C\,\theta_t \\ &= C\,(\theta_1 - \theta_2) \\ &= \frac{C\,g\,p \sqrt{[f^2 + p^2]}}{\sqrt{[r^2 + s^2 p^2]}} \sin (p\,t + \phi) \quad \text{......(8)} \end{aligned}$$

where
$$\tan \phi = \frac{r\,f + s\,p^2}{(s\,f - r)\,p} \quad \text{..........................(9)}$$

By substitution,

$$T_V = \frac{C\sqrt{J_2^2 p^2 + k_2^2} \times T_0 \sin (p\,t + \phi)}{\sqrt{[p\,J_1 J_2 (p^2 - p_c^2)]^2 + \{k_1 J_2/J_1 [J_1(p^2 - p_c^2) + C]\}^2 + \{k_2 J_1/J_2 [J_2 (p^2 - p_c^2) + C]\}^2 + k_1 k_2 (k_1 k_2 p^2 + 2\,C^2)}} \quad \text{.........................(10)}$$

and

$$\tan \phi = \frac{p\{k_1 J_1 [J_2^2 (p^2 - p_c^2) + k_2^2] + C\,(J_2^2 k_1 + J_1^2 k_2)\}}{p^2 J_1^2 [J_2^2 (p^2 - p_c^2) + k_2^2] - C\,J_1 k_2 (k_1 + k_2)} \quad \text{.........................(11)}$$

also
$$\tan \psi = -\frac{k_2\,p}{C} \times \frac{1}{\left(1 - \frac{J_2\,p^2}{C}\right)} \quad \text{.............(12)}$$

and
$$\frac{|\theta_1|_{max}}{|\theta_2|_{max}} = \sqrt{\left[\left(p^2 - p_c^2 + \frac{C}{J_1}\right)^2 + \left(\frac{k_2\,p}{J_2}\right)\right]\left(\frac{J_2}{C}\right)} \quad \text{....(13)}$$

It will be noted from Fig. 23 that on account of the phase angle ψ, *which is completely independent of engine damping, there is, in general, no true nodal point in the system,* only a position of minimum amplitude given by the point, E, the foot of the perpendicular from 0 on A B.

It is clear that the line, A B, in Fig. 23, can be taken to represent Fig. 22 to a different scale from that used for the vibration amplitudes. Then it can be shown that the distance, B E, of the point of minimum amplitude, E, from the propeller is found from

$$\bar{y} = \frac{C\,J_2}{k_2^2 + J_2^2 p^2} \times L \quad \text{......................(14)}$$

Also the value of the minimum amplitude at E is given by

$$O\,E = \theta_r = \frac{C\,k_2}{p\,(k_2^2 + J_2^2 p^2)} \times |\theta_t|_{max} \times \cos (p\,t + \phi) \quad \text{.........(15)}$$

which is 90 deg. out-of-phase with the twist amplitude, θ_t.

It will be clear from Fig. 23 that the phase angle, ϵ, between the amplitudes, θ_F and θ_A, at any two points on the line of shafting may vary between zero and ψ.

It will also be seen that the amplitudes, θ_1 and θ_2, can each be regarded as made up of two components, one in phase with the twist amplitude θ_t and the other 90 deg. out-of-phase, the latter being of the same amount for each and as given by equation (15). In other words, the system can be imagined to be oscillating as a rigid mass with amplitude, θ_r, on which is superimposed, 90 deg. out-of-phase, the relative or twist motion, θ_t, between J_1 and J_2.

Special case of resonance

It is sufficiently accurate to assume that resonance occurs at the undamped natural phase velocity, p_c. Thus, putting $p = p_c$ in equations (10), (11), (12), and (13):—

$$T_V = \frac{\sqrt{[J_2^2 p_c^2 + k_2^2]} \times T_0 \sin (p_c\,t + \phi_c)}{\sqrt{\left[\left(k_1 \frac{J_2}{J_1}\right)^2 + \left(k_2 \frac{J_1}{J_2}\right)^2 + \left(\frac{k_1 k_2 p_c}{C}\right)^2 + 2\,k_1 k_2\right]}} \quad \text{.....(16)}$$

$$\tan \phi_c = \frac{\left[\left(\frac{J_2}{k_2}\right)^2 \frac{k_1}{J_1} + \frac{J_1}{k_2} + \frac{k_1}{C}\right] p_c}{\left(\frac{J_1}{J_2} - \frac{k_1}{k_2}\right)} \quad \text{...........(17)}$$

$$\tan \psi_c = \frac{J_1}{J_2} \times \frac{k_2\,p_c}{C} \quad \text{......................(18)}$$

$$\frac{|\theta_1|_{max}}{|\theta_2|_{max}} = \sqrt{\left[\left(\frac{J_2}{J_1}\right) + \left(\frac{k_2\,p_c}{C}\right)^2\right]} = \frac{J_2}{J_1} \sec \psi_c \quad \text{......(19)}$$

From (18) it is clear that ψ_c increases from π when $k_2 = 0$ towards $\frac{3\,\pi}{2}$ as k_2 increases, also as J_2 is reduced relative to J_1. This trend may also be inferred from equation (15), since in general k_2^2 is very small compared with $J_2^2\,p_c^2$.

Thus, the out-of-phase effect becomes increasingly important as the propeller damping factor increases, or as the ratio of propeller to engine moment of inertia is reduced. Hence, it would be expected to be greatest in heavy oil-engine installations, particularly where the critical occurs near the service speed, giving a high propeller torque and therefore large k_2 value.

If θ_1 is measured and the vibration torque derived in the conventional manner, viz.:—

$$T'_V = J_1\,p_c^2\,\theta_1 \quad \text{............................(20)}$$

then from the relation,

$$T_V = C\,\theta_t = C\sqrt{[\theta_1^2 + \theta_2^2 - 2\,\theta_1\,\theta_2 \cos \psi_c]} \quad \text{.............(21)}$$

it can be shown that the error in the derived vibration torque is given by the ratio,

$$\frac{T'_V}{T_V} = \left(1 + \frac{J_1}{J_2}\right)\sqrt{\frac{\left(\frac{J_2}{J_1}\right)^2 + \left(\frac{k_2\,p_c}{C}\right)^2}{\left(1 + \frac{J_2}{J_1}\right)^2 + \left(\frac{k_2\,p_c}{C}\right)^2}} \quad \text{.........(22)}$$

For the S.S. *Bendoran* this error is negligible, being less than + 1 per cent for the 3rd order critical. However, taking a typical oil engine installation with $J_1 = 220{,}000$ units, $J_2 = 58{,}500$ units, $C = 23 \times 10^6$ units, $p_c = 22{\cdot}3$ rads. per sec., $k_2 = 208{,}000$ units for 3rd order critical at 71 r.p.m., then T'_V/T_V from equation (22) would amount to 1·24. Further, if C is reduced to $16{\cdot}4 \times 10^6$ units (say, by lengthening shafting), giving $p_c = 18{\cdot}84$ rads. per sec. and $k_2 = 352{,}000$ units for 1½ order critical at service speed of 120 r.p.m., then the error would be as much as + 73 per cent. The error becomes markedly greater where, say, θ_F is measured at a single point and θ_1 is derived from it using the undamped swinging form. Thus, if in the oil-engined vessel this procedure had been followed at a point situated 0·868 L from the propeller, the derived 3rd order vibration stress would be exactly double the value obtained when taking into account the phase angle, ψ_c.

The necessity for care when interpreting vibration records taken with seismic type torsiographs is thus apparent, more especially where it is impracticable to record at the engine itself. In such cases the following method suggests itself:—

In Fig. 23, draw $O\,A = \theta_1 = 1$ unit of length; calculate ψ_c from equation (18) and $|\theta_2|/|\theta_1|$ from equation (19). It is then possible to draw the triangle A O B. Suppose now an absolute amplitude, θ_F radians, is measured at a single point, F, situated at a distance from the propeller given by $\frac{B\,F}{A\,B} = \frac{y_F}{L}$

Join O F. Then the value of the vibration torque in the line shafting is obtained from:—

$$\begin{aligned} T_V &= C\,\theta_t \\ &= C \times \left(\frac{A\,B}{O\,F}\right) \times \theta_F \quad \text{.......................(23)} \end{aligned}$$

The method has been given for the particular case of resonance, but is quite general and can be applied equally well at non-resonant frequencies.

From the foregoing the following conclusions emerge:—

1. For normal type marine steam reciprocating installations, torsiograph measurement at a single point is usually sufficiently accurate.
2. For heavy marine oil-engine installations serious errors may arise unless the amplitudes and their relative phase are recorded simultaneously at two separate positions. Alternatively, if measurement is made at one position only, these errors may be minimized by using the grapho-analytical method described herein. It is of course realized that its validity hinges to some extent upon the assumption that engine damping can be taken as an equivalent viscous torque, although since the expressions for ψ_c, θ_r, and T'_V/T_V are independent of k_1, any error from this cause should be unimportant in practice.
3. Where reliability and freedom from distortion can be guaranteed, the electrical resistance strain gauge method has the advantage that the twist amplitude is recorded directly.

(2) *Without Propeller Damping*

In this case, as arranged in Fig. 13(*a*), ϵ will be 180 deg.

Thus, $$\bar{\theta} = \theta_F + \theta_A$$

Where only the forward Geiger is used, the equation (1), previously given, may be applied, i.e.,

$$f = \pm \frac{C_F\,\theta_F}{Z\left(1 - \dfrac{C_F}{J_2\,p_c^2}\right)} \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots(1)$$

The values of C_F were calculated as $41{\cdot}5 \times 10^6$ and $31{\cdot}8 \times 10^6$ (lb.-in. per rad.) for the S.S. *Bendoran* and S.S *Clan Macbeth* respectively. Thus, inserting known values in equation (1),

For S.S. Bendoran [Fig. 14, (B), (C) and (D)]:

$$f = \pm\ 130{,}400\ \theta_F \text{ lb. per sq. in.)} \ldots\ldots\ldots\ldots(27)$$

For S.S. Clan Macbeth [Fig. 14(E)]:

$$f = \pm\ 83{,}600\ \theta_F \text{ (lb. per sq. in.)} \ldots\ldots\ldots\ldots(28)$$

Details of the calculations for vibration stress are tabulated below:—

Item	Fig. 14				
	(A)	(B)	(C)	(D)	(E)
Order	3	3	3	2	2
With propeller damping	Yes	No	No	No	No
x_F	3	6·67	4·3	8·6	22·4
M_F	⅓	⅓	⅓	⅓	⅔
θ_F	0·0262	0·0583	0·0376	0·0753	0·098
x_A	4	15	—	—	—
M_A	1	1	—	—	—
θ_A	0·01167	0·04375	—	—	—
cos ϵ	− 0·974	− 1·00	—	—	—
θ	± 0·0377	± 0·102	—	—	—
f [from eqn. (26)]	± 2,960†	± 8,020*	—	—	—
f [from eqn. (27)]	—	± 7,620*	± 4,900	± 9,850	—
f [from eqn. (28)]	—	—	—	—	± 8,200

* Average stress = ± 7,820 lb. per sq. in.

† For comparison, the calculated stress by equation (16), Appendix I, using values of T_0, k_1 and k_2 given in Appendix III(*a*), is ± 3,020 lb. per sq. in. with a phase angle ϕ = 88½ deg. from equation (17), Appendix I.

APPENDIX II

CALCULATIONS FOR DETERMINING TORSIONAL VIBRATION STRESSES FROM TORSIOGRAPH RECORDS (FIG. 14)

Notation

Let subscripts F and A refer to forward and after Geigers respectively.

θ = recorded amplitude of vibration (± rad.).
x = double amplitude on Geiger record (± mm.).
M = effective magnification of Geiger torsiograph.
ϵ = recorded phase angle between θ_F and θ_A.
L_1 = equivalent length of 13½-inch diameter shafting between the Geiger instruments = 702 inch.
d_1 = diameter of shaft driving Geigers = 13·5 inch.
d_2 = diameter of screwshaft = 15·25 inch.
G = modulus of rigidity of shaft material = $11{\cdot}8 \times 10^6$ (lb. per sq. in. per rad.).
f = amplitude of torsional vibration stress in 15¼-inch diameter screwshaft (± lb. per sq. in.).

Then, $$\theta_F = \frac{x_F}{25{\cdot}4\,M_F\,d_1}; \quad \theta_A = \frac{x_A}{25{\cdot}4\,M_A\,d_1}$$

(1) *With Propeller Damping*

In this case there will, in general, be a phase angle ϵ between θ_F and θ_A which is not 0 deg. or 180 deg.

Amplitude of twist on length L_1 inch.

$$\bar{\theta} = \pm\sqrt{\theta_F^2 + \theta_A^2 - 2\,\theta_F\,\theta_A \cos\epsilon} \text{ (rad.)} \ldots\ldots\ldots(24)$$

and $$f = \pm\frac{G\,d_1\,\bar{\theta}}{2\,L} \times \left(\frac{d_1}{d_2}\right)^3 \ldots\ldots\ldots\ldots\ldots\ldots(25)$$

Substituting known values in equation (25),

$$f = \pm\ 78{,}600\ \bar{\theta} \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots(26)$$

APPENDIX III

GENERAL CALCULATIONS ON TORSIONAL VIBRATION

(*a*) *Estimation of Engine Damping Factor*

For this purpose the 3rd order 1-node stress measured on the S.S. *Bendoran* at sea is used as a basis, in conjunction with harmonic analysis of indicated torque diagrams and derivation of mean propeller torque and damping factor from Troost's B.4 Series.

(*i*) *Propeller Damping*

Let K_2 = propeller damping coefficient (lb.-ft. per rad. per sec.).
ω = mean angular velocity of propeller (radians per sec.).
Q = mean propeller torque (lb.-ft.).
D = propeller diameter (ft.).
N = propeller r.p.m.
n = propeller r.p.s.
V_K = propeller speed of advance (knots).
V_A = propeller speed of advance (ft. per min.).
v = propeller speed of advance (ft. per sec.).
ρ = density of sea-water (lb. mass per cu. ft.) = 1·99.

$$\lambda = \frac{V_K \times 101\tfrac{1}{3}}{N\,D} = \frac{V_A}{N\,D} \doteq \frac{v}{n\,D}$$

C_2 = propeller torque coefficient.

$$= \frac{Q}{\rho\,D^5\,n^2}$$

In Troost's diagrams 10 C_2 is plotted against λ for a series of constant pitch ratios, i.e., 10 $C_2 = f(\lambda)$.

Then, $$Q = \frac{\rho}{10}\,D^5\,n^2\,f(\lambda) \ldots\ldots\ldots\ldots\ldots\ldots(29)$$

$$\text{and } K_2 = \frac{\partial Q}{\partial \omega_{(v)}} = \frac{\partial n}{\partial \omega} \times \frac{\partial Q}{\partial n_{(v)}}$$

$$= \frac{1}{2\pi} \times \frac{\rho D^5}{10} \times \frac{\partial}{\partial n_{(v)}} [n^2 f(\lambda)]$$

$$= \frac{\rho D^5 n}{20\pi} [2 f(\lambda) - \lambda f'(\lambda)]$$

$$= 9{\cdot}55 \frac{Q}{N} \left[2 - \frac{\lambda f'(\lambda)}{f(\lambda)} \right] \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots(30)$$

Alternatively, it may be preferred to use the relation

$$K_2 = \frac{\rho D^5 N}{1{,}200\,\pi} [2 f(\lambda) - \lambda f'(\lambda)]$$

$$= 5{\cdot}28 \times 10^{-4} \times D^5 N [2 f(\lambda) - \lambda f'(\lambda)] \ldots\ldots\ldots\ldots\ldots(31)$$

These relations are convenient for obtaining the propeller damping coefficient direct from diagrams such as used by Troost. All that is required for a given value of λ is the corresponding value of the coefficient, $10\,C_2$, and the slope of the appropriate $10\,C_2$ curve of constant pitch ratio relative to the λ-axis. It is then possible to plot the non-dimensional quantities.

$$\left[2 - \frac{\lambda f'(\lambda)}{f(\lambda)} \right] \quad \text{or} \quad [2 f(\lambda) - \lambda f'(\lambda)]$$

to a base of useful λ values for a series of common pitch ratios.

For the ballast condition of S.S. *Bendoran,* take ship speed as 12¼ knots at 76·5 r.p.m., also Froude wake friction as, say, 0·365.

$$\text{Therefore} \qquad V_K = \frac{12{\cdot}25}{1{\cdot}365} = 8{\cdot}97 \text{ knots}$$

$$\text{Therefore} \qquad \lambda = \frac{8{\cdot}97 \times 101\tfrac{1}{3}}{76{\cdot}5 \times 18{\cdot}5} = 0{\cdot}642$$

For this value of λ and a pitch ratio of 0·865 the values of $f(\lambda)$ and $f'(\lambda)$, measured from the B.4 Series curves, are:

$$f(\lambda) = 10\,C_2 = 0{\cdot}228$$

$$f'(\lambda) = -\,0{\cdot}411$$

Thus, from equation (31),

$$K_2 = 63{,}000\text{lb.-ft. per rad. per sec.}$$

Also, from equation (29),

$$Q = 160{,}000\text{lb.-ft.}$$

Hence, from equation (30),

$$K_2 = 30{\cdot}1 \frac{Q}{N}$$

$$\text{d.h.p.} = \frac{160{,}000 \times 2\pi \times 76{\cdot}5}{33{,}000} = 2{,}330$$

$$\text{Assume } \frac{\text{d.h.p.}}{\text{i.h.p.}} = 0{\cdot}88 \times 0{\cdot}985 = 0{\cdot}867 \text{ (say)}$$

Therefore i.h.p. = 2,330/0·867 = 2,680

Therefore frictional horsepower = 2,680 − 2,330 = 350

However, on account of poor immersion of propeller, probably only about ⅝ of disk is effective; thus estimated actual

$$\text{d.h.p.} = 0{\cdot}625 \times 2{,}330 = 1{,}456$$

$$\text{and} \qquad \text{i.h.p.} = 1{,}456 + 350 = 1{,}806$$

It will also be assumed that the propeller damping coefficient is reduced in the same ratio as the mean torque, i.e.,

$$k_2 = 0{\cdot}625 \times 63{,}000 \times 12$$

$$= 472{,}000\text{lb.-in. per rad. per sec.}$$

(ii) *Energy Input from Cylinders*

Harmonic analysis of typical indicator cards, giving a total i.h.p. of 2,150 at 70·5 r.p.m., yielded the following values of the 3rd order sin and cos components of the steam torque harmonics, in units of 1,000lb.:—

High pressure		Medium pressure		Low pressure	
Cos	Sin	Cos	Sin	Cos	Sin
− 61·6	+ 65·3	− 61·9	+ 45·8	− 55·6	+ 79·1

The weights of the reciprocating masses of the cylinder lines were taken as follows:—

High pressure ...	4,000lb.
Medium pressure	4,850lb.
Low pressure ...	8,330lb. (includes 1,480lb. for pumps, etc.)

$$\text{Thus, with a ratio } \frac{\text{connecting rod centres}}{\text{crank throw}} = 4$$

the values of the harmonic torques due to the acceleration of these masses have been calculated as under, in units of 1,000lb.:—

Order	r.p.m.	High pressure		Medium pressure		Low pressure	
		Cos	Sin	Cos	Sin	Cos	Sin
2	120	0	−469	0	−568·5	0	−977·5
3	76·5	0	−73·8	0	−89·4	0	−153·7
3	80	0	−80·7	0	−97·8	0	−168·5

It was considered reasonable to reduce the 3rd order steam harmonics in the ratio of the referred mean indicated pressures in the two conditions, i.e., by the factor

$$\frac{1{,}806}{2{,}150} \times \frac{70{\cdot}5}{76{\cdot}5} = 0{\cdot}774$$

Thus, estimated resultant 3rd order harmonic torques at 76·5 r.p.m. on S.S. *Bendoran* due to steam pressure and inertia forces are as shown in the following table, in units of 1,000lb.:—

Cylinder	3rd order cos	3rd order sin
High pressure	− 47·7	+ 50·6 − 73·8 = − 23·2
Medium pressure	− 47·9	+ 35·4 − 89·4 = − 54·0
Low pressure	− 43·1	+ 61·3 − 153·7 = − 92·4

Let θ = relative amplitude of vibration of a given mass for ± 1 rad. at No. 1 mass (see Table IV), taking subscripts H, M, L for h.p., m.p., and l.p.

T = resultant 3rd order harmonic torque for each cylinder (lb.-in.).

T_3 = resultant 3rd order harmonic torque for whole engine (lb.-in.).

f = measured 3rd order vibration stress in screwshaft (± lb. per sq. in.) = ± 3,000.

From Table IV,

$$\theta_H = 0{\cdot}9999,\ \theta_M = 0{\cdot}9867,\ \theta_L = 0{\cdot}9573$$

$$\text{Therefore} \quad T_H\,\theta_H = (-\,47{\cdot}7 \cos 3\,\omega t - 23{\cdot}2 \sin 3\,\omega t) \times 10^3$$

$$T_M\,\theta_M = (-\,47{\cdot}3 \cos 3\,\omega t - 53{\cdot}3 \sin 3\,\omega t) \times 10^3$$

$$T_L\,\theta_L = (-\,41{\cdot}3 \cos 3\,\omega t - 88{\cdot}5 \sin 3\,\omega t) \times 10^3$$

Since cranks are at 120 deg., the 3rd order vectors coincide in phase, and it is only necessary to add the above values algebraically; thus

$$\Sigma (T\,\theta) = (-\,136{\cdot}3 \cos 3\,\omega t - 165{\cdot}0 \sin 3\,\omega t) \times 10^3$$

$$= -\,214{,}000 \sin (3\,\omega t + \beta)$$

Also mean engine relative amplitude = 0·9813 rad.

Therefore $$T_3 = \frac{\Sigma\,(T\,\theta)}{0{\cdot}9813}$$
$$= 218{,}000\text{lb.-in. (amplitude)}$$

(iii) Applying the energy equation,

$$\pi\,T_3\,\theta_1 = \pi\,k_1\,p_c\,\theta_1^2 + \pi\,k_2\,p_c\,\theta_2^2$$

from which it follows that

$$k_1 = \frac{J_1\,p_c\,T_3}{Z\,f} - \left(\frac{J_1}{J_2}\right)^2 k_2 \ldots\ldots\ldots\ldots (32)$$

$$= \frac{52{,}725 \times 24{\cdot}1 \times 218{,}000}{694 \times 3{,}000} - \left(\frac{52{,}725}{154{,}000}\right) \times 472{,}000$$

$$= 133{,}300 - 55{,}300$$

$$= 78{,}000\text{lb.-in. per rad. per sec.}$$

Assuming that the input energy and propeller damping energy for the vessels Nos. 3, 5 and 6 of Table III will not vary appreciably, then, based on these three vessels only, the average value of k_1 would amount to 61,700lb.-in. per rad. per sec.

(*b*) *Calculation of 2nd and 3rd Order Stresses during Racing Test on S.S.* Bendoran

Having regard to the light load on the engine and the virtual absence of propeller damping, it is assumed that the energy input is derived entirely from inertia forces of reciprocating masses and energy is dissipated in engine damping only.

The vector diagram for the 2nd order is a 120 deg. star, and, taking the values of the harmonic inertia torques as already calculated at 120 r.p.m., the resultant 2nd order torque vector, T_2, at mean engine amplitude is determined as ± 433,000lb.-in., allowing for relative amplitudes given in Table V.

From equation (32), or equation (16), Appendix I, putting $k_2 = 0$, the 2nd order stress

$$f_2 = \pm\frac{J_1\,p_c\,T_2}{Z\,k_1}$$

$$= \pm\frac{52{,}725 \times 24{\cdot}9 \times 433{,}000}{694 \times 78{,}000}$$

$$= \pm\,10{,}500\text{lb. per sq. in.}$$

This compares reasonably closely with the value of ± 9,850lb. per sq. in. measured from the torsiograph record [Fig. 14 (D)], remembering that the passage through the critical was fairly rapid.

An estimate of the value of the 2nd order stress during racing at sea, for average installations, might thus be:

$$\pm\,9{,}850 \times \frac{78{,}000}{61{,}700} = \pm\,12{,}400\text{lb. per sq. in.}$$

Making a similar calculation for the 3rd order stress during the run-up, the value of the equivalent resultant 3rd order inertia torque harmonic is found to be ± 345,000lb.-in.

Thus, 3rd order stress during run-up

$$f_3 = \pm\frac{52{,}725 \times 24{\cdot}9 \times 345{,}000}{694 \times 78{,}000}$$

$$= \pm\,8{,}350\text{lb. per sq. in.}$$

Again this is in fair agreement with the value of approximately ± 8,000lb. per sq. in. derived from the torsiograph records, Fig. 14 (b).

From the foregoing it is suggested that for continuous operation a suitable average value of the engine damping factor (assuming viscous type damping) for this class of steam reciprocating installation would be approximately 0·48lb.-in. per rad. per sec. per square inch of total piston area per inch of crank throw, although it may well vary by as much as ± 30 per cent even in apparently identical installations.

APPENDIX IV

CALCULATIONS ON BENDING FATIGUE

(*a*) *Ballasted, Smooth Water*

Assumed Data:

Draught aft	14ft. 7in.
Draught, mean	11ft. 5in.
Corresponding displacement	5,170 tons
Speed of ship, V_s	11·5 knots
Engine r.p.m., N	72
Froude wake fraction	0·365
Pitch ratio, $P/D = \frac{16}{18{\cdot}5}$	0·865
D.A.R.	0·434
Modulus of shaft section in bending ...	347 inch^3
Weight of propeller	10 tons

$$V_K = \frac{11{\cdot}5}{1{\cdot}365} = 8{\cdot}42\text{ knots}$$

$$\frac{V_A}{N\,D} = \frac{8{\cdot}42 \times 101\tfrac{1}{3}}{72 \times 18{\cdot}5} = 0{\cdot}64$$

Interpolating for D.A.R. 0·434 between Troost's B.4.40 and B.4.55 series, the thrust coefficient $C_1 = 0{\cdot}146$. Thus, total thrust,

$$T = \rho\,D^4\,N^2\,C_1$$

$$= 1{\cdot}99 \times (18{\cdot}5)^4 \times \left(\frac{72}{60}\right)^2 \times 0{\cdot}146$$

$$= 49{,}000\text{lb.}$$

Assume that thrust due to one blade is ¼ of total thrust if propeller were fully immersed, also that centre of thrust is at, say, 0·65 of tip radius, i.e., at 0·65 × 111 = 72 inch.

Therefore bending moment in vertical axial plane is

$$M = 12{,}250 \times 72$$

$$= 882{,}500\text{lb.-in.}$$

It is found that vectorial effect of additional bending moment due to unbalanced torque in horizontal plane is equivalent to about 10 per cent increase in M.

Thus, bending fatigue stress in 15¼-inch diameter screwshaft, assuming one blade in top vertical position is completely unloaded, is given by

$$f_B = \pm\frac{882{,}500 \times 1{\cdot}1}{347}$$

$$= \pm\,2{,}540\text{lb. per sq. in.}$$

Estimated axial distance of propeller centre of gravity from section at top of cone

$$= 22\tfrac{1}{2}\text{ inch, say}$$

Therefore bending fatigue stress due to weight of propeller and shaft,

$$f_B = \text{say} \pm 1{\cdot}1 \times \frac{22{,}400 \times 22{\cdot}5}{347}$$

$$= \pm\,1{\cdot}1 \times 1{,}455$$

$$= \pm\,1{,}600\text{lb. per sq. in.}$$

Total bending fatigue stress

$$= \pm\,(2{,}540 + 1{,}600)$$

$$= \pm\,4{,}140\text{lb. per sq. in.}$$

(*b*) *Ballasted, Propeller Racing*

Assumed data: as in (*a*) except that:

Ship speed, V_s ... ... = 6 knots
Engine r.p.m., N (instantaneous) ... = 100
Distance of centre of thrust from shaft axis ... ... = 0·8 R
= 88·8 inch

$$V_K = \frac{6}{1{\cdot}365} = 4{\cdot}4\text{ knots}$$

$$\frac{V_A}{N\,D} = \frac{4{\cdot}4 \times 101\tfrac{1}{3}}{100 \times 18{\cdot}5} = 0{\cdot}24$$

From Troost's B.4 series curves, thrust coefficient $C_1 = 0{\cdot}287$.

Thus, total thrust, if propeller were completely immersed, would be given by

$$T = 1{\cdot}99 \times (18{\cdot}5)^4 \times \left(\frac{100}{60}\right) \times 0{\cdot}287$$

$$= 187{,}000\text{lb.}$$

Assume only one half of lower blade re-entering water is effective, then bending fatigue stress due to thrust and torque unbalance

$$f_B = \pm \frac{187{,}000 \times 88 \cdot 8 \times 1 \cdot 1 \text{ (say)}}{2 \times 4 \times 347}$$
$$= \pm 6{,}600 \text{lb. per sq. in.}$$

Add stress due to dead weight of propeller, then total bending fatigue stress

$$= \pm (6{,}600 + 1{,}600)$$
$$= \pm 8{,}200 \text{lb. per sq. in.}$$

Appendix V

PRECAUTIONS AGAINST THE DAMAGING EFFECTS OF TORSIONAL AND FLEXURAL VIBRATION

(*a*) *Avoidance of continuous operation at the 3rd Order, 1-node critical speed*

The following is a brief discussion of three methods of achieving this object, substantially as recommended by the American Bureau of Shipping in April 1948:—

(1) *New propeller*

The existing propeller, 18ft. 6in. diameter and 16-feet mean pitch, will absorb 2,500 i.h.p. or approximately 2,150 delivered horsepower at the maximum service speed of 76 r.p.m. and approximately 1,750 i.h.p. or 1,400 d.h.p. at 66 r.p.m.

By increasing the mean pitch to 17ft. 3in. with the same diameter, 18ft. 6in., it is possible to absorb approximately 2,200 i.h.p. or 1,850 d.h.p. at a lower maximum service speed of 66 r.p.m.

If propulsive efficiency does not change appreciably, the resulting decrease in ship speed at respective maximum service revolutions

$$= 100 \left[1 - \sqrt[3]{\frac{1{,}850}{2{,}150}} \right]$$
$$= 5 \text{ per cent.}$$

The limiting factor is the mean torque in the shafting, which is approximately the same with the new propeller at 66 r.p.m.

The position of the critical speeds and magnitude of the vibration stresses will not be appreciably affected by fitting the new propeller, the advantage obtained being that under conditions of racing, the critical is approached or passed through under "transient" conditions instead of being "dwelt upon" continuously.

At the time of writing it is understood that designers on both sides of the Atlantic are optimistic that a more efficient propeller can, in fact, be produced and that thereby the above loss of speed can be avoided.

(2) *Increased Stiffness of Line Shafting*

By increasing the diameter of, say, three length of intermediate shafting from 13½ to 16 inch, or of two lengths to 18½ inch, it is possible to raise the position of the 3rd order critical speed to approximately 86 r.p.m., but with little change in the magnitude of the vibration stresses in the screwshaft. Thus, under conditions of racing the advantage obtained is again that the critical is approached or passed through under "transient" conditions instead of being "dwelt upon" continuously.

By suitable machining and use of tapers, it may be possible to avoid the renewal of plummer blocks.

This method has the added advantage that the more severe 2nd order critical will also be removed further from the service speed, an undoubted benefit when racing, even though the magnitude of the stress is not reduced.

(3) *Increased Moment of Inertia of Engine*

By fitting a heavy flywheel having a moment of inertia of about 41,400lb.-in.-sec.^{2} units (78·5 per cent of total engine moment of inertia) the total moment of inertia of the complete engine-propeller system will be increased by 20 per cent. Further, the 1-node natural frequency will be reduced to 189 v.p.m., giving the 3rd order at 63 r.p.m. and the 2nd order (with propeller dry) at about 99 r.p.m.

Estimated d.h.p. at 63 r.p.m. in ballast condition

$$= 1{,}456 \times \left(\frac{63}{76 \cdot 5} \right)$$
$$= 815$$

Thus, i.h.p. $= 815 + 350$

$$= 1{,}165$$

Reduction factor for 3rd order steam harmonic torques

$$= \frac{1{,}165}{2{,}150} \times \frac{70 \cdot 5}{63} = 0 \cdot 607$$

Reduction factor for 3rd order inertia harmonic torques

$$= \left(\frac{63}{76 \cdot 5} \right) = 0 \cdot 68$$

Taking common factor of, say, 0·64 for both, then approximately, new value of

$$T'_3 = 218{,}000 \times 0 \cdot 64$$
$$= 140{,}000 \text{lb.-in.}$$

Now propeller slip at 63 r.p.m. will not be appreciably different from that at 76·5 r.p.m., assuming the Ⓒ curve is reasonably flat. Thus, the value of λ will not change and $f(\lambda)$ and $f'(\lambda)$ will be as before and new propeller damping factor

$$k'_2 = 472{,}000 \times \frac{63}{76 \cdot 5} = 389{,}000 \text{lb.-in. per rad. per sec.}$$

Thus, from equation (32)

$$f' = \pm \frac{J'_1 \, p'_c \, T'_3}{Z \left[k_1 + \left(\frac{J'_1}{J_2} \right)^2 k'_2 \right]}$$

$$= \pm \frac{52{,}725 \times 1 \cdot 785 \times 19 \cdot 8 \times 140{,}000}{694 \left[78{,}000 + \left(\frac{94{,}125}{154{,}000} \right)^2 \times 389{,}000 \right]}$$

= ± 1,675lb. per sq. in. for S.S. *Bendoran*,

and for the average vessel in which a stress of about ± 3,400lb. per sq. in. may be assumed, corresponding to an engine damping factor of 61,700 units.

$$f' = \pm 1{,}800 \text{lb. per sq. in.}$$

With full propeller immersion, and thus greater propeller damping, it is probable that these stresses would be still further reduced, despite the corresponding increase in energy input from the cylinders at the higher m.i.p.

In the case of the 2nd order critical, assuming zero propeller damping, $k'_2 = 0$, and that engine damping, k_1, remains as before, then estimated stress for S.S. *Bendoran* when racing.

$$f'_2 = \pm 9{,}850 \times 1 \cdot 785 \times \left(\frac{198}{238 \cdot 3} \right)^3$$
$$= \pm 10{,}100 \text{lb. per sq. in.}$$

This result indicates that the fitting of a flywheel, sufficiently heavy to increase the engine moment of inertia by 78·5 per cent, may even slightly augment the 2nd order stress when racing and precautions against inadequate propeller immersion should therefore be more than ever necessary where this arrangement is adopted, in view of the closer proximity of the critical to the service speed. It is known that speed surges, without the heavy flywheel, can range from 40-140 r.p.m. Thus, for the same initial acceleration torque and period of race, it would be reasonable to expect a speed rise of approximately $\frac{(140 - 40)}{1 \cdot 2} = 83 \cdot 3$ r.p.m., or say, probable maximum speed = 125 r.p.m., i.e., beyond the 2nd order at 120 r.p.m.

(*b*) *General precautions*

Although the provision of an efficient governor will not prevent racing, it will undoubtedly limit the resulting speed rise and will also eliminate to some extent the human element. For these reasons it is considered all vessels of this class should be fitted with a robust and readily adjustable governor, and particularly, that engines should be adequately shut in when liable to race.

One of the difficulties with the average steam reciprocating engine is that it is not easy to estimate instantaneous engine speeds. In the author's opinion it would repay owners to fit a suitable tachometer to the "Liberty" engine. This would be of considerable value to the engineers when in heavy weather, since it would enable the adjustment of engine controls and governor setting to be accurately suited to the prevailing conditions, thus helping to ensure that maximum speed surges are always well below the dangerous 2nd order critical.

Experience has shown that in many cases (see, for example, Fig. 12), the detection of fatigue cracks in their early stages has defied the usual methods of visual examination. In these circumstances the use of magnetic methods of crack detection is becoming increasingly common when screwshafts come under survey. In fact, having regard to the prime importance of this business end of a ship's machinery, it would seem well worthwhile to instal in every dry dock of any size suitable equipment for magnetic crack detection which would be used as part of the normal docking services.

Among the many smaller but important precautions against screwshaft failure, the following are suggested as worthy of consideration: —

(i) The fitting of an outside sealing gland to the propeller boss, preferably additional to the original arrangement.

(ii) Check that the fit of the cone guard for the propeller-nut is watertight in order to prevent leakage along the shaft from the small end of the taper.

(iii) Verify that when the propeller is hard up on the taper, the square end of the key in the propeller keyway is flush with the bottom of the recess in the boss.

(iv) For replace shafts omit the forward key-retaining bolthole (see Figs. 5 and 8) and allow a greater distance between the end of the keyway and the end of the shaft liner.

Discussion

DR. W. KER WILSON, Wh.Ex. wrote that the author had made an extremely welcome contribution to the literature on marine propeller shaft vibration, emphasizing, as he did, the fact that under certain circumstances an apparently simple vibration problem might develop deep and dangerous undercurrents.

Above all, the paper provided yet another demonstration of the wisdom behind Admiral D. W. Taylor's famous injunction —"The time for pessimism is in the drawing office, not on the trial trip".

He thought that a brief description of a somewhat similar trouble which beset the coned end of an aero-engine crankshaft a few years ago might not be out of place.

Fig. 24 (Plate 5) showed the coned end of the crankshaft of a 4-cylinders-in-line aero-engine crankshaft. The propeller hub was mounted on the coned end of the crankshaft and a single key was provided to assist in transmitting the engine torque from the shaft to the hub. The hub was held in place by a nut and a second nut was provided just above the larger end of the cone to retain a ball thrust bearing. The thrust bearing retaining nut was locked by a tab washer, one of the locking tabs being accommodated in a narrow groove milled in the shaft from the end of the main keyway. The cone was about 2-inch long with a taper of 1 in 10.

The prototype engines successfully completed their development and acceptance tests, during which they were, as usual, subjected to some lengthy and gruelling running. In spite of this, however, when the production models went into service, fatigue failures of the coned end occurred of which the photograph shows a typical example.

The fatigue cracks invariably originated at the relatively sharp edge of the keyway and there was also evidence of appreciable fretting between the hub and the shaft in the region containing the starting point of the crack.

Seven failures of this type were experienced over a period of about a year.

Fig. 25 (Plate 6) showed a modified design in which certain improvements were made to overcome the trouble. These modifications might be summarized as follows: —

(*a*) The keyway was milled parallel to the taper to provide a somewhat stronger shaft section at the larger end of the cone.

(*b*) The narrow slot for the tab of the retaining nut locking washer was eliminated to remove one source of stress concentration.

(*c*) A flat was milled at the top end of the keyway to prevent fretting between the hub and the shaft in the region where fatigue failures were most likely to occur.

(*d*) The included angle of the conical bore in the hub was made slightly smaller than the included angle of the shaft cone. The difference between the two angles was of the order of 2 minutes which correspond to a tighter fit at the larger end of the cone of about 0·001 inch. Since the larger end of the cone was 2 inch in diameter this change produced a tighter fit at the larger end of about 0·0005 inch per inch diameter.

This was done to make sure that the hub gripped the larger end of the shaft cone before contact was made at the smaller end.

The hub was then forced into close contact when the retaining nut was tightened. In this way the possibility of initial slackness of the fit between the hub and the shaft, particularly at the critical region towards the larger end of the cone, was minimized.

After these changes had been made several thousand engines were put into service and no repetition of the early failures was experienced.

There were, however, one or two isolated cases of fatigue failures originating at the screw thread which retained the thrust bearing, and in the latest designs the coned end had been abandoned altogether in favour of a parallel splined end with the hub located concentrically by a narrow conical collar at each end of the spline.

This experience indicated in no uncertain manner the paramount importance of attention to detail in the design of parts operating under necessarily high dynamic loadings.

Referring now to Fig. 5, which showed details of the "Liberty" propeller shaft tail end, there would seem to be a certain amount of scope for improvement of mechanical design.

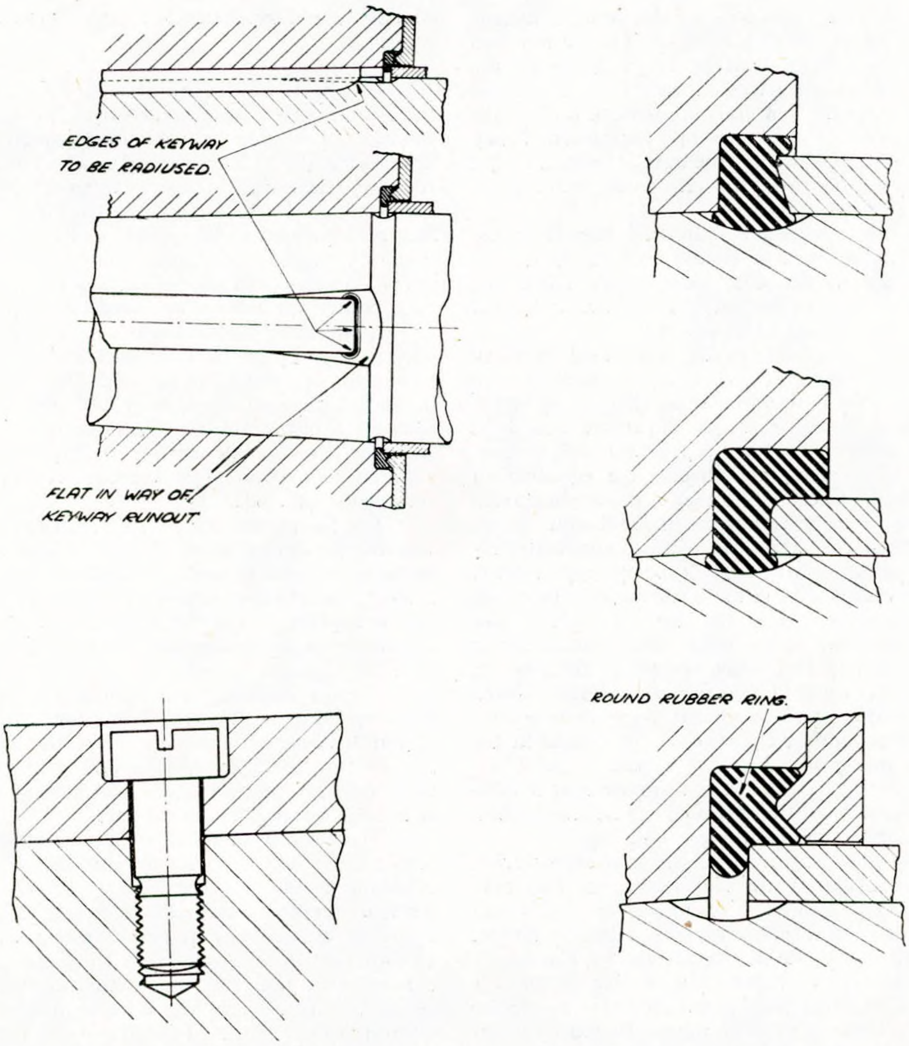

FIG. 26—*Suggested improvements for design of Liberty propeller shaft*

Fig. 26 showed one or two suggestions for improvement, namely:

(*a*) Provision of a flat at the top end of the keyway, to prevent fretting in this sensitive zone.

(*b*) Provision of well-formed radii on the edges and corners of the keyway.

(*c*) Provision of a slight differential fit between the hub and the shaft cones, to make sure that a good fit was obtained, particularly at the larger end.

(*d*) Provision of an undercut at the larger end of the cone, coupled with the provision of an overlap for the shaft liner and the hub. This over-lapping of the tightly fitted parts prevented a fatigue crack from starting in what would otherwise be regions of heavy clamping pressure.

(*e*) Provision of an improved rubber seal. Three designs were shown in the sketch. The upper arrangement was that used on "Liberty" shafts.

(*f*) Provision of a plain counterbore, with well-formed edge radius, to receive the tap bolts which secure the key. This should reduce the risk of cracks in this region.

The author rightly mentioned the necessity for care when interpreting the records obtained from seismic torsiographs. Since this matter was apt to be overlooked, it was well to keep in mind the remark made by Captain L. P. Small, of the American Bureau of Ships, namely:

"It may be stated that as long as measuring devices (and all our calculations are based on the data obtained from them) are subject to the inaccuracies which only recently upset the whole structure of carefully accumulated data of one of the engine builders, or—to put it in other words—so long as torsiographs are what they are, and even the best of engineers can be misled by their indications, it is only prudent for the Bureau to be conservative in assigning the value of the allowable vibratory stresses".*

With this forthright pronouncement in mind, therefore, it was pleasing to note that the author took the precaution of using electrical strain gauge equipment to check the results of the torsiograph measurements and to give direct readings of the shaft strains.

In conclusion he would like to ask if the author could give some particulars of the types of governor that were available for controlling racing. According to published information it would appear that one of the most effective "governors" for this purpose was an exhaust steam turbine. Such an arrangement not only prevented racing, but also increased torque when the propeller was immersed and the ship was labouring in heavy weather.

MR. A. R. GATEWOOD (Member) (read by Mr. W. Low) said that the paper was a lucid and comprehensive analysis of a problem which was important in itself, and whose investiga-

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**Evaluation of Effects of Torsional Vibration*, Society of Automotive Engineers, New York, 1945, p. 11.

tion had resulted in practical advances in the design, manufacture, and installation of propeller shafting. The author had made particularly useful contributions in the analysis of the bending stresses to which shafting was subjected when a vessel was operated with the propeller not fully immersed, and in the discussion of stresses encountered during racing in heavy weather. Another point mentioned by the author, which might well be emphasized, was the need for effectively sealing the after-end of the propeller assembly.

Before discussing the author's conclusions regarding the effects of insufficient propeller immersion and of racing, it might be desirable to fill in the data missing in Table III. The *Ira Nelson Morris* was in ballast during the tests, her draught being 8 feet forward and 14ft. 6in. aft.

Operation with the propeller partly immersed was, of course, not unusual with single screw vessels in general. In reviewing the data of Table I, he noted that there were forty-four vessels listed whose draught aft at departure could be ascertained. Of these, seven exceeded Sir Amos Ayre's recommended minimum of 20ft. 7in., and twenty-six equalled or exceeded Bocler's figure of 16 feet. Discussion proceeding from these figures would appear to be somewhat inconclusive.

In order to ascertain the significance of insufficient propeller immersion, the records of fifty-four "Liberty" type tankers were examined. These vessels were fitted with engines, shafting, and propellers of the same design as the dry cargo ships, but they were not usually operated in the light condition except in the brief interval immediately preceding arrival at the loading port. On these tankers a total of eighty-one propeller shafts were drawn for examination and seventeen were replaced for various reasons. Of these, nine were rejected for cracks in the keyway. No wheels in this group were lost at sea.

On the basis of these figures, it did not appear that insufficient propeller immersion was the primary cause of failure, although it might well be a contributing cause.

The author's investigation of propeller shaft stresses during racing had done much to clarify the significance of this type of loading in propeller shaft failures. There was no doubt that repeated heavy overloads caused by racing might start a fatigue crack, which could then propagate at normal service loading.

In attempting to assess the importance of this factor, the records of "Liberty" ships fitted with governors of a particular design were examined. These governors were actuated by water pressure, not by overspeeding, and were designed to operate before the propeller emerged from the water. The primary function was thus to anticipate and control racing in heavy weather. The records of 161 vessels fitted with these governors were examined, and it was found that twenty-four typical propeller shaft failures occurred after the governors were installed. In five instances the propeller was lost at sea. In the other nineteen cases, the shaft was condemned because of cracks at the forward end of the keyway. (Failures from other causes were not considered typical.) In three cases new propeller shafts had been fitted when the governor was installed, or immediately prior thereto; and in two cases, the shaft had been drawn for examination when the governor was installed.

From this it might be inferred that racing was not the principal or a necessary factor in these failures. This conclusion found supporting evidence in the surprising freedom from failure of the other shafting. As was mentioned by the author, a search of the records of all "Liberty" ships classed with the American Bureau of Shipping revealed only nine crankshafts that failed by loosening of the shrink fit. The strength of the shrink fit had been adequately treated by the author, and it need only be remarked that these crankshafts were assembled by heating the webs in oil, and that the interference was 19·5 ± 1·5 thousandths, which was somewhat less than had been allowed in the author's calculations.

The foregoing considerations, coupled with the practical difficulties encountered in altering ballasting conditions, led to the decision by the American Bureau of Shipping to reduce the engine speed to 66 r.p.m. under all loading conditions, unless alterations were made to the main propulsion system which effectively removed the 3rd order critical from the operating range.

DR. S. LIVINGSTON SMITH, F.C.G.I. (Member) said that the author was to be congratulated, not only upon the systematic manner in which he had investigated the principal factors underlying the "Liberty" ship screwshaft failures, but also upon the very clear and logical manner in which he had presented the results. To him the conclusions as to the causes of these failures appeared to be sound, and in particular the author had certainly made a good case for paying more attention to propeller immersion in the ballast condition. There were, however, two points on which he would like to make some remarks.

In the first place, it was interesting to note the difficulties encountered by the author in attempting to assess the relative strengths of round-ended and sled-runner types of keyway under fatigue loading—a very difficult thing to do. The fatigue tests of Thum and Peterson afforded some guidance to the designer, but, as the author rightly pointed out, their results suffered from numerous limitations when applied to the full size propeller shaft.

The figures the author had used and which involved clamping stresses seemed to be all right. But he was not sure whether those experiments were carried out with the torque applied through the stress concentration or with the stress concentration in the middle of the shaft and the torque applied at the end, because he was certain that the results would be quite different in the two cases.

Lloyd's Register of Shipping were at present carrying out a comprehensive investigation for the British Shipbuilding Research Association on the torsional fatigue strength of full-size shafting, and the relative strengths of the round-ended and sled-runner type of keyway was one of the problems which it had been decided to investigate.

The second point to which he would refer concerned the strength of shrink fits. Under the heading, "Discussion of Available Evidence", the author gave the results of calculations made to determine the relative strengths of the shrink fits of the after l.p. crank webs in two different installations. It seemed to him that in all probability the figures arrived at would give a reasonably true picture of the relative strengths in the two cases, but he thought the author himself would agree that the absolute values obtained could only be regarded as very approximate. The fact was that the necessary experimental data was lacking to make such calculations with any pretence to accuracy even when the shrink allowance and conditions of assembly were fully known.

The grip strength depended primarily upon two factors, namely, the inter-face pressure obtained and the "overall coefficient of friction"—the latter taking into account such factors as surface finish, lubricant, geometrical proportions, etc. Much uncertainty attached to the calculation of inter-face pressure, and data was lacking as to the effect of the various factors comprising an overall coefficient of friction. Russell's work, which the author had made use of in his calculations, was carried out at the Royal Technical College many years ago, and whilst it probably represented the best available information on the effect of lubricant on the coefficient of friction, it dealt with only one aspect of the problem.

During the past two years or so the Royal Technical College have been carrying out a comprehensive investigation for the British Shipbuilding Research Association into this problem, with particular reference to failures in built-up crankshafts. These experiments had involved both small-scale and full-scale tests—the latter on full-size marine engine crankshafts. This investigation was by no means completed, and indeed, it was only now entering upon the most interesting stage, but there was every hope that it would place the design of shrink fits upon a more rational basis. At this stage he would merely draw attention to two interesting points which had already emerged. The first concerned the effect of shrink fit allowance upon grip strength. The small-scale experiments had shown that there was a zone of uncertainty of shrink fit allowance where it appeared the elastic limit was exceeded and yielding and subsequent plastic flow occurred in the bore of the assembly. Further, the shrink fit used with full-size crankshafts appeared to correspond with the zone of uncertainty as determined on small-scale tests. Secondly, these experiments, as indeed was the case with those of previous workers, were carried out under static loading conditions. As yet little or nothing was known about the effect of pulsating load conditions such as occur in marine engine crankshafts. Experiments were therefore now being made under such conditions.

Finally, it might interest the author to note that in "pushing out" tests on the crank-pin of a full-sized crankshaft values of the coefficient of friction were obtained, based upon the observed inter-face pressure between 0·28 and 0·36. This crankshaft had seen five years service in a marine Diesel installation and the pins were of 21½-inch diameter.

PROFESSOR L. C. BURRILL, M.Sc., Ph.D. (Member) said that the breakage of propeller shafts in, and the loss of propellers from, "Liberty" ships had presented many people with considerable problems in the past two or three years. He had personally been concerned with the design of replacement propellers, and although he could not contribute very much to the technical criticism of the paper, it might be of interest if he added some notes on what had happened so far as propeller manufacturers were concerned relative to these breakages.

In the first place, it became evident that there was a torsional critical present within the working range in these ships. Thus, the first replacement propellers were made so that they developed the full power of 2,500 i.h.p. at 87 r.p.m. Some of these propellers must have been in service now for 2-2½ years, and so far as was known they had been perfectly satisfactory.

The later approach was that suggested by the American Bureau of Shipping. They stated that the revolutions should be reduced to 65. He was now able to say that they had designed propellers—and about thirty have been fitted so far—which develop the service power of 2,200 i.h.p. at 65-67 r.p.m. These propellers were a little larger than the original propellers, but were working at lower revolutions; and it was therefore possible to get a closer approach to the optimum diameter with this bigger screw. Despite that, the weight was about a ton less than that of the original propeller and, by altering the radial distribution of thickness, the radius of gyration had been reduced, so that the mass moment of inertia of this new propeller, working at 66 r.p.m., was about 83 per cent of the mass moment of inertia of the original propeller, excluding entrained water in each case.

The engine builders had naturally co-operated in this matter, and they had stated quite clearly that this engine could run at 66 or 65 r.p.m. for the full power, particularly if some slight change was made in the cut-off. So far as was known—and the experience was fairly broad, covering quite a number of shipowners and shipping lines—this later solution was the better one, i.e., the engine had been made to take up the higher torque, the screw was running more slowly, it was more efficient from the propeller designer's point of view, and the smaller moment of inertia propeller had put up the natural frequency. At the same time, there was no need to reduce the service power.

He understood from the author that, if these ships were properly ballasted, then changes in revolutions would not be required; but he still felt that the solution he had mentioned above was perhaps the best.

Turning now to a point in Appendix III of the paper concerning $\partial Q/\partial\omega$, the propeller damping factor, he was interested to read this section and there were two points from the propeller point of view to which he would like to refer briefly. First, it had been assumed, he thought, that the power was reduced directly according to the loss of area (i.e., if the screw had a certain area out of the water, then the percentage power loss was equivalent to the area loss). That was not so. Experiments showed that the loss of power was almost linear with the loss of immersion, or with the loss of diameter, and not with the area. In other words, even if 1/10th of the diameter protruded out of the water, which represented a very small area loss, about 1/10th of the power was lost. He thought that this question might be looked into further, and the author might find it convenient to consult the work of Dr. Kemp and Miss Smith Rean on this subject.

The second point was that the author had calculated the d.h.p. by using an assumed wake and Troost's curves and, assuming the d.h.p. /i.h.p. was as indicated (giving certain fractions), a certain answer was arrived at. The author then compared the d.h.p. from the assumed wake with that answer and deduced a frictional h.p. He suggested that the author was deluding himself in thinking he had deduced anything, because it had only been assumed that the d.h.p. had a certain relationship. He did not quarrel with the author's calculation, other than to say that it was not considered reasonable in the propeller world to assume the wake and then work out the d.h.p. It was better to assume the d.h.p. and work out the analysis wake from the propeller curves.

He felt that from this whole investigation there was still something missing. He did not consider that either the suggestion that it was the occurrence of torsional vibrations or the alternative suggestion, that it was the lack of immersion of the propellers, that was the root cause of all this trouble, was correct. They had heard very little about the inefficient rubber ring and sealing devices on these propellers. Dr. Ker Wilson had shown that the end of the liner did not fill the recess in the forward end of the boss when squeezed up. That left quite a large gap; and the liner was of much smaller diameter than the recess. Quite a number of these propeller shafts had been gouged out by corrosion in way of this recess, and he had himself seen circular gougings right round the shaft when inefficient sealing had been used. That was a very important factor. Another matter which had not been considered was that the tapers of these shafts were not always accurate and that the propeller did not mate all the way up the shaft taper.

Finally, they were told that the rudder was of the "Goldschmidt" type, which he thought was a reaction rudder. He did not like a reaction immediately behind the propeller blades and would keep away from it as much as possible. When the ship was in the ballasted condition, with the propeller half out of the water, it had to be remembered that the propeller was only effective in the bottom half. That was an important matter, as the reaction was also limited to the lower half of the disk, and vibration forces might be set up by this arrangement, if the blades were too close to the reaction..

MR. J. M. MURRAY, M.B.E., B.Sc. said that the author had examined all the relevant factors entering into the loss of propellers in "Liberty" ships, and had made a strong case that racing of the propeller through insufficient immersion in the ballast condition was a very important factor. He was not sure that the author had proved, however, that insufficient immersion was the primary factor. There were many other circumstances contributing to these failures, as had been pointed out already in the discussion, and it must be a matter of extreme difficulty to segregate and separate them. He thought that more experience of the effect of the remedies which had already been applied would be necessary before it could be said what were the primary and what were the secondary factors. He felt that the proportion of cracked shafts discovered in the future would be the proper criterion on which to judge whether these remedies were in themselves sufficient.

The figures given in Table I showing the ballast draughts when leaving the ports indicated that in many cases a considerable improvement could be effected. Apart altogether from the point of view of stressing propeller shafts, these draughts could with very great advantage to the ship be increased in many cases. The author's analysis of ballast conditions of "Liberty" ships and "Ocean" ships was interesting, but it was suggested that in order to obtain better water ballast capacity it would be preferable to increase the size of No. 3 deep tank rather than build tanks at the side of No. 5 hold, the solution advocated by the author. He thought that any arrangement of ballasting which increased the hogging moment should be avoided, and this was especially true in the case of "Liberty" ships.

The same point arose with "Ocean" ships. In condition 2 for the "Ocean" ships, Table 6, the deep tank aft was left empty and, as the author said, the stresses were reduced thereby. He thought it should be appreciated that these "Ocean" ships were sufficiently robust to stand up to the sagging stresses that would be imposed with the deep tank full.

Fig. 20, showing the engine speed during the test on the *Bendoran,* was exceedingly interesting. It should be noted that the pause at 80 r.p.m. where third order vibration of the shafting was excited coincided with the ship itself passing through a hull critical; Prohaska's correction for depth of water gives the primary vertical hull frequency in this region. Quite severe hull vibration was induced, and it was not surprising that the engineer controlling the engines did not run through this critical as rapidly as through others. Altogether the experiment was most spectacular in the engine-room and even more so at the propeller.

MR. J. NICOLAISEN said that he had recently designed a torsional vibration damper for the United Steamship Co. Ltd.'s "Liberty" ship S.S. *Oregon.*

The damper was of unique design; it weighed about two tons and was clamped on the thrust shaft without keys, and without necessitating lifting of the shaft. The damper contained no wearing parts, and no maintenance was anticipated.

According to tests carried out by the United Steamship Co.'s engineers and consultants the maximum vibration stress in the intermediate shaft at 3rd order resonance was found to be about 165 kg. per sq. cm. in ballast with draught 9ft. 0in. forward, and 14ft. 8in. aft. (The corresponding stress in the propeller shaft was about 1,630lb. per sq. in.)

No tests had been made at 2nd order resonance, but he expected that the damper would be very effective also for this condition.

MR. G. DANIEL said that the author thought it best that "Liberty" ships should be ballasted to a certain fairly deep condition as these shaft failures were more likely to occur in a light ballast condition than in the loaded condition. That was a deduction which did not appear to be very conclusive, and as mentioned by Mr. Gatewood did not seem to appeal very much to the Americans.

One feature needing attention in connexion with ballasting was the rather vague or crude attitude generally adopted in defining what the ballast condition should be. There were some who said they would like to have 75 per cent of the propeller immersed, others thought 80 per cent, whilst someone else wanted full immersion. It seemed that there was a lack of scientific information as to what was an adequate propeller immersion for the proper manœuvring and efficient service of cargo ships. Associated with sufficient propeller immersion there must, of course, be suitable stem immersion.

There should be research to ascertain the most practical and satisfactory immersion of a propeller in the ballast condition and what the associated immersion at the stem should be, having regard to all the conditions which are required to be met. Fuel consumption was concerned as also was the propeller efficiency when the vessel was loaded and when in ballast. It might not be possible to determine the matter by pure mathematical investigation for there were unpredictable factors. Failing mathematical information it might be possible to carry out tank tests under wind and wave conditions, and if they were not conclusive or not quite practicable, perhaps tests or observations on ships at sea such as were now carried out by the British Shipbuilding Research Association for various purposes might form a line of attack.

It was considered in the early days of convoys in ballast during the war, that 80 per cent immersion would be satisfactory as Mr. Bocler proposed in his paper,* but it was found to be insufficient in practice. Another 1,000 tons of ballast was therefore added in the standard ships, but there were still difficulties in maintaining course and speed. Then it was agreed that the propeller tips ought to be immersed about two feet. There seemed something rather unsatisfactory about this. At present one had to go to the practical seaman for advice on suitable immersion of the ship forward and aft, but instead one should be able to go to the naval architect or the scientific investigator for a determination of the best ballast condition to meet the appropriate circumstances, and the sailor should use his practical experience to make adjustments afterwards as he thought necessary.

There seemed, therefore, to be a necessity for research into the problem of what was a satisfactory propeller immersion for ships in ballast, from the point of view of maintaining course and speed and efficiency in a sea way.

MR. J. MORRIS said that he had been concerned with the investigation of coupled engine and propeller vibration in aircraft, which was due to the interaction of a variety of factors. In the early stages, when resonant torsional vibration was suggested as the cause of a number of failures in aircraft engines, investigators began to look into details to find the reasons, but after a time they concluded that there was more in it than minutiæ of design. But they made a great mistake in maintaining that the propeller could be treated as a rigid body, as a sort of flywheel; and they did not get very far. He insisted that the flexibility of the propeller must be taken into account.

Eventually he arrived at an analytical solution in which he took into account the flexibility of the propeller—its twist, and so on—after which fewer failures were experienced, because appropriate variations were made in the parameters affecting the phenomenon.

He did not know whether anything similar could be done in the marine line. The aircraft propeller blade was somewhat like a rod with a straight flexural axis and was more amenable to analysis. But in any event the problem was one affecting the combination of the engine and the propeller, and in his opinion so long as the two were kept distinct the problem would not be solved.

MR. J. N. JARVIE (Member) said that no engineer could possibly reject the conclusion reached by the author, because it had been shown definitely that the heavy stresses referred to did exist, but the engineer who ran the ship and was responsible for their maintenance, was sure to look in other directions. The engineer knew from experience the conditions in which he must link up and stand by his throttle. But what was of even greater importance was the fact that the ship under his care was probably built in British yards and engined in a British yard; the propeller key was carefully fitted and the propeller tried and tried again on its cone to ensure that if there were any slightly tighter place than any other it would be at the top of its cone and not at the point of the shaft.

That care meant a great deal in the life of a shaft. Within the last six months he had seen a shaft break because the propeller had been driven hard on to the end of the cone and was barely gripping at the top.

He wanted to emphasize that there was no intention to cast aspersions at or to lay blame on American yard practice. The ships and engines in question were built under very exceptional circumstances, and nobody would deny that they more than fulfilled their purpose. But, because it was expedient to do so at that time, it may be that hundreds of their propellers were mass-produced and pushed on to any shaft that happened to come along. That practice, with perhaps negligible fillets in the key-ways, indifferently fitted keys and bad sealing, was in his opinion the main contributory cause of ultimate failure.

In explaining the comparative immunity from shaft trouble of the Canadian-built ships, and particularly the "Ocean" type

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*North-East Coast Institution of Engineers and Shipbuilders, 1942.

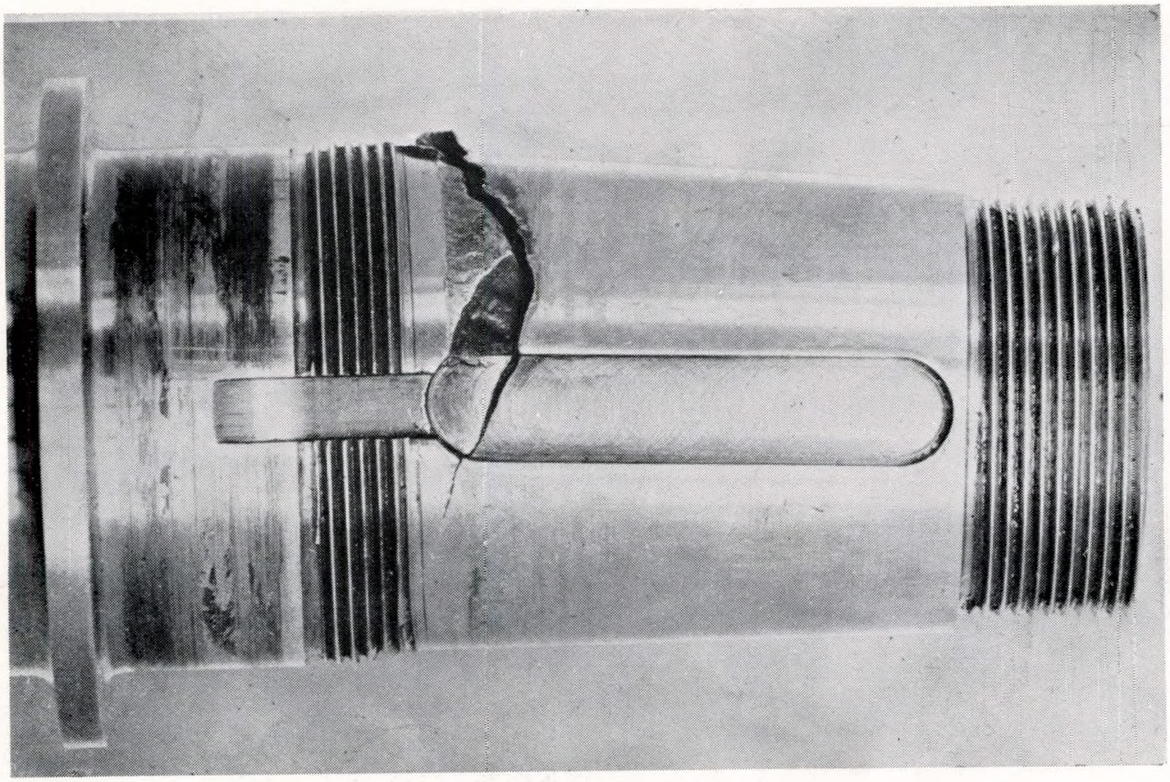


FIG. 24—*Aero-engine crankshaft*

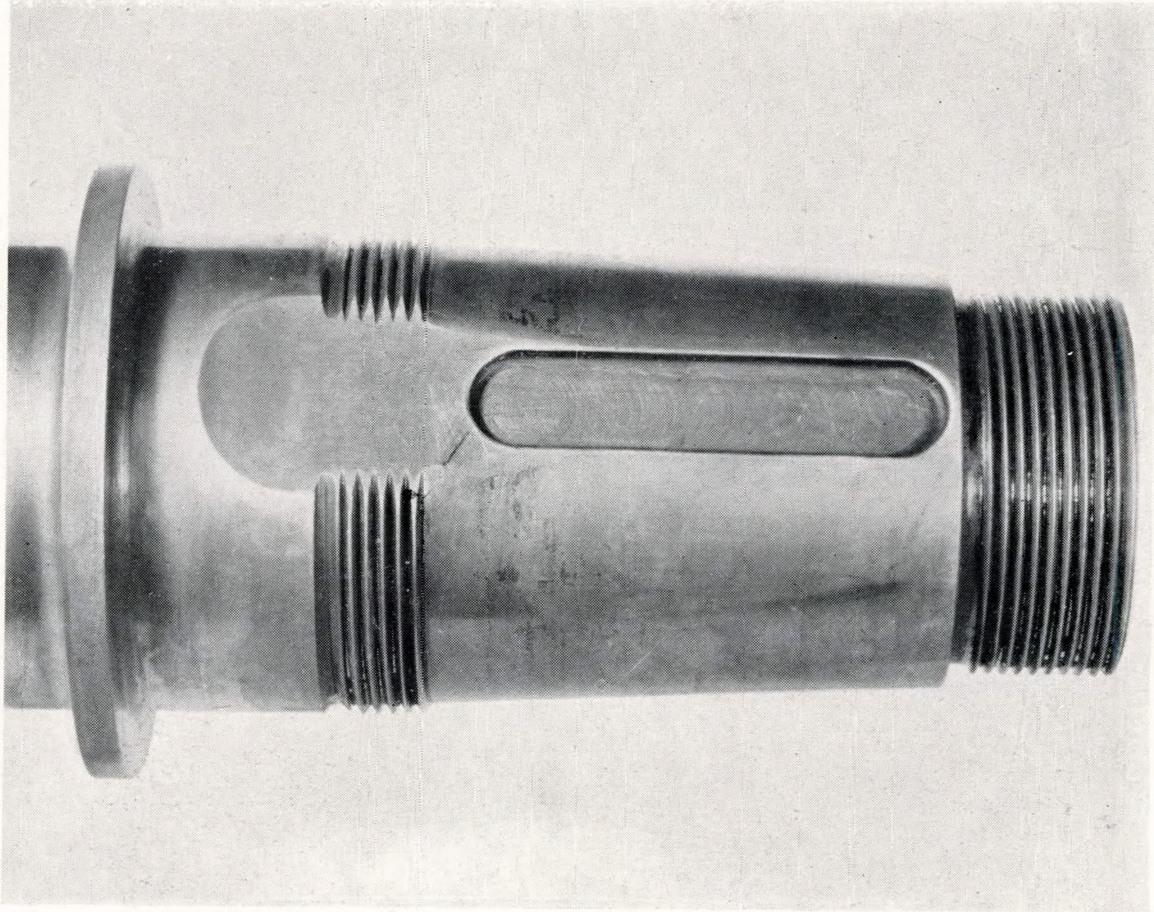


FIG. 25—*Modified crankshaft as shown in Fig. 24*

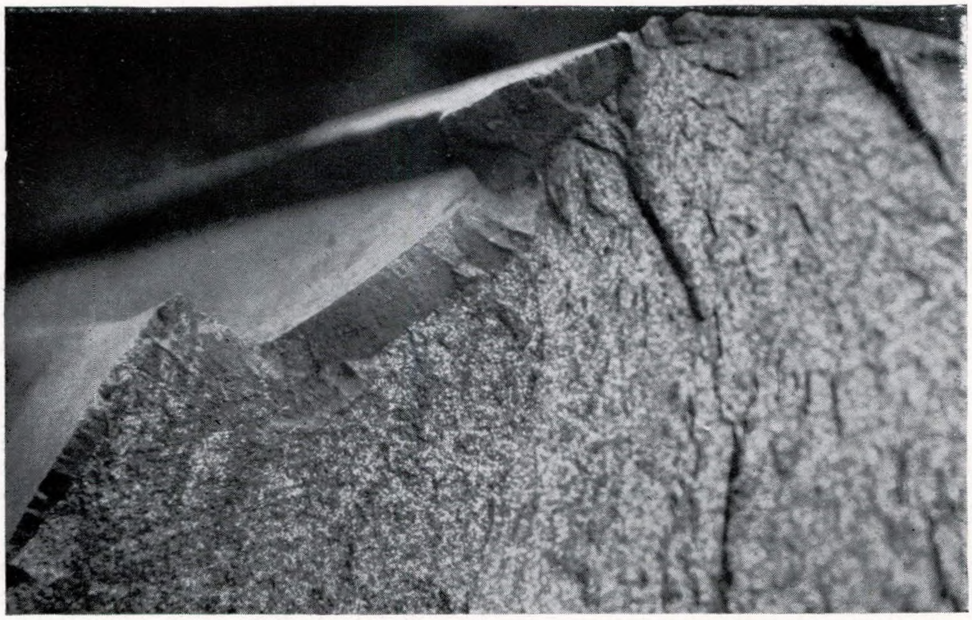


FIG. 29

FIG. 30

FIG. 31

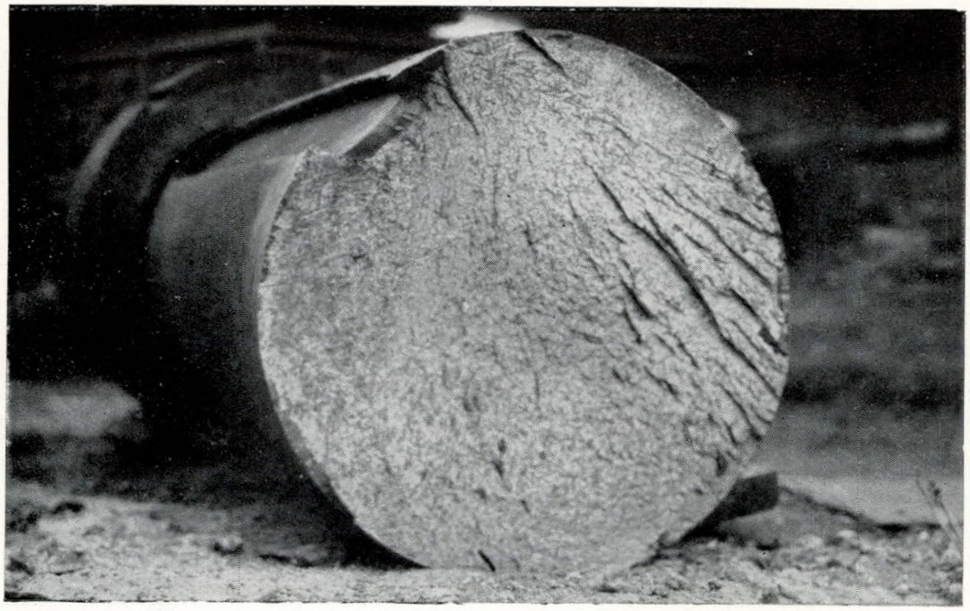


FIG. 32

ships which were built in America, similar in nearly every respect to the "Liberty" design, the author very modestly attributed it to the fact that those types were coal-burners and were never so light, therefore, as the oil-burning "Liberty" ships. He wondered how far that could be accepted.

The immersion of the propeller was very important. But what difference did two-feet immersion make in regard to racing? The effect would be small, and if the ship had no governor or was not effectively controlled in North Atlantic seas, the peak torsional vibration stress to damage the shaft would not be lessened. He had made many passages in a tanker in loaded trim when the emergency pawl of an Aspinall governor had been thrown into operation two and three times an hour, or more. The fact was that "Ocean's" and Canadian ships were built and engined under a more particular survey—Lloyd's survey—then were the "Liberty"'s.

In his opinion, the quality of the shaft material was not in doubt, though it had not been so easy to prove that the forging practice, annealing, etc., was perfect. With a propeller properly fitted on cone and key, the elimination of all stress raisers (amongst which the author placed bad fillets, tight keys, badly shaped key-ways and all marks penetrating the skin of the shaft) and perfect sealing, there should not be great trouble. Perfect sealing was very important, for there could be no doubt that, if sea water had access to a stressed shaft, the endurance limit of the material was reduced. If the three points he had mentioned had been watched as they would be watched under ordinary peace-time conditions, there would have been perhaps no unusual spate of shaft failures, even though the revolutions for which the engine and propellers were designed, namely, 78 maximum when light and 74 when loaded, had been maintained, so long as care was taken to prevent extreme racing at any time. It was interesting to note that there was a record of at least one shaft failure that had occurred at the reduced speed of 68 r.p.m., which proved that, if the shaft had been damaged by being subjected to very high vibratory stresses during extreme racing, even for a short period, the ordinary torsion and bending would finish the job. That applied to any propeller shaft.

It was not irrelevant to point out that the Transport Ministry felt so keenly the need to prevent racing under the conditions prevailing during the war, when so many voyages in ballast had to be made, that governors were bought and fitted at Government expense.

Mr. W. J. Ferguson, M.Eng. (Member of Council) said that one point had arisen which he felt should not go without explanation. Professor Burrill rather suggested that there was a difference of opinion between those concerned on the two sides of the Atlantic, but he felt the measure of agreement was most marked.

Surely the story was that on both sides of the Atlantic engineers were aware of the torsional vibration stresses; they found that stresses occurred such as one would not accept at the design stage. Failures occurred and continued to occur, but they could not reconcile the magnitude of the stresses with the number of failures, and they looked for some additional explanation.

They knew that bad workmanship had explained a number of failures, or had masked the true reason. They knew that, above all, the leakage of rubber sealing rings had accounted for failures. But in cases where these odd defects had been absent, where the propellers had been well fitted, the sealing rings had been right, etc., there had still been failures, and they were faced with the difficulty of trying to reconcile low stresses with failure.

The author had shown how these vibration stresses could be increased when propeller immersion was inadequate, and they felt this provided the explanation. There should be no question of which takes priority.

Referring to the last paragraph of the author's conclusions, it would be noted that adequate propeller immersion was advocated plus avoidance of continuous operation at or near the existing 3rd order critical speed.

Correspondence

Mr. A. Logan (Chairman of Council) wrote that as one of those engaged in tanker operation his experience with propeller shafting had been fortunate, and that good fortune could no doubt be accounted for by the fact that a tanker could always increase the amount of ballast carried to bring the propeller immersion to a stage where with efficient governing, engine racing could be kept within safe limits. This would tend to confirm the author's contention that lack of propeller immersion in the particular ships under review was a primary factor in causing screwshaft failure.

It had often been noticed when a ship's master, who had spent years in tankers, was transferred to a case oil carrier, where conditions were akin to those of a dry cargo vessel, that he had become so accustomed to trim a ship exactly as he wished that he was definitely worried about lack of propeller immersion or even a slight list condition, which a master used to dry cargo vessels would take in his stride.

They had experienced some screwshaft failures, but these had shown up on survey only and in the main had been due to ill-fitting sealing rings. The importance of fitting the correct size of sealing ring to prevent sea water entry between shaft and liner could not be over-stressed, for even recently a screwshaft had to be replaced due to the fact that at the previous survey greasy packing had been used instead of a rubber sealing ring.

When taking over a number of T.2 tankers a year or two ago, the screwshaft survey revealed several defective shafts due to liner fractures having caused corrosion underneath. These failures had been attributed to the nature of the material which lined the stern tubes; the suggestion being that the synthetic preparation of canvas and resin caused overheating of the liner and shaft. This material was replaced by lignum vitæ. Fractures of screwshafts in the T.2 ships had also been reported, which ran obliquely from the end of the keyway and extended to the brass liner; also hair fractures had been located at the small end of the cone in way of the keyway. Some of these failures could no doubt be attributed to faulty assembly.

In this rapid summary of their experiences, he would suggest that the immunity from screwshaft failures at sea could be attributed in a large measure to the fact that by the immersion of the propeller, excessive racing was prevented, thereby the stresses in the shafting were kept within the designed limits.

Captain J. P. Thompson (Associate) wrote that he commanded a tanker of 8,500 tons deadweight from 1921 to 1924; the vessel was engaged in the Gulf-U.K. trade, making about twenty Atlantic crossings from east to west in ballast.

The vessel's propelling machinery was situated aft and was similar in all material respects to the machinery in use on the

"Liberty" vessels, and revolutions were 70 to 75 per minute.

The draught leaving U.K. would be about 16 feet forward and 20 feet aft. This draught would be maintained until fine weather was reached, usually in the vicinity of Bermuda, when ballast was reduced to give a draught of 8 feet forward and 18 feet aft. No difficulty ever occurred with the shaft or propeller, and this vessel, under the same draught conditions, continued to run for another twelve years without trouble.

This could be said of seven other tankers, likewise built in U.S.A. in 1921, and known to him. Here there was even greater evidence of immunity from shaft or propeller trouble. In his notes on a "Liberty" vessel, dated July 1948, he found the following:—

"I am of the opinion that the 'Liberty' vessel did not use to full advantage the whole of her tanks. If we take, say:—

Water ballast, fore peak ...	145 tons	
Water ballast, No. 1 deep ...	228 tons	
Water ballast, No. 2 deep ...	420 tons	
Water ballast, No. 3 deep ...	760 tons	
Water ballast, void tank ...	100 tons	
Water ballast, aft peak ...	155 tons	
		1,808 tons
Fuel oil D.B., No. 1	132 tons	
Fuel oil D.B., No. 2	314 tons	
Fuel oil D.B., No. 3	231 tons	
Fuel oil D.B., No. 5	232 tons	
Fuel oil D.B., No. 6	108 tons	
Fuel oil E.R., settling tanks ...	100 tons	
		1,117 tons
		2,925 tons

"and, disregarding boiler water in No. 4 D.B. (130 tons) and domestic water (56 tons), we have a deadweight of 2,925 tons, which I estimate would give a draught in salt water of 14ft. 1in. forward, 14ft. 3in. aft. If, for operational reasons, or a winter month's north Atlantic passage, then the extra dry ballast required would only be about 700 tons, and this could be stowed, say, half in one hold aft at the sides of the tunnel, and the other half between shifting boards either in No. 2 or No. 4 'tween decks, whichever suited the trim best. I think the G.M. would be satisfactory".

This was on the assumption that the bunkers were stowed in the double bottom tanks, which would be transferred to the settling tanks, and when empty, the double bottom tanks would be filled with water.

In turning to Table I, which included a list of 101 failures, he noted that there were thirty-four in which the draught had been recorded, both fore and aft. The average draught forward of ten of these was 6ft. 8in. and to attempt an Atlantic crossing from east to west with such a draught was, to his mind, courting disaster. A cargo vessel meeting head gales—when trimmed to such a light draught forward—would pound in a fearful manner, setting up shudders far in excess of normal vibrations.

It was mentioned that the operators of "Liberty"-type vessels did not put water in their double bottom fuel oil tanks on account of the risk of contamination by water in the fuel. He had discussed this with a Marine Superintendent, who commanded a "Liberty"-type vessel. His practice was to use the deep tanks as water ballast and to carry the oil fuel bunkers in the double bottom tanks, and as each double bottom tank became empty, it was filled with water and the vessel was kept to a draught of about 12 feet forward, and the best possible draught aft, perhaps 15 feet. Good passages were made with this draught, and no difficulty was ever experienced with water in the fuel.

A reference was made to the case of the coal-burning "Ocean" class vessel, which had been almost immune from screwshaft failures. This vessel would be able to use all her ballast tanks and obtain more immersion. The case, as set out by the author, all pointed to the cause of the failures being insufficient immersion, with which he was in full agreement, particularly forward.

PROFESSOR G. VEDELER, D.Tech. wrote that to judge from the upper diagram in Fig. 3 the problem had been given sufficient consideration already to reduce the casualties for this type of ship down to perhaps a similar percentage as for other ships.

But still the number of screwshafts renewed each year for all types of ships was very large and had been so for many years. Dr. Dorey dealt with the question in a paper before the war. As far as he remembered, Dr. Dorey mentioned that tailshafts must break sooner or later, say after three or four years, if the shafts started to corrode due, for instance, to leakage at the rubber ring between liner and propeller. He was not aware that the usual type of a single rubber ring packing between propeller and liner had been improved lately. But he thought that this question was so important at least to ship owners, that marine engineers should try to find a design by which water admittance to the shaft would be avoided entirely, even if there was carelessness in the fitting of the rubber ring when fitting the propeller.

Would it be advisable to try a thin coat of some plastic over the shaft cone or at least over the first inch of the shaft abaft the liner, or could the opening between liner and propeller be sealed in an efficient way, for instance by metal spraying after the propeller had been fitted?

He wondered if some suitable body might be willing to offer a premium for the best solution of this technically small but rather important question? There might be many people employed by marine engineering firms and by dockyards who had given the question some thought and one of whom might be able to suggest just the right solution.

MR. H. BOCLER wrote that in the preparation of his paper on "Ballasting of Cargo Ships" to which the author referred, the main considerations he had in view were effective control of ship in all weathers and sufficient immersion of propeller for reasonable propulsive efficiency. The author had added another important consideration, the avoidance of shaft breakages.

It seemed to him shipowners might have to accept some increase of cost in the direction of increased size of ship to meet desired cargo capacity requirements in conjunction with larger water ballast tanks to secure deeper immersion of ship in the ballast condition. In the case of oil burning ships facilities should also be provided to use the fuel tanks alternatively for water ballast, and engineers would have to face any complication of operations involved thereby.

Care must be exercised that with any loading arrangement devised to secure propeller immersion sufficient draught forward was also maintained.

It would be beneficial if the whole question of ballasting were now considered by some competent body and general instructions drawn up for the guidance of those designing and operating cargo ships.

MR. L. BAKER, D.S.C. (Member) wrote that the author had given a complete summary of the work in which many companies had contributed their experiences. The Blue Funnel Line had indeed been among the earlier ones in trouble.

It might be of interest, however, to know that the limitations imposed on the ships resulted in the elimination of further lost propellers and since that date the opportunity had been taken to replace the tailshafts by ones of slightly larger diameter and the propellers by solid ones of manganese bronze and considerably lighter construction than those originally fitted.

The following table gave some service experience of the old and new conditions.

	Speed, knots	I.H.P.	R.P.M.	Slip, per cent	Draught	Fuel Consumption, tons day
Voyage 1, June 1947 (unestimated) ...	12·09	2,319	70·4	−4·9	22 ft. 11 in.	27·93
Voyage 2, March 1948 (restricted) ...	11·63	2,240	68·8	−3·6	20 ft. 9 in.	25·04
Voyage 4 (modified)	11·61	2,210	68·0	2·5	21 ft. 7 in.	26·83

In connexion, however, with the American built ships, it might be of some significance to record that considerable trouble had been experienced by his company with the liners on American tailshafts. In a very large number of instances these had been found to be either porous or split axially, suggesting that in some instances at least, the workmanship had left something to be desired.

Dr. G. H. Forsyth (Local Vice-President) wrote that the conclusions arrived at were precisely the same as found "by luck" during the war on a small triple expansion engined vessel, on which four crank failures had occurred. The word "luck" was used because it so happened that the trials were carried out in very bad weather, light ship without even bunkers, excepting those for trial purposes. The results indicated clearly that when the propeller came out of the water, two things happened:—

(1) The engines raced, the normal speed of 65 r.p.m. increasing to 154 r.p.m.

(2) During this racing, the line shafting went through all the various critical speeds, the magnitude of the resulting stresses being greatly increased owing to the absence of propeller damping.

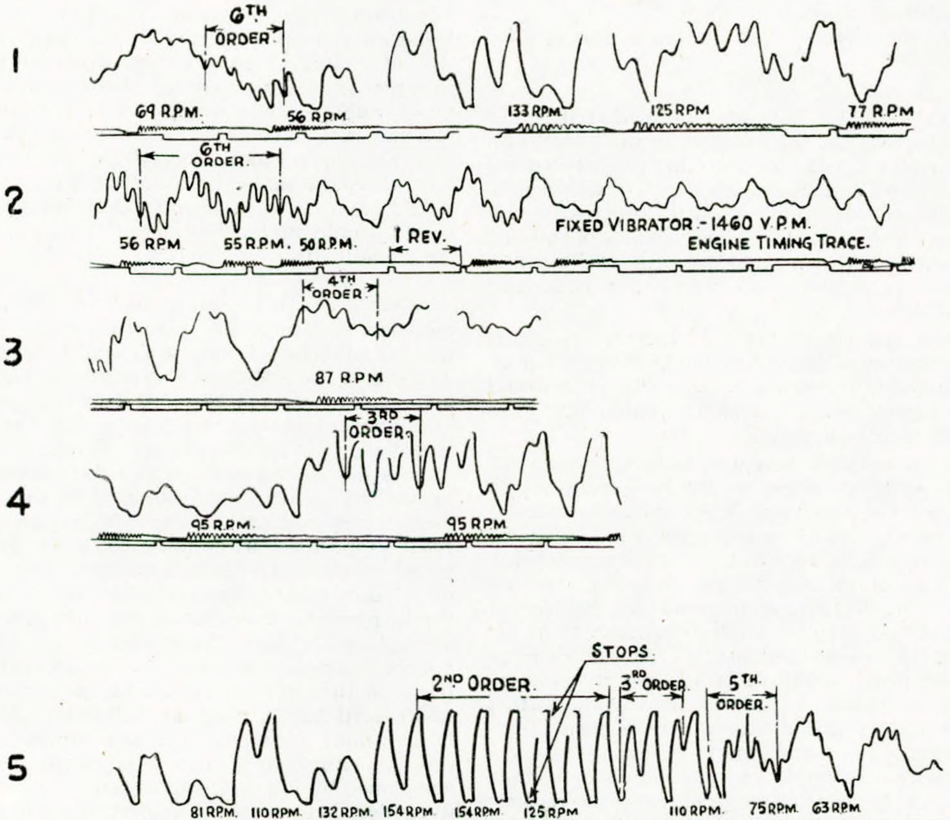


Fig. 27—*Geiger torsiograph records taken under conditions of rough weather on a vessel fitted with steam reciprocating triple expansion engines*

All readings were taken just aft of the thrust block with rapidly varying r.p.m. and the propeller not fully immersed. 1, 2, 3 and 4 show third, fourth and sixth orders of vibration; 5, 6 and 7 show second, third and fifth orders of vibration. Shaft diameter, 10 inch. Magnification 1: 1.

The natural frequency of oscillation between the engine and screw shaft amounted to 339 v.p.m. and the third order, also the flank of the second order vibration, were apparent during this racing, and seemed to build up in a matter of two or three oscillations to maximum amplitude.

The maximum stress in the 10-inch diameter intermediate shaft was in excess of 10,000lb. per sq. in., the instrument at this point failing to record the maximum vibration because it was "hitting the stops".

Taking the acceleration of the speed surge into account, this stress was further increased to something greater than 13,800lb. per sq. in. On returning to port, it was found that the shrink grip of the l.p. crank had failed, the records therefore represented actual conditions which had produced failure.

This vessel was subsequently fitted with a governor, the setting of which was such as to prevent the third or fourth order vibrations being reached during racing.

No special arrangements were made for ballasting this vessel, and it is interesting to note that, as mentioned by the author, no further failures occurred.

Fig. 27 indicated the vibration records taken during these trials.

MR. G. M. SELLAR, B.Sc. (Member) wrote that it might be of interest to supplement the information in the paper with the record of 70 "Canadians" which were built to British Corporation class. All ships had Scotch boilers. Twelve were fitted with ordinary streamlined rudders and the remainder with Goldschmidt rudders. During 1946-47 twelve cases of cracked tailshafts were reported, of which eight were in coal-burning ships and four in oil-burning ships: all twelve ships concerned had Goldschmidt rudders.

The rubber sealing arrangement of the "Liberty" tailshafts was described in the paper as being of standard type, but so far as observed at surveys, it consisted originally of a round rubber ring in a square recess, which combination was uncommon in British merchant practice.

It was suggested further that there was, in fact, no standard sealing arrangement, although there would be advantages in adopting as a standard for this vital detail the most efficient and foolproof arrangement of the many types in use. Some types had the rubber ring fully enclosed, some partly enclosed, and others left the ring unprotected. When designing a sealing arrangement it was desirable to have in mind (*a*) that rubber which was stretched would in time develop permanent set; (*b*) that rubber was practically incompressible; (*c*) that rubber rings would be renewed and fitted to place at three-year intervals by repair staff at different drydocks throughout the world; (*d*) that the replacement rubber rings would vary widely in their "free" dimensions, as allowances for fitting might be minus 1½ inch, or ½ inch on the diameter for stretch, and plus ⅛ inch or $\frac{1}{16}$ inch for axial compression depending on the practice of the repairer concerned.

The ill-effects of (*a*) would be minimized by confining the rubber ring in a recess in the propeller boss and by adopting a ring of section which would completely fill the recess. The results of (*b*) and the conditions of (*c*) and (*d*) required that provision be made for the possibility of an oversize ring being supplied at some time.

The sealing device of the "Liberty" tailshaft was shown at the foot of Fig. 5, from which it could be seen that there was an annular gap leading from the recess containing the rubber ring. This gap provided relief for any excess rubber trapped in the recess and was $\frac{3}{16}$ inch deep in the "Liberty"s. In other ships with comparable arrangements this gap varied from ⅛ inch to ⅜ inch according to the practice of different builders. The larger size would seem to be preferable.

From the foregoing considerations there was much to commend the adoption of an external packing gland for sealing purposes as this fitting was more likely to maintain the shaft watertight with a ring which was undersize, oversize, or with one which has been used previously. It also eliminated the possibility of a slack propeller resulting from the fitting of an oversize ring.

Reference was rightly made in the paper to limitations of the non-anticipating type of governor as confirmed during the trials of the S.S. *Clan Macbeth.* It was known that anticipating governors had been fitted to some of the "Liberty"s and many of the "Canadians" and it was understood that they had proved satisfactory in eliminating racing from the viewpoint of the ship's staff. It would be of interest if the author would state his own views on the effectiveness of an anticipating governor of the hydrostatic type in relation to this screwshaft problem, and whether shaft failures had recurred in ships so fitted.

There was also reference to clamping stresses at the top of the shaft taper due to liner and propeller boss. In this connexion the installation of an oil lubrication system for the stern tube bearing and the fitting of a bare tailshaft 1½ inch larger in diameter suggest themselves as a means of removing the liner clamping stress and of greatly improving tailshaft stress conditions in general. In the event of such a conversion being considered it should not be overlooked in the design of the oil sealing device that sea temperature variations resulted in differential expansion of ships hull and line of shafting, which might amount to ½ inch or more variation in the distance between face of propeller boss and after-end of stern tube.

The conclusions reached by the author from his most comprehensive investigation of the causes of screwshaft failures would seem to be justified, and if adopted for "Liberty" type vessels would undoubtedly result in more reliable service from the screwshafts.

MR. A. HILEY wrote that of the several precautionary measures set out at the end of the paper, the one dealing with the curtailment of keyways would carry its special appeal wherever a possibility of effecting any reduction of risks to the vital part of the tail shafting came into sight.

The circumstances under which these large vessels were laid down in such considerable numbers would entail a choice of sites in tidal locations which were usually regarded as liable to variable settlements, especially when appreciable loadings were superposed, as on the keel blocks closest to the water. Piled support would doubtless be given in all instances as was usually essential for the launchways. For the remainder, owing to the demands of time and economy, expansive areas of berth would normally be rafted as the shipwright might arrange, to average out between 2 and 3 tons per sq. ft.—which meant, possibly, double that intensity would occur towards the stern frame of the ship. If these conditions were found to result in undue settlements, reaching well over 1 inch, troubles through local strains to which the hull girder must submit, partly through efforts to correct alignments, might often be largely occasioned by non-uniform nature of weak pockets prevalent in waterlogged substrata. The persistance of internal strains starting through such uncertain causes might account in some measure for those unfamiliar influences referred to by Mr. Low, which, as he remarked, took their toll in a way that, seemingly, calculation fails to forewarn. Of course, it might also be thought that unusual strains tending to become intensified by conspicuous rigidity restricting the axis of the shafting, as at the stern-post bossing, could only be detected by strain gauges of the piezo-electric quartz crystal type. As he knew by experience, these gauges were indispensable in investigation of impact forces such as also must be withstood by the tail shaft. A description of these gauges will be found in Technical Paper No. 20, issued by the Building Research Station, Garston, in 1937 and obtainable from H.M. Stationery Office. In regard to assessing what influence the ground qualities and bearing capacity on building berths might exert, as a minor but possibly important factor, it would be of interest to learn if any steps were taken to preserve the designed alignment of shaft and likewise the line of keel. It was true that a considerable amount of attention has been given in this country during recent years to the design of apparatus capable of determining the available bearing capacity of ground; also to the provision of bases prepared accordingly, at fixed points from which accurate levels could be checked.

Should a marine engineer who was concerned with the design of propeller shafts desire a knowledge of natural settlements in relation to typical ground resistances as tested *in situ,* concise information, as from one shipbuilder to another, would be found in *The Structural Engineer* of January 1938.

With regard to the question of curtailing the keyway recess in a "Liberty" ship's shaft and using a shorter length of key in the propeller boss, the need for this, as a practicable step to conserve better what safety was possible was shown by the indisputable evidence given by Figs. 8 and 12; and, to inquire further into the possibilities that emerged when a start was made in this direction, he would, with due respect to orthodox practices, submit the following comments.

Taking first an ordinary step, and assuming that the keyway showed signs of cracks, and the key itself was somewhat crushed and, therefore, had become slack: a new shaft would then have to replace the original and a new key would also be provided. Would the author please say whether, if, in the event of renewals, the new keyway and its key were shorter than before by an amount equal to one-fourth of the full diameter of shaft, this reduction would be acceptable, generally speaking, or could it with advantage go much farther? The enormous reduction in the factor of safety of the tail shaft caused by the mere presence of any recess of any kind that can augment the ordinarily estimated torsional stress, raised problems of the most troublesome kind; for the torsional stress was then no longer uniform around the shaft, which was thereby exposed to an inevitable intensification of shear-stress concentration that must be made still worse near its extremity, where violent side pressure could be delivered by the key. For some extreme conditions which propeller shafts must endure in service, as when propellers break surface for more than half of their disk area, the only satisfactory means of ensuring safety would appear to be a changed form of constructional detail.

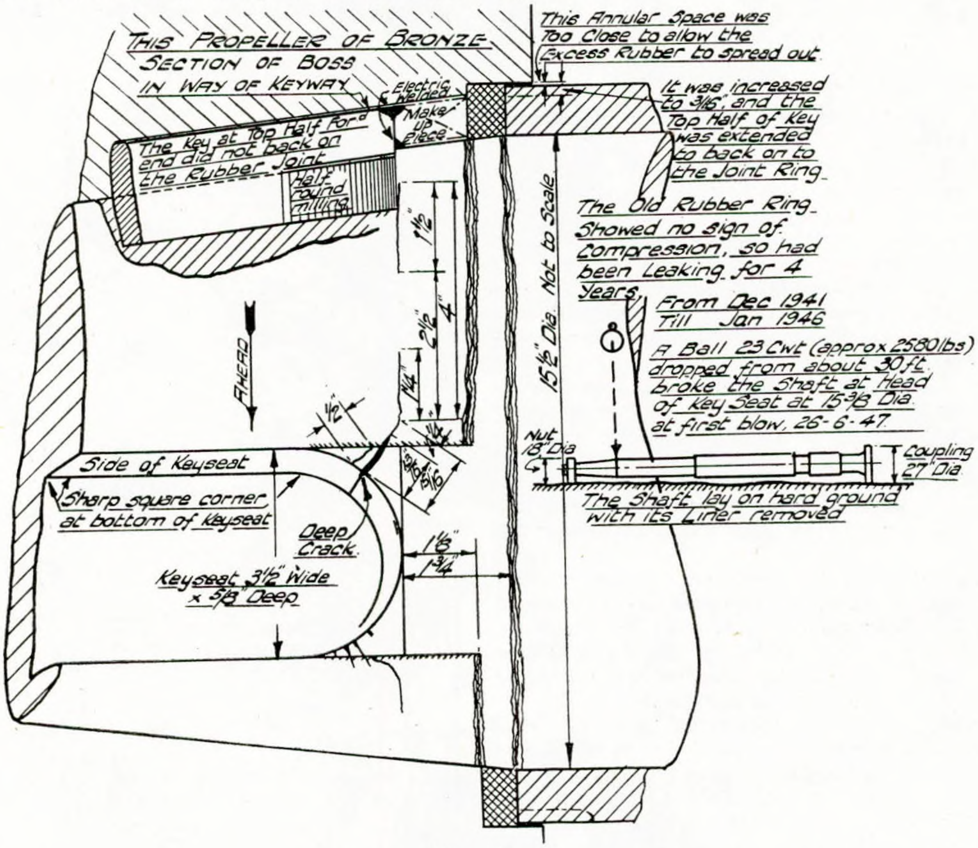


FIG. 28

An example which could be followed with confidence of success, and had completely overcome difficulties of a similar kind experienced in connexion with extractor gear, might, he suggested, be considered in the following form.

Proceeding on this principle, all torsional shear stresses (and, for the most part, thrust as well) would be provided for by fitting a forged steel torque-disk to the rear face of the hub of propeller casting to transmit these known forces, as through a ring of electric welded connection around the periphery of the protruding tail shaft. Neither grooves nor keys would then be needed, and the tapered shaft would be of plain circular section throughout the length of the hub. The latter, some 30 inch in extent, would be made an accurate fit around the tapered shaft for some 6 inch nearing the full 15¼ inch diameter. This would permit relative tangential movement between the circular hub-bearing and the shaft, and confer that freedom which was so very necessary. Forced lubrication within the hub would be desirable.

The crudity and unsuitability of keys of rigid steel to cope with torsional strains of a fluctuating character, and their liability to develop slackness in contrast to the method proposed would be manifest. The steel disk forging would be formed with its inner edge, contiguous around the shaft, suitably bevelled to receive the welding deposit. When this was completed and cleaned off the propeller end-nut would be replaced.

The chief duty that would be required of this nut would have been discharged when it had (just previous to the welding being performed) served to push the hub firmly up to its correct position on the shaft.

This end nut, it would be seen, would require to be shortened somewhat to allow of thickness of the torque disk being interposed, securely attached to the hub and virtually integral with it. To transmit a turning moment of 160,000lb-ft., eight steel stud-bolts would suffice, arranged in the disk on a pitch circle of a convenient 12-inch radius.

Favourably to this suggested detail, the "Liberty" ships happened to have features of stern frame and type of rudder which could not be better to facilitate the handling or removal of the propeller and tail-shaft assembly when required.

There was no technical difficulty in removing the ring of weld material to detach a shaft from the disk or connecting the parts as before.

If, on survey, some "Liberty" ship was found to have her screwshaft preserved in good condition, an enterprising owner might possibly agree to test this proposal and, if so, would not take much risk by utilizing the old tail shaft, despite the handicap from the presence of its existing recess and bolt hole.

Where propellers run well submerged (and the submergence should include an allowance of, say, 1 per cent of a vessel's length to cover waves and pitching) the general practice in fitting a keyed connexion to the shafting, using longitudinal keyways, would not usually be called into question. Where, however, extremes prevailed, and in a variety of adverse conditions were productive of the hammer-blow effects to be expected when propeller blades came above the water-line, the safer annular form of torsion key should be preferred. It should be borne in mind, too, that mechanically enmeshed teeth can be employed on new shafts to facilitate assembly, instead of the welded shear ring proposed as an emergency fitting on existing propeller shafts. If the author had statistics available to indicate which breakages of shafts could be ascribed to inadequate propeller-blade immersion, perhaps he would care to draw the line showing the extent of immersion that corresponded with the safe employment of common keyways, and beyond which the suggested torsion key might be used.

Mr. A. Watt (Member) wrote that the type of cracking shown on Figs. 8, 10, 11 and 12, was not confined to "Liberty" or "Ocean" classes of ships. It had been found in other merchant ships and in tankers having much larger diameter shafts. The cause, however, was usually a defective joint arrangement between the propeller boss and the aft end of the tail shaft liner.

Fig. 5 showed the detail of the sled runner keyseat at its forward end. The propeller keyseat has a radius of $\frac{3}{32}$ inch but the sled runner end apparently had a sharp corner at each side. This lack of side radius was confirmed from an examination of Figs. 8, 10, 11 and 12.

When leakage of sea water past the joint ring took place—especially with a bronze propeller—corrosion cracking commenced at the sharp corners at the ahead side of the keyseat as indicated clearly on Fig. 10 and 11, and showed the futility of a sled runner end when the sharp corners were left at the periphery of the shaft taper. In fact it showed that not sufficient thought and care was given to the design of this part of the machinery.

Fig. 5 also showed a detail of the recess in the propeller boss for the joint ring. With a deep recess as shown, it would be much safer to have a recess of larger diameter than that shown—it appeared to be about $\frac{1}{4}$ inch larger in diameter than the aft end of the liner. If good pliable rubber was used, and the thickness had been carefully measured, this $\frac{1}{8}$ inch annular space should be sufficient to permit the excess rubber to spread out. But in many cases where the recess was a closer fit over the liner it had not allowed the propeller to grip hard on the taper of the shaft. In other cases the rubber joint had been just a neat fit in the recess, but not tight enough to prevent water eventually leaking past, and so starting corrosion cracking at the big end of the taper. An example of this was shown in Fig. 28 in the case of a 15½-inch diameter shaft. Figs. 29, 30, 31, 32 and Plates 7 and 8 showed the fractures.

Sometimes the keyway of the propeller had cut through into the recess so that two pin holes were left for water to enter.

In his opinion, the details of this joint recess should be shown on the plan of the tail end of the screw shaft, also a view showing the forward end of the keyway in the propeller and recess diameter, so that this could be considered and dealt with when the shaft was submitted for approval by the owners and the surveyors. This also applied to the more usual key having a half-round seat at each end of the tail end.

With both the sled-runner end and the half-round end keyseats, the sharp corners at the big end of the tail shaft cone should be well rounded in order to reduce the stress concentration at the forward end of the keyseat due to the combined bending and torsion.

This was shown on Fig. 33, Type A and Type B keyseats, and it would be noted that the forward end of the keyseat in both these designs was shown well abaft the end of the tail shaft liner.

It was noted with interest that the author—under (C) "The Strength of the Keyed Propeller Assembly in Fatigue" made a definite suggestion that the minimum distance between the end of the keyway (or keyseat) and the end of the liner should be, say one-sixth of the screw shaft diameter. This figure would be 2¾ inch for the 15¼-inch diameter screw shafts of the "Liberty" and "Ocean" class ships. That was about 2 inch greater than that shown on Fig. 5, and was a step in the right direction.

The 2-inch grip of the propeller boss on the uncut portion of the tapered tail end should help considerably to reduce the bending stress at the weakest and most sensitive point on the shaft.

It was also noted that the author considered the round-ended keyway in a tail shaft was slightly stronger than the sled-runner type.

The round-ended keyseat also had the advantage that it was simpler to machine and it was easier to fit the key in its seat.

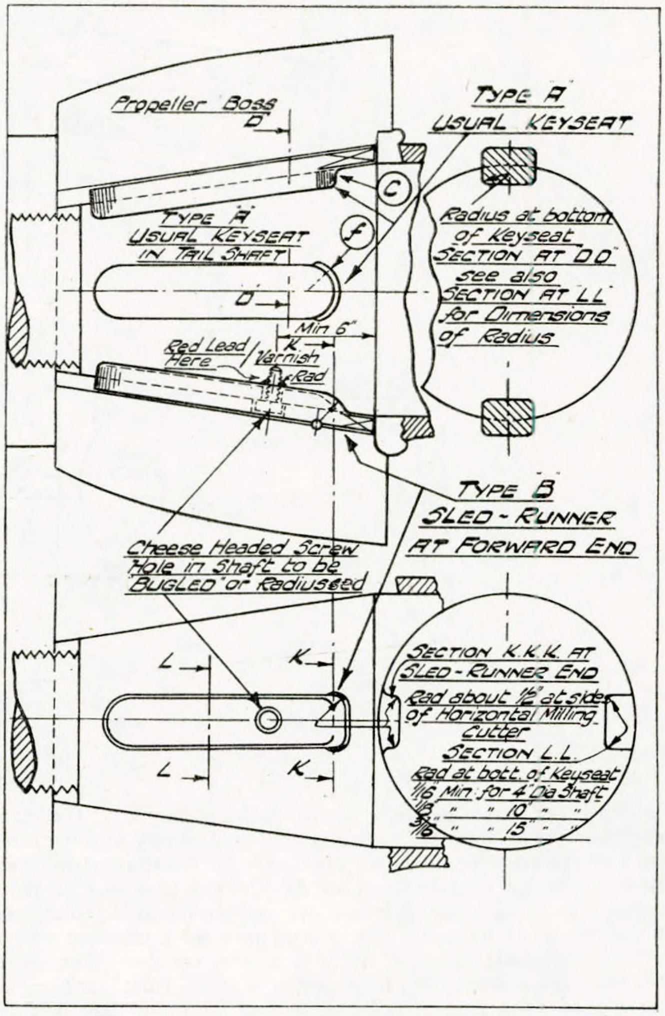


FIG. 33—*Type A and B keyseats*

Under "General Precautions", the author suggested (1) the fitting of an outside sealing gland to the propeller boss, preferably additional to the original arrangement. This, in his opinion, was not a desirable fitting as additional to a good design of joint embodying the usual rubber joint ring. The gland was liable to be disturbed should fouling by a wire rope occur. In such case, the surveyor was in doubt as to the watertightness of the joint.

The other three suggestions were important precautions which should be taken by all responsible people.

Author's Verbal Reply

MR. ARCHER said that would be difficult to make detailed reply there to the many points raised, and he would, therefore, endeavour to deal only with those questions to which he could perhaps give fairly ready answers.

He thought most people would agree with Dr. Ker Wilson that the marine propeller shaft could with advantage be improved in the direction of reducing stress concentrations, but to derive the maximum benefits would require most careful tests on a large scale. As Dr. Livington Smith had pointed out, such tests were in progress or shortly to be commenced. Then Dr. Wilson asked for particulars of the types of governors that were available for controlling racing. Of course, they were all familiar with the common reciprocating inertia type of governor, whose limitations were obvious in that it so very often "shut the stable door after the horse had gone!" The drawback to this and similar devices was the amount of steam in the engine receivers which had to be used up before there was any appreciable speed reduction. Dr. Wilson's suggestion to fit an exhaust steam turbine was a thoroughly promising one, and he left it to the superintendent engineers to look into that problem, because he believed that thereby they might be able to combine shaft protection with improved efficiency.

Mr. Gatewood suggested that Table I did not include sufficient statistics to be conclusive. He would admit that they were not able in all cases to obtain as much information as was wanted. However, the Table could be regarded as indicative, in a representative way, of the service conditions of the "Liberty" class as a whole; i.e., it did not follow that because a ship had a draught of 20 feet on her casualty voyage she had the same draught on previous voyages. The lesson to be learned from Table I was that the class as a whole had been operating under too light draughts. As regards "Liberty" tanker statistics, he felt it was not possible to assess the weight of Mr. Gatewood's evidence sufficiently without further details of the actual service conditions. For example, it was known that these tankers liked to discharge their ballast in good time before arrival in port, and they might, therefore, be very light over a period of some hours. They knew that the weather off the American eastern seaboard could be very onerous at times, and he was not altogether convinced that some damage could not have been done in that period. Again, they did not know the condition of the shafts, whether there was corrosion fatigue and possibly there were stress raisers of one kind or another. Perhaps Mr. Gatewood would be able to give more information on these points.

Another question concerned governors. How effective were they? He would like to know whether any tests had been carried out with the anticipating type of governor described by Mr. Gatewood, using tachometers under racing conditions at sea, because they had had similar types from which much had been anticipated, but which for some reason or another in most cases had never really caught on. Could not the occurrence of casualties after fitting these governors be equally well taken as evidence of prior damage to the shafts or of ineffectual governing? Then there was the matter of "Liberty" crankshafts. Mr. Gatewood pointed out that the shrinkage allowance was somewhat less than assumed in the paper. He himself estimated it would amount to about 20 per cent less grip, but the conclusions arrived at would not thereby be seriously affected.

He was sure that members would note with approval Dr. Livingstone Smith's statement on the very close collaboration between the British Shipbuilding Research Association and Lloyd's Register of Shipping, particularly in the large-scale fatigue testing of shafting.

On the efficiency of current shrinkage allowances for crankshafts, Dr. Smith had suggested that for marine shafts there might be plastic flow at certain values. He would, however, point out that the allowances generally adopted today had resulted from years of practical experience and it might well be Dr. Smith would ultimately find that in this case experience was the best mentor.

Reference was made by Professor Burrill to two different propeller designs, one with a service speed above the present 76 r.p.m. and the other below. His own personal preference was for the second design. With the higher service speed in heavy weather one was always liable to be forced down on to the critical. He noticed that the mean torque in the shaft in the second design with 2,500 i.h.p. had been increased above that approved in the original installation. He had not worked out the increase, but his opinion was that a slightly augmented mean stress would be acceptable.

It was interesting that Professor Burrill had been able to effect a reduction in moment of inertia; this, of course, was a good thing, particularly since it also increased the propeller damping somewhat. He was also grateful to Professor Burrill for pointing out that the propeller power loss varied as the depth of the immersion and not the area. Professor Burrill then referred to the gap left outside the liner in the propeller boss recess. He would say that without such a gap the rubber would definitely become volume-bound; it was most important to ensure that there was sufficient gap left outside the liner to permit the rubber to flow, and the propeller boss to seat properly on the shaft taper.

A gouging effect underneath the rubber ring was also mentioned; that was, of course, an all too common result of corrosion fatigue.

Not being a rudder designer, he did not propose to enter into a discussion of the Goldschmidt rudder, which had come in for quite a lot of criticism; but he would only point out that the "Ocean" class ships were fitted with identical rudders, and so far as he knew, there had been no screwshaft failures in that class, apart from the one ship which was converted to oil-fuel burning, so that he could not believe the rudder in itself had an effect in causing these failures.

Mr. Murray felt that more experience should be obtained of the speed reductions and various other measures which had been recommended before it could be said what was the cure. He would like to point out, however, that these recommendations had now been in operation for twelve months or more, and he considered enough information had been gathered in that time to see that the avoidance of operation at the 3rd order critical was having an effect, even though there had been a very substantial reduction in the number of "Liberty" vessels in service. At the moment only about 60 per cent of the original fleet was in operation, and that fact of course should be borne in mind. Another factor might be that during the 1948 winter the weather generally was probably not nearly so severe as that in 1947.

He did not intend to argue with Mr. Murray about the relative advantages of increasing the length of the deep tank abaft the machinery space or of fitting the deep tank abreast the tunnel. However, it was known that some of these after-hold tanks were fitted in a number of fairly similar vessels in the British emergency programme and, so far as he was aware, proved to be quite satisfactory.

Then Mr. Murray mentioned the hull vibration during the racing test. One of the reasons he felt that a lower service speed was advantageous was that the speed of 72-74 r.p.m. was right on a hull critical when in ballast, and it could be rather unpleasant. By running at 66 r.p.m. that was avoided.

He was interested in the information given by Mr. Nicolaisen concerning the special type of damper which he developed and fitted on a "Liberty" ship. But he would like him to make certain in the designs that the securing arrangements were adequate to deal with the high 2nd order stresses which might be imposed on the damper.

Some weighty remarks on ballasting were made by Mr. Daniel. He felt rather diffident about taking up with him at this stage the criticisms made. He would like, however, to support the suggestion that there was a decided need for investigation into this important matter. He would remind members that in 1903 a Select Committee of the House of Lords was set up to examine the practicability of applying a light load line; but he understood that finally it was considered to be impracticable. It was difficult to forecast what the results might be if such a committee were set up again today.

He did not think that tank tests would be sufficient in themselves to prove what was the optimum propeller immersion. There could be no doubt that sea conditions were the best criteria of that. Personally he would like to see complete propeller immersion, preferably in the arrival, rather than the leaving condition.

Mr. Morris was well known as an authority on coupled modes. He could not agree, however, that the question of propeller blade flexibility really entered into the problem, as there had been no doubt as to what the critical speed had been; it was possible to judge that with a fair degree of accuracy, and he did not know of any appreciable number of propeller blade fractures. Perhaps Mr. Low could give information from America on that matter.

Mr. Low interjected that they had had some but they were boss fractures

Mr. Archer, continuing, said that there was no fundamental difference of opinion between the two sides of the Atlantic.

He had been asked by Mr. Ferguson to confirm that it was his conclusion that, given adequate propeller immersion, no restriction of revolutions would have been necessary. His own feeling was that if he were a shipowner he would nevertheless avoid running at the critical speed. But one had to remember that the "Ocean" class ships had not been subject to this trouble, and they had been running on the same critical. However, it was really in the design stage that the problem of the coincidence of critical and service revolutions should be dealt with.

He was greatly indebted to Mr. Jarvie for his very kind and stimulating remarks. It was interesting to know that Mr. Jarvie, presumably a North Atlantic man himself, realized what it meant to operate a throttle in heavy weather. He also had seen how these "Liberty" ships' screws raced in high seas; he had spoken to the masters and engineers of quite a number of these ships, and he was sure there was sufficient evidence that this class of vessel was unusually prone to severe racing. He would not minimize for one moment the importance of hand-fitting and attention to machining details in the propeller assembly; these were matters of the utmost importance, and surveyors and engineers could not pay too much attention to having them just right. However, they had studied the problem very closely and had now endeavoured to put forward what appeared to be a logical explanation for this extraordinarily high casualty rate; it was only by hard practical experience that they would be able to judge whether or not it was justified.

He had no information at the moment on the steel used in these screwshafts, beyond that which was given in the paper, but in view of the fact that so many different forges and steelworks were involved, he did not believe that the quality of the steel could be one of the primary factors.

This paper was really born at a meeting held at Lloyd's Register of Shipping about eighteen months ago, when Dr. Dorey and Mr. Harry Hunter, also Mr. Adam and Mr. Sellar of the British Corporation Register of Shipping were present. The question of these "Liberty" failures was keenly discussed, and it was then agreed that Lloyd's Register of Shipping should be used as a sort of clearing house for all information on the subject. In presenting this paper he had tried to summarize and analyse the information which had come into their possession, and if that could lead to a satisfactory solution of the problem their task would have been accomplished.

Author's written reply

Whilst the good record of the "Ocean" vessels indicated that the design of the "Liberty" tailshaft was, *per se,* probably no worse than that in many other vessels, there was little doubt that the usual form of propeller assembly was by no means ideal. Dr. Ker Wilson had put forward some very interesting suggestions for improving the mechanical design of this vital part of the tailshaft, probably to some extent inspired by experiences such as that quoted from aero-engine service. It was concluded the fatigue loading in that case was principally due to torsional oscillation. The origin of the crack at the sharp edge of the keyway in Fig. 24 (Plate 5), and the presence in that region of frettage corrosion, bore striking resemblance to a number of cases of marine tailshaft fractures where there had been other evidence to show that the fit of the propeller at the large-end of the cone had been poor. For this reason he would support Dr. Wilson's suggestion (*a*) to provide a slight flat at the top end of the keyway. The importance of suggestion (*b*) should be sufficiently obvious, but unfortunately was often ignored. Proposal (*c*) for a slight differential fit between hub and shaft cones raised interesting considerations. Reflection showed that with equally tapered bore and shaft and uniform boss thickness, then on the assumption that the compression of the shaft was negligible compared with the expansion of the boss (E for bronze was only about 45 per cent of that for steel), the fractional interference fit resulting from a given axial movement of the boss on to the shaft varied inversely as the diameter, d, along the taper. Further, if the Lamé formula was taken as approximately applicable and neglecting the effect of blades, the ratio of the grip pressures per square inch of contact surface at large and small ends of the cone was given by

$$\frac{p_1}{p_2} = \frac{d_2}{d_1} \times \left(\frac{1 - 1/k^2_1}{1 - 1/k^2_2}\right)$$

where the subscripts 1, 2 referred to large and small ends respectively and k was the ratio, outside radius of boss to shaft radius at any section. Clearly, k_1 was less than k_2, and thus the bracketed term was less than unity, as also of course was d_2/d_1. Thus, if equal tapers were used with uniform boss thickness, the specific grip pressure at the large end of the cone relative to that at the small end was reduced on two counts, namely, that resulting from a smaller interference fit and that due to a smaller ratio of boss thickness to shaft radius. If it were accepted as desirable that the final specific grip pressure obtained after hardening up the propeller should be fairly uniformly distributed along the shaft, it would appear preferable to taper the boss thickness down towards the after end in such a manner that the product of the two factors in the above equation tended to unity, and this would then permit the use of equal tapers for boss bore and shaft. In the case of the "Liberty" propeller, for approximately the same mid-length boss thickness, this would call for about 15 per cent reduction of the boss outside diameter at the after end, together with an increase of about $8\frac{1}{2}$ per cent at the forward end, i.e., a steeper taper on boss thickness. With such a design a uniformly distributed grip pressure was theoretically obtained independently of the axial load forcing the boss on to its seat. With a differential taper this would not, in general, be true and, further, there was a greater danger of setting up excessive clamping stresses at the more sensitive forward end of the shaft cone. Also, by tapering the boss to give a greater thickness at the large end, the hoop stress was reduced at that section and thus the danger of causing a slack fit of the key in the driving end of the boss keyway was minimized. There would appear to be a fruitful field for research here, making use of modern strain gauge technique.

The provision of a circumferential groove to relieve clamping stresses, as suggested in (*d*) and shown in Fig. 26, was interesting, but the cavity under the end of the liner was not favoured on account of inspection difficulties. It was felt that almost as good results could be achieved by a carefully executed, very slight bell-mouthing of the propeller boss bore, also of the liner end before shrinking. It was, however, strongly agreed that the dangers of clamping stresses were insufficiently appreciated. In particular, the effect of a line of clamping stress discontinuity intersecting the sharp upper edges of a shaft keyway was believed to be fraught with danger, and, as suggested in the paper, it would seem only prudent to arrange for the shaft keyway to terminate some way aft of the clamping edge of the propeller boss recess, and in any case the latter, together with the edges of the keyway, should be well rounded off. The desirability of more attention to these points might be emphasized by reference to Fig. 12, showing the location of the cracks in the S.S. *Bendoran's* shaft, which was typical of a number of others. He was glad to note Mr. Watt's agreement in these matters in view of the very wide experience Mr. Watt had had in the survey of screwshafts; also the emphasis on the need for adequate fillet radii at the forward end of both round-ended and sled runner key seats; and his other practical suggestions.

On the question of Dr. Wilson's item (*e*) there seemed to be some division of opinion as to what constituted an improved rubber seal. British practice was on the whole, but by no means exclusively, in favour of the design shown in Fig. 5, whereas on the Continent and in the United States the external gland was in common use, and, in fact, the American Bureau of Shipping had recommended that for "Liberty" ships the sealing arrangement should be changed to that method. In view of Mr. Watt's objections on the score of vulnerability to fouling by wire ropes, etc., which incidentally could be minimized by suitable shrouding or, for example, by the use of sunk-in Allen screws, it would seem well worth while, and at very little extra expense, to provide a double seal using both methods conjointly as recommended in Appendix V(*b*)(i). This extra insurance against leakage was practised by at least one well-known Continental firm of shipbuilders.

Mr. Sellar's analysis of the practical and design requirements of an efficient sealing arrangement would repay careful study. It was significant to note the preference for the external gland with its less exacting demands on the skill and care of shore repair staffs. As regards the type of rubber ring used for the "Liberty's" it was understood these were originally round but later changed to square section. Professor Vedeler's suggestion to coat the shaft surface with a protective film of plastic was a good one, and to his own knowledge was practised by at least one famous British company in combination with an external gland, the coating being confined to the forward end only. It would be interesting to see if some public-spirited body would take up the challenge contained in Professor Vedeler's final paragraph and, if so, whether the necessary "brain wave" would be forthcoming.

With Dr. Wilson's final suggestion (*f*) he was in full agreement, but would only repeat the opinion that the forward key-retaining bolt hole could well be omitted altogether.

Mr. Watt suggested that the round-ended keyway was believed by him to be slightly stronger than the sled runner type. It should be pointed out, however, that this was merely a tentative inference from some rather inconclusive small scale tests in reversed bending and, as stated in the paper, until large scale tests proved otherwise, there seemed little to choose between the two types.

As Dr. Livingston Smith had indicated, the vexed question of the relative torsional fatigue strength of the round-ended and sled-runner types of keyway was to be investigated by Lloyd's Register of Shipping as part of the research programme of B.S.R.A. Such tests would, of course, be inconclusive for the marine propeller application unless the effect of clamping stresses was simultaneously explored, and Dr. Smith rightly pointed out that the torque would require to be applied as in the actual assembly, it being fully agreed that very different results would be expected if the torque was merely passed through the shaft with the clamping applied to the keyway in the middle. Both Petersen and Thum used the latter method in their reversed bending tests quoted in the paper, which made their results even more inconclusive when referred to an actual full-size propeller assembly. In reply to Mr. Hiley's question on the minimum length of keyway required, it might interest him to know that theoretically it would be possible to halve the length of the keyway and still leave sufficient shear area, but the bearing stress on the sides of the keyway would then be excessive unless its depth were increased. This latter would, however, have the undesirable effect of increasing the stress concentration in the shaft. As stated in the paper, it was considered that a shortening of the shaft keyway by about 2 inch at the forward end would definitely increase the strength of the assembly, but that was as far as he would be prepared to go. With a well-fitting and correctly assembled propeller the key should not normally be called upon to withstand torque stresses at all, and its function could be regarded as in the nature of an insurance against shocks, etc., similar to the dowel in a shrunk crank-web. That this was so could readily be inferred from the following rough calculation:—

Mean torque corresponding to 2,200 s.h.p. at 76 r.p.m. $= 1{\cdot}825 \times 10^6$lb.-in.

With a 1 in 12 taper on shaft the magnifying or "drift" factor would be just over 12, i.e., if it were a flat wedge, an axial load of 1 ton on the propeller boss would give a force of just over 12 tons normal to the wedge.

The sectional area of propeller screw at base of threads = 104 sq. in. approximately.

Then, assuming a mean cone radius of $7\frac{1}{4}$ inch and a coefficient of static friction of 0·2, the minimum mean tensile stress required in the threaded portion of the propeller screw to avoid slip was given by

$$f = \frac{1{\cdot}825 \times 10^6}{7{\cdot}25 \times 0{\cdot}2 \times 12 \times 104}$$
$$= 1{,}045\text{lb. per sq. in.}$$

If a torsional vibration stress of ± 4,000lb. per sq. in. were superimposed on the full load mean transmission stress, the additional torque would amount to

$$\frac{\pi \times (15{\cdot}25)^3}{16} \times 4{,}000 = 2{\cdot}78 \times 10^6\text{lb.-in.}$$

and maximum torque $= (2{\cdot}78 + 1{\cdot}825) \times 10^6$
$= 4{\cdot}605 \times 10^6$lb.-in.

and corresponding maximum value of

$$f = 1{,}045 \times \frac{4{\cdot}605}{1{\cdot}825}$$
$$= 2{,}630\text{lb. per sq. in.}$$

With a 2nd order torsional vibration stress of ± 12,500lb. per sq. in. acting alone, as when racing, the value of

$$f = 2{,}630 \times \frac{12{,}500}{4{,}000} \times \frac{2{\cdot}78}{4{\cdot}605}$$

$$= \pm\ 5{,}000\text{lb. per sq. in.}$$

The important part played by the frictional forces in a well-fitted propeller assembly was thus apparent, and he firmly believed that in many cases it was only when the propeller assembly was subjected to unusually heavy impacts or vibratory effects that the frictional grip might be overcome with resulting transfer of the dynamic loads to the keyed connection, and, if sufficiently severe and prolonged, might lead in turn to fretting corrosion, progressive slackening, and stress concentration at the ends of the shaft keyway, finally resulting in fatigue cracking and failure.

Mr. Hiley's proposals for improving the marine tail end design were very ingenious but hardly likely to appeal to the practical marine engineer, who would not view with favour the necessity for removing a ring of welding each time the propeller was backed off. Nor would he wish to provide forced lubrication to avoid fretting corrosion at the forward end and thereby lose the whole benefit of the excellent frictional grip inherent in a conical seat.

The question of governors was raised by several contributors. Dr. Wilson asked what types of governors were available for controlling racing. These might, broadly, be divided into two classes, the non-anticipating and the anticipating. The former of course required for their action the prior development of a definite rise in speed. Both the reciprocating inertia and the rotary or Watt types were too well known to require description. The latter was rarely seen in main steam installations, but with spring control was in common use for oil-engine governing.

The underlying controlling influence with most anticipating governors was the change of inclination of the vessel. In one type this was utilized directly to cause a ball to roll along a pivoted beam, thereby opening or closing a ball valve placing the condenser vacuum in communication with one side or other of a throttle controlling piston through a vacuum-operated relay. In another variant the change in pressure head of water at the stern of the vessel was utilized to operate the throttle valve through a relay. In one design a sea cock at the stern admitted water into an air vessel and compressed the air to a pressure equivalent to the head of water outside the ship. The rise and fall of a float in an open-topped cylinder or well having a lower connexion to the sea had also been tried as a control mechanism in conjunction with an electrical relay on the engine. In some anticipating designs an emergency governor actuated by rise of speed was also incorporated to deal with certain conditions of short steep following seas or, for example, a broken shaft. Theoretically, the anticipating governor might be expected to be more efficient, as the throttle could be made to close before the speed surge started. However, with a propeller already little more than half immersed he had yet to be convinced that any considerable advantage was obtained over the speed governor when correctly adjusted. A direct comparison in high seas on a ballasted "Liberty" vessel fitted with both types and with an accurate tachometer to record speed surges would go far towards clearing up this controversial question. Mr. Gatewood claimed that because a number of failures occurred after installing the anticipating governor, and in three cases new shafts were fitted concurrently, it might be inferred that racing was not the principal or a necessary factor in these failures. He himself strongly disagreed with this inference and would suggest that the reason was rather to be found either in previous fatigue damage during earlier racing or in insufficiently effective governing, probably when inadequately ballasted, with or without some predisposing influence such as stress raisers or corrosion fatigue. Mr. Gatewood's supporting evidence of freedom from failure of other shafting was no more convincing when it was remembered that the torsional fatigue strength of an intermediate shaft was of the order of ± 15,000lb. per sq. in. (see Bibliography, reference (4)) and the torsional grip strength of a shrunk crank web was probably at least 18,000lb. per sq. in. in terms of screwshaft shear stress, allowing for the smaller shrinkage quoted by Mr. Gatewood and neglecting the dowel strength. These figures allowed ample margin for very severe screwshaft stresses without endangering the remaining shafting. Direct corroboration of this argument could be adduced in several cases where tailshafts coupling bolts had worked slack during heavy weather with clear indication of hammering in their bolt holes. In this connexion the following extract from the log of an oil-burning Canadian "Victory" type vessel during a North Atlantic ballast voyage might not be out of place:—

13th Dec.: 0800 Wind SW force 6. S/S rolling, pounding, pitching and labouring heavily . . . 0900 revs. reduced to 54 . . . 2000 Wind SW force 7-8. S/S pitching, pounding and labouring heavily . . . S/S steering badly and falling off at times.

16th Dec.: 1200 Wind S force 7-8. S/S pitching, rolling and pounding heavily . . .

19th Dec.: 1200 Wind NNW force 6-7. S/S rolling and pitching heavily . . .

25th Dec.: 0800 Wind NNW force 6-7. S/S rolling, pitching and pounding heavily . . . 1000 engine stopped, engineers repairing tailshaft. 1100 under way full ahead.

26th Dec.: 0400 Wind SSE force 7. S/S rolling and pitching heavily, ship's head falling off course. 0800 Wind S force 8. Gyro swing between 240 and 280. S/S rolling and pitching very heavily, helm hard to port, ship not steering all watch . . . N.U.C. signal hoisted.

27th Dec.: 0400 Wind NW'ly force 6-7. S/S rolling and pitching heavily in trough . . . N.U.C. lights burning. Gyro swinging between 180 and 220. 0500 full ahead.

The tailshaft repair mentioned was undoubtedly the hardening up of the coupling bolts, as these had subsequently to be renewed owing to slackness with evidence of hammering.

He was glad to note Dr. Smith's concurrence with the principal findings of the investigation and was grateful for the results of the "pushing out" test on a large built-up oil engine crankshaft, from which it would seem that the figure of 0·25 assumed in the paper for the static coefficient of friction in the shrink grip was conservative, and perhaps Mr. Gatewood would note this. As Dr. Smith rightly pointed out, it was quite possible that different results would be obtained under dynamic conditions of pulsating torques, and the results of the further experiments on this point would be keenly awaited by marine engineers. Since, however, his own calculation was purely comparative as between the "Liberty" crankshaft and that quoted by Dr. Forsyth, the absolute value of the frictional coefficient was not considered important for that purpose.

Professor Burrill seemed to think that the root cause of the screwshaft breakages was neither torsional vibration nor lack of propeller immersion. However, as repeatedly stated in the paper, the evidence clearly pointed to the combination of a number of factors, and, in his opinion, the principal causes were those mentioned.

Mr. Jarvie held the view that the principal factor had been bad workmanship, and attempted to explain the immunity of the "Ocean" class as by virtue of their having been built under Lloyd's survey. He was grateful for this graceful tribute, but, unfortunately for this tempting argument, the first intimation of trouble reached the Society on account of a spate of failures in oil-burning vessels of the Canadian "Victory" and "Canadian" types, of which a considerable proportion were built and engined under Lloyd's survey. There had also been sufficient numbers of other cases reported with no evidence of leaky seals, bad workmanship or fitting. It was for this reason it was found necessary to probe more deeply into the whole matter and to bring the results and conclusions to public notice.

Mr. Jarvie seemed to doubt whether the differences in draught between the "Oceans" and "Liberty"s as calculated in Table VI were of sufficient consequence to matter. As pointed out in the paper, however, the comparison would show even more to the advantage of the "Ocean"s where, as was usual, coal bunkers were taken at the start of the ballast "leg" (say United Kingdom or Continent) instead of refuelling at the loading port as was normal practice with the "Liberty"s.

Calculations based on these more realistic service conditions showed the following results:—

"Liberty"

(a) Condition X

Taking Condition II as a basis but substituting 760 tons S.W. ballast in No. 3 deep tank for the 700 tons of oil fuel (less fuel was required for North Atlantic short voyage). Also assumed that 270 tons oil fuel were consumed from No. 2 D.B. tank on loaded passage from United States. This was ballasted departure condition from United Kingdom or Continent to Eastern United States ports.

(b) Condition XI

As in (*a*) but with remainder of No. 2 and whole of No. 3 D.B. tanks burnt out, equal to 275 tons oil fuel consumption, and not replenished with S.W. ballast. This was common ballast arrival condition in United States ports.

"Ocean"

(a) Condition IV

Taking condition I as a basis but omitting 955 tons coal in No. 3 lower hold (short voyage). This was ballasted departure condition after bunkering at discharging port, say, United Kingdom or Continent, for round voyage to United States eastern ports (6,000 miles plus seven days reserve).

(b) Condition V

As in (*a*) but with 400 tons coal burnt from cross-bunker (ten days steaming at 11 knots). This was ballasted arrival condition at, say, United States loading port.

Condition	"Liberty"		"Ocean"	
	X	XI	IV	V
Ballast displacement (tons)	5,927	5,652	6,679	6,279
Mean draught (S.W.)	12ft. 10in.	12ft. 4in.	14ft. 4in.	13ft. 7in.
Trim by stern ...	6ft. 3in.	6ft. 10in.	10ft. 0in.	10ft. 2in.
Draught forward ...	9ft. 9in.	8ft. 11in.	9ft. 4in.	8ft. 6in.
Draught aft	16ft. 0in.	15ft. 9in.	19ft. 4in.	18ft. 8in.

On the basis of these further figures it must surely be concluded that in the case of the "Ocean" class racing would be considerably less frequent or prolonged and maximum speed surges much lower and more easily controlled than in the "Liberty" class.

Some criticism had been levelled at the statistical data of Table I. Mr. Gatewood pointed out that in seven cases out of forty-four ascertainable the leaving draughts aft exceeded Sir Amos Ayre's recommended minimum of 20ft. 7in., and in twenty-six out of forty-four they equalled or exceeded Bocler's figure of 16 feet. It was instructive to examine these figures more closely, remembering that the Table not only illustrated the starting conditions of the casualty voyage but equally gave a picture of the minimum ballast draughts prevalent in the class as a whole. The seven cases quoted by Mr. Gatewood were all in the loaded or part-loaded condition and therefore had no bearing on the question of the prevailing minimum *ballast* draughts. The reason fractures occurred in these cases was, in his opinion, because fatigue damage had been done during racing on previous ballast voyages. Deducting these seven cases from Mr. Gatewood's twenty-six, left nineteen cases with draughts aft equal to or more than 16 feet. Of these, one case carried 2,000 tons of sand and another 1,166 tons cargo, and were thus not properly representative of ballast conditions. Another four vessels had forward draughts varying between 6 feet and 7ft. 2in. only with draughts aft of between 16 feet and 17ft. 6in. Clearly for manageability under North Atlantic conditions trimming down by means of the forward ballast tanks would be essential with probable loss of immersion aft. Thus, strictly, only about thirteen cases out of thirty-five applicable equalled or exceeded Bocler's figure. Of these thirteen nothing was known of the forward draughts in six cases, so that there are possibilities that similar trimming action would have been needed. In any case he was not at all convinced that Bocler's standard was edequate for this class of oil-burning largely welded vessel, and, for safety, draughts should probably be increased towards Sir Amos Ayre's recommended minimum figures. In reply to Mr. Daniel's complaint about lack of precise information on this important subject it was agreed there was a definite field for practical research.

It was clear that the necessary propeller immersion was inter-related with stem immersion and the two must be considered together. For a given displacement the best compromise must be struck by the master to suit wind and weather. The sailor was powerless, however, if the design was at fault, with inadequate or incorrectly distributed ballast capacity.

Model tests might be useful for preliminary study purposes, but the final requirements ought to be established by scientific observations of behaviour at sea. In the meantime for the "Liberty" vessel he would suggest that 90 per cent screw immersion on arrival coupled with a stem immersion of not less than 35 per cent of loaded mean draught would be a desirable target, i.e., say, 18 feet aft and at least 10 feet forward. Comparison with the "Ocean" class and the U.S.M.C. wartime ballasting practice for "Liberty"s would seem to indicate that such a target was not unreasonable. In this connexion it was interesting to note Mr. Murray's opinion that, apart altogether from propeller shaft stresses, the draughts could in many cases be increased with very great advantage to the ship. On the practical side the author was very grateful to Captain Thomson for his corroborative evidence on the subject of "Liberty" ship ballasting, particularly as to the dangers of attempting an east to west Atlantic crossing with forward draughts of 6 to 7 feet. Although Captain Thomson quoted a case of oil-fuel tanks being replenished with S.W. ballast, this practice was by no means common on account of practical difficulties of tank cleaning and oil-fuel contamination.

He noted with interest Captain Thomson's considered opinion that better ballasting in these vessels was the way to end screwshaft troubles. He was also grateful to Mr. Bocler for his support on the need for improved ballasting.

Mr. Logan's wide experience in the operation of oil tankers shed further light upon the problem, and it was significant that their comparative immunity was attributed to the better propeller immersion obtainable with that class of vessel. In fairness, however, he would point out that with steam reciprocating machinery fitted aft, the 2nd and 3rd order torsional critical speeds would usually lie well above the maximum likely speed surge, and only the higher and less severe torque harmonics were likely to achieve resonance when racing. On the other hand, it was certain that the better screw immersion would protect the shafting from undue bending stresses.

Dr. Forsyth's valuable contribution provided excellent corroboration of the dangers of uncontrolled racing due to the loss of propeller damping, and also showed the extreme rapidity with which the various critical vibrations built up, which was confirmed by himself during the synthetic racing test on the S.S. *Bendoran*.

N.B.—Dr. Forsyth stated that crank failures had occurred on *four* occasions, but according to his paper (Bibliography, Ref. No. 10) the number was given as *six*.

It should be noted that in Dr. Forsyth's vessel the shrinkage grip of the after l.p. crank web was increased by some 20 per cent in addition to fitting a governor.

These precautions, coupled with the fact that the screwshaft was appreciably oversize, doubtless accounted for the vessel's subsequent immunity from further shaft failures.

He was grateful to Mr. Gatewood for supplying the missing data in Table III, and was glad to note agreement with him that repeated heavy overloads during racing may cause fatigue damage capable of resulting in failure at normal service rating. In this connexion it was significant that Mr. Jarvie was able to report failure in at least one shaft at the reduced revolutions of 68 per minute, and others were known to himself.

In one "Liberty" ship a new shaft was fitted in November 1947. On her first voyage (return trip to Europe) her log showed an average speed of 74 r.p.m. from port to port (22 days). On subsequent voyages engine speed was reduced to approximately 66 r.p.m. The propeller was lost at sea on 8th May 1948. The engineer's log contained the entry, "engine racing at intervals", for each of the four days preceding loss of the screw. The fatigue crack was subsequently found to have started just at the end of the liner. There was no evidence of corrosion.

The two points raised by Professor Burrill had been considered, and on the question of loss of propeller torque due to incomplete immersion [Appendix III(*a*)] the following calculated results were given for the S.S. *Bendoran's* draught of 13 feet aft:—

Torque reduction based on loss of area = 33 per cent.
Torque reduction based on loss of diameter = 36·5 per cent.
Torque reduction assumed in Appendix III(*a*) = 37·5 per cent.

The difference was thus seen to be negligible, as would be expected at immersions approaching the half disk condition.

Professor Burrill also criticized the method of calculating the propeller d.h.p. given in Appendix III(*a*). He would point out, however, that the only reliable data at his disposal were the vessel's ballasted speed and corresponding revolutions, the indicated power being unknown. There would, therefore, seem to be no option but to assume a wake fraction suitable to the vessel's lines and ballasted displacement. Naturally, if the power had been known, a more accurate estimate of the propeller torque would have been possible.

Regarding the vibration damper designed by Mr. Nicolaisen, he was informed that the torsiograph tests and service performance of the first damper were so successful that a repeat order had been placed by the United Steamship Co. for a sister vessel. He hoped that Mr. Nicolaisen would later publish a description of the damper in the technical press, since he believed it might be found to be capable of application to other types of machinery.

The comparative service results quoted by Mr. Baker with the old and new designs of propeller were welcomed, but it would perhaps be premature to judge the results on so few voyages.

Finally, his thanks were due to Mr. Ferguson for his support, particularly as confirming the large measure of agreement between the two sides of the Atlantic on the practical methods of overcoming the troubles experienced.

CORRIGENDUM

Page 64, equation (24) should read:—

$$\bar{\theta} = \pm \sqrt{[\theta_F{}^2 + \theta_A{}^2 - 2\,\theta_F\,\theta_A \cos \epsilon]}\ \text{(rad.)}\ldots\ldots\ldots(24)$$

Fig. 21 (d) and (b) should be transposed in lettering above the diagram.

INSTITUTE ACTIVITIES

MINUTES OF PROCEEDINGS OF THE ORDINARY MEETING HELD AT THE INSTITUTE ON THE 8TH NOVEMBER, 1949.

An ordinary meeting was held at the Institute on Tuesday, 8th November 1949 at 5.30 p.m. G. Ormiston (Vice-Chairman of Council) was in the chair. A paper entitled "Boiler Corrosion and Boiler Feed Water Treatment" by H. Hillier, O.B.E., M.I.Mech.E. (Member) was read and discussed. 180 members and visitors were present and eight speakers took part in the discussion.

T. A. Bennett, B.Sc. (Member) proposed a vote of thanks to the author which was accorded with acclamation.

The meeting terminated at 7.30 p.m.

JUNIOR SECTION

Lecture at Acton

An appreciative audience of approximately 220 attended Mr. J. E. M. Payne's lecture on "Marine Engine Room Auxiliaries" at Acton Technical College on the 10th November 1949.

Mr. Payne introduced his subject by stating that marine auxiliaries were actually installed in ships before the advent of their propulsion by motive power, and to emphasize his point he mentioned such equipment as bilge pumps, winches, etc., used in sailing ships. He then went on to describe in detail the progress in connexion with the design, etc., of steam, electrical and Diesel marine auxiliaries over the past sixty years—the various points raised being illustrated by a very full range of lantern slides which commenced with details of the auxiliary equipment installed in the S.S. *Great Eastern* and terminated with details of auxiliaries installed in the latest addition to the Cunard fleet—the *Caronia.*

A lively question session ensued and Mr. Payne spent a full hour answering questions on such widely divergent subjects as two-stroke Diesel engines and torque converters, etc.

A vote of thanks was proposed by a member of the staff of Acton Technical College, which was enthusiastically received by the audience.

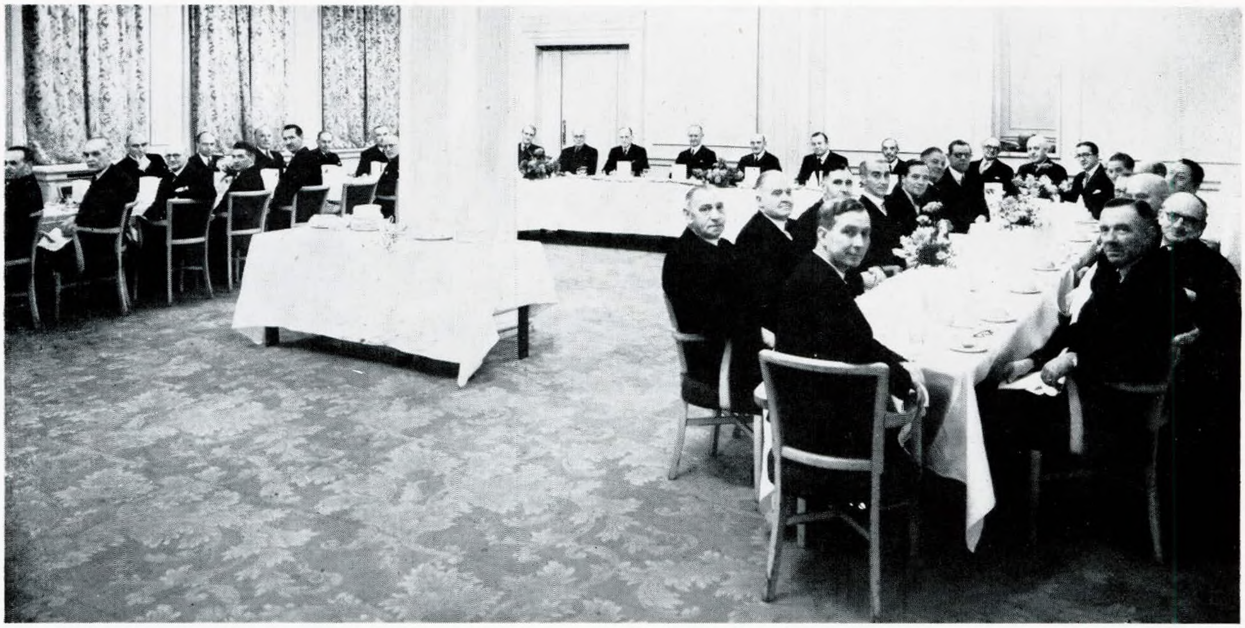
The meeting terminated at 10.20 p.m. at which late hour Mr. Payne was still besieged by questioners.

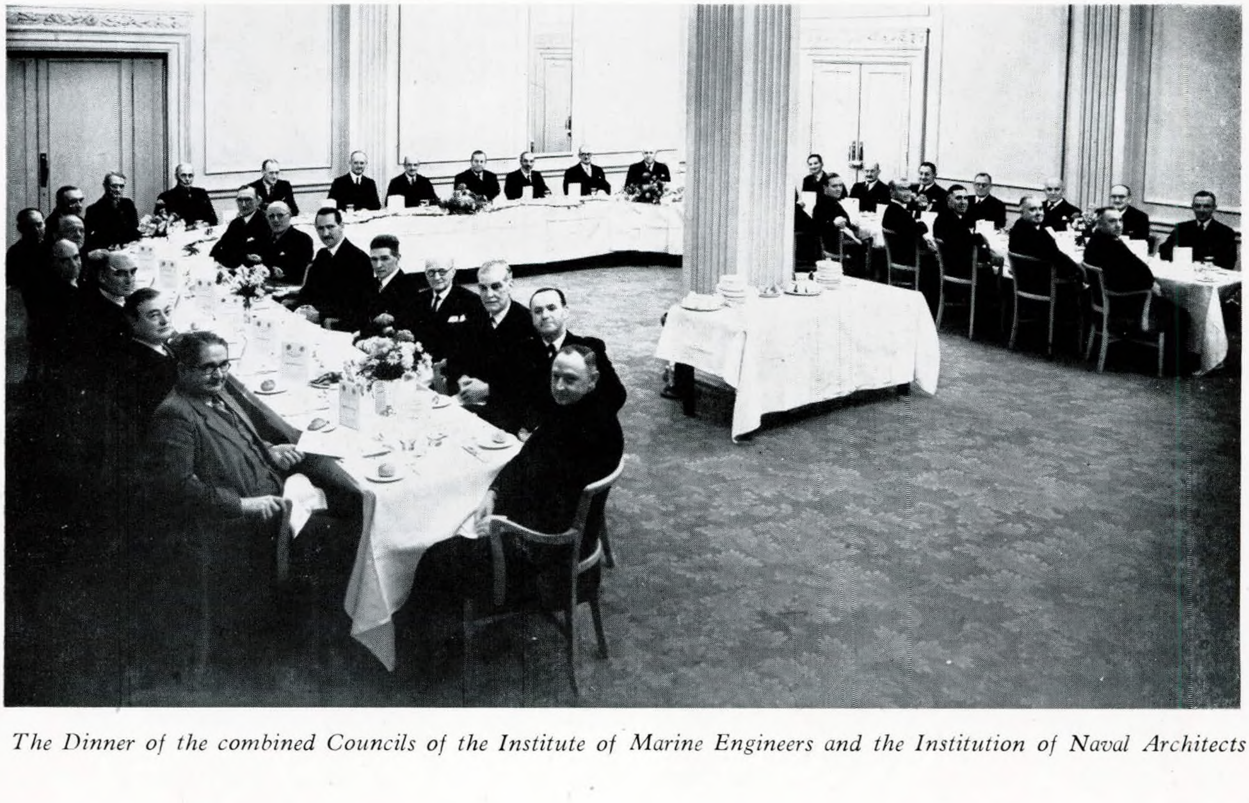
Lecture at Greenock

In conjunction with the Greenock Association of Engineers and Shipbuilders, Mr. Broatch, B.Sc., on behalf of the Institute delivered a lecture on "Modern Inspection Technique" at Watt Hall, Greenock, on Tuesday, 13th December. Lantern slides were used to illustrate the lecture and showed the various types of measuring tools used to give the high degree of accuracy required for the various components in mass production. A very interesting film supported the lecture and emphasized the necessity of keeping tools in perfect condition to ensure accuracy in the products. Mr. Broatch ably dealt with the many questions which ensued.

Mr. A. G. Akester, President of the Greenock Association was in the Chair and Mr. D. R. Hutcheson proposed a vote of thanks. Mr. M. McAffer (Local Vice-President) represented the Council.

Annual Conversazione at Connaught Rooms





The Dinner of the combined Councils of the Institute of Marine Engineers and the Institution of Naval Architects