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*Some Notes on Welded Ship Construction.

By JAMES TURNBULL (Member of Council).

The Author's Reply to the Discussion.

It is often said that the value of a paper is largely dependent upon the discussion that follows, and if that be accepted as the criterion in this case the author has reason to feel satisfied that the time spent in the preparation and presentation of the paper has been fully justified.

The intention of the opening remarks regarding important developments which have taken place in recent years was to record the accelerated rate of development in welded ship construction during the last three years, and anyone who is intimately associated with ship construction cannot fail to agree that the present stage would normally have taken many more years to reach. A well advanced state of development has existed in this country as a result of the progressive outlook of shipbuilders, and Mr. Norman Hunter's remarks on this subject have provided the answer to Mr. Burn's criticism of them.

Mr. Nelson endorses the statement in the paper that a good welded joint is superior to a good riveted joint and for various reasons he strongly advocates the adoption of welding. While he has obviously experienced irritating troubles through faulty riveting it is well to remember that poor quality welding is more dangerous than poor quality riveting, and therefore a wholesale adoption of welding can only be justified when good quality work is assured. Welding is, of course, especially attractive to tanker owners if only for the saving in the "out of service time". Mr. Nelson's interesting figures relating to the economics of weight saving from the owner's point of view demonstrate most ably the need for technical advisers to consider seriously the cost of construction in relation to possible increased earning capacity of a new ship before the type of construction is decided. Similarly Mr. Oxburgh's remarks have a bearing on this although practical difficulties may be met in fitting a tee-bar for the margin shell connection.

Mr. Norman Hunter considers that in welded construction a shift of plating butts should be arranged as for riveted work, a view which is now shared by many authorities on the subject, but it would seem that there is no need for the shift to be as great as two frame spaces.

The author agrees with Sir Amos Ayre that there is hardly any part of the ship's structure at which welding is to be preferred to hydraulic riveting on builders' economic grounds, except perhaps in the case of automatic machine welded bulkheads.

There has been general agreement with the items recommended to be of welded construction, and thanks are due to contributors for usefully detailing the list. Mr. Norman Hunter's list is of especial interest since it represents the practice in a particular shipyard where there has been economic justification for the welding of the items mentioned, although the author considers it is extremely doubtful if it is, at the present time, economical for any shipbuilder to weld seams as well as butts of all deck plating irrespective of thickness. A question of a similar nature, but with regard to machine welding, has been raised by Mr. Shepheard, but the author is still of the opinion that, where plating is over ½ inch in thickness it will generally be found more economical in this country at the present time to rivet seams and weld butts. It is certain that in course of time developments will take place to justify modification of the statement.

Attention has been drawn by Mr. Cocks to what appears to be an anomaly in the recommendations in the paper. It is true that an arrangement of double bottom structure has been suggested in which the frames are riveted to the bottom shell plating, and that all-welded construction is recommended in the slamming area. This latter recommendation, which includes the welding of the floors to the shell, is based on the satisfactory behaviour of welded construction under most severe service conditions, whereas with riveted construction trouble commonly occurs no matter how massive the structure. On the other hand the suggestion to rivet tank frames to the bottom shell plating clear of the slamming area was made in order to simplify construction and repair in a region where either method of construction is equally effective. It will be observed then that the recommendation to weld in the slamming area is one that should be adopted if inconvenience to builders and cost to owners are not too great.

Both Mr. Shepheard and Mr. Cocks consider that too much stress has been laid on the avoidance of welding inside confined double bottom spaces, whereas Mr. Chipchase and Mr. Patton are in favour of combined riveted and welded design to obviate welding inside the tank, and under present conditions, simplify the problem of repairs. While it is agreed that efficient welded joints can be made inside the double bottom, the author is of the opinion that welding in confined spaces should be avoided so far as is reasonably practicable, as there is no doubt that uncomfortable conditions for the operators do not have a beneficial effect on the general quality of the work produced.

Mr. Chipchase and others suggest that when new designs are being prepared, consideration should be given to the question of repairs, particularly in the case of the double bottom structure. This feature was considered in the suggested arrangements shown in Figs. 3 and 4, and in this connection attention is drawn to the existence of a design of all-welded double bottom structure, details of which have already been published†. It is thought that the form of construction referred to would simplify the repairer's problem and also accord with the suggestions put forward by Dr. Telfer and Mr. Day.

The arrangement of tank margin connection in Fig. 1, in which the flat bars shown are separated by a continuous margin plate, has been referred to by Dr. Telfer, Mr. Chipchase, Mr. Day and Mr. Cocks. There is no serious objection of having welded angles instead of welded flat bars, although the author considers the flat bar arrangement to be the better. Cracks that are experienced in riveted margin plates are mainly, if not entirely, due to the movement of the connecting angle bars relative to the margin plate when the rivets are being subjected to tension and compression along their shanks. The adoption of an intercostal margin plate would result in a better joint, but would introduce difficulties in construction not warranted because the more practical arrangement shown has already proved, and is likely to continue to be, satisfactory in service. The author does not share Mr. Cocks' opinion that no great advantage is to be gained by the adoption of welded flat bars in place of hydraulically riveted angles.

The question raised by Mr. Day regarding damage where gusset plates are welded direct to the knuckle of the cold flanged margin plate cannot be answered by the author because he has no experience of that particular type of connection, but considers it one likely to be

* Published in Transactions, Vol. LVI, Part 4, pp. 41-54.

†(See author's contribution to discussion on Mr. J. L. Adam's paper presented to the Institution of Engineers and Shipbuilders in Scotland in November, 1942).

a potential source of trouble. There is, however, no objection to butt welding gusset plates to the tank top, but the welded connection should be positioned on the flat of the tank top clear of the knuckle and with that arrangement the gusset plates should be continuous.

Mr. Patton has asked for the author's opinion on which of two specified methods in the repair of damage to double bottom floors is the better. The method preferred is to remove completely the fillet welds, fair the slightly deformed floors, and when re-welding this attachment to the bottom shell plating take precautions to ensure that any part of the floor plate which is grooved, due to burning at the time of removal, is made good by welding. There is no objection to welding over old weld metal as such, but care must be taken to ensure that the surfaces of the old metal are smooth and do not preclude a satisfactory joint being made.

Dr. Telfer has asked why there is no saving in weight by the elimination of overlapped seams on strength decks. The scantlings of strength deck plating in riveted ships have been proved by experience to be just adequate for service conditions over a long period of time, and there is no evidence to support the suggestion that in the more rigid all-welded ship a reduced effective sectional area would be satisfactory. The statement may, however, require modification from time to time as a better understanding of the present problems is acquired, but at the present stage in development it would definitely be imprudent to save weight by reducing material in such an important member of the main hull girder. The Classification Societies give consideration to sheer strain and, in fact, it might be gathered from the paper that certain items, where riveting is recommended, indicate awareness of this factor. In this connection it is noteworthy that the "Guidance Notes on the Application of Electric Arc Welding to Ships' Structures", recently published by The British Corporation Register of Shipping, encourages the adoption of riveted side shell seams in ships of 400ft. length and above. A similar suggestion has been made by Mr. Jaffrey.

The author is not prepared to agree fully with Dr. Telfer's views regarding the distribution of stress across the strength deck, as quite a large amount of data has been examined which does not bear out the contention that the maximum stress is in all cases and in all circumstances at the gunwale in spite of the fact that present widely accepted theories support Dr. Telfer. Progress towards totally satisfactory all-welded ships is likely to be more rapid when the whole subject of strains and stresses (more particularly strains) has been thoroughly investigated and understood.

Mr. Burn in all his arguments appears to have considered only one of the main design problems. Hull designers realise the great importance of fatigue, but have also to consider the shipbuilders' facilities and the purpose for which a ship is built.

The arrangement of gunwale shown in Fig. 5 has stimulated great interest and led to a large amount of comment. Dr. Telfer, Mr. Burn and Mr. Riddell suggest that the gunwale be rounded. During the preparation of the paper it was realised that a rounded gunwale was scientifically superior to the arrangement shown in Fig. 5, but its adoption was not considered to be justified at the present time as the additional work involved would be considerable and there is already sufficient evidence to support the practical suggestion shown in the paper. The Memorandum issued by the Admiralty Ship Welding Committee, with which some of the contributors are familiar, shows a similar arrangement. It should be specially noted, answering Mr. Jameson's question, that the failures which have occurred with the arrangement in which the sheerstrake projects above the strength deck could not be attributed to any fault in the joint of the stringer plate to the sheerstrake, but could be attributed to notches in the projection which, as Sir Amos Ayre and Mr. Shepheard emphasise, should always be avoided. The major reason for the inverted angle bar is to obviate this troublesome projection, and to produce a notch free arrangement.

Many contributors have supported the recommendation to rivet the gunwale bar in large ships, but Mr. Norman Hunter is doubtful of the complete efficiency of the suggested arrangement because of the slight stretching of the stringer angle during the process of riveting. The author is of the opinion that this stretching will have no appreciable effect under service conditions because of the accommodating behaviour of the riveted connection. Mr. Nelson too is not entirely in favour of the arrangement and would seem to prefer a welded connection, but experience has shown that, in large-welded ships with the present day distribution of structural material, the danger of failure is reduced by introducing, in the correct location in the main structure, a limited amount of riveting. His remarks regarding the avoidance of an overhead run of welding in strength deck stringer to sheer strake connections are not understood, as a butt joint in such a position would be considered unsatisfactory if a back run were not fitted, and in fact veed butt joints in any part of a ship should have a backing run, and the success which has attended the arrangement shown in Mr. Nelson's Fig. 11 for 2nd deck stringer connection can only be due to the low stresses generally experienced in that locality.

Practical experience has not shown that any great benefit is likely to be obtained from having symmetrical rolled sections. The one sided section is, in almost every case, much more suitable for the builder at any rate at the present time, and there is no evidence that tripping on account of the lack of symmetry will occur under ordinary service conditions. There is no doubt, however, that new rolled sections specially suitable for welded construction are desirable.

Mr. Norman Hunter's and Mr. Day's remarks regarding Fig. 6 are correct. The welded chocks between frames could be made in one piece, and in way of tanks the frame faying flange should be removed for a short distance on each side of the chock. The arrangement shown in Mr. Nelson's Fig. 10 would undoubtedly prove satisfactory in service, but it does not appear to be as simple a job for the builder as the arrangement where the chocks are intercostal between the frames. Both Mr. Oxburgh and Mr. Patton have had actual evidence of the success of the arrangement and it would appear to be well on the way to establishing itself as standard practice.

Mr. Camps, Mr. Jaffrey and Mr. Allan have drawn attention to the need for taking precautions to minimise locked up stresses by the adoption of correct welding procedures and erection sequences. They, and others similarly interested, are referred to the recently published Memorandum issued by the Admiralty Ship Welding Committee, which sets out the basic principles involved and also gives some examples of their practical application.

Dr. Telfer's suggestion to fit rounded hatchway corners in strength decks is endorsed, but the author cannot subscribe to the suggestion to fit a riveted doubler where it has been necessary to have a square corner. A similar arrangement has been tried and found unsatisfactory.

The great economy in weight which can be effected by the adoption of corrrugated bulkhead stiffeners has attracted much attention. As Mr. Cocks states, these bulkheads result in broken stowage on both sides of the bulkhead, but it is not thought that the third corrugation shown in Fig. 7 is greatly objectionable on that score and, considering all factors, it appears to be the most suitable form of corrugation for ordinary cargo ships. It is particularly interesting to learn from Sir Amos Ayre that it has been found cheaper and easier to rivet the joints connecting these corrugations to each other. Mr. Burn may be correct in his suggestion that the joints should be placed at or near the neutral axis, but the practical overrule the theoretical considerations, including fatigue, which is not in this case a major factor.

There would be a difference in the plating thickness of the various types of corrugation, assuming each to have the same moment of inertia, but that difference would depend to a large extent on the location of the bulkhead and the work it had to perform besides resisting simple pressure at right angles to the surface. Bulkhead plating in welded construction is usuallv of the same thickness as for riveted construction, as the panel of plating when pressure is applied to the stiffener side is not, as stated by Mr. Corin, 3in. greater in width than in the case of welded construction. The width is approximately the same, and in passing it should be mentioned that the Classification Societies' requirements for ordinary subdivision bulkheads are not appreciably in excess of those given in the Bulkhead Committee's report. It is agreed that, with longitudinal stiffening on the deck and shell, the end rotations are likely to be reduced and in consequence a higher degree of fixity obtained.

Mr. Spanner's "Rigidfix" connection is similar to a type of bracket now becoming a common feature in tankers. The "Uniflex" diaphragm system of bulkhead stiffening has some good features, but the author considers that, where the depth of stiffener is great in relation to the beam or frame spacing, the system loses its value.

Support for the adoption of the angle bar strut brackets shown in Figs. 8 and 9 has been given by many contributors, including Dr. Telfer who, however, asks if the arrangement has been properly tried in service. Sufficient experience has already been gained to show that it is unlikely that the struts will be severely damaged due to cargo handling gear. The possibility of such damage has been kept in mind when arriving at the scantlings of these angle struts. Mr. Burn is the only contributor who disagrees with the arrangement shown in Fig. 9. His suggested positioning of the lower end of the strut would undoubtedly result in cracking of the tank top plating.

The recommendation to dress smooth the exposed welding in way of stern frames, rudders and adjacent shell plating applies irrespective of the material of which the propellers are made. In reply to Mr. Cocks, it seems that corrosion of welds is accelerated by the flow of water towards and through the propeller, and incidentally, it should be mentioned that where overlaps of underwater shell plating have been welded, particularly in the region of the stem or stern, accelerated corrosion takes place in the weld metal.

In recommending welding of engine seatings, it was intended to include the heavy seating plate usually fitted under Diesel machinery. Although there may be something to be said in favour of the principle involved in Mr. Jaffrey's suggestion to rivet the ends of engine seating, experience so far does not indicate any necessity for this precaution.

Mr. Burn's advice to adopt machine welding in all cases in preference to manual welding requires careful qualification. Machine welding can be the more efficient when operating under favourable conditions, but several authorities have found it necessary to ban the use of machine welding for certain important joints. It is, however, probable that machines will improve and justify wider adoption.

The author agrees with Mr. Stephens' remarks regarding curved sections, and also that generally the yard should be adapted so far as practicable to suit changes in structural design.

The importance of supervision of welders has been emphasised. The Classification Societies are encouraging a high standard of supervision so far as they are allowed and Mr. Riddell may be sure that they would welcome any steps which would automatically remove some of the existing dangers. As Mr. Day has remarked, it has been suggested in some quarters that there should be one supervisor for every six operators. This question is not one that can be settled quite so simply; the number of operators to be placed under one supervisor depends upon the location of the work, degree of reliance which can be placed upon, and skill of, the particular operators.

Mr. Riddell considers that the yield point should be taken as the criterion for strength calculations. Data based on uniaxially loaded test specimens supports his remarks, and as applied to riveted ships has been found within limits to be compatible with experience. The author is hesitant, however, to apply these to a welded ship without qualification, in view of the absence of reliable strain data of welded ships as girders stressed in a seaway and of welded constructions under fatigue and multiaxial stress conditions. It does seem, however, that weld metal, or rather a welded joint, having the same mechanical properties as the parent metal is the ideal to be aimed at. It is agreed that the statement, based on ultimate tensile strength, that the weld is the strongest part of the joint is a misleading selling argument.

It is surprising that Mr. Jaffrey has found porosity in weld metal to cause serious leakage. This is a condition which should not exist to-day, and his suggestion to have a final run of dense non-porous weld metal is certainly not good practice. All weld metal should be non-porous. The trouble experienced in securing tightness at the tank corners would probably be overcome by chipping out the irregularly shaped end of each run before the succeeding run is laid down. This method is far more likely to be successful than endeavouring to fill a hole with weld metal.

In conclusion the author would like to record his deep appreciation of the kind references made by the contributors, to each of whom he is grateful for providing such interesting and stimulating criticism on a subject about which there is still a great deal to be learned.

EDITORIAL NOTE.—It is regretted that the sketches Figs. 12 and 13 on pages 52 and 53 *ante*, illustrating Mr. E. F. Spanner's contribution, have been inverted in printing.

Boiler Feed Water Regulation.

Read by H. HILLIER, O.B.E., M.I.Mech.E.

On Tuesday, May 9th, 1944, at 5.30 p.m. at 85, Minories, E.C.3.

Chairman : S. A. SMITH, M.Sc. (Chairman of Council).

Synopsis.

Reliable and efficient boiler operation requires correct regulation of the feed flow to the boiler. Manual control of the feed supply entails constant attention and intelligent supervision to obtain satisfactory boiler operation and any neglect is liable to allow the water level to fall too low or rise too high with possible serious consequences.

Automatic boiler feed water regulation eliminates such possibilities of neglect and maintains the boiler water level between predetermined limits. The benefits obtained are a reduction of operating personnel, lower maintenance costs of boilers, piping and engines and better operating efficiency.

Various types of boilers, feed pumps, feed systems and feed flow regulators have different characteristics, and, to obtain efficient and satisfactory boiler feed flow control, the relevant characteristics must be known and the controls designed and arranged to suit. The characteristics and their interactions are explained and illustrated.

The principles underlying the design and operation of the commonly used types of boiler feed pumps and boiler feed flow regulators and the arrangement of boiler feed regulators on the boiler to obtain satisfactory operation on board ship under the conditions of motion to be met at sea are discussed and illustrated.

General.

It is desirable to maintain the water level in a boiler within the limited range of the level that is visible in the boiler water level gauge glass.

Manual control of the feed supply entails constant attention and intelligent supervision to obtain satisfactory boiler operation, and any neglect is liable to allow the water level to fall too low or rise too high.

Too low a water level may cause damage to the boiler tubes through overheating, thereby causing increased boiler outage and maintenance costs, while an undue high level may cause priming with a high carry-over of moisture and solid matter which increases the maintenance required for the superheaters, steam lines and turbines or engines.

Automatic boiler feed water regulation eliminates such possibilities of neglect and maintains the boiler water level between predetermined limits. The benefits obtained are a reduction in the number of operating personnel, lower maintenance costs of boilers, piping and turbines, and the better operating efficiency which results from a more regular feed supply and better stabilised conditions of boiler operation. In many modern boiler plants, the volume of water in the boiler is small compared with the evaporative capacity of the boiler so that, if neglected, the water level can fall to a dangerous level in a few minutes. Reliability is, therefore, of paramount importance in a boiler feed water regulator, whether human or automatic, since the safety of the boiler depends upon the water level being maintained within prescribed limits. Automatic boiler feed water regulation operates continuously in adjusting the feed supply to suit the boiler steam load and, by reducing to a minimum the time lag between a change in the steam demand and the corresponding required correction of the feed supply, ensures the safety of the boiler plant.

Boiler Equilibrium.

When a boiler is in equilibrium, the heat transfer from the fuel to the water generates steam at a rate equal to the rate at which steam is taken from the boiler, and the feed flow into the boiler is equal to the rate at which the steam leaves the boiler.

The heat transferred from the fuel across the boiler heating surface adds sensible heat to the feed water entering the boiler until it reaches boiling point, then adds the latent heat to the feed water to cause the water to evaporate into steam. If the boiler is provided with a superheater, a further amount of heat is added to superheat the steam before it leaves the boiler. The heat acquired by the steam comprises three portions, therefore; sensible heat, latent heat and superheat.

If, for a constant rate of fuel combustion the rate of the feed flow into the boiler is varied, the amount of sensible heat required will be reduced if the feed flow is reduced and increased if the feed flow is increased. An increase in feed flow, therefore, leaves less heat available for latent heat and superheat, with the result that a smaller weight of steam is generated with an increased feed flow, and vice versa.

The equilibrium of a boiler can be disturbed by an increase or decrease in the rate of feed flow, as just described, or by an increase or decrease in the rate at which steam is taken from the boiler or by an alteration to the rate at which fuel is burned in the combustion chamber. Normally, the rate at which steam is drawn off from the boiler is varied to suit the demands of the prime movers. This is followed by a changing rate of feed flow, as will be described later, and the fuel combustion must, of course, be adjusted as quickly as possible to match any change in the rate of steam "draw off" so as to produce equilibrium again as quickly as possible between the various factors at the new rate of steam "draw off".

If the fuel combustion rate exceeds the steam demand, the boiler pressure rises and, conversely, the boiler pressure falls when the steam demand is greater than the rate of fuel combustion.

Water and Steam Volumes below the Boiler Water Level.

When a boiler is in operation, the gauge glass gives an indication of the approximate water level inside the boiler. The water in the boiler is not solid but is a mixture of water and steam bubbles in circulation, and the amount of steam contained below the boiler water level increases as the boiler load increases.

For any given rate of steam generation, there is a corresponding volume of steam below the water level, and the volume of steam below the water level increases as the steam load is increased.

When an increase in steam generation occurs, due either to increased fuel combustion or to increased steam "draw off" the first effect is to raise the water level due to an increase in the volume of steam below the water level. When the steam "draw off" is reduced, the water level falls due to a reduction in the volume of steam below the water level. These conditions are illustrated by Fig. 1. Automatic feed water regulators, controlled from the boiler water level and designed for continuous feeding, operate at a high water level H at zero boiler load and a lower water level L at maximum feed flow with an intermediate water level for any corresponding intermediate feed flow.

Assuming H L represents the boiler water level, as it falls with increasing boiler loading, at an intermediate boiler load O A, the water level is C and the volume of water in the boiler can be shown by the ordinate A B and the volume of steam below the boiler water level by the ordinate B C. At the maximum boiler evaporation, the water is shown by M K and the steam below the water level by K L. It will be seen that the actual volume and, therefore, weight of water in the boiler is gradually reduced as the boiler load increases, due to the action of the feed regulator and the

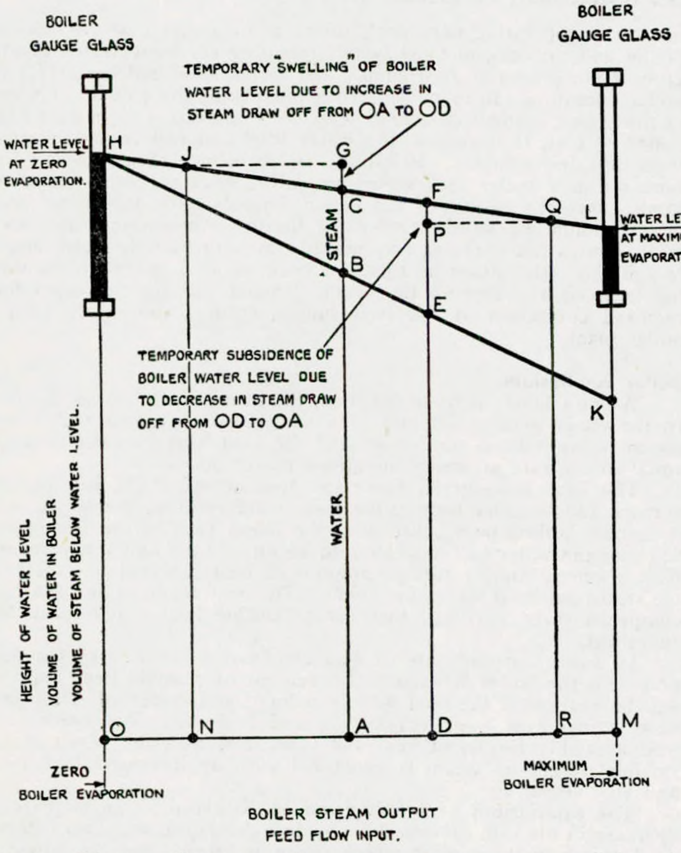

FIG. 1.—*Diagram illustrating the water and steam volumes below the boiler water level.*

fact that the volume of steam below the water level increases as the boiler load increases. This effect is considerable in water tube boilers and particularly in boilers, such as Naval boilers, which are small in size and are operated at high rates of evaporation. In the latest Naval boilers, the volume of water in the boiler at full power is of the order of two thirds of the volume of water in the boiler at zero load. In the case of such a boiler containing, say, twelve tons of water when standing, a change of load from zero to full power involves the evaporation of four tons of water while an abrupt change of power from full ahead to stop requires a feed discharge into the boiler of four tons of water.

Assume that the boiler is operating at an evaporation equal to O A. The water level for steady steaming will be shown by C and the water steam volume ratio below the water level by the lines A B, B C. If the steam "draw off" is now increased to O D, the first effect is to increase the volume of steam below the water level to E F which corresponds to the new increased rate of evaporation. This occurs with practically no time lag, with the result that, temporarily, the water level is raised to G where B G is equal to E F and the feed regulator reduces the feed flow to that corresponding to the level J G which is equal to a boiler load of O N. The temporary rise in boiler water level is usually termed "swelling". The evaporation is now taking place at a rate equal to O D while the feed flow only equals O N, so that the water level falls steadily and the feed regulator gradually increases the feed flow as the water level falls from J to F until the feed flow and steam "draw off" are in equilibrium at O D. The volume of water in the boiler has been reduced from A B to D E, the volume of steam below the water level has been increased from B C to E F and the water level has first risen from C to J quickly and then fallen gradually to F. The first and a temporary effect, therefore, of an increase in steam "draw off" is to reduce the feed flow until the evaporation lowers the water level to that associated with equilibrium between the fuel rate and the steam rate. During this period, the boiler pressure falls and the reduced feed flow and the storage effect of the water in the boiler enable the increase in steam demand to be met until the fuel rate is increased to suit. The reduction in feed flow assists the boiler to supply the increase in steam demand during the period of lag in changing the rate of fuel combustion, since less sensible heat for feed heating is called for. As the boiler water level falls, the rate of feed flow is gradually increased until equilibrium is established again between the increased steam rate, the rate at which fuel is burning and the rate at which feed water is supplied to the boiler.

If the steam demand is reduced from O D to O A, the immediate effect is to cause the water level to fall from F to P; this effect, which is temporary, is usually termed "subsidence" and is caused by the reduction in steam volume below the water level from E F to E P where E P is equal to B C, which is the steam volume below the water level corresponding to the new rate of steam "draw off". The feed regulator allows an increased feed flow to the boiler of O R, corresponding to the water level P Q, so that the water level rises steadily from Q to C and the feed regulator gradually reduces the feed flow until the feed flow and steam "draw off" are in equilibrium at O A.

If the steam demand is decreased, the immediate and temporary effect is to cause the water level to fall and the feed regulator to allow an increased feed flow into the boiler with an increased demand for sensible heat which assists in absorbing the furnace heat until the rate of fuel combustion is reduced as required to re-establish equilibrium. The increased rate of feed flow gradually raises the boiler water level and is itself gradually reduced until the feed flow, fuel combustion and steam rate are in equilibrium at the required reduced rate of steam demand. This characteristic of a continuous feeding type of feed water regulator materially assists in obtaining smooth boiler operation, minimises boiler pressure fluctuations during changing steam conditions and is a considerable offset to the time lag which occurs between a change in the steam demand and the corresponding change in fuel combustion becoming effective.

Automatic Boiler Feed Regulation.

To ensure reliability, an automatic boiler feed regulator should be simple and robust in design and should have ample power to overcome any frictional or hydraulic resistance to the movements of the moving parts. While reliability is the primary consideration in a boiler feed water regulator, it is also important that the regulator should control the feed supply so that the feed water flows continuously into the boiler while it is steaming and so that the feed water flow is varied in accordance with the predetermined requirements for any given boiler installation. These requirements vary with different types of boiler plants, and the type of automatic feed water regulator fitted to a given boiler plant must be correlated to the characteristics of that boiler plant.

For all boilers, the feed regulator must possess some feature which responds to changes in the boiler water level. In the case of small boilers, a float can be used to operate directly a steam or a water valve. In the case of boilers of moderate and large size, the feature which responds to changes in the boiler water level should preferably bring into action and control the action of a secondary source of power which is powerful enough to regulate the movements of a valve which controls the motive power of the boiler feed pump or the feed flow from the boiler feed pump into the boiler.

In single boiler installations or installations in which each boiler has its own feed pump, it is sufficient to control only the motive power of the boiler feed pump.

Where the boiler installation comprises two or more boilers with common feed pumps, it is essential to control separately the feed flow into each boiler and, in conjunction therewith, the motive power of the feed pumps must also be controlled either manually or automatically.

The above remarks apply to automatic feed regulators mounted on the boiler. Where there is one boiler of the tank type, the regulation of the feed flow can be effected from any water level on the suction side of the feed pump which varies with the boiler steaming as will be described later. The fundamental essentials as outlined above will be traced in the various arrangements described herein.

Boiler and Feed System Characteristics.

Boilers can be classified into two broad types; tank boilers and water tube boilers. Tank boilers are generally of fire tube construction with a very large water capacity in relation to their steaming capacity, such as the Scotch, Cornish, Lancashire, Economic and similar boilers. In water tube boilers, the weight of water contained in the boiler is very much smaller relative to the steaming capacity of the boiler, and it is much more important to maintain steady and reliable feeding conditions for a water tube boiler than in the case of most tank boilers.

In marine water tube boilers, particularly in Naval vessels, the proportions of the boilers are reduced to a minimum to meet weight and space considerations, and the weight of water in such boilers is smaller still in relation to the rate at which steam is generated in the boiler.

Feed water regulation for tank boilers is comparatively simple. The water level changes are slow and gradual, due to the large water capacity of the boiler, and changes in the rate of evaporation do not seriously affect the water level. Feed water regulation may be effected by controlling the boiler feed pump motive power supply or the discharge of water from the boiler feed pump either adjacent

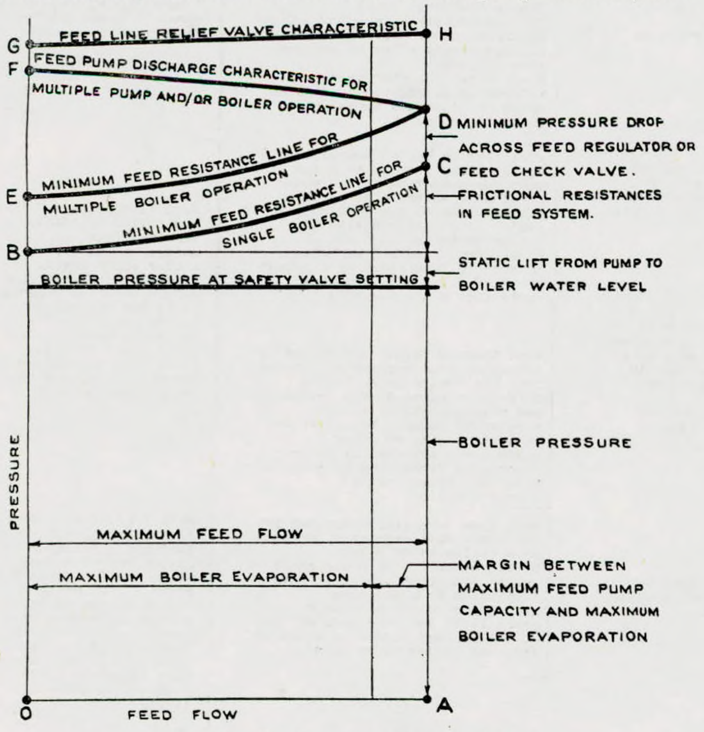

FIG. 2.—*Feed system characteristics.*

to the feed pump or at the feed inlet to the boiler. The control may be effected from the water level in the boiler or a suitable water level somewhere in the system on the suction side of the boiler feed pump, dependent upon the character of the installation.

To ensure the feed supply to a boiler, the feed pump must have a discharge pressure equal to the pressure in the steam drum, plus the static lift from the feed pump to the boiler water level, plus the frictional resistances in the feed system between the feed pump and the boiler. This discharge pressure is a minimum when the feed inlet to the boiler is wide open, and this minimum can only be obtained in the case of one feed pump supplying one boiler without an automatic feed water regulator in the feed pump discharge line and with the motive power of the boiler feed pump controlled to give a feed supply varying in accordance with the demands of the boiler or the return of the feed water from the engine system.

Where an automatic feed water regulator is fitted in the feed pump discharge line, the pressure drop across such regulator must be added to the other resistances to be overcome by the feed pump, as mentioned above.

Fig. 2 shows these points graphically. B C is the minimum feed resistance curve, the point C being the minimum feed pump discharge pressure at the maximum feed flow to the boiler as indicated by O A. This minimum feed discharge line B C can be obtained only with single boiler operation. Where two or more boilers are in service with a common feed supply, it is necessary to provide a distributing throttling in the feed discharge to each boiler, and the line E D shows the minimum feed resistance line for multiple boiler operation where the ordinate between the curves B C and E D represents the minimum throttling necessary across the feed check valve or feed regulator on each boiler to ensure satisfactory feed distribution between boilers and satisfactory operation of the feed regulators, so that, when an automatic feed regulator is installed, the feed discharge pressure at maximum evaporation is increased by C D. In general, feed pumps operating in conjunction with automatic feed regulators are controlled either inherently by their design or by automatic control of their motive power to follow a discharge pressure characteristic similar to F D which falls continuously from F to D. At any given feed flow, the ordinate between the lines F D and B C is the pressure drop across the feed water regulator. As is shown by Fig. 2, the use of a throttling type of automatic feed flow regulator—involves the use of higher feed discharge pressures than the minimum necessary simply to cause the water to flow into the boiler. The difference is the price paid in feed pump operating power for the benefit of automatic feed regulation and is a very small price compared with the advantages obtained. We shall see how the price varies with the different types of feed pumps. Where the feed system is fitted with a relief valve, the discharge pressure characteristic, when the relief valve opens, is shown by the line G H where G is a pressure slightly greater than F, the maximum feed discharge pressure under normal operating conditions.

Feed Pump Types and Controls.

Feed pumps are broadly divided into two types, Reciprocating and Centrifugal. Reciprocating pumps may be of the Simplex or Duplex steam driven direct acting types or they may be of the multi-throw type with one, two or more plungers or buckets driven through a crank or other reciprocating mechanism which is driven by a steam turbine, steam engine or an electric motor.

Reciprocating pumps of the usual type must be varied in speed to effect a variation in feed flow, and the output is directly proportional to their speed. With an electric motor drive, D.C. current is often used to obtain a variable speed, and the speed variation obtainable is usually limited to the range between half speed and full speed. With A.C. motors, the motor must be operated intermittently or a byepass used. Automatic control of D.C. or A.C. motors driving reciprocating pumps is difficult except with small units. Steam drives possess the advantage of being much more flexible and provide an infinitely variable control of speed between zero output and maximum feed flow. Lack of flexibility in a feed pump necessitates the use of a byepass from the pump discharge back to the suction to pass the difference between the pump output and the feed demand at any given moment.

Complete variability of flow from practically zero to maximum can be obtained with electrically driven reciprocating pumps by the use of a variable speed electric drive such as, for example, the Ward Leonard system, or by phase changing mechanism, or stroke variation by lever mechanism, but such arrangements are correspondingly complicated and expensive particularly if automatic control of the motive power supply to the driving unit is provided. In the case of phase changing mechanism and stroke variation, any other source of power such as compressed air or high pressure oil may be used to effect the necessary mechanical motions to change output.

Centrifugal feed pumps operate at a constant speed or with a comparatively small variation in speed. The feed flow is varied by throttling the feed discharge to the boiler or controlling the motive power supply to the driving unit. The steam drive for a centrifugal pump is extremely good from the point of view of control and permits the output of the pump to be varied easily, automatically, rapidly and with infinite gradualness between zero and maximum feed flows. Electric drives with D.C. or A.C. motors drive centrifugal pumps at practically constant speed and the feed flow is varied by stopping or starting the pump or by throttling the feed flow to the boiler. The diagrams, Figs. 3 and 4 show various drives used for the different types of feed pumps and the types of controls used in connection therewith. It will be appreciated that feed flow can be controlled by hand or automatically as desired and as may be practicable and that the arrangements for single and muiltiple boiler operations can be the same for similar types of boilers with the exception that distributing control is necessary between two or more boilers.

GENERAL PUMP TYPE	TYPE OF DRIVE AND MOTIVE POWER	INDIVIDUAL TYPE OF PUMP	ELECTRIC SUPPLY	SPEED AND VARIABILITY CHARACTERISTICS	TYPES OF CONTROLS FOR SINGLE BOILER OPERATION	TYPES OF CONTROLS FOR MULTIPLE BOILER OPERATION
RECIPROCATING FEED-PUMPS, BUCKET OR PLUNGER TYPES, WITH OUTPUT DIRECTLY PROPORTIONAL TO PUMP SPEED	STEAM DRIVE DIRECT-ACTING	SIMPLEX DUPLEX		**Infinite speed variability with infinite variability in pump output.**	**Feed flow to boiler not controlled. Automatic control of the steam supply to the pump operated from the boiler water level or a water level on the suction side of the feed pump which varies in accordance with the boiler steaming** **or** **Feed flow controlled by an automatic feed regulator operated from the boiler water level. Steam to pump controlled by a discharge pressure governor.**	**Feed flow to each boiler controlled by automatic feed regulator on each boiler operated from the boiler water level. Steam controlled by a discharge pressure governor or speed governor.** or (For Tank Boilers) Automatic control of the steam supply to the pump operated from a water level on the suction side of the feed pump which varies in accordance with the boiler steaming. Feed flow to each boiler controlled by automatic feed regulator on each boiler operated from the boiler water level.
	STEAM DRIVE TURBINE or ENGINE	7 - THROW 5 - THROW 3 - THROW		**Infinite speed variability with infinite variability in pump output.**	**Feed flow to boiler not controlled. Automatic control of the steam supply to the pump operated from the boiler water level or a water level on the suction side of the feed pump which varies in accordance with the boiler steaming.** **or** **Feed flow controlled by an automatic feed regulator operated from the boiler water level. Steam to pump controlled by a discharge pressure governor or speed governor.**	
	ELECTRIC MOTOR DRIVE	7 - THROW 5 - THROW 3 - THROW 2 - THROW 1 - THROW	A. C. CURRENT	Speed, zero and maximum. Stop and start giving intermittent operation, i. e. zero and full output. or No speed variation - constant speed. Continuous operation at full output.	**Feed flow to boiler not controlled. Motor switch gear to stop and start pump operated from boiler water level or a water level on the suction side of the feed pump which varies in accordance with the boiler steaming. Horse power absorbed by motor inherently determined by constant speed of motor and feed line resistance characteristic.** **Feed flow to boiler controlled by automatic feed regulator operated from boiler water level. Difference between pump output and requirements of boiler discharged through bypass valve to suction side of pump. Horse power absorbed by motor inherently determined by constant speed of motor and bypass valve characteristic.**	**Feed flow to each boiler controlled by automatic feed regulator on each boiler operated from boiler water level. Difference between pump output and requirements of boiler discharged through bypass valve to suction side of pump. Horse power of motor inherently controlled by speed of motor and bypass valve characteristic.**
			D. C. CURRENT	Speed, zero and maximum. Stop and start giving intermittent operation, i. e. zero and full output. or **Manually operated step controller giving constant speed at any chosen step. Extreme range of speeds maximum to 50 per cent maximum. Output proportional to speed at which pump is operated.**	**Feed flow to boiler not controlled. Motor switch gear to stop and start pump operated from boiler water level or a water level on the suction side of the feed pump which varies in accordance with the boiler steaming. Horse power absorbed by motor inherently determined by constant speed of motor and feed line resistance characteristic** **or** **Feed flow to boiler controlled by automatic feed regulator operated from the boiler water level. Difference between pump output and requirements of boiler discharged through bypass valve to suction side of pump. Horse power absorbed by motor inherently determined by speed of motor and bypass valve characteristic**	**Feed flow to each boiler controlled by automatic feed regulator on each boiler operated from boiler water level. Difference between pump output and requirements of boiler discharged through bypass valve to suction side of pump. Horse power of motor inherently controlled by speed of motor and bypass valve characteristic.**
RECIPROCATING FEED PUMPS BUCKET OR PLUNGER TYPES, SPECIAL VARIABLE SPEED ELECTRIC DRIVE, PHASE CHANGING MECHANISM OR STROKE VARIATION BY LEVER MECHANISM.	ELECTRIC MOTOR DRIVE	7 - THROW 5 - THROW 3 - THROW	A. C. or D. C. CURRENT	Infinite speed variability with infinite variability in pump output.	**Feed flow to boiler controlled by automatic feed regulator operated from boiler water level. Motive power to the motor which varies the speed, phase or stroke controlled by a governor responsive to the discharge pressure of the feed pump or the automatic feed regulator operated from the boiler water level.**	Feed flow to each boiler controlled by automatic feed regulator on each boiler operated from boiler water level. Motive power to the motor which varies the speed, phase or stroke controlled by a governor responsive to the discharge pressure of the feed pump. or the automatic feed regulator operated from the boiler water level.

FIG. 3.—*Characteristics of reciprocating feed pumps.*

GENERAL PUMP TYPE	TYPE OF DRIVE AND MOTIVE POWER	ELECTRIC SUPPLY	SPEED AND VARIABILITY CHARACTERISTICS	TYPES OF CONTROLS FOR SINGLE BOILER OPERATION	TYPES OF CONTROLS FOR MULTIPLE BOILER OPERATION
CENTRIFUGAL FEED PUMPS	STEAM TURBINE DRIVE		Speed substantially constant under control of pressure governor. Infinite variability of pump output. or Speed substantially constant under control of speed governor. Infinite variability of pump output.	Feed flow to boiler not controlled. Automatic control of the steam supply to the turbine operated from the boiler water level or a water level on the suction side of the feed pump which varies in accordance with the boiler steaming. or Feed flow controlled by an automatic feed regulator operated from the boiler water level. Steam supply controlled by a discharge pressure governor or speed governor.	Feed flow to each boiler controlled by automatic feed regulator on each boiler operated from the boiler water level. Steam supply to turbine controlled by a discharge pressure governor or speed governor.
	ELECTRIC MOTOR DRIVE	A. C. CURRENT	Constant speed. Infinite variability of pump output.	Feed flow controlled by an automatic feed regulator operated from the boiler water level. or Feed flow controlled by automatic regulator operated from a water level on the suction side of the feed pump which varies in accordance with the boiler steaming. Horsepower absorbed by motor inherently determined by constant speed of motor and pump characteristic.	Feed flow to each boiler controlled by automatic feed regulator on each boiler operated from the boiler water level. Horsepower absorbed by motor inherently determined by constant speed of motor and pump characteristic.
		D. C. CURRENT	Constant speed at each step of control. Infinite variability of pump output.	Feed flow controlled by an automatic feed regulator operated from the boiler water level. or Feed flow controlled by automatic regulator operated from a water level on the suction side of the feed pump which varies in accordance with the boiler steaming. Controller step fixed by manual operation. Constant speed at controller step in use. Horsepower absorbed by motor inherently determined by chosen step and pump characteristic.	Feed flow to each boiler controlled by automatic feed regulator on each boiler operated from the boiler water level. Controller step fixed by manual operation. Constant speed at controller step in use. Horsepower absorbed by motor inherently determined by chosen step and pump characteristic.

FIG. 4.—*Characteristics of centrifugal feed pumps.*

Feed Pump Characteristics.

Fig. 5 shows the Discharge Pressure Capacity characteristics of Direct Acting Feed Pumps. The minimum feed resistance line is shown by G H which corresponds to B C or E D of Fig. 2 dependent upon whether single boiler or multiple boiler operation is under consideration. When the pump discharge pressure corresponds to the minimum feed resistance line, the minimum expenditure of motive power is obtained. With a direct acting pump, manual or automatic control of the steam supply to the pump is highly desirable. If the steam supply valve is adjusted and set to permit the pump to operate at the point H, corresponding to maximum feed flow, any reduction in feed flow caused by the closing of the feed check valve or the action of an automatic feed flow regulator will cause the discharge pressure to rise along the line J H where J is the shut discharge valve pressure of the pump. The point J will depend upon the ratio between the steam cylinder and the pump barrel and may be as high as 1·8 to 2 times the steam pressure. For example, if the feed flow to the boiler is reduced to O B, the feed check valve or feed regulator opening will be reduced until the feed resistance line intersects the pump discharge line at K. Since a pressure equal to M will allow the required feed flow to pass into the boiler, the additional pressure M K is unnecessary and wasteful. If, now, the opening of the steam supply valve is reduced, the pump pressure could be reduced to M but, since exact setting of valves is difficult, it will probably be adjusted to pass sufficient steam to give a slightly higher pressure than M, say L, and the feed check valve or feed regulator opening would be adjusted as necessary to give a feed flow resistance as shown by G L. With careful manual operation of the steam supply, the pump discharge pressure can be caused to follow the line G H at all loads, although this would be operating perfection. Manual operation is the usual method adopted but, where desired, the steam supply can be automatically controlled from the discharge pressure of the pump so that the discharge pressure follows a characteristic such as N H which falls slightly from N to H, in which case the feed regulator or the feed check valve must give an opening which causes a feed resistance line G P for a feed flow of O B and the automatic steam control will adjust the steam supply to the pump to give a discharge pressure of P. Where J may be an excessive discharge pressure, a relief valve should be provided in the feed discharge line and set to open at some pressure, such as R,

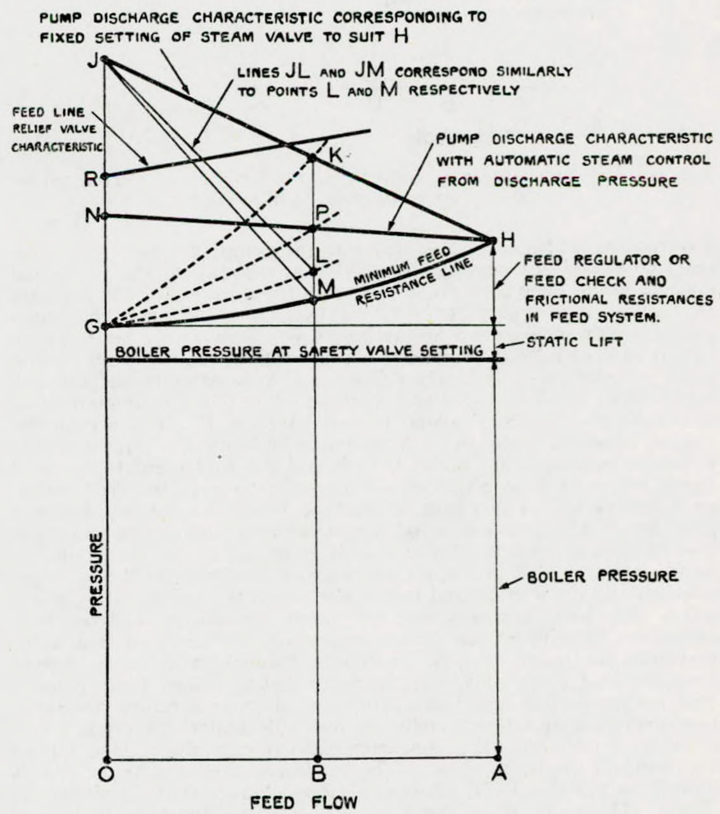

FIG. 5.—*Discharge pressure capacity characteristics of direct acting feed pumps.*

slightly in excess of the pressure N.

Fig. 6 shows the pressure capacity characteristics of electric motor driven reciprocating feed pumps of normal design. They are usually operated with single boiler installations. In small sizes, A.C. or D.C. motors are used with stop and start mechanisms and, in such cases, the pumps operate at the point H when discharging into the boiler. With D.C. motors, the pump can be set to operate at any one of a number of stops between about half load and maximum capacity and the difference between the capacity at the stop in operation and the feed required by the boiler must be discharged through

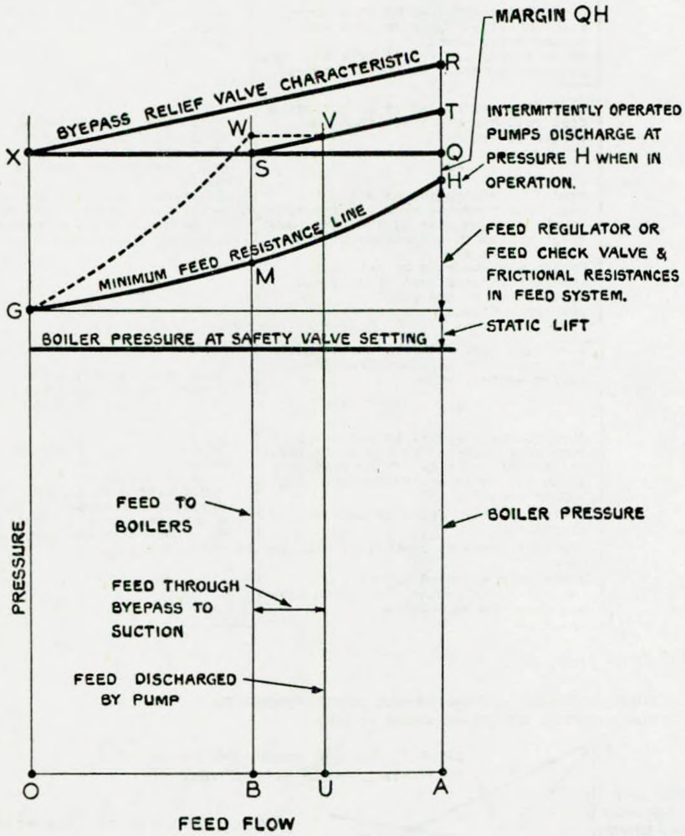

FIG. 6.—*Discharge pressure capacity characteristics of electric motor driven reciprocating feed pumps.*

a spring loaded byepass relief valve to the pump suction. This valve must be capable of passing the maximum capacity of the pump and must be set to open at a pressure Q slightly in excess of the pressure H. When discharging through the byepass valve only, the pump characteristic is as shown by the line X R. Now, if the boiler steam rate is as shown by O B and the nearest stop on the electrical control gives a feed flow of O U, the difference B U must be passed through the byepass valve to the pump suction. The pump characteristic is shown by the line S T which is parallel to X R. The actual discharge pressure is the point V at pump capacity O U, the quantity of water passing to the boiler is O B and the feed regulator or feed check valve must be adjusted as necessary to give the feed resistance line G W. The result is that the pump discharges against a pressure of M W in excess of the possible minimum resistance and also handles a quantity of water B U in excess of the feed required by the boiler. This illustrates the lack of flexibility in the arrangement but, where it is desired to use electricity, the reciprocating pump offers the best arrangement for small quantities and/or high pressures. Reciprocating feed pumps specially designed and automatically controlled to give an infinite gradual variation in output from no load to full load can be controlled to follow the minimum feed resistance line for single boiler operation or a falling discharge pressure capacity characteristic for multiple boiler operation.

Fig. 7 illustrates the characteristics of centrifugal feed pumps operating at constant speed. The minimum feed resistance line is shown by the line E D_4 while the pump characteristic is shown by F D_4. The point D_4 is the only one at which there is no loss of pumping energy. If the boiler is operated at a partial load, the feed regulator must close to suit. At loads 1/4, 2/4 and 3/4, the regulator must close until the feed resistance line follows the line E D_1, E D_2 and E D_3 respectively, and the ordinate between the line E D_4 and F D_4 at any given boiler load represents the additional pressure drop which must be carried across the feed flow regulator. This, of course, is the amount of discharge pressure above the minimum which would feed the boiler and can, therefore, be said to be lost energy. This is the price paid for the simplicity of the A.C. motor drive with its automatic variation in motive power without any control equipment.

Steam turbine driven centrifugal feed pumps controlled by a

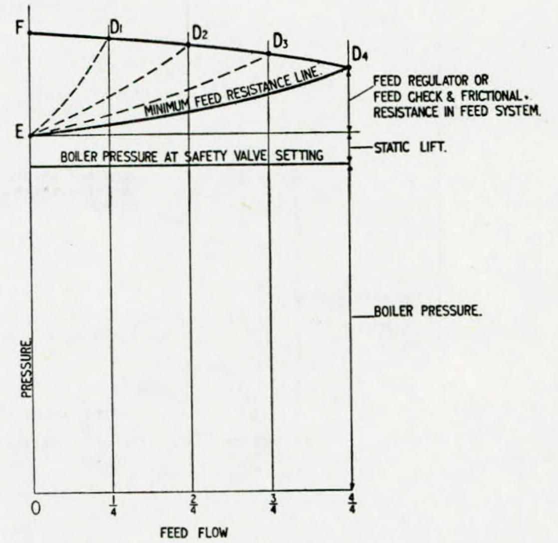

FIG. 7.—*Discharge pressure capacity characteristics of constant speed and pressure controlled centrifugal pumps and variable speed reciprocating pumps.*

speed governor or a pressure governor give a pressure capacity characteristic similar to FD_4, and behave generally in a similar manner to A.C. motor driven centrifugal feed pumps.

Fig. 8 shows the characteristics of centrifugal feed pumps when driven by a D.C. motor with speed variation by step control. The

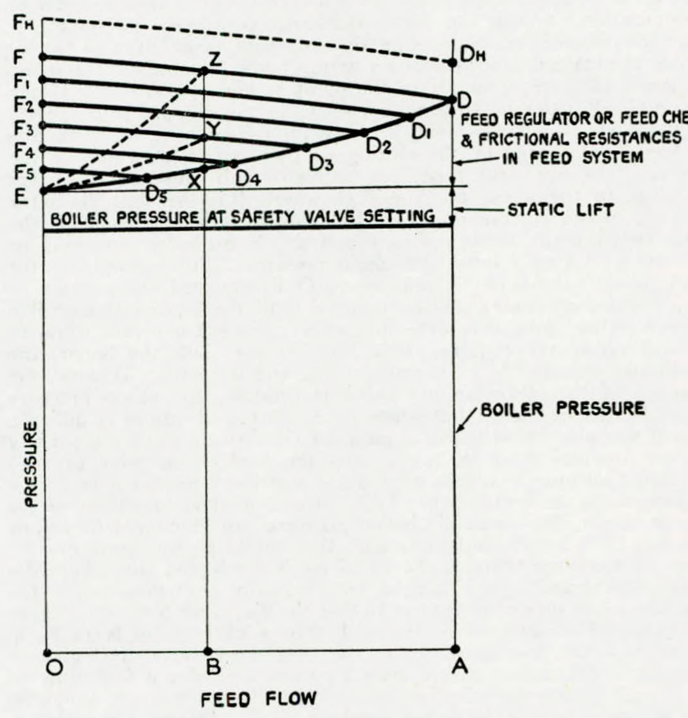

FIG. 8.—*Discharge pressure capacity characteristics of D.C. motor driven centrifugal feed pump with step control.*

discharge pressure characteristic is shown by F D at the speed necessary to discharge the maximum feed flow into the boiler. When a D.C. motor is started, a considerable time elapses before the motor reaches a steady temperature and, as the temperature rises, the speed rises. It is necessary, therefore, to design the unit so that it can generate a characteristic as shown by F D when the motor is cold. As the motor rises in temperature, its speed will rise until it reaches a steady temperature when the pump characteristic will be that shown by F_H D_H, and its speed will require to be adjusted by the resistance control to the desired speed. With step control, the speed of the pump can be set to characteristics F D, F_1 D_1, F_2 D_2 and so on.

Assume, as an example, steady motor temperature has been reached and the pump is operating along the line F_3 D_3. If the boiler is operating at a capacity O B, the feed flow regulator will adjust its opening to cause the feed resistance line to follow E Y. The minimum resistance line is E D and the additional pressure drop across the feed regulator is, therefore, X Y as against X Z for an A.C. motor unit. This illustrates the saving in power which can be obtained by the provision of speed adjustment.

Small Boiler Installations.

Small boilers such as used for auxiliary or harbour duties have a comparatively small water level area and, in general, the feed tank has a relatively large capacity. For such boilers the feed flow is preferably controlled from the water level in the boiler. For boiler capacities up to 10,000lb./hour and boiler pressures up to 200lb. per square inch, a direct operated boiler feed regulator can be used in which the float operates directly the valve controlling the steam or the water supply.

Where a steam driven feed pump is used with a single boiler, the automatic regulator can be operated from the boiler water level

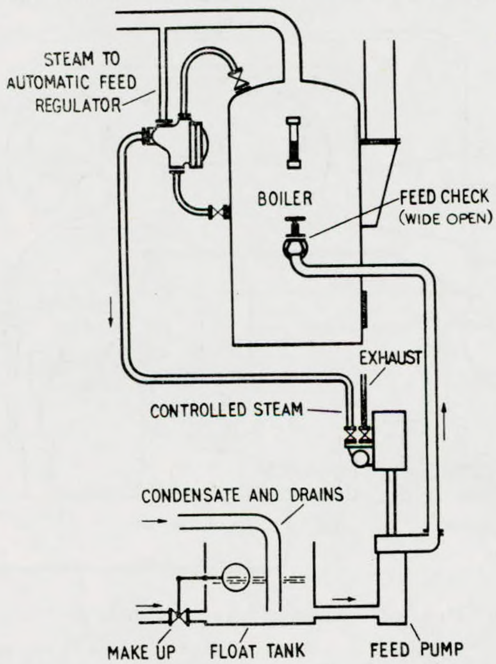

FIG. 9.—*Small single boiler installation with D.A. feed pump.*

to control the steam supply to the pump. Fig. 9 shows this arrangement. The feed check valve is left wide open thereby obtaining the minimum resistance to feed flow as shown by B.C. in Fig. 2 with the corresponding lowest possible steam consumption for the feed pumping duty. The feed tank can be provided with a float controlled make-up valve connected to the reserve feed tank which makes the feed system completely automatic. The feed flow into the boiler is regulated in accordance with the boiler requirements and the feed pump is regulated to give the necessary feed flow at the lowest steam consumption possible.

Fig. 10 shows a single boiler installation with an electrically driven reciprocating feed pump. The feed regulator for such an installation comprises a float box and a float lever operating a relay switch which opens and closes the main switch on the motor driving the feed pump. A tumbler arrangement is provided in the relay switch whereby, on a rising water level in the boiler, the pump is stopped at a predetermined high level and with a falling water level in the boiler, the pump is started at a predetermined low water level.

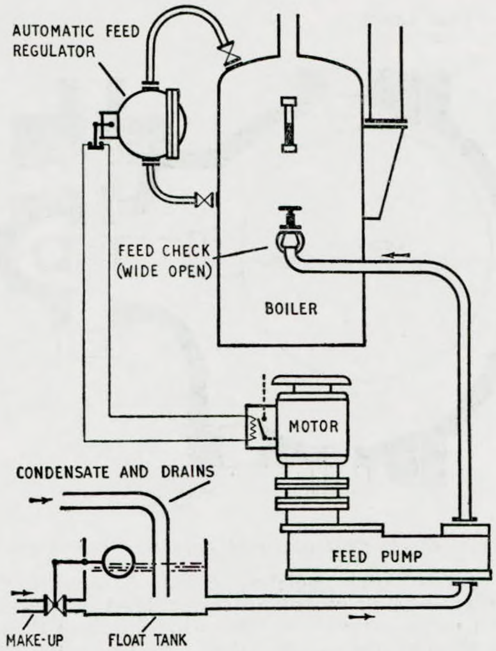

FIG. 10.—*Single boiler installation with electrically driven feed pump.*

The pump, therefore, operates intermittently at full output and is stopped for such periods as necessary for the intermittent discharge of water from the pump to balance the continuous steaming rate of the boiler.

When the pump is in operation it discharges water at the pressure H as shown in Fig. 6 corresponding to the minimum feed resistance at the maximum feed flow, the feed check valve on the boiler being left wide open so that the pump has a minimum power consumption for the feed pumping duty to be performed although such power consumption corresponds to maximum feed flow.

With two or more boilers, the total feed flow must be distributed between the boilers and each boiler is provided with an automatic feed regulator in which the float is actuated by the water level in the boiler and controls the flow of feed water into the boiler. Fig. 11 shows this arrangement.

The steam supply to the feed pump may be manually controlled to suit the total feed flow or it may be automatically controlled from the discharge pressure of the pump, the characteristics of the system being as shown in Fig. 5. Fig. 12 shows a section through a

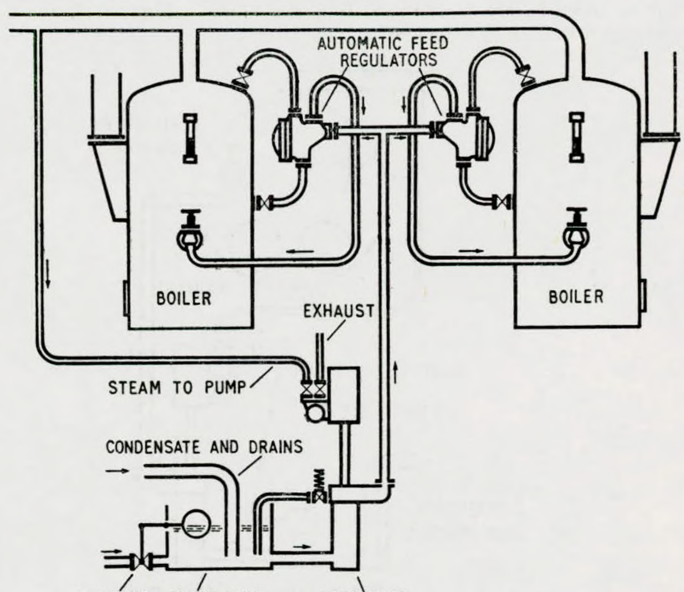

FIG. 11.—*Small multiple boiler installation with D.A. feed pump.*

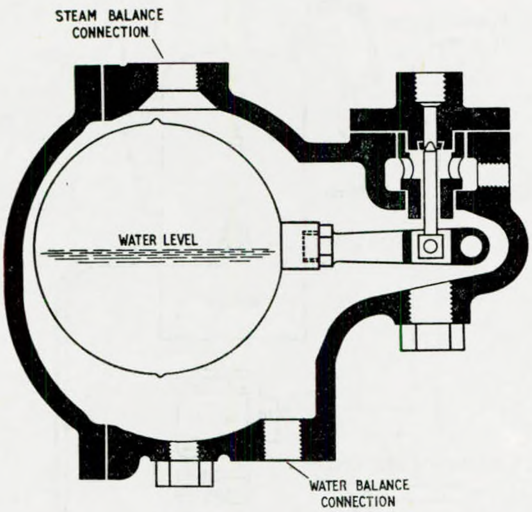

FIG. 12.—*Small direct float operated regulator for steam or water.*

direct float operated feed regulator. Briefly, it consists of a simple float operated needle or control valve regulated by the water level so that a fall in the water level lowers the float and thus increases the opening through the needle valve. A rise in the water level reduces the opening through the needle valve. For a single boiler installation the needle valve is used to control the flow of steam to drive the feed pump. For a multiple boiler installation the needle valve controls the flow of feed water into the boiler.

When electrically driven reciprocating pumps are used with a multiple boiler installation, the total feed flow must be distributed between the boilers as shown in Fig. 11 and the difference between the feed water discharged by the pump at the speed at which it is running and the steaming requirements of the boilers must be discharged through a spring loaded relief valve to the suction side of the pump. For such an arrangement, characteristics are as shown in Fig. 6.

Typical Feed Systems for Large Boilers.

In a tank type boiler, the quantity of water available between the permissible high and low water levels is usually so considerable that it is not essential to provide a large capacity feed tank and it is usual to arrange on the suction side of the feed pump, a small float tank into which all condensate and drains are discharged by the air pump, feed heating system, etc.

The float tank has no storage capacity and is arranged to control the steam supply to the feed pump so that water entering the float tank is immediately discharged into the boiler. Fig. 13 shows this arrangement with a direct acting pump.

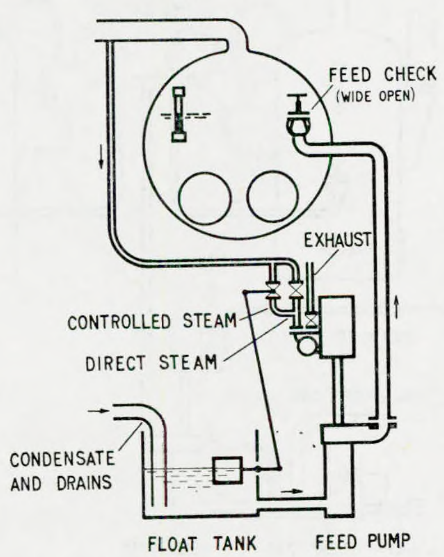

FIG. 13.—*Simple open system for tank boiler.*

The feed inlet check valve on the boiler is left wide open and the speed of the feed pump is controlled by the float which admits more steam to the feed pump as the water level rises in the float tank and vice versa. This arrangement gives the minimum discharge pressure on the feed pump at all times, as shown by B C on Fig. 2, and, therefore, requires the minimum feed pump steam consumption.

To make good feed losses from the system, a connection is provided from the reserve feed tank to the condenser. This connection is manually opened periodically or is adjusted for continuous flow as necessary to enable make up feed water to be passed into the system by way of the condenser to ensure that the boiler water level does not fall below a predetermined low level.

The flow through the make up connection can be adjusted from time to time to give the desired water level and as the variations between the steam drawn off from the boiler and the condensate, drains, and make up supplied are small and the water capacity of the tank type boiler is large, any rise or fall in the boiler water level takes place very slowly and adjustment of the make up can be made at infrequent intervals. This arrangement can, of course, be used with a steam turbine centrifugal feed pump in which the steam supply to the feed pump turbine is controlled by the float in the float tank and the pump discharge pressure follows the minimum discharge pressure corresponding to the line B C in Fig. 2. This gives the lowest possible feed pump steam consumption.

If the installation has two or more boilers, the total feed flow must be distributed between the boilers by manual control of the feed check valve on each boiler or automatically by an automatic feed water regulator. If the openings of the check valves are adjusted to give the minimum throttling necessary for correct feed distribution, the feed pump discharge pressure will follow the minimum resistance curve for multiple boiler operation shown by E D in Fig. 2. This gives the lowest possible feed pump steam

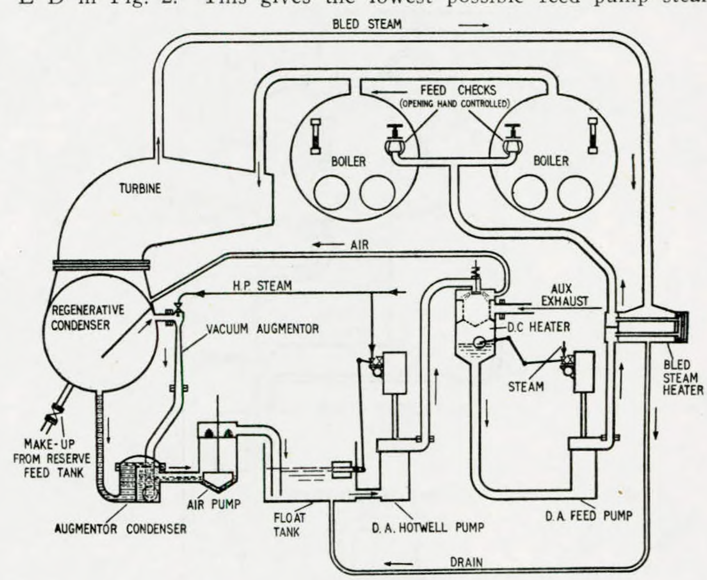

FIG. 14.—*Feed system with engine-driven air pump.*

consumption for multiple boiler operation. Fig. 14 shows an installation similar to Fig. 13, but with two boilers of the tank type.

The feed system shown in Fig. 14 is widely used in ships fitted with reciprocating engines and exhaust steam turbines where the highest possible vacuum is required. The air pump is of the Edwards type or a three valve type driven by levers from the main engines, and a steam operated air ejector is fitted to operate in conjunction with the engine driven pump. The ejector steam is condensed by the condensate on its way to the reciprocating air pump, and the air pump discharges the air with the condensate to the float tank.

In many boiler installations it is considered desirable to de-aerate the feed water to eliminate corrosion in feed lines, boilers, steam lines, etc.

In Fig. 14 to obtain de-aeration of the feed supply, the feed water is passed through a direct contact heater which is connected to the main condenser or atmosphere by a vent pipe which carries away the air released from the feed water in passing through the heater.

Modern ships are almost invariably fitted with a second stage feed heater utilising steam bled from an intermediate receiver or stage of the turbine to give a final feed temperature in the region of 300° F.

The use of a direct contact heater to obtain de-aeration necessitates the use of a pump to deliver water from the float tank into the heater.

A direct-acting feed pump draws water from the D.C. heater and discharges through the bled steam surface feed heater and the feed check valve into the boiler. In this arrangement, the water level in the D.C. heater on the suction side of the feed pump is used to control the steam supply to the feed pump. The water level in the float tank is used to control the steam supply to the hotwell pump. Water entering the float tank is immediately discharged by way of the hotwell pump, D.C. heater and feed pump to the boiler, both floats operating to increase the steam supply to their respective pumps when the water level rises, and vice versa.

Both the hotwell pump and the feed pump can be of the centrifugal type, if desired, with the steam supply controlled by the floats in the same manner.

With feed regulators of the continuous feeding type, the minimum throttling necessary for correct feed distribution is effected automatically at all rates of feed flow and the line E D in Fig. 2 with the corresponding minimum feed pump steam consumption is obtained automatically. Further, the regulators automatically distribute the correct feed flow to each boiler when the boilers are steaming at different loads. To obtain this desirable control, the make up must be adjusted as necessary to allow the feed regulators to operate at about the low predetermined water level, thereby offering the minimum distributing throttling to the feed flow.

If the feed regulators are of the intermittent type, it is necessary to limit manually the opening of the feed regulator valve to obtain the distributing throttling effect, and the correct total feed flow to each boiler is obtained by the intermittent closing of the feed regulator on any boiler into which the rate of feed flow is greater than that corresponding to the boiler steam load. Automatic adjustment of the feed flow to each boiler is thereby obtained when the boilers are steaming at equal or different loads. During the periods when one of the regulators is closed, the feed discharge pressure is increased above the minimum feed pressure possible with a corresponding increase in feed pump steam consumption.

Independent air pumps have the advantage of flexibility and enable the vacuum to be maintained when the main engines are not running. Fig. 15 shows a feed system with an independent air pump of the Paragon type as used in conjunction with a reciprocating engine exhaust steam turbine combination or a straight turbine arrangement.

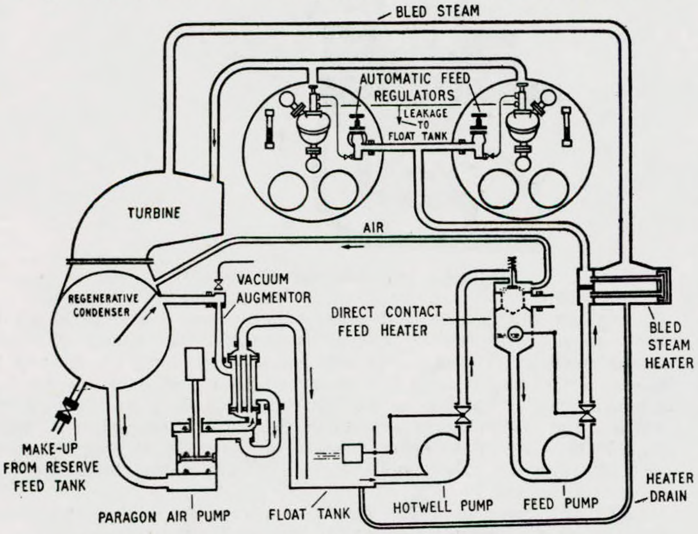

FIG. 15.—*Feed system with independent air pumps.*

The steam supply to the hotwell pump and the feed pump of the Centrifugal type can be controlled by floats in the float tank and the D.C. heater respectively or, alternatively, the floats can be used to control discharge valves arranged in the discharge lines of the hotwell pump and feed pump respectively, the speeds of the hotwell pump and feed pump being controlled either by speed governors, pressure governors, or electric motors.

With float controlled steam supplies, the pumps can be operated against the minimum feed discharge pressures corresponding to single boiler operation or multiple boiler operation as the case may be.

If Centrifugal pumps are used as shown in Fig. 15 and the pumps are controlled by speed governors or pressure governors, the discharge capacity characteristics of the pumps will be similar to F.D. in Fig. 2 and the steam consumptions will be higher than those which are obtained when pumping against the minimum feed resistance line possible.

The float controlled discharge valves on the hotwell pump and the feed pump are used to absorb the difference in pressure of the pump discharge as shown by the line F D in Fig. 2 and the minimum feed resistance line possible. A rise in water level in the float tank or the D.C. heater causes the float controlled discharge valve to open, and vice versa.

An alternative method, widely used, of de-aerating the feed supply is the modern closed feed system in which all the feed water passes through the condenser wherein it is de-aerated before it is withdrawn and discharged to the boilers through a closed pipe line at a pressure above atmospheric pressure so that all possibility of the feed water absorbing air is prevented.

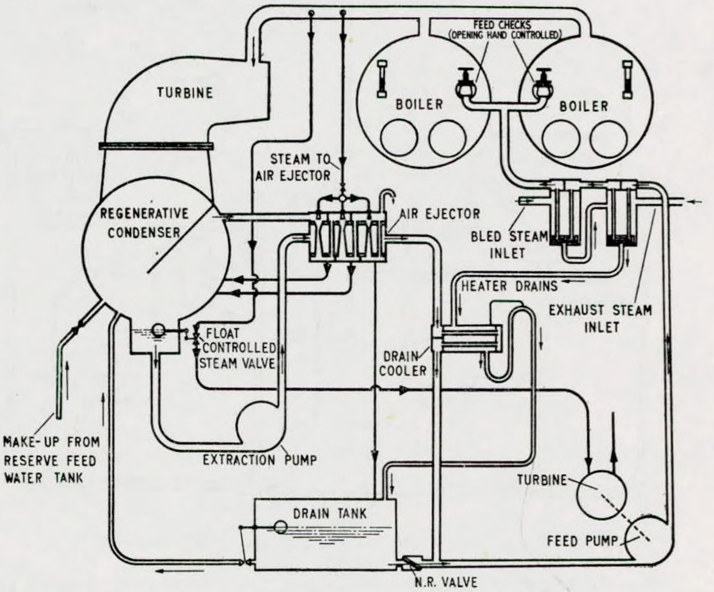

FIG. 16.—*Closed feed system for tank boilers.*

Fig. 16 shows a closed feed system for boilers of the tank type. The drainage from the auxiliary exhaust and feed heating system is led to the feed tank which is provided with a float-operated valve which passes the drainage water to the condenser when the water level rises in the feed tank. As the water level in the condenser rises and falls, it actuates a float which controls the steam supply to the feed pump, the steam supply being opened as the water level in the condenser rises, and closed as the water level in the condenser falls. The whole of the steam condensed in both the main and the auxiliary systems is passed through the condenser wherein it is de-aerated before it is passed to the boilers. Make-up feed passes from the reserve feed tank to the condenser under control of a manually-operated valve, which is opened as necessary to maintain the desired water level in the boilers. Where one boiler only is in use, the feed check valve is left wide open and the discharge pressure of the feed pump follows the minimum resistance line corresponding to B C in Fig. 2, so that the minimum possible steam consumption for the feed pump duty is obtained.

It will be seen that, with this arrangement, for a single boiler, the feed flow to the boiler is not controlled and the steam supply to the feed pump is automatically controlled from a water level on the suction side of the feed pump, such water level being provided in the main condenser so that, as the water level in the condenser rises, the steam supply to the feed pump is increased and vice versa.

If the installation has two or more boilers, the total feed flow must be distributed between the boilers by manual control of the feed check valve on each boiler or automatically by an automatic feed water regulator.

If the openings of the check valves are adjusted to give the minimum throttling necessary for correct feed distribution, the feed pump discharge pressure will follow the minimum resistance curve for multiple boiler operation shown by E D in Fig. 2, thereby obtaining the lowest possible feed pump steam consumption for multiple boiler operation.

Feed regulators can be of the continuous feeding type or the intermittent type, and their action will be the same as described in relation to Fig. 7.

In all the arrangements described in relation to the operation of tank type boilers, human supervision of the boiler water levels is necessary, entailing manual operation of the flow of make-up feed water into the system as necessary to ensure that the boiler water level does not fall below a predetermined low level.

In the case of water tube boilers, the quantity of water available in the boiler is generally comparatively small compared with the steaming rate of the boiler and it is, therefore, necessary to provide a large capacity feed tank, the flow of water into the boiler being controlled from the boiler water level either manually or automatically by means of an automatic feed regulator which operates to effect a restricting control over the feed flow into the boiler so that such feed flow corresponds to the boiler requirements.

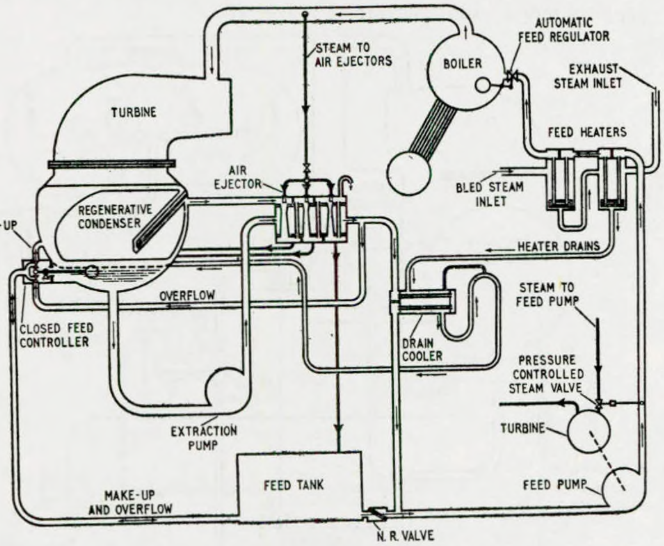

FIG. 17.—*Closed feed system for watertube boilers.*

Fig. 17 shows a closed feed system for water tube boilers in which the water content is relatively small. A great advantage obtained with the system is that all units are entirely automatic at all powers. The controlling water level in the system is the water level in the boiler which is maintained between predetermined levels, the level at full power being two to three inches lower than the level at no evaporation. Variations between the rate at which water is fed into the boiler and steam is condensed in the condenser, alter the water level in the base of the condenser. The water level in the condenser is automatically regulated by a closed feed controller which comprises two valves actuated by a common float, one valve being an overflow valve and the other a make-up valve. If the water level rises in the base of the condenser, the overflow valve opens and water is discharged to the feed tank, while a fall of water level in the condenser causes the make-up valve to open and water is drawn into the condenser from the feed tank. Under steady running conditions, the make-up valve is slightly open, just sufficient to pass the quantity of water necessary to make good losses from the system.

The feed pump shown is of the turbine-driven centrifugal type, the pump being controlled by a pressure governor which controls the steam supply to the turbine of the pump from the pressure in the feed pump discharge line. With this arrangement, the feed pump discharge pressure follows the characteristic shown by F D in Fig. 2.

The feed regulator should be of the continuous flow type operating to close the valve as the water level in the boiler rises and vice versa.

The minimum feed resistance line will follow the line B C in Fig. 2, and the difference in pressure between the characteristic of the feed pump shown by F D and the minimum feed resistance line B C must be absorbed across the feed regulator.

Pressure control of the steam supply to the turbine-driven centrifugal feed pump is the usual method of controlling the feed pump, but the pump could be controlled by a speed governor and would operate in a similar manner.

Direct-acting feed pumps are sometimes used in closed feed systems with water tube boilers, and the steam supply to the direct-acting feed pump can be hand controlled if desired, in which case the feed pump discharge pressure would follow the characteristics outlined in Fig. 5 by J H or J L or J M, depending upon the setting of the steam valve. Alternatively, the steam supply to the direct-acting feed pump can be controlled automatically by a pressure governor from the discharge pressure of the pump, in which case the discharge pressure characteristic of the feed pump would be as shown by the line N H in Fig. 5.

In these cases, the difference in pressure between the discharge pressure characteristic of the pump and the minimum feed resistance line must be carried across the automatic feed regulator which will automatically adjust its opening to suit. This same arrangement can be used for two or more boilers in multiple boiler operation in which each boiler is fitted with an automatic feed water regulator. Human supervision of the water level in the feed tank is necessary, and water must be drawn from a reserve feed tank as necessary to maintain the feed tank about two-thirds full under normal running conditions.

In many installations, evaporators using either boiler steam or bled steam are fitted in the feed system and are operated as necessary to maintain the feed tank water level.

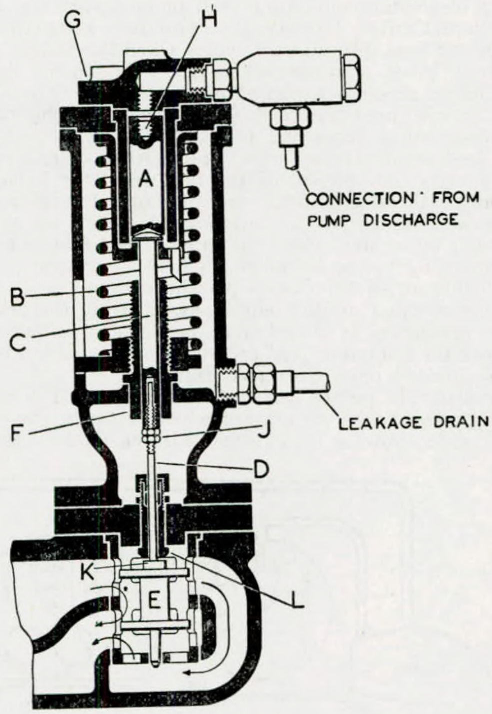

FIG. 18.—*Pressure governor.*

Fig. 18 shows a typical pressure governor for controlling the steam supply to a turbine driven centrifugal feed pump. The discharge pressure of the pump acts on top of piston A against the bias of the spring B until the spring load and the pressure load are in equilibrium. As shown, the throttle valve E is in the full open position and the pressure applied to the top of the piston A causes the throttle valve E to move downwards partially closing the steam ports through the valve cage thereby controlling the steam supply to the pump turbine.

For any given load, the discharge pressure of the pump finds a position of equilibrium against the spring load with an opening of the throttle valve ports which passes the quantity of steam required to drive the feed pump at the discharge pressure corresponding to the given feed flow. The characteristic of the spring is arranged in relation to the area of the throttle valve ports to give a pressure output characteristic which falls slightly from no load to full load in order to ensure stable operation of the feed pump, the discharge characteristic obtained being similar to F D_4 in Fig. 17. This characteristic is, of course, in excess of the minimum feed resistance line E D_4, with a corresponding waste in pumping pressure, at all loads below the maximum.

In some installations where multi-stage feed heaters and economisers are fitted, the frictional resistance between the feed pump and the boiler drum is considerable and where such installations are required to operate on partial loads for any considerable length of time, considerable advantage can be obtained if the steam

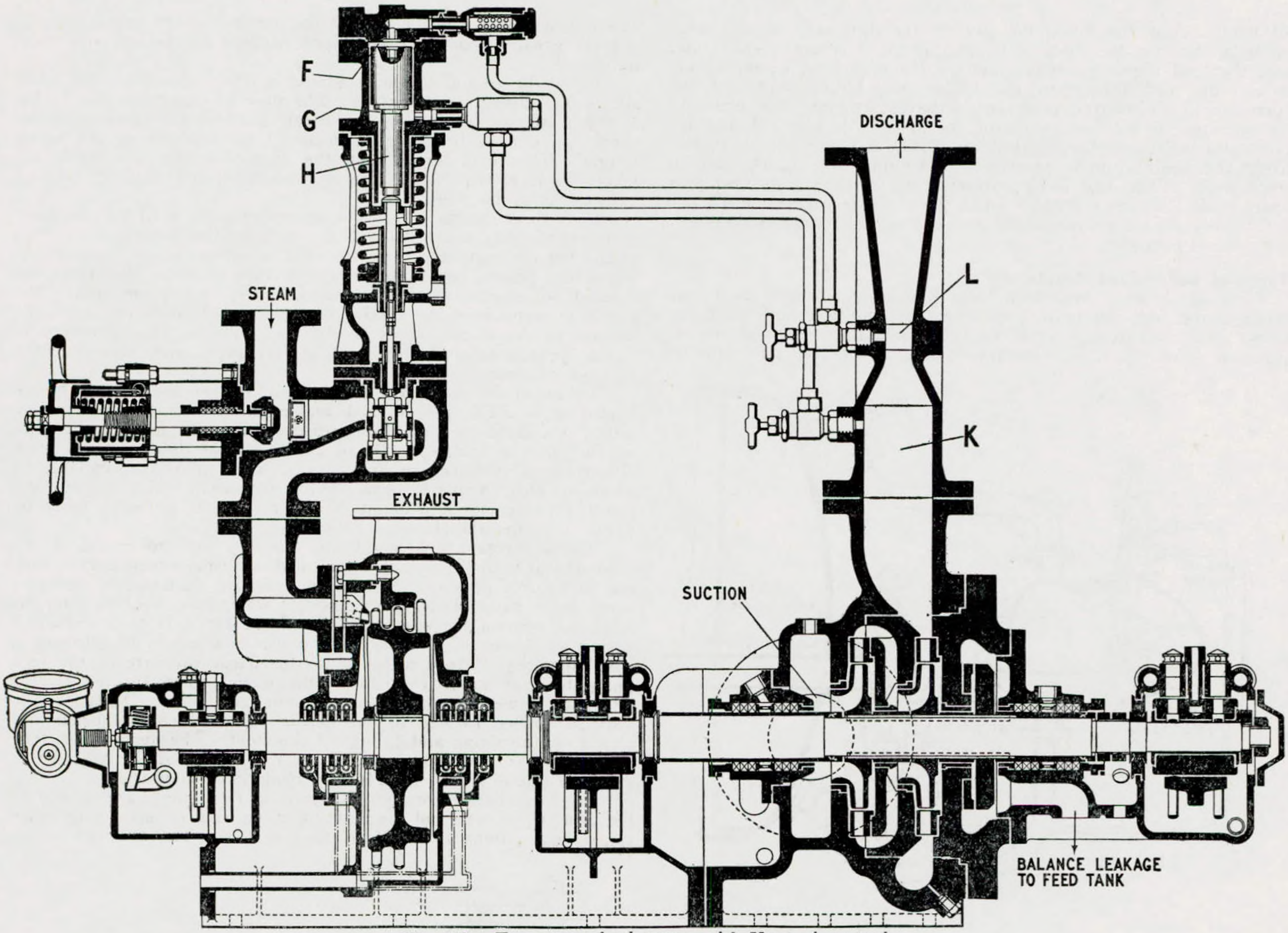

FIG. 19.—*Two-stage feed pump with Venturi control.*

supply to the centrifugal feed pump is controlled to cause the discharge pressure of the pump to follow the possible minimum feed resistance line with an automatic feed flow regulator operated from the boiler water level.

Fig. 19 shows a multi-stage turbine driven centrifugal feed pump with the steam supply to the turbine controlled by a differential pressure governor in conjunction with a Venturi tube in the discharge line of the pump, the arrangement being such that the discharge pressure of the pump is automatically controlled to follow the minimum feed resistance line such as E D_4 in Fig. 7.

Pressure from the Venturi throat at L is taken to the top of the differential piston at F while pressure from the Venturi inlet at K is taken to the annulus under the differential piston at G. At light loads, the two pressures at K and L are approximately the same and since the pressure at G balances the pressure on an equal area at F the only unbalanced pressure is that on the piston H whose underside is open to the atmosphere. At light loads, therefore, the governor controls exactly in the same way as a pressure governor of the type shown in Fig. 18 having a piston of the same area as H. As the load increases, the differential piston is subjected to an increasing difference of pressure between the points K and L and by choosing suitable dimensions for the Venturi throat this differential pressure is arranged to give the desired pressure output curve corresponding to the minimum feed resistance line. The resultant force on the differential piston falls gradually with increasing feed flow through the Venturi thereby enabling the spring to open gradually the area for steam flow through the ports in the valve cage. For any given load, equilibrium is obtained between the resultant force on the differential piston and the load of the spring giving an opening of the throttle valve ports to pass the quantity of steam required to drive the feed pump at the discharge pressure on the minimum feed resistance line corresponding to the given feed flow.

Fig. 20 illustrates the action of the feed regulator and the discharge pressure characteristic obtained with the Venturi control. F D_4 shows the discharge pressure characteristic which is obtained with a pressure governor control of the type shown in Fig. 18 while E D_4 shows the discharge pressure characteristic with the Venturi control and differential governor shown in Fig. 19. The Venturi control effects a saving in steam consumption of the feed pump at

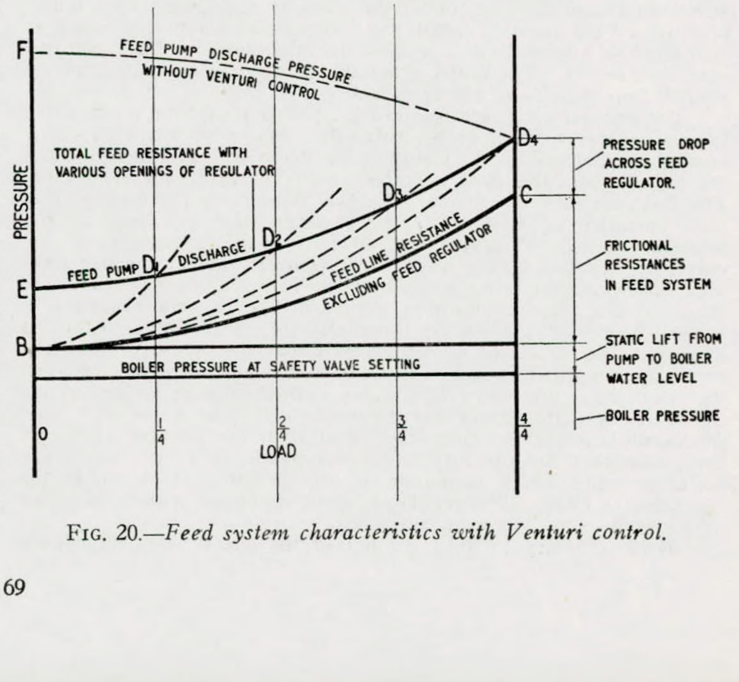

FIG. 20.—*Feed system characteristics with Venturi control.*

all loads below the maximum due to the difference in discharge pressure between the lines F D_4 and E D_4. At any given boiler load the feed regulator must close until equilibrium is reached between the feed flow into the boiler, the boiler load and the corresponding discharge pressure of the feed pump. For example, the openings of the feed regulator for loads 1/4, 2/4, 3/4, and 4/4 give total resistance curves of B D_1, B D_2, B D_3 and B D_4 respectively and equilibrium is reached at the points D_1, D_2, D_3, and D_4 respectively. The line B C represents the resistance to feed flow apart from the feed regulator while the ordinate between E D_4 and B C represents the pressure drop across the feed regulator necessary for control purposes.

Types of Boiler Feed Regulators.

Fig. 21 shows a Mumford Feed Regulator which is used with direct-acting feed pumps in both single and multiple boiler installations. It is used to control the feed flow into the boiler and, in conjunction therewith, it is necessary to control the feed pump motive

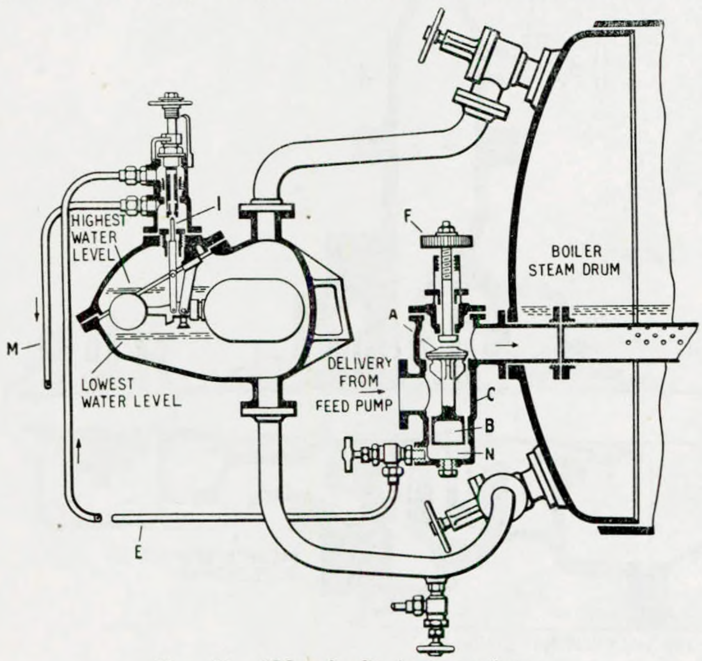

FIG. 21.—*"Mumford" feed regulator.*

power supply either manually or automatically.

The regulator comprises a float gear which is responsive to changes in the boiler water level and a feed check valve mounted on the boiler to regulate the feed flow into the boiler.

The feed check valve A is combined with a piston B having the same diameter so that it is balanced in respect of the feed pump discharge pressure. The top of the valve is subjected to the boiler pressure all the time. Beneath the piston is a chamber N which is connected by a small bore pipe E to the inlet side of a float operated needle valve (1). The outlet from the needle valve is connected by a small bore pipe M to the feed tank.

The annular clearance between the piston B and the check valve body C forms a small orifice through which feed water can flow from the upper side of the piston to the chamber N and then through the pipe E past the float operated needle valve to the feed tank. The float actuates the needle valve which controls the flow of this small quantity of feed water the pressure of which supplies the secondary source of power used to control the movements of the valve A. By this means, very slight changes in boiler water level applied to the float bring into play large forces applied to the check valve so that the regulator is very sensitive to small changes in boiler water level. When the float falls, the needle valve closes and stops the flow of water with the result that the pressure under the piston rises until it is sufficiently above the boiler pressure to open the check valve and allow feed water to flow into the boiler. When the float rises, the needle valve opens and the area for flow past the needle valve is so large compared with the annular area past the piston that the pressure below the piston falls well below the boiler pressure which, acting on top of the check valve, causes the regulator to close. The regulator opens or closes with a variation of water level of less than ½in.

It is necessary to limit the lift of the check valve by manual adjustment of the wheel F so that the opening of the check valve is slightly greater than that necessary to feed the boiler with continuous flow.

It will be seen that the regulator is either closed or full open and is intermittent in its action. The flow of water into the boiler is either stopped or flowing at the rate fixed by the opening of the check valve fixed by the adjustment of the position of the valve spindle. The regulator cuts off the flow of water into the boiler when the level rises above the working level and requires manual adjustment as described.

With direct-acting pumps, the intermittent action of the regulator gives satisfactory operation with the reciprocating action of the feed pump, but the regulator is not so suitable for operation with centrifugal feed pumps, because of its intermittent action. The closer the manual adjustment of the lift of the check valve, the nearer the regulator approaches to continuous feeding. Manual or automatic control of the steam supply to the feed pump is also necessary to avoid unnecessarily high discharge pressures and, consequently, wastage of steam.

Taking a 2in. bore Mumford regulator for use with a boiler operating at 200lb. per sq. inch and a feed discharge pressure of 250lb., it will be seen that the pressure under the piston, when the needle valve is shut, can rise to 250lb. so that there is a pressure difference of 50lb. per sq. inch, i.e. a force of 150lb. available to cause the check valve to open. When the needle valve is open, the piston pressure falls to about 50lb. per sq. inch, giving a force of 470lb. to ensure the closing of the valve.

These forces are brought into play by the movement of the float of less than ½in., which equals a float displacement corresponding to a force of about 3·8lb. These figures demonstrate how the small force available by the change of water level operating on the float can control the operation of a large force from a secondary source of power. This source of power is obtained by allowing a small quantity of feed water at the discharge pressure of the feed pump to flow in series through the fixed orifice formed by the clearance between the check valve piston and the valve body and the variable orifice formed by the needle valve which is opened and closed by the rising and falling of the float. The quantity of feed water used for this purpose is about 1 per cent. of the capacity of the feed pump and is, therefore, a negligible amount.

For large boiler installations, the boiler feed pumps are usually of the centrifugal type and are frequently working in series with other centrifugal pumps in a closed feed system. The flow of water

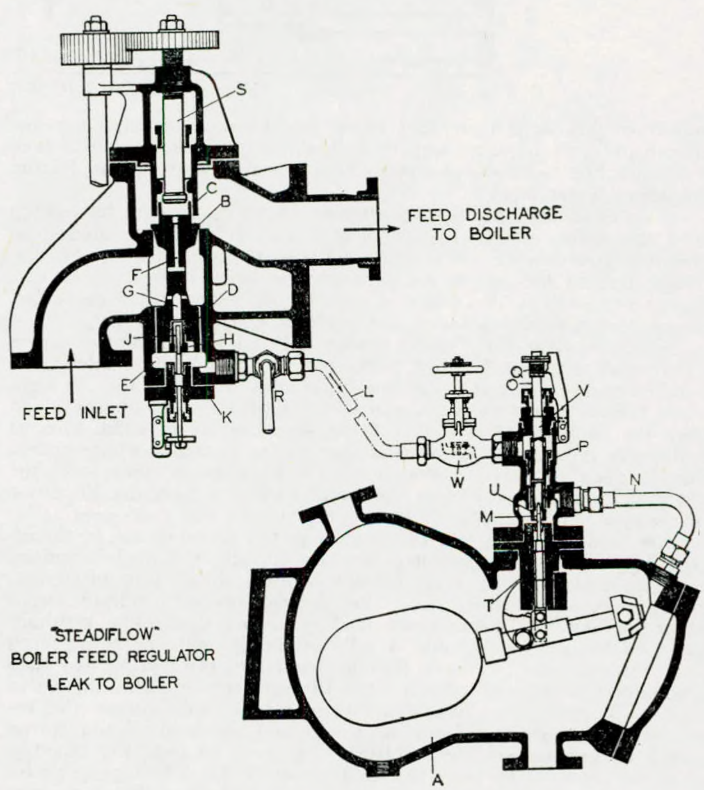

FIG. 22.—*"Steadiflow" boiler feed regulator leak to boiler.*

through these pumps must be steady if the various pumps in the system are to operate efficiently and without trouble. Further, heaters are usually arranged in the feed line to heat the feed water before it is discharged into the boilers. The heating steam for such heaters may be steam bled from a main turbine, the exhaust steam from auxiliary machinery, evaporator vapour or low-pressure steam from other sources. Wide and rapid variations in the flow of water through these heaters cause undesirable fluctuations in the quantities of steam condensed in the heaters and in the pressure of the steam in the heaters. Such pressure variations in the auxiliary exhaust system seriously affect both the mechanical working and the efficiency of the whole of the auxiliary machinery. The boiler feed regulator for such installations must control the feed water supplied to the boilers so that the flow of feed water through the pumps and heaters is steady and continuous under all conditions except when no evaporation is taking place in the boiler.

Fig. 22 shows the Steadiflow regulator which is very similar to the Mumford regulator just described but with modifications to eliminate the intermittent action and obtain continuous feeding. The operation is based on the continuous flow of a small quantity of feed water at feed pump discharge pressure through two variable orifices in series, the water, after leaving the second orifice, passing into the boiler.

The regulator comprises a float box A working in conjunction with a check valve which is mounted on the boiler drum. The check valve consists of a skirted check valve B with an upper piston C of smaller diameter and a lower piston D of the same diameter as the valve. The upper piston is about half the area of the lower piston, and moves in a cylinder formed in the valve chest cover. A hole F is bored through the valve B and the piston C so that feed discharge pressure is maintained at all times on top of the small piston C. Another hole G is bored through the valve stem and the lower piston D whereby feed discharge water can flow into the cylinder E formed by the valve chest round the lower piston D. The outlet of the hole G into the lower piston chamber E is provided with a renewable orifice plug H which is parallel in its bore but is provided with an adjustable leakage control valve J which has an inverted taper and is secured in the cover K on the bottom of the valve chest and arranged so that it can be raised or lowered, thereby enabling the annular orifice between the leakage control valve J and the renewable orifice plug H to be varied while the regulator is in operation. This annular orifice forms the first of the two variable orifices in series. The inverted taper provides an orifice which reduces the orifice area as the check valve rises. Water flows into the chamber E on the underside of the lower piston D through this orifice and the quantity of leakage water is reduced as the check valve lifts. There is also a flow of water through the clearance between the piston D and the check valve chest into the piston chamber E.

The lower piston chamber E is connected by a pipe L to the top connection of the float gear unit in which a float controlled tapered needle valve M is operated to close gradually as the water level falls and to open as the level rises. An outlet connection N is arranged to lead the water which leaks past the float needle valve M into the float box A. This float controlled tapered needle valve M forms the second of the two variable orifices in series through which leakage feed water flows from the inlet of the feed check valve to the boiler.

The quantity of operating leakage water varies gradually from a maximum when the feed check valve is closed to a minimum when the feed check valve is full open. The maximum flow of leakage water corresponds to a high level position of the float, while a predetermined low level position of the float is associated with the minimum quantity of leakage water. At any intermediate water level, the float adjusts the float needle valve to pass the corresponding intermediate quantity of leakage water and the check valve is moved hydraulically until the area through the first orifice is modified to that which gives equilibrium. The water level, therefore, varies from a high level when no evaporation takes place to a low level when the boiler is working at full power. At any intermediate boiler load, the regulator maintains a correspondingly intermediate water level.

The action is as follows:—

On a fall in water level, the float needle valve M closes, and builds up the pressure in the bottom piston chamber E and the main valve B rises and partially closes the orifice past the leakage control valve J until the leakages balance one another. On a rise in water level, the reverse happens, the float needle valve M opens and permits the leakage water to pass to the float box through a non-return valve W, and as the pressure underneath the piston E falls, the main valve closes down correspondingly until the leakage past the leakage control valve J and piston D balances the leakage past the float needle valve M, and the main valve B is in equilibrium in the required position for the evaporation rate corresponding to the float position.

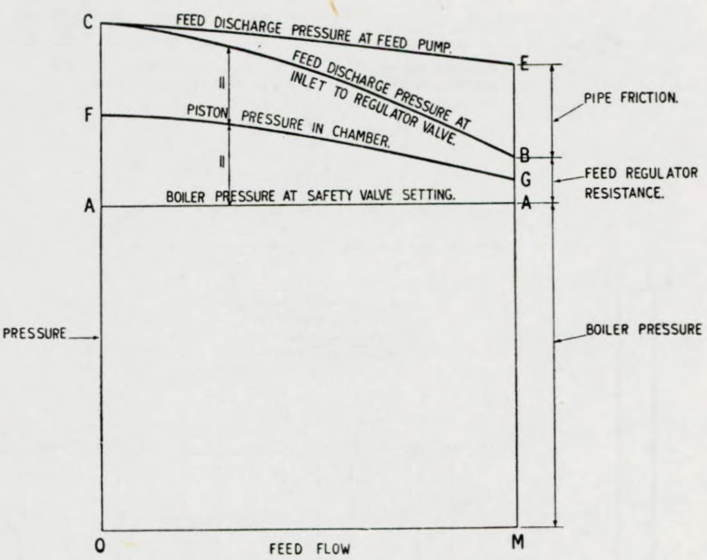

FIG. 23.—*Pressure capacity characteristic relative to a "Robot" or a "Steadiflow" feed regulator.*

The following gives an explanation of the forces acting on the Steadiflow regulator. Referring to Fig. 23, the line AA shows the boiler pressure, C B the falling characteristic of the feed pump discharge pressure, measured at the inlet to the Steadiflow regulator, while F G is the equilibrium pressure below the regulator lower piston and is half way between AA and CB.

The operation of the regulator is based on the continuous flow of a small quantity of feed water at feed pump discharge pressure through two variable orifices in series, the water, after leaving the second orifice, passing into the boiler.

The annular orifice between the inverted taper leakage control valve and the orifice plug forms the first of the variable orifices in series. The inverted taper provides an orifice which reduces the orifice area as the check valve rises. There is also a flow of water through the clearance between the piston and the check valve chest into the piston chamber. The second variable orifice is formed by the float controlled tapered needle valve, and this orifice is increased by a rising float and reduced by a falling float.

The quantity of operating leakage water varies gradually from a maximum when the feed check valve is closed to a minimum when the feed check valve is full open. The maximum flow of leakage water corresponds to a high level position of the float, while a predetermined low level position of the float is associated with the minimum quantity of leakage water. At any intermediate water level, the float positions the float needle valve to a given orifice area which will pass the corresponding intermediate quantity of leakage water and the check valve is moved hydraulically until the area through the first orifice is modified to that which gives equilibrium. The water level, therefore, varies from a high level when no evaporation is taking place to a low level when the boiler is working at full power. At any intermediate boiler load, the regulator maintains a correspondingly intermediate water level.

It will be seen that the regulator automatically adjusts itself to take care of the feed pump characteristics, the frictional losses in the feed system, and also takes care of any variations in boiler pressure.

The small upper piston is subjected to the feed discharge pressure all the time and its area is chosen so that the piston pressure in the chamber under the lower piston is approximately half way between the line CB and the line AA on Fig. 23 when the regulator is in equilibrium at any given rate of evaporation.

The quantity of operating leakage water is proportional to the pressure drop across the annular clearance between the lower piston and the cylinder in which it works and the area past the inverted tapered needle valve engaging with the renewable orifice plug in the lower piston.

The operating leakage water in the regulator varies from less than one per cent. of the maximum flow of water into the boiler at maximum feed flow to about twice the quantity at no evaporation.

Fig. 24 illustrates the forces available to control the movements of a Steadiflow regulator. At the top of the figure is shown diagrammatically the flow of the operating water which passes

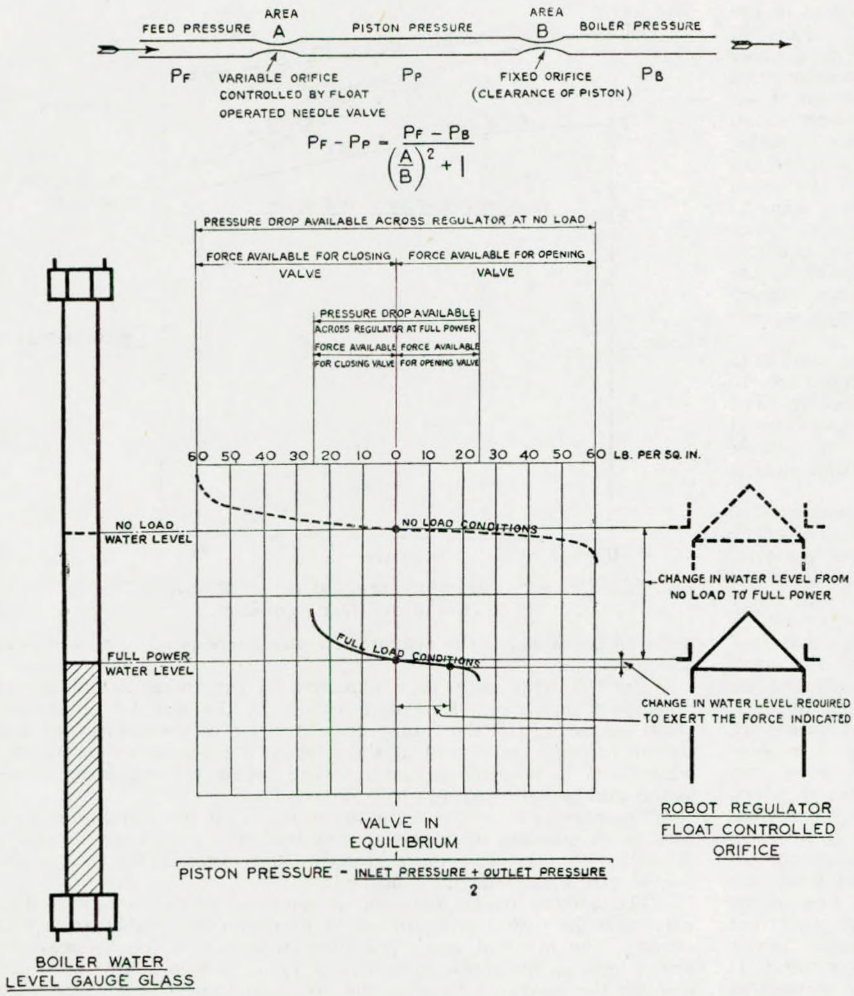

FIG. 24.—*Diagram showing forces available for movement of "Steadiflow" regulator valve.*

through the two orifices in series, as previously described. The piston pressure is between the two orifices and is controlled by the opening and closing of the needle valve which is operated by the float and varies the area of the second orifice to give the area necessary to maintain equilibrium of the regulator valve for steady conditions and as necessary to effect the desired movements of the regulating valve to suit a change of conditions.

The pressure in the piston chamber can be determined from the formula :—

$$P_F - P_P = \frac{P_F - P_B}{\left(\frac{A}{B}\right)^2 + 1}$$

where P_F = the feed discharge pressure at the inlet to the Steadiflow regulator.

P_P = the pressure in the piston chamber.

P_B = the boiler pressure.

A = the variable area past the check needle valve plus the piston clearance.

B = the variable area past the float controlled needle valve.

From Fig. 23, it will be seen that $P_F - P_B$ varies from a maximum AC at no load to a minimum AB at maximum feed flow. Generally, AB is of the order of 50lb. per square inch and AC may be 120lb. per square inch or more, dependent upon the feed pump characteristic and the frictional resistance in the feed system.

For equilibrium, the piston pressure under the lower piston is about half way between CB and AA in Fig. 23.

For a given change in water level from X to Y, the needle valve rises and reduces the area B to such an extent that the piston pressure is raised by about 15lb. per square inch; in fact, however, the regulator moves with a very much smaller pressure change in the piston chamber. In the opposite direction, a rise of water level would cause the needle valve to open the orifice B, and the maximum increase in piston pressure which can be obtained is about 25lb. per square inch. The full line in Fig. 23 shows the forces available at full load with the corresponding change in water level required to give such forces, while the dotted line shows the forces available at no load for any changes in the water level from the equilibrium point Z.

It will be seen from the diagram that the regulator valve is hydraulically constrained to follow the movements of the needle valve because any movement of the float controlled needle valve alters the area of the orifice B and brings into action an hydraulic force to move the regulator valve to a position relative to the check needle valve which corresponds to equilibrium pressure in the piston chamber.

Fig. 25 shows a Robot Feed Regulator which allows a steady continuous flow of feed water to enter the boiler at all rates of evaporation and operates equally well with direct-acting, reciprocating or centrifugal feed pumps.

Referring to Fig. 25, the feed regulating valve A is similar to an ordinary non-return valve with a piston B arranged above the valve and working in a cylinder C formed in the valve casing. The piston B is about twice the area of the valve A and is an easy fit in the cylinder C. The discharge from the feed pump enters the regulator by the branch F on the underside of the valve A and, after passing through the valve, flows directly to the boiler from the branch G. A passage K is provided through the valve and piston to the piston chamber D. The entrance to this passage K is controlled by the needle valve H which is actuated by the float J. The float box P is connected to the boiler by pipes coupled to the branches R and S so that the water level in the float box is the same as the water level in the boiler. The movements of the feed regulator valve A are controlled by a small quantity of boiler feed water flowing from feed pump discharge pressure through two orifices in series to the boiler. The first orifice is formed by the opening of the needle valve at the entrance to the passage K and is variable depending upon the relative movements of the needle valve H and the valve A. The second orifice is a fixed orifice and is formed by the annular clearance between the piston B and the valve casing C.

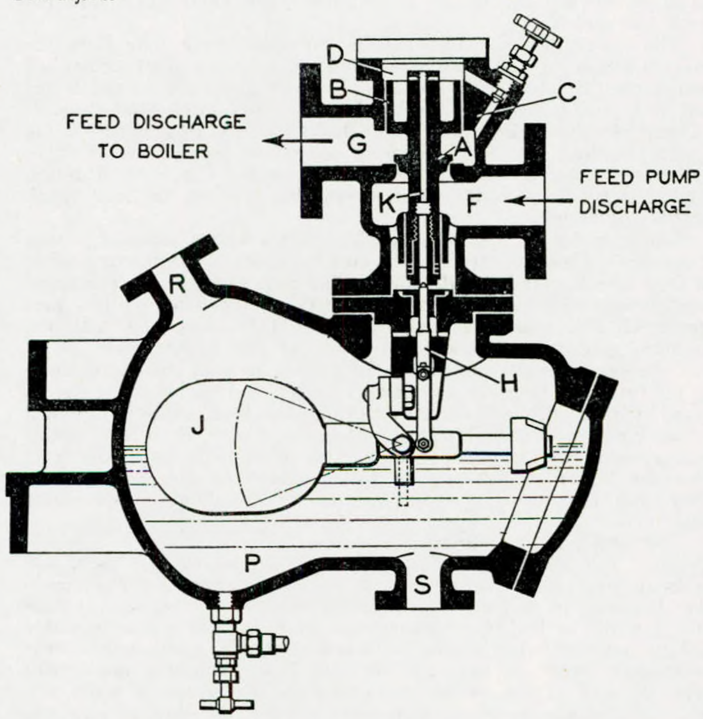

FIG. 25.—*The Weir "Robot" boiler feed regulator.*

The feed discharge pressure acts on the area of the underside of the valve A. The boiler pressure acts on the top of the valve A and the underside of the piston B. When the pressure in the chamber D on the top of the piston B is about half-way between these two, the valve A is in equilibrium and the quantity of feed water flowing to the boiler is maintained at a steady rate.

The pressure in the chamber D is varied by movement of the needle valve H and, since the position of the needle valve H is controlled by the position of the float J, the position of the float J determines the position at which the needle valve A is maintained in equilibrium and therefore the position of the valve A.

The regulator maintains a working water level in the boiler at a predetermined high level when no steam is being generated and the water level falls to a predetermined low level when the boiler is operating at its maximum rate of evaporation. At any intermediate rate of evaporation between no load and the maximum, a steady water level is maintained at a correspondingly intermediate water level.

Assume that the regulator as shown in Fig. 25 is controlling a quantity of water flowing to the boiler corresponding to half load and the boiler water level is rising due to the boiler load being decreased. The float rises and the needle valve H opens the passage K, increases the pressure on the top of the piston B, thereby causing the valve A to close until the valve and needle valve resume their relative positions for equilibrium with equilibrium pressure in the chamber D, at which point the valve A will be passing a reduced quantity of feed water into the boiler corresponding to the reduced boiler load.

If the float falls due to the boiler load being increased, the needle valve closes the opening of the passage K, the pressure in the chamber D falls, and valve A will rise until the equilibrium relative positions of the valve A and the needle valve H are restored with equilibrium pressure in chamber D. The movement of the needle valve H is the master movement, while the movement of the valve A is the slave movement. The position of the needle valve H is determined by the float in accordance with the water level in the boiler, and the valve A is hydraulically compelled to take up the same position at all times relative to the needle valve. For any movement of the needle valve, the main valve must move in exactly the same direction and to exactly the same extent and, whenever the needle valve is stationary, the main valve is held in equilibrium in a partially open position corresponding to the needle valve position.

It is of interest to examine the various forces acting upon a Robot regulator.

Referring to Fig. 23, the line AA shows the boiler pressure, CB the falling characteristic of the feed pump discharge pressure measured at the inlet to valve A, while FG is the pressure on top of the feed regulator piston and is half way between AA and CB. The quantity of leakage water is a function of the pressure drop across the fixed orifice, i.e., the ordinate between FG and AA and, therefore, falls slightly as the feed flow increases. The quantity of this leakage water is less than half of one per cent. The regulator automatically adjusts itself to take care of the feed pump characteristic, the frictional losses in the feed system and also takes care of any variations in boiler pressure.

Fig. 26 illustrates the forces available to control the movements of the Robot regulator.

At the top of the figure is shown diagrammatically the flow of the operating leakage water which passes through the two orifices in series, as previously described. The piston pressure is between the two orifices and is controlled by the opening and closing of the needle valve which is operated by the float and varies the area of the first orifice to give the area necessary to maintain equilibrium of the regulator valve for steady conditions and as necessary to effect the desired movements of the regulator valve to suit a change of conditions. The first orifice is, therefore, formed by the positioning of the float controlled needle valve relative to the needle valve seat in the feed regulator, movements of the float causing the needle valve to increase or decrease the available area for flow of operating leakage water through the needle valve seat to the feed regulator piston chamber. The second orifice is a fixed orifice and is provided by the annular clearance area between the feed regulator piston and the cylinder in which it operates. The quantity of water flowing through the clearance between the piston and the regulator valve chest varies with the pressure difference across the piston as described in relation to Fig. 23 and, for equilibrium purposes, the needle valve adjusts the area of the variable orifice so that the feed regulator is in equilibrum with a pressure on top of the piston approximately half way between the feed discharge pressure measured at the inlet to the regulator valve and the feed discharge pressure as measured at the outlet from the feed regulator valve. The pressure in the piston chamber can be determined from the formula:—

$$P_F - P_P = \frac{P_F - P_B}{\left(\frac{A}{B}\right)^2 + 1}$$

where P_F = the feed discharge pressure at the inlet to the Robot regulator.

P_P = the pressure in the piston chamber.

P_B = the boiler pressure.

A = the area past the needle valve.

B = the clearance area past the regulator piston.

From Fig. 23, it will be seen that PF−PB varies from a maximum AC at no load to a minimum AB at maximum feed flow. Generally, AB is of the order of 50lb. per square inch and AC may be 120lb. per square inch or more, dependent upon the feed pump characteristic and the frictional resistances in the feed system.

For equilibrium, the piston pressure is about half way between CB and AA as shown by FG. If it is assumed that the boiler is operating at maximum load, the point H in Fig. 26 shows the equilibrium position for water level and piston pressure.

For a given change in water level from H to J, the needle valve rises and reduces the area A to such an extent that the piston pressure

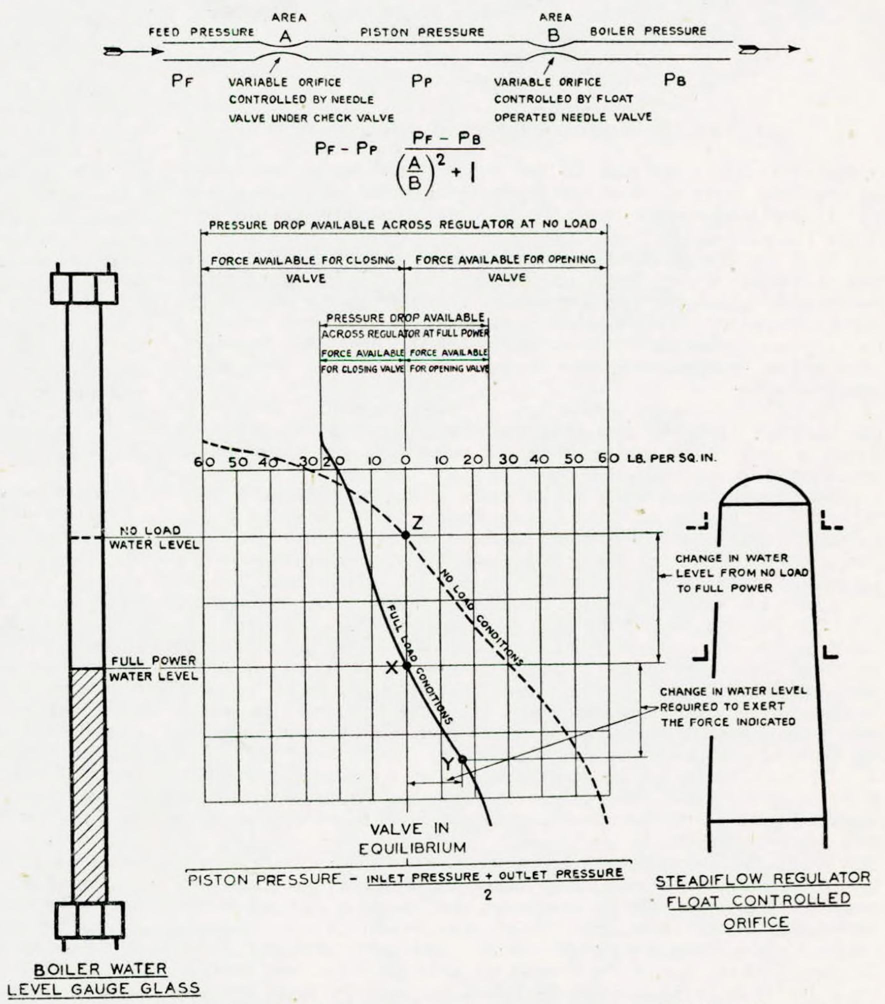

FIG. 26.—*Diagram showing forces available for movement of "Robot" regulator valve.*

is reduced by about 15lb. per square inch; in fact, the regulator moves with a very much smaller pressure change in the piston chamber. In the opposite direction, a rise of water level would cause the needle valve to open the orifice A and increase the piston pressure up to a maximum of 25lb. per square inch. The full line in Fig. 26 shows the forces available at full load with the corresponding change in water level to give such forces, while the dotted line shows the forces available at no load for any change in water level from the equilibrium point K. It will be seen clearly from this diagram that the regulator valve is hydraulically constrained to follow faithfully the movements of the needle valve because any movement of the needle valve alters the area of the orifice A and brings into action an hydraulic force to move the regulator valve to the position relative to the needle valve which corresponds to equilibrium pressure in the piston chamber.

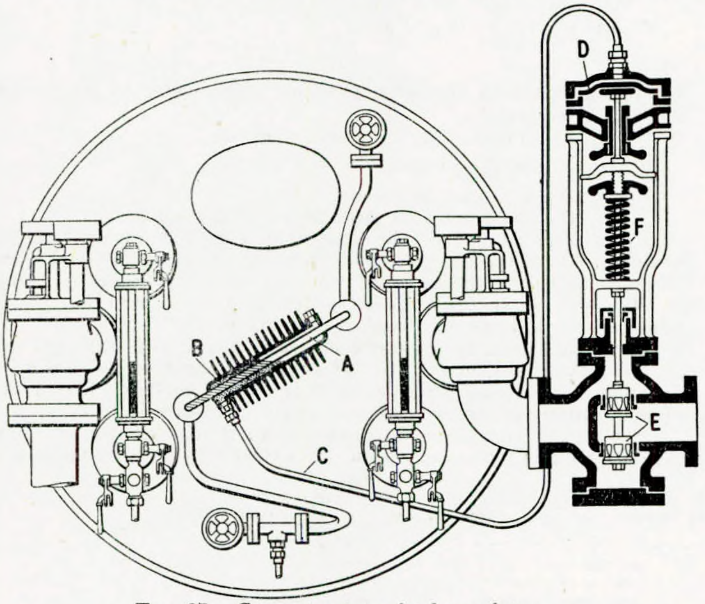

FIG. 27.—*Generator type feed regulator.*

With the Robot regulator, a very small change in water level and, therefore, a very small movement of the needle valve creates the maximum possible pressure change available on the area of the regulator piston. The regulator, therefore, is extremely sensitive to very small changes in the movement of the float and, therefore, exercises an extremely sensitive control over the flow of feed water into the boiler.

Fig. 27 shows a Generator type of feed regulator. Its operation depends upon the fact that the volume of a given weight of steam is very much greater than the volume of the same weight of water at the operating pressure.

The generator consists of an inner and outer tube, the inner tube A being connected to the boiler drum so that the upper end is in communication with the steam space of the drum while the lower end is connected to the water space, so that the water level in the inner tube is the same as the water level in the boiler drum.

The outer tube B forms a closed vessel with a connecting pipe C to the diaphragm D of the feed regulating valve E. Fins are provided on the outer diameter of the outer tube. The space between the inner and outer tubes A and B is filled with water when the feed regulating valve is closed and the diaphragm is in its outer position. The feed regulating valve is of the balanced type which may have ports, as shown, or skirts to give a gradual opening for the flow of feed water as the valve opens.

Pressure on top of the diaphragm acts in opposition to a spring F to open the valve, the spring load and the diaphragm pressure load being in equilibrium for any steady feed flow corresponding to a given boiler load.

When the regulator is placed in operation, the steam in the upper portion of the inner tube causes the water in the space surrounding the inner tube to evaporate and steam is formed in the upper part of the outer tube. Generation of this steam sets up a pressure in the generator system and such pressure is applied through the pipe C to the top of the diaphragm and causes the feed regulating valve E to open until the feed flow through the valve matches the load of the boiler. The lower the water level falls in the inner tube A the greater the amount of steam generated in the outer tube B and the greater the pressure exerted on the diaphragm D of the feed regulator. When the water level in the boiler rises, some of the steam in the outer tube condenses, due to radiation from the tube and its attached fins, thereby lowering the pressure in the generator system so that the spring forces the diaphragm and the valve into such position that the flow through the feed regulator is reduced until it is again in equilibrium with the boiler load. It will be seen that there is a continuous transmission of heat from the inner tube to the water surrounding it and from the outer tube and its fins to the surrounding atmosphere and the heat flow varies with the water level in the inner tube. Due to this heat flow, steam is generated to a greater or lesser extent inside the outer tube dependent upon the level of the water in the inner tube and the radiation to the atmosphere.

The water level in the boiler varies from a predetermined high level at no load to a predetermined low level at maximum boiler evaporation, the volume of steam inside the outer tube increases from a minimum to a maximum and minimum and maximum pressures are developed in the generator system which correspond to the closed position of the check valve and the full open position respectively.

This variation in pressure in the generator system is matched against the characteristic of the spring and the valve port areas so that for any intermediate boiler load, an intermediate water level is obtained in the boiler and the feed regulating valve passes a steady flow of water to the boiler at an intermediate opening corresponding to the intermediate boiler load. The regulator is, therefore, of the continuous feeding type and small changes in the boiler water level are used to generate steam as a secondary source of power to apply the necessary force to the diaphragm operating the regulating valve.

The pressure in the generator is always appreciably lower than the boiler pressure dependent upon the design of the generator. The generator pressure is of the order of 50 to 100lb./sq. inch. The diaphragms vary in size with different sizes of regulators. Generally, diaphragms are between 4in. and 5in. in their effective diameters giving an area of 12 to 20 square inches on which the generator pressure is exerted. A typical spring design is such that a load of 200lb. is required to compress it 1in. Very slight changes in the boiler water level cause the generator pressure to vary by a pound per square inch which immediately brings into play a force of 12 to 20lb. to move the feed regulating valve into a new position. Care must be taken in the operation of this type of regulator to ensure that all joints on the generator system are completely tight since the water level in the boiler drum depends upon the amount of water in the generator system and any leakage of water from the generator system results in a correspondingly lower water level being carried in the boiler drum.

In the arrangement of this regulator all the necessary precautions are required to ensure that the water level inside the inner tube of the generator is the same as the water level in the boiler drum and the generator must be so placed on the boiler that rolling or heeling of the ship does not vary the water level inside the generator inner tube. This type of regulator can be used equally well with direct-acting, reciprocating or centrifugal feed pumps.

Arrangement of Feed Regulator on Boiler Drum.

A boiler feed regulator can only operate in accordance with the water level established in the regulator. It is important, therefore, that the water level in the float box, or other device which is responsive to water level changes, should reflect the changes of the boiler water level faithfully and accurately. The steam and water connections between the boiler and the float box should, therefore, be as short as possible and should be the full area throughout their length. Isolating valves in these connections should be of the full way type. The steam connection should be free from water pockets, while the water connection from the float box to the boiler drum should be so arranged as to avoid any vapour pockets which might cause a vapour lock. In marine boilers, these requirements must be met with the ship heeled at any angle up to, say, 20° on either side of the normal horizontal. The float box should also be arranged on a marine boiler so that rolling does not interfere with the operation of the regulator. The steam drums of marine boilers are usually arranged in a fore and aft direction, and the float box should be placed on the centre line of the steam drum in the fore and aft line. The necessity for this is illustrated by Figs. 28 and 29; Fig. 28 shows the effect on the float box if it is offset from the centre line of the boiler drum as shown by the line FF.

Assuming that the feed regulator is of the continuous flow type, with a stroke between no load and maximum feed flow, the point B shows the float centre at no load, while A represents the float centre at full power with C the level at which the float will allow the

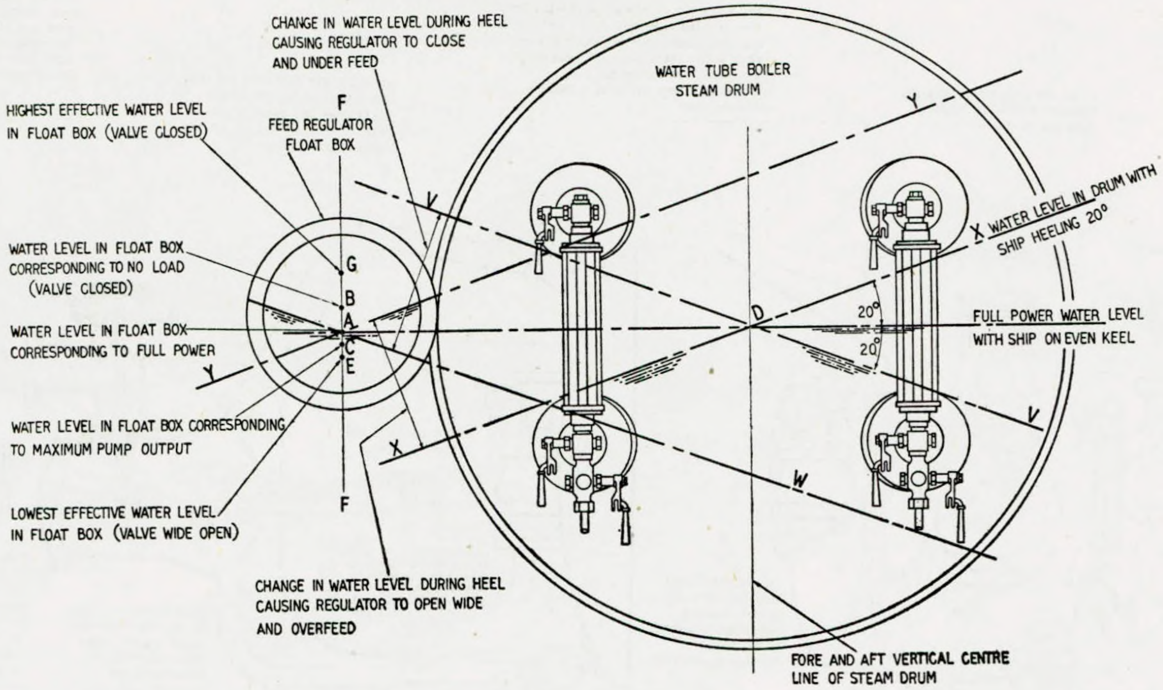

FIG. 28.—*Diagram showing effect of rolling ship on the action of a boiler feed regulator with float box offset from centre line of drum.*

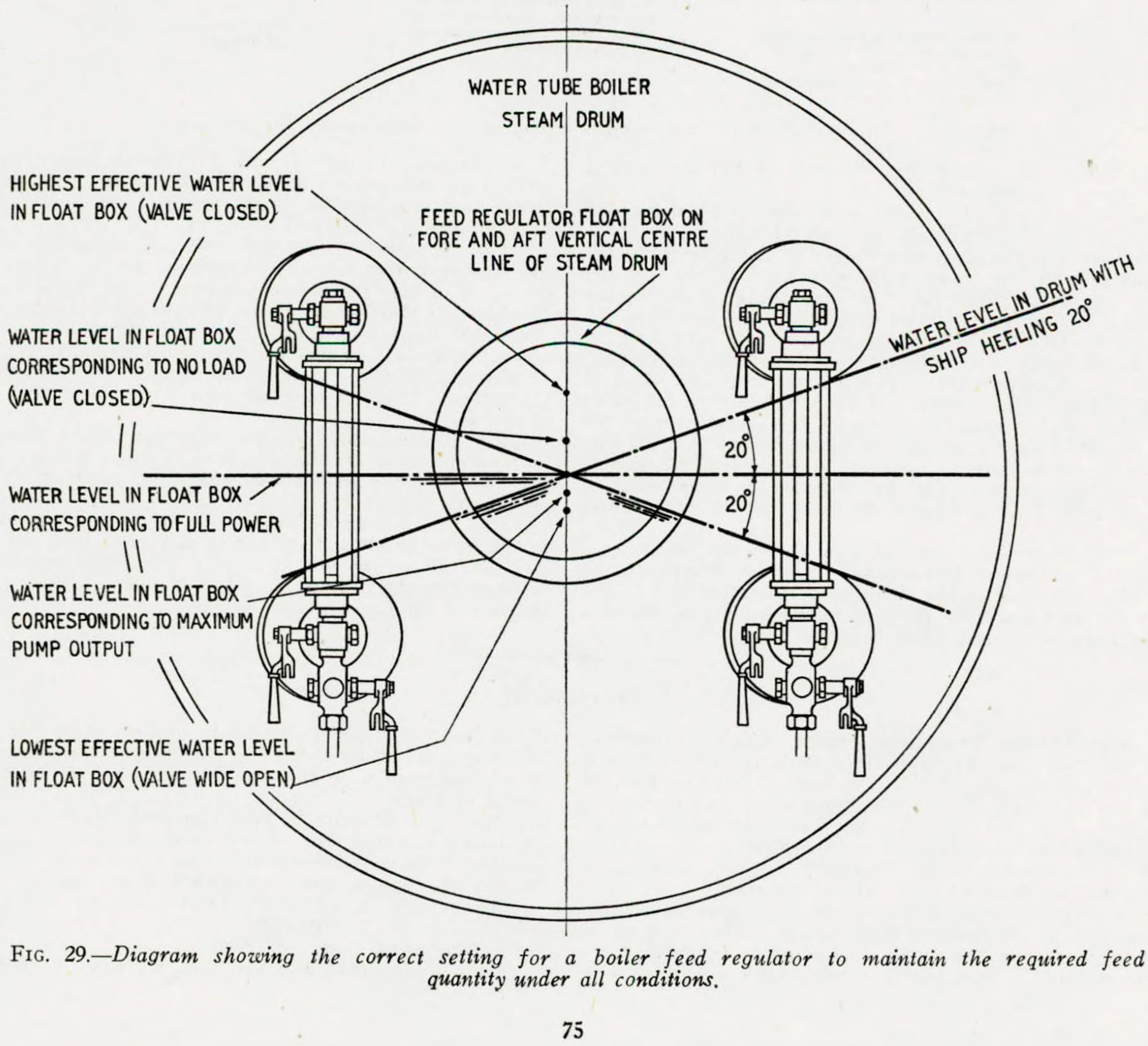

FIG. 29.—*Diagram showing the correct setting for a boiler feed regulator to maintain the required feed quantity under all conditions.*

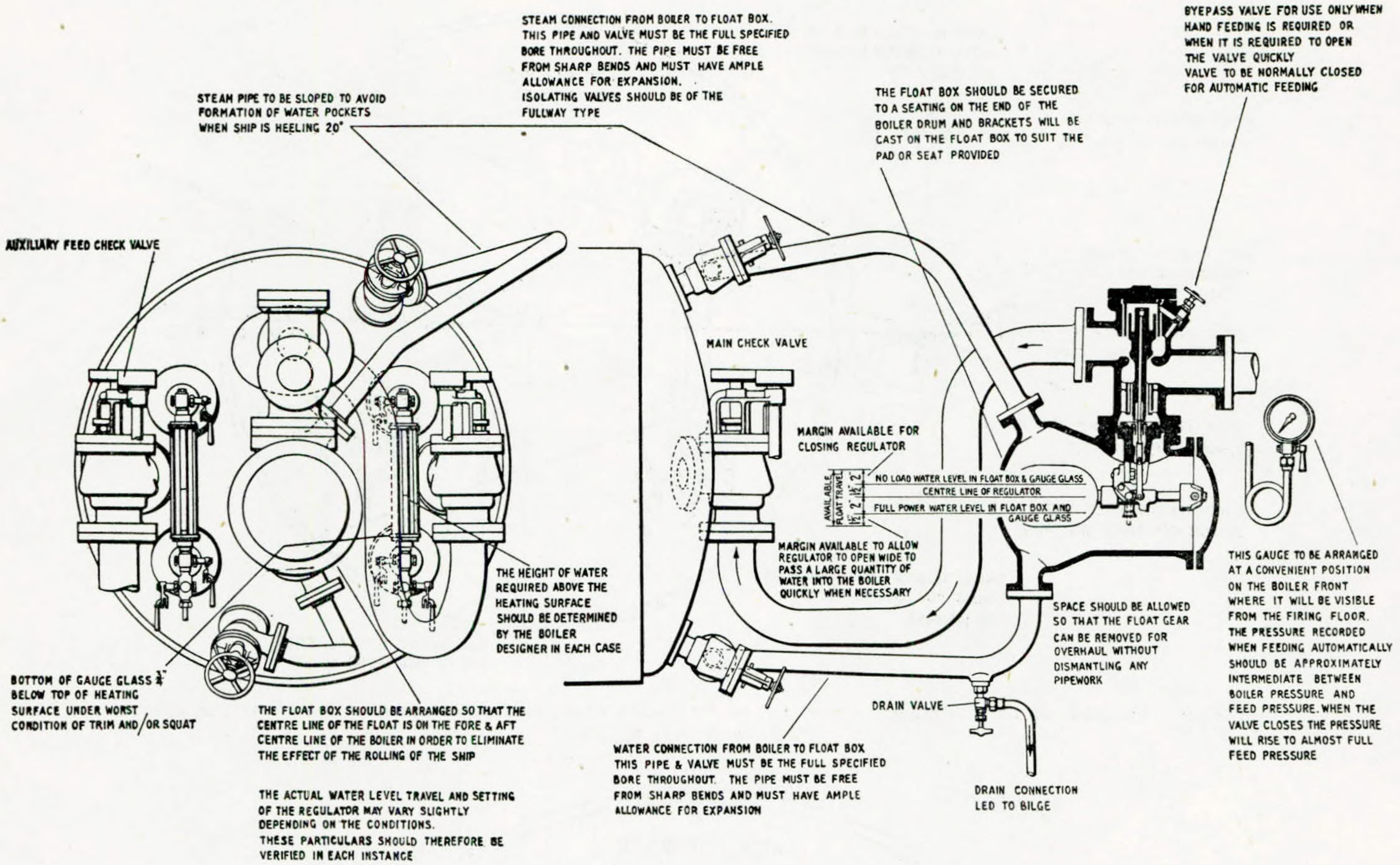

FIG. 30.—*"Robot" feed regulator; diagram for arrangement purposes.*

maximum feed flow to enter the boiler. When the boiler is operating at full power, the water level will be AD.

Now, when the ship rolls, the water level in the steam drum will oscillate above the fore and aft line passing through the point D, while the water level in the float box oscillates about the fore and aft line passing through A. Assume the ship heels 20° so that the water level in the steam drum AD takes up the level XDX, the immediate effect is that water flows from the float box into the steam drum and the level in the float box falls rapidly. As the level in the float box reaches C, the regulator opens wide and permits the feed flow into the boiler to increase to the maximum rate, which is in excess of the boiler requirements, and the water level in the boiler rises gradually to YAY, when the regulator has again reduced the feed flow to the full power requirements of the boiler. During this period, a considerable amount of additional water above the steaming requirements has been pumped into the boiler, as represented by the rise in level from XX to YY and the level may be dangerously high.

A heel of 20° in the opposite direction causes the boiler level to swing to VDV and water flows from the steam drum into the float box so that the level in the float box rises rapidly. As the water level in the float box rises, the feed regulator cuts off the feed flow until the flow is stopped when the level passes the point B. A considerable period of time will elapse before the water level VDV falls and permits the feed regulator to open and pass the full power feed requirements of the boiler at the level AW which may be a dangerously low level.

Fig. 29 shows clearly how these disadvantages are avoided if the regulator float box is arranged on the centre line of the steam drum. With this arrangement, rolling of the ship does not affect the operation of the feed regulator because the water level in the steam drum and the water level in the regulator float box both oscillate about the same fore and aft line at all times, there is no transfer of water between the steam drum and the float box, and the regulator passes a steady continuous feed flow independent of the rolling of the ship.

Fig. 30 illustrates how the steam and water connections should be arranged to ensure that the flow of steam and water to and from the float box can be effected satisfactorily under all conditions of operation.

The float box should, of course, be positioned vertically relative to the boiler drum and the gauge glasses to give the desired water level at full power and so that a reasonable margin is available between both the full power and no load water levels and the bottom and top of the gauge glasses respectively.

Discussion

Engineer Vice-Admiral Sir George Preece, K.C.B. (President), said that he was very glad to have the opportunity of welcoming his old friend Mr. Hillier to the Institute, and of thanking him, on behalf of all the members, for the most instructive and interesting paper which he had given. To every engineer concerned with the operation of highly forced boilers the feed regulator was an item of equipment of the most vital importance, on the satisfactory behaviour of which his peace of mind largely depended.

He thought that the author had modestly understated the really remarkable capacity of the modern feed regulator when he said on page 59 that "if neglected, the water level can fall to a dangerous level in a few minutes". In a highly forced boiler of Naval type, if the feed supply were completely shut off when it was steaming at full power, the water level disappeared from the gauge glass in some 21 seconds, whilst in 68 seconds the steam drum was completely devoid of water.

The chief difficulty he found in studying the behaviour of feed regulators was that they seemed to be affected by so many different variables in an installation comprising a battery of boilers, more than one set of engines and a multitude of associated or independent auxiliaries, that it was very difficult to find which factors were individually, or it might be collectively, responsible for certain phenomena which were observed.

He had no doubt that the speakers who followed would raise various points in which they were particularly interested, so he would confine himself to a few relatively minor points of detail. For instance, in Fig. 1, was the line HL indicating the water level in the gauge glass actually a straight line function of the boiler output, or was this an assumption and just taken as an example? Then again, in the same figure, was it an established fact that if HL were a straight line HK would also be a straight line or, in other words, did the ratio of the volume of steam below water level to the total volume of water in the boiler actually vary with the boiler output in this simple way? Finally, as regards the same figure (1), was it strictly correct to use this diagram to illustrate the effect of varying the "draw off" of steam from the boiler, considering that HL and HK, he imagined, referred to conditions of steady boiler pressure, whilst under the conditions discussed there must be some rise or fall in the boiler pressure?

There was one other point of detail on which he would welcome the author's opinion. He had remarked that the float box was connected to the boiler by pipes so that the water level in the float box was the same as that in the boiler. Would not the water in the float box be quiescent and free from steam bubbles whereas the water in the steam drum would, be imagined, be very much less dense and so presumably the water level in the boiler—if it could be described as a level at all—was appreciably higher than that in the float box and incidentally in the gauge glasses. On occasions he had drawn some admittedly slender comfort from this supposition, which he hoped was correct.

Although it was almost a subject for a paper in itself, he was sure that the author's opinion would be welcomed on the so-called "hunting" which sometimes occurred in boilers. One could see that if a large quantity of relatively cold water was admitted suddenly to a highly forced steaming boiler, a sort of rake's progress began. The cold water collapsed the steam bubbles below the surface of the water in the boiler, causing a lowering of the water level and causing the feed regulator to open still more and allow still more water to enter. If feed heaters were fitted, the increased flow of feed water caused the feed temperature to fall, thus aggravating things inside the boiler. Indeed, he saw no end to this, short of the whole water in the boiler going on collapsing until it was all "solid" were it not that two remedial measures automatically came into operation, the first being that the steam pressure in the boiler dropped and so its output was reduced, whilst in an installation where the feed was heated by the "closed exhaust" those auxiliary engines, not governor controlled, tended to speed up under the reduced back pressure and passed more steam to the feed heater and thus heated up the feed water going to the boiler again to a higher temperature, so that ultimately the heat input again began to win. Then in certain cases a swing or surge took place in the other direction, and the water in the gauge glass climbed upwards and then the cycle repeated itself, the period being sometimes of the order of 4 to 5 minutes.

He remembered that when much younger he took a trip on an express locomotive the driver had remarked that he took care to have a good fire and a full gauge glass of water when starting up an incline so that he would not have to admit any feed during the ascent, but the full significance of this was not apparent to him at the time!

The speaker expected and hoped that others present would discuss this phenomenon of surge, so he would not say more than that he would welcome the author's opinion as to whether any regulator could entirely cope with this problem in which there were so many variables. It would be of great interest to hear the author's opinions as to whether it was possible to specify any limiting set of conditions inside which the feed level should not surge in this way. For example, it seemed possible that, even in a single boiler under test, swings of the nature referred to could be originated by too drastic manipulation of the steam outlet valve when attempting to keep a steady boiler pressure. It would appear, however, that the problem would become easier and possibly disappear altogether if it could be ensured that under all conditions the feed water entered the boiler at the saturation temperature of the steam being produced.

Engineer Rear-Admiral S. R. Dight, C.B.E. (Member), said he would like to pay tribute to the magnificent work Mr. Hillier had done in connection with the development of feed regulators and the problems appertaining to them. He had given a very clear exposition of what happened in a watertube boiler when there was a change of output. With the swelling and depression of the water level, and the consequent following of the feed regulator, the watertube boiler was given, in effect, a steam reservoir capacity. When the change over was made from tank boilers to watertube boilers one felt that quite a lot of reservoir capacity was lost, but the action described by Mr. Hillier gave an effect which was equivalent to reservoir capacity which was of very great value.

On the question of hunting, if the feed regulator should suffer from corrosion or dirt or any foreign material coming over with the feed, it might cause the feed regulator piston to stick. There would then be a delay action and a certain amount of hunting. Mr. Hillier had spent many hours on trials of new ships dealing with these hunts, and wondering whether it was the fault of the regulator, of the design, of dirt, or just pure cussedness.

Having made a regulator which would give steady working, there were other peculiar phenomena to be dealt with. If one had a regulator which would feed quite steadily with one boiler in operation, something often happened when two boilers were in operation, so that at one time one boiler was getting a large charge taking up all the feed while the other was getting little, if any, feed. After a certain interval they changed over, and the result was to upset all the conditions in the feed heating system, varying the exhaust pressure and causing the fans and other auxiliaries to hunt. This could be quite an alarming experience. Perhaps Mr. Hillier would enlarge on this.

Commander (E) H. F. Atkins, D.S.O., D.S.C., R.N. (Visitor): Admiral Preece had said that even taking the simplest case of a boiler under test there might be hunting. It had been found, in a series of trials, just completed on a Foster-Wheeler boiler, the simplest possible case of a boiler on shore in a test house, that with a drowned feed pipe there was hunting; it could not be stopped. Mr. Hillier was present and saw that there was no mechanical or other defect, but until their so-called feed pots were fitted, which sprayed the water into the steam space so that the water was at saturation temperature before it entered the down-comer tubes, the hunting went on. Mr. Sampson evolved another scheme which consisted of a feed pipe just above the water level with a sort of weir. It was found that if they put a steam pipe in the steam space they cured the hunting but got a water hammer, and the little trough or weir ensured that the feed pipe was full of water at all times; the water was sprayed upwards and controlled by a hood, so that it did not get mixed up with the steam too much.

Mr. Hillier was proposing to go one stage further and to try the effect of a feed pipe about 3in. below the water level and spraying upwards into the steam space, but the result of that procedure was not yet known.

There was one other point; he would have thought that the thermal type of water level regulator would have been extraordinarily sensitive to draughts, and that when the speed of the fan was increased the air would take away more heat and alter the water level. He would like to know whether the regulator had to be shielded carefully and its surroundings kept at a constant temperature.

Engineer Rear-Admiral Dight: With regard to the point which Commander Atkins had raised about some of the trials, when one was running a boiler on test many conditions did not correspond to conditions of service in the ship. On that particular trial the output of the boiler was controlled by the stop valve, so that if the steam pressure in the boiler tended to rise, the stop valve opening was increased. The result of that was to lower the steam pressure in the boiler and raise the water level, so reducing the amount of feed and thereby increasing the generation of steam, and it was necessary to go on opening the stop valve to a much greater amount to get control. If the steam pressure was going down in the boiler, and the stop valve opening was decreased, the water level was depressed, more feed was consequently put into the boiler, with a further reduction in steam pressure and a further closing of the stop valve to correct it. This method of control defeated the feed regulator completely. His contention was, and his experience of boiler control on boiler trials was that the boiler stop valve should be set to the correct opening and the output controlled on the oil and air supply to the furnace. In that way a neutralising effect was obtained, and steady conditions resulted.

Commander Atkins: At the same time there was a regular hunting.

Engineer Rear-Admiral Dight: It might not completely eliminate hunting which might be due to other causes, but it would make the conditions steadier by eliminating that cause of hunting.

Mr. S. B. Jackson (Member) said he would like to thank Mr. Hillier for his masterly exposition of boiler feed water regulation. Fig. 2 provided the basis of a new boiler feed water regulation now being developed. It would be seen that the constant speed pump characteristic was given by FD and that the highest pressure was necessary at no load. With simple speed regulation of the pump the discharge characteristic was given by BC, which included the static lift from the pump to the boiler water level. Speed control eliminated CD, which was the pressure drop across the regulator; hence the maximum pressure was C at full load, which was considerably below F at no load. Therefore it would seem that such a system of pressure regulation was to be desired.

If one considered a boiler of 300,000lb. per hr., in which he was interested, the discharge pressure at no load on conventional control was 825lb. per sq. in., but on a variable speed controlled pump only 720lb. at full load. Thus lower design pressures of the pump, feed lines and fittings might be employed.

One of the advantages of varying pump speed was that of pump efficiency, for if the pump efficiency was, say, 75 per cent. at full speed, if the speed could be varied, the pump efficiency remained constant right down to no load. In the case referred to this resulted in substantial economy of pump power, averaging 11·5 per cent. over the whole range of pump load; and at 80 per cent. boiler rating, nearly 17 per cent. in addition to the avoidance of excess feed line pressures. As the boiler was not always operating at full load, the general pressures must be even lower than the maximum of 720lb. per sq. in.

The second point about variation of pump speed was related to the pump characteristic. It was found that the resistance of the feed system as a whole varied in accordance with the quantity delivered, and hence to the speed. As the pump quantity delivered varied with the speed, it followed that the natural resistance law followed the same law as the pump with speed. Therefore, the most logical method of controlling boiler feed water levels was obviously by pump speed variation, and proposals, notably from America, for water quantity variation by variable pitch axial flow pumps could not produce the optimum efficiency as was given by speed variation.

The generally accepted speed variation device had been the D.C. motor. The A.C. motor at constant frequency did not lend itself to variation of speed, and hence the development, at almost prohibitive cost, of the commutator motor. However, an electric coupling was now available for use with the constant speed A.C. motor giving the same characteristic as the hydraulic coupling. This enabled the use of relatively cheap squirrel-cage motors and their high reliability in addition to which commutators were eliminated. The Ward-Leonard system was comparatively expensive and could not be justified. The relative prices were as follows:—

	Price.
(1) 3 phase, 50 cycle, 725 r.p.m. motor with electric coupling	1·0
(2) D.C. motor, 725 r.p.m. with rheostats	1·25
(3) 3 phase, 50 cycle, 725 r.p.m. commutator motors... ...	7·0
(4) Ward-Leonard scheme	8·5

The electric coupling consisted of an outer member, the inner periphery of which was slotted; the inner member carried a glass insulated circumferential coil in a recess of the outer periphery supplied by D.C. The outer periphery of the inner member was perfectly smooth. The outer member was driven by the motor and the application of excitation to the coil induced eddy currents in the teeth of the inner periphery of the outer member which created a magnetic flux to interact with the inner member which drove the pump causing rotation. The torque was proportional to the excitation which was only about 0·3 per cent. of the total motor power. The advantage was that the motor could be started on no load and infinite and inertialess variation of the excitation brought the speed up very gradually, lending itself to precision means of control.

This scheme was being developed to suit boiler level control, the initial operating current being the output of 0-10 milliwatts from an amplifier controlling the water level.

Mr. A. B. S. Laidlaw, M.A., M.I.Mech.E., M.I.E.S. (Visitor), said he wished to revert to the question of hunting. Having spent a good deal of time working on feed water regulators he would like to ask boiler designers to try to find some method of getting the water into the boiler so that it did not interfere with the ebullition. He had had a certain amount of experience trying it out and had come to the conclusion that if they were to get a steady effect entirely irrespective of the feed water tube, they had to get the water introduced into the boiler so that the ebullition was not interfered with. He had only found one case in which it was possible to get a perfectly steady effect; in this case, when the water supply into the boiler was increased, one did not get a decrease in the artificial water level or an increase when the feed was decreased; that was in a Lewis boiler, wherein the feed was introduced down the centre of the tubes, and evidently did not interfere at all with the ebullition. One would think that a large tank boiler would be perfectly free from interference with the ebullition in view of the fact that for its water content its evaporation was relatively small compared with a watertube boiler, but this was not so. Tests carried out in a vertical boiler of 5,000lb. evaporation gave a period of something of the order of three minutes; similar tests carried out on a 3-flue wet back boiler with an evaporation of about 20,000lb. gave a period of six minutes. These periods appeared to be independent of anything except of the fact that when the feed regulator came into operation or when there was a change in the rate of supply of feed water there was invariably a corresponding artificial drop in water level. This occurred even in a tank boiler! Boiler manufacturers would be well advised to study the problem because he was quite sure that it was one of the feed water regulator manufacturer's bugbears and that he was trying to overcome a fault which lay inherently in the boiler.

He was brought into touch with this problem by virtue of being a combustion engineer, and one of the first troubles he had in feed regulation was due to the variation in the rate of firing of his combustion apparatus on account of automatic burner controls, and was not due to the feed water regulator. It was, in fact, quite possible for the feed regulator to hunt about, entirely due to changes in the rate of firing the boiler. He would say, therefore, to boiler manufacturers, that they should endeavour to introduce feed water into their boilers so that ebullition was not interfered with.

Mr. Hillier had not given any indication as to whether in modern practice thermostatic regulators of any type had been employed in ships. In ordinary power station practice they had Cope's regulators which were standard on most boilers; he was wondering, therefore, whether the thermostatic regulator had proved too slow in operation for marine practice.

Mr. W. Sampson (Member): The author's clear and detailed paper on the designs and characteristics of feed regulators and feed pump governing arrangements was most timely as just now there was such a swing over to watertube boilers in the Merchant Marine.

It would be a surprise to many engineers to realise how many variables the automatic governing of feed had to take into account to produce the apparently simple result of maintaining a desired water level in the boiler drum. To do this automatically was not such a simple matter as it might appear and in order that due appreciation be given to the author and his firm for their designs and products and for the amount of experiment and study which they had given to feed regulators and pump governing devices, the following requirements were listed, the author making it clear in his paper that regulators and pump governing devices fulfilled these requirements with an amazing degree of reliability.

To maintain a water level meant: (1) varying the rate of feed to adjust the quantity of water in a boiler so as to maintain a constant volume of the mixture of water and steam in the boiler, the volume at any instant depending on

(*a*) the pressure, which might fluctuate quickly,
(*b*) the rate of firing or degree of forcing.
(*c*) the rate of steam take-off,
(*d*) the temperature of the entering feed, which governed the amount of sensible heat added in the boiler;

(2) that the regulator had to be extremely sensitive to all the quickly changing rates of feed required caused by the variations above, and the regulator must be not only sensitive but quick in action; (3) that feed regulators and pump governing devices must be as simple as possible in construction, easily understood by those in charge of them, and utterly reliable; (4) that feed regulators and pump governing devices must fulfil their duties under conditions of roll and pitch, with consequent surge of the boiler water; (5) that they must maintain the volume of steam and water mixture, i.e. constant level, under crash shut-down conditions when the rate of feed might require to be increased enormously for a short period; (6) that they must maintain the volume of the mixture of steam and water under a sudden collapse of pressure in the drum from any cause with a consequent sudden increase in the volume, and under these conditions prevent flooding and priming with possible dangers to machinery.

In the ordinary mercantile boiler at conservative ratings one was used to seeing the regulators maintaining their steady levels under varying conditions, but in highly rated boilers the changes were all of much greater magnitude and happened much more quickly, and those who had seen the regulators described by the author in action under the most severe conditions learned to have utter confidence in automatic feed regulators.

The feed pump governors and feed regulators could not be expected to compensate and keep the boiler in equilibrium without some time lag, although in practice this time lag was extremely small; it could, under adverse conditions, create "hunting" feed level, i.e. a water level which rose and fell rhythmically following the variations in the rate of feed. These adverse influences included such variables as

(1) varying back pressure on feed pump exhausts,

(2) fluctuations or pulsating firing rates of the fuel,
(3) surging or quick fluctuations in the amount of steam take-off from boiler,
(4) loss of water, if caused by intermittent priming and carry over.

Although the means for the correction of these disturbing influences were provided within the feed pump governing and feed regulating devices, it was the time lag which was very hard to overcome, but a cure could be effected by attention to the following points:—

(*a*) by reducing to the smallest possible degree the amount of sensible heat to be added to the feed in the boiler, which meant arranging for the highest possible feed temperature either by heaters or by economisers, and introducing the feed into the boiler by spraying it into the steam space so that it reached saturation temperature before entering the water and steam mixture in the drum and the boiler circulation system in such a manner that there was no sensible heat to add in the water body; spraying the water into the feed drum was the logical thing to do. There were difficulties in that and there were various designs used for the purpose. The reasons for this were made very clear in the author's paper.

(*b*) every device to ensure steady fuel firing conditions should be incorporated in the boiler design, and care should be taken to have the correct proportions of steam drum volumes and steam release areas; fittings should be incorporated in the drums to prevent surge movements under roll and pitch conditions.

There was one point he wanted to make very clearly; most engineers having years of service with feed regulators could testify to their great reliability, and it was interesting to know that nearly all the troubles with regulators seemed to occur either on trial trips or on the first voyage of a vessel. The history of these troubles seemed to indicate that they were nearly always due to foreign material in the feed lines and feed system generally, and a plea was made here for the same attention to be given in new ships to secure cleanliness of the feed systems before they were put into use; in fact, that the feed systems be treated with the same care in this respect as was given to turbine lubricating oil systems, interiors of turbines, etc.

The author described three types of feed regulators, and at the bottom of page 70 mentioned, in describing the Mumford regulator "that the leak-off for this purpose is about 1 per cent. of the capacity of the feed pump and is, therefore, a negligible amount". In certain large vessels using this type of regulator it was found that the leak-off quantity was actually higher than this 1 per cent., possibly averaging about 3 per cent., and it must be remembered that this leak-off, delivering feed at final feed temperature to the feed tank, meant a loss of heat overboard in the main condenser in a closed feed system, and it was a question whether this amount could truly be called "negligible". As the author mentioned that the Robot regulator worked satisfactorily with both reciprocating and turbo feed pumps and had no heat loss through leak-off to a feed tank, could the author indicate under what conditions the Mumford type of regulator would be fitted in preference to the Robot?

Finally, it was thought that the paper would well repay detailed study by engineers, particularly those taking over watertube boilers for the first time.

Mr. H. J. Wheadon (Member of Council): Marine engineers who had experienced the task of hand controlling the water level in a highly rated watertube boiler under manœuvring conditions would agree with him that a reliable automatic feed water regulator was a most desirable fitting, and with the ever increasing rates of evaporation and decreasing water capacities must now be regarded as essential for safety of operation. Many owners who in the past considered watertube boilers insufficiently robust for their particular class of vessel had had them under their wartime management and the experience they and their engineers had gained was, he believed, such that many of them were not likely to revert to the Scotch boiler when they were again able to build steamships to their own specifications. In his view, therefore, the growing confidence in the watertube boiler would enable it rapidly to encroach into the field previously served by the Scotch boiler, and for that reason Mr. Hillier's paper would be of particular and timely value to the seagoing members.

To refer to the paper, it would be observed that Fig. 11 included a relief valve on the discharge side of the pump, whilst in Fig. 9 it was omitted. He had recently experienced trouble with a small installation similar to Fig. 11, except that the pump discharged to a single boiler, which experiences were, he believed, relevant to and of sufficient interest to be included in this discussion. The steam supply to the pump was, as would be seen from Fig. 11, uncontrolled, and when the feed regulator on the discharge side of the pump closed, the direct acting pump would be brought to a standstill (provided the bucket rings were absolutely tight), with a condition of equilibrium between the steam load on the piston and the hydraulic load on the bucket. In this connection it was commonly understood by marine engineers that the greatest hydraulic pressure that could be attained under such conditions was equal to the steam pressure multiplied by the ratio of the steam piston area to the pump bucket area. This ratio for a Weir pump was normally of the order of 2 : 1 and so with a steam pressure of, say, 100lb. per sq. in. the average engineer would expect the greatest hydraulic pressure attainable to be about 200lb. per sq. in. The chief engineer of the installation to which he referred found that under certain conditions of very low steam demand the discharge relief valve would lift and waste feed water to the bilge, and under the impression that he could do no harm with a direct acting pump he screwed the relief valve down until the leakage was arrested. The pressure continued to build up, however, and eventually became so great as to burst the flange joints. The conclusion reached was that under such conditions—that is, with uncontrolled steam pressure on the piston and with the feed regulator closed—the pump would slowly creep, due to leakage past the bucket rings. In effect, therefore, although the bucket was creeping, the actual water displacement was, because of leakage past the rings from one side of the bucket to the other, only that due to the entry of the bucket rod into the chamber. Consequently the rod acted in effect as a hydraulic ram and under such conditions as those described he believed the pressure reached was equal to the steam pressure multiplied by the ratio $\frac{\text{steam piston area}}{\text{bucket rod section area.}}$ This being so, it would mean that with a 6in. piston and a $1\frac{1}{2}$in. bucket rod a maximum hydraulic pressure of some 1,600lb. could be exerted with 100lb. steam pressure on the piston. It was because of that, he believed, their joints were failing. He would like to know if Mr. Hillier agreed with the reasoning, and if that was why he provided a relief valve on the installation shown in Fig. 11, and not that shown in Fig. 9.

The only feed regulators of which he had any extensive experience was that of the Weir Robot type, as illustrated in Fig. 25. He had found it to be very reliable and sensitive to changing conditions in operation. He would suggest to the author, however, that an improvement would be effected from the maintenance point of view if it were redesigned so that the needle valve guide could be renewed without dismantling the chest and the pipe connections. He believed this could be done by arranging the guide to be screwed in instead of being fastened by countersunk screws as at present.

The watertube boiler was rendered considerably more reliable by the addition of an automatic feed regulator, but even so the best of feed regulators would not prevent damage to the boiler if the feed pump or the extraction pump broke down. In this connection he would suggest that the author should complete his excellent resumé of the subject by a short treatment of the means of cutting off fuel to the burners with dangerously low water levels.

Commander Atkins said he agreed with Mr. Wheadon that one certainly could get a very high pressure in a feed pump. He remembered during his first job at sea that he was surprised to find 2,000lb. per sq. in. pressure in a feed pump because the rings were absent. With regard to Mr. Hillier's Fig. 1, the American Navy did not use feed regulators, and it was an interesting fact that they had trained their stokers to do the opposite of what the water level did. During manœuvring if the water level rose owing to the lighting of a sprayer, the stokers instead of shutting off, opened out on the check valve or on the steam to the pump, and vice versa. That must take a lot of nerve, but they were trained to do so.

The Author's Reply to the Discussion.

Mr. Hillier thanked the various speakers for their kind remarks and for their contributions to the discussion, which he considered added considerably to the interest and value of the paper.

Sir George Preece: The diagram Fig. 1 was intended to illustrate the general character of changes which took place when the steam draw off was increased or reduced. The skirt of a feed regulator valve was designed in relation to the pressure drop available across the valve so that the feed flow increased substantially as a straight line function as the water level fell from H to L.

For illustration purposes, the line HK had been shown as a straight line. The ratio of steam to water varied with different types of boilers. A considerable amount of work had now been done on the theory of boiler circulation, and it was possible to estimate the volume of steam at any load for a given boiler design and so draw a diagram similar to Fig. 1 showing how the steam and water volumes varied with boiler loading for a particular boiler. For a naval boiler which had been examined, the steam volume increased rapidly at low load and less rapidly as the boiler load approached full power, the line HK being a curve convex downwards. The lines HL and HK were associated with the steady normal boiler pressure but variations in the rate of steam draw off were accompanied by variations in the boiler pressure. When the steam draw off was increased, a fall in boiler pressure occurred while the storage capacity of the boiler supplied the increase in steam demand until the fuel combustion rate was increased to restore the boiler pressure to normal at the increased rate of evaporation required. When the steam draw off was decreased, the boiler pressure rose with an accompanying storage of heat until the rate of fuel combustion was decreased to suit the reduced steaming rate and the boiler pressure fell to normal.

He agreed that the water level inside the boiler was undoubtedly higher than the water level shown in the gauge glass or the water level in the feed regulator float box because of the difference in density between the respective water columns due to temperature differences and the presence of steam bubbles in the water inside the boiler. This difference in level varied with the boiler load and the boiler design. In a particular design of boiler it has been estimated from gauge glasses arranged with their bottom connections at different heights on the boiler drum that the ratio of the density of the water column inside the boiler drum to the density of the water column in the gauge glass was of the order of 0·66 to 1·0, at full load.

The phenomenon of hunting was very complicated but it could be stated quite definitely that if the feed water were heated to the saturation temperature corresponding to the boiler pressure before entering the circulation circuit of the boiler, hunting would not occur. Various feed distribution arrangements were in use in which the feed water was sprayed into the steam space and with such arrangements hunting did not occur. A distinction should be drawn between rhythmical hunting and variations in water level which were caused by variations in the rate of steam draw off in the normal operation of a boiler. The rise and fall of the water level caused by swelling and subsidence when the rate of steam draw off was varied were caused by factors entirely beyond the control of the feed regulator and the magnitude of such variations in water level were approximately proportional to the amount by which the steam draw off was changed, and varied with different boilers. The change in water level caused by swelling or subsidence with a large change in the rate of steam draw off was frequently much greater than the total travel of the float of the feed regulator and each boiler had its own characteristic in this respect, but for the greater part of the changes in the rate of steam draw off normally experienced in boiler operation, a feed regulator should maintain the water level within the limits of about three inches.

With a single boiler on test, any manipulation of the steam outlet valve altered the rate of steam draw off with a corresponding swelling or subsidence and a corresponding change in boiler pressure.

The changes in boiler water level caused by varying the opening of the steam outlet valve were entirely beyond the control of any boiler feed regulator. He considered the steam outlet valve for shore tests should be specially designed to permit of very fine control of the opening through the valve for the outflow of steam, the valve to be as free as possible from backlash and designed so that extremely fine variations in the valve opening could be effected with ease.

He considered it had now been proved conclusively that if the feed water were heated to the saturation temperature of the steam before it entered the circulation circuit of the boiler, water level variations were reduced to a minimum and rhythmical hunting was avoided.

Engineer Rear-Admiral Dight: Admiral Dight had suggested that he should enlarge on the question of rhythmical hunting between two boilers, and he thought this would be of interest. Normal changes in the rate of steam draw off and the rate of fuel combustion, or dirt or foreign material causing stickiness of the feed regulator, usually caused somewhat irregular variations in the water level. In considering fluctuations in the water level in a boiler it was important to distinguish clearly between irregular changes in the boiler water level and the rhythmical and continuous variations in water level which occurred in a definite periodic cycle which could truly be called hunting.

In a typical case where rhythmical hunting was experienced, the main turbines were running at a substantially constant power with two boilers in operation, the rate of oil fuel burning was maintained substantially constant on each boiler but the water level in each boiler varied with an amplitude of 10 to 12 inches in the gauge glass as compared with the float travel of the feed regulator of about 5in. between no flow and maximum feed flow. The feed flow, therefore, varied between no flow and the maximum feed flow which would pass through the feed regulator lying wide open with a consequent variation in the evaporation of each boiler so that the steam drawn off from each boiler varied between a minimum and a maximum although the combined steam draw off remained substantially constant. The time taken for the water level to travel from the lowest level to the highest level and back to the lowest level was about three minutes and the cycle repeated itself continuously. The rhythmical variation in feed flow through the feed heaters caused corresponding variations in the exhaust pressure of the various auxiliaries which varied in speed accordingly. A high water level and maximum steam draw off for one boiler were associated with a low water level and minimum steam draw off for the other boiler and vice versa. The two boilers were, therefore, continuously out of step. For steady boiler operation with a constant rate of fuel combustion, the following factors must match at a given water level; the rate of feed flow, the rate of steam generation, the rate of boiler circulation and the volume of steam below the water level. The hunting was due to a continual transfer of steam draw off from one boiler to the other and because the rate of steam generation, the rate of boiler circulation and the volume of steam below the water level did not change anything like so quickly as the feed pump and feed regulator changed the rate of feed flow. There was a continuous time lag between the rate of feed flow corresponding to a water level and the rate of steam generation, the rate of boiler circulation and the steam volume below the water level which should have been associated with that rate of feed flow and that water level.

In these boilers, the feed distributing pipes inside the boilers were arranged below the water level. The feed distributing pipes were re-arranged to spray the entering feed into the steam space so that the feed was heated to boiling point before it entered the circulation circuit. This ensured that for any constant rate of fuel burning there was a constant rate of steam generation with a substantially constant volume of steam below the water level and any variations in feed flow could not affect the steam volume below the water level with the result that the hunting was completely eliminated. On board ship, boilers which hunted when two or more boilers were operating in parallel were invariably quite steady when operated solo.

He agreed with Admiral Dight that, with a single boiler on shore test, manipulation of the steam outlet valve tended to set up hunting and that it was preferable for the steam outlet valve to be set at the correct opening for the evaporation required, and the output of the boiler controlled on the oil and air supply to the combustion chamber.

Commander Atkins: The thermal type of regulator was one which was widely used in land power stations where the surrounding air temperatures were more stable than those experienced under marine conditions, particularly in Naval vessels.

It was not usual to shield the regulator or to take any precautions to maintain its surroundings at a constant temperature, but there was no doubt that any variation in the radiation from the regulator due to variations in the air flow past the regulator would alter the water level accordingly.

Mr. S. B. Jackson: The particulars given with regard to the development of magnetic couplings as a means of varying the speed of feed pumps were interesting. A certain amount of consideration was at present being given to the use of magnetic couplings with A.C. motor driven centrifugal feed pumps in land power stations. A magnetic coupling had a slip of 3 to 4 per cent. at the maximum driven speed with a corresponding loss at full load. At lower speeds the loss in power was proportional to the slip between the driving member and the driven member so that slightly less than half the possible power saving was obtained at any given feed flow.

He agreed that the development of the magnetic coupling offered a promising means of achieving variability in the speed of electrically driven feed pumps where it could suitably be applied and the savings obtained in running costs justified the additional capital charges and complication involved.

With regard to the relative prices of various electric drives he agreed that there was a considerable difference between the various drives mentioned, but in the various cases which had come within his knowledge the variation was very considerably less than that given.

Mr. A. B. S. Laidlaw: The author agreed that the boiler designer should arrange the entry of the feed water into the boiler so that it could not interfere with the ebullition.

Mr. Laidlaw raised the question as to whether, in modern practice, thermostatic regulators of any type had been employed in ships. Thermostatic regulators have been tried in foreign navies but had not come into common use. Float operated regulators responded to water level changes more definitely and quickly than thermostatic regulators and were almost exclusively used in naval ships both here and abroad. Thermostatic regulators had been fitted to watertube boilers in a considerable number of merchant ships which had been built in the United States during the present war, various kinds of thermostatic elements being used. The operating results obtained with these ships would no doubt receive close attention from marine engineers in this country.

Mr. W. Sampson: Mr. Sampson's outline of the requirements to be met by feed water regulators and feed pump governing apparatus and the factors to be considered in their operation was a very welcome amplification to the paper. He would like to associate himself very strongly with Mr. Sampson's plea that every precaution possible should be taken in the erection of the feed lines and feed system generally to ensure the exclusion of dirt and any foreign material which might interfere with the satisfactory operation of the boiler feed regulators and the feed governing devices, the feed systems to be treated with the same care in this respect as was given to turbine lubrication oil systems, interiors of turbines, etc. It was not uncommon to find pieces of wire, swarf, welding flash and nuts inside feed regulators. In some shipyards, the practice was adopted when a ship was being prepared for basin trials, of breaking the feed pipe line adjacent to the feed regulator and flushing the entire feed system through to bilge before putting the feed regulator and the boiler into operation. This was a practice strongly to be recommended.

With regard to the conditions governing the choice of a Mumford regulator, he said that it was simpler and cheaper than the other types and was quite suitable for small boilers with reciprocating feed pumps, but, in general, was not so suitable for centrifugal feed pumps and large boilers. The quantity of operating leakage water in a Mumford regulator was dependent upon the size of the regulator piston, the clearance between the piston and the check valve body, and the pressure difference across the piston. When the regulator was open the pressure difference across the piston was of the order of 50lb. per sq. inch but when the regulator was closed the pressure difference might be of the order of 250lb. per sq. inch with a corresponding increase in operating leakage water. If the lift of the feed check valve were restricted so that its closed periods were reduced to a minimum, the amount of leakage water flowing past the piston would also be a minimum and would be of the order of 1 per cent. If the closed periods of the regulator were considerable, the leak-off quantity would, of course, be increased accordingly.

Mr. H. J. Wheadon: With regard to Mr. Wheadon's comment on the omission of the relief valve on the discharge side of the feed pump in Fig. 9, the author said it had been omitted from some of the diagrams for the sake of simplicity. It was a requirement of the Classification Societies that a relief valve should be fitted on a feed system to limit the pressure to which the system could be subjected.

The phenomenon of an extremely high pressure being established in a feed pump was undoubtedly of general interest and Mr. Wheadon's description was quite correct. It arose due to a combination of unusual circumstances. For it to occur, the feed discharge system must be tight or the discharge isolating valve on the feed pump closed and tight. The pump suction valves must also be tight or the suction isolating valve closed and tight. Given these conditions, the volume of water within the pump was practically fixed. If these conditions occurred when the bucket was above the bottom of its stroke and steam were applied to the piston (this could occur with a leaky steam valve) the steam pressure on top of the piston caused the pump rod to act as a hydraulic ram and a very consider-

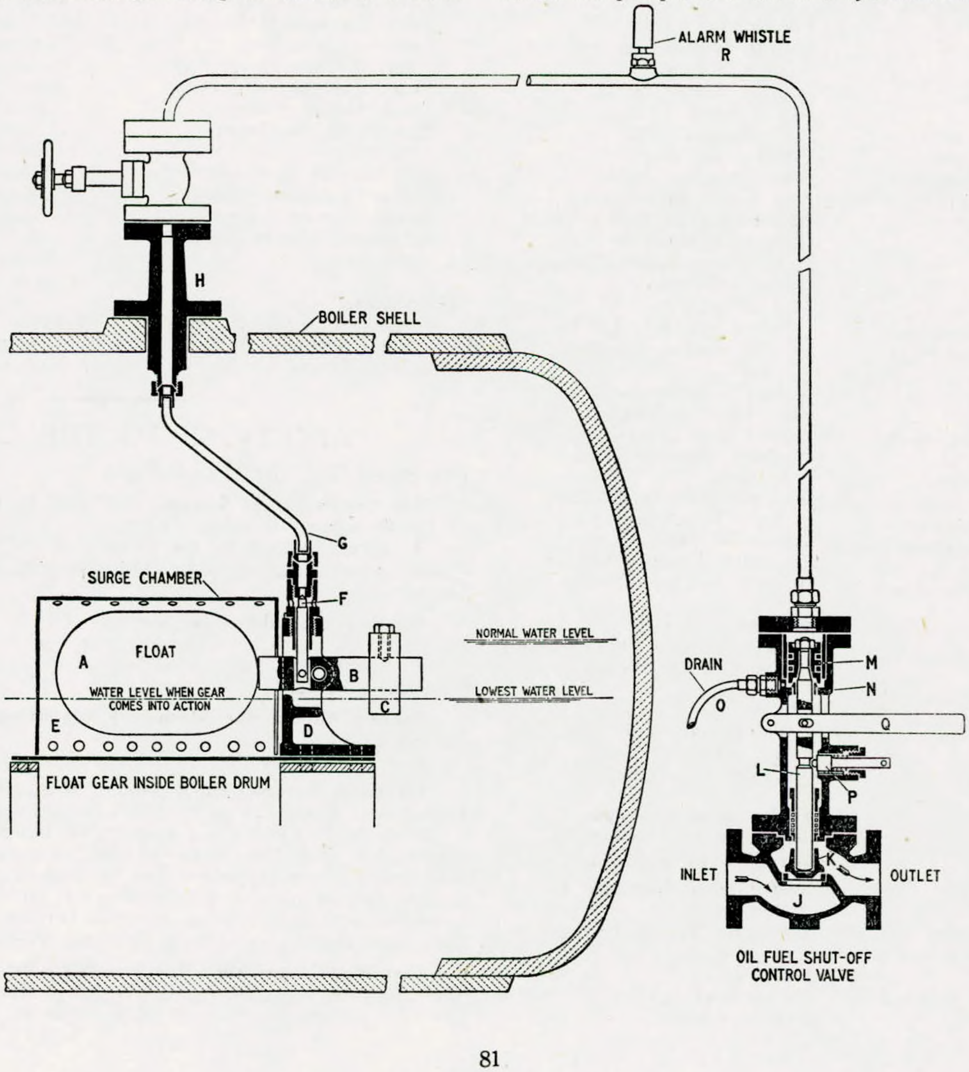

FIG. 31.
Arrangement of low water level alarm and oil fuel control gear.

able pressure could be established in the pump. The bucket could be disregarded since leakage past the bucket rings equalised the pressure above and below the bucket.

Undue pressure could be prevented by fitting a small sentinel valve on the discharge chamber of the pump but it was usual to depend on a small hole drilled in one of the suction valve decks since the suction valve was usually left open or leaked sufficiently to prevent an undue pressure being established.

The author was grateful to Mr. Wheadon for the suggested improvement to the design of the needle valve guide in a Robot regulator. As suggested by Mr. Wheadon he had prepared a diagram, Fig. 31, showing means for cutting off the oil fuel to the burners in the event of a dangerously low water level occurring in a boiler.

Referring to Fig. 31, an automatic shut-off valve J is fitted in the oil fuel discharge line between the oil fuel pump and the burners. Float A with lever B and balance weight C are arranged inside the boiler drum and are supported on a bracket D at a predetermined height. A surge chamber E prevents the float from being affected by surges arising from pitching and rolling. If the water level falls to a predetermined level below the normal working level, the float A opens the needle valve F and admits steam through the pipe G to the top of the piston M so that the valve K is closed, thereby cutting off the oil fuel supply to the burners. At the same time, an alarm whistle R is sounded by the steam from the pipe G. A spring operated trigger P engages in a groove in the valve spindle L when the valve K is closed and maintains the valve closed. To reset the valve K, the trigger is withdrawn and spindle L and valve K are lifted by means of the lever Q into the position for further automatic action. The construction is simple and the action is positive while the boiler is under steam. In the event of the water falling to the predetermined low level the oil fuel supply to the boiler is cut off and an alarm whistle is sounded. When the oil fuel has been cut off it is impossible to restart the oil supply to the burners until the water level in the boiler has been restored to a safe level. The whistle continues to give audible warning so long as the water is below the danger level.

Mr. A. F. C. Timpson, M.B.E. (Member) said that the paper was extremely sound and thorough, and it had provoked an excellent discussion. Mr. Hillier had replied to the various points raised in the discussion in a masterly manner. He had great pleasure in proposing a vote of thanks to the author for his excellent paper.

Mr. H. J. Wheadon seconded the vote of thanks, which was accorded with acclamation.

MEMBERSHIP ELECTIONS.

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David Ridgway Leak, Acting Lieut.(E.), R.N.

Transfer from Associate Member to Member.

Gerald Stanley Mead.

Transfer from Associate to Member.

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Robert McDonald.
Robert Newall.

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John Sharp.
Walter Scott Steel.
Jacob Bernard van Wijk.

Transfer from Student to Associate.

Joseph Laudells Black.

ADDITIONS TO THE LIBRARY.

Presented by the Publishers.

The British Patent System. Published by the Chartered Institute of Patent Agents, London, 10 pp.

A report adopted by the Council of the Chartered Institute of Patent Agents, dealing with certain aspects of the British Patent System.

Bonding of Rubber to Metals. By S. Buchan, M.A., B.Sc., Ph.D., F.I.C., A.I.R.I. Andre Rubber Co., Surrey, 14 pp., 5 illus.

A reprint of a paper read before the Institution of the Rubber Industry, 1941-42.

Bonded Rubber for Machinery Mountings. Pamphlet. Andre Rubber Co., Ltd., Surrey.

Reprinted from "The Oil Engine", February, 1942.

Standards Review, Vol. I, No. 1, May, 1944. British Standards Institution. Price to non-members 7s. per annum.

"Standards Review" is now to be issued in response to many suggestions that the work of the Institution be given greater publicity. It will, however, not be limited to that work, but will include information as to the progress of the movement in other countries, and as to the co-operation between the British Standards Institution and other National Standard Bodies.

Post War Cargo Ship. The Search for Efficiency. Stanley Hinde. The Shipping World, London. 12 pp., 2 illus.

This publication consists of a series of articles reprinted from

"The Shipping World" in which the author discusses various aspects of the post-war cargo ship and suggests, from a practical angle, how it might be made into a more efficient vehicle than before. He starts from the premise that most war built ships will be unsuitable for restoring Britain's maritime supremacy, which can only be done with vessels of the highest practical efficiency. In seeking that end he poses a number of challenging questions for the shipbuilder, naval architect, marine engineer, and the makers of all kinds of auxiliary equipment.

Engineering as a Career. Pegson, Ltd., Coalville, Leicestershire.

The Pegson Apprentice Training Scheme is described in this brochure. In common with most other undertakings, Pegson, Ltd., at their works at Coalville, offer facilities for technical education to young men who intend to adopt engineering as a career.

Electric Power Stations. By T. H. Carr. 2nd edition. Vol. I. Chapman and Hall, Ltd., London, 1944, 507 pp., 249 illus., 32s. 6d. net.

Upon power station design, construction, and operation, only very few may be considered expert in all the extensive field of engineering involved. Power plants are usually out of date when commissioned, owing to the rapid advance in the science and practice of power generation, and if there are omissions the author may be pardoned because of this state of flux.

The volume presents a comprehensive review of the subject in its major aspects, and its treatment of the various branches of a complete power system is concisely interconnected. The work fulfils a want felt by all those interested in electric power stations. Copies should be in the libraries of all technical colleges, as the book forms a satisfactory basis of study, being essentially the work of an authority on this subject.

The reviewer's experience in electric power plants justifies his recommending this book as giving excellent value for the price. Naturally there are omissions, but having regard to the wide range covered the author has executed his task very well.

Mechanical Testing of Metallic Materials. By R. A. Beaumont. Sir Isaac Pitman & Sons, Ltd. Aeronautical Engineering Series. Second edition, 1944, 141 pp., 94 illus., 8s. 6d. net.

This little book does not set out to describe all the mechanical tests applied to metallic materials, but concentrates mainly upon tests for their proof strength. Thus the chapters are arranged to describe tensile tests, proof stress determination, types of tensile testing machines and instruments, hardness tests, impact tests, the calibration of testing machines, tests on sheet strip, wire and tube, and also radiological testing and crack detection. In view of the particular references which have been made to proof testing, the addition of this latter chapter appears to be a trifle irrelevant.

This book is, however, written in a nice style which is easily followed and gives a very good outline of the several methods of testing a metallic material for its proof stress, hardness and impact strength. The subject is dealt with entirely from the practical aspects of operating the testing machines and the author does not touch upon the subject of materials and their properties.

For this reason this book is suitable as a text-book for those operatives requiring a general knowledge of the methods of mechanically testing metallic materials. No detailed exposition of the merits and demerits of individual machines is given, which accordingly detracts from its value as a reference book for actual engineers. There is no doubt, however, that this work is very valuable as a text book for students.

The Analysis of Engineering Structures. By A. S. S. Pippard, M.B.E., D.Sc. and J. F. Baker, M.A., D.Sc. London, Edward Arnold & Co. 8in. by 5½in. by 1in. 627 pp., price 30s.

Whilst written particularly for civil and structural engineering students, this book can be read with the utmost profit by students of shipbuilding. The presentation differs from that of the usual standard text-books on the theory of structures. These are usually content with the exposition of the classic theories of the subject and cause the casual student to doubt their possibility of practical application. The present book, however, removes all doubt on this score, since, although not shirking any of the most modern theory, it gives not only the usual numerical or practical examples, but also many experimental tests of structural theory. These tests have been largely taken from Dr. Baker's work on the Steel Structures Research Committee, and they show how closely the modern theory of framed structures, based upon the early work of Clerk Maxwell and Castigliano and developed by Southwell, Hardy-Cross and others, interpret and predict the stress distribution in quite complicated frameworks. Some of these, for example, of the skyscraper type, closely resemble the framework of a multi-deck ship and would have direct application thereto.

The chapter dealing with experimental stress analysis is particularly valuable in this connection, and deals with the technique of the elastically or sufficiently elastically similar model. Various contributions to this technique have been made familiar to naval architects by Professor Coker in his work on photo-elasticity, but it is clearly not appreciated how widely useful this subject has become. It has also provided the introduction to the more practical subject of elasto-plastic behaviour, to which the authors devote their final chapter. This subject effects the complete marriage of theory and experience, and whereas the practical constructor could rightly object to the limitations of pure elastic theory, yet none could disparage the splendid progress made in elasto-plastic theory since this does respect the facts of structural behaviour.

The study of joints and connections is particularly illuminating and can be read with profit by all naval architectural students.

The book is well produced and well illustrated. The first edition was published in 1936; this, the second edition, is extended in practical scope and should merit acquisition and study by a wide circle of readers.

Purchased.

Diesel Engine Catalog. Vol. 8. Edited by Rex W. Wadman. Diesel Engines, Inc., New York, 408 pp., copiously illustrated, $5.

Details of practically all the types of oil engine on the American market are to be found in this one volume—the products of exactly 50 makers are included. This presentation is aided by the large-sized page (13½in. by 10in.) which permits of excellent large scale illustrations (both sectional and photographic) of each class of Diesel motor to be given. This book, which is highly informative and useful, covers industrial, marine, transportation and aviation types.

Fuel Economy in the Operation of Boiler Feed Pumps. Messrs. G. & J. Weir, Ltd., Cathcart, Glasgow, S.4, have prepared as a series of lantern slides, the illustrations in their booklet "Fuel Economy in the Operation of Boiler Feed Pumps".* Anyone arranging a talk on Fuel Economy can obtain these slides on loan by applying to the Publicity Department, Messrs. G. & J. Weir, Ltd., at the above address.

*See notice in TRANSACTIONS, Vol. LV, No. 12 (January 1944 issue), p. 211.

British Coal. By H. L. Pirie, M.I.Mech.E., M.I.Mar.E. Sir Isaac Pitman & Sons, Ltd., London, 168 pp., 12 illus., 15s. net.

This book describes in the most informative way the nature and characteristics of British coals according to the districts in which they are found, the sale, distribution and purchase, delivery and storage, utilization of raw coal, service, etc.

Short descriptions of the work of the Fuel Efficiency Committee, the White Paper on Coal, and some other war-time measures, are included. For the benefit of those who wish to pursue the subject further, a list of publications that may profitably be consulted is given at the end of each chapter.

England's Sea-Officers. By Michael Lewis. George Allen & Unwin, Ltd., 1939, 307 pp., 10 illus., 10s. 6d. net.

The officers of the "Silent Service", as a body, have hitherto had no biographer, but they have found one now in the Professor of History at Greenwich, the University of the Navy.

England's Sea Officers is the story of the naval profession, its origins, and its growth from earliest times to the present day. Its heroes are not individuals like Nelson and Blake, but the Admiral, the Captain and the Lieutenant, the Engineer, the Surgeon, the Chaplain, the Paymaster, etc. It reveals the whole English process of creation in all its fascinating illogicality.

We see the machine being built up cog by cog, as the ranks and branches—even the Admiralty itself—develop through the centuries.

Abstracts of the Technical Press

Dufty Aero-dynamic Governor for Marine Engines.

The recently introduced Dufty aero-dynamic anti-race governor for marine engines differs from other governors in the principle feature of its operation. The latter is based on the fact that a fan rotating in a closed chamber opposite a vane or impulse wheel will transmit to the wheel a torque whose magnitude is proportional to: (*a*) the square of the speed of the impeller, and (*b*) the pressure (and hence the density) of the gas (*i.e.*, air) in the chamber. An impeller only 7in. in diameter running at a speed of 3,000 r.p.m. will transmit to the impulse wheel a torque of 410 cm. gr. when the pressure in the chamber is equal to 18in. of water gauge. The general arrangement of the aero-dynamic governor is shown diagrammatically in Fig. 1, whilst Fig. 2 is a front view of the impulse-wheel cam control. The

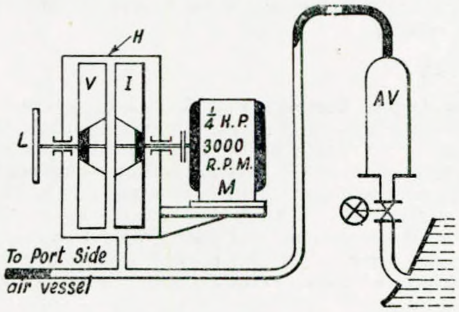


FIG. 1.

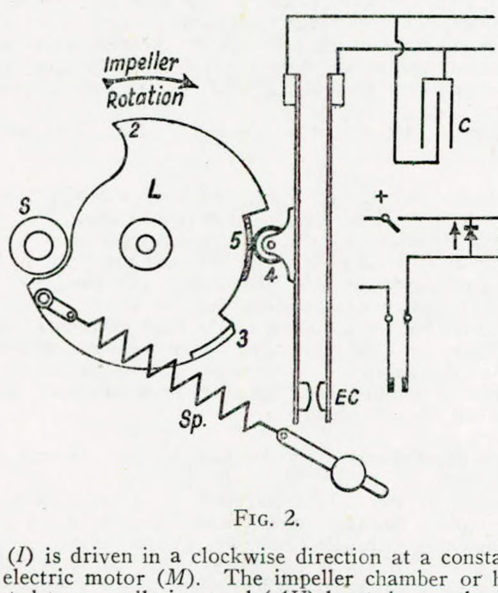


FIG. 2.

impeller (*I*) is driven in a clockwise direction at a constant speed by a small electric motor (*M*). The impeller chamber or housing (*H*) is connected to a small air vessel (*AV*) located near the stern of the ship and in communication with the sea water. When the sea-cock to the air vessel (*AV*) is opened, the air in the latter as well as that in the pipe line and impeller chamber (*H*) is compressed by the head of water under which the ship's propeller is rotating and a corresponding clockwise torque is transmitted from the impeller (*I*) to the vane or impulse wheel (*V*), tending to turn the cam plate (*L*) attached to the impulse-wheel spindle, in a clockwise direction. This rotation is resisted by a spring (*Sp*) which is adjusted so that the lower left-hand contact lip of the cam plate (*L*) is forced up against a buffer stop (*S*) so long as the ship's propeller is fully submerged. If the stern rises owing to pitching of the ship, the air pressure in the air vessel (*AV*) and impeller chamber (*H*) is reduced, in consequence of which the torque transmitted to the impulse wheel (*V*) is decreased and the spring (*Sp*) is then able to turn the cam plate (*L*) anti-clockwise, causing the lower right-hand lip (*3*) to engage an insulated roller (*4*) and thus close an electrical contact (*EC*). By suitable telemotor control, the closing of this contact is arranged to operate on the steam valve of the engine, or on the fuel-injection pumps in the case of a Diesel engine, thus cutting off the power when there is any danger of the engine racing. Adjustment of the spring (*Sp*) provides means of timing the power cut-off point, the adjustment being capable of variation to suit the rate of pitching of the ship. A condenser is connected across the contacts (*EC*) to prevent sparking and a lubricating pad (*5*) is provided for the insulated roller (*4*). The output of the motor (*M*) is under ¼ h.p. at a constant speed of about 3,000 r.p.m. An alternative method of protecting the contacts (*EC*) is to place a dry-plate rectifier as a shunt across the magnet operating the steam governor or the injection pumps. This rectifier, by absorbing the inductive energy stored in the magnet winding, effectively quenches the arc when the contacts are opened. A no-volt relay in series with the driving motor opens the electric circuit from the contacts (*EC*) and thus puts the governor out of action in the event of a failure in the ship's current supply. The impeller spindle of the governor runs in ball bearings with labyrinth sealing glands, the construction of which is shown in Fig. 3. The impulse-wheel boss (*6*) has a series of concentric labyrinth grooves cut on its end face and these bear against a carbon ring (*7*) which is countersunk in the front cover plate (*8*). The sharp labyrinth grooves cut into the face of the carbon ring until the face of the boss (*6*) closes up to the face of the bush (*B*), thus giving a leak-proof seal which is allowed to cut its own clearance in the carbon ring before the cam plate (*L*) is fitted on the outer end of the spindle. The governor is claimed to be instantaneous in action, free from hunting, and capable of adjustment. In the case of a multiple-cylinder steam engine, any governor must have an anticipating characteristic if it is to be efficient, as the steam in the receivers, cylinders and pipes leading to the condenser (already beyond the control of the governor valve) is sufficient to cause serious damage.—*"The Marine Engineer", Vol. 67, No. 801, April, 1944, pp. 105-106.*

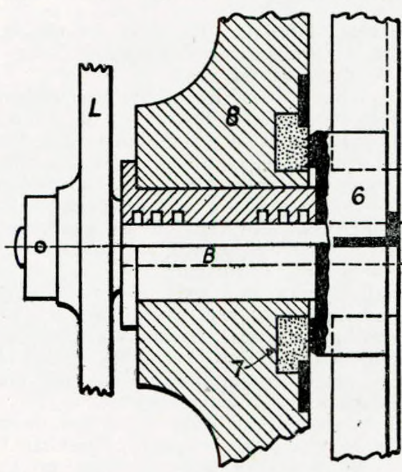
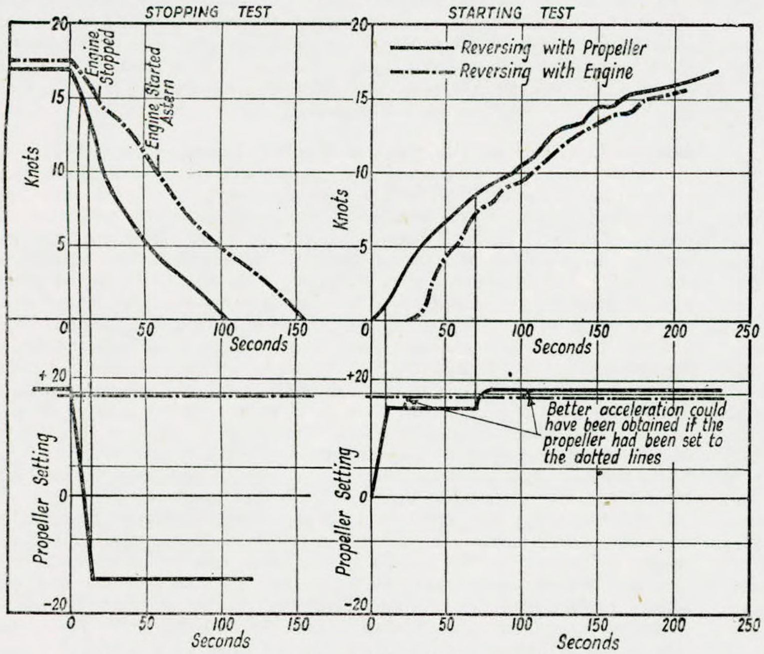


FIG. 3.

Sea Trials of First Ocean-going Motorship with Variable-pitch Propellers.

The 7,400-ton twin-screw passenger and cargo liner "Suecia", recently built by the Götaverken for the Johnson Line, has successfully completed her acceptance trials off Gothenburg. The vessel is claimed to be the first ocean-going motorship to be fitted with Kamewa adjustable-pitch propellers, these being directly driven by two sets of 6-cylr. two-stroke Diesel engines of the Götaverken's own design. The total output of the propelling machinery is 7,000 b.h.p. at 125 r.p.m., which is estimated to give the ship a speed of 17 knots in a fully-loaded condition. Although the propellers are normally adjusted and reversed from the bridge by means of telemotor gear, the main engines are fitted with the standard form of reversing mechanism, but a locking device on the latter prevents the engines from being run in the astern direction when manœuvring with the Kamewa propellers. The engine governors have somewhat larger pendulums than usual, and limit the variation in speed to

5 per cent.from full load to no load. The governors can be adjusted from the bridge to enable the speed of the engines to be varied as necessary between full speed and half speed. The engines can also be stopped from the bridge but not started, as this would involve too great a risk for the E.R. staff. For ordinary manœuvring in port, estuaries, etc., the main engines are run at a constant speed in the ahead direction, all manœuvring being effected by the control levers of the v.p. propellers. When the engines are cold, however, they should not be loaded up to normal full output and special stops can be put into operation by the E.R. staff to limit the loading of the engines until the machinery is thoroughly warmed through. Indicator lamps are fitted on the bridge to show when the P. and S. engines are carrying full load, or at least a load corresponding to good economy. The lamps normally come into operation when the mean pressure in the engine cylinders attains a value of about 90lb./in.^{2}. When full speed is required the speed of the engines is first set at the 125 r.p.m. adjustment, and the propeller pitch is then increased until the lamps light up. To facilitate accurate adjustments of the blade positions, suitable tables are displayed on the bridge, showing the blade adjustment for any given displacement and ship-speed. The fuel consumption per mile and per hour is also given. The same procedure is followed for lower speeds, the propeller pitch being increased until the lamps are burning, while for still lower speeds the operating levers are put in the maximum position and the speed of the engines reduced. When the lowest possible number of revolutions on this setting has been attained and it is desired to have a still lower ship speed, the propeller pitch is reduced. Circumstances made it necessary for the "Suecia" to carry out her sea trials in light condition and in unfavourable weather. Stopping and starting tests were made first by adjusting the propeller blades and also by reversing the engines in the ordinary way. The results are shown by the accompanying curves. The propeller-blade adjustment necessary for these tests was made in 15 seconds, and the ship was then stopped in 1 minute 46 seconds. The E.R. staff were not familiar with the engines and therefore took rather longer to reverse the engines than had been anticipated. The tests were subsequently repeated, and it then took 50 seconds to reverse the engines and exactly two minutes to stop the ship (as against 2 minutes 35 seconds on the first occasion). The graphs indicate that manœuvring by means of the v.p. propellers is quicker than with fixed-pitch propellers and engine reversing, although the difference is not so great as might be expected. Had the ship been fully loaded, however, the Kamewa propellers would undoubtedly have shown up better in this respect. Apart from this, the trials demonstrated that manœuvring with these pro-

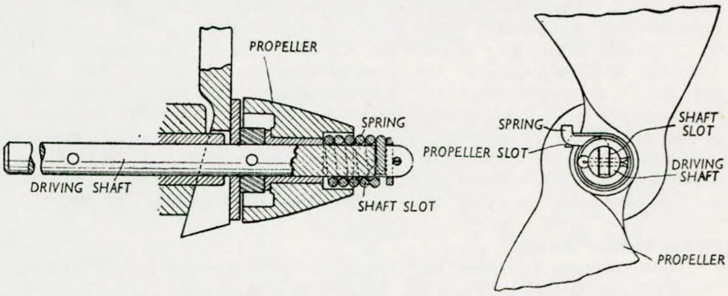


Ship stopping and starting trials of the m.s. "Suecia".

pellers was smooth and convenient. Progressive speed trials were subsequently run by the "Suecia" at the same displacement, and a maximum power output of 9,450 i.h.p. from both engines was obtained with a speed of 18·52 knots, about 7,370 i.h.p. being required to maintain 17·52 knots and just under 6,250 i.h.p. for 16·52 knots. The normal designed output of the two engines (7,000 b.h.p.) for 17 knots with the ship fully loaded is stated to be 8,800 i.h.p. The trial speeds attained by the "Suecia" correspond very closely with the results obtained by a sister ship with ordinary propellers and are in agreement with the predictions of the experimental tank staff at Gothenburg. At the conclusion of the acceptance trials, the captain of the "Suecia" berthed the ship alongside the quay at Gothenburg by means of the propellers only with a minimum of trouble.—*"The Marine Engineer", Vol. 67, No. 801, April, 1944, pp. 89-97.*

Spring Drive for Propellers of Outboard Motors.

The accompanying illustrations show a patented device which is claimed to provide a cushioned drive for the propeller of an outboard



Spring drive for outboard engine propeller.

motor, an advantage of the arrangement being that no shear pin is needed. If the propeller encounters an obstruction the spring binds tightly round the shaft. The strength of the spring is chosen so that it will fracture at a certain load, and it thus fulfils the functions of the normal pin, which breaks in order to prevent damage to the propeller or transmission. The final drive from the motor is transmitted through a close-coiled helical left-wound spring, one end of which forms a hook that fits into a slot in the propeller, the other end of the spring being turned over and taken through a slot cut in the end of the propeller shaft.—*"The Motor Boat", Vol. LXXVII, No. 1,918, April, 1944, p. 107.*

Calculation of Marine Propeller Performance Characteristics.

The paper is concerned mainly with the development of a complete strip-theory method of calculation for marine propellers which includes the correction necessary to take account of the differences between wide-bladed marine propellers and normal airscrew types, in such a way that satisfactory agreement is obtained between theory and practice. The usual momentum and thrust equations are modified to take account of slip-stream contraction consistent with the inflow velocities. A series of standard curves are given which enable the lift and drag characteristics to be calculated for aerofoil and round-back sections. The effect of blade interference is presented by the author in a manner intended to be convenient for the purpose of calculation. —*Paper by L. C. Burrill, B.Sc., read at a meeting of the N.-E. Coast Institution of Engineers and Shipbuilders on the 3rd March, 1944.*

Synthetic Resin Bearings for Engines.

In Germany, a considerable amount of research work has been carried out with synthetic resin bearing surfaces for main bearings and crankhead bearings. It has been shown that the load-carrying capacity exceeds that of a tin-base bearing, when the bearing material is anchored to the crank-pin journal, the bearing housing then being a plain bush which can be of soft carbon steel. This type of "wrapped" bearing is made up of a number of thin strips of fabric impregnated with synthetic resin which are wrapped around the journal to a maximum thickness of 0·08in. and then pressed and hardened.—*"The Oil Engine", Vol. XI, No. 131, March, 1944, p. 295.*

Fast-running Engines' Difficulty.

It is probable that the minimum limits for the weight and space taken up by marine engines and boilers have not yet been reached, and there are several indications of the general lines upon which further progress in this direction may be anticipated. At the present time there is little to choose between watertube boiler and geared turbine installations and d.a. 2-stroke Diesel engines as regards weight, but it has been suggested that further reductions in machinery weight might be effected by the adoption of high-speed multiple oil engines, geared mechanically or by means of electric couplings, to the propeller shaft; it has even been said that the fast marine Diesel engine of the future will be constructed on the lines of the present-day aeroplane engine, modified to run on Diesel oil instead of petrol, and with a reduced speed. Unfortunately, engines of this type have a very limited life, and are rarely capable of operating for more than 1,200-1,500 hours. This contrasts unfavourably with the life of a slow-speed Diesel engine, but it must be recognised that there is still room for improvement in certain features of this type of machinery with a view to prolonging the life of the cylinder liners in particular. The wear of the latter is, of course, largely dependent on the amount of manœuvring which has to be performed, the type of oil used and the treatment of the surfaces of the cylinder liners. It has often been found that this wear is actually greater in a slow-speed Diesel engine than in a high-speed unit, lubrication and other factors being equal. Even if the reliability of the small high-speed oil engine was equal to that of an orthodox Diesel unit, a case for the adoption of the former could scarcely be established at the present time owing to the fact that the fuel consumption of the multiple-engine machinery would be higher than that of the other. However, it is probable that improvements are likely to be made in this respect also, and some marine engineers believe that the future development of propelling machinery will be in the direction of the small, high-speed oil engine. —*"Fairplay", Vol. CLXII, No. 3,177, 30th March, 1944, p. 486.*

Single Helical Gears.

The pinions and gear-wheels of the reduction gears in marine practice are usually cut with double helical teeth. Helical teeth possess the advantage that there is always a number of teeth meshing at a given time, so that the load is gradually applied to the full width of the tooth and the induced stress remains moderate. On the other hand, since the teeth are inclined to the centre line of the shaft, the pressure on the teeth develops an end thrust in the direction of the shaft. This is, however, overcome by the use of double helical teeth in which two end thrusts equal in magnitude but opposite in direction are developed, these necessarily balancing one another. This form of tooth is more difficult to machine, particularly where the two helices meet, unless, as is frequently done in marine gears, a considerable space is left between them. To overcome this difficulty, a well-known Continental firm who specialise in the construction of marine equipment, have adopted a design of single helical gearing instead of the so-called herring-bone wheel, for geared installations of moderate power. The end thrust is taken up by a Michell type thrust collar on the end of the pinion shaft in the vicinity of the teeth themselves, thus preventing any transmission of thrust to the shafts and making it possible to utilise couplings with axial clearance. It is claimed that single helical gears of this design are smaller and lighter than double helical gears, and that they run more quietly because they can be more accurately machined.—*"Shipbuilding and Shipping Record", Vol. LXIII, No. 13, 30th March, 1944, p. 291.*

Geared Turbine Propulsion.

Speaking at the annual meeting of Cammell Laird & Co., Ltd., at Birkenhead, Sir Robert Johnson, chairman and managing director, suggested that geared turbines would be a serious competitor with Diesel engines for ship propulsion after the war. One of the difficulties with which shipbuilders and shipowners in this country were faced, he declared, was to know what was likely to happen in connection with the vast number of war-built American and Canadian merchant vessels, and what effect they would have on the construction of new ships in Great Britain. If they were to be used in competition with British ships, then the great activity in shipbuilding that had been forecast after the war might be jeopardised. He did not think some of these ships would cause anxiety, but the more modern vessels with geared turbines and watertube boilers would be a different proposition, as this form of propulsion was likely to be highly developed immediately after the war. He was fairly certain that it would become a serious competitor to Diesel propulsion. It also had the advantage of being practically free from noise and vibration, so that it was likely to be in great demand for passenger and cargo liners. The speaker concluded by referring to the facilities possessed by Cammell Laird for producing ships with this form of propulsion, as well as for the manufacture of the watertube boilers, which would almost certainly be required for the generation of steam at high pressure and temperatures to suit the new conditions.—*"The Syren", Vol. CXCL, No. 2,485, 12th April, 1944, pp. 47-48.*

Fabricated Steel Plunger Barrels for Marine Boiler-feed Pumps.

Among the prize-winning papers submitted to the Jas. F. Lincoln Arc Welding Foundation, was one by Mr. E. F. Wright, of the reciprocating pump department of the Worthington Pump and Machinery Corporation, describing the construction of all-welded steel cylinders or plunger barrels for the boiler-feed pumps in the C-3 passenger and cargo vessels of the U.S. Maritime Commission. The feed pumps of these ships are of the vertical single-acting triplex type, and deliver feed water at a temperature of 240° F. against a rated pressure of 575lb./in.^{2}. The three plunger barrels are arranged in a single block or housing, the lower portion of which forms a sump for the oiling system of the entire unit. The plungers are 4½in. in diameter and have a stroke of 8in. They are driven by a variable-speed electric motor through double-reduction gearing which is enclosed in a housing on one end of, and extending above the pump frame. The crankcase and crosshead guides are in this frame directly above the pump barrels or cylinders, all the valves and pipe connections being arranged in a chest bolted to the front of the cylinders. The latter therefore support the entire pump and motor. As originally designed, the cylinders and valve chest were high-duty iron castings, but as the result of experience gained early in the present war it was found necessary to employ a material more resistant to shock than cast iron for all machinery parts subjected to relatively high internal pressure, and it was accordingly decided to use steel castings for the cylinder block and valve chest. The Worthington Pump Corporation supplied a considerable number of feed pumps with cast steel cylinder blocks and valve chests for the Maritime Commission's ships before considering the possibility of replacing these components with fabricated units. An experimental cylinder block of fabricated construction was first produced about three years ago and underwent a prolonged series of rigorous tests under conditions approximating a severe overload. The results offered fairly conclusive evidence that the fabricated cylinder could be used continuously at the rated pressure of the pump, since no leaks or signs of structural weakness developed during the tests. Early in March, 1942, the test cylinder block was utilised to replace a casting that had failed after 18 months' service and a second unit, to be held in stock, was ordered; before fabrication was actually begun, however, this was sold for use on board a ship employed in the war area. A third fabricated cylinder block has since been put in hand for emergency use. The value of the fabricated design for emergency conditions became very apparent on this second order, as the steel foundries were not in a position to deliver castings for several months, whereas only a fortnight was required for the fabrication and machining of a complete cylinder block ready for service. The actual cost of the second cylinder block, which was made in April, 1942, was very slightly less than the current cost of a steel casting as received from the foundry, but it is anticipated that the production costs of fabricated units made on a commercial basis would be substantially lower than those of cast steel cylinder blocks. The finished weight is approximately the same, since both designs utilise steel of about the same strength. During the past few months fabricated steel cylinder blocks have been accepted as a standard replacement unit for defective cylinder castings of feed pumps already in service.—*"Canadian Shipping and Marine Engineering News", Vol. 15, No. 8, March, 1944, pp. 35-37 and 43.*

Marine Activities of the General Electric Company in 1943.

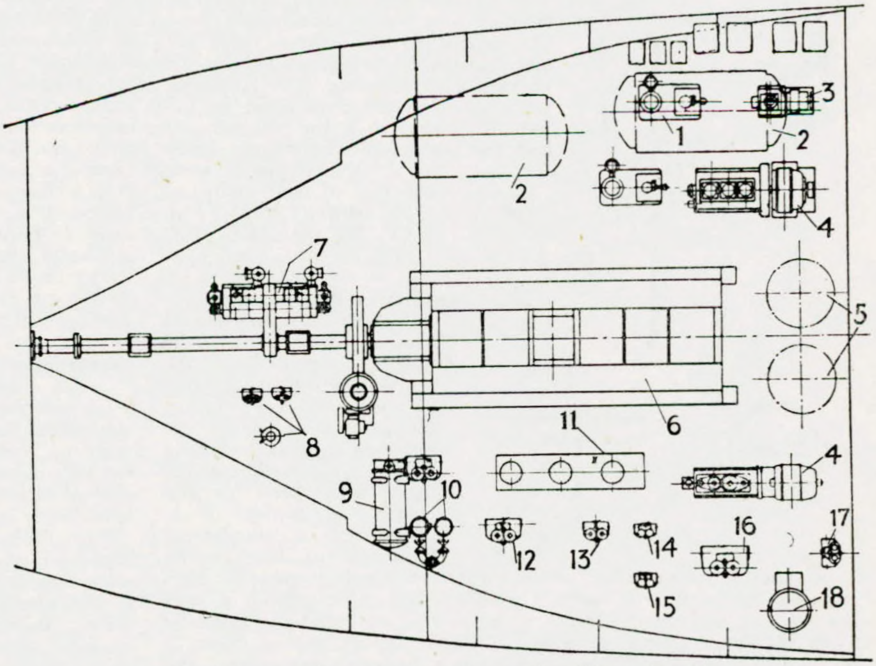
The total horse-power of the marine turbines delivered by the (American) General Electric Co. last year was 11,170,000, including turbo-electric propulsion equipment for tankers which amounted to nearly 900,000 h.p., and approximately 1,000,000 h.p. for geared-turbine installations for cargo vessels. About half the propelling machinery for merchant ships completed by the firm during 1943, was shipped to the Pacific Coast. Reports on the performance of marine installations supplied by the company during the previous year indicate that operating results have proved highly satisfactory under the most arduous conditions. For example, a vessel of the C-3 class steamed nearly 70,000 sea miles in 1943 with her geared turbines developing 10 per cent. more than their normal power during 80 per cent. of her time at sea, whilst two similar ships made a 17,000-mile round voyage together in record time, leaving and returning to their home port in sight of each other. They were unescorted throughout the voyage and steamed at high speed all the time, with their geared-turbine machinery subjected to a 10 per cent. overload. The output of heavy components for turbine installations from the four large works belonging to the company has been greatly increased. For example, whereas two years were required to complete the first 100 sets of turbines for cruisers and destroyers in one of the new turbine shops equipped for the repetitive manufacture of such plant, the second 100 sets were produced and shipped in seven months, although the original production schedule at these works called for only 48 sets per annum. Despite this increased rate of production, manufacturing costs have been reduced and a complete set of turbines and gears for a destroyer escort was being turned out in 1943 at two-thirds of the 1939 cost. More than a third of all the U.S. destroyer escort vessels have been or are being equipped with G-E turbo-electric propelling machinery and auxiliaries. The design of this machinery permits unit installation, the main condenser forming the foundation for the complete unit. The turbines and generators are mounted directly on the condensers, which, in their turn, are supported by the hull structure. The main propulsion motors are likewise designed for direct mounting on supports forming part of the ship's structure. Two-motor, one-generator operation is thus possible from either control station, permitting the use of twin screws from one generator when cruising at economical speed or in the event of damage to one engine or boiler room. The first of the high-speed turbo-electric tankers built by the Marinship Corporation are now in service. Their boilers operate at a steam pressure of 590lb./in.^{2} with a superheat temperature of 815° F., and their G-E. propulsion motors, rated at 10,000 h.p., are among the largest ever built for marine service. They are installed on board as a complete unit by means of two 50-ton cranes and a spreader, instead of being put into place in component parts; the conventional method is to begin by securing the stator frame in the hull and then threading the rotor into it, a difficult procedure involving considerably more time. Among the improvements in the design of G-E. auxiliary machinery effected in 1943, is one relating to an electric cargo winch of smaller size enclosed in a watertight steel casing which also contains the control cabinet. Previously, the control gear, consisting of panels and resistors, was located in an adjacent deck-house. The horse-power of the Diesel-electric propelling machinery delivered by the company during 1943 was 62 per cent. more than that of their 1942 output. Apart from a large number of U.S. submarines, many auxiliary vessels of the U.S. Navy are now equipped with Diesel-electric machinery. These ships include minesweepers, net tenders, salvage craft, petrol tankers and various tugs. A pair of induction-synchronous electric couplings was built last year to permit two Diesel engines to transmit their output to the pinions of a set of gears, the main gear-wheel being coupled direct to the propeller shaft. Thus, the couplings allow two or more prime movers to drive a single load. The electric couplings have neither bearings nor shafts, but are interposed between the coupling flanges of the engines and gear pinions. The engines are connected to the load by exciting the fields of the electric couplings, operation ahead or astern being possible with either or both engines simply by starting the latter in the desired direction and energizing the corresponding couplings.—*J. W. Belanger, "Marine Engineering and Shipping Review", Vol. XLIX, No. 1, January, 1944, pp. 168-170.*

Standardising Part Names.

Differences in the nomenclature employed by individual designers and draughtsmen for similar machinery parts frequently prove puzzling to engineers, and there appears to be a clear case for some authority to decide on the standardising of part names. Until quite recently, for example, the reversing link in Walschaert valve gear might be described as such or called the radius link or the quadrant; the valve rod was often referred to as the radius rod; the return crank was frequently called the eccentric crank, and so on. So confusing did the position become that a ruling was ultimately made by those in authority, and one name only is now recognised for any particular detail. Another instance concerns references in certain documents to what was termed a "cod piece", which the writer was unable to identify without examining the drawing. The part in question proved to be one of the small die plates used at the extremities of a clutch fork lever, and an appeal to four different draughtsmen produced a different name in each case! The problem is, of course, easier to criticise than to solve, but many components used in different branches of engineering —clutches, gearboxes, valves, etc.—consist of parts which function alike and differ only in detail design. It is therefore suggested that the British Institute of Standards, or some similar body, should specify the names to be used in such cases, in exactly the same manner as has already been done in the electrical industry.—*G. W. McArd, "Mechanical World", Vol. 115, No. 2,988, 7th April, 1944, p. 389.*

The Portuguese Motor Tanker "Sam Bras".

A recent issue of the Portuguese *Revista da Ordem Dos Engenheiros* gives a detailed description of the single-screw motor tanker "Sam Bras", 4,915 tons d.w., built in the new State Arsenal at Alfeite, for service as a fleet auxiliary of the Portuguese Navy. The yard in question was constructed and equipped under the German reparations scheme of the last war, and although not in a position to undertake extensive marine engineering work, the extent of the shipbuilding facilities it provides is demonstrated by the fact that the "Sam Bras" went on her maiden voyage on 14th November, 1942, only nine months after her keel was laid. The main and auxiliary machinery was supplied from Denmark by Burmeister & Wain, who also gave technical assistance to the Portuguese shipbuilders at Alfeite. The tanker has a length of about 356ft. b.p., a moulded beam of 51ft., a moulded depth of just over 23½ft. and a mean draught of 18ft. on a full-load displacement of 7,375 tons. The 14 cargo tanks have a maximum capacity of 3,850 tons of oil, in addition to which there are tanks for 40 tons of aviation spirit, 264 tons of fresh water, and 60 tons of lubricating oil. The fuel tanks for the ship's own use hold 500 tons of Diesel oil, which is sufficient for 12,000 miles at 12 knots in a fully-loaded condition. The cargo-pumping installation consists of two 250-ton steam-driven pumps located in a midships pump room and connected to a ring system of 8-in. C.I. piping with 6-in. branches to the various tanks. The piping system on deck likewise consists of 8-in. pipes with all the usual connections and fittings. In addition to these piping systems for handling the cargo, there are the usual three separate piping systems for cleaning the tanks, fire extinguishing by means of steam, and heating the cargo tanks by steam, the latter system being made of 2-in. tubing. There is also the usual gas-line system with the piping fitted into coamings and connected to gas vents on the two masts about 50ft. above deck level. Gauges indicating the gas pressures prevailing in the various tanks are fitted on the bridge. The usual steam ejector installation is also provided. In addition to cargo oil tanks the ship has a large compartment for dry cargo forward. The propelling machinery, which is aft, consists of a 5-cylr. s.a. two-stroke Burmeister & Wain Diesel engine developing 2,150 b.h.p. at 115 r.p.m., but with an overload capacity of 2,500 b.h.p. at 120 r.p.m. for four hours. The engine-room lay-out of the ship is shown in the accompanying drawing, in which the unusual arrangement of the pumps driven from the main engine will be noted. These comprise cooling-water, S.W. sanitary, bilge and lubricating-oil pumps. Most of the E.R. auxiliaries are motor-driven, but there is also a steam-driven 50-kW. 110-volt turbo-generator, a 95-ton cooling-water pump for the refrigerating plant, two vertical air compressors each of 88 cu. ft./min. capacity and pumps for the lubricating-oil, sanitary and fresh-water services. The steam requirements are met by an oil-fired boiler of 1,175ft.^{2} heating surface operating at 180lb./in.^{2} pressure with Diesel oil as fuel. It was originally intended to employ a waste-heat exhaust boiler with additional oil firing, but this plan had to be abandoned "because of the international situation",



Engine-room lay-out of motor tanker "Sam Bras".

1.—Air compressors. 2.—Starting-air reservoirs. 3.—6-kW. Diesel generator. 4.—50-kW. Diesel generators. 5.—Diesel oil service tanks. 6.—Main Diesel engine. 7.—Pumps driven by propeller shaft. 8.—Boiler feed pumps. 9.—Lubricating-oil cooler. 10.—Lubricating-oil filter. 11.—Centrifugal oil purifiers. 12.—Auxiliary lubricating-oil pump. 13.—General service pump. 14.—Fresh-water pump. 15.—Sanitary services pump. 16.—Ballast pump. 17.—Diesel oil transfer pump. 18.—Bilge oil separator.

it is stated. The boiler equipment comprises a forced-draught fan, two steam-driven and one motor-driven feed pumps, a condenser of 323ft.^{2} cooling capacity, a circulating-water pump for the latter and a 50-ton evaporator for make-up feed purposes.—*"The Shipping World", Vol. CX, No. 2,651, 5th April, 1944, pp. 370-371.*

Safety Device for Propeller Shafts of High-speed Small Craft.

The wear on the stern-tube bearings of high-speed small craft is always heavy, and when these vessels operate in shallow water the wear is increased by the action of water-borne sand and silt. When the stern-tube bearing is worn the propeller shaft drops, thereby straining the whole of the shafting and propelling machinery. This condition, if not remedied by lining up the bearing, may lead to a fracture of the shaft, and if the fracture occurs in or near the stern tube, it is probable that the after end of the shaft together with the propeller will be lost and that water will enter through the stern tube. A safety device to prevent such a mishap has recently been developed and patented in this country. It consists primarily of a flange-shaped fitting clamped on to the propeller shaft and completely enclosed in a strong watertight casing which is secured to the hull of the vessel adjacent to the stern-tube bearing. The inner end of this casing may be continued to the next bearing. The casing is provided with a watertight inspection door and has a facing or raised machined surface on the forward side of its after end. For normal working the clamped collar or flange on the shaft is arranged to run just clear of the facing, but in the event of the shaft being fractured and moving aft, the clamped collar comes into contact with the machined surface of the casing and thus prevents the inboard end of the shaft from going out of the bearing at the stern end. If the shaft breaks or becomes detached anywhere between the collar and the stern tube, water is prevented from entering by the remainder of the shaft and the close contact between the clamped collar and the casing. The hole for the shaft at the inboard end of the casing is made slightly larger than the shaft diameter, so that it offers no resistance to the rotation of the shaft. At the same time it prevents the occurrence of any excessive leakage. Provision is made for the space inside the casing to be grease-lubricated, and another small collar can be fitted on the shaft to prevent the escape of grease from the casing.—*"The Journal of Commerce" (Shipbuilding and Engineering Edition), No. 36,240, 6th April, 1944, p. 7.*

Spark Extinguishing Device for Motorships.

A recently published British patent covers a steam-operated device for extinguishing sparks in the exhaust from a ship's Diesel engine. The arrangement, shown in Fig. 4, comprises a steam scavenging pipe (*1*) located in the exhaust pipe (*2*) and capable of being rotated by the operation of a handle (*3*). A distributor (*4*) for steam is fitted in the exhaust pipe above the silencer (*5*) and further steam distributing pipes (*6, 7, 8*) are placed near the top of the funnel.—*"The Motor Ship", Vol. XXV, No. 291, April, 1944, p. 32.*

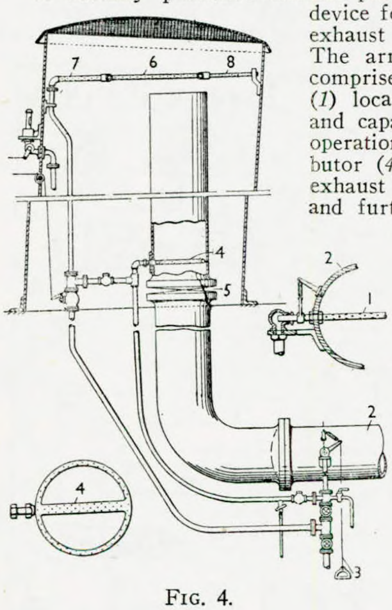


FIG. 4.

An Engineering Paradox.

With certain Diesel engines for which the cooling water and the lubricating oil are supplied by engine-driven rotary pumps, the design of the latter is such that if they run at considerably below their rated speed for any length of time, it is necessary to supplement their output by starting up the corresponding steam or motor-driven stand-by pumps. This condition may well arise when a moderately fast motorship is in a slow convoy, and on one occasion when this occurred the stand-by circulating-water pump developed a slight defect, which made it necessary to stop the pump for a time in order to make good the defect. The chief engineer duly notified the captain and asked that the ship's speed be increased temporarily to ensure an adequate supply of cooling water whilst the stand-by pump was out of action. The increased speed of the ship made it necessary for the vessel to make circles at relatively high speed until it was possible to start up the stand-by circulating-water pump again, when the speed was reduced to correspond with that of the convoy. It happened that the commodore of the convoy was on board this particular ship, and when the incident was closed, he asked the chief engineer about the trouble, and remarked that during his life at sea he had never before experienced a case in which trouble in the engine room made it necessary to run the engines at a higher speed whilst effecting repairs.—*"The Motor Ship", Vol. XXV, No. 291, April, 1944, p. 4.*

Free-piston-engined Air Compressors.

The two forms of free-piston-engined air compressor shown in Fig. 3, include improvements in this type of machine effected by a Portuguese engineer and recently patented in this country. Referring first to the upper diagram, the engine cylinder (*1*) contains two opposed pistons which uncover, respectively, the inlet port (*3*) and the exhaust port (*4*) at the outer ends of their stroke. One piston rod is attached to the air-compressor plunger (*5*), which charges a receiver (*16*) on one side and supplies the engine cylinder casing (*7*) with air on the other. The plunger (*8*) on the right acts as an air cushion, storing energy and returning it to the engine pistons on the return strokes. The fuel pump (*13*) is actuated by a lever (*14*) and supplies the injector (*18*), the quantity of fuel being regulated by a spindle (*15*) according to the air pressure in the receiver (*16*). In order to increase the output of the plant, a supercharging blower (*19*) is provided to supply air to both sides of the compressor plunger (*5*). This blower is driven by an exhaust-gas turbine (*20*) having three inlet pipes controlled by a slide valve (*22*), the position of which is determined by the air receiver pressure acting on a piston (*23*). In the lower diagram, the supercharging blower (*19*) supplies air only to the inner end of the cylinders (*6*) which delivers air to the engine cylinders. The outer chamber of the compressor is fed by a turbine (*29*) driven by air from the receiver (*16*), so that the air for this part of the machine is in a closed circuit, with the pressure regulated by a valve (*30*) connected to a reservoir (*32*) or to the atmosphere, as necessary. In order to regulate the power of the turbo-blower unit (*19, 20*) simultaneously, the right-hand side of the piston (*23*) is subjected either to the pressure in the conduit (*31*) or to that in the supply pipe (*33*) of the turbine (*29*).—*"The Oil Engine", Vol. XI, No. 132, April, 1944, p. 328.*

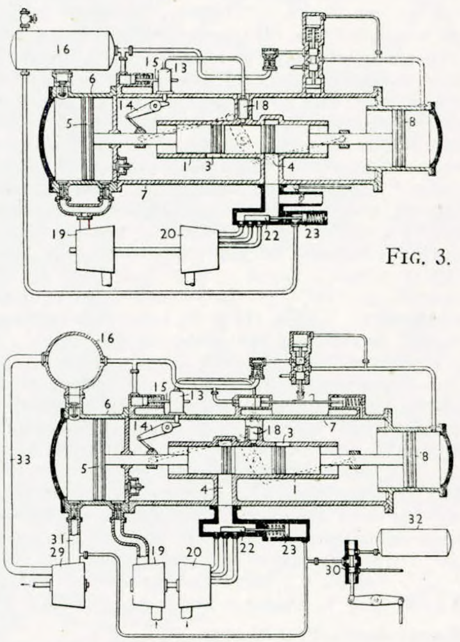


FIG. 3.

Independent Automatic Superchargers.

The employment of a directly-driven supercharging blower for a four-stroke Diesel engine may not be entirely advantageous because some of the additional power obtained from the engine by its use is required to drive the blower, and because of this, the maximum engine output is not reached. It is partly for this reason that independently operated superchargers are now being fitted in certain new ships, including some of the 50 large ocean-going tugs of the "Seguin" class, building for the U.S. Maritime Commission. These tugs have single screws operating in a Kort nozzle and driven through electro-magnetic slip couplings and 2·5 to 1 reduction gearing by two 8-cylr. supercharged four-stroke Diesel engines, each rated at 1,250 b.h.p. Independent motor-driven blowers are installed for these engines, their output and operation being automatically controlled according to the actual requirements of the main engines. The blowers, which are of the positive pressure cycloidal type, are driven by their respective motors at constant speed. A relay control switch is provided for each motor, and if the engine control is set so that the b.m.e.p. is below 78lb./in.^{2}, which corresponds to the normal supercharging output of the blower, the switch is not closed and the blower motor does not start. When the control is advanced, however, the relay is actuated and the blower motor starts up, providing the necessary supercharging air.—*"The Motor Ship", Vol. XXV, No. 291, April, 1944, p. 3.*

High-speed Opposed-piston Diesel Engines.

In one of the earliest ships to be driven by a Doxford opposed-piston engine, the auxiliary engines driving the dynamos were also of the opposed-piston type and ran at a speed of approximately 300 r.p.m. No further engines of this class were constructed and since that time no progress has been made in this country with the building of fast-running opposed-piston engines, despite the fact that in some respects they would appear to lend themselves to operation at moderately high speed. Opposed-piston engines of about 1,200 b.h.p. running at 700 r.p.m. have been built in America for the propulsion of submarines and other craft, and, in view of the probable future demand for fast-running two-stroke engines, both for propulsion and auxiliary service on board ship, it might seem desirable for British engine-builders to undertake some development work with the object of producing a satisfactory design of high-speed opposed-piston engine.—*"The Motor Ship", Vol. XXV, No. 291, 1944, p. 5.*

New Developments of the Marine Gas Turbine.

It is reported that the U.S. Navy Department has placed contracts with four manufacturers for marine gas turbine installations, but no details of these have been released. The experimental gas turbines which have already been built elsewhere are based on initial temperatures of slightly over 1,000° F., and it is believed that if this temperature can be raised to 1,200° F., it may become possible to manufacture gas turbines competitive in fuel consumption, weight, space, and general performance with the most efficient steam installations. If boiler oil can be used—and this is not the case at present—marine propulsion by gas turbines might even become competitive with propulsion by Diesel engines. If the initial temperature could be increased to 1,500° F. with a single reheat of the gases to this initial temperature after passing half-way through the turbine, and with a single stage of inter-cooling in the compressor, the fuel consumption could, it is claimed, be reduced to 0·4lb./b.h.p.-hr. It should be remembered, however, that there are definite limits to the thermal efficiency of the gas turbine, no matter what progress may be made, and hopes should not be raised too high, although it may be anticipated that during the next few years it might become possible to develop a design of gas turbine of the simplest construction, operating with a lower initial temperature and having an efficiency equal to that of a steam installation, but lower than that of corresponding Diesel plants. Such a gas turbine will, however, possess the advantage of economy in weight and space, although it will require special arrangements for astern running, a possible design comprising a two-unit installation, one turbine driving the compressor directly, and the other connected to the propeller through electric transmission or gearing. The employment of a variable-pitch propeller with gas turbines has also been suggested. At the present time there would appear to be a wide divergence in the ideas of designers of gas turbines, since the running speeds vary from 2,930 to 13,000 r.p.m., the pressure ratios between 4·6 and 7·8, the maximum temperatures between 1,200° and 1,800° F., the anticipated thermal efficiency between 15 and 31 per cent., and the maximum output per unit 2,500 and 4,000 h.p. In addition, consideration is being given to the employment of a Diesel engine driving its supercharger and exhausting into a gas turbine, presumably along the lines adopted by Sulzer Bros.—*"The Motor Ship", Vol. XXIV, No. 290, March, 1944, p. 385.*

Diesel Starting Systems.

An article in a recent issue of *Power* discusses the various systems employed for the starting up of Diesel engines and describes six of them, *viz.:* compressed-air starting and cranking by electric motor, petrol engine, air motor, motorised generator, or manual effort, as the case may be. To start ignition in the cylinders, the engine speed should be brought up to at least 10 to 15 per cent. of normal, starting being facilitated by heating the intake air or cylinder-jacket water, or both. A common method of starting up Diesel engines is to admit compressed air at a pressure of about 250lb./in.^{2} to two or three of the cylinders, but the provision of starting-air valves on every cylinder is preferable, since it ensures correct starting from any crank position. On the other hand, the admission of air to only a few of the cylinders avoids cooling the others by the expansion of the starting air. The capacity of the starting-air bottles or reservoirs may be about 30 times the displacement volume of one engine cylinder, and where engines start infrequently the air compressor may be of a size enabling the air bottles or reservoirs to be recharged in from 10 to 20 minutes. Petrol engine and electric motor drive for the compressor makes it possible to start up the Diesel engine from a complete shut-down. Power-driven and manual cranking arrangements are suitable for small Diesel engines. Diesel-electric sets can be started by running the generator as a motor from accumulators, which must be capable of delivering the heavy current required. Some of the latest air motors for starting Diesel engines consist of two small two-cylinder engines 90° apart on a common crankshaft, driven by compressed air from an external source. Hand-cranking is only suitable for the smallest Diesel engines and requires the provision of a decompression device in addition to a heavy fly-wheel.—*"The Power and Works Engineer", Vol. XXXIX, No. 454, April, 1944, p. 96.*

New Polar Engine Design.

The most recent development in two-stroke oil engines for marine service reported from Sweden relates to an improved type of Polar engine produced by the Atlas Diesel Co. It incorporates a new type of fuel-injection valve by means of which practically the entire charge of fuel is injected and dispersed before ignition takes place. Combustion is prolonged without the burnt gases formed at the beginning of the combustion process mingling with the unconsumed air in the outer part of the cylinder chamber. The inconvenience of ignition delay has been overcome by mechanical devices. Exceptionally quiet running of the engine is achieved by the employment of a wave damper in the fuel system, consisting of a double valve which permits surges of fuel to pass through the pipes, but prevents them from being reflected against the discharge valves. Other features of the new engine are improved flexibility, a low fuel consumption and an ability to burn fuels of varying characteristics. Engines of this type have been running on boiler oils with a very low cetane number, and it is claimed that heavy asphalt oils may be used without any risk of choked sprayers.—*"The Motor Ship", Vol. XXV, No. 291; April, 1944, p. 23.*

Engine Costs.

Reports in the American press concerning a "poor man's Diesel" to cost about £3 per h.p., suggests a reversal of the usual trend of development making for higher cost and increased complication. The figure quoted is only a fraction of that normal for present-day engines, more especially those of the slow-running marine type, but it is not always realised that the aero engine, in spite of its refined design and close tolerances leading to high cost per lb., is a relatively cheap prime mover reckoned in £ per h.p., and that the total cost of certain engines of this type may quite possibly approach the figure quoted. This applies to the ordinary piston engine which, even when conservatively rated, has a very short useful life as judged by marine standards. It has been suggested that the simplicity of the jet engine, together with the resulting saving in weight and first cost, will constitute a more attractive proposition in this respect, since the reduction in the number of wearing surfaces will increase the durability and decrease the maintenance costs. Both the aero jet engine and its marine counterpart, the internal-combustion turbine, suffer in their simplest forms from the disadvantage of low thermal efficiency at the working temperatures attainable with available blade materials, but there should be a fair margin in hand for the addition of such complications as may be required to reduce the fuel consumption without making the weight and cost of the unit non-competitive.—*"Shipbuilding and Shipping Record", Vol. LXIII, No. 16, 20th April, 1944, p. 363.*

Applications of the Velox System of Supercharging.

An article by Dr. W. G. Noak in a recent issue of the German *Z.V.D.I.* calls attention to the fact that a Velox installation has a number of points in common with a straightforward pressure-charging system of an internal-combustion engine, and that Velox units can be and are, therefore, used for a number of purposes outside the usual field of steam generation. Typical applications include air heating for blast furnaces and chemical works, as well as superheating and reheating the steam in large power stations. It is claimed that the relatively small specific weight and o.a. dimensions of a Velox unit make it particularly suitable for marine purposes, and that when used for superheating and reheating the steam in a turbine installation, it can be fitted close to the turbine, thereby eliminating long connecting pipes with their attendant losses. The unit takes up little space and requires no special attention, since its operation is automatic. The Velox unit for heating steam differs very little from the Velox boiler, the main differences being that the former does not include an evaporator section, and that the automatic control gear is governed by the degree of superheat instead of by the amount of steam pressure.—*"The Marine Engineer", Vol. 67, No. 801, April, 1944, p. 127.*

Continental Practice in Diesel Engines.

The German technical periodical *Schiff und Werft* (which is now the title of the amalgamated (*Schiffbau* and *Werft*Hafen*Reederei*) recently published an article dealing with developments in pre-war marine Diesel engines. It is suggested that the Diesel engine is most suited for powers above 4,000 i.h.p., speeds above 14 knots and a radius of action above 4,000 miles. The 4-stroke engine, it is stated, has by the application of supercharging regained some of the ground it had previously lost, whilst the single-acting 2-stroke engine is becoming more popular at the expense of the d.a. engine. Cylinder diameters, which reached a maximum of 900 mm. in certain Sulzer designs and 875 mm. in the case of Krupp engines, are becoming stabilised at about 760 mm., which is sufficient to give an output of 1,000 h.p. per cylinder in a s.a. 2-stroke engine. An objection to cylinders of very large diameter is the length of time necessary for overhauls and repairs, especially with the limited facilities available on board ship. Continental 4-stroke engines are nearly always of the trunk type, whereas s.a. 2-stroke units are generally built with cross-heads. The M.A.N. firm, however, make a trunk type of 2-stroke engine developing 335 i.h.p. per cylinder at 225 r.p.m., which is designed specially for use with reduction gearing. For large cargo vessels and tankers making long voyages, a s.a. 2-stroke engine of such a type running at 100 to 150 r.p.m. is recommended, and a 10-cylr. unit of this kind, built by Krupp, has an output of 7,000 i.h.p. The Burmeister and Wain d.a. 2-stroke engine, with exhaust pistons at the ends of the cylinders, is given as an example of recent practice in this type of machinery. As far as can be gathered from the article, the development of the Diesel engine in this country has not lagged behind Continental practice, and there is every reason to believe that in certain types we have a definite advantage. Nevertheless, it is reasonable to suppose that progress has been made on the Continent during the war years, and that post-war designs will incorporate improvements. The article suggests that the post-war Diesel engine is likely to be of the high-speed type, with either electrical or mechanical gearing, an opinion which is held by some authorities in this country also, despite the various difficulties which have still to be overcome before this system of drive can compete with the direct drive in reliability and fuel consumption. It is probable that the thermal efficiency of the modern heavy oil engine is now approaching a limit, but further reductions in weight and costs are still possible, together with simplified operation and maintenance. —*"Fairplay", Vol. CLXII, No. 3,179, 13th April, 1944, p. 550.*

Producer Gas for Belgian Canal Craft.

It is reported that owners of Belgian canal craft are to be encouraged to convert their Diesel propelling machinery to producer-gas operation by loans to cover the entire cost of conversion. Owners who do not voluntarily have their vessels converted may be compelled to have the work carried out by order of the Central Bureau for Producer-gas Installations in Brussels. An improvement of 7 per cent. in performance is claimed to have been attained with two vessels which were recently converted. These craft have a cargo capacity of about 300 tons and are equipped with engines of about 80 h.p. After the latter had been modified to run on producer gas, the two vessels underwent successful trials, during which they reached a speed of 7½ m.p.h.—*"Lloyd's List and Shipping Gazette", No. 40,357, 29th March, 1944, p. 9.*

Ten Years' Operation of Diesel-electric Ferries.

The two double-ended Diesel-electric ferries "Queen Margaret" and "Robert the Bruce", which carry passengers and motor vehicles between North and South Queensferry, have now been in service for 10 years. The service operates seven days a week and 16 hours per day throughout the year, and, except for an annual dry-docking and refit of about a fortnight's duration, the two ships have been run continuously for the past decade. Built by Wm. Denny & Bros., Ltd., Dumbarton, they are paddle vessels 149ft. in o.a. length, with a beam of 28ft., a breadth over sponsons of 47ft. 8in. and a depth of 7ft. 10in. The "Robert the Bruce" was probably the first all-electric-welded vessel built in Scotland, the "Queen Margaret" being of riveted construction. The propelling machinery, in each case, consists of two 8-cylr. Paxman oil engines developing 200 b.h.p. at 750 r.p.m. and each directly coupled to a 110-kW. d.c. generator and 9-kW. exciter. The generators supply current at 440 volts to two 135-b.h.p. propulsion motors, normally running at 270 r.p.m. and each driving a paddle wheel through Bibby flexible couplings and twin duplex Renold roller chains. The normal paddle-wheel speed is 45 r.p.m. A 12-kW. auxiliary generator, directly driven at 1,000 r.p.m. by a 3-cylr. R.N. oil engine, supplies direct current at 220 volts for lighting and auxiliary purposes. The E.R. staff of each vessel consists of two engineers, each of whom does about 8 hours' duty per day, and of a maintenance engineer, who goes aboard at night and does whatever happen to be the most urgent jobs at the moment, in addition to routine examinations and overhauls. Both hulls have given good service and there seems to be very little to choose between them. The welded "Robert the Bruce" shows up rather better as regards corrosion, whilst the riveted hull appears to be less susceptible to vibration, but both hulls are in an excellent state of preservation. The main electrical machinery has run almost entirely free of trouble throughout the 10 years it has been in service, whilst the main Diesels have acquitted themselves well in spite of the arduous conditions under which they have been operating. Although they are only run for about one-third of the time the vessels are in service every day, they are started and stopped at least 32 times a day. The cylinders are cooled with salt water and silting in the jackets is considerable, about 6in. of sludge having to be removed from the bottom of the jackets every six months. Running conditions are not favourable for the maintenance of a uniform and efficient cylinder temperature, as the engines are run for 10-12 minutes on the actual passage and are then shut down for 18-20 minutes, as a result of which the jackets are always cold when they are started up and, as there is no after-cooling circulation, the jackets become very hot immediately after the engines are stopped. These conditions are not conducive to minimum cylinder liner and piston ring wear, but even so, liner wear has not been abnormal, although a number of replacements have been made. If a new ship were being built for this service, fresh-water cooling would be adopted. The original water-cooled exhaust manifolds were in one piece and their removal for dismantling a single cylinder cover used to be a laborious job, since it involved breaking eight exhaust joints and eight water joints. The water-cooled manifolds were therefore replaced by uncooled ones made in three sections and fitted with diaphragm type expansion pieces at the joints of the sections. This arrangement has proved far more convenient when overhauling, as it is now only necessary to remove the manifold from three cylinder heads at most when removing a cover. The abnormal amount of starting and stopping used to cause frequent breakages of the spindles of the centrifugal cooling-water and bilge pumps, for which reason independent motor-driven pumps were installed, although the lubricating pumps remain engine driven. The original hand steering gear had to be replaced by electrical steering gear, and the resulting extra load added to that imposed by the new motor-driven pumps, made it necessary to replace the original twin-cylinder auxiliary Diesel engines which drove the auxiliary generators as well as an air compressor and a general service pump, by 3-cylr. engines of the same make. Trouble with the fuel-injection pump drive of the main engines made it necessary to fit an improved and more flexible type of drive. The main engines are started by compressed air at a pressure of 250lb./in.² admitted directly into the cylinders in the usual way, but it has been found that when the engines are warm, a pressure of 125lb./in.² is adequate, so in order to reduce the stresses induced by the frequent starts called for by the particular service, a reducing valve was fitted in the starting-air pipe system. One of the starting-air pipes close to the starting-air control valve on the engine, was replaced by a pipe of about 11in. bore which acts as a receiver for the reduced-pressure air and stabilises the pressure on the engine side of the reducing valve. A by-pass is fitted to the latter to enable the full starting-air pressure to be used when required. Originally, the inlet and exhaust valves of the main engines were of the type where the valve is guided on the valve stem only, *i.e.*, without a separate piston-type guide. Wear on the valve stems was heavy and to save the valve cages, the old guide bosses were machined off and loose renewable guides fitted. The driving chains of the paddle wheels are standing up to their work remarkably well, and three of the four chains fitted 10 years ago are still in service. The paddle boxes have had to be strengthened to withstand the rough treatment to which they are liable to be subjected when berthing in heavy weather. The motor-driven deck machinery has proved generally satisfactory, but the original metal-to-metal joints between the motors and the machines they drive have had to be replaced by insertion joints. All exposed shafts have also been well protected.—*"The Marine Engineer", Vol. 67, No. 801, April, 1944, pp. 107-112 and 120.*

A Diesel Engine Laboratory.

Much has been said and written recently about the need for research in industry, but the fact is sometimes overlooked that if research work is to be successful, it is essential to have well-trained research workers and well-equipped laboratories in which they can work. Some idea of what constitutes a well-equipped laboratory can be gathered from the description recently published of a Diesel engine research laboratory which has just been placed in operation in the U.S. It has a number of test bays with dynamometers capable of testing engines rated from 20 up to 2,000 h.p. and equipment is available for making fuel and lubricating-oil consumption tests, engine heat-balance tests, noise tests and studies of the vibration characteristics of an engine as a whole as well as of its component parts. Investigations can be made into the pressure changes in the suction and exhaust systems as well as in the cylinders of the engines, while tests can also be carried out on such details as fuel, water and lubricating-oil pumps, including their calibration and adjustment, nozzles, governors and general engine accessories. Throughout the establishment no windows have been employed, lighting being obtained by a fluorescent system, while the ventilation equipment, in addition to regulating the temperature of the air, filters it so as to exclude all dust and dirt. Special provision has been made for investigations into the noise made by engines during operation, a number of sound-proof bays being available for these tests.—*"Shipbuilding and Shipping Record", Vol. LXIII, No. 16, 20th April, 1944, p. 363.*

Control of Hand-fired Coal-burning Boilers.

The main causes of boiler inefficiency in ships equipped with hand-fired coal-burning boilers are unburnt fuel in the ash, radiation from boiler surfaces, unburnt gases and carbon in the products of combustion, and heat lost to the atmosphere with the flue gases passing out of the funnel. The first two of these items are largely unavoidable and beyond the control of the operating personnel, but the last two are easily regulated and therefore vital to the problem. The extent of the controllable factors may range from 18 to 34 per cent. in the total heat account of the fuel, so that the gross boiler efficiency can vary from 60 to 75 per cent., according to the care that is exercised, whilst the wastage of fuel may amount to as much as 20 per cent. The carbon content of good steam coal amounts to some 85 per cent., and this can combine with the air to form carbon dioxide when there is a specific liberation of about 14,500 B.Th.U. per lb. of fuel. A pure carbon fuel combining with the exact amount of air to form CO_2 would yield 21 per cent. of this gas in the products of combustion, but all fuels contain some hydrogen which combines with some of the oxygen in the air to form water, so that the optimum percentage of CO_2 resulting from the combustion of the coal is only about 18 per cent. Moreover, in practice, it is necessary to supply a certain amount of air in excess of the theoretical requirements to obtain proper combustion, and this dilutes the funnel gases so that the percentage of CO_2 is further reduced. Air admitted to the furnaces in excess of what is required will reduce the temperature of the funnel gases, so that the loss of heat from this cause is apparently diminished, but, on the other hand, a relatively greater aggregate is passing away to waste. For example, the heat that passes to waste when satisfactory combustion is obtained with 50 per cent. excess air will equal about 15 per cent. of the heat in the fuel, while the funnel loss will be double this when 200 per cent. excess air is supplied. It is therefore essential to keep the amount of the excess air down to a minimum in order to secure the best results. Moreover, the excess air entering the furnaces reduces the temperature of the fires, whereas a high furnace temperature is desirable for promoting the transmission of heat in the most efficient manner. Any shortcomings in combustion of the fuel will also produce heavy deposits of soot on the heat convection surfaces of the boiler, and soot has such high non-conducting properties, that a 1-in. coating of soot will reduce the heat transmission to about 1/20th of the value for a perfectly clean tube, apart from the mechanical damage that can result. Thus, a falling-off in boiler-room efficiency is reflected by a reduced power output of the engines and a loss of ship speed, so that the cumulative effect of all these deficiencies can assume serious proportions. The only reliable way of controlling combustion in boilers is by measuring the percentage of carbon dioxide in the funnel gases and their temperature at the funnel base, in order that the air supply to the furnaces may be regulated on this basis in relation to the load at which the boilers are being worked. To reduce the loss as much as possible, the temperature of the funnel gases should be kept at a minimum, whilst the percentage of CO_2 which they contain must be kept at a maximum. With good quality coal and a reasonable standard of hand-firing, the amount of excess air to be supplied for complete combustion can be reduced to about 80 per cent., corresponding to about 10 per cent. of CO_2 in the funnel gases. Nothing of all this is new, but the means available for ensuring such conditions are, perhaps, not as widely known as they deserve to be. Various types of apparatus have been developed for the analysis of funnel gases, but many marine engineers are under the impression that such appliances are unsuitable for use at sea because they are thinking of the older forms of CO_2 recorders that were a maze of glasswork and rubber tubing. Modern instruments are both reliable and robust enough to meet all the requirements of sea service if reasonable care is exercised. It has also been suggested that the value of these devices is discounted with hand-fired boilers because the condition of the fires is constantly changing, but this is the very factor that has to be brought under control and which provides a strengthened reason for their intelligent use. An inherent difficulty connected with the use of the older forms of gas-analysing apparatus with hand-fired coal-burning boilers, was the time lag involved; a delay of up to 20 minutes might be inevitable between the time the sample was taken and when the results were available. The use of improved forms of aspirator with modern CO_2 recording instruments has eliminated this difficulty, and the time lag has now been reduced to about 30 seconds, so that a close indication of what is happening at the moment is provided. Comprehensive instructions concerning the burning of each class of coal taken into the ship's bunkers should always be furnished to the chief engineer; this information should state when fires should be cleaned, the quantity of air to be admitted above and below the firegrate, and the most suitable thickness and disposition of the firebed. The firemen attending the boilers must be able to observe the changes that are taking place in the furnaces with every alteration in the conditions of firing and air supply, by means of readings indicated upon a conspicuous gauge in the stokehold. A recording gauge providing a continuous time record of the furnace performance is also useful, as it shows the chief engineer when the firing has been taking place, how long the furnace doors were left open and how the fires are being cleaned. A self-recording pressure indicator, in synchronism with the combustion indicator, will provide a most helpful stimulus to the firemen in maintaining improved conditions in the stokehold. Efficiency in the boiler room can only be kept up by taking continuous measurements of the factors involved and acting upon them. The saving of fuel and improvement in working conditions will soon pay for the cost of the necessary instruments and for the extra care that has to be taken.—*"The Journal of Commerce" (Shipbuilding and Engineering Edition), No. 36,228, 23rd March, 1944, p. 1.*

Modern Apparatus for Boiler Flue Gas Analysis.

Various types of instrument have been developed for the analysis of funnel gases. Chemical instruments depend for their action upon the absorption of carbon dioxide by substances such as caustic potash and soda, or lime, while physical instruments rely upon the observation of physical characteristics that vary with different percentage diffusions in the contents of the flue gases being analysed. One simple form of chemical recorder consists of a chamber containing a porous pot in which the absorbent is placed. Samples of the gas being analysed are drawn through the chamber, and some of it penetrates into the porous pot where it is absorbed by the reagent, thereby creating a partial vacuum; the difference of pressure across the walls of the pot constitutes a measure of the CO_2 that can be read off directly on a water-gauge column. An alternative design of instrument provides for the measurement of a quantity of the flue gases at regular intervals, the gases then passing into a tank containing caustic potash which absorbs the CO_2, the remainder of the gases then being measured in another chamber. The difference between the quantity of the sample and the remainder provides the measure of carbon dioxide that is present in the flue gases. An electrical method for indicating the CO_2 content of the products of combustion is based upon the principle that the temperature of a platinum wire, which is surrounded by a gas and connected to a source of constant electromotive force, will rise until a condition of equilibrium is reached where the dissipation of thermal energy equals the electrical energy being supplied. The thermal energy is dissipated by convection, radiation and conduction, but the apparatus is arranged so that the loss from the first two of these causes is negligible, the final temperature of the wire being also entirely dependent on the thermal conductivity of the surrounding gas. Flue gases consist chiefly of nitrogen and carbon dioxide, with small quantities of oxygen, water vapour and carbon monoxide. Nitrogen, oxygen and carbon monoxide have about the same thermal conductivity, so that variations of these gases have no influence on the readings, and arrangements are provided for counteracting the effect of water vapour, when the temperature of the wire, which can be determined by measuring its electrical resistance, gives a direct indication of the percentage of CO_2 that is present. Another instrument of the physical type relies for its operation on the principle that a fan rotating opposite a vane wheel is an enclosed chamber will transmit a torque of a magnitude depending upon the density of the contained gas. Two chambers are used, one containing air as a standard for comparison, while flue gas is passed through the other, the vane wheel systems of the two chambers being connected by linkage so that the relative magnitude of their respective torques is indicated on a scale to give a direct reading of the CO_2 percentage. Readings of carbon dioxide alone can be deceptive, since the percentage present will be steadily increasing up to the point where there is just sufficient excess air for complete combustion and, during this time, there will be a corresponding reduction in the amount of carbon monoxide being formed. With a greater quantity of excess air, there will be a falling off in the percentage of CO_2 through dilution of the flue gases. Any readings of carbon dioxide, below the maximum that is possible, may be due to too little excess air when the loss of heat through incomplete combustion will be much greater than the loss occasioned by a surplus and, for this reason, it is sometimes urged that readings for carbon monoxide should also be recorded, but such a procedure would introduce complications and is unnecessary with hand-fired coal-burning boilers, where the quantity of excess air required is such that this situation is unlikely to arise.—*"The Journal of Commerce" (Shipbuilding and Engineering Edition), No. 36,228, 23rd March, 1944, p. 1.*

Welding of Boiler Tubes.

There are no actual compelling reasons why an assembly of tubes welded into the drums or headers of a boiler should not operate satisfactorily, and welded connections are in fact used for the tubes of certain special types of boilers. However, there are some difficulties to be overcome in the substitution of welded connections for the expanded joints of the tubes in the conventional types of boilers in use at the present time. The principal difficulty arises from the fact that in welding in the tube the tube-plate or header is heated and may thereby become distorted and, what is even more serious, locked-up or unrelieved stresses may be set up in the tubes or headers by the heating process. Where a boiler is built up in sections, it may be possible to assemble a complete section by welding and to stress relieve the assembly on completion of the welding process, but a procedure of this kind would involve some extra expense. Another possible difficulty arises when tubes have to be cut out and replaced. The removal of a tube that was welded in would be extremely difficult without damaging the tube seat, and even if this were overcome, the welding in of the new tube might heat up the adjacent parts of the boiler and give rise to local distortion or other damage. Such difficulties might, in some cases, be overcome by care and the use of special apparatus. However, since the practice of welding in tubes does not save any appreciable weight or expense and since an expanded tube can be removed and replaced more easily than a welded-in tube without requiring any special tools except expanders, there is not much advantage to be gained by employing welded-in tubes. —*"Marine Engineering and Shipping Review", Vol. XLIX, No. 1, January, 1944, pp. 222-224.*

Merchant Shipbuilding in America.

Up to the end of January, 1944, the total tonnage put into service by American shipyards since the U.S. entered the war was 26,625,263 d.w. tons, or over 17,300,000 tons gross. A large part of this tonnage was made of Liberty ships, of which there were about 1,700 in service at the beginning of December. Not only did these vessels contribute so successfully to the building up of so vast an amount of tonnage, but the simple propelling machinery installed in them required a minimum of skilled personnel in the engine room. The E.R. complement of a Liberty ship numbers only 15 officers and men, and in order to provide the necessary skilled supervision recourse was often had to special measures, such as granting a chief engineer's certificate to a second engineer after only six month's service as such. In January, 1944, American yards delivered 71 Liberty ships, 9 standard C-type ships, 11 standard tankers, 3 tankers converted from Liberty ships, and 2 coastal tankers, in addition to a number of miscellaneous vessels of special types for the U.S. Army and Navy. Actually merchant ship production in January was the lowest since the previous February, and only amounted to 124 vessels, as against 208 ships in December. As U.S. shipbuilding was planned with the sole object of obtaining maximum production regardless of cost, with an inevitably high expenditure on wages and other items, and as many of the firms engaged on the mass production of marine engines and boilers had no previous experience of this class of work, American shipbuilding costs are far higher than those in this country, despite economies achieved by the employment of welding and prefabrication for the construction of hulls. The cost per ton deadweight of the 60 Ocean-type cargo steamers built in the Tod and Richmond yards for the British Government was about £45, *i.e.*, approximately three times the cost in this country. It has also been freely stated that it takes twice the number of man-hours to produce a ton of shipping in the U.S. to what it does in Great Britain.—*"The Marine Engineer", Vol. 67, No. 800, March, 1944, pp. 76-77 and 82.*

A Fabricated Shaft Bracket.

A constructional engineering firm on the Clyde recently made a fabricated propeller-shaft bracket to replace the cast-steel starboard shaft bracket of a ship which had sustained damage. The constituent parts of the fabricated bracket were assembled by electric welding. The boss, which was prepared first, has a uniform taper towards aft, and is formed from two 2-in. plates, butt-welded at the ends of a diametrical section. The boss carries two heavy internal slab rings, 13in. in length, which were flame cut to size and welded to the boss plates. The after edge of the arm of the bracket is formed by a 15-in. nosing piece, whilst the upper and lower surfaces of the bracket consist of 2½-in. plates with internal stiffening members. The inboard end of the arm is secured to a heavy plate, 2in. thick, conforming to the shape of the hull and attached to the latter by a welded assembly of heavy plates in which both horizontal and vertical stiffening members are incorporated. The total weight of the fabricated bracket is 14½ tons.—*"The Shipbuilder", Vol. 51, No. 418, March, 1944, pp. 94-95.*

Safety Standards in New Construction.

In many respects the ships now being built have been modified to meet war-time hazards, thus providing a standard of safety—in relation to peace-time conditions—far higher than was usual in pre-war days. It has been suggested, however, that there is still room for improvement in this direction, more especially as regards increased W.T. sub-division of the hull. The safety of a ship can be enhanced by increasing the number of transverse bulkheads, but if this is done, the value of the vessel as a cargo carrier is reduced on account of the difficulties involved in the way of stowage and the handling of cargo. The sub-division of the shelter-deck ships being constructed at the present time is such that the vessel will remain afloat with any one compartment flooded, but this standard of safety has made it necessary to extend the W.T. bulkheads to the shelter deck, thereby increasing the tonnage. Unless the existing tonnage regulations are altered after the war, it may be anticipated that tonnage openings will be cut in 'tween-deck bulkheads in order to exempt the 'tween-deck spaces from tonnage measurement, with the result that the present-day standard of safety will be lowered. It would be possible to retain this standard with an open shelter deck by a slight reduction of the permissible draught, as has been done in the case of the American standard cargo ships, but such a procedure involves some sacrifice in the d.w. carrying capacity. The compressed-air method of keeping a damaged ship afloat, which has proved so successful in oil tankers, is scarcely applicable to ordinary cargo vessels owing to the difficulty of making the various deck openings, including the large cargo hatchways, airtight. Even the provision of steel hatch covers would not entirely overcome this difficulty. Generally speaking, however, safety devices which have proved their value in war-time are likely to be equally useful in peace-time, so that any improvement which may be effected in this direction must be regarded as being of enduring, and not merely transient importance.—*"Fairplay", Vol. CLXII, No. 3, 174, 9th March, 1944, p. 384.*

"Some Notes on Welded Ship Construction".

The author points out that the application of electric-arc welding to ship construction has aroused the interest of all associated with ships and shipping, and that shipowners and their technical advisers who may be considering plans for the post-war re-establishment of their fleets are naturally anxious to know as much as possible about the process and the advantages to be gained by its use. Welding is a means of improving the hull structure, but due to the main factors involved it is not practicable to lay down hard and fast rules as to economic justification for the welding of any particular item, as the available plant and facilities vary so widely in different shipyards. However, the views expressed by the author in this paper are based on knowledge derived from observations of shipbuilders' practice and from records of service experience of riveted and all-welded ships, and he therefore suggests that they may be acceptable as a general guide. He concludes by declaring that, from the evidence available, a hull structure properly designed for welding and having a high standard of workmanship should prove more satisfactory in service than a riveted hull.—*Paper by Jas. Turnbull, "Transactions of the Institute of Marine Engineers", Vol. LVI, No. 4, May, 1944, pp. 41-45.*

Fabricated Deck Fittings.

The production of fabricated deck fittings such as masts, derrick posts, bollards, etc., by various structural engineering firms with no previous experience of shipbuilding work has now been developed in this country to such an extent that it has liberated a substantial amount of skilled shipyard labour and specialised plant for other important branches of the shipbuilding industry. Apart from this, however, recent increases in the scantlings of masts and derricks to deal with the many relatively large lifts required in war-time, as compared with those to be handled under normal peace-time conditions, have frequently proved beyond the capacity of the plant ordinarily used in some of the shipyards for the manufacture of these items. The increased thicknesses and diameters now necessary would require the employment of plate rolls—normally reserved for heavy work—for this incidental purpose, with a correspondingly adverse effect on ship production. These considerations have likewise made it convenient to entrust the production of masts and similar deck fittings to other establishments which employ prefabrication methods for this purpose. The design of all such equipment has been standardised to allow constructional firms to plan their own production programmes, and to facilitate the replacement of damaged items from similar components in stock for new ships. The standard tubular masts for cargo vessels are 49ft. in o.a. length, with an outside diameter of 38in. at the foot and 28¼in. at the head. They are designed to house inside 9-ft. 'tween decks and a 7-ft. mast-house and are built in five welded sections, telescoped 2ft. at each joint. The topmost tube is reinforced by an interior doubling plate which covers the final 3ft. of the mast. The joints between the sections and the attachment of the doubling plate to the topmost tube are the only riveted connections in the assembly. The longitudinal seams are welded and the riveted joints are sealed by a fillet weld along the upper edge of the outer sleeve, the steel being sheared down to the depth of the weld to facilitate welding and to prevent water from lodging on the exposed edge when the mast is erected and thus setting up corrosion of the weld. The plates for the mast are first trimmed to size in a planing machine, which is also used to bevel the edges to be welded. The plates are then rolled to a circle or semi-circle in a large rolling machine and the longitudinal seams electrically welded, after which the welded sections or tiers are heated to 600° or 700° C. in a furnace to anneal the welds and relieve any locked-up stresses. The various sections are given a final rounding before being telescoped the required amount—in most cases adjacent sections being shrunk-on. The rivets are then driven and the sealing runs completed. As the sections are progressively built up to provide the required strength at particular points, redundant material is eliminated and the completed mast is appreciably lighter in weight than a corresponding mast of conventional design as it would normally be constructed in a shipyard. Several hundreds of these fabricated masts have been fitted in various types of merchant ships built in this country and have proved highly satisfactory in service. An exactly similar form of construction is used for the fabricated 5-ton and 10-ton derrick posts, for which, however, only two sections are required to cover the total length involved (41ft. 6in.). In the 5-ton posts, the outside diameter of the lower section is 28¼in., whilst that of the upper one is 27¼in., with thicknesses of $\frac{5}{16}$in. and ⅜in. respectively. The corresponding diameters in the 10-ton posts are 29in. and 27½in. with thicknesses of ⅝in. and ½in. As in the masts, the sections overlap 2ft. at the joint, and there are four rows of nineteen ⅞-in. diameter rivets, with a sealing weld at each joint. An internal doubling plate, 3ft. long and ⅜in. thick, is riveted to the top of the upper section of each post. The upper and lower sections are each formed from one plate, rolled to circular form, the abutting edges being welded. In the final assembly, the upper and lower tubes are so arranged that the longitudinal welds are on opposite sides of the post. Standard 12-in. diameter bollards are built up of mild steel plates, welded throughout. The flat base, 5ft. 3in. by 1ft. 6in., is flame cut from 1¼-in. plate and supports the two cylindrical portions, 1ft. 10½in. high and 12½in. in outside diameter, formed from ½-in. plate, rounded and longitudinally welded. The circular 16-in. diameter tops are of ⅝-in. plate. It is claimed that this form of construction effects important economies in material, labour and time, in comparison with the requirements of a cast unit.—*"The Shipbuilder", Vol. 51, No. 419, April, 1944, pp.119-120.*

Structural Failures in Liberty Ships.

A somewhat sensational article in a recent issue of the *Seattle Post-Intelligence* declares that ten Liberty ships converted into troop transports and valued at 21 million dollars, are now laid up near Seattle because the U.S. Navy and Army authorities have refused to make use of them for this purpose after one of the vessels in question, the "Christopher Greenup", had developed serious hull defects while in port at Cold Bay, Alaska, on 24th January. The ship was one of the 20 or more vessels of the Liberty type which had already had to undergo extensive hull repairs. Built by Mr. Kaiser's Oregon Shipbuilding Corporation, the "Christopher Greenup" first developed cracks in her hull plating on 5th March, 1943, while in Alaska waters. In addition to breaks at both Nos. 2 and 3 hatches, the vessel, then employed as a cargo carrier, developed cracks in her deck and shell plating in way of No. 5 hold, her port sheer strake opening vertically downwards for a length of 38in., whilst the deck stringer plating parted in an arc from the point where the cracked sheer strake met the stringer, and extending inboard for 36in. In spite of this failure, the War Shipping Administration ordered the vessel's conversion to a troopship, and it was while she was employed as such that the second mishap occurred. She was full of troops and was about to sail from Cold Bay, in calm weather, when an underwater crack developed in the shell plating of the engine room and water began to stream into the latter. All the troops were hurriedly put ashore and the ship proceeded to Seattle for repairs, arriving there on 10th February, after a voyage during which all the available pumps were used to pump water from the E.R. bilges. A technical expert of the W.S.A. was immediately sent from Washington to investigate the trouble and arrange for the necessary repairs. He recommended that all the converted Liberty ships should have their sides strengthened by the addition of a 6-in.×6-in. steel gunwale bar running fore and aft for four-fifths of the ship's length, but this proposal was summarily rejected by the U.S. naval and military authorities, who have stated that they will refuse to employ these vessels as transports even after the proposed strengthening of their hulls has been completed. Alternative suggestioos for strengthening the ships have been put forward by other technical experts, including some from the fighting services, but the W.S.A. have now given orders for five of the laid-up Liberty transports to be taken in hand for strengthening according to their technical expert's recommendations. This work will take several months and will cost millions of dollars. It is pointed out that the conversion of these Liberty ships to troopships has already cost more than $3,800,000 and that the time taken to complete this work was three times as long as that required to build them. They are now destined to be laid up for several more months.—*"The Shipping World", Vol. CX, No. 2,651, 5th April, 1944, p. 369.*

Ship Structural Members—Part II.

This paper is a continuation of an earlier one (*see abstract on p. 108 of* TRANSACTIONS, *July, 1940*) which gave the results of a series of tests forming part of the programme laid down by the Welding Research Council. The authors now give (*1*) the results of tests on compound girders formed of channel and bulb angle sections riveted to plating and having riveted bracket and lugged end connections of the type commonly fitted in ships; (*2*) the effects of welding, in part and in whole, the end connections of such girders; (*3*) the results of tests on a 12-in. bulb plate, *i.e.*, a bulb angle with the flange removed and web welded to plating; (*4*) the results of tests on compound girders formed of welded sections on plating, with varying face areas and thickness of plating; (*5*) the effects of varying the rigidity of the base structure; and (*6*) the amount of plating which contributes to the strength of the girders. The authors' conclusions are that a considerable reduction in scantlings can be obtained by the use of all-welded construction in girders, etc., but that care must be taken in applying this principle to structures subjected to shock or to fluctuating loads that deflection does not become excessive; until further knowledge is obtained, it seems advisable to maintain the present length-depth ratio of the girder. In the case of structures such as watertight bulkheads, whose primary function is to retain or exclude water, welding is undoubtedly a more effective medium for making a connection or joint than riveting and caulking, and it is hoped that further research both on the testing machine and on actual ship structures will provide a guide to a suitable standard of stiffness consistent with safety.—*Paper by C. S. Lillicrap, C.B., M.B.E., R.C.N.C. and C. J. G. Jensen, B.Sc., read at a meeting of the Institution of Engineers and Shipbuilders in Scotland on the 7th March, 1944.*

Novel Features in Design of American Post-war Ships.

It is reported that the Black Diamond Steamship Corporation, who used to maintain a service between U.S. ports, Belgium and Holland in pre-war years, intend to operate an express cargo-passenger line between New York and Antwerp after the war. The New York *Journal of Commerce* recently published some details of a design of vessel which the company are said to be desirous of building for this special purpose. The main dimensions are: length o.a. 489ft. and length b.p. 470ft.; beam 63ft.; depth 37ft.; and draught 27ft. at a displacement of 14,000 tons. The block coefficient is 0·615. The total cargo capacity is 559,000 cu. ft. (bale) and 602,000 cu. ft. (grain), and there is accommodation for about 50 passengers. The propelling machinery consists of cross-compound turbines with reduction gears, supplied with steam at a pressure of 400lb./in.^{2} and 715° F. total temperature by watertube boilers located above the turbines—an arrangement which increases the cargo-carrying capacity of the ship by 75,000 cu. ft. The service speed would be 18 knots, with a maximum of 20 knots. All the auxiliaries are electrically operated and one of the ship's generators is driven off the low-speed pinion shaft. Permanent steel grain fittings are provided in Nos. 2, 3, 4 (deep tank) and 5 lower cargo holds, the middle sections of these fittings in way of the hatches being portable. Up to 4,280 tons of grain can be carried in the above spaces. Patented fastening devices in the upper and lower 'tween deck spaces allow the carriage of about 140 unboxed motor vehicles. Cargo separation fittings are also provided in these spaces. A mechanical ventilation system for all the cargo spaces is to be installed, the air ducts being built on the bulkheads and carried down to the bottom of each compartment. The hatch to No. 3 cargo hold and lower 'tween deck is trunked through the insulated upper 'tween deck. The refrigerating machinery is to be fitted in the latter, between the forward bulkhead and the trunk hatch. Cold air drawn from the refrigerated space in the upper 'tween deck and re-circulated, will be introduced into the lower 'tween deck and lower cargo hold spaces by air ducts, thus making these compartments suitable for the carriage of perishable cargo not requiring refrigeration. The No. 3 lower 'tween deck and lower cargo hold can also be used for general cargo and grain. The president of the Black Diamond Line has declared that ships of the above design "are the only class of vessels offering much hope of profitable operation in the face of foreign competition".—*"The Shipping World", Vol. CX, No. 2,652, 12th April, 1944, p. 392.*

Conversion of the "Mormacmail".

Some details of the first conversion of one of the U.S. Maritime Commission's C-3 type cargo liners into an aircraft carrier, have recently been released. The vessel in question was the "Mormacmail"—now known as the U.S.S. "Long Island"—and the conversion work was carried out by the Newport News Shipbuilding and Dry Dock Co., who actually began the preliminary design work necessary before the ship was brought to their yard in March, 1941. The work involved included the removal of part of the superstructure and all the original cargo-handling equipment, the installation of a flight deck for aircraft landing and a hangar deck for aircraft stowage. Additional electric generators were fitted, practically all the existing electric wiring was removed and complete new systems were installed. Petrol tanks and piping, an aircraft lift from the hangar deck to the flight deck, arresting gears and barriers were also installed. Suitable living accommodation for a large complement of naval personnel had likewise to be arranged for. The entire work was performed at such a speed that the completed aircraft carrier was able to leave the yard 77 days after her arrival there.—*"The Syren", Vol. CXCI, No. 2,486, 19th April, 1944, p. 78.*

Foundry Work at Cape Town.

War-time conditions have made it necessary to carry out many repair jobs at South African ports which would have been sent to Europe in times of peace. Because of this Gearings (1878), Ltd., of Cape Town, the oldest and largest firm of marine engineers and ship repairers in the Union, have had to cast, machine and fit a complete cylinder and valve chest, weighing over 11 tons and representing the heaviest job of its kind ever undertaken in the engineering history of the country. Another big job was the making of a 2½-ton Diesel-engine cylinder liner, which was cast, machined and fitted. Several large centrifugal pump casings have been made, as has a 10-ton flywheel. Rudders have been manufactured for 10,000-ton ships, one 9-ton rudder being made in five days! Among the ship repair jobs carried out by the firm was the complete rebuilding of the stern of a 10,000-ton ship, entailing the fitting of a new rudder-post and rudder, a new tail-end shaft and a new propeller. The entire job was carried out on a floating pontoon, as the extent of the work involved and the time required for its completion made it undesirable to have the graving dock monopolised by one ship for so long. Complete new bows have been built on a number of vessels which had suffered damage in collision or by enemy action. The ships dealt with in this manner ranged up to 18,000 tons in register, so that it is clear that the construction of large ships is not beyond the capacity of the Union's technical resources, although it might not be economic. The firm also converted a large passenger liner into an armed merchant cruiser, complete with offensive armament. An armed merchant cruiser which had suffered damage in an engagement was repaired and thoroughly refitted. An unusual job was the straightening of the 12-ton bronze propeller of a 10,000-ton ship by putting it under a 200-ton press. This particular job was the first of its kind to be tackled in the Union, and its successful completion put another ship to sea, for there was no spare propeller available, and one could not have been made in South Africa. Apart from their marine activities, the firm have, for the past 40 years, specialised in the production of windmills, of which a large number are in operation in all parts of South Africa. The firm recently established a new factory devoted solely to the making of windmills. All the materials for this factory were produced in the Union, existing machinery being adapted for the purpose.—*"Foundry Trade Journal", Vol. 72, No. 1,439, 16th March, 1944, p. 224.*

Electric Arc Welding.

The British Corporation Register of Shipping and Aircraft have issued a publication entitled *Guidance Notes on the Application of Electric Arc Welding to Ships' Structures,* which is primarily intended for the use of their surveyors, but which also contains valuable information for shipyard managers, shipowners and their superintendents. The management committee of the Corporation recently approved certain amendments to *Section 24*—electric arc welding—of the rules for the construction of steel ships and a revised list of electrodes and processes approved for use in shipbuilding. In the guidance notes now published attention is called to a number of features which have sometimes given rise to trouble, particularly in ships of or above 400ft. in length, or where thick material is used, and some suggestions are made concerning preventive measures. It is pointed out that a structure may prove unsatisfactory in spite of sound welding, but that a structure cannot be satisfactory if the welding is unsound. One of the greatest dangers in welding is that wrong methods can often be adopted without producing any immediate drastic consequences, but that these follow when some serious failure occurs. This may be due to neglect of proved precautionary measures and lack of attention to the application of sound rules and principles. It is strongly emphasized that faulty welding is more dangerous than defective riveting, and that unless welding is carried out to a correct design with the proper equipment, controlled procedure, proper sequences and under proper supervision, it is better not to use welding for the main hull structure. The importance of climatic conditions must also be borne in mind. High winds may have a very detrimental effect on welds made under conditions where no effective screening is provided. Welding during frosty weather may occasion trouble if proper precautions are not taken; without considering the question of preheating, an effective measure, especially with thick plating or tied welds, is to fire the plates—as was common practice at one time—after sprinkling them with sawdust and inflammable oil.—*"The Syren", Vol CXC, No. 2,481, 15th March, 1944, p. 423.*

"Flame Cutting in the Shipyard".

A paper bearing the above title was presented at a recent meeting of the Institute of Welding by Mr. G. M. Boys, who explained that the development of electric welding in shipyards has stimulated the use of flame cutting to such an extent as to make the process an important factor in production. In American yards flame cutting has almost eliminated mechanical cutting, mainly because the plant and labour are available, and that the process is extremely suitable for the preparation of edges for welding. The degree of accuracy required for welded construction is more readily attainable by flame cutting than by any other process, and the equipment now available and in use in this country constitutes a marked advance on that common before the war, when flame cutting played a relatively minor part in shipbuilding. The author recommends that in a shipyard, flame cutting should be divided into two classes—blacksmith work, comprising such items as eye-plates, derrick fittings, fairleads, bollards, etc., and platers' work, such as floors, intercostals, shell, deck and bulkhead plates with their stiffening members. When the volume of work warrants it, the ideal arrangement is to have the gas laid on, which means using oxygen supplied in liquid form and conveyed from a central point by pipes to the individual machines. For the fuel gas, coal gas is convenient and cheap, can be readily distributed, and is preferable under present conditions to acetylene, which is made from imported materials. Moreover, it has been found that oxy-coal gas cutting is cheaper than oxy-acetylene, oxygen being the principal item of cost. Since the cost of labour has not a determining influence on the total cost of cutting, and increasing the speed of cutting only reduces labour at the expense of oxygen and fuel gas, it is not generally economical to aim at a mere increase in the speed of cutting. The supports for the work in the cutting shop should be rigid and easily replaceable, and must offer as little interference to the work as possible. The underside of the work must also be readily accessible to permit of inspection and to facilitate the clearing away of scrap and slag. Machines are now available for all classes of work, and vary from the small portable power-driven types, running either on rails or directly on the work, to the large planing machines designed to deal with the edges of large shell or deck plates. As with welding, distortion problems arise with flame cutting, but, if proper precautions are taken, the inaccuracies due to this can be kept within very close limits, and are not, in general, troublesome in shipyard work.—*"Fairplay", Vol. CLXII, No. 3,172, 24th February, 1944, pp. 322 and 324.*