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Recent Developments in Marine Propelling Machinery.

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READ

On Tuesday, September 29, *at* 6.30 *p.m.*

CHAIRMAN: JAS. CARNAGHAN (Vice-President).

THE past ten or fifteen years have witnessed many changes and considerable development in all branches of engineering. Marine engineering, distinguished for a conservative as well as a progressive attitude in the past, has shared inevitably in these changes too. The engineer who went to sea twenty years ago knew what type of machinery he would have to handle, for the steam reciprocating engine, in one or two variants, was in universal use. In twenty years from now it is difficult to say what type of machinery will predominate, but more than one type will certainly be in use. The future, however, depends upon the present and to the present we shall mainly direct our attention this evening.

Means of Propulsion—The Screw.—The screw propeller is almost universally in use as the means by which the power of the propelling machinery is transformed into propulsive effect. Strangely enough, too, it has changed very little in form with eighty years' use. Any improvements that have been effected are due mainly to rational design resulting in the adoption of better proportions; and to better manufacture.

From time to time, however, many proposals have been made to improve the type of screw propeller and two of these may be referred to briefly. One is the shrouded propeller and the other is the Star Contra propeller. The shrouded propeller is not a new idea, for good results were obtained from small diameter screws fitted in a cylindrical casing on the torpedo boat *Lightning* in 1881.

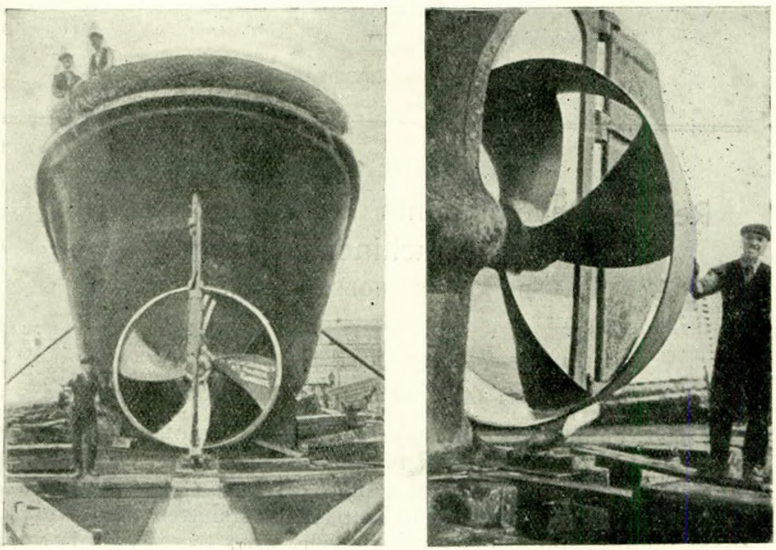


Fig. 1.—Gill Shrouded Propeller.

The propeller shown in Fig. 1 has been developed after careful investigation, and promises some measure of advantage over the plain propeller. It has not been tried in the largest sizes yet, but it has been made and used satisfactorily in a number of small vessels. If a propeller of this form can be shown to give at 20% higher revolutions, an efficiency equal to a plain propeller, then its adoption in many cases will be an advantage, for as will be seen later, high revolutions are the pathway to low initial cost. On the other hand, should this shrouded propeller require a smaller diameter to give the same efficiency as a plain propeller, the advantage will in several instances be well worth having, for there are a number of ships whose draft of water limits the size of propeller that may be fitted.

In the other type referred to, and illustrated in Fig. 2, claims for an improved efficiency of about 10% are put forward, which, if substantiated in a number of other ships, will be worth having too, in spite of the disadvantages of fixing "stars" to the rudder post. It is probable that both of these types of propeller will have a restricted range of application, as in the larger sizes one doubts if they would stand up to rough weather conditions.

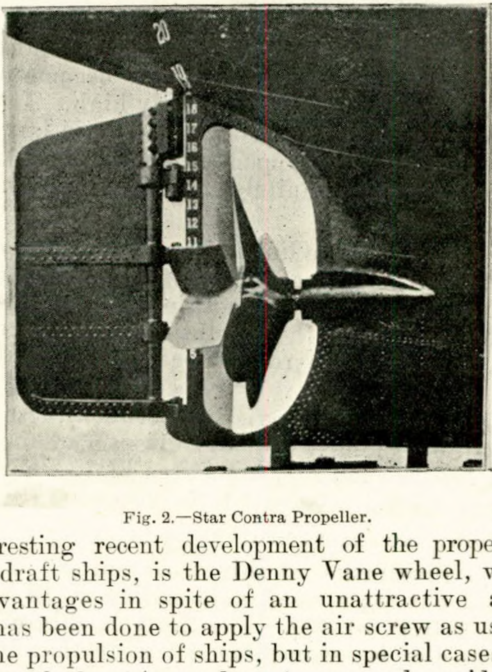


Fig. 2.—Star Contra Propeller.

*An interesting recent development of the propeller applied to shallow draft ships, is the Denny Vane wheel, which offers definite advantages in spite of an unattractive appearance. Not much has been done to apply the air screw as used on aeroplanes to the propulsion of ships, but in special cases this forms a practical method. †A word or two may be said in passing, upon the use of jets for ship propulsion. Inherently this is an inefficient method, but there are cases where, as in very shallow draft ships, the system has all the advantages. Its simplicity and the ease and rapidity of reversing the propulsive effort on the ship commend it. In large ships it has little or no prospects of ever being adopted, but for ships sailing in weedy waters, it forms almost the only possible means of propulsion.

* Note also the proposed Midship Propellers, by Ing. G. de Meo., to eliminate vibration.

† Paper No. 21. Transactions, September, 1890.—J.A.

Choice of Propeller Revolutions.—Before passing from this introductory portion of the subject, it is desirable to indicate briefly the effects of the propeller upon the propelling machinery.

The propeller must in every case be made to suit the ship to which it is to be fitted. It must first of all be able to deliver the thrust necessary to propel the ship at whatever speed is required or decided upon. To do this, it must be of a certain diameter and pitch corresponding to the number of revolutions per minute at which it runs. The larger the propeller diameter is, the more efficient it is , but nevertheless it is desirable that its diameter should be something less than the draft of the ship, so that it may be reasonably well immersed in water at all times. Propellers seldom exceed 20 or 21 feet in diameter, for when they do, the machinery revolutions become very low and the machinery weights are consequently high. In other words the machinery limits the propeller diameter in large ships, and quite often the opposite happens in small ships, the diameter of the propeller being definitely limited by the available draft of the ship.

Assuming then that the limitations of propeller diameter are defined and that the thrust horse-power required from the propeller is fixed to give a certain ship a certain speed, it then becomes necessary to fix the revolutions at which the propeller turns and from that the revolutions of the main propelling engines.

The general relationship existing between these factors may be illustrated by Table I, which has been made up for a vessel of 500 feet length, and a speed at sea of 18 knots, for which an E.H.P. of 8,100 is assumed to be necessary.

Fig. 3 shows the same results in the form of a curve.

In further illustration of this, a second set of figures have been worked out for a small ship sailing at 10 knots and requiring an E.H.P. of 300. The draft of this ship is 11 feet and the propeller is assumed at not more than 9 feet in diameter. Table II and Fig. 4 show these figures. It is to be noticed that owing to the limitation in propeller diameter, the minimum value of the H.P. curve is well along the revolution scale, particularly for the single screw ship.

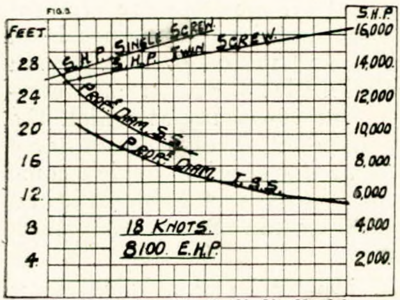
Before passing away from this section, reference may be made to the special case of fast ships where minimum weight of machinery is of the utmost importance and where in the ordinary course the high ship speeds are favourable to high rates of propeller revolutions. It is found in such cases that the ordinary propeller law, viz., Horse Power varies as $(\text{revs.})^3$ or Horse Power x Speed varies as $(\text{revs.})^4$, holds for a time

TABLE I.

18 KNOTS – 8,100 E.H.P.

	REVS PER MINUTE	SHAFT HORSE POWER	PROPELLER DIAMETER
SINGLE SCREW	60	13,320	29'-3"
	80	14,000	25'-0"
	100	14,620	22'-0"
	120	15,050	20'-0"
	150	15,900	17'-9"
TWIN SCREW	80	13,410	21'-0"
	100	13,780	18'-10"
	120	14,170	17'-0"
	140	14,550	15'-7"
	180	15,300	13'-6"
	200	15,530	12'-10"
	250	16,280	11'-5"

Note how power required depends upon Revolutions chosen, and propeller diameter.



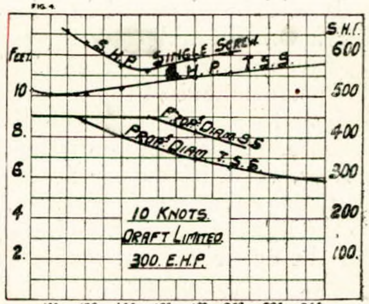


Fig. 3. Fig. 4.

Relation between Revolutions per minute of Propeller and horse powers required; also propeller diameter.

and then a change point occurs as the power increases. This point is usually termed the cavitation point and is of special importance in such cases, for it limits the revolutions that would otherwise be chosen. Some typical results of ships' progressive trials are shown (plotted on logarithmic scales) in Figs. 5 and 6.

TABLE II.

10 KNOTS - DRAFT LIMITED - 300 E.H.P.

	REVS PER MINUTE	SHAFT HORSE POWER	PROPELLER DIAMETER
SINGLE SCREW	110	654	9'-0"
	120	623	9'-0"
	140	576	9'-0"
	154	565	9'-0"
	180	591	8'-3"
	200	609	7'-8"
TWIN SCREW	90	511	9'-0"
	100	505	9'-0"
	114	508	9'-0"
	120	511	8'-9"
	140	523	8'-0"
	180	548	6'-10"
	200	560	6'-6"
	250	580	5'-10"

Note how power required depends upon Revolutions chosen, and propeller diameter.

It will be observed in Fig. 5 that all the lines are straight for a considerable length, but that certain of the lines bend upwards at the upper end. This bend indicates a change in the performance of the propeller. In Fig. 6, similar characteristics are apparent in the lines, but the point at which the change occurs is more distinctly marked. The values of horse power, revolutions per minute, and the speed at which some of these change points occur, are tabulated in Table III, and to the figures given have been added certain recognised criteria for the occurrence of cavitation.

The latest American cruisers, it may be mentioned, have been designed with their propeller revolutions so well up to the limiting safe value that the results available indicate that their propellers are just reaching the cavitation point at 33 knots, and this in spite of the fact that mechanical reduction gearing has been adopted. In other words, the designer has been pushed to the utmost safe limit of revolutions by the call for high power on minimum weight.

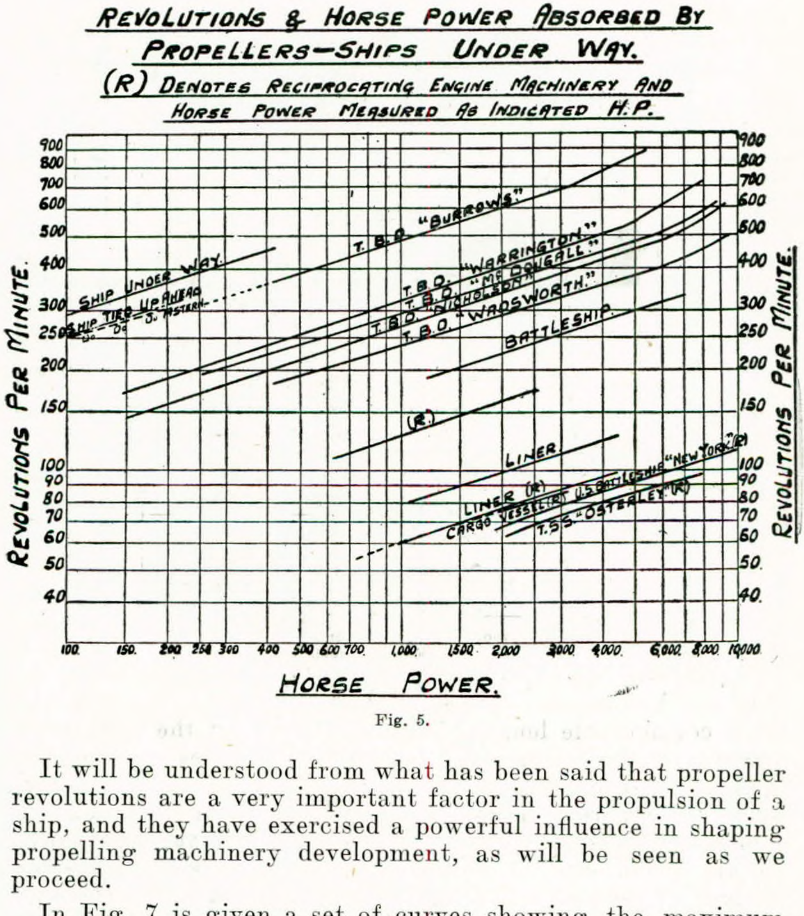


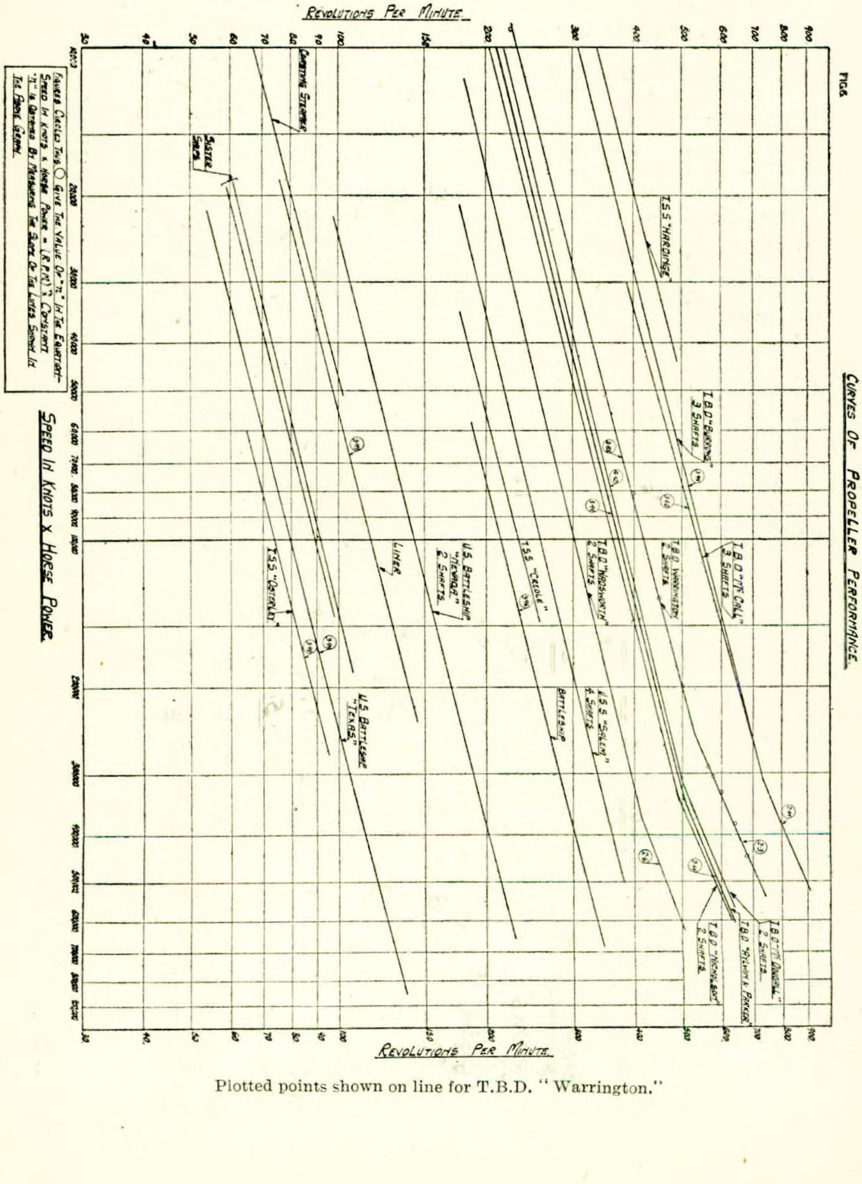
Fig. 5.

It will be understood from what has been said that propeller revolutions are a very important factor in the propulsion of a ship, and they have exercised a powerful influence in shaping propelling machinery development, as will be seen as we proceed.

In Fig. 7 is given a set of curves showing the maximum desirable revolutions that should be chosen for propellers fitted in single screw ships.

Reciprocating Steam Engines.—This is the best known type of marine propelling engine and, 20 years ago, it was practically the only type to be taken into account. It passed through the stages of compound, triple and quadruple expansion from

1860 to 1890. Compound engines are still used in the smallest boats, but quadruple expansion engines are seldom constructed now, though at least one notable company still pins its faith to this type.



Plotted points shown on line for T.B.D. "Warrington."

TABLE III.

Propeller Particulars						Critical Point in Propeller Performance.						
Ship	Diam.	Pitch	Proj. Area.	Developed Area	No. of Shafts.	S.H.P. per Shaft	Speed	Revs	Prop.r Tip Speed	$\frac{\text{S.H.P.}}{\text{Speed} \times \text{Proj. Area}}$	Speed of Prop.r — Speed of Ship	$\frac{\text{Revs} \times \text{Pitch}}{\text{Speed}}$
T.B.D. "Burrows"	5'-3¼"	4'-10"	13·63	15·26	Three	3,530	28·4	735	Ft./Sec. 202	9·14	11·0 Ft/Sec	125
T.B.D. "McCall"	5'-3¼"	4'-10"	"	"	Three	3,530	28·7	735	202	9·04	10·5 "	124
T.B.D. "McDougall"	7'-4	6'-8"	25·4	28·69	Two	5,600	27·4	500	192	8·06	9·2 "	122
T.B.D. "Nicholson"	7'-8½"	6'-8"	28·2	31·5	Two	6,080	26·5	490	197	8·13	9·6 "	123
Alwyn & Parker	7'-8½	6'-8"	"	"	Two	6,220	27·43	508	207	8·04	10·0	123·5
T.B.D. "Warrington"	6'-8"	6'-2·1"	20·41	23·17	Two	4,520	26·8	530	184	8·63	9·1 "	122·5
T.B.D. "Wadsworth"	7'-7⅝	8'-7·5	25·42	29·54	Two All Geared	6,200	28·5	412	164	8·55	9·0 "	124·3
Cruiser "Richmond"	11'-4"	12'-0"	61·32	69·62	Four Geared	About 18,450	About 33·2	About 336	199	9	11·1 "	121·4

Cavitation point for some Propellers.

At the termination of the war there was a swing over towards geared turbine machinery, but since that a distinct reaction has occurred, and certain superintendents have been frightened back to the well tried and cheaper reciprocating triple expansion engine. The advent of superheated steam has helped the

FIG. 7.

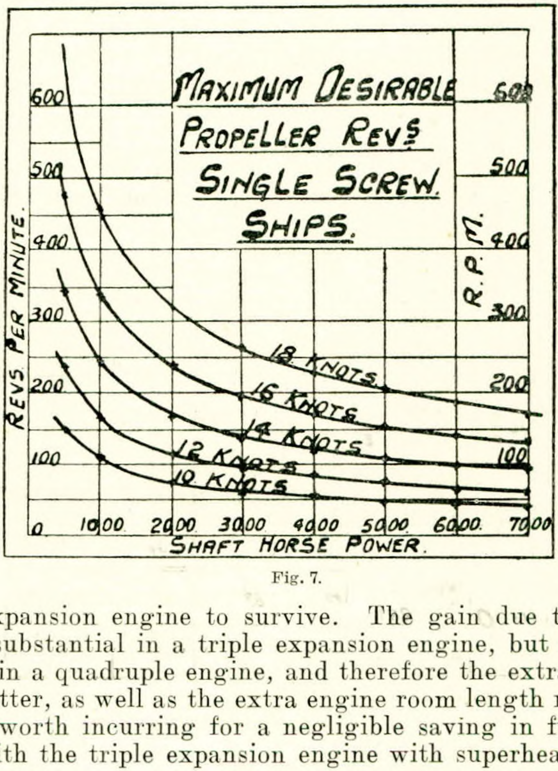


Fig. 7.

triple expansion engine to survive. The gain due to superheat is substantial in a triple expansion engine, but not very marked in a quadruple engine, and therefore the extra weight of the latter, as well as the extra engine room length required, are not worth incurring for a negligible saving in fuel compared with the triple expansion engine with superheat.

The triple expansion reciprocating engine is not passing without a struggle. The design of valves is being improved, and the steam passages are being made shorter and more direct. Improvements in piston packing rings are increasing the heat and mechanical efficiencies and reducing the wearing of the cylinders. One cannot foresee much further improvement in the steam engine, except in the adoption of the uniflow principle. In this case the complete expansion of the steam takes place in one cylinder. All pistons and cylinders of an engine would in that case be of the same size and would lend themselves to cheaper production methods than at present, but in other respects such a type would be more expensive to construct and the long piston would call for better workmanship than suffices at present. A four-crank compound engine in which cam-operated poppet valves are used, has been put forward and actually fitted. The fuel consumption claimed for this type of engine equals that of a geared turbine installation, so that it appears a very attractive proposition, but the behaviour of poppet valves under continuous sea-going conditions forms an interesting field of speculation for marine engineers.

Table IV shows some results of ships fitted with steam reciprocating engines. These are good average results, and from them the improvement produced by the use of superheated

TABLE IV.

Ship	Type of Machinery	Dispt.	Speed	Ind. Horse Power	Coal per Day Tons	Coal per I.H.P. per Hour	Coal Coefft.	Admt. Coeff.
1885	Triple Expansion Saturated Steam		12·25	1,650	34·45	1·95		
1916	Triple Expan. Saturated	3,927	10·94	1300	24·31	1·75	13,450	251
	Superheated	3,945	11·56	1,440	22·3	1·45	17,300	269
1920	Triple Expan. Saturated	12,000	12·63	3,896	70·2	1·7	15,090	271
	Superheated	12,265	12·55	4,090	60·5	1·37.1	17,535	258

NOTE.—Admiralty Coefficient =

$$\frac{(\text{Ship Displacement})^{\frac{2}{3}} \times (\text{Speed in Knots})^{3}}{\text{Horse Power.}}$$

Fuel Coefficient =

$$\frac{(\text{Ship Displacement})^{\frac{2}{3}} \times (\text{Speed in Knots})^{3}}{\text{Fuel in Tons per 24 hours.}}$$

steam will be seen. These results are specially interesting because various elements, whose effects are usually indeterminate, are eliminated by the comparison being made in the same ship. The improvement which is shown may therefore be ascribed entirely to the use of superheated steam. The progress made between the years 1885 and 1916 may be noted in passing.

In spite of this there are some superintendents who still adopt a conservative attitude to the use of superheated steam. If, however, the machinery receives the attention it merits reliable service results indicate that the use of superheated steam is fully justified.

Boilers.—Very little change has taken place in the design of boilers or in the type of boilers favoured for marine use. The Scotch or Multitubular boiler is very largely used. The main reason for its popularity is its robust construction—it cannot be made in any other way—and the fact that it lends itself to

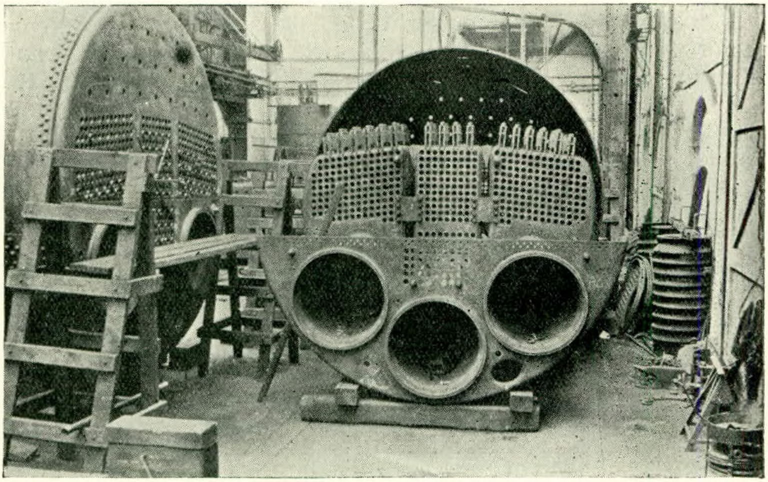


Fig. 8.—A Marine Cylindrical Boiler under construction.

repeated repair. It has many faults, but some redeeming features, and these have been sufficient to account for its remarkable popularity. The cylindrical boiler lends itself to the installation of superheaters and the latest developments in the preheating of air promise to render it still more efficient. A good efficiency, with coal burning, under Howden's ordinary system of forced draught, is about 70%. The latest experiments that have been quoted by Mr. J. Howden Hume give, for oil burning, with preheated air, efficiencies of 85% and 88%. If it is found under service conditions that efficiencies approximating to these can be obtained, then the extra complication and cost would appear to be justified. It remains to be seen if any disadvantages are revealed in service under reasonable sea-going conditions.

For all ships where light weight of machinery is required, this type of boiler is entirely superseded by the water tube boilers of Yarrow, White Forster or Babcock and Wilcox type. These have given some remarkable steaming results when fitted for oil fuel burning.

Table V shows the progress that has been made in boilers fitted on warships, where high output and light weight are of

TABLE V.

PROGRESS IN WARSHIP BOILERS.
FOR LIGHT, FAST, TYPES OF VESSELS.
FROM 1890 TO 1915.

YEAR	1890	1895	1900	1905	1910	1920
TYPE OF BOILER	LOCO-MOTIVE	WATER TUBE	WATER TUBE	WATER TUBE	WATER TUBE	WATER TUBE
KIND OF FUEL USED	COAL	COAL	COAL	COAL	OIL	OIL
WORKING PRESSURE	130	175	250	250	220	250
HORSE POWER PER BOILER	1,000	1,040	1,700	2,200	2,500	9,000
HEAT^G SURFACE PER HORSE POWER	1·75	2·05	1·8	1·8	1·4	0·88
APPROX. HORSE POWER PER TON OF BOILER R^M MACHINERY INCLUDING WATER	39·5	65·5	85	85	93	140

the greatest importance. The growth in size of boiler, measured by heating surface, is very noticeable, and the effects of changing from coal to oil are to be noted in:—

(a) Size of boiler;

(b) Heating surface per horse-power;

(c) Weight per horse-power.

There is a growing realisation that cleanliness in engineering work is as necessary as in the human machine, and when once boiler waters are kept really clean and are freed from air by methods now available, marine engineers will get over their traditional dislike to the water tube type of boiler.

Internal Combustion Engines.—Oil engines are claiming a great deal of attention at the present time, and up till now they are of the reciprocating type. Reciprocating engines have served very well in the past, and are serving well at present, but a reciprocating movement is not by any means an engineering ideal, though nature has used it very widely.

The development of the internal combustion oil engine for marine use is of recent date. Prior to 1914, there was a distinct leaning towards this type of prime mover, but several disappointing experiences delayed its successful development about that time. Then the war intervened, and except for submarine purposes, the internal combustion engine was not pushed. Since then there has been remarkable development and the engineers of this country have not been behind-hand. The four-cycle engine has attained greatest popularity, or at least it has been most frequently fitted on board ship up to the present. Its design and construction are simpler than the two-cycle engine, and rightly or wrongly it has acquired a reputation for greater economy. Many of the previous failures with oil engines arose from the tendency of the builders to over-rate the engines. This was a natural enough inclination when one considers—

(a) the cost of the engine and the desire to get the maximum return from it; and

(b) the ease with which overloading may be carried out.

On the other hand, the success of the well-known and extensively fitted Burmeister and Wain engine has largely depended upon the precautions which the builders have taken to ensure a low rating of their engine in service, and to this end their use of the term "indicated horse power" has been a distinct commercial asset.

In steam engines the ratio of brake horse power to indicated horse power is practically constant for all sizes and types, but the ratio applicable to one oil engine is no guide to that of another. The mechanical efficiencies vary with the cycle adapted and also depend upon the auxiliaries driven by the oil engine, for as will be readily understood, high pressure air compressors or scavenging air blowers consume a considerable amount of power. As a result the mechanical efficiencies of oil engines presently on the market vary from about 72% in one type to 90% in another. That means that the rating of oil engines by indicated horse power instead of brake horse power is misleading. After all, to the marine engineer it is not the indicated power but the brake horse power which is the power delivered to the propeller, that is of practical significance.

The conditions existing in an oil engine cylinder are very arduous. The cylinder becomes a furnace, but remains the working surface for a piston. The combustion does not take place continuously, but is continually interrupted and recommenced. The pressure conditions are much more severe than are ever experienced in boilers and yet the materials most suitable for withstanding high pressure and temperature are not well suited for rubbing parts. We are all familiar with the internal combustion engine in small sizes. The internal conditions are similar to those described, but their smallness makes a great difference in the problems presented. It has already been shown that the revolutions of the propeller are very important in marine work, and if direct coupling of the engine to the propeller is adopted, the engine speed must always be low relatively. To develop large powers, it follows that a large number of cylinders must be fitted or the dimensions of the cylinders made large. The former is a choice no one would willingly take if the other alternative carried no disadvantages, but the most difficult constructional problems arise as the cylinder dimensions increase. As a consequence, the demands in marine work are for large sized engines, and the development of engines to meet this demand has not been an easy or remunerative task.

The development has, however, been steady, and it now seems to be possible to construct cylinders of 33 inches diameter with success. A year or two ago, engines of 5,000 H.P. per shaft were dismissed as impracticable. This year several large ships are under construction with engines exceeding this power, and the largest powered set develops 7,500 B.H.P. per shaft and 15,000 B.H.P. total. It is interesting to notice that marine engineers who have been very conservative in regard to piston speeds on steam engines are accepting speeds of 1,100 feet per minute on oil engines. This line of development has been forced upon them by oil engine builders in their endeavour to increase the power output of the engines without unduly increasing cylinder diameters, the tendency being rather in the direction of long strokes and higher revolutions.

The development which has taken place has been marked, not by any epoch-making discovery, but by gradual steps and improved design. Materials have not altered to any extent, but the quality and fitness of the material for its duty have been improved by trial and research. Cast iron is almost universally used for cylinders and pistons.

The oil engine cycle has not changed much. The air is compressed to about 500 lbs. per sq. inch, and oil is then injected and burned at such a rate as to maintain a constant pressure during injection. Two types of engine do not follow these lines—the Doxford and the Scott-Still, and both of these are of solid or airless injection type and work with compression pressures between 300 and 400 lbs. per sq. inch. Compressed air is most usual for injecting oil into the cylinders, but air at 1,000 lbs. per sq. inch is rather a danger substance, and some people object to its use on this account, though good combustion is more easily obtained with it than without it.

In the generation of power the engineers' problem is to produce a machine in which the waste of heat is reduced to the practicable minimum. In spite of all efforts to prevent it, a huge proportion of heat inevitably passes with the exhaust steam to the condenser in steam engines and to the atmosphere in the high temperature exhaust of oil engines. Attempts to conserve some of this heat have been made by compounding the oil engine, but the more general method is to use the exhaust gases for heating purposes or for generating steam in a low pressure boiler for heating and auxiliary purposes. In the Scott-Still marine oil engine, this process has been carried a step farther. It is well known that the temperature of combustion in an oil engine necessitates the circulation of cold water through the cylinder jackets and that this water then passes overboard, so that the heat which it picks up in its course is lost. No doubt the members of this Institute are familiar with the operation of the Scott-Still engine from published accounts, and know that this engine is in a limited sense a combination of an oil engine and a steam engine. The products of cylinder combustion pass through a boiler and generate steam which is usefully employed to supplement the horse power of the main engines. The exhaust gases then pass through a feed heater and thence up the funnel. On the water side, the boiler is in constant circuit when under way, with the cylinder jackets. As a result of these arrangements the Scott-Still engine has the distinction of being the most economical prime mover for marine propulsion in use at the present time, having an economy of at least 10% over the best oil engine hitherto designed.

It is, as has been mentioned, easier to construct successful four-stroke cycle engines than two-stroke cycle engines, yet the number of builders in this country who have tackled the latter exceeds the number building the four-stroke cycle engine. Curiously enough and quite contrary to the belief of many, the two-stroke engine is proving to be more economical than the other, because it has only about one half the frictional losses of the four-stroke engine. On the other hand the two-stroke engine does not give twice the power from the same size of cylinder as the four-stroke engine. It works out about 1·7 times, but even this is a very substantial difference.

It is not surprising that builders have been discontented with the return obtained from either the four or the two-stroke engine, and are trying to produce double acting engines, and with such success that powers of 1,100 B.H.P. per cylinder have been obtained. Engines of this size and type are being fitted in some of the large powered ships already mentioned.

A study of patent specifications dealing with double acting engines shows that a great deal of thought is being given to this aspect of the subject, and at the same time it makes one amazed at what is being attempted.

A full treatment of this section of marine propelling machinery is, however, out of the question this evening, and would in any case be superfluous considering the attention this Institute has given to the subject of oil engines.

Steam Turbine Machinery.—We come now to the type of marine propelling machinery which adapts itself to the greatest range of powers and is capable of giving the very largest powers yet projected. Turbine machinery is the only machinery possible to-day for the largest passenger steamers as well as the fastest naval vessels. The whole development of this type of machinery has taken place within the last 25 years.

Some Turbine Characteristics and Limits.—It is unnecessary to refer, to any extent, to the characteristic features of steam turbines or to deal at any length with their design, but a few remarks on the limiting factors in turbine design will assist in following the stages through which marine turbine machinery has passed. Two types of turbines are in use, the reaction turbine and the impulse turbine. There are various varieties and combinations of these in use, but their names and designations do not matter at this stage.

The economy of a turbine depends, other things being the same, upon the value of (Pitch Circle Diam. of Blades)^2 $\times$ (R.P.M.)^2 $\times$ No. of stages or expansions $\div$ heat units available in the steam in expanding from the inlet to the outlet pressure. As the available heat units are for the usual pressure conditions in marine work almost constant, we may leave this term out of consideration. Now the size of the turbine depends upon the P.C.D. and No. of Stages fitted, and the weight varies with the $(P.C.D.)^2 \times$ No. of Stages. It follows, therefore, that for light weight and small dimensions it is essential to keep up the r.p.m. of the turbine. In other words, as regards cost the only factor in the design and economy of the turbine which has not to be paid for, is the term $(R.P.M)^2$ and it is to be noted that this term is to the second power. Hence high speeds for the turbine are essential and, as has been seen, low speeds are required for the propeller, at least speeds relatively very much lower than are called for by the turbine.

Another important factor in turbine design is the power to be developed. There is a limit to the length of blade which may be fitted in a rotor wheel, and further there is a limit to the speed at which this limited length of blade may be run if it is to remain in the wheel. It follows, therefore, that there is a limited area available for steam to pass through the last rows of blades without incurring undue losses. For every diameter of rotor there is an appropriate maximum power and appropriate maximum revolutions. No two manufacturers may agree upon what these limiting values are, but they do form the limits in design. Of course, in special cases, one is always forced to consider what losses may be permissible at the last stages (or the L.P. end) of a turbine, and so we find that in a destroyer set of machinery 2,000 r.p.m. may be adopted to pass steam giving 14,000 S.H.P., while in a cargo steamer such a speed may be associated with 4,000 S.H.P. The fact that the latter runs all its time at this power has a predominating influence on one's decision, while in the case of the destroyer the machinery may seldom or never run at the full power for more than an hour or two. Another feature to be noted is that at the high pressure end of the turbine, where the steam volumes are small, the rotor diameter should preferably be small and the r.p.m. high, and when this can be done, the parts subjected to the high pressure may be more easily and cheaply constructed to withstand the pressure. In other words, the ideal arrangement in a turbine would be to divide it into a number of stages increasing in diameter towards the L.P. end and reducing in speed, that is r.p.m. Radial flow turbines such as the Ljungstrom, satisfy the former condition, but of course do not meet the latter condition. Turbines divided into two, three or four casings may, to a limited extent, meet both conditions.

In land practice very high blade speeds are now adopted, up to 750 and 800 ft. per second. In marine practice, so far, speeds of 500 ft. per second are seldom exceeded. In power stations on land some disastrous breakdowns have occurred involving almost complete wreckage of the machinery. Breakdowns have occurred at sea, but, so far as I know, no serious damage has been done to even the turbine casing on such occasions. As will be understood, the energy in a turbine rotor increases very considerably with increases in the peripheral speed, and the higher the rotational speeds adopted, the higher of necessity must the stresses be. Marine practice is, therefore, conservative in this respect.

Illustrations of some turbine parts are given in Fig. 9, 10 and 11.

Fig. 12 shows a set of single reduction turbines of 4,000 S.H.P., coupled to their gearing.

Stages in Turbine Development.—Marine turbines when first made, were fitted to drive the propeller shafts direct. The conditions best suited for this were ships of high speed where high propeller revolutions were permissible as well as desirable. At the same time, reciprocating steam engines were not too well suited for such conditions. Here was an opening and the steam turbine stepped into it and very soon it was recognised as a very suitable means of propulsion for destroyers, fast cruisers and fast cross-channel steamers. In several cases, large powers, as in the *Lusitania*, were called for and here again the direct turbine found an opening. Several instances followed in which direct turbines were fitted under circumstances not well suited for them, and considerable disappointment was the result. The struggle in every case was to get the required value of " K " without involving too high propeller revolutions or turbines of too large a size.

Some turbines of very large dimensions were constructed in the early stages, H.P. rotors being 96 inches and L.P. rotors being 140 inches in diameter; but even with these large sizes the propeller revolutions were as high as 275 per minute in the *Tiger*, for example. As was to be expected the construction of units of such a size was a difficult problem and warping of the casings under heat had to be provided for as far as possible. Some very large sizes of workshop machines were needed to manufacture such casings and rotors and nowadays in certain marine engine works large machines specially provided for this class of work may be seen working on parts much below their capacity.

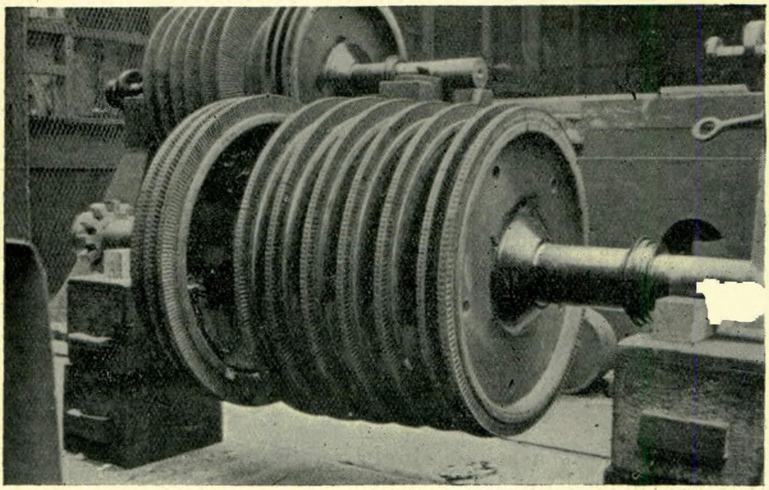


Fig. 9.—High pressure turbine rotor, 2,000 H.P. and designed for 3,500 r.p.m. Astern wheel with three rows of blades on left. Low pressure rotor in background, also designed for 3,500 r.p.m. Wheels are solid with spindle in H.P. rotor.

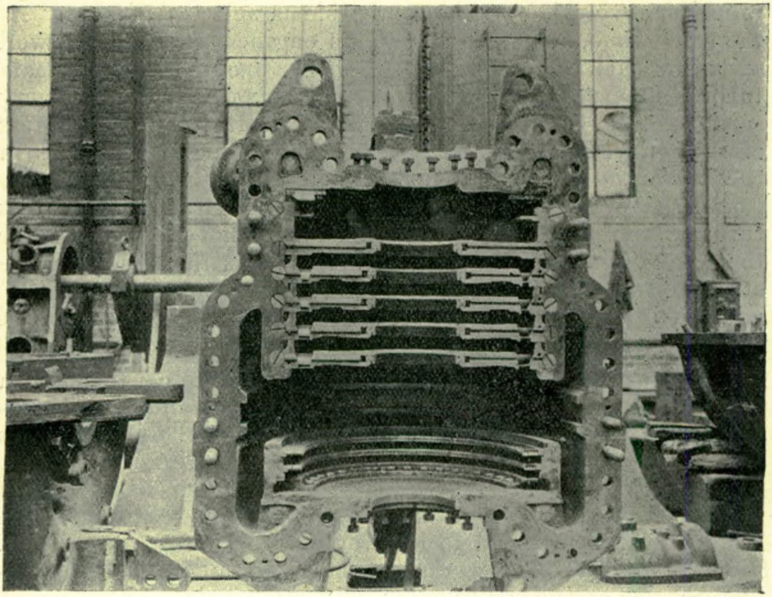


Fig. 10.—Top half of turbine casing with diaphragms in position. The upper portion is for the ahead and the lower portion for the astern turbine. The diaphragm separating the two is not in position.

The degree of accuracy in machining that was called for served a good purpose in preparing for the still higher standard of accuracy now required in turbines, gearing and oil engine manufacture.

As will readily be understood a high value of " K " was not provided in any of these direct sets and, as a consequence, economy at reduced powers was not good. In the naval ships this was a distinct disadvantage but in ships like the *Lusitania*

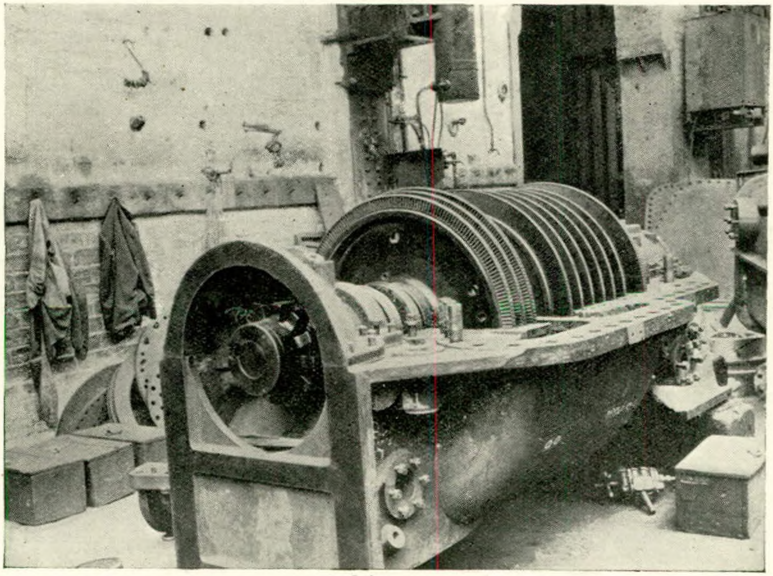


Fig. 11.—Lower half of turbine casing with rotor in position. Nearest wheel is for astern running. Part of " claw " coupling is visible.

it was of much less importance, for they run at nearly constant power. As a consequence, various arrangements of extra turbines were installed to improve the consumption at cruising speeds. They all complicated the arrangements of machinery, and increased the weight and space required. It was impossible therefore to avoid facing the question of employing some means of stepping up the turbine speed and at the same time lowering the propeller speed.

Geared Machinery.—As was mentioned, some form of speed reduction gear was urgently called for to improve the steam turbine in its application and to widen its scope of application. The schemes which have been brought to a successful issue are :—

1. Mechanical gearing.
2. Föttinger transformer.
3. Electrical propulsion.

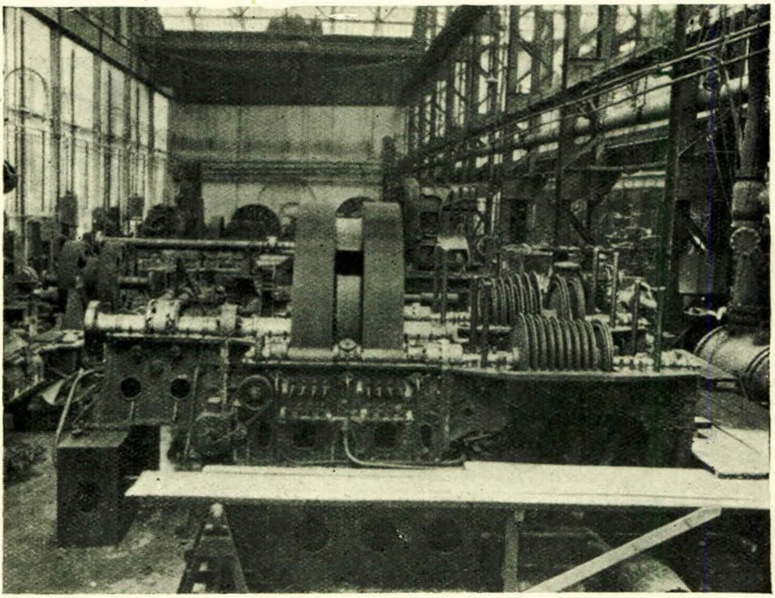
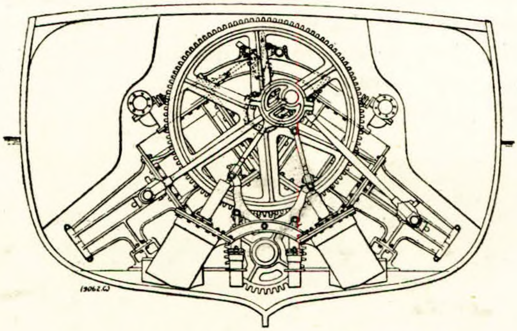


Fig. 12.—A set of single reduction gears coupled to turbines of 4,000 S.H.P. High pressure turbine is in front.

As a side light upon this point it is interesting to look at an installation of machinery by Scotts of Greenock, built in 1850. It is shown in Fig. 13. The machinery was of 250 H.P., and the ship to which it was fitted was 260ft. in length. The cylinders were 52in. dia. x 45in. stroke, and were fitted with two piston rods in each cylinder. The gear wheels were $137 \cdot \frac{1}{2}$in. and 54in. diameter, giving a propeller speed of $2\frac{1}{2}$ times the engine speed. No doubt when this particular type of machinery was superseded it was considered that gearing would never be used again for main propelling machinery, but it has been re-introduced on a greater and more extensive scale.

In the Navy, mechanical gearing was adopted by stages — first to couple cruising turbines to the main propelling turbines; second, to couple the high pressure turbine to the direct driving low pressure turbine, and then finally as the only means of drive between the turbines and the propeller.



DOUBLE-GEARED ENGINE FOR EARLY ATLANTIC LINER.

Fig. 13.

In order to make the turbine applicable to cargo ships a trial was made of an installation of geared turbines fitted to the SS. *Vespasian*, in which reciprocating machinery had previously been installed. The movement spread gradually. The first Atlantic liner to have geared turbine machinery, viz., T.S.S. *Transylvania*, was built by Messrs. Scotts' for the Cunard Co., and the SS. *Diomed*, belonging to the Blue Funnel Line, also built by Scotts, was the largest powered single screw vessel of its day to be fitted with turbines and gearing. Both ships were sunk during the war.

The application of gearing continued to increase in the Navy as well as the mercantile fleets. H.M.S *Hood* represents the latest and finest development in the Navy. In merchant work, however, it was soon found that there was a limited ratio of reduction possible, and in many cases the maximum size of gear wheel that could be constructed was being used. A large main gear wheel of a double reduction set of 6,000 S.H.P. is shown in Fig. 14.

Double Reduction Gearing.—Immediately after the war and as a result of the excellent service obtained from gears during the war, a large amount of merchant tonnage was placed where turbine machinery was adopted, in which the gear reduction was carried out in two stages, or by what is known as double reduction gearing.

Mr. Fred Samuelson, the Chief Mechanical Engineer of the British Thomson-Houston Co., of Rugby, remarked several years ago, when single reduction gears were coming into vogue and doing well, that it would be a great surprise to him if a period of gearing troubles was not experienced, for no engineering step of such magnitude had ever been taken in the past without having to pay the price for it. His words were prophetic, for the first steps into the double reduction field were disastrous to some, disappointing to others, and gave great anxiety to many more.

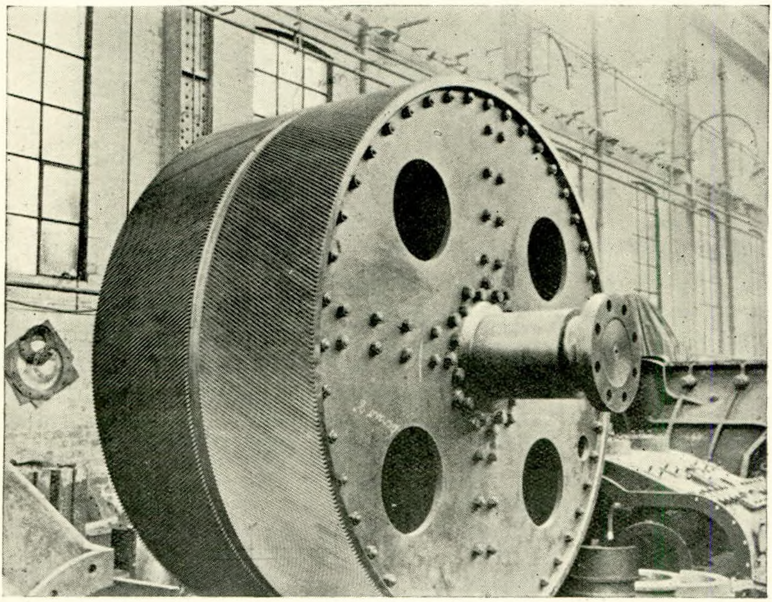


Fig. 14.—Main gear wheel for single screw ship of 6.000 S.H.P.

It was difficult to determine why this should be. The differences between single and double reduction gears were anxiously examined and compared to find out what was wrong. Metallurgists blamed the engineers, and the engineers blamed the metallurgists. Each was persuaded the other was wrong in his diagnosis, for strong evidence in support of both was forthcoming in different cases.

When the accuracy of the gear cutting began to be carefully examined, the boasted accuracy of the gear cutters was found to exist mainly on paper and in their imaginations. The impression that gears were cut to thousandths of an inch was strongly impressed on people's minds, but this was found to be anything but true. Great improvements have been made in large gear-cutting machines in the last few years, and gears cut by reputable firms can now be depended upon to be as accurate as can reasonably be looked for.

The results of a test of the accuracy of a gear wheel and one of its engaging pinions are shown in Fig. 15. The wheel was

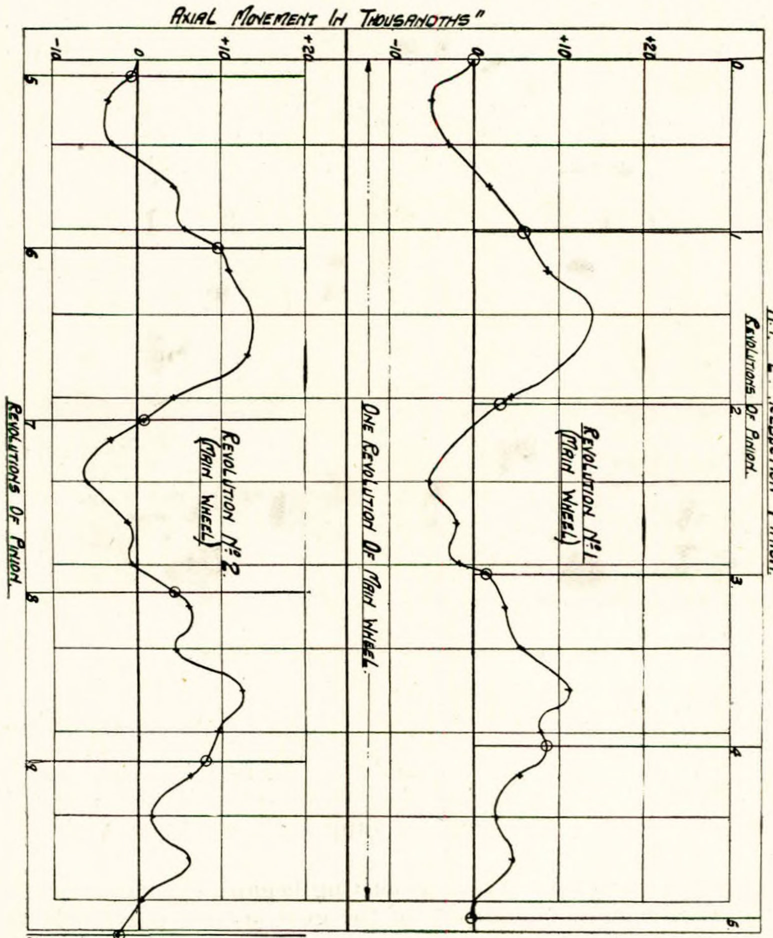


Fig. 15.—Test for accuracy in gear cutting.

slowly rotated and the difference in axial movement of wheel and pinion was noted at intervals of the revolution. Had the wheel and pinion been accurately cut, the relative axial movement would have been zero. It was, however, as shown.

Nearly all, but not all, of the first double reduction gears were noisy and generally unpleasantly noisy. Single reduction gears have not been so. In some cases wear has been very severe, in many more it has been quite normal. In a number of cases of double reduction gears, parts of teeth have broken out. Where these pieces have fallen clear no harm has resulted, but where they do not get clear a serious accident may happen. Many gears, however, have given no trouble at all.

In the writer's view, materials have been at fault, partly due to the designer's failure to allow for the difficulty of heat treating pinions of large dimensions, but principally due to the limi-

FIG. 16

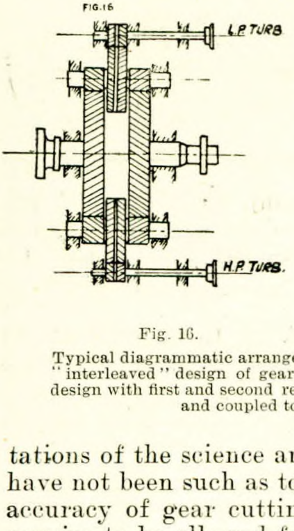


FIG. 17

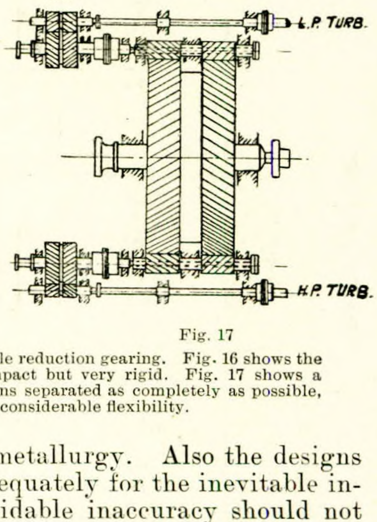


Fig. 16. Fig. 17

Typical diagrammatic arrangements of double reduction gearing. Fig. 16 shows the "interleaved" design of gear which is compact but very rigid. Fig. 17 shows a design with first and second reduction portions separated as completely as possible, and coupled together to give considerable flexibility.

tations of the science and art of metallurgy. Also the designs have not been such as to allow adequately for the inevitable inaccuracy of gear cutting. Avoidable inaccuracy should not require to be allowed for, but it is wise to provide for some of this if at all possible. Synchronism of vibrations in the parts of gearing may have accounted in one or two cases for troubles, but it is doubtful if nodalising the gearing is of any real value. Want of flexibility between the first and second reduction gears is one of the greatest defects in design that occur in double reduction gearing.

Fig. 18 shows the gear wheel and pinion of a double reduction gear. The proximity of gear wheel and pinion will be noted. In Fig. 16 a typical arrangement is shown for double reduction gearing. Notice the extreme rigidity of wheel and pinion of intermediate shaft. In Fig. 17 is shown the latest design of double reduction gear that Messrs. Scotts' Shipbuilding and

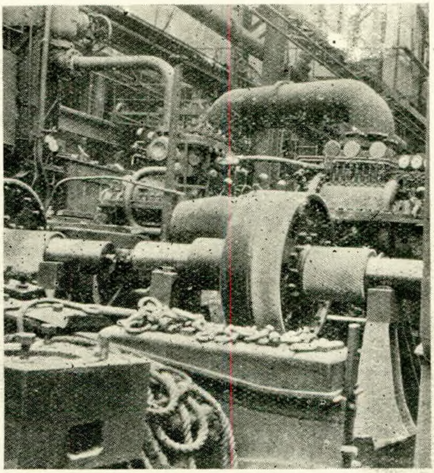


Fig. 18.—Intermediate wheel and pinions of the "interleaved" type of double reduction gearing.

Engineering Co., Ltd., have fitted to ships. Note the extent to which the two gears are made independent, and how the turbine rotors are flexibly connected to the gearing. A photograph of one set of gears constructed on this principle is shown in Fig. 19.

Results with Turbines and Gearing.—In view of what has been said, it may be asked: "Are gearing and turbines worth fitting to a ship?" It is known that some installations of turbines have not justified themselves. In some cases they have not been looked after. In others the installations have not been well designed. It has to be realised that turbines are sensitive to ignorant inattention. Leakage of air into them soon spoils the vacuum in the condenser and eats into the economy. It is easy to pass steam uselessly through a turbine if valves and seats are not attended to. With intelligently applied attention, turbine installations have given very good results, and it is the writer's opinion that double reduction geared turbines may be accepted by shipowners where the conditions are suitable without any misgiving.

Table VI. shows results for similar single screw ships—practically sister ships. The improvement in results is quite distinct. Table VII. shows results for Atlantic liners. The ships are not so closely similar as those given in Table VI. Table VIII. shows how oil and coal compare in the same ship, fitted with D.R. geared turbines and superheated steam.

Hydraulic and Electric means of speed reduction. — Time does not permit of dealing with the other methods of speed reduction which have been tried. The Fottinger transformer has been used in several cases with a fair measure of success. The *Empress of Australia* is the largest ship so far fitted with

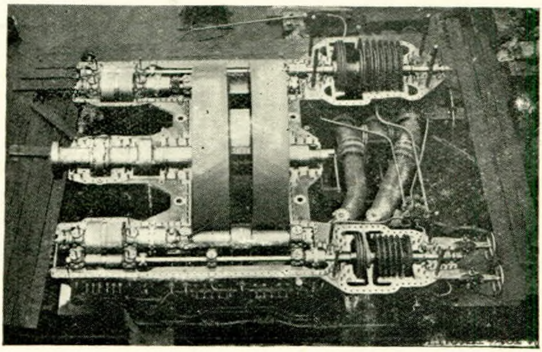


Fig. 19.—Double reduction turbines and gearing—4,000 S.H.P. at 80 r.p.m. of propeller. A small speed reduction is provided in first step and a large speed reduction in second step. Great flexibility is provided between turbines, first and second reduction gears.

this gear, but what results have been obtained are not known. This system is unlikely to be used to any extent as it provides a limited reduction ratio and is not so efficient as is desired.

Electric propulsion is a more serious proposition. It is well adapted for battleships, large cruisers and for merchant ships having large electrical installations for ship and cargo purposes. An electric propelling installation cannot be regarded as simple; it has its own peculiarities of behaviour and it is not as economical as a geared turbine installation, but it has some good features. Steam turbines have usually been the source of power but oil engines may also be utilised where suitable.

Higher Steam Pressures. — Steam pressures on board ship have advanced slowly but steadily to what they are, but in land power stations very great advances in steam working pressures have taken place in the last year or two, and pressures as high as 1,200 lbs. are actually in use. Various proposals have been made in the direction of higher steam pressures for marine propelling machinery, and it is of significance that an instalation of 4,000 S.H.P., working at 500-550 lbs. per sq. inch work-

TABLE VI.

Ship	Type of Machinery	Disp^{T}.	Speed	Revs per Min.	Shaft Horse Power	Coal per Day Tons	Coal per S.H.P. per Hour	Coal Coef^{t}.	Adm^{ty} Coef^{t}
A	Triple Expansion Saturated	12,400	13·77	74·7	5070	87·1	1·60	16,000	275
B	Single Red^{n} Turbines Saturated	12,800	13·3	88·8	3650	68·5	1·75	18,800	352
C	Single Red^{n} Turbines Superheated	14,530	13·82	111·0	5440	74·0	1·27	21,200	289
D	Double Red^{n} Turbines Superheated	13,960	14·77	93·8	5900	75·0	1·19	24,900	316

In ship A the power is indicated horse power and the coal consumption rate is in lbs. per indicated horse power. In B, C and D, shaft horse power is used. Shaft horse power is about 0·9 times the indicated horse power in steam reciprocating engines.

TABLE VII.

Ship	Type of Machinery	Disp^{T}.	Speed	Revs per Min.	Horse Power	Coal per Day Tons	Coal per H.P. per Hour	Coal Coeff^{t}.	Adm^{ty} Coef^{t}
E	Quadruple Expansion Saturated	15,940	13·95	79·0	6,880 I.H.P	132	1·79	13,100	250
F	Single Reduction Turbines Saturated	17,000	15·5	123·2	8,100 S.H.P	150	1·73	16,500	304
G	Double Reduction Turbines Superheated	14,810	14·1	78·7	6,480 S.H.P.	74 Oil	1·07	✓ 22,800 * 16,500	261

✓ For Oil Fuel; * Equivalent for Coal.

ing pressure, is now under construction at Messrs. Denny's and the Parsons Marine Steam Turbine Co., and will be fitted in a steamer to ply on the Firth of Clyde. High steam pressures are associated with the use of the water tube type of boiler and in the ship referred to, water tube boilers of Yarrow type are to be fitted. As this installation is to have stage heating and preheated air, it will prove a most interesting experiment, and the results will be awaited with the keenest interest. A paper entitled "Relative commercial efficiency of internal combustion and steam engines for high speed passenger vessels" by

TABLE VIII.

Ship	Type of Machinery	Fuel Used	Dispt.	Speed	Revs per Min.	Shaft Horse Power	Fuel per Day	Fuel per S.H.P. per Hour	Coal or Oil Coefft.	Admy. Coef.
H.	Double Reductn. Turbines	Coal	13,051.	14·93.	93·55	5,421.	76.	1·31.	24,200	340.
			14,200	13·65	85·86	4,239.	67·2	1·48.	22,250	353.
	Superht. Steam.	Oil.	13,163	14·45	91·04	5,330	52·7	·923	31,900	315
			14,200	14·77	92·15	5,516	50	·843	37,950	342

Sir John Biles, read before the Institution of Naval Architects this year deals with the question of high steam pressures. A reference to the same subject will be found in a brochure issued by the Parsons Marine Steam Turbine Co., on high efficiency steam turbines for marine work.

Auxiliary Machinery.—Developments of auxiliary propelling machinery have been continuous. It is always the case that development in one direction must be assisted by development in another. The turbine required efficient condensing plant, and as a result the design of condensers and the effectiveness of air extracting apparatus have been very much improved by comparison with what they were. In the same way, the new demands for auxiliary machinery for internal combustion engines have produced quite a variety of motor driven machines which have a general application as well. The use of electricity for auxiliary purposes has also been extended, and as a consequence a new point of view has been presented regarding the efficiency of all ships' auxiliary machinery which will more and more affect future developments.

Several recent developments have been left untouched and many things have been left unsaid, but in closing we ask, what has the future in store for the shipowner, and by what means will the world's trade and commerce be transported over the oceans of the world? The answer lies in the future, but the future is taking its course from the present.

Thanks are due to Mr. James Brown, one of your Vice-Presidents, who is managing director of Scotts' Shipbuilding and Engineering Co., Ltd., Greenock, for permission to publish the data and illustrations given in this paper.

Oil Engine Details and their Working.

The following contribution has been received from J. C. Phillips, Member:—

Introduction.—The writer, being alive to the requirements of the Diesel age in which marine engineers find themselves, believes that a simple imaginary conversation (between a second and a junior engineer) might be appreciated by many of the juniors, and in a minor way, assist those that are about to take charge of motor vessels. In humbly submitting my experience in this simple, practical paper I have in mind the fact that of all the highly technical and masterly works that form part of the Diesel engineers library, my five years' motor ship experience of six different vessels has taught me that there is still room for something which is not exactly technical but sufficiently practical to help and instruct those of the operating staff who have not had the opportunity to learn the importance of first-class workmanship and what it means to the successful running of these "very practical engines."

To my mind there is nothing gained by hiding the fact that some ships have more success than others of similar design and age. There must be a reason for this. Is it the engine or the operators? Take an example: Two second engineers of similar ships know all the details concerning their engines (auxiliary machinery in particular), why has one more success than the other? It is because the successful one understands the qualifications necessary for the efficient handling of men. He thoroughly realises that many parts peculiar to this engine call for *constant observation of detail*, and almost ceaseless attention to the engines as a whole. This enables him to gain the confidence of his staff, all of whom are quick to realise that his methods, although strict, are the keynote to successful running, with the result that a more comfortable time is spent, since good workmanship has cut down field days to a minimum.

Of course, comfortable conditions in the engine room may not exist in some ships that have been running a number of years, since these ships, being the experimental units, contained many parts of weak design which claimed and still claim much of our attention. Nevertheless these weaknesses are gradually being eliminated, so those who have helped in this direction can always look back on their troubles and trials as something well worth the effort and care expended, since they obtained what every engineer desires—experience—and this cannot be bought at any figure, it must be worked for. Perhaps some learned advocates who are inclined to treat the *operating* of this engine from a scientific standpoint will not agree with the term " A very practical engine " nor with these criticisms.

The various points which occur, experience are embodied in the following questions and answers. Here some explanation is necessary. Undoubtedly the engine is a scientific engine, but after construction, when it becomes operative in marine conditions, the only science required in the engine room is ability to overcome difficulties experienced from faulty working parts which can in almost every case be traced to inferior workmanship. This is not intended as a reflection on engine makers and contractors, although it can be truthfully said that none of us have yet reached the infallable stage.

Cylinder Cover Water Ring Joint.

Second Engineer: When you take out the brass ring, examine the outside groove to insure the metal at the top and bottom of the groove is of uniform thickness all the way round.

Junior Engineer: And what would happen if it was not of uniform thickness?

S.E.: You will see at the test. The *thinned* part is not strong enough to force the rubber ring into position and yields to the rubber ring when the brass ring is forced into position. It was formerly the practice to press this ring into position with the cylinder cover, since this method was discontinued we now know our position before a cover is put on. We have here eight shores of wood about 18in. long by 4in. square with a $\frac{1}{4}$in.

plate secured at each end. When the brass ring is thoroughly cleaned (with emery paper) both in the groove and at the angle part inside, put on the rubber rings and force the brass ring into position, using a shore at each stud and a $\frac{5}{8}$in. bolt and nut between the shores and underneath each stud nut to give the

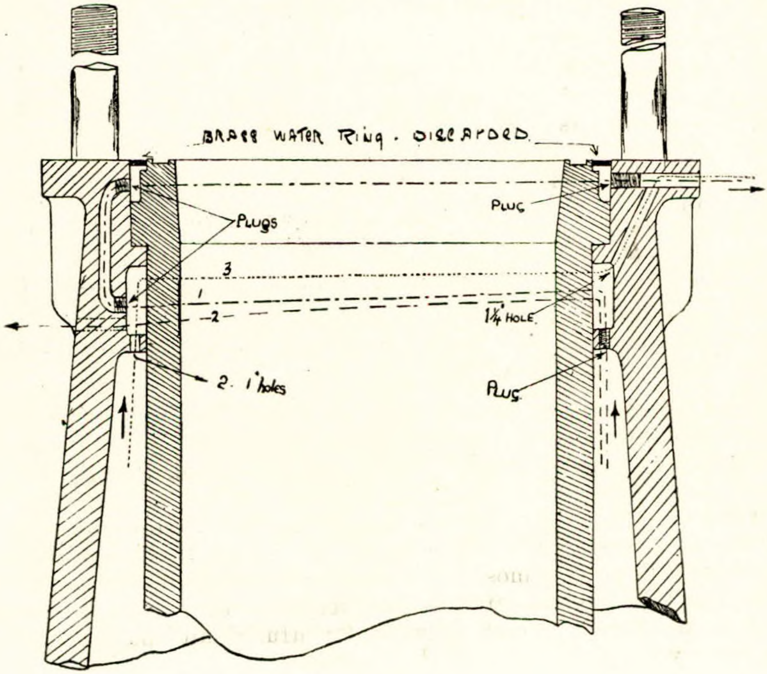


FIGURE I

Sketch of main cylinder water jacket, showing original and alternative methods of circulating water flow. Note the brass ring, now discarded.

(1) Original flow of water before brass ring was discarded.

(2) Alternative method of water flow. This method leads to the back of the cover, and promotes better circulation.

(3) Alternative method of water flow. This method leads to the front of the cover and causes faulty circulation.

desired pressure. When satisfied that the ring is properly home, put on the circulating water and test for tightness.

J.E.: The pressure is now 30 lbs. per square inch and there is a leak at the point aft on the brass ring exactly where the metal was thinned or weakened.

S.E. : Now to prove that the brass ring is responsible for the leak. Ease the water pressure, lift up the ring, and give it half a turn and force it into position again.

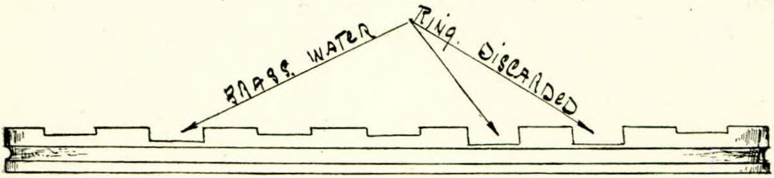


Figure (2)

This brass ring shows weak points which interfere with proper functioning of rubber ring which fits in groove. It can now be dispensed with by altering the flow of water as seen in Figure 1.

J.E. : I have done so and now find a leak forward and at the same point on the brass ring.

S.E. : Exactly. Now it is impossible to make a job with this ring.

J.E. : Why? Can't we try a quarter turn now and see if it will leak? Because it might be the badly pitted state of the cylinder jacket (against which the rubber ring fits) that is causing the leak.

S.E. : That is not so, because since the ring leaked at a point aft on the first test, but at the same point on the ring, thus it is impossible to blame the condition of the jacket surface. If the jacket surface was responsible then we would have the leak at the same point, no matter how many times we shifted the position of the ring.

J.E. : Now, what must I do to rectify this weakened ring?

S.E.: We will require to examine the spare rings and fit the best that we can find. The old ring could be re-inforced but they never make a job.

J.E. : It must be the state of all the brass rings which causes most of the water leaks.

S.E. Not in every case. Many of the rings have been quite tight but being left in position too long the rubber perished, consequently leaks occurred which were allowed to continue, resulting in pitting of the jacket surface. Small cracks on the cylinder jacket adjacent to the cover studs are also responsible for many leaks. Experience has taught us that it is not necessary to have the cooling water at such a high point on the liner. By withdrawing the liner and plugging up holes in the jacket ribs to stop the water going to the top of the liner, we can do away with the brass ring altogether. Modern covers have no such rings fitted.

J.E.: So the space originally water cooled is now open to the atmosphere?

S.E.: Yes.

J.E.: Is there more than one method of plugging these holes?

S.E.: Yes. Looking at the sketch you will notice how the water is led to the cover by the original arrangement. There are two alternative methods shown, one of which leads the water to the back of the cover.

J.E.: Does not the fitting of one or the other of these methods influence the temperature of the cylinder covers?

S.E.: Yes. When we lead the water to the back of the cover we find we get better circulation than when using the original fitting to the front of the cover.

J.E.: How do you prove this temperature difference?

S.E.: When the water is led to the *front* of the cover the temperature of the water from the small air cock adjacent to the inlet valve is from 20° to 90° more than the water leaving the exhaust valve, and when the water is led to the *back* of the cover the temperature from this small cock falls below the temperature of the water leaving the exhaust valve.

J.E.: Do you think the faulty circulation had anything to do with the cracking of cylinder covers?

S.E.: That is a question best left to the designer and metallurgist. In my opinion cracked covers are mainly caused by pre-ignition in the cylinders when entering or leaving port. It is recognised that the proper opening of the fuel valve should be 6° before the top centre (tested with blast air at 60 atmospheres). With an engine compressing up to 33—34 atmospheres when under normal load and well warmed through, and with the fuel valve opening as stated—bearing in mind of course that all other conditions are correct—we obtain a diagram the contour of which is almost perfect. But as an engine tuned thus is working under first-class conditions at sea, certain adjustments will be necessary to obviate high maximum pressures when manœuvring.

J.E.: If the engine is perfectly tuned up for sea conditions can't the same tuning do for manœuvring when we want the conditions almost perfect?

S.E.: Unfortunately no. Conditions have altered. At sea the speed of the piston is so great that the crank is over the top centre in a much shorter period of time than when starting up from rest when manœuvring in and out of port. Bearing this point in mind it follows that when starting up from rest the slow moving crank is not over the top centre when the fuel valve opens and combustion begins, therefore it is necessary for us to compensate for this by retarding the fuel valve opening. Some engineers, in place of retarding the fuel valve opening, adopt the practice of working with a low blast of about 45 atmospheres, but this is just as ineffective as it is dangerous. I have a set of diagrams here taken from one cylinder where the fuel valve opened at $5\frac{1}{2}°$ before the top centre (tested with blast air at 60 atmospheres pressure). Before taking the compression diagram the engine was set with the inlet valve for this particular cylinder just about to open, and from this position the engine was started on air.

Diagram No. 2 was taken on starting from rest at a blast pressure of 60 atmospheres, and No. 3 was also taken from rest immediately afterwards at a blast pressure of 42 atmospheres. No. 4 was taken 24 hours after the ship left port.

J.E.: These diagrams are really astonishing since they demonstrate—more so than any verbal explanation—the enormous shock that pre-ignition can create to the engines. To obviate these high maximum pressures you mention the retarding of the fuel valve openings; how is this carried out?

S.E.: By increasing the clearance at the cam roller. Some prefer to make the valve open on the dead centre, which is correct, and keep the blast air at 50 atmospheres. But it is the case in many ships that the valve opening so late cause sluggish starting, and it is to insure a good start that many engineers prefer to leave the settings undisturbed, much to the detriment of the cylinder covers, etc.

J.E.: When the fuel valves are opening on the dead centre instead of 6° before the dead centre, why should this cause sluggish starting?

S.E.: The piston almost comes to the top of its stroke about 23° before the crank is on its dead centre. It is obvious that the heat generated by the slow moving piston to ignite the charge of fuel is rapidly absorbed by the cold piston and sur-

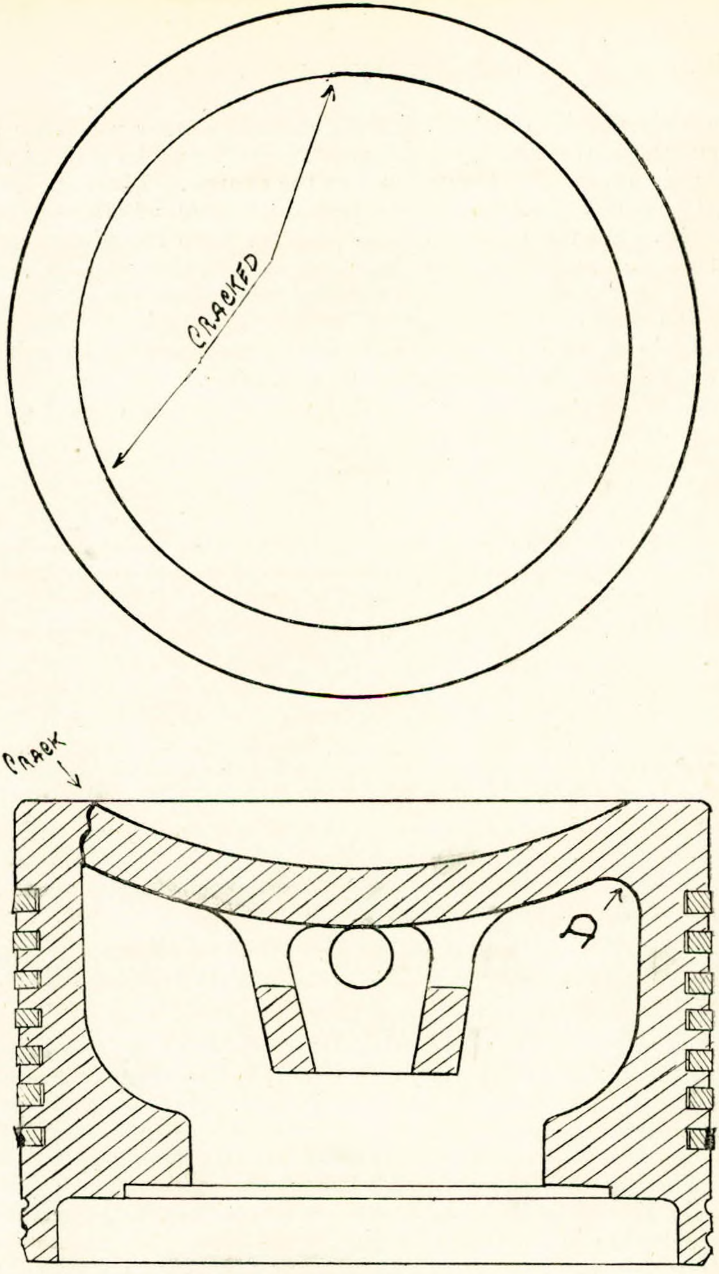


FIGURE (3)

Showing cracked piston. Note where the crack developed at the edge of the bowl or dish, and extended into the wall of the piston. For a distance of 1/3rd of the circumference (red line) there was not sufficient reinforcement as at A.

rounding walls of the cylinder, and the longer we delay the opening of the fuel valve the greater chance we have of finding the air has become too cold to fire the charge. This, of course, only applies to engines where piston clearances have been neglected. On the other hand, in engines were the piston clearance is treated with consideration no difficulty will be found in firing the charge of oil when the fuel valves are set to open on the dead centre. Diagram No. 5 was taken at starting with this setting and Diagram No. 6 during manœuvring. The blast pressure was 50 atmospheres in both cases.

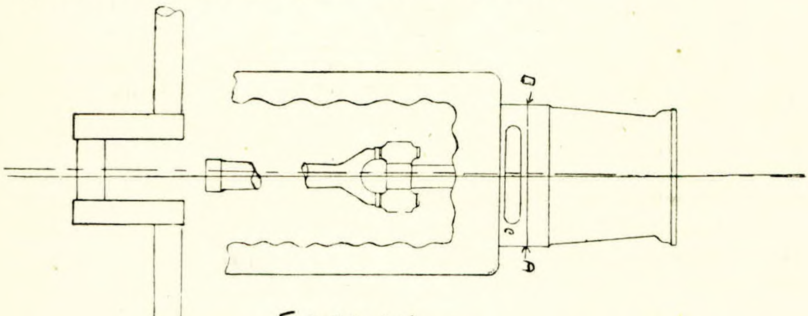


Figure (4)

Shows a cylinder out of line. (Exaggerated.) Machining the distance piece C from something to nothing in the direction from A to B is the only possible solution for true alignment. Packing with liners is an impossible method.

J.E.: I am not quite clear as to how you alter these settings, increasing the roller clearance to retard the valve opening from 6° to nothing will reduce the lift and period of opening of the valve. Won't this effect combustion?

S.E.: Yes. But not to the detriment of the engine while manœuvring. But we must try and satisfy two conditions.

J.E.: What are they?

S.E.: The particular setting referred to for sea conditions, and the particular setting referred to for manœuvring conditions. To have one we must sacrifice the other since altering the roller clearance is effective only for two or three degrees at the most. We are compelled therefor to strike a happy medium by making the sea setting 3° before the top centre, which is easily altered to the dead centre by increasing the roller clearance before manœuvring begins.

J.E.: When the valves are set for 3° before the top centre for sea conditions, do you carry the same blast pressure as you did when the setting was 6°?

S.E.: No. By increasing the blast pressure another two or three atmospheres, we obtain a diagram almost similar to our ideal.

J.E.: I am sure that anyone examining diagram No. 2 will be convinced that certain conditions can easily turn the unit into an explosion engine when manœuvring.

S.E.: Yes. That and the other diagrams certainly give us some food for thought.

Pistons.

J.E.: Does pre-ignition cause cracked pistons?

S.E.: It helps. When a piston crown cracks at the edge of the bowl it can generally be recorded as a faulty casting. Pistons have cracked at the edge of the bowl for a distance of $\frac{1}{3}$rd of their circumference. On disconnecting the rods to examine inside, these cracks were sometimes found to develop at a point where the inside wall of the piston formed a complete right angle with the underside of the crown metal. A sketch will explain matters. Cracks often develop in the centre of the bowl—termed surface cracks, since they don't penetrate the metal, and such pistons will continue to work in this state for a considerable period. I believe our troubles would be eliminated if pistons, when new, were thoroughly inspected for detail faults, such as shown in the sketch, and if a hole were drilled and plugged in the centre of the bowl to help to reduce stresses due to contraction and expansion. To insure these plugs being quite tight I would suggest the hole being screwed with a square thread, and a plug fitted with the threads a tight fit vertically but a shade slack diametrically.

J.E.: Are there any other important points to know about pistons?

S.E.: Yes. Alignment, lubrication, piston cooling and care of piston rings. It has been my experience that much trouble and annoyance is caused by inattention to cylinder lubrication. The oil should be introduced at a point between the two top rings when the piston is at the bottom centre. It is certainly no use looking at the oil globules going up the glass, when the oil that you see going up is pushing out a similar amount on to the cylinder walls which the piston, on its downward stroke, is going to push off again.

J.E.: I see quite well that oil should be introduced when the piston is on the bottom, which would trap the oil between the rings and cylinder walls, but I am not quite clear which is the next stroke of the piston. Is it the exhaust or the compression stroke?

S.E.: Compression stroke. When setting forward lubricators, proceed as follows:—(1) Fix the spur wheel in its proper place on the cam shaft without regard to the cam setting. (2) Turn the engine crank (I) 50° ahead over the bottom centre on the *compression* stroke. (3) Turn the lubricator shaft until the lubricating plungers for No. 1 cylinder are moved out as far as possible. (4) Turn the spur wheel until the pin hole corresponds, and fix it to the lubricating shaft. For the after lubricators repeat as for the forward lubricators but with engine crank (IV) and lubricating plungers for cylinder (IV).

J.E.: Does the firing order of the engine make any difference to the cam settings on the lubricating pump cam shafts?

S.E.: Yes. Assuming the engine to fire 1, 4, 2, 6, 3, 5, the forward lubricating shaft cams 1, 2, and 3 would be set at 120° to each other in that order, but the after lubricator shaft cams would be set for No. 6 to follow No. 4 at 120°, so the cam settings would then be 1, 2, 3, 4, 6, 5. If the engine fired 1, 5, 3, 6, 2, 4 then the cam settings for the forward lubricators on that engine would be 1, 3, 2 and for the after lubricator 4, 5 and 6.

J.E.: How much oil do you consider necessary for the efficient lubrication of cylinder per 24 hours?

S.E.: A well managed engine may use from three to four gallons per 24 hours, but much depends on alignment, state of piston rings, and whether the combustion is good or bad. When conditions are good one drop per minute is ample.

J.E.: Does that mean that the piston is without oil for an interval of one minute?

S.E.: No. If it takes nine strokes to form and liberate one oil drop it means that oil equal to 1/9th of the drop is being pumped each stroke of the plunger.

J.E.: How do you test for alignment?

S.E.: The best practical method is to look through the distance piece from the middle platform. Shut one eye. With the open eye catch some object on the ship's side where an imaginary line drawn from this object to the eye would pass the side of the working piston rod. This will enable you to detect the least " out of truth " movement.

J.E.: How can the alignment be rectified?

S.E.: It cannot be rectified without the expenditure of much time and patience. The fault is generally caused by the cylinder not being fitted absolutely vertical to the centre line of the crank shaft when under erection in the shop. Let us suppose that we detect with the eye a "Wobbly" piston rod. Now to test for truth would mean dropping a line down the cylinder through a prepared small hole set up between the centres of the crank shaft. We would then set the line centrally at the small hole, and also at the top of the cylinder, and by testing the bottom of the cylinder or stuffing box at four points we would find that the cylinder is lying over at a slight angle either athwartships or fore and aft. To correct this fault would mean disconnecting the cylinder to enable the distance piece (which fits between the cylinder and the engine framing) to be machined the small amount necessary to overcome the alignment error.

J.E.: Is it not possible to rectify this error by fitting a liner under one side of the cross head brasses?

S.E.: No. Although many believe that it can be done but they are only making matters worse. Examine the sketch of a cylinder out of line (exaggerated) and see how your suggested liners could be tolerated.

J.E.: What is there to know about piston rings?

S.E.: All pistons are fitted with a scraping ring top and bottom, and it is really astonishing how we find these rings fitted upside down. The bottom ring is the most particular and should be fitted with its scraping edge looking upwards. The top ring should be an ordinary ring without a scraping edge since we find that in fitting a scraping ring, the scraping edge wears too quickly. Some engineers believe that it pays when overhauling a piston to renew the two top rings, others believe in removing the third and fourth rings up and fitting the new rings in the vacated spaces. When a piston is opened up strict attention to detail is necessary. In many pistons the carbon behind the rings is the principal cause of trouble. It takes a little extra time to clear away carbon but like everything else in a Diesel engine, it is always the particular methods that give results. Granting alignment, lubrication, and combustion is correct and the rings are fitted into the bottom of the cylinder with the necessary clearance allowed (four to six thousandths of an inch), the running of a Diesel piston will give no more trouble than a steam piston running under superheated conditions.

J.E.: Is it not a fact that the dowel pins which are fitted into the piston to keep the rings from turning have broken or worked loose and got behind the rings, causing a scored liner and eventually a stop at sea, necessitating the withdrawal of the piston?

S.E.: Yes. But we now find that it is unnecessary to fit dowel pins. When lowering a piston into position in a cylinder, care must be taken that it is not allowed to turn, this ensures that the rings will be in their proper position.

Piston Cooling.

J.E.: Do piston cooling pipes require much attention?

S.E.: No. With proper workmanship once in a six months' voyage will suffice for an examination. It is the practice in some ships to examine the plunger pipes once in six months. If, when the plunger pipes are examined, the intermediate pipes are examined also, there should be no doubt about running them six months or longer.

J.E.: What precaution is necessary when re-placing these pipes?

S.E.: To see that the "Woodite pistons" on the plunger pipe are floating, and 1/16th inch vertical clearance is about the amount that should be allowed. When replacing the intermediate pipe it is important not to screw it up too tightly, since the ported nut which is brazed to the pipe is easily twisted. The jamb nut which backs the ported nut should also have very little strain when screwed home.

J.E.: Is not there a special tool supplied by the engine makers for use with these pipes?

S.E.: Yes; and it is the abuse of such a tool which causes bent pipes, etc. There are two tools supplied, a male and female. The female tool fits in two slots in the outer pipe, while the male tool fits into slots on the ported nut of the intermediate pipe. Two men are necessary to use this tool, so anyone using it by himself is only tending to twist the pipe.

J.E.: At sea, when the piston cooling return from a particular piston is cold, what does that signify?

S.E.: It signifies the plunger or inner pipe being carried away. This deserves immediate attention, since the piston will in a very short time become overheated and eventually seize.

J.E.: Is there any method by which a cracked piston can be detected when at sea?

S.E.: Yes. By the odour of the piston cooling return which smells like burning vulcanite. Difficulty in keeping down the temperature of the return water is another sign of a cracked piston.

J.E.: Is not there a positive method of detecting a cracked piston?

S.E.: Yes. When in port put the particular piston on the top centre and open the blast air. If the piston is cracked part of this blast air will be noticed coming from the piston cooling return pipe.

Cylinder Liners.

J.E.: What do you consider the life of a cylinder liner?

S.E.: Anything from four to six years.

J.E.: What increase in diameter would you expect at the end of that period?

S.E.: Anything from four to five m/m.

J.E.: Is this at the top or the bottom or is it the average increase?

S.E.: At the top.

J.E.: Would not this increase in diameter affect compression pressures?

S.E.: No. We compensate for that by lifting the piston $\frac{3}{4}$ m/m. for every one m/m. increase in diameter.

J.E.: Would not larger piston rings be necessary to make up for the increase diameter of the liner?

S.E.: No. But the standard rings will require renewing more frequently.

J.E.: Does the water space surrounding the liner accumulate much sediment?

S.E.: The amount of sediment that accumulates in the various spaces of a Diesel engine depends on the care bestowed on the iron gauze wire which is fitted to the rose plate of the main injection. Every three to four weeks this gauze should be renewed. Even with this attention much sediment is sure to find its way to the water spaces. There is generally a high suction fitted to be used when rivers are being navigated. On the side of the ship on which this valve is fitted, particular attention should be paid to the rose plate on which the gauze wire is fitted because the fitting of this rose plate in a careless manner will allow the water from the high injection to pass to the engine without being strained.

J.E.: How is this sediment got rid of?

S.E.: Mud doors are fitted which enable a portion of the spaces to be cleaned, but the most effective method is to fit a hose connection to the exhaust valve branch piece, and by sending a force of water through the system in a reverse direction tend to clear away much of the sediment. This operation can be carried out in port at night time, for the operation calls for little or no attention after the connection has been made.

J.E.: What are the actual connections?

S.E.: A valve and cock fitted on the sanitary line to which we connect a hose, the other end of the hose having suitable fittings to connect to the exhaust valve. The water of the sanitary system flows through the valve, into the cover, down through the liner jacket into the guides and finally flows to the bilge through the inlet pipe to the guide, which of course has the joint broken.

J.E.: For what length of time do you allow the water to run?

S.E.: Until the discoloured discharge changes to clear water. This may take six, eight or ten hours.

J.E.: Do you have similar connections for the auxiliary engines?

S.E.: No. We make use of the overboard discharge. Homewards, when the ship is loaded, the discharge for the auxiliaries is under water, so we allow the water to flow back from the sea into the system finally going to the bilges through a joint broken at the oil cooler.

J.E.: Is it difficult to draw the liners when they have been in position for some time?

S.E.: Occasionally one is found difficult to withdraw due to tightness in fitting when new. It is now the practice to fit them easy, which gives no trouble in withdrawing.

J.E.: Isn't it often necessary when a liner is being fitted in position to withdraw it again because the rubber rings which form the water joint at the bottom are leaking?

S.E.: Yes. But carelessness in not thoroughly cleaning out the ring grooves was responsible for the trouble. Before replacing the liners these grooves should be freed from all scale and then rubbed clean with a piece of emery cloth.

J.E.: If your method of cleaning the water spaces is satisfactory, why frequently withdraw the liners?

S.E.: To note the scale thickness at the top of the liners.

J.E.: In a well managed engine what would be the thickness of scale after a period of three years?

S.E.: Not more than one m/m.

J.E.: Is it a Board of Trade or Lloyd's rule that these liners should be withdrawn for a periodical survey?

S.E.: No. But when one is up the surveyor likes to examine it. When ships are in a home port and pistons and liners are opened up for survey, it is generally a last minute rush to close up again, with the result that many details are overlooked, which has an effect on the engine at later periods.

J.E.: So you think that lifting pistons and liners in a home port disorganise other detail work?

S.E.: Yes. Only the most necessary big work should be done in a home port, because this class of work is best left to the operating staff who can safely cope with these jobs in a foreign port when time permits. A record could then be kept of this work, which could be furnished each voyage to the various classification societies. Of course, at the present moment we are having trouble with water leakage at the piston rod flange joint, but this is now effectively cured by fitting special plates between the rod and piston, which makes a good joint. When all the pistons have been treated in this manner and all the top water spaces of the cylinder liners are altered we will then be in the position of eliminating much of the big work in a home port. Modern Diesels have not this weak design to contend with.

J.E.: If time did not permit when abroad would not this big work require attention in home ports?

S.E.: On vessels trading on the South American Coast there are many ports in close proximity to one another, and in doing these ports the vessel often leaves one at night time to make the next port at daylight. This is often done at a speed of six or seven knots, since one engine is sufficient for this speed any big overhaul on one engine could be undertaken with confidence.

J.E.: Your suggestion might be considered for a cargo ship, but what would be done in the case of a mail ship running against time when practically everything would require attention at the home port?

S.E.: That would be a different proposition. A mail ship would have one particular home port with a definite number of days in which to execute repairs. These repairs would be under the supervision of an experienced engineer whose interest in the job would be equal to that of the operating staff. Furthermore, the fact of having a definite number of days each voyage would enable the work to be systematically carried out.

J.E.: So you infer that a cargo ship with no set home port is at a disadvantage with the mail ship as regards the overhauling of machinery?

S.E.: Yes. But granting the suggested method of doing pistons and liners abroad, then the operating staff could easily undertake the changing and overhauling of valves and other detail work which would be necessary previous to an outward voyage.

J.E.: It seems to me that this method of running a cargo ship would be very economical and advantageous to the owners?

S.E.: I believe it would.

Exhaust Valves.

J.E.: Do exhaust valves require very much attention?

S.E.: The main point about exhaust valves or any other valve for that matter, is seeing that the faces are properly undercut. I'm afraid that some engineers treat this important point lightly.

J.E.: What is the limit a false seat will run before renewing?

S.E.: It is advisable to renew the false seat of the valve spindle when reduced to $\frac{1}{8}$in.

J.E.: What precaution is necessary when renewing these false seats?

S.E.: To see that the spindle head on which fits the false seat is free from all indents, otherwise the hot gases will have an easy passage between the indents whilst the valve is shut—which will in a very short time destroy the entire valve. (See sketch).

J.E.: Do springs weaken due to overheating?

S.E.: No. It is rarely that an exhaust valve spring causes any trouble.

J.E.: Then why do exhaust valves after running a few thousand miles begin to "bounce," *i.e.*, the valve closes, then appears to open again slightly as if the spring had weakened?

S.E.: The inlet valve opens a few degrees before the top centre, and the exhaust valve closes a few degrees after the top centre. If the slits on the inlet valve strainer are not kept clear a partial vacuum is formed by the out-going piston, which tends to open the exhaust valve until the internal and external pressures equalise. If the cam peak had worn and been filed and the circular part had not been reduced in conformity, it often occurs that the exhaust valve closes before the inlet valve opens. This will also cause "bouncing," whether the inlet slits are clear or stopped up.

J.E.: How many times are exhaust valves changed in a voyage of say 20,000 miles?

S.E.: Twice a voyage. If by any chance a valve gives trouble during the voyage it will be due either to inefficient lubrication or an overloaded engine.

J.E.: What do you call inefficient lubrication?

S.E.: That will be dealt with at a later period.

J.E.: When changing exhaust valves do you overhaul the old valves and get them off the platform as soon as possible?

S.E.: No. It is a mistake to rush at overhauling these valves just because they are an eyesore when lying about on the platform, because in the hurry many valves escape a proper inspection before closing up and are put in the racks in an unsatisfactory state. It is much better to put the valves in their respective racks to remain there until an opportunity presents itself for thorough overhauling.

J.E.: What do you mean by a thorough overhauling?

S.E.: I mean that these valves should be perfectly cleaned as well as properly ground. The springs should be dropped a few times on the platform to free any scale, the spring chambers should be examined and freed of all carbon deposit, the piston should have all oil grooves cleaned, and finally, a broom handle with a mop at the end saturated with paraffin should be pushed through the guide hole to clear any sticky substance lodging there. All this, of course, takes a little time and trouble, but in the end it is time well spent.

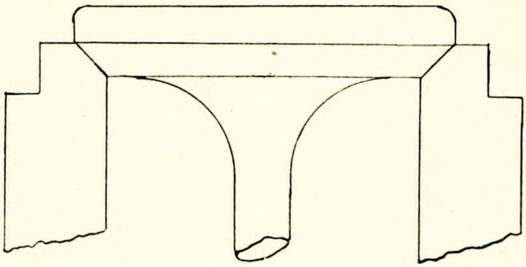


Figure (5)

Shows a valve spindle head and body seat when new.

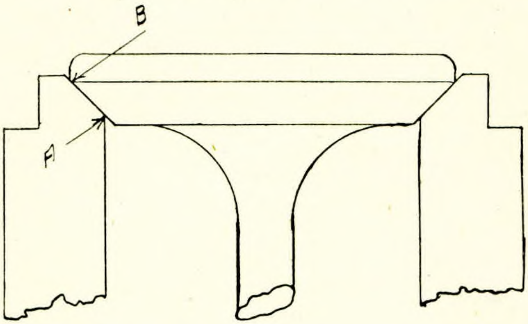


Figure (6)

Shows the same valve when valve and body seats have worn 1 m/m. each. Note how the valve head is buried in the body seat, if allowed to work in this condition the valve and body would soon be rendered useless through the point of the body (A) digging into the valve face, and the point (B) of the valve face digging into the body face.

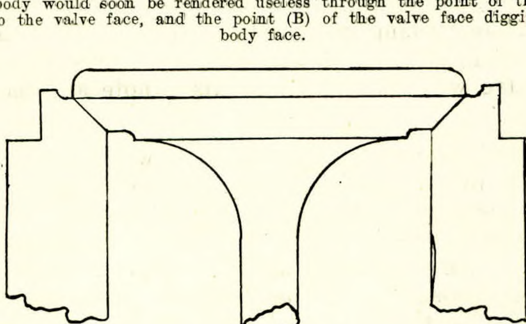


Figure (7)

Shows Figure No. 6 faces *undercut*, a simple operation and a very important one since the valves last much longer and tight faces are assured over a greater period of time.

J.E.: To be as particular as this, how many valves can be overhauled a day?

S.E.: Two in one working day is good work. Without the aid of a junior engineer, the donkeyman, when properly trained, can manage this amount, which is always inspected by the second engineer before being closed up. This method ensures that no valve can be put in the rack which is not in good order. In the early days when all hands were put on exhaust valves, they were generally in their racks inside twenty-four hours, consequently the rush caused a great deal of valve trouble at sea.

J.E.: If, for example, something went wrong with a valve at sea and the valve for this particular cylinder had not yet been overhauled, would we not be in a predicament?

S.E.: No. The chance of any of these valves giving any trouble is very remote indeed. Due to our thoroughness in overhauling, it is only something very unusual which could happen. We could put one of the old valves back, since system changed the valves, not necessity.

Inlet Valves.

J.E.: Do these valves require as much attention as exhaust valves?

S.E.: No. Every six months will suffice for an examination. If time will not permit renewal these valves can run for twelve months without detrimental effect on the engine.

J.E.: When leaving port and the engines are cold, I have often heard a hissing noise through the air inlet strainer when the fuel valve was firing. Does this denote a defective inlet valve?

S.E.: Not necessarily. There are two types of cylinder covers fitted to the engines we are discussing. These are called the fuel valve, air-cooled type, and the water-cooled type. The fuel valve pocket of the first mentioned type is common to the air inlet passage, and on the latter is isolated from the air inlet passage by a brass liner inside of which the fuel valve is fitted. In the air-cooled type, if a fuel valve pocket seat were leaking, a hissing noise would be heard through the air inlet strainer every time the fuel valve fired.

J.E.: How could we prove that it was not the inlet valve leaking?

S.E.: By rejointing the fuel valve seat and putting the engine on the top centre (firing stroke) and opening the blast air to the cylinder. These leaks are not serious, since they take up in a few minutes when the engine is warm.

J.E.: At manœuvring into port, between the intervals of starting and stopping, smoke often issues from the air inlet strainer. Where does this smoke come from?

S.E.: If a fuel valve with a false end were fitted in an air-cooled type of cover, and this false end was leaking, the blast air trapped in the fuel valve (after the engine had stopped) would be saturated with fuel oil, and this leaking through the defective joint would cause much smoke to emerge through the inlet strainer.

J.E.: Since these two fuel valve defects are readily noticed in an air-cooled cover, how can they be detected in a water-cooled type of cylinder cover?

S.E.: It is the practice to treat the head of each fuel valve with a thin rubber ring in manner similar to what is done with the starting valve. This rubber serves as a joint to keep any water or oil which may lodge on top of the cylinder covers from finding a passage into the fuel valve pocket. This joint also serves as a tell-tale for any gas leaking at the pocket seat and false end joint, since any pressures in the valve pocket would make itself known by creating a squealing noise before eventually bursting the rubber ring joint.

Fuel Valves.

J.E.: It seems that there is quite an amount to know about the overhauling of these valves.

S.E.: Any engineer who has never seen a fuel valve before, is quite in a position to overhaul one satisfactorily if he accepts and appreciates the advice of those responsible for his training. Many parts of a Diesel engine call for particular workmanship and fuel valves are no exception to this rule. The screwed cone nut which covers the pulverisers should be taken off each time the valve is overhauled, to thoroughly clean the carbon which at times adheres to the passages. The clearance of this cone nut when in position should be ½ m/m., which can be tested by using a piece of 1/16th inch lead wire.

J.E.: Why is it necessary to have this ½ m/m. clearance?

S.E.: It is not necessary. Theoretically, the cone should bear against the valve body, but as this is expecting too much from practice the next best thing to do is allow a little clearance which does not interfere with good pulverising.

J.E.: If you made the clearance $\frac{1}{8}$in., what would happen?

S.E.: Bad pulverising. The oil would be blown through the $\frac{1}{8}$in. clearance space, which of course would be the line of least resistance. Sketch the end of the valve and see for yourself what would happen.

J.E.: A fuel valve has two faces. Is it the outer or inner face that should be ground?

S.E.: The inner face. We could grind up both faces together which would be the ideal method, but this would take considerable care and time. There is no necessity for anxiety so long as the inner face is a good job with the outer face a few thousands of an inch clear.

J.E.: Should the valve spindle head be "proud" of the body, under the flush, or should the end of the head be flush with the end of the body?

S.E.: That is a very important point. The valve head should always be "proud": buried heads have caused the valves to leak after a few days' run, due to a small particle of burnt carbon being trapped on the outer face as the valve was closing, consequently the inner faces were kept apart due to this defect.

J.E.: Sometimes fuel valves are fitted with false ends. Do these require special care in fitting?

S.E.: Yes. A main fuel valve spindle does not work in the centre of the body, so it follows that a new end requires special care to ensure that the centre of the new end is in line with the rest of the body. Proceed as follows:—Screw on the false end until it is nearly home, using the facing tool or cutter (which is supplied for refacing body seats) for finding the true centre. The true position is found by unscrewing or screwing-on the false end until the facing tool revolves freely in the hole. When the true position is found, draw a thin line across the body and the false end, then screw home to these marks, using the thin copper washers as make up.

J.E.: How do you know when a new end is required?

S.E.: By the increased diameter of the orifice behind the inner seating. As the orifice increases the inner seating decreases.

J.E.: Does the continual undercutting of both valves and body seats affect the compression of the valve spring, and if so, how?

S.E.: From the underside of the valve spindle nut to the top of the bridge is a certain measurement. If the valve and body faces have worn an amount which allows the valve head being buried $\frac{1}{8}$in., it follows that the certain measurement has been increased $\frac{1}{8}$in., and to keep the original compression, an $\frac{1}{8}$in. washer must be fitted between the nuts and the spring.

J.E.: Is it good practice to measure the length of spring each time the valve is overhauled?

S.E.: Absolutely essential. Many valves have been burnt out by this neglect. It is good practice to have gauges made for both main and auxiliary springs which should always be used when these valves are being overhauled, this will ensure that no spring can be put back which is not correct to gauge measurement.

J.E.: When the engine is working how can a weak spring be detected?

S.E.: By a hissing noise through the air inlet strainer.

J.E.: What causes the hissing noise?

S.E.: The blast air (overcoming the spring pressure) blowing into the cylinder whilst the inlet valve is open.

J.E.: What precaution is necessary when this occurs?

S.E.: Immediately by-pass the fuel valve until assisting gear is rigged up to compensate.

J.E.: I'm afraid these valves cause quite an amount of trouble if improperly treated. What is your recommendation when overhauling one?

S.E.: Disconnect the various parts. Before thinking of grinding up the faces put the spindle in a vice and screw the nut hard home. At all times be careful to see that the *top* of the spindle is hard up against the *nut*. If the pin holes do not coincide, never turn the nut back, file a little off the spindle head and give the nut one half turn more.

J.E.: How can the taper pin be a proper fit if the nut is given a half turn from its original position when the pin was fitted?

S.E.: It is not a hole in the top of the spindle, it is a parallel slot, so that the taper pin can be fitted from either side. Although this is not an ideal job it is the best that can be done with such fittings. Good strong split pins are just as effective as taper pins and much easier to fix.

J.E.: Your idea of making sure that the top of the spindle is hard up against the nut is to ensure that the spindle takes the shock instead of the pin, when the valve is opening?

S.E.: Exactly. This point is important. When this end of the spindle is completed, next put the spindle in a lathe to undercut the faces if necessary. While in the lathe clean out the oil groove and with the oil stone run over that part of the spindle which works in the stuffing box. If the valve body does not require undercutting you can commence grinding the faces. When satisfied with this operation, examine the N.R.V. by-pass valve and pulveriser, and clean out the various passages with the air hose, after which the valve should be ready for assembling. When replacing the pulverising tube be careful to see that it is thoroughly clean and do not forget the $\frac{1}{2}$ m/m. clearance at the cone end. The spindle can now be smeared over with cylinder oil, and after packing, and before the spring is put on, make sure you can move the spindle. A blow with the end of a hand hammer shaft ought to be sufficient to move it, if it does not, slack back the gland nut until it does. Sometimes a copper washer is used to joint the N.R.V. case, it is an easier and quite a good job to substitute lead for copper, which gives no trouble in withdrawing. The spring can now be measured for length and, if correct, the valve can be completed ready for testing.

J.E.: What is the best packing to use in these valves?

S.E.: "Palmetto" and "Garlock" are two well-known packings that give good results if properly treated.

J.E.: What do you mean by properly treated?

S.E.: I mean that good packing and *inferior* workmanship will not give results obtained with inferior packing and *good* workmanship.

J.E.: Surely it is a simple operation for any engineer to pack a valve and make a good job?

S.E.: I agree that it is a simple operation, but for successful results one must be very particular indeed. One engineer can pack a valve that will run a week, whilst one done by another engineer will run for months. Before a good job can be made a special tool is required to push the packing home in the stuffing box, and another tool is required which cuts the turns of packing the exact length. The first mentioned tool is a long steel tube bored out 1/64th inch larger than spindle diameter, and neatly turned on the outside to 1/32nd inch less than the stuffing box diameter. The latter tool is a hard wood arrangement and is merely a template to ensure that each turn of packing is cut the exact length. With this template the packing can be cut and collected in a box to be ready for use when required. This in itself is a particular point as it keeps the packing from lying about on the vice bench where much foreign matter collects.

J.E.: How are these valves tested for tightness?

S.E.: By air at 60 atmospheres pressure. A special bracket is fitted in the engine room with all necessary connections. When the connections are made, the air is put on and the spindle is hit a few times to ensure that it does not stick open. It sometimes happens when the N.R.V. is being tested that only once in a dozen times will the valve show tightness. This proves that it is only little particles of dirt that cause the leakage, which will soon disappear when the valve is in use.

Starting Valves.

J.E.: Do the starting valves give much trouble?

S.E.: Like everything else they will give trouble if not properly treated. At one time they developed defects but these were overcome. We found that many of the body seats were a shade eccentric and this was cured by turning up a mandril to fit the face end of the valve body. (See sketch). The long shank left on this mandril allowed the tool easy access to the seat. With this useful medium a body seat could be tested and rectified in less than ten minutes. We next found that the spindle heads were slightly bent, and when these were straightened, valve troubles vanished.

J.E.: I have heard it mentioned that nothing is more deceptive than a starting valve face?

S.E.: That is common knowledge. But if everyone took a little extra time to test the faces, this deception would not arise.

J.E.: When you assemble these valves, what oil do you use?

S.E.: Cylinder oil only.

J.E.: Is it important to clean out the oil grooves on the air piston?

S.E.: Very. On these pistons are three small holes to feed the oil to the grooves. These holes are often closed by carbon deposit, consequently those overhauling the valves overlook them if not previously informed to see that they are clear.

J.E.: What treatment do these valves receive at sea?

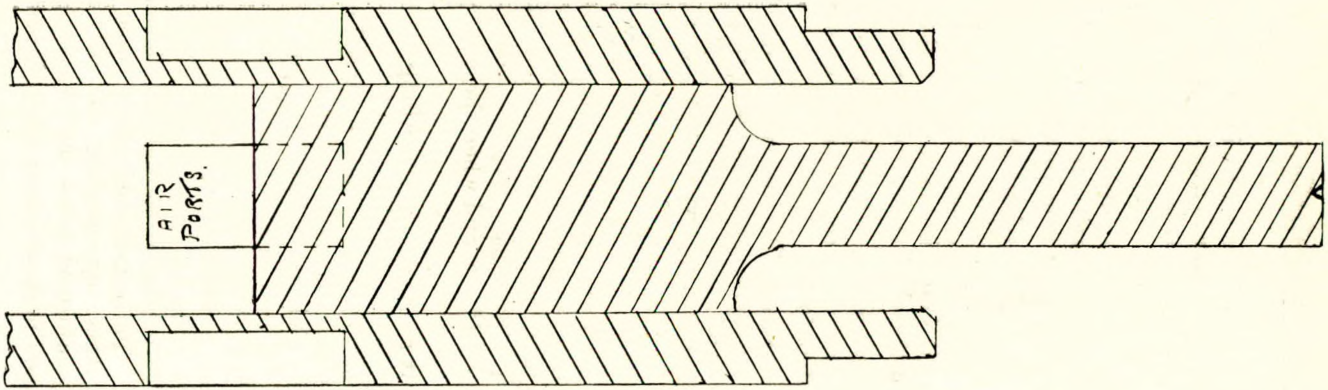


FIGURE (8)

Shows mandril for use in the lathe in *truing-up* starting valve seats. This valve will deceive the engineer more than any other valve, therefore particular attention should be given to these valve spindles, since their heads are often found slightly bent if working in a seat that was not perfectly true.

S.E.: They are oiled and turned on their faces once a week at sea. When oiling, a can similar to a sewing machine oil can should be used. The best lubricant is one of paraffin to two of compressor oil.

J.E.: I have noticed after these valves have been oiled and turned on their faces that some particular valve begins to give trouble, necessitating a stop at sea to enable it to be changed.

S.E.: That was due to the eccentricity of the body seat, or a slightly bent valve spindle head. Before attempting to turn these spindles at sea, the condition of each valve must be known beforehand. Therefore, it is advisable to treat the faces (as previously suggested) at the earliest possible opportunity.

J.E.: I have noticed two grooves about ½in. wide by 1/32nd inch deep cut on the circumference of the lower part of the valve body. What is that for?

S.E.: At one time we had considerable trouble in withdrawing these valves from the cylinder cover, and these grooves were turned, the keen edges of which helped to cut away through any dust or scale likely to deposit in the pocket.

J.E.: What caused the scale to accumulate in the valve pocket?

S.E.: At one time none of us were sufficiently experienced to see that the starting air was properly drained, consequently much water was blown into the valve pocket when the engines were being manœuvred. Reservoirs are now blown through daily and the air slide drains each watch and *always* before manœuvring.

J.E.: Was it this water in the starting air that caused the early troubles with "sticking" air slides?

S.E.: No. The seizing of the air slide was caused by carbon escaping from a leaky air starting valve. Drains are now fitted to the air starting line which are always open and any gases from a leaky valve pass through these drains, the ends of which are led to the control platform.

J.E.: When explaining the overhaulings of air starting valves you mentioned that the small holes for lubricating the air piston were generally found carbonised and choked. Was that caused by the gas from one of the valves?

S.E.: From several. The leak is so minute that it has no detrimental effect on the valve face, but sufficient gas escapes which collects in the pipe line and forms a sooty deposit. Since there is no pressure in the line the sooty deposit stops there to be blown into the starting valves when manœuvring of the engine takes place. But in the Diesel engine there is generally a remedy for all defects. On entering port, after the telegraph has rung stop, reverse the engines so that they come to rest quickly. When at rest, open the starting air to the air slide and give the manœuvring handle a few quick movements to supply just sufficient air to clear out through the drains all the sooty deposit.

J.E.: What would happen if the telegraph rung for an order before you had the line clear?

S.E.: It is for this reason that the engine should be brought to rest by reversing. There is nothing that can be done now but obey the telegraph and blow all the deposits into the air pistons. It is a few manœuvres like this that dry up the pistons, and block the small lubricating holes. Of course it is only in long runs that carbon can accumulate in any quantity and this can be partly cleared at sea by slightly easing the starting air to create a pressure just sufficient to help and clear the line.

J.E.: How long do you allow these valves to run?

S.E.: A round voyage of four to five months. Occasionally one or two may be a little defective due to an overdose of carbon, these should be taken out at the first opportunity.

J.E.: I notice that these valves have grease cups fitted to lubricate the spindle. They don't look an efficient fitting.

S.E.: They are not and for that reason they are not used. If care is used in oiling the spindles you will encounter no difficulty with sticking spindles.

J.E.: What is the usual jointing medium used for all cylinder cover valves?

S.E.: For good results grinding gear is necessary for all body seats and for pocket seats in cylinder covers. Where cylinder covers are not cracked these seats should be jointed with nothing more than boiled oil. The usual practice is to use graphite or a thin red lead and varnish paste, but this generally covers a multitude of sins and should not be adopted. Recently a friend of mine took charge of an installation with joints of this kind. After getting some tools for improving some of the defective faces he arrived home with not one valve, either in compressor, auxiliaries, or main motors that was not a metal to metal joint with nothing else but boiled oil as a jointing medium. I merely mention this in order to show you what can be done in a motor ship with good tools and good workmanship.

Exhaust Manifolds.

J.E.: How often should these manifolds be examined?

S.E.: After the first two years' running, one manifold should be opened up and scaled each voyage.

J.E.: Where is most of this scale found?

S.E.: At the top of the cover between the strengthening ribs.

J.E.: What happens when the scale is not sufficiently cleaned off?

S.E.: There is a danger of the inner box being cracked by unequal expansion of the metal forming the top and bottom side of the box.

J.E.: Do these cracks develop at the middle or at the end of the boxes?

S.E.: At each end near the flanges.

J.E.: How can you test if these boxes are cracked?

S.E.: At the forward end of each box an adapter is fitted through the water space into the exhaust side, and into this adapter is fitted a small cock, the opening of which will prove if any water is in the exhaust side of the box.

J.E.: Are not these cocks often found choked up?

S.E.: Yes. They are too small and often are neglected. Cocks are not necessary and the hole through the adapter should be at least $\frac{3}{8}$in. in diameter and this should be at the *after* instead of the forward end of the box.

J.E.: Why should no cocks be fitted and why should the adapter be fitted aft instead of forward?

S.E.: Because we want a *tell-tale* on the boxes and if we have a clear opening of $\frac{3}{8}$in. or more (which would not require constant attention due to choking up) from the exhaust box to the atmosphere, any water finding its way to the exhaust side would immediately run out through the hole and be seen by some member of the engine room staff, who happened to pass underneath it. All vessels are generally a little down by the stern and if these adapters were fitted aft there would be no accumulation of water in a cracked box since it would run out through the hole as fast as it could run in.

J.E.: At sea would not the exhaust gases blow through such a big hole?

S.E.: Yes; but being colourless and odourless they do no harm.

J.E.: What might happen if a crack developed on an inner box of an engine where these small cocks were neglected?

S.E.: If the ship was in port three or four days, the exhaust box would fill up with water, which would eventually find a passage into several of the cylinders, and if this was not noticed previous to leaving port serious consequences may occur.

J.E.: Assuming a box was cracked, and we were aware of it, could we still manœuvre in and out of port?

S.E.: Before leaving port the circulating water control valve should be opened just sufficient to flood the cylinder covers without overflowing into the water space of the exhaust box, and when " under way " the normal water pressure can be carried because the pressure of the exhaust will keep the box dry. On entering port more care is required. After the first stop of the engines (the exhaust pressure being released) the water will at once leak into the box. To keep the water pressure off the box the control valve should be opened and shut as the engines are starting and stopping.

Main Compressors.

J.E.: Do the pistons of these compressors require much attention?

S.E.: No. Once in two years will suffice for an examination of piston rings and gudgeon pin brasses.

J.E.: Why is the H.P. piston a floating fit?

S.E.: To correct any error in alignment.

J.E.: There is a core plug in the centre of the H.P. piston, has this been known to slacken back and cause a complete breakdown.

S.E.: It is a mistaken idea to think these core plugs slacken back. If the screw threads of the core hole and plug are made of dissimilar metals and the plug screwed in not a particularly tight fit, unequal expansion of the metals may slacken the plug sufficiently for it to be blown out by the high pressure air which can easily find a passage under the flange of the floating piston. These should be examined each time the H.P. valves are overhauled.

J.E.: How often are the compressor valves changed?

S.E.: When a good job is made of these valves, *previous to being fitted into the compressor*, no trouble will be found in running them ten to twelve thousand miles.

J.E.: Then why do some ships have trouble with these valves during the voyage?

S.E.: Because several of the staff overhauled the valves between them. It is better to make this a one man job. He must be shown exactly how to undercut the bearing faces and be told every little particular detail in connection with the overhaulting. If this method is followed compressor troubles will vanish. Eagerness to learn and strict attention to duty cannot be denied in the majority of young men who can be made either efficient or inefficient Diesel engineers, depending upon the training they receive. Now let us suppose we are in port and that all the compressor valves have been changed. We would then be left with over thirty valves which require overhauling. At one time we treated these valves like exhaust valves, we could not get at them quickly enough, with the result that the lack of organisation caused many of the valves being put together in a manner likely to cause disaster, for of all the valves hurriedly put together less than half could possibly be well done. Experience of this taught us that we were working on wrong lines.

J.E.: Then what is the best method?

S.E.: Making it a one man job. The refrigerating engineer is best fitted for this purpose since we generally pick a particular junior for the care of this machine. The second engineer shows him one of the valves opened out, explaining exactly what is required, and when the first valve is completed the second examines it again and if satisfactory the junior is left to carry on at his leisure without any hurry whatsoever.

J.E.: But won't it take a long time for one man to get through all these valves?

S.E.: Yes; but we can afford this man all the time he wants, since he overhauls the valves and attends the "freezer" at the same time, and now and again does other particular jobs which the second engineer may have in hand for him.

J.E.: Supposing a few valves were required in the meantime, what would happen if none were overhauled and ready?

S.E.: Exactly what would happen if a main exhaust valve failed. Put back the valve that came out since system changed them, not necessity.

J.E.: So you believe that the chances of any of the valves giving trouble is indeed remote?

S.E.: Yes. You can take it for granted that all valves put on the board by the one man are a good job. When the time arrives again for another change of valves it is advisable, although not necessary, to change one stage at a time. This can be carried out when the runs between ports are of short duration. The suction and delivery valves for the H.P. stage should be typed 1, 2 and 3 from their respective positions and immediately put in the prepared rack from which were taken the valves to be fitted, which were already typed for their respective places. Only the H.P. suction valves can be prepared for dropping into position, the delivery valves necessitating grinding into the seats of the cylinder head. A special tool for facing the valves in the cylinder head is supplied by the engine makers.

J.E.: Isn't there more than one tool for refacing the delivery valve seats?

S.E.: Yes. A lip is formed which wears after a certain period. A special tool will be necessary for undercutting to reform the lip again. This is a very important point and must not be overlooked, since the seat, if allowed to go below flush, will round the edges of the valves, causing them to be changed more frequently. (See sketch).

J.E.: Name and explain the use of the several parts forming one of these L.P. suction or delivery valves?

S.E.: The " brass seat " on which works the " steel plate " valve. The " stop ring " which keeps the valve central. The " spring," the inside diameter of which should be a neat fit for the diameter of the stop ring. The " elastic discs " (two) which are necessary to absorb the shock of the steel plate valve when it opens. The " stop disc " on which fits the elastic disc and which is really a nut to bind the whole in position.

J.E.: The underside of the stop disc is rounded off. Why?

S.E.: If it were flat the " elastic discs " would be useless, since no clearance would be there to form the cushion. Carelessness in not fitting the elastic disc over the spigot formed on the stop disc is responsible for many broken elastic discs and the faulty working of valves.

J.E.: If the L.P. or N.P. receiver pressures are above or below normal, what would be the cause?

S.E.: Examine the sketch of a simplified three-stage air compressor where the pressure from the various stages can be readily followed. If the stage pressures vary it is obvious from

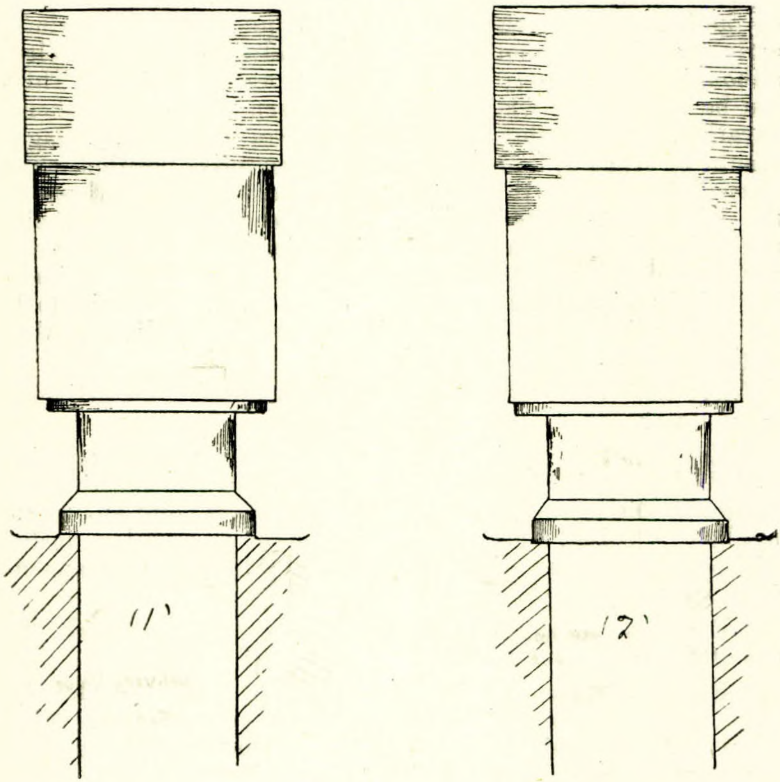


Figure (9)

Shows main engine H.P. compressor valves. Note the bearing lip on No. 1. When this lip is worn with continual grinding the valve is buried as shown in No. 2. This turns up the edges of the valve face with the result that the valves are never tight. A special tool must be designed to reform the lip again.

this sketch that the defective valve or valves can be easily found. When the M.P. receiver pressure is raised above normal and the air from the H.P. cooler to the blast bottle is at the usual temperature, the H.P. suction valve is at fault. On the other hand, if the air is cold from the cooler and gradually heats up when nearing the blast bottle, the fault is at the H.P. discharge valve.

J.E.: What makes the air heat up when nearing the blast bottle?

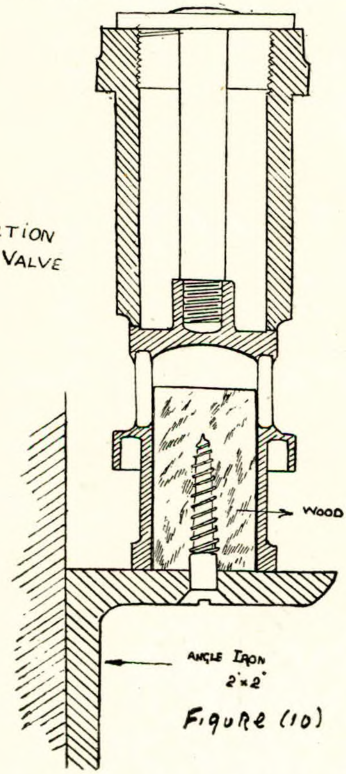


Figure (10)

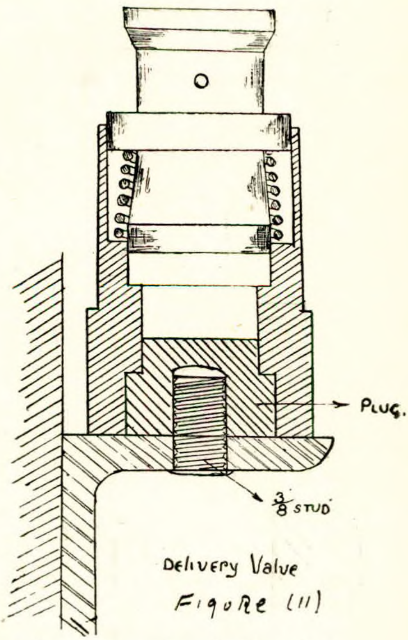
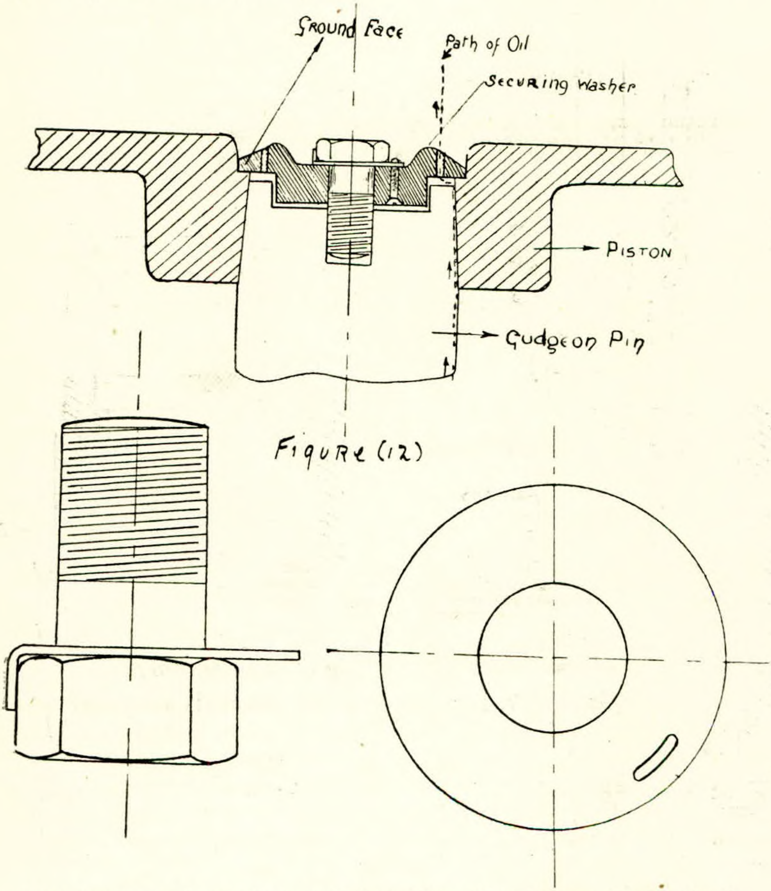


Figure (11)

Shows rack for spare H.P. suction and delivery valves. These valves should receive special care and attention and not be left in drawers or lockers to be knocked about. Figure 10 is a suction valve ground in ready for immediate use; the two parts are held together by the temporary screw shown. Figure 11 is the discharge valve. This must be ground in position when valves are being changed. There should be a double set of valves for both compressors, each typed to their respective positions.

S.E.: To say the air heats up when nearing the blast bottle is not technically correct. What actually happens when a faulty H.P. discharge valve becomes defective is that the good working compressor is still pumping up to 60 atmospheres, and the defective one, if tested by itself, would be found to be pumping to only 50 atmospheres or thereabouts, therefore it is obvious that the good working compressor must help the defective one if each engine is to be supplied with air at 60 atmospheres; and to do this the air in the blast bottle into which the defective compressor is pumping is again being compressed (through the cross connection on the regulating head) by the good working compressor.



Shows auxiliary gudgeon pin fastened in position. Note the ground face, also the tapping of washers adjacent to the ground face. If the taper end of the gudgeon pin is a slack fit, lubricating oil finds an easy passage through these tapping holes which should have been drilled only half-way through. Note the dowel pin in the washer to keep the copper washer from turning when tightening up the fastening pin. If the head of this fastening pin is left as shown in *position*, it will elongate the dowel pin hole in the copper washer as shown in the sketch. Take the edge off this pin as shown in the sketch and this trouble will disappear. It is important that this small hole in the copper washer should not be elongated; for it is this that causes these pins to slacken back.

J.E.: So you infer that the heat is in the bottle and is tending to travel up the discharge pipe?

S.E.: Yes. The pipe is cold as it leaves the compressor, but gradually warms as it is nearing the bottle. This is the proof that work is being done on the air in this bottle.

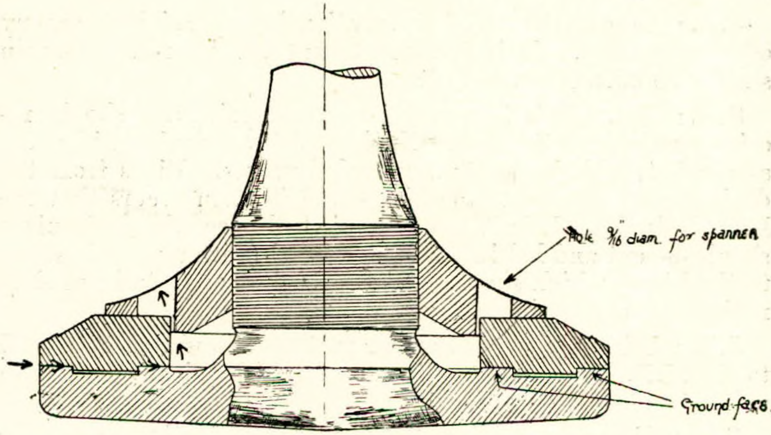


FIGURE (131)

Shows a main engine exhaust detachable seat. When this seat is renewed, care should be taken to free any scale lodging on the seat of the solid head, since neglect of this creates easy passages for the gases which in a short time will entirely destroy the valve.

J.E.: How often should the cooling coils be tested?

S.E.: Once in twelve months. When the coils are taken out and cleaned it is recommended that they be weighed, but this is hardly a practical operation with the class of scales generally found in an engine room. Take this opportunity of thoroughly cleaning the H.P. cylinder walls which should afterwards be coated with a protective paint before replacing the coils again. Before replacing the hood see that the inside is cleaned and painted. Soft copper rings should always be used as the jointing medium for the coil joints.

J.E.: There is a brass ring or bush in the centre of this hood. What is its use?

S.E.: This brass bush in conjunction with a rubber ring around a groove at the top of the H.P. cylinder head forms the water joint. At one time our inexperience caused us to lift this hood many times before a tight joint could be made.

J.E.: And what did you eventually find?

S.E.: That we were not sufficiently particular to observe that a little piece of emery cloth was necessary to clean the ring of any scale and dirt.

J.E.: I see that lack of observation is responsible for many of our troubles. Is it important that the inter-cooler drain should be open when at sea?

S.E.: Yes. This is a very important point and should receive particular attention. Neglect of these drains will cause as much trouble as badly overhauled valves. Pipes from the drain valves are led to the bilges and it often occurs that the L.P. drain gets choked, to make this fool-proof the drain pipes should be cut and led into a large funnel (with a pipe attached to the bilge) which should be fitted high enough to be readily noticed by the engineer on watch.

J.E.: What is your reason for cutting off the compressor oil to the M.P. stage and stopping the forced lubricating oil from going to the piston guides?

S.E.: Because sufficient lubricating oil is thrown by the crank on to the guide faces, a small amount of which manages to feed the M.P. stage without any detrimental effect on rings and valves. It is impossible to advise one how to manage a compressor, *observation* of valves when changing them is one's only guidance.

Fuel Pumps.

J.E.: How often do you think it necessary to regrind the suction and delivery valves of these pumps?

S.E.: Once in two years will suffice if thoroughly done.

J.E.: Does this also apply to the auxiliary engine fuel pumps?

S.E.: No. The auxiliary pumps require more frequent attention; this will be dealt with at a later period.

J.E.: What parts are deserving of special attention?

S.E.: The alignment and repacking of plungers and the resetting of suction valves.

J.E.: How often should these plungers be repacked?

S.E.: Everything depends on the method adopted. If you attempt to take out the old packing with the aid of the

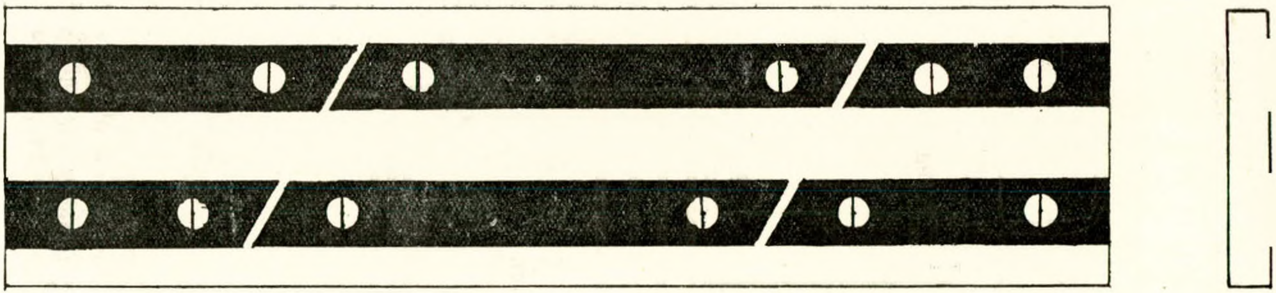


Fig. 14.
Shows a simple template for ensuring that packing be cut to the proper size; very amateurish, but very necessary when working at the packing of fuel pumps and fuel valves.

tang end of a file you can't expect good results. With good workmanship and good packing (Garlock packing is very successful) no difficulty should be found in working them twelve

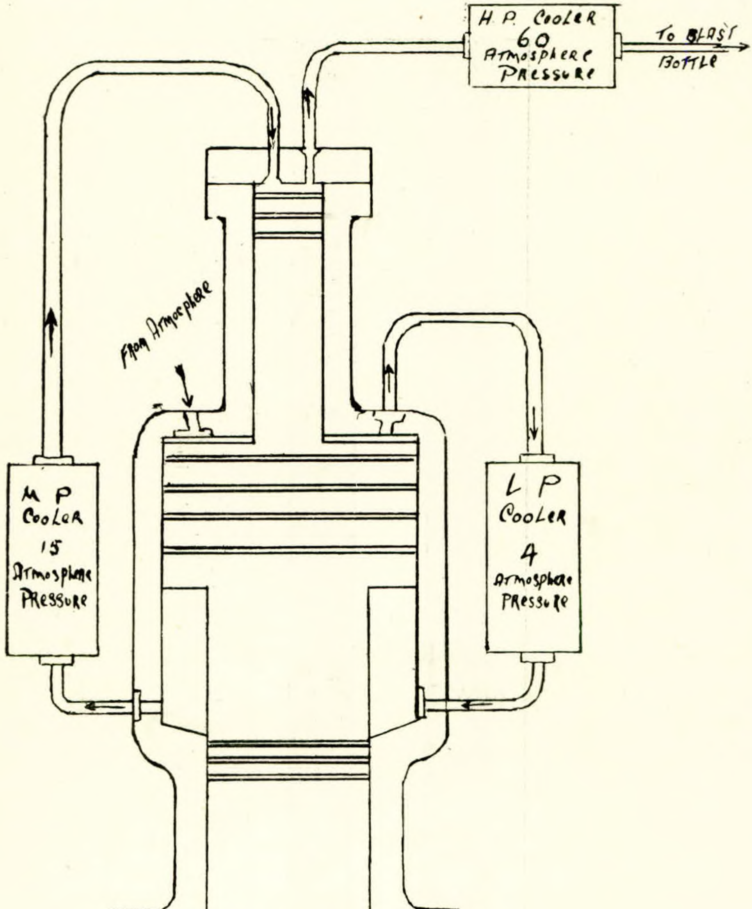


FIGURE (18)

Shows a simplified three-stage air compressor. When the various stage pressures vary, it is obvious from this sketch that the defective valve or valves can be readily located.

months. Let us assume that we are overhauling a pump. To test for alignment begin by using feelers to find out whether or not the guide bars require attention, after which withdraw the plungers. These plungers develop a slight ridge top and bottom, which should at all times be buffed off, after which they should be put in the lathe to be given a touch up with an oil stone. Since the crank shaft is out of the way repacking becomes a very simple operation. When cutting packing, use the template and since there is plenty of head room make use of a fuel valve packing tube to force each turn properly home.

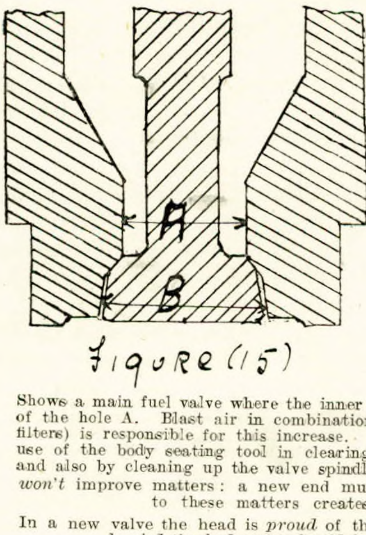


Figure (15)

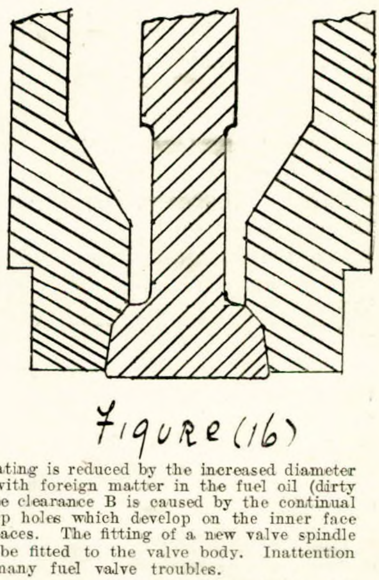


Figure (16)

Shows a main fuel valve where the inner seating is reduced by the increased diameter of the hole A. Blast air in combination with foreign matter in the fuel oil (dirty filters) is responsible for this increase. The clearance B is caused by the continual use of the body seating tool in clearing up holes which develop on the inner face and also by cleaning up the valve spindle faces. The fitting of a new valve spindle *won't* improve matters: a new end must be fitted to the valve body. Inattention to these matters creates many fuel valve troubles.

In a new valve the head is *proud* of the body seat as shown. When the head is *buried* the body seat should be *undercut* as shown at Figure 15.

Don't fill stuffing boxes too full, but just sufficient for gland nuts to be entered half of their depth without the aid of a spanner. The crank shaft can now be replaced and before re-jointing the plugs at the bottom of the pump drive up the plungers into the crosshead with the aid of a copper drift. This is a much more satisfactory method than trying to get the old packing out with the aid of a file tang.

J.E.: Yes, but isn't it a much longer method?

S.E.: What is an extra half day when you are getting a pump to run for a year. It is workmanship of this class that gives us results. At the present moment engineers in various ships are experimenting with different classes of packing for these pumps. Probably a long neck bush of white metal made a working fit on the plunger and a slack fit for the stuffing box (to rectify misalignment) will be the successful packing. This has given satisfaction in many auxiliary pumps.

J.E.: What is the clearance between suction valve and tappet spindle?

S.E.: About 4/1000th inch.

J.E.: When clearances are so fine what is done to compensate for the wear of the various pins and bushes?

S.E.: Even if the total slackness amounted to 1/16th inch it would not interfere with the proper functioning of suction valve. But certain precautions would be necessary. When setting the pump see that the pump crossheads are always at their *highest* point and the tappet spindles at their lowest points. When turning the pump crank shaft always turn in the one direction, if you happen to go too far at any time continue turning another revolution rather than turning back, this ensures the cross head always being at its highest point. When altering the right and left-handed nuts on the regulating rod behind the pump for the desired clearance necessary at the suction valve, never try this clearance without first bringing the manœuvring handle back to stop position and then on again to setting position, which is midway on the quadrant between stop and start. In other words, the manœuvring handle should always be moved from setting position to stop and back to setting position again, after each alteration of regulating lever.

J.E.: Why is this shifting of the manœuvring handle necessary?

S.E.: To ensure that the tappet spindles are in their *lowest* position.

J.E.: Why is it necessary to have the pump crossheads at their *highest* position, and the tappet spindles at their *lowest* position when the clearance is taken?

S.E.: To get as near as possible to actual working conditions which exist when a pump has a few worn pins and bushes, etc. The tappet spindles are worked by a series of levers connected to and driven by the plunger crossheads, therefore it is easy to follow that the friction of the packing in the plunger and also in the tappet spindle stuffing boxes must be taken into consideration when the fuel pump is being set.

J.E.: Many times I have heard it remarked that half-a-dozen men could set these pumps and each would get a reading different. I now see the reason why. I suppose it is the inaccurate setting of these pumps which causes one cylinder to develop more power than the others?

S.E.: Not always.

J.E.: If when taking indicator diagrams you found one cylinder developing more power than the rest, would you alter the setting of the pump to decrease the amount of fuel oil delivered?

S.E.: No. Once the pump was properly set I would do anything rather than alter the setting. When taking indicator diagrams there are many things to be taken into consideration when the mean pressure is being worked out. For instance one diagram may give a lower mean pressure than another receiving an equal amount of fuel, but this will be dealt with later.

J.E.: You say two years would suffice for a grinding-in of valves. Isn't it possible these valves would commence leaking before that time?

S.E.: Possibly; but a small leak would be harmless, so long as the N.R.V. in the fuel valves and the intermediate N.R.V. in the pipe line between pump and fuel valves are tight.

J.E.: How often should the intermediate N.R.V. be overhauled?

S.E.: Once each voyage of six months. It is advisable to have three of these complete valves as spare, which should be overhauled and tested to 60 atmospheres, after which they should be placed in a prepared rack ready to be used when required. It is then a very simple matter keeping them all in good condition.

J.E.: Even with these valves in good condition is it possible to get an air lock in the fuel pump?

S.E.: Air locks have been known to occur after the repacking of the fuel pump. This can be attributed to some foreign matter being left in the pump in the process of repacking. Then sooner or later a particle is trapped on the faces of pump discharge valve, intermediate N.R.V., and fuel valve N.R.V., simultaneously, which creates an opening for the blast air to reach the fuel pump.

J.E.: It seems an impossibility for such a coincidence to happen to three valves at almost the same moment?

S.E.: It seems impossible, but nothing else can actually cause it. Some believe that an oil with a watery instead of a viscous nature is responsible for many air locks, but if this were true, it must follow that air locks would be happening continuously, whereas they only happen at long intervals on some ships, and never on others.

J.E.: Is it possible for an air lock to be caused by one of the auxiliary engines?

S.E.: Yes. That will be fully explained later when dealing with auxiliary engines.

J.E.: If the auxiliary engines are capable of causing air locks, how do you know when one occurs if the fault is at the auxiliary or the main engine?

S.E.: If the auxiliary should cause an air lock, the engine room lighting would be affected, due to the auxiliary slowing down. But if the main engine slowed down without any effect on the lighting you will know at once that the trouble is with the main engine.

J.E.: If the air lock is in one engine, does this also affect the other?

S.E.: The engine causing the air lock will slow down first and quite a few minutes will elapse before the other is affected, but if you are smart enough you can locate the trouble in a moment and rectify things before the engines have time to stop.

J.E.: What would you tell a junior engineer to do if such an occasion arose on your watch?

S.E.: I would expect him to carry out the instructions I had given him when he first came on watch with me. I consider it a senior watch keeper's duty to explain to his junior what is expected from him when certain occasions arise and not wait until they actually happen. If this were done many of the trivial incidents peculiar to a Diesel engine room could be avoided. In the event of one of the engines slowing down, first go to the cylinder tops and begin to by-pass the fuel, starting from No. 1. If oil flows from No. 1 it will be alright, so pass on to No. 2 and so on until you come to one where air is passing. This one should be left open, at the same time shouting down the number of the cylinder. While you are carrying on this testing I would have the manœuvring handle full over on fuel with the test cock open on the *fuel oil filter*, this would tend to carry away much of the air which happened to pass through the pump into the filter, after which I would proceed to open the air screws which are on top of each discharge valve plug of the fuel pump. Since these operations will take less than two minutes it is readily seen how easily an air lock can be dealt with without a stopping of the engine.

J.E.: Having found the cylinder causing the trouble, what must be done to cure it?

S.E.: Now the trouble is located the manœuvring handle can be put back to the previous position and the filter cock shut. It is more than likely that some foreign substance, in all probability a few shreds of packing have worked through, which may take a little time to clear. At any rate the cylinder can be put in and a careful watch kept on the pyrometer. If there is any sign of the pyrometer going back, which would mean that the line is not yet clear, it is advisable to disconnect the suction valve regulating gear for this particular cylinder, so that a full charge of oil can be pumped through the line for a few strokes only; this will cure the trouble.

Pyrometers.

J.E.: Why is it that pyrometers generally give readings that cannot be depended upon?

S.E.: There is supposed to be a certain length of wire connected separately to one common ammeter from a pyrometer of the thermo-couple type fitted to each exhaust pipe. When these wires are not accurately measured the indicated readings are not to be depended upon. Some adopt the practice of inserting resistance wire to give a common reading, but this is not satisfactory, because the readings very soon alter again. It is much better to work with the original wire without altering them in any way, and once the engine is balanced and tuned up a note should be taken of the various readings. Suppose, instead of getting a common reading, of, say, 350 for each cylinder, we read 360, 330, 350, 300, 320 and 300 respectively, we know by our tuning that all should be alike, so all that is necessary is to make a card with a note of these readings and fix it on to the ammeter so that the watch keeper will know what reading to expect when trying the ammeter switch.

Indicator Diagrams.

J.E.: What should be the actual height of an indicator diagram?

S.E.: The general practice is 37·5 m/m. (one m/m. in height being equal to one atmosphere) when running at normal load, but this is not necessarily a hard and fast rule. Our aim is for smokeless combustion and if 35 m/m. can give us this condition by all means keep the diagram at 35 m/m. height. Where an engine (when well warmed through) is compressing up to 34 atmospheres 35 m/m. is the ideal diagram, although much depends upon the power developed and the class of fuel oil in use.

J.E.: When dealing with the fuel pump you remarked that if one of a set of cards gave a higher mean pressure than others you would not alter the pump setting?

S.E.: That is so. For it does not always follow that the cylinder from which this particular card was taken was getting a greater charge of oil than the others; it all depends upon the shape of the card. If the contour is not as it should be some external influence is the cause.

J.E.: Suppose the contour is all that could be desired would you then have altered the pump setting?

S.E.: If the height of the card was in keeping with the others and the exhaust thermometer and pyrometer readings were higher than the others then there would be no alternative but to alter the pump; but this would not be necessary if the pump had been properly set in the first instance.

J.E.: If any of a set of cards gave a lower power than the others would you increase the fuel supply?

S.E.: If I had previously taken a set of cards and found the engine in an unbalanced state I might be tempted to alter the pump setting, but not before I had taken careful note of such things as the air inlet strainers, thermometers and pyrometers, state of exhaust valve, leaky pump glands and blowing of piston.

J.E.: How could you tell the state of the exhaust valve and the extent of a leaky piston?

S.E.: Only by the height of the compression line of the diagram. If an exhaust valve is very bad, "shooting" will be noticed at the plug on the exhaust pipe.

J.E.: Are the slits on the air inlet strainer a source of annoyance in having to clear them continually?

S.E.: Yes. But we now fit washers on the bottom studs between the flanges, this gives us all the air necessary without continually clearing the strainers.

J.E.: Can an indicator diagram be absolutely relied upon?

S.E.: It takes some practical experience to know when the indicator is deceptive. After having such experience you can safely rely upon the diagram taken off. After indicator diagrams have been taken off, and the engine balanced according to the mean pressure registered, a trace of smoke is often seen at the funnel and traced to some particular cylinder which gave a good shape diagram. The trouble here is that the fuel valve is opening too soon, and when diagrams have been taken off the lift of the valve was reduced in conformity with the height of the diagram. This to all appearances rectified the diagram, but the lift of the valve being reduced too much did not allow of sufficient blast air to enter the cylinder for complete combustion. Since the cam cannot be retarded until an opportunity presents itself in port, the right and left-handed regulating screw at the foot of the push rod should be lengthened to retard the opening, after which the valve can be given more lift to supply a little more air to the cylinder. Shortly before arrival in port it will be necessary to put the screw back to its original position, so as not to interfere with the astern opening of this valve.

J.E.: Are the draw cord diagrams as helpful as the mean pressure diagrams?

S.E.: These draw cord diagrams are most useful in checking the fuel valve timings and compression. It takes little or no experience for anyone to take off a draw cord diagram.

J.E.: There is sometimes a droop in the drawn line between compression and firing of a draw cord diagram. What is the cause of it?

S.E.: I have found that in some cases, although the fuel valve timing was correct, even with good pulversing of fuel, the droop was still in evidence. I have seen two cylinders from which draw cord diagrams were taken, one of which showed a pronounced droop. When we reached port it was decided to change the fuel valve from one cylinder into the other and *vice versa*. This did not alter the droop in any way, but it proved pulverising and timing was not the fault.

J.E.: Then what do you think was the cause?

S.E.: A leaky piston. The piston on its upward stroke comes almost to rest when the crank is somewhere about 23 degrees from the dead centre. Since the fuel valve begins to open at six degrees before the top, there is an infinitesimal period of time to be taken into consideration during which the piston is almost stationary, when the crank is travelling through the 17 degrees before the fuel valve opens. Bearing in mind an old engine where a few of the cylinder liners are scored it is an easy matter to lose a little of the compression pressure. Air escapes past the piston rings much more easily than gas from combustion; to prove this, put one of the worst blowing pistons on the top dead centre on the firing stroke, open the blast air to the open fuel valve and you will find the air escaping past the ring as fast as you can supply it. Watch this same piston when under working conditions and you will find the " blow past " nothing compared to the " blow past " on air.

J.E.: In that case, how do you manage to compress to 34 atmospheres with a clearance volume equal to that of a cylinder where the liner is not scored?

S.E.: We make the clearance volume less with a scored liner to compensate for the inefficient suction power of the piston, and loss of air on compression stroke.

J.E.: Isn't there a limit to which the cylinder clearance is adjusted?

S.E.: There is a designed clearance to which we must pay attention, but when wear takes place we must suit our requirements, and so long as we do not interfere with the opening and closing of the inlet and exhaust valves we can with safety go on decreasing the clearance volume until the indicator diagram tells us to stop. But when these alterations take place one must know one's indicator and also the condition of the pistons, since " blow past " has a decided effect on the diagram. When scored or enlarged liners create a continual " blow past " which cannot be cured, the liner should be renewed, because apart from the loss of efficiency, the machinery adjacent to this cylinder is always in a dirty state which is not encouraging in promoting cleanliness.

J.E.: Is it necessary to take frequently the timing of the engines?

S.E.: No. But it is advisable on joining a vessel to take the timings and compare the figures with the official readings recorded and supplied by the engine makers.

J.E.: Is there some quick method by which these timings can be taken without much turning of the engine?

S.E.: Yes. By making a timing diagram on a piece of cardboard the timing can be taken in two revolutions of the engine. Procure a piece of cardboard about 6in. square. In the centre draw a circle of 4in. diameter and mark it off into 360 degrees

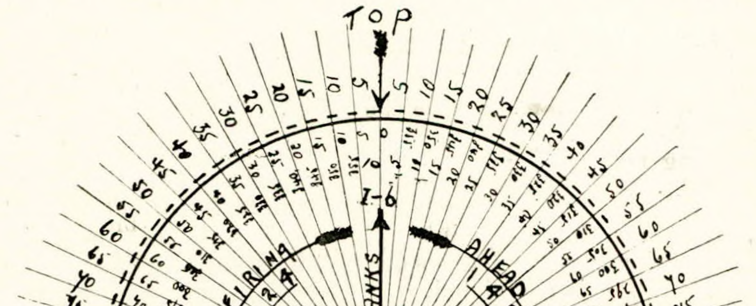


FIGURE (14)

Shows a timing diagram for the main motors to be used in combination with the motor flywheel readings. Cut around the dotted line and paste the square and disc on to a piece of cardboard, then replace disc in position again, making it free to revolve on the centre.

in spaces of five degrees. Through the centre draw a vertical line to represent the top and bottom centres. From another piece of cardboard cut a disc 4in. diameter, also marking it off into 360 degrees in five degrees spaces. Fasten this disc to the square of cardboard so that the disc can be made to revolve in the circle. This disc will represent the fly-wheel of the engine, and if three heavy lines are marked at 120 degrees to each other on this disc to represent the cranks 1 and 6, 4 and 3, and 2 and 5, their respective positions can be seen for any given figure on the engine fly-wheel reading. (See sketch).

J.E.: What do you mean by any given figure on the fly-wheel reading?

S.E.: Cut on the periphery of the wheel are 360 lines denoting degrees in one revolution and since only the top and bottom centres of the cranks are typed on the periphery of the wheel it is always difficult for the engineer who is attending the turning engine when timings are being recorded to know the actual position for any particular valve. This little diagram will simplify matters if we take the top centre of 1 and 6 crank as our starting point. From this zero point we will chalk-mark every five degrees on the wheel until 360 degrees or one revolution is completed.

J.E.: What is the sequence of the cycle when recording the timing?

S.E.:

Inlet opening ...	1	5	3	6	2	4
Fuel opening ...	6	2	4	1	5	3
Exhaust closing ...	1	5	3	6	2	4
Starting opening ...	6	2	4	1	5	3
Exhaust opening ...	3	6	2	4	1	5
Starting closing ...	3	6	2	4	1	5
Fuel closing ...	6	2	4	1	5	3
Inlet closing ...	4	1	5	3	6	2

The above is the ahead sequence for an engine firing:—

1 5 3 6 2 4

and the following is the ahead sequence for an engine firing:—

	1	4	2	6	3	5
Inlet opening ...	1	4	2	6	3	5
Fuel opening ...	6	3	5	1	4	2
Exhaust closing ...	1	4	2	6	3	5
Starting opening ...	6	3	5	1	4	2
Exhaust opening ...	2	6	3	5	1	4
Starting closing ...	2	6	3	5	1	4
Fuel closing ...	6	3	5	1	4	2
Inlet closing ...	5	1	4	2	6	3

It will be noticed that the ahead sequence for an engine firing:—

1 5 3 6 2 4

is also the astern sequence for an engine firing:—

1 4 2 6 3 5

and *vice versa*.

J.E.: Referring to the first mentioned sequence: If the No. 3 starting valve had closed when the fly-wheel had turned through 15 degrees after zero point, what would be the closing point for this valve?

S.E.: Set cranks 1 and 6 on the disc to 15 degrees after top centre, this will show No. 3 crank at 135 degrees after its own top centre, which will be the closing point for this particular valve. If No. 4 inlet valve was being timed and the order was given to stop turning after the flywheel had turned through 76 degrees after zero point, the diagram would show the No. 4 crank at 16 degrees after the bottom centre. This latter figure would then be the recorded closing point for this particular valve.

Auxiliary Motors.

J.E.: What do you consider the ideal dynamo engine? Is it the two, three or four-cylinder type?

S.E.: Each has its advantages and disadvantages. Having sailed with each type, I much prefer the four-cylinder units, because this is a slower running engine than the other types. From an overhauling standpoint the two-cylinder unit has the advantage over the others. Usually four of these engines are fitted, and since two are generally in use, two are left as stand by, this enables one to be thoroughly overhauled from top to bottom without any undue haste, consequently it is an easy matter to keep them in condition. The three-cylinder type, three of which are fitted, can also come under this heading, because only one is necessary for carrying the sea load. The four-cylinder type, although a much superior unit in every way, has the disadvantage of one stand-by engine only.

J.E.: I believe these four-cylinder units are very reliable and give good service.

S.E.: If carefully overhauled, yes. It is then an easy matter to run one for many weeks without a stop. But it is possible that careful and particular workmanship may not be expended in an engine of this description, because the work of overhauling the one and only standby engine is generally carried out in a hasty manner, indirectly affecting the unit at a later period. In new vessels building, a huge success could be made of them if an extra two-cylinder unit were installed to act as a standby. This small unit would be capable of taking the sea load, would run the heating and lighting in port when cargo is not working, and above all would give the operating staff confidence at sea when overhauling one of the four-cylinder units. Of course this extra unit would entail additional expense, but this would be repaid with interest before the plant was many years old.

J.E.: Some engineers believe that the auxiliary engines are the bane of a Diesel engineer's existence. Is that so?

S.E.: They certainly deserve close attention, but good workmanship and observation with regard to details will eliminate most of the troubles.

J.E.: What is the principal trouble?

S.E.: Being sufficiently particular to see that the engines are kept in proper tune. Apart from the tuning of the engine much trouble and annoyance is caused by inferior design and workmanship of engine seatings, and also in fly-wheels not being perfectly balanced; this is more noticeable in the two than in either the three or four-cylinder units.

J.E.: Taking a three-cylinder unit as an example, how would you go about the tuning?

S.E.: By making the cylinders develop a power equal to each other when working at no load. It often happens that after the valves and other parts have been overhauled difficulty is found in getting the engine to run without one or more of the cylinders miss firing, when all starting levers are in line with each other.

J.E.: What do you mean by *in line* with each other?

S.E.: Suppose No. 1 cylinder starting lever was in three notches, the other two starting levers should be in three notches also. Inattention to this important point is the principal cause of erratic working in these engines. This engine at 38 to 40 atmospheres blast pressure on no load, should run on all three cylinders without any miss firing when the starting levers are in three notches from neutral position.

J.E.: Assume an engine was tested under these conditions and one of the three cylinders was continually miss firing, what would cause this?

S.E.: If compression and the fuel pump setting were correct it would most likely be faulty pulverising, but this can be proved in the following manner:—Commence raising the blast to 60 atmospheres, as the pressure is rising *cut out* one cylinder at a time by by-passing the fuel and bringing the starting lever back into neutral position. The engine will now be working on one cylinder only, the starting handle of which will be full over.

J.E.: Is this starting handle moved over to full from the third notch to give more fuel to the working cylinder?

S.E.: This starting lever has nothing whatever to do with the *fuel supply*, it merely controls the lift and *opening period* of the fuel valve. From neutral position the more notches you move the starting lever the more *blast air only* you supply to the charge of oil delivered by the fuel pump, which is controlled by a governor coupled up by a series of levers to the pump tappet spindle shaft. Now that this engine is running on one cylinder I am going to leave you to it, and anything you want to know I will tell you.

J.E.: There is a heavy *knocking* in the cylinder.

S.E.: What is the blast pressure?

J.E.: Fifty-eight atmospheres.

S.E.: Is there any smoke coming from the test plug on your exhaust pipe?

J.E.: Yes; but not much.

S.E.: Is it black, blue, or white?

J.E.: It is black.

S.E.: Increase the blast pressure further until the knocking stops. Now, before we begin testing let me impress upon you that black smoke means that more blast air is required, and white smoke means that too much blast air is being supplied. Blue smoke is the most difficult to rectify because there are so many different things which account for it; of course we all know the usual blue smoke caused by miss firing, but it is the faintest blue smoke that is sometimes seen at the test plug that we are concerned about, and lubricating oil escaping into the combustion space and a leaky fuel valve are two of the most important contributors.

J.E.: The knocking has stopped and the blast is now 62 atmospheres.

S.E.: Go up to the switch board and switch the engine on to the voltmeter.

J.E.: I have done this and the voltage is 150.

S.E.: What should be the voltage?

J.E.: 220 volts.

S.E.: Well, make it 220 volts now.

J.E.: What shall I do to make it 220 volts?

S.E.: Each engine has a resistance fixed at the back of the switch board which is controlled by a handle at the front of the board to enable us to *take out* or *put in* resistance as required. Now turn the handle to cut out some resistance, an amount sufficient to increase your voltage from 150 to 220 volts.

J.E.: I have cut out some resistance and the voltage is now 220.

S.E.: And what about the resistance?

J.E.: It is about 3/4ths out.

S.E.: Now the resistance is 3/4ths out, the blast is steady at 62 atmospheres, the starting lever is full over, the voltage is 220, and the revolutions are about 306 per minute. It is obvious that if this working cylinder gives us satisfactory conditions for these figures, we must expect similar conditions from the other two cylinders.

J.E.: For these figures No. 2 cylinder was equal to No. 1 cylinder, but No. 3 cylinder only gave 300 revolutions and 190 volts with all other conditions correct.

S.E.: We can now give this No. 3 cylinder more fuel, which will speed up the engine and incidentally rectify your revolutions and your voltage. But since the fuel pump was set before we started the testing, it can hardly be possible that this weak cylinder is receiving less fuel than the other two. It is a very bad habit to jump to conclusions when testing, always try and look for something which will prove one thing against another. In this case if we *change* the fuel pipes from *No. 1 cylinder* to *No. 3 cylinder* and *vice versa*, and the working conditions of No. 3 are now similar to the working conditions of No. 1 cylinder, it's obvious that the fuel pump was inaccurately set as this simple test of crossing the pump line has shown.

J.E.: This changing of the pipe line is certainly a true check on the setting of the fuel pump.

S.E.: After you try each cylinder separately try two together, when you have recorded figures for this running you can try the engine on all three cylinders and commence tuning up on *no load*. For this last condition the blast will only be about 38 to 40 atmospheres, resistance about six studs out from *all in* and the speed will be somewhere about 320 revolutions at 220 volts with all starting levers in three notches from neutral position.

J.E.: Why are you so particular about all levers being in three notches?

S.E.: When an engine is on load and has not been properly tuned it often happens that one lever is in three notches, one in four notches and another full over. This is very misleading to the inexperienced engineer who generally finds himself in a muddle at not knowing what to do to steady up the engine if knocking takes place, whereas, if the engine was properly tuned with the levers in three notches he would find no trouble in moving about his levers to suit the varying loads, because if he had to move No. 1 lever two or three notches he would also move the others a corresponding amount.

J.E.: Supposing the resistance was *full out* on no load, with engine revolution at the desired speed. What would be done to correct this?

S.E.: A little more compression should be put on the governor spring. This will increase the fuel supply to each cylinder, the effect of which will be to speed up the engine and incidentally raise the voltage, which can be reduced by turning the handle to *put in* more resistance.

J.E.: What causes one of the cylinders to suddenly stop firing?

S.E.: Sometimes a piece of grit sticks on the face of the N.R.V. in the fuel valve immediately after a delivery of oil, and this gets washed off again at the next delivery. In the time between these two delivery strokes of the fuel pump a small amount of blast air escaped past this valve face into the pipe line. In a few more strokes of the pump this air is compressed and expanded in the pipe line, resulting in no discharge of oil for this particular cylinder.

J.E.: How do you get rid of this air?

S.E.: By opening the air screw of the fuel pump discharge valve plug or by by-passing the fuel for a few minutes.

J.E.: Why didn't this escape of air cause an air lock in the pump?

S.E.: Because the amount of air that passed before the valve tightened itself again was not strong enough to force a passage into the suction side of the pump.

J.E.: Is this phenomenon peculiar to the auxiliary engines?

S.E.: Yes. But at manœuvring of the main motors you no doubt have experienced a few false starts when leaving port. In almost every case this can be traced to the air not being effectively cleared from the fuel valve pipe line. When pumping through you should never be satisfied until a solid stream of oil flows from all the by-pass pipes. This is a simple operation but it should always receive the serious consideration it deserves.

J.E.: Are air locks more frequent in the auxiliaries than the main motors?

S.E.: Yes, but it is generally caused through neglect. When an auxiliary is shut down it should be always left in a starting position with the fuel pump handle at the *on* position.

J.E.: Why at the *on* position?

S.E.: To ensure that the pump suction valves are on their seats.

J.E.: What would happen if the suction valves were off their seats?

S.E.: If an N.R.V. was leaking slightly on any of the fuel valves and someone by mistake opened the valve on the blast bottle, which supplied blast air to the fuel valve, a portion of this air would pass the leaky N.R.V. into the fuel pump through the suction valves which were held *off* their seats by the fuel pump starting lever being carelessly left at the *off* position.

J.E.: If an N.R.V. is leaking slightly, will it have an effect on the working of that particular cylinder?

S.E.: Not if the leak is slight. But on starting up it may take some little time to clear the air out of the pipe line; this generally takes some little time with the hand plunger pump fitted for priming purposes, but it can be successfully cleared by keeping this particular cylinder running on air with the fuel by-pass valve open and the air screw open on the pump discharge valve plug. When a solid stream of oil is seen you can shut down the air screw and by-pass valve, and throw over the lever, when the cylinder will commence firing immediately.

J.E.: You say I can start up an engine with a slightly leaky N.R.V., provided I take this course of clearing the line?

S.E.: Yes. Supposing the leak was in No. 3, keep No. 2 and 3 on air, pick up on No. 1, put in No. 2. When No. 2 has picked up put No. 3 starting lever back into neutral position and shut off the starting air. You are now at liberty to clear the air from No. 3 pipe line. I hope I have made everything quite clear to you.

J.E.: I understand you exactly regarding the starting of the engine, but not with regard to the stopping?

S.E.: To stop the engine put the fuel pump handle into stop position.

J.E.: But what stops the engine?

S.E.: Putting the fuel pump handle into the stop position operates the suction valve tappet spindle shaft, the effect of which is to hold all suction valves off their seats, consequently the fuel oil instead of being discharged into the fuel valve takes the line of least resistance and is discharged back again to the pump suction chamber.

J.E.: On several points I am not clear. The engine has stopped due to the suction valves being mechanically held off their seats. The blast air valve from the blast bottle is still open. One of the N.R.V. valves is leaking. Why doesn't the blast air create an air lock if there is an open passage from the blast bottle to the suction side of the pump chamber?

S.E.: There is not an open passage. The fuel pump discharge valve stops the air from going to the suction side of the pump.

J.E.: Why didn't you mention this discharge valve before?

S.E.: It is such an important valve that I took a roundabout way of drawing your attention to it. When these discharge valves are leaky you must be very particular to see that the fuel pump handle is put at the *on* position as soon as the engine has stopped.

J.E.: What about the blast air bottle valve, is that to be left open?

S.E.: That valve should always be shut before the engine comes to rest. This ensures the air being cut off from the fuel valves, and also helps to pump up the blast pressure, since the rotating engine is still pumping into the blast bottle.

J.E.: Everyone seems to shut down these engines in a different manner. How would you suggest shutting one down?

S.E.: I would blow through the bottle drains to make sure that all water was cleared, then shut up the compressor drains and put the starting levers into the second notch and bring the blast pressure up to 60 atmospheres. When this pressure was reached I would shut off the fuel and open the by-pass valves, and before bringing the starting levers into neutral position I would put them full over so that the high blast air blowing into the fuel valve would help to clear the pulveriser. A few revolutions would suffice for this. Once I had brought the levers back into neutral position I would go down and shut the blast bottle valve to the fuel valve and wait there until the engines stopped before shutting the other bottle valve. I would then put the fuel pump handle at *on* position and then turn the engine into starting position, after which I would open the lubricating by-pass valve and also the compressor drains. I would not shut down the circulating water until about 30 minutes after the engines had stopped.

J.E.: There are three valves on each blast bottle. (1) Inlet valve from the compressor. (2) Outlet to the fuel valve blast air line. (3) By-pass valve to the main line. On taking over the watch it is customary to examine the blast bottles of the stand-by engines, to ensure that the blast pressure is at 60 atmospheres. Which two of these valves should be opened when testing this pressure.

S.E.: To each bottle is fitted a pressure gauge, and on the end of the main pumping line another gauge is fitted, called a master gauge. If valves 1 and 3 are opened the pressure of the bottle will be shown on the master gauge, but not on the bottle gauge, and *vice versa* if 2 and 3 are opened.

J.E.: Which are the correct valves to open?

S.E.: 1 and 3.

J.E.: Why?

S.E.: Because opening No. 2 would fill the blast air line on the engine, which might create an air lock to the whole system if the engine conditions were as previously mentioned. So always remember never to open this valve without first looking at the position of the fuel pump handle. This also applies when starting up an engine.

J.E.: How would you start up an engine?

S.E.: I would open the fuel pump stop valve and commence pumping through.

J.E.: Why didn't you mention the shutting of this stop valve when you were explaining your method of stopping an engine?

S.E.: Because this is another important valve to which I wish to draw your attention. No doubt you have experienced much *banging* (lifting of escape valves) when an auxiliary engine has been started up, and perhaps you never gave a thought to the detrimental effects which the engine was likely to suffer from such violent explosions. Probably you would put this down to over-charging when pumping through?

J.E.: I should imagine that would most likely be the cause.

S.E.: That of course is one of the causes, but there is a more serious one. If the fuel valve by-pass valves were left *shut* and the fuel pump stop valve left *open*, and the springs of the fuel pump discharge valve, and N.R.V. in the fuel valves had not received attention for some time, there is nothing to prevent the fuel oil from the highly placed settling tank finding an easy passage to the fuel valve. When this fuel valve is full, the oil will overflow into the blast air line, the serious consequences of which will be obvious. This has happened on more than one occasion with disastrous effect to the engine. It is, therefore, absolutely necessary to always close this pump stop valve and to see that the by-pass valves have not been left shut by mistake. After pumping through I would see that the compressor drains and lubricating by-pass valve were open. Blow down the blast pressure from 60 to 45 atmospheres, the lower pressure being most effective for starting. All auxiliary engines have certain peculiarities with regards to firing. No matter how well balanced one of these engines are you often find that one may be a better firing cylinder than the other. When you know of any weak firing cylinders always keep them on starting air until the *last*, when the increased speed will help matters considerably. If this engine is going on the switchboard in parallel with another machine raise the blast to the same pressure as on the other before paralleling up, after which the blast can be altered on each machine to suit the load carried.

J.E.: On entering or leaving port when the manœuvring and starting air compressor is about to be started there is usually a shouting of orders from one engineer to another, which often confuses one or the other of them. Is this shouting necessary?

S.E.: Certainly not. Where the staff is efficiently organised there is no need for it. If you are the man to start the compressor you take command of the job. There will be two machines running, one on load and one running light, both of which are attended by one engineer. These we will call 1 and 2 respectively: Proceed in this manner: Tell the engineer that you are going to put No. 2 on the switch-board in parallel with No. 1. This will make him prepare No. 2 for the extra load by increasing his blast and giving an extra notch to the levers. *You, standing there, will see him do this.* It is now time for you to signal the electrician on the switch-board (who is watching you) to parallel up, after which you immediately go to start up the compressor. Do not be in a hurry to close the compressor cooler drain, because the noise from them blowing is heard on the other side by the engineer attending to Nos. 1 and 2 machines, and if he is not alert enough to notice the No. 2 machine taking the load, informing him that the compressor is started, he will certainly hear the noise, which will advise him to give the levers extra notches if you have been too quick for him before he got the blast pressure up. While the drains are blowing look at the gauges to make sure that the *water* and *oil* pressures are correct. You can now close the drains and walk round to the other side to see how the engineer has treated his engines. If he is contented, tell him you are going to increase the load on his machine, and before you have returned to load the compressor he will have moved the levers of both machines full over and will now be regulating his blast air in conformity with the load carried.

J.E.: These instructions seem easy enough for one to carry out. Now what about stopping the compressor?

S.E.: Before stopping the compressor ease the load and run it light. Then tell the electrician what you are going to do, and also tell him the number of the machine that you are going to stop. Then go round and tell the engineer on Nos. 1 and 2 that you are going to stop the compressor and inform him that when he hears the noise which made him throw the levers over when the compressor was started, this time it will be the signal for him to stop No. 2, since that was the machine started up to help to take the load of the compressor. In this case there is no need to signal the electrician, the dropping of the load on the ammeter when you stop the compressor will be his signal to pull out the machine you told him of.

J.E.: Suppose I forgot to open the cooler drains, how would the engineer know that the compressor was stopped?

S.E.: He would be waiting on the platform of No. 2, and the misfiring and erratic working of the engine running light under loaded conditions would inform him that something was wrong, and without another thought he would immediately stop his machine.

J.E.: Doesn't No. 1 machine require any attention after No. 2 is taken off the switch-board?

S.E.: No. This will make it quite clear to you. Before No. 2 was started No. 1 was the working machine at say, a load of 200 amperes. Since it takes about 200 amperes to drive the compressor we would have on the ammeter of each machine a load of 200 amperes. Now if you had to stop the compressor and left both machines running you would now have 100 amperes on each machine. If the machines were allowed to run for a time on this load it is obvious that we would require to lower the blast and bring the levers in a notch or two to suit the reduced load. If, on the other hand, we stop the compressor and immediately pull out of parallel No. 2 machine, that would leave No. 1 running on the same load, as when the compressor and two machines were running. Therefore it is obvious that No. 1 can look after itself.

J.E.: When working cargo and the load is fluctuating between 200 and 500 amperes, do you believe in running one or more machines?

S.E.: This is an argument that has many sides to it. These three cylinder machines are on full power at 450 amperes, and a well balanced machine can take a fluctuating load of 150-550 amperes without any trouble whatsoever, but against that we have the opinion of many that it is more beneficial (although not so economical) to run two machines in parallel.

J.E.: Why is it more beneficial?

S.E.: Because a Diesel engine has a longer life running at a light or medium load than when running loaded.

J.E.: Do you hold that opinion?

S.E.: When speaking of an engine working by itself, Yes. But if in parallel with another machine then my opinion is No.

J.E.: For what reason?

S.E.: When two machines are working in parallel it is sometimes difficult, if they are not properly tuned, to keep them from miss fire when the load has momentarily dropped from, say, 250 to 50 amperes on each machine.

J.E.: So that if one machine only was working the load would momentarily drop from 500 to 100 amperes?

S.E.: That is so, but there would not be any miss firing.

J.E.: That is curious, it appears to me that if the two machines in parallel miss fire at 50 amperes the other machines will miss fire at 100 amperes.

S.E.: The machines in parallel do not miss fire at 50 amperes, only *one* of these machines miss fire. This is what actually happens. When the load drops, one of the machines, due to a sluggish governor, drops or holds the load more than another, consequently we find that instead of 50 amperes on each machine one has 100 and the other nothing, with the result that miss firing takes place on the lightest running machine. It is for this reason that I am in favour (where the load will allow) of running one machine only.

J.E.: Is miss firing very detrimental to the engines?

S.E.: Very. It is mainly responsible for the several ailments which affect the engine at later periods, and is the principal cause of twisted cranks.

J.E.: What is the cause of the governors of these engines becoming sluggish?

S.E.: Slack connections in the various levers from governor to fuel pump, which should always be attended to at the first opportunity. The governor pawls (the ends of which work on rollers) should never be allowed to have flats worn on them, because these flats are one of the main causes of a sluggish governor. When flats do occur the ends should be turned down and screwed 3/8 in. and fitted with new ends, it is then a simple operation to renew the ends when they begin to wear. Cheese-headed pins are used to fasten the governor hood to the base, to facilitate overhauling these should be replaced by hexagon-headed pins.

J.E.: When you spoke about the main fuel pumps you said that slackness of connections, etc., would not matter much?

S.E.: That is so. But the auxiliary fuel pump is a totally different proposition and deserves very close attention, since the levers are connected directly to the governor, the least slackness of any connection affects the working of the engine considerably. At the overhauling of the pump it should always be disconnected from the engine and taken to the vice bench, where a thorough examination can be made of the various parts.

J.E.: Have you any means of testing the suction and delivery valves after they have been ground in?

S.E.: Yes. Adapters should be made, and with the use of the armoured air hose (which should be fitted by the engine makers to one of the starting air reservoirs in all Diesel ships) each valve can be tested with air at a pressure of 25 atmospheres.

J.E.: Why should these discharge valves require more attention than those on the main motors?

S.E.: On the main we have an extra N.R.V. fitted on the pipe line between the pump and the fuel valve, which is not fitted on the auxiliary motors.

J.E.: When setting the auxiliary fuel pumps must we be careful to set them in a manner similar to the mains?

S.E.: Yes. But as I remarked previously, on the auxiliaries you can always cross over the pipes at any time to prove the pump.

J.E.: Have you ever seen one of these machines running when one of the cylinder escape valves would suddenly lift. What is the cause of this?

S.E.: Lubricating oil escaping into the combustion chamber. When this happens the particular cylinder affected is not taking its share of the work due to the escaping lubricating oil creating imperfect combustion, consequently much carbon is deposited on the piston crowns and becomes red hot. After a time this red hot carbon is liberated, resulting in the escape valve being lifted due to pre-ignition. The faint bluish smoke to which I previously drew your attention would be emitted from the test plug of an engine working under this condition.

J.E.: Can't this imperfect combustion be rectified?

S.E.: Yes, by increasing the fuel supply to this particular cylinder. This will then help the combustible mixture which will not be influenced to the same extent by the escaping lubricating oil, since it is burnt instead of being deposited on the piston crowns. Bear in mind that this is only a temporary measure, because this cylinder, when it begins to fire properly, will be doing more than its share of the work.

J.E.: The black oil sometimes seen running down the exhaust pipes. Is that fuel oil or lubricating oil?

S.E.: Lubricating oil. You see that only when a machine is running for a time on light load. If the same machine were on 3/4ths or full load all that black oil you see running down the pipes would be burnt instead of wasted.

J.E.: I noticed when you were explaining the starting and stopping of an auxiliary engine how particular you were at opening up this by-pass valve on lubricating oil filter.

S.E.: Yes. That is a little valve which creates much trouble if neglected. So you now know that it should be regulated to 6 or 7 lbs. pressure, which is quite sufficient, and that the valve should always be tried to see that it is full opened out before the engine is started.

J.E.: Does this lubricating oil escape past the piston rings into the combustion chamber?

S.E.: Only when the splash guards are fitted in *two halves.* The oil is thrown on to the cylinder walls through the small space left open by the two halves not being fitted *closely* together. Most of the lubricating oil escaping into the combustion chamber is caused by badly fitted gudgeon pins, the design of which leaves much to be desired. Engine makers should never allow these gudgeon pins to be passed until their tightness is tested to at least 10 lbs. per square inch water pressure. This would probably find out many defectively fitted gudgeon pin *feathers* through which most of the oil passes.

J.E.: Isn't there some defect with the washer which fastens the gudgeon pin in position?

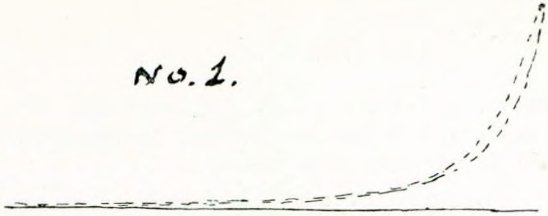
S.E.: Yes. There are three tapped holes in this washer which are bored right through adjacent to a ground face. When the taper end of gudgeon pin is slack the oil will travel along the taper and work through these holes, which leaves open a clear passage for the oil. (See sketch.)

J.E.: How do you manage to stop the oil from escaping past these defectively fitted pins?

S.E.: By using a mixture of thin red lead and varnish. This keeps the pin tight under water test, but it is not a job because in a few weeks, leakages commence again.

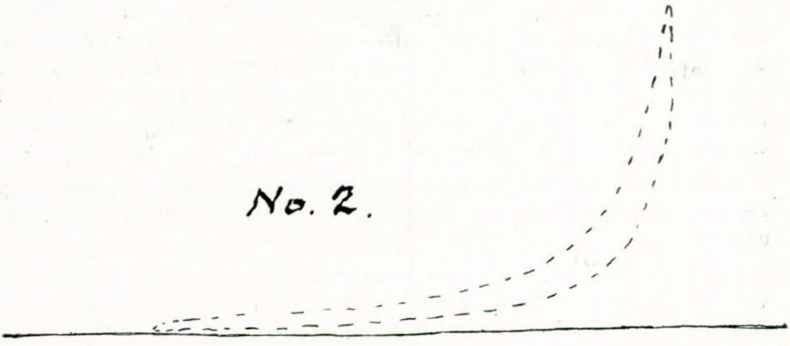
J.E.: These gudgeon pin washers which fasten the gudgeon pin in position often slacken back. How is that?

S.E.: The old story. Lack of observation with regard to detail. A small dowel pin is fitted in this washer to hold in position a copper washer through which the gas screwed pin is fitted. When this pin is hardened home the copper washer is bent up to help to keep the pin from slackening back, it has been found that when tightening up this pin the sharp edges of the hexagon head cut into the copper washer, which tends



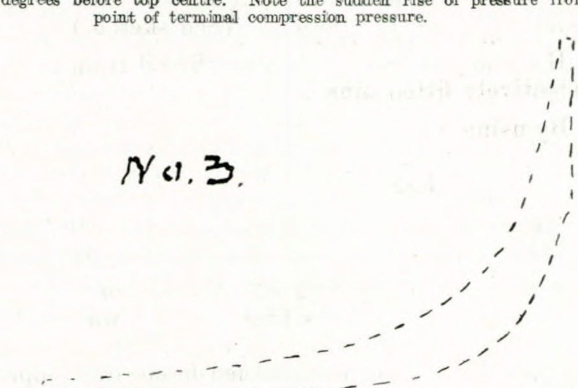
Compression. Cold Engine leaving Port.

Compression diagram starting from rest. The engine was set to enable the diagram to be immediately taken in order to show the compression pressure obtained from a slow moving piston.



42 Blast. Cold Engine. no 2 Stard.

Firing diagram starting from rest. Blast pressure 60 atmospheres. Fuel valve opening 5½ degrees before top centre. Note the sudden rise of pressure from the point of terminal compression pressure.



60 Blast. Cold Engine. Starting starting. no 2 Star'd.

Firing diagram starting from rest. Conditions similar to No. 2, but with blast air pressure 42 instead of 60 atmospheres.

to elongate the dowel pin hole. If this edge is rounded off, the turning effort of the pin working on the copper washer is considerably reduced. (See sketch.)

J.E.: That long bolt which is fitted to keep the wedge in position on top of gudgeon brasses often breaks. How do you account for that?

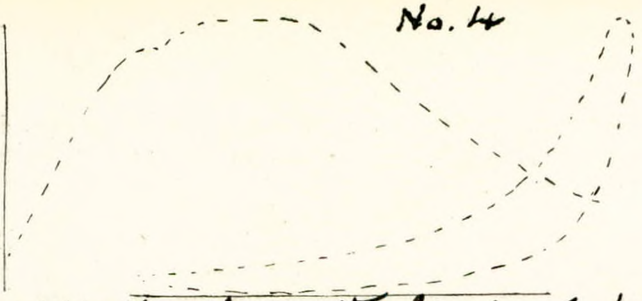
S.E.: That bolt keeps in position the *wedge piece* which is necessary to bind the gudgeon pin brasses in position. When the gudgeon pin is being fitted and is a little too tight for the brasses some would slacken back on this bolt rather than fit a thin liner between the two half brasses. To make a liner for the brasses would take at the most an extra hour or so and would make a *solid job*. Compare the time to make this liner and the time of having to take out the piston again to renew a broken bolt which was caused in the first instance through the wedge piece not being screwed home *solid*.

J.E.: How can we obviate the continual *firing up* of the L.P. receiver on the blast air compressor. Is this caused by lubricating oil?

S.E.: By more frequent attention to the L.P. discharge valve. Where splash guards are inefficiently fitted much lubricating oil escapes up the piston guides into the L.P. receivers and carbonises on the walls and also in the holes of the L.P. discharge valve, consequently a point is reached where the reduced area of these holes is not sufficient to take away the air, resulting in the compressor firing up. Since it is a very simple operation to change this valve it should be changed frequently. When the valve is withdrawn the receiver walls should be examined for carbon and if there is any quantity it should first be scraped off with special scrapers made for the purpose, after which the engine should be run on starting air when every particle scraped off will be blown out. This should always be carried out at sea where plenty of starting air is available. When a compressor begins to fire up and it is not convenient to shut it down, disconnect the L.P. drain and let the air blow freely, at the same time working the vacuum lubricator to give a little extra oil to the H.P. piston. This will cure the trouble and allow you to carry on until you are in a position to stop the engine.

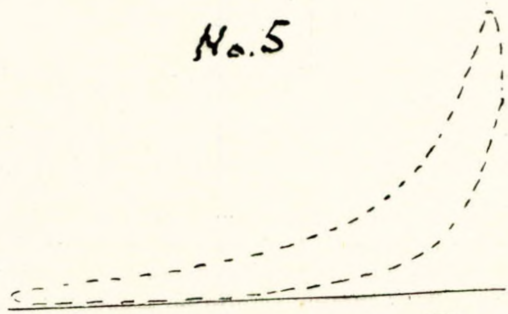
J.E.: When fuel oil contains water, has this any effect on the working of the auxiliary motors?

S.E.: When the fuel oil contains water the *after* engine is the most affected. The symptoms are the lifting of the cylinder



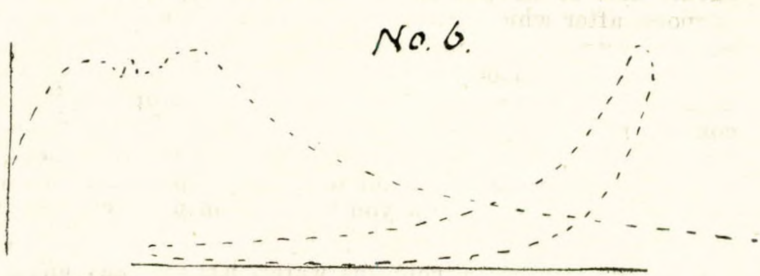
no 2 star'd. 24 hours after leaving Port.
Blast 60. Fuel valve opening – 5½° B.T.
Same valve setting as for No 2.

Working diagram obtained from the same conditions as No. 2 after the engine was thoroughly warmed and working at normal load. Compare this diagram for the *sea load* with No. 2 when starting from rest.



50 Blast. Valve opening on top centre. Starting from rest.

From the same cylinder. Starting from rest, blast air pressure 50 atmospheres, fuel valve opening on top centre.



50 Blast. Valve opening on top Centre. Engine running.

After firing twice, this diagram was taken on the third firing, conditions similar to No. 5.

These diagrams demonstrate the valve settings for manœuvring as compared to the settings for a sea load. No. 5 and 6 show the advisability of opening the fuel valves on the top centre for manœuvring purposes.

escape valves in a manner similar to what happens when much lubricating oil escapes into the combustion chamber, but with this difference, all the escape valves lift, since all three cylinders are affected.

J.E.: Why should the after engine be the only one affected?

S.E.: Since the ship is generally down by the stern all the water which separates from the oil is lodged in the lowest part of the oil feed line. It is now the practice to fit from the settling tank to the auxiliaries a separate feed line on some part of which a filter and water collector is fitted. This fitting is a most useful one for water troubles on the auxiliaries, which are now non-existent, where ships have not yet been fitted with this separate line the lowest part of the line feeding the *after* auxiliary engine should be broken and a cock fitted to drain away occasionally, any accumulation of water.

J.E.: I thought settling tanks were responsible for getting rid of most of the water?

S.E.: So they are. But sometimes careless handling of the oil when starting on a full double bottom tank puts much water into the settling tank. When starting on a full tank the transfer pump should first pump into the *deep tank* for at least ten minutes, after which a glass of oil should be taken from a cock which should always be fitted to the discharge pipe of the pump. This sample should be placed on top of one of the hot cylinder covers to settle for inspection.

Operating.

J.E.: I believe that the economical operating of a Diesel ship depends to a great extent on the capabilities of the engineers?

S.E.: That is largely true. But the methods which were satisfactory when dealing with steam engines are not applicable to oil engines. A more accurate method is necessary, and much closer attention to small details is required in dealing with Diesel engines which will not respond to any rough and ready treatment, and they can only be expected to give good results when they are cared for by those who understand them.

J.E.: You said that men are appointed for special work?

S.E.: Yes. When at sea each of the senior watch-keepers will overhaul the spare main fuel valves between them without in any way interfering with each other's valves, and one of these senior watch-keepers will also overhaul the auxiliary fuel valves, and the other the main starting valves. The refrigerating engineer will do the compressor valves and a junior engineer will overhaul the auxiliary exhaust valves, and the main exhaust valves will be done by the donkeyman and passed before assembling by the second engineer. By making each of these engineers shoulder a certain amount of responsibility they feel that if anything goes wrong with any of their valves that it is themselves who are to blame, since no one else interfered with their work. This is one of the successful methods of overcoming valve troubles. In the early days everybody had a hand in the overhauling of the various valves, with the result that we were never out of trouble.

J.E.: It seems that successful running depends much on having sufficient spare gear and tools?

S.E.: Naturally. It is no secret that the lack of spare gear and good tools creates a *hard worked and dispirited staff*. This is one of the many points that a chief engineer must be alive to, because it is one of the deciding factors of economical operating.

J.E.: So before expecting good and particular workmanship from the staff you believe in providing efficient tools to aid them in their efforts?

S.E.: Yes. This not only tends to invite particular workmanship, but also gives just cause for complaint when workmanship is not of a high standard. In some ships, engineers are working all day long on some little job which could easily be done in a few minutes if the proper gear had been in the ship. It is work of this nature which overworks and disorganises the engine-room personnel and causes the junior engineers to be exhausted with "field days." I don't believe in any of the staff wasting time by making many of the small parts peculiar to this engine, which can be supplied by the engine makers at a very moderate figure.

J.E.: Do you believe in the present practice of stowing spare gear and tools away in lockers and drawers?

S.E.: The fitting of lockers and drawers in a Diesel engine-room store and workshop is a waste of time, material and space. All particular spare gear and tools, etc., should be fitted in specially prepared racks surrounding the workshop and store and other suitable parts of the engine-room. Even taps and dies and other special tools which are supplied by the engine makers in special boxes should be fitted into special racks to enable those responsible for their care to see that all are kept in good condition.

Reminiscences of Early Diesel Engine Troubles and Mishaps.

CONTRIBUTED BY ROBERT McKINNON (Member).

WITHIN recent years, many valuable contributions have been made to our knowledge of the prime mover that is revolutionising marine transport; and which, in view of the limited supply of fuel available for the increasing needs of mankind, is destined to play an important part in its conservation, and, therefore, in the prolongation of our civilisation.

The following remarks, which in certain respects have a general application, relate particularly to the Burmeister and Wain single-acting four-stroke type of Diesel engine, to which the writer's practical Diesel experience has been confined; and while it is not expected that they will add materially to the acquired knowledge of the Diesel engine—their interest being chiefly of an historical character—it is hoped that the narration of troubles experienced in the earlier stages of its development, and of the means subsequently adopted to obviate them, will be of interest and profitable to those engaged in the work of perfecting, and of manipulating, one of the greatest inventions of modern times.

Fuel Oil.—The fuel oil question being of paramount importance, we shall consider it in the first instance; as, no matter how excellent the design of an engine, its mechanical qualities will show to little advantage if fuel oil of an unsuitable nature be used in it.

About five years ago the writer was appointed to a new motor ship—the twin engines of which developed 3,600 I.H.P. —and joined her ere she left the builder's yard on the Clyde. A supply of Mexican fuel oil was taken at the yard; but, as the quantity available did not meet our full requirements, the remaining bunker space was filled with a Persian oil, obtained from a ship belonging to the same Company, which arrived at this yard prior to her departure to undergo extensive repairs and alterations. These different classes of oils, of which there were about equal quantities, were not mixed together in the bunkers.

At the outset, the settling tanks were filled with the Mexican oil, it having been decided to use up this class first; and as all went well during the trial trip, which was of short duration, the builder's staff were disembarked; the vessel thereafter proceeding to Middlesbrough by the Northern route.

About four or five hours after leaving the Clyde, however, trouble was experienced with a number of the main engine exhaust valves, which became so sluggish in operation that it was found necessary to assist them to regain their seats by means of spare springs suspended from overhead beams and connected to the ends of the valve spindles. This improvised arrangement enabled us to effect a non-stop run to Middlesbrough; although some difficulty was experienced, under the circumstances, in manœuvring the vessel into dock.

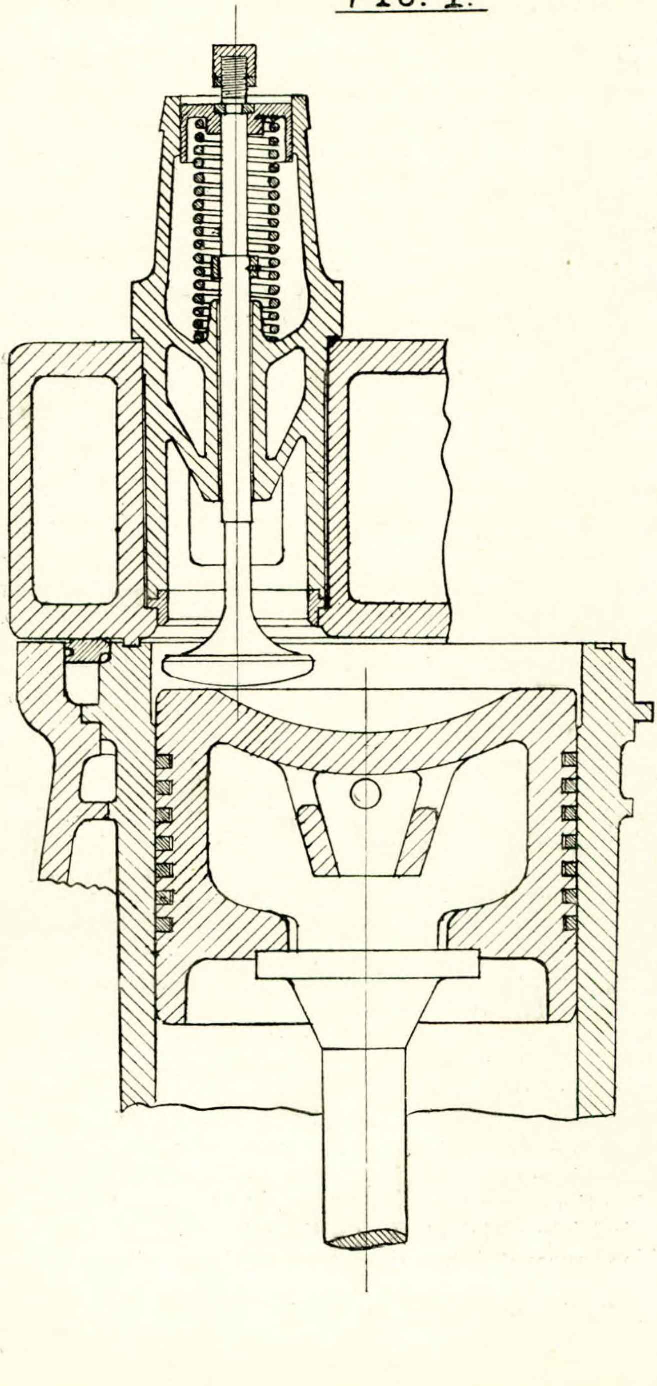
On removing these valves, great difficulty was experienced in withdrawing the spindles from the valve casings owing to the presence of a stiff and somewhat gummy deposit in way of the spindle guides, while almost all of the twelve exhaust valve spindles were found to be bent.

The cause of this was at first not quite apparent; it being supposed that the valves were received in such a condition from the builders; however, it was observed later that the under side of the head of each valve, with bent spindle, was considerably indented in one place near its circumference; from which it was inferred that the excessively sluggish movement of the valves, due to the deposit in the spindle guides, resulted in their being overtaken by the engine pistons; the ensuing impacts causing the spindles to bend. (See Fig. 1.)

The spindles of these valves were, accordingly, straightened; and an arrangement of strong-backs provided to enable the ship's engineers to straighten them in event of the trouble recurring. It was also deemed advisable to have the Mexican fuel oil analysed, as, in our opinion, it was unsuitable for use in a Diesel engine.

The analysis revealed this fuel to have an asphalt content of 14 per cent. and 2 per cent. of sulphur, its specific gravity being 0·91 and flash point 175°F.; and as it was obvious that a considerable portion of the former constituent left the cylinders in an unburned state, and was deposited in the exhaust valves, we suggested that the Mexican oil be sold as boiler fuel, and oil of a more suitable nature obtained before proceeding to the East. On the ground of expense, however, this suggestion was not carried; and we were obliged to make the best of an unenviable situation by pumping equal quantities of the Mexican and Persian oils into the settling tanks, thereby reducing the percentage of asphalt. This expedient was found to work fairly satisfactorily; although the trouble recurred in a milder degree at intervals so long as the Mexican oil lasted.

FIG. 1.



In the latter circumstances, the percentage of asphalt in the fuel would be approximately 7% to 9%; and, as the trouble was not entirely eliminated under these conditions, it would appear that fuel oil having an asphalt content in excess of, say, 6 per cent. should be considered unsuitable for use in engines of the four-stroke type.

Although the introduction of the centrifugal separator to the motor ship calls for the modification of this view; nevertheless, the supply of fuel oil of high asphalt content to vessels fitted with engines operating on the four-stroke cycle is to be deprecated; while it would appear desirable that motor ships be furnished with the equipment necessary to determine the chemical properties of the fuel oils offered at the various bunkering ports.

It should, perhaps, be mentioned that the asphalt did not act deleteriously on the piston rings; although, as the purely Mexican oil was used for a short period only, it should not be assumed that the rings would have been unaffected by it, had this oil been used unalloyed over an extended period.

The danger to the ship from the use of such a fuel will be apparent; as, should the exhaust valve spindles get bent in this way when manœuvring in restricted waters, or when passing through, say, the Suez Canal, the loss of compression in the cylinders, due to the valves not lying squarely on their seats, would probably result in continuous miss firing of the fuel charges, and consequent failure of the engines to respond to the telegraph.

Explosions.—The engineer of a motor ship has under his control a vast quantity of potential energy, stored in the fuel oil; and, unless great care and intelligence be exercised in dealing with it, an inordinately large amount of this energy is liable to be released instantaneously, with disastrous results.

About six years ago, when the M.S. ——— was proceeding to an anchorage in Hong Kong harbour, a terrific explosion, which was heard all over the harbour, occurred at the first manœuvre with the port engine.

The writer, who was on the manœuvring platform at the time, narrowly missed being struck by a large fragment of the casing of the starting air slide valve of that engine, situated about ten feet above the manœuvring platform in line with the control levers, and which controlled the compressed air supply from the main starting receivers to the starting valves on the cylinder heads. Fortunately, no one was injured by the explosion; and, as the engines were stopped and the anchors cast shortly after its occurrence, no mishap resulted to the vessel.

On investigation of the matter being made when the smoke had cleared, the casing of the starting air slide valve was found to have been completely shattered, while a number of pipes and handrails in the vicinity were much distorted. Fortunately, we were in possession of a drawing of this complicated part, so that, thanks to the skill and industry of the Chinese, the ship was ready to proceed on her voyage five days after the mishap occurred.

Several theories were advanced to account for the explosion; but none appeared to meet the case; and the problem, therefore, remained unsolved at that time.

About six months later, in the same ship, another violent explosion occurred in the starting air system of the main engines, at the first manœuvre with the starboard engine, when entering another Eastern port.

On this occasion, the air cylinder of the starting valve on the after cylinder head of the starboard engine, which formed a dead-end of the starting air range, burst; no one, fortunately, sustaining injury.

The trouble was thus assuming serious magnitude; and still no clue to the cause was apparent: however, we began to suspect the presence of fuel oil in the starting air system, and, therefore, decided to take steps to ascertain whether this was the case, prior to manœuvring the engines on approaching the next point of call.

Accordingly, the engines were stopped while still several miles from port; and the starting air valve on the after cylinder of one of the main engines withdrawn. On admitting air to the starting system of this engine, our suspicions were confirmed; a considerable quantity of fuel oil being ejected from the range—steps having been taken, before opening the air control valve, to prevent any fuel, if present, being blown into the cylinder from which the starting valve was removed, by fitting a wood plug at the bottom of the starting valve orifice in cylinder head. Before proceeding into port, a similar procedure was adopted with the other main engine; in the starting air system of which a small quantity of fuel oil was also located.

The problem that then arose was to determine how the fuel was finding its way into the starting air range; and this we solved shortly after the discovery of the cause of the trouble.

Figure 2 (a) shows the type of starting valve employed in the cylinder heads; from which it will be seen that the removable casing "C" is held against its lower seat by means of a stout bridge "B"; the lower joint being metal to metal, while a ring of slightly compressible material, such as Klingerit jointing, was inserted between the bridge spigot and bottom of recess at top of the starting valve orifice in cylinder head to prevent the escape of starting air.

It will be observed that extreme care had to be exercised in jointing up this starting valve; as, if too thick jointing material were used at top, the bottom jointing face of valve casing would not abut against the corresponding face on cylinder head; while, if too thin jointing were used at the top, it would not be compressed, with the result that there would be leakage of starting air from the system when admitted for manœuvring purposes.

It will also be apparent that, when under way at sea with the starting air shut off, fuel oil, accumulating on top of a cylinder head due to a slight leakage past the fuel valve spindle gland, would readily find admittance to the starting air system by way of the top starting valve joint, were this too thin; and, passing down the narrow space between starting valve body and wall of cylinder head orifice, would enter the annular space around lower part of the valve stem and accumulate on the head of the valve, while a quantity might pass into the starting air range through air passage "P" in the cylinder head.

After the detection of fuel oil in the starting air range, close examination of those joints revealed the source of the trouble, as, in consequence of a few of them not having been sufficiently compressed, oil from the fuel valve spindle glands had been admitted to the air starting system by way of them.

The danger of explosions from this cause was subsequently minimised by the substitution of a round rubber ring for the thin flat jointing material at the top joint of starting valve—see Fig. 2 (b). The trouble, of course, could have been obviated by the formation of a ridge or lip around the top of starting valve orifice in cylinder head—see Fig. 2 (c); but such precautions are generally suggested only by dear-bought experience.

FIG. 2. (a)

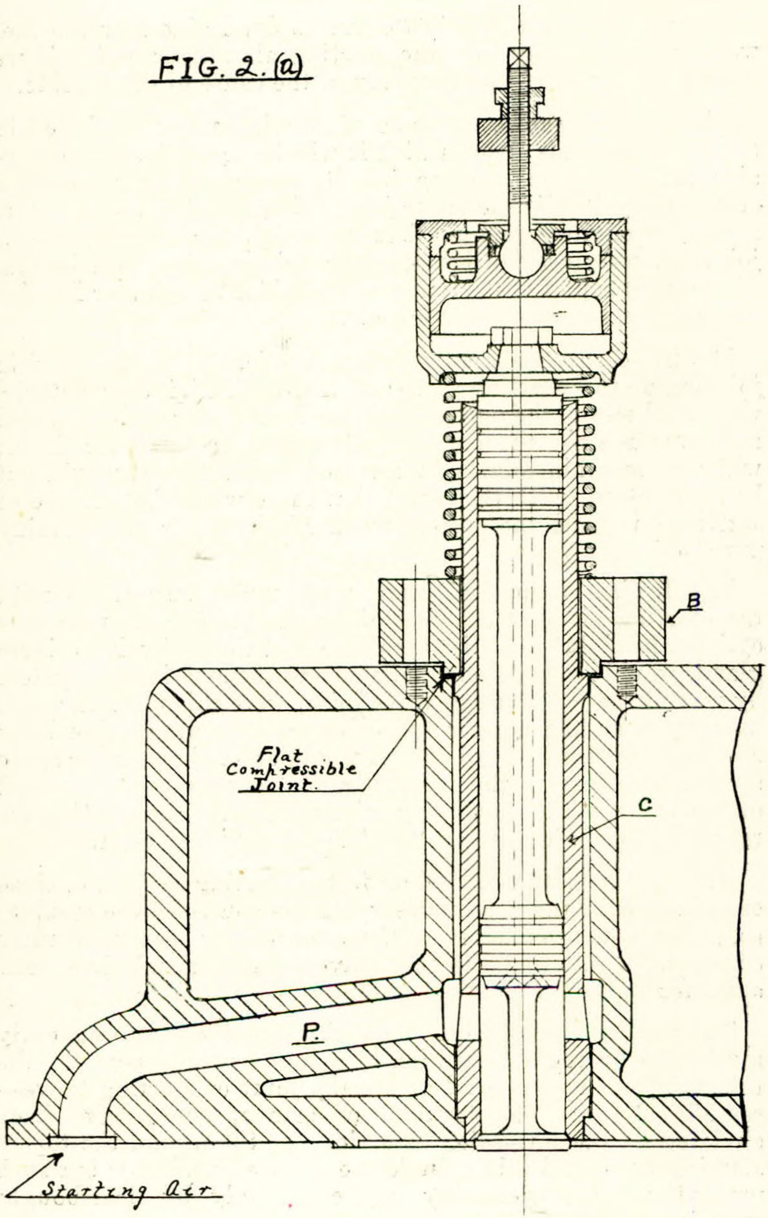


FIG. 2. (b).

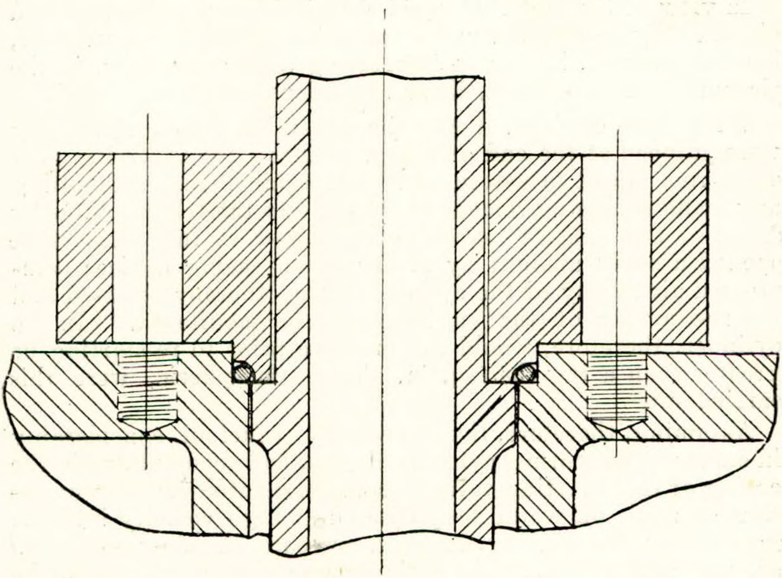
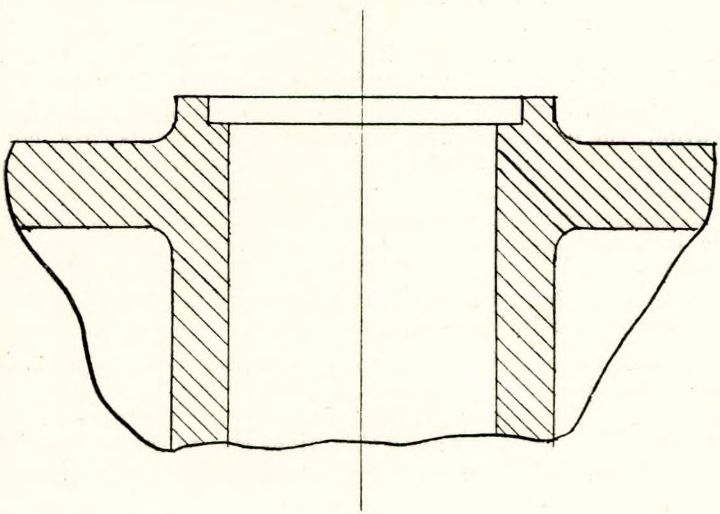


FIG. 2. (c).



In view of the fact that rapid chemical union of fuel oil and air can only be effected under conditions of high temperature, local or general, it may be interesting to consider how, in the circumstances described, these explosions occurred.

With these engines, it was the general practice, when they were stopped at the end of a run, to throw the reversing gear over from ahead to astern and back again before attempting any manœuvring movement in response to the telegraph, as by doing so the exhaust valves, lying closed at the time, were opened a small amount by means of an easing device; compressed air in any of the cylinders being thereby released and the re-starting of the engines greatly facilitated. In no cylinder, therefore, would the pressure prior to re-starting be greater than a few lbs./sq. in. above atmospheric, were this carried out.

Assuming, therefore, that a quantity of fuel oil, considerably in excess of the normal cylinder charge, had accumulated in the casing of that starting valve to come first into operation on re-starting: this oil would be projected at high velocity into the cylinder by the starting air at a pressure of about 350 lbs./sq. in.; and, although the cylinder temperature would not be sufficiently high to ignite the oil, it is possible that the heat generated by the impact of the oil and air on top of the piston would be great enough to bring this about; while, owing to the presence of a large volume of starting air at high pressure, it is quite conceivable that the whole of the entrained fuel would be burned almost instantaneously, and an exceedingly high pressure generated.

On the other hand, supposing the engine had been turned through a half or a full revolution on starting air before the starting valve, in which the fuel oil had accumulated, came into operation, the corresponding cylinder would, in this case, contain air at high pressure and temperature—almost the maximum compression temperature; as each starting valve opens a few degrees after the completion of the compression stroke, when in operation—and, on the starting valve being opened, the fuel would immediately be brought into contact with the highly heated air in the cylinder; in which case an explosion would almost inevitably result.

In regard to the mishaps cited, it is worthy of note that, in either case, disruption occurred at, what might be considered, a dead-end of the system; remote, in the first case at any rate, from the explosion centre: and this gives rise to the suggestion that the failure of these parts was brought about not directly by the excessive pressure generated, but by dissipation of the momentum of the products of combustion, impelled at an enormous velocity from the explosion centre along the starting air range.

Considering the large volume, or weight, of air in the starting system, it will be evident that, were a sufficient quantity of fuel oil to find its way thither, a disastrous explosion might readily be brought about; safety valves being of little use as safeguards against such contingencies.

On another occasion, about seven years ago, while the writer was engaged in the testing department of a large Diesel engine works, a terrific explosion occurred when a large marine engine of the Burmeister and Wain type was started. Investigation of the matter revealed that the explosion had occurred in the blast air system; the solid drawn steel blast air main—lying horizontally along the front of the engine about ten feet above the manœuvring platform—having been shattered locally near the middle of its length. Fortunately, there were no casualties, although several persons narrowly escaped serious injury.

No satisfactory explanation of the cause of the explosion was offered at that time; and, as the writer was not engaged on this particular engine, no definite statement regarding the circumstances attending it can be made; however, there is little room for doubt that the mishap was due to the blast air pressure falling below the maximum compression pressure in the engine cylinder.

This may have been due to defective compressor valves, or to the hand-operated master valve at manœuvring station—controlling the blast air supply from the reservoirs to the fuel valves on cylinder heads—not having been opened sufficiently. In either case, when the fuel valve to come first into operation was opened, instead of the fuel being injected into the cylinder in a finely divided state by the blast air at a pressure of about 900 lbs./sq. in., the highly compressed and heated air in the cylinder would enter the fuel valve casing and there ignite the fuel.

As combustion under such circumstances would, in all probability, take place instantaneously, an exceedingly high pressure would be generated; and the products of combustion driven at a high velocity from the fuel valve casing. Owing to the resistance that would be offered to their passage into the engine cylinder by the pulveriser of the fuel valve, the movement of the gases along the blast air range would be much less restricted, and, consequently, the greater portion would be driven along the latter.

Here again it would appear that, as the part of the system at which disruption occurred was considerably stronger than the

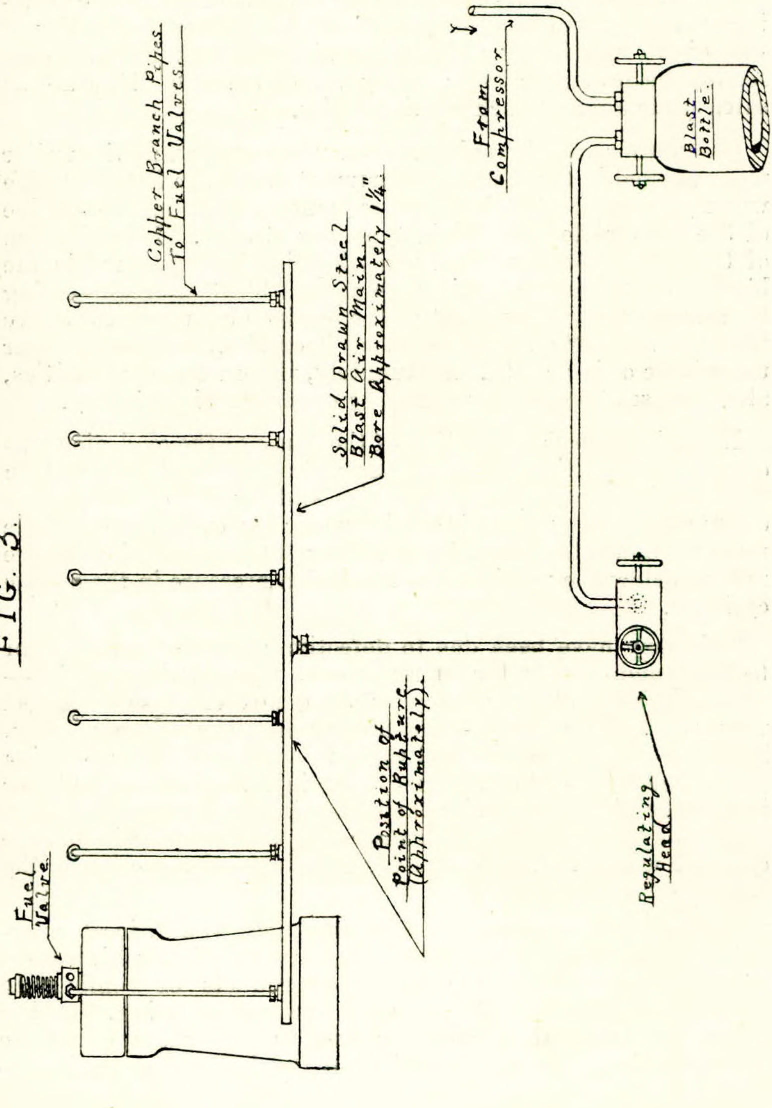


FIG. 3.

remainder, the failure was brought about not by an excessive uniform pressure, but by an action analogous to that of water-hammer in a range of steam piping. (See Fig. 3.)

That the mishap resulted from ignition of fuel oil there can be no doubt, as a considerable quantity of this oil was found in the burst blast pipe.

An alternative explanation of the cause of this explosion might be offered. Assuming that the blast air pressure was adequate to inject the fuel into the cylinders, it is possible the explosion originated in one of the cylinders, in consequence of the fuel valve on its head having been primed unduly with fuel oil prior to starting—it being the practice to prime each fuel valve independently, by means of a small hand pump incorporated in the fuel pump chamber, to facilitate the starting of these engines.

In these circumstances, the pressure generated in the cylinder might possibly have attained a considerably higher value than that of the blast air; and, consequently, before the fuel valve closed, a portion of the products of combustion, together with any unburned fuel in the valve, would be driven into the fuel valve and thence along the blast air range; disruption occurring in the manner previously suggested.

Another mishap of a somewhat similar nature, which occurred fully a year ago on a large motor ship, with twin single-acting Burmeister and Wain engines developing 6,400 I.H.P., was brought to the writer's notice.

In this instance, the casing of one of the main engine fuel valves burst while the engines were being manœuvred (see Fig. 4); and, although no particulars of the case were submitted, it is not improbable that this failure also was due to the ignition of the fuel charge in the casing of the fuel valve, in consequence of the blast air pressure falling below the maximum compression pressure in the engine cylinder.

Great as has been the responsibility of the marine engineer in the past, reflection upon the matters enumerated in this section will reveal the extent to which his responsibility has been augmented by the advent of the Diesel engine; the greater and ever present danger associated with the new system of propulsion calling for the further development of those qualities, that have distinguished him in the past, and to which the pre-eminence of Great Britain as a maritime power is so largely attributable.

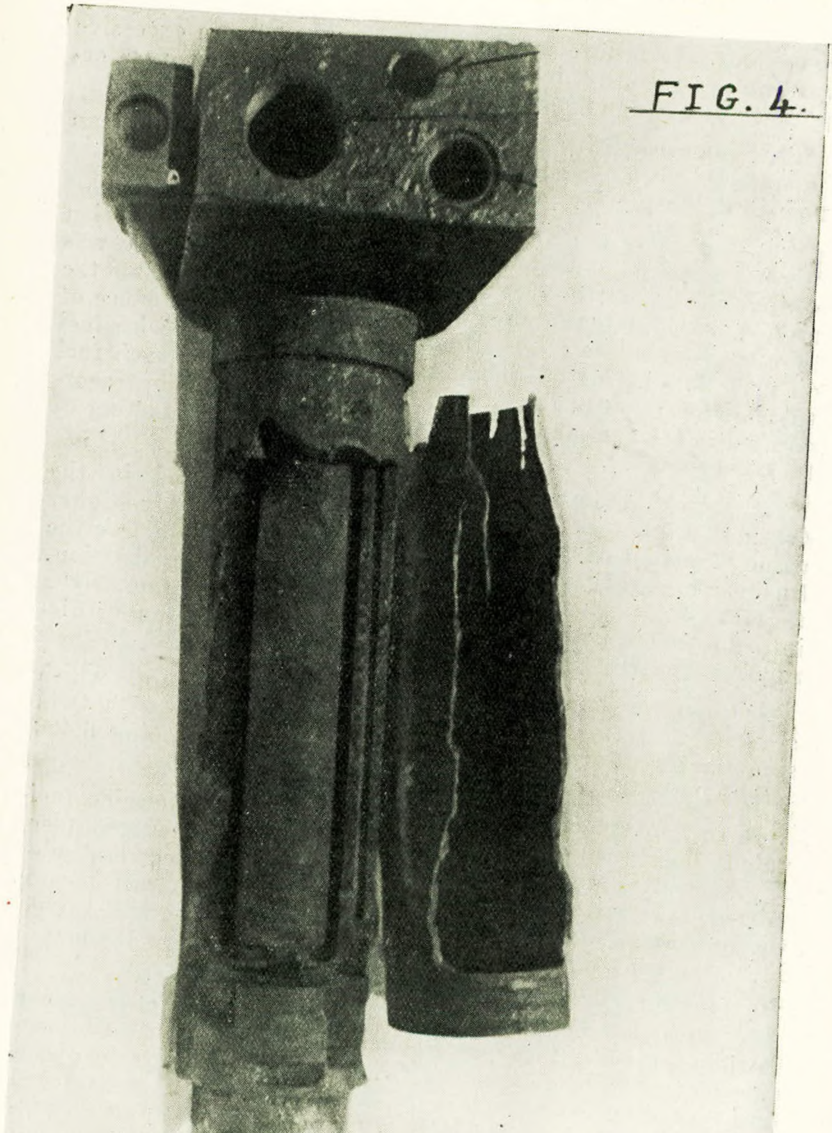


FIG. 4.

(A) *Mechanical Troubles.*—The great attention paid, within recent years, to the mechanical details and arrangement of the Diesel engine has resulted in the elimination of many of the troubles associated with the earlier stages of its development; and, while there is much that might yet be accomplished in the way of simplification and standardisation, the reliability of the best types can now bear comparison with that of the steam engine.

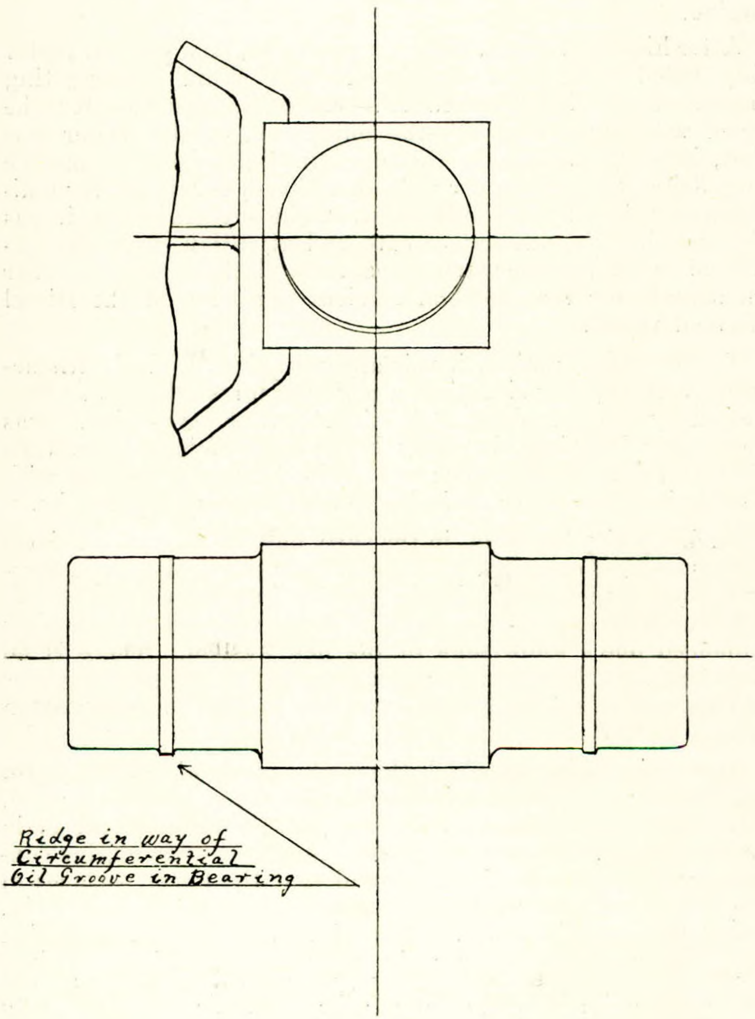
After his first voyage, fully six years ago, on a pre-war motor ship fitted with twin Burmeister and Wain single-acting engines developing 3,000 I.H.P.—one of the first vessels to be fitted with these engines—the impression of the writer was that, if the tenure of the Diesel engine in the field of marine propulsion was to be other than short-lived, considerable modification of the system would require to be effected; while it was obvious that the amount of care and attention, normally bestowed on the average marine steam installation, was altogether inadequate for the safe and efficient operation of the Diesel engined vessel.

To the writer's mind, it was apparent that, to deal satisfactorily with the new situation, a policy of forestallment, as suggested by the old adage, "A stitch in time saves nine." was necessary: it being obviously better to anticipate, and take steps to evade trouble, then to acquire facility in dealing with it after its occurrence; desirable as the latter quality may be.

Such a policy involved, in the early days at least, a considerable amount of exertion, both mental and physical, which, however, was amply repaid in the greater immunity from breakdown acquired; while the thought thus expended considerably enhanced one's knowledge of the new system. There is no doubt that much of the trouble experienced in the earlier stages of this engine's development, was due to lack of appreciation of this matter.

The motor ship, on which the writer made his first voyage, had been in commission for several years ere he joined her; and, at the end of that voyage, an extensive overhaul of the main engines was undertaken. On opening up the main engine connecting rod bearings, it was found that the lower surfaces of the crosshead pins were worn from 1/8 in. to fully 3/16 in. below their original size. (See Fig. 5.) Forced lubrication was employed throughout; the main and auxiliary engines being supplied from the same double-bottom tank, while the crosshead bearings received their oil supply by means of oil holes through the centre of the connecting rods; a circumferential oil groove, about $\frac{1}{4}$ in. deep, being cut in the middle of each white metal lined crosshead bearing, to allow of the passage of oil, through holes in the crossheads, to the guides.

FIG. 5.



Accordingly, the crossheads were taken ashore and the pins turned; as several had to be reduced about $\frac{3}{8}$ in. below their original diameter, it was considered expedient to shrink steel sleeves over these to maintain the normal area of bearing surface. The white metal lining of the whole of the crosshead bearings was also renewed.

Regarding the cause of the excessive wear, it transpired that the lubricating oil, then in the double-bottom tank, had not been renewed for several years—centrifugal separators had not been introduced on motor ships at that time; the supply having been maintained by the addition of small quantities of fresh oil from time to time.

As previously stated, the main and auxiliary engines were served by the same lubricating oil pump; the oil returning from their respective crank chambers to the double-bottom tank, whence it was drawn by the pump.

The cylinders of the main engines were isolated from the crank chambers by means of distance pieces, through which the piston rods passed; but the auxiliary engine cylinders were open to the crank cases, their pistons being of the trunk type. Consequently, small particles of unburned constituents of the fuel, passing the piston rings, were probably deposited in the auxiliary engine crank chambers from time to time, and thence carried by the lubricating oil to the double-bottom tank.

It would appear that the whole of these particles of foreign matter did not settle to the bottom of the tank; some being carried with the oil through the bearings, and there acting as an abrasive.

To prevent the recurrence of this trouble, the auxiliary lubricating system was isolated from that of the main engines; each auxiliary engine—four in all—being fitted with a geared-wheel pump and small separate lubricating oil tank.

Although fairly effective, lubricating oil filters were employed, these alone cannot be relied upon to remove exceedingly minute particles of foreign matter from the oil; and, therefore, to reduce wear at the bearings to a minimum, periodic renewal of the oil, together with the regular use of centrifugal separators, is necessary.

This experience serves to show the costliness of lack of foresight, both in design and management; and emphasises the importance of maintaining the lubricating oil in a thoroughly clean state, as, had this matter received due attention in the case referred to, the effect of the faulty arrangement would have been considerably mitigated.

(B) *Air Compressors.*—Air compressors being important adjuncts to the Diesel engine, a word or two concerning them may not be out of place.

In the case of the auxiliary Diesel engines of the ships on which the writer served, the fuel injection and starting air was supplied by single-acting vertical two-stage compressors, each driven from an overhanging crank at the forward end of its engine crank shaft.

Air was compressed from atmospheric pressure to eight atmospheres in the L.P. cylinder, and from eight to sixty atmospheres in the H.P.

Not infrequently, trouble was experienced owing to the failure of the valves, which were automatic in action; the H.P. delivery valve being the most frequent defaulter, while the failure of this valve was immediately indicated by the violent escape of air past the safety valve on the L.P. receiver.

It will be apparent that, when an H.P. delivery valve broke, the load sustained by the L.P. piston, and consequently by the compressor crank, would, if the safety valve on the L.P. receiver were overloaded, or of inadequate relieving capacity, be considerably in excess of the normal load: indeed, if that safety valve had become inoperative as the result of, say, carbonised lubricating oil from the engine crank chamber choking its orifice, the full blast pressure would be borne by the L.P. piston, as, in these circumstances, the top of the H.P. suction valve would be exposed continuously to the full pressure of the air in the blast receiver; and consequently the pressure of the air discharged from the L.P. cylinder would require to be slightly in excess of the blast pressure before the H.P. suction valve could be lifted and the air allowed to pass from the L.P. to the H.P. cylinder. Under such circumstances, the breaking of the compressor crank pin or the bursting of the L.P. cylinder are not unlikely contingencies; and, although no such mishaps occurred in the writer's experience, he learned, about two years after leaving the first motor ship on which he served, that the compressor crank pin of one of her 200 H.P. auxiliary engines had broken.

On several occasions, when an H.P. delivery valve broke, the pressure gauge on the L.P. receiver was observed to register from twelve to fourteen atmospheres with the safety valve blowing freely; and although, in the first instance, this would not be likely to result in breakdown, its frequent recurrence might fatigue the material of the compressor crank and give rise to incipient cracks, which, if undetected, would ultimately result in failure.

The most common cause of the breaking of the compressor valves was excessive lift; and the writer found that, by reducing the lift of these valves. in certain instances, by about a millimeter only, their durability was increased tenfold.

When it is considered that the sudden failure of a single auxiliary engine compressor valve might result in serious damage to the ship, when manœuvring in restricted waters, the importance of frequent inspection and careful adjustment of the lift of these valves cannot be overestimated; while the attention required by the compressor safety valves needs no further emphasis. The adjustment of these valves to blow at pressures not exceeding, say, 10% in excess of the maximum working pressures would appear good practice; while the necessity for the provision of ample relieving capacity is obvious.

The stepped pistons of these auxiliary compressors being of the trunk type, the under side of the L.P. piston was open to the engine crank chamber; consequently, no matter how carefully the piston rings were fitted, a small quantity of oil vapour, from the engine bearing oil in crank chamber, found its way into the L.P. cylinder during the suction stroke (special compressor oil, in small quantities, was used for piston lubrication). This vapour carbonised readily; the bulk of it being deposited in the L.P. receiver where, occasionally, it was ignited by the hot air, thus causing the receiver walls to attain an unduly high temperature. It was, therefore, found necessary to keep the receiver drain slightly open in order to reduce the accumulation of carbonised oil, and also to clear the receiver frequently, otherwise burning of the oil and overheating of the walls resulted.

Although the writer did not encounter such a mishap, fracturing of the receiver, under such conditions, is liable to occur; while an explosion in the L.P. cylinder or receiver is not an unlikely contingency.

In view of this, the substitution—in the case of single-acting compressors—of pistons attached to piston rods and crossheads, with diaphragms between them and the crank chambers, for those of the trunk type, appeared a desirable rearrangement.

(C) *Crank Shaft Failure.*—The following breakdown did not come within the scope of the writer's experience; but, as it occurred on a ship belonging to the Company, with whom he served,

FIG. 6. (a).

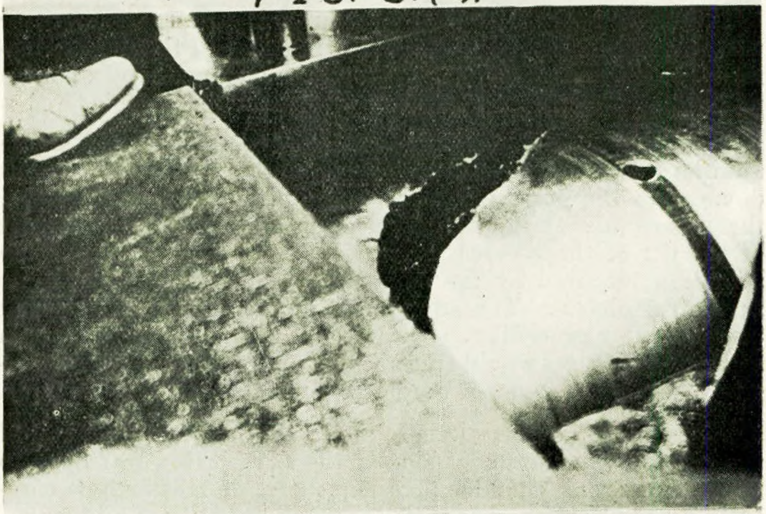
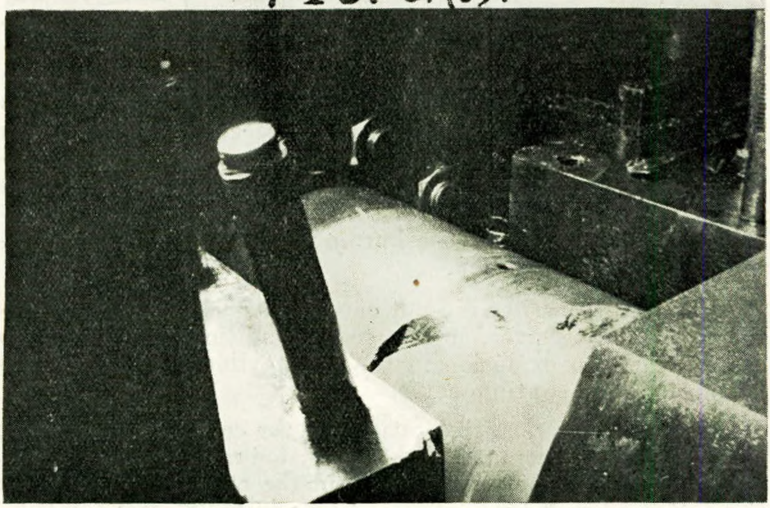


FIG. 6. (b).



it will, in view of its important character, be recounted and commented upon here. While crossing the Indian Ocean, homeward bound from the East, the crank shaft of one of the main engines of the M.S. ——— broke in way of No. 6 bearing, near the after end of the engine. (See Fig. 6, a and b). This vessel is fitted with twin engines of the Burmeister and Wain single-acting type, each having eight cylinders developing 3200 I.H.P., while the crank shafts are 460 millimetres, approximately 18 inches, in diameter.

Although the other engine was unaffected, it was considered expedient to put back to Colombo for survey, where, as equipment for handling so big a job was not available, permission was obtained to proceed home under the power of one engine.

Accordingly, the vessel resumed her voyage and reached this country safely in due course: she maintained a speed on the homeward journey of about eleven knots; her full speed under both engines being from fourteen to fifteen knots.

Not being in a position to discuss the circumstances attending this breakdown, the writer can only offer comments.

So far as could be gathered, the material of the broken shaft appeared to be flawless; while, as will be seen from the photographs, Figs. 6 (a) and 6 (b), the form of the fracture is such as would be caused by a combined bending and twisting moment.

It appears to have been established, in the case of the ordinary materials of construction, that, although having met the specified mechanical test requirements when manufactured, they are liable to fail in service if exposed, over a more or less lengthy period, to stresses, which, although in excess of the maximum safe working stresses, are considerably lower than those that represent the ultimate strengths of the materials; especially if the load be of an alternating nature.

As is generally known, the principal stresses sustained by the crank shaft of a Diesel engine are torsional and tensional in nature; and these attain their combined maximum intensity when the crank has moved approximately thirty degrees from the top dead centre on the expansion stroke. The torsional stresses alternate each power stroke, the reversal taking place at the end of the compression and beginning of the expansion strokes; while the tensional stresses are set up by the large bending moment produced by the pressure on the crank during the latter part of the compression and first part of the expansion strokes.

The normal working conditions of these shafts are, therefore, severe; and, should the working stresses be augmented to a considerable extent in consequence of, say, a leaky fuel valve, it is quite conceivable that the continuance of such a condition over a prolonged period might fatigue the shaft and give rise

FIG. 7 (a).

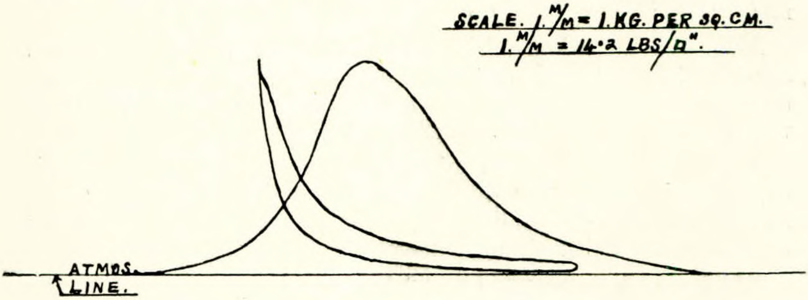
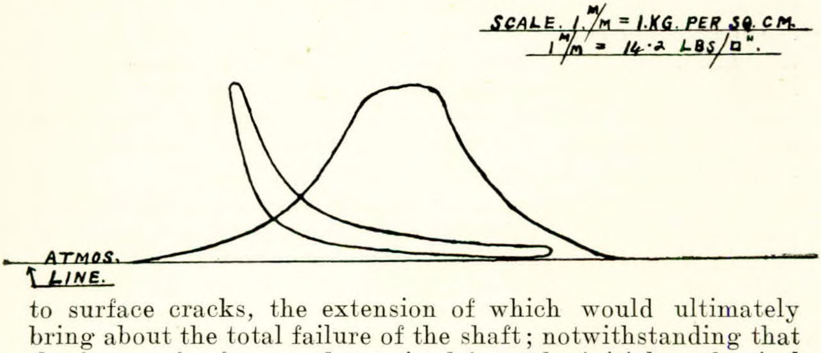


FIG. 7 (b).



to surface cracks, the extension of which would ultimately bring about the total failure of the shaft; notwithstanding that the factor of safety, as determined from the initial mechanical test results, appeared adequate to meet such contingencies.

The following indicator diagram Fig. 7 (a), which was obtained at sea by the writer while indicating the main engines of M.S. ———, will serve to illustrate this point.

It will be seen that the maximum pressure in the cylinder, from which the diagram was taken, was 610 lbs./sq. in.; this being nearly 100 lbs./sq. in. above the normal maximum cylinder pressure, as shown in diagram Fig. 7 (b).

Inspection of the superimposed draw card, Fig. 7 (a), will show that the undue rise in pressure commenced before the end of the compression stroke; clearly indicating that oil leakage past the fuel valve had been taking place.

On this valve being changed, the normal diagram Fig. 7 (b) was obtained; while examination of the faulty valve showed that the leakage could only have been slight.

On this occasion, there was no external evidence that leakage past the fuel valve was occurring; although, on another, similar trouble was, in the first instance, suggested by moderate gas leakage past the piston rings of the affected cylinder; the indicator diagram, which was subsequently taken, proving the correctness of the inference.

It will, therefore, be apparent that, with a greater fuel valve leakage than that indicated, the crank shaft stresses would be augmented to such an extent that, if the leakage were not detected, failure of the shaft might ultimately occur.

A small spring loaded safety valve was fitted to each cylinder head; but the tendency was for these to be adjusted to lift at pressures considerably in excess of the maximum working pressure, to obviate leakage and consequent burning of the valve faces and seats.

On starting up these engines, pressures exceeding the normal maximum pressure are frequently generated in the cylinders, as a result of the fuel valves being primed by small hand pumps —incorporated in the fuel pump—prior to starting; and, were the relief valves set to lift at pressures only slightly in excess of the normal cylinder pressure, they would lift frequently on the engines being started. Should, then, any particles of carbon find lodgment between the valve faces and seats, continuous leakage and consequent burning of the valves and seats takes place: the temptation, therefore, is to so adjust these relief valves that they will not lift when the engines are started, unless unduly high pressures are generated.

This practice, however, is to be deprecated; and it would appear sound policy, in view of the danger attending fuel valve leakage when under way, to limit the loading of such valves so that they will lift at a pressure not greater than, say, ten per cent. in excess of the normal maximum cylinder pressure.

From the foregoing, it will be obvious that the duty of maintaining the crank shaft in accurate alignment is one of the most important devolving on the engineer of a motor engined ship; as the detrimental effect of a leaky fuel valve on the shaft will be greatly accentuated should it be inaccurately aligned.

While the development of the internal combustion engine since its inception has been phenomenal, there is still considerable scope for exploration in this field; and, although all of the views expressed in this paper may not receive general acceptance, it is hoped that they may be provocative of further thought and discussion to the enhancement of the reliability of the marine oil engine.

———o———

Notes.

One of our members in China, where he has been for some years, writes as follows for our information:—

"The unfortunate boycot of British shipping in this part of the world by a Chinese section, *inspired by Moscow*, must inevitably mean months, if not years, before we can resume normal trade. I hope the Institute will lend its weight in any way that may occur to dispel the mischievous lies that appear to be current in many quarters at home, *i.e.*, that the trouble is simply an ordinary strike caused by oppression of the working classes by Imperialists. It is a deliberate attempt to oust British trade and interests from China, and is as the thin end of the wedge to destroy British prestige throughout the world."

Evidences are not lacking from other quarters, which indicate that the mischief-maker is at work in many places where the truth is hidden by forceful persuasion.

Apropos of the Note from China.—The reminiscences and comments in "Six prisons and two revolutions" by Oliver Baldwin, is gruesome reading in some pages, but conveys lessons to ponder over.

The following from "Lloyd's List and Shipping Gazette," Sept. 30th, forms a contrast, and a possible lesson, to those interested in the cutting from the New Zealand Press, quoted in our Sept. issue, page 282.

Big Steam Roller Contract.—A Gainsborough firm, Marshall, Sons and Co., Ltd., have secured a contract for the construction of 100 steam road rollers for the Greek Government. It is one of the largest contracts of the kind placed in this country since the war. A feature of the construction will be that each of the road wheels will have its hub and spokes combined in one steel casting. The engines will have compound cylinders, and the boilers will be fitted with anti-incrustation fire boxes with cambered crown-plates.

The following is from "The Iron and Coal Trades Review," September 25th:—

New Marine Engine.—A new American internal combustion oil engine for marine use is reported to bid fair to mark a development in prime movers. The engine designed by W. B. Smith Whaley, a well-known mechanical engineer and inventor, is described as a 750 h.p., two-cycle, single acting unit, operating on a principle entirely different from the Diesel, and weighing only about 68,000 lbs., as compared with about 200,000 lbs. for a Diesel of similar power. In addition to being lighter in weight; the engine, it is declared, can be operated much more economically and efficiently than the Diesel. The reduced weight will mean lower initial cost, and a great saving in space, as compared with other engines of the internal combustion type. The Whaley engine is to be built for both marine and land use.

Boiler Explosions Acts, No. 2712.—The SS. *Trelyon* was on the eve of leaving Boston U.S.A., on the morning of February 27th, 1924, when at 5.30 a.m. the 2nd engineer, who was on watch, heard the sound of a violent explosion from the boiler top, and investigation led to the discovery that the port boiler stop valve chest had burst and killed one of the greasers. There were three boilers abreast in the stokehold, the pressure being 180 lbs. The steam pipes leading from the cast iron stop valve chests to the main engines were of iron, lap welded, 5½in. diam., fitted with screwed flanges, each outside boiler having a separate pipe bolted to a three-flanged main stop valve chest. The boiler stop valve chests, with 5in. valves, operated by the usual screwed spindles, were bolted direct to the shell plates with drain valves and pipe ½in. diam. The stop valve chest, when new in February, 1919, was tested to 300 lbs. hyd. pressure, the thickness being 13/16ths inch. It was secured to the boiler shell by eight bolts of one inch diam. The vessel, when the explosion occurred, was about 6ft. 6in. down by the stern, and berthed at the Army Base Pier. Whilst in port steam had been maintained on the port boiler. On February 26th, fires were lighted on the starboard and centre boilers about 8 a.m. Watches were set at 4 a.m. on the 27th; steam was ordered for main engines at 7 a.m., when the bursting sound was heard at 5.30, and it was realised that the greaser was on the boiler top, efforts were made to reach him, but unhappily these failed until the steam was cleared away, when his body was found. The fireman who was in the stokehold at the time, escaped into the cross bunker and then on deck with slight injury to his leg. It was found afterwards, from evidence, that all the main stop valves on the boilers were open to some extent, and the drain cocks closed, so also were the auxiliary stop valves; there was no record of the steam pressure. On examination of the chest by Mr. E. Taylor, B.T. Surveyor, Cardiff, he found it to be a good casting of even thickness, clear of faults and porous places, such as are often discovered. The position in which the body of the greaser was found led to the supposition that he had been operating the stop valves on the centre and starboard boilers and that the port boiler valve had probably not been opened, thus leading to water hammer action, and when the chest burst the valve would be eased from the seat. The engine stop valve chest with the three branches was not fitted with drains; the drainage was accomplished by easing the main valve and allowing the water to pass through to the H.P. casing drain cock and the reversing engine. The observations of the engineer surveyor-in-chief are " There was no direct evidence obtainable regarding the circumstances and events immediately prior to this explosion, but from its nature there can be little doubt that in the report it is correctly attributed to water hammer action. This view is confirmed by another fatal explosion that also occurred recently, from the port main stop valve of an exactly similar installation, in which case the evidence was conclusive in indicating water hammer as the cause. In both cases no drain cock was provided at the after parts of the pipes, consequently when the vessel was down by the stern a large accumulation of water was possible, even when the drains provided at the stop valves were full open and passing steam owing to the disturbance of the water level which would probably occur if the water were removed rapidly through a large orifice, it would be dangerous to use the engine stop valves for draining the pipes while the main stop valves were open to steam. It is clear, therefore, that drain cocks should be provided in this and similar ships by means of which the steam pipes can be effectively drained under all conditions of trim, and the stop valves admitting steam to the pipes should not be opened to any extent until it has been shown that the pipes are entirely cleared of water."

The following is from "Engineering and Boilerhouse Review" of September:—

"A Death Trap Warning.—We publish in this issue a letter from one of our readers drawing attention to the sad catastrophe at Hawarden Bridge, Flintshire, where four men lost their lives as a result of admitting live steam to an empty boiler in which these unfortunate men were working. Five men were actually inside the boiler at the time, and four of them received injuries from which they died, after removal to the infirmary. When one boiler of a battery is shut off for cleaning, the danger to the lives of those working inside is obvious, but this unfortunate accident shows that it is a matter in which nothing should be left to chance. We are giving prominence to this particular case so that it may be a warning to all boiler-house engineers to see that it is impossible for live steam to enter a boiler when a human life is inside.

The writer recalls a personal experience at sea, revealing another kind of danger, where the boilers were changed over without warning. The engine driving the main ammonia compressor in the freezer actually ran away as a result of this change over, and although the occurrence only occupied seconds it was merely good fortune that a burst ammonia pipe did not result."

Another instance may be cited, in which case the chief engineer during the meal hour went into a boiler to further examine a small defect which his attention had been called to. The doors were all on, except the top manhole, and the fires ready to be set away. The top door was jointed, fitted and screwed up while the chief was inside. Fortunately when he discovered he was locked in, he had a hammer in his hand and made use of it to make himself known, and after a short delay the door was off and the chief liberated.

We are indebted to John F. Smail (Member) for presenting a dozen portable ash trays of well polished brass, mounted on brass tubes and with winged brass feet, for use in the Premises, to take the place of those formerly used, than which they of the new set are more appropriate and efficient.

Election of Members.

List of those elected at Council Meeting of 5th October, 1925.

Members.

Archibald Caldwell, "City Line," 75, Bothwell Street, Glasgow.

Walter Pattison Douglas, Singapore Slipway and Engineering Co., Ltd., Tangong Rhoo, Singapore.

William Galbraith, B.I. Engineers' Club, Box 296, Calcutta.

Herbert Reid Howells, *c/o* Lloyd's Register of Shipping, 71, Fenchurch Street, London, E.C.3.

Bruce MacKay, 12, Rosedale Avenue, Gt. Crosby, Liverpool.

Angus William Maclean, 13, Coleridge Avenue, Manor Park, E.12.

George Albert Madden, 51, Winter Road, East Southsea, Hants.

Stanley Gordon Martin, 33, Carlton Mansions, Portsdown Road, Maida Vale, W.9.

Thomas William Arthur Masterman, Lieut., R.N., 3, Kedale Road, Seaford, Sussex.

Kenneth Alexander Mitchell, *c/o* Howrah Jute Mill, Calcutta.

Walter G. Murray, 4, Park Terrace, Maryport, Cumberland.

Walter Henry Purdon, 32, Ossian Road, Stroud Green, N.4.

Gifford James Reyburn, 16, Keston Avenue, Mosman, Sydney, N.S.W.

Companion.

Frederick Manley, 24, Vanbrugh Fields, Blackheath, S.E.3.

Associate Members.

Frank Hood Edwards, "Westcombe," Wrotham Road, Gravesend.

John Johnston Robertson, 14, Hendford Grove, Yeovil, Somerset.

Hylton Widdrington Young, 59½, Southwark Street, London, S.E.1.

Associate.

Frederick Oliver Harding, *c/o* 24, Prestwick Street, Dunedin, N.Z.

Student-Graduate.

Malcolm Waters, 28, Oaklands, Gosforth, Newcastle-on-Tyne.

Graduates.

Leslie Evelyn Goodwin, 34, Penerly Road, Catford, S.E.6.
Richard Leslie Holland, 1, York Drive, Hyndland, Glasgow.

Transferred from Associate to Member.

B. W. Grearson, "Ruberslaw," Millpark Avenue, Hornchurch.

It was with great pleasure that the acceptance of the office of President, by Lord Inverforth was received, and on the occasion of his address, it is fitting that a brief outline of his progressive advancement in life should be given, as an example of what can be done by work and energy rightly directed.

Born at Kirkcaldy and educated at the High School there—made famous by Thomas Carlyle as Headmaster—he began his business career in the Commercial Bank of Scotland at Sinclairtown, and after spending four years there he obtained an appointment in a shipowner's office in Glasgow.

Later on, he commenced business on his own account, and in 1885 he founded the Bank Line, now so well known by its ships in every port of the World.

The days of the sailing ship were numbered and the Bank Line changed to steam, and now another passing away is predicted, and the Bank Line has prepared by providing a fleet of oil-engined vessels to carry on the race for industry and commerce.

The National work carried on by our President during the war is well known ; he was Surveyor-General of Supply at the War Office and Member of the Army Council from 1917-1919. Minister of Munitions, January 1919 to March, 1921 ; first Chairman of the Disposals Board and Liquidation Commission to May, 1921. The duties in connection with these offices were executed to the National gain.