The INSTITUTE of MARINE ENGINEERS

Founded 1889. Incorporated by Royal Charter, 1933.

SESSION 1942. *Transactions* Vol. LIV. Part 3.

Patron : HIS MAJESTY THE KING. President: The RT HON. LORD MOTTISTONE, P.C., C.B., C.M.G., D.S.O.

*"Steam Tugs, Past, Present and Future," by G. T. SHOOSMITH, M.A. (Member): Discussion and Author's Reply.

Mr. Julian S. Tritton, M.Inst.C.E., wrote :—Tugs and kindred craft have not in the past received the scientific attention they deserve, either when being designed or under service conditions. Some of the points the author makes are extremely good, especially the possibility of mechanical stoking, bridge control of engines and the more general use of welding.

There are, however, two points on which the writer would join issue with the author. They relate to recent developments of Kort nozzle propulsion, which has come to the fore so much of late in tug design. As the writer's firm has been responsible for the design of numerous tugs for port trusts and other clients the most searching enquiries have been made into the claims put forward by the Kort people in London. In these enquiries we had to consider the two important questions raised by the author, namely, those of capital cost and vulnerability. On both these issues we decided in favour of the Kort nozzle and in 1937 adopted this system for a twin-screw 900 h.p. tug for the Madras Port Trust. Our investigations, prior to drawing up the specification, showed that this 900 h.p. tug would have the potentiality, so far as bollard pull was concerned, of a normal tug of practically double her horse power. When the tug was built, the general results were so satisfactory that we included Kort nozzles in another tug of 800 h.p. for service in Chittagong. The point is that for these harbour tugs, where free speed is not of consequence, it is possible with Kort nozzles to reduce the installed horse power by close on 50 per cent. and still obtain the same performance. Under these conditions the capital cost of the Kort nozzle is much more than offset by the saving in first cost of engines, boilers and hull. There is also a very reduced fuel bill, available for the whole of the vessel's life.

The writer admits that for river towing such as Mr. Shoosmith has to deal with, the savings in horse power would not be of such a high order, but it is interesting to record that at trials which the writer witnessed in Grimsby between two single-screw tugs owned by the L.N.E.R. similar in every respect except that one had a nozzle and one an open propeller, at towing speeds of 2 and 4 knots the nozzle tug had a performance equivalent to that of a normal tug of 74 per cent. and 48 per cent. more power respectively. The writer is satisfied that these figures are correct and it seems, therefore, that the capital cost involved, even under London towing conditions, would be more than offset by savings in first costs and fuel consumption.

Regarding vulnerability of the nozzle structure, the writer holds the view that the nozzle with its very rigid internal stiffening and rigid attachment to the sternframe is one of the strongest components of the whole vessel. In addition, the attachment to the hull is spread over a large area, thus distributing any shock loading due to grounding or "striking a submerged object". In such cases damage to the nozzle is localised and unlikely either to distort the nozzle or upset its alignment with the propeller to which it affords very useful protection. There is little doubt in the writer's mind that had this protection been available in the case of the river tug quoted in the paper, the damage to the propeller, sternframe and propeller shaft cone would have been very much less.

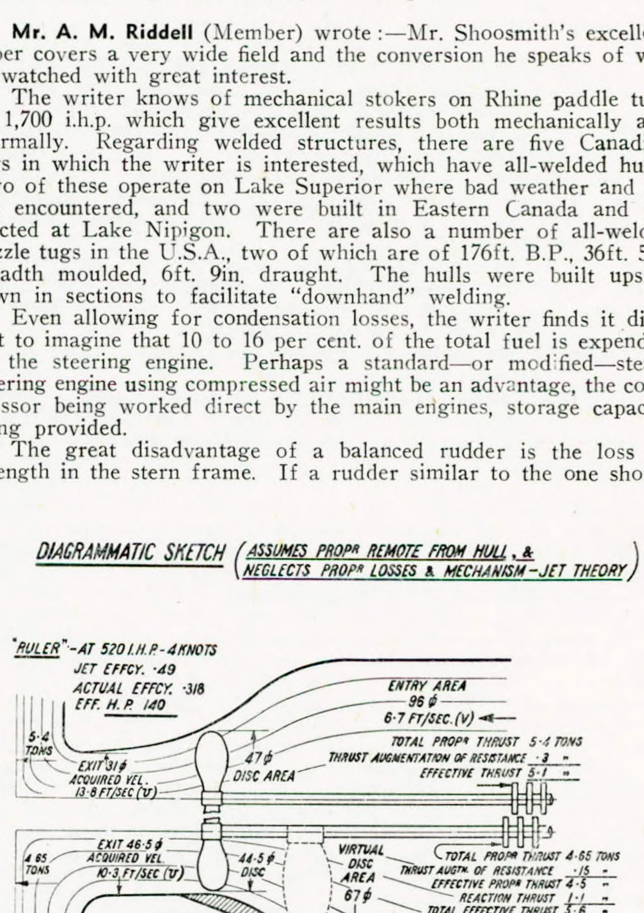
The author's remarks on propeller design are very apt. In the past, too many specifications have limited propeller efficiency by specifying an unwarranted free running speed, to attain which the propeller designer has had to sacrifice efficiency at towing speeds. It is better for owners when ordering new tugs to specify the duties required in order of priority. The designer can then provide the best possible compromise in propeller design.

Mr. A. M. Riddell (Member) wrote :—Mr. Shoosmith's excellent paper covers a very wide field and the conversion he speaks of will be watched with great interest.

The writer knows of mechanical stokers on Rhine paddle tugs of 1,700 i.h.p. which give excellent results both mechanically and thermally. Regarding welded structures, there are five Canadian tugs in which the writer is interested, which have all-welded hulls. Two of these operate on Lake Superior where bad weather and ice are encountered, and two were built in Eastern Canada and re-erected at Lake Nipigon. There are also a number of all-welded nozzle tugs in the U.S.A., two of which are of 176ft. B.P., 36ft. 5in. breadth moulded, 6ft. 9in. draught. The hulls were built upside down in sections to facilitate "downhand" welding.

Even allowing for condensation losses, the writer finds it difficult to imagine that 10 to 16 per cent. of the total fuel is expended on the steering engine. Perhaps a standard—or modified—steam steering engine using compressed air might be an advantage, the compressor being worked direct by the main engines, storage capacity being provided.

The great disadvantage of a balanced rudder is the loss of strength in the stern frame. If a rudder similar to the one shown



*Paper published in January, 1942 TRANSACTIONS, Vol. LIII, No. 12, pp. 185-192.

were to be fitted, but extending only to just forward of the pintle and the balanced part were to be fixed as a permanent fin, this would give strength to the stern frame, and in the writer's opinion would not lose anything in steering moment. In addition, the fin would give a better "divide" for the propeller feed with the engines running astern, with a consequent improvement in astern steering. Such a rudder would have the same protection from the counter as the balanced rudder, but the torque would be increased. However, this would be less than for an ordinary rudder capable of the same steering moment.

Mr. Shoosmith suggests that the larger area of race with nozzle is beneficial. This is quite true, but is not to be compared to the benefit due to the more efficient propeller and the nozzle reaction. Assuming the "Redeem" to be fitted with a nozzle, the hand sketch on p. 27 shows the effect of the nozzle in increasing the area of race. The races are both assumed to impinge on a flat surface perpendicular to the race, the total impact being equal and opposite to the total propeller thrust. In practice, only a small fraction of the energy is lost by impact on the leading barges, but the sketch will indicate approximately the relative impact either on the flat plate or on the leading barges. The following is an extract from a letter signed by the Superintendent of the Wm. Froude Laboratory dealing with tank trials of a tug with and without nozzle :—

"With regard to the evaluation of the towing forces measured in the Tank. These were taken by a string at the after end of the nozzle without any barges being present. In any actual towing work there would be such barges present, and a proportion of the towing force in the towing rope would be wasted by impact upon the barges behind the tug. The proportion so lost, however, would be practically constant, whether a Kort nozzle was being used or whether the screw was running in open water, so that the ratio of these two forces with the two different stern arrangements can be accepted as representing the probable value of the towing forces actually obtained in practice".

Trials carried out in the Thames proved that for exactly similar speeds towing the *same* craft, the pull without nozzle was 3¾ tons and with nozzle only 3¼ tons, proof that some of the artificial resistance due to impact is reduced by the nozzle.

Regarding astern thrust, the nozzle unaccountably increases this by about 30 per cent.

Mr. Shoosmith mentions the quite appreciable "capital cost involved" as a disadvantage, but to mention cost without mentioning available benefit creates an entirely false impression. If the "Redeem" were to be fitted with a nozzle she would give a performance as good as that of a tug with an open-propeller of 580 i.h.p., that is, 10 per cent. better than the "Redeem" with 20 per cent. less fuel. The cost of the nozzle would be more than offset by the benefit. In the case of new structures a saving of 160 h.p. would bring about a reduction in first cost of engines and boilers considerably in excess of the cost of the nozzle. In both of the foregoing cases the saving in fuel alone over a few years would pay for the fitting of a nozzle and, as steam tugs like "old soldiers never die", the saving in fuel would be very considerable taken over a period of, say, 30 years.

The writer's Company will shortly be fitting a nozzle to a 520 h.p. Diesel tug engaged in London river towing. The towing speed is under 3 knots and the writer assesses that the towing performance will be equal to that of a normal tug of about 750 h.p.

Mr. Shoosmith also mentions the vulnerability of the nozzle structure to damage, but with the very rigid internal stiffening and the fact that the nozzle is attached to the hull by means of vertical plates coinciding with the hull framing, the writer cannot imagine that the nozzle could be "started" or distorted to such an extent as to jam the propeller.

Mr. Alfred Carter (Member) wrote :—With regard to the merits of Diesel and steam tugs, on Mr. Shoosmith's own showing a good case is made out for a steam-driven tug fitted with a Kort nozzle for the class of work he is interested in.

The initial cost of the steam tug and nozzle would be less than that of a Diesel tug. The cost of upkeep, according to Mr. Shoosmith, would be materially less. The length of service obtainable from the steam tug would be greater. As Mr. Shoosmith says, "Steam tugs never die, they only fade away", and they do even that with seeming reluctance.

The writer questions Mr. Shoosmith's statement that the "quite appreciable capital cost involved" in the fitting of a Kort nozzle is a disadvantage. The writer has witnessed very interesting tests carried out on three tugs fitted with Kort nozzles the results being that the pull was increased by from 50 per cent. in each case, in other words, from approximately 6½ tons to 11 tons. These tugs were all about 30 years old, in good condition, and no alterations were carried out to hulls or rudders. In one of the three cases the old propeller was used as the owners wished to see exactly what difference the nozzle itself made. Manœuvring and speed trials were also carried out, the free speed and manœuvring ability in each case being somewhat better with the nozzle. The writer appreciates that the service on which these tugs are engaged is vastly different from the service in which Mr. Shoosmith is interested, but he has in his possession an article which appeared in the *"Journal of Commerce"*, dated 16th June, 1939, which shows that the fitting of a Kort nozzle to a tug owned by the L.N.E.R. Company resulted in an increase in performance equivalent to that of an ordinary tug of 74 per cent. more power at 2 knots and 48 per cent. at 4 knots. Assuming the service conditions which Mr. Shoosmith has in mind to be around 3 knots, the fitting of a nozzle would mean that an existing tug could be converted to give the same performance as an ordinary tug of 60 per cent. more power. Taking a steam tug of 300 h.p., the fitting of a nozzle would give this vessel the potentiality of a tug of 480 h.p. Even if a conversion were not considered, the saving in first cost of machinery represented by the additional 180 h.p. should more than offset the cost of the nozzle.

Another important point is that on a powerful tug the reduction in power made possible, assuming the life of the tug to be 30 years, would result in the saving of fuel considerably in excess of the cost of the nozzle.

Mr. Shoosmith thinks that the nozzle would be likely to receive severe damage due to contact with low-lying barges, etc. The writer does not think this is probable as the nozzles are strongly constructed and the outer plating could be indented rather badly without disturbing the true inner circle. The writer would imagine that a contact heavy enough seriously to damage a nozzle would have disastrous effects on a tug not fitted with a nozzle. The writer knows of instances where the nozzle has prevented the propeller being broken without any material damage to the nozzle.

Mr. C. B. L. Filmer (London Manager, The Adelaide Steamship Co.) wrote :—Under the heading "Propulsion" the author refers to the Kort nozzle and suggests some possible disadvantages, although he states *inter alia* that this nozzle has been fitted to a large number of tugs.

Is there any data that the sternway of a tug is adversely affected when Kort nozzle is fitted? It would seem from the experience of the writer's particular friends that the sternway was not adversely affected in the ship-handling tugs concerned.

With regard to the disadvantage owing to the extra capital cost involved, very obviously any additional capital cost is a disadvantage unless it can show a suitable return. The writer's friends are satisfied that the fitting of a nozzle is worth while. It was tried out first on a comparatively new tug of about 700 i.h.p. Recently it was necessary to consider the question of having available a tug more powerful than the one in service, a vessel of some 1,100/1,200 i.h.p. and 26 years old. The vessel was in good condition and it was decided to install a Kort nozzle. She now satisfactorily fulfils requirements, and a third tug engaged chiefly on long distance tows is now being fitted. From our point of view the additional capital cost involved is very small compared with the first cost of the tug or, what might have been necessary, the cost of a new tug. It is also hoped that there will be a saving in fuel costs when the full power available is not required.

With regard to the possible disadvantage of damage from barges, the writer certainly has no experience of barges loaded down to the freeboard mentioned, but he should have thought that if anything the nozzle, if reasonably solidly constructed, would be of assistance in avoiding the damage to which reference is made; alternatively, if a low riding barge struck the nozzle with sufficent force to distort the nozzle to an appreciable extent, it would, in the case of a tug not so fitted, have damaged the propeller and/or rudder.

Capt. E. C. Goldsworthy wrote :—It is unquestionable that for some years after the war raw materials of all descriptions will be in short supply and shipping space at a premium. Coal will have to be used internally to the fullest extent to provide the necessary motive power and to ensure work for a maximum number of people. This provides a chance for the steam engineer to prove that the modern steam engine is comparable in revenue-earning to any other method, and the author's contribution to this effort deserves the highest praise and should be widely known to all shipowners.

With regard to the remarks made by the author on the propulsion of tugs by the Voith-Schneider propeller, it is interesting to note that he is technically in favour of this method. He states that his investigation in 1938 of a steam tug with a Voith-Schneider propeller gave a prohibitive first cost; but this may have been due to the shipyard taking the cost of tug with normal screw and rudder, adding the cost of the Voith-Schneider installation plus a handling charge and giving the result as the cost of the V.S.P. tug.

There should not have been any great difference in the overall first cost if proper allowances had been made by the shipyard. The elimination of thrust block, steering gear, rudder, stern frame, ordinary screw and stern tubes should have gone a considerable way to offset the initial cost of the Voith-Schneider propeller. An engine of less first cost might also have been employed rather than one specially designed for the revolutions required and it is conceivable that an appreciable amount could be saved.

The author does not give any details of the tug which was investigated, but it is possible that further savings might have been effected by using less draught, which would have been possible with the Voith-Schneider but not with a screw having the same swept area. This would have increased the sphere of operation of the tug which might have been of advantage to the owner.

The adoption of the Voith-Schneider propeller would have eliminated the problem of the design of the screw propeller, which the author states as being "one of considerable difficulty", since the variable pitch of the former enables the full engine power to be absorbed under any towing conditions. Was the cost of a possible replacement of the screw propeller with one of different design offset against the cost of the Voith-Schneider tug?

Bridge control gear for a tug is essential, not only to eliminate the possibility of delays and errors in engine movements but to enable the engineer to look after his engines rather than waste his time operating speed and reverse levers. The author fully appreciates these points and his arrangement of bridge control gear is of the greatest interest and practical use; he is to be congratulated on the design which has been proved in service, no doubt to the considerable benefit of the owners. Nevertheless, this control gear cost money and even then is only a control of the engine. There is still the time lag of altering the speed or direction of the engine before the action of the propeller can be effective. With the Voith-Schneider propeller the control gear is included in the price, but it is possible this was not taken into consideration when making the comparison of first cost. It also operates direct on to the propeller and, as the engine runs at constant speed at all times, action is instantaneous.

The advantages which Voith-Schneider propulsion gives to tugs through manœuvrability, simplicity and instantaneous control, and the great value of optimum thrust under any conditions of load, make it desirable that the owner should not be misled by the bald statement of the initial cost of the installation but should insist that the shipyard make out their detailed costs separately when asked to quote, for it is only by this method that the true first cost of the vessel can be given. From careful investigations in the past it is confidently anticipated that post war the cost of a Voith-Schneider vessel will be comparable to that of any other method of propulsion having bridge control and power-operated steering.

Mr. C. Davis (Associate Member) wrote:—The writer considers that the scope of the paper covers many improvements to overcome defects arising from daily wear and tear.

The author expresses a preference for welded boiler shell seams. Would this accord with B.o.T., Lloyd's and other classification societies requirements? And is this preference prompted by a desire to overcome a defect with which the author has been plagued or for simplicity in construction?

The proposal to adopt a dry-back type boiler is worthy of consideration if only for reasons of simplicity in construction and easier access for cleaning and inspection; in addition, it would be less liable to defects than the ordinary marine type. The writer considers that an efficient circulator should be fitted to boilers having more than one furnace to assist in the prevention of corrosion of the lower part of the shell ring.

The provision of forced-lubrication engines is almost a necessity. Whilst owners would be inclined to regard the initial cost as unduly high, they would eventually consider this justified by reduced overhaul costs, and, very possibly, maintenance costs, the latter being largely dependent on the type of man running open type engines. The proposed chain drive for the pumps simplifies matters considerably and should reduce the possibility of damage due to defects or maladjustment.

The adequate and efficient lagging of both boilers and engines is desirable, not only for the prevention of heat losses but also for the comfort of the men in charge of the engine room; for if they are subject to discomfort, due to high temperatures, they will constantly be "having a breather" and the machinery will suffer in consequence. In this connection, design of the boiler and engine-room casings should receive adequate attention as well as the placing of ventilators. In making these remarks, perhaps it should be mentioned that the writer has in mind tugs which are to be used in tropical waters.

The writer is rather exercised in mind regarding the proposal to fit double-bottom feed tanks, but doubtless the author has considered a floor depth which will enable cleaning and inspection to be done without difficulty. One great advantage of such tanks would be freedom from contamination of the feed water which would occur in side feed tanks were the side plating damaged in any way.

The provision of battery lighting would be a great convenience and no doubt would serve the navigation lights as well as below decks. In putting this forward it is assumed that the author has compared the cost of such an installation against the cost of oil lamps over a period. In proposing a welded hull it is presumed that flush joints are implied to avoid wasting and rubbing of the landing edges which take place to a great extent not only in tugs but also barges and similar craft.

Bulwarks and fendering are liable to considerable damage. Bulwark plating should be well set in, especially at the counter, and supported by bulb angle stanchions at close intervals; the bulwark rail should be substantial and preferably of bulb angle. Even so, bulwarks are frequently damaged and the sheer strake should be carried up sufficiently high to enable the bulwark plate to be easily removed, faired up and replaced or renewed as the case may be without difficulty. It often happens that the fender angles cover the riveting of the bulwark and sheer strake joint and involve a lot of tedious work both in renewal and replacement, especially in those shops where only hand labour can be employed.

Fendering is the *bête noire* of all concerned with its upkeep. Its width should be such as will protect the bilge plating at the maximum angle of roll. This factor may not be great in river work where the water is usually quiet, but a roll of up to 15 degrees should be allowed for open harbour or bay work. The fendering is usually one of the harder woods with a suitable convex bar firmly screwed on to it. The convex bar is, in the writer's opinion, a doubtful advantage, for a few blows are sufficient to start the screws and distort the bar with consequent need for constant attention. On one tug the writer inserted the wood in a $\frac{3}{16}$-in. steel sheathing riveted to angles which, in turn, were riveted to the hull; this stood up well for several years and needed but little attention; suitable treatment of the plating in way of the fendering precluded any serious corrosion. A doubling plate in way of the fendering would be a great advantage unless the fendering is regularly removed for cleaning and examination of the plating.

Wasting of the keel bar and stern frame are common, and these should be of corrosion-resisting steel in tugs which are expected to give a long life.

The form of a tug will, of course, depend on local conditions and the service it has to perform, so that it may not always be possible to employ the ideal design. Tugs also vary considerably in their behaviour. As an illustration of this, the writer had the misfortune to have a vessel of some 170-tons displacement stuck on the slipway above high water mark, due to the car leaving the rails, so that it could be neither hauled up nor towed off. The writer rigged gear and forced the vessel and car down until she was all but buoyant. The question of a tow arose and, as our tugs were considered to be too light for the job, the engineer of the company to whom the vessel belonged proposed that a sister ship fully laden should be used, the idea being that the heavier moving mass would pull the vessel off the cradle blocks. The fully loaded displacement of the towing vessel was just over 400 tons. Accordingly, the gear was re-rigged to assist by pushing and at spring tide the attempt was made. The push was not sufficient to give the vessel buoyancy and the tow was a failure, although as much of a running start as possible was given to the towing vessel. Being anxious not to miss the tide, the writer borrowed a tug heavier than our craft from a neighbouring company, but this tug also failed, although given a running start. Neither tow had moved the vessel, and, unfortunately there was no swell which could have assisted in lifting the inert mass. By this time the tide had turned and there was no time to re-rig the gear to continue pushing, but as a last resort the writer put on what was considered to be the best of our tugs. The usual running start was given; the tow-rope tautened and brought the tug up, but the vessel moved and as the tug kept going ahead, was finally towed off—to the writer's great relief. Neglecting the sister ship, the sizes of the tugs used were 65ft. long, engines 12in.×24in.×18in., i.h.p. 120 (which failed), and 55ft. long, engines

9in. × 18in. × 12in., estimated i.h.p. 80 (which succeeded, and which was, incidentally, nearly 40 years old at the time).

We had three tugs of the same dimensions all built by Messrs. David Rowan & Co., and fine little tugs they were in spite of their age, yet each of them behaved differently. In one case the propeller race had to be partially blocked under the counter and the height of the rudder increased so that the vessel should behave better under helm. This fact should lend weight to the author's argument for improved rudder design. Since a tug should respond readily to its helm and be able to turn in as small a circle as possible, would it not be preferable for the width of the rudder in line with the propeller boss to be such as to cover the path swept by the tip of the propeller when the helm is hard over? This would ensure maximum thrust and a quicker turning movement, which would be assisted if the dead wood were cut away.

The author has expressed a preference for cast-steel propellers, and the writer would be glad to know the reason. Although our tugs often had to work in rough weather in the dark, little propeller trouble was experienced. The propellers were of cast iron.

The writer agrees with the author that a tug should be kept fully employed to obtain economical maintenance costs, but if heavy repair bills are to be avoided it is desirable that the crews should be as permanent as possible and the tug stopped for overhaul and cleaning at regular intervals.

Paddle wheel tugs are mentioned. The writer believes the Admiralty still have a number of these in use which are of ancient vintage.

The Author's Reply to the Discussion.

It gives great pleasure to the author to find that his paper has raised the many interesting points which have been dealt with in the correspondence.

A number of contributors have come forward to espouse the claims of the Kort nozzle. Mr. Tritton states that the Kort nozzle tug has the potentiality on a bollard pull of a normal tug of practically double the horse power. The author has not been able to find any figures which fully substantiate this. He recollects that the tug "Sir Francis Spring", fitted with nozzles, developed a bollard pull of 16¼ tons at 697 s.h.p., equivalent to 42·5 s.h.p. per ton. In a published description of this tug a statement was made that the pull with normal propellers would have been 11·2 tons. This latter figure would have been equivalent to 62·5 s.h.p. per ton. There are, however, a number of tugs with normal open propellers which develop at the bollard a pull of 54 s.h.p. per ton and better, and on this basis the gain at the bollard due to the nozzles fitted to the "Sir Francis Spring" was less than 25 per cent. as against the 50 per cent. claimed. It should be noted in this connection that the fitting of the 10ft. diameter nozzles on this tug limited the propeller diameter to 7ft. 6in. whereas without nozzles the propellers could have been of appreciably larger diameter.

When it comes to considering barge-handling tugs of the type used on the Thames, the circumstances are quite different from those applicable to ship-handling tugs. Free running speed is a factor of importance for the former, as such tugs frequently have to run light over the tide to pick up their tow. In this connection the results of the L.N.E.R. tugs are of interest. Of these two identical tugs No. 5 without nozzles ran at 9·27 knots when developing 107 i.h.p. and No. 10 with nozzles at 9·23 knots when developing 143 i.h.p.

The figures for the tug "Scud" show that the nozzle reduced the free running speed from 9·25 knots before to 8·75 knots after fitting.

Another point of extreme importance for barge handling is manœuvrability, particularly the control when in stern way. The author has experience of two single-screw nozzle tugs on which it is almost impossible to tell which way the stern will swing when going astern. It is appreciated, however, that more recent developments may have improved the control in this respect.

On the score of vulnerability, there are two sides to the question and again the author knows of a nozzle equipped tug which has been in trouble due to driftwood jamming between the propeller and the nozzle structure.

The author is not quite clear as to the figures given in Mr. Riddle's sketch. The term "virtual disc area" is presumably the disc area of the nozzle structure. If a structure of such increased diameter can be accommodated, it is probable that by modification of the stern frame a propeller of something approaching the nozzle diameter could also be accommodated.

Coming back to the "Sir Francis Spring", the nozzles fitted to this tug were 10ft. diameter and the propellers 7ft. 6in. diameter for a tug drawing 12ft. to 12ft. 6in. aft. On the Thames, many tugs carry propellers up to 9ft. diameter on 10ft. 6in. draught, and with a tunnelled stern they could carry propellers even larger; it is undoubtedly a fact that sufficient attention has not been given to this aspect of tug design in the past.

The conditions met with in barge handling, where two swim head barges abreast are towed right up astern, bear little resemblance to conditions met with elsewhere. It is impossible to reproduce them artificially or to attempt calculations based on results of bollard pulls or stern-to-stern tests as carried out at Grimsby. It is all the more interesting, therefore, to hear that accurate comparative towing tests are shortly to be made on the Thames with a tug before and after fitting a Kort nozzle. The author wishes to emphasize that it is only by means of such a trial, taken in conjunction with accurate figures of comparative free-running speed and handiness that any definite conclusions can be drawn.

The author is interested in Mr. Goldsworthy's remarks on the Voith-Schneider propeller and agrees that it is possible that a true comparison in costs between this and normal propulsion has not always been obtained in the past. Presumably an appreciable saving in first cost would result from the Voith-Schneider unit being constructed in this country, which it is hoped will be the case after the war. The fact that the Voith-Schneider propeller enables a large swept area to be developed on a restricted draught is a point very much in its favour for barge-handling tugs. Disregarding the question of cost, the author is of opinion that this form of propulsion has great possibilities for barge handling. It can develop a high thrust on limited draught, the full engine power can be employed under all conditions from free running to heavy towing. The manœuvring is superior to any other type of propulsion and the control just as exact when going astern as when going ahead. The wide flat stern which is an inherent feature of the design is also a protection against damage from other craft.

In reply to Mr. Davis and the question of welded boiler shells, the preference for welded construction was expressed since it is general experience with Scotch boilers that the majority of their troubles occur at the riveted seams. A marine boiler, 15ft. 6in. diameter, 180lb. per sq. in. working pressure was built by Messrs. R. & W. Hawthorn Leslie & Co., in 1920, of completely welded construction. In this boiler the end plates were let into grooves planed in the shell and filet welded thereto. This boiler gave very satisfactory service. It will be appreciated that apart from saving in maintenance costs, such a construction should result in an appreciable saving in first cost as it eliminates the costly dishing of the end plates. Codes covering the design and construction of welded fired pressure vessels have been published by the A.S.M.E. and by authorities in Germany and Switzerland, whilst rules have also been laid down by Lloyd's for land purposes.

The author agrees that an efficient circulator would be an advantage but, unfortunately, he has not come across a circulator which remains efficient for any length of time in service. The circulating troubles would be considerably reduced by pre-heated feed.

The author is in entire agreement with Mr. Davis' remarks regarding bulwarks and fendering. Fortunately, it is possible with barge-handling tugs to eliminate timber fenders with a considerable reduction in upkeep costs.

On the question of rudder design, both tests and theory indicate that additional width of rudder does not give a corresponding increase in turning moment, whilst it has the disadvantage that the trailing edge of the rudder is rendered more vulnerable to damage.

The advantage of the cast steel propeller is that defective blade tips, erosion, etc., can be easily repaired by electric welding. Wasting of the keel bar and stern frame, also fractures of the stem bar, can now be cheaply and easily dealt with by electric welding.

Molecular Aspects of Heat Transference.

By R. A. Collacott, B.Sc. (Eng.) Hons, A.I.Mar.E., S.I.N.A.

All substance is assumed to consist of small moving particles. Temperature is proportional to the mean molecular kinetic energy of translation (not rotation) of these particles or molecules. Therefore, from the very first conception of heat and heat transmission, it is evident that a molecular aspect to the theory of heat transference must exist.

Molecular theory of matter.

Molecules, small moving particles, possess mass and therefore exhibit mutual attraction. These forces draw the molecules together in solids and they are arranged closely together so that the distance separating them is so small that they orientate to form a "space lattice" with the dispositions of individual molecules depending entirely upon the substance itself. Liquids are composed of molecules with freedom of movement within the mass, but sufficient internal attraction to prevent them escaping and obtaining complete freedom of movement within space. Gaseous molecules have complete freedom in space, they move with constant velocity until they strike the containing walls. Change of momentum of the molecules sets up the pressure of the gas upon the containing walls.

Conduction.

Heat transference through a solid is essentially a molecular process. Molecules in the region of high temperature move with greater translational kinetic energy than those in the region of low temperature. Since the molecular weight remains constant, the molecular velocity in the region of high temperature must be greater than in the region of low temperature. By virtue of their greater velocity, the fast moving molecules will collide frequently and so transmit momentum to other molecules which therefore manifest temperature, themselves transmitting momentum and temperature to other molecules. Continued exchange of momentum in this manner represents the transference of heat by conduction from the hot to the cold region.

It is significant that metals and alloys which are good thermal conductors are also good electrical conductors. The electron theory attributes both phenomena to the movements of "free" electrons loosely attached to the atoms. Although this does not lie within the scope of this discourse, the theoretical Lorenz equation, $k=CjT$ where k and j are thermal and electrical conductivities, T the absolute temperature and C a constant, which holds approximately at normal temperatures, supports the theory relating conduction to the molecular structure of matter.

Corrosion.

When heat is transmitted between two fluids, either liquid or gas, on either side of a metal wall, corrosion is formed in layers on the wall. If corrosion does not attack the metal, the anti-corrosion media itself must be responsible for the formation of an impervious film over the metal wall. It will be seen from Fig. 1 that the corrosion layer or the protecting film plays an important part in the transference of heat from the fluid.

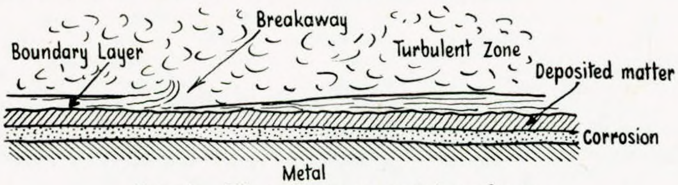


Fig. 1.—*Film effects on metal surfaces.*

Electrolytic corrosion is the principal form of surface deterioration. Impurities on the surface of a metal react with acid moistures or vapour so as to form a small, local, galvanic cell. The current which is formed is discharged through the circuit and electrolyses the poles. These products of electrolysis form the corrosion layer. Commercial metals are impure, thus steel (which is a molecular dispersion of carbon in gamma iron) will also contain silicon, sulphur, phosphorus, manganese or other elements which will form chemical constituents within the steel. At the surface, the junction of these patches of impurities comprises the zone of electrolytic activity at which corrosion attacks occur.

It has been found that thin oxide films over the metal ward off this corrosion attack. Although this film is only one or two molecules thick it presents almost as high a thermal resistance as the corrosion products themselves. This layer adheres to the metal by virtue of the absorption of its molecules to the surface molecules of the metal, so that it is probable that the molecular structure and activity of these surface films do not follow precise laws of conduction for the simple, solid material of which the film consists.

Fluid flow.

Fluid flow, whether laminar, turbulent or transitional is of prime importance. In each of these states, the motion of the fluid molecules is the criterion. During laminar motion, the molecules move along straight paths, with linear translational velocity and rotation about their axis only. When the dynamic forces keeping this system in equilibrium fail, the translatory linear motion is affected and the molecules describe a radial path; turbulent motion ensues. Between these two conditions for which no sharp line of demarcation exists, the motion is in a state of transition.

The physical properties of density and viscosity control the nature of the motion as deduced from the classic Reynolds' Number

$$R_e = \frac{\rho V.d}{\mu}$$

where ρ and μ represent density and viscosity, V the mass velocity and d some typical dimension. Of the former variables, density depends upon the number of molecules per unit volume, while viscosity depends upon the attraction between individual molecules. So far then as the motion and heat transference of a fluid is controlled by physical factors, molecular construction is a criterion.

Boundary layers.

Across the stream of moving fluid, the velocity of individual molecules varies. Thus, for laminar flow a parabolic law governs the distribution of velocity across the stream as in Fig. *2a* while turbulent flow presents a velocity distribution such as Fig. *2b*. It will be seen from Fig. *2b* that turbulent flow comprises a laminar layer at the pipe walls which changes to turbulence. This is known as the "boundary layer".

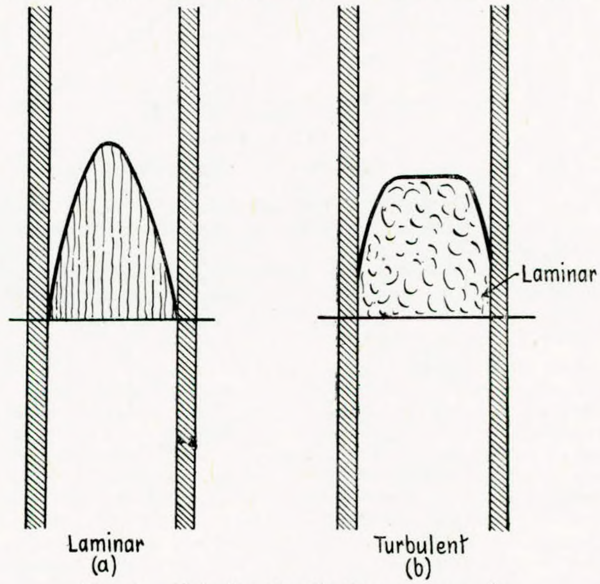


Fig. 2.—*Velocity distribution across a pipe.*

Adhesion of this layer to the pipe walls is a very complex phenomenon, although it is largely a function of the molecular attraction between the metal and fluid molecules. In such circumstances it would be anticipated that the boundary layers on different materials would be of different thicknesses under identical conditions. Data relating to the fluid friction of pipes of various material indicate that the friction factor—which depends upon boundary layer thickness—varies, but the variation is usually ascribed to surface roughness only. It has been found that the physical properties of the fluid control the thickness, so that it is probable that while the molecular forces due to various materials do exist, they are very small compared with the molecular forces of the fluids.

Because heat must traverse this boundary layer, and because this film of stationary fluid presents a high thermal resistance, these considerations are of prime importance in the study of heat transference. It will be seen that as the nature of the fluid is of vital importance, it is not surprising that the heat transfer coefficients vary.

Convection.

The mechanism of convection heat transference may be briefly stated as the motion due to change in density due to local temperature differences throughout the fluid. Although convection currents may be set up at the source of heating, local convection will also occur during the passage of this fluid through the pipe since the temperature distribution varies across the section in a similar manner to the velocity.

The velocity, density, viscosity and specific heat of the fluid as well as the size, shape and temperature of the body transmitting heat influence the convection heat transfer rate. Density, viscosity and temperature are established molecular phenomena; specific heat is also dependent upon the molecular structure. Thus, the specific heat of calcium chloride brine varies in the manner of Fig. 3.

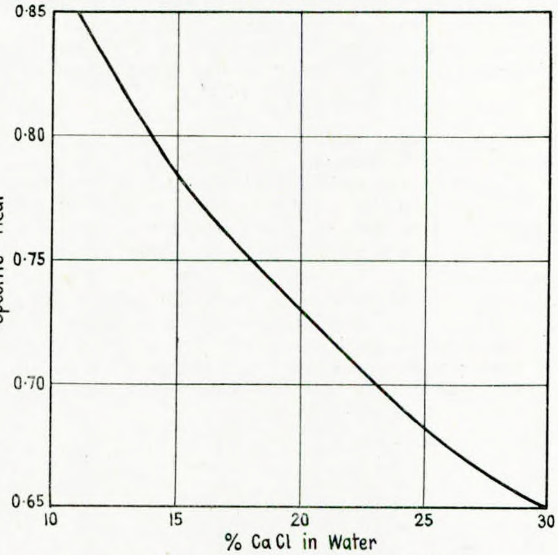


FIG. 3.—*Specific heat of calcium chloride brines*

When heat is added to the brine molecules they tend to vibrate faster, *i.e.,* to gain temperature, depending upon the proportion of heat energy absorbed in translation.

The conditions necessary for a substance to have a high specific heat in which a great amount of translational energy is received in preference to rotational energy depends upon the chemistry of the space-lattice for each substance.

Latent heat transference.

Many heat exchangers convert the state of a substance from the liquid to the vapour or *vice versa.* Throughout this change the latent heat of the substance is exchanged. Latent heat is a function of molecular activity; heat supplied for evaporation is used in doing work against the force of attraction between molecules. When also the state of a substance is changed it commences to exhibit other phenomena lacking in its former state; thus the surface molecules of condensed moisture drops possess surface tension, the molecules of a vapour produce pressure. With such additional complexities the mechanisms of latent heat transference are greatly complicated.

Condensation.

According to the condition of a plate a vapour may condense as either a film or in drops. Drop condensation requires between four and eight times as much heat removal as film condensation, so that this phenomenon deserves considerable attention.

With drop condensation it would appear that the surface attraction of the metal was insufficient to link with the molecules on the

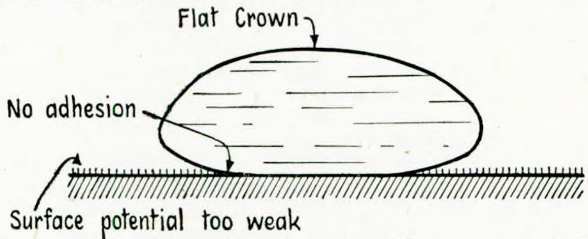


FIG. 4.—*Drop-wise condensation of liquid upon a metal plate.*

surface of the drop and therefore the drop remained isolated (Fig. 4). If other drops form in the vicinity they may be allowed to neutralise each other's surface attraction, unite and form a layer over the surface. Although this layer may not "wet" the surface, if it is not drained, heat transference will be reduced and the temperature fall so that the surface molecules may then be able to absorb the moisture molecules.

When absorption is complete, a liquid film will be formed. If, as the drops fall, the surface is in a molecularly strong condition, the moisture molecules may be immediately absorbed in the form of film condensation. To continue this theory it would be necessary to examine the molecular attraction of surface molecules. It is significant, however, that clean surfaces, which promote film condensation, are also necessary to produce the "wringing" together of certain types of gauges.

Evaporation.

When a liquid is boiling vigorously there is a very rapid rate of heat transference. Observations of boiling liquid on a hot plate indicate that columns of bubbles arise from separate points on the

FIG. 5.—*Point-evaporation of a boiling liquid.*

surface, as in Fig. 5, which move from time to time. The explanation of this phenomenon is closely connected with the vapour pressure.

Investigations remain very incomplete but it is known that vapour pressure of a liquid is the result of escaping molecules which, although they may leave the surface, just have not enough force to remain entirely free and are drawn back again into the liquid. With such a foundation, the author believes it may be possible to correlate some of the data at present available in contradictory confusion.

Conclusions.

Many problems of the transference of heat remain unsolved and appear sometimes inexplicable. To practical men the abstruse physical explanations of the molecular and other theories often appear elaborate and unnecessary, yet from the considerations so far evolved it is evident that more attention might be paid to this, the molecular aspect of heat transference.

JUNIOR SECTION.

Naval Architecture and Ship Construction (Chapter XVI).

By R. S. HOGG, M.I.N.A.

Resistance and Propulsion.

The resistances experienced by a ship when proceeding through the water may be classified under two heads. The first is termed *frictional resistance* and the second *residuary resistance.* Residuary resistance, which includes both wave and eddy effects, follows the laws of dynamic similarity. As, however, frictional resistance does not do so, it becomes necessary to study and compute the two factors separately.

If R = total resistance in pounds

R_f = frictional resistance

R_r = residuary resistance

$R = R_f + R_r$.

Skin Friction.

Although a good deal of experimental work has been done on this subject, the laws governing it are still but imperfectly understood. It would appear, however, that at very low speeds a thin film of water clings to the surface of the ship, and before motion can take place this film must be sheared from its contiguous layer. The flow would then be termed viscous, and viscous resistances vary directly as the speed. For all ordinary ship speeds the effect of this film on the neighbouring water is to produce innumerable whirling eddies. The motion is then said to be turbulent. If the flow was purely turbulent, the resistance would vary as the square of the speed, but in practice an index rather less than two may be expected.

William Froude carried out a most exhaustive experimental inquiry into the resistance of thin planks towed through the water at varying speeds. He employed planks 21in. deep, $\frac{3}{16}$in. thick, of lengths varying up to 50ft., fitted with a cut water at the fore end in order to eliminate all resistances other than frictional. The speeds covered a range up to 800ft. per minute, and the planks were coated variously with calico, fine, medium and coarse sand, varnish and paraffin wax. He concluded that frictional resistance could be expressed by a formula of the type $R_f = fSV^n$, where

R_f = resistance in lbs.
if S = wetted surface in square feet,
V = speed in knots,
f = a coefficient depending upon length of surface and the nature thereof,
n = speed index, which appeared to be fairly constant over the whole range of experiments.

The values of f for paraffin wax and for varnish were much the same, indicating the probability that the nature of the resistances experienced by a wax model when being towed in a tank are comparable with those experienced by the actual ship.

For lengths up to 10 or 20ft. f may be taken as 0·01.

For lengths about 400ft. f may be taken as 0·0088. n may be taken as 1·825.

Dr. G. S. Baker (National Physical Laboratory) has verified many of these experiments, but suggests that they should be extended to cover lengths up to 800 or 900ft. and the speed should be increased four-fold. Further, he doubts whether results obtained for plane surfaces are strictly applicable to solid forms. Froude was of course aware that there was considerable variation in the velocity of the water relative to the ship for different points on the immersed surface, and that therefore the specific resistance (resistance per square foot) would vary from point to point. He thought, however, that the differences would cancel one another, and that the assumption of uniform velocity of flow over the whole of the surface would prove valid.

Dr. Kempf tested this conclusion by fitting Pitot tubes at selected points on the immersed hull of a German cruiser. He was thus able to measure the velocity of the water relative to the ship at each of these points, calculate the specific resistance for the areas in the immediate vicinity, and by a process of summation estimate the total skin friction for the ship. His results were some 10 per cent. in excess of those obtained from the Froude formula. Although the concensus of opinion seems to indicate that Baker and Kempf are right in their views, more experimental work of a full-scale character must be done before final judgment can be given.

Wetted surface can be calculated by employing Simpson's Rules, or it can be estimated from one of several empirical formulæ.

Kirk gives wetted surface $S = 2LD + \frac{V}{D}$

Denny „ „ „ $S = 1{\cdot}7LD + \frac{V}{D}$

Where S = surface in square feet.
L = length of ship in feet.
D = mean draught in feet.
V = underwater volume in cu. feet.

Denny is probably more correct.

Example 65.

Find the frictional resistance of a ship 400ft. long, 52ft. beam, 24ft. mean draught, block coefficient 0·7, speed 15 knots. Take $f = {\cdot}0088$, $n = 1{\cdot}83$, and use Denny's formula for wetted surface. What horse power is necessary to overcome friction at this speed?

$$S = 1{\cdot}7LD + \frac{V}{D} = 1{\cdot}7LD + \frac{0{\cdot}7L \times B \times D}{D}$$
$$= L(1{\cdot}7D + 0{\cdot}7B)$$
$$= 400(1{\cdot}7 \times 24 + {\cdot}7 \times 52)$$
$$= 30{,}880 \text{ sq. ft.}$$
$$R_f = fSV^n = 0{\cdot}0088 \times 30{,}880 \times 15^{1{\cdot}83} = 38{,}650\text{lb.}$$
$$\text{Frictional horse power} = R_f \times V \times \frac{6{,}080}{60} \times \frac{1}{33{,}000}$$
$$= \frac{38{,}650 \times 15 \times 6{,}080}{60 \times 33{,}000} = 1{,}782 \text{ h.p.}$$

Wave Resistances.

There are three systems of waves formed as a vessel makes its way through the water.

(1) Bow divergent system.
(2) Stern divergent system.
(3) Transverse system.

These are shown diagrammatically in Fig. 188.

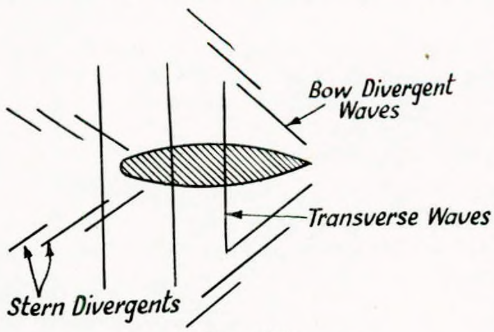


Fig. 188.

The divergent systems are in evidence at all ordinary speeds, but the transverse system does not assume any importance until comparatively high speeds are reached. Incidentally, Froude discovered that when the speed was such that the crest of a transverse system appeared at the stern, the stern divergent waves were hampered in their formation, and a serious augmentation of resistance occurred. This phenomenon shows itself by a "hump" on the "Speed-Power" curve and indicates that at the speed at which it occurs the vessel will prove most uneconomical.

By *eddy resistance* is meant that resistance which is offered by projecting parts, such as "A" brackets, stern bossings, sternpost and bilge keels.

As already mentioned residuary resistances, made up of wave and eddy, follow the laws of dynamic similarity as enunciated by Lord Rayleigh and as applied to the ship by Froude.

The Laws of Dynamic Similarity.

The resistances of similar ships will vary as the cube of the linear dimensions provided the speeds are in the ratio of the square root of the linear dimensions.

(The law does *not* apply to frictional resistance).

Stated symbolically $\frac{R_r}{r_r}=\frac{L^3}{l^3}$ if $\frac{V}{v}=\sqrt{\frac{L}{l}}$

or in other words $R_r \propto L^3$ if $\frac{V}{\sqrt{L}}$ is constant.

Further, since for similar ships the *displacement* will also vary as the cube of the linear dimensions, it follows that

$$\frac{R_r}{r_r}=\frac{\text{displacement }(W)}{\text{smaller displacement }(w)} \text{ if } \frac{V}{\sqrt{L}} \text{ is constant.}$$

When $\frac{V}{\sqrt{L}}$ is constant the ships are said to be running at their corresponding speeds.

Example 66.

The proposed speed of a vessel 400ft. long is 12 knots. At what speed should the tank model be run if it is to have $\frac{1}{16}$ the linear dimensions of the ship?

$$\frac{l}{L}=\frac{1}{16} \therefore \frac{v}{V}=\sqrt{\frac{1}{16}}=\frac{1}{4}$$

$$\therefore v=12\times\tfrac{1}{4}=3 \text{ knots.}$$

Example 67.

The residuary resistance of the model in the previous question was 7lb. What would be the residuary resistance of the actual ship?

$$\frac{R_r}{r_r}=\left(\frac{L}{l}\right)^3=16^3$$

$$\therefore R_r=16^3\times r_r=16^3\times 7=28{,}672\text{lb.}$$

The Effect of Density.

If the model is run in fresh water, some allowance must be made for the fact that the ship will be running in salt water.

It may be assumed that resistance $\propto$ density and therefore if in the above example 28,672lb. is the correct value for water at 1,000oz. per cu. ft. the residuary resistance in water at 1,025oz. per cu. ft.

$=28{,}672\times 1{\cdot}025=29{,}389$lb.

Example 68.

A vessel of 15,000 tons has a speed of 16 knots. What would be the corresponding speed of a similar ship of 12,000 tons displacement?

$\frac{v}{V}=\sqrt{\frac{l}{L}}$, but if $\frac{w}{W}=\frac{l^3}{L^3}$ where $w=$ displacement, it follows that $\frac{l}{L}=\left(\frac{w}{W}\right)^{\frac{1}{3}}$

$$\therefore \frac{v}{V}=\sqrt{\left(\frac{w}{W}\right)^{\frac{1}{3}}}=\left(\frac{w}{W}\right)^{\frac{1}{6}} \quad \text{i.e. } v=16\times\left(\frac{12{,}000}{15{,}000}\right)^{\frac{1}{6}}$$

$=15{\cdot}4$ knots.

Preliminary Estimates of Horse Power.

It is easy to see that if resistance $R \propto$ wetted surface and wetted surface $\propto$ (linear dimensions)$^2 \propto$ (displacement)$^{\frac{2}{3}}$ then in symbols S (surface) $\propto L^2 \propto W^{\frac{2}{3}}$

$\therefore R \propto W^{\frac{2}{3}}$ if speed V is constant.

If W is constant, resistance $(R) \propto$ (speed)^2 i.e. $R \propto v^2$.

Now if $R \propto W^{\frac{2}{3}}$ when V is constant, and $R \propto V^2$ when W is constant,
then $R \propto W^{\frac{2}{3}}V^2$ when both are variable.

Again horse power $(H) \propto R\times v$

$\therefore$ h.p. $\propto W^{\frac{2}{3}}V^2\times V \propto W^{\frac{2}{3}}V^3$.

It is usual to write h.p. $=\frac{W^{\frac{2}{3}}V^3}{C}$, where

C is termed the Admiralty coefficient.

Unfortunately C varies for different speeds in the same ship, and also for different displacements.

In estimating preliminary horse power from this formula, great caution is necessary. Before a reliable C value can be assumed, data for a ship of a similar character must be available, and the operating conditions must not depart sensibly from those of the known ship.

Example 69.

Make a first approximation to the indicated horse power necessary to drive a ship of 12,000 tons at 12 knots if the value of the Admiralty coefficient may be taken as 300.

$$\text{I.h.p.}=\frac{12{,}000^{\frac{2}{3}}\times 12^3}{300}=3{,}000 \text{ h.p. (approx.).}$$

Example 70.

A vessel of 7,000 tons requires 10,000 horse power to drive it at 20 knots. What is the value of the Admiralty coefficient?

$$\text{I.h.p.}=\frac{W^{\frac{2}{3}}V^3}{C}$$

from which $C=\frac{W^{\frac{2}{3}}V^3}{\text{I.h.p.}}=\frac{7{,}000^{\frac{2}{3}}\times 20^3}{10{,}000}=293.$

At the higher speeds it is not always true to say that resistance varies as the square of the speed. It may vary as the cube, in which case the horse power would vary as the fourth power of the speed. If it is proposed to design a new ship on similar lines to one where it is known that horse power is varying as some index of the speed other than 3, the Admiralty coefficient formula would prove unreliable, and the following procedure as illustrated in the example should be used.

Example 71.

The indicated horse power necessary to drive a vessel of 10,000 tons at 25 knots is 24,000. If at this speed the horse power is varying as the fourth power of the speed, what power would be necessary to drive a similar vessel of 12,000 tons at 28 knots?

Corresponding speeds $\propto W^{\frac{1}{6}}$

$\therefore$ speed of existing vessel corresponding to 28 knots in the proposed vessel $=28\times\left(\frac{10{,}000}{12{,}000}\right)^{\frac{1}{6}}=27{\cdot}16$ knots.

Horse power is varying as V^4

$\therefore$ h.p. of existing ship at $27{\cdot}16$ knots $=24{,}000\times\left(\frac{27{\cdot}16}{25}\right)^4$
$=33{,}470$ h.p.

In similar ships resistance $\propto$ (linear dimension)^3 if speed $\propto \sqrt{\text{linear dimension}}$.

Further displacement $\propto$ (linear dimension)^3

$\therefore R \propto L^3 \propto W$ when $V \propto L^{\frac{1}{2}} \propto W^{\frac{1}{6}}$

Hence if new ship is similar to existing ship, and is run at a speed corresponding to 27·16, *viz.,* 28 knots, then required h.p. $\propto RV \propto (W\times W^{\frac{1}{6}}) \propto W^{\frac{7}{6}}$

hence required h.p. $=33{,}470\times\left(\frac{12{,}000}{10{,}000}\right)^{\frac{7}{6}}$
$=41{,}400$ h.p.

Fuel Coefficient.

The economy of operation in respect of fuel consumption may be expressed conveniently in terms of a *fuel coefficient.*

The horse power developed must be proportional to the fuel consumption per unit of time, and hence if h.p. $\propto W^{\frac{2}{3}}V^3$, so also will fuel consumption per unit of time $\propto W^{\frac{2}{3}}V^3$, and it is usual to write fuel consumption

per 24 hours $= \dfrac{W^{\frac{2}{3}}V^3}{k}$

or $k = \dfrac{W^{\frac{2}{3}}V^3}{\text{fuel consumption per 24 hours}}$

Example 72.

A vessel burns 40 tons of fuel per day when steaming at 16 knots. If the displacement is 11,000 tons, what would be the fuel coefficient?

$$k = \frac{W^{\frac{2}{3}}V^3}{\text{consumption per 24 hrs.}} = \frac{11{,}000^{\frac{2}{3}} \times 16}{40}$$
$$= 50{,}640$$

The Determination of Horse Power from Model Experiments.

The foregoing estimates of horse power required to drive a new ship can only be regarded as first approximations, and it is now common practice to have a model experiment carried out at the Teddington Tank (National Physical Laboratory). Several of the larger shipbuilding firms have their own experimental tanks, and in addition there is the Admiralty establishment at Haslar.

The William Froude Tank at Teddington is nearly 500ft. in length, 30ft. wide, with a depth of water of 12·5ft. The models used are made of paraffin wax. They are finished accurately to scale, and measure anything from 10 to 20ft. in length according to the size of the ship and the scale employed. They are towed by means of a travelling bridge from which they are suspended in such a way as to be free to roll, pitch or yaw. An automatic record is taken of total resistance and speed. For more detailed information regarding the construction of the model and the experimental procedure, reference should be made to a paper read by Dr. G. S. Baker before the Institution of Naval Architects in 1911.

Needless to say the tank authorities have developed their own technique for recording and analysing the results of the experiments, but the basic principles will be understood from the following description :—

(1) The model is run at the correct corresponding speed

i.e. $v = V\sqrt{\dfrac{l}{L}}$ where $\begin{cases} v, l \text{ are for model} \\ V, L \text{ are for ship} \end{cases}$

(2) The total resistance for the model (r) is recorded from the experiment.

(3) The frictional resistance for model is calculated from the formula $r_f = fsv^{1 \cdot 825}$
where f = coefficient (0·01 approximately),
s = wetted surface of model.

(4) The residuary resistance of the model (r_r) is obtained by subtracting (3) from (2)
i.e. $r_r = r - r_f$

(5) The residuary resistance of the ship R_r is obtained from the law of dynamic similarity,

i.e. $\dfrac{R_r}{r_r} = \dfrac{L^3}{l^3}$

or $R_r = \left(\dfrac{L}{l}\right)^3 \times r_r$.

(6) The frictional resistance of ship is calculated from $R_f = fsv^{1 \cdot 825}$
where f = coefficient (0·0088),
s = wetted surface of model,
v = ship speed in knots.

(7) Total resistance of ship $= R = R_r + R_f$ lb.

(8) The effective horse power of ship (e.h.p.)

$$= \frac{R \times V \times \frac{6{,}080}{60}}{33{,}000}$$

The value for R_r should be corrected for density as previously explained.

(9) No mention has been made in these discussions of air resistance. It is of great importance, but experimental data is rather scanty, and it is the usual practice to add on a percentage, say 10 per cent. for a slow speed cargo vessel, up to 30 per cent. for a high speed liner, to cover this effect.

(10) The ratio $\dfrac{\text{e.h.p.}}{\text{i.h.p.}}$ is called the propulsive coefficient.

Example 73.

The total resistance of a model 16ft. long is found to be 4lb. The wetted surface is 50 sq. ft. Estimate the effective horse power necessary to drive a vessel designed on this model, if the length of the vessel is 400ft. and speed of model was 2·5 knots. Take

$f_{\,model} = 0{\cdot}01$, $f_{\,ship} = 0{\cdot}0088$, $n = 1{\cdot}825$.

(1) Frictional resistance of model at 2·5 knots
$= r_f = fsv^n = 0{\cdot}01 \times 50 \times 2{\cdot}5^{1 \cdot 825}$
$= 2{\cdot}66$lb.

(2) r_r for model $= r - r_f = 4 - 2{\cdot}66 = 1{\cdot}34$lb.

(3) Residuary resistance of ship $R_r = \left(\dfrac{L}{l}\right)^3 \times r_r$

$$= \left(\frac{400}{16}\right)^3 \times 1{\cdot}34$$
$$= 20{,}940\text{lb.}$$

Correcting for density
$R_r = 20{,}940 \times 1{\cdot}025 = 21{,}464$lb.

(4) Frictional resistance of ship
$= R_f = fSV^n = 0{\cdot}0088 \times S \times V^{1 \cdot 825}$
Now wetted surface on similar ships $\propto$ square of linear dimension
$\therefore$Wetted surface of ship S

$$= 50 \times \frac{L^2}{l^2}$$
$$= 50 \times \left[\frac{400}{16}\right]^2 = 31{,}250 \text{ sq. ft.}$$

Corresponding speed for ship V

$$= v\sqrt{\frac{L}{l}} = 2{\cdot}5\sqrt{\frac{400}{16}} = 12{\cdot}5 \text{ knots.}$$

$\therefore R_f = 0{\cdot}0088 \times 31{,}250 \times 12{\cdot}5^{1 \cdot 825}$
$= 27{,}630$lb.

(5) Total resistance $R = R_r + R_f$
$= 21{,}464 + 27{,}630 = 49{,}094$lb.

(6) $\text{E.h.p.} = \dfrac{R \times V \times 6{,}080}{33{,}000 \times 60} = \dfrac{49{,}094 \times 12{\cdot}5 \times 6{,}080}{33{,}000 \times 60}$
$= 1{,}884$ h.p.

Propulsion.

The overall efficiency of propulsion will depend upon three factors, *viz.:* The engine efficiency, hull efficiency and the propeller efficiency.

The engine efficiency will depend upon the type of engine employed and the conditions under which it is run. It will be considered here only insofar as it affects the revolutions of the propeller.

Wake.

As the vessel proceeds through the water there is a large mass of water which follows the ship known as the wake. The wake obviously has a forward velocity and therefore the rate of progress of the screw through the wake water is less than the speed of the ship. This has a beneficial effect, and must be considered when estimating efficiency of propulsion.

Let v_1 be the speed of the propeller through the wake;

Let v be the speed of the ship.

Froude suggested that $v = v_1 \ (1+w)$ where w is to be known as the wake factor.

Thrust Deduction Factor.

The action of the propeller in disturbing the water around the stern of the ship produces some augmentation of resistance, and is therefore harmful. It is convenient to regard this as a thrust deduction effect and if R = total resistance of ship (which would otherwise equal the thrust)

$R = T \ (1-t)$ where T is the thrust which must be developed to drive the ship at a speed v, and t is called the thrust deduction factor.

$$\text{The thrust horse power} \propto (T \times v_1) \propto \left(\frac{R}{1-t} \times \frac{v}{1+w}\right)$$

$$\propto \left(\frac{Rv}{(1-t)\ (1+w)}\right)$$

now $R \times v$ is a measure of the useful work done in towing the ship and is therefore proportional to the effective or tow rope horse power.

$$\text{Hence if } k \ (TV_1) = k\left(\frac{RV}{(1-t)\ (1+w)}\right)$$

$$\frac{RV}{TV_1} = (1-t)\ (1+w)$$

The ratio $\dfrac{\text{effective horse power}}{\text{thrust horse power}}$ is termed hull efficiency.

$$\therefore \text{Hull efficiency} = \frac{\text{e.h.p.}}{\text{t.h.p.}} = \frac{RV}{TV_1} = (1-t)\ (1+w).$$

Froude and others have determined the values of t and w separately for different types of ships, under varying working conditions, and the value $(1-t)\ (1+w)$ is generally of the order 98 per cent. or thereabouts. It would seem then that the hull efficiency factor is not of much importance to the designer, but a knowledge of the separate factors is rather necessary when designing the blade itself.

Dr. Baker puts forward the following simple and useful analysis :—

$$\text{Propulsive efficiency of screw } (\eta) = \frac{\text{E.h.p.}}{\text{S.h.p.}} = \frac{E}{S}$$

which may be split up as follows :—

$$\eta = \frac{E}{S} = \frac{RV}{S} = \frac{R}{T} \times \frac{V}{V_1} \times \frac{TV_1}{S_1} \times \frac{S_1}{S}$$

The product of the first two factors, viz. $\dfrac{RV}{TV_1}$, is hull efficiency, $\dfrac{TV_1}{S_1}$ is the ratio of the work done by the screw when turning in the wake compared with the work which would be done by the screw if operating in open water under the same revolutions and thrust. It is, of course, the screw efficiency in open water.

$\dfrac{S_1}{S}$ is called the relative rotative efficiency and may be taken for all practical purposes as unity.

Hence propulsive efficiency of screw = hull efficiency × screw efficiency in open water.

Overall efficiency = hull efficiency × screw efficiency in open water × engine efficiency.

From the point of view of the naval architect the most important factor in this equation is the *screw efficiency* in the open. From the engineer's standpoint engine efficiency is of paramount importance.

The work of propeller design is difficult and elusive, and the results are not always satisfactory. No known theory completely fits the facts, the usual method of design being based on the experimental work of W. Froude which was very ably extended by his son, Dr. R. E. Froude. The problem is dealt with exhaustively in Sir John Bile's book on resistance and propulsion (Griffin) and in Dr. G. S. Baker's book "Ship Form and Screw Propulsion" (Constable).

The Propeller

In the following notes some of the more important definitions will be dealt with, and certain characteristics discussed, but no attempt will be made to deal with design.

The driving face of a propeller is the surface which is seen by the observer when looking from aft.

The back of the blade is that which is seen when standing forward and looking aft.

The leading edge or forward edge is that which cuts the water when proceeding ahead.

The trailing edge is the following edge.

A right-hand screw is one which is turning in a clockwise direction when viewed from aft and when the vessel is moving ahead.

Blade area is the actual developed area of the face of the blade.

Developed area of the propeller is the sum of the true areas of each of its blades.

The disc area is the area of a circle swept out by the tip of a blade.

The disc area ratio $= \dfrac{\text{developed area of propeller}}{\text{disc area}}$.

Projected area is that obtained when the blade surfaces are projected on to a transverse vertical plane.

Blade width ratio $= \dfrac{\text{maximum true width of blade}}{\text{radius of propeller}}$

$$= \frac{B}{R}$$

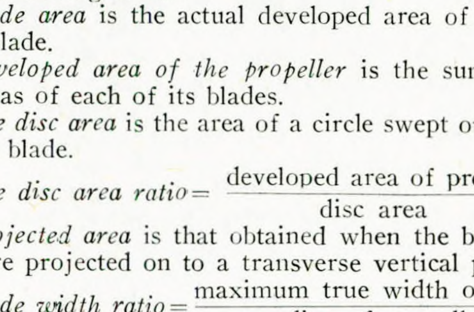


FIG. 189.

Root thickness is the effective thickness at the boss $= t$.

Blade thickness ratio =

$$\frac{\text{blade thickness if continued to centre of disc}}{\text{diameter of propeller}} = \frac{t_1}{2R}$$

Pitch is the distance any specified point on the face of blade would move forward in one revolution if turning in a solid nut.

The simplest and on the whole most satisfactory propeller has true helical face pitch. The pitch is then the same for any section of the blade.

$$\textit{Pitch ratio} = \frac{\text{face pitch}}{\text{diameter}} = \frac{P}{D} = p.$$

Axially increasing pitch indicates that pitch is increasing from leading to trailing edge.

Radially increasing pitch means that pitch is increasing towards the tip of the blade.

Slip in its popular interpretation $= \dfrac{RP - V}{RP}$

where R = revolutions per minute
P = pitch in feet
V = ship speed in feet per minute.

The correct name for this however is *apparent slip*.

The actual speed of the propeller through the wake has already been shown to equal v_1 where $v_1 = \dfrac{v}{(1+w)}$

v_1 may be about $0{\cdot}9\,v$.

Hence the *apparent true slip* $= \dfrac{RP - V_1}{RP}$.

The face pitch of a blade is not the true pitch, owing to the influence of blade thickness on the behaviour of the screw. The true or *analysis pitch* $= 1{\cdot}02 \times$ face pitch (approx.).

Hence *real slip* $= \dfrac{1{\cdot}02RP - 0{\cdot}9V}{1{\cdot}02RP}$ (approx.).

Example 74.

It is desired to work a propeller at 20 per cent. slip. The revolutions are fixed at 80 per minute and the speed of the ship is to be 12·5 knots. Approximate to the face pitch. Use the apparent slip formula.

$$\frac{RP - 101V}{RP} = \text{slip fraction. [Note : } \frac{6{,}080}{60} = 101 \text{ app.]}$$

$$\frac{80P - 101 \times 12{\cdot}5}{80P} = 0{\cdot}2$$

$$80P - 1262{\cdot}5 = 16P$$

$$64P = 1262{\cdot}5$$

$$P = 19{\cdot}75\text{ft. (app.).}$$

Example 75.

A propeller is turning at 60 r.p.m. when the ship is proceeding at 10 knots. If the pitch of the propeller is 17ft. and the speed through the wake is 0·9 of the ship speed, determine the apparent true slip.

In this instance use the formula

$$\text{Apparent true slip} = \frac{RP - V_1}{RP}$$

then $\dfrac{60 \times 17 - 0{\cdot}9 \times 10 \times 101}{60 \times 17}$ = apparent true slip

Apparent true slip $= \dfrac{1{,}020 - 909}{1{,}020} = \dfrac{111}{1{,}020} =$ 11 per cent. nearly.

Rake.

A blade is said to be raked forward or aft, according as the centre line at the blade tip is before or abaft the centre line at the root.

The Geometry of the Blade.

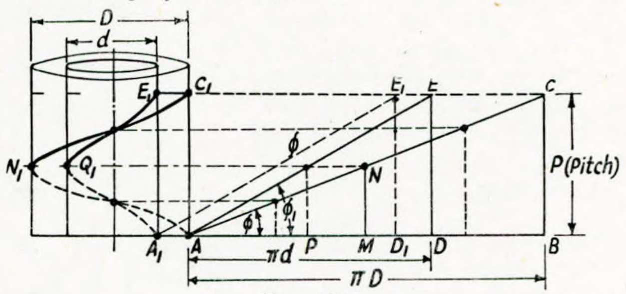


FIG. 190.

Consider a right-angled triangle ABC whose base measures the circumference of a cylinder of diameter D. If this triangle be wrapped round the cylinder, the hypotenuse will trace out a true helix. The quantity BC will be the pitch of the helix. If now another triangle $A_1D_1E_1$ of the same pitch, but of base πd, where d is the diameter of some smaller cylinder, be drawn, it will, if wrapped around the smaller cylinder, trace out a helix $A_1Q_1E_1$ such that the surface containing both helixes will be a true helical surface. The face of a constant pitch propeller will be a portion of such a surface.

$\tan\phi = \dfrac{P}{\pi d}$ where P = face pitch
D = tip diameter
ϕ = pitch angle of tip.

$\tan\phi_1 = \dfrac{P}{\pi d}$ where d = root diameter
and ϕ_1 = pitch angle at the root.

For intermediate sections the pitch angle would be between the values ϕ and ϕ_1. It should be noted that pitch is constant over the surface, but pitch angle decreases for each section as the tip is approached. For convenience in drawing the propeller, it is better to make a drawing as shown in Fig. 191 (d).

The height of the triangle is made equal to $\dfrac{P}{2\pi}$ and the several bases are section radii.

Thus for the tip $\tan\phi =$

$$\frac{P}{2\pi} \times \frac{1}{R} = \frac{P}{2\pi R} = \frac{P}{\pi D} \text{ as before.}$$

Procedure.

Fig. 191(a) shows the developed blade contour as determined from the design data. Draw a series of parallel lines *aa, bb, cc,* etc., at distances R_1 R_2 R_3, etc., from the disc centre. In Fig. 191(d) make height of triangle $= \dfrac{\text{P (pitch)}}{2\pi}$ and set out OR_1 OR_2 OR_3, etc., as measured from Fig. 191(a). Join R_1M, R_2M, etc.; then MR_1O, MR_2O, etc., are the pitch angles for the various sections.

In Fig. 191(b) the root thickness is arranged symmetrically about the centre line of the boss. So that x = half root thickness. Draw a line nm passing through c_1 at a distance x from the centre line in Fig. 191(c). Now draw in lines representing the various pitch angles all intersecting at c_1, Fig. 191(c). Make $a_1a_1 = aa$, $b_1b_1 = bb$, $c_1c_1 = cc$, etc. Then the curve drawn through $a_1b_1c_1$, etc., represents the edge of the blade, and the surface depicted is the face of the blade. For convenience a root section has been shown.

For the guidance of the pattern makers, cross sections should be made for each section of the blade. This can be done by picking up the thickness from Fig. 191(b), such as t_3, and setting it out as at cc in Fig. 191(a).

The curve cTc may be drawn to the eye, but bear in mind that sharp edges are advantageous. The projections in Fig. 191(a) and 191(b) are obtained in the usual way, and have no particular practical value.

To Measure Pitch from the Actual Blade.

There are several pitchometers available for measuring pitch. They all, however, depend upon the same fundamental fact, and it is proposed therefore to describe an elementary but successful procedure which must necessarily be the basis of any more elaborate method.

Assume the propeller blade to be in a horizontal position. Pass a plumb line round the blade so that it hangs vertically from a point p on the trailing edge Fig. 192(b). Measure the distance of this plumb line from the centre line of shaft, *i.e.* obtain the value of r in Fig. 192(a). Set up a horizontal batten in a fore and aft plane which just touches the plumb line and the leading edge of the blade. Now by any convenient means measure the distances b and a, Fig. 192(b).

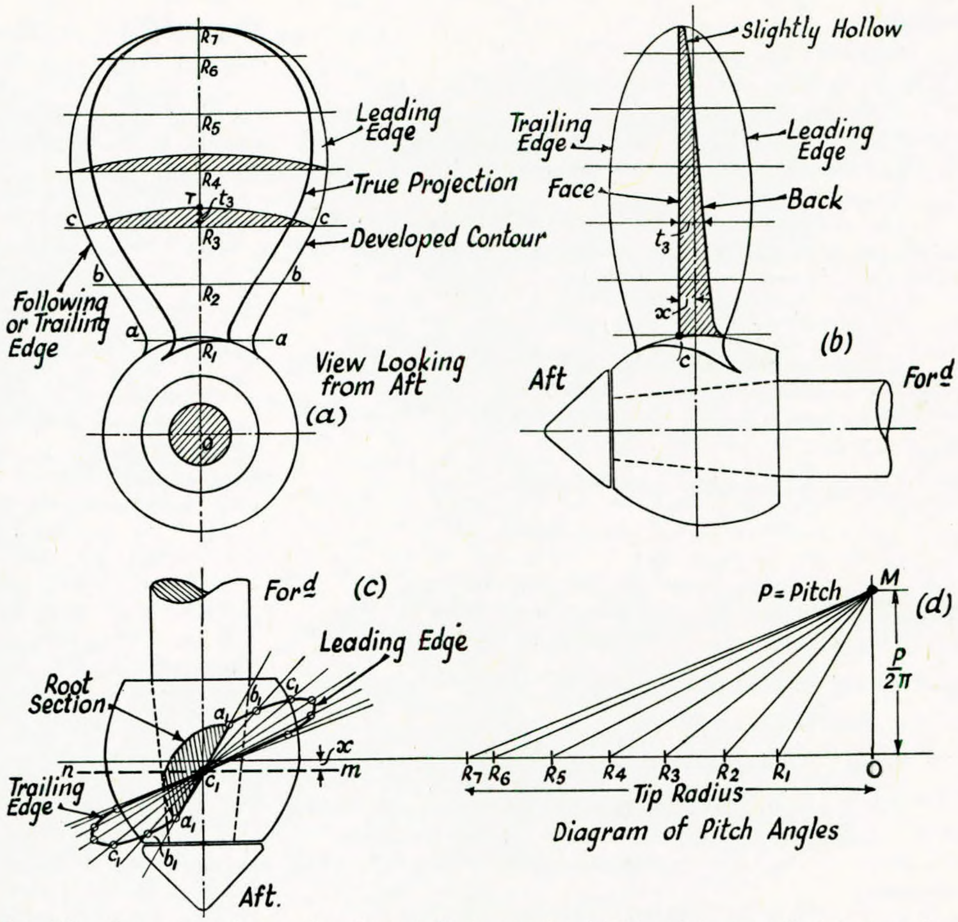


FIG. 191.—Geometry of propeller (true helical screw); right-handed, three-bladed, low disc area ratio.

Clearly $\frac{b}{a} = \tan\phi$ where ϕ is the pitch angle for the section.

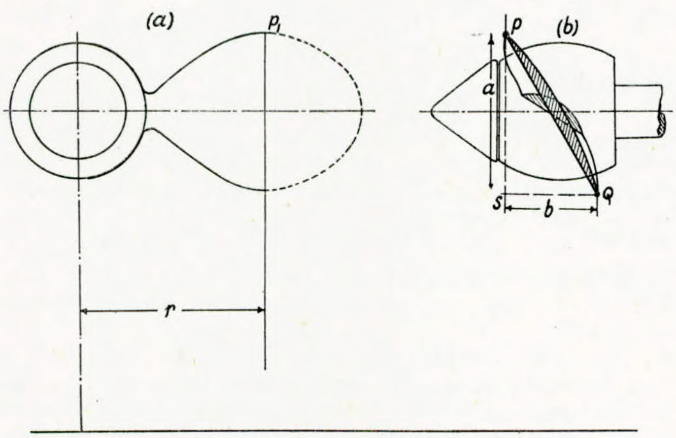


FIG. 192.

It has been shown previously that $\tan\phi = \frac{\text{Pitch}}{2\pi r}$ where r is the radius of the section.

$$\therefore \frac{\text{Pitch}}{2\pi\times r} = \frac{b}{a}$$

or Pitch $= \frac{2\pi rb}{a}$ where r, b, a have been measured.

Some General Remarks on Propellers.

The highest propeller efficiencies are in the neighbourhood of 75 per cent. Efficiency is very sensitive to slip, and the best values are obtained for slips in the region of 20 per cent. Dimensions and pitch can be varied over a fairly wide range without sacrificing much efficiency. Revolutions are fixed by the type of engine, and hence this must always be the starting point in fixing dimensions. Diameter should be large for slow-speed ships, but can be smaller for high speeds. The blade tip should be at least 1ft. clear of the hull. The centre of the propeller should be immersed to a depth equal to $0{\cdot}8\times$ radius of the propeller. Wide tip blades develop more thrust but lower efficiency slightly. Since maximum intensity of thrust is experienced near the broadest edges of the blade there will be some tendency for broad tips to break. Blades should therefore be well rounded.

There is a minimum blade area consistent with the avoidance of cavitation. Barnaby suggested as the pressure limit for cavitation pressure per sq. in. of projected area should not exceed $10{\cdot}85+\frac{3}{8}h$ lb., where h is immersion of blade tip in feet. G. S. Baker suggests it should not exceed 13lb. per sq. in. unless the blade is exceptionally well buried.

Blade surfaces should be machined very carefully and made as smooth as possible. A rough surface is very inefficient. Blades should be as thin as possible consistent with strength. It is for this reason, coupled with the fact that more accurate casting can be obtained, that bronze propellers should be preferred.

Disc area ratios vary considerably, but a ratio of 0·5 is typical for a medium-speed ship. In modern designs pitch ratios of from 0·8 to 1·2 are commonly employed.

Revolutions should be low in low-speed ships. They can be increased as speed increases without sacrificing very much efficiency. In high-speed motor boats the best efficiencies occur when revolutions pass into the thousands.

Cavitation.

The following remarks are based on the opinions quoted by G. S. Baker in his book on resistance and propulsion. "Cavitation is the name given to the phenomenon which makes itself felt by the absence of proper increase in screw thrust with increase of torque".

Causes.

(1) An advancing and rotating screw produces in front of it a suction which causes the water to move in towards the screw. If this suction should tend to be stronger than the sum of the still water pressure plus atmospheric pressure, the water will not be able to follow up, and cavities will form both on the back of the blade and may be on the face near the tip.

(2) Irregularity in rotation as is experienced in reciprocating engines causes slip variations during the revolution. This may produce areas of discontinuous flow round the blade tips.

(3) Insufficient clearance between blade tips and hull may also cause eddying near the blade, and the water is thus unable to follow up properly.

Erosion is the wastage of material which results from the collapse of the cavitation areas. The energy of collapse is all expended over a small area of the blade, and exhibits itself in a hammer effect. For a clearer and more detailed exposition see G. S. Baker ("Resistance and Propulsion").

Miscellaneous Problems on Propulsion.

Example 76.

A propeller 20ft. pitch, 16ft. diameter, 60 r.p.m., steam pressure 60lb. If the pressure remains the same, but the revolutions increase to 70, find the new pitch.

Thrust $\propto$ resistance, and

thrust $\propto$ (speed)^2.

Then assuming no losses, thrust × revolutions × pitch = resistance × ship speed

$$TNP = RV$$

I.h.p. $\propto V^3$

A = area of lower pressure cylinder,

p_m = mean referred pressure,

$A \times p_m \times R \propto I.h.p. \propto V^3$

$$\therefore \frac{A p_m N_1}{A p_m N_2} = \frac{v_1^3}{v_2^3} = \frac{(N_1 P_1)^3}{(N_2 P_2)^3}$$

$$\therefore \left(\frac{N_2}{N_1}\right)^2 = \left(\frac{P_1}{P_2}\right)^3$$

$$\left(\frac{70}{60}\right)^2 = \left(\frac{20}{P_2}\right)^3 \qquad \text{from which } P_2 = 18\text{ft.}$$

Example 77.

It is proposed to alter revolutions from 58 to 64. Diameter of blade is 17ft. 6in.; original pitch 21ft. Mean referred pressure 28lb. New mean referred pressure 24lb. Find the new pitch.

$$\frac{A p_m N_1}{A p_m N_2} \propto (\text{speed})^3 = \left(\frac{N_1 P_1}{N_2 P_2}\right)^3$$

$$\text{i.e. } \frac{28 \times 58}{24 \times 64} = \frac{58^3}{64^3} \times \frac{21^3}{P_2^{\,3}}$$

from which $P_2 = 18{\cdot}67$ft.

A Note on Speed Trials.

The object of the trials is to determine useful data in respect of the economy of operation of the ship, and to accumulate technical experience for guidance in producing future designs.

The trials should be carried out on a special *measured mile* with buoys marking the distance. A day should be chosen when the least interference from wind, sea, currents, etc., is to be expected, and the trial should be run where the depth of water is such as to eliminate effects due to shallowness.

The vessel is run at a series of speeds from comparatively low to the maximum for the ship. There should be at least four runs at each speed, *viz.*, two with the tide and two against.

To eliminate tidal effect what is known as the *mean of means* is employed.

If V_1 V_2 V_3 V_4 are the successive speeds with and against the tide, the mean is taken as follows:—

$$\begin{array}{llll} V_1 & & & \\ & \dfrac{V_1+V_2}{2} & & \\ V_2 & & \dfrac{V_1+2V_2+V_3}{4} & \\ & \dfrac{V_2+V_3}{2} & & \dfrac{V_1+3V_2+3V_3+V_4}{8} \\ V_3 & & \dfrac{V_2+2V_3+V_4}{4} & \\ & \dfrac{V_3+V_4}{2} & & \\ V_4 & & & \end{array}$$

The final quantity is the correct mean. This statement can be justified mathematically. It is necessary on each run to record i.h.p. (s.h.p.), revolutions and time of run.

Several cards should be taken so as to obtain a good result. The revolutions are handy in order to get slip values and compare at the different speeds. Attention should be paid to the maintenance of uniform boiler pressure, steady revolutions, and the helm should be used as little as possible. Indicator cards may be in error from 2 to 5 per cent. Should revolutions in a twin-screw ship differ, it is wise to take the average. Fuel expenditure should also be recorded.

The following curves are plotted from the results of the trial:—

(1) I.h.p. against speed.

(2) I.h.p. against $v/\sqrt{L}$.

(3) I.h.p. against Admiralty coefficient or fuel coefficient.

The complete analysis of trial data may involve a considerable amount of work, and what has been stated here represents the minimum that should be attempted.

A typical speed - power curve is of the form indicated in Fig. 193.

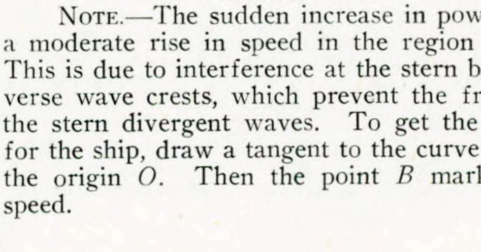


Fig. 193.

Note.—The sudden increase in power necessary for a moderate rise in speed in the region of the point A. This is due to interference at the stern by the bow transverse wave crests, which prevent the free formation of the stern divergent waves. To get the economic speed for the ship, draw a tangent to the curve passing through the origin O. Then the point B marks the economic speed.

OBITUARY.

MR. GEORGE WILLIAM BUCKWELL.

It is with deep regret that we record the death of Mr. George William Buckwell (Past Vice-President and Member No. 224), who passed away at Liverpool on Friday, 20th February, 1942, aged 78 years.

Of a seafaring family, Mr. Buckwell was born at Brighton and educated at Brighton Grammar School, where he displayed the marked ability and diligence which were to result eventually in his being one of the three youngest students to be awarded a Whitworth Scholarship. After an all-round training in craftsmanship as a pupil of the late William Stroudley, locomotive superintendent of the London, Brighton and South Coast Railway, Mr. Buckwell spent some time on experimental work and attended the Brighton School of Art. During the next two years he had a variety of experience in several works, including positions of chief draughtsman and works foreman. At the age of twenty he went to sea, and a fortnight after passing his twenty-sixth birthday he obtained his Extra-First Class Board of Trade Certificate. Eight months later, in 1890, he sat the examination for Board of Trade Surveyor and gained first place.

Mr. Buckwell served as a surveyor at Liverpool, Sunderland and Barrow, became senior surveyor at Glasgow in 1912, and was promoted to the position of principal officer of the Eastern District of England, stationed at Hull, in 1919, where he remained until his retirement in 1928 at the age of 65, having been in the service of the Board for over 38 years.

In 1890 Mr. Buckwell was elected a Member of The Institute and thenceforward he maintained a keen interest in its work. During 1905-10 he held the office of Vice-President for the Barrow area and was later elected as Vice-President for the Hull area, which office he held during 1922-27.

Mr. Buckwell was a Member of the Institution of Civil Engineers, the Institution of Mechanical Engineers, and the Institution of Naval Architects, and of two masonic lodges, having joined the craft in 1884. His death will occasion widespread regret among his many friends in the marine engineering world.

The late Mr. James George Hawthorn.

MR. JAMES GEORGE HAWTHORN.

The deep regret which the Council feel in recording the death of Mr. James George Hawthorn (Past Member of Council and Member No. 6), who passed away at Forest Gate on Tuesday, 17th February, 1942, will be shared by the wide circle of The Institute membership to whom he was so well known.

Mr. Hawthorn was the last survivor of that group of marine engineers, conscious of the need for the intellectual and social advancement of their profession, who initiated the discussions which resulted in the foundation of The Institute in 1889. Towards the end of 1888, at a meeting held in Mr. Hawthorn's marine engineering school in East India Dock Road, an organizing committee was formed, on which Mr. Hawthorn served, to propagate the idea of forming such a society amongst engineers of ships in the London docks. The work of this committee resulted in the foundation of The Institute on the 2nd February, 1889, and Mr. Hawthorn was elected a member of the first Council. His interest in The Institute was unflagging, and he served almost uninterruptedly on the Council until, in 1927, ill health compelled his retirement, since when he retained a lively interest in The Institute's activities.

Born at Devonport in 1853, Mr. Hawthorn was educated at the Royal Naval School, Greenwich, from which he passed out in 1869 with the First Medal in Navigation and Nautical Astronomy. He was nominated by the Admiralty for examination for engineer student in H.M. Dockyard, and came out first in this all-England examination. He then served in H.M. Dockyard, Sheerness, which he subsequently left to take up civil engineering. Later he joined the Royal Engineers with a view to going to India in the Madras or Bombay Staff Corps, but the outbreak of the Ashanti war prevented this object being achieved. After three years' service he left the Royal Engineers and took an appointment with Messrs. Earle's Shipbuilding and Engineering Co., Ltd., of Hull. Soon afterwards he commenced his sea career and very quickly obtained an Extra-First Class Board of Trade certificate. In 1888, desiring to live ashore, Mr. Hawthorn established a school in London to coach marine engineers for the Board of Trade examinations. This business prospered so well that he withdrew his candidature for a Board of Trade surveyorship for which he had passed the examination and for which he was first reserve for a subsequent vacancy. Mr. Hawthorn carried on his business as a coach until 1928, when ill health compelled his retirement. During this period he had over 6,000 successful pupils.

Mr. Hawthorn, who leaves a widow, was of adventurous disposition, and once successfully undertook a walking tour across Europe. He will long be remembered for his sterling character and forceful personality. The Chairman of Council and the Secretary represented The Institute at the funeral, which took place at Wanstead Cemetery on Monday, 23rd February, 1942.

ELECTION OF MEMBERS.

List of those elected by the Council during the period 22nd January to 26th February, 1942.

Member.
Thomas Martin Buchanan.
William George Findlay.
Robert Haswell Gardner.
Frederick Hurst.
Frederick Cavill Lloyd.
John Evelyn McTaggart.
William Albert Nevin.
Adrianus Dirk Olie.
Edward Stephenson.
Eric Francis Truscott, Lt.(E), R.N.V.R.
Thomas Alfred Mansfield Searle.
Henry Foxwell Sherborne.
Arthur James Tweddle.
Patrick Kilgour Wiener, Sub.-Lt.(E), R.N.V.R. (S.A.).

Associates.
Joshua Brewer.
James Adam Barbour-Caldwell.
George Frederick Castle.
Harry Edgar Durrant.
Norman Vernon Laskey.
Alan Mitchell McPhee.
John Richardson.
Charles William Thomas Saunders, Lt.(E), R.N.

Graduate.
Norman Blair.

Student.
Alistair Maurice Archibald Denny.

Transfer from Associate Member to Member.
Wilfred Ernest Grant, Temp'y. Lt.(E), R.N.
Alfred James Bescoby.

Transfer from Associate to Member.
George Dick Pounder.

ADDITIONS TO THE LIBRARY.

Purchased.

Kempe's Engineer's Year Book, 1942. Morgan Bros. (Publishers) Ltd., 35s. net.

Abstracts of the Technical Press

"The Use of Aluminium Alloys in Vessels of the U.S. Navy".

The booklet bearing the above title contains pertinent information and instructions regarding aluminium alloy fixtures and fittings on board U.S. naval vessels. The chapter on the design of such components and structures deals with mechanical failures as well as failures through corrosion of the material, whilst another chapter is devoted to fabrication, *i.e.*, to lay-out, forming, machining, erection, riveting, bolting, welding, preservation, and contact with dissimilar metals and wood. Other chapters concern the repair and upkeep of aluminium alloy fittings and structures, the manufacture of castings of this material in repair shops and tenders, supply procedure for replacement of stocks, and general information. An appendix of six sections is devoted to technical data.—*"Technical Bulletin No. 1", by Lt.-Cdr. S. N. Pyne, U.S.N., Lt. P. W. Snyder, U.S.N., and H. D. McKinnon, issued by the Bureau of Ships, U.S. Navy Department, September, 1940.*

"Nomenclature of Naval Vessels".

This handbook has been prepared primarily for use in the training establishments at the various Navy Yards and Stations of the U.S. Navy. The Service designation of the various parts and components of the ship's structure is given in general terms, the nomenclature of decks and numbering of watertight compartments is explained, and a complete glossary of the shipbuilding terms current in the United States is appended. Lists of the symbols used in specifications and of the abbreviations commonly employed in the preparation of marine engineering and shipbuilding drawings are likewise given.—*U.S. Navy Department publication, June, 1941.*

High Pressure Steam Machinery of New German Tug.

The Teltow shipyard recently completed the construction of a twin-screw tug of an entirely new design for service on the Elbe and the canal system of Central Germany. Hitherto it has been the practice to employ Diesel-engined tugs, capable of towing up to 2,000 tons at a speed of 4·0-4·5 m.p.h., for this service, but the growing shortage of oil and oil products has now made it necessary to have recourse to home-produced fuel. There could be no question of using low-pressure steam machinery, as the weight involved would have been inadmissible for the dimensions of a tug which has to traverse the locks of the Central Canal. The problem was therefore solved by the installation of a Schmidt-Hartmann indirectly-fired high-pressure boiler. The new tug has an o.a. length of just over 106ft. and a beam of about 19ft., the total length of the machinery space being 31ft. Contrary to the usual practice, and in order to maintain the correct fore-and-aft trim of the vessel as the fuel on board is consumed, the engines are located forward of the steam generator. The Schmidt-Hartmann boiler differs from all other watertube boilers in that it does not have to be fed with distilled water, and for this reason it is particularly suitable for a small vessel carrying a minimum E.R. staff of limited technical skill. The heat required for evaporation is obtained indirectly through the agency of a fluid which, in this instance, is distilled water. The combustion chamber is surrounded by the tubular coils of the primary evaporating system, in which distilled water is circulated, and the steam generated in this system is passed through the internal coils of a large cylindrical heat exchanger, in which it gives up its heat to the surrounding water of the secondary evaporating system, thereby turning the latter into steam and being itself condensed. As the distilled water circulates in a closed circuit, scarcely any replenishment is necessary after the initial filling, whilst fresh water of any reasonable degree of purity can be utilised for the make-up feed of the secondary evaporating system. The boiler supplies steam at a pressure of about 767lb./in.² and total temperature of round about 850° F., to two sets of quadruple-expansion three-crank engines equipped with live-steam reheating coils between the second- and third-expansion stages. The engines exhaust to a surface condenser cooled by chemically-treated circulating water drawn from the canal. Air is excluded from the cooling-water system by a steam jet which is led into the main inlet, with a coke filter to stop any oil from being brought over by the steam. The feed water drawn from the condenser is heated up to about 212° F. in a smoke-box economiser, after passing through a feed heater in which it is heated by the exhaust steam from the turbine used for driving the forced-draught fan of the boiler. This fan supplies combustion air to the furnace at a pressure of 1·4 to 1·6in. w.g. The boiler furnace is fired by a Weck mechanical stoker, driven by a chain from one of the propeller shafts. The fuel normally used is Westphalian coal (No. 4 size nuts or fine coal), and ash separation is effected automatically. The maximum steam output of the boiler, when clean, is from 1·0 to 1·2 tons/hr., with a primary steam pressure of 1,350lb./in.². The safety valve on the primary steam system is set for a pressure of 1,563lb./in.² and never lifts in service. The total water capacity of the primary and secondary circuits is only 121 gallons, but no difficulty is experienced in preventing the secondary steam safety valve from lifting even if the main engines are stopped suddenly. In service, no regular feed make-up is required for the primary system, as it is only necessary to make up losses at intervals, when the water level drops below a certain point. The boiler attendant has, in practice, only to deal with the make-up feed required for the secondary system. The main engines have two single-acting cylinders of 5·5in. and 8in. diameter, respectively, and two double-acting cylinders of 8·86in. and 19·68in. diameter, respectively, with a combined output of 150 i.h.p. at 220 r.p.m. Piston type slide valves with spring rings are fitted to all the cylinders. A summary of the trial results obtained is given in the following table:—

Secondary steam pressure ...	766lb./in.²
Temperature of water at economiser outlet ...	310° F.
Steam temperature at primary steam superheater outlet ...	810° F.
Steam temperature at superheater outlet ...	897° F.
Hourly evaporation of steam ...	2,178lb.
Hourly consumption of fuel ...	285lb.
Calorific value of coal used ...	12,692 B.Th.U.
CO_2 content of uptake gases ...	13·6%
Water evaporated per lb. of coal ...	16·9lb.
Boiler efficiency ...	81·96%
Receiver pressure at H.P. cylinders ...	210lb./in.²
Steam temperature at H.P. receivers ...	853° F.
Vacuum in condenser ...	25·7in.
Total power developed ...	132 s.h.p.
Hourly coal consumption per s.h.p. ...	1·06lb.
Hourly steam consumption per s.h.p. ...	8·1lb.
Overall thermal efficiency of engines and boiler	18·6%

—*"Journal de la Marine Marchande", Vol. 23, No. 1,143, 6th November, 1941, p. 871.*

Temperature-indicating Paints.

Shortly before the war a firm in Germany developed a series of paints which changed colour when exposed to specific temperatures and which were used for the purpose of determining temperature differences in the cylinders and pistons of aircraft engines (*see abstract on p. 191 of* TRANSACTIONS *for September, 1939*). British paints of this kind are now available under the trade name of Thermindex temperature-indicating colours. The use of these paints is, of course, not confined to the example cited above. Apart from a large number of industrial applications, these products should be useful in such mechanisms as air-cooled air compressors, where they might enable the designer to dispose his ducts to the best advantage, and conversely, with electrical and other heating apparatus, how best to utilise the available radiation. Thermindex colours are produced in the form of a paint suitable for direct application, and the makers' full range comprises 16 original colours and 40 temperatures, the latter ranging from 80° C. (176° F.) to 800° C. (1,472° F.).—*"Engineering", Vol. 152, No. 3,958, 21st November, 1941, p. 416.*

Symposium on Electric Drive Held in New York.

At a symposium on electric drive for ships, held in New York on the 4th December, under the auspices of two prominent technical societies, there were altogether five speakers. A. Kennedy, Jr., read a paper on "Electric Propulsion Principles for Turbine Electric Drive", in the course of which he suggested that simplicity and reliability in a marine installation were even more important than efficiency. Among the several advantages which he claimed for the turbo-electric drive is the elimination of astern turbines and the possibility of obtaining auxiliary power from the main generator, thereby dispensing with the need for large auxiliary sets for special port services. J. S. Newton read a paper on "Steam Turbine Principles for Turbine Electric Drive", in which he remarked that the turbo-electric drive turbine is, fundamentally, a variable-speed unit, control of the speed of the ship being obtained by varying the speed of the main turbo-generator set. Because of this variable speed operating feature, the glands must be of the carbon type which will seal against leakage of steam at all speeds, instead of the water glands generally used in power-producing turbines ashore. Marine turbines for electric ship propulsion are usually lighter than land machines of comparable output, slightly lower in efficiency and somewhat higher in cost, mainly because of the variable speed feature. H. C. Coleman's paper dealt with "Electric Propulsion Principles for Diesel Electric Drive", and he pointed out that this system is non-mechanical because the engine transmits its power through the air gaps of the generator and motor. Although the d.c. system has been generally used because of the operating advantages it offers and its innate simplicity, recent increases in the running speeds of Diesel engines have necessitated the development of special speed-control arrangements. During the past 20 years over 200 American ships have been equipped with this type of drive, in addition to which about 140 further installations are in hand. The author also referred to the employment of a.c. for electric propulsion, and mentioned the submarine tender with a.c. propulsion motors of 11,800 s.h.p. and eight Diesel-generators at present under construction for the U.S. Navy. H. C. Lenfest's paper on "Diesel Engine Principles for Diesel Electric Drive" dealt with various features of Diesel engine design, whilst L. M. Goldsmith read a paper entitled "Why Electric Drive", in which he stressed the advantages of this form of propulsion and criticised the design of governors and oil coolers. He emphasised the value of electric drive for tugs.—*"Marine Journal", Vol. 68, No. 12, December, 1941, pp. 14, 15 and 29.*

"Electric Drive for Ships"—Some Criticisms.

According to some of the speakers at the recent symposium on the subject of electric propulsion, this form of drive, whether turbo-electric or Diesel electric, possesses many advantages over other types of drive, as proof of which the number of electrically-driven ships now under construction was cited. In actual fact, however, the reason why electric drive has been adopted in the present emergency is that the gear industry has not been able to produce reduction gears in sufficient numbers. One speaker stated that the 2 per cent. loss in reduction gearing necessitated cumbersome oil coolers, but omitted to mention that the 6 to 8 per cent. loss in electric transmission, with large fans hidden in the generator and a separate, water-cooled air cooler, more than offset this. Moreover, the propulsion motor also requires cooling and a separate motor-driven fan is often used for passing the air through the cooler, the latter again being water-cooled. In a 5,000-h.p. electric-drive ship the total surface of the two air coolers is 4,600ft.^{2}, in addition to which an oil cooler of 59 gall./min. capacity must also be provided in the case of a turbo-electric drive. The oil cooler required for a 6,000-h.p. geared turbine ship must be able to deal with 220 gall./min. of lubricating oil, but no air coolers are necessary. It is hard to see how the electric-drive ship has any advantage over the geared turbine vessel in this respect. Another speaker emphasised the simplicity and reliability of electric propelling equipment. As regards simplicity, the facts just related speak for themselves, and to this should be added the complicated governing system used in conjunction with the variable-speed turbines employed in turbo-electric marine installations. A statement made by one of the speakers concerning the low fuel consumption of electric-drive ships as compared with geared turbine ships, calls for analysis and amplification. Carefully conducted tests of eight C-2 ships showed an average fuel consumption for all purposes of 0·5455lb./s.h.p.-hr., while one vessel, the "Santa Rita", has a record fuel rate of 0·5280lb./s.h.p.-hr. of oil fuel of 18,500 B.Th.U. calorific value. The geared turbines of these vessels are designed for steam at 440lb./in.^{2} and 740° F. total temperature at their rated output of 6,000 s.h.p. As against this the 5,000-h.p. turbo-electric tanker "J. W. Van Dyke" has a fuel consumption for all purposes corresponding to 0·6121b./s.h.p.-hr. of similar fuel, although the turbines of this vessel were manufactured by the same firm as those of the C-2 ships and her H.P. turbine takes steam at the higher pressure of 600lb./in.^{2} with a total temperature of 825° F. It is believed that the auxiliary requirements of the C-2 ships and of the "J. W. Van Dyke" are relatively and approximately the same, so that a comparison of the fuel rates is quite instructive. The turbo-electric ship, in spite of about 35 per cent. higher steam pressure and 85° F. higher steam temperature, has a fuel consumption about 12 per cent. higher than geared turbine vessels of slightly greater power. Relative machinery weights were also discussed, the implication being that electric propulsion equipment is lighter than geared turbine machinery. In comparing the two systems of drive, it seems relevant to include only the propelling machinery, *i.e.*, the turbo-generator, motor, wiring and switchboard equipment in the one case, and the turbines and reduction gear in the other. On adding these separate weights together, it will be found that the 5,000-h.p. turbo-electric installation of the "J. W. Van Dyke" weighs 211,700lb., or 42·3lb./s.h.p., whereas the corresponding weight of a 6,000-h.p. geared turbine installation, including the main thrust bearing, is about 194,000lb., or 32·3lb./s.h.p. So much for the 40 to 50 per cent. greater weight of geared turbine machinery claimed in L. M. Goldsmith's paper! It was also asserted that gears get out of alignment, causing serious wear aand increased reduction in efficiency. With properly designed foundations and gear cases, this does not occur, and considering the great number of geared turbines which have been in constant operation for many years, instances of gear wear are almost negligible. In the very few cases where wear has taken place, it has nearly always been due to the oil used, so that change-over to a better grade of lubricating oil was all that was required to eliminate further wear. As to loss of efficiency after the gears have been in service for some time, it suffices to say that there are no cases on record where such a loss of efficiency has occurred.—*A. Peterson, "Marine Journal", Vol. 68, No. 12, December, 1941, p. 30.*

New Tube-ice Machine.

A new method of producing hard, clear ice at once in small pieces of almost any desired size, has been evolved in the form of a tube-ice machine described in a paper recently presented by B. F. Kubaugh, at a meeting of the American Society of Mechanical Engineers, in Louisville, Ky. The new process is claimed to combine the advantages of both the plate and the can ice manufacturing methods. The tube-ice machine consists of a number of tubes of any diameter or length set on end vertically and enclosed in a shell, in the same manner as an ordinary vertical tubular open-ended condenser. This vessel is suitably supported over an inclined chute, consisting partly of solid metal and partly of screen or perforated metal. Below this chute is a water tank from which the water to be frozen is withdrawn by a pump and discharged into the top of the vessel, where it is distributed evenly to the several tubes. Each tube is fitted with a metal distributor which causes the water passing through its orifice to be projected against the inner surface of the tube, to which it clings in its descent. The circulation of the water provides the agitation which is required to produce clear ice. Below the vessel and above the chute is a rotating cutter, which not only nicks and breaks the ice as it descends, but also measures the length of the protruding pieces. Just below the lower tube head is a shear plate, attached to the tube plate by a ring, which forms a closed chamber. The latter serves as an auxiliary thawing chamber through which a very small quantity of warm water, at 85° to 90° F., is discharged at the desired moment to thaw the lower ends of the tubes, which protrude a short distance below the tube plate and are extended to and through the shear plate by means of copper ferrules to form the thawing chamber. The vessel itself is filled with the refrigerant—usually ammonia—to the required height, determined by a float-controlled liquid feed valve. Near the top of the vessel is a horizontal transfer drum into which the refrigerant from the freezer is periodically discharged and from which it is expelled and returned to the freezer. An accumulator or liquid trap is attached to the transfer drum. At the beginning of the freezing operation a timing device, which controls all the functions, is started. On single-vessel installations, the timing gear also controls the water pump, starting and stopping it periodically, but in the case of multiple-vessel installations, the water pump runs continuously, and the timing device merely controls the admission of the water alternately to the two or more vessels which comprise the unit. With the water in circulation, the timer opens the suction valve and the liquid feed, and the freezing process proceeds for a period predetermined for the suction or evaporating pressure employed, and for the thickness or kind of ice desired. At the termination of the freezing period, the timer shuts off the suction gas, the liquid feed and the water flow, and then admits high-pressure evacuating gas from the condenser, which displaces or removes the liquid refrigerant from the freezer into the transfer drum, and, at the same time, thaws the ice from the tubes. A few seconds after the beginning of the evacuation, the timing device starts the rotating cutter and admits warm water to the thawing chamber. Simultaneously with the removal of the liquid, the ice in the tubes drops by gravity on to the rotating cutter and is delivered to the chute in pieces of the desired size. All these operations take place in a matter of seconds; the whole process from the moment when the suction valve is closed to that when the last piece of ice has dropped into the chute takes barely a couple of minutes, regardless of the size of the freezing vessel. The discharged ice passes over the perforated portion of the chute into an ice storage tank. The general arrangement of a tube-ice machine of this kind is shown diagrammatically in Fig. 1, whilst that of a multiple-vessel unit is shown in Fig. 2, in which the shell-to-shell evacuating-system can be seen. The thickness of the ice made varies from a very thin shell to a solid cylinder, according to the duration of the freezing period, which is regulated by a simple speed control. The freezing period varies from 10 min. for crushed shell ice at low evaporating pressure (around 15lb./in.^{2}) to about 37 min. for solid ice at high evaporating pressure (about 30lb./in.^{2}). Where crushed ice is desired, the ice cylinders are relatively thin—about $\frac{1}{2}$-in.—and short, whereas solid ice is produced in the form of cylinders of about 1$\frac{3}{8}$in. diameter by 2$\frac{1}{2}$in. long. Other lengths are obtainable by merely fixing the measuring plate of the cutter at the

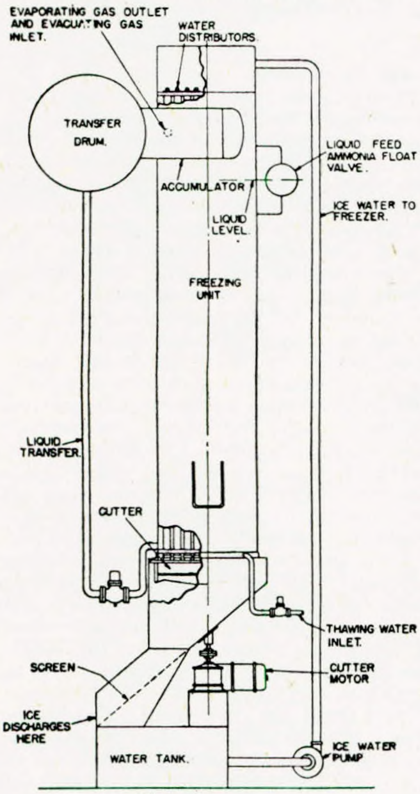


FIG. 1.

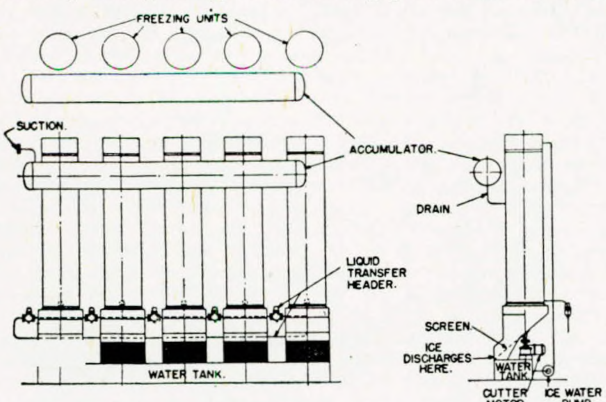


FIG. 2.

required distance. Single freezer units have capacities varying from a fraction of 1 ton to 29 tons in "solid ice", and to 40 tons in crushed ice. A 10-ton single-vessel unit occupies a space of about 6ft.×7ft.; a 40-ton single-vessel unit, 8ft.×9ft.; a 65-ton two-vessel unit, 8ft.×12ft.; and a 200-ton five-vessel unit, 9ft.×30ft. or 18ft.×20ft. The tube-ice machine may be attached to an existing refrigerating plant, or it may be installed independently with its own "high side", compressor, condenser, receiver, etc. The operation of the machine is fully automatic and only requires occasional supervision, as nothing leaves the apparatus but the ice to be used. All small particles which chip off in the cutting process return to the pumping tank, contributing refrigerating effect to the circulating water. The timing mechanism consists of several cams of plastic material mounted on a horizontal shaft driven by a very small motor. Tube ice has many uses, and the machine may serve the dual purpose of cooling water only during a part of the day and freezing ice during the remainder by merely setting the suction valve to produce the required evaporating temperature for either purpose, or arranging two valves in parallel lines, each set for its respective pressure with the necessary stop valves for bringing each into service.—*"Modern Refrigeration", Vol. XLV, No. 526, January, 1942, pp. 9-10.*

Welding Ships' Funnels.

Although less widely used than metallic arc welding in the fabrication of ship assemblies, resistance welding has proved adaptable for the rapid and economical fabrication of many ship parts. At the Philadelphia Navy Yard resistance welding is employed for fabricating the funnels of naval ships. These funnels are of cylindrical or elliptical shape, from 8ft. to 14ft. in diameter, and from 25ft. to 40ft. in height. They are built up of galvanized or stainless steel plating, from 0·032in. to 0·096in. thick, stiffened by flat bars, angles or channels running circumferentially and longitudinally in the interior of the funnel. The former method of assembly used at the yard was by riveting, but welding has proved to be quicker and more economical. The resistance-welding equipment required consists of a pedestal type press welder of a capacity depending upon the total thickness of the material to be welded, a small 150-kVA capacity seam welder being used to seam or intermittently spot weld small-gauge plate assemblies. Several push guns are of value in making welds that are inaccessible with the pedestal type equipment and also for use aboard ship. The main difficulties to be overcome in the fabrication of large funnels by spot welding are the handling of the sub-assemblies prior to final welding and the selection of a proper welding sequence to avoid distortion in the completed funnel. The erection sequence of a large funnel involves special care owing to the employment of two separate welding methods during assembly. The primary strength joints having been assembled by spot welding, the structure is then rigid, much as a riveted structure would be, and when the abutting edges of the outer plates are arc welded, considerable distortion is liable to take place. Care must be taken to allow for this contraction across the arc-welded seams by leaving adequate gaps or by arranging to heat the plates while assembling the structure.—*H. O. Klinke, "Canadian Shipping and Marine Engineering News", Vol. 13, No. 5, December, 1941, pp. 19-20.*

Non-reversible Engines for Motor Trawlers.

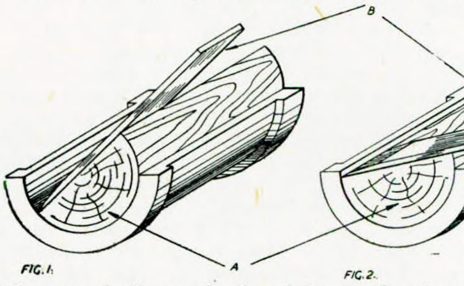
It is considered probable that Diesel engine propulsion will become increasingly popular for trawlers after the war, and it has been suggested that even when machinery of 800-1,000-h.p. is in question—and this power will be required for the larger types—Diesel engines with reversing gears may be employed. It has hitherto generally been considered desirable to use direct-reversing engines when the power is over 500 b.h.p., but this view will not necessarily be adopted after the war (*see abstract on p. 161 of* TRANSACTIONS *of January, 1942*). The makers of the Ruston marine Diesel engine have now pointed out that Ruston engines of up to 840 b.h.p., running at 400 r.p.m. and giving a propeller speed of 133 r.p.m. through oil-operated reverse reduction gears, are already in service. The idea that the direct-reversing system is preferable for powers above 500 b.h.p. was, it is stated, abandoned some time ago.—*"The Motorship", Vol. XXII, No. 263, December, 1941, p. 275.*

The Machinery of Channel Packets.

The fourteenth Thomas Lowe Gray Lecture, which was devoted to the above subject, was delivered on 23rd January by Major Wm. Gregson, M.Sc., at a meeting of the Institution of Mechanical Engineers. The first part of the lecture dealt with the historical and technical development of steam-driven cross-channel packets, whilst the second part was devoted to a consideration of three main categories for the classification of modern propelling-machinery installations for such vessels; *viz.* (*a*) geared turbines with oil-fired boilers; (*b*) geared turbines with coal-fired watertube boilers and mechanical stokers; and (*c*) direct Diesel drive. The lecturer suggested that future developments might include the employment of the turbo-electric drive and the evolution of what he termed a "Pullman sea coach" with superlatively light machinery and a sea speed of something like 40 knots.—*"Engineering", Vol. 153, No. 3,968, 30th January, 1942, p. 92.*

Scraping Bearings.

The use of a round-nosed scraper for scraping half-round bearings has the disadvantage of cutting a series of furrows axially along the bearing, in addition to which a cut cannot be taken from one end to the other with any degree of uniformity. A half-round scraper also leaves the job far from perfect because it is not a tool to which a great deal of pressure can be applied owing to the manner in which it has to be handled. Furthermore, it is practically impossible to apply the operating force at a constant angle to the surface which is being scraped, neither can the cutting angle be maintained constant to the axis of the cylindrical bore of the bearings. A useful method of scraping small and medium-sized bearings is illustrated in Figs. 1 and 2, which show a piece of wood (*A*),



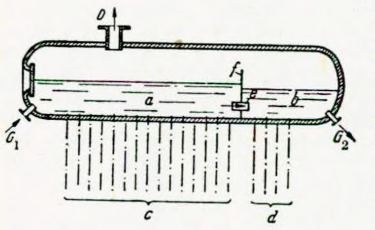
slightly over half-round, placed in the bearing while the latter is being held firmly in a vice. A flat scraper (*B*) rests with its cutting edge against the surface of the bearing, and when the handle of this scraper is lowered, the wooden rest turns in the bearing. As the flat side of the scraper (*B*) is pressed against that of the block of wood (*A*), it becomes possible to apply a considerable amount of pressure to the cutting edge of the tool and to take long, wide and useful cuts.—*T. Harris, "Practical Engineering", Vol. 5, No. 106, 29th January, 1942, p. 41.*

New Variable-delivery Pump.

A well-known New Jersey firm of pump manufacturers are now producing an improved variable-delivery high-pressure pump which, they claim, can deliver from 0 to 6 gall. (U.S.) per minute at a pressure of 5,000lb./in.^{2}. This is accomplished, while the pump is running, by altering the angle of a driving member trunnioned on the driving shaft to produce a corresponding stepless change in the stroke of the pump plunger. The stroke can be varied from zero to 4in. The stroke-control shaft projects outside the pump housing and is arranged for operation by either manual or automatic pressure control. The pump is stated to be particularly suitable for use with hydraulic machinery or appliances in which a sudden advance must be followed by a slow movement at high pressure, and for boiler feed purposes where it is necessary to deal with sudden changes of output which may affect the water level in the boiler. The pump is driven by a 25-h.p. motor, which is so mounted that the entire unit is only 4ft. high and takes up a floor space of only 4ft. by 4ft. Similar pumps for designed pressures ranging from a few hundred up to 10,000lb./in.^{2} can be supplied by the makers.—*"Marine Engineering and Shipping Review", Vol. XLVI, No. 12, December, 1941, p. 59.*

Reduction of Salt Content of Steam.

An article in *Zeitschrift des V.D.I.*, reproduced in a recent issue of *Combustion*, concerns a device for reducing the salt content of steam. The accompanying illustration shows diagrammatically the division of the boiler water circulation into two independent circuits, one being fed by overflow from the other in order to reduce the salt content of the steam by the two-stage evaporation thus effected. If more than two circuits are arranged in this manner, there is an increase in concentration from stage to stage and the blow-off water is taken from the last stage. The steam spaces of the several stages may be interconnected, steam being taken from the first stage where the risk of contamination is least. Referring to the arrangement shown, the first- and second-stage water spaces (*a, b*) served by the steam-generating tubes (*c, d*), are separated by a wall (*f*) with an overflow or transfer opening (*e*). Steam leaves the boiler at *D*, feed water is admitted at *b*, and a blow-down connection is fitted at *G*. It is claimed that with this arrangement 3 per cent. blow-down, as compared with 10 per cent. for normal construction, is



sufficient to keep the solids content of the steam down to 22 grains/cu. ft. with feed water containing 7 grains/gall. of solids and with 5 per cent. of moisture in the steam. Water-level indicators must be provided for each stage and kept under observation, and tests show that a definite minimum alkalinity must be maintained in the first stage.—*"The Power and Works Engineer", Vol. XXXVII, No. 428, February, 1942, p. 70.*

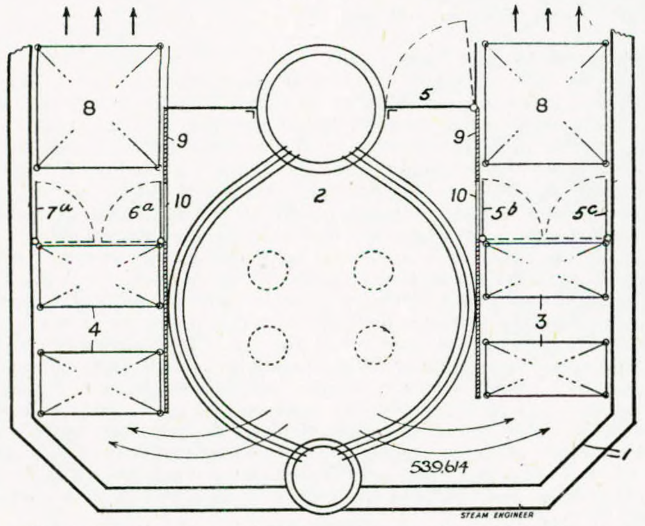
Marine Steam Engineering—The Next Phase.

Among the improvements that may be expected to take place in the design of marine propelling machinery after the war is an extension of the employment of superheat in conjunction with still higher steam pressures, although it is unlikely that the latter will exceed 1,000lb./in.^{2} during the next few years, so that there will be plenty of scope left for even higher pressures if such an advance should prove to be justifiable by actual practice. The employment of higher steam pressures will, of course, involve other problems and solutions, and, incidentally, will bring the uptake economiser into the picture once more. Lower pressures in marine boilers did not favour the economiser, but land power-station practice made increasing use of the economiser as pressures rose until it became as much an essential part of the land boiler as the superheater and air-heater, taking its rightful place as an efficient feed-water heater and leaving the boiler proper to perform its correct function of merely supplying the latent heat necessary for converting the fully-heated water into steam in readiness for passage to the superheater. Thus each part of the steam-generating plant fulfils the function for which it is best adapted without encroaching on the rightful duty of the other parts. In fact, water-heater, transformer, and steam-heater, might not unhappily replace the names of these three components hitherto, and very misleadingly known as economiser, boiler and superheater. It will be a new experience for marine engineers to handle the modern economiser, which will have to be proportioned to work in conjunction with the air-heater, since both obtain their heat from the exhaust gases of the boiler. The output of hot air for the furnaces from the air-heater must be regulated to suit furnace conditions varying from the old-fashioned firegrate to oil burners, *i.e.*, from about 250° to 400° F. The higher exhaust-gas temperatures supply heat to the economiser tubes, raising the temperature of the feed water by "sensible heat" up to approximately that of the saturated steam corresponding to that in the boiler which then proceeds to evaporate steam at the same temperature, while the lower ranges of flue-gas temperatures are more suitable for heating furnace air. The regulation of these separate functions of the air-heater, economiser, steam generator (boiler) and superheater, will involve close attention on the part of the boiler attendant to suit the varying conditions of manœuvring and working demand. In the utilisation of higher steam pressures for turbines and engines it must be realised that the engine is not in effect a prime mover, as it merely converts the potential energy of the steam produced by the generator into the kinetic energy of motion. Parsons always emphasised that his turbine was essentially and merely a pump on the same principles as the water turbine, although specially adapted to utilise the immensely higher velocities due to the change from water to steam pressure. Higher pressure therefore means greater power transmitted to the turbine blades, or the engine pistons, as the case may be; and the higher pressures envisaged will of course result in a greater output of power per pound weight of steam or, alternatively, smaller turbines and engines of equal power. This is not the least of the incidental advantages to be derived from the use of higher pressures, to say nothing of the all-round diminution in the size of pipes and steam connections. The specific volume of steam at 800lb./in.^{2} as compared, *e.g.*, with 400lb./in.^{2}, is as 0·626 to 1·206, practically half; and therefore the internal areas of pipes, steam connections and pressure joints can be correspondingly reduced in size. This simplifies the problem of nut-and-bolt fastenings, which may naturally cause some concern to designers, although this difficulty has been foreseen in the research work recently carried out on "creep". The effect of creep due to stress under prolonged tension and heat on steel is now pretty well understood, and a slight increase in the diameter of bolts reduces the stretch to infinitesimal proportions. Moreover, it has a beneficial effect in a ring of bolts where it is impossible to tighten them all up by the same amount manually, so that if one bolt should be over-stressed to start with it will gradually stretch and transmit its excess load to the adjacent bolts until all bear equally. The design of joints, stuffing boxes and glands will exercise the ingenuity of draughtsmen, but these fittings have been vastly improved in recent years and should present no difficulty. There will be plenty of scope for new and stronger designs of various other accessories, such as boiler mountings; and here also many years' experience with hydraulic connections, to say nothing of heavy-oil I.C. engines in which higher pressures and temperatures are encountered than for the augmented steam pressures contemplated, will simplify the designer's task. In the light of the foregoing developments, it may be assumed that the boiler itself is likely to undergo a still further reduction in size for the power output required. This was a rather unexpected result of the last great increase in pressure, and paradoxically, the final result was to build much bigger units and fewer of them. The number of boiler units was frequently halved, and the same effect will probably be experienced again. The water-tube boiler is capable of almost unlimited expansion in size and power output, so that we may soon expect to see two boilers where six or eight would have been installed a decade ago for the same total horsepower. This again will have a profound effect on general ship design, for even a complete boiler room may be eliminated as redundant, and, indeed, we may see the engine and boiler rooms combined in one compartment for relatively high powers. All this follows from an advance in one direction only, the doubling of existing boiler pressures. A marked reduction in fuel consumption will also be realised, and much of the space now required for the carriage of fuel will become available for other purposes. Having regard to the progress made in this direction in the past 15 years, it may confidently be anticipated that a fuel consumption of 0·5lb. of oil fuel or 0·75lb. of coal per s.h.p.-hr. will come to be considered as reasonable, anything more being looked upon as extravagant.—*J. Hamilton Gibson, O.B.E., Wh.Ex., M.Eng., "The Journal of Commerce" (Shipbuilding and Engineering Edition), No. 35,562, 29th January, 1942, pp. 1-2.*

Improved Superheating and Steam Reheating Arrangements for Boilers.

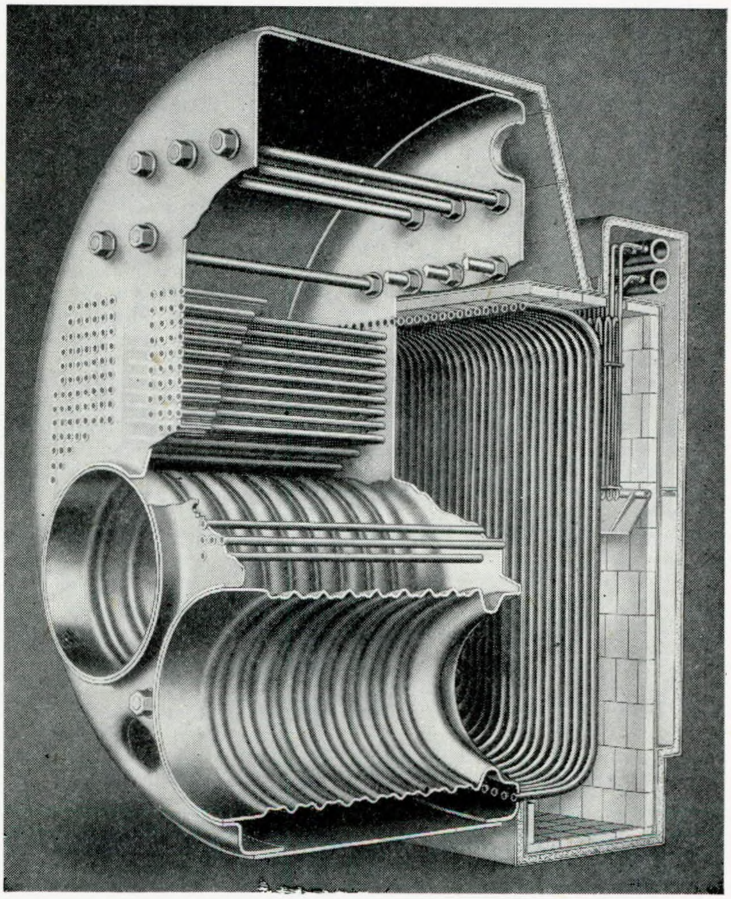
A recent British patent granted to Mr. J. Johnson covers an invention relating to installations comprising double-flow watertube boilers with a single furnace or combustion chamber where, as in the case of marine boiler installations, there is a liability of overheating the superheater or re-superheater elements during steam raising, at intermediate powers, or when running astern. Fig. 1

FIG. 1.



is a diagrammatic view of one arrangement suitable for use on board ship, with a two-drum double-flow type boiler having a partially air-jacketed casing (*1*) which encloses the boiler (*2*), a superheater (*3*), and a reheater (*4*). The superheater and reheater are each surmounted by an economiser (*8*) within the casing (*1*). Partitions (*9*) of suitable steel and brick construction with an opening (*10*), extend to some distance short of the bottom of the casing (*1*), separate the superheater and reheater from the boiler and protect these components from the effects of the combustion gases when by-passed by the latter. A damper (*5*) is associated with a pair of dampers (*5b, 5c*) arranged above the superheater (*3*) behind and below the opening (*10*) in the adjacent partition (*9*). Similar dampers (*6a, 7a*) are arranged above the reheater (*4*) and behind and below the opening (*10*) in the adjacent partition (*9*). When raising steam, the damper (*5*) is turned up to its vertical position, whilst the dampers (*5b*), (*5c*), (*6a*), (*7a*) are held in the horizontal position, both the superheater and the reheater as well as the economisers being thus by-passed. When the engines are manœuvring or going astern, the dampers (*5*), (*6a*) *and* (*7a*) are set horizontal, whilst the dampers (*5b*) and (*5c*) are set to give the required superheat temperature, the reheater still being by-passed, with the opening (*10*) above the reheater free for the escape of the gases through the associated economiser (*8*). Normally, when steaming ahead, the damper (*5*) will be horizontal, the dampers (*5b*) and (*6a*) will be vertical, and the dampers (*5c*) and (*7a*) set to give the required superheat and reheat temperatures, the gases flowing in controlled quantities beneath the partitions (*9*).—*"The Steam Engineer", Vol. 11, No. 125, February, 1942, p. 137.*

Superheater for Howden-Johnson Boiler.



The accompanying illustration shows the arrangement of the superheater incorporated in the latest type of Howden-Johnson accelerated-circulation boiler. It is claimed that by placing the superheater elements between the water tubes and the dry back of the boiler, they are afforded better protection and maintained in a clean condition longer than might otherwise be the case, in addition to which they can easily be inspected or removed from the outside of the boiler. The superheater headers are located away from the air space common to each side of the boiler, thereby eliminating the possibility of air leaks into the combustion chamber. This superheater arrangement enables steam temperatures of up to 800° F. to be made possible with the medium pressures (up to 300lb./in.²) used in this type of boiler.—*"The Marine Engineer", Vol. 65, No. 775, February, 1942, pp. 33-35.*

Cost of High-speed Ships.

A good example of the extra capital expenditure involved when high speeds are desired was recently given by a ship-owner. The type of ship which he was accustomed to order was a good-class cargo carrier of about 9,000 tons d.w. In 1910 he paid £65,000 for a vessel of this description, the machinery being steam triple-expansion, and the speed, which was higher than was usual at that time, about 11 knots on trial. In recent years he turned to Diesel propulsion, and just before the war he took delivery of a ship of about the same size, having a slightly higher speed. The cost was £130,000, or exactly double that of the old ship. The construction of a 16-knot vessel of the same cargo capacity under present-day conditions would, he estimated, cost something like £300,000, owing to the greater dimensions which the hull would have to possess on account of its much finer form and the increased machinery weight. —*"Fairplay", Vol. CLVIII, No. 3,064, 29th January, 1942, p. 174.*

Propelling Machinery of New American Tankers.

It is reported that the turbo-electrical propelling machinery for the 35 oil tankers to be built for the U.S. Maritime Commission by the Sun Shipbuilding and Dry Dock Company at Chester, Pa., is being manufactured at the South Philadelphia and East Pittsburgh works of the Westinghouse Company. The propelling machinery of these tankers will be of the direct-drive type with a high-speed turbo-generator supplying current to a 6,600-h.p. electric motor running at about 93 r.p.m., which will drive the propeller. The direct-drive system has been adopted because machinery of this type can be produced more quickly than that which includes reduction gears. The new tankers will each carry approximately 140,000 barrels (about 18,900 tons) of oil.—*"Mechanical Engineering", Vol. 64, No. 1, January, 1942, p. 20.*

Corrosion in Feed Systems.

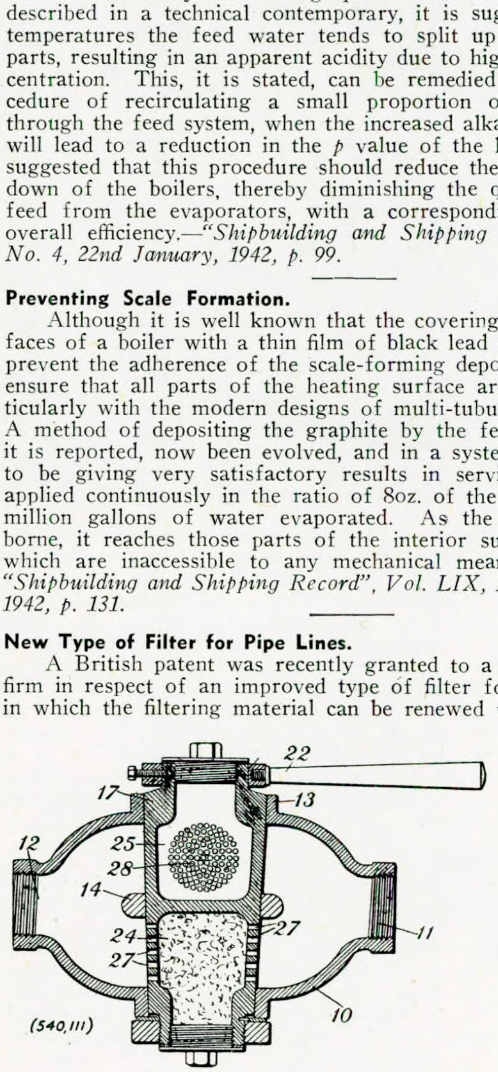
The use of high-pressure steam on board ship has necessitated the adoption of accurate methods for controlling the amount of impurities in feed water in order to prevent the formation of scale and the possibility of corrosion in the boilers, but it is not generally realised that the feed water itself may be responsible for corrosion in the feed pumps and pipe lines betwen the pumps and the boilers, on account of the high pressure and temperature at which it is maintained. In an investigation into the causes of excessive corrosion in the feed systems of high-pressure boilers which was recently described in a technical contemporary, it is suggested that at high temperatures the feed water tends to split up into its constituent parts, resulting in an apparent acidity due to high hydrogen ion concentration. This, it is stated, can be remedied by the simple procedure of recirculating a small proportion of the boiler water through the feed system, when the increased alkalinity of the former will lead to a reduction in the *p* value of the latter. It is further suggested that this procedure should reduce the amount of blowing down of the boilers, thereby diminishing the quantity of make-up feed from the evaporators, with a corresponding improvement in overall efficiency.—*"Shipbuilding and Shipping Record", Vol LIX, No. 4, 22nd January, 1942, p. 99.*

Preventing Scale Formation.

Although it is well known that the covering of the heating surfaces of a boiler with a thin film of black lead or graphite tends to prevent the adherence of the scale-forming deposits, it is difficult to ensure that all parts of the heating surface are so protected, particularly with the modern designs of multi-tubular steam generator. A method of depositing the graphite by the feed water itself has, it is reported, now been evolved, and in a system which is claimed to be giving very satisfactory results in service, the graphite is applied continuously in the ratio of 8oz. of the compound to every million gallons of water evaporated. As the graphite is water-borne, it reaches those parts of the interior surface of the boiler which are inaccessible to any mechanical means of application.—*"Shipbuilding and Shipping Record", Vol. LIX, No. 5, 29th January, 1942, p. 131.*

New Type of Filter for Pipe Lines.

A British patent was recently granted to a London engineering firm in respect of an improved type of filter for use in pipe lines, in which the filtering material can be renewed without interrupting the flow. Referring to the accompanying sectional drawing, it may be seen that the device comprises a casing *(10)* of generally elliptical shape, having an inlet *(11)* and an outlet *(12)* at opposite ends, by which it can be connected to the pipe line. The casing *(10)* has a central cylindrical portion *(13)*, the axis of which is perpendicular to the centre line of the inlet and outlet, and this cylindrical portion is divided by a transverse partition *(14)* so that two passages, each of a cross-sectional area equal to that of the pipe line, are presented between the inlet and outlet. A slightly tapered plug *(17)* fits in the cylindrical portion *(13)* and carries a handle *(22)* for turning it through 90°. The plug *(17)* is recessed at each end to form two chambers *(24, 25)* separated by a partition. The wall of the lower chamber *(24)* is perforated at opposite ends to allow a flow through it when the perforations *(27)* are in line with the pipe. The wall of the upper chamber *(25)* is similarly perforated, but the perforations *(28)* are at right angles to those in the lower chamber. With the plug *(17)* in one position, the flow is through one chamber, while, when the plug is rotated through 90°, the flow is through the other chamber. The two chambers *(24, 25)* are filled with fibrous material such as cotton, loosely packed, and the chambers are closed by screwed caps which can readily be removed to give access to the filtering material. It is thus possible to renew the filtering material in either chamber by diverting the flow through the other.—*"Engineering", Vol. 153, No. 3,971, 20th February, 1942, p. 160.*



Insulation by Frozen Lard.

It is reported that a large shipment of frozen meat recently arrived in this country from the United States in a non-refrigerated ship. According to the American Meat Institute, the method used was to freeze the meat, box and all, to an extremely low temperature before stowing it in the ship's holds, the temperature of which was reduced by the effect of the frozen cargo. Instead of the usual cork or slag wool insulation, the internal surfaces of the vessel's holds were insulated with boxes of frozen lard, and the compartments were sealed by placing more of the frozen lard on the top of the cargo. No refrigerating machinery was used in the ship, and the temperature in the holds must have risen steadily during the voyage, but in spite of this the meat, which is described as mild-cured bacon, arrived in good condition. As an adjunct to the insulated-cargo tonnage available, this method of transporting certain varieties of frozen meat products across the Atlantic is valuable under the present circumstances, although it is unlikely to be utilised to any extent in normal times.—*"Fairplay", Vol. CLVIII, No. 3,063, 22nd January, 1942, pp. 146 and 148.*

Marking-off by Spot Light.

In the marking-off of holes on plates or castings which have to be drilled, it is sometimes difficult to ensure that the markings should be clearly visible when the job is in the machine, and it often involves loss of time to ensure that the drill shall descend accurately on the desired spot. To surmount this difficulty, a British firm of lighting specialists have devised a spot-light attachment which can be fitted to a machine in such a manner that an intense beam of light is directed on the spot where the tool will make contact with the job. Actually, as fitted to a spot-welding machine, the current for the lamp is obtained through a step-down transformer, the lamp with its lense system being carried in a metal housing secured in such a manner that the brilliant spot of light marks the point where the upper electrode makes contact on the job. A development of the device using two spots of light enables a considerable increase in output to be made where a number of spot welds in line have to be made. In this case it is only necessary to mark out the line on which the welds are to be made and to set the two spot lights at the desired pitch when, by placing the job so that one spot of light falls on the weld just made the other indicates the position for the next.—*"Shipbuilding and Shipping Record", Vol. LIX, No. 5, 29th January, 1942, p. 131.*

Carbon in Diesel Engines.

A saving in fuel and maintenance costs of at least 10 per cent. is said to be obtainable by adding a carbon-inhibiting preparation known as "Dieslip" to the fuel and lubricating oils of Diesel engines. The addition of 2 per cent. of this liquid to the lubricating oil is said to prevent the sticking of piston rings and the formation of sludge in the crankcase, whilst the addition of 5 gallons of Dieslip to every 1,000 gallons of fuel oil is claimed to improve the working of the fuel pumps, thus reducing wear and preventing dribble and direct wastage of fuel. Among other advantages claimed for the preparation is its solvent action on gummy and carbonaceous matter resulting from incomplete combustion. This prevents the sticking of piston rings and valve stems, and also checks the formation of hard carbon on the piston and cylinder walls. Independent tests carried out on a 6-cylr. two-stroke engine rated at 165 b.h.p. at 500 r.p.m. showed that after 824 hours' running all the piston rings were free, whereas they had formerly begun to stick after only 200-250 hours' running. Carbon deposits on the fuel nozzles were much reduced, and such carbon as was deposited was of a softer nature than the extremely hard deposits previously encountered. After the addition of the inhibitor to the fuel and lubricating oils, the fuel consumption improved by 9 per cent., but this eventually fell away to normal, so that the average improvement throughout the run of 824 hours was 6½ per cent. The constant reduction in lubricating-oil consumption during the trial was 9½ per cent. The actual cash saving on fuel and lubricating oil during the test, after allowing for the cost of the inhibitor, was 45s., whilst the saving in labour charges for cleaning and maintaining the engine was estimated at two-thirds. —*"Lloyd's List and Shipping Gazette", No. 39,689, 28th January, 1942, p. 5.*

Ignition Quality of Diesel Fuel.

In the specification of fuel for Diesel engines, many qualities, both physical and chemical, call for determination, among which are specific gravity, viscosity, flash point, sulphur content, and so on. These, however, do not give an accurate indication of the suitability of the fuel for use in the cylinders of a Diesel engine, particularly as regards ease of starting and smoothness of running. With the development of the high-speed Diesel engines now being increasingly employed for the propulsion of small craft and for driving the electrical generators of larger ships, these properties become particularly important, and it is now being suggested that what has been termed the ignition quality of the fuel should be specified. Good ignition quality denotes easy starting, and the property of igniting regularly and rapidly so as to eliminate the possibility of combustion knock, but the difficulty is to find some property which can be easily assessed and which will command general acceptance as an indication of the ignition quality of the fuel. It is already recognised that oils having a paraffin base make good Diesel fuels.—*"Shipbuilding and Shipping Record", Vol. LIX, No. 5, 29th January, 1942, p. 131.*

Emergency Quick-starting Power Units.

Although oil-engined generators are often used as a stand-by, it is sometimes necessary to employ steam-driven dynamos for this purpose, and the English Electric Company recently built a 500-kW. emergency quick-starting turbo-alternator for a large power plant overseas where there is no interconnection with external sources of supply. It was essential to reduce to a minimum the period of any inadvertent shut-down, and this new emergency unit has therefore been designed for entirely automatic operation by remote control, and for running up to full speed in the shortest possible time. The turbine, of the impulse type, takes steam at 200lb./in.^{2} pressure and 750° F. total temperature, and runs at 6,000 r.p.m., driving the alternator, through reduction gearing, at a speed of 1,500 r.p.m. The alternator, of the salient-pole type, is designed for three-phase, 50-cycle supply, and has its exciter mounted overhung on the extended alternator shaft. A flexible coupling is fitted between the turbine and gear pinion, but the alternator rotor is solidly coupled to the slow-speed gear wheel, the weight of the rotor being shared between the main gear bearing and the alternator outboard bearing. The turbine being of the impulse type with only one velocity-compounded stage, large blade clearances are permissible without undue sacrifice of efficiency and, with all high steam pressures and temperatures confined to the nozzle belt before the main nozzles, the unit is particularly suited for quick starting without distortion of the turbine casing and consequent misalignment troubles. The turbine shaft glands, of the labyrinth type in which there are no rubbing parts, are fitted with a small steam-jet ejector which comes into operation automatically when the main steam valves are opened. The ejector withdraws the leakage steam from the glands and discharges it into the turbine exhaust trunk. Apart, therefore, from the closing of the drain valve once the turbine has reached its normal temperature and the routine inspection of bearing temperatures, etc., the unit requires no attention in service. The turbine is started by remote control from a special panel in the main control room, the small amount of auxiliary power required for this purpose being provided by storage batteries. In the unlikely event of even this supply being unavailable, the unit can still be started up by hand from the turbine platform with the aid of a steam-driven auxiliary oil pump provided to cover such a contingency. The arrangement of the control system is shown in Fig. 3, from which it may be seen that control of the various sequences of operations is effected by oil under pressure, supplied in normal circumstances, by a motor-driven oil pump on starting up, and maintained, when the unit is up to speed, by the main gear oil pump driven from the turbine shaft. The control panel carries, in addition to an electrical speed indicator actuated from a tachometer generator on the turbine, the starting switch for the motor-driven oil pump with three positions marked "Hand", "Off" and "Auto", and red and green signal lamps. On starting, the main steam and cooling-water isolating valves on the set being always open, the motor switch is turned to the "Hand" position. The pump starts up, the red signal lamp glows, and as soon as the oil pressure is high enough, the hydraulically-controlled emergency stop valve and the main throttle valve on the turbine begin to open, as does the oil-pressure-controlled water-inlet valve on the oil cooler. The turbine begins to turn as soon as the emergency stop and throttle valves leave their seats and full speed is reached within about 30 seconds from the time the motor-driven oil pump is started up. When full speed has been attained the turbine governor takes control and keeps the speed constant, but if necessary the speed of the set can be regulated by remote control of the governor speedometer, and in the same way the unit can be paralleled with any supply available later. When the turbine is running steadily under the control of the governor, the switch of the motor-driven oil pump is turned through the "Off" position to "Auto", when, providing the main gear oil pump driven from the turbine is functioning properly, the motor-driven pump shuts down, the red signal lamp is extinguished, and the green lamp lights up. The motor-driven oil pump is then in a position to start up automatically should the relay oil pressure fall below normal. Under ordinary running conditions the stand-by steam-driven oil pump is isolated from the main steam supply and therefore inoperative, but with hand control, it can be used for starting and when stopping the set should the motor-driven oil pump be out of service or fail

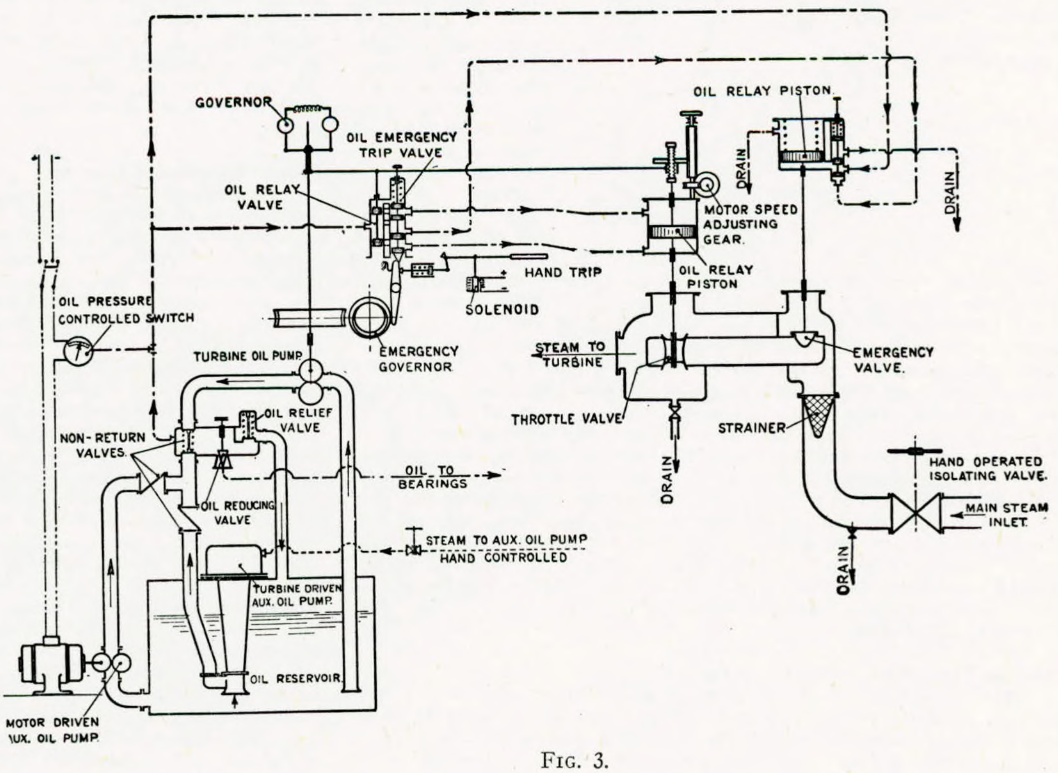


FIG. 3.

to function when required. For shutting down the set from the control panel, a "Stop" button is provided, by means of which a solenoid fitted to the turbine emergency trip valve is energised, thereby causing it to trip the governor and stop the set. The emergency trip gear can also be operated by hand from the turbine platform. Before despatch from the makers' works, the unit with its control equipment and pumps, etc., was erected and subjected to a series of starting-up and running tests under the designed steam conditions. From the moment of starting the motor-driven oil pump, about 15 sec. were required for the pump to raise enough pressure to open the steam valves, and the unit was up to speed and under the control of the turbine governor within 36 seconds. Even under such onerous starting conditions, the balance and running of the set was very satisfactory.—*"Industrial Power", Vol. XVII, No. 195, December, 1941, pp. 153-155.*

A Magnetic Tank-level Indicator.

Among the various appliances made for indicating the amount of liquid in lubricating-oil, fuel-oil, water and other service tanks, is a magnetic tank-level indicator, the construction of which is shown in the accompanying sectional drawing (Fig. 2). The operation of the device is dependent on the movement of a float which rests on the surface of the liquid inside the tank. Any movement of this float is transferred to the arm shown in the illustration, and transmitted by suitable gearing to a cobalt permanent magnet. The latter is separated from a magnetic needle by the solid head of the indicator, and actuates this needle, which, in its turn, rotates over a graduated scale and thereby indicates the quantity of liquid in the tank. There is no connecting shaft or pipe between the interior and the exterior of the tank, as the indicator is fitted by cutting a hole in the shell of the tank to receive the float mechanism, the mounting ring being riveted and sweated in place. The float arm is then passed into the tank, and the head of the indicator secured by studs and nuts to the mounting ring, a jointing washer being used to seal the tank opening. It is claimed that this method of assembly eliminates any possibility of leakage of the liquid or vapour, whether the indicator is fitted to the top, side or base of a tank. Moreover, a positive reading is available at all times. An alternative form of this tank-level indicator has been developed by the makers for certain applications where the previous pattern would not give the required visibility. The method of actuating the magnetic needle and graduated scale is similar to that already described, but, in this case, the float acts through a coarse-threaded screw cut in the float tube. As the float and spindle rise, a projection engaging in the thread causes the float and spindle to rotate, thus turning the cobalt permanent magnet.—*"The Shipbuilder", Vol. XLIX, No. 391, February, 1942, p. 49.*

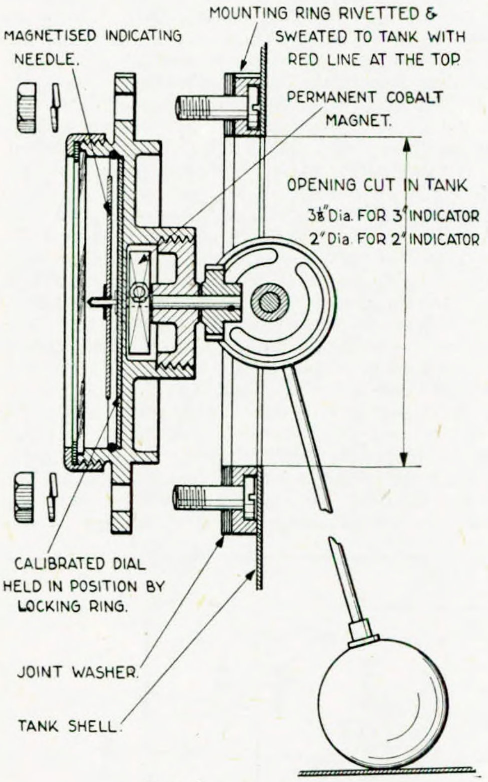


FIG. 2.

"Steam Tugs, Past, Present and Future".

The author traces the history of steam tugs from their introduction down to the present day, and outlines the gradual development of tug machinery and the improvements which have been effected in its design. The partial conversion and modernisation of an existing tug are described, and the author's proposals for the design of a modern steam tug are discussed. These include a cylindrical return-tube boiler of the Howden-Johnson dry-back type with shovel-type mechanical stokers, supplying saturated steam to a slow-speed totally-enclosed four-crank triple-expansion engine with forced lubrication throughout and automatic steam traps to all cylinder drains. The paper concludes with some notes on tug-boat propulsion and its possible improvement.—*Paper by G. T. Shoosmith, M.A., "Transactions of the Institute of Marine Engineers", Vol. LIII, No. 12, January, 1942, pp. 185-192.*

Direct-drive Diesel Tug.

Built originally as a steam tug, but recently reconstructed and converted to Diesel-engine propulsion in a Brooklyn shipyard, the "Barryton" is the largest commercial tug with direct Diesel drive under the American flag. She is 150ft. in length, with a beam of 37·7ft. and a depth of 14ft., the maximum load draught being 17ft. She is now equipped with an 8-cylr. Fairbanks-Morse Diesel engine rated at 1,400 h.p. at 300 r.p.m. Electric current for the auxiliary machinery and extensive pumping installation of the tug is supplied by two 60-kW. generators directly driven at 900 r.p.m. by two 6-cylr. 90-h.p. F.M. engines.—*"The Shipping World", Vol. CVI, No. 2,537, 28th January, 1942, p. 105.*

A Geared Diesel Grain Carrier.

A special type of vessel designed to carry either grain in bulk or oil was recently completed at Albany, N.Y., in a shipyard improvised by her owners, Cargill Inc., who built the ship themselves in order to avoid the delay in construction which would have resulted if she had been built by any of the American shipbuilding firms. The new ship, launched as the "Carlantic", but now named "Victoria", is operated by a Buenos Aires shipping firm on behalf of her U.S. owners as an ocean grain carrier, although she can be quickly converted for the transport of oil, if required. The vessel has a gross tonnage of about 7,000 tons and a d.w. capacity of 12,200 tons, her main dimensions being an o.a. length of 437ft., a beam of 60ft. and a draught of 28ft. 3in. The propelling machinery is located aft and consists of two 6-cylr. Fairbanks-Morse two-stroke engines, each developing 1,050 b.h.p. at 300 r.p.m. and driving a single propeller through electric slip couplings and reduction gear arranged at the forward ends of the two sets of main engines, the propeller shaft being led aft between the latter. The cargo pumps are driven by a.c. motors, and when the ship is in port the necessary current for these is supplied by a 300-kW. three-phase 60-cycle 480-volt alternator located on a flat above the main engines, either of which can drive it by means of a chain. The electric current normally required at sea is furnished by two 100-kW. d.c. generators driven at 400 r.p.m. by 4-cylr. F.M. engines, in addition to which there is a 20-kW. emergency dynamo driven by a 40-h.p. high-speed engine. All the auxiliary machinery in the engine room is electrically operated.—*"The Motor Ship", Vol. XXII, No. 264, January, 1942, p. 341.*

Porous Aluminium Castings.

Porosity in aluminium castings can be detected by filling them with petrol or methylated spirit which has been coloured with some distinctive dye such as methylene blue. These liquids have great penetrating ability, and if there is any porosity a blue colour will show outside the casting in at most 15 minutes. In the case of plates or castings which will not hold liquids it is sufficient to paint one side with the coloured fluid, and if there is any sponginess the colour will show on the other side after a short time. The wet side must be kept wet during the whole time of the test. Porosity can be remedied by doping with a hot mixture of four parts of water and one part of sodium silicate, and then subjecting the casting to an air-pressure test of 20 minutes' duration at as high a pressure as the article will stand. The pressure should then be released and the doped parts washed with hot water, any white deposits which may have formed being brushed off. If the porosity is too great to be cured by these means the casting should be rejected, as considerable porosity always means mechanical weakness.—*"Mechanical World", Vol. CXI, No. 2,874, 30th January, 1942, p. 105.*

Alternative for Copper Tubing.

According to the American technical periodical *Automotive Industries*, a well-known manufacturing concern in Michigan has developed a flexible, semi-transparent synthetic resin tubing material as an alternative to copper tubing, called "Saran". It is claimed to be resistant to acids, alkalies and water, and besides being tough, it is said to be capable of withstanding temperatures of up to 250° F. for short periods. Special couplings for this tubing have been designed by another firm, and tests have proved that the assembly can deal with pressures of up to 1,500lb./in.^{2} without leakage or breakage. During the fatigue tests to which a sample length of Saran was subjected, the material was flexed through 15° at the rate of 1,750 per minute for 2,500,000 cycles without breakdown, whereas copper failed at 500 cycles.—*"Practical Engineering", Vol. 5, No. 105, 22nd January, 1942, p. 20.*

Atlas Diesel-engine Coupling Arrangement.

A method of coupling Diesel-engine crankshafts together, in respect of which a British patent has been secured by the A.-B. Atlas Diesel, of Stockholm, is illustrated in Fig. 1. The arrangement with

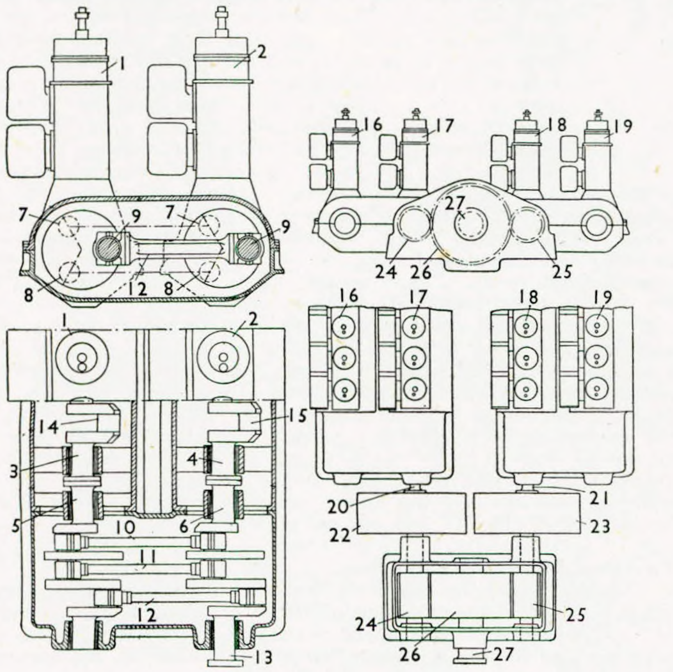


FIG. 1.

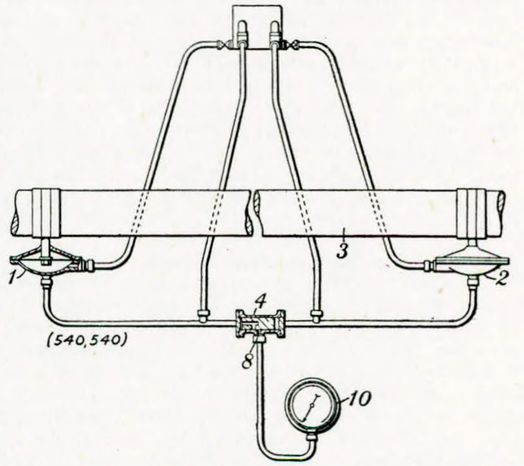
two engines shows the rows of cylinders (*1, 2*) with the crankshafts (*3*) coupled to extensions (*5, 6*). Each of the extended parts consists of a three-throw crankshaft with the crankpins (*7, 8, 9*) set at 120° and coupled by means of rods (*10, 11, 12*) so that the whole assembly rotates together and the power is transmitted finally through the main shaft (*13*). The crankpins (*14, 15*) of the engine crankshafts occupy equal positions. For example, two opposite cranks would be upright rather than one up and the other down, so that the moments of each shaft occur simultaneously and uniformly in the whole transmission. By this means, single-node vibrations of the shaft system are prevented. The arrangement in general has a special application to an installation comprising two double-row engines (*16, 17, 18, 19*). Of these four engines, two have their shafts (*20, 21*) transmitting power through couplings (*22, 23*) to the pinions (*24, 25*) of a reduction gear. The gear wheel (*26*) takes the final drive to the propeller shaft (*27*). No flywheels are required. The employment of twin engines with connecting rods for the crankshafts is of special importance with marine propelling-machinery installations because critical numbers of revolutions may make it impossible to connect the shafts by means of gearing.—*"The Motor Ship", Vol. XXII, No. 265, February, 1942, p. 380.*

Hydraulic Gear for Ships.

In a paper entitled "Hydraulic Automotive Apparatus for Ships", recently read before a meeting of the Institution of Engineers and Shipbuilders in Scotland, the author, E. Bruce Ball, Jr., described the general principles employed in this system and the manner in which it can be applied to various services on board ship. The system involves the use of pressures of the order of 700-800lb./in.^{2}, which permits of small-scale apparatus being employed, and since this pressure is only called for when the operation—say, of closing a watertight door—is in actual progress, the required pressure can be readily developed by a hand-operated pump. There is no pressure in the working system when the controls are released, the operating piston being locked in its cylinder by an automatic mechanical device which can be supplemented by a hydraulic lock when heavy reactive loads are carried. A notable feature of the automotive system is its simplicity, the component parts comprising so-called slave cylinders which provide the operating effort; a transmitter or pump of the rotary type—operated by the simple turning of a handwheel—which provides the pressure energy; selector valves by means of which the power generated in the transmitter is directed to the desired slave cylinder; and special tubing for the hydraulic circuits. The fluid used in the latter is a specially prepared mineral oil having a very high flash point and a very low freezing point, which is neutral to the rubber of the packings in the slave cylinders and to the metal parts with which it comes into contact. Various instruments, including hydraulic and electric position indicators, signal lamps, etc., are mounted on the control panel which also carries the driving wheel of the transmitter pump. The author gives details of these fittings and enumerates some of their applications to the equipment of a ship. He describes how all the sluice valves in a large oil tanker with 10 cargo tanks can be operated from one panel by means of a single transmitter, indicators showing the position of every valve in the ship, whether open or shut. The system can also be applied to the operation of the rudder of ships up to 120ft. in length, as well as to the engine-reversing gear of small motor vessels.—*"Shipbuilding and Shipping Record", Vol. LIX, No. 5, 29th January, 1942, p. 133.*

Hydraulic Torsionmeter.

A world-famous Glasgow firm of steam-valve manufacturers recently patented an improved type of hydraulic torsionmeter. As may be seen in the accompanying drawing, the apparatus comprises two hydraulic diaphragm pumps (*1, 2*) driven from axially-spaced cams on a torque - transmitting shaft (*3*), a piston valve (*4*) and a torque-indicating instrument (*10*) actuated by the pumps. The pump chambers are connected by pipes with the opposite ends of the piston-valve cylinder. The latter has a passage which communicates with a cylinder port (*8*) connected to the instrument (*10*) which indicates the phase of the pumps and thus the torsional deflection of the shaft (*3*). Since the reading of the instrument (*10*) is a function of this torsional deflection in a known length of the shaft, the shaft horse-power can readily be calculated.—*"Engineering", Vol. 153, No. 3,970, 13th February, 1942, p. 140.*



"The Normands of Havre: Shipbuilders and Engineers".

The paper bearing this title is devoted to a brief history of the family of master shipwrights, naval constructors and marine engineers, who have contributed so much to shipbuilding and marine engineering at Honfleur and Havre, and whose direct descendents are still connected with the famous Havre firm known as "La Société Anonyme des Chantiers et Ateliers Augustin Normand". The first member of this virile family to be connected with the industry, François Normand (1661-1715), was one of the leading master shipwrights of his time. Among his descendants were Augustin Normand (1792-1871), who started shipbuilding at Havre in 1816 and who did pioneering work on the construction of steam vessels, screw propulsion and iron shipbuilding; Charles-Benjamin Normand (1830-88), known for his work on compound and triple-expansion marine engines; and Jacques-Augustin Normand (1839-1906), who was recognised as one of the foremost naval constructors in the world. The author gives a short account of the technical achievements of these and certain other members of the family, and concludes his paper with a list of the torpedo boats and destroyers built by the Normand yard between 1877 and 1910.—*Paper by Eng. Capt. E. C. Smith, O.B.E., R.N., read at a meeting of the Newcomen Society, on the 14th January, 1942.*

An Oil-retaining Gland for Propeller Shafts.

A stuffing box or oil-retaining gland incorporating a ball-and-socket joint located between an outer and an inner ring is shown in Fig. 2. The outer ring (*1*) is fixed on the propeller shaft and is in

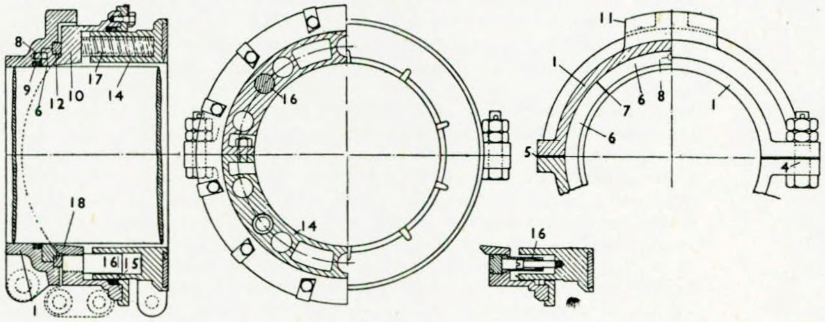


FIG. 2.

halves, clamped together by bolts (*4*). There is a bearing ring in sections (*6*) free of the shaft and with the pieces clear of each other, a pin (*8*) being provided to ensure that the joints (*7*) do not register with those (*5*) of the ring (*1*). Packing rings (*9*) are located in the position shown in the diagram. The inner ring (*10*) is also in halves, but is clear of the shaft and rotates with the outer ring, spaced projections (*11*) being provided. Between the two principal parts is a packing ring (*12*). The outer end of the sleeve (*14*) is mounted on the shaft against the end face of the propeller-bracket boss. Pins (*16*) are fitted for the assembly and drive of the sleeve. The inner ring (*10*) is pressed against the outer ring (*1*) by circumferentially spaced springs (*17*). The inner ring is formed with an annular projection (*18*) which presents the segment of a sphere, bearing on the ring (*6*). Thus, the sleeve (*14*) and the inner ring (*10*) can vibrate together without affecting the packing gland (*15*) through which the sleeve slides. The various parts may be cast steel, sprayed with aluminium on the machined surfaces to prevent corrosion.—*"The Motor Ship", Vol. XXII, No. 265, February, 1942, p. 380.*

A 3,850-b.h.p. Air-injection Engine.

In this country the air-injection Diesel engine is considered completely out of date, but a certain number of engines operating on this principle are still being built in the United States, and many of them are of large size. A 9-cylr. two-stroke crosshead-type Nordberg air-injection engine has just been installed in the power station of the City of Grand Haven, Michigan. It has cylinders 21½in. in diameter, with a piston stroke of 31in., and develops 3,850 b.h.p. at 225 r.p.m. It is stated that one of the reasons for employing air injection is that it gives better results with a low-grade fuel oil than mechanical injection. Among the engine-driven auxiliaries of this Nordberg unit are a blast air-injection compressor and a chain-driven rotary scavenge blower.—*"The Oil Engine", Vol. IX, No. 105, January, 1942, p. 232.*

A Refrigerated Coaster.

Although the design of the hull and machinery of the 693-ton motor coaster "Empire Atoll", recently completed by the Ardrossan Dockyard to the order of the Ministry of War Transport, is similar to that of the Coast Lines' m.v. "Moray Coast", delivered by the same firm nearly a year ago, the new ship is different in one important respect, being specially equipped for the carriage of refrigerated cargoes. She is actually the first vessel of her kind to be equipped in this manner. The main dimensions of the hull are 201·3ft.×32ft.×21·1ft., and the maximum designed draught is 13ft. 7¼in. The machinery is aft and there are two holds and 'tween decks forward, with a total capacity of about 50,600 cu. ft. The vessel is intended mainly for the carriage of frozen meat, of which she can carry 530 tons when fully laden (allowing 95 cu. ft./ton). Up to 295 tons of water ballast can be carried in the fore and after peaks and three D.B. ballast tanks, in addition to 35 tons of fuel oil in a tank under the engine room and 8½ tons of gas oil, in a separate tank forward of the main fuel tank, for the generator engines. The propelling machinery consists of a 7-cylr. standard Polar Diesel engine developing 1,120 b.h.p. at 240 r.p.m., whilst auxiliary power is supplied by three Diesel-engined 40-kW. generators. The bulkheads and sides of the four cargo spaces are insulated with 9in. of granulated cork, and air is circulated around them through 6-in. air trunks by motor-driven fans. The refrigerating plant consists of a twin-cylinder single-acting NH_3 compressor driven by an electric motor and rated at 12,000 B.Th.U./hr. at plus 5° F. evaporation temperature and 86° F. condensation temperature. Thermometers are fitted in the holds and 'tween decks, with electric temperature recorders in the engine room.—*"The Motor Ship", Vol. XXII, No. 264, January, 1942, pp. 320-322.*

Diesel-engined Escort Ships.

More than a year ago the U.S. Navy Department announced that orders had been placed for a number of steel motor vessels of about 170ft. in length, intended for anti-submarine and escort duties. The first of these ships has now entered service and will shortly be followed by about 30 more, in addition to which 36 similar vessels have been laid down. These ships are reported to cost about half-a-million dollars each, and their internal arrangement is said to resemble that of U.S. destroyers of some years ago. They have a complement of three officers and some 50 men, and their speed is about 20 knots. Various types of engines are installed, including G.M., Hooven-Owens-Rentschler and Fairbanks-Morse units, the first-named being a two-stroke exhaust-valve design. In minesweepers with similar hulls Cooper-Bessemer and Busch-Sulzer engines are used. Apart from these steel-built vessels, the U.S. Navy had 45 of the 110-ft. wooden-hulled submarine chasers on order at the time America entered the war. These craft are each equipped with two G.M. light-weight engines of special design. They are V-type 12-cylr. units designed to develop 750 to 800 b.h.p. at 750 r.p.m. and are fitted with a new type of reversing gear and variable-speed gears to permit economic and efficient operation over a wide range of speed. In each cylinder head are two exhaust valves, and the injection valve is combined in a single unit with the fuel pump, in accordance with the normal G.M. practice. The weight of the engines without gearing is about 21lb./b.h.p., and the type is similar to that which has been standardised in America for large Diesel locomotives. The first of the 110-ft. submarine chasers equipped with the new machinery ran trials recently.—*"The Motor Ship", Vol. XXII, No. 265, February, 1942, p. 369.*

Two New Diesel Marine Engines.

Shortly before the war, the Götaverken produced a new type of single-acting two-stroke engine of their own design, and a number of such engines were subsequently constructed. The design has now been modified to such an extent that it is to all intents and purposes an entirely different model. An 8-cylr. engine of this improved type develops 5,600 b.h.p. at 112 r.p.m. and has a cylr. diameter of 680 mm. with a piston stroke of 1,500 mm. The mean indicated pressure is about 92lb./in.^{2}, which is substantially higher than that of any normal single-acting two-stroke engine with port scavenging. In the original design, the scavenging air was delivered by a tandem-piston scavenge pump located at the forward end of the engine and driven off the crankshaft, whereas in the latest design the scavenging air for each cylinder is supplied by an under-piston scavenge pump. There is also a supercharging pump which provides an additional supply of scavenging air and gives the engine an increased output. These changes have resulted in a reduction of about 4ft. 6in. in the length of the unit, together with a saving in weight of approximately 7 per cent. The exhaust valve for each cylinder is vertical and actuated through a cross-beam driven by two vertical levers lifted by cams on the crankshaft. The cylinder covers are circular and the fuel valve is located at one side of the cover, the relief valve being likewise fitted in the cylinder head. No camshaft is required apart from a short shaft, chain-driven from the crankshaft, for the actuation of the fuel pumps. The scavenging-air pressure is about 1·6lb./in.^{2} and the fuel consumption is reported to be 0·355lb./b.h.p.-hr., the mechanical efficiency being something like 82 per cent. The manœuvring wheel is interconnected with the telegraph and the operation of a single wheel suffices to start, stop or reverse the engine, as well as to control the running speed. In addition to this 5,600-b.h.p. model, a smaller engine of similar design is being produced, with a cylr. diameter of 630 mm. and piston stroke of 1,300 mm. This unit runs normally at 125 r.p.m. and is built with five, six and eight cylinders, developing respectively 2,900, 3,500 and 4,700 b.h.p. Electric welding is extensively used in the construction of the framework of both types, in consequence of which their respective weights are reported to be some 25 to 30 per cent. lower than those of corresponding single-acting crosshead engines built some years ago. The first engine of the new design was constructed in 1939, and up to the present time more than 40 units have been built, with a total output of 170,000 b.h.p. Apart from these engines, the Götaverken have also developed a single-acting four-stroke pressure-charged engine of improved design, and the new

20-knot passenger motorship "Saga", designed for the Gothenburg-Tilbury service and now approaching completion, is equipped with four of these units. They have cylinders of 500 mm. diameter with a piston stroke of 700 mm. and are each rated at 1,700 b.h.p. at 270 r.p.m. All four engines drive a single propeller at 115 r.p.m. through A.S.E.A. electric slip couplings and reduction gearing. The outstanding features of this type of engine include a new combined fuel valve and pump, also a system of partial supercharging. A single air-admission valve is used, but there are separate inlets for the air drawn in by suction and compressed air from an air compressor located at the end of the engine and driven by the crankshaft. The mean indicated pressure is about 120lb./in.^{2}.—*"The Motor Ship", Vol. XXII, No. 265, February, 1942, pp. 364-366.*

Meetings of German Shipbuilding Institutions.

At a meeting of the Hamburg Tank Society in Hamburg on October 17th, attended by 400 members, several papers were read, including one on "Recent Experience with Gas-engined Vessels". Another dealt with "The Design of a 72-ft. Standard Motor Fishing Boat", and a third was entitled "The Use of Plated Steel Plates in Shipbuilding". The Schiffbau Technische Gesellschaft held three meetings during the past few months. On November 17th a paper on "The Salient Features of Some Foreign Battleships and Aircraft Carriers of the Present War" was read, and on November 28th, at a meeting held in conjunction with the V.D.I. in Hamburg, Herr Bleicken, superintendent engineer of the Hamburg-American Line, read a paper on "Alternating Current on Board Merchant Ships". At a meeting held in Danzig on December 9th, a paper was read on "The Development of Warship Types Since the Outbreak of War".—*"The Motor Ship", Vol. XXII, No. 265, February, 1942, p. 374.*

Paddle-wheel Motorship for Germany.

At a recent meeting of the Hamburg Tank Society, held at Düsseldorf, a paper was read on the subject of Diesel- and gas-engine drive for paddle ships. The author strongly advocated this system of propulsion for the larger type of inland water tugs, cargo ships and passenger vessels, and particularly favoured the adoption of the present-day gas engine, with Diesel-oil pilot ignition. He stated that the employment of the new designs of laminated-disc clutch developed by the Demag concern, in conjunction with flywheels of suitable dimensions, ensured the provision of a flexible drive. The gear is hydraulically operated and controlled from the bridge by oil pressure. A special type of Deutz engine has been developed, and a complete unit designed in which the exhaust gases are passed through a waste-heat boiler to raise all the steam required for auxiliary use on board. It is claimed that the fuel consumption is only 0·536lb./b.h.p.-hr.—*"The Motorship", Vol. XXI, No. 265, February, 1942, p. 373.*

560-h.p. Twin-screw Gas-engined Passenger Vessel.

Among the numerous gas-engined craft built in Germany during the past year is the twin-screw passenger vessel "Wilhelm Boltz", which was placed in service in Hamburg harbour quite recently. She is equipped with two 280-b.h.p. Deutz gas engines, with relatively high compression and Diesel-oil pilot ignition. The engines are fitted with electrically-operated reversing gear. The gas producer is of a new type developed by the Klöckner-Humboldt-Deutz A.-G., with a rotating grate. It incorporates a new system of water cooling, the cooler being of small size designed for intensive cooling. The Deutz Company are now reported to be manufacturing high-compression gas engines of the two-stroke type, also with Diesel-oil pilot ignition.—*"The Motor Ship", Vol. XXII, No. 265, February, 1942, p. 374.*

Gas and Dual-fuel Motorships and Craft.

For some years before the war, gas-engined tugs were in regular use on the Rhine, and proved the value of this form of propulsion under arduous conditions. Two of the first tugs of modern design for gas propulsion to be built in Germany were the "Harpen I" and "E. Tengelmann" (*described in J. F. Gibbon's paper on "Gas Engines for Small Craft" in* TRANSACTIONS *of November, 1938*) and the present shortage of oil fuel in Germany has led to a marked increase in the use of gas engines for the propulsion of tugs and other vessels employed on the inland waterways of that country. The Deutz Company were reported to have developed an engine for such craft which could be driven by almost any kind of fuel, whilst other firms are said to be adapting petrol engines for gas operation by altering the compression ratio from 5·5 :1 to about 8 :1. The Deutz concern have now developed a unit with two types of cylinder head and piston, with an injection pump and magneto, which is readily convertible to either oil or gas operation. The Bussing-NAG concern have converted their engines from oil to other fuels by plugging the pre-combustion chamber and altering the piston, whilst the Mercedes-Benz Company are effecting similar conversions by replacing the ante-chamber and using a shorter piston. A considerable amount of research and development work has also been carried out in this country on dual-fuel engines designed to run on gas or oil, as required. In the case of engines for marine use, a dual-fuel engine would normally be adapted to run on oil or producer-gas, and would be equipped with the usual compression-engine fuel pump and nozzles, the latter being located in the cylinders between the usual inlet and exhaust valves. The air inlet is so designed that it can be employed to admit an air-gas mixture, or air only, if desired, and a compression ratio of about 14 :1 is used. About 10 per cent.—but never less than 5 per cent.—of the power is due to oil, and a thermal efficiency of about 35 per cent. is attained. It should be possible to instal dual-fuel engines of up to 1,500 h.p. in the smaller types of vessel, as the value and efficiency of gas-engined ships have been proved, and the time may be ripe for further developments in gas/oil propulsion. Among the advantages to be gained by its use are the following: *(1)* liquid or solid fuel, as available, may be employed; *(2)* home-produced fuels such as coal, coke or anthracite, could be used in river, coastal and even small ocean-going craft with advantage to the national economy and freedom from dependence on imported fuel; *(3)* ships could be built more rapidly and at a lower cost than steam-operated craft; *(4)* additional cargo space would be available as compared with that of a steam-operated vessel, as much as 12,000 cu. ft. of cargo space being saved in a gas-oil engined ship of 7,000 tons; and *(5)* economical running costs. Ships equipped with dual-fuel engines would be designed on the basis of the power obtainable with producer-gas plus the determined quantity of ignition oil, and, of course, when operating on oil only, there would be a considerable reserve of engine power. The gas generator would preferably be of the down-draught or a double-draught type, with automatic fuel feed, rotary grate, and carefully designed cooling and cleaning arrangements, so that bituminous coal could be used as fuel. A simplified plant could be installed in river and coastal craft serving districts where anthracite and coke are readily available. The fuel consumption per b.h.p.-hr. of a vessel equipped with a modern dual-fuel engine would work out at about 0·66lb. of solid fuel plus 0·04lb. of fuel oil for starting and ignition purposes. An important feature of producer-gas plant for marine work is that the design of the gas-cooling and cleaning arrangements is facilitated by the fact that an adequate supply of water is available for these purposes. Space is limited, but spray and circulating washers and coolers can be employed and the gas freed from moisture by dry-plate, sawdust-tray scrubbers or other well-known methods. The air required by the gas generator could be drawn from above deck through pipe-lines, so that fouling of engine rooms, etc., would be avoided.—*N. C. Jones, "The Shipbuilder", Vol. XLIX, No. 391, February, 1942, pp. 37-38.*

The Axial Vibration of Diesel Engine Crankshafts.

The paper describes an investigation carried out to determine the magnitude of axial vibration of engine crankshafts, with the object of establishing that axial resonance may occur within the normal running range of an engine. The author gives the results of tests on various makes of engine, using a portable vibrograph to record the axial movement of the crankshaft. In some cases the torsional oscillation was measured simultaneously by means of a universal vibrograph. The vibrograph records indicate that at certain speeds some engines are subject to a very marked increase in axial vibration of a magnitude consistent with a condition of axial resonance. The author describes a test in which the natural frequency of axial vibration of a crankshaft is determined experimentally by using an a.c. electromagnet as a means of applying an alternating force. Expressions are derived for calculating the axial deflection of a crankshaft due to the piston load, and also for calculating the axial deflection under an axial load. It is suggested that the relation between these two deflections determines the magnitude of the axial force set up by the piston loading. From a series of load-deflection tests carried out on a number of crankshafts, the author determines the value of empirical factors to be used in estimating the axial stiffness.—*Paper by R. Poole, "Journal and Proceedings of the Institution of Mechanical Engineers", Vol. 146, No. 4, February, 1942, pp. 167-182.*

Cleaning Condenser Tubes.

Condensers supplied by river water vary a great deal in their cleanliness at certain times of the year. In the autumn when dead leaves are about, and at times of flood, these are liable to get drawn into the condenser tubes and to choke the circulating-water inlets, whilst in the summer months, when the river temperature is high, minute growths attach themselves to the condenser tubes These are of a slimy, greasy nature and lower the heat-transmission rate to a great extent. This growth can be inhibited by the use of chlorine. The procedure is to admit small quantities of chlorine gas from the cylinders in which it is kept to a certain part of the circulating water, where it is absorbed. This water is then returned and mixed with the main-inlet circulating water. If, however, the employment of chlorine gas is not deemed to be justifiable for financial reasons, condenser tubes can be cleaned in the usual way by means of wire brushes or by condenser-tube bullets. Wire brushing is a laborious and lengthy job, which calls for close supervision. A quicker method is to use rubber or gauze bullets which are propelled through the tubes by means of a jet of compressed air or water. A further advantage of this method is that the tube is cleaned by the bullet, and that it does not depend on whether a man uses a brush effectively or in a half-hearted manner. By means of bullets, a condenser of a 20,000-kW. generating set, with over 6,000 tubes, has been cleaned in 48 man-hours. One man or men can be placing the bullets in the tubes whilst the other man or men are firing them. After the bullets have travelled through the tubes they are collected from the water-box and washed ready for further use.—*A. S. Hollin, "The Steam Engineer", Vol. 11, No. 125, February, 1942, pp. 115-116.*

Two New Fittings for Oil Engines from America.

Two interesting oil-engine devices have recently made their appearance in America. The first is a simple little fitting, known as a "visco-meter", which is inserted in the forced lubrication system of an engine for giving, on a dial gauge, a constant indication of the viscosity of the lubricating oil. The gauge dial is divided into three sections, marked "thin", "ideal" and "heavy", respectively. Although the device does not correct the defect when the gauge pointer leaves the "ideal" zone, the instrument should prove useful, as it is of robust construction and has only one working part. The second device is known as a "cetane selector" and permits of the timing of the fuel pumps being adjusted by a handy control while the engine is running so as to secure the most efficient utilisation of the particular fuel being used. A quadrant device at the end of the engine provides visual indication of how far the timing of the injection pumps has been advanced or retarded (all the pumps are, of course, altered simultaneously), and it is claimed that the range of timing variation provided by the device is great enough to deal with the widest limits of fuel ignition characteristics likely to be encountered in practice.—*"Gas and Oil Power", Vol. XXXVII, No. 437, February, 1942, p. 40.*

Metal Spraying by the Wire Process.

The paper deals with some of the developments of this comparatively new process of metal spraying. The basic principles of the wire-spraying pistol are explained, and the use of the metallic films produced as a means of protection against corrosion is described. Particular notice is given to aluminium coatings and to the application of the aluminising process for protection against oxidation at high temperatures. The theory underlying the use of deposits of the harder metals for the reclamation of worn parts is discussed, and suggestions are made with regard to the issue of specifications for sprayed coatings. The methods used in the preparation of surfaces for receiving the spray, including shot-blasting and rough turning, are described. Some particulars of the mechanical properties of the deposits are given, and their structure is discussed. The concluding portion of the paper is devoted to notes on methods of finishing, and descriptions of the use of the process in its application to specific problems in marine engineering, including the reclamation and protection of propellers and the building-up of propeller shafts, bracket bushes, rudders, turbine shafts and rotors, crankshafts, main bearings, cylinder liners, piston valves and rods, gudgeon pins, boiler valves, steam joints, pump parts and armature shafts. The protection of ships' hulls, water tanks, winches and boiler mountings is likewise discussed. The authors point out that metal spraying frequently meets the demand for a "putting-on tool", but that it is not necessarily applicable to every case of wear, and that parts subjected to heavy shock loads are not, generally speaking, ideal subjects for spraying.—*Paper by W. E. Ballard, G. F. Fairbairn and F. S. Pilkington, "Transactions of the Institute of Marine Engineers", Vol. LIV, No. 1, February, 1942, pp. 1-6.*

Marine Steam Plant.

The boiler exercises the greatest retarding influence on progress in modern marine steam engineering practice, and minimises the great possibilities of marine steam engineering practice, and minimises the great possibilities of marine steam prime movers. The Scotch boiler has given excellent service for some 60 years, but unfortunately, it has undergone no major improvement during that period. Its construction is such that any increase in steam pressure or output involves a disproportionate increase in weight, so that a maximum working pressure of 300lb./in.^{2} only is practicable. Since the thermal efficiency of a steam prime mover depends upon the difference between the initial and final temperatures of its working medium, any increase in efficiency, nowadays, must be sought at the higher end of the cycle, but the limitations of the Scotch boiler have tended to compel designers to adopt the cheaper method of obtaining increased efficiency by the employment of superheaters, and they are restricted by the properties of ordinary steel to a maximum temperature of 750° to 800° F. Where reciprocating steam engines are utilised, it is usually necessary to work at an even lower temperature, in order to avoid lubricating troubles. Whatever the type of steam prime mover employed, however, the enhanced efficiency gained by superheat is largely due to the subsequent reduction of condensation losses, so that the principal improvement—particularly in the case of turbine machinery—occurs at the L.P. end of the cycle. To obtain the maximum advantage it is necessary to utilise higher steam pressures as well as higher temperatures, and these higher pressures are beyond the range of the Scotch boiler and reciprocating engine. A weight reduction of the order of 50 per cent. can be achieved by the employment of high-pressure geared-turbine machinery, the cost of which is, however, higher than that of normal low-pressure plant; but against this must be placed the fuel saving effected, as well as a greater carrying capacity during the life of the ship. It is obvious that if higher efficiencies with steam propelling machinery are to be achieved, the Scotch boiler must be replaced by the watertube boiler, despite the misgiving with which the latter is still viewed by certain engineers. During the early stages of its development, the watertube boiler undoubtedly gave trouble, but some of this was due to insufficient care in the matter of feed-water purity. It has been proved that where a closed feed system is employed, supplemented by a colloidal treatment to ensure that the small amount of residual oxygen in the feed water is absorbed, the water and superheater tubes of watertube boilers will give many years of service. Furthermore with tight condensers and adherence to the principles indicated, it has been found quite safe to run such boilers for six or even twelve months without internal examination and cleaning. It has also been contended that there are not sufficient experienced marine engineers, at the present time, to operate watertube boilers in particular, and high-pressure steam systems in general. This aspect of the matter would seem to require serious consideration in the near future, in view of the present course of the war. We are expecting to obtain a number of new steamships from the U.S.A. from time to time; but, as hostilities progress, materials of construction will become more scarce. American engineers already favour the use of oil-fired watertube boilers for all but moderate powers; so that it is more than likely that the demand for steel and fuel conservation will lead to an even more widespread adoption of marine boilers capable of giving a maximum output for a minimum weight and fuel consumption. The low-pressure Scotch boiler does not show up well in this respect, and the practical solution is the adoption of oil-fired watertube boilers and geared turbines working at pressure and temperature conditions considerably in excess of those now current in the British merchant service. —*S. D. Scorer, "The Shipbuilder", Vol. XLIX", No. 391, February, 1942, pp. 36-37.*

Neither The Institute of Marine Engineers nor The Institution of Naval Architects is responsible for the statements made or the opinions expressed in the preceding pages.