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Gearing as Applied to the Marine Steam Turbine

BY MR. JOHN HOUSTON (Member).

READ

Tuesday, May 13, *at* 6 *p.m.*

CHAIRMAN: MR. B. P. FIELDEN (Chairman of Council).

The CHAIRMAN: We recently had a paper on Air Pumps and on that occasion the efficiency of turbines was disoussed. To-night Mr. Houston will deal with part of the turbine itself, viz., gearing, and as geared turbines are being largely adopted for the propulsion of ships the paper will be most useful and instructive to all of us.

The improvement of the marine engine depends largely upon two things "Failures" and "Publicity." We learn most through having troubles which in other words is known as experience, and our Institute is the means whereby experience is made public.

With the advent of the direct coupled steam turbine into the Merchant Service, the desirability of having some form of reduction gear between the high-speed turbine and the propeller was soon obvious, especially for vessels with a speed of 15 knots or under, and suggestions for effecting this have been formulated in three ways (*a*) with mechanical reduction and transmission gear; (*b*) with hydraulic reduction and transmission gear; (*c*) with electric reduction and transmission gear.

Though the possibilities of such intermediate methods are obviously greater in the case of slow and medium speed vessels, it is not to be inferred from the foregoing remarks that these methods cannot be successfully applied to the propulsion of high-speed vessels, as at the present time many of our fastest vessels are being so fitted.

Before discussing in detail the mechanical form of gearing, which has almost superseded the hydraulic and electric forms of transmission, I propose dealing with these latter in a few sentences only.

One form of the hydraulic gear, namely, the Fottinger hydraulic and transmission gear, consists of a primary water wheel driven by the steam turbine shaft, which transmits water at a certain velocity through guide vanes to a secondary wheel in the same casing, and in the same axial plane. This secondary wheel is mounted on a secondary spindle connected to the propeller shaft, the diameters of the wheels being so arranged that ratios up to 10 to 1 may be obtained between the revolutions of the turbine and propeller. One of the distinct advantages of this gear is the possibility it affords for obtaining a full power reversing arrangement without special astern steam turbines, by incorporating a second set of wheels of opposite blade angle in an extension of the same casing as the ahead wheels, the water being admitted to the ahead or astern wheels as required by suitable valve arrangement.

There have been many proposals for interposing a generator and motor between a high-speed turbine and the propeller. It would appear from the Press that the United States naval authorities are concentrating on this method of propulsion, but it appears to me that there are several objections to it, and these mostly of a practical nature, such as the necessity for the use of high voltages to keep the generators and motors of reasonable size and weight, and high voltages are particularly dangerous on board ship, also the danger of failure owing to the access of water to the electrical machinery. Then the complications arising from having generators and motors to look after as compared with a gear box are considerable, especially when one remembers that Marine Engineers, as a rule, only have a knowledge of electric light installations, and that on a comparatively small scale. They would have to be specially trained in the control of a plant generating and using several thousand horse-power with its complicated accessories. Like the hydraulic system, however, it has the advantage of not requiring a special astern turbine.

The U.S. method is to have the high-speed turbine coupled direct to the generator, and the induction motors are connected direct to the propellers. By cutting out of poles on the motors, the speed of these can be controlled, and this gives the reduction as between the turbine and the propeller.

I recently read a paper on electric propulsion of ships, which was given before the Society of Naval Architects and Marine Engineers in New York, but the writer, in my opinion, did not show any real saving over mechanical gearing either in upkeep or in fuel consumption.

A friend of mine quite lately had the privilege of being on board the U.S. Battleship *New Mexico*, but I do not think he was greatly impressed by the possibilities of this mode of propulsion, at least for the merchant service. One vessel in the United Kingdom fitted with turbo-electric motors is the S.S. *Wulsty Castle*, and in addition, mechanical reduction gear is interposed between the motor and the propeller, the ratio of reduction in her case being 9·4 to 1.

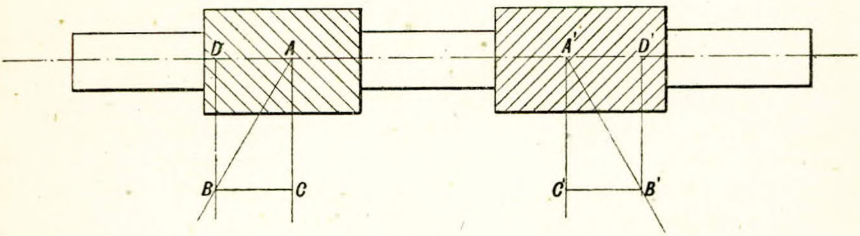
This briefly describes the hydraulic and electric forms of transmission, but the modern trend of thought in this country is particularly towards the mechanical reduction and transmission gear, and this brings us to our subject proper.—The helical spur wheel reduction gear as applied to the marine steam turbine.

The success or failure of a geared turbine installation depends on three important factors, viz., design, workmanship, and materials, and it is proposed to deal with these in seriatim.

Design: Of the mechanical form of gearing there are at present two types, commonly known as the "rigid" and "floating" frame gears. Of the latter I cannot say much as I have never actually seen an example, but from what I have read about it, it is claimed that there is a considerable saving in weight, and also that it adjusts itself more readily to the correct pinion alignment. The flexible coupling in the rigid gear allows for this, but perhaps the discussion will give us more information as to the relative merits of the two types.

In order to avoid end thrust on the pinion shaft or on the gear, two pinions and wheels are required, one wheel and pinion having right-handed helices, and the other pair left-handed, the helices being at an angle usually of 30° to the axis of the shaft, although an angle of 45° is favoured by some designers. By having a larger angle of helix a greater number of teeth are in mesh at one and the same time.

To allow of both pairs of pinions and wheels engaging properly, and of the total forces on the teeth being evenly distributed—a most important point to keep in view—a slight end play is necessary on the pinion shaft. This can easily be seen by giving attention for a few moments to the following figure:—



Let the total forces at the tooth contacts, which will be nearly at right angles to the teeth at A and A^1 be represented by A B and A^1 B^1, and they will be at an angle of 30° to the vertical. Complete the parallelograms A C B D and A^1 C^1 B^1 D^1, and it will be seen that they are similar. The only exial forces acting on the pinions are A D and A^1 D^1, and if these are unequal the pinion shaft will at once move longitudinally, which it is free to do, until they are equal and the two parallelograms of forces then become equal in every respect.

By this means and with the flexible couplings attached to the pinion shaft, errors of alignment adjust themselves automatically. There are other errors of alignment which may take place, but they will be dealt with later.

As a minimum amount of noise is desirable, a small pitch is essential, and this necessarily means broad teeth, in view of the enormous powers which have to be transmitted with a high pitch line speed, and with a limiting pressure not yet exceeding 1,000 lbs. per inch of tooth width.

As already stated the full breadth of tooth required is divided into two, having right and left-hand helices, and this arrangement also allows of three journals on the pinion shaft, which reduces the tendency to cross-bending action.

The proportioning of the pinion shaft journals is, as a rule, determined by the torque on the shaft due to the power transmitted, and by the deflection due to the forces acting on the pinions.

This diameter to a certain extent determines the pinion diameter, but the limiting pressure per inch of tooth width has to be taken into consideration, in conjunction with the revolutions of the pinion shaft, keeping in view of course, the ratio of reduction desired. The number of teeth also must be such that there is no interference. The pressure per inch of width on pinion teeth varies nearly as $\sqrt{\text{pinion diar. in inches}}$, and it has been found in practice that load per inch width of pinion in lbs. $\div$ by $\sqrt{\text{Diar. of pinion in inches}}$ should be approximately 200.

There is a relationship also between the diameter of pinion and width of face of the teeth, as the deflection becomes greater the wider the face, and to decrease the deflection the diameter would require to be increased.

Having decided upon the load per inch of tooth width and the diameter and revolution per minute of the pinion, the width of teeth face necessary to transmit the required S.H.P. can then be obtained.

Any power, however large, could be transmitted, if the reduction gear were made large enough, but if it is too large and heavy it would be useless for the propulsion of ships. It calls for a keen judgment on the part of the draughtsman to determine the proportions to give to the gearing to obtain the desired results, and at the same time to keep the weight down to a minimum. As in the designs of most other things, there will be one set of proportions which will be better than any other.

The following simple formula will at once present itself:—

Let D = Dia. of pinion in inches at pitch line.
L = total length of tooth face in inches.
p = average pressure per inch width of tooth face in lbs.
R = revolutions per minute of pinion.
P = S.H.P. transmitted.

$$\text{Then } P = \frac{\pi D\ R\ p\ L}{12 \times 33000}$$

$$\therefore\ P = \text{Const.} \times D\ R\ p\ L.$$

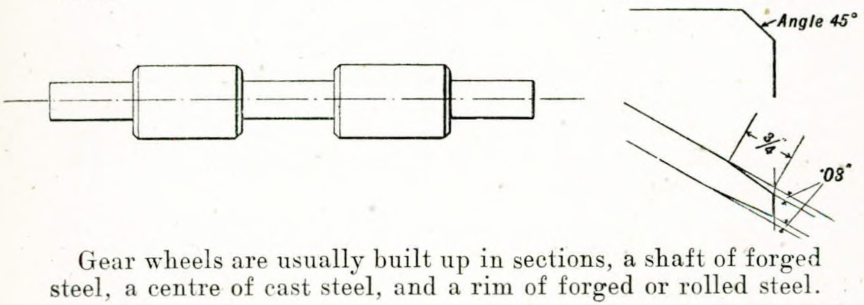
from which any one unknown quantity can be obtained, given the others.

Let us consider for a moment the action of the teeth when in mesh. No two bodies bearing on one another have point or line contact; if they had, the intensity of stress would be infinite. There is always sufficient distortion to bring areas into contact, and in the case of teeth there is actually a very narrow strip in contact. As this narrow strip is difficult to measure, the stress on the teeth is usually not considered in designing helical gearing, the maximum load per inch of width permissible being taken, and this has been arrived at from practical running experience. The presence of lubricating oil keeping the surfaces from actual contact, diminishing friction, and increasing the effectual breadth of the above strip, will greatly increase the allowable load.

The pinion teeth should be chamfered away at the ends of the tooth faces, and the flanks of the teeth relieved in region of the chamfering, and this work should be carried out strictly to gauge.

The idea of this is to overcome the danger of the extreme ends of the teeth being overloaded at the moment of coming into mesh, but this is only considered necessary in the pinions.

To return to our formula, the calculations need only be made for the pinion, as ample rigidity can readily be given to the gear wheel. Its teeth need only be made as strong as those of the pinion, as it is the pinion which will be most affected by wear.



Gear wheels are usually built up in sections, a shaft of forged steel, a centre of cast steel, and a rim of forged or rolled steel.

In addition to rotational stiffness they should be stiff in a fore and aft direction, so that no displacement of the helices will occur. The limiting speed for the gear wheel at the P.C.D. is about 140ft. per second.

The gear casing generally, must be so designed that there is rigidity throughout, otherwise some form of misalignment will

NOTE.—The ¾ on right hand side of above block should read ⅜.

take place. The bearings must receive proper consideration with a view to oiling and bearing surface, to meet the requirements of the high peripheral speed of the shafts.

It has been mentioned that a small pitch of teeth is essential, and gears which have come under my notice have a normal pitch of ·5833″ which gives a circumferential pitch of ·8159″. It has been found with this pitch that the smallest number of teeth in the pinion should not be less than 25, but 30 is safer. With a less number of teeth than 25 and with a normal pitch of ·5833″, interference of the teeth would take place with the resultant undercutting. In designing pinions therefore, attention must always be given to this point. The shape of the teeth is involute, and as is well known with involute teeth, the line of action or contact is a straight line tangent to the base circles of the gear and pinion.

It is also well known that the ratio of the radii of the base circles of two wheels gearing with one another is the same as the ratio of the radii of the pitch circles. If the centres of the wheels be pushed further apart or closer together the wheels will still work correctly together, because this is equivalent to altering the radii of the pitch circles, without altering their ratio.

This is of special advantage in cases where the distance between the centres of two wheels cannot be maintained constant, as might happen in the type of gear we are considering, owing to the wearing of the bearings, or even to the heating of same. If the temperature of the pinion or gear, or both, rises above that of the bedplate the expansion will cause the designed pitch circles to overlap slightly, but this would produce no error in the action of involute teeth.

As temperature has been mentioned, I cannot do better than follow on with a few remarks as to the system of oiling of the gears. It is necessary that the supply of oil should be copious, and directed to the proper place. It is usually applied through nozzles and sprayers under a pressure of about seven to 10 lbs., on the gearing at the meshing of the pinion and wheels, so that the oil has not time to be thrown off owing to centrifugal force. In consequence of the high revolution speed of the pinion it will naturally incline to heat, and as is well known, a plunging supply of oil will carry off heat better than an oil bath, and the oil is also kept cooler. An oil cooler is an essential fitting, as if the oil becomes heated, carbonisation sets in and particles will gradually be deposited on the tooth with the resulting clogging up of same.

One breakdown of gearing which I heard of in a naval vessel was traced to this cause. The tooth clearances became nil, with the result that the points of the teeth fouled the bottom, the bedplate was fractured, and the pinion wheel teeth distorted and broken.

Mention has been made previously of the cross-bending action on the pinion shaft, but by the universally adopted method of fitting three bearings, at least in single reduction gears, any trouble likely to arise from this cause is very remote. The same can be said about the torsional yield of the pinion shaft. No doubt there is a slight torque on this shaft, but errors due to bending or twisting are infinitesimal, compared with the errors of inaccurate gear cutting, and this brings us to *Workmanship*.

It has sometimes been supposed that it is sufficient to give the pinion shaft end play, but as we have seen, this only divides the power equally between the two helices, it does not otherwise alter the distribution of tooth pressure. In cutting the teeth of gear wheels and pinions, very great precision is demanded in order to secure silent running and a proper distribution of the load across the whole width of the engaging faces. Extreme accuracy of tooth is therefore another requisite in reduction gears. With the present day demand for reduction gears, it behoves manufacturers to give their attention to the tuning up of their gear-cutting appliances and machines, and I have no doubt that the importance of this is not forgotten.

The teeth are cut usually by a milling process in a "hobbing" machine. The cutting is done by a milling tool called a "hob"; this is a worm, in the threads of which longtitudinal gashes are made to form cutting edges. A standard hob is $5\frac{1}{2}$in. on the P.C.D., the pitch of worm being 7/12in. which gives the normal pitch of ·5833 to the helical teeth.

During construction arrangements should always be made whereby every gear cut can be accurately gauged, and the error, if any, recorded.

Even with the greatest care there will very often be an error to record, but if this does not exceed ·001″ on a length of 20″ or thereabouts of tooth width with teeth set at an angle of 30° to the axis of the shaft, it is found that the gear meshes and runs without trouble, and as this error is greater than any due to bending and twisting, these latter can be neglected. As a rule a very much smaller error than this is recorded. Gears should always be given a bench trial under load, and if the hard spots visible on the subsequent examination are found to be evenly distributed over the teeth very little need be done to them, a slight scraping being all that is considered necessary. Should the hard bearing places be found in line with one another this would point to considerable error, and something drastic would have to be done, but the necessity of accurate cutting of the teeth is understood, and the greatest possible care is taken. It is not advisable that scraping of the teeth should have to be resorted to, to any extent, as the proper rolling circle could not be maintained.

Pitting of the teeth has been noticed in some instances, in gears which have been running over a period, and this in my opinion, is due to the intense pressure on the hard points, causing the lamination of the material, and the consequent flaking off of same, leaving small cavities or hollows. This in turn will bring other points under hard pressure with the same cycle of action.

This theory of mine may be entirely wrong, but if it is right, it proves the necessity for a smooth finish to the machining of the teeth.

In the machining of the bearings for the pinion shaft and the main gear wheel, every care must be taken to ensure that the axes of both shafts are exactly parallel. Stiff boring bars must be employed and any slight error should be corrected in the final bedding of the bearings and the shafts in place.

As the forces on pinion and gear shafts are much more nearly vertical than horizontal, the rate of vertical wear will be much more than that of side wear, hence we may conclude that if once set correctly the alignment of the pinion shaft will remain fairly correct; this proves the necessity of having the axes of both shafts exactly parallel to begin with.

Special attention must also be given to the adjustments of the bearings, as slackness in these would contribute to want of alignment and would lead to unequal pressure on the teeth. This is a point which applies more particularly to gears which have been running rather than to new machinery, as the question of proper adjustment in the shops will no doubt receive its due consideration.

Another point which might be emphasised is the balancing of the gear wheels, and also the pinions. Just as the most careful balancing of the turbine rotors is essential in order to eliminate vibration, so is the balancing of the gear wheels necessary for the same reason.

But the best of designs and the most satisfactory workmanship will be brought to nought, if the materials are not up to the requirements of the work they are called upon to do. To make use of a metaphor recently originated by a gentleman of whom we have all heard, "One cannot make an 'A1' machine out of 'C3' materials."

For a gear therefore to have a reasonable length of useful service, the very best of materials must be employed. The pinion shaft and pinions are as a rule, made of nickel steel, oil-hardened, with a tensile strength of about 45 tons per square inch, and an elongation of 20 per cent. on a standard test piece. The tyre of the gear wheel can be of ordinary 28-32 ton steel, but usually a higher tonnage, say 31-35 tons, is asked for. The other materials used in building up the gear wheel must of course be of the best, and free from all flaws, and the scantlings sufficiently heavy to give the necessary rigidity to the wheel as a whole. There are many proved whitemetals for bearings, any of which will answer, provided the oil supply to them is kept up.

The life of a gear depends as much on the quality of the materials used in its construction as on the other points which I have emphasised.

Every endeavour should be made to have the materials of as homogeneous a nature as is possible, and absolutely without flaw. Pitting of teeth already mentioned may be due to a lack of homogeneity in the material at the tooth surfaces.

As the pitch is small the teeth are more liable to injury, if roughly handled, than would otherwise be the case, and dents on the pinion teeth will make themselves repeatedly felt in the gear wheel teeth.

Those who are entrusted with the running of gears must exercise the greatest care that no damage is done to the teeth in overhauling.

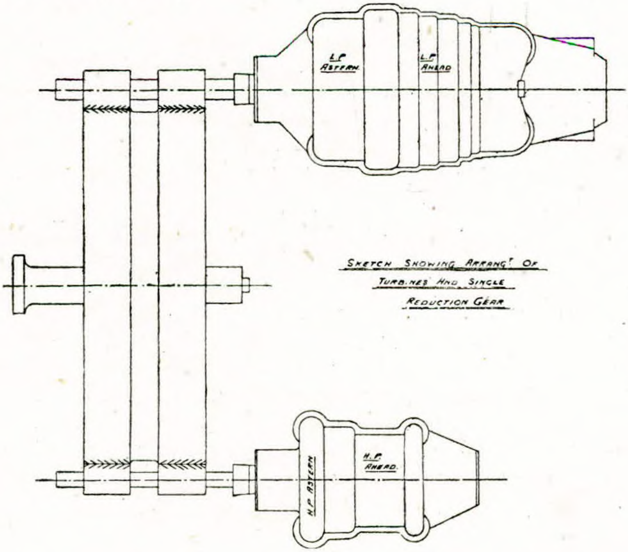
The necessity of keeping all foreign bodies outside the gear box need scarcely be emphasised.

Should the lubricating oil supply fail, the teeth will soon be scored and cut, but so long as the necessary oil film is present to prevent metallic contact, the teeth should run well.

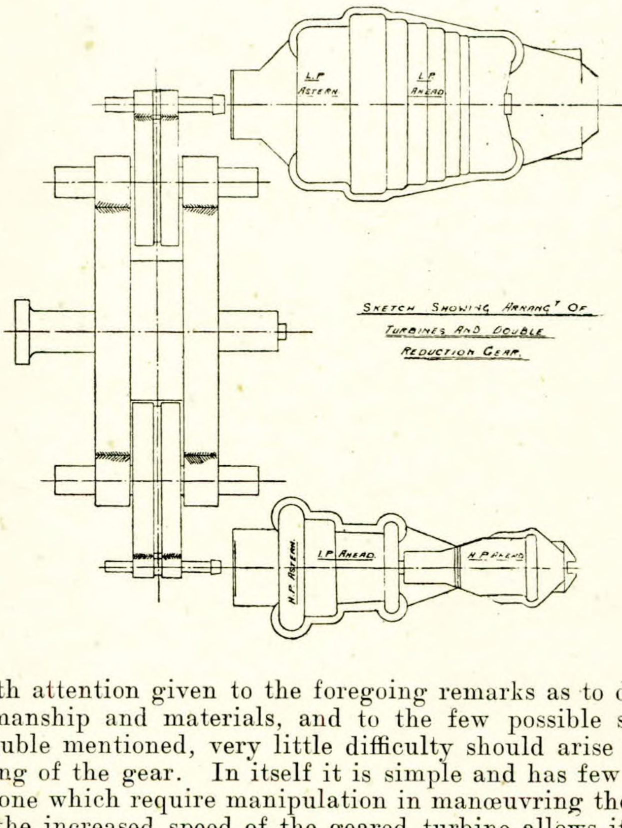
In writing the whole of the foregoing I have been thinking principally of the single reduction gear, but these remarks apply equally to the double reduction gear, which is rapidly coming into use. In the latter, the tyre of the first reduction wheel is also usually of nickel steel, oil-hardened, and in the second reduction pinion and wheel some makers employ a larger pitch of tooth, which is 1·2in. normal as compared with ·58in. in the first reduction.

In it also the width of face of the teeth does not require to be as much as in the single reduction, which allows of a more compact arrangement, and the centre bearing of the pinion shafts can be dispensed with.

The sketches shown give an idea of the arrangement of the two forms of gearing.



The S.S. *Vespasian* was the first vessel to be fitted with geared turbines, and she was converted from reciprocating engines for experimental purposes. The ratio of gearing in her case was 20 to 1, the revolutions of the turbines being 1,460 per minute, and the S.H.P. about 1,100. Lately the double reduction has been coming more to the front, with ratios of 45 and 50 to 1, with a corresponding saving in weight and in fuel consumption.



SKETCH SHOWING ARRANG^T OF TURBINES AND DOUBLE REDUCTION GEAR.

With attention given to the foregoing remarks as to design, workmanship and materials, and to the few possible sources of trouble mentioned, very little difficulty should arise in the running of the gear. In itself it is simple and has few parts, and none which require manipulation in manœuvring the ship. Also the increased speed of the geared turbine allows it to be greatly reduced in size, and in the rows and number of blades. This combination makes the machinery less complex and simpler, and therefore there is less liability to breakdown. The turbine is highly economical, the gear absorbs little power (about two per cent.), and the propeller can be run at its maximum efficiency.

This high efficiency leads to diminished weight in boilers, and this will allow of a decrease in the weight of fuel for a given voyage. These savings may be utilised to carry more deadweight on a given displacement, or to decrease the displacement for a given tonnage carried, all of which tends to economy.

The weight of machinery of a naval vessel of 40,000 S.H.P. which came under my notice lately, may be of interest. With steam up in the boilers (water tube, oil fired) and including all auxiliaries, stores, etc., but excluding the fuel oil, the total weight amount to 920 tons only, which gives a weight of $51\frac{1}{2}$ lbs. per S.H.P., a combination of weight and H.P. hitherto undreamt of.

From these results one might say with truth that the mechanically geared turbine along with oil fuel fired boilers is the marine prime mover of the moment, and likely to oust all competitors.

The foregoing is "Gearing as applied to the Marine Steam Turbine" as I know it, and I am only sorry that I cannot write on the subject from a practical point of view—I mean from experience gained by actual seagoing conditions. I have had to look at it from the draughtsman's standpoint, and in bringing this part of the paper to a close I am conscious of my limitations, and of my inability to deal with it as fully as it deserves.

The advent of the geared turbine marks a new epoch in Marine Engineering, and as it goes far into one of the most important questions of the day, namely, the conserving of our natural resources, I do not think it could be considered inconsistent with the preceding part of this paper if I add a few remarks on fuel consumption, and draw a comparison between the geared turbine and its predecessors, or even with its contemporary, the Diesel engine.

When one considers that the burning of 1lb. of Welsh coal per hour generates sufficient heat for the evaporation of about $15\frac{1}{2}$ lbs. of water, which is equal to 15,000 units of heat, or 11,580,000 ft. lbs. per hour or nearly 6 H.P., the loss of heat energy will be better understood, and one can only come to the conclusion, that there is room for improvement.

As even the best of reciprocating engines requires about $1\frac{1}{2}$ lbs. of coal per I.H.P. per hour, it is clear that almost 90 per cent. of this available heat is wasted in that type of engine. Compared with this, the geared turbine consumes about $1\frac{1}{4}$ lbs.

per S.H.P. per hour, but before a proper comparison can be made between these, some idea of the relation between I.H.P. and S.H.P. must be arrived at.

It need scarcely be said that I.H.P. is the gross power obtained without allowing for power absorbed by friction, and S.H.P. is the power transmitted to the shaft after overcoming friction. As it is generally taken that about 15 per cent. of the I.H.P. is required to overcome the friction, in a reciprocating engine, leaving only about 85 per cent. of the power to be transmitted to the shaft, then the $1\frac{1}{2}$ lbs. per I.H.P. corresponds to about $1\frac{3}{4}$ lbs. per S.H.P., which shows a saving to the credit of the geared turbine of at least 25 per cent.

It will be seen therefore from the point of view of fuel economy, that the geared turbine is a very considerable improvement on its predecessor. Where oil is used instead of coal the consumption of oil is about 1lb. per S.H.P. per hour, which is a further saving still.

Or let us consider the matter from the steam consumption side of the question. The ordinary high pressure triple expansion engine requires in the neighbourhood of 16 lbs. of steam per I.H.P. per hour = to about $18\frac{1}{2}$ lbs. per S.H.P. per hour, the direct coupled turbine requires about 13 lbs. per S.H.P. per hour, the geared turbine about 11 lbs. per S.H.P. per hour. Expressed in adiabatic efficiencies these would be 36 per cent., 51 per cent., and 61 per cent. respectively.

The mention of fuel oil naturally makes one think of the Diesel engine, and one must confess at the outset that from the consumption per S.H.P. or B.H.P. point of view, the Diesel engine compares very favourably even with the geared turbine, the consumption being only about ·47 lb. per B.H.P. per hour.

The greatest loss of heat energy in the steam engine is of course in the condenser, and this is where the Diesel engine has the advantage, developing as it does its power from the fuel directly in the cylinder.

But the Diesel engine has not yet been brought to that state of perfection where the weight of the machinery works out at about 50 lbs. per B.H.P.—in its case it is in the region of 200 lbs. per B.H.P. for mercantile vessels—and I am sure you will all agree that where large powers are required the geared turbine either in single or double reduction is an easy winner.

With superheated steam again coming into general use in combination with geared turbines, with the consequent economy in fuel and water consumption, a great attraction is held out to builders and owners to fit their vessels with this system, and this is apparent by the number of vessels on order being thus equipped. It will be interesting to watch their performances on service, and if the results of these were published the information would be useful and instructive to one and all of us. It may be interesting in passing to observe that there is a saving in consumption of 1 per. cent. for every 10° of superheat. This is a generally accepted figure up to about 120° of superheat, when the saving gets gradually less. At 200° superheat the saving is about 18 per cent.

The CHAIRMAN: Pitting of the teeth of the gears has been mentioned, and Mr. Houston attributes this to intense pressure on the hard points causing the lamination of the material. He may be right but there should not be any hard points. Later on he draws attention to the fact that the material should be homogeneous, and that pitting may be due to the lack of this.

The pitting of gears I have seen was on every tooth for their whole length, and I am inclined to think air bubbles in the oil were the cause of the pitting, but it did not appear to me to be a very serious trouble, in fact one might look upon a moderate number of pit holes as being receptacles to hold oil.

The power, when using the Fottinger Transformer, is transmitted through water (which, by the way, becomes heated, and the heat is transferred to the feed water) and with mechanical gears we should endeavour to get as much of the load carried by the oil as possible and thus relieve the metal.

Mr. Houston's figures of comparison between geared turbine and Diesel engines do not go far enough and are hardly fair to the Diesel engine. When comparisons are made one has to bear in mind weights of fuel carried, as the efficiency of a ship is to leave a port with the maximum amount of cargo.

Let us take Mr. Houston's figures for two vessels of equal dimensions, one geared turbine, one Diesel, each with 3,000 b.h.p.

The weights of machinery will be:—

G.T. $= 3{,}000 \times 50 = 150{,}000$ lbs.
Diesel $= 3{,}000 \times 200 = 600{,}000$ lbs.

The fuel consumption will be:—

G.T. $= 3{,}000 \times 1{\cdot}25 = 3{,}750$ lbs. per hour
Diesel $= 3{,}000 \times {\cdot}47 = 1{,}410$,, ,,

And if the ship is say 500 hours on the voyage, the total fuel burnt will be:—

$$\text{G.T.} = 3{,}750 \times 500 = 1{,}875{,}000$$
$$\text{Diesel} = 1{,}410 \times 500 = 705{,}000$$

so that the ships would leave port with:—

G.T. Weight of machinery	... ...	150,000 lbs.
G.T. weight of fuel	... ...	1,875,000 ,,
	Total ...	2,025,000 ,,
Diesel weight of machinery	... ...	600,000 ,,
Diesel weight of fuel	... ...	705,000 ,,
	Total ...	1,305,000 ,,

or a saving in favour of the Diesel engine of 720,000 lbs. or 321 tons.

We are indebted to Mr. Houston for his paper, and we shall be glad to have criticisms from our seagoing members as they can very probably enlighten us on many of the points raised.

Mr. J. SHANKS: In order to save valuable time in discussing this excellent paper, I may start the ball rolling in the hope that members present who have experience with geared turbines will give us the benefit of their knowledge and thereby add to the value of the paper. It was a happy thought of our Hon. Secretary to arrange the reading of this paper to-night in view of the visit of members to the Power Plant Works at West Drayton on Saturday next, where they will have an opportunity of seeing the most modern methods of gear cutting.

Personally, I have no experience with geared turbines, but the rapidity with which gearing has been successfully applied in marine propulsion within recent years has made it apparent to all marine engineers that it is a subject they must become thoroughly conversant with. The author has drawn our attention to the great importance of accuracy in the pitching of the gear in order to ensure successful results and even then a slight end play must be allowed on the pinion shaft to make alignment perfect, and this of course necessitates the introduction of a flexible coupling between the pinion shaft and the turbine. I should like the author, or any member present, to give us their experience as to tear and wear on the gearing itself and the flexible coupling. Personally, I have heard most contradictory reports, some describing tear and wear to such an extent as to cause serious trouble within a year or two, and others that there was no perceptible wear after years of work; probably both are true. But no doubt, with more experience and improvements in gearing, present difficulties, where they occur, will be overcome. Up to the present the turbine has only been applied to ships with speeds over 15 knots, but with the introduction of a reliable double reduction gear having a ratio of 45 or 50 to 1, I can foresee a great future for their adoption even in tramp steamers.

The author has drawn our attention to the gain in weight of machinery and economy in fuel as compared with the reciprocating engine, and few, I think, will seriously dispute his figures; but first of all we must have reliability; when that is assured, the coming of the turbine with double reduction gear in all class of ships will follow. I beg to thank the author for his valuable paper and look forward to an interesting and instructive discussion.

Mr. R. Balfour: Some of us have been anxiously waiting for a paper on the subject so ably presented by Mr. Houston to-night. At no time in our history has gearing claimed our attention more than at present, especially in connection with the advent of steam turbine power.

My experience has been confined to single reduction gear in land installations requiring only moderate speeds; still it has been sufficient to convince me that extraordinary care and accuracy are needed to ensure success. Assuming that the materials and workmanship are of the best, and the adjustments are mathematically correct before leaving the factory, we are yet confronted with serious trouble as regards the durability of the teeth. In my opinion some of the causes of failure are due to rough pitch, excessive tooth pressure, want of definite forced lubrication at point of meshing, quality and temperature of oil, and inaccuracy of adjustment of pinion shafts in relation to wheel shafts whether with floating or rigid gears.

I merely got up to congratulate my colleague for his able and interesting paper, and I think the thanks of the members are due to the Committee of Lloyd's Register for again allowing one of their staff to come and give us the benefit of his knowledge and experience.

Mr. Calderwood (Power Plant Company): No mention has been made of the gear shaping process of generating teeth. This method compares favourably with the hobbing method, and up to certain sizes seems likely to supersede the hob, and it is worthy of being noted in a discussion on the subject introduced by the author. No details or description of the methods used to determine the amount of error in the teeth, either for pitch, shape or lead are given, these are important and of interest to engineers. It may be doubted as to the correctness of the remark that pinions are usually made of nickel steel and oil hardened, owing to the extreme difficulty of preserving the teeth from warping during hardening. My experience is that pinions are made of nickel steel, heat treated to toughen them before cutting the teeth. Mention had been made that the pinion has lateral play when under load. It seems to me that this can only arise in the case of faulty cutting where the plane containing the apices of the teeth is not at right angles to the shaft axis. If the cutting and helices are accurate the pinion should remain without movement. Mention has been made of the floating frame for pinions; I presume that the Melville-MacAlpine system has been in view, and I venture to think that this system is somewhat costly to instal, and intricate in operation. Our Company is introducing a spring bearing of simple design, of which the same results as were obtained by the floating frame are expected. Such systems demanded flexible couplings, and if I might reply to a previous remark in the discussion I do not know of a failure with the claw type flexible coupling on shipboard. As it is the intention of the members to visit the Power Plant Company's Works on Saturday, I hope to show to the visitors gearing of the double reduction type under construction (as shown on the blue print handed round by the author) and also the gear shaping process in operation.

Mr. J. Thom: The application of gearing for large powers is comparatively a new industry, at least where sizes have to be kept down as much as possible owing to confined space on board ship. The deep cut gearing requires large diameter wheels to give the best results, the opposite of what is possible on ship board.

The smaller the teeth, which will give the strength necessary, the more satisfactory the gearing will run. I am of opinion when hardening or toughening the teeth in large wheels has been perfected the life of the wheels will be greatly increased, even so to harden the pinions alone would give much improved results.

Lubrication under pressure is absolutely necessary after the oil has been cooled. We are looking forward to our proposed visit to West Drayton where doubtless we will see this kind of work carried out with mathematical precission.

Mr. W. McLaren: The author brought forward a paper on Gearing as applied to Marine Steam Turbines, which is certainly of special interest to Marine Engineers. But I would beg to remind him that, although Marine Engineers, as a class, have not had actual trade experience with heavy electrical power units, they have not failed to overcome the running and maintenance of many complicated machines on board ship, electrical or otherwise.

We had a valuable paper read on electrical ship propulsion by one of our members, Mr. Durtnall, several years ago, and I hope and trust it will be found as time goes on the Marine Engineer is not found wanting, for his opportunities, considering the special gearing dealt with by the author as pinion and spur gears. I should like to know which side of the gear he recommends, the flexible couplings, that is the turbine side, or the propeller shaft side, as I understand the flexible coupling is close on the gears and would say the rigid gear with three bearings where there is a right and left-hand and helices, pair of pinions, but if in one pinion, certainly a bearing each side of pinion, and when I say rigid I mean that the pinion and spur wheels are in one independent frame or bedplate, with adjustment for alignment of pinion and spur bearings when wear takes place. Having since been privileged to visit the Power Plant Company's Works and seen their production to the methods they apply to actual gear cutting for heavy power units, land and marine purposes, I cannot see the need to introduce a floating gear pinion especially with a helical cut gear as any allowance for a floating action must be in the flexible coupling on the turbine side, as end play is not desirable with a turbine rotor, yet has no disadvantage on the propeller or thrust shaft side. I cannot agree with the author in his fuel or steam consumption with the turbine engine, $15\frac{1}{2}$ lbs. water per 1 lb. of Welsh coal is more theoretical than practical. I may mention we have had Durham coal with as high a calorific value as Welsh coal. Has this steam consumption been corrected by the weight of condensed water? and also is the consumption of steam and weight of auxiliary machinery taken into account that is necessary for the circulating and vacuum pumps with Naval and Mercantile Marine practice?

Mr. Houston: I think Mr. Shanks set the ball a rolling, so I shall begin my reply by discussing the various points he raised, and then proceed with the questions of the other gentlemen who have spoken. Mr. Shanks asked about the fore and aft allowance of movement in the pinion shafts; what wear there is on the teeth, and also what wear, if any, takes place in the flexible couplings. I think I made it clear that the fore and aft movement given to the pinion shafts is to allow the pinions to accommodate themselves to the opposite angle of helix of the gear wheels, but this allowance is a mere fraction about ·020 to ·030. This movement also takes up slight inaccuracies in the cutting of the teeth, mentioned by Mr. Shanks. The wear that takes place on the teeth is negligible if the oil supply is kept, up, and with a properly designed flexible coupling of the claw pattern, there should be little or no wear. Mr. Shanks enquired as to whether double reduction gears were being fitted to tramp steamers. In my experience I have only come in contact with single reduction gears for low powered vessels, but it is quite within the bounds of possibility that double reduction gears will be fitted into tramp steamers just as they are at present being fitted into liners, as the saving in weight and in consumption is very considerable. I recently saw a design for a 750 S.H.P. double reduction set, which was got out for demonstration purposes, and the weight of machinery and the room it occupied were almost absurdly small as compared with a reciprocating engine.

The gentleman from the Power Plant Co., had some very interesting remarks and queries to make, and in view of the proposed visit to this Company's works on Saturday his presence here to-night is very opportune. He raises the point of the generation of the teeth by the shaping process, as against the "hobbing" method. I've no doubt the shaping process gives satisfactory results, or it would not be employed, but, as I have not seen the operation I am not in a position to compare the two methods. The method of testing the accuracy of the gear cutting could be better explained by practical demonstration than by description, but I shall endeavour to give some idea of how it is done. The pinions to be gauged are set up in V blocks on a surface table, and two points are selected, say 20 pitches or so apart, on the teeth, and on the line of the axis of the shaft. These points can then be measured by a Vernier gauge, and the distance apart compared with the measurement made up by the axial pitch of one tooth multiplied by the number of pitches taken. In the case of the pitch of the teeth which we have had before us, viz.: ·5833 normal and ·8159 circumferentially, the axial pitch works out at 1·1429in. This multiplied by say twenty pitches give a measurement to compare with a Vernier reading over twenty pitches, and the error, if any, can then be seen and recorded. It was interesting to hear about the Power Plant Company's spring bearings and to learn that they give satisfaction. This idea adds to the different designs already quoted; the rigid which we have considered, and the floating frame which was mentioned.

Mr. Newton and Mr. Thom both enquired about the oil hardening or toughening of the nickel steel pinions, and were of opinion that any heat treatment would cause distortion of the teeth. I am in agreement with them on that point, but the nickel shafts and pinions are treated when rough turned and machined almost to finished sizes, and before the actual cutting or shaping of the teeth takes place. Mr. Thom also enquired about the amount of end play on the pinion shaft. This question was answered in my reply to Mr. Shanks.

Mr. Maclaren raised the point as to the advisability of fitting a flexible coupling between the main gear shaft and the thrust shaft, but I have never heard this suggestion being put forward as necessary. The flexible couplings between the turbines and the pinion shafts are fitted primarily to allow of the slight fore and aft movement of the pinion shafts, whereas there is no necessity for any movement in a fore and aft direction of the main line shafting. Rather, the main line shafting and the main gear-wheel shaft should be so lined off and coupled together to ensure perfect rigidity throughout. He also rather doubted the consumption of 11 lbs. per S.H.P. as compared with 18½ lbs. for reciprocating engines, but that is an actual figure obtained on the full power trial of a 40,000 S.H.P. naval vessel. In fact it is an "all on" figure, including auxiliaries, the turbine consumption was about 9 lbs.

To those of us who have been writing and speaking on this subject from the theoretical side, it was interesting to have the remarks of the gentleman who had been at sea in a geared turbine vessel, and who could speak with first-hand knowledge. He puts forward the suggestion that the pitting of teeth sometimes met with, is due to the presence of salt water in the oil, and he explained that the oil is often collected from the bilges and separated, and a certain amount of salt water goes through with it. If this is actually done it is a most reprehensible practice in my opinion, as in most instances of break downs of the gearing the cause can usually be traced to a defect in the oil supply system, or in the class of oil used. It is probably not too much to say that the maintainance of a copious supply of oil on the gearing is the most important matter necessary to the successful operation of a marine geared-turbine installation.

In his remarks he stated that the temperature of the oil was kept at about 110°, which would be quite satisfactory, but even this temperature shows the necessity of passing the oil through oil coolers.

Mr. Balfour was of the opinion that the wear on the teeth and on the flexible couplings was due to the load not being constant or steady, and he attributed this to the number of blades on the propeller. Personally I do not see that this has much bearing on the point, but I am in agreement with him on the desirability of having a steady load. Shocks or a repetition of shocks, as one would get in an uneven load, will always bring wear in their train, but I contend that under normal running conditions the load is fairly constant. This brings us to a point raised by Mr. Corns when he enquired if any bad effects were met with, due to the engines racing.

It was my good fortune to be on board the SS. *Vespasian* on several occasions not very long after she was converted with geared turbines, and whilst she was trading across the North Sea in all weathers between the Tyne and Hamburg. The engineers reported that they did not know they had engines on board in heavy weather; rather a difference from the hair-raising experiences some of us no doubt have had with reciprocating engines. I can corroborate Mr. Corns's statement that the machinery of the SS. *Vespasian* is now in a tramp steamer called *Lord Byron*, as this vessel was recently in Barrow, when I took the opportunity of calling on board knowing that the transfer of machinery had taken place. I was informed that it was still going strong.

In reply to Mr. Beckett, the turbine itself is prevented from fore and aft motion, thereby bringing the rotary and fixed blading into contact, by fitting a small thrust block, usually nowadays of the Michel type, on the rotar shaft. The main line shafting still requires of course the thrust block in the usual way. The main line shafting may be slightly altered from true alignment owing to the method of loading the vessel, but this slight alteration in the shape of the vessel should not affect the gear case which, in a fore and aft direction occupies comparatively little space.

With reference to our Chairman's remarks, the only vessel which I know of fitted with a "Fottinger" transmission gear, was a little vessel called *Holzapfel I.* Her motive power was a suction gas engine, but I do not think it gave satisfaction, and I understand it ultimately gave way to a steam engine, and I suppose the "Fottinger" transmission gear would go with it. It is difficult to say off-hand as to the amount of wear permissible on gear teeth; one would have to consider each individual case on its merits. As to the method of lubricating the gear, in my opinion the best method is to supply the oil under pressure, through nozzles or sprayers distributed along the width of tooth face, and immediately at the meshing of the gear and pinion. Mr. Fielden advances the theory that pitting of teeth may be due to air, but I am inclined to keep to the opinions expressed in the paper, as air is present in all gear cases, but pitting does not take place in every instance. The reason for employing harder steels in the pinions than in the gear wheels is because of the heavier work coming on the pinions due to the difference in speeds and diameters.

I am in agreement with Mr. Fielden in his remarks about the Diesel engine, but you will notice that I exclude the fuel oil from the weights, and I distinctly say where *large* powers are required. The highest powered Diesel engined vessel that I have heard of is in the region of 6,000 B.H.P., so I still contend that the geared turbine is the winner. I think these remarks of mine deal with the various points raised, and I am glad we have had such a good discussion.

Capt. P. T. Brown (By correspondence): The subject of geared turbines is so important that I think Mr. Houston has performed a splendid service in bringing it before the Institute, and I hope, with him, that the discussion may be the means of inducing those in the know to impart to us a little of their experience.

In regard to the hydraulic system I trust some member may be able to give some detailed information. Previous hydraulic systems of propulsion, *e.g.*, H.M.S. *Waterwitch*, were not successes, but I am of opinion that a useful turbo-hydraulic system would possess certain peculiar advantages over mechanical gearing. Weight would probably be reduced, and the simplicity in construction of water wheels as compared with gear-wheels would help from the cost point of view. Running costs —in view of the lubrication saving—would also probably be considerably less.

Electrical systems for mercantile service are, I fear, without the range of practical consideration. Without doubt successful running would entail the necessity of carrying electrical as well as steam engineers, and we all know the view a ship-owner takes when increase in personnel is asked.

The author's descriptive matter of mechanical gearing leaves little room for criticism. He has expounded the fundamental principles of design most clearly, but I should have liked a little more about the lubrication system and also about the oil. The noise made by gearing is a very important consideration in passenger ships. I have been told that the engine-room of the *Vespasian* was not exactly quiet, in fact the reverse. Some 5 or 6 years ago I surveyed, during construction, the machinery of three geared turbine ships. The noise in the engine-room on the trial trip of the first was very great. Experimenting with oil gave splendid results, and the noise on the trials of the third ship was no greater than one would get in an ordinary turbine engine-room.

I should be glad if Mr. Houston would confirm his figure for the limiting speed at P.C.D. of the gear-wheel. 140 ft. per second seems to be rather high. Mr. Houston's theory as to the pitting of teeth is very interesting, but in searching for an explanation of the phenomenon I would rather he had eliminated the possible chemical effect of the oil.

I am sorry to disagree with anything in a most valuable paper, but I fear the author's figures for consumption of other types of propelling machinery are rather higher than is found in really good practice. There are many reciprocating jobs running on less than $1\frac{1}{2}$ lbs. per I.H.P. hour, and in regard to the use of an exhaust L.P. turbine I know a case where, for all purposes, on a round voyage the consumption worked out at not far from 1 lb. per S.H.P. hour. Large sets of marine Diesel engines in my experience use only ·39 lb. per B.H.P. hour.

I would especially congratulate the author for his discrimination in pointing out that to effect economy is to "conserve our natural resources." That is the real spirit. Saving of fuel is of more importance to us as a nation than the saving of a shipowner's coal bill. But our problem needs to be analysed further. We must simplify the means by which we effect the saving and my own idea is that we shall only make *real* progress when we cut out links in the chain from fuel to propulsive effort. The geared turbine introduces a link, the oil engine cuts one out.

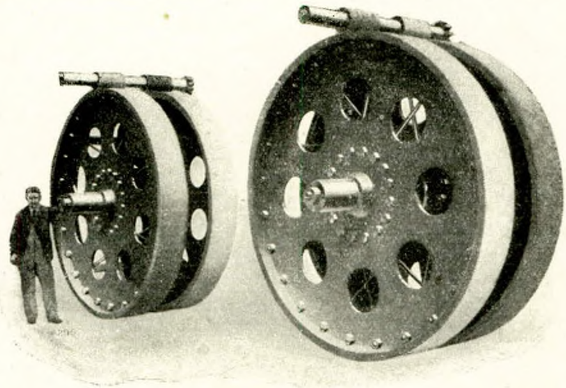
Workmanship in helical gearing, as the author points out, must be of the highest order, and the extension of this system will, I hope, induce marine engine builders to adopt better machine and workshop practice. The almost insane competition amongst marine engine builders and others recently has led to the introduction of rough methods and practice which needs reform. To carry the quoted metaphor still further one cannot construct an A1 machine by C3 methods.

In conclusion, I would like to congratulate the author on his paper and to thank him for it.

Visit to Power Plant Works.

A visit was paid to the Power Plant Works on the 17th May, when we had an excellent opportunity of seeing gearing in the process of making and of completion. We were very cordially received by the Managing Director, who, with his staff, kindly devoted the afternoon to show the visitors through the works and explain the details of the different processes and the machines used to accomplish the accurate finish requisite for smooth working and absence of noise associated with spur gearing. After inspecting the various departments and the machines at work cutting the gearing on pinions and wheels, the Michel thrust block, with the details of its construction exposed to view, was examined with interest. A plant was then set to work by electric motor to illustrate the advantage gained by the systematically cut gearing for reducing speed, with the result that smooth and noiseless working was demonstrated.

The methods of gear cutting which were shown included several processes, all being of considerable interest. The end milling process, where a small milling cutter having the shape of a space between two teeth, cuts out the spaces and leaves the teeth in correct profile on the blank. The machines on which this type of cutting is performed is arranged to cut straight, double helical or triple helical teeth in ordinary spur wheels



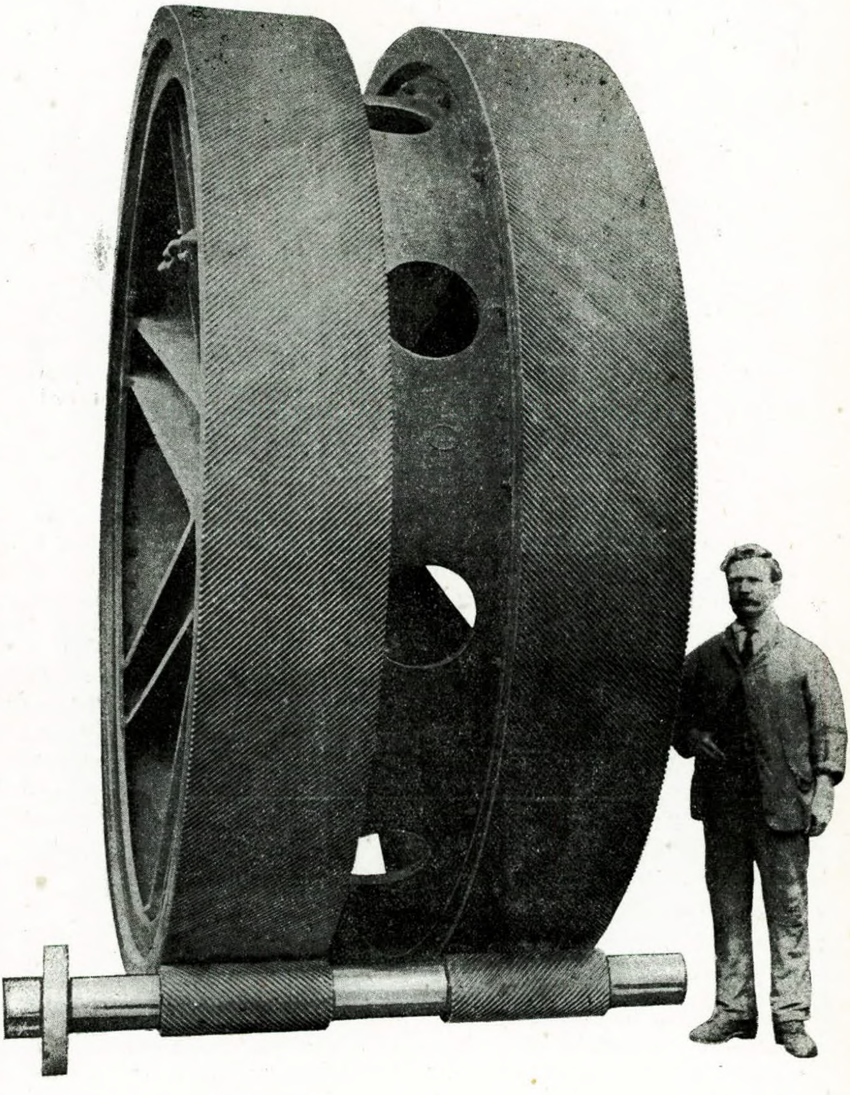
or in bevels of any angle. The machines were seen cutting special double helical bevels and a very large spur wheel, all with double helical chevron type teeth.

The hobbing process is on the Wust principle, which was the first system of cutting double helical gears on a true generating process from a solid blank. This system was introduced into this country by the Power Plant Co. fourteen years ago, and at that time they were the only double helical gear manufacturers in the world outside of Switzerland. A number of these machines were seen working with wheels from a comparatively small diameter up to 12 ft. in the process of being cut.

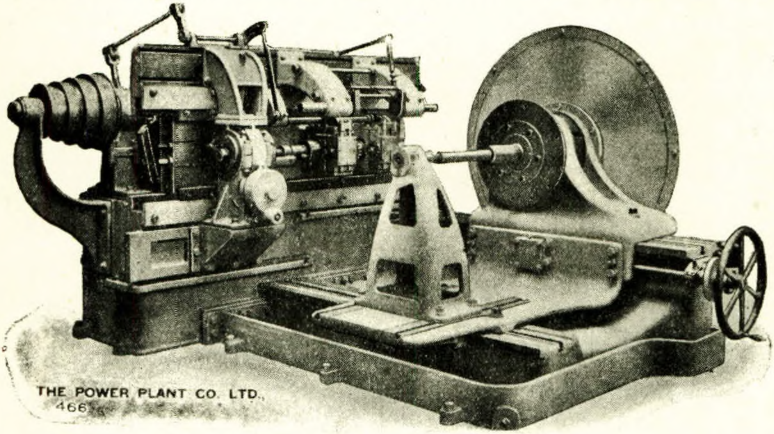
The Sykes' Shaping Process. This is carried out on a machine designed and patented by Mr. W. E. Sykes of the Power Plant Company, and ordinary straight spur wheels and double helical wheels were shown in the process of cutting. As this process is comparatively new, considerable interest was evinced in regard to it. The system is being rapidly developed by the Company to take the largest size of turbine reduction gears. The first reduction wheel for a 3,000 H.P. set of gears was being cut in the course of our visit.

The various methods of producing the tools used in gear cutting, and also a large number of finished wheels, gears in cases, were shown and explained. The pattern shop and its appliances with the interesting patterns and core boxes were viewed and examined with appreciation, reminiscent to some of the visitors of their early experience.

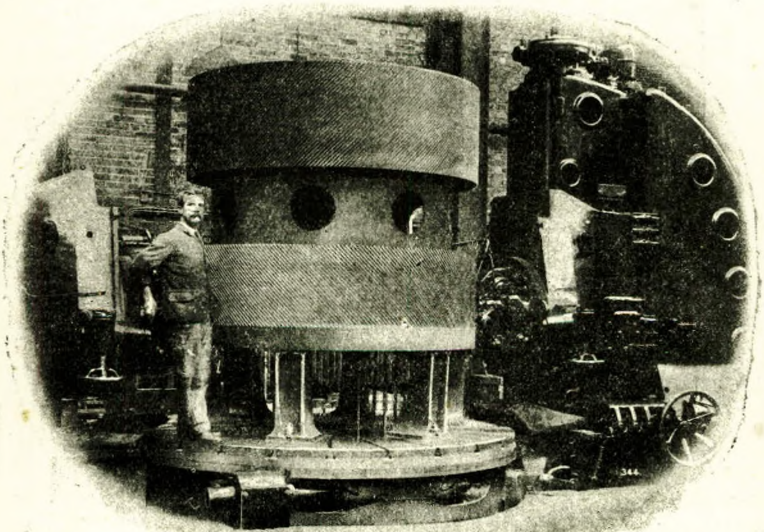
After examining the works, we assembled around a hospitable table and enjoyed the repast provided by the Company for their visitors. A vote of thanks to our hosts, the Managing Director, Mr. R. J. McLeod, and his staff, who had proved such kindly conductors, was proposed by Mr. B. P. Fielden,



and seconded by Mr. George Adams in terms which were fully endorsed and expressive of the educational value attaching to



the visit, and the lessons conveyed to the minds of all who had been able to avail themselves of the opportunity afforded, and of thanks for the cordial reception extended to all.



The Managing Director, Mr. R. J. McLeod, replied on behalf of the Company and staff, and the visit terminated amid congratulations about 6 p.m.

Notes.

The following article on oil engines and description of an oil-fuel sprayer appeared in *The Shipbuilding and Shipping Record* of August 14th, and are reproduced by kind permission:—

HIGH-POWERED OIL ENGINES FOR SHIP PROPULSION. — The exceptional fuel economy of the Diesel motor has, in a great measure, been the cause of its adoption as a ship's engine, but expenditure in other directions has sometimes proved excessive. Economical consumption of fuel, important though it be, is not everything, and that benefit must be supported by other advantages, *e.g.*, reasonable first cost and general maintenance charges, safety of operation, and robust construction throughout the fixed and moving parts. Yet, while it cannot be contended that such desiderata have invariably been forthcoming, the Diesel motor as a prime mover in the sphere of ship propulsion enjoys a position of constantly extending strength and utility. It is now opportune for the British engineer to enter upon the construction of high pressure heavy-oil marine engines without doubt of success, provided he be resolved to become associated only with a first-rate design.

Since the early Diesel motors were made for industrial purposes on land, it is advisable that the newcomer should approach the matter from a different standpoint, profiting as far as possible by the experience of his predecessors. For, whereas a sound marine model of the high-pressure type is readily adaptable to land purposes, the translation of an industrial engine into one for ship propulsion is unlikely to prove satisfactory. In a modern marine engine, the greatest reliability being a *sine quâ non*, the components must be of substantial design and as simple as possible. Ease of approach to the entire machine is required, as are the utmost facilities for taking down those parts subject to wear and misplacement. In this connection, particularly, too much attention can scarcely be paid to the design of the fuel-nozzle and to all other valves. The slightest derangement of the fuel-feeding apparatus, which is so delicate, will render the whole engine liable to be thrown out of gear. Restriction must be placed upon any tendency to superfluous weight or overall measurements, and the various parts must be few in number, while ensuring real interchangeability, easy, rapid renewal, and inexpensive repair. A low consumption of both fuel and lubricant have to be striven for—the first point will be found far easier to accomplish than the second—and, moreover, the motor must be made so as to be capable of turning at slow revolutions, and also possessing rapidity of reversing and of manœuvring.

It is not commonly known outside the circles of engine-room staffs that with this type of prime mover trouble with the valves is of somewhat frequent occurrence: the exceedingly high pressures and temperatures inherent in Diesel engines tend to militate against the completely satisfactory use of mushroom poppet valves. Hence, there is considerable scope for the development of a balanced slide-valve mechanism. The adaptation of a piston or tube valve should not prove insuperable, if it were arranged to slide in a well-lubricated and water-cooled chest. Something of the kind was attempted both in England and Germany before the war.

In the larger sizes a material augmentation of both the dimensions and inertia forces become evident, large and massive framing consequently being demanded and, owing to the necessarily wide range of torque, it is difficult to keep the normal revolutions low. The cylinder then tends to become too cool for efficiency, resulting in missfiring. However, by the adoption of six or more cylinders and corresponding cranks, lower speeds are possible, but this method is open to the objection that it increases dimensions, weights, working parts, and running costs, not to mention the initial outlay.

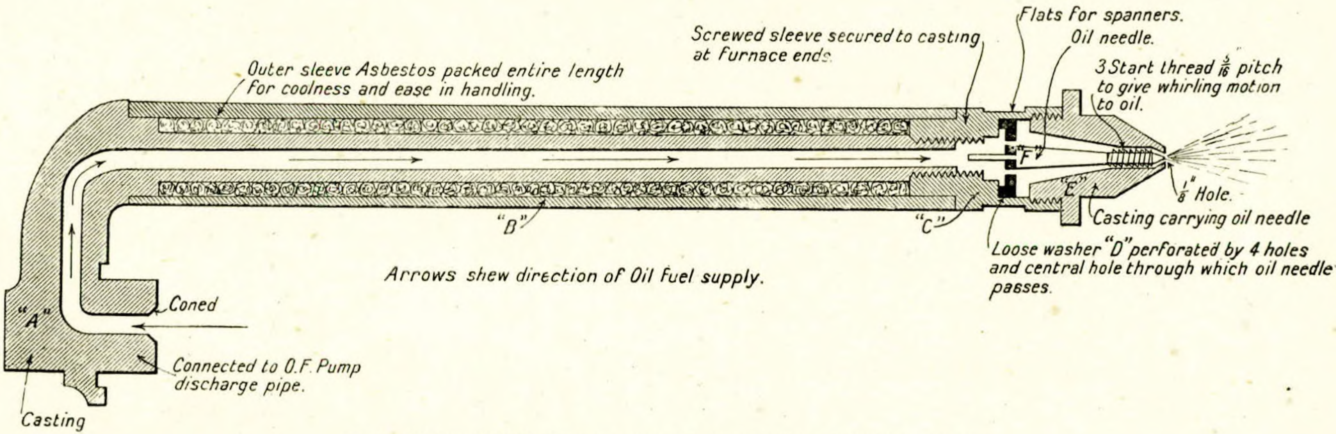
Having due regard to such considerations, it is not surprising that many advocate the employment of the two-stroke Diesel engine for marine work. It is claimed that the greater average pressures in the Diesel engine afford a marked restriction of crank-shaft speed without appreciable loss of turning moment. As a result the frame is relatively of light weight and the overall dimensions are decreased. On the other hand, an air-scavenging pump of large capacity is a necessary accompaniment, while the quantity of fuel burned exceeds that of a four-cycle engine by some 25 per cent. at the least. Experience has shown that this defect arises from incomplete combustion following upon the effect of insufficient air-scavenging. Where some of the exhaust remains in the cylinder the proper volume of uncontaminated air is unable to enter, and thus we get a rise of temperature and delayed combustion, which produces a perceptible fall in the average effective pressure.

Now, seeing that whether their action is rapid or tardy, the area of cam-lifted valves is small, the volume of the air-scavenging by this means is inadequate, nor is it diffused to the required extent. Instead of swift and complete expulsion of the exhaust, both the discharge and dilution effects are meagre, which means a lean charge for the power stroke and late combustion occasioned by the high temperature of the cylinder. It is plain that the solution of the difficulty rests to a great degree in the air-scavenging valve, which should possess a much larger area than has heretofore been provided.

Another characteristic, the significance of which should be realised fully, is the subject of lubrication. In any Diesel-type motor this must be carried out carefully in order to obtain not only satisfactory performance, but safety. It is not generally recognised that in all engines of this type there exists the risk of the lubricating oil working past the piston in greater quantity than is required to promote lubrication. When this occurs the oil mixes with the highly-compressed air, causing premature combustion. Accordingly, in the endeavour to obviate this drawback, a very thorough method of correctly lubricating the piston is absolutely essential. Then, too, the air-compressors, both for starting and feeding the cylinders when running, call for specialised experience in their design, for upon their efficiency of the Diesel engine largely rests, so that the type chosen should act on the multi-stage principle and be thoroughly water-cooled between each stage.

Reverting to the important question of the large two-stroke engine, an eminent authority has lately declared that (1) the working stroke should be allowed longer expansion before release; (2) the scavenger air should be strong and variable in volume (either by forcing-in or by the introduction of means for scavenging by exhaustion); and (3) not before the return or inward stroke should a substantial charge of undiluted air be admitted to the cylinder rapidly under strong pressure. Those who are conversant with the practice of Diesel engineering will appreciate that the conditions laid down present many interesting problems.

Oil-Fuel Sprayer as Fitted in German Torpedo-Boats.—We show herewith the design of an oil-fuel sprayer as fitted in German torpedo-boats. The fitting is very simple, and admits of no adjustment of spray orifice after connecting up, as is now done in the best British practice. The device consists of a solid iron casting A, connected to the oil-fuel pump discharge pipe at one end, the other end terminating in a long tube with a fine thread at the end. Over this end a tube B is fitted, the space between being packed with asbestos, as shown on the sketch.



Sectional View of Oil-Fuel Sprayer as used in German Torpedo-Boat Destroyers.

This loose tube is secured by means of the screwed portion C. A loose washer D fits into a recess in C. This washer carries the spindle of the oil needle, and, in addition, is perforated by four equidistant holes. The end portion E screws into C, and is shaped as shown, the orifice through which the oil is pumped into the furnace being $\frac{1}{8}$in. in diameter. When E is screwed home there is very little play for the needle F, which is shaped as shown, the outer collar resting against the loose washer D. The parallel portion of the needle is provided with a three-start thread, 5-16in. pitch, which gives the necessary whirling motion to the oil and assures the breaking up of the particles for efficient combustion.

The air cones used are unlike those employed in British practice, in which the air slots are cut round the circumference of the cones. In the German practice two cones are fitted, one inside the other, and each have slots cut, and the air can be adjusted by rotation of the outer cone.

Coal Production.—Owing to the great interest taken in the subject of coal, dealt with in our June and July issues, we have been in communication with the City and County of Norwich Librarian, who has devoted special attention to books and articles on coal, and has compiled a list of books bearing on the subject in the *Readers' Guide*, published by the Library Committee at 1d.; so that any members who wish to study the subject further, including the geological formation, may examine the list and note the books they may wish to read.

Election of Members.

Members elected at a meeting of the Council held on Tuesday, 2nd September, 1919:—

As Members.

Fredk. George Archbold, c/o Lloyd's Register of Shipping, 167, Yamashita Cho., Yokohama, Japan.

Colin Wilson Bain, "Woodbine," Free School Lane, Dunfermline.

Arthur Charles Bowden, Lloyd's Register of Shipping, Waterloo Mansions, Bombay.

David Bramah, 2, Alexandra Road, Leyton.

Finlay Munro Curror, 21, Queen Street, Waterloo, Liverpool.
John Hall Davidson, 51, Cameron Street, Stonehaven, Kincardineshire.
Robert Cecil Dodgson, 2, Levenford Terrace, Dumbarton.
Alexander Ewing, Box 48, Yokohama, Japan.
Hugh Maclean Ferguson, 54, Randolph Gardens, Broomhill, Partick, Glasgow.
George Thomas Gillespy, "Kinvarra," Horley, Surrey.
John George Pattison Fowler, 183, Taylor Street, South Shields.
Joseph Hanna, "Channel View," Boscombe, Bournemouth.
Harry Hargreaves, 9, Sherfield Road, Grays, Essex.
George Herbert Montgomery Hutchinson, 27, Park Avenue, Hull.
Frank Riley Lindley, c/o Casebourne & Co., 78, Gracechurch Street, London.
William Harry Lloyd, 13, South Terrace, Aberystwyth.
Cecil John Mansfield Lowe, "Glenfauldt, Craigmore, Rothesay.
Herbert McLean, "Casilla," 239, Punta Arenas, Chile.
Wm. Geo. Matthias, 11, Yew Tree Road, Walton, Liverpool.
Cecil Bertram Mawer, "Toynton," 30, Victoria Park Road, Cardiff, W.
Fredk. Stanley Millward, "The Haven," Luton Avenue, Broadstairs, Kent.
Robt. Paterson Palmer, 74, Windsor Road, Forest Gate, E.
Alfred Norman Renton, c/o G. Swainston & Co., 116, Fenchurch Street, London, E.C.
Thos. B. Ross, 55, Queenswood Road, Forest Hill, S.E.23.
Edwin Shaw, Walford Trading Co., 72, Bute Street, Cardiff.
Owen George Smak (Engr. Lt.-Comdr. R.N.), H.M.S. *Serene*, G.P.O., London.
Percy C. Strother, 2, Bentinck Terrace, Newcastle-on-Tyne.
Wilfred Thos. Townend, "Harringay," Elmfield Road, Bromley, Kent.
Wilton Eldon Warren, "Los Pinos," Reading Road, Fleet, Hants.
Wm. Pritchard Watkins, 54, Roseneath Road, S.W.11.
Gavin, Watson, 9, Mansfield Road, Ilford, E.
John Keith Lindsay Woebling, 97, Highgate, Roslyn, Dunedin, N.Z.

Companion.

Robert Alexander, 87, Bishopsgate, E.C.2.

Associate-Members.

Hugh Fredk. Brown, 13, Lake House Road, Wanstead.
Charles James Hampshire, 43, Eleanor Road, West Ham, E.15.
John Mathias, 11, Yew Tree Road, Aintree, Liverpool.
Fredk. Arthur Newrick, 31, Scotland Green Road, Ponders End, Middlesex.

Associates.

William Boak (junr.), 270, Mansion House Chambers, Queen Victoria Street, E.C.
Fredk. Wm. Laverick, 46, Manchester Street, Southampton.
Percy Raymond Moore, 46, Pease Street, Eastbourne, Darlington.

Transferred from Associate-Member to Member.

G. V. Cole, Chichester Hospital, Horsham.
G. A. Roper, Heiston Road, Kelvingrove, Brisbane, Queensland.

From Associate to Associate-Member.

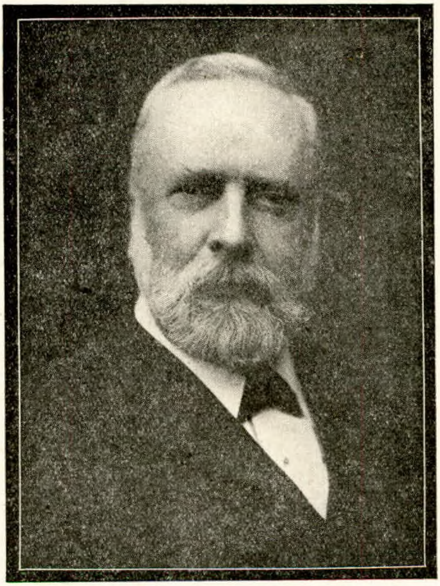
Percy Sampson, 33, Oakdale Road, Liverpool.

From Graduate to Associate-Member.

E. Banner, 30, High Street, Wimbledon Common, S.W.

Graduate to Associate.

J. L. Rutherford, 19, Claremont Road, Forest Gate, E.7.



JOHN INGLIS, LL.D. (Past President).

* By the death of John Inglis, LL.D., head of the shipbuilding and engineering firm of Messrs. A. and J. Inglis, Ltd., of Pointhouse Shipyard and Warroch Street Engine Works, Glasgow, which took place at the Ayr residence of his son, Mr. Quentin G. Inglis, on July 13th, the domain of British shipbuilding sustains the loss of one whose training and career as a shipbuilder over a period of half-a-century were distinguished by scientific method, prompting and guiding practical achievement. Dr. Inglis had all but completed his 77th year, and the interment of his remains in Glasgow Necropolis on July 16th took place on his 77th birthday.

Born in Glasgow in 1842, John Inglis, almost from his birth, was destined for the engineering profession, his father, Anthony Inglis, and uncle, John Inglis, having established themselves as marine engineers in Warroch Street Foundry in 1847. The firm's first important marine contract, obtained in 1850, was for the machinery for the Clyde Trustees' steam tug, *Clyde*, a vessel which, after $6\frac{1}{2}$ years of regular service on the river, with her original engines was only superseded in 1912. These engines having many notable features are still preserved, and may be seen on the river front at Renfrew, having been presented to the

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Burgh by Mr. William Brown, of Messrs. William Simons and Co., Ltd., whose father, Mr. Andrew Brown, was manager for the Inglis firm when they were constructed. In 1855 the firm built engines of 3,000 I.H.P. for the *Tasmanian*, one of the pioneers among screw steamers, and one of the largest and fastest vessels built up to that time. The success of the *Tasmanian* was such that the reputation of her engineers became firmly established, and other orders came along in quick succession.

In 1867 John Inglis, junr., then 25 years of age, took charge at Pointhouse. His education and training had been of the most thorough kind, both in the scientific and the practical side. He was educated at Glasgow Academy and at Glasgow University, entering the latter at 15 years of age, and having the advantage of completing his studies under such distinguished teachers as William Thomson (afterwards Lord Kelvin), Maquorn-Rankine, Ramsay and Blackburn. He greatly distinguished himself in mathematics, engineering and natural philosophy. His student days over, he was apprenticed in 1859 to practical marine engineering in Warroch Street Works, "serving his time" among the other youths who were graduating as journeymen engineers. While having great respect for the "rule of thumb" methods obtaining in the shops, especially so as he found that the science which he brought to bear on work done proved for most part the soundness of the practice obtaining, he yet resolutely set himself to welding the practical experience of the firm in designing engines and ships to the best scientific methods.

One of the first vessels built at Pointhouse was the *Erl King*, notable in the story of ocean navigation as the first ship to steam all the way from London to Shanghai. This, of course, was prior to the opening of the Suez Canal, and when steam communication with the East was unknown. Her arrival and performances caused a sensation in the Far East, and resulted in a long series of orders being sent to the Clyde for river and other steamers for Chinese waters. Ever since these early successes the firm have specialised in light craft paddle steamers for home, as well as foreign rivers. For service on the Irrawaddy in Burmah and for South American waters, a great number of elaborately equipped paddle steamers have been produced. Some of the fastest and best of the Clyde passenger steamers have emanated from Pointhouse, as have also channel and coasting steamers innumerable. Variety and individualism in products—with at the same time standardisation and repetition of types—have been distinguishing features of Pointhouse work all through. Steam and sailing yachts, river steamers, cargo steamers, coasters, liners, train ferries, dredgers, barges, etc., have followed each other, or been concurrently brought into being, on the stocks.

In the design and construction of all types of vessels, John Inglis' genius made itself particularly felt, but perhaps in no kind of work were his scientific methods more successfully applied than in the building of yachts—at first sailing craft and latterly steam. He himself was a keen yachtsman, and made yacht designing and building a hobby; the firm under his direction turning out many splendid steam yachts. The placing with them in March, 1905, of the contract for the royal yacht *Alexandra* was a tribute to the designing skill of Dr. Inglis, and to the excellence of Pointhouse workmanship.

That the scientific side of shipbuilding always received special attention from Dr. Inglis is evidenced by the close way in which his name is associated with a number of research subjects. He was the first shipbuilder on the Clyde to follow the practice of inclining vessels so as to ascertain their stability, and he was one of the earliest to apply the correct method of estimating, and allowing for, the longitudinal strain to which ocean steamers are subject to in service. Contemporaneously with his friend, Mr. William Denny, of Dumbarton, he adopted the practice of carefully testing new vessels, at progressive speeds, over the measured mile, and tabulating and analysing the data thus obtained. He interested himself also, although not to the same extent as Mr. Denny, in Dr. William Froude's methods of comparing results obtained from tank experiments, with models with the actual speed and other data resulting from trials of the vessels.

The honorary degree of LL.D.—Doctor of Laws—was conferred upon Mr. Inglis by the Glasgow University in 1898 in recognition of his scientific, commercial, literary and artistic attainments. But his sympathies were not bounded by his profession; he was well known for the support he gave to the cultivation of music in Glasgow, and for his keen interest in social and economic questions. Music was Dr. Inglis's favourite hobby in his later years, as yachting was in earlier life. As is better known he was a member of many professional and scientific societies, and served on a number of Government Commissions of Enquiry on questions concerned with shipping, shipbuilding, and engineering. In 1893 he was President of the Institution of Engineers and Shipbuilders in Scotland, and in 1898 of the Institute of Marine Engineers, to both of which bodies he delivered notable addresses. He was also a Vice-

President of the Institution of Naval Architects and a member of the Worshipful Company of Shipwrights. He was a member of Lloyds' Register Technical Committee, and for many years a highly useful member of the Clyde Navigation Trust and Clyde Lighthouses Trust. He was identified prominently with many other Glasgow and West of Scotland bodies—commercial and social—and delivered from time to time notable addresses on economic and labour questions.

Dr. Inglis is survived by his wife (a sister of Mr. James Denny, the present head of Messrs. Wm. Denny and Bros., Dumbarton), and by six sons.

In his Presidential address to the Institute of Marine Engineers, which contained many thoughtful paragraphs presented in his own pregnant way, the following may be quoted:—

"Nothing could be further from my intention than the discouragement of any effort to improve the means of instructing the young, or even the elderly, among us. I rather hold that the acquisition of knowledge is in itself an end, the attainment of which will repay all our exertions; and I wish to warn against the expectations which may not be realised when that wisdom, whose price is above rubies, is sought for merely as an instrument for the opening of some imaginary strong box, which shall disgorge itself upon us in golden showers. Pecuniary gain does not always attend superior mental attainments any more than prosperity invariably follows piety. Not only the wicked, but occasionally also the ignorant, flourish like the proverbial green bay tree. This, however, is no more an excuse for ignorance that it should be an incentive to wickedness.

"About a twelvemonth ago, at the annual dinner of a kindred society, I was privileged to hear the eulogy of James Watt pronounced by the illustrious Lord Kelvin. We were then reminded of the often told tale of the jealousy of the trade guilds, which hindered the young instrument maker from commencing business in his own workshop within the burgh of Glasgow; of the refuge afforded him by the venerable University, and of the work entrusted to him—the repair of a model steam engine belonging to the apparatus room of the natural philosophy class—and of how while engaged in this work, he was so fortunate as to conceive the idea of the separate condenser, the first of the long series of contrivances which make James Watt the most famous inventor in ancient or modern times.

"While listening to the eloquent address of the great philosopher, this somewhat irreverant question would keep intruding itself: Why did not the professor of natural philosophy himself invent the separate condenser instead of the young workman from Greenock? James Watt, with all his industry and capacity, cannot have been the equal of the Professor in knowledge of physics, and yet, for one person who has heard of his patron, the founder of the Andersonian University, tens of thousands are familiar with the name of the author of the modern steam engine. This brings us back to the proposition already faintly indicated, that we cannot produce to order a genius or an epoch-making invention by any process of culture yet discovered, and it is vain and unprofitable to feel discouraged if such flowers do not always spring up to adorn the fields under our various systems of tillage."

WM. CRINGLE ROBERTS, R.N.R. (Past Chairman of the Council).

The death of Wm. Cringle Roberts, in the 77th year of his age, on July 24th, caused many expressions of regret from a large number of engineering and other friends, by whom he was highly esteemed for his personal qualities. His birthplace was Falkirk, where his father, Henry Roberts, was an auctioneer.

During his school days William leaned to engineering for the future course of his life and became an apprentice with Messrs. Barclay, Curle & Co. He afterwards served with Messrs. Crawhall & Campbell, Glasgow. With a view to closer touch with marine engineering, he was employed in the works of the London and Glasgow Engineering Co., and in 1865 obtained an appointment as junior engineer in the SS. *Therese*, sailing between Grangemouth and Rotterdam. His next appointment was in Messrs. T. Wilson & Son's service in the Baltic trade. From this he transferred to Messrs. Malcolmson Bros.' employ, and he was engaged in steamers sailing to America, the East Indies, and the Mediterranean. Having gained the qualifying experience, he obtained his chief's certificate in 1868 and was appointed chief engineer in one of the passenger steamers sailing to Dublin. The London and Glasgow Engineering Co. were then building steamers for the Glen Line, and in 1872 he was appointed chief engineer of one sailing between London, China and Japan in the tea trade. Subsequently he was appointed superintendent engineer of the Glen Line, and held that appointment for about 44 years, retiring in 1916. He was a member of the Institution of Naval Architects, N.E. Coast Institution of Engineers and Shipbuilders, also of the Institute of Marine Engineers, in which he took a special interest and devoted himself to duties in connection with it as member and then as chairman of Council in 1901. He held the rank of hon. Engineer Commander, R.N.R.