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Notes on Reduction Gear.

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READ

Tuesday, November 9, *at* 6.30 *p.m.*

CHAIRMAN: Mr. R. S. KENNEDY (Chairman of the Council).

The CHAIRMAN: I have pleasure in introducing Mr. Guthrie, whose association with the well-known firm of Richardsons, Westgarth & Co., Ltd., is an assurance of his capability to deal with the important subject of marine reduction gears. Outside this special subject, Mr. Guthrie must know a great deal about the application of gears generally. It is a subject which has been very prominent lately, both in the technical press and in papers read before the various engineering societies, so that while Mr. Guthrie has no doubt endeavoured to give us points which have not been touched upon previously, he will perhaps have to a certain extent to go over old ground as well.

THE use of reduction gear in conjunction with turbines enables vessels of all speeds to be propelled with higher overall efficiencies than are possible with the direct turbine drive, and renders practicable the use of high pressures and temperatures whereby present day efficiencies may be further improved.

The nett overall efficiency of a marine turbine installation aft of the turbine stop valve, is the product of three factors.

(*a*) The efficiency of the turbine in developing the required power in terms of steam consumption.

(*b*) The efficiency of the propeller.

(*c*) The efficiency of transmission between turbine and propeller.

SECTION I.

Steam Turbines.—Steam turbines develop their power most economically at high revolutions, especially in the case of high pressure turbines. The efficiency of a turbine depends on the blading efficiency:—on the degree to which the mean velocity ratio (the ratio of mean blade speed to steam speed) approaches the velocity ratio corresponding to maximum efficiency:—on the leakage and windage losses, and finally on the mechanical efficiency.

In marine turbines considerations of reliability necessitate relatively large clearances, and the leakage losses become excessive at low speeds of revolutions, whilst the large diameters required for a suitable mean velocity ratio increase windage losses and make the turbine large and heavy. There is, of course, a limit to the speed of a turbine designed for a given power, and this is determined by the area required at the low pressure end to permit the steam to exhaust to the condenser without excessive leaving losses, and by the strength of the materials of construction. Fig. 1 shows the variation in efficiency of two separate series of turbines of constant normal power, but different speeds. It will be noted that there is in each case a definite point where the efficiency falls rapidly as the revolutions are reduced.

The desire for improved efficiency has led to the adoption of higher pressures and temperatures and the results obtained in service with the T.S.S. *King George V.*, the turbines of which are designed to work with steam of 500 lbs. pressure, superheated to 750° F. will be watched with interest by all engineers and will, undoubtedly, largely influence the future application of high pressures to marine propulsion.

The reason for the adoption of higher pressures will be evident from an inspection of Fig. 2 which shows the increased proportion of the total heat available for useful work as the pressure is increased, and it will be noted that while the latent heat of evaporation decreases as the pressure rises, the total

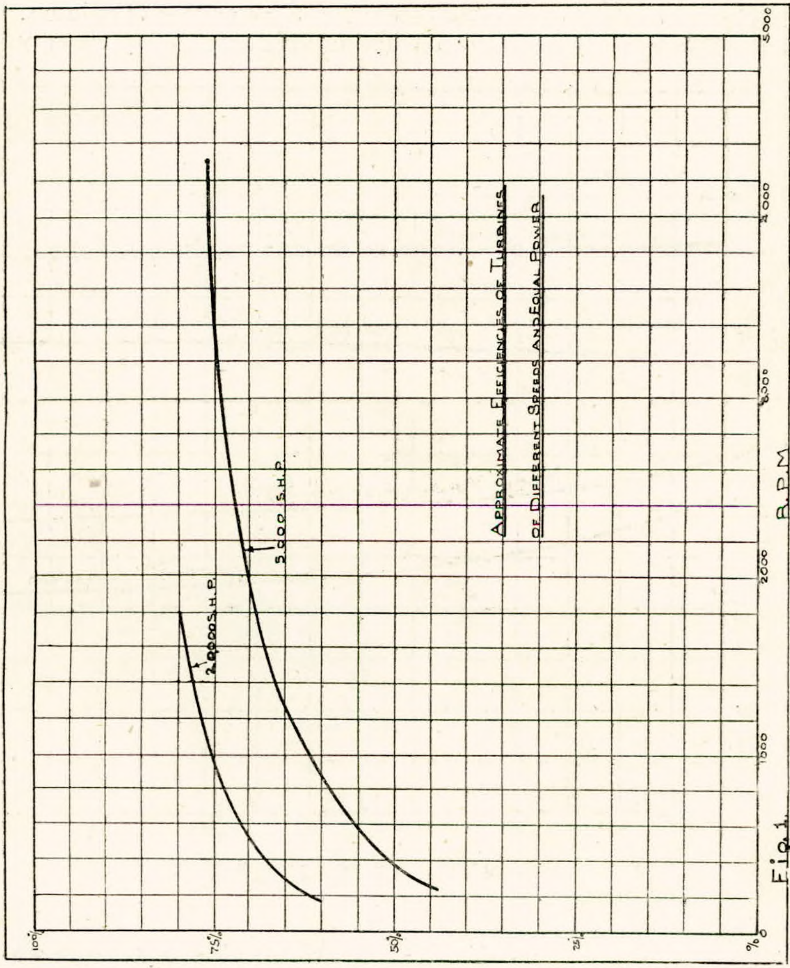


Fig. 1.

heat and consequently the fuel required for generating the steam varies only slightly throughout the range, and is actually becoming less at the higher pressures.

The reason for the adoption of high temperatures is that high pressure steam is unsuitable for use in a turbine unless it contains a sufficient degree of superheat to enable it to expand throughout its range and exhaust to the condenser with a reasonably high dryness fraction.

The upper limit of temperature is largely a matter of opinion, but with the materials of construction available to-day we may

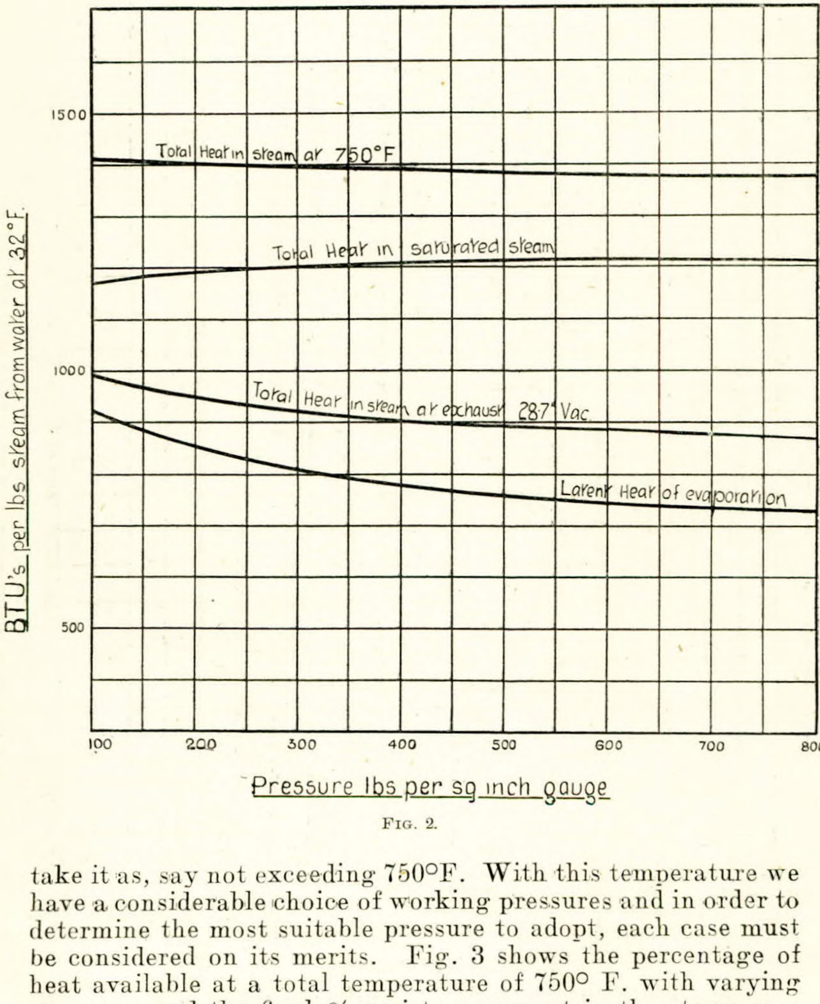


FIG. 2.

take it as, say not exceeding 750°F. With this temperature we have a considerable choice of working pressures and in order to determine the most suitable pressure to adopt, each case must be considered on its merits. Fig. 3 shows the percentage of heat available at a total temperature of 750° F. with varying pressures and the final % moisture present in the steam as a result of adiabatic expansion of the steam at each pressure.

The less the degree of superheat in the steam the greater will be the percentage of moisture in the steam when it finally exhausts to the condenser and the lower will be in consequence the efficiency of the turbine. The overall efficiency is of course the product of the turbine efficiency and the ratio of available heat to total heat.

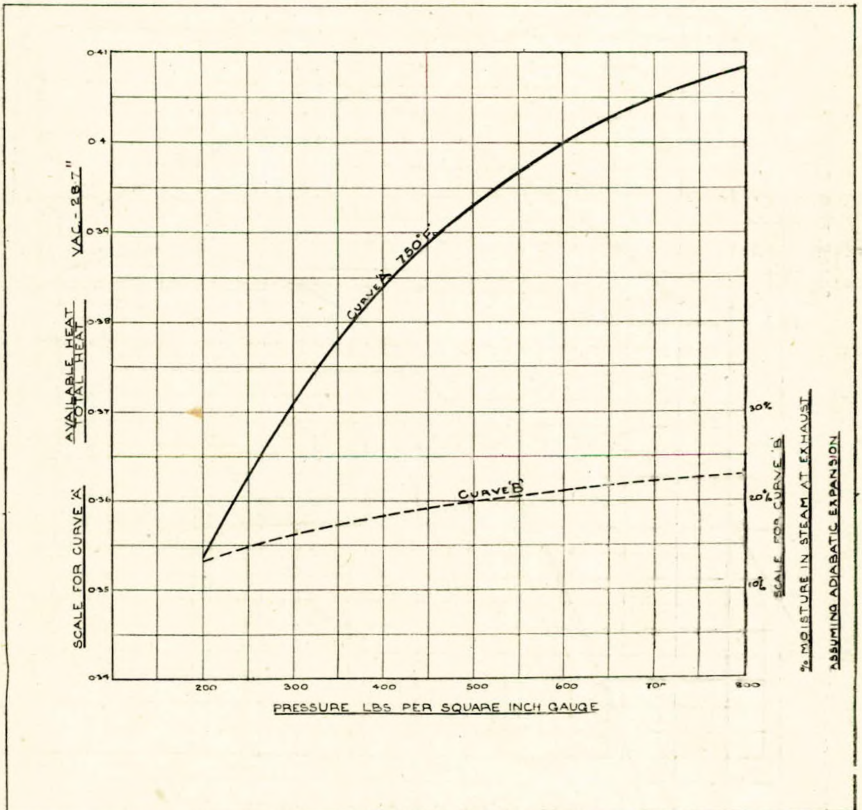


FIG. 3.*

Fig. 3a shows how these vary with varying pressures and a curve has been added showing the combined efficiency at each pressure.

To reduce or eliminate the moisture in the steam interstage heating may be adopted.

HIGH PRESSURE TURBINES.—In any turbine the energy absorbed in a stage per lb. of steam varies as the mean blade

*NOTE:—Actual Moisture in Steam o.c. % Moisture from Curve 'B' x by Turbine Efficiency.

speed^2 and if the turbine efficiency is to remain constant, increased pressures must necessarily be accompanied by increased speeds of rotation or alternatively by an increased

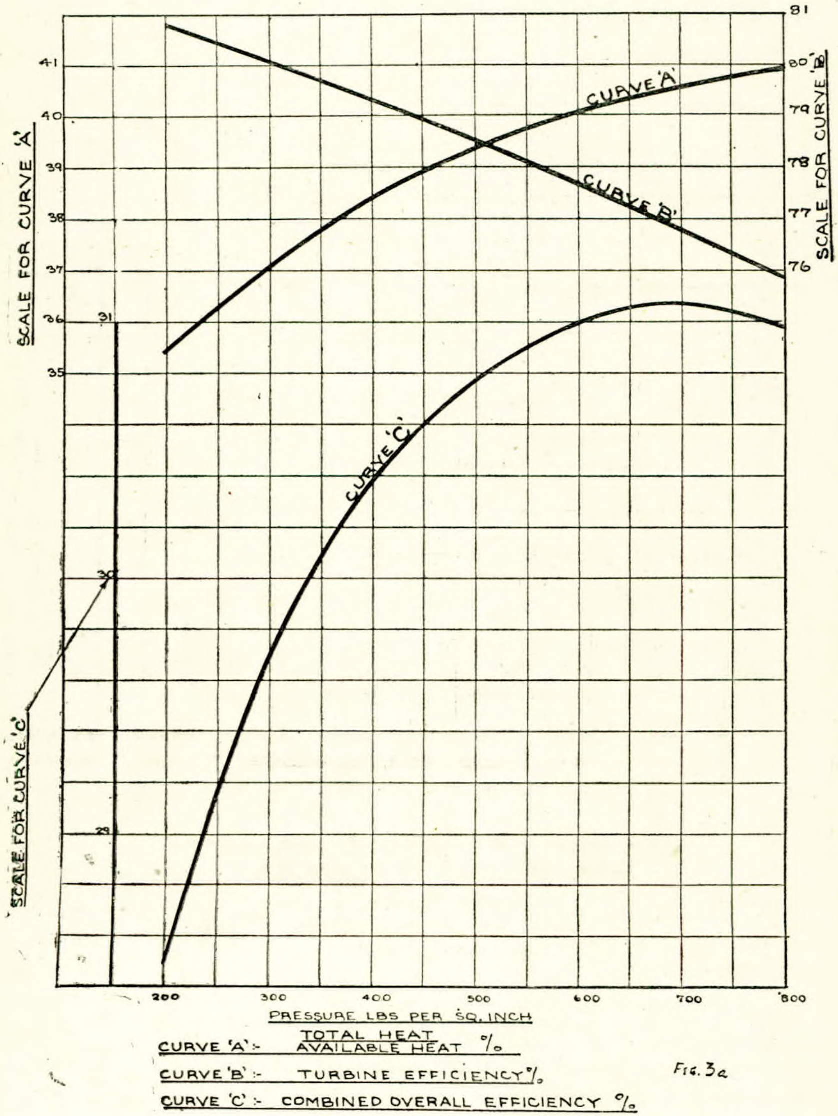
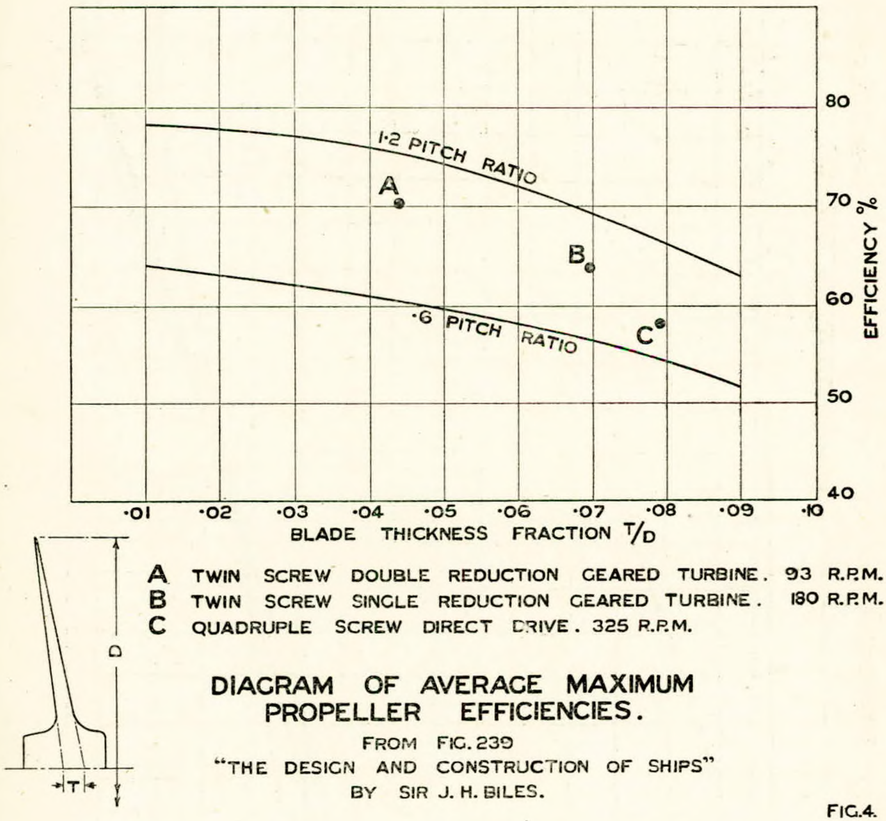


FIG. 3a

number of stages. Wherever possible high revolutions is the better method, as it enables the turbine to be of small dimensions, thus reducing windage and gland losses.

SECTION II.

Propellers.—Propellers if they are to be efficient must run slowly. The permissible revolutions vary with the type, and

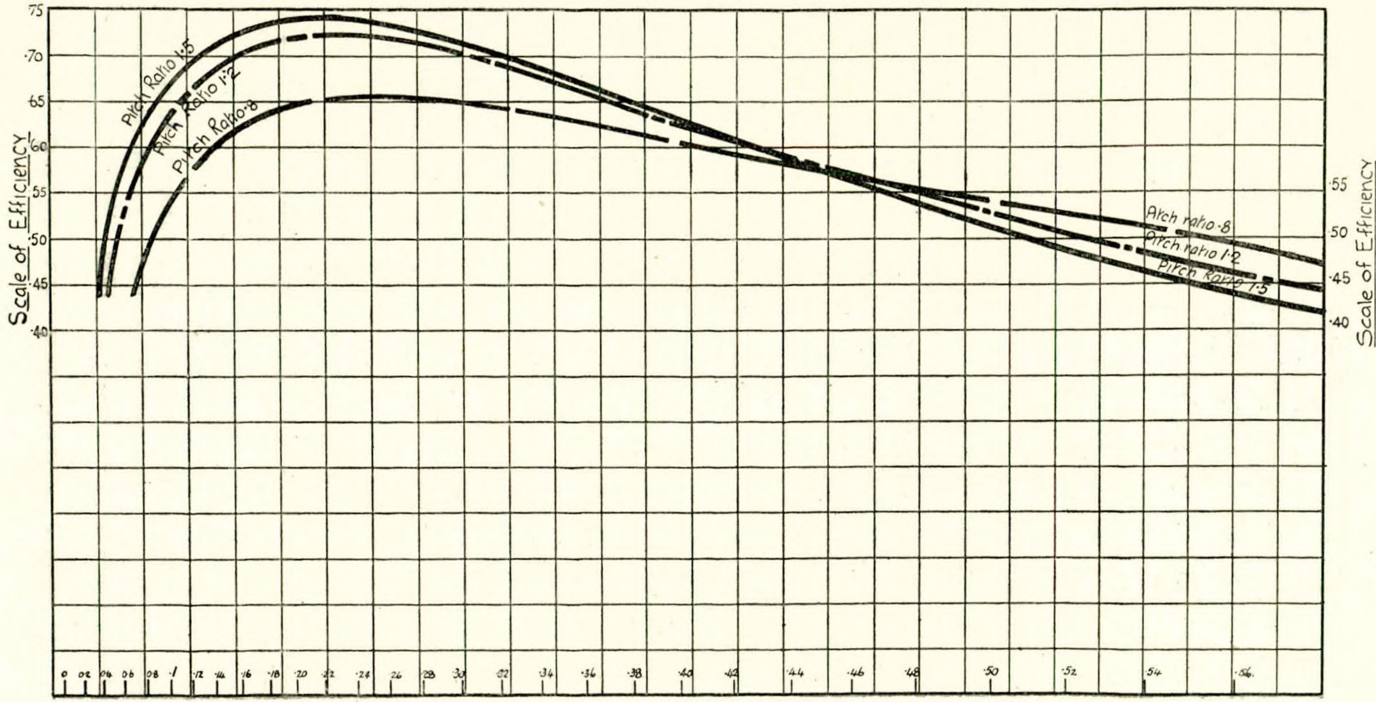


A TWIN SCREW DOUBLE REDUCTION GEARED TURBINE. 93 R.P.M.
B TWIN SCREW SINGLE REDUCTION GEARED TURBINE. 180 R.P.M.
C QUADRUPLE SCREW DIRECT DRIVE. 325 R.P.M.

DIAGRAM OF AVERAGE MAXIMUM PROPELLER EFFICIENCIES.
FROM FIG. 239
"THE DESIGN AND CONSTRUCTION OF SHIPS"
BY SIR J. H. BILES.

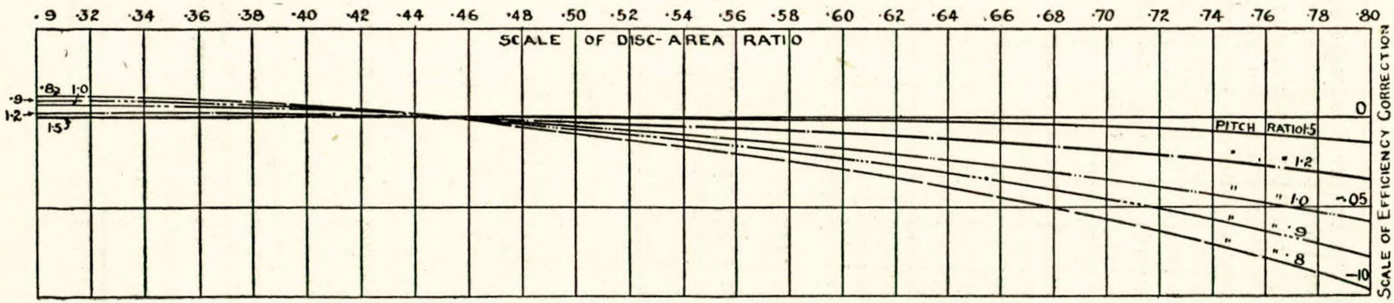
FIG. 4.

generally speaking increase with the speed of the vessel. High revolutions are necessarily associated with low pitch ratios, large disc area ratios, and high blade thickness ratios each of which as will be found from an inspection of Figs. 4, 4a, and 4b to be detrimental to efficiency.



Slip Ratio

Curves of Efficiency



VARIATION OF EFFICIENCY WITH DISC AREA RATIO

From considerations of Sections 1 and 2 it is evident that the efficiency characteristic of a turbine rises as the revolutions are increased while the efficiency characteristic of a propeller falls. The efficiency corresponding to that point at which the characteristic curves intersect is the maximum combined efficiency that can be obtained with a direct drive, whereas a much higher combined efficiency is possible by the interposition of reduction gears between the turbine and propeller, provided always that the loss in such transmission gear is relatively small.

Three systems of reduction gear have been used in marine practice, viz. :—

(*a*) Mechanical, i.e., toothed gearing;

(*b*) Hydraulic gearing;

(*c*) Electrical gearing.

Of these, mechanical, or toothed gearing, is at once the simplest and most efficient type. Its reliability to-day whether of the single or double type is unquestioned, and it is the only system that will be dealt with in these notes.

SECTION III.

Gearing Efficiencies.—The efficiency of reduction gearing has been accurately determined by driving two similar gears in opposition by a motor interposed between one set of pinions and placing a generator between the other set of pinions. The motor and generators are preferably of the direct current type. The efficiency of the motor and generator can be accurately determined and hence this method lends itself to precise measurements. Only the losses are measured and these form a very small percentage of the total power transmitted.

Efficiencies as high as 99%, including bearing losses have been recorded with a single reduction gear, and 98% with double reduction gears. The loss, therefore, due to the interposition of gearing is exceedingly small and is far outweighed by the improved overall efficiencies attainable when the turbines and propellers are designed without reference to one another and each arranged to run at speeds conducive to maximum efficiency. The temperature rise of the oil in its passage through the gears is sometimes used as a rough and ready means to check the efficiency and in this connection it is con-

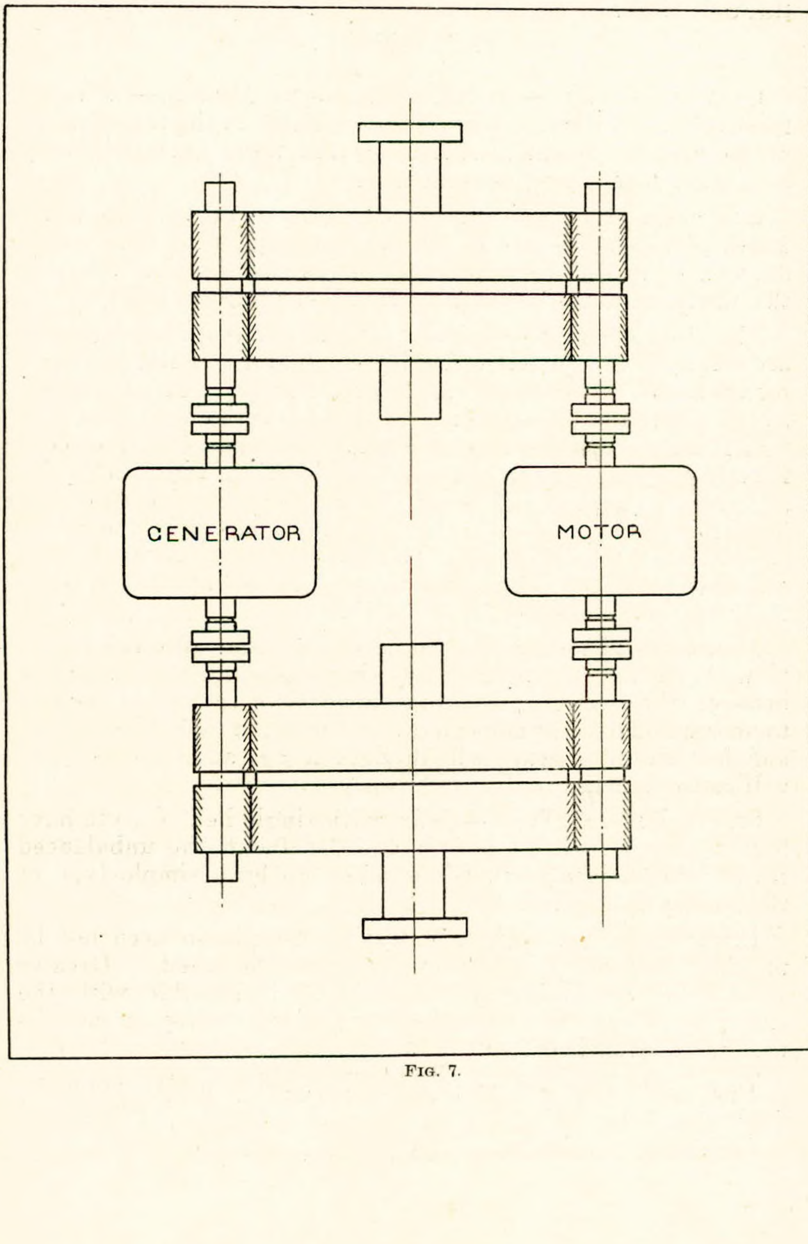


FIG. 7.

venient to remember that one brake horse-power dissipated as heat will raise the temperature of one gallon of lubricating oil through 10°F.

SECTION IV.

GEARING DESIGN.—GENERAL POINTS OF DESIGN.—So much information on details of design is available in the transactions of the various technical societies* that these notes will only deal with design in a general way.

Gear teeth profiles are almost invariably of the involute form. Teeth of this shape are easily produced and with them small differences in the distance between the centres do not impair the efficiency of the gears nor affect their uniform rotation.

The teeth are formed spirally, and to eliminate axial thrust, are arranged in two helices having equal and opposite angles of inclination. Helical gears are superior to spur gears inasmuch as the continuity of engagement is greater, the duration of contact is longer and the rate of loading and unloading the teeth is more gradual. Each of these considerations makes for quiet and smooth running and permits of greater tooth pressures and higher peripheral speeds. The additional strength due to the spiral form of tooth enables smaller pitches to be used, thus reducing back-lash, and allowing an increased number of teeth to be in mesh together.

Variations in the pitch or form of the teeth cause the pinion to move to and fro in an axial direction and the coupling between the pinion and turbine must have sufficient freedom to accommodate this movement, which in the case of badly cut and fast running gears will produce noise, and, if excessive, will cause damage.

SINGLE HELICAL TYPE.—Gears with single helical teeth have been made in America and on the Continent, the unbalanced thrust, due to the teeth being taken up by a simple type of thrust block or collar.

In this type axial displacement of the pinion need not be provided for and a solid coupling can be used. Greater accuracy of machining and assembling is possible with the single helix type and consequently a given degree of smoothness in running is easier to obtain.

To provide some degree of flexibility and to partly damp out fluctuations due to tooth errors, a "quill drive" is often used.

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* See List of References.

between the turbines and pinions and to compensate for errors of tooth alignment floating frame gears of the "Macalpine" type are frequently adopted. Single reduction gears are used for speed reduction ratios not exceeding, say, 20 to 1, and for higher ratios double reduction gears are used. The latter may be of the three box type in which two complete single reduction gears are connected in series, or the gears may be of the interleaved type with all gears incorporated in one casing. Whichever type of gear is adopted it is desirable to interpose a stiff cast iron bedplate between the gear case and the engine seating as shown on Fig. 5. This not only adds to the rigidity of the seating and gears, but enables the gears to be assembled in the ship in exactly the same alignment as they were assembled in the shops.

It is sometimes said that double reduction gears are noiser than single reduction gears, but with accurately cut gears the difference, if any, is very slight. On the other hand the effect of any inaccuracies causing momentary positive or negative accelerations of the gears will be more noticeable in the case of double reduction gears, owing to the increased number of freedoms which this type possesses when compared with single reduction gears, and the effect of any lack of dynamic balance will be more pronounced owing to the higher speed of gear wheels and increased flywheel effect of the turbines.

Opinions differ as to the desirability of having a hunting tooth in the large wheel, that is, one tooth more or less than an exact ratio between the wheel and pinion. The object is to ensure the complete interchange of all the teeth, but in the writer's opinion it is doubtful if there is any advantage in this except in the case of wheels cut with multiple threaded hobs.

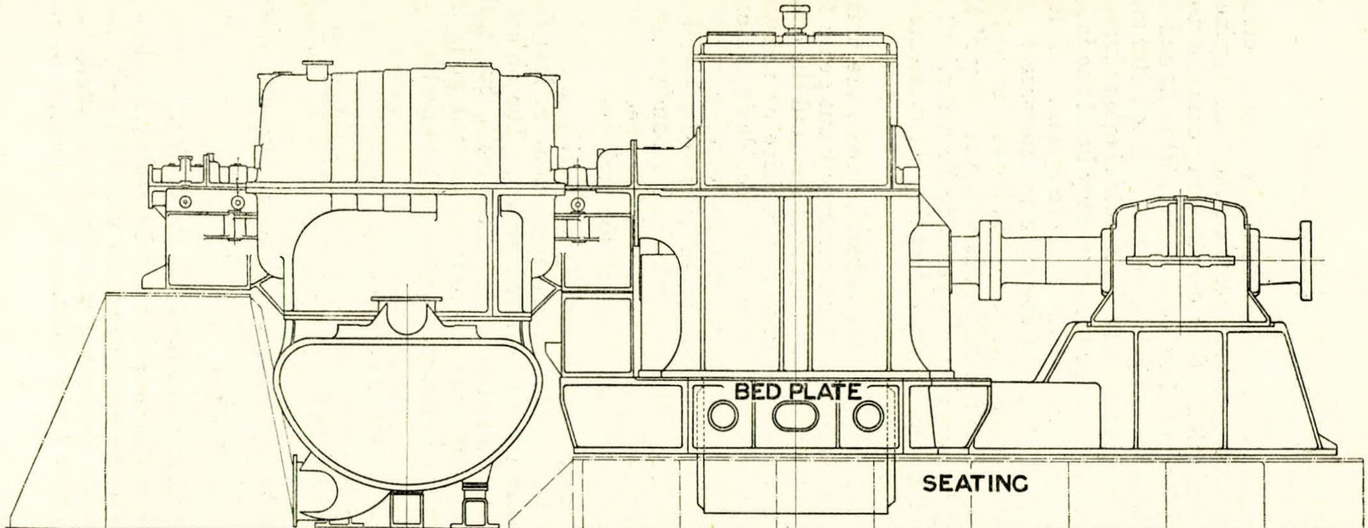
SECTION V.

Gears should run smoothly and without noise and will do so if:—

(*a*) The peripheries and pitch circles of pinions and wheels are truly concentric with their journals.

(*b*) The pitch of teeth in pinion and wheel correspond and are as uniform as possible and the teeth are correct to form.

(*c*) The centre of gravity of pinions and wheels coincide with the geometrical axis of rotation and there are no unbalanced couples. That is, the pinion and wheel must be in dynamic balance.



ARRANGEMENT OF GEARED TURBO DRIVE SHOWING GEARING MOUNTED ON BEDPLATE

FIG. 5.

The importance of the wheels and pinions being in dynamic balance is not always appreciated as it should be. If the wheel is not in dynamic balance the out of balance weight will, when the wheel is revolving, develop a centrifugal force which will cause the wheel to oscillate. In the case of an unbalanced and freely revolving wheel the oscillating displacement in a given plane can be readily calculated. It varies directly as the resolved centrifugal force in that direction and inversely as the weight of the wheel. It should be noted that the amplitude of the oscillating displacement is entirely independent of the velocity of rotation of the wheel. This is because, whilst the force producing the displacement varies as the square of the circumferential velocity, the time during which the force acts in a given direction varies inversely as the speed^2. In actual practice the case is somewhat complicated, as we have not freely rotating wheels to deal with, but wheels, the axis of which are constrained to some extent by bearings housed in a gear case which is firmly bolted either directly or through a bedplate to the ship. The extent to which this constraint reduces the amplitude is indeterminate, but nevertheless considerable amplitudes may and do exist, and remain undetected as it is only when displacements, due to lack of balance, are associated with frequencies of a much higher order than are met with in gear wheels, that they become perceptible as vibrations.

SECTION VI.

Lubrication.—Gears must be effectively lubricated. The most suitable oil for lubricating gear teeth is a straight mineral oil with a high viscosity. On the other hand the viscosity of the oil for lubricating the high speed bearings should be low in order that the heat generated by the friction losses may not be too great. It is obviously impracticable to have two separate oil systems and the problem is usually solved in practice by using one quality of oil and supplying it to the gear teeth and bearings at different temperatures and consequently different viscosities.

SECTION VII.

Gear Cutting.—Gears are usually cut by hobbing and the hobbing machine should possess a high degree of accuracy. Any periodic errors in the dividing wheel will be reproduced in an aggravated form in the gears cut unless some differential gear such as the Parsons Creep mechanism, is interposed between the dividing wheel and table. This mechanism, while it does not eliminate the errors, destroys their periodicity by distributing them over the periphery of the wheel in a manner, depending on the speed ratio of the additional gear. Whether such gear is an advantage or not depends entirely on the degree of accuracy of the primary dividing wheel of the machine. Incidentally it may be mentioned that the tooth spaces cut on a machine fitted with "creep" are necessarily wider than they would be if cut with the same hob on a machine not so fitted.

The machine must always be accurately adjusted. The points which require special attention are the index worm, which must be concentric with its driving spindle, and the thrust washers of the worm, hob and feed screws, which must be truly parallel and at right angles to the axis of rotation. Inaccuracies at any of these points will result in periodic errors in the tooth spacing. Friction brakes on the hob spindle and other driving gears are advantageous, as they damp out fluctuations and tend to smoothness of cutting.

The hobbing process produces work quickly with a high degree of accuracy, but it has the disadvantage that while the blank is rotating continuously, the hob, owing to the spaces between its teeth, is cutting intermittently, and slight feed marks are left on the bottoms of the tooth spaces and faces of the teeth.

The points of teeth produced by hobs, the teeth of which are of true rack form, are thicker than they should be and the hob teeth should be modified to correct this.

In the past, errors in hobs were responsible for many cases of faulty gear cutting and for accurate work the tooth surfaces of hobs must be ground after hardening. Hobs should preferably be of the single threaded type, as the smaller the lead angle the nearer will the cut teeth approach the true involute form and the smaller will be the amount of sliding in the gears. Multiple threaded hobs are less likely to produce true curves and have the additional objection that they impose greater strains on the dividing mechanism of the machine.

It is advisable to cut the gear teeth in two stages, using roughing and finishing hobs, the latter cutting on the faces of the teeth only. Partial depth cuts are not satisfactory, as during the first cut the hob is trying to insert a greater number of tooth spaces in the blank than that corresponding to the dividing mechanism of the machine.

SETTING.—The greatest care should be taken in locating the gear blank accurately on the machine and in fastening it to the table without distortion. The relative position of the gear blank and table should be marked so that both helices will be cut in the same position relative to the dividing wheel.

When cutting, a uniform temperature should be maintained and the power absorbed by the machine should be kept constant. Any variation in the temperature will be accompanied by a variation in the phase of the gears of the machine and a change in the viscosity of the oil lubricating the slide faces carrying the table which will cause a difference in the power absorbed.

In the early days of gear cutting, engineers frequently complained of noise in the gears, partly from considerations of comfort, but mostly in the belief that the efficiencies of such gears were very low. It is true that the frictional losses in the teeth will vary to some extent with the noise, but the frictional losses form such a very small percentage of the total power transmitted that their effect on the overall efficiency of the gears is very small. A few notes, however, on this matter may be of interest.

Noise in gears is caused by vibrations set up in the material of the gears and gear case, the vibrations being mainly caused by faulty meshing of the teeth due to:—

(1) Variations in the gear centres occurring during a revolution which tend to cause momentary accelerations or decelerations of the gears, lead to unequal loading of the teeth and give rise to vibrations.

(2) The effect of irregularities of pitch or lack of correspondence of pitch between wheel and pinion also lead to unequal loading of the teeth and in extreme cases causing momentary separation of the teeth, resulting in the load being transferred from tooth to tooth by a series of impacts.

(3) Lack of exact correspondence of pitch of the teeth in wheel and pinion.

(4) The involute curve not extending throughout the full working depth of the tooth. This only occurs in the case of pinions having relatively few teeth and is detrimental to smoothness of action.

SECTION VIII.

GEAR TROUBLES.—PITTING.—Pitting of gear wheel teeth is frequently noticed near the pitch line soon after the gears have been first put into service. Various theories have been advanced to account for this phenomenon, but as it almost always occurs where the relative movement of the teeth is a minimum and where consequently effective lubrication is difficult it would appear reasonable to suppose that it is caused by partial seizing or welding and subsequent tearing apart of the tooth surfaces in action.

The pitting gradually gets less and less as the feed marks on the tooth surface left by the hobbing machine are smoothed out in service, and as it normally ceases after a time, it is not now regarded as important.

FRACTURE OF TEETH.—Instances have been recorded of pinion teeth fracturing shortly after the gears have been put into service, but the number of such failures is insignificant when compared with the total S.H.P. transmitted by gears.

A probable explanation of such fractures may be found in any or all of the following three causes:—

(*a*) Excessive loads on the gear teeth.

(*b*) The distribution of tooth stress being such that the maximum stress differs widely from the mean.

(*c*) The condition of the material being such that it is unable to withstand the stresses which ordinarily would be well within its capacity.

(*a*) The load stresses on gear teeth are:—

(1) Stresses due to the power transmitted by the tooth.

(2) Stresses due to irregularities of pitch, bad form of tooth, faulty alignment, etc.

(3) Stresses due to forces brought into play by lack of dynamic balance.

(4) Stresses caused by torsional oscillations.

The stresses under (1) are easily calculated, but those under (2) and (3) are indeterminate. They vary with the degree of accuracy of cutting and assembling the gears and for a given standard of workmanship and a given pitch of tooth vary as the square of the peripherial speed of the gears. The speed should, therefore, be considered when determining the load coefficient. Stresses of any magnitude due to (4) will only arise in exceptional cases and will not be considered here.

However great the total stresses may be, they are sensibly the same for the teeth of the wheel and of the pinion, and the magnitudes and velocities of impact are of course identical. Although a tooth in the pinion is loaded and unloaded much more frequently than a tooth in the wheel rim this is immaterial as the capacity of the material to resist varying dynamic loads is not affected by the rate of stress repetitions.

The designed margin of strength in the teeth is great and the fact that the fractures are almost entirely confined to pinion teeth and that replace pinions of the same design and cut under similar conditions have given satisfactory service would seem to indicate that the failures were not caused by excessive tooth loads.

(*b*) The maximum stress may differ widely from the mean especially in the case of the harder alloy steels. For example, the concentration of stresses at a mathematically sharp corner causes an intensity of stress which is infinitely great. Fortunately such corners are never met with in practice, but a hair crack produced during hardening or by segregation or a tool scratch approximates to it very closely.

With steels of reasonable ductility as used for wheel rims local intensity of loading gives rise to plastic flow or distortion of the steel and automatically leads to a safer distribution of stress.

On the other hand with harder steels in which plastic flow does not occur so readily the stress is not so effectively distributed and the tendency to fracture is consequently greater.

The importance of making all changes of Section gradual and well rounding the bottoms of the tooth spaces, will be evident from an inspection of figure 6, which is reproduced from a report of the Advisory Committee for Aeronautics. The effect of the size of fillet on the fatigue range is clearly shown by the full line (the dotted portion has been added by the writer) and the rapid decrease in the fatigue range with decrease in the radius of the fillet should be noted.

SECTION IX.

METALLURGICAL.—Because of wear considerations, pinions are almost invariably made of alloy steel. A typical specification of the material is given in Appendix I.

To obtain the desired physical properties it is necessary to subject the forgings to heat treatment, that is, the forging is first hardened by heating it uniformly to a specific temperature and quenching it in oil. It is then tempered by heating

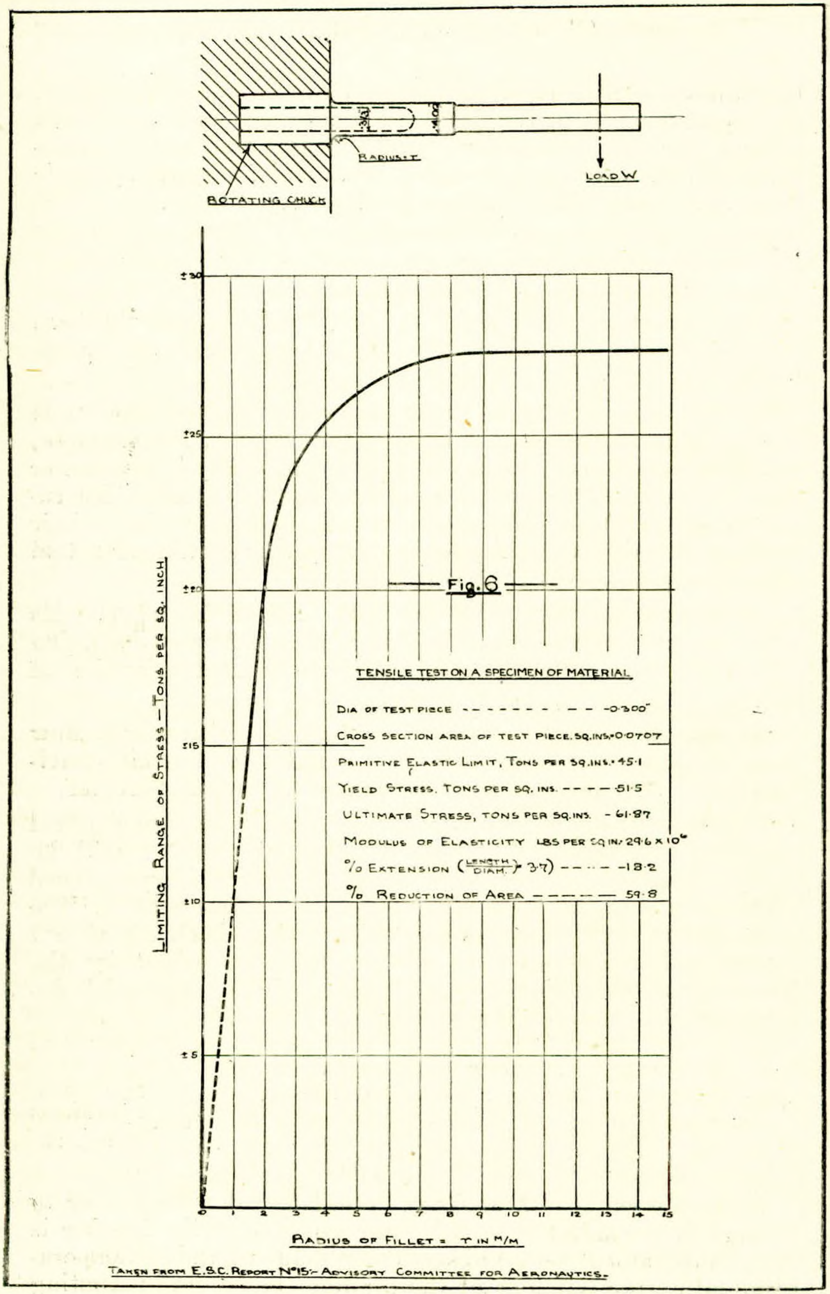

ROTATING CHUCK
RADIUS = r
LOAD W
±30
±25
±20
±15
±10
±5
0 1 2 3 4 5 6 7 8 9 10 11 12 13 14 15
LIMITING RANGE OF STRESS — TONS PER SQ. INCH
Fig. 6
TENSILE TEST ON A SPECIMEN OF MATERIAL.
DIA OF TEST PIECE - - - - - - - - - -0·300"
CROSS SECTION AREA OF TEST PIECE. SQ.INS.-0·0707
PRIMITIVE ELASTIC LIMIT, TONS PER SQ.INS.-45·1
YIELD STRESS. TONS PER SQ. INS. - - - - 51·5
ULTIMATE STRESS, TONS PER SQ.INS. - 61·87
MODULUS OF ELASTICITY LBS PER SQ IN. 29·6 x 10^6
% EXTENSION (LENGTH/DIAM. = 3·7) - - - - 18·2
% REDUCTION OF AREA - - - - - 59·8
RADIUS OF FILLET = r IN M/M
TAKEN FROM E.S.C. REPORT No 15 - ADVISORY COMMITTEE FOR AERONAUTICS.

FIG. 6.

to a lower specific temperature and again quenching in oil. The temperatures at which the steel is hardened and tempered are important and in carrying out the heat treatment the temperature of the furnace must not be appreciably higher than the specified temperature, otherwise the outer portions of the forgings will reach the hardening temperature while the inner portions are still at a lower temperature. Even if the forging has been uniformly heated there is still the difficulty of cooling.

Every steel has a critical rate of hardening, if the actual rate of cooling is equal to or greater than this, the steel will be homogeneous throughout. If it is less, the steel will be different at the inner and outer portions of the forging. The forging as a whole may possibly be in a state of internal stress, and in extreme cases minute or hair cracks may have formed. For this reason when dealing with forgings of large and variable mass it is advisable to select a steel having a low critical rate of cooling, for example, a steel which to some extent possesses air hardening qualities and then accelerate the cooling by quenching in oil. It is also highly desirable to hollow bore all pinion forgings, and thus enable the quenching medium to have access to both inner and outer surfaces at the same time.

Mass effects are of no importance in tempering, neither is the rate of cooling, except in the case of nickel alloy steels, which under certain conditions, are liable to "temper brittleness." The phenomenon of temper brittleness is not yet clearly understood, but it is definitely known that it can be avoided in nickel alloy steels—tempered at temperatures above 550° C. if the rate of cooling is sufficiently rapid. Steel suffering from "temper brittleness" will give good test results as regards ultimate strength, elastic limit, elongation per cent. and reduction of area per cent. The thermal conductivity and the microstructure as revealed by the microscope will be satisfactory and unless a notched bar test is made the brittleness may remain unsuspected. This test is an important one as the notched bar value is a measure of the capacity of the steel to resist the formation and growth of cracks, and affords some indication of the efficacy of the heat treatment and the condition of the microstructure of the material. Most pinion fractures known to the writer have been fatigue fractures, that is, they have been caused by the effective fatigue range of the material having been exceeded. In almost every case mechanical tests subsequently made of the broken teeth and adjoining portions of the pinions showed the material to

NICKEL CHROMIUM STEEL.

OIL HARDENED AT 800/820 DEGREES CENTIGRADE & REHEATED AT INTERVALS OF 50 DEGREES CENTIGRADE AT TEMPERATURES INDICATED.

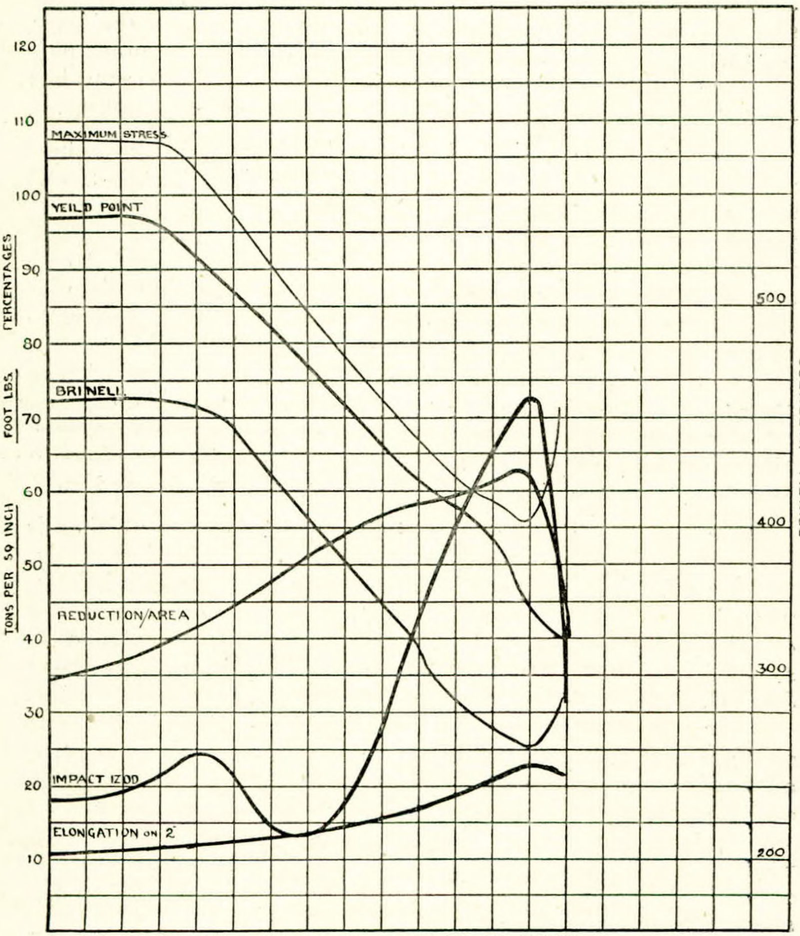


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FIG. 8

be satisfactory, and the fractures were probably caused by the normal fatigue range of the material having been exceeded largely due to severe concentration of stresses arising from some cause or another. In some cases the notched bar value was low, indicating lack of toughness in the material, but this in itself would have no effect on the fatigue range, although it would indicate that the material was not in its best condition. It should be noted that notched bar tests are quality tests only, and as pointed out by Gough in his "Fatigue of Metals" it does not follow that if two different materials A and B absorb, say 10 and 1ft. lbs. respectively to fracture, using test pieces of one particular shape and size that material A possesses 10 times the shock-resisting properties of B.

Fig. 8 shows the variation in the physical properties of an alloy steel with different tempering temperatures.

APPENDIX I.

Specification of Gear Pinions.—1. The steel to be made in an electric furnace or by the acid open hearth process, from selected scrap, which is reasonably free from rust and other surface impurities.

2. The steel is to contain about but not less than 3·5% of nickel and 0·24 to 0·35% carbon. The phosphorus and sulphur not to exceed 0·04% and the manganese 0·5 to 0·8%.

3. The shafts are to be gradually and uniformally forged from ingots from which not less than 40% of the total weight of the ingot is to be discarded from the top end of the ingot, and not less than 5% of the total weight of the ingot from the bottom end.

4. Effective means are to be taken to detect any segregation or discontinuity of surface.

5. Photographs of polished and etched discs taken from the forgings are to be supplied by the steel makers.

6. The forging before heat treatment is to be reduced in diameter one-and-a-half inches, by machining, and one inch is to be machined off all vertical faces, leaving quarter of an inch on all diameters and quarter of an inch on all vertical faces for finish machining by engine builders.

Axial holes through all pinion shafts are to be bored from the solid, the holes to be concentric with the outside diameter and to be finished smooth and true to dimensions given.

7. The parts of the pinion shaft on which the teeth are to be cut are to be machined smooth and a sulphur print taken from the surface by the steelmakers.

8. The forging after machining and after the hole has been bored is to be heated to a temperature of 825 degrees to 875 degrees C and then quenched in oil, this is to be followed by tempering at 550 degrees to 660 degrees C.

During both treatments care is to be taken that the temperature of the furnace is not higher than necessary, and the duration of heating is not to exceed by more than half an hour the minimum required to raise the centre of the forging to the specified temperature.

9. The forging is to be free from all flaws and defects when examined after final machining by the engine builders.

10. Test pieces cut from the material in longitudinal and transverse directions after the heat treatment are to fulfil the following requirements:—

Tensile Strength: 40 to 45 tons per sq. inch in both directions.

Elastic Limit: To be desirably 30 tons, but not less than 28 tons per sq. inch.

Elongation: Not less than 20% in two inches longitudinal, and 16% transverse.

Bending Test: Test pieces three-quarter inches wide and three-eights inches thick to be bent over a three-quarter inch radius through 180 degrees without facture.

Sankey and Izod test pieces to be cut from the forging in longitudinal and tranverse directions after heat treatment. If the results from these test pieces are abnormally low a further microscopical examination to be made.

11. The forging to be open to the inspection of the engine builders at all stages of manufacture.

12. Test pieces to be machined to comply with requirements of British Standard test pieces "C," "D," or "E."

13. The manufacture, inspection and testing of the forgings to be to Lloyd's requirements.

14. The forging to be subject to the usual guarantee of the engine builders for a period of six months.

APPENDIX.—LIST OF REFERENCES.

General.

Title.	Author.	Publication.	Date.
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Reduction Gearing on Steam Ships	... ...	Engineering	7 May, 1920
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APPENDIX.—LIST OF REFERENCES—*continued.*

GENERAL.

TITLE.	AUTHOR.	PUBLICATION.	DATE.
SECTIONS IV/VIII.—GEARING—*continued.*			
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The Engagement and Loading of Involute Gearing	A. F. Ainslie	Engineering	28 Dec., 1923
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The Mechanical Properties of Steel with Some Considerations of the Question of Brittleness	Dr. W. H. Hadfield, D.Met., A.M.I. Mech.E.	Excerpt Minutes of Proceedings of the Meetings of the Inst. of Mechanical Engineers in London, Sheffield and Manchester	1919
Steels Used in Aero. Work	Dr. W. H. Hadfield, Brown and Firth Research Laboratory, Sheffield	Paper read before Aeronautical Society of Great Britain ...	—
The Fatigue of Metals	H. S. Gough, M.B.E.B.Sc. ...	... ...	1924

The CHAIRMAN: We have listened to a paper which I am sure you will agree has been very interesting. There is plenty of material in it worthy of further consideration and discussion, both on the theoretical and the practical sides. In this Institute we are always glad to hear the practical side. My own experience of reduction gears has been principally on the ship-repairing side. We have been very fortunate for the last three or four years, but in the earlier days gearing failures caused some anxiety. The only case we had, however, which was never clearly explained, was a gear wheel which became slack on the cone. We were unable to trace the cause; it was brought up on the cone again, and it has run satisfactorily ever since. I think we all have every confidence in gears to-day. The author's remarks on materials are, I think, very true, especially as regards the pinion.

Mr. W. HAMILTON-MARTIN: The author mentioned smoothness of running and noise in gears, the latter being usually more noticeable in double reduction gearing, and pointed out the necessity for carefully balancing such gears.

The importance of obtaining highly accurate balance during construction or overhaul of such parts is being more fully realised to-day as speeds and operating powers increase.

The comparatively heavy and fast running intermediate wheel of a double-reduction gear, interposed between a pinion connected to a high speed rotor of considerable mass and the main gear-wheel, which is rigidly connected by lengthy shafting to a heavy propeller, acts as the weakest link in the transmission.

Any lack of balance in this system is bound to express itself at this wheel; that is to say, at the contact surfaces of its teeth with those of the pinion and main wheel. Any change in motion which might tend to follow from such unbalance is always energetically opposed by this intermediate gear, which will strongly endeavour to maintain any momentary speed it possesses. In single reduction gears, the main wheel teeth or those of the pinions may likewise suffer.

To ensure complete freedom from resulting wear, noise, vibration and untimely repair or damage, the most perfect balance of all parts of the transmission is essential.

Any money spent on careful balancing during the construction of such parts will repay itself many times over in their wear-and-trouble-free service.

Gear wheels, rotors and propellers require a highly accurate static balance, and where possible their component parts should be successively and collectively balanced, when couples of undesirable magnitude will be avoided. Large masses revolving at high speed and held in bearings will not be adversely affected by any residual couples remaining, which are too small to set up any rock or cause vibration or wear. Small residual static moments, however, left in by the usual balancing methods have shown, and will show up considerable vibration, and it behoves us for this reason to eliminate statical errors to a very high degree of accuracy. *Fractions of ounces of static unbalance* have only too often proved fatal in large rotary bodies; this has not always been sufficiently realised, and as a result damage has often too readily been put down to dynamic unbalance. The following case illustrates this: I had occasion to be dealing with a ship belonging to well-known owners in which the high pressure rotor had twenty sets of bearings replaced during two years' running. This was considered to be due to a gross lack of dynamic balance, and the owners decided to remove the turbine to trace the cause. The rotor was taken out and balanced on a static balancing machine, when it was found to be about 2 ft.-lbs. out statically. This was corrected and the turbine replaced on board, since when the ship has been in service for several years, running as smoothly and perfectly on the same bearings as one could wish. This shows that available balancing methods did not allow one to eliminate the static unbalance sufficiently close enough, and the unbalance remaining proved enough to do the harm. This was then quite naturally ascribed to dynamic unbalance, and has no doubt often been the case with both gearing and rotors.

Simple British machines are available to-day, however, to enable one to eliminate practically all static unbalance. Fractions of ounces or less than inch-pound moments are quickly detected and located in amount and angular position in the largest rotors.

Several geared turbine installations statically balanced as indicated, showed no vibration, appreciable wear, or noise in the gears or rotors after years of service, no money being expended during that time on their upkeep, proving that it is *imperative to obtain as perfect a static balance as possible* on all such rotary parts. As speeds increase this is becoming more necessary, and it is more than likely that before long we shall see insistence on limits of unbalance for gears, rotors, propellers, shafting, etc., according to operating conditions.

The effect of an unbalanced propeller acting at the end of a long shaft may become much magnified at the teeth-faces and result in damage.

Some gearing of large lathe headstocks was carefully balanced, when it was found that the former noise, which the makers had always attributed to the more or less imperfect shape of machine-cut gearing, had disappeared entirely, while moreover, the power to drive them was reduced perceptibly as well as the wear of the bearings. A similar improvement will be found in marine reduction gearing if subjected to a highly accurate balance as outlined, as well as in the rotors and propeller to which they are connected.

Quill-drives, floating-frames, and elastic couplings may be effective for isolating shocks from the teeth, but they do not remove their cause, thus being a localising and partial cure, for if the unbalance happens to be in the gears themselves, the adoption of these devices will not overcome the trouble at all.

An absolutely preventative method is needed, and the only way is to remove the cause by completely balancing all parts of the transmission so far as practicable and thus eliminating any dangerous accumulative moments which might be built up. It is the degree of static balance obtained which counts, the need of which is now acknowledged by several navies and well-known builders in their having adopted the necessary plant.

Incidentally, all rotating parts of internal combustion engines require a similar careful balance, which will enable material reductions to be made in their weights, an all-important matter.

The author has done a real service by pointing out to us that much closer attention will have to be given to the balancing of gearing, and I would like to add my appreciation of his highly interesting paper.

Mr. Alex. F. Ainslie (Power Plant Co., Ltd.), Visitor: I have listened to this paper with very considerable pleasure, because to me it is one bristling with questions, and that, I think, is a very fair criterion of a good paper.

I would like first to ask whether the author could give us any figures for the efficiencies which he mentions. He states the points very well, but he has omitted to give us any particular values from which we can obtain appropriate combined values.

Referring to Section I, what is the present limit to the circumferential speed of turbine blades? I know that it has been increased recently.

As regards gearing efficiencies we can confirm that they are very high, and that the main loss occurs not in the teeth but in the journal bearings. Certainly efficiencies approaching 99½% have been obtained.

There is one point in Section IV which I should like to mention, as I think it has been misunderstood. The author says that in helical gears the duration of contact is longer than in straight gears. If we consider the action in a plane at right-angles to the shaft, the duration of contact is the same, depending on the addenda, the pressure angle and the reduction ratio. The line of contact is certainly longer, but I do not think that that is what is meant.

Again in Section IV, the author says that there are various types of gear in use. Most of the troubles experienced with double reduction gearing occurred with the interleaved type. When my Company discovered its advantages, they set out and designed another type with the high speed casing on top of the low speed casing, with the result that the alignment was not upset by the working of the ship. Flexibility was obtained by a quill attachment in the intermediate shaft with a pin type coupling instead of the usual claw. I believe I am correct in saying that if that type had been copied and adopted when it was introduced, the strong discussions of three or four years ago would never have arisen.

In Section VI it is suggested that the lubrication should be carried out at two temperatures. It has been done, but the general opinion is that this is an unnecessary complication.

In Section VII the author says, "The points of teeth produced by hobs, the teeth of which are of true rack form, are thicker than they should be, and the hob teeth should be modified to correct this." . . . . . . it was not because there was any error in the form of the tooth, but because the sliding velocity was greater at the tips and the material flowed and so formed a wire edge along the top. For cutting it is certainly advisable to mark the blank and the table so that they come into the same relative position when the wheel is turned upside down to cut the second face; but it is not necessary in a double hobbing machine on which right and left helices are cut at the same time.

Another point, which interests me particularly, because I have made experiments on the strength of teeth, is the author's suggestion that the stress on pinion and wheel teeth is the same. It is almost invariably the rule that the pinion teeth are those which break. I suggest that as the bending is greater in the pinion than in the wheel that the combined stress is therefore greater in the pinion than in the wheel. That, of course, is assuming that the material is perfectly homogeneous. Referring to the last Section, No. IX, I think that a straight nickel steel is quite good for pinions. With alloy steels the difficulty is to avoid segregation, and we can rely upon obtaining more nearly uniform material with nickel.

Mr. R. J. McLeod (Managing Director, The Power Plant Co., Ltd.): It has been a great pleasure to me to listen to Mr. Guthrie this evening, particularly so because the subject he has dealt with is one with which I am intimately associated in the course of my business.

Referring to Section IV, Mr. Guthrie gives a ratio of 20 to1 as the maximum for single reduction; this is, however, purely an arbitrary statement, because fundamentally a pinion meshing with a rack gives the very best engagement conditions, consequently the larger the wheel in relation to the pinion, the nearer we get to the rack conditions of engagement. The limiting factor in determining single or double reduction should be size of wheel, whether it can be accommodated or transported to its destination.

A single reduction will always be more efficient than a double reduction, and it has the virtue of greater simplicity; however, in first cost, above approximately 15 to 1 double reduction becomes increasingly cheaper relative to a single reduction of the same ratio.

Dealing with double reduction gears as fitted on shipboard, Mr. Guthrie mentions the "interleaved type." This design was first initiated by my Company, and copied by practically every designer both in this country and abroad. The compactness of the arrangement appealed to everyone. Experience with this design has, however, shown that it lacks essential flexibility in regard to angular displacements which are imposed on the system through the action of the propeller, and it admits of too much flexibility as regards the casing, in this way destroying the correct alignment of the gears.

For the above reasons we have abandoned the interleaved design and introduced an overhead design, where, by raising the intermediate gears so that they are disposed almost on top of the low speed wheel, we obtain a casing with a deep web, thus using the analogy of beams varying as bd^2. By reducing the overall breadth and increasing the depth a very much stiffer casing is obtained, so much so that it becomes to a large extent independent of the deflections of the ship's seating.

I have referred to the need for angular flexibility, and the lack of it in the interleaved design. To understand how important this is we must consider the action of the propeller and the torsional oscillations set up by the blades striking the water, or, when fully submerged, by the blades in turn passing from minimum to maximum pressure due to head.

The oscillations due to any propeller have a periodicity of N x R, where N is the number of blades in the propeller and R is revs. per minute; thus, with four blades and 90 revs., the periodicity has a value of 360. If the total displacement of the propeller at a mean radius, r, be X, then total displacement of low speed wheel would be (neglecting effect of the tunnel shafting):—

$$d = X \frac{R}{r} \qquad (1)$$

where R = pitch radius of wheel.

Now the pinion which drives this wheel would be displaced exactly the same amount as the wheel, neither more nor less, although due to smaller diameter the angle of displacement would be greater. If this pinion be rigidly attached to a primary wheel, as in the case of the interleaved design, then the increased displacement of this primary wheel would be exactly proportional to its radius and that of the pinion to which it is attached, thus:—

$$D = X \frac{R.R_1}{r} \qquad (2)$$

Where R_1 = radius of primary wheel.

D therefore equals the maximum displacement due to torsional oscillation with a rigid double reduction, and d the maximum displacement with a single reduction.

This explains why a rigid single reduction gear will work better than a rigid double reduction gear.

If we introduce a flexible shaft between the first and second reductions, we limit the displacement to the second reduction, so that like the single reduction its value remains d.

In the overhead design we place flexible shafts between the first and second reductions, and thus obtain the necessary angular flexibility; also as explained above, by making the casing rigid the alignment of the gears is preserved independent of the deflections of the vessel.

Section VI.—I fail to see the necessity for two qualities of oil in any system of gearing; to my mind it introduces useless complications. The principal function the oil performs is that of a cooling agent, and only a very small proportion is used as a lubricant.

Section VII.—On the question of gear cutting, it is well understood by gear specialists that the master dividing wheel must be accurate; it must also be large in relation to the work, so as to diminish errors. Introducing creeping tables operated by a differential motion, whilst claiming to average errors, reproduces errors in the creep motion gears on the work, therefore it can only be considered as a poor substitute for an accurate dividing wheel.

In conclusion I wish to express my own personal thanks to Mr. Guthrie for his paper this evening, and hope he will favour the Institute with further papers as time and opportunity afford.

Mr. A. F. Evans: May I be allowed to add a discordant note to this otherwise friendly discussion. I happen to have in my pocket a letter from a well-known firm of engine builders in a large way of business, relative to a proposed marine installation involving the use of reduction gearing, and the following is an extract from same: " . . . the revolutions of the engine being too high, and knowing the trouble that has originated in the past from, and with regard to gearing, and also the general dislike by marine engineers for gearing, we cannot adopt the scheme as put forward."

That decision came to me as a shock; I have very little experience of large marine reduction gear problems and perhaps my principal association with the subject was in connection with a vessel in Germany some few years back, when I was sent over there to observe a geared installation which consisted of two propeller shafts and two Diesel engines geared to each.

These engines were nearly 3,000 h.p. each and turned at 180 r.p.m., while the propeller shafts ran at 70. The engines were rigidly connected to the pinion shaft, no quill or other flexible couplings. I took with me one of those engineer's stethoscopes, in spite of my colleagues amusement at the idea, and I assure you that this instrument was necessary to pick up any noise from the gear box.

I was told that a sister ship fitted in this manner had run very well indeed and that they were encouraged to go still further, double acting engines next and then perhaps four engines per shaft.

The vessel I refer to was the *Monte Olivia*, built by Blohm and Voss and fitted with M.A.N. engines.

For what reason is oil used in gears? Is it for plain lubricating or does it act as a dash-pot to absorb vibrations, or is it for boundary lubrication or a combination of all three.

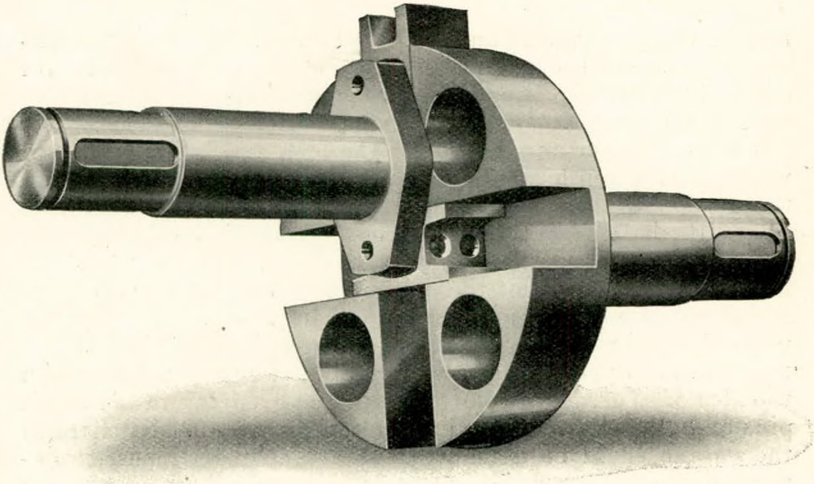
To deal with the first two cases there is no doubt that an oil with considerable viscosity would be the most effective, while to deal with boundary lubrication a thin organic oil would be most suitable. Perhaps the author can tell us how this problem is really dealt with; do you compromise matters by adding a small quantity of organic oil to a medium mineral oil?

Mr. W. Wilton (Royal National Lifeboat Institution), Visitor: I have been interested in the statements relating to inaccuracies of gearing. I would like to ask the author whether the question of grinding has been considered.

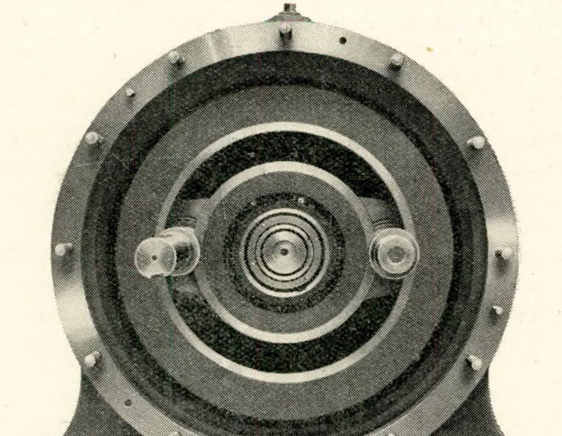
Mr. F. C. Ashby (Royal National Lifeboat Institution), Member: My experience with reduction gears has been only in connexion with comparatively small horse powers relative to the large horse powers mentioned this evening, but I trust I shall be in order in introducing three types of reduction gear, which I think would be of interest to Members.

The Royal National Lifeboat Institution is now adopting the policy of having higher speed engines fitted with reducing gears, with the main object of reducing the weight of machinery per B.H.P., and incidentally giving more operating space in the engine room, and retaining propeller efficiency.

The first type of reducing gear I should like to mention, is the "Burn," which has proved perfectly reliable, efficient, and silent—(one 60 H.P. model 1200/600 R.P.M. having just completed 100 hours non-stop test). The principle upon which this gear operates may be gathered from the illustration which shows the simple 2 to 1 gear. The driving shaft, shown on the left, carries a double crank, forming a tee head, the cranks being of equal throw and each crankpin carrying a slipper. These slippers engage with two slots cut at right angles in the



1. Burn 2/1 Reducing Gear



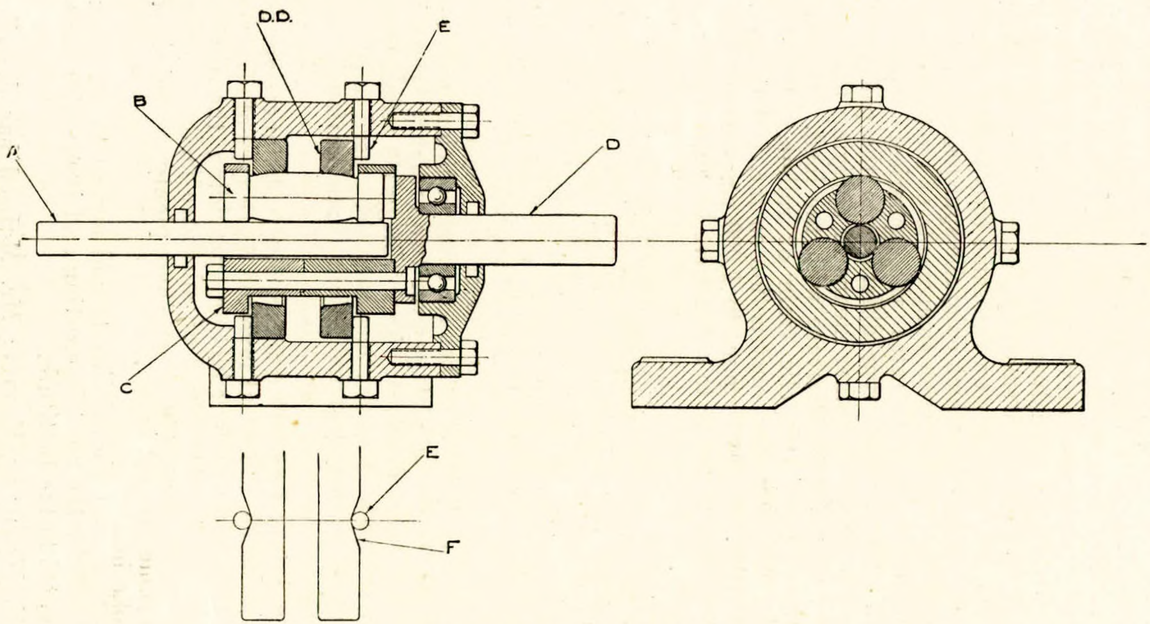
2 Garrard Gear—Parallel Type.

face of the disc carried on the driven shaft. Assuming clockwise rotation of the driving shaft when viewed from the left of the picture, the top slipper presses against the top right quadrant of the disc, causing the driven shaft to rotate clockwise. The lower slipper now travels to the left, and by pressing upwards on the top left hand quadrant takes its part in driving the disc. By the time the lower slipper has reached the top, *i.e.*, after half a revolution of the driving shaft—the slot in which it travels, and shown horizontal in the illustration, will now be vertical, so that the driven shaft will have carried out a quarter of a revolution, thus the 2 to 1 ratio. The normal method of lubricating this gear is by splash or jet, but in setting out this type of gear for the Lifeboat engines, I hit upon the idea of so arranging the oil holes and grooving in the crank pins and slippers, that the lubrication pressure feed system in the engine could be extended right up to the slippers, automatically cutting off the supply of oil as the slippers pass by the inner ends of the cross slots in the driven member, and thus avoiding any " open ends " in the lubrication system. I understand the " Burn " Reducing Gear is made for Horse Powers up to 300 and in units of 2 to 1, and 4 to 1 ratios, up to speeds of 3,600 r.p.m. A mechanical efficiency of 98% per 2 to 1 ratio unit is obtained. I have seen one small model of this gear which, I was informed, had actually done 24,000 r.p.m. ! (*no* load !).

The second type I think would be of interest is the " Garrard " Gear, a 40 H.P. model, reducing from 3,000 to 900 r.p.m. having just completed satisfactorily its preliminary tests for Lifeboat service.

The principle of this gear is that it uses hardened and polished steel surfaces for the transmission of power instead of toothed gears, in just the same way as a heavy locomotive depends on a smooth wheel running on a smooth rail to haul its load. It is a gear of very high mechanical efficiency (99%), absolutely silent in operation, and of very small dimensions when compared with a spur wheel type designed for transmitting the same torque. There are two types—the "Parallel" and the "Spherical."

In the illustration of the Parallel type, the end cover being removed, it can be seen that there is a driving, a driven, and an idler roller, which are located by bearings placed in a straight line, and are surrounded by a floating ring supported by the two outer rollers. This ring moves from its central



3. Garrard Gear—Spherical Type.

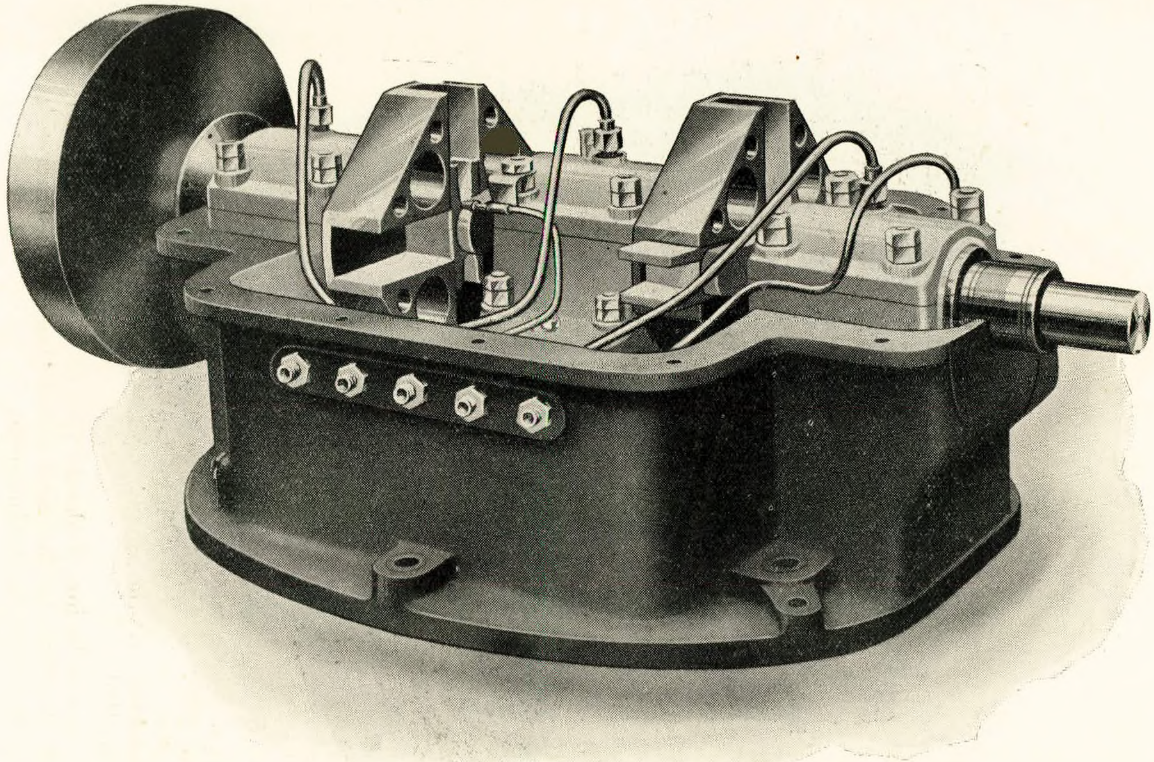
position as the load varies and exerts automatically the necessary adhesion to transmit the load positively with pure rolling contact entirely free from slip. Epicyclic trains—simple and compound—can very easily be arranged in a similar manner, giving all the advantages of epicyclic gearing (*e.g.*, big reduction ratios in a small space; and the keeping of the driving and driven shafts on the same axis), without the low mechanical efficiencies which epicyclic spur gearing inherently has, due to the large friction components of the tooth pressures. The line diagram illustrates the spherical type of the "Garrard" Gear, which type differs from the parallel, inasmuch as it embodies another means of applying the surface pressures, which again increase with the load. It also illustrates the simple epicyclic train of rollers. The largest "Garrard" Gear under construction so far, is I believe, one for 1,000 H.P. at 6,000 r.p.m. to 1,000 r.p.m.; and I understand, reduction ratios up to 250/1 have been designed and run successfully.

The third type of reducing gear I have in mind is the "Brigg's." This consists of a driving toothed pinion meshing into a driven internally toothed wheel, and on the outside periphery of the latter is arranged a series of "Michell" bearing pads—copiously lubricated, which take the torque reaction loads directly over the line of application. It is obvious that this ideal way of looking after the bearing loads of an internally toothed wheel makes for a very efficient gear of this kind, and gives the advantage over two ordinary spur wheels of a closer centre distance. The "Brigg's" reducing gear, however, so far as I know, has only been used on Aircraft, but nevertheless, may yet find a field of application in Marine work.

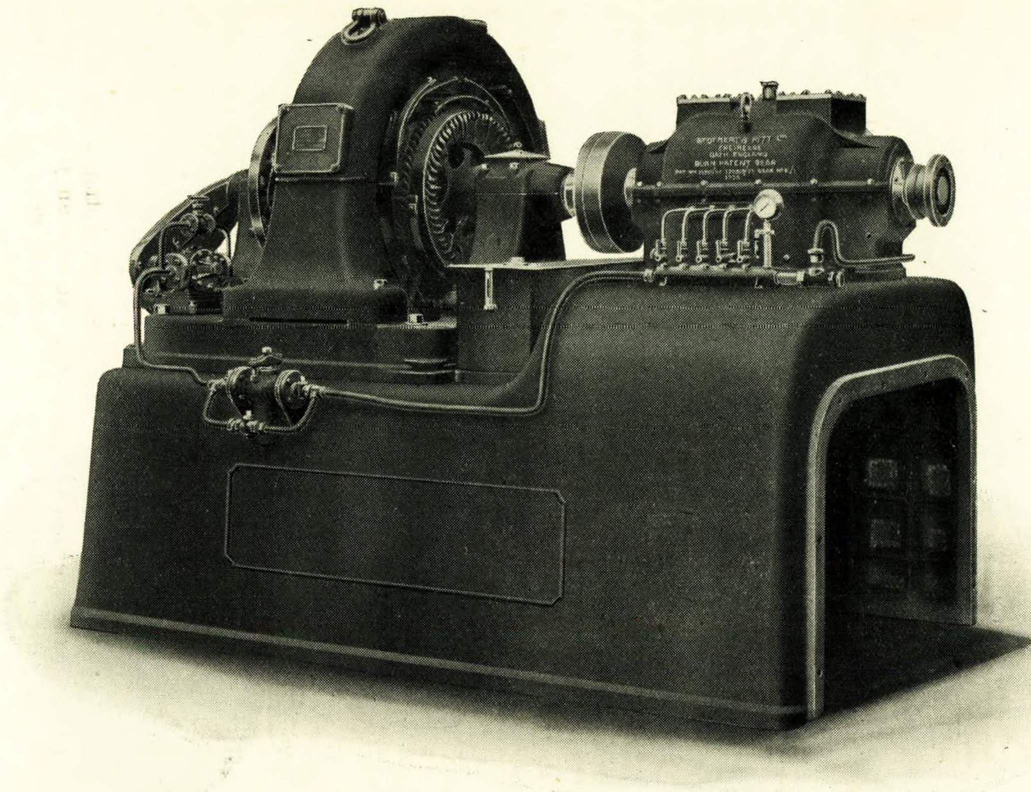
Finally, might I add, that where centre distances are not of great importance, the silent inverted tooth type of chain offers an excellent means of reduction, providing the peripheral speeds are not too high, and an adequate factor of safety is allowed for the chain stresses, and plenty of lubricant is supplied.

In conclusion, I should like to add my quota of thanks to Mr. Guthrie for the most interesting and instructive paper he has presented to the Institute.

Mr. W. Hamilton Martin: Mr. McLeod has described a means of diminishing the effect of the beats of the propellers. I would like to mention that in this year's volume of Trans-



4, Burn 4/1 Reduction Gear—225 B.H.P. 3600/900 r.p.m.



5. Burn 4/1 Reduction Gear—225 B.H.P. 3600/900 r.p.m.

actions of the Institution of Naval Architects I have contributed some particulars of trials of fairly large propellers with respectively 2, 3, 4 and six blades, which were carried out with the idea of finding the best one for some light high-speed vessels. The six-bladed propeller proved the best of the lot in the official trials which were made for the Dutch Colonial Government, although there was probably not 10% difference between the best and the worst type. One of these propellers, which was 7 ft. 6 in. in diameter, broke later on in service and a four-bladed propeller was substituted temporarily. Soon after the new six-bladed propeller arrived and was fitted, the captain of the ship expressed his satisfaction as he said when comparing the two, that it seemed as if the ship had passed again from water into syrup, so much smoother was the running with the six-bladed screw.

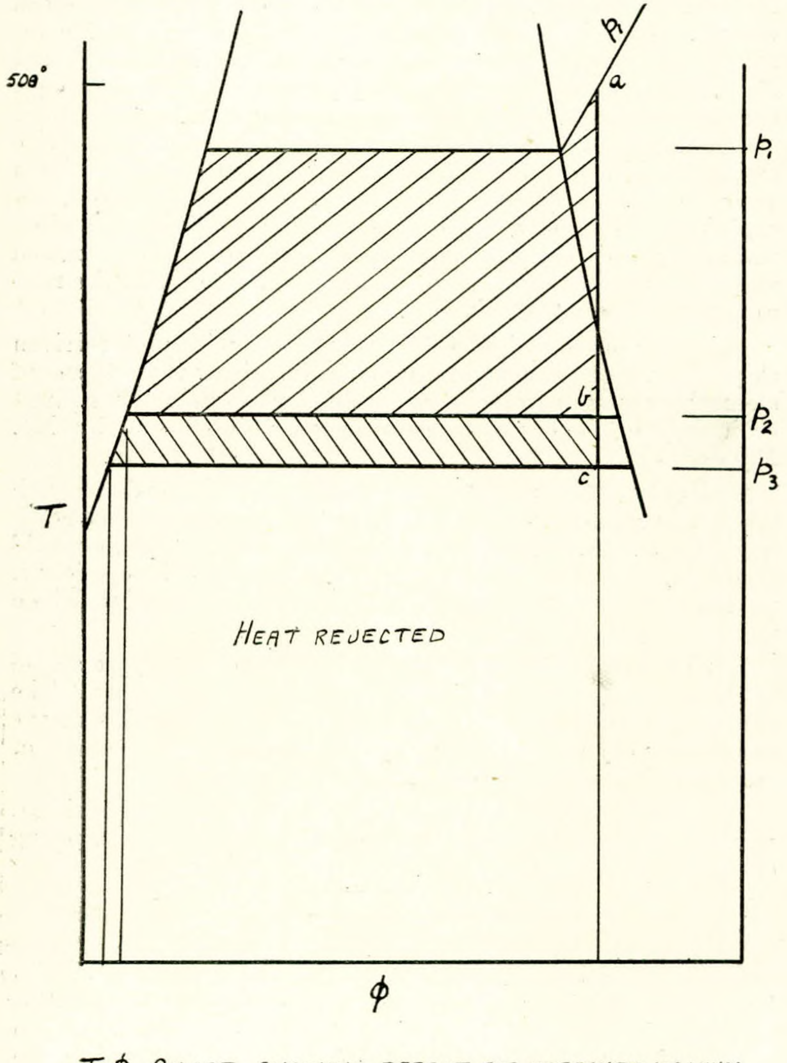
This may prove of interest in connection with motor-driven ships, in which somewhat higher revolutions than those of steamships may at times be desirable (for example, 220 r.p.m.) or it may be beneficial to gearing as pointed out by Mr. McLeod.

It might be well worth while to carry out tank tests on such a series of model propellers to ascertain whether higher revolutions with more blades of less width in the axial plane would result in smoother operation, without sacrifice in efficiency, for application to double-reduction geared or oil-engine driven vessels of moderately high speed.

THE CHAIRMAN: It gives me great pleasure to propose a vote of thanks to Mr. Guthrie for his very interesting paper. The subject is rather a difficult one to discuss, and I think we have benefited very considerably by the Author's treatment of certain points which deserve careful consideration.

The vote of thanks, which was carried with enthusiasm, was briefly acknowledged by the Author and the meeting closed.

CONTRIBUTION BY MR. J. WARD, B.SC.: In section 1, the Author refers to the method of improving the efficiency of a turbine by adopting higher pressures and temperatures. I am rather surprised he does not mention the increase in efficiency due to lowering the exhaust pressure, *i.e.*, by improving the vacuum. Quite as much work has been done by the designer of condensing plant to improve turbine efficiency as by the designer of high pressure and temperature boiler plant. If one examines curve A. on fig. 3, one notices that the rate of increase of thermal efficiency decreases as the pressure increases.



T ϕ CHART SHOWING EFFECT OF IMPROVED VACUUM.

Fig. 1

It is interesting to show that the rate of increase of thermal efficiency increases as the exhaust pressure decreases. Consider the case of turbine supplied with steam, say at 200 lb. per square inch absolute and superheated to 230° C. This initial condition is represented by the static point A. on the temperature entropy chart fig. *i*. If expansion takes place to a pressure p_2, the final condition represented by b, the heat equivalent of the work done is the shaded area. If the exhaust pressure is lowered to p^3 the gain of work will be the shaded strip and the point c represents the final condition.

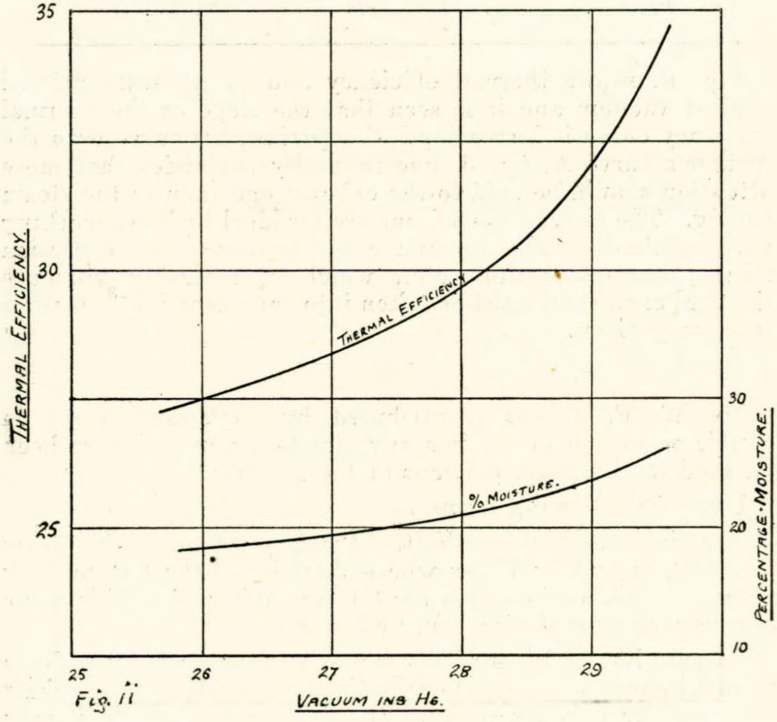


Fig. II

The accompanying table gives the thermal efficiencies and final percentage moisture for various exhaust pressures.

Exhaust pressure lb per sq. absolute.	Vacuum ins Hg.	Heat equivalent of work done C.H.U. per lb.	Thermal Efficiency.	% Moisture.
0·2	29·6	236	·347	26
0·4	29·2	218	·325	24
0·5	28·98	214	·321	23·5
0·6	28·78	209	·316	23
0·8	28·37	200	·304	22
1·0	27·96	195	·296	21
1·5	26·94	183	·284	19
2·0	25·91	175	·274	18

Fig. ii. shows thermal efficiency and % moisture plotted against vacuum and it is seen that the slope of the thermal efficiency curve is increasing. Comparing this curve with the Author's curve A, fig. 3, one naturally concludes that more attention should be paid to the exhaust end than to the steam supply. The above comparisons are for ideal turbines working with adiabatic expansion and other factors such as friction reheat, supersaturation, etc., which considerably influence the final condition, must be taken into consideration to make a true comparison.

Mr. W. F. Jacobs (contributed by correspondence): In double reduction gears, has any standard nomenclature been adopted for the main portions of the gearing?

I use the following terms:—

For pinion on turbine shaft, "Primary pinion"; this gears into "Primary wheel" on whose shaft is cut the "Secondary pinion"; this meshes with the "Great Wheel." (This name is borrowed from clock work, by the way.)

I would like to know if any investigation has been made as to the chances of torsional oscillations being set up in the shaft carrying the primary wheel. If this does take place, it may cause very irregular loading on the primary pinion and so tend to fracture.

In the case of long ships I am in favour of fitting a flexible coupling between the great wheel and tunnel shaft in order that any stresses on the shaft due to the ship working should not have any effect on the gearing. I would like to know how the various parts of the gears are dynamically balanced; are the gears placed in a frame and run up to working speed to check the dynamic balance?

I consider that some of the causes of fracture of pinions in a double reduction gear may be that (1) in a double reduction gearing, the speeds of the turbine and primary pinion are from 2 to 4 times that of the corresponding single reduction gear, therefore the stresses due to lack of balance, etc., are 4 to 16 times greater in the double gear than in the single gear.

(2) Again, we can expect that the life of a single reduction primary pinion is longer in terms of time than that of a double reduction primary pinion, assuming that the final speed in revolution of the propeller is the same in both cases, for the teeth are only in engagement from $\frac{1}{4}$ to $\frac{1}{2}$ the number of times per unit of time in a single reduction gear compared with a double reduction gear, or, if a pinion of a double reduction gear is worn out or fractures in, say, one year, then we can expect that in the corresponding single reduction gear the pinion will last two to four years.

Contribution from Mr. R. J. Willis: Referring to the Paper entitled "Notes on Reduction Gear" which was read recently before the Members of your Institute by Mr. W. J. Guthrie, I have studied with intense interest the various points raised, but in two instances I am completely at a loss to grasp the true significance. In the hope that you may be able to throw some light on these points I take the liberty of bringing same to your notice, assuring you that I would be most grateful if you could explain in a clear and concise manner the idea or purpose which Mr. Guthrie wished to convey.

Under the sub-title "Gearing Efficiencies and Design," Mr. Guthrie states:—

"The efficiency of reduction gearing has been accurately determined by driving two similar gear in OPPOSITION by a motor interposed between one set of pinions and placing a generator between the other set of pinions."

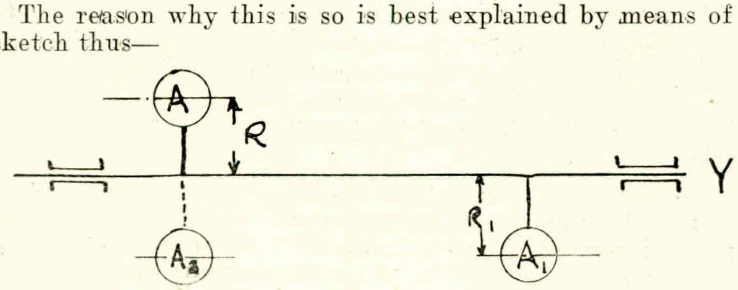
Could you indicate, by means of a rough sketch, the general lay-out of such an arrangement and demonstrate the manner in which gears are driven in opposition. Again, I am led to believe that second gearing and pinion system would be required to bring about "opposition." Is this correct? If so, could not efficiency be determined without resorting to duplicate gearing, etc.? Again thanking you and trusting to hear from you later.

Mr. Guthrie: I appreciate the manner in which these notes have been received and thank those gentlemen who, by taking part in the discussion, have materially increased the scope and enhanced the value of the paper.

The Chairman mentioned a case of a gear wheel which became slack on the cone and after being refitted gave no further trouble. It would appear probable that the wheel had been improperly fitted to the shaft in the first instance and that instead of the conical surfaces of the wheel boss and shaft being in intimate contact throughout their length and especially so at the large end of the cone, the area of contact when first fitted had been mainly confined to the small end of the cone. Cases have been known where this was found to be the explanation of propellers becoming slack and tail shafts breaking in the coned portion of the shaft.

Mr. W. Hamilton Martin rightly points out that smooth and silent running is only possible when all rotating parts of a system possess a high degree of running balance. It may be well to point out, however, that this condition is not necessarily complied with when all rotating parts are in a state of static balance. Improving the static balance may make the running balance worse instead of better, particularly if the axial length of the body is considerable and the speed of rotation high.

The reason why this is so is best explained by means of a sketch thus—



X-Y is the axis of a rotating body having an out of balance weight A ozs. acting at radius R, and having a moment relative to the axis X-Y of AR inch/oz. To produce static balance all that is necessary is to place a weight A_1 diametrically opposite at such a radius R_1 that its moment will be equal to AR, *i.e.*, $AR - A_1R_1$ and this will apply to whatever point the weight is applied along the axis of X-Y, whereas the only point at which it can be applied to ensure that there is neither static nor dynamic out of balance, is as shown by the dotted lines.

In the case of bodies whose length is relatively short, static balancing will generally yield satisfactory results especially if the weight is applied in such a way that it has no axial moment about the centre of gravity of the body.

Mr. Martin also, I think, suggests that if all parts were in perfect balance quill drives, elastic couplings and floating frames would be unnecessary, but here again I think some slight qualification is desirable. The real reason for adopting the quill drive or what amounts to the same thing, an elastic coupling, has been clearly explained by Mr. McLeod, and the reason put forward for the adoption of the floating frame is that it automatically adjusts the alignment and so enables a higher tooth pressure to be used.

Mr. Ainsley has raised several points well worthy of attention. In the first place he asks for definite values for combined efficiencies obtainable with high pressure turbines. No results are yet available for high pressure marine turbines, but in power station work it is claimed that the use of high pressures has enabled efficiencies of 27 or 28% to be obtained from steam to buss bars.

The maximum circumferential blade tip speed of which I have any record is 1,050 feet per sec.

As regards duration of contact, I regret, if in attempting to be brief, I have not made myself quite clear. What I did mean to convey was that due to the helical form of the tooth any particular tooth was longer in engagement than in an ordinary spur gear.

Mr. Ainsley points out that bending stresses may cause the teeth of the pinion to be more severely stressed than the teeth in the wheel and in this I concur. A further disparity of stress is also due to the fact that the contours of pinion teeth are further removed from the true rack form than are the teeth of the wheel, but these facts were in my mind when I said, not that the stresses were the same, but that they were sensibly the same. In any case giving the disparity its utmost value the differences in the stresses are nothing like 45 to 28, which is the ratio of the ultimate strengths of the material of pinion and wheel rims, and any difference in this respect can therefore have no direct bearing on the vexed question as to why nearly all tooth failures are confined to the pinion.

Mr. McLeod's contribution to the discussion is a valuable one and he has dealt with the question of torsional vibrations in a very clear and lucid manner.

Mr. Evans quotes an objection raised by a firm of engine builders to the adoption of reduction gears. As one who is himself associated with a firm of engine builders, I am unable to understand this attitude. No more convincing testimony to the reliability of reduction gears could possibly be forthcoming than the important repeat orders for geared turbine installations recently placed by various shipowners, all of whom have had such installations, and are well qualified by experience to judge the merits of them.

The oil is not used in any way to 'damp' out vibrations, but simply to lubricate the teeth and carry off the heat generated by friction. In selecting a suitable viscosity the oil must not be too thin, otherwise it will be thrown off the teeth and it must not be too thick, otherwise it will not find its way properly between the tooth surfaces.

In reply to Mr. Wilton I may say the only ground gears known to the author are those of the Maag type having straight spur teeth.

Mr. Ashby's remarks are very interesting, but deal with a type of gear of which the author has had no experience.

Mr. Jacobs asks if there is a standard nomenclature for reduction gears. I do not know of any, but the terms used in the paper are those in common usage by marine engineers.

In reply to Mr. Willis, if a motor is coupled to one pinion of a single set of gears and a dynamo is coupled to the other pinion, the efficiency of the gears can be determined by comparing the input power to the motor with the output power from the generator. The difference when corrected for the efficiencies of motor and generator is the lost power due to frictional losses in the gears. The motor and generator may be coupled electrically as in the well-known "Hopkinson test." The disadvantage of this method, however, is that the gears can only be tested at the loads and revs. for which the motor and generator are suitable.

When two sets of gears are available it is preferable to run the two gears together as shown, as the frictional losses which form a very small percentage of the total power are thus doubled and more easily determined.

Another method frequently adopted is to run two gears in opposition, the generator in the sketch being replaced by a transmission dynamometer. This enables the teeth to be loaded to any pressure and the output required from the motor is only that required to overcome the frictional losses. If a dynamometer is not available the tooth pressure may be obtained by twisting the two pinion shafts relative to one another and coupling them together while in this strained condition.

Mr. Ward deals with the effect of increased vacuum, and provided the turbine is so bladed that the leaving loss percentage is not large, any increase in vacuum will result in a gain in economy.

This aspect of the subject was not touched upon in the paper however, because much has been already written on the effect of vacuum on Turbine Economy and because vacuum has no direct influence in increasing the revolutions at which the turbines are designed to run. On the other hand within limits, the greater the pressure the greater the revolutions for which the turbines should be designed and consequently the greater the necessity for the use of reduction gears.

Mr. Ward's diagrams show the effect of vacuum on the thermal efficiency very clearly and in this connection it is indeed a striking fact that nearly as much heat energy can be obtained from the expansion of 1 lb. of steam from 2 lbs. per sq. in. abs. to 1lb. per sq. in. abs., as from the expansion of 1 lb. of steam from 200 lbs. per sq. in. abs. to 100 lbs. per sq. in. abs.

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Notes.

Report No. 5, Marine Oil Engine Trials Committee. — This Report was discussed at a meeting held at the Institution of Mechanical Engineers on December 10th. It embraced the trials of the *Cape York*, and the following summary of the Report and the discussion is from "Lloyd's List and Shipping Gazette," of December 15th:—

The report covered the testing of the engines ashore and the engines and auxiliaries at sea, including a series of progressive trials on the measured mile.

The trials at sea were carried out using fuel of which the qualities were practically identical with those of the fuel used ashore, and sufficient records of the performance of the main engines were made to enable the results of the tests ashore to be utilised in deducing the power actually delivered to the propeller shaft. The *Cape York* was built by Messrs. Lithgow, Ltd., and was designed to the following general particulars: Length b.p., 410 ft.; moulded breadth, 53 ft. 9 in.; and mean draught to the bottom of the keel, 26 ft. ½ in. The main propelling engines, which are of the Werkspoor type, were built by Messrs. R. and W. Hawthorn, Leslie and Co., Ltd., at Newcastle-on-Tyne. The rating of each engine is 1020 b.h.p. at 125 r.p.m., the power being developed in six cylinders acting on the four-stroke cycle, single acting, with six cranks. The method of fuel admission to the cylinders is by blast air from an engine-driven compressor, and firing is obtained by the temperature of compression. The cylinders are 22.05 in. in diameter, and the piston-stroke is 39.37 inches. The fuel is injected at a pressure of 925 lb. per square inch by means of a plunger pump, the supply to the cylinders being regulated by means of a hand-control, and also by the engine governor. The lubrication is effected from a gravity tank and can also be delivered by a force pump.

Tests Ashore.—The first series of tests were carried out ashore at Messrs. Hawthorn, Leslie and Co.'s works, and were conducted to a schedule of loads and speeds calculated from the simple approximation to propeller law. The constant in the relation, Torque = constant × (square of revs. per minute), was determined from the condition of rated full power of the engine, namely 1020 b.h.p. at 125 revolutions per minute. The procedure adopted was that the attendants were informed what brake-load and speed would be required for each particular test, no other conditions of running being laid down. The timing of the fuel admission to the cylinders was that at which it was intended the engines should be run at sea. Tests of over-loading, friction, slowest speeds, etc., were also carried out in addition to the main tests. Samples of exhaust gas were analysed, and samples of fuel were analysed at the Fuel Research Station, Greenwich.

Curves of i.h.p., thermal efficiency, fuel consumption, mean pressures, etc., were plotted, and a most exhaustive series of tables and diagrams compiled. The data so obtained are very clearly set forth, and the deductions made follow immediately from the observed particulars, sufficient information being recorded to permit of further deductions being made. The weight of fuel per b.h.p. hour at full load is about .43 lb., and the thermal efficiency on a b.h.p. basis is about .30. It is interesting to note how these figures vary as the load and speed are altered.

Sea Trials.—The trials at sea were carried out in the Clyde estuary, and consisted of measured-mile runs, starting air trials, manœuvring trials, slow speed tests, and fuel consumption trials. The vessel's draught was 8 ft. 6 in. forward and 16 ft. 5 in. aft, some 2,276 tons of fuel and ballast being on board at the time. The moulded displacement at the trial draught was 5,510 tons, as against 12,355 tons under full-load conditions. A progressive series of runs from half the designed number of revolutions of the engines by stages to the full number were made on the measured mile. A most comprehensive and complete set of observations were recorded, and the data so obtained were utilised to calculate the horse-power, torque and efficiency under varying conditions. Direct measurements of the torque transmitted by the port engine were made at the same time, but no attempt was made to obtain direct measurements of the thrust.

It is seldom that an opportunity is so afforded as to measure directly and indirectly at the same time the torque on the shaft, and the results obtained were approximately the same, the slight difference being due to losses on shaft transmission. Interesting features noted were that the number of starts possible with the starting air capacity installed was 43, and that the slowest possible number of revolutions was about one-third of the number at full speed. The Trials Committee simply record the observations obtained at the trial and make no comments upon the information gathered.

Discussion.—Mr. C. W. J. Taffs, in presenting the report, said it followed very closely in its arrangement on that of the four previous reports in order that comparisons could be readily made as and when opportunity arose. Although the fifth report was now under discussion an opportunity to discuss comparisons would be afforded later on when a paper correlating results would, it was hoped, be presented. The principal points that called for discussion in the descriptive part of the report were the advantages of electricity as the mode of drive for the deck and other machinery, and the advantages of driving multifarious pumps from the main engines; whether it was economical to use air at a relatively low pressure for manœuvring purposes, and whether the electric generating machinery should be driven by oil engines. There was nothing that called for particular comment in the shore trials themselves except that the mechanical efficiency was lower than might be expected, but perhaps that abnormality was only apparent seeing the number of pumps the engine was called upon to drive.

That trim might have some effect on the data deduced from engine observations, particularly at the lower powers. The principal results of the sea trials, other than those shown in the table already referred to, were the over-all consumptions of fuel and the consumption of lubricant in the main engines. A new type of torsion-meter was mounted on the port shaft, and this was found to give very consistent readings. Unfortunately, no records of the thrust were made as the Michell block installed was not adapted for such measurements.

Sir Archibald Ross said a word of special thanks was due to Sir James Lithgow, who readily consented to place the *Cape York* at the disposal of the Committee when his company approached him on the subject. The value of the reports could not be overstated, and though one might be inclined to say from the shipowner's point of view that the only three initial points which were worthy of his consideration were cost, weight (which affected carrying capacity), and fuel consumption, there were, of course, other factors, such as maintenance and reliability. The objects of the Committee were technical and scientific, and all the reports were conspicuous by a very wise and necessary absence of bias towards any one particular type or peculiarity, but from the shipowners' point of view in particular, and also from the Institutions', any supplementary information which could be obtained from actual sea performances over long periods would be of great value. One realised that the impartial attitude of the Trials Committee did not permit of comparisons being made by them, but they were glad to know that Professor Dalby intended reading a paper shortly, based on the now completed five reports.

Need of Thrust Records.—It was from the naval architect's point of view, perhaps, regrettable that trials carried out so thoroughly in all respects as far as machinery was concerned did not in all cases include reliable thrust records, which, after all, for ships' designers were the most important factor. There would appear still to be difficulty in establishing the actual indicated horse-power, a question of primary importance, not only in estimating the mechanical efficiency of the engine, but in ascertaining the maximum pressures. Even with the best indicators obtainable, inaccuracies were on evidence. He had no doubt that the difficulties associated with indicating the power of internal combustion marine engines would be dispelled, but at present there could be little doubt but that they did exist. He concluded by paying a tribute to Sir George Goodwin, who had done so much to instil enthusiasm and obtain thorough knowledge of an engine which had got to be mastered.

Mr. Narbeth, speaking as a naval architect, observed that we in this country lagged very much behind in adopting the Werkspoor engine, which was an engine possessing very excellent merits, and because of its great vogue abroad there must be admirable features which the trials had not brought out.

Mr. T. Clarkson said all would agree that the report was a most valuable contribution to the investigation of the marine oil engine. One criticism he would venture to express was that there was nothing to indicate that any steps were taken to recover the very large amount of heat which was going away in the exhaust gases.

Mr. W. A. Tookey, who exhibited some slides, drew attention to the peculiarities in the indicated horse-power shown in the report, showing that there was something wrong with the indicating gear.

Mr. Baker said that the report was a little better than before with regard to the thrust measurements. It looked at last as though there was a torsion meter which was satisfactory. The shipowner was concerned only with thrust measurement; the others were purely engineering points. The trials were to supply the owners with some data to enable them to get more confidence as to what an oil engine would do.

High-Speed Engines.—Sir Ernest Petter suggested that it was a pity that an opportunity could not be taken of comparing the results of slow-speed engines with engines of higher speed. His opinion was that the internal-combustion marine engine would go through very much the same stage as the steam engine, and while marine engineers of to-day would only look at speeds corresponding with steam-engine speeds, the time might come—and might not be far distant—when they would exhibit engines to give higher revolution speed. If they got rid of a prejudice it would be worth investigating, and he did not know whether the Committee could carry a test in engines of higher speed. Apart from the question of propeller efficiency, there was a great deal to be said for higher speed in oil engines. The real point was how much thrust power they were going to give the shipowner.

After some further discussion, Mr. Taffs dealt with points which had been raised by the several speakers.

Owing to the enquiries made regarding the use of pulverised fuel attention is directed to the Paper read by Dr. W. Lulops, Amsterdam, at the International Conference on large electric systems held in Paris in June, 1925.

"The Machine Tool Review," of September and October, 1925, contained a full report of the Paper from which the following points are selected to indicate the scope of it:—

The main object of the author was to make a comparison between the two most important systems in powdered fuel stoking; the Central System and the Unit System.

Starting with the comparison between the mechanical stoker and the firing of powdered fuel in general it must be borne in mind that the combustion of the coal is merely a chemical reaction between the oxygen of the air and the carbon, hydrogen, etc., of the coal. Looking closely into this matter it is obvious that the conditions under which this chemical process takes place must be far less ideal for the mechanical stoker.

As the coal burns away steadily on the mechanical stoker whilst it is transported, first the volatile and then the carbon, leaving only the ash at the end, the thickness of the layer of coal along the stoker gradually diminishes towards the end of the stoker and the quality of the fuel diminishes correspondingly.

This has the effect, assuming an equal draught over the whole stoker, that the ratio of quantity of air per lb. of combustible matter varies considerably along the grate, causing an excess of air at the end, a shortage of air at the front part of the stoker, also that the stoker at the end has to deal with very inferior quality of coal containing no volatile and an ever increasing percentage of ash, theoretically up to 100%.

This results in a low percentage of CO_2 at this end of the stoker, on an average not more than 5% and a great difficulty in burning out the coal completely. Here we have two sources of loss. The uneven distribution of air makes it impossible to run at a high average CO_2 without CO; in fact this is the explanation why with the mechanical stoker one cannot go beyond 13-14% CO_2 without CO, whereas with powered fuel 16-17% may be obtained.

The other loss is a certain percentage of carbon left in the ash, also due to the cooling effect of the boiler on the grate, 5-10% of carbon on an average being a low figure. In fact one has to compromise between these two losess. A high percentage of CO_2 means high loss by unburnt carbon in the ash, a low loss figure in the ash necessitates a lower percentage of CO_2. The unavoidable percentage of unburnt carbon in ash and flue dust is largely dependent on the character of the coal. For instance if the melting point of the ash is low it is practically impossible to avoid a considerable percentage of carbon in the ash; particles of coke and even coal are shut in by the molten ash and so become partly or entirely shut off from the oxygen, which means that this part of the coal remains unburnt.

Another sure loss when using fine coal is to be found in the particles which are blown off the grate. In order to burn fine coal, especially if it contains a high percentage of ash, a low percentage of volatile, and if it is of a slow burning nature, one has to use forced draught, forcing the air through the layer of coal; but with increasing force it stands to reason that an increasing number of coal particles are blown off the grate and are carried away with the gases through the boiler. Some stop there and tend to clog up the boiler and economiser, leaving in the former big pieces of clinker as a result of slow dry distillation; owing to the higher temperature, the finer particles leave the chimney.

Owing to the defects which are inherent to the nature of the mechanical stoker it requires much skill to operate it efficiently, and therefore taking into consideration the average attention with which one must reckon, it means that the actual loss will be greater than that previously mentioned.

And this is one of the reasons why there is, in general, such a vast difference between the test trials run with mechanical stokers by expert attendants sent there on purpose by the manufacturers of the stoker and the results obtained in ordinary running condition.

To this must be added extra stand by and starting up losses on account of the lack of adaptability of the chain grate stoker to variations in the load and the considerable amount of time it takes from starting up to efficient running condition.

In view of the enormous number of mechanical stokers now in daily use it may prove of value to point out how the adaptation of powdered fuel in combination with existing mechanical stokers can reduce the aforementioned principal defects and by so doing tends to bring the mechanical stoker more up to the standard of powdered fuel firing. At the same time this combination may overcome some of the difficulties met with in the latter method of firing.

This combination has been effected by inserting a number of small burners (or jets) of powdered fuel at the back end of the stoker in such a manner that a row of small flames from these burners are directed, almost parallel, to the mechanical grate but pointing slightly towards it. This results in three distinct advantages over the mechanical stoker alone, viz.:—

A. Increased efficiency.

B. Increased steam production of the boiler.

C. Increase in the adaptability for a varying load.

Re A. 1. This row of small flames acts as a perfect form of back arch with all its advantages and none of its disadvantages especially in respect to the upkeep and repair of brickwork.

2. The amount of excess air at the back of the mechanical stoker is utilised for completing the combustion of this powdered fuel by regulating the amount of air supplied in the mixture to the burners.

3. As the flames from the burners are at right angles to the flames from the mechanical stoker a whirling motion is set up in the combustion chamber by which means excess air and combustible matter are brought in close contact with one another.

Re B. Increased evaporation is obtained not only by the extra amount of BTU's set free from the combustion of the powdered fuel, but also through a higher temperature in the combustion chamber. More fuel can be burnt per square foot of grate surface, the boiler being capable of absorbing these extra BTU's on account of this higher temperature. This is a well-known fact and is also the reason why the temperature of the superheated steam is not increased by the higher temperature in the combustion chamber. Actual tests have proved that with a 20% increased evaporation the temperature of the exit gases from the economiser was not raised. Even 30% increased evaporation was reached, but could not be continued on account of the boiler beginning to prime.

Re C. By this means the production of the boiler can be instantaneously adjusted by regulating the supply of powdered fuel and further the starting up of the boiler takes considerably less time because the powdered fuel flames, once started, bring up the temperature of brickwork and combustion chamber very rapidly.

On the other hand the mechanical stoker will assist the powdered fuel firing; firstly the ignition is always sure and secondly the ash from the powdered fuel is removed as it falls on the mechanical stoker.

It will be obvious from paragraphs B and C that the further advantage of this adaptation of powdered fuel is that the steam production of an existing boiler house may be thus increased without any extension of boilers.

Returning now to burning powdered fuel proper I would mention that the systems of which one hears the most at present, as Lopulco, Fuller, Holbeck, are all based on the principle of the Central System and if I am well informed this system is by far the most popular in America, which is the cradle of pulverized fuel. Although in the beginning the Unit System equally was used it seems that in the further development it has been left behind more and more.

Considering that the suppliers of pulverized fuel plant are, nearly without exception, representatives of American industry in this branch and are therefore influenced entirely by experience obtained in that country, the Central System is more and more brought to the front. There are not many who have had the courage to gain experience by making tests and trials themselves, although this is the only way, in present circumstances, to form an unbiased opinion on this important question.

About four years ago when I thoroughly studied the various systems then arising, I immediately became convinced that, however important the differences are between the various systems such as Lopulco and Fuller the most important of all is the problem of choosing between the Central and Unit Systems. A thorough study of the advantages and disadvantages of each convinced me that in principle the Unit System was to be preferred to the Central System.

So I confined myself to going thoroughly into this system and not without results—results in two directions. In the first place, as I shall presently prove, this system has been adapted for practical use with most satisfactory results; secondly, this strengthens my conviction that the Unit System will in most cases compete successfully with the Central System for many reasons into which I hope to go into more fully presently.

The peculiar difficulties which arise in pulverized fuel stoking, especially if one has to depend entirely on personal initiative, demand perseverance and patience. For about three years I learnt nothing excepting "how not to do it," until at last an installation, in which all objections have vanished, has been working regularly for eight months.

BOILER EXPLOSIONS ACTS, 1882 AND 1890.—Report No. 2770. While the *Lyntown* was on her way from Algiers in ballast on July 9th, 1925, and two hours after leaving, a leak was noticed in the main steam pipe near the main engine stop valve chest, leading from the port boiler, adjoining the branch from the starboard boiler. The leak gradually became greater. It was decided to return to Algiers in order to investigate the cause, and find a remedy.

A clip was round the pipe and on its removal, a circumferential crack was traced about 6 inches long. Prior to the ship's departure from Cardiff on the voyage, a slight leak was noticed and on examination a pin point drop was found, this led to the clip being bound round the pipe to cover the spot, and evidently it had extended as the heat and the cold led to expansion and contraction.

There were two boilers, pressure 160 lbs. From each of the stop valves, the steam pipe was led to the engine stop valve chest on which there was a branch for each. The connecting pipes were led on a line with the backs of the boilers, then with bends aft to the engine room at right angles, without expansion bends or joints to relieve the strain. The pipe from the port boiler had been repaired by brazed sleeve joint at the engine stop valve chest. The sleeve was made of solid drawn copper 4 inch internal diameter, brazed to the flange and expanded for a length of $3\frac{3}{4}$ inches to allow the pipe to enter. It was afterwards brazed. The pipes were tested by hydraulic pressure to 320 lbs. on April 22nd, and the vessel sailed from Cardiff on June 25th. The attention of the owners was called to the advisability of providing for expansion. The investigation of the cause of the failure was carried out by Mr. J. A. Whyte, Board of Trade Surveyor, Liverpool, and the closing observations of the Engineer Surveyor-in-Chief, were to the effect that the formation of the pipes was such that considerable alternating stresses were imposed on the necks at the flanges, and also to that due to the racing of the engines in heavy weather. The attention of the owners having been called to the subject, they would no doubt give careful consideration to the subject. When the defect was seen in its initial stage, it is surprising that a clip was considered a sufficient safeguard before the vessel sailed.

CARRYING COALS TO NEWCASTLE.—Sir Wm. Noble, Chairman of the Tyne Commission, stated that during the period covered by the stoppage of the coal production from the home mines, 354,392 tons were imported from abroad.

CARRIAGE OF FRUIT.—The following is from the "Times" of January 10th :—

DISEASE IN APPLES.—PREVENTION OF BITTER PIT.—A review of the problem of bitter pit in apples, issued by the Department of Scientific and Industrial Research (H.M. Stationery Office, 1s.), shows that it is not known whether it invariably originates on the tree or whether sometimes it may develop *de novo* in storage. Consequently it cannot be said definitely whether orchard practices which have been found effective to some extent in preventing bitter pit actually control the cause of the disease, or merely influence the susceptibility of the fruit.

A survey of the published data yields two fairly definite results: that severe pruning on the one hand, and heavy irrigation towards the end of the season on the other, are circumstances predisposing to bitter pit. This has suggested the conclusion that the relations of water to the growing fruit are concerned with the disease. In certain varieties bitter pit attacks the fruit of the tree, but in others it seems to be inhibited while the fruit is on the tree, although it may develop rapidly after picking. In the latter case it has been found to develop in storage much more among immature fruit than in fruit allowed to ripen on the tree. Cold storage has been found to retard it in some varieties which develop it in storage rather than on the tree, but contradictory results have been obtained in other cases. The retarding effect of cold storage, where present, is apparently reduced by a delay in storing.

The theories advanced to explain bitter pit are discussed. It is pointed out that no direct experimental verification has been obtained, or, in most instances, even sought, while most of the theories are open to grave objection. The commercial importance of the problem lies largely in the development of bitter pit in export fruit during its carriage oversea. It is tentatively suggested that a combination of later picking with some rapid refrigeration may be found to give the best results and further experimental work along these lines is recommended.

Our present knowledge of the disease is critically discussed with particular reference to some hitherto unpublished results dealing with its development in stored fruit. The temperature of storage is usually thought to have a marked effect on the rate of development of bitter pit, in the sense that low storage temperatures are considered partially or completely to arrest the disease. Experiments in Australia with the varieties Ribston Pippin, Cox's Orange Pippin, and Cleopatra showed a marked contrast in behaviour between Ribston Pippins and Cox's Orange Pippins on the one hand and the Cleopatras on the other. The Tasmanian varieties showed the expected retarding effect of cold storage. The Cleopatras, which were badly pitted to begin with, developed bitter pit rapidly in cold storage but hardly at all at ordinary temperatures.

Books added to the Library.

Purchased: Culliford's "Commercial Shiprepairing and Estimating." Published by E. Culliford & Co., Liverpool.

Election of Members.

List of those elected at Council Meeting of January 10th, 1927 :—

Members.

James McLean Anderson, 109, Arkleston Road, Paisley, Renfrewshire.

Thomas Tangaroa Bollons, *c/o* New Zealand Office, 415, Strand, W.C.2.

Edward Alfred Copp, *c/o* London Scaling Co., Ltd., 1, Pier Head Chambers, Cardiff.

Robert Arthur Stewart Duncan, Engineers' Quarters, Thornhill Road, Stanhope Lines, Aldershot.

Thomas Henry Faulks, 18, Merches Gardens, Cardiff.

John James Harding, 6, Azalea Avenue, Sunderland.

Stewart Hogg, 124, Tressilian Road, Brockley, S.E.4.

Abraham Hamilton Kemp, Capt., R.A.O.C., Ordnance Depôt, Dover, Kent.

Louis Adam Macfarlane, China Navigation Co., *c/o* Messrs. Butterfield & Swire, Hong Kong.

Finlay Sinclair McLay, *c/o* Messrs. Butterfield & Swire, Hong Kong.

Alexander McVean, China Navigation Co., *c/o* Butterfield & Swire, Hong Kong.

Robert Meldrum, *c/o* Messrs. Butterfield & Swire, Hong Kong.

Archibald Miebach, 45, Hampden Avenue, Beckenham, Kent.

Frank Murray Paskins, Murray House, Victoria Road, Cowes, Isle of Wight.

Fred Rowland Platt, 113, Pall Mall, Leigh-on-Sea, Essex.

Hugh Purdie, *c/o* Maclay & McIntyre, 21, Bothwell Street, Glasgow.

Angus Rankine, "Southview," Corringham Road, Stanford-le-Hope, Essex.

Robert Scott Sinclair, China Navigation Co., Messrs. Butterfield & Swire, Hong Kong.

George John Tagg, "Rangemore," East Molesey, Surrey.

Tom Roberts Thomas, *c/o* British Corporation Registry, 6, Billiter Street, E.C.3.

Andrew Lees Thomson, 3, Corstorphine Bank Drive, Corstorphine, Midlothian.

Robert Tosh Young, 18, Woodland Park Road, Newport, Mon.

Associate-Members.

Walter Clarence Baker, *c/o* Burns, Philp & Co., Ltd., Bridge Street, Sydney, N.S.W.

George Alexander Day, 108, Fillebrook Road, Leytonstone, E.11.

Gordon Hugh Murdoch McKenzie, "Inver," 96, Chasefield Road, Tooting, S.W.17.

Transferred from Graduate to Associate-Member.

Angus M. Campbell, 36, Muswell Avenue, London, N.10.

Transferred from Student-Graduate to Associate.

David Murray, Greahan Estate, Post Bangoen Poerba, E.C. Sumatra, Dutch East Indies.