The INSTITUTE of MARINE ENGINEERS

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

The Presidential Address.

By The Rt. Hon. LORD MOTTISTONE, P.C., C.B., C.M.G., D.S.O.

[Delivered by the President at a meeting of The Institute held at the Institution of Mechanical Engineers on Wednesday, 16th September, 1942, at 5.30 p.m.]

The President, in opening the meeting, thanked the President and Council of the Institution of Mechanical Engineers for their kindness in permitting the Institute to meet in their hall, the Institute's own hall being not available at present.

He also offered a welcome to the distinguished visitors who were present, and who included Admiral of the Fleet Lord Chatfield, P.C., G.C.B., O.M., K.C.M.G, C.V.O., D.C.L., President of the Institution of Naval Architects; Sir John Thornycroft, K.B.E., President-Elect of the Institution of Civil Engineers; Colonel S. J. Thompson, D.S.O., President of the Institution of Mechanical Engineers; Sir Noel Ashbridge, President of the Institution of Electrical Engineers; Sir Richard Allen, C.B.E., Past President of the Institution of Mechanical Engineers; Engineer Vice-Admiral Sir George Preece, K.C.B., former Engineer-in-Chief of the Fleet; and Professor A. C. G. Egerton, M.A., F.R.S., who represented the Royal Society.

The President then delivered his Presidential Address.

PRESIDENTIAL ADDRESS.

Owing to the war, the Council of the Institute have thought it well that I should give my Presidential Address on this occasion, rather than that there should be a special meeting for the purpose. In spite of my high sense of the honour which you have conferred upon me by making me your President, I intend to be brief, partly because brevity is always desirable and partly because we are all anxious to hear the Parsons Memorial Lecture by Dr. Dorey which is to follow.

There are three subjects of outstanding importance to which, as your President, I wish to refer. It so happens that I have been concerned in or have some knowledge of all of them. The first is the great advance which has been foreshadowed to us in the consideration of the future status of marine engineers and of the Merchant Navy generally. As the representative of this great Institute, I wrote to Lord Leathers, the Minister of War Transport, after a personal interview with him, the following letter:

"The Council of the Institute of Marine Engineers, of which I am President, have asked me to forward the enclosed proposals to you. They were anxious to put their views forward in the right manner, and I told them it would be right for them to send their proposals to you as Minister in the first instance.

"The Institute has a membership of over 4,000 merchant marine engineers, of whom nearly 50 per cent. are actually sea-going officers at the present time. Being a scientific body, they only include in their membership those with certain attainments.

"The Council wish me to take this opportunity of expressing their cordial thanks for the great help which many high officials of your Ministry have given and are giving to them, especially on the scientific side of their work.

"Knowing how anxious you are to secure good conditions of service for the merchant marine engineers both during this war, in which they are rendering such exceptional service, and in the days to come, I do not hesitate to bring these proposals to your notice, in the hope that you may be able to incorporate some of them in the great scheme which you will in due course lay before Parliament".

I was able to write that because Lord Leathers had stated publicly in the House of Lords, and had also informed me personally, that he was bringing forward at the right time some far-reaching proposals, of which he said to me: "I believe they will not be unsatisfactory to the Merchant Navy, and not least to that important branch of it with which you are concerned as President of the Institute of Marine Engineers".

Lord Leathers acknowledged that letter, asking me to thank the Council for their appreciation of the help given by his officials and saying that he would write to me again when he had fully considered the proposals I had sent to him. This is his reply:

"My dear Mottistone,

"I am writing to thank you for your letter of July 23, with which you enclosed a memorandum on the training, grading and conditions of service of engineers in the Merchant Navy that had been prepared by the Council of the Institute of Marine Engineers.

"You may remember that last February, during a debate in the House of Lords on the future of the Merchant Navy, I announced that I had asked the National Maritime Board to expedite their consideration of plans for continuity of employment and for the improvement of the methods of entry and training of sea-going personnel after the war. I am glad to say that the National Maritime Board and the Merchant Navy Training Board have already made considerable progress.

"In the detailed discussions about reconstruction which must presently take place, the suggestions put forward by the Institute of Marine Engineers, which are now being examined very closely by my technical advisers, will be of great value, and I shall be glad if you will convey my thanks to the Council of the Institute and assure them that their proposals are being considered very carefully".

That is expressed in official language, but it means that he will probably be able to adopt a good many of them. He concludes:

"May I add my good wishes for the continued prosperity of the Institute".

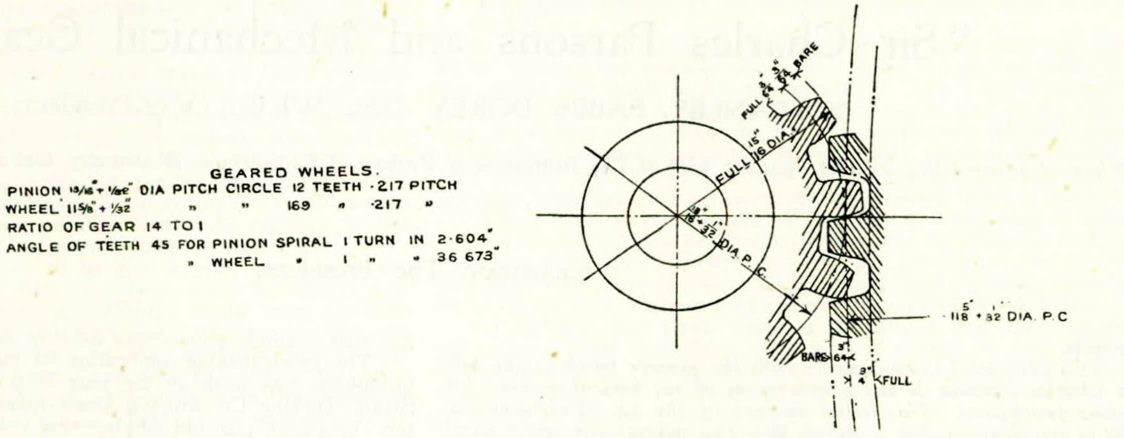
I think you will agree with me that that is a highly satisfactory letter, which Mr. Curling will be able to keep in the archives for future reference. I hope and believe that the status of the marine engineers will be carefully considered and improved.

My next point will show, I think, that high on the list of those

The success of the "Vespasian" heralded a new era in turbine propulsion, which in a correspondingly short space of time was to cause an almost complete change-over from direct to geared-turbine drive.

So far as naval vessels are concerned, in the 1910 programme two torpedo-boat destroyers were ordered, namely, the "Badger" and the "Beaver", in which the H.P. and cruising turbines were in tandem and delivered 3,000 s.h.p. through a single-reduction gear to the propeller shafting to which the L.P. turbine was directly coupled. In 1912, the Admiralty went a step further in placing the order for the machinery of the torpedo-boat destroyers "Leonidas" and "Lucifer", each of 22,500 s.h.p. In these vessels, completed in August, 1914, the whole of the power was transmitted through single-reduction gearing, the pinions being arranged on the port and starboard sides of the main wheel and delivering power from the H.P. and L.P. turbines. This represents present-day naval practice, excepting that in some cases a cruising turbine is fitted as an extension to the H.P. spindle.

Before the completion of the torpedo-boat destroyers "Leonidas" and "Lucifer", two light cruisers, the "Calliope" and "Champion", each of 40,000 s.h.p. were ordered, and these were all-geared installations. The "Calliope" was a quadruple-screw vessel and the "Champion" a twin-screw vessel and both were successful in service. In about three years, therefore, from the original adoption of gearing a figure of 20,000 s.h.p. transmitted through one set of gearing was reached. During the 1914-1918 war geared turbines were not totally adopted for naval purposes; all warships, however, during that period were fitted with cruising turbines which delivered power through mechanical gearing. No doubt the adoption of gearing at this time was limited by the number of gear-cutting machines available rather than by considerations affecting the reliability of geared drive. After 1916 practically all warships, excepting special types, were fitted with mechanical gearing, and the powers transmitted rose to 36,000 s.h.p. per gear set. Something of this phenomenal development in geared drive can be realized by considering the total shaft horse power transmitted through mechanical gearing, which by 1920 reached a grand total for naval vessels of nearly 7 million s.h.p. This total does not include the shaft horse power transmitted by cruising turbines. The extended use of mechanical gearing over the comparatively short period of 10 years, starting with Parsons' classical trials with the "Vespasian", indicates something of the debt owed by the Royal Navy to the genius behind these achievements.



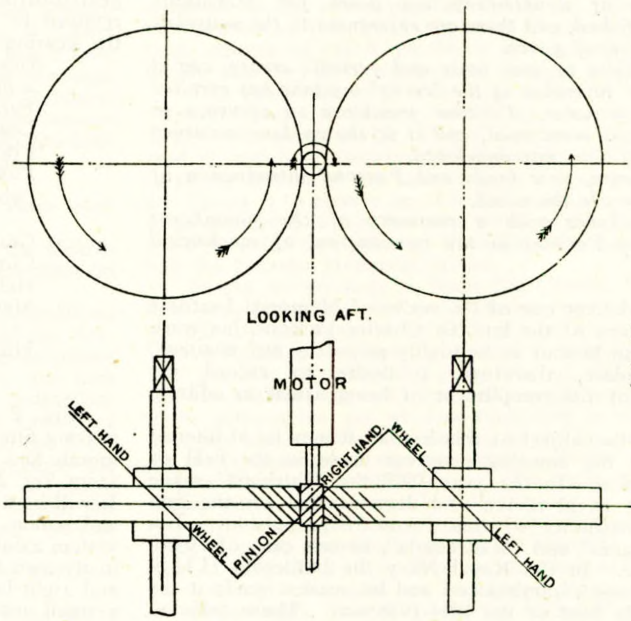
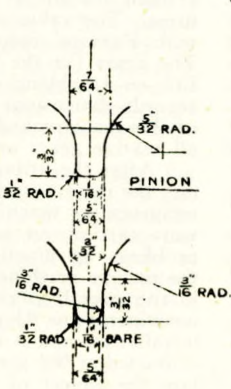


DIAGRAM SHOWING ANGLE TEETH ARE TO BE CUT OUT



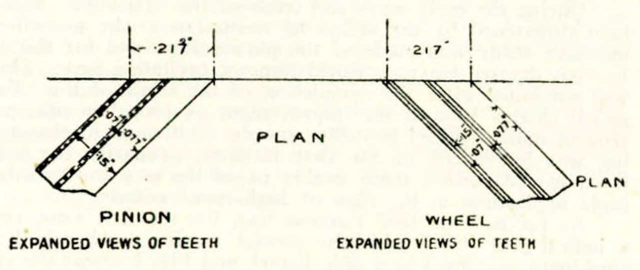


FIG. 1.—*Details of first marine reduction gear fitted by Parsons in 22-ft. launch.*

The value of Parsons' work shines even more brightly when it is remembered that much of the progress was made during a critical period for this country when the failure of gears might well have had disastrous effects on the efficiency of naval vessels.

In the Merchant Service, the success of the geared system in the "Vespasian" led to its adoption in Channel steamers, and subsequently in cargo vessels. Lloyd's Register's annual returns for the year 1913-1914 showed that 23 vessels were fitted with mechanical gearing. The year 1914 also saw the completion of the Anchor Line turbine steamer "Transylvania", a vessel of 14,000 tons. By August, 1914, geared turbines amounting to 260,000 s.h.p. had been built or were building with single-reduction gears.

During the 1914-1918 war the application of geared turbines, or indeed turbines, to merchant vessels, was retarded by the demands of naval construction. It is interesting to note, however, that in America both single- and double-reduction gears were employed for ship propulsion during this period.

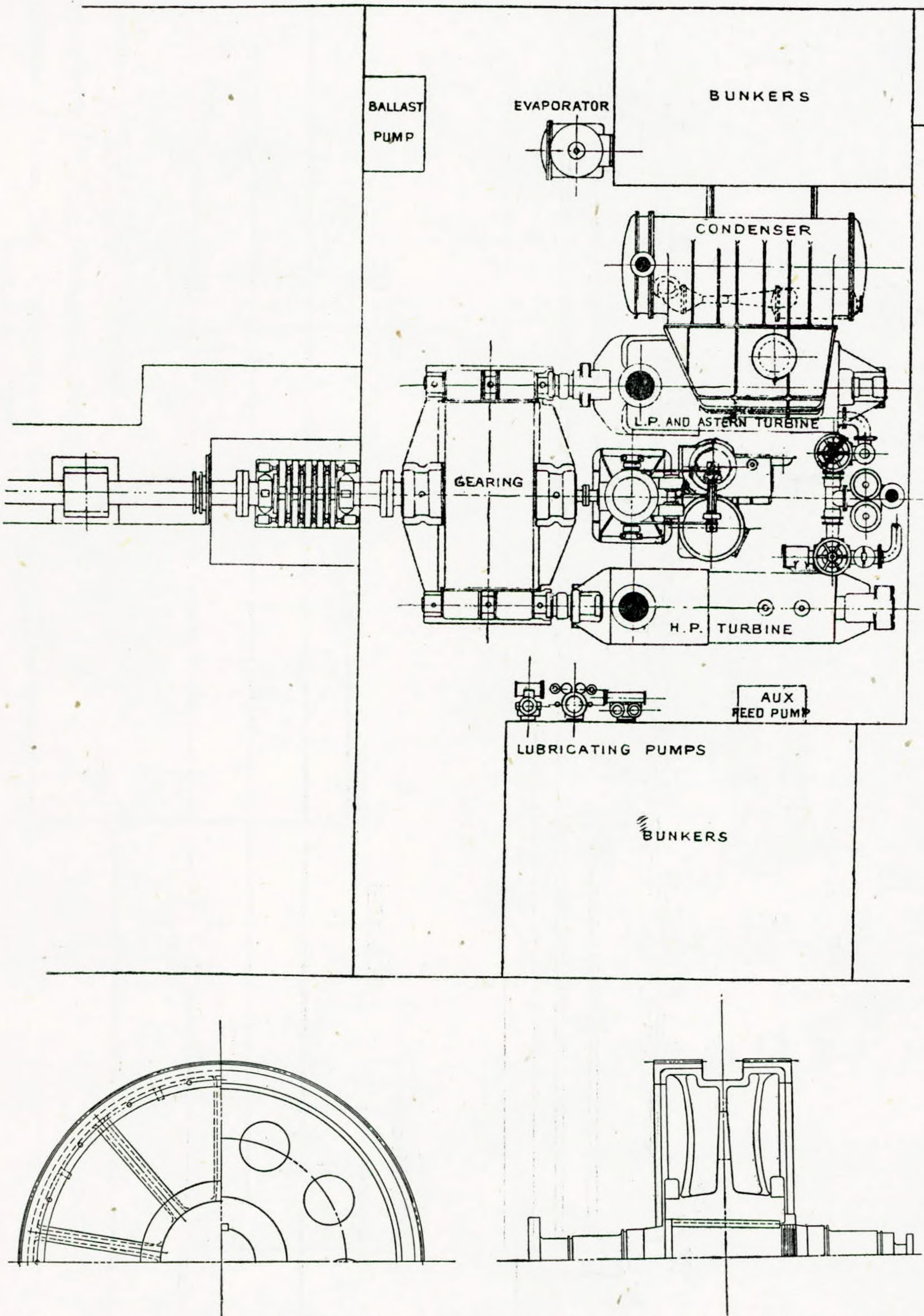


FIG. 2.—*Arrangement of the single-reduction geared turbines for the s.s. "Vespasian" and details of main wheel construction.*

Double-reduction Gears.

The first vessel fitted with double-reduction gearing as shown by the records of Lloyd's Register of Shipping was the s.s. "Pacific" of 6,000 gross tons completed in 1915 by the Union Iron Works Co. of San Francisco, the machinery installation being constructed by the General Electric Co., New York. Details of this gear installation are given in Fig. 4.

In this country the first vessel fitted with double reduction gearing was the s.s. "Somerset" of 9,770 gross tons, built in 1918 for the Federal Steam Navigation Co. This was a single-screw vessel of 4,500 shaft horsepower, with propeller running at 85 r.p.m. Fig. 5 shows the plan and details of the gearing, which is of the "interleaved" type. It is of interest to note that the gears in this vessel were in service until 1933 when they were reconditioned, the secondary gears then being cut with the all-addendum tooth.

The success of the gears in the "Somerset" led to the gradual introduction of the double-reduction system up to 1922-3, when over 1,000,000 s.h.p. had been fitted in this country. Subsequently, however, records from Lloyd's Register indicate that owners showed a preference for the single-reduction gear. In the United States, by 1921 over 2,000,000 s.h.p. had been fitted in about 750 ships with double-reduction gearing, other vessels built up to this period being fitted with single-reduction gears. It seems true to state that one of the difficulties with the double-reduction gear is the heavy mass concentrated at the secondary pinions, such mass being due to the first reduction wheel being rigidly attached to the same shaft. Axial movement of the secondary pinions necessitated by gear-cutting inaccuracies involves much higher inertia loading than is the case with the primary pinion having an adjacent flexible coupling.

The troubles which arose in the early days with double-reduction gearing were mainly those connected with excessive pinion wear, particularly that of the 2nd reduction. The remedies suggested for obviating this trouble were to reduce the load per inch face width in the case of double-reduction gearing, whilst in the United States the "floating frame" type of gear was suggested as a solution to the trouble. Parsons, on the other hand, maintained an uncompromising adherence to the rigid frame type of gear, and held that gear-cutting inaccuracies should be reduced to a minimum, attention being paid to alignment and the maintenance of the necessary stiffness of the second-reduction pinion shafts. Development since that date shows how right Parsons was and that, no doubt, he realized well the effects of increased inertia of the pinion shafts already mentioned. At this stage it may be mentioned that measurements have actually been made by Mr. Forsyth on my staff of the shuttle action of the 2nd reduction pinion shafts, *i.e.* port and starboard, in the case of a gear in which failure of the 2nd reduction pinions had twice occurred. The shuttle action was measured under ordinary running conditions, and was almost identical on the port and starboard sides, indicating errors in the main wheel. The main wheel was re-cut and no further trouble was experienced with these gears. It is interesting to note from this example that it was the 2nd reduction pinions which failed as a result of the inaccuracies in the main wheel.

As regards the "floating frame" it has been definitely established that excessive elasticity in conjunction with gearing can, with the incidence of resonance, set up resonant vibrations leading to excessive gear loads and excessive noise from the gear, and there seems little doubt that Parsons' adherence to rigid frame gears was thoroughly justified.

Whilst the application of double-reduction gearing to slow-speed merchant vessels in this country since about 1923 has not fulfilled the early expectations, it is probable that this must in the main be

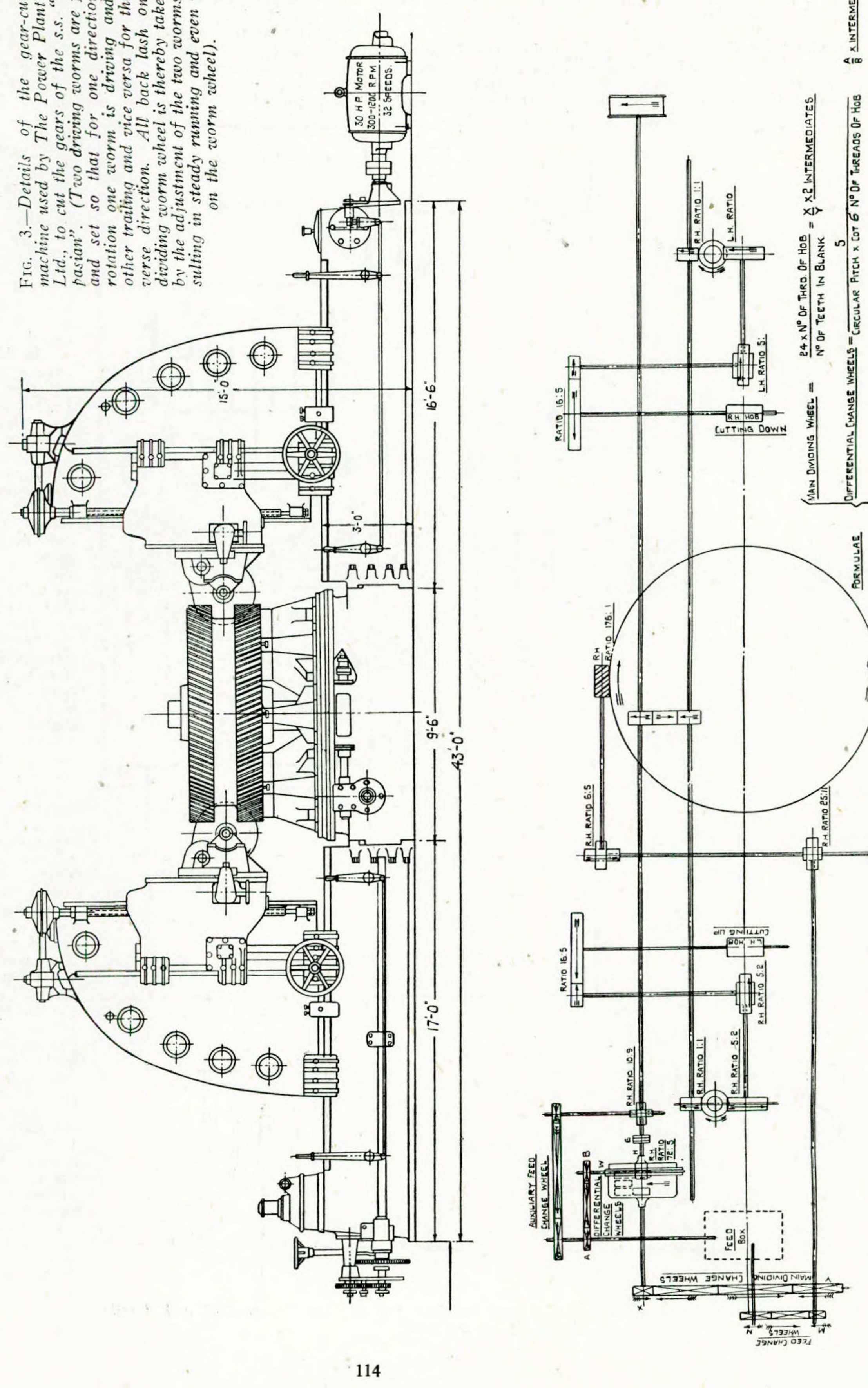


FIG. 3.—*Details of the gear-cutting machine used by The Power Plant Co., Ltd., to cut the gears of the s.s. "Vespasian". (Two driving worms are fitted and set so that for one direction of rotation one worm is driving and the other trailing and vice versa for the reverse direction. All back lash on the dividing worm wheel is thereby taken up by the adjustment of the two worms, resulting in steady running and even wear on the worm wheel).*

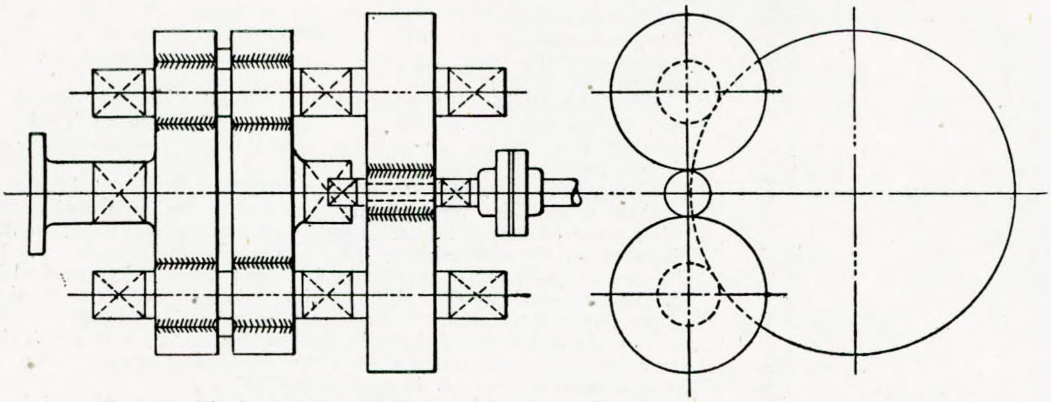


FIG. 4.—*Single turbine double-reduction two-plane type of gear for s.s. "Pacific".*

attributed to the increased economy obtainable with the heavy-oil engine rather than to inherent lack of possible reliability in the double-reduction gear.

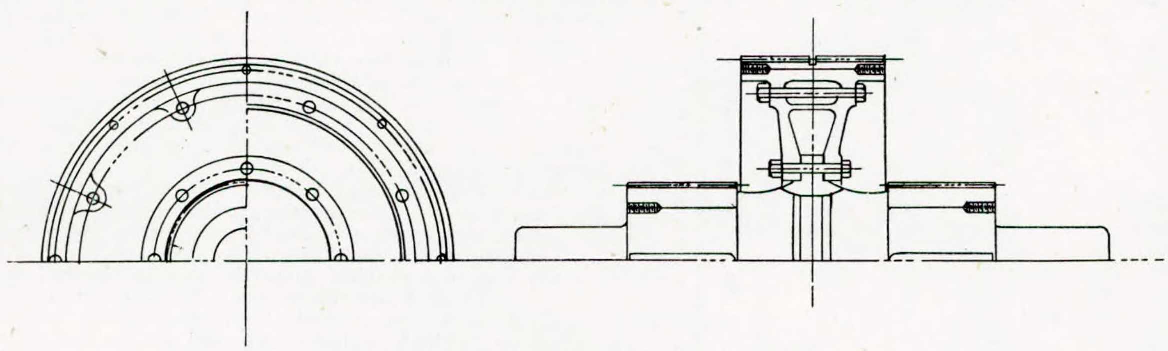
Materials for Mechanical Reduction Gearing.

It is a very remarkable fact that the materials originally selected by Sir Charles Parsons for the gearing of the "Vespasian" were substantially identical in physical properties with those which are in general use to-day, and this in spite of the enormous strides which have been made in the progress of the science of metallurgy during the past 30 years, more especially as regards alloy steels and their heat-treatment. That the original materials used have been shown now by long experience to meet the exacting requirements of gearing bears eloquent testimony to the sound judgment and engineering "instinct" which characterized Parsons in every phase of his work.

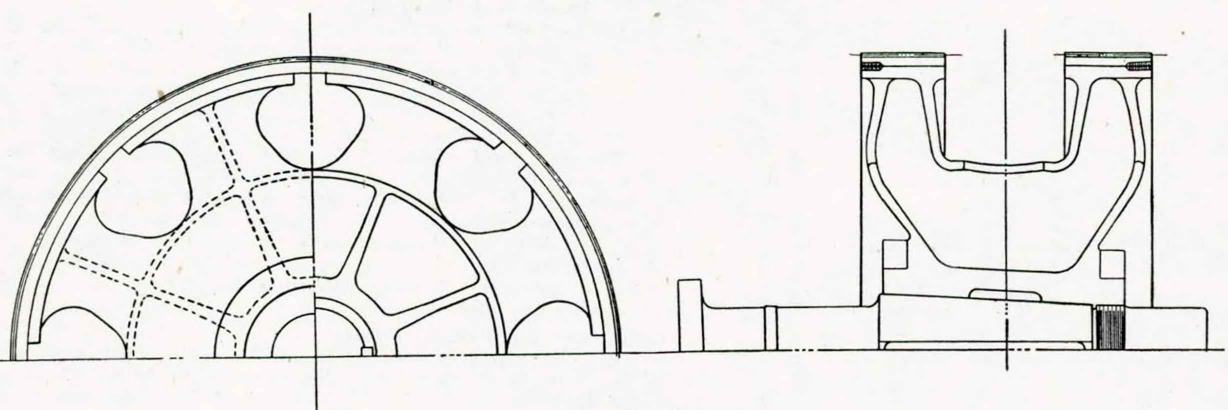
The properties of materials which experience has shown to be necessary in mechanical gears may be summarized as follows. For pinions, it is necessary to have satisfactory tensile strength to prevent fracture of the teeth at the root, sufficiently high yield strength to minimize the danger of permanent deformations which might arise from abnormal conditions of loading due to impact, vibration, misalignment or other causes, adequate ductility to assist in the more uniform distribution of stresses, an appropriate degree of hardness to give resistance to wear, and fatigue strength, not so much to withstand the root stresses in the teeth as to enable the tooth flanks to resist the repeated application of high surface stresses. These latter qualities are particularly important in the case of the pinion material, since the surface stresses are higher consequent upon the smaller radius of curvature of the pinion teeth relative to those of the wheel and, further, since the number of applications of load may be anything from six to 12 times as great as on the wheel teeth.

A further important requirement for main gears is that the necessary properties should be obtained without a final heat-treatment, subsequent to the cutting of the teeth.

Turning now to the material of gear wheels, the original construction adopted by Sir Charles Parsons in the case of the "Vespasian", namely, forged-steel rims shrunk and pinned on a cast-iron centre, is still in the main that adopted in the merchant service, although cast-steel or bolted-plate centres are preferred in naval work. For the rims, which are usually of rolled or



First-reduction wheel and second-reduction pinion.



Main wheel.

FIG. 5.—*Gearing details of s.s. "Somerset".*

	1st Reduction.			2nd Reduction.	
2 wheels	59·827″ p.c.d.	277 teeth	1 wheel	111·875″ p.c.d.	305 teeth
H.P. and I.P. pinions	8·423″ ,,	39 ,,	2 pinions	22·742″ ,,	62 ,,
L.P. pinion	12·527″ ,,	58 ,,			
Width of face H.P. and I.P.	16″		Width of face	32″	
L.P.	21″				
Circular pitch	0·678527″			1·152349″	
Helical angle	30°-46·3′			29°-57·0′	
Hob	7/12″ lead			1″ lead	
Pressure angle	16°·36′·9			16°·35′·7	

TABLE I.

Part.	Specification.	Type of material.	Mechanical Properties. U.T.S. tons/ sq. in.	E.L. tons/ sq. in.	Y.P. tons/ sq. in.	Elong. (2in.) %	R.A. %	Izod impact ft.-lb.	Bend test.	Chemical Composition %. C.	Si.	Mn.	S.	P.	Ni.	Cr.	Heat treatment.
Pinions.	S.s. "Vespasian"	Forged Ni-Cr steel.	33·8	9·3	19·8	34	66	—	—	·16	·13	·461	·019	·018	3·46	·53	Annealed.
	Typical modern specification.	3½% Ni steel.	40 min. (L.&T.)	—	24 min. (L.&T.)	20(L.) min. 16(T.) min.	—	35 min. (L.)	† 180° (L.&T.)	·24 to ·35	·25 max.	·50 to ·80	·04 max.	·04 max.	3·25 to 3·75	—	Normalized.
Wheel Rims.	S.s. "Vespasian".	Forged Carbon steel.	30 to 35	—	—	—	—	—	—	—	—	—	—	—	—	—	—
	Typical modern specification.	Rolled Carbon steel.	31 to 35	—	—	26	—	—	180° ‡	*	—	—	—	—	—	—	Normalized.

†Bend test—$\frac{3}{4}$in. × $\frac{3}{8}$in. over $\frac{3}{4}$in. radius. ‡Bend test—1in. square over $\frac{3}{8}$in. radius. *Carbon content as required to give mechanical properties.

Note.—In American practice for shrunk rims of 1st and 2nd reduction wheels the carbon content and ultimate tensile strength are somewhat higher, a typical specification being as follows, viz.: U.T.S. 36 tons/square in., Y.P. 20 tons/square in., elongation (2in.) 22%, R.A. 40%, B.H.N. 160-190. Heat-treatment: Annealed or normalized and tempered.

hollow-forged construction, it is a first requirement to use a material which can be readily forged, having a moderately high tensile strength and good ductility, the necessity for hardness and high fatigue strength being less than for the pinion material for the reasons already stated.

For the purpose of comparison, particulars of the materials used in the gearing of the "Vespasian" are shown in Table I, together with typical present-day specifications for materials of pinions and wheel rims.

The pinion forgings are made with sufficient material on the portions from which the teeth are cut to allow 2in. on the diameter being removed before heat-treatment, and leaving about ¼in. on the diameter for the machining of all surfaces after heat-treatment. In the case of naval gears, the forging is heated to 850° C., quenched in oil and tempered at 600° C. The time taken to raise the forging to the required temperature should not exceed half-an-hour.

The figures herewith indicate that fundamentally very much the same types of materials are used for mechanical gears to-day as were adopted by Parsons in the "Vespasian".

In connection with the use of nickel-steel forgings for pinions, it may be mentioned that some troubles were experienced in the post-war years of 1921-2 due to the cracking of nickel-steel pinions. It was at first thought that this was brought about by contraction stresses in quenching and tempering, but it is probable that the troubles experienced were initiated by defects in the material which showed itself particularly liable to develop hardening cracks. Whatever the true cause may have been, it may be noted that since then the great majority of nickel-steel pinions for the Mercantile Marine have been normalized only, although for naval gears the oil-hardening and tempering process has been continued. A very important point with regard to the material of pinions is the amount of discard to be removed from the ingots before forging. In average practice to-day at least 40 per cent. of the total weight of ingot is removed from the top end and these proportions are usually sufficient to ensure freedom from piping and segregation. A further precaution against the occurrence of such defects is the boring of the pinion forging for its whole length prior to heat-treatment, and this has the added advantage of promoting a more homogeneous structure, particularly in pinions of large diameter in which mass effects may be important.

As regards alternative materials for gear wheels, in some cases, particularly in American practice for 2nd reduction gears, the wheels are steel castings without special rims. Among the advantages that are claimed for this material are:—

(1) Improved physical properties can be obtained as compared with rolled material, particularly as regards hardness.
(2) Reduction of strength due to the occurrence of blow-holes or cavities in well-designed steel castings is less serious than if the same flaws were rolled out in the form of cleavage planes from ingot castings.
(3) Internal stresses in large castings can be eliminated by annealing and normalizing, whereas with shrunk rims internal stresses are always present, and further the distribution of shrinkage grip is never uniform around a solid centre.
(4) Reduced weight and cost.

For such steel castings, it is usual to employ a carbon content of between 0·20-0·35 per cent. with the addition of from 0·40 to 0·60 per cent. of molybdenum. It is claimed that molybdenum confers increased hardness, whilst reducing mass effect in heat treatment, and improves the machining qualities of the casting. The physical properties of a cast carbon molybdenum steel would be U.T.S. 31 tons, Y.P. 20 tons, elongation (2in.) 22 per cent., reduction of area 35 per cent.

Gear Noise and Periodic Errors.

The first two gear-cutting machines were installed in the works of the Parsons Marine Steam Turbine Co. in 1910. These were "solid table" hobbing machines and judged from present-day standards the worm periodic error in the gear was somewhat pronounced, amounting, as stated by Parsons, to a double-amplitude of 4/1,000 inch. These machines were made by Messrs. Wm. Muir & Co., Ltd.

Making use of a microphone and oscillograph, Parsons established that the noise frequency corresponded to the number of teeth in the worm wheel of the gear-cutting machine multiplied by the revolutions of the wheel. By 1913 Parsons had introduced his ingenious "creeping" table machine and there is little doubt that even to-day the advantages to be gained by the use of this machine, together with those accruing from the provision of longitudinal freedom given by a flexible coupling, have made the best contribution towards the reduction of gear noise.

In the case of gears cut on solid-table machines, the predominant frequencies in the noise are:—

(1) Tooth contact frequency.
(2) Worm wheel tooth frequency.
(3) Tooth frequencies of the worm wheel of previous hobbing machines used in the manufacture of the gear-cutting machine which cut the gear.
(4) Feed screw periodic, due to one revolution of the feed screw.

Regarding frequencies under (3), these have been spoken of as the "hereditary taint" and actual noise analysis has definitely established the truth of this statement. The tooth contact frequency is given by the rate at which the teeth run into mesh.

The worm wheel frequency results from a periodic error in the worm, with its abutments, used for driving the work table. This results in a periodic motion of the table superimposed on its mean rotation during gear-cutting. It is obvious that on a helical gear such periodic movements will give rise to undulations along the tooth faces. Furthermore, it can be seen that the crests and troughs of such undulations from tooth space to tooth space will lie in the same axial plane. With a worm error of the order mentioned by Parsons of $\pm 2/1{,}000$in. the meshing of the teeth occurs more like a spur gear having the same number of teeth as the worm wheel of the gear-cutting machine. The advantages of the helical form are thus lost. In a single-helical gear the undulation gives rise to a torsional and axial vibration of the gear wheel. With the double-helical gear, the vibration is again torsional and axial, but the magnitude depends upon the phase relation of the undulation on the two helices. The magnitude of the movements may be greatly increased by the incidence of resonance in the case of gears rigidly connected to mass-elastic systems. As a result of such resonance, tooth loads can easily exceed the fatigue strength of the teeth and failure occurs. The resonance is also invariably associated with intense or "harsh" gear noise. Nor is it a matter of coincidence that resonance occurs. There are present many torsional and axial frequencies in the range of frequencies which may be excited by gear-cutting inaccuracies in the case of a rigidly coupled system, when both mass and elasticity of shafting are taken into account. Parsons' idea of introducing a flexible coupling between the pinion and turbine did much towards eliminating the danger of resonance and consequent heavy tooth loading. Although the main gear wheel was rigidly coupled to the line shafting, the forces exerted on the wheel, due to the "shuttle" action of the pinion, were greatly reduced by the pinions being flexibly connected.

The undulation introduced by periodics in the gear-cutting machines can now be accurately measured by an *instrument developed at the National Physical Laboratory. The wave length of the undulation due to the worm error in a solid-table machine is given by the formula, *viz.*:—

$$\lambda = \frac{n}{m} p_c \operatorname{cosec} \delta$$

where λ = wave length in inches.
δ = spiral angle.
n = number of teeth in the gear.
m = number of teeth in the worm wheel of the gear-cutting machine.
p_c = circular pitch of gear.

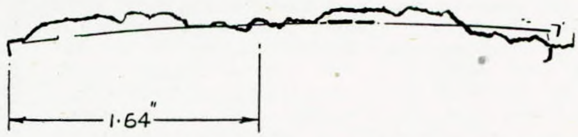
Fig. 6 shows a record taken by this instrument, giving the worm periodic in a solid-table machine.

The feed screw periodic generally gives a much shorter wave length undulation than that caused by the worm error. Since the axial length is equal to one pitch of the feed screw, the wave length along the tooth space is given by the formula:—

$$\lambda = p_f \sec \delta \text{ (where } p_f = \text{pitch of feed screw).}$$

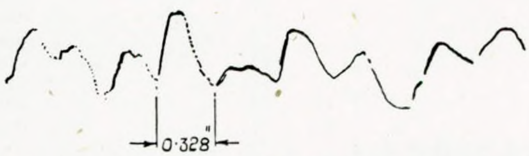
The phase of this undulation is nearly constant round the wheel. The crests and troughs of these undulations may or may not lie in the same axial planes; in both above respects they resemble the cumulative worm wheel undulation of a creep gear, as will be shown later.

*This instrument was designed by Dr. G. A. Tomlinson, Metrology Dept., N.P.L.



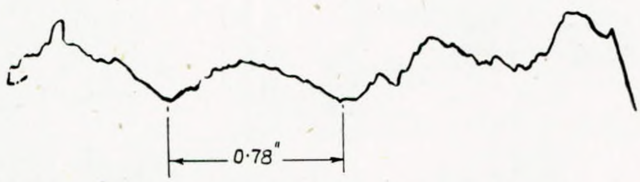
RECORD SHOWS WORM ERROR UNDULATION IN GEAR CUT ON SOLID TABLE MACHINE, DOUBLE AMPLITUDE OF UNDULATION 0·0004", WAVE LENGTH 1·64"

(a)



RECORD SHOWS CUMULATIVE ERROR UNDULATION IN R.H. HELIX OF GEAR CUT ON CREEP MACHINE, MAXIMUM DOUBLE AMPLITUDE 0·0012", WAVE LENGTH 0·328".

(b)



RECORD SHOWS UNDULATION DUE TO WORM ERROR IN L.H. HELIX OF SAME GEAR AS (b) ABOVE *i.e.* CUT ON CREEP MACHINE, MAXIMUM DOUBLE AMPLITUDE 0·0009", WAVE LENGTH 0·78".

NOTE:- THE CREEP ARRANGEMENT IN THE MACHINE WAS AS SHOWN IN FIG. 10 (b) AND THE WORM WHEEL AND SPUR CREEP HAD AN EQUAL NUMBER OF TEETH. CONSEQUENTLY THERE WAS AN EXACT WHOLE NUMBER OF REVOLUTIONS OF THE WORM PER REVOLUTION OF THE WORK, THUS THE EFFECT OF CREEP IN REDUCING THE WORM ERROR AND DISPERSING IT IN A SPIRAL WAS LOST.

(c)

FIG. 6.—*Undulation records taken with instrument designed by Dr. G. A. Tomlinson, Metrology Department, National Physical Laboratory.*

In solid-table machines the errors most likely to occur are those due to the worm periodic, which includes the axial drift of the worm once per revolution, and the feed screw periodic. There are other possible causes such as eccentricity in some wheel of the drive near to the worm and the periodic motion of the hob, either rotative, axial or eccentric. The cumulative division error of the worm wheel gives an undulation along the helix, the length in the axial direction being equal to the lead of the spiral of the gear. It is important that the curve of the cumulative division error of the worm wheel, from tooth to tooth, should be as smooth as possible. In this way the cumulative error will not set up undulations demanding rapid accelerations and decelerations of the gear wheel, but will appear more as a helix angle error.

Much has been done in recent years to eliminate the periodic errors in solid-table machines; it is true to say that undulation records indicate that such errors have almost been eliminated by a few firms. On the other hand, creep machines generally indicate a greater magnitude of undulations than gears cut on good solid-table machines. In this connection it is interesting to study the undulation records in Fig. 6. In spite of this, however, it seems generally agreed that gears cut on creep machines are more silent than those on solid-table machines. The conclusion to be drawn is that the value of the creep machine consists in placing the perodics spirally round the gear instead of in the same axial plane, rather than in reducing the magnitude of the undulations due to periodics.

Solid-table machines have also been made with a large number of teeth in the worm wheel with a view to raising the frequency of the noise even above the audible range of frequency. Such attempts have succeeded in reducing noise level, but frequencies have been present corresponding to the number of teeth in the worm wheels of the gear-cutting machines used in producing the worm wheel with the larger number of teeth. It seems, therefore, that the elimination as far as possible of the periodic error is the correct solution, whether or not the number of teeth in the worm wheel is increased.

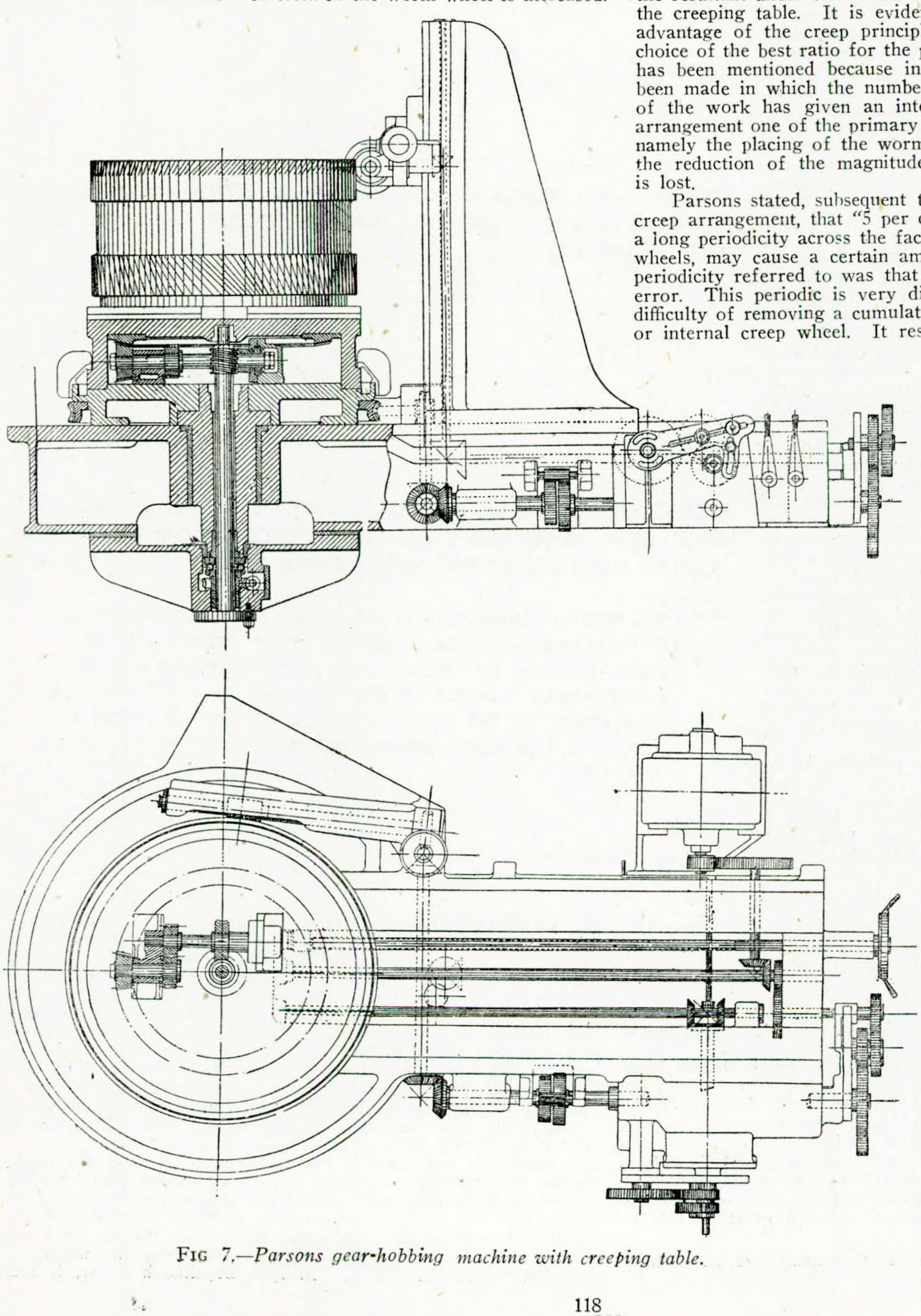


FIG 7.—*Parsons gear-hobbing machine with creeping table.*

Fig. 7 shows details of the first gear-hobbing machine fitted with a creeping table developed by Parsons and his colleagues. In this machine the work table is given an advance of 1 per cent. in relation to the worm wheel, there being 160 teeth in the worm wheel. This resulted in (160-1·616) cycles of worm error being registered round each complete feed spiral, the fractional part of this number, *viz.* 0·384, giving the change in phase at any tooth space after the completion of a feed spiral. The feed spiral had a lead of only 1/20in. and it is obvious, therefore, that the rapidly changing phase of the worm error for an advance of only 1/20in. axially down the gear results in successive hob cuts, after each revolution of the work, reducing the magnitude of the worm error undulation. Parsons showed that with the above ratio of change of phase of the error, the resultant undulation amounted to only one fifth of that without the creeping table. It is evident, therefore, that to gain the full advantage of the creep principle, attention should be paid to the choice of the best ratio for the phase change per creep spiral. This has been mentioned because in recent years creep machines have been made in which the number of cycles of error per revolution of the work has given an integer (see Fig. 6*c*). With such an arrangement one of the primary advantages of the creep mechanism, namely the placing of the worm error spirally round the gear and the reduction of the magnitude of the error to about one fifth, is lost.

Parsons stated, subsequent to the adoption of the one per cent. creep arrangement, that "5 per cent. creep is better, for it destroys a long periodicity across the face of the wheel, which, with narrow wheels, may cause a certain amount of vibration". No doubt the periodicity referred to was that due to the worm wheel cumulative error. This periodic is very difficult to eliminate, because of the difficulty of removing a cumulative error from a large worm wheel or internal creep wheel. It results in the formation of the well-known creep markings on the gear. With one per cent. creep the phase shift of the cumulative error is 0·01 times the pitch circle circumference, and the feed advances f across the gear, where f is the feed per revolution of the work. When the phase has moved an amount equal to the pitch circle circumference, the feed will have advanced an amount equal to the lead of the creep spiral, *i.e.* $100 \times f$ for one per cent. creep. Fig. 8*a* shows the form and phase change of the undulation due to the cumulative error. It has been calculated for a case having the following particulars, viz. :—

Pinion: p.c.d.—27·63in., normal pitch 7/12in., number of teeth 41, face width 6in., spiral angle 30°, f—1/20in.

It will be noted that the periodic causes an undulation 5·77in. in length along the tooth space. With 5 per cent. creep the lead of the creep spiral becomes $20 \times f$ and the form of the undulation is shown by Fig. 8*b*. This diagram indicates a much smaller length of undulation, *viz.* 1·155in. and this was found by Parsons to be better than that shown by Fig. 8*a*, in the case of a gear wheel having a narrow face.

There is another interesting point regarding Fig. 8*a* and *b*; the lines *AB* and *CD* join points in the undulations 180° out of phase respectively in adjoining tooth spaces. It will be noted that these lines are nearly parallel with the axis of the gear. In Fig. 8*c* the lead of the creep spiral has been made equal to the axial pitch of the gear with the result that lines *AB* and *CD*

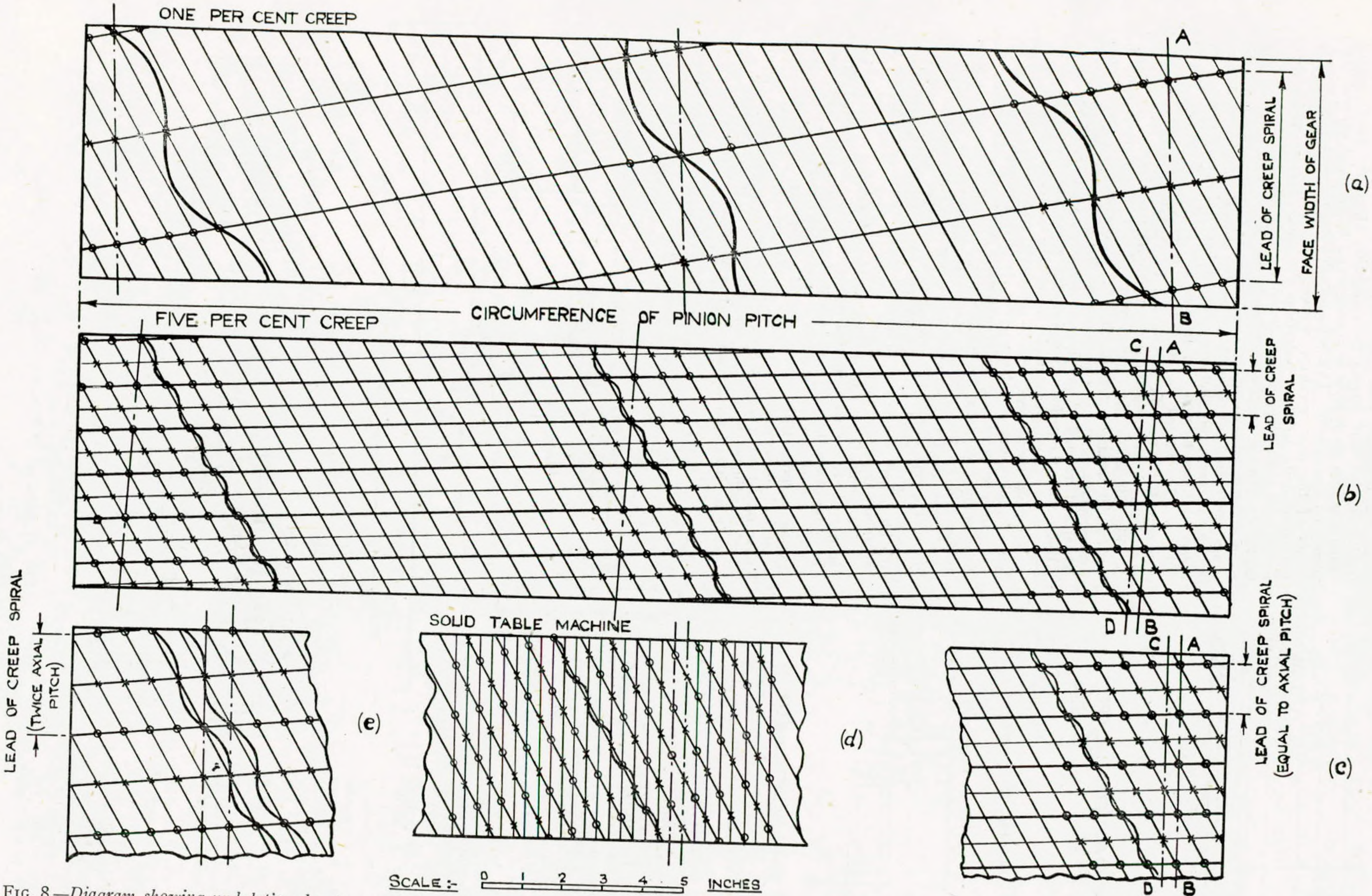


FIG. 8.—*Diagram showing undulation due to cumulative worm error with one and five per cent. creep, also undulation with lead of creep spiral equal to one and two axial pitches. Undulation with solid table included for comparison.*

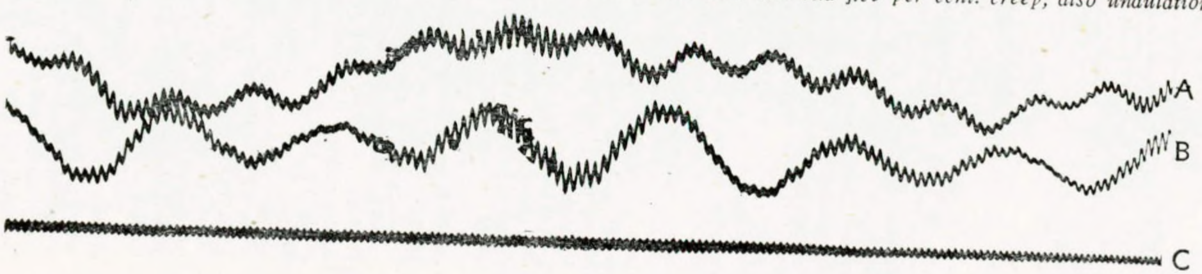


FIG. 9.—*Vibration records of torsional and axial movement of a single-helical pinion. (A=axial record; B=torsional record; C=fixed vibrator. The record was taken with a Sperry M.I.T. set; frequency of both torsional and axial vibration approximately 400 v.p.s.; frequency of fundamental in noise analysis from gear also 400 v.p.s.; range of axial movement of pinion 7/1,000in. at maximum amplitude; range of torsional movement of pinion 0·0095° at maximum amplitude).*

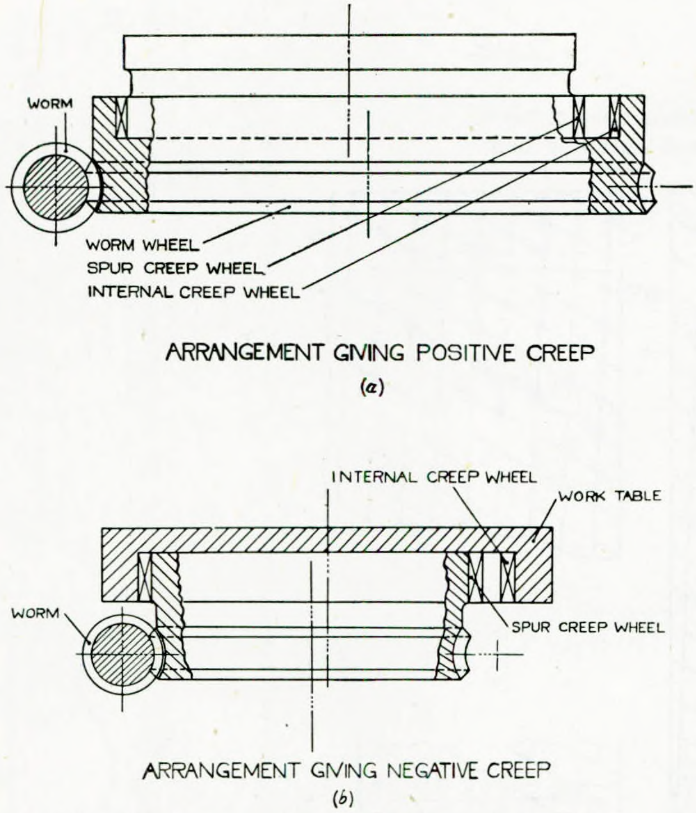


FIG. 10.—*Creep arrangement in gear-cutting machines.*

are in fact parallel with the axis of the gear. The pattern of this undulation, as far as axial planes are concerned, should be compared with that of Fig. 8*d* for a gear cut on a solid-table machine. It will be noted that in both these cases the undulations are all in phase with one another as they run into mesh, such an arrangement causing periodic movements both torsional and axial as the gear runs into mesh, with noise corresponding to the frequency of meshing of the undulations. In Fig. 8*e* the lead of the creep spiral has been made equal to twice the axial pitch and co-axial planes now link up undulations alternately 180° out of phase with one another. So far as uniform motion of the gear wheel is concerned, this arrangement is ideal provided the full width of the gear is not too small. From the foregoing it is obvious that the creep arrangement requires very careful consideration, and for a particular gear to be cut the value of the feed f must be considered so as to avoid as far as possible the condition indicated by Fig. 8*c*. Furthermore, the creep ratio should be such as to reduce the magnitude of the worm error by arranging a large phase shift of the error at the completion of each feed spiral. Although this was pointed out by Parsons in 1913, many creep machines have been made since that date in which the creep ratio is such that a whole number of periods of worm error are registered round the gear wheel during one revolution of the work.

So far as gear noise is concerned, the danger of resonance has already been indicated; this should be considered in the design of double-helical gearing by maintaining a stiff construction, both torsional and axial, between the two helices of the gear. Fig. 9 shows the torsional and axial vibration of a single-helical gear, rigidly coupled, and having an undulation along the tooth spaces which ran into mesh at a frequency equal to an axial natural frequency of the system. This set up both torsional and axial movements of the gear which were exactly equal to the fundamental frequency in the noise analysis from the gear. This certainly illustrates how right Parsons was in adhering to the principle of the fitting of flexible couplings between the pinion and turbine, and maintaining maximum stiffness otherwise in the construction of the gears and casing, as opposed to flexible mountings advocated elsewhere. It seems, too, that much can be done to improve present-day creep machines by adhering to principles explained by Parsons in 1913, namely, that of the correct spiral arrangement of the worm and cumulative periodic errors, and by the elimination, as far as possible, of inaccuracies in the gear-cutting machine.

Fig. 10 shows the arrangement of the creep mechanism in present-day machines, and of these (*a*) represents the more common arrangement. With this arrangement the periodicity of the worm wheel and internal creep wheel is one per revolution of the worm wheel. Since the periodicity of both the worm wheel and the internal creep wheel is the same, their cumulative errors combine to form a resultant cumulative error. If these errors are 180° out of phase, a minimum cumulative error results. The cumulative of the spur creep wheel is one per revolution of the work and results in a long undulation along the tooth space, the lead of which is equal to the lead of the spiral of the gear. Similarly, in case (*b*) the spur creep wheel and worm wheel form a resultant periodic, and the internal creep wheel the long undulation. The creep mechanism does not affect the lead screw periodic; any periodic movements of the hob affect the profile of the involute, but do not produce undulations along the tooth space.

To eliminate the undulations mentioned, lapping processes have been resorted to both in this country and in the United States. In some cases, the gears have been simply meshed and run with a light load using glass powder as a grinding medium. Since tooth action includes both rolling and sliding contact, care has to be exercised to avoid destroying the involute profile. The Parsons Co. have introduced a method of lapping pinions shown by Fig. 11. The load is adjusted by a brake mechanism B. The pinion and lap are run in mesh and the lap can be moved axially along the bed of the

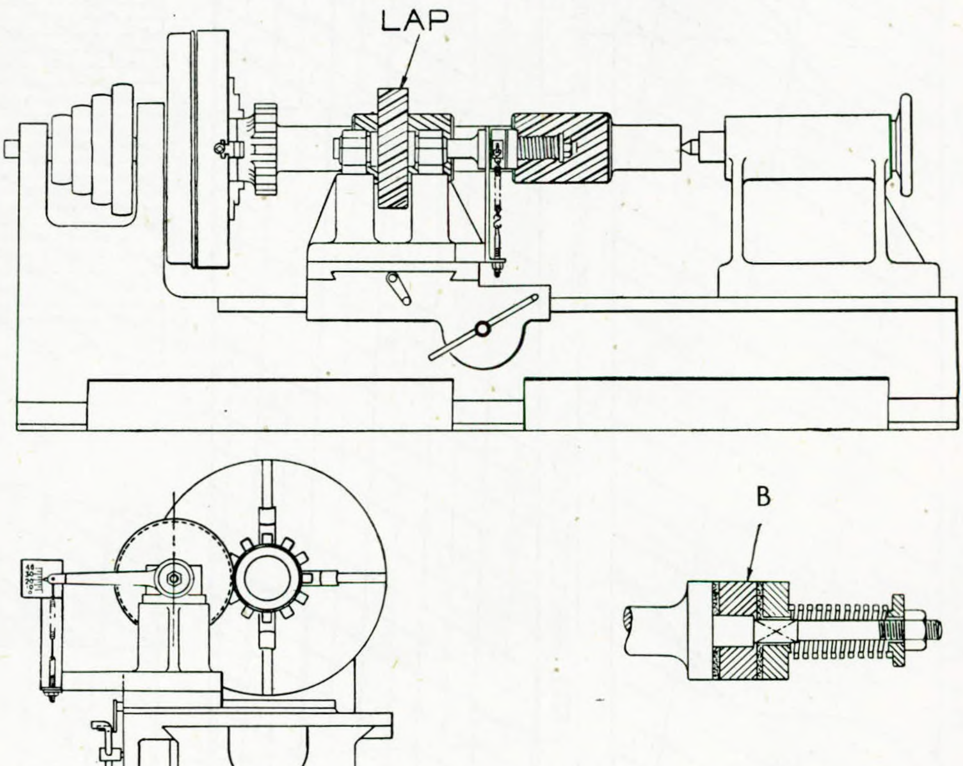


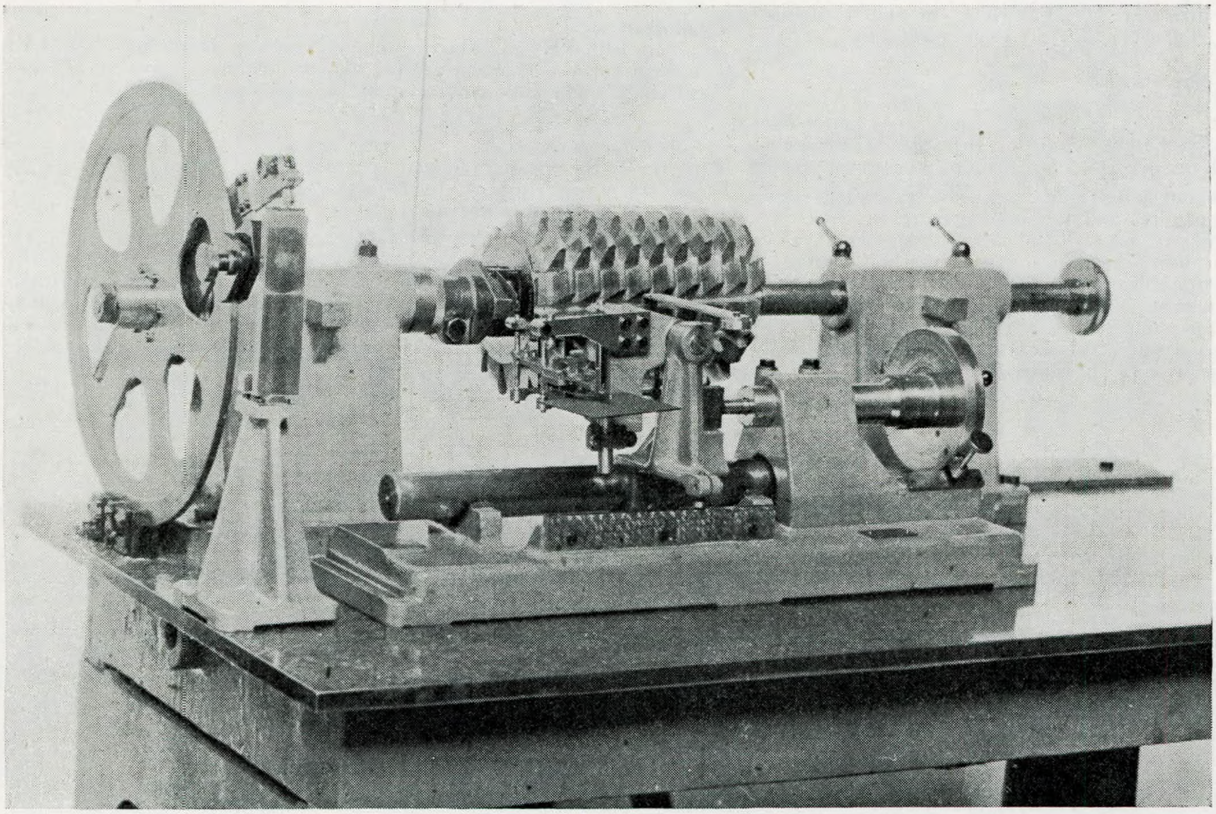
FIG. 11.—*Details of Parsons' pinion lapping machine.*

lathe. In the U.S.A. other methods of lapping are in use and large wheels have been lapped. This seems a definite step in the right direction, because it is the large wheels which have undulations of increased magnitude as compared with pinions.

There seems little doubt that for increased load and reduced weight of gear for a given power transmission, undulations must be removed or in any case reduced, otherwise excessive surface stresses will result. Further, lubrication difficulties will be increased with increase of load unless undulations are eliminated and surface finish improved. Whether solid-table or creep machines will succeed in producing the best gears in the future is an open question, but at present, as already mentioned, creep cut gears generally have been found more silent than those cut on solid-table machines, although the undulations may be greater, and the surface finish inferior.

Accuracy of Gear-cutting Machines.

As has already been mentioned, Parsons preferred to eliminate as far as possible gear-cutting inaccuracies, rather than introduce flexibility in the form of floating frames to reduce inertia loading due to the inaccuracies present. A good example of the attack made by the Parsons Company in this direction is illustrated in a paper given by *L. M. Douglas of that Company, wherein it was indicated that by the use of an angular pitch gauge and by grinding of individual errors, by a grinding wheel mounted in a slotting machine, and finally by the use of a crown lap the pitch error was reduced to 1/1,000in. in the case of an 80in. diameter spur wheel. The error was in the form of a first harmonic due to either a deviation in pitch or eccentricity. Work such as this is the only way in which to reduce the cumulative periodic already mentioned for the case of creep machines.



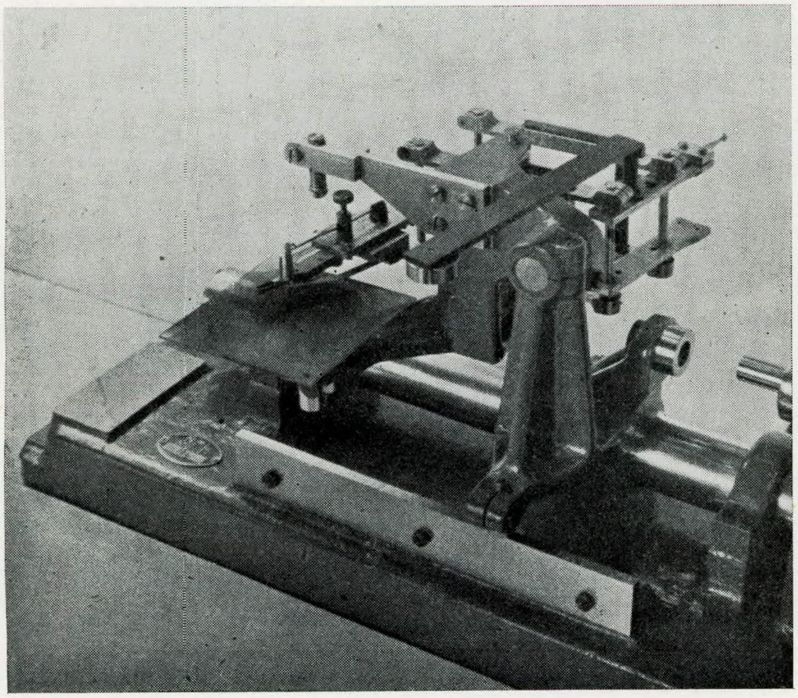


FIG. 12.—*Pantograph used by the National Physical Laboratory for the gauging of hobs.*

The Parsons Company also arranged their gear-cutting machines in one shop with temperature control so that diurnal errors could be obviated. In the early days of hobbed gears, the hob itself introduced errors which affected the form of the involute. A system of gauging was introduced at the National Physical Laboratory for checking the pitch, the flank angle, the pitch circle diameter, and the form of the flanks. Fig. 12 shows the pantograph used for making these measurements. Fig. 13*a* shows the results obtained for a hob in 1925 and Fig. 13*b* those for a similar hob in 1942. The following are the results obtained for these two hobs, which are representative of the best practice for the two dates mentioned, *viz.*:—

Description (applicable to either hob) :—

Single start hob	
Pressure angle (in axial plane) ...	14½°
Nominal pitch diameter	5·5in.
Number of flutes	14
Linear pitch	7/12in.

1. Spacing of Teeth.

1925.	*1942.*
Pitch curve shows marked periodic error of magnitude about ±0·001in. Cumulative error is small (see Fig 14*a*).	Pitch curves show no periodic or cumulative errors. Variations from mean pitch do not generally exceed ±0·0002in. (see Fig. 14*b*).

N.B.—At this time cumulative error of about 0·001in. over 100 teeth was not unusual.

* "Mechanical Gearing for Large Power Transmission", by L. M. Douglas. Trans. N.-E. Coast Inst. of Engs. & Shipbuilders, Dec., 1940.

2. Flank Angle.

1925.	*1942.*
α flank 14° 28′.	α flank 14° 31′.
β flank 14° 32′.	β flank 14° 28′.
N.B.—These angles were rather better than most at the time. Errors of ±7′ were quite usual.	*N.B.*—Very few modern hobs have errors greater than ±3′.

3. Pitch Diameter.

1925.	*1942.*
Variation in pitch diameter amounts to about ±0·002in.	Variation in pitch diameter less than ±0·0003in.

4. Tooth Form.

1925.	*1942.*
Form good, and tooth flanks uniformly straight throughout (see print of pantograph traces).	Form good, and tooth flanks uniformly straight throughout (see print of pantograph traces).

A further point about hobs is the accurate mounting of the hob on the arbor or mandrel and the making sure of its concentricity. Hobs nowadays are generally made with a ground surface at the ends for this purpose. End movement of the hob should also be eliminated. Lack of accuracy of the hob, or in the mounting, would give rise to imperfect involute profiles which would show a general waviness, the form depending on the errors present.

Factors affecting axial pitch include accuracy of the feed screw, the straightness or otherwise of the slide, and the correct alignment of the work when set up in the machine. Instruments are now available for the measurement of axial pitch within very fine limits and it is no longer necessary to rely on marking with the gears in mesh.

In Douglas' paper already referred to it was pointed out that theoretically the index wheels for cutting a pinion should be the same diameter as for the wheel. This desideratum is not fulfilled in practice, but it does indicate the advisability of using machines as large as possible for work on pinions and further of using extra care in the maintenance of these machines. Experience certainly indicates the necessity for taking such precautions.

Lubrication.

In 1916 *H. M. Martin applied Osborne Reynolds' theory of lubrication to mechanical gearing and his analysis is worthy of careful study. The following relationship was obtained, *viz.:*

$$p \propto \frac{u_0\sqrt{R}}{h_0^{1\cdot5}}$$

where R = the mean radius of curvature of the pinion teeth
u_0 = mean linear velocity of rolling
h_0 = minimum film thickness
p = maximum pressure in the oil film.

This relationship is important because R is proportional to the pinion p.c.d. for a given ratio of addendum to dedendum, and for a given pressure angle. Further p is proportional to the load P which can be carried per inch face width. It is thus seen that for a given value of the film thickness the permissible value of P will vary as

*See "Engineering", August 11th, 1916.

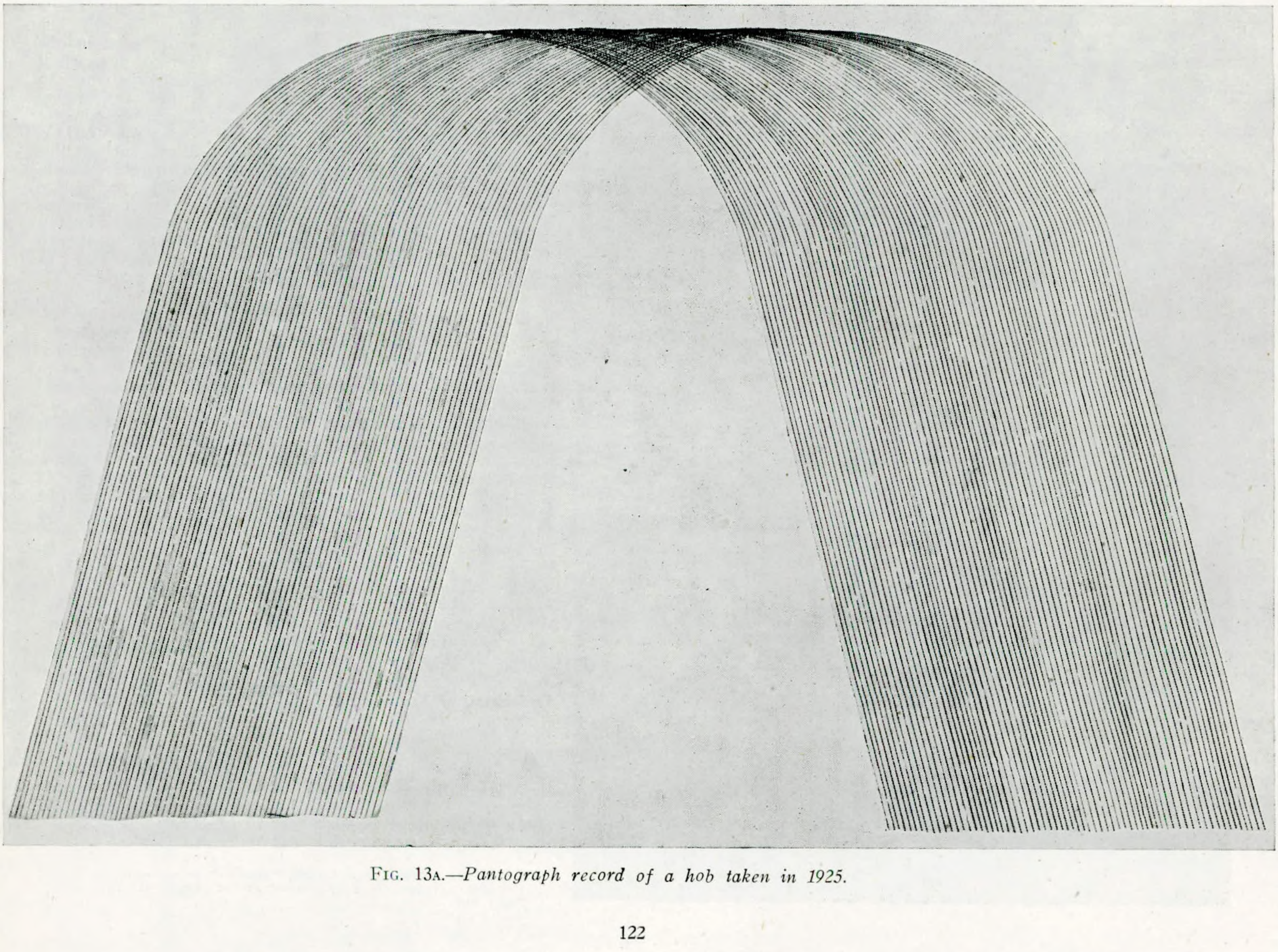


FIG. 13A.—*Pantograph record of a hob taken in 1925.*

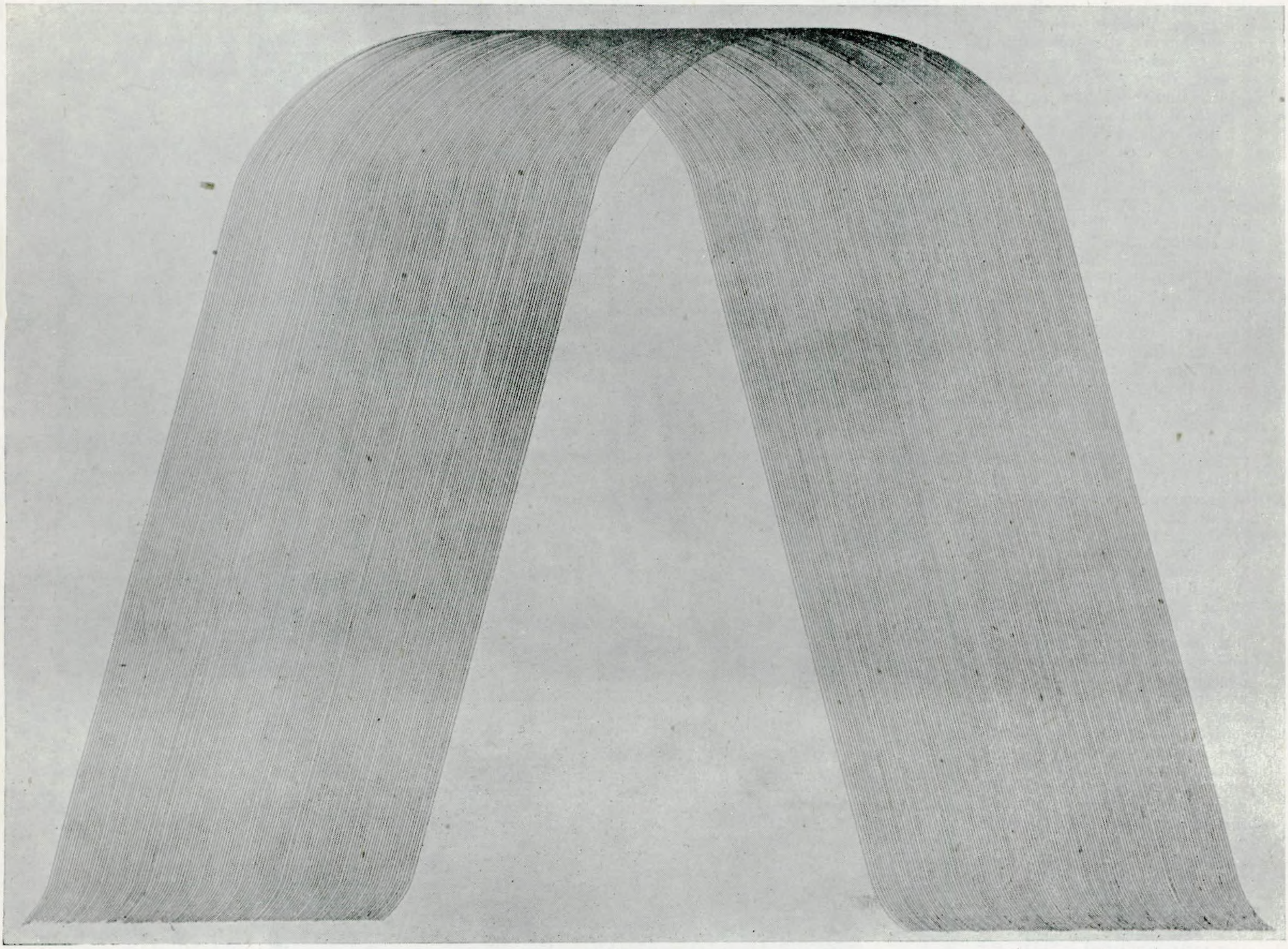


FIG. 13B.—*Pantograph record of a hob taken in 1942.*

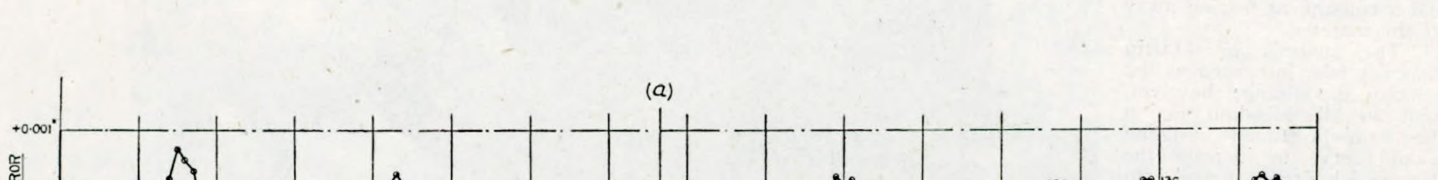


FIG. 14.—*(a) Graph showing pitch error derived from record shown in Fig. 13 (a); (b) graph showing pitch error from record shown in Fig. 13 (b).*

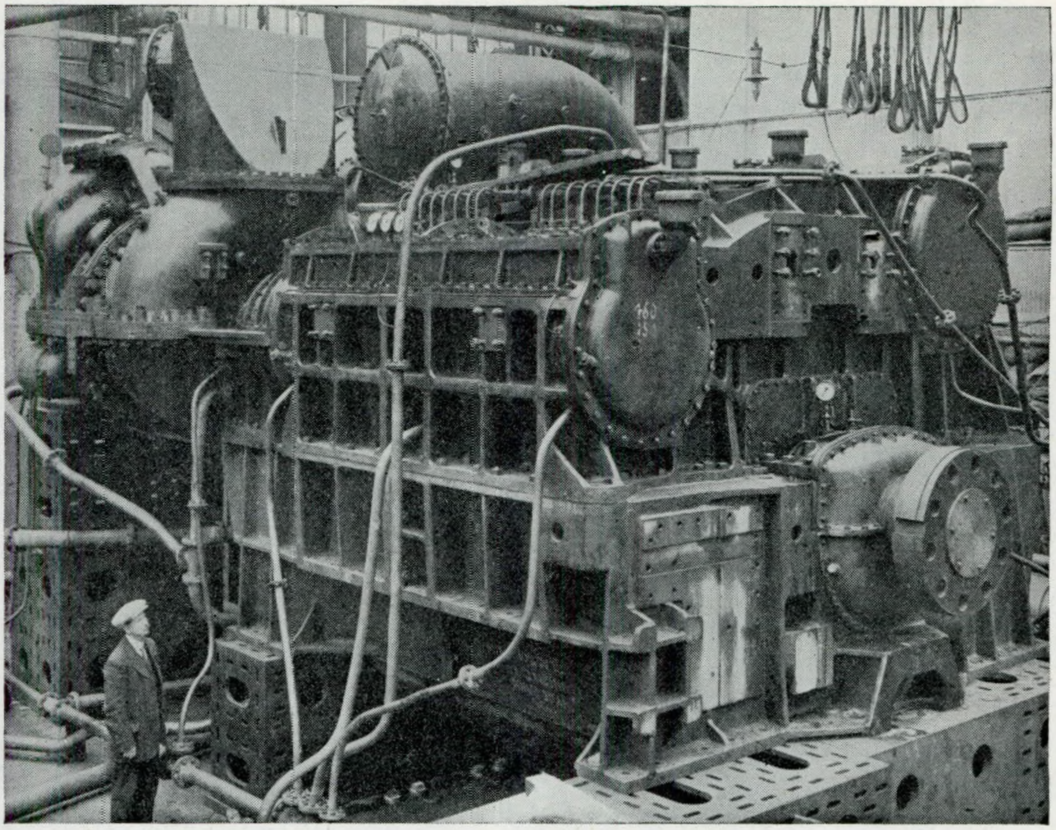
the square root of the p.c.d. Martin applied his results to a pinion having the following particulars, *viz.:*—

p.c.d., 14in.
pitch line speed, 62ft. per second.
pitch, 0·822in.
helix angle, 45°.
load per inch face width, 520lb.
„ „ lineal inch, 267lb.

The oil-film thickness was calculated at 0·000125in., maximum oil pressure 6,376lb. per sq. in. and the width of the bearing pressure zone 0·0866in. In making these calculations the temperature of the lubricating oil was taken at 90-100° F., with a viscosity of 0·7 dynes per sq. cm.

As the amount of the addendum of the pinion is increased, the curvature of the pinion becomes less sharp, that is, the value of R in the above equation increases, as does the value of u_n. The permissible value of P is thus increased for a given value of film thickness. Theoretically, therefore, an increase in the ratio addendum to dedendum permits a higher load from considerations of lubrication. As will be seen later, it also brings about a reduction of the surface stress. Parsons took advantage of this by introducing all-addendum pinions in 1932, and proved by experiment that an increase in load was possible. This led in some cases to the elimination of the centre pinion bearing, which had been provided from considerations of lateral stiffness where the axial length over the gear face exceeded three times the pinion p.c.d. The consequent saving in weight was very great. However, the velocity of sliding during the meshing of teeth was considerably increased, and in the case of small pinions "scuffing" of the gear occurred. This so-called "scuffing" is a result of seizure of the tooth surfaces, and a consequent tearing away of the material.

The analysis by Martin does not take into account the velocity of sliding; however, with an all-addendum pinion the uni-directional sliding should serve to increase the drag which draws the oil into the region of high pressure, resulting theoretically in improving oil film lubrication. In this respect it may be mentioned that with an all-addendum pinion the sliding velocity is sufficient for the formation of an oil film at normal running speeds, if the temperature rise in the oil film above that of the oil circuit is taken to be only of the order of several degrees, as considered probable by Osborne Reynolds. Furthermore, so far as the oil film formed by the sliding velocity is concerned, its thickness or load-carrying capacity will vary as d, the pitch circle diameter,



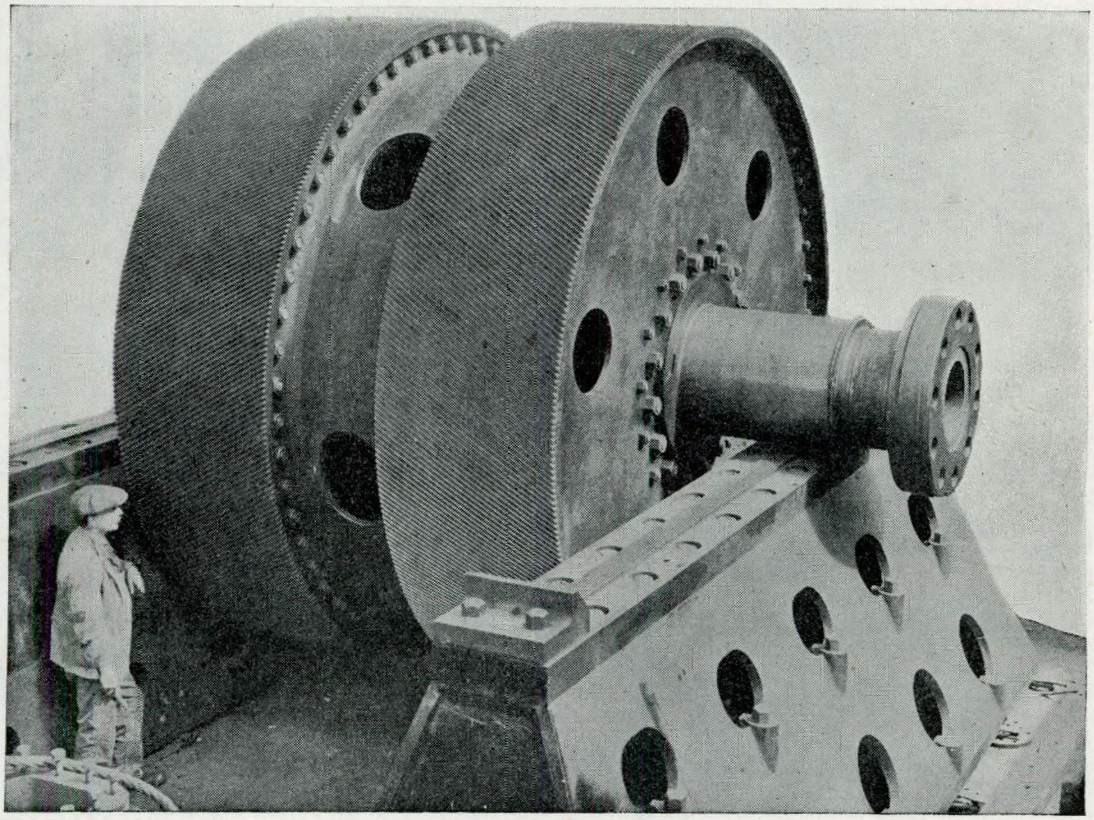


FIG. 15.—*Gearing of H.M.S. "Hood".*

Number of sets in ship	4	Pitch of teeth	1in. normal
Number of pinions per set	2 (1 H.P. and 1 L.P.)	Width of face (total)	75in.
P.c.d. of wheel	143·78in.	Spiral angle	about 30°
P.c.d. of pinions	H.P. 20·174in. L.P. 27·51in.	Total s.h.p.	144,000
Number of teeth in wheel	392	Propeller r.p.m.	210
Number of teeth in pinions	H.P. 55, L.P. 75.	Speed of vessel	32 knots

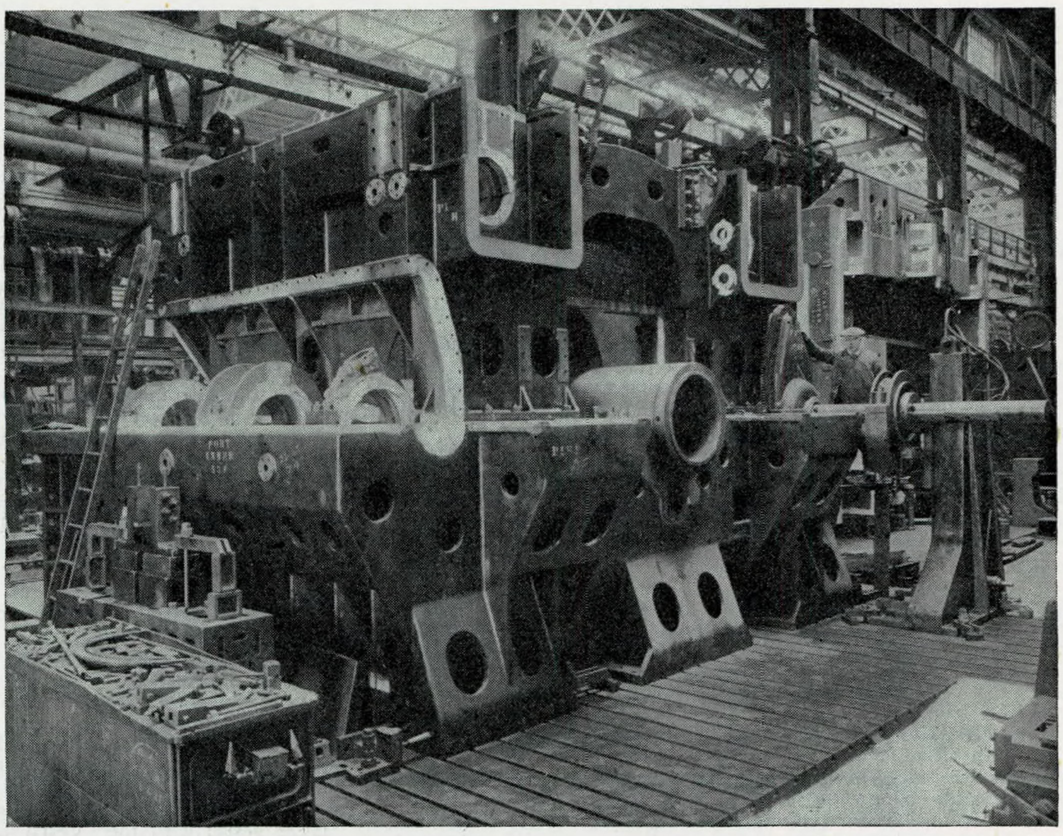
instead of $\sqrt{d}$ for rolling contact. It is interesting, however, to consider the calculated film thickness, *viz.* 0·000125in. in conjunction with the phenomenon of "scuffing". The small magnitude of the film thickness would demand a high degree of surface finish for film lubrication to be the only governing factor. Undulations, cusps and facets present in a new gear would suggest that in the first hours of operation abrasion occurs. Under these conditions of surface imperfection, pure rolling contact seems preferable to sliding and no doubt the increased sliding velocity with an all-addendum pinion results in "scuffing", whereas otherwise pitting might or might not occur. On the other hand, once the tooth flanks have obtained that polished mirror-like surface after a period of operation, there seems little doubt, as Parsons showed, that both from a point of view of surface stress and lubrication, the A.A. pinion teeth having less sharp curvature will successfully withstand an increase of load as compared with the pinion having the normal ratio of addendum to dedendum. Messrs. Parsons insist on the lapping of all-addendum pinions and this is probably done to prevent "scuffing" action, which would otherwise be likely to occur in the first hours of life of the gear. The theory of oil film lubrication certainly indicates the necessity for a running-in period for a gear before fully loading it. Until this running period is successfully bridged, oil film lubrication is not fully applicable, nor are the advantages accruing from the all-addendum gear.

Increased pressure angle from, say, $14\frac{1}{2}°$ to $22\frac{1}{2}°$ also increases both the mean radius of curvature of the pinion tooth profile, and the relative speed of the two surfaces, with increased oil drag. However, the length of the path of contact is reduced.

From the foregoing it is seen how from considerations of oil film lubrication the following relationship is obtained, *viz.*:—

$$P=K\sqrt{d}$$

where K is a constant and d is the pitch circle diameter, and the speed is taken as approximately constant for marine gears. Parsons adopted this relationship before Martin made his analysis, and the value of K adopted was 180 to 220 for gears for the Merchant Service and for Naval work 200 to 225, and up to 310 with A.A. teeth. A speed factor has so far been omitted; this is probably due to compensating effects, because although an increase in speed increases the possible load for the same oil-film thickness, the accelerations and decelerations due to gear-cutting inaccuracies also increase, resulting in an increase of inertia loading.



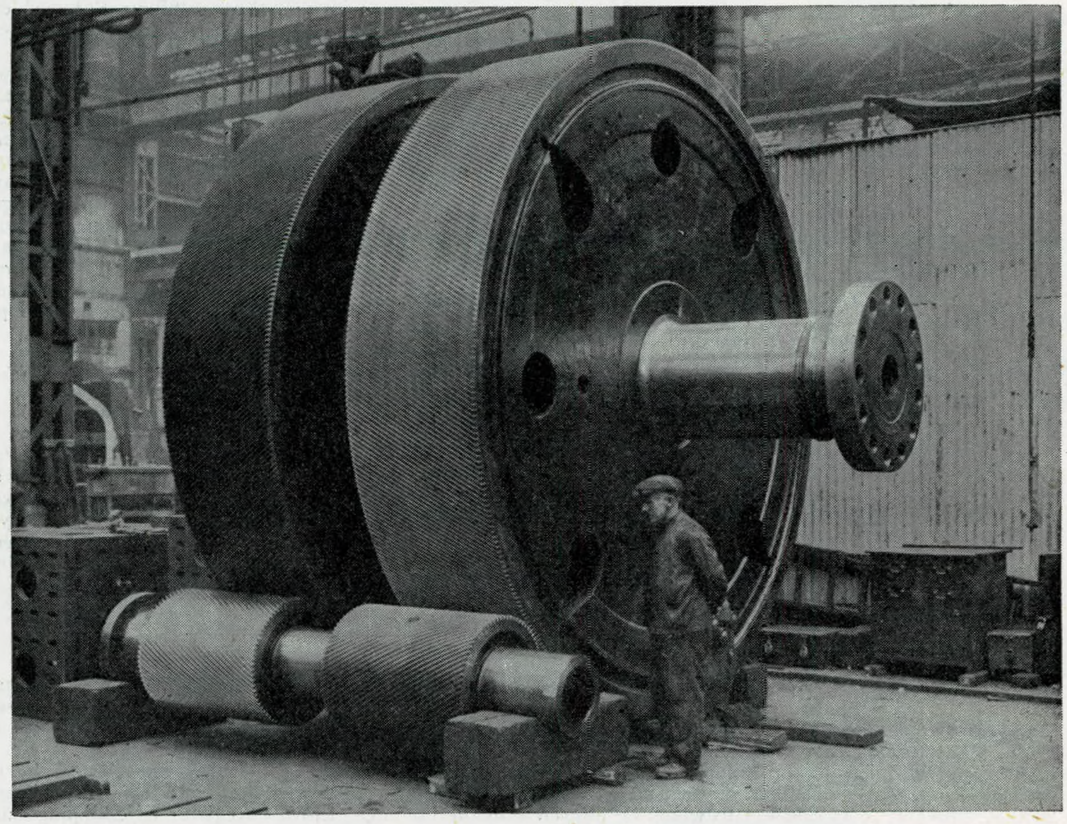


FIG. 16.—*Gearing of Q.S.S. "Queen Mary".*

Number of sets in ship	4	Number of teeth in pinions	H.P. and 1st I.P. 52, 2nd I.P. and L.P. 77
Number of pinions per set	4 (1 H.P., 1 1st-I.P., 1 2nd-I.P. and 1 L.P.)	Pitch of teeth	1in. normal
P.c.d. of wheel	162·49in.	Width of face (total)	57in.
P.c.d. of pinions	H.P and 1st I.P. 19·07in., 2nd I.P. and L.P. 28·24in.	Spiral angle	about 30°
		Total s.h.p.	158,000
		Propeller r.p.m.	180
Number of teeth in wheel	443	Speed of vessel	—

Gear Loads.

As already explained in the section dealing with lubrication, gear loading depends to a large extent on the ability of the gear teeth to maintain oil film lubrication during operation. Regarding the bending stress, Parsons showed that under static conditions a load of 40,000lb. per inch of face width was required to break the teeth of a 6in. diameter pinion, when meshed with a gear wheel of 26in. diameter, the normal pitch being 7/12in. and the spiral angle 30°. This experiment, together with the practice of rounding the bottom of the interspaces between the teeth, certainly eliminated bending fatigue stress as being a controlling factor in gear loading. Furthermore, it is borne out in practice, since only in cases of defective material or excessive resonant vibration has failure occurred by tooth fracture.

Apart from bending stress on the teeth, they are subject to surface stress, and Hertz has shown that the following relationship holds, *viz.* $P=k\times d$, k being a constant and P and d being, as before, the load per inch width of face and the pitch circle diameter, respectively.

d appears in this formula for a similar reason to that for which it appears in that dealing with oil film lubrication, namely, because it directly affects the mean radius of curvature of the profile of the pinion tooth. In the early days, for pinions 10in. in diameter and less, the value of k was taken as 60 to 75 with a view to keeping down the surface stress. Above 10in. diameter the radius of curvature was such that with the increased radius oil film lubrication became the limiting factor rather than surface stress, and as already mentioned in dealing with lubrication, the gear size was calculated using the formula

$$P=K\sqrt{d}$$

where K had the values 180 to 220 for gears for the Merchant Service, and 200 to 225 for Naval work and up to 310 with AA teeth.

With all-addendum pinions it is clear from Hertz's work that the surface stress is considerably reduced, and it is of interest to note that the value of k in the formula $P=k\times d$ has in some cases reached about 120 for pinions of about 5in. pitch circle diameter.

General.

In the early gears, Parsons used 7/12in. and 1in. lead hobs for the 1st and 2nd reduction gears, respectively, and these leads are in very common use to-day for the production of marine gears. The largest gear ratio used by Parsons was 26:1 in the case of the s.s. "Cairnross" fitted with single-reduction gears. In naval practice, gear ratios of single reduction vary from about 5 to 13, whilst for the early merchant vessels fitted with single-reduction gearing the ratio varied from about 14 to 20. In the case of double-reduction gears, the first-reduction ratio at the H.P. and L.P. pinions was about 8 and 6, respectively, and the second-reduction ratio about 6 to 1. The general practice is to balance pinions over 6in. diameter dynamically and the wheels statically. The necessity for good balance is obvious from considerations of satisfactory meshing of the gear teeth.

At the ends of the tooth faces the teeth are relieved so as to prevent excessive local stress at the root section. To do this the teeth are cut back at an angle of 45° to the pitch line, and are further tapered on each side of the tooth to an extent of 30/1000in. to nothing at a distance of ⅜in. from the top of the tooth.

Backlash is also allowed between the engaging gears and this is sufficient to allow for differences of temperature between the gear case and the gear wheels. Such clearance is generally determined by measuring the axial "float" of the pinion shaft. Typical values for this axial clearance are as follows, viz.:—

Single-reduction gearing, 7/12in. normal pitch.
Axial float 80-100/1,000in.

Double reduction gearing, 7/12in. normal pitch throughout, or with 1in. normal pitch at 2nd reduction.
Total axial float at 1st-reduction pinion (*i.e.* due to 1st and 2nd reduction axial floats) ... ... 180-200/1,000in.
Axial float at 2nd-reduction pinion ... 80-100/1,000in.

Summing up Parsons' work on mechanical gearing, it is of interest to indicate what his work achieved. Not only did it make possible the turbine propulsion of slow-speed merchant ships, but actually increased the power and speed of naval vessels and passenger liners due to the weight reduction obtained. Further, the application of gearing resulted in considerable saving in steam consumption, because both propeller and turbines could be operated at their most economical speed. Single-reduction gearing showed, in the case of vessels of moderate size and speed, a saving of some 5lb. of steam per s.h.p. hour as compared with direct-turbine drive, whilst the introduction of double-reduction geared turbines showed steam consumption of 10lb. per s.h.p. hour with saturated steam and 8lb. per s.h.p. hour with 200° superheat, giving a fuel economy some 20 per cent. over that obtained with steam reciprocating machinery.

At the time of the first application of mechanized gearing to ship propulsion, both electrical and hydraulic methods of transmission were under consideration, and much time and money was spent in Germany on the development of the latter means of gearing. As has been shown, Parsons' immediate success opened up the almost universal application of mechanical gearing to marine turbine propulsion. In tracing the evolution of mechanical gearing for marine propulsion it is fitting to conclude with some remarks on powers transmitted in the case of notable naval and mercantile vessels. Fig. 15 shows the gearing fitted over 20 years ago in H.M.S. "Hood", where a total of 150,000 s.h.p. was transmitted to four propeller shafts running at 210 r.p.m. Fig. 16 shows the gearing fitted in the Cunard Liner "Queen Mary", the service shaft horse power being 158,000, again on four shafts, the revolutions per minute being 180; there is, however, an ample margin of power. Comparing this latter vessel with the "Vespasian", it is seen that the power transmitted through a single gear has increased from 1,100 to about 40,000 s.h.p., thus fulfilling Parsons' early prediction that so far as the gearing was concerned, there seemed no limit to the power which could be transmitted.

With increased power transmitted and size of gear, machine index errors, however, are correspondingly increased, and it becomes more difficult to eliminate undulations along the tooth space. In paying tribute, therefore, to the magnificent achievements of Parsons, it seems fitting to mention the co-operation and encouragement given by the Engineer-in-Chief's Department of the Admiralty on matters concerning the reduction of gear-cutting inaccuracies. Without such work and endeavour, the powers to-day successfully transmitted through gearing would not have been possible.

Although Parsons' work on mechanical gearing envisaged turbines as the prime mover, there seems little doubt that the future will show it to have a most important bearing on geared oil-engine propulsion.

Acknowledgments.

I desire to acknowledge the assistance given by Mr. S. S. Cook, F.R.S. of Messrs. Parsons Marine Steam Turbine Co., Ltd., by Sir Stephen Pigott of Messrs. John Brown & Co., Ltd., and by Mr. R. McLeod of Messrs. Power Plant Co., Ltd., West Drayton, in matters relating to the historical details and for illustrations kindly given. I also wish to express my indebtedness to Dr. G. A. Tomlinson of the Metrology Department, National Physical Laboratory, for information regarding the checking of hobs, and also for the assistance given by members of my staff at Lloyd's Register of Shipping.

VOTE OF THANKS TO DR. DOREY.

Admiral of the Fleet Lord Chatfield, P.C., G.C.B., O.M., K.C.M.G., C.V.O., D.C.L., who proposed a vote of thanks to Dr. Dorey for his Lecture, said his pleasure in doing so was not lessened by the fact that Dr. Dorey was a member of the Council of the Institution of Naval Architects, an Institution to which he himself had the honour to belong, as well as a Vice-President of The Institute of Marine Engineers. The standard of the Parsons Memorial Lectures was a very high one, as was only fitting, since they were given in memory of so great a man, and there would be general agreement that Dr. Dorey had well maintained that standard. The Lecture must have been of absorbing interest to students of the subject with which it dealt, and was of interest also even to laymen who, like himself, had profited by being able to use the machines which others had invented, designed and constructed.

Dr. Dorey was one of the most distinguished marine engineers now living, and in his position as Chief Engineer Surveyor of Lloyd's Register of Shipping had no doubt unique opportunities of co-ordinating the theory and practice of marine engineering. The good use which Dr. Dorey had made and was making of those opportunities had been shown by his Lecture that day and by the large number of papers of great importance which he had contributed to many Institutions.

Personally, he had had, like the President, the great privilege of knowing Sir Charles Parsons. When Controller of the Navy, fifteen or sixteen years ago, Sir Charles frequently came to see him with ideas and suggestions which he wished to induce the Engineer-in-Chief of the Navy to adopt, principally in regard to extremely high boiler pressures and superheated steam.

Dr. Dorey had rightly said that the Navy owed a great debt of gratitude to Sir Charles Parsons. The Navy recognized this, or at any rate the old Navy did; and one advantage of such Lectures as that which Dr. Dorey had given was that they enabled one to realize the immense energy, as well as the genius, which lay behind the fruition of Sir Charles's work.

It was difficult to imagine what the Navy would have done in this war and the last without the work of Sir Charles Parsons, particularly when the submarine menace made it necessary always to steam at high speed, and when it was necessary to steam economically so as to save fuel, which at the time of the last war was sometimes coal. Moreover, Sir Charles's work made it possible for the time spent on repairs and maintenance to be reduced to a minimum, and in the last war failures in marine engineering were the least of all the mechanical troubles experienced. The wonderful steaming of the Fleet in the present war—which, ship for ship, probably exceeded by a great deal what was done in the last war—was again an example of what the Navy owed to Sir Charles Parsons. From the Admiral and the Captain on the bridge down to the stoker and the engine-room artificer below, everyone in the Navy owed him an immense debt.

He was glad that Dr. Dorey had paid a compliment to the department of the Engineer-in-Chief at the Admiralty for their collaboration and help in the great days when Parsons was, as it were, moving the world. The responsibility which fell on the Engineer-in-Chief of the day was very great; he had to decide what should be done, and the fate of the British Fleet, and perhaps of the Empire and of the world, might turn on some decision taken in his private office. Perhaps those who owed most to Sir Charles were the members of that very distinguished branch of the Service to which Sir George Preece had the honour to belong.

Sir Richard W. Allen, C.B.E., who seconded the vote of thanks, praised the decision of the Royal Society to perpetuate the memory of Sir Charles Parsons by annual Memorial Lectures. Many young engineers in particular were inclined to take things for granted, and coupled the name of Sir Charles Parsons solely with the steam turbine. A year ago a remarkable Memorial Lecture was delivered by Sir Stanley Goodall, the Director of Naval Construction, who dealt with Sir Charles's research work on the propeller; this year, the President and Council of The Institute had been very fortunate in securing the services of Dr. Dorey, and in selecting the subject of mechanical gearing. So far as he knew, there had until now been no such concise history of the matter as Dr. Dorey had given, showing the extraordinary vision of the future which Parsons must have had and what a wonderful brain he had, enabling him to see that the gearing was the solution of the problem of ship propulsion in the future. Personally, he had been told by Sir Charles that there were at the time many people who opposed that view, and one well-known Liverpool shipowner, when Sir Charles suggested that the new gear should be used in a ship of his, replied "What? Drive my ship with a cogwheel? No fear!"

Often, when butterfly-hunting, one found on opening the net that the butterfly one thought one had caught had escaped. Gearing was something like that; one encountered difficulties and hoped to overcome them by changes in design, but on "opening the net" one was disillusioned. Dr. Dorey had clearly shown that it was only by dogged perseverance, trying first one thing and then another, that the results obtainable to-day were achieved.

Sir George Preece had remarked to him how remarkable it was that these gears, designed and constructed before the war, should have stood up so well to war-time stresses, when ships were developing higher speeds and more power than had ever been contemplated, and doing so under the terrible conditions of the war. Another remarkable fact, mentioned by Dr. Dorey, was that the steel chosen by Sir Charles Parsons for gear wheels, pinions and so on was still being used for those purposes to-day.

Everyone who had heard it had enjoyed Dr. Dorey's Lecture, and it was striking that so busy a man as Dr. Dorey should have found the time to prepare it in these strenuous days.

The vote of thanks was carried unanimously, with acclamation.

Dr. S. F. Dorey, in reply, mentioned that he had had the pleasure of meeting Sir Charles Parsons on one occasion only, at the time of the trials which were carried out on the *King George V*. He was proud, he said, to have met so great a man.

The proposer and seconder of the vote of thanks had said a good deal about himself, but he would like to say that without his staff, and without the staff of Lloyd's throughout the world, it would have been quite impossible for him to prepare not only the Lecture which he had given that day, but the many other papers which he had read elsewhere. They represented merely the accumulation of the experience and effort of his colleagues throughout the world, and it was only right that their able assistance should be recognized.

PRESENTATION OF THE PARSONS MEMORIAL MEDAL TO DR. DOREY.

Professor A. C. G. Egerton, M.A., F.I.C., B.Sc., F.R.S., then, on behalf of the Royal Society, presented the Parsons Memorial Medal to Dr. Dorey. The presentation, he explained, should have been made by Professor G. I. Taylor, but Professor Taylor was absent on a mission to America. Personally, he was very pleased that the duty had fallen to him, because he knew Sir Charles Parsons very well and had been very devoted to him.

Dr. Dorey had received many medals. He had twice received the Denny Gold Medal of The Institute of Marine Engineers, which showed in what high esteem his work was held by the engineering world. He was a great authority on marine machinery, and he had given that evening a delightful and most interesting account of one of the facets of Sir Charles Parsons' work. The Parsons Memorial Lectures gathered together the progress in engineering science which had been made along the lines which Sir Charles had followed and which he had done so much to develop. It was indeed wonderful to see how, as Dr. Dorey had shown, Sir Charles was able to reach the goal he had in view without being swamped in mathematical analysis or led up blind alleys.

The Medal was then presented, amid applause.

Dr. Dorey, in response, said it had been stated that he had received many medals, but no-one could receive a medal of greater value than that which had just been handed to him, connected as it was with the memory of Sir Charles Parsons.

VOTE OF THANKS TO THE PRESIDENT.

Mr. H. J. Wheadon (Chairman of the Council) proposed a vote of thanks to the President, Lord Mottistone, for his informative Presidential Address and for occupying the Chair that evening. Lord Mottistone's many activities, he said, were so well known that there would be a general feeling of gratitude that, despite the many calls on his time, he had spared an evening to attend a meeting of The Institute.

In the course of his Address, the President had touched on several phases of The Institute's activities and of the work which was being done by the Council at the present time, chief among them being efforts to implement The Institute's duty under the terms of its Royal Charter, namely to maintain and improve the status of the marine engineer and of the marine engineering profession. Certain of the Council's recommendations must, on account of their nature, be submitted for approval to Parliament or to the appropriate Government department before they could become effective; and in that respect The Institute was fortunate in having as its leader one who was extremely skilled in Parliamentary procedure and who had long been keenly interested in maritime affairs. Lord Mottistone loved the sea and admired the men who sailed upon it; and therefore The Institute could rest assured that he would continue to support, to the utmost of his power and skill, any measures intended to improve the conditions and status of those fine men who had taken to the sea for a career, whether they served on the deck or in the engine-room, whether they were officers or ratings, and whether they served in the Royal Navy or in the Merchant Navy.

Mr. A. F. C. Timpson, who seconded, recalled that the President at the outset of his Address had referred to the virtue of brevity. Personally, therefore, he would be very brief, and would content himself with formally seconding the vote of thanks.

The vote of thanks was carried unanimously, with acclamation.

The President, in reply, said that no thanks were due to him; as Mr. Wheadon had said, he loved the sea and those who sailed upon it. He would endeavour to be worthy of the confidence which The Institute had placed in him.

The meeting terminated at 7.40 p.m.

Informal Dinner to the President.

Following the Parsons Memorial Lecture, a number of Members of Council and Vice-Presidents entertained the President at an informal dinner at St. Ermin's Restaurant, Caxton Street, S.W.1. The hosts were: Messrs. H. J. Wheadon (Chairman of Council), W. S. Burn, M.Sc., A. E. Crighton, James M. Dewar, J. D. Farmer, Sterry B. Freeman, C.B.E., M.Eng., W. Dennis Heck, B.Sc., F. M. Jones, B.Sc., R. S. Kennedy, Major E. W. B. Kidby, O.B.E., Geo. A. Laing, A. F. C. Timpson, M.B.E., W. T. Williams, O.B.E., B.Sc., Wh.Ex., Arthur R. T. Woods, F. W. Youldon, A. Robertson, C.C. (Hon. Treasurer) and B. C. Curling (Secretary).

In addition to the President, the following guests were present: Stanley F. Dorey, D.Sc., Wh.Ex., Sir John E. Thornycroft, K.B.E. (President-elect, Institution of Civil Engineers), Sir Noel Ashbridge, B.Sc. (President, Institution of Electrical Engineers), Col. S. J. Thompson, D.S.O. (President, Institution of Mechanical Engineers), Eng. Vice-Admiral Sir George Preece, K.C.B., Professor A. C. G. Egerton, M.A., F.I.C., B.Sc., F.R.S. and Mr. Thos. A. Crompton.

The element of austerity now imposed by war restrictions for such functions was greatly outweighed by the spontaneous geniality which prevailed throughout the proceedings.

The Loyal Toast having been drunk, the **Chairman of Council, Mr. H. J. Wheadon,** rose and said: My Lord President and Gentlemen, I rise on behalf of the Council to extend to you, Sir, and to the Council's other distinguished guests, a very hearty welcome to our table this evening, and to express our appreciation of the honour you have bestowed upon us in accepting our invitations. The opportunities for social intercourse are so rare in these austere days that it occurred to me that Members of Council might welcome the opportunity to take advantage of the occasion of the Parsons Memorial Lecture to meet our President, together with a few representatives of kindred institutions who were attending the lecture, at an informal dinner party afterwards. (That so many are present is a measure both of the esteem we have for our President and indirectly of the unfailing high quality of Dr. Dorey's lectures). In passing I would like to say on behalf of the London Members of Council how pleased we are that several of our provincial colleagues have been able to join us, some of whom have travelled considerable distances to attend. Mr. Sterry Freeman, one of our Liverpool Vice-Presidents, is here, also Major Kidby representing Swansea, Mr. Laing from Hull, and our old friend Mr. Arthur Woods.

The majority of us sitting around this table tonight, my Lord President, are corporate members of one or more technical institutions—we are professional engineers of one branch or another covering in the aggregate I suppose practically the whole field of science and technology. It appears to me to be not out of place, therefore, to express the hope that in these days of the slowly awakening consciousness of the lay public and others, of the vital part to be played by technical men in defeating our highly scientific enemy, that we co-operate to the full in matters affecting our common interests or which tend to raise the mean level of our scientific or technical knowledge. We should see to it by all the means within our power that the products of our knowledge and labour are not wasted, but on the contrary are made available to those who can best use them in time of war to the discomforture of our foes, and when peace returns to the greater benefit of mankind.

The President, Lord Mottistone, suitably responded and all the other guests, in a series of brief speeches, echoed the note of warm appreciation of the welcome accorded to them on this occasion, and particularly expressed support for the Chairman's appeal for closer co-operation between the institutions represented at that gathering.

The proceedings ended at 10 p.m.

ELECTION OF MEMBERS.

List of those elected by the Council at the Meeting held on Monday, 28th September, 1942.

Members.
Henry John Cooper, Temp. Engr. Lt.-Com'r., R.N.R.
Joseph Lawrence Dunn, Lt.-Com'r.(E.), R.C.N.R.
James Gilchrist.
Thomas Irving Grainger.
Thomas Hughes.
Norman Clifford James.
Jack Silvester Mason.
James Murray McNeish.
Sydney Townsend.

Associates.
John Francis Boylan.
William Walter Alan Brownsea.
James William Chase, Temp. Sub. Lt.(E.), R.N.R.
George Donald.
William Leslie Espin.
Robert Guthrie.
Erick Manners.
Leslie Randolph Moores.
James Milne Watson.

Transfer from Associate Member to Member.
William Leonard Evans.

Transfer from Associate to Member.
Patrick Dalrymple Davidson.

Transfer from Graduate to Associate.
Douglas Evan Arthur Coombs.

Transfer from Student to Graduate.
Jack Henry Maycock.

ADDITIONS TO THE LIBRARY.

Presented by the Publishers.

The Mechanical Testing of Metals and Alloys. By P. Field Foster, B.Sc.(Lond.), M.Sc.(Wales), Wh.Ex. Sir Isaac Pitman & Sons, Ltd., 3rd edn., 317pp., 234 illus., 18s. net.

This book describes modern testing equipment with its method of use, and embraces in a practical way the theory underlying present-day developments in the testing of metals and alloys.

A full and generally favourable criticism of the second edition of the book was published in the August, 1940 issue of the Transactions (*Vol. LII, Part 7, p. 140*), and the fact that this edition has been exhausted so rapidly is an indication of the growing demand for information on the testing of metals and alloys and of how well this book is meeting the demand.

In this new edition descriptions of new apparatus have been added, the chapter on notched-bar impact testing has been revised in the light of recent research, and a short account has been given of damping capacity in metals. Other chapters in the book deal with: elasticity; the structure of metals; universal testing machines; tension and bending tests; testing machine accessories; extensometers and records; torsion testing machines; repeated stresses; elastic constants—testing of wire and sheet metal; and some test phenomena and results. Some tables of properties of metals and alloys, a bibliography and an index complete the book.

Marine Engineering, Vol. I. Edited by H. L. Seward, Ph.B., M.E. The Society of Naval Architects and Marine Engineers, New York, 353 pp., copiously illus., $6 post free U.S.A.

The success of their publication "Principles of Naval Architecture", issued in 1939, has encouraged the Society to issue this companion book on marine engineering.

In the book on naval architecture each chapter is written by a separate author. Since this not only distributes the work but also affords an opportunity, by a careful selection of authors, to obtain the advantages of professional specialization, a similar course has been followed in this book on marine engineering.

The editor and authors have assumed that the reader is reasonably familiar with the underlying scientific principles and also that he is not entirely unacquainted with merchant ships. The basic subjects of thermodynamics, theoretical and applied mechanics, machine design, etc. are only referred to on occasion or to develop a particular scientific analysis when necessary. Where descriptions are given or designs are described, they represent conservative modern practice.

Vol. I deals with the main propulsion units after some general principles of design are presented. It includes chapters on: propelling machinery; power and revolutions; procedure in general design; boilers; reciprocating steam engines; steam turbines; Diesel engines; reduction gears; propellers and shafting; materials and metallurgical engineering. Vol. II, which will be published at a later date, will deal with auxiliaries after some special marine developments in the basic sciences have been presented.

Each author is a recognized authority in his field, and the editor has kept in mind the possibilities open to a resourceful teacher of using the tabulated matter or the illustrations as the basis of numerical problems.

Abstracts of the Technical Press

Machinery of French Motor Liner "Maréchal Pétain".

Most of the main and auxiliary machinery of the triple-screw passenger motorship "Maréchal Pétain" (*described in abstracts on pp. 130 and 136 of* TRANSACTIONS, *October and November, 1941*) has now undergone its shop trials. The main propelling machinery, consisting of three French-built 11-cylr. single-acting two-stroke Sulzer Diesel engines, has a normal output of 25,000 b.h.p. at 131 r.p.m., but is to be capable of developing 31,000 b.h.p. at 141 r.p.m., for a continuous period of 24 hrs. The cylinders have a diameter of 720 mm. and a piston stroke of 1,250 mm. Each engine drives its own scavenging-air pump, which is located at the end of the crank-shaft. The shop trials, carried out at the engine-builders' works at Saint Denis, are reported to have been highly satisfactory, and no difficulty was experienced in maintaining the specified output at both normal and maximum speed. The fuel-oil consumption varied from 0·36lb./b.h.p.-hr. at half power to 0·366lb./b.h.p. at normal full power, the consumption at maximum output being in the neighbourhood of 0·387lb./b.h.p.-hr. The electrical equipment of the vessel includes four Schneider-Westinghouse 220-volt three-phase 50-cycle a.c. generators, each rated at 900 kVA., and each directly driven at 250 r.p.m. by a 6-cylr. single-acting two-stroke Sulzer Diesel engine. These engines have likewise undergone exhaustive shop trials at various loads with very satisfactory results. The fuel-oil consumption per kW.-hr. is stated to have been 0·638lb. at 50 per cent. load and 0·563lb. at full load.—*"Journal de la Marine Marchande", Vol. 24, No. 1,159, 26th February, 1942, pp. 231-232.*

The French Collier "Paul de Rousiers".

The cargo steamer "Paul de Rousiers", built by the Chantiers et Ateliers de Provence, Port-de-Boue, and launched in May, 1941, has now been handed over to her owners, the Union Industrielle et Maritime. The ship is the first merchant vessel of any size to be completed in France since the armistice, and her construction was delayed by the lack of materials. Furthermore, some of the auxiliary machinery and equipment, which was to have been manufactured in the U.K., had eventually to be replaced by French products, the supply of which involved considerable difficulties. The "Paul de Rousiers" is a single-screw vessel about 375ft. in o.a. length, with a breadth of 48½ft. and a moulded depth of 24ft. 4in. She is designed for the carriage of coal and similar bulk cargoes, and the total capacity of the four holds is 4,700 tons, although the maximum load draught of the ship is only 20ft. The hull is of welded construction and a double bottom extends from the fore peak to the after peak. The cargo hatches are of exceptionally large size and are served by three 3-ton and one 5-ton derricks. The propelling machinery, which is located amidships, consists of a triple-expansion engine and Götaverken exhaust turbine developing 2,000 i.h.p. supplied with steam at a pressure of 256lb./in.^{2} and superheat temperature of about 670° F. by two coal-fired Hudson-Capus boilers. The designed speed is about 12 knots. Two similar vessels are building in the same yard for the French Government. It is pointed out that before the war France possessed 86 colliers with a total gross tonnage of about 250,000 tons, but that she now has only 40 vessels of this kind with a tonnage of 135,000 gross tons.—*"Journal de la Marine Marchande", Vol. 24, No. 1,163, 26th March, 1942, p. 344.*

The Training of Marine Engineers.

The Merchant Navy Officers' Training Board, whose members include representatives of the Navigators and Engineer Officers' Union, is reported to be considering the problem of training for officers and men of all departments of the Mercantile Marine. It has been suggested that one of the basic features of a satisfactory training scheme for deck and engineer officers should extend over a period of four years, and include a course of study of six or twelve months at a central Merchant Navy Officers' College maintained either by the Government itself or by the shipping industry with Government supervision and financial assistance. Several of the leading maritime nations follow this procedure, which, it is considered, has much to commend it, *more especially if during this period of shore training, future deck and engineer officers of contemporary ages can be brought together* in the social and disciplinary life of the College and, where necessary, in training classes in subjects of common interest. It is believed that such courses would provide far sounder theoretical training in many subjects than can be conveniently imparted afloat, and that such a college, if well run, would develop an *esprit de corps* throughout the Service as it gradually became officered by men who had all been through the same "mill".—*"Merchant Navy Journal", Vol. IV, No. 4, April, 1942, p. 64.*

Training New Personnel for the American Merchant Marine.

On the 1st March, 1942, the administration of the U.S. Merchant Marine Cadet Training System was transferred from the Maritime Commission to the U.S. Coast Guard. The period of training for cadets has now been reduced to 16 months, this being made up of two months' preliminary shore training at one of the basic schools at New York, New Orleans or San Francisco, followed by six to eight months afloat in an ocean-going merchant vessel, and completed by six to eight months of advanced courses at the U.S. Merchant Marine Academy at Great Neck, Long Island, N.Y. Graduates are then eligible to be examined for certificates as Third Mates or Third Assistant Engineers. Instruction in naval subjects is also provided at the Academy, and on qualifying as junior officers of the above grades, graduates are eligible for enrolment in the Merchant Marine Reserve of the U.S. Navy as Ensigns, although they are not called upon to serve as such while they remain in merchant vessels. Candidates for appointments to the U.S. Merchant Marine Cadet Corps must be unmarried citizens of the U.S., native born or naturalised for at least ten years, between the ages of 18 and 23 years, with a high school education or its equivalent, in good health and of good moral character, and able to qualify for enrolment in the M.M. Naval Reserve. While undergoing training cadets are paid $65 per month by the Government while ashore and by the shipping companies while afloat. They are also provided with uniform, textbooks, board and lodging, and medical attendance. While his ship is in port, each cadet is examined by a Cadet Training Instructor and is assigned studies to be carried on at sea. The cadets are quartered with the ships' officers who also supervise their practical work. The U.S. Merchant Marine Academy, at which the cadets undergo their final eight months' course, is now training 2,400 cadets at a time, whilst the basic schools have also been expanded to accommodate over 300 cadets each. The Academy has three training ships attached to it. As a result of this intensified training, approximately 6,000 cadets of the U.S. Merchant Marine Cadet Corps are expected to qualify as junior deck or engineer officers during the years 1942, 1943 and 1944. During this period, it is anticipated that about 1,200 cadets will also graduate from the four State Maritime Academies which provide similar courses of instruction and are likewise supervised by the U.S. Coast Guard. The regulations for admission to these establishments are practically identical with those for appointments to the Cadet Corps, but certain fees have to be paid to the respective States by their cadets. The course of training comprises 12 months ashore and six months afloat in one of the Coast Guard training ships. The U.S. Maritime Service also maintains two officers' schools at Fort Trumbull, New London, Conn., and Government Island, Alameda, Cal., at which seamen and E.R. personnel with a minimum of 14 months' sea service, are given a four months' course to qualify them to take the examinations for certificates as Third Mates or Third Assistant Engineers. These two schools are now turning out 4,200 qualified deck and engineer officers each year, and this number is to be increased. Third Assistant Engineers are likewise being recruited from students of approved engineering colleges who have completed a three-year course of study. They are eligible to be examined for certificates after completing three months' service afloat in a merchant vessel. It is proposed to enrol 1,500 such students in the Coast Guard Reserve within the next two months. They will remain at their homes until they are sent for to undergo a short course at Hoffman Island, in New York Harbour. This course is to be of three weeks' duration and will provide instruction in lifeboat work and other subjects, after which the students will join various merchant vessels as junior uncertificated engineers to put in the necessary sea service to qualify them to take the examination for a Third Assistant Engineer's certificate.—*T. Knight, "The Nautical Gazette", Vol. 132, No. 6, June, 1942, pp. 20-23.*

Laminated Plastic Materials in American Ships.

Laminated plastics, such as micarta, manufactured by the Westinghouse Company, are used for a variety of purposes in the latest American ships. Micarta consists of layers of fibrous substance impregnated with synthetic resins, hot-pressed into a dense material of high mechanical strength, with excellent resistance to corrosion and low moisture absorption. The resiliency of micarta makes it specially suitable for applications where severe shock loads and fatigue stresses are encountered. When submerged in water and operated against bronze, brass, Monel metal or steel, the coefficient of friction is only of the order of 0·002, and because of this marine micarta is being employed in lieu of lignum vitæ for stern-tube and A-bracket bearings. A micarta-lined stern-tube bearing fitted to the American s.s. "Steel Exporter" in 1933, had worn only $\frac{3}{32}$in. when examined more than four years later, and was still reported to be in excellent condition just before the war. The record of a micarta stern-tube bearing fitted to one of the propeller shafts of an 8,000-ton twin-screw ship running between the East Coast and Valparaiso, is claimed to be of particular interest. The micarta bearing was originally fitted in the starboard stern tube in August, 1936, when the original clearance was 0·06in. Lignum vitæ strips were fitted to the port stern-tube bearing at the same time, and when the shafts were drawn for examination in April, 1939, the wear of the micarta bearing was found to be only 0·045in., whereas that on the lignum vitæ bearing was 0·375in., and the wear on the shaft liner was between 0·125 and 0·190in. No appreciable wear could be measured on the shaft liner run in the micarta bearing. New micarta bearing strips were thereupon fitted in the port stern tube in place of lignum vitæ. During a recent voyage, one of this vessel's propeller blades broke off, and she had to do 6,850 miles under those conditions. Although the speed of the engines was adjusted to keep the vibration down as much as possible, there was still a considerable amount of pounding in the stern-tube bearing, and this eventually caused the stern tube to crack at the bottom of its outer end over a width of 6in. and a length of 24in. The laminated plastic bearing strips, however, suffered no damage whatever. The maximum wear that could be measured on the bearing strips was 0·16in. after more than five years of continuous service under very adverse conditions. This wear had only occurred on the bottom half of the bearing, there being practically no wear on the top half. The bearing in question had four keeper plates about 2in. wide. The micarta was fitted in small strips about 9in. long with water grooves between each set of strips. It was found that the strips at the outboard end had actually been pounded into the bearing shell, and it became necessary to cut most of them out with the aid of cold chisels. The shaft liner, however, was in good condition. Micarta-lined pintle bushings are fitted in many American vessels, including cargo vessels of the C-1, C-2 and C-3 classes. It is claimed that apart from the long life of these micarta pintle bushings, the press fits do not crack micarta bushes, whereas lignum vitæ frequently cracks or chips when forced into position. Although the first cost of micarta is slightly higher than that of lignum vitæ, it is easier to handle and machine, so that the actual cost of finished micarta-lined bearings is about the same. Micarta laminated bearings have given satisfactory service over two years in the main condenser circulating-water pumps of several U.S. battleships. Other machinery parts for which micarta is utilised include valve seats and discs, plunger rings for reciprocating pumps, stuffing-boxes for pumps, anti-vibration mounting blocks for engines (particularly in tugs), gears for cargo winches, guide bearings for the control rods of bilge valves, wearing rings for centrifugal pumps and insulated bulkhead glands for refrigerating pipes. The high dielectric strength of paper-base micarta makes this material useful for a variety of electrical purposes on board ship. These include switchboard panels, contactors and control apparatus.—*C. E. Enz and F. P. Hunsicker, "Marine Engineering and Shipping Review", Vol. XLVII, No. 6, June, 1942, pp. 121-123.*

Revised Formula for Sea Speed of U.S. Maritime Commission's Ships.

The U.S. Maritime Commission announced that a revised formula for determining the sea speed of its ships would be employed after the 15th May. The new regulations state that the "sea speed" of any vessel or group of sister ships, shall be the speed attained over a measured mile in deep water at 80 per cent. of the normal power of the engines, with a clean bottom, on an even keel, at a draught corresponding to the international summer load line, with no current, and the proper correction made for wind.—*"Marine Engineering and Shipping Review", Vol. XLVII, No. 6, June, 1942, p. 114.*

War Problems of the American Merchant Marine.

The nation-wide celebration of National Maritime Day on the 22nd May last, when 27 cargo ships were launched in 18 shipyards, included a dinner held by the Propeller Club, in Boston, at which the principal address was given by Rear-Admiral Emory S. Land, chairman of the U.S. Maritime Commission. In the course of his speech, he pointed out that the construction of the Victory Fleet by the shipyards and factories of the U.S.A. means the realisation of the greatest shipbuilding programme in the history of the world, involving the production of 2,300 merchant vessels, aggregating twenty-three million deadweight tons, in the two-year period ending with 1943, in addition to the building of about 700 smaller auxiliary craft. In 1937 the American shipbuilding industry comprised only 10 yards with 47 building slips, most of them engaged in the construction of naval ships. Apart from tankers, only two ocean-going cargo vessels had been built in the 15 years prior to 1937. At the present time there are approximately 60 shipyards on the Atlantic, the Pacific, The Gulf of Mexico and the Great Lakes already participating in the Maritime Commission's programme or under construction. Two-thirds of these yards, with a capacity of 295 ways, are engaged in the building of ocean-going merchant vessels averaging 10,000 d.w. tons. Since February, 1941, the Maritime Commission has authorised the building of 17 of those yards. Nearly 300,000 men are already employed in the mercantile shipyards of America, and 200,000 more are being or will be trained as rapidly as the shipbuilders can get them or train them. In addition, over 500 factories all over the country are producing materials, parts and supplies for merchant ships. These establishments employ nearly a million workers. The production of merchant ships rose from 28 in 1938 and 40 in 1939, to 103 in 1941. So far (*i.e., up to the 22nd May*), the 1942 output has amounted to 138 completed vessels. The stage of delivering—not launching—two ships a day has already been passed, and three ships a day will be put into service by the autumn. The 2,300 large merchant ships under construction include some 1,500 vessels of the EC-2 type known as Liberty ships, about 300 tankers with an average capacity of 135,000 barrels (=about 18,000 tons), approximately 500 of the Maritime Commission's standard design C-type ships and a few special designs. The additional 700 smaller vessels are made up of harbour service and sea-going tugs, coastal cargo vessels and tankers, and barges of various types. Some of the shipyards building Liberty ships are now turning them out in 90 days or less. One Pacific Coast yard launched a vessel of this type 43 days after its keel was laid, the ship being ready for service barely a month later. Apart from the immense programme of new construction, the shipyards of America are undertaking an enormous amount of maintenance and repair work on both U.S. and Allied vessels. During the year ending the 1st April, 1942, 450 ships were de-gaussed, 560 were armed and provided with quarters for guns' crews, etc., and 120 further ships were in hand for repairs and equipment. Thirty-one sabotaged German and Italian ships requisitioned by the U.S.A. in American ports were repaired, de-gaussed and equipped for war service at a cost of ten million dollars, in addition to which 65 Danish, Finnish, French and other foreign-owned vessels were taken over, armed and reconditioned. When the Maritime Commission came into existence in 1937, the remainder of the old World War laid-up fleet consisting of 198 ships, was taken over. Many of them were reconditioned and placed in service, the last one being completed in April, 1942. The average cost of this work was about $250,000 per ship. Many more went to the Allied Nations. Up to the 1st May, 1942, there had been acquired for the U.S. Army and Navy a total of 285 merchant vessels aggregating over three million tons d.w. for service as supply ships, transports and other auxiliaries. The vessels in question include the "America", which is now the Navy's "West Point", the "Manhattan", the "Washington", and more than half the new C-type ships built by the Commission since 1938. Furthermore, the Commission has acquired and turned over to the armed forces, including the Coast Guard, well over 800 small craft, among them many of the finest yachts in America.—*"Marine Engineering and Shipping Review", Vol. XLVII, No. 6, June, 1942, pp. 95-97.*

U.S. Marine Engine Production.

Official encouragement in many forms is given to the competitive rivalry between the various American shipyards engaged in the building of Liberty cargo vessels, and the following instance of the competitive spirit on the marine engineering side was recently recorded in the American Magazine *Time* in an article describing how Mr. Chas. E. Moore had tackled the sub-contracting problem. "A year ago, seizing on the bankrupt Joshua Hendry Iron Works, a dilapidated foundry in the middle of a pear orchard near Sunnyvale, Cal., Moore cleared off 34 acres of trees, put up 300,000ft.^{2} of buildings, and began to turn out mighty, two-storey 271,000-lb. triple-expansion engines for Liberty ships". Enough Hendry engines, it was added, will have been produced by the end of 1942 to drive a million tons of cargo shipping. A fortnight later the magazine published a telegram from the works manager of the Hoover-Owens-Rentschler Division of the Hamilton General Machinery Corporation, the builders of the "Ocean Vanguard's" engines. The telegram "noted with interest" the effort of the Joshua Henry Iron Works, and added: "This is good news to us, but in fairness to a thousand or more hard-working employees of the plant, will say that a little more than a year ago we delivered here the construction and tool-up drawings for these engines, and subsequently furnished same to all other engine builders, and for record we want you to know that our Company has shipped as of to-day enough of the same type triple-expansion engines to propel 850,000 tons of shipping, and from March 30 to the end of 1942 we propose to ship enough of these engines to propel an additional 1,550,000 tons of Maritime Commission Liberty ships, and this production we will also take in stride. Come on, Messrs. Kaiser and Moore!"—*"The Shipping World", Vol. CVI, No. 2,558, 24th June, 1942, pp. 435-436.*

How Far is Collaboration Between Shipyards Possible?

It has frequently been suggested that some form of collaboration between shipyards building the same type of vessel would facilitate an increased output. In point of fact, some collaboration in regard to drawing-office work is already taking place, but its extension to embrace building methods would be attended with serious difficulties for reasons of transport. The object of such collaboration would be to reduce time and cost by the adoption of some form of mass production, one yard taking the responsibility for the manufacture of certain components which would then be duplicated for other yards. In practice, it might be easier to divide a job up in such a way that each of, say, four yards took over a fourth part of it, but manufactured for all four ships. This should involve no more work for each of the yards concerned, and might effect a considerable saving of time. Difficulties of transport would, unfortunately, make it well-nigh impossible to realise such a scheme, although it could, perhaps, be applied to certain important phases of production. By mutual arrangement, the preliminary working drawings could be divided out between the different drawing offices, one dealing with the shell, plate edges and model, another with the framing, another with the deck plating and tank tops, and so on, each yard supplying its quota of working drawings and ordering the materials for the others as well as for themselves, the rolling mills duplicating the order for each yard. The latter would undertake to supply all the templates, marked battens, etc., for that part of the work undertaken by them. This would not be difficult, since duplicate templates could be made in the rough and packed together under the original template, which would serve as a jig for boring the others—as when drilling a packet of duplicate shell plates. In the case of four yards, each yard would only have to prepare one-fourth of the necessary templates, thus saving a considerable amount of time and labour. Where other yards followed behind the first, and transport was possible, the templates could be loaned to the others as desired or convenient. It might also be possible where yards are in close proximity, and the one yard possesses ample equipment for handling some particular job, for this yard to furnish a certain quantity of machined material to the others. The different series of templates would, of course, have to be supplied to each yard in the order required, and for an ordinary transverse-framed ship this would be somewhat as follows:—Vertical keel, with flat keel and bottom shell, floors and intercostals, tank top and margin plates, upper frames, bulkheads, deck beams and plating, stem and stern frame, hatchways, casings, and so on. The sharing out of the preparation of the templates so as to make them available at the proper moment and in the correct sequence would, of course, require a certain amount of consideration. For instance, the information for marking off the keel and bottom shell could be supplied quickly as it would only consist of marked battens, whereas for the end floors several templates would be required, as also for the margin plates at the ends. The yard supplying the simple marked battens for the keel and bottom shell might perhaps undertake, when these were finished, the preparation of the more elaborate templates for the upper frames at the ends, as by the time these templates were ready some of the upper frames would be already turned and ready for marking off, though it would, of course, be more convenient for the yard marking off the frames to mark off the shell at the ends also. The yard which made the templates for the floors might take over a later job such as the deck plating and beams, which would mainly consist of simple marked battens. Such a scheme of collaboration would naturally involve a certain amount of liaison work to prevent—at any rate at the outset—misunderstandings, but this should not be a difficult matter. Collaboration with regard to the ordering of equipment and the placing of sub-contracts, should assist both the yards and the sub-contractors, and this would also apply to the manufacture of standardized fittings. As regards labour, collaboration in the sense of making the best use of the local labour available by the transfer of redundant labour to other yards, would be of advantage to all parties, but would have to be done through the local labour exchanges. There would remain, however, another very important element to be considered—the human element. Even supposing that the collaborating yards were equally skilled in the fabricating methods involved, it is doubtful whether all of their managements would care to accept the responsibility of preparing templates for others to work to, or to incur the responsibility of marking off thousands of pounds worth of their own materials from templates prepared outside their personal supervision.—*"The Journal of Commerce" (Shipbuilding and Engineering Edition), No. 35,687, 25th June, 1942, p. 1.*

Salvage of Stranded Ships.

An improved method of refloating stranded ships by means of pontoons and hauling tackle is covered by a new British patent and is illustrated in the accompanying diagrams (Fig. 1). A ship (A) is represented aground, and is connected to the nearest pontoon (B) by a number of ropes (E) attached to the bow or stern. The first pontoon (B) is secured to the intermediate pontoon (C) by different tackle falls (F), the end of the cable being led to a winch (G). When this winch operates the tackle serves to draw the pontoons together. The intermediate pontoon (C) is similarly connected to the outer pontoon (D) by tackle falls (H), but the end of the cable, in this case, is taken to a second winch (G_1) on board the stranded vessel. The outer pontoon (D) is anchored. By periodically applying a steady strain to the ropes and then leaving the ship for a while, she will gradually become "alive" and work in the direction of the strain; by taking up and increasing this strain at intervals, it is claimed that the vessel will finally work herself off into deeper water without straining the hull. The hauling action may be spread over a long period. Gear of the kind described may be left in position afloat for a considerable time without loss of efficiency and anchors may be run from the pontoons to assist in holding the tackle in position and check broadside drift. For the purpose of holding the pontoons in their proper relative positions when the strain of the pull is eased, or in the event of any of the hauling tackle becoming broken or disconnected, the pontoons may be connected together by an anchoring chain (M) extending between them. One end of this

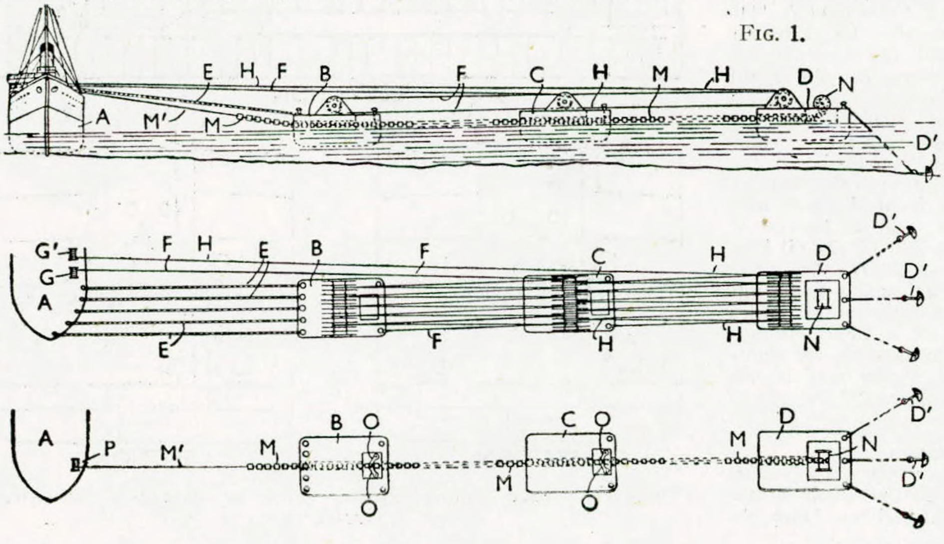


FIG. 1.

chain cable is wound upon a drum (N) on the outer pontoon and then passed through stoppers (O) on each of the other pontoons, the other end being connected by a cable (M_1) with a winch (F) on board the ship.—*"The Motor Ship", Vol. XXIII, No. 270, July, 1942, p. 131.*

Repairing Leaky Economiser Tubes.

An article in a recent issue of the *Electrical World (Vol. 117, No. 22, p. 85)* describes the procedure adopted in the boiler house of an American utility undertaking for carrying out welded repairs to economiser tubes which develop leaks at the header due to the loosening of the tubes by corrosion. Up to about 18in. is cut from the ends of a defective tube and replaced by welding a corresponding single length of tube to one end of the remainder. A specially machined finned casting is then shrunk on to the added length of tube to replace the fins removed from the cut-away lengths. The accompanying diagram (Fig. 2) shows successive steps in the repair. After "burning" the leaky tubes out of the header, the fin castings are knocked off for a distance of about 6in. at one end and 12in. at the other, the damaged ends of the tubes then being cut off and trimmed. The end to which the new length of tube is to be added is scarfed, the extension piece is welded on, and the fin casting is counterbored to a depth of about 3in. to clear the bead of the weld, the remaining 3in. of the casting being bored 0·0015in. smaller than the outside diameter of the tube, on to which it is then shrunk in the usual manner. The cast end fitting of the tube is counter-bored and shrunk on in the same way, a sufficient length of tube being allowed to project at each end to enable the tube to be rolled into the header. This treatment has been applied successfully to about 700 tubes during the past three years, and the effective life of the tubes is thus doubled at a cost of something like \$15 per tube, whereas new tubes would cost about \$50 each..—*"Boiler House Review", Vol. 56, No. 4, August, 1942, pp. 111-112.*

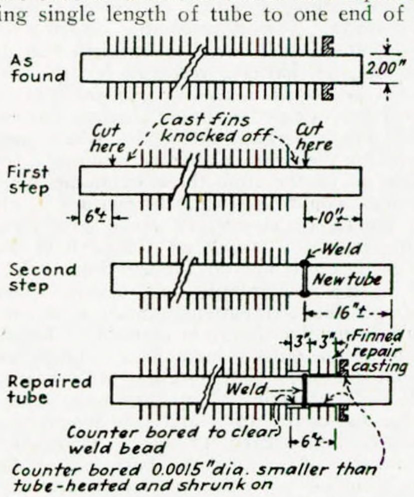
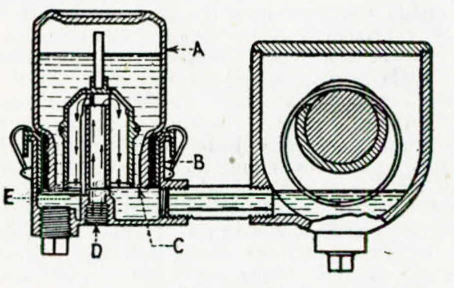


FIG. 2.—*Diagram showing steps in repair of economiser tubes by welding.*

Constant-level Lubricator.

Automatic oil feed as required to maintain a constant level in the bearing is obtained on the barometric principle in the "Oil-Rite" lubricator shown in the accompanying illustration. The glass reservoir *A,* provided with a metal collar fixed by cement impervious to heat, water, acid and oil, is filled with oil and inverted on its base *B.* Oil runs out until the level in the well reaches the mouth *C* of the reservoir and the edge *E* of the dome. A liquid seal is then effected at this point and the flow of oil from *A* ceases until the seal is broken by the fall of the oil level in the bearing. When this happens, air enters at *D,* through a filter if desired, and following the path of the arrows, it flows into the reservoir *A* from which a corresponding volume of oil passes down into the well to restore the level. Felt wick on the dome edge serves as a stopper when refilling. "Oil-Rite" lubricators are manufactured by a firm in Milwaukee, Wis.—*"The Power and Works Engineer", Vol. XXXVII, No. 434, August, 1942, p. 227.*



Harland and Wolff Pressure-charged Power Station.

The accompanying diagram (Fig. 4) shows a cross-section through a Diesel-engined power station, in which the whole of the machinery (B) is housed under air pressure, this scheme being the subject of a recent British patent granted to Harland & Wolff, Ltd., Belfast. If the engines are of the two-stroke type, they do not require to be equipped with scavenging air pumps, whilst four-stroke engines can be pressure-charged without blowers. A fan intake (E) with a duct (D) leading to the fan (F) is arranged outside the station. Air supplied in this manner reaches the engine air inlets (L). Where necessary, air filters (P) can be fitted in conjunction with the fans (F) to exclude any grit or dust which may be present in the outside air. All the engine-room entrances and exists are provided with double doors (H) and air locks (G). The air pressure to be maintained in the engine room may be somewhat higher than that usually found in a boiler room equipped with a forced-draught installation.—*"The Oil Engine", Vol. X, No. 111, July, 1942, p. 80.*

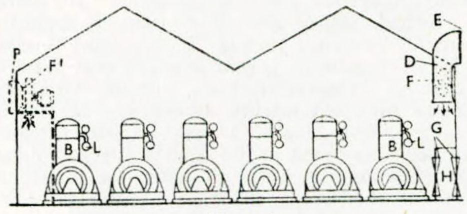
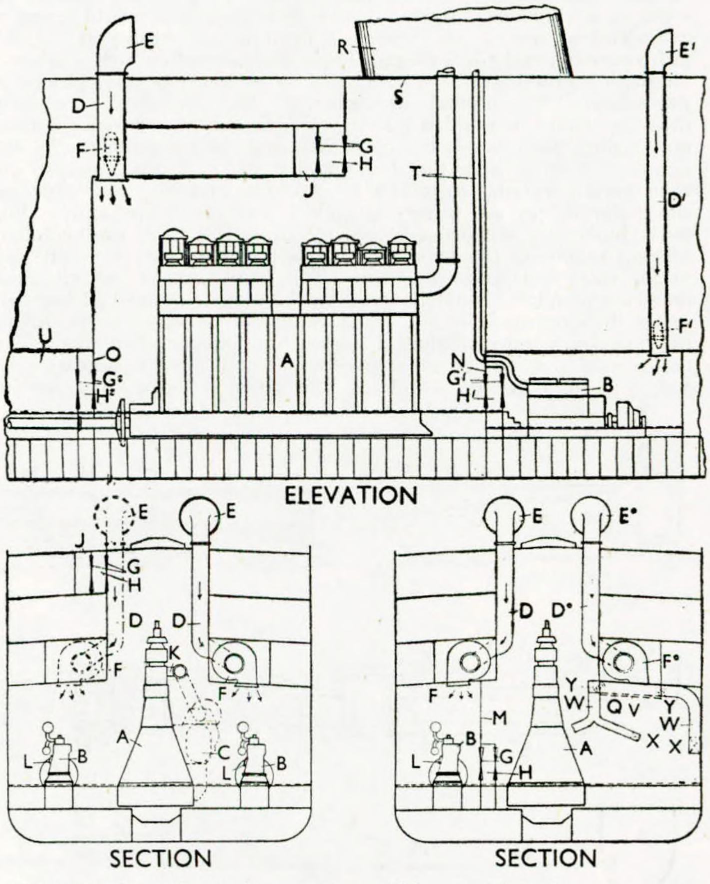


FIG. 4.

A Pressure-charged Engine Room.

A development of the Harland & Wolff patented design for a pressure-charged power station (*see preceding abstract*) is represented by the adaptation of the same principle to motorships' engine rooms. It is claimed that the maintenance of an air pressure in the engine room not only makes it possible, in some circumstances, to dispense with scavenging-air blowers, but also prevents any seepage of lubricating oil at the joints of the crank chambers. The main parts of three relatively large engine rooms designed on this system are lettered in the accompanying illustrations. A compartment housing the main engine (A) is shown, together with a separate auxiliary engine room, in the side elevation, containing the generating sets (B). Referring to the left-hand cross-section, the customary scavenging-air blower (C) is indicated in dotted lines and, instead of such a blower, air supply pipes (D) drawing air from inlets (E)



Pressure-charged engine rooms with double-acting two-stroke machinery.

produced with due regard to the avoidance of this phenomenon. The ship was dry-docked and the propeller was removed to the works of a local firm where it was carefully examined by Professor W. Kerr, of the Royal Technical College, Glasgow, who has carried out a good deal of research work in connection with "singing" propellers. The examination disclosed that although the propeller was in accordance with the design as regards general features, the edges of the blades were not streamlined quite correctly. This was rectified under the supervision of Professor Kerr and the propeller was then refitted to the ship. Subsequent service proved that the "singing" had been cured at all conditions of loading, and a similar correction is now being made to the propellers of other ships of the group which have been reported as having "singing" propellers as well as to other propellers intended for steamers of this class. The British-built vessels of this type have cast-iron propellers, whereas those constructed in the U.S.A. to designs supplied from this country have bronze propellers.—*"The Journal of Commerce" (Shipbuilding and Engineering Edition), No. 35,699, 9th July, 1942, p. 2.*

Adjustable-blade Propellers for Swedish Motor Liner.

A twin-screw passenger and cargo liner of about 7,400 tons d.w. being built by the Götaverken for the South American service of the Swedish Johnson Line is to be equipped with Ka-Me-Wa variable-pitch propellers. They will be made of stainless steel and will each have a diameter of about 15ft., which probably makes them the largest with adjustable blades hitherto ordered. Each propeller will be driven by a Diesel engine of 3,500 b.h.p. The object of fitting Ka-Me-Wa propellers to the new Johnson liner is to test the advantages claimed for this type of screw for large ships. Among these advantages are an increased and more reliable manœuvring capacity due to the control being direct from the bridge without any intermediary and the fact that the speed of the ship can be maintained at any value down to zero. The speed of the engines remains constant while the vessel is manœuvring, and this operation is therefore quieter and smoother because no critical engine speeds have to be passed. Furthermore, engine wear is reduced because reversing of the engines is eliminated and the entire power output can be effectively utilised at any draught and speed.—*"The Journal of Commerce" (Shipbuilding and Engineering Edition), No. 35,711, 23rd July, 1942, p. 7.*

Design of Propeller Shafts.

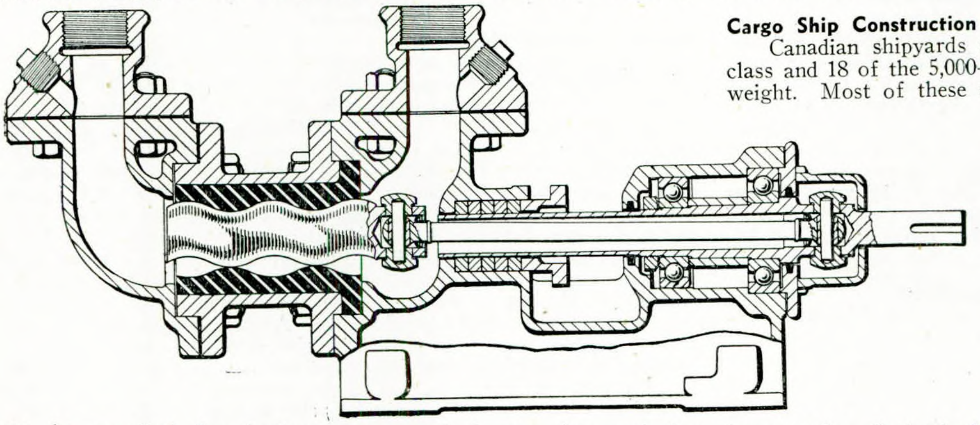
Considerable attention has been attracted to a paper entitled "Factor of Safety and Working Stresses of Marine Propulsion Shafting", by R. Michel, which appeared in a recent issue of the *"Journal of the American Society of Marine Engineers and Naval Architects"*, and in which the author, by evaluating the additional stresses, reduces the factor of safety to the order of 2. He points out that most failures of propeller shafts have been due to fatigue aggravated by the influence of corrosion on the endurance limit of the shaft material, and that the primary failure always started at a point of high localised stress due to a discontinuity or sudden change of cross-section. In addition to the steady shearing stress due to torque and the steady compressive stress due to thrust, there are alternating stresses due to vibration and a variable compressive stress due to the interaction of the propeller and the stern of the ship. It is now possible to evaluate most of these variable stresses with a considerable degree of accuracy. Thus, even with electric drive, geared turbine or Diesel drive, for which the driving torque can be considered constant, the vibratory torque varies from 7½ to 15 per cent. of the mean torque, depending on the shape of the stern. With direct-coupled reciprocating steam or oil engines the shear stress varies considerably, due to the fluctuating driving torque, and dangerous torsional critical speeds may exist within the operating range. Having determined the combined effect of the steady shear stress for full-power conditions and the steady compressive stress due to maximum thrust, the author proceeds to show how the alternating resultant stress can be evaluated in terms of stress concentration factors for bending and for torsion. The points of maximum stress concentration occur at the corners of keyways and at the fillets that join the coupling flanges with the body of the shaft, and charts are given showing the value of these stress concentration factors as a function of the ratio of the fillet radius to the diameter of the shaft. Thus, if the ratio of the diameter of the flange to the diameter of the shaft is 1·6, and if the ratio of the radius of the fillet to the diameter of the shaft is 0·0125, the shear stress in the flange fillet will be 2·4 times the maximum shear stress in the shaft body. Thence, having carefully evaluated the highest stress at these points of stress concentration, it is suggested that a factor of safety of 2 for variable stresses, based upon the endurance limit for line shafting not subject to corrosion, is ample. The possibility of corrosion in the case of the stern-tube shaft calls for a somewhat higher factor of safety, and here a value of 2·25 is used. In addition, it is necessary to check for whirling speed, which should be at least 15 per cent. above the running speed. The author claims that the adoption of this procedure should result in a light yet safe design, provided that care is taken to reduce to a minimum all points of stress concentration.—*"Shipbuilding and Shipping Record", Vol.. LX, No. 4, 23rd July, 1942, p. 76.*

Marine Propulsion Shafting.

Among the contributions to the written discussion of Mr. R. Michel's paper (*see preceding abstract*) is an article by Captain H. C. Dinger, U.S.N.(ret.), which appears in the latest issue of the *"Journal of the American Society of Naval Engineers"*, and in which he remarks that up to recent years the design of propeller shafts was based upon the tensile strength of the material, an allowance of from 4 to 8 factor of safety being made upon this. The main reason for this procedure was that almost the only definite thing known about the material was its tensile strength, the yield point being somewhat variable and not always a definite figure. Since the yield point of steel is approximately half the tensile strength, a so-called factor of safety of 6 on the tensile strength is really a factor of safety of only 3 on the yield point. Among the stresses, mostly variable, other than that caused by the engine torque transmitted through the shaft, the most important and destructive are those due to vibration. When the nature of the vibration and the causes that give rise to it are known, particularly the critical speed, it is possible to make allowances for them by calculation with reasonable accuracy. However, there are various factors tending to cause vibration that may change during operation, among which are: (*1*) misalignment; (*2*) movement of shaft bearings due to deterioration of foundations; (*3*) incorrect setting of propeller blades or damage to same; and (*4*) lack of balance due to eccentric borings, etc. In cargo vessels, the tendency and nature of the vibration may also be affected by the status of loading and the movement of fluids in the tanks in heavy seas. Therefore, although the calculation of the maximum stress due to possible vibration may be based on what is estimated as being the most severe condition, there is still the possibility that, due to some changed condition for which no allowance has been made, the actual stress may build up beyond this calculation. Accordingly, it is necessary to have some margin, even though all known possibilities of stress have been allowed for. The endurance limit for fatigue and corrosion fatigue of most shafting materials has now been determined as being approximately one-third to one-half of the ultimate strength, so that a factor of safety of 3 on the tensile strength would allow only for the normal torque and for fatigue. In order to be quite safe it would also be necessary to allow for maximum stress due to vibration, and to make certain that the design and construction of the shaft and its appendages did not produce any stress concentration beyond this. Stress concentrations arise mainly at the sharp corners of keyways, crank webs, coupling flanges, etc., and if these are carefully filleted the stress concentration can be removed. However, stress concentrations are also caused by hidden locked-up stresses set up in casting, forging or welding operations and not properly stress-relieved, as well as by hidden defects, small blow-holes, metallic inclusions, etc. In sound, carefully processed material such hidden defects are rare, but there is never any definite certainty about that and some allowance should be made on this account. Reverting to the factor of safety of 6, reduced to 3 by reason of yield point, endurance limit and expected vibration, if some allowance is made for imperfections in the material and inaccuracies in the machining of parts, it is clear that a factor of safety of 3 is actually reduced to 2 without any appreciable margin. With reference to Mr. Michel's remarks concerning the need for a higher factor of safety for shafting operating in sea water, Capt. Dinger points out that it has been the practice, for almost a century, to allow for a double bending moment on all outboard shafting. Originally this was a guess to allow for difficulties and deterioration caused by the fact that the shaft had to operate in water instead of air, but modern research concerning the effect of salt water on the fatigue limit of such a shaft and recent corrosion fatigue discoveries prove that this somewhat generous allowance was fully justifiable. Until recent times the danger of sharp corners was not properly sensed, although they were probably more responsible for casualties in stressed machinery parts than any other particular point of bad design.—*"Shipbuilding and Shipping Record", Vol. LX, No. 4, 23rd July, 1942, p. 81.*

Mono Self-priming Rotary Pump.

It is reported that the Mono pump, an ingenious invention in which rotary motion produces an effect equivalent to that of a piston moving continuously and with uniform velocity in a cylinder of infinite length, and which was first introduced about four years ago, has now been adopted by the Admiralty, War Office, and other Government departments and public authorities, after standing the test of everyday use in a wide variety of industries all over the world.. Standard models are now available in capacities up to 150 g.p.m. (40 tons/hr.) for delivery heads up to 400ft., and smaller units are being produced for pressures up to 300lb./in.^{2}. The pump



can be supplied for horizontal or vertical operation and for stationary or portable use, and simple modifications of the standard pump allow it to be driven direct by any kind of prime-mover, or by flat or Vee-belt. A standard range is also made with direct-coupled compressed-air motors. As the pump is valveless and able to operate with suction lifts of 25ft. or more as a self-priming appliance, even if air enters the suction, it is claimed to be particularly useful for pumping out tanks containing either water or other fluids of heavy gravities. The straight-through nature of the flow makes it suitable for dealing with viscous fluids; for corrosive liquids the rotor casing may be constructed of stainless steel, Monel metal or other special material. The stator is normally made of rubber, not because it deforms appreciably, but in order to secure the closest contact with the rotor and to take advantage of the remarkable wear-resisting properties of this material. The stator has an internal double-start thread and the rotor is in the form of a helical scroll having half the pitch of the stator. It is driven through a universal connecting rod, permitting angular flexibility, and, guided by the stator, it performs a hypocycloidal motion which results in a continuous and positive displacement of fluid through the stator. A uniformly high performance is claimed over a wide range of capacities and pressures. As the line of contact between the rotor and the stator is constantly changing its position on both elements, there is little tendency for particles of grit to become embedded and the pump is therefore able to handle abrasive liquids. Moreover, the water space through the stator is constant in size and shape, so that small solids can be handled with ease. The accompanying sectional view shows one of the latest standard designs, incorporating several mechanical improvements.—*"The Power and Works Engineer", Vol. XXXVII, No. 434, August, 1942, pp. 225-226.*

Fast Cargo Ships Advocated.

In a recent letter to the *"Daily Telegraph"* Professor T. B. Abell, of the University of Liverpool, urged the construction of fast cargo ships in order to increase delivery of supplies to this country and to decrease the number of casualties, whether from air attack or submarines, by shortening the duration of voyages. He enumerated the advantages to be derived from the employment of higher speeds at sea, but admitted that it would involve greater fuel consumptions and more powerful and elaborate propelling machinery. He also recognised the fact that the fast vessel would have to be larger than the slower one, and that the time of construction would probably be increased in the ratio of the larger amount of cargo carried. There is, however, another aspect of the matter to be considered; facilities for engining the smaller number of fast ships would probably be hard to find under present conditions, whilst the question of finding suitable engine-room personnel to man them might present a serious difficulty. The majority of sea-going engineers are not experienced in the running of high-speed Diesel engines or geared turbines, and the complexities of such machinery would impose an additional strain on the already fully-occupied E.R. staffs of the Merchant Navy. Because of this, the same degree of reliability as is now maintained could hardly be expected, whereas under the conditions called for by the convoy system, the need for reliability becomes all the more imperative. It is well known that the Admiralty have arranged for the construction of relatively high-speed cargo ships wherever facilities for building them are available, although the bulk of the new tonnage will perforce have to be of the slower type. Nevertheless, it should be remembered that the term "slow" is merely relative and means something very different to what it represented a generation ago.—*"Fairplay", Vol. CLIX, No. 3,090, 30th July, 1942, p. 154.*

Cargo Ship Construction in Canada.

Canadian shipyards are building 154 vessels of the 10,000-ton class and 18 of the 5,000-ton class, or a total of 1,630,000 tons deadweight. Most of these ships will be completed before the end of 1942, no fewer than 20 of the larger type having already been delivered before the end of last June. These ships are almost identical in design to the E.C.-2 or Liberty ships of the U.S. Maritime Commission, except for their boilers. The original design—to which the ships can be readily converted after the war—called for a 25½ft. draught, 6,800 gross tons and 9,300 tons d.w., but the tonnage opening has been temporarily closed and the main bulkheads extended to the upper deck, thereby not only making the ship more seaworthy and safe under wartime conditions, but also increasing the cargo-carrying capacity. Instead of the two oil-fired watertube boilers installed in the Maritime Commission's vessels, the Canadian-built ships have three coal-burning cylindrical boilers, each 14ft. 9in. in diameter and 11ft. 9in. long, arranged athwartship forward of the engine room and separated from the latter by a spectacle bulkhead located about 14in. from the rear end of the boilers, and fitted snugly around the latter. A cross bunker of 505 tons' capacity extends right across the ship forward of the stokehold, and if necessary, the fuel capacity of the vessel can be increased by 1,000 tons by utilising No. 3 hold as a coal bunker. Access to this space from the boiler room is provided by a W.T. tunnel running under the cross bunker. Quick-closing watertight doors are fitted to the W.T. bulkheads, and escape hatches at four separate locations in the machinery spaces provide quick access to the upper deck for the personnel in the engine and boiler rooms. The cargo-handling equipment comprises ten 5-ton derricks operated by ten steam winches. Another steam winch (of the double drum type) is fitted aft for warping and auxiliary steering purposes, whilst a steam windlass is installed forward for working the anchor cables. Two refrigerated provision rooms—of 250 and 350 cu. ft. capacity, respectively—are located amidships on the main deck. Cooling is effected by a Freon commercial type machine. The ship carries a complement of 47 officers and men, in addition to a party of 10 naval ratings or soldiers for defence purposes. The E.R. personnel consists of four engineers, a donkeyman, twelve firemen and three greasers. The vessel's armament includes naval guns, Oerlikon A.A. guns and machine guns. Armour plating is provided for protecting the vital parts of the ship. The 5,000-ton vessels differ from the larger ships in having a forecastle deck and only four cargo holds. These are specially adapted for the carriage of timber, steel shapes and similar bulky cargoes. No 'tween decks are fitted. The four large cargo hatches are served by six 5-ton derricks and six steam winches, the remaining deck machinery comprising a steam windlass on the forecastle deck and a warping winch on the poop deck. The propelling machinery consists of a triple-expansion engine with cylinders of 20in., 55in. and 31in. dia. and 39in. stroke, the L.P. cylinder being in the middle. This arrangement gives a better balance to the engine and allows the poppet valves of the H.P. and M.P. cylinders to be kept relatively cool, in addition to which they are very accessible. A balanced slide valve is used on the L.P. cylinder. Steam at a pressure of 200lb./in.^{2} is supplied by two coal-fired Scotch boilers, 13ft. 6in. in diameter and 11ft. 6in. long, located forward of the main engine and flanked by wing bunkers, the stokehold being at the forward end of the boilers. The auxiliary machinery includes a forced-draught fan for the boilers, a 12-ton evaporator, a 10in.×11in.×10in. ballast pump, an 8in.×6in.×15in. G.S. pump, an auxiliary condenser of 400ft.^{2} capacity, a two-stage feed-water heater, two 12½-kW. 120-volt dynamos, a 4in.×5in. ash hoist and a 9-in. circulating pump.. The vessel carries a crew of 30 officers and men, including three engineers, a donkeyman and six firemen, in addition to eight defence personnel for the ship's guns.—*A. C. MacNeish, "Canadian Shipping and Marine Engineering News", Vol. 13, No. 12, July, 1942, pp. 23-38 and 56-58.*

A Pump for Varied Cargoes in Oil Tankers.

An improved design of cargo oil pump for use in tankers has been developed by an engineering firm in Boston, Mass., and forms the subject of a recently published British patent. The construction of the pump, which is of the rotary displacement type, is shown in Fig. 2. Where fluids of widely different characters have to be handled, cargo oil pumps frequently need repairs, and these must be carried out with a minimum of delay. Among the advantages claimed for the new pump to facilitate maintenance are the special arrangements made regarding the packing and other features. Referring to the diagrams, the bed (*1*) is formed with an inlet (*21*) and the rotor casing (*30*) has a discharge connection (*40*). A cover plate (*33*) gives access to the rotors and these are withdrawn without disturbing the packing or shaft bearings. The rotors (*50*) are diametrically undercut (*52*) to a size equal to the diameter of the hub and the stepped portions (*35*) form a joint. The bosses (*34, 36*) are prevented from acting as supports for the overhanging ends of the rotors. The projected area of each boss, however, subtracted from the total projected area of the rotors, results in a reduction

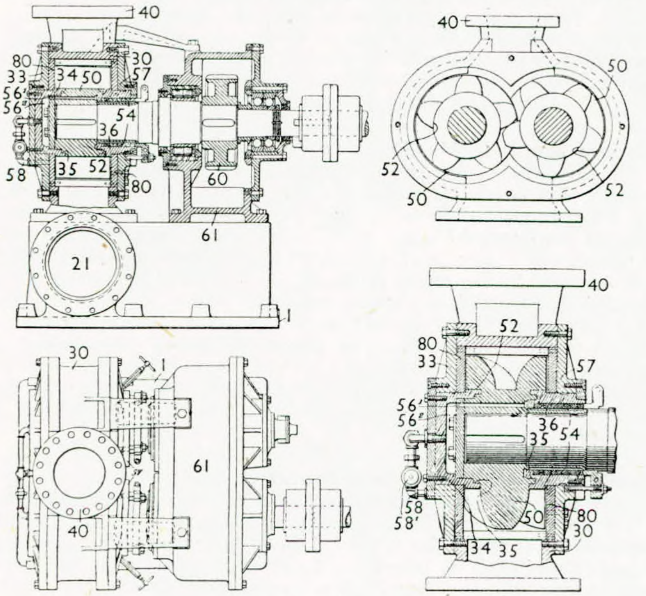


FIG. 2.

of the latter, reducing the amount of shaft deflection and load to be supported by the bearings. Further, the bosses enable the casing to take up internally developed pressures without stress developing through the rotor on its bearings. In order to allow for the wear set up when pumping volatile fluids or water ballast, vents (56^1) lead from the boss (*36*) forming the stuffing-box for the packing (*54*). The chamber (56^2) has a pipe (*58*) connected to the inlet side whereby the pressure on the packing may be regulated by means of a valve (58^1). This makes it unnecessary to tighten the packing hard and thus reduces the wear. Pinions (*60*) take the drive between the two rotor shafts, which are relieved of transmission duty on the blades. Liners (*80*) are slipped over the bosses (*34, 36*) through apertures and are held in place by the bolts which fasten the end plates (*33, 57*) to the casing (*30*). The gearbox (*61*) forms an independent unit.—*"The Motor Ship", Vol. XXIII, No. 271, August, 1942, p. 167.*

Monel Metal-lined Trawlers' Holds.

Four new trawlers recently added to the fleet of the General Seafoods Corporation, a well-known fishery undertaking operating on the eastern seaboard of America, have wood-sheathed fish holds with a lining of 20-gauge Monel sheet instead of the nickel-clad steel plate used in earlier craft. Both nickel-clad steel and Monel possess the same hygienic and corrosion-resistant qualities, and eliminate the danger of contaminating the fish by rust and other by-products of corrosion. The smooth, hard surfaces of these lining materials are easily kept clean and are stated to be free from the cracks and crevices where bacteria might collect and breed. Prior to the introduction of these sanitary holds, it was frequently found that spoilage due to contamination by bacteria-infested wooden holds, warm weather and other conditions, was responsible for catch losses running up to 20 per cent. or more. Lower ice requirements due to the introduction of metal-lined holds have also made it possible to increase the fish-carrying capacity of the newer trawlers by upwards of 30 per cent.—*"Modern Refrigeration", Vol. XIV, No. 532, July, 1942, p. 129.*

Propeller-shaft-driven Generators.

The generating installation of the average motor cargo liner of, say, 9,000 tons d.w., usually comprises three Diesel-engined dynamos, each of about 100 kW. At sea, the electrical load is moderate, and in some vessels, more particularly tankers, shaft-driven generators have been installed in order to avoid running a separate Diesel engine. An improved design for an installation of this kind, operating in conjunction with a Diesel-engined generator, has been developed by the makers of the A.S.E.A. electric slip couplings. The arrangement is shown in Fig. 1, and utilises an electric slip coupling which is interposed between the shaft and the generator. The latter only supplies current to the ship's mains when the propeller shaft is running at approximately full speed, and in order to reduce the cost, size and weight of the shaft-driven dynamo, the drive is taken through speed-increasing gearing. Referring to Fig. 1, the propeller shaft (*1*) runs in the thrust bearing (*2*) and has mounted on it a gear wheel which, together with its associated pinions, is enclosed in a gear case (*3*). The generator drive is taken through the electro-magnetic slip coupling (*4*) to the dynamo (*5*), which is provided with an exciter (*6*). The electrical circuit is shown diagrammatically in Fig. 2. The main Diesel engine (*1*) is coupled to the propeller shaft, on which is mounted the generator gearing (*2*). The shaft-driven dynamo (*4*) is driven by the electro-magnetic slip coupling (*3*), whilst an auxiliary generator (*5*) is driven by a Diesel engine (*6*). The circuit breakers for the slip coupling (*3*), shaft-driven generator (*4*) and auxiliary generator (*5*) are shown at (*7*), (*8*) and (*9*) respectively. A battery (*10*) is also provided for emergency use, and may be charged from the ship's mains. The emergency lighting circuit from this battery is shown at (*13*) and the battery-charging connection at (*14*). When the ship is in port and also whilst manœuvring, current is obtained from the auxiliary generator (*5*), the circuit breaker (*9*) being closed and the breakers (*7*) and (*8*) opened. When the ship's speed increases and the propeller shaft is running at normal revolutions, the circuit breaker (*7*) is closed and the slip coupling (*3*) becomes magnetised. The voltage of the shaft-driven dynamo (*4*) then rises until it reaches that of the ship's mains (*12*), whereupon the circuit breaker (*8*) is closed and the load is transferred from the auxiliary generator (*5*) to the shaft-driven generator (*4*) through field-resistance or other regulation. As soon as the dynamo (*4*) is taking the required load, the Diesel-driven set (*5*) is shut down. The current generated by this system is obtained at a lower fuel cost than that supplied by a Diesel-driven dynamo. Even if the load represented the full output of the Diesel-driven generator there would be a saving of at least 10 per cent. in the fuel consumption, since the propelling engine has a consumption of, say, 0·35lb./b.h.p.-hr., as compared with the 0·38 to 0·39lb. of the average high-speed Diesel generating set. If, however, the load is only half the designed output of the Diesel generator, the saving in fuel would be further increased to something between 15 and 20 per cent. It is not possible, however, to save the cost of one Diesel-engined generator in this way, since it is essential to have sufficient generating plant available for the operation of cargo winches or

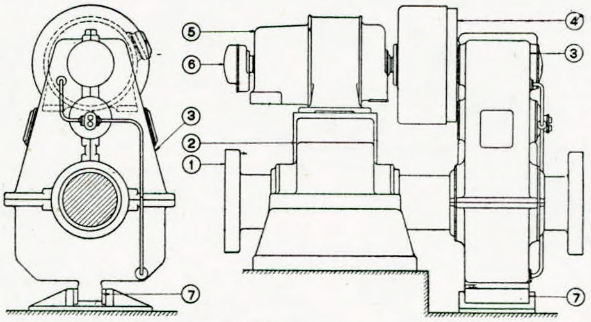


FIG. 1.—*Propeller-shaft-driven generator with electric slip coupling.*

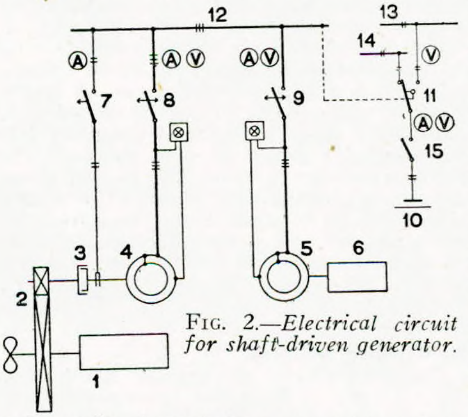


FIG. 2.—*Electrical circuit for shaft-driven generator.*

pumps when the ship is in port and the shaft-driven dynamo out of use.—*"The Motor Ship", Vol. XXIII, No. 271, August, 1942, p. 163.*

Ventilating Ships when Blacked Out.

Fittings for ships' scuttles and ports which enable ventilation to take place under black-out conditions form the subject of a recently published British patent, and are illustrated in Fig. 3. The rings *(6)* in the ships' side *(5)* are fitted with a casing *(7)* having a flange *(8)*. The cone *(9)* has an extension *(10)*, and a cover *(11)* is held in place by brackets *(12)*. An internal baffle *(13)* is spaced clear of the cone by brackets *(14)*. The plate *(15)* forms a wind vane and handles *(16)* are provided for fixing and removing the device. The ship is assumed to be proceeding in the direction of the arrow *(17)*. In the lower left-hand illustration the circular fitting is replaced by a rectangular casing *(17a)*, whilst an alternative arrangement includes a series of springs *(18)* stamped from the metal of the annulus *(7)*. These springs allow a certain amount of variation in the diameter of the port. The lower right-hand diagram shows a hollow frustum *(19)* attached to a mask *(13)* that closes it at the top. It is claimed that the use of this device enables the free admission of air to continue irrespective of changes in wind direction or alteration of the ship's course.—*"The Motor Ship", Vol. XXIII, No. 271, August, 1942, p. 167.*

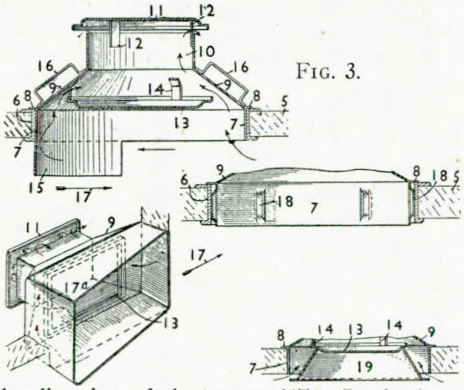


FIG. 3.

Aluminium Foil for Marine Insulation.

The makers of "Ardor" aluminium foil for the insulation of refrigerated holds and cold storage spaces have recently issued an illustrated brochure in which typical methods of the application of this material are described and shown by means of line drawings. The insulating medium in question is a patented form of bright aluminium foil of high purity and consists of two sheets of foil, one of which is corrugated to a depth of ½in. and the other left plain; the two sheets are secured together by wire staples. The material is available in standard rolls of 50ft.×2ft. by ½in. thick, with or without 1in. extra sidelap on the plain sheet. The weight of the complete roll being 3lb. An alternative size measures 6in. wider and weighs ¾lb. more per roll. The thermal conductivity of this insulating material is stated to be 0·21 B.Th.U./ft.^{2}-hr. for 1in. thickness and 1° F. temperature difference.—*"Modern Refrigeration", Vol. XIV, No. 532, July, 1942, p. 129.*

Metallic Copper as an Anti-fouling Medium.

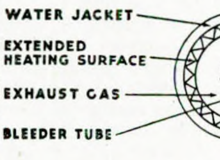
Paint surfaces for the protection of underwater parts of a ship's hull have to satisfy a variety of important requirements, and the diversity of these renders the problem a difficult one. The primer and top coat must fulfil their functions as well as be mutually compatible in forming a coherent film. Research work recently carried out in America is reported to have shown that the most promising results are obtained with paints formulated with pure electrolytic flake copper in one of the class of varnish vehicles with short oil length, containing coumarone-indene and phenolic resins. The anti-fouling compositions used in some of the tests contained a copper bronze pigment incorporated in paste form, the paste being made up of 78 to 80 per cent. of electrolytically-refined oxygen-free copper coated with a polishing agent of stearic acid and oily matter, and 10 to 12 per cent. of high-solvency petroleum spirit. Each composition tested contained 3lb. of this pigment paste per gallon of paint. A quick-drying red lead paint was used as a primer. A large-scale test of an anti-fouling composition of this description is now in progress, the 3,000-ton cargo steamer "Lake Traverse" being used for the purpose. The hull of this vessel, which is 250ft.×40ft., was sand-blasted to remove all the old paint and rust. The surface was very clean after this treatment, but was marred by numerous pits approximately $\frac{1}{16}$-in. deep. Three coats of the red lead alkyd primer were sprayed on, each coat being allowed to dry overnight. Two coats of metallic copper anti-fouling paint were then applied, the total area covered being about 20,000ft.^{2}. The amounts used were 60 gall. of red lead for the first coat, 50 for the second, and 45 for the third, 52 gall. of copper paint being required for each of the two upper coats. The latter leafed perfectly and gave the appearance of a brilliant sheet of metallic copper. After five months' general cargo service between various ports in North and South America and the West Indies, the ship had to be dry-docked to make good some minor damage sustained in a collision, and it was then found that the hull was in perfect condition, with no sign of fouling of any kind. There was a complete absence of film failure, except for the scraped portions, which were retouched by spot priming with the same red lead primer as was used initially. The entire hull was given a single coat of copper anti-fouling paint, and the ship was then put into charter service in Carribean waters. When a report on the test was made six months later, the vessel was still in service without any increase in fuel consumption for the maintenance of her usual speed.—*"Metallurgia", Vol. XXVI, No. 154, July, 1942, p. 115.*

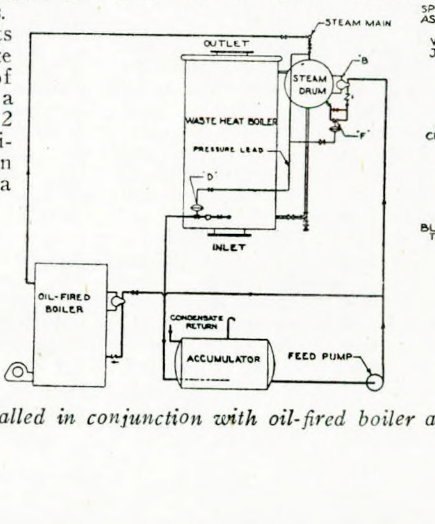
Shipyards and Scrap Metal.

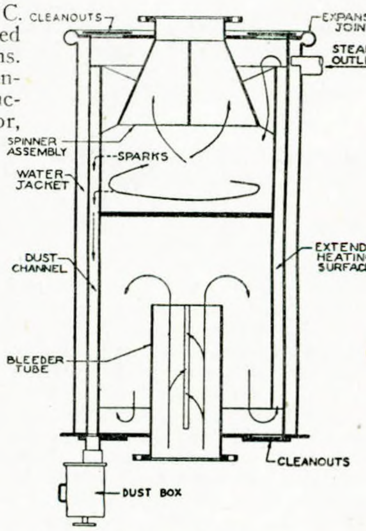
A Ministry of Works inspection of N.E. Coast shipyards has revealed that nearly 200 additional tons of scrap metal per week are now being recovered from 25 shipyards, compared with the amounts collected four months ago. The Tyneside and Tees shipyards alone are saving the nation an extra 10,000 tons of iron a year. This has been achieved by the co-operation of all concerned, and many of the shipyards have appointed special recovery officers and squads whose job it is to find the scrap and keep it moving in a regular steady flow to the furnaces. Most of the scrap comes from repair and conversion work—old plates, old boilers, bars, machinery and gear, as well as many tons of smaller stuff, such as cuttings, punchings and broken rivets.—*"The Engineer", Vol. CLXXIV, No. 4,516, 31st July, 1942, p. 104.*

Something New in Waste-heat Recovery.

The heat input to an exhaust-gas boiler, and therefore the steam or hot water delivered by it, is not controlled by a heating or steam load but by a power loading, and in order to regulate the output of such a boiler it is necessary to provide some form of by-pass arrangement whereby any required proportion of the total weight of exhaust gases in a given period is passed through the boiler, the remainder going direct to the silencer. Where automatic control gear in a waste-heat recovery system is employed, the by-pass device (usually some form of gate valve) requires a fair amount of power to move it; and should the valve ever become jammed by heavy carbon deposits, the working of the automatic control gear is rendered ineffective. In order to overcome this difficulty, the Maxim Silencer Company have developed a method of controlling the output of an exhaust-gas boiler which not only dispenses with the usual by-pass arrangement, but likewise eliminates the need for a secondary silencer, thereby saving weight and expense. An installation of this kind fitted in the twin-screw motor tanker "William C. McTarnahan", is illustrated by the accompanying diagrams. The Maxim waste-heat silencer-boiler, which also functions as a spark arrestor,







Maxim silencer installed in conjunction with oil-fired boiler and steam accumulator.

takes the exhaust from the four aftermost cylinders of the port main engine, which has an output of 1,700 b.h.p. and is pressure-charged by the Büchi system. The construction of the silencer-boiler is shown in the right-hand sectional view, whilst the small left-hand cross-section shows how the extended surface of the silencer element serves to heat the water circulating through the annular space formed by the inner and outer shells of the boiler. The lay-out of the automatic control system is shown in the circuit diagram, from which it may be seen that the installation comprises a waste-heat boiler with an external steam drum as well as a small oil-fired boiler and a steam accumulator. If steam in excess of the demand at a particular moment is being generated by the waste-heat boiler, the pressure-regulated valve (*F*) in the feed line closes at a predetermined pressure (16lb./in.^{2} in this particular case) and shuts off the water supply to the boiler. If the steam demand is reduced, pressure will continue to build up in the boiler, and when it reaches 18lb./in.^{2}, the reverse-acting pressure-regulated valve (*D*) at the bottom of the water space of the boiler, opens and allows water from the boiler to flow into the steam accumulator, thereby uncovering part of the extended heating surface in the boiler and decreasing the generation of steam to meet the reduced demand between the pressure setting of the two valves referred to. The feed-water regulator (*B*) serves as a high-limit control in the system, which it will be noticed, incorporates a feed pump between the accumulator and feed-water regulator (*B*). The steam from the waste-heat boiler in the "William C. McTarnaham" is used for the ship's heating system, and the vessel operates on a service between a northern port where, during winter months. the heating load is high, and a southern port where it is practically nil. With normal heating loads, the pressure in the system varies from 15 to 20lb./in.^{2}. Where there is no heating load on the circuit, the main valve from the boiler is closed and the pressure hunts over a range from 15 to 25lb./in.^{2} due to the fact that when the small amount of feed water required to maintain steam is admitted to the boiler by the valve (*F*), there is a slight drop in pressure due to the low temperature of the water. After a while it builds again as steam passes through the pressure valve (*D*) at a point where the water level is slightly below this connection, *i.e.*, the pressure-regulated valve (*D*) acts as a safety valve and allows a small amount of steam to be discharged into the accumulator, where it is condensed during its passage through the submerged heater indicated in the diagram by a dotted line near the bottom of the accumulator. It has been found in practice that this waste-heat boiler has an output of 740,000 B.Th.U./hr. at a pressure of 15lb./in.^{2} when the engine is developing about 800 b.h.p. in its four after cylinders. This is claimed to be an excellent performance, as it approximates to 1lb. of steam per b.h.p./hr., the silencing qualities of the boiler being extremely good. It has also proved to be an efficient spark arrestor.—*W. R. Williamson in "Diesel Progress", of April, 1942, summarised in "The Marine Engineer", Vol. 65, No. 781, August, 1942, pp. 168-169.*

Batteryless Submarines?

It has been reported that the Germans have perfected a submarine which needs no electrical accumulators for underwater propulsion, but can cruise on a mixture of hydrogen and oxygen, generated by electrolysis. Whether this is true or not, there is nothing impossible in the idea, and the technical aspects of this form of drive are worth considering. Very little information about the supposed new type of machinery is available, but if the explosive gas mentioned in the reports is actually hydrogen and oxygen. which does not seem unlikely, facilities for producing it on board must presumably be provided in the shape of an electrolysing plant of some kind supplied with current from a generator driven by the main engines, to be operated when on the surface, just as in the case of battery charging. The carriage of a large quantity of gas compressed to the high pressure needed for storage in a reasonable space would scarcely be an economical proposition, owing to the weight of the containers which would be required for such a purpose, but some reserve of gases may well be kept for emergency use. Electrolysis of sea water would be impracticable on account of the large quantities of salts in solution, producing as by-products of the electrolytic action quantities of acid and alkali in the tanks—in fact, this is a standard commercial method of manufacturing caustic soda. Fresh water would therefore probably have to be used, with a small quantity of suitable electrolyte added; only a limited amount of fresh water can, of course, be carried. but this can be made up from condensation of the engine exhaust and perhaps other expedients, such as recovery of atmospheric moisture and evaporated vapour, *i.e.*, low-temperature distillation. For a cruise of four months' duration a very small daily rate of recovery should be sufficient. Compression of the oxygen and hydrogen might be effected simply by carrying out the actual electrolysis under high pressure, rather than by a separate compressor, and this may prove to be quite a simple and economical process as regards power consumed. Whether the gases are utilised in the main engines, with or without modification, or how this modification is carried out with the requisite instantaneity, is not clear, but it seems fairly certain that if they can only be burnt in a special engine for underwater use, most of the advantage of the novel scheme would vanish. The advantage of substituting one system of energy storage for another must lie either in reduced weight or in increased overall efficiency embracing generating, storage losses and reconversion, or in a combination of both, apart from any incidental gains of a subsidiary character. So far as efficiency is concerned, generating and storage losses should be quite low, but the efficiency of the heat engine cycle in reconverting to mechanical power will tend to reduce the overall figure, even assuming a hydrogen consumption equal to or less than the specific fuel consumption of any known prime mover. Nevertheless, the overall result may not be unfavourable when compared with that of the Diesel generator-battery-motor system currently in general use. The burning of hydrogen in a closed chamber such as a submerged submarine is made possible by the fact that the product of combustion is simply water vapour. A slight incompleteness of burning, and a release of small quantities of oxygen and hydrogen inside the hull would not be fatal, as these gases are not harmful, though of course, highly explosive under suitable conditions; in any case, precautions can be taken. A more serious problem may be fumes of partly burnt lubricating oil which finds its way into the cylinders. If the change-over from surface to submerged propulsion is in fact simply a change of fuel, from oil to hydrogen, there is still the problem of reconciling engine speeds, mean effective pressures and horse-powers in the two conditions with the attainable water speed as influencing the working of the propellers. Variable-pitch screws may have been introduced and may even have been combined with a variable reduction-gear ratio from engine to screw. Some reduction gearing, implying light high-speed prime movers, seems a fairly safe guess, and for that matter an installation of this character without other changes of a more radical nature, such as have just been discovered, might alone be enough to account for the enhanced performance of which the latest enemy submarines appear and are believed to be capable.—*J. Lockwood Taylor, D.Sc., "Shipbuilding and Shipping Record", Vol. LX, No. 4, 23rd July, 1942, p. 79.*

Condensation in Exhaust Pipes.

All Diesel fuels and most fuel gases contain hydrogen which, during combustion, unites with the oxygen in the air to form water. Due to the heat of combustion this water is converted into steam and passes into the exhaust system. Where the exhaust pipe is short the gas leaves it at a high temperature and the steam escapes uncondensed, but if a long exhaust pipe is necessary it is safer to include a moisture trap or arrange for the pipe to slope away from the engine. Some unusual instances of damage due to condensation have recently come to the writer's notice and may prove of interest to all engine users. Three gas engines were connected to one common exhaust main, on which a heat exchanger was fitted. Under normal running conditions only one or two engines were running at the same time. Each engine was fitted with an isolating valve of the butterfly pattern which was kept closed when the engine was not running. After the plant had been in operation for some little time, it was reported that the water-cooler exhaust manifold on one engine was leaking internally and that water was entering the engine through the valve. When the engine was opened up for cleaning, the exhaust manifold was examined and tested for leaks. The engine proved to be badly rusted, but no leaks were found in the manifold. Further investigation showed that the butterfly valve was not gastight and that the exhaust gases from the two running engines were leaking into the standing cold engine and condensing. It is very difficult to make and maintain a butterfly valve gastight, especially when operating at a temperature up to 1,000° F., and in this instance the pulsating exhaust gases from the other engines had caused the valve to hammer on its seatings, thereby aggravating the trouble. The butterfly valves were discarded in favour of simple gastight sliding dampers, and no more trouble has been experienced with this plant. In another case the exhaust pipe from a small auxiliary engine was led into the main engine exhaust pipe. A plug cock was fitted to isolate the small engine when it was idle, but, after a short time, this plug cock became tight due to heat and, as a consequence, was always left open. Exhaust gas from the main engine condensed in the small engine and complaints were constantly received that the cylinder head must be leaking because the cylinder filled with water and the engine could not be started. A more suitable cock was then fitted and a moisture trap inserted in the pipe, after which no further complaints were made. The main engine should also be protected in this manner if the auxiliary engine is likely to be running for long periods by itself. An unusual case recently arose in a marine installation, which, while not caused by the engine exhaust condensate, is worthy of inclusion as a warning against an easily overlooked pitfall. A Diesel engine was installed on board a steamship for auxiliary service, and after erection the engine stood for some time before being run. When an attempt was made to start it up, serious damage was incurred due to the presence of water in some of the cylinders. No leaks were found in the engine itself and the water was traced as coming from the exhaust pipe. This pipe was led to atmosphere alongside the ship's funnel and terminated just below the main boiler blow-off pipe. Every time the boilers were blown off, the Diesel exhaust pipe received its full share of the condensed vapour. A moisture trap had been fitted in the exhaust system but was not of sufficient capacity to take this unexpected flow. A bend on the exhaust pipe outlet solved the trouble. This particular accident could have been avoided if the Diesel engine had been barred round by hand before the attempt to start it up was made.—*F. D. Langley, "Gas and Oil Power", Vol. XXXVII, No. 442, July, 1942, pp. 113 and 120.*

Büchi Turbo-blower Assembly.

A recently patented method of assembling the combined rotor unit of a Büchi turbo-blower is illustrated in Fig. 2, in which the complete unit is also shown by two small diagrams. The two rotors of the blower are inter-connected by a sleeve which is free to expand under the influence of heat and centifugal force. The right-hand diagram shows the central part enlarged and fully sectioned. The turbine rotor *(1)* is connected to a sleeve *(2)* and has a central stud *(3)*. The blower rotor *(4)* has a hub *(5)* which is screwed into the sleeve and the joint tends to tighten in operation. A hole in the mounting shaft *(10)* is provided to take a pipe *(14)* which supplies cooling fluid to a space *(13)* in the stud *(3)*, the discharge being led outside the pipe down the annulus *(14')*. Although the turbine outlet casing *(16, 17)* is in two sections, the inlet casing *(19)* is integral and supports a rim *(20)*. This rim carries the inlet nozzles for the rotor. Cooling spaces *(21, 22)* are provided in the outlet casing. In order that the uncooled inlet casing shall remain concentric with the other at all times, keys *(23, 24)* are arranged in crossed relation to the three parts *(16, 17, 19)*. Passages *(28)* through the sleeve *(2)* lead to the interior so that air by-passed from the blower rotor *(4)* can be conveyed to the bearings *(29)*; the air also cools the sleeve *(2)* which is provided with fins *(33)*. In order to prevent gases from passing into the blower from the turbine, packing *(34)* is provided. Lubricating oil to the bearing *(36)* is supplied through a hole *(37)* and returned through a collecting passage *(38)*. A vented space *(41)* is provided to which air is supplied through a hole *(42)*. Oil or water drops that may have formed can thus escape through an outlet *(43)*.—*"The Oil Engine", Vol. X, No. 112, August, 1942, p. 108.*

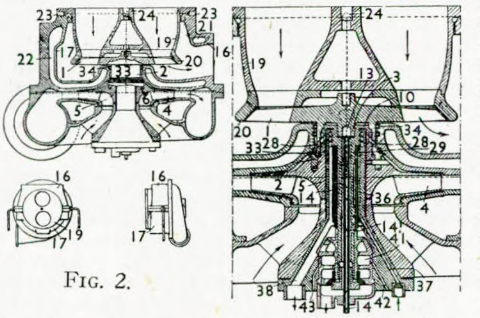


FIG. 2.

Powder Device for Starting Heavy Oil Engines.

A Swedish firm manufacturing I.C. engines for fishing vessels, etc., the Bolinder-Munktell A.B., have patented a powder starting device for heavy oil engines, which is not only effective and economical in use, but also enables the usual starting equipment for engines of this type to be dispensed with. The device comprises a steel cylinder from which the pressure produced by the combustion of a special powder charge is admitted to the combustion chamber of the engine cylinder. The cylindrical container, attached to the cylinder head, has a holder for the cartridge at one end. The cartridge is fired by a hammer blow on the ignition pin. The chief advantage claimed for powder starting, which can be used as the main starting device or as an auxiliary to compressed air, is its reliability. Whereas the admission of compressed air to the combustion chamber of an engine lowers the temperature, the combustion of powder generates heat and thereby facilitates starting. The Bolinder-Munktell starting device is stated to be absolutely safe, the quantity of smokeless powder in the cartridge being only six grammes (0·2oz.) and the combustion rate of this powder low enough to eliminate any risk of explosive action. Tests carried out with container completely closed so as to prevent all the generated gas from escaping have shown that no damage from excessive pressure need be feared. Starting tests have been made with the piston in all positions, and even with the piston at the top dead centre no abnormal strain on the cylinder head is said to have been observed.—*"The Journal of Commerce" (Shipbuilding and Engineering Edition), No. 35,699, 9th July, 1942.*

Over-speeding of Oil Engine Due to Fuel Flooding.

A mishap which resulted in the over-speeding of an oil engine is described in the July issue of *"Vulcan"*. It occurred in connection with a horizontal unit having a cylinder 11in. in diameter and a piston stroke of 19in. Owing to the failure of one of the welded ends of a cylindrical fuel tank above the engine, the oil poured out on the floor plates, partially submerging the exhaust silencer and the air intake silencer. The engine then gathered speed to a dangerous extent, but was brought to rest without damage. It was thought that some of the fuel oil might have entered in liquid form through the open drain hole at the bottom of the air silencer, but it seems more likely that vapourised oil entered the silencer through the normal inlet, after having been in contact with the hot exhaust pipe. Thus, the engine would obtain fuel not under control of the governor, and so long as any combustible mixture was being drawn in through the air silencer the engine would be liable to race. The obvious moral is that care should be taken to ensure that in no circumstances should fuel oil be able to gain access to the air silencer or inlet pipe.—*"The Oil Engine", Vol. X, No. 111, July, 1942, p. 58.*

Removing Big Ends through Cylinders.

A method of designing the crankhead ends of I.C. engine connecting rods so as to enable them to be removed through the cylinders has been developed and patented by the Swiss Locomotive Works, Winterthur, and is shown in Fig. 1. A crankpin *(3)* of relatively large diameter can be used with this arrangement. The two connecting-rod bolts *(7)* are inclined so that their heads *(8)* bearing on the cap *(2)* are spaced farther apart than the ends carrying the nuts *(9)*. The two parts *(1, 2)* are located in their relative positions by means of dowels *(12)*. With a standard arrangement the width (B) would represent the outside dimension of the crankhead, and as shown in the left-hand diagram, this may be greater than that of the cylinder bore *(13)*. The top half of the bearing fits on the face *(6)* of the connecting rod *(5)* in the arrangement shown on the left, whereas the right-hand diagram indicates a bearing with a single joint passing through the axis of the crankpin *(3)*. Two bearing brasses *(14, 15)* are provided and inclined bolts are fitted as before. The width (B) would again be taken up by the bearing if parallel bolts were fitted.—*"The Oil Engine", Vol. X, No. 112, August, 1942, p. 108.*

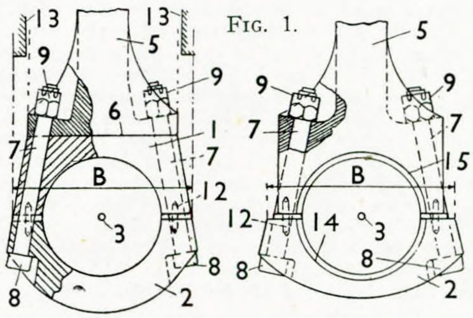


FIG. 1.

Future of Mild Steel in Ship Construction.

With reference to the proposals put forward in many quarters concerning the employment of aluminium magnesium alloys for ships' superstructures, it is pointed out that stainless steel having an ultimate tensile strength of about 80 tons/in.^{2} (which is about three times that of mild steel), can compete with most of these alloys when weight comes into the question, as scantlings made of stainless steel can, within limits, be reduced, whilst providing strength equal to that of aluminium alloy scantlings many times thicker. The relatively high cost of high-tensile steel has hitherto outweighed the advantages gained by the saving in weight which can be effected by its use, and the general reliability of mild steel in ship construction has not encouraged a demand for a higher quality substitute. Nevertheless, it is probable that mild steel, in its present form, may, to some extent, be superseded by improved materials such as plastics, aluminium alloys and H.T. steel, the odds being in favour of the latter.—*"Fairplay", Vol. CLIX, No. 3,088, 16th July, 1942, p. 102.*