SESSION 1910-1911

President: Sir David Gill, K.C.B.

The Application of the Marine Steam Turbine and Mechanical Gearing to Merchant Ships

By the Hon. C. A. PARSONS, C.B., Past Pres. I. Mar. E., Vice-President, Inst. Nav. Arch.

[*Read at the Spring Meetings of the Fifty-first Session of the Institution of Naval Architects, March* 18, 1910, *and reprinted by kind permission of the Council of the Institution, and of Mr. Parsons.*]

The steam turbine has not, as yet, been applied to vessels of slow normal speed on account of the high initial cost and inferior economy in steam; further, no promising scheme has, as yet, been evolved having for its object the modification of the turbine or propeller, so as to reduce the efficient speed of revolution of the former, and increase that of the latter for vessels of 12 knots sea speed and under, and the only approach to meeting these conditions (if we except gearing propositions) has been in the combination system, where the turbine plays a secondary part in the equipment, by utilizing the lower portion of the expansion of the steam between the low-pressure cylinder of the reciprocating engine and the condenser. This system was fully explained in the paper by Mr. Walker and myself at the meeting of this Institution in March, 1908.

The complete and most satisfactory solution for slow-speed vessels would appear to be by means of gearing, provided the losses in transmission, first cost, and cost of maintenance are not too great. Many forms of gearing—mechanical, electrical, and hydraulic—have been proposed or applied on a small scale.

I believe the first application of helical spur gearing to drive

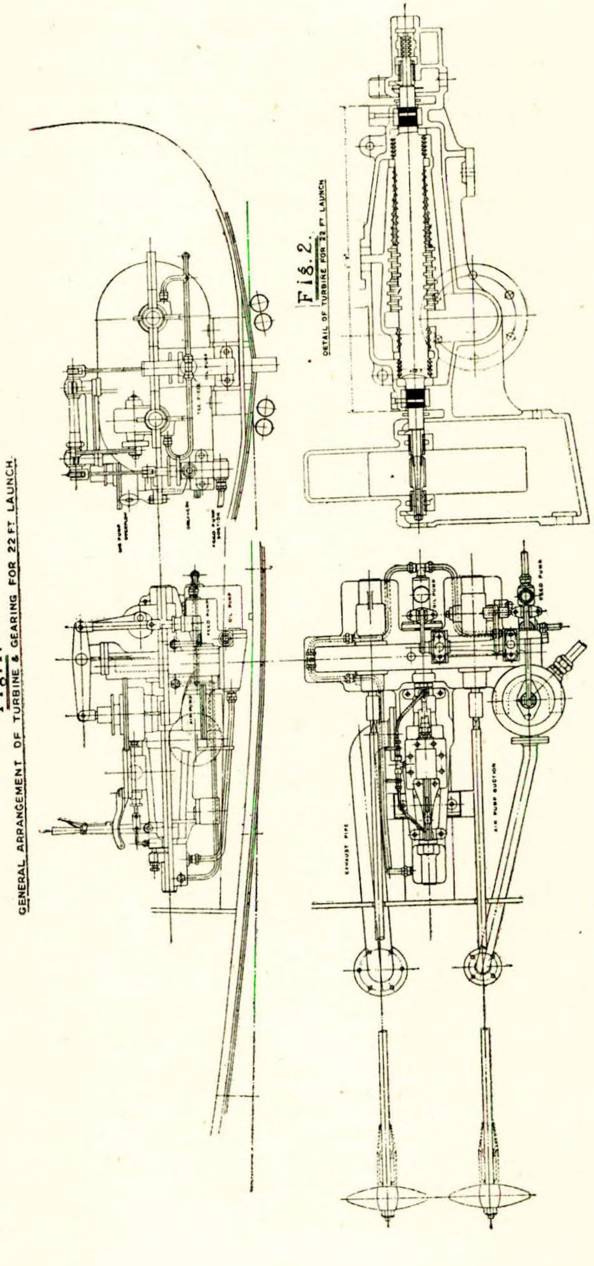


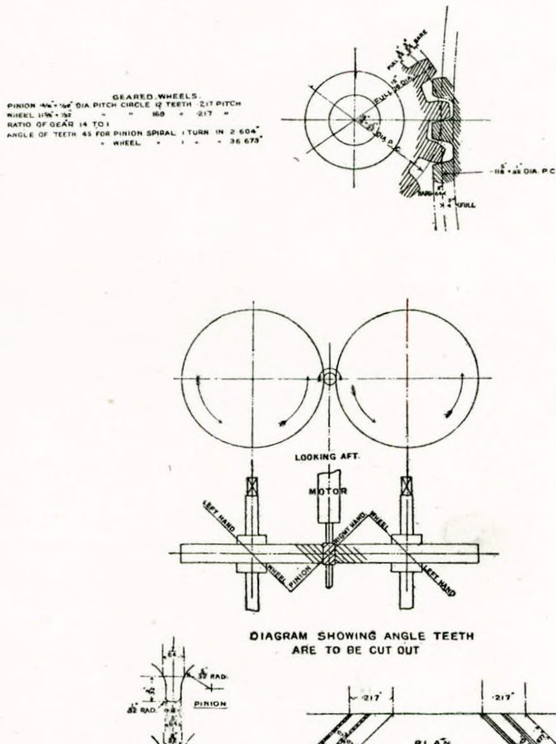
Fig. 1.
GENERAL ARRANGEMENT OF TURBINE & GEARING FOR 22 FT LAUNCH.

Fig. 2.
DETAIL OF TURBINE FOR 22 FT LAUNCH

a propeller was made by The Parsons Marine Steam Turbine Company, Limited, in 1897. The turbine was of 10 H.P. geared to two wheels, each wheel driving a propeller shaft, as shown in Fig. 1 (Plate I). The revolutions of the propellers were 1,400 per minute, and the ratio of the gear 14 to 1. The turbine was of the Parsons type, with a reversing turbine on the same shaft incorporated in the same casing (Fig. 2). The

Fig. 3

DETAIL OF GEAR.



gear was single helical, as shown in Fig. 3. The turbine took part of the thrust of the propeller, the remaining thrust being taken on the thrust bearing in the gear casing. The air, circulating, and oil pumps were driven by worm gearing off one of the screw shafts, as shown in Fig. 1. The launch was 22 ft. over all, and attained a speed of 9 miles an hour. She was built to the order of Mr. F. B. Atkinson, for his yacht *Charmian*. The launch was sent to the Turbinia Works in 1904, and the turbine was generally overhauled and cleaned. The gear was found to be in perfect order, and did not require any repair.

Helical and double helical gear of fine pitch suited to high speeds of rotation was, I believe, first introduced by Dr. De Laval, of Stockholm, and has been extensively used in connexion with his turbine for many years with entire success, and at a moderate cost of maintenance. I have had several experimental sets constructed. One of these was a double helical gear of the De Laval type made in 1897, gearing from 9,600 revolutions of the turbine to 4,800 of the dynamo, transmitting 300 H.P. The efficiency was estimated by the method of heat loss to be above 98 per cent. This gear was cut in an ordinary universal milling machine without any special precaution as to accuracy, and I was much impressed (in spite of the obvious irregularity of the teeth) by finding how well it ran, except that it made a considerable noise.

Gears that have been recently cut by the Power Plant Company and by special machinery run with very little noise. A recent experimental set of gearing cut by Messrs. D. Brown, of Huddersfield, from 2,000 to 400 revolutions, transmitting 300 H.P., on a Heenan-Froude water brake dynamometer, gave a total loss in the gear case, including friction of gear and bearings of $1\frac{1}{2}$ per cent.

In the summer of last year the directors of the Turbinia Works Company decided to test turbines mechanically geared to the screw shaft of an existing typical slow-speed vessel, and a cargo vessel named the *Vespasian* was purchased for this purpose.

The *Vespasian* was built in 1887 by Messrs. Short Brothers, of Sunderland. Her dimensions are :—Length on load water line, 275 ft. ; breadth moulded, 38 ft. 9 in. ; depth moulded, 21 ft. 2 in. ; mean loaded draught, 19 ft. 8 in. and displacement, 4,350 tons.

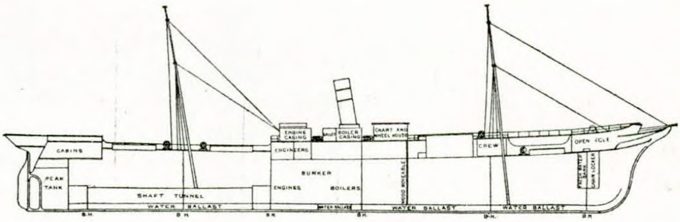
Previous to installing turbine engines with reduction gear, the vessel was fitted with an ordinary triple-expansion surface-condensing engine by Mr. G. Clark, of Sunderland, with cylinders $22\frac{1}{4}$ in. by 35 in. by 59 in. and 42 in. stroke. The air, circulating, feed, and bilge pumps were driven from the intermediate-pressure crosshead, with the usual arrangement of levers and links. The condenser was cast with the back columns of the main engine, and had a cooling surface of 1,770 sq. ft. The boilers—two in number—are 13 ft. diameter by 10 ft. 6 in. long, with a total heating surface of 3,430 sq. ft., and grate area of 98 sq. ft., working under a pressure of 150 lbs. with natural draught. The propeller is of cast iron, and has four blades, having a diameter of 14 ft., pitch 16·35 ft., and expanded area of 70 sq. ft.

With a view to obtaining comparative data between the turbine installation and the reciprocating engine, it was decided to run trials of the vessel with her reciprocating engine previous to its removal and the installing of the turbines and gearing.

Fig. 4 (Plate II) shows a profile of the vessel, and Fig. 5 a general arrangement of the reciprocating engine and boilers.

Fig. 4.

PROFILE



Before proceeding on the voyage from which data regarding the performance of the reciprocating engine were taken, the propelling machinery was completely dismantled and overhauled. The high-pressure piston valve chamber was rebored and new valve rings fitted; slide valves were replaned and faced up; bearings were renewed, and other repairs carried out wherever necessary to bring the machinery into an efficient condition and first-class working order. To obtain reliable measurements of water consumption, two tanks were fitted, each of 400 gallons capacity, with suitable change cocks and connexions for the air pump to discharge through these measuring tanks.

It was necessary, for the purpose of obtaining data under service conditions, that the vessel should be run at her loaded

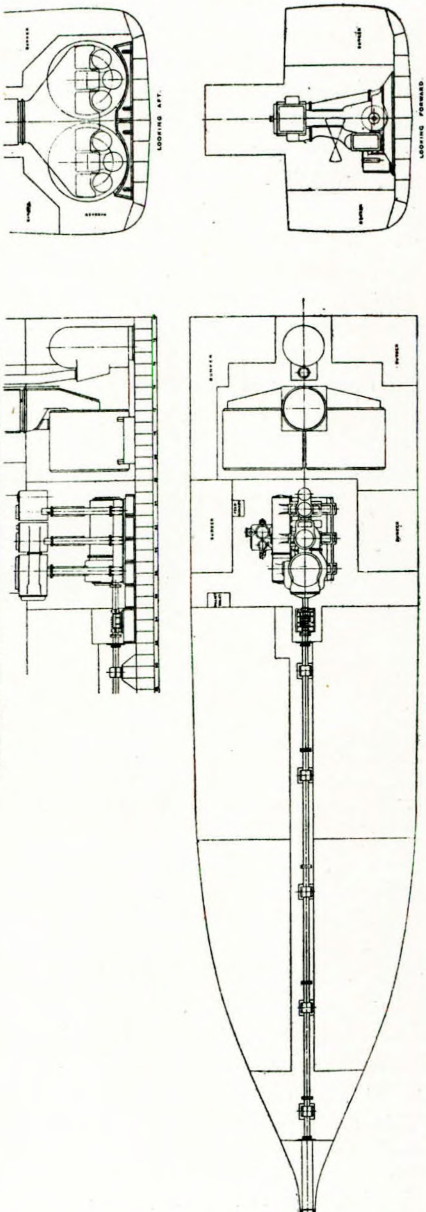


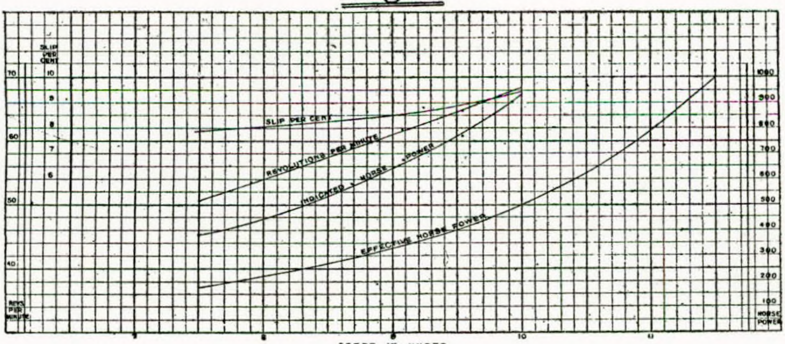
Fig. 5.
S.S. 'VESPASIAN'. GENERAL ARRANGEMENT OF RECIPROCATING ENGINES & BOILERS.

condition. Arrangement were consequently made with a local firm of shipbrokers to take a cargo of coal from the Tyne to Malta, and on June 26, last year, the *Vespasian* left the Tyne in a loaded condition with a special recording staff on board, and on this voyage careful measurements of coal and water consumption were made.

The data and results of a progressive trial carried out on the Whitley Bay mile are given in the Appendix, from which, together with the data taken on the voyage referred to, the curves in diagrams 6, 7, 8, 9, and 10 (Plate IV) have been plotted.

Diagram 6 (full lines) shows the results of the progressive trial to a basis of speed of vessel. The effective horse-power shown on this diagram is calculated from the resistance as

Fig. 6.



obtained from a model experiment of the vessel to a scale of $\frac{3}{8}$ in. to the foot.

Diagram 7 (full lines) shows the results as obtained on the voyage, plotted to a basis of revolutions. For the sake of comparison, the indicated horse-power taken on the progressive trial is also shown, together with the speed corresponding to revolutions taken on the measured mile.

Diagram 8 (full lines) shows the water consumption per hour for main engines only and also for all purposes to a basis of revolutions. The difference between these two curves represents the consumption of steam of the steering engine, the exhaust of which was led to a separate measuring tank.

Fig. 7.

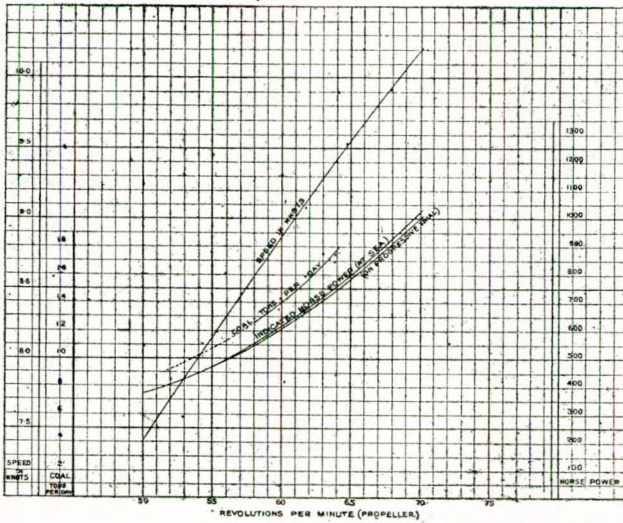


Diagram 9 shows the water per indicated horse-power also plotted to a basis of revolutions ; and—

Fig. 8.

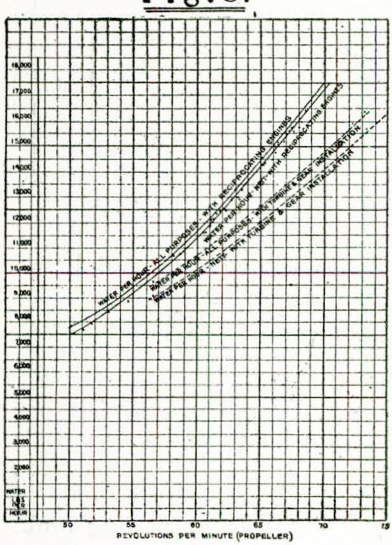
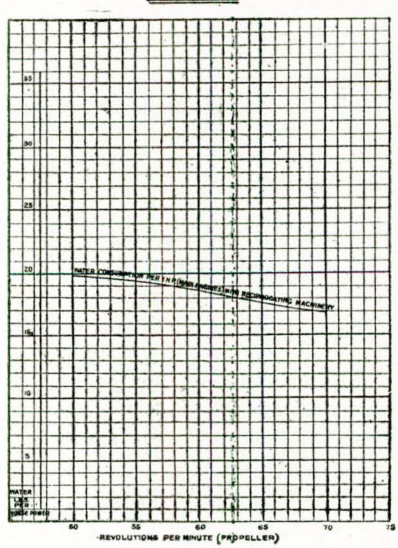


Fig. 9.



S S "VESPASIAN" GENERAL ARRANGEMENT OF TURBINES & GEARING

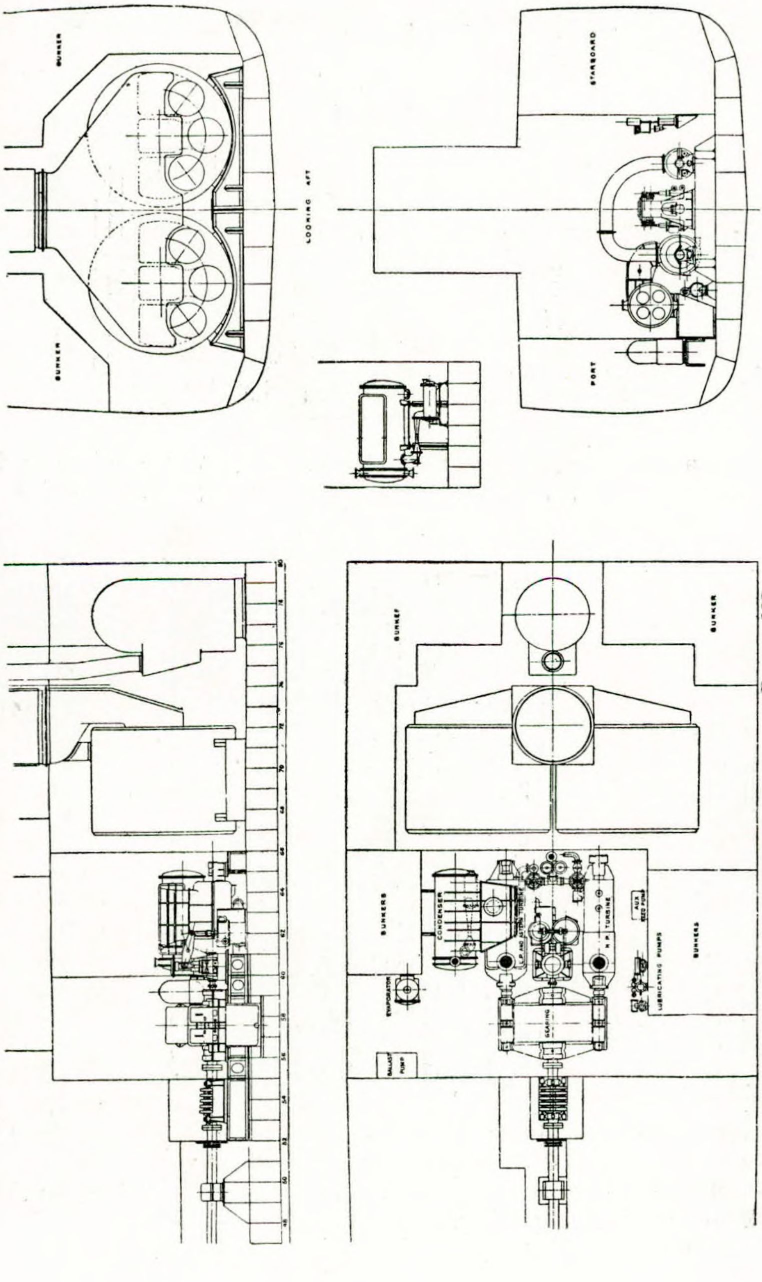
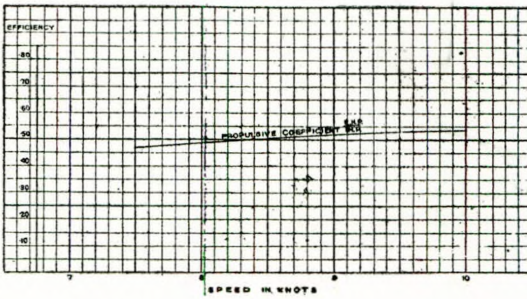


PLATE III.

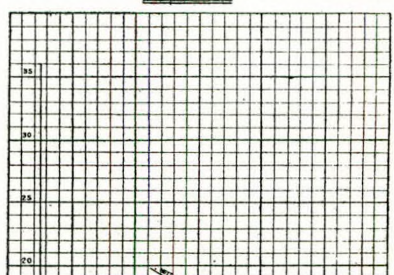
Diagram 10 shows the propulsive co-efficient plotted to a basis of speed.

Fig. 10.



On the completion of the voyage, the vessel returned to the Turbinia Works, where her reciprocating engine was taken

Fig. 11.



out, engine seats re-modelled, and preparations made for the reception of the turbines and gearing.

Plate III shows the general arrangement of the turbine machinery and gearing.

The only alteration made to the vessel was in the type of propelling engines ; the boilers, propeller, shafting, and thrust blocks remained the same as for the reciprocating engine.

The propelling machinery consists of two turbines in "series," viz., one high pressure and one low pressure, the high-pressure turbine being placed on the starboard side of the vessel and the low-pressure on the port side. At the after end of each of the turbines a driving pinion is connected with a flexible coupling between the pinion shaft and the turbine, the pinion on each side of the vessel being geared into a wheel, which is coupled to the propeller shaft.

A reversing turbine is incorporated in the exhaust casing of the low-pressure turbine. The air, circulating, feed, and bilge pumps are of the usual design for tramp steamers, and are driven by means of a crank and connecting rod coupled to the forward end of the gear wheel shaft. The turbine and pinion shaft bearings are under forced lubrication, similar to ordinary turbine practice. The teeth of the pinions and of the gear wheel are lubricated by means of a "spray" pipe extending the full width of the face of the wheel. Independent oil pumps are fitted for supplying oil to the bearings and gear wheel, with a view to the possibility of experimenting with different lubricants for the gear wheel, the oiling system for the bearings being separate from that of the gear wheel.

The high-pressure turbine is 3 ft. maximum diameter by 13 ft. over all length, and the low-pressure 3 ft. 10 in. diameter by 12 ft. 6 in. length. The turbines are similar in design to a land turbine, being balanced for steam thrust only, the propeller thrust being taken up by the ordinary thrust-block of the horse-shoe type, which is fitted aft of the gear wheel.

A new condenser, together with a vacuum augmentor, was fitted with the turbine installation. The cooling surface of the condenser is 1,165 sq. ft.

The gear wheel is of cast iron, with two forged steel rims shrunk on. The diameter of the wheel is 8 ft. 3½ in. pitch circle, having 398 teeth—double helical—with a circular pitch of ·7854 in. The total width of face of wheel is 24 in. ; inclination of teeth 20° to the axis.

The pinion shafts are of chrome nickel steel, 5 in. diameter pitch circle, with 20 teeth ·7854 circular pitch. The ratio of gear is 19·9 to 1.

On the completion of the erection, on board, of the turbine-gearing installation at the end of February of this year, the vessel was loaded to the same draught and displacement as on the trials referred to in the Appendix. As already mentioned, the propeller has not been touched or altered in any way. In the short interval since the completion of her alterations, the vessel has been out to sea on four occasions.

The following table gives the data and results of a run made off the Tyne on the 11th of this month (April, 1910) at varying revolutions :—

Speed in knots	8·4	9·56	10·5	10·66
Revolutions per minute	56·5	65·0	71·3	73·3
Boiler pressure	145	144	140	145
Initial pressure H.P. turbine	60 lb.	86 lb.	110 lb.	121 lb.
Initial pressure L.P. turbine	15·2 in.	12·5 in.	7·1 in.	5·5 in.
Vacuum	28·8 in.	28·8 in.	28·7 in.	28·5 in.
Barometer	29·9 in.			
Shaft horse-power	456	740	980	1,095
Water consumption per hour, main engines	9,070 lb.	12,000 lb.	14,480 lb.	15,670 lb.
Water consumption, all purposes	9,670 lb.	12,620 lb.	15,120 lb.	16,370 lb.
Water consumption, per shaft horse-power, main engines	19·8 lb.	16·2 lb.	14·8 lb.	14·3 lb.

The water consumptions per hour at the several rates of revolutions have been plotted on Diagram 8, shown in full lines for the reciprocating engines and in dotted lines for the turbine geared engines.

It will be noted that under normal full speed steaming conditions an increase of about one knot is obtained with the same coal consumption.

Diagram 11 shows the water consumption per shaft horse-power for the geared turbines.

It will be noted that the observed mean speeds on the measured mile given in the above table correspond to the speeds obtained with the reciprocating engines at the same revolutions, thus eliminating any necessity for allowances, the weather conditions in the two cases being very similar.

It may be mentioned that the turbines and gearing have given no trouble, and have worked satisfactorily with very little noise or vibration throughout the trials. Further, there is no appreciable wear on the teeth or bearings.

It is proposed to put the vessel into commission and run extended trials.

APPENDIX.

SS. *VESPASIAN*. PROGRESSIVE TRIAL ON HARTLEY MILE, WITH RECIPROCATING ENGINES.

Draught taken at Whitley Bay. Draught, forward, 19 ft. 1 in.; draught, aft, 20 ft. 3 in.; draught, mean, 19 ft. 8 in. Dimensions, moulded, 275 ft. by 38·75 ft. by 21·16 ft. Displacement, 4,350 tons; co-eff., ·754. Propeller, solid cast iron, 4 blades; diameter, 14 ft.; pitch, 16·35 ft.; expanded area, 10,100 sq. in. Measured mile at Hartley. High-pressure cylinder, diam., 22¼ in., mean cut off, 71%; intermediate cylinder, diam., 35 in., mean cut off, 63%; low-pressure cylinder, diam., 59 in., mean cut off, 64%; stroke, 42 in.

Direction of run	S	N	Mean	S	N	Mean	S	N	Mean	S	N	Mean
Speed	7·438	7·563	7·50	8·531	7·860	8·195	9·278	8·090	8·684	11·009	9·399	10·204
Revolutions per min.	49·9	51·27	50·58	55·1	55·5	55·3	58·4	59·3	58·85	70·0	70·1	70·05
Boiler pressure in lb.	126	130	128	138	129	133·5	135	143	139	152	149	150·5
H.P. Receiver pressure	73	72	72·5	82	84	83	95	95	95	128·5	127·5	128
I.P. Receiver pressure	17·5	17·25	17·37	20·5	22	21·25	26	26	26	44	45	44·5
L.P. Receiver pressure	−4·75	−6	−5·37	−4·5	−4·75	−4·62	−3·37	−2·75	−3·06	3	3·62	3·31
L.P. Exhaust pressure	27·0″	27·0″	27·0″	27·0″	27·0″	27·0″	26·8″	26·8″	26·8″	25·2″	25·2″	25·2″
Vacuum	28·25″	28·37″	28·31″	28·5″	28·5″	28·5″	28·25″	28·25″	28·25″	26·5″	26·5″	26·5″
Barometer	29·96″	—	—	—	—	—	—	—	—	—	—	—
Mean H.P. Pressure	30·75	30·35	30·55	34·25	34·85	34·55	37·3	38·15	37·72	47·6	47·95	47·77
Mean I.P. Pressure	13·8	13·85	13·82	14·8	14·6	14·70	17·25	17·45	17·35	24·8	24·65	24·72
Mean L.P. Pressure	3·91	4·125	4·01	4·8	4·87	4·83	5·61	5·91	5·76	8·86	9·51	9·18
I.H.P., H.P.	124	126	125	152·5	156·5	154·5	176	183	179·5	269	272	270·5
I.H.P., I.P.	139	143·5	141·2	164·5	164	164·2	204	209	206·5	351	348	349·5
I.H.P., L.P.	113	122	117·5	153	156·5	154·7	189·5	203	196·2	359	387	373
I.H.P., Total	376	391·5	383·7	470	477	473·5	569·5	595	582·2	979	1007	993
Temp. Circ. Inlet.	55·5	—	—	—	—	—	—	—	—	—	—	—
,, ,, Disc.	76	81	78·5	87	87	87	91	90	90·5	106	107	106·5
,, Hotwell	73	71	72	81	85	83	91	92	91·5	118	119	118·5

Direction of run	S	N	Mean	S	N	Mean	S	N	Mean	S	N	Mean
Speed	10·778	9·068	9·923	10·315	8·759	9·537	9·809	8·824	9·316	8·824	9·326	9·075
Revolutions per min.	67·4	68·27	67·83	64·9	64·36	64·63	62·0	62·1	62·05	62·1	61·3	61·7
Boiler pressure in lb.	142·5	130	136·2	143	132	137·5	128	128	128	128	120	124
H.P. Receiver pressure	120·5	115	117·75	113	107	110	101	102	101·5	102	100	101
I.P. Receiver pressure	40·0	42	41·0	36·5	35·5	36	31	30·5	30·75	30·5	32	31·25
L.P. Receiver pressure	1·5	2·75	2·12	−·75	−1·0	−·875	−1·5	−1·5	−1·5	−1·5	−2·5	−2
L.P. Exhaust pressure	25·75″	25·5″	25·62″	26·2″	26·0″	26·1″	26·25″	26·25″	26·25″	26·25″	26·25″	26·25″
Vacuum	27·25″	26·87″	27·06″	27·67″	27·37″	27·52″	27·5″	27·75″	27·62″	27·75″	27·67″	27·71″
Barometer	—	—	—	—	—	—	—	—	—	—	—	—
Mean H.P. Pressure	44·12	43	43·56	41·85	42·25	42·05	39·3	40·5	39·9	40·5	40·5	40·5
Mean I.P. Pressure	23·2	24·67	23·93	21·7	22·0	21·85	20·25	19·7	19·97	19·7	20·55	20·12
Mean L.P. Pressure	8·42	8·8	8·61	7·18	6·96	7·07	6·32	6·5	6·41	6·5	5·83	6·16
I.H.P., H.P.	241	237·5	239·2	220	220	220	197·5	203·5	200·5	203·5	201	202·2
I.H.P., I.P.	316	340·5	328·2	284·5	286	285·2	253·5	247	250·2	247	254·5	250·7
I.H.P., L.P.	328·5	344	336·2	269·5	259	264·2	227	233·5	230·2	233·5	206·5	220
I.H.P., Total	885·5	922	903·7	774	765	769·5	678	684	681	684	662	673
Temp. Circ. Inlet.	—	—	—	—	—	—	—	—	—	—	—	—
,, ,, Disc.	101	104	102·5	97	98	97·5	94·5	94	94·2	94	94	94
,, Hotwell	112	116	114	98	101	99·5	94	93	93·5	93	94	93·5

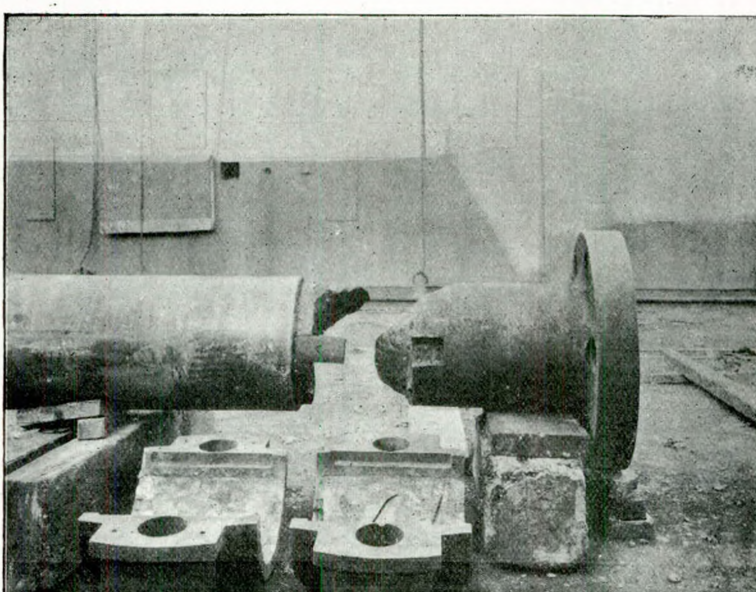
President : Sir David Gill, K.C.B.

Repairs to Tail-end Shaft of ss. "Anglian"

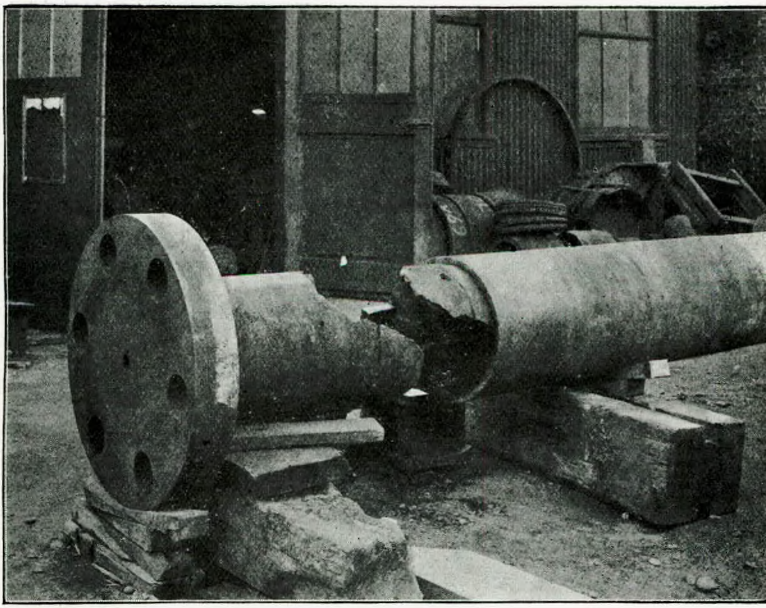
Contributed by Mr. E. W. RUTTER, R.D., R.N.R. (Member).

June, 1910.

The Leyland Line ss. *Anglian*, while on a voyage from Boston to London, fractured her "tail end" shaft during a heavy gale on April 8, 1910. The damage consisted of a circumferential fracture about one inch forward of the stern



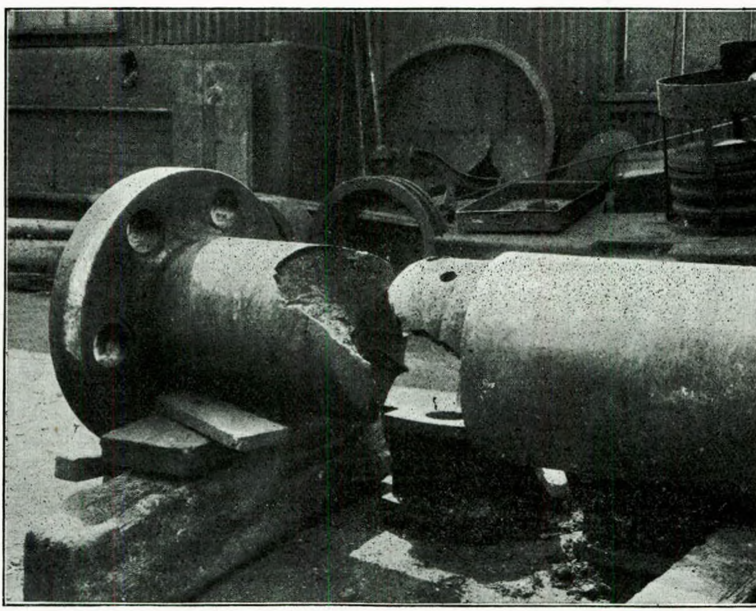
gland, at the forward end of the continuous liner, running off longitudinally forward about 7 inches and after to about the same extent inside the liner, though this latter could not be ascertained at the time; the stern gland was also broken. To get abaft the circumferential fracture as far as possible so as to make effective repairs, it was necessary to draw the stern gland, take out the packing, and then screw the gland up as far as possible. The liner being bulged forward of the gland, about one inch had to be cut off before the gland could be



drawn. After the latter was drawn the packing was taken out and the gland screwed up another $1\frac{1}{2}$ inches; a further portion of the liner was then cut off, exposing the shaft $2\frac{1}{2}$ inches abaft the circumferential fracture. A steel key 2 in. × $3\frac{1}{2}$ in. × 6 in. was then fitted in the shaft in the only good part available for this purpose, and a pair of "bottom end" brasses bolted over all by two 4 in. main bearing bolts; the "bottom end" bolts being too long for clearance, the difference between the sizes of bolts and holes—4 in. and $4\frac{1}{2}$ in.—was made up by bushes of $\frac{1}{4}$-inch Muntz metal and the bolts thus made a driving fit. On these repairs the *Anglian* steamed

540 miles at an average speed of seven knots, arriving at Queenstown and anchoring in the harbour there without assistance, the repairs still holding good.

The *Anglian* left Queenstown for London with two tugs in attendance at 4 p.m, April 19, steaming at a speed of $7\frac{1}{2}$ knots. All went well until 10 a.m. on the 20th, when, in response to an urgent order from the bridge, accompanied by a verbal message from the Master, which under the circumstances it was impossible to disregard, the engines were re-



versed, with the result that the shaft parted and slipped aft, the broken ends fouling, bending intermediate shaft, and breaking two tunnel-bearing pedestals. The remainder of the voyage to London was performed in tow of the two tugs attending, but had it not been for the order to go "Full speed astern," there is not the slightest doubt the repaired shaft would have taken the ship to her destination, as up to the time of going astern, it remained in as good order as when the repair was completed, and up to the time of attempting to carry out the emergency order and go full speed astern, it had taken the ship 670 miles.

The following members were elected at the meeting of Council held Thursday, June 3; 1910.

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Mr. J. Howden Hume, Glasgow.
Mr. T. W. Mallett, London.
Mr. J. Morrison, Felixstowe.
Mr. J. J. Murdoch, Liverpool.
Mr. Charles Spink, Hull.
Mr. A. E. Starkey, R.N.R., Belfast.
Mr. Charles Stewart, Trieste.

As Associate.

Wm. E. Batt, London.

INSTITUTE OF MARINE ENGINEERS

INCORPORATED

SESSION 1910–1911

President: Sir David Gill, K.C.B.

Vol. XXII

LECTURE ON THE STABILITY OF SHIPS,

By Mr. EDWIN TATE,

Read Monday, April 4th, 1910.

Chairman: Mr. J. T. MILTON (Chairman of Council).

PAPER OF TRANSACTIONS NO. CLXVII.

INTERNAL COMBUSTION ENGINES FOR MARINE USE,

By Mr. W. R. CUMMINS (Member),

Read Saturday, June 25th, 1910,

In the Hall of the Garden Club, Japan-British Exhibition, Shepherd's Bush, W.

Chairman: Sir WM. HALL-JONES, K.C.M.G. (High Commissioner for New Zealand).

REPORT ON BURST STEAM PIPE,

Contributed by Mr. J. M. BUCHANAN (Member).

BOARD OF TRADE CERTIFICATES:

Correspondence with Regard to Proposed Alterations to the Regulations.

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INCORPORATED

SESSION 1910-1911

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Hon. Minute Secretary: J. G. Hawthorn.

Hon. Financial Secretary: E. W. Ross.

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Jas. Adamson, Awards.
A. E. Battle, Papers.
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D. Hulme, Reading Room.
A. H. Mather, J. McLaren, Recreation.
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J. F. Redman, Library.

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