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President: STEPHEN J. PIGOTT, Esq., D.Sc.

The White Combined Steam Engine.

READ

By W. A. WHITE (Member).

On Tuesday, April 13th, 1937, at 6 p.m.

CHAIRMAN: Mr. R. RAINIE, M.C. (Chairman of Council).

Synopsis.

THE paper deals with modern steam machinery for marine service, with particular reference to the White combined steam engine, its development and design, and discusses torsional characteristics, etc., also the application of the engine to cargo vessels and their performance in service.

The thermal efficiency of a heat engine depends chiefly upon the temperature range in accordance with Carnot's principle "The temperature at which heat (steam) is supplied, and the temperature at which it is rejected should be as wide apart as possible". The author in his engine applies the maximum range obtainable when using normal steam pressures and high superheat temperatures, by supplying the ingoing steam (at a pressure of 240/250lb. gauge) at the maximum superheat temperature obtainable with Scotch boilers, and rejecting it at the lowest temperature obtainable in a low pressure turbine exhausting into maximum vacuum.

In the development of the engine the constant endeavour has been to extend the thermodynamic cycle and obtain as nearly as is possible the maximum efficiency of conversion of the available energy in the steam, while keeping within the limits of the pressures used in everyday practice.

Regarding the economy to be gained by using high steam pressures, with the high superheats now obtainable, the author does not consider the added gain obtainable from steam pressures above 240/260lb. gauge sufficient to warrant the commercial expenditure entailed in providing for their use in merchant vessels (provided weight of machinery is not of paramount importance). The author rather advocates high superheat temperatures and has not encountered any troubles when using superheats of 700° to 720° F. This no doubt is largely owing to the small diameter of the high-pressure cylinders required with his engine being eminently suitable for high superheats.

Of the various methods which may be applied to improve the thermal efficiency of a steam unit, the method most generally adopted is that of increasing the temperature of the feed water by using steam tapped from the engine after it has done a definite amount of work in the engine. This method is not applied in the White engine as there is not a sufficient material gain indicated, the small steam consumption per unit of power in the engine making it a better commercial proposition to convert the entire steam into work or power.

Advantage is taken in the engine of the economy to be obtained from reheating the steam after partial expansion has been effected. The exhaust steam from the h.p. cylinders is reheated in its passage to the l.p. cylinder, the amount of heat transferred being regulated to the amount required to ensure dry or slightly superheated steam being exhausted from the l.p. cylinder. The exhaust steam from the reciprocating engine is also reheated in its passage to the turbine.

In the reciprocating engine the reheating is effected by extracting a percentage of the heat from the highly superheated steam in its passage from the engine stop valve to the h.p. valve chest and giving it to the l.p. cylinder steam. The re-heating of the turbine steam is effected by extracting a percentage of the heat from the superheated auxiliary steam (before it branches off to the various auxiliaries), and giving it to the reciprocating engine exhaust steam in its passage to the turbine. This systematic reheating throughout the engine ensures a higher dryness fraction at the reciprocating engine and the turbine exhausts, and consequently a higher overall thermal efficiency is obtained.

The exhaust steam from the l.p. cylinders is passed through a specially designed separator (fitted between the engine and the turbine) where any entrained oil or water is extracted before the steam is passed through the reheater.

For 50 years the reciprocating engine had been regarded as all that could be desired for the tramp or cargo-carrying vessel. It was simple and reliable, and could be subjected to long periods of service treatment without seriously affecting its performance. These features in more prosperous times offset to a very large extent its heavy fuel consumption. In the recent slump in shipping and shipbuilding, the need of a steam unit which had not so heavy a fuel consumption and yet possessed the advantages of the triple- or quadruple-

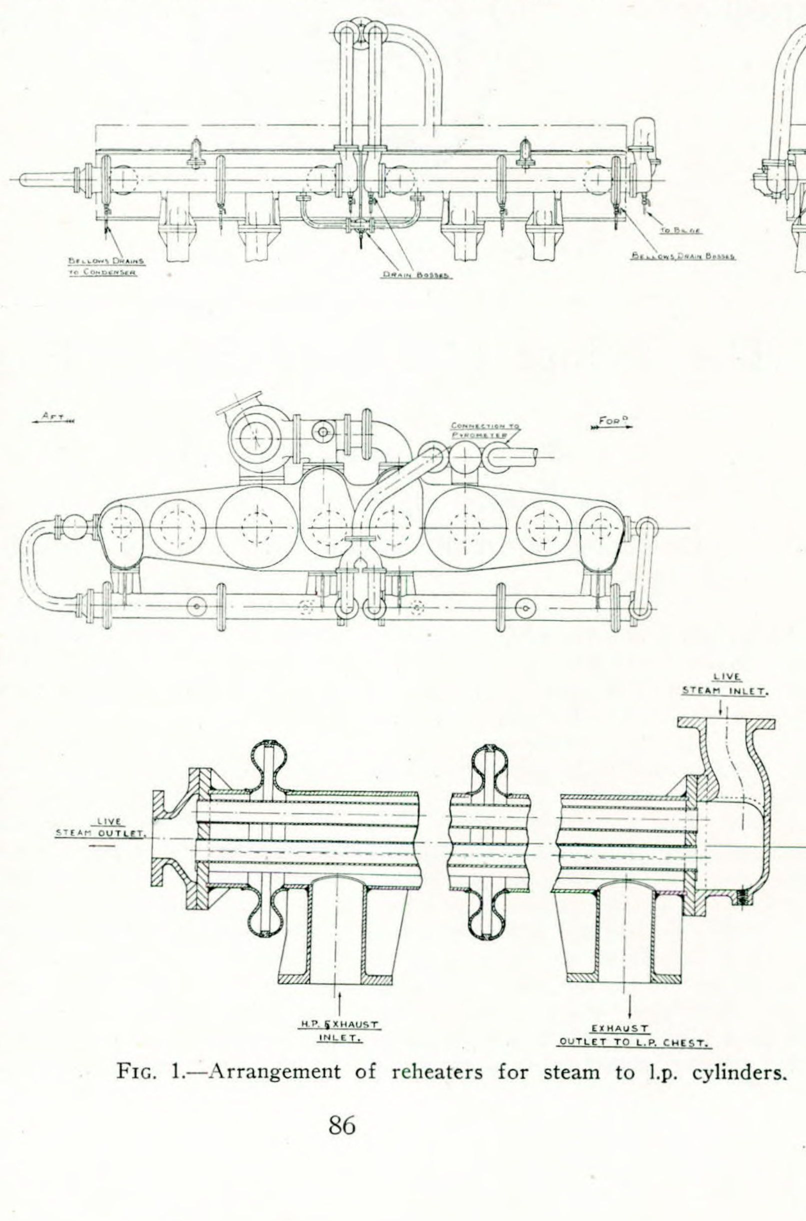


FIG. 1.—Arrangement of reheaters for steam to l.p. cylinders.

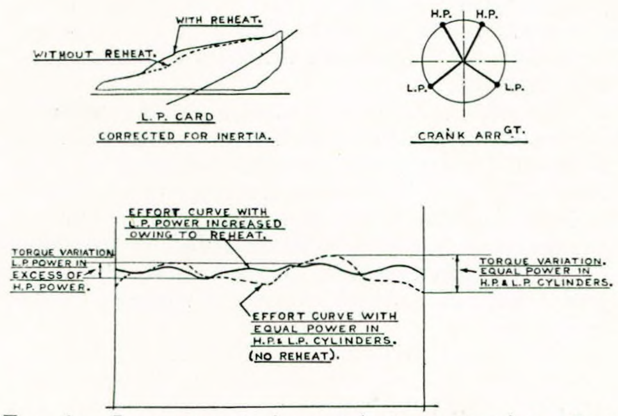


FIG. 1A.—Improvement from reheat on turning effort diagrams.

expansion engine became apparent, and the author applied himself to an investigation of the high-speed reciprocating engine with its high efficiency and reduced weight, running in conjunction with a low-pressure turbine.

Ten or twelve years ago it would have been regarded as impracticable to combine a high-speed engine with reduction gearing, but, as the author observed, Diesel engines were being coupled to the propeller shaft through gearing and giving satisfactory results, for example the twin-screw vessels "Havelland", "Munsterland" and "Vogtland". The author contended that a well-balanced steam engine would not be so harsh in its action nor yet have such large variations in turning effort, and should naturally, as far as the gears were concerned, be even more satisfactory than the geared Diesel engine. It was decided to build a small engine of about 500 i.h.p. to demonstrate and test the possibilities of this type of steam unit. The engine was built and from a mechanical standpoint functioned perfectly and left little to be desired. There were still those who held the opinion that a high-speed engine would not work in conjunction with gears and predicted heavy wear of the teeth; but the smooth running of the trial engine proved that there was nothing to fear, and the teeth of the gearing gave no evidence of shock or wear whatever.

The average efficiency ratios of an ordinary triple-expansion engine can be taken as h.p. cylinder 85 per cent., m.p. cylinder 75 per cent., and l.p. cylinder 50 per cent. Allowing one-third of the available heat drop in each cylinder, there is an

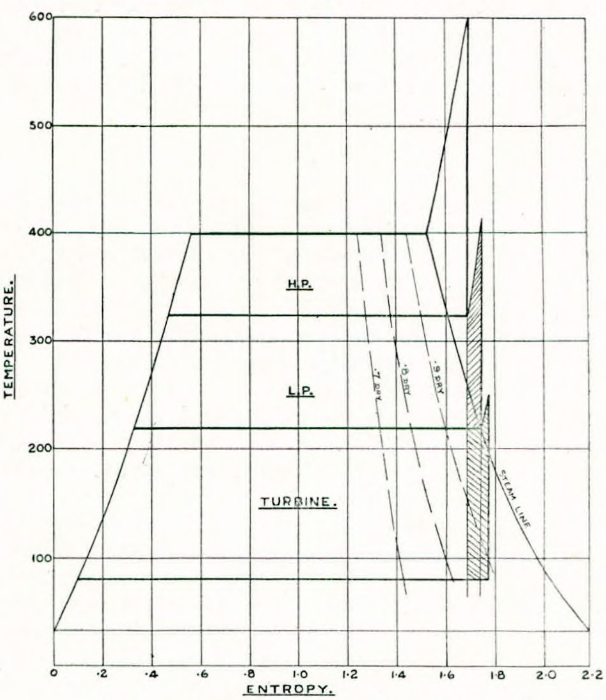


FIG. 3.—Entropy diagram for reheat.

overall efficiency ratio of $(\cdot 85 \times \cdot 33) + (\cdot 75 \times \cdot 33) + (\cdot 5 \times \cdot 33) = \cdot 7$, representing a card factor of around ·66 to ·67. The triple-expansion engine in the test set had efficiency ratios of ·93, ·9 and ·66, representing an overall efficiency of ·83 and a card factor of about ·79. With the double-compound engine the efficiency ratios are further improved, the h.p. cylinder being around ·9, and the l.p. cylinder around ·84, giving an improved efficiency ratio in the neighbourhood of 5 per cent. in the reciprocating engine, and an overall gain in the engine and turbine of about 7½ per cent.

Opinions differ regarding the efficiency ratio of low-pressure turbines, estimates varying as widely as ·65 to ·85 being given. The efficiency is materially reduced by reduction in vacuum, which to ensure maximum efficiency of the turbine must be kept as high as possible. 72·5 per cent. would appear to repre-

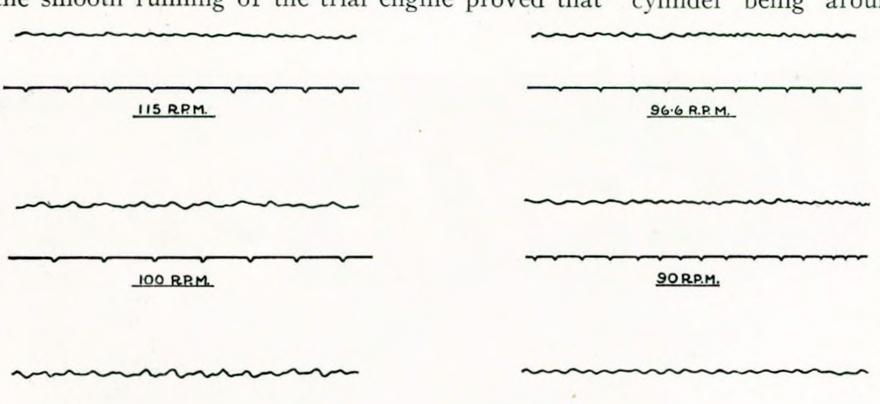


FIG. 2.—Torsiograph records taken from main shaft of White engine.

sent a fair average efficiency of the low-pressure turbine with inlet pressures up to about 20lb. absolute and exhausting between ·5lb. and ·7lb. absolute; with reheat this may reach 75 to 78 per cent.

In the later designs of the engine two stages of reheat are introduced, one between the h.p. and l.p. cylinders and one between the reciprocating engine and turbine. These have the effect of further increasing the efficiency ratio of the l.p. cylinders and turbine by from 3 to 5 per cent. each, and an overall efficiency ratio of around 83 per cent. is being obtained.

Design.

In designing the engine for marine use, the aim has been to develop a propelling unit which would be of robust construction, reliable and highly economical. At the same time constant regard has been paid to the all-important factor of first cost to ensure that this be kept sufficiently low to make the project a commercial one.

Particular care has been taken in design to minimise the effects of unbalanced inertia forces, and the dynamic characteristics of the installation as a whole have been adjusted to provide smooth operation at all speeds. In the several units built almost perfect balance has been obtained, with no evidence of vibration at revolutions of over 350 per minute. With the double-compound engine there are four exhaust periods per revolution, ensuring a steady steam flow to the turbine, and the disposition of the cranks and the weights of the running gear have been adjusted to provide the highest possible degree of engine balance.

The revolving weights of the crankshaft and connecting rods are wholly balanced in the usual manner by balance weights forged integral with the crank webs, which cancel all inertia forces due to rotating masses for each cylinder individually and remove a large proportion of the inertia loading from the crankshaft and engine frame. Primary inertia forces originating from the reciprocating parts are balanced by the disposition of the cylinders, by suitable adjustment of the reciprocating weights of the h.p. and l.p. engines, and by proper choice of crank angles which are determined solely from consideration of balance. Secondary inertia forces are balanced in the same way, the only residual unbalanced effect being a small secondary inertia couple.

Since an unbalanced *couple* is less effective in causing vibration than an unbalanced *force* of equal magnitude, and since a *secondary* couple is less effective than a *primary* couple of equal magnitude, it would require an unusual degree of resonance to produce any perceptible vibration with the engine thus balanced.

To ensure further as small a speed fluctuation as possible, a flywheel (used for turning wheel) is mounted between the engine and gears. The dimensions of this flywheel have been chosen very carefully to match the dynamic characteristics of the system as a whole, with the object of providing a degree of speed and energy fluctuation comparable with that of steam engines driving electric generators.

The large inertia of the turbine ensures that the turbine end of the installation revolves with sensibly constant angular velocity, and any periodic torque variation either at the propeller or at the reciprocating engine is prevented from causing undesirable torque fluctuations at the gear teeth by carefully adjusting the dynamic characteristics of the system to suit the desired operating conditions.

In the s.s. "Adderstone" the oscillating system comprising the propeller and the propeller shafting is tuned to a frequency of 132 vibrations per minute, and since the propeller is four-bladed, critical conditions arise at a propeller shaft speed of $\frac{132}{4} = 33$ r.p.m. The torque curve indicates that not only is the critical speed well below the operating speed, but, in addition, the damping action of the propeller is sufficient to prevent excessive vibration torque and stress.

The oscillating system comprising the engine masses and quill shaft is tuned to a frequency of 420 vibrations per minute, the principal critical speeds occurring at 210, 140 and 100 r.p.m. The normal running speed is therefore situated in the quiet region between the first order critical speed at 420 r.p.m. and the second order at 210 r.p.m. The torque curve shown in Fig. 5 indicates that the critical speeds are well below the normal operating speeds, and the magnitude of the vibration torques and stresses well within permissible limits, so that there are no difficulties in running up to speed or in operat-

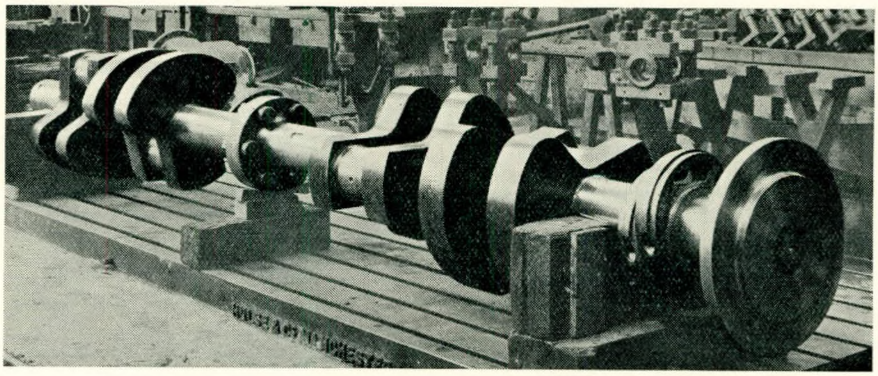


FIG. 4.—Crankshaft.

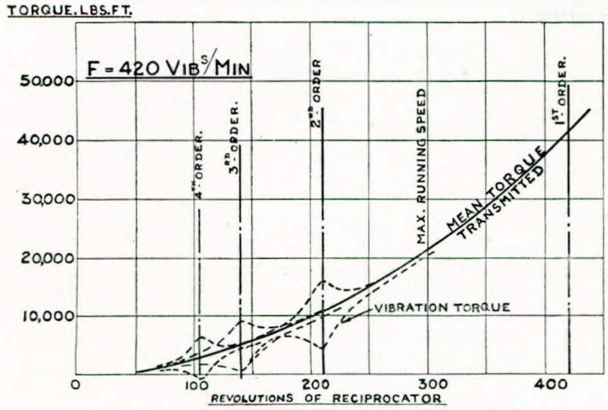


FIG. 5.—S.s. "Adderstone"—torque characteristics of engine shaft system.

ing at low speeds when manœuvring.

Application.

The pronounced success of the trial engine encouraged and decided the author to demonstrate its efficiency and reliability for larger classes of vessels, and he purchased the s.s. "Boswell", a turbine-driven 8,000-ton deadweight steamer of 11,523 tons displacement, 400ft. long × 52ft. breadth × 25ft. 3in. draft, built by Messrs. Harland & Wolff in 1920. She was renamed the s.s. "Adderstone".

The "Boswell" was a single-screw double-reduction geared turbine job, with the impulse type of high-pressure and the reaction type of low-pressure turbine, using saturated steam of 200lb. pressure which was supplied by three main boilers 15ft. 6in. diameter by 11ft. 6in. long, and forced draught.

In making the conversion the high-pressure turbine and its gear wheels were removed, and a high-speed four-cylinder double-compound engine having cylinders 12in. × 22¼in. × 22¼in. × 12in. over a 15in. stroke, connected to the original main gear wheel through a single-reduction pinion, fitted in its place. The engine was designed to run at 280/300 r.p.m. under the normal sea speed agreed upon. The low-pressure turbine was re-bladed to meet the new steam conditions, its original gearing being left in place untouched. As the main gear wheel naturally showed more wear on the ahead driving face side of the teeth, it was reversed, and the original astern driving face of the teeth used for ahead driving.

As it was estimated that two of the main boilers would easily supply all the steam required for the full power (2,200 s.h.p.), the centre boiler was removed from the ship. The two wing boilers were fitted with superheaters, to give a superheat temperature of 600° F. at the engine stop valve. A rotary multi-stage feed pump was fitted to the forward end of the turbine, a rotary lubricating oil pump to the after end of the reciprocating engine pinion shaft, and the main condenser was converted to the regenerative type. No other changes were made in the machinery, except the re-arranging of certain of the auxiliaries.

The cross-bunker deep tank was converted into a cargo hold, increasing the cargo-carrying capacity by approximately 580 tons, and while in dry dock a solid bronze propeller was substituted for the original bronze blade built-up one; the rudder post was streamlined in the now common manner.

ABSTRACT FROM ENGINEERS LOG.

OF S.S. "ADDERSTONE"

STEAMING TIME:- 43 DAYS 10 HRS. 38 MINS. MEAN DRAFT:- FOR^{D}. 24'-7½" AFT. 25'-6½"

DATE. 1936. NOON to NOON	STEAM PRESSURES		TEMPERATURES.				VACUUM	REVS PER MIN	DISTANCE IN KNOTS			SLIP %	OIL CONS. PER DAY.	REMARKS ENGINES, BOILERS WEATHER ETC
	BOILER	H.P. CHEST	SUPERHEAT AT ENG STOP VALVE	FEED HEATER OUTLET	FUNNEL PYROR	SEA			SHIP	PROP	PER HOUR			
Dec 7th	200	190	600	230	365	48	29.2"	64.6	156	193.6	7.7	19.4	11.6	Very heavy head sea vessel plunging
" 8th	"	"	"	"	375	53	"	64.2	200	228.3	8.33	12.4	14.3	Mod: S.W. Gale & heavy sea
" 9th	"	"	"	"	"	54	"	66.4	213	235.9	8.88	9.7	14.0	" Wind heavy confused sea
" 10th	"	"	"	"	"	55	"	66.5	220	235.4	9.2	6.5	14.3	" " Confused swell
" 11th	"	"	"	"	"	59	28.9"	66.7	218	235.8	9.13	7.6	"	" " & Swell
" 12th	"	"	"	"	"	63	"	67.1	222	237.3	9.28	6.4	14.2	" " " "
" 13th	"	"	"	"	"	64	28.8"	67.4	226	237.9	9.66	5.0	14.5	" " " "
" 14th	"	"	"	"	380	74	28.5"	66.2	216	234	9.06	7.7	"	Mod: Head wind & swell
" 15th	"	"	"	"	"	75	28.4"	66.1	222	233.4	9.28	4.9	14.2	Light wind & mod: swell
" 16th	"	"	"	"	"	78	28.3"	66.1	218	233.3	9.16	6.6	"	" " " Swell
" 17th	"	"	"	"	"	80	28.2"	66	"	233	"	6.4	14.4	" " " Sea
" 18th	"	"	"	"	385	86	28.0"	65.5	221	230.7	9.24	4.2	13.9	" " " "
" 19th	"	"	"	"	"	"	27.8"	65	228	229	9.54	.44	13.8	" " " "
" 20th	"	"	"	"	"	73	28.5"	60.1	102	212.7	4.27	51.9	12.9	Fresh gale heavy sea V/L hove too
" 21st	"	"	"	"	"	82	28.1"	63.1	167	223.6	7.0	25.3	14.1	" " " Breaking sea
" 22nd	"	"	"	"	"	"	28.2"	64.1	190	226.4	8.0	16.1	13.7	Mod beam wind & sea
" 23rd	"	200	"	"	"	78	28.3"	66.2	201	233.3	8.44	13.8	"	Fresh beam wind & mod: swell
" 24th	"	"	"	"	"	81	28.0"	66	190	232.8	8.0	18.4	14.1	Strong wind & heavy swell
" 25th	"	"	"	"	390	84	"	66.4	233	233.9	9.75	—	14.4	Light wind & sea
" 26th	"	"	"	"	"	81	"	66	194	233.1	8.16	16.8	14.3	Strong wind mod: sea contrary currents
" 28th	"	195	"	"	"	"	28.1"	65.9	165	179	9.02	7.9	10.1	Fresh head wind & sea
" 29th	"	"	"	"	"	"	"	64.8	190	230.3	7.9	17.5	14.1	Strong head wind & heavy sea
" 30th	"	"	"	"	"	80	28.2"	65.6	214	231.9	8.98	7.7	"	Mod: Head wind & sea
" 31st	"	"	"	"	"	"	"	65.5	194	231.6	8.12	16.2	"	Fresh " " " " Contrary currents
Jan 1st 1937	"	"	"	"	"	"	"	63.7	184	225	7.7	18.2	"	Strong head wind. Heavy breaking sea
" 2nd	"	"	"	"	"	79	"	60.7	143	214.9	6.0	33.5	14.0	" N.E. Gale " " "
" 3rd	"	"	"	"	"	78	28.4"	60.8	145	215.4	6.08	32.7	"	Mod: Head gale " " "
" 4th	"	200	"	"	"	77	"	64.1	182	226.8	7.61	19.7	14.1	Strong head wind " " "
" 5th	"	"	"	"	"	75	"	64.7	191	228.6	8.02	16.5	"	Fresh head wind heavy swell V/L plunging
" 6th	"	"	"	"	"	74	28.5"	64.5	177	227.6	7.44	22.2	14.2	Strong wind & heavy sea V/L plunging
" 7th	"	"	"	"	"	72	"	63.2	161	223.3	6.83	27.9	14.1	Mod: Head gale heavy breaking sea
" 8th	"	"	"	"	"	70	"	60.9	100	215.3	4.14	53.5	14.0	Fresh head gale very heavy break^{g} sea V/L pl^{g}.
" 9th	"	"	"	"	"	68	28.4"	65.5	200	231.3	8.4	13.5	"	Gale & heavy swell moderating thro' day
" 10th	"	"	"	"	"	67	"	67.5	212	239	8.9	11.3	14.3	Mod: wind & confused swell.
" 11th	"	"	"	"	400	"	28.5"	67.2	206	236.7	8.66	13.0	"	" " " Heavy confused swell
" 12th	"	"	"	"	"	62	28.9"	65.9	184	232	7.73	20.7	14.1	" Gale & heavy breaking sea
" 13th	"	"	"	"	"	"	"	66.8	214	234.9	9.0	8.9	14.0	Fresh wind confused swell V/L rolling
" 14th	"	"	"	"	"	60	29.0"	67.2	218	236.2	9.2	7.6	14.2	Mod gale heavy breaking sea
" 15th	"	"	"	"	"	"	"	67.1	200	235.8	8.44	15.2	14.1	Strong gale & mountainous sea Eng^{g} racing
" 16th	"	"	"	"	"	56	29.2"	66.8	208	234.6	8.8	11.5	"	Strong wind & heavy confused swell
" 17th	"	"	"	"	"	-"-	-"-	67.7	219	237.9	9.24	7.9	"	Fresh wind & confused swell
" 18th	"	"	"	"	"	54	"	70.5	223	247.4	9.42	9.92	14.7	Strong variable winds & heavy sea
" 19th	"	"	"	"	"	50	"	70.5	225	"	9.5	9.0	"	Strong beam winds & heavy sea

DECEMBER TO JANUARY, ATLANTIC VOYAGE.

PERFORMANCE OF S.S. "ADDERSTONE"

THROUGH EXCESSIVE BAD WEATHER.

NOTE:- APPROXIMATE CONSTANT REVOLUTIONS OF PROPELLER WITH EXCESSIVE SLIP.

FIG. 6.

ABSTRACT FROM ENGINEERS LOG OF S.S. "ADDERSTONE".

Run on one boiler. Boiler: 15′ 6″ dia. × 11′ 6″ long.
Steaming time: 10 days 9 hr. 4 min. Mean draft: Forward 24′ 8½″; aft 25′ 9½″.

Date: 1934	Steam Pressures.		Temperatures.						Distance in knots.					Remarks.
	Boiler	H.P. Chest.	Super-heat at Eng. Stop Valve.	Feed heater outlet.	Fun-nel Pyro'r	Sea.	Vac-uum.	Revs. per min.	Ship.	Prop.	Per hour.	Slip %	Oil cons. per day.	Engines, Boilers, Weather, etc.
July 1st	195	185	570	230	370	57	28·75″	66	247	237	10·1	−4·0	13·2	Fine weather.
,, 2nd	200	190	590	,,	,,	,,	,,	65·1	237	236	9·71	—	13·0	,, ,,
,, 3rd	,,	,,	600	,,	360	,,	,,	65·4	246	234	10·15	4·8	13·2	,, ,,
,, 4th	,,	,,	,,	,,	,,	61	28·7″	65·5	225	233	9·34	3·4	13·0	,, ,,
,, 5th	,,	,,	,,	,,	,,	,,	,,	66	234	234	9·75	—	12·7	,, ,,
,, 6th	,,	,,	,,	,,	,,	65	28·6″	66·3	239	233	10·0	−2·5	,,	,, ,,
,, 7th	,,	,,	,,	,,	,,	,,	,,	66	227	232	9·58	2·1	12·5	,, ,,
,, 8th	,,	,,	,,	,,	,,	76	28·4″	65·8	194	236	8·14	16·3	12·8	Fresh head wind and sea.
,, 9th	,,	,,	,,	,,	,,	74	28·5″	67	231	236	9·73	2·1	12·9	Fresh head wind & sea. Moderating to fine.
,, 10th	,,	,,	,,	,,	,,	,,	,,	67·6	221	237	9·26	6·7	,,	Fine.

Total fuel consumed on passage, 128·9 tons. Consumption tons per day, 12·5 tons. Average speed, 9·58 knots. Total distance run, 2,387 miles. Mean r.p.m., 66·1. Average slip, 1·2 per cent.

FIG. 6A.

When the conversion was completed the machinery was given the usual mooring trial and the vessel placed in commission, since when she has been in constant service, mostly trading in the North Atlantic.

In regard to the main purpose for which the combined engine was introduced, namely to demonstrate the economy of such a combination, expectations have been fully realised.

The following is a summary of two logs of the s.s. "Adderstone" before and after conversion, taken under nearly similar conditions of loading and weather, which was mostly bad with strong head winds and seas.

Voyage:	Before conversion. Buenos Aires-Liverpool.	After conversion. Montevideo-Sharpness.
Draft	24ft. 10½in.	25ft. 6¼in.
D.W. cargo	7,001 tons	7,705 tons
Speed port to port ...	9·27 knots	9·21 knots
Oil per day	28·7 tons	13·82 tons
,, ,, s.h.p./hr. ...	1·165lb.	0·665lb.

$$\text{Saving} = \frac{1{\cdot}165 - 0{\cdot}665}{1{\cdot}165} = 43 \text{ per cent.}$$

For the periods covering the winters 1934/35 and 1936/37, the vessel was trading in the North Atlantic and encountered most of the severe gales which were responsible for so much loss and damage

ABSTRACT FROM ENGINEERS LOG OF S.T. "WHITE PIONEER".

Date.	Steam Pressures.		Vacuum.	Temp. at Engine Stop Valve.	Prop. r.p.m.	Speed.	Trawling or steaming.	Position of Links.
	Boiler.	H.P. Cylinder.						
Feb. 19th	210lb.	170lb.	28″	470°	108	10	S.	Links ¼in. Wheel out.
,, 20th	210 ,,	,,	,,	490°	74		T.	Links ¼in.
,, 21st	220 ,,	160 ,,	28½″	480°	76		T.	Wheel one turn.
,, 22nd	215 ,,	165 ,,	28″	470°	76		T.	,, ,, ,,
,, 23rd	210 ,,	180 ,,	,,	480°	104	10·6	S.	Links.
,, 27th	210 ,,	170 ,,	,,	470°	76		T.	Links ½in. Wheel out.
,, 28th	210 ,,	160 ,,	,,	490°	76	3	T.	Links ½in. Wheel one turn.
Mar. 1st	215 ,,	180 ,,	,,	480°	83	4·2	T.	Links ½in. Wheel out.
,, 2nd	215 ,,	120 ,,	28½″	490°	75	5	S.	Links ½in. Wheel one turn.
,, 3rd	220 ,,	180 ,,	28″	480°	81		T.	Links ½in. Wheel out.
,, 4th	215 ,,	,,	,,	470°	80		T.	Links ½in. Wheel one turn.
,, 5th	215 ,,	,,	28½″	480°	105	10·2	S.	Links ¼in. Wheel out.

FIG. 7.

to shipping. During one of the heaviest gales, as will be seen from the accompanying log, the propeller slip reached the abnormal figure of 53·5%, the engines having been kept at the original setting with the stop valves full open. It will, therefore, be appreciated that the engine has undergone a thorough testing.

Service Performance.

To apply the test engine to a commercial use, and with the object of proving the reliability of the engine in the roughest of service, and to give the gears and machinery the most severe testing and roughest treatment it was considered possible to obtain in any kind of vessel, the gears and turbine of the test set, with a new single-compound engine, were installed in a standard size trawler 125ft. 6in. long × 23ft.4in. beam × 13ft. 6in. depth, built for the purpose, the machinery being installed in the same position and space as is occupied by a standard trawler triple-expansion engine of 500 h.p.

This vessel has now been in service two years, trawling off the Faroe Islands, encountering the very severe weather experienced during the last two winters, and the machinery has functioned perfectly and withstood the very severe treatment with no indication of trouble. The machinery has been handled the entire time by non-certificated men. She has proved a very satisfactory vessel for trawling, on many occasions having demonstrated her ability to keep trawling because of the non-racing of the engine, when the other trawlers have had to abandon trawling owing to the severity of the weather. This lack of racing also accounts for her staying at sea when other vessels have had to make for shelter.

From the above it will be appreciated the machinery has had the most thorough and severe testing, and from the fact that no astern element is fitted to the turbine it will be realised that the gears have throughout been subjected both to extreme loads and the most severe treatment. When opened out recently for examination, the gears and the machinery were found to be in excellent condition.

General.

The principal features of the machinery have been profusely dealt with in the technical press and need not be referred to here. Some departures from general marine design may, however, be noted.

It has already been indicated that the compound engine is a more efficient unit than the triple-expansion engine to combine with an exhaust turbine, the double compound engine with two h.p. cylinders and two l.p. cylinders with the large reciprocating weights in the centre giving the desired condition of almost perfect balance, an essential feature where high revolutions are concerned. These two facts led to the author adopting the double-compound engine as a standard for merchant ships using powers up to, say, 5,000 i.h.p., while for smaller powers up to, say, 800 i.h.p., the single-compound is adopted, as exemplified in the trawler engine.

From the superintendent's and shipowners' point of view, the criterion of performance of any engine is its earning power, and whilst low consumptions are a commendable feature, upkeep costs must also be taken into consideration. During the initial investigations, this point received special attention. It was recognised that the reciprocating engine, turbine and gears were definitely established, and their reliability unquestioned. There only remained the combining of them in a manner to obtain the best features of each. It is very satisfactory to record that expectations have been fully realised in this regard, both from a mechanical and thermal standpoint.

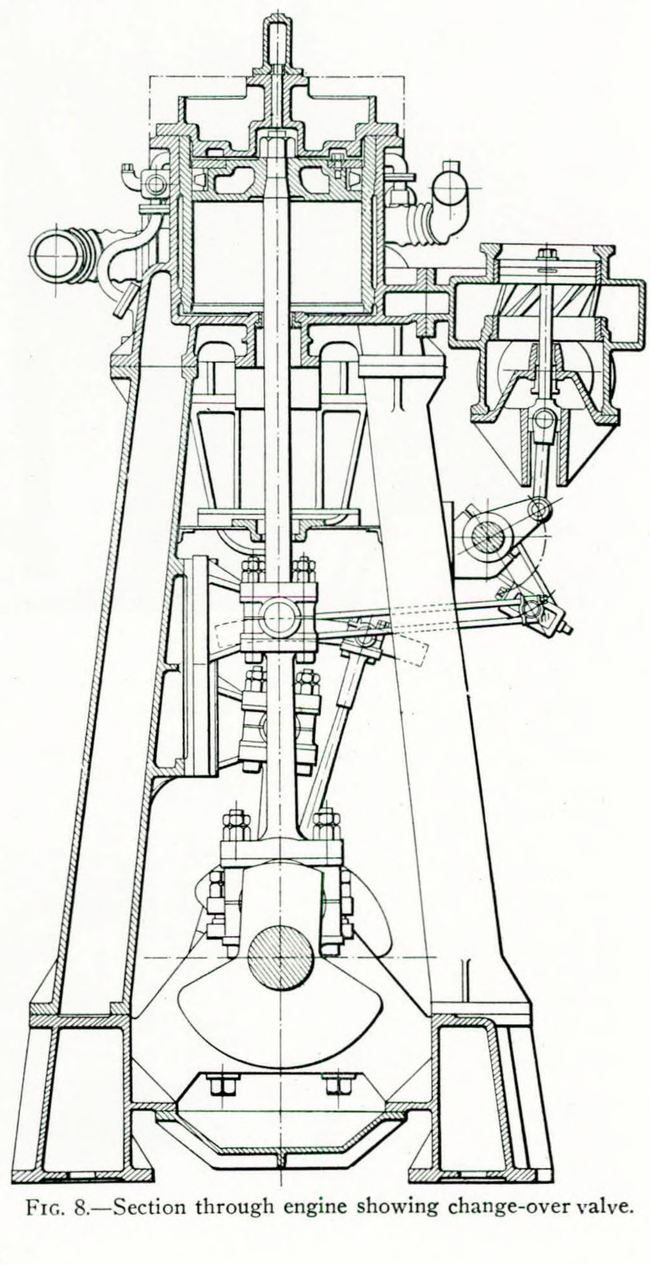


FIG. 8.—Section through engine showing change-over valve.

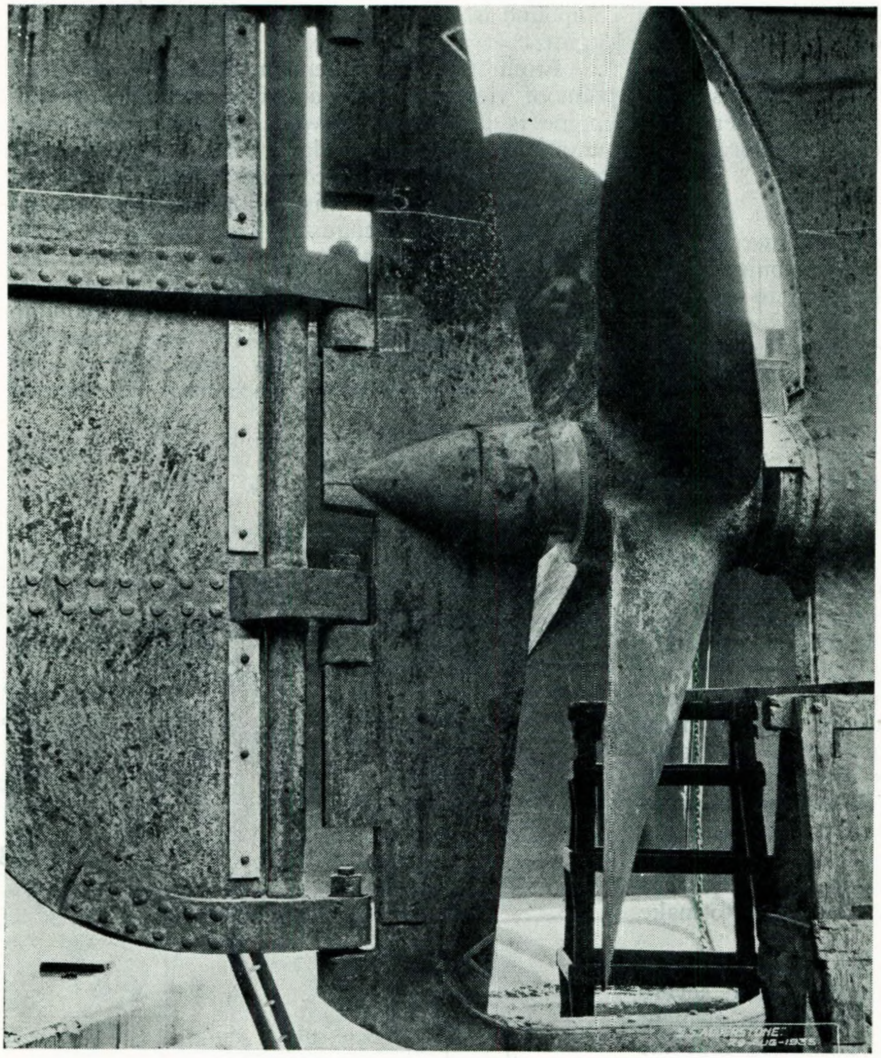


Fig. 9.

Regarding the question of upkeep, the s.s. "Adderstone", which was the first vessel to be fitted with the engine, has now been running continuously for three years since the machinery was installed in March, 1934, and has steamed over 160,000 miles, and up to the present has not required any repairs to the main engines, her repair bills for the entire period for the engine department being practically nil.

Recently, while discharging in a home port, her engines were opened up for examination. Her gears revealed no signs of wear whatever, the surface of contact on the teeth being uniform and showing a high polish. The cylinders, valve liners, pistons and valves were in excellent condition. The main bearings and bottom ends of the reciprocator showed only a slight wear amounting to ·001in. to ·0015in., which can be regarded as negligible as it was too little to allow of the removal of even the thinnest liner. This remarkably small wear is entirely attributable to all running parts being lubricated with oil under a pressure of 8 to 10lb. Up to date it has not been possible to take even the thinnest liner out of any of her bearings or guides.

A feature of the engine design is the facility with which the reciprocating engine can be disconnected from the turbine. When it is desired to operate with the reciprocating engine alone, it is only necessary to disconnect the pin connecting the weigh shaft arm with the change-over valve, and lock the valve in position to exhaust the steam into the condenser direct. 65 per cent. of full power can then be developed by the reciprocating engine.

The engine is extremely simple to handle, and very quick in responding. All valves controlling the engine exhaust or steam to the astern turbine are

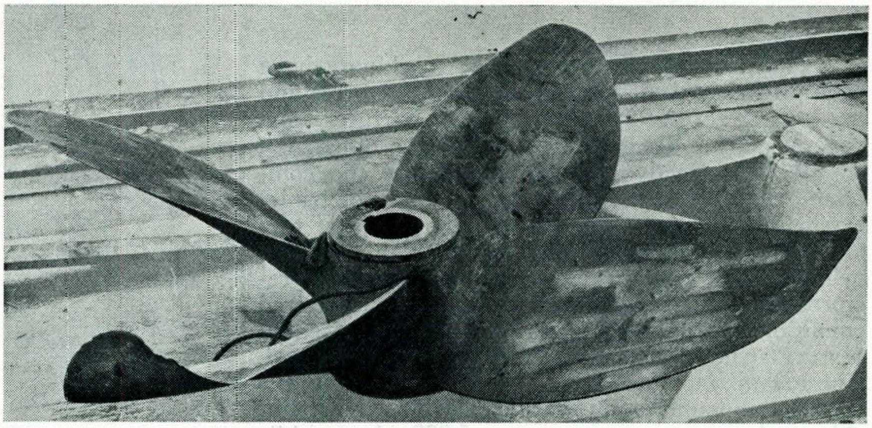


Fig. 10.

mechanically operated from arms on the weigh shaft, the valves opening as the engine is being reversed, giving a positive and simple arrangement which has given entire satisfaction, and removes the objection of valve lag usual with fluid operated valves.

The engine is extremely sensitive and rapid in its response to any variation of the links or stop valve. On occasions the main shaft revolutions have been maintained as low as three per minute.

Reversal from full ahead to full astern can be effected in 17 seconds without increasing the normal loading of the gears, and in emergency this time can be reduced to 12 seconds without exceeding the overload margin.

As evidence of the manœuvring qualities of the engine, the author cites one charter covering four return voyages North America to Northern Spanish ports. During each voyage the vessel entered and left 11 ports under her own steam, on occasions doing two ports in one day.

The inertia effect of the large revolving masses of the turbine and gears prevents any excessive increase in propeller revolutions when the vessel is pitching, with the result that a higher average speed is maintained during bad weather.

The following tabulation for a 1,850-h.p. vessel operating normally at between 1,450 to 1,500 i.h.p. indicates the small variation in steam consumption with the varying powers, and demonstrates the wide range of economical speeds obtainable with the engine.

FIG. 12.

POWER AND WATER CONSUMPTIONS.

I.h.p.	Water to main engine.	Water per i.h.p. main engine.	Auxiliary water.	Total water.	Water per i.h.p. all purposes.
1,450	10,000	6·9	2,200	12,200	8·41
1,745	12,400	7·1	2,600	15,000	8·60
1,850	13,450	7·27	2,925	16,375	8·85

Figs. 9, 10, 11 and 12 are photographs of a damaged propeller and engine pinion of the "Adderstone" taken to demonstrate the condition of the teeth after the severe shock caused by the propeller striking an obstruction, which bent the blade over, causing the blade to strike the rudder post fin, and cutting through the material as shown. The material of the fin is ⅜in. thick plate.

Fig. 13 shows a general arrangement of the machinery for a 1,800 h.p. installation as fitted in the vessels completed to date.

The accompanying extract is taken from the log of the s.s. "Llanashe". This vessel has now completed her first round voyage, Tyne to Gibraltar loaded, Gibraltar to River Plate in ballast, and River Plate to Avonmouth loaded, at an average speed for the round voyage of 10·05 knots, and an average daily consumption of 15 tons. The

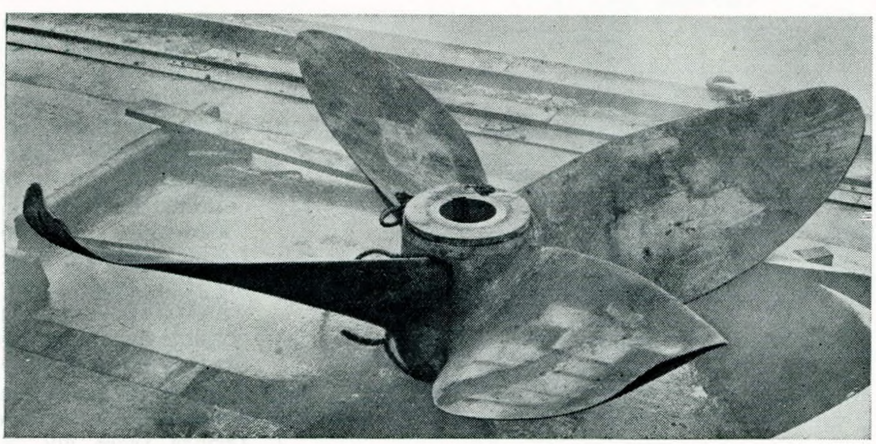


FIG. 11.

total coal burned for the round voyage, including all sea and port consumptions, was 973 tons of unscreened north country coal. The vessel commenced her voyage from the Tyne with 1,040 tons of bunkers on board, and had 67 tons on her return, after berthing at Avonmouth.

The following tables give the general performance of the several vessels completed to date, and are of special interest in view of the high fuel coefficient which has been consistently maintained.

S.s. "Llanashe".
Load Ship Trials.

Displacement	12,240 tons
Speed	10 knots
S.h.p.	1,230
Consumption	13·25 tons per day
Coal per i.h.p. all purposes	·91b.
Water per i.h.p. all purposes ...	8lb.
Fuel coefficient...	40,000

Fuel Co-efficients.
$\text{Speed}^3 \times \text{Displacement}^{\frac{2}{3}} \div \text{tons per day}$.

Oil.
S.s. "A".

22. 7.36-17. 8.36.	Loaded co-eff.	41,500	Equiv. coal	29,000
19. 8.36-22.11.36.	„ „	34,500	„ „	24,200
3.12.36-10.12.36.	„ „	41,250	„ „	28,700
19.12.36-19. 1.36.	„ „	30,800	„ „	21,600
23.10.36-22.11.36.	„ „	52,000	„ „	36,300
„ „ (run on one boiler)	„	47,500	„ „	33,200

S.s. "B".

31. 8.36- 9. 9.36.	Loaded co-eff.	58,500	Equiv. coal	40,800
9. 9.36-15. 9.36.	„ „	47,500	„ „	33,200
17. 9.36-22. 9.36.	„ „	43,000	„ „	30,000

S.s. "C".

3.10.36-20.10.36.	Loaded co-eff.	51,500	Equiv. coal	36,000

Comparative Performance.

	S.s. "Adderstone"	*S.s. "Llanashe"*
S.h.p.	1,840	1,230
Boiler pressure	200lb.	230lb.
Temperature °F. ...	590	660
Feed °F.	220	220
Vacuum	28½in	28¾in.
Heat drop	403	430
Lb. per theor: h.p. $\left(\frac{2,545}{\text{B.T.U.}}\right)$	6·3	5·9
Coal per hour		1,300lb.
Water main engine ...	15,500lb.	9,600lb.
Water per s.h.p.	8·45	7·8
Water per i.h.p.	7·6	7·0
Efficiency % to s.h.p. ...	75	75·64
Efficiency % to i.h.p. ...	83	84

It will be noted that the aforementioned results are being obtained in everyday service; and they possibly constitute the best, and the consumption the lowest, yet obtained with steam propelling machinery of the average horse powers employed in cargo liners and tramp vessels.

It will be appreciated that in the new vessels now in service all the auxiliaries are independent. In the further installations now under construction, it is intended to drive several of the auxiliaries direct from the main engines with the object of further reducing the all-purposes consumption. The amount of auxiliary exhaust steam required for heating the feed-water, of course, limits the number driven off the engine.

The author looks forward with confidence to the day when a satisfactory material (obtainable at a commercially satisfactory price) will be produced, to enable engineers to take advantage of the gain to be obtained by using the waste heat in the funnel gases for heating the feed water, when it would be possible even with our present-day knowledge to produce one horse power in marine service from ¾lb. of coal and under for all purposes.

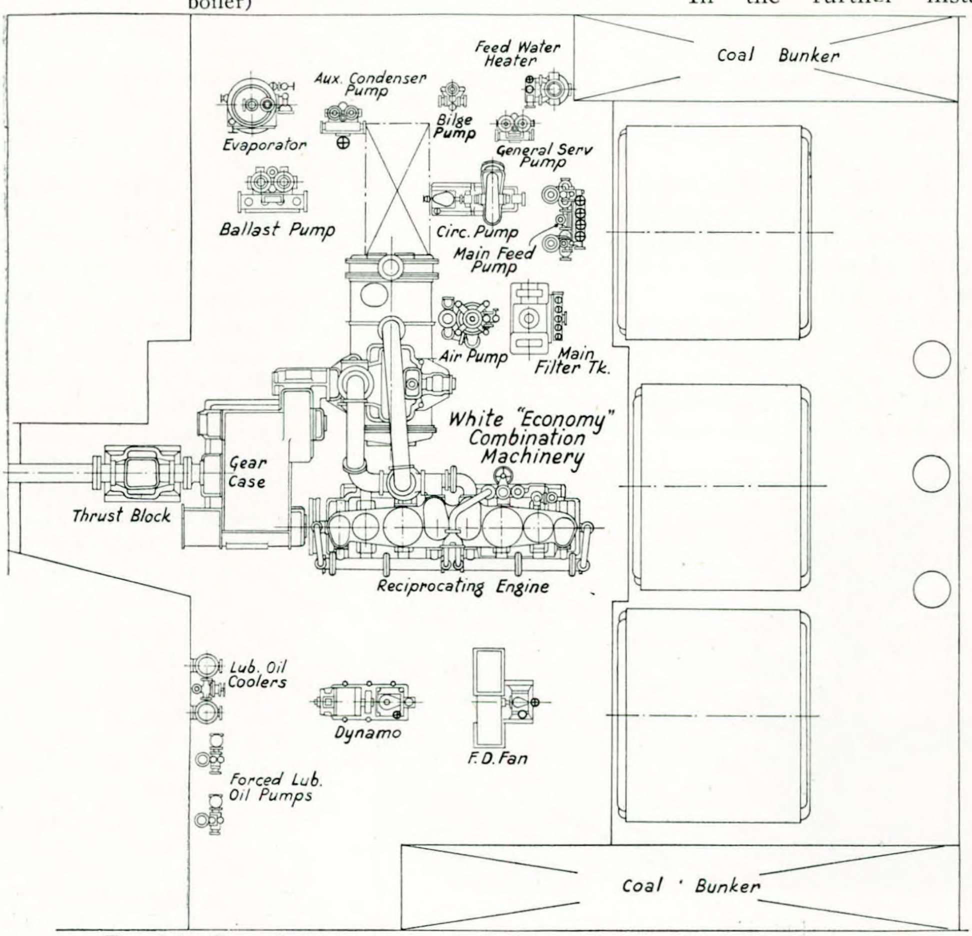


Fig. 13.—General arrangement of machinery for a 1,800 h.p. installation.

Discussion.

Mr. Harry Hunter (Member), opening the discussion, said that the author stated he had experienced no trouble when using superheats of 700° to 720° F. Was that the temperature on admission to the cylinder or the temperature before the reheater, and was the experience obtained in regular service?

With regard to the reheating system, it seemed to be on parallel lines with that disclosed in Mr. John Neill's paper read before The Institute in 1933, when he described the North Eastern Marine Engineering Co.'s patented reheater engine.

The author's reference to the high efficiency of the high-speed reciprocating engine rather gave the impression that he considered that the high-speed engine was of higher efficiency than the low-speed engine. In his researches in connection with a paper read two years ago before the North East Coast Institution of Engineers and Shipbuilders, for which it was necessary to obtain data on reciprocating engines over a period of fifty years, he (the speaker) had been rather surprised to find that the most efficient reciprocating steam engines were large pumping engines (such as those at Hampton Court) running at 25 to 30 r.p.m. The high-speed engine was undoubtedly efficient, but the large low-speed engine was at least equally efficient, and it was not correct to assume that the high-speed steam engine had a greater efficiency than had the low-speed engine. Indeed, if the speed was such as to cause excessive dynamic forces, then the high-speed engine was definitely less efficient than was a lower speed engine; the engines described by Mr. White, however, did not seem to run at a speed sufficiently high to prejudice economy.

The speaker suggested that the advantage of the geared high-speed engine in this particular application rested on its reduced weight and also on its more even turning moment which permitted it to be coupled to the turbine without any damping device.

Fig. 5 showed a torque variation with revolutions and the vibration torque was also dotted in. These were critical speeds due to the propeller impulse. The speaker considered that the propeller excitation rarely gave torsional oscillation. Were these torsional vibrations ascertained at sea or calculated? They seemed to be calculated with possibly an insufficient allowance for mechanical and metallurgical damping.

On page 93 the times for reversal from full ahead to full astern were given. He presumed these were given to reassure them when the engine had to slow down the shafting system against the flywheel effect of the turbine. Were the times given for reversal from full ahead revolutions to full astern revolutions? There was no doubt that, as Mr. White said, the flywheel effect of the turbine was very helpful in maintaining propeller speed in weather so heavy that the propeller was lifting. There was, however, a liability attached to this feature at the moment when the propeller after being partly out was suddenly submerged; the turbine with its inertia tended to keep up its speed, but in so doing the torque in the system was increased above normal—possibly to the extent of 50 or 75 per cent., depending on the degree and suddenness of the re-immersion of the propeller. The liability could of course be met by including a sufficient margin in teeth loads, etc., and certainly, according to the paper, Mr. White's experience with the gears seemed highly satisfactory.

On the same page the i.h.p. for the water consumption was given. How was this arrived at? It was an unusual thing for a turbine to have a specific consumption reducing with reducing power. He did not think that had been published before. The i.h.p. of the turbine must have been deduced by some indirect means. What was that means?

On page 94 the s.h.p. was given. Was that by torsionmeter or was it assessed in some other manner?

Had a noisy propeller ever been experienced with the combination machinery? There had been a recent paper on this subject, but the author of that paper did not give any cases of ships fitted with combination machinery.

Mr. G. Ridley Watson, B.Sc. (Member) said that the author was to be congratulated on having the courage of his convictions in carrying out, at his own expense, what were really full-sized experiments with the White combined steam engine, first of all in the "Adderstone" and then later in the trawler "White Pioneer". The success of the operation of these two vessels had fully justified the author's early belief in the proposed arrangement of machinery.

Could the author let them have the percentage saving in machinery weight in a vessel of the "Llanashe" type, compared with ordinary modern triple-expansion machinery fitted for using superheated steam?

He thought it was generally agreed that much criticism that was levelled against the White combined steam engine was in connection with the gearing, and he was very pleased to note that the author was now able to show, as the result of the operation of the "Adderstone", that there was nothing to fear in this direction.

It would be interesting to have the author's views on the cause of gearing troubles in some of the earlier double-reduction geared turbine sets, as he believed this had been considered to be largely due to the inefficient h.p. turbine, and the more or less unknown quantity of power developed by that unit.

On page 93 the author referred to the inertia effect of the large revolving masses of the turbine and gears preventing any excessive increase in propeller revolutions when the vessel was pitching. Did this not result in additional loads on the gearing, and if so, it was satisfactory to note that the gearing had been able to withstand all such severe loads.

It had been noted that recently the Coal Utilisation Council had been studying the problem of increasing the use of coal for marine purposes, and were strongly advocating the water-tube boiler and mechanical stokers for cargo ships. Presumably there was no reason why this type of boiler and stoking plant could not be utilised in conjunction with the White machinery, and he would value the author's remarks in this connection.

Mr. W. A. Christianson (Member) said that although the paper was on the White engine, it would not, he thought, be out of place to mention that the firm with which he was associated, which developed a special type of double-compound engine over ten years ago, saw the special suitability and advantages of the double-compound type for combination with an exhaust turbine, and gave prominence to this at the North East Coast Institution's symposium on special types of reciprocating engines in 1931 and subsequently in the technical press in an article entitled "Double-compound Engines in conjunction with Exhaust Turbines".

As the early published information on the White system prior to the above-mentioned publicity showed triple-expansion engines only, it would appear that the speaker's firm had had something to do with the adoption of the double-compound type in the latest form of the White system.

With an exhaust turbine there was no necessity for three stages of expansion in the reciprocating engine. Compound expansion in the latter was quite sufficient, and furthermore the double-compound engine carried with it many other practical advantages.

Like a previous speaker, he desired to know how the i.h.p. and b.h.p. were arrived at. In the White system, as the reciprocating engine ran at comparatively high revolutions, it would not be easy to indicate, if at all, and furthermore, as the exhaust turbine was permanently coupled and not declutchable, the total equivalent i.h.p. could not be arrived at as in systems with declutchable turbines, e.g., the Bauer-Wach system. In the latter, the total equivalent i.h.p. was obtained by indicating the engine portion both with the turbine in and out of action, and then by calculation from the corresponding revolutions. It would be appreciated that the correctness of much data in the paper was dependent on the correctness of the quoted i.h.p's and b.h.p's in the first instance.

Referring to the table of power and water consumption on page 93, the auxiliary water consumption appeared high in relation to the consumption of the main engine. It might be that the total water consumption was correct and that any inaccuracy was just a case of the division of this into the two classes.

Mr. J. McAfee (Member) said that the author must be heartily congratulated on designing an engine which in average service propelled a vessel of the size of the "Llanashe" at a speed of 10 knots on a consumption of only 15 tons of coal per day. This figure was the more remarkable when it was remembered that steamers of similar size built in the immediate post-war period were burning double this amount. The results shown by the author were indicative of the great strides made in marine steam engineering in recent years.

In designing his engine the author had rightly extended the temperature in preference to the pressure range in his search for economy. No doubt the small size of the cylinders—the h.p. cylinder in the "Llanashe" was only 10½in. diameter—was largely responsible for the absence of trouble when using a steam temperature in the neighbourhood of 650° F. It would be of interest, however, to know why piston valves had been used instead of some form of drop valve, as these would require less lubrication. They would also result in a shorter engine, which might be of some importance since it appeared from the plan of arrangement in the "Llanashe" that the length of the reciprocating engine was the main factor in determining the length of the engine room.

This question of lubricating oil was of great importance in an installation using superheated steam, where from 1 to 2 pints per 1,000 i.h.p. might be fed into the cylinders per 24 hours. The utmost precautions must be taken to prevent this oil entering the boilers.

In a number of cases of vessels using superheated steam, trouble had occurred with the furnaces which had deflected badly in service—sometimes in ships only a few months old and where there was no question as to the suitability of the scantlings to meet the working pressure. The trouble had usually been traced to the presence of oil in the boilers, but it was usually difficult to detect this for the very reason that it was burnt off when the overheating took place, though sometimes a trace had been found on the underside of the lower rows of tubes. It was noted that in this engine an oil filter was placed in the reciprocating engine exhaust, and the plan showed the usual filter tank. It would be of interest to know if any other precautions were taken, and if a trap was fitted in the auxiliary line.

The author referred to the absence of propeller racing due to the inertia effect of the turbine and gearing, and this was an argument in favour of installations where a reciprocating engine and turbine worked on the same shaft. It must not be overlooked, however, that this damping effect had to be transmitted through the gearing which might be subjected to severe loading when the vessel was pitching heavily in a seaway, but the particulars given in the paper suggested that these factors had been fully taken into account.

He would suggest that the author's remarks on page 88 regarding the relative importance of out of balance forces and couples required some qualification. Their effect in producing vibration was dependent usually on the position of the engine with respect to the nodes about which the vessel vibrated. An out of balance force had its maximum effect when located midway between the nodes about which the vessel vibrated, its influence being a minimum when applied directly at the nodes. A couple on the other hand had its maximum effect when applied at the nodes.

Had the author carried out any comparative tests in order to ascertain the economy due to the fitting of reheaters. The improved performance to be obtained by the use of drier steam was of course well recognised, but it must not be overlooked that the higher adiabatic efficiency ratios in the l.p. cylinder and turbine were due not to some inherent characteristic in the engine itself but to the addition of heat to the steam at an intermediate point in the expansion. This was really a denial of the Carnot principle to which the author referred in his opening remarks.

In theory the reheat cycle was not an efficient one, as a glance at the entropy diagram on page 89 indicated that more heat per lb. of steam was lost to the condenser when the reheaters were in use. Theory and practice did not always agree—or perhaps it should be said that their theory was incomplete—but as long as their practice led them to the results which the author had shown in this paper it was obvious that they were on the right path.

Eng. Capt. H. B. Tostevin, D.S.O., R.N. ret. (Member) said that he had been privileged by the kindness of Mr. White and the respective shipowners to attend the trials of some of the new ships fitted with the installation under review, and discussions with certain superintendents had elicited criticisms which Mr. White might care to deal with at this meeting. There was an impression that the installation represented an increase in complexity which was not warranted by gains in other directions and they required further data regarding improvement in economy. Increase in complexity was somewhat difficult to understand seeing that the Bauer-Wach installation with its reciprocating engine, turbine and gears was accepted with complacency. The heat interchangers were also looked upon as another complexity, although it was not considered that these should give trouble. It must be admitted that the first sight of an engine room in these ships from above gave the impression of an inordinate number of steam pipes and might prejudice the conclusion arrived at.

Regarding complexity, it was interesting to see that in future installations several of the auxiliaries would be driven direct from the main engine. The speaker was hopeful that this modification would eliminate a disadvantage open to criticism. He thought he was right in saying that in the present installations there were 8 reciprocating auxiliaries using superheated steam. Each of these had a mechanical lubricator pumping oil into the steam, and with the best of designs this must result in a quantity of oil out of all proportion to the power having to be dealt with in the condensate, and probably as much would be used in the auxiliaries as in the main engine.

The author would probably prefer criticism to adulation at this meeting, and he (the speaker) would like to be clear on one or two points or omissions in the paper. Fig. 1a showed the improved turning effort of the reciprocating engine using reheat, and it would be of interest to know if this was just accidental to the design and was not put forward as an implied advantage of reheat in general. In Fig. 3 attention was called to the ideal entropy diagram for the cycle and it was questioned whether for completeness the drop of temperature from the initial steam condition to the state at entry to the h.p. cylinder should not also be included. To avoid any misconception it also appeared desirable to state whether it could be accepted as roughly to scale as regards the heat interchanges, the values of which were nowhere referred to in the paper.

While not questioning the efficiency ratio figures given on page 87, it would be interesting to know if they were the results of direct experiments and had been confirmed in service installations.

The results from the s.s. "Adderstone" before and after conversion were very interesting, but one was inclined to question whether the machinery before conversion was in a comparable condition with that afterwards in view of the overhaul and the new machinery items necessarily fitted. The results with this vessel during the three years operation had been known to be very satisfactory and to have strengthened the case for further installations. The absence of any repairs to the main engines was referred to on page 92. Did this cover renewals of piston rings, as if such had been unnecessary it confirmed that fewer difficulties attended the lubrication of the smaller type reciprocating engine used?

On page 91 the author stated that it would be realised the gears in the "White Pioneer" had been subjected to extreme loads and severe treatment because no astern element was fitted to the turbine. While the implication behind this statement could be guessed at, an amplification appeared desirable to clarify this point. In another part of the paper a reference was made to the overload on the gears not having been exceeded, and it would be interesting to know if the arrangement called for a greater or less allowance for overload than was current in normal turbine or other l.p. turbine drives.

During discussions on this installation it was found that potential users and engineers interested generally requested some idea of the lubricating oil consumption and perhaps Mr. White would give some information on this feature. It was thought also that in the plan of the installation shown in Fig. 13 a lubricating oil purifier should be indicated.

Mr. John Reid (Visitor) said that he had been associated with the author in the past when Mr. White, at New York, was turning out with somewhat primitive resources oil burning equipments for ships of considerable number and size for which the speaker had been furnishing the forced draught gear.

The author had now gained prominence in another field and there was little to criticise in the White combination machinery as described in the paper, particularly as all reference to the boilers had been left out. From the particulars which had been published, the boilers seemed to be on the small side, although the machinery certainly took an extraordinarily low amount of steam per h.p. However, the plant was designed to burn both coal and oil, and if the furnace was small there might be difficulty in getting proper combustion in the furnace with coal.

The author stated that he had experienced no trouble when using superheated steam at 700 to 720° F. It was a mystery how superheat at such a high temperature could be fairly taken from a Scotch boiler, and it could only be obtained at the expense of some difficulty with the evaporation. Had there not been some trouble with the superheat equipment in these ships? He had heard that the trawler fitted had given a little trouble when manœuvring.

The author several times used the expression "tuning". Mr. White had evidently made a useful study of this subject and applied it to his work. The speaker, however, had difficulty in assuming that all was well, because the machinery side could not be tuned up without the boiler also being tuned. Mr. White being an expert on oil burning would be interested to know that the speaker had been trying to tune up oil combustion in Scotch boiler furnaces. His object was to obtain the highest possible boiler economy for the benefit of what happened on the other side of the screen bulkhead. He therefore regretted that the boiler side had not been dealt with in the paper.

From Fig. 13 it seemed again that the boilers were a little small. They would notice how the main boilers looked not much bigger than the donkey boiler. Would the author kindly state the size of the furnace diameters; he was interested in this particularly with reference to improving coal burning efficiency.

He knew that the "Llanashe" on coal was doing the work which the author claimed. In spite of these successful developments, however, it should be remembered that the same type of ship with a comparatively simple "up and downer", reasonable superheat of 600° F., average Welsh coal, no auxiliary turbines and ordinary auxiliaries, had been in service for years at 9·9 knots loaded on a consumption of no higher than 16 or 17 tons per day. That did not detract in the least from the author's work and figures, but it did show that in order to get one or two tons off the consumption very radical departures from the normal dependable and lower-priced machinery had to be made.

With regard to the final paragraph of the paper, he sincerely hoped that a metal would never be discovered which would fill the chimney with any more gadgets. There were far too many already, and they could not be properly looked after. For a trifling economy the engineers were given a great deal of trouble and worry, and it should be remembered that a breakdown which held up the ship for weeks would offset this trifling economy for quite a long period.

On the proposal of **Mr. Wm. McLaren** (Member) a vote of thanks was heartily accorded to the author.

By Correspondence.

Mr. J. Hamilton Gibson, O.B.E., M.Eng. (Vice-President) wrote that the main interest of this paper from the shipowner's point of view was the possibility of still further economy in fuel consumption.

For a small engine, such as that in the s.s. "Llanashe", of only 1,230 s.h.p. Mr. White returned the remarkable figure of 0·9lb. of coal per i.h.p. for all purposes. The hourly consumption was given as 1,300lb. and $\frac{1,300}{0 \cdot 9} = 1,445$ i.h.p. Now $\frac{\text{s.h.p.}}{\text{i.h.p.}}$ i.e. $\frac{1,230}{1,445} = 85$ per cent. mechanical efficiency. But the author told them that 90 per cent. or slightly more was a usual all-round figure: and $\frac{1,230}{0 \cdot 9} = 1,365$ i.h.p., which gave 0·95lb. per i.h.p./hour instead of 0·9—an increase of $5\frac{1}{2}$ per cent.

In the absence of reliable figures for i.h.p. one must be careful in assessing consumption ratios, and the above short analysis made one doubt the figure of 0·9 given in the paper. At the same time one must admit that anything less than 1·0 (and there seemed to be no doubt that the White combination machinery had attained that economy) was a wonderful achievement for such a small engine, and that larger installations would show correspondingly better results.

The effect of size and, of course, higher initial pressure was strikingly illustrated by the Battersea power station. At a recent Coal Utilisation Conference it was stated that the consumption there was as low as 0·9lb. *per kW.-hour.* Referring that figure to i.h.p. one got $0 \cdot 9 \times \cdot 746 = 0 \cdot 672$. In terms of oil fuel that came very near to good Diesel performance.

The author's final paragraph breathed a fine spirit of optimism in visualising $\frac{3}{4}$lb. of coal *and under* for all purposes. There could be no doubt, however, that a considerable reduction in coal consumption was quite possible on the lines indicated, viz., by utilising the waste heat in the funnel gases for feed heating. Their American friends had been doing this for years, and it was understood that already over 60 steamers in the U.S.A. Navy and Mercantile Marine were now fitted with Foster gilled-tube economisers in the uptakes.

The author deserved every credit for his untiring efforts to make engineers realise the importance of arresting heat leakage and delaying condensation in the steam cycle, and had shown that without going in for high-pressure water-tube boilers, elaborate bleeding systems, etc., there were considerable running economies still available for exploitation.

The Author's Reply to the Discussion.

The Author, in reply to Mr. Hunter, said that the temperatures given were temperatures recorded in the h.p. steam chests during service running.

The author still maintained that the high-speed engine was of higher efficiency than the low-speed engine, and did not agree with or accept the view that large low-speed engines were even equally efficient. He would point out in support of his contention that these low-speed pumping engines were proved to be the most benefited by steam jacketing, which reduced the condensation loss resulting from the interchange of heat between the steam and the cylinder. From this point alone, the high-speed engine must conduce to economy of steam, as there was less time for the interchange of heat between the steam and the cylinder. The thermal efficiencies of well over 90 per cent. which were being obtained in both the high-pressure and low-pressure cylinders of high-speed non-condensing engines had never been attained in slow-running engines to the author's knowledge under equal steam conditions.

As marine practice was not likely to necessitate such excessive high-speed engines as would cause excessive dynamic forces, there was no necessity to discuss this question.

Reheating as described in the paper and applied in the author's engine was quite old, and had been common knowledge for a number of years. Among various citations, reference to it might be found in Kent's Engineers' Handbook and in D. A. Low's book on "Heat Engines".

Fig. 5 applied to the engine shaft system and not the propeller shaft, and it was intended to demonstrate that gears could be used in conjunction with a reciprocating engine without any harmful effect. The "Adderstone" was, however, examined from the aspect of torsional oscillations due to the propeller, and it was found that the anticipated results were obtaining in service; the excellent condition of the teeth of all the gear wheels and pinions after three years' service running also gave definite practical evidence on this point.

With reference to the torque in the system when the propeller after being partly out was suddenly submerged, the figures cited by Mr. Hunter were given by the author during the discussion, and of course applied to all geared jobs, owing to the inertia stored in the gears and turbine. The author desired to draw attention to the extent of the increase in torque in the line shafting of any vessel under certain conditions of draft, and particularly geared engine vessels, with the vessel pitching heavily.

The author was at a loss to know what indicated horse power for the water consumption was referred to, as no separate figures were given for turbine consumption. It might, however, be stated that the consumption of a turbine would reduce with reducing power, if the vacuum into which it was exhausting was increased.

The shaft horse power referred to was taken and agreed from two torsion meters, one of which was a Siemens Ford sight reading, and the other a continuous reading power meter.

In the vessels fitted so far with the White engine, a noisy propeller had not been experienced. The author would not, however, consider this was because of the type of machinery.

In reply to Mr. Watson, the saving in weight with the White machinery compared with the ordinary modern triple-expansion machinery was approximately 20 per cent. As explained during the oral discussion, the author considered that the probable cause of a large percentage of the troubles experienced with early double-reduction marine gears was that the extent of the increase in the torque with the vessel pitching heavily, and when the propeller was quickly immersed after it had been out of the water, was not fully recognised. The continuous recording power meter recorded this increase of torque, and indicated that it increased by over 60 per cent. on the vessel in question, which was in ballast trim and pitching heavily. This condition would not, of course, exist when the vessel was fully loaded. To take for example an engine designed for 1,800 i.h.p. at full power, when in ballast and developing 1,400 i.h.p. the increase in torque on the gears with the vessel pitching heavily and the propeller being suddenly immersed after emerging, would subject them to over 2,200 i.h.p. or 20 per cent. above their designed load assuming that a 10 per cent. to 15 per cent. margin only had been allowed for in their design, and as there was little doubt that the l.p. turbine would be doing well over 50 per cent. of the total power, the l.p. reduction gear would under the conditions stated become overloaded to the breaking point. The gear makers worked with the author and guarded against this possible excess load when designing the gears for his several vessels.

There was no reason why the water-tube boiler and mechanical stoking should not be used to generate the steam required for White machinery. In fact the author would advocate using the water-tube boiler but for its unavoidable high cost, as he considered this boiler to be better adapted for superheating.

Mr. Christianson's conclusions were not correct. When the idea of combining a high-speed high-pressure reciprocating engine with a low-pressure turbine was first conceived, the first desideratum was an engine that would give the most perfect balance, and also give as constant a flow of steam as was possible to the turbine. It was recognised that this would be best obtained with a four-crank four-cylinder engine having two low-pressure cylinders, and the cranks arranged to give the resulting four impulses of exhaust steam at approximately equal intervals during each revolution, and the author worked on these lines.

When consideration was given to the building of a test set, the author appreciated that using a triple-expansion engine would be a more severe test on the gears and also on the turbine, and accordingly the test set was arranged with a triple-expansion engine.

It would appear that Mr. Christianson was forgetting many of the early pioneers in high-speed engine design, such as Willan, Morcom and Dr. Bauer, who gave a description of this type of engine in the 1905 edition of his book on "Marine Engines and Boilers".

The author's staff had experienced no trouble in obtaining accurate indicator diagrams from the reciprocating engine when using the high-speed engine indicators now made by several firms. There were in fact a number of the actual diagrams taken from the engine when running at from 300 to 320 revolutions per minute on view at the time the paper was read. The enclosed set (Fig. 14) taken off the "Llanashe" were a typical example.

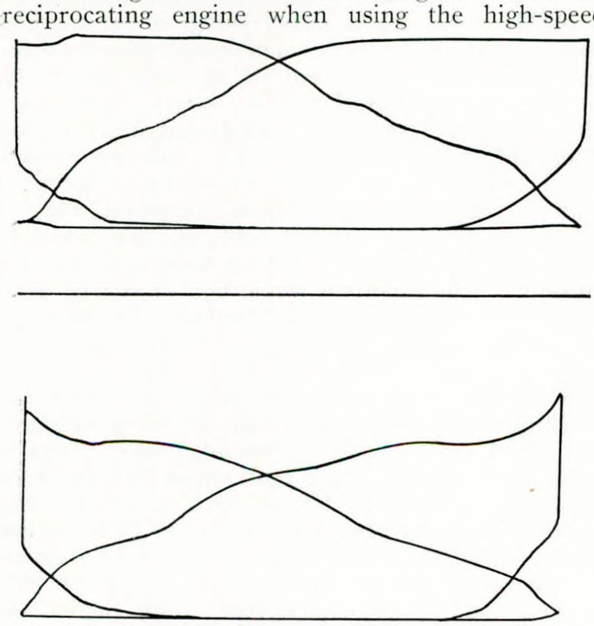


FIG. 14.—S.s. "Llanashe". Indicator cards—top h.p.; bottom l.p.

As previously stated, the b.h.p. of the complete combination was recorded by two independent makes of torsion meters, and a mechanical efficiency of 90 per cent. was used to convert the b.h.p. into indicated horse power.

Incidentally the author ran the load ship voyage of the "Adderstone" on one boiler (abstract of the log was given in Fig. 6A), and gave the dimensions of the boiler purposely so that anyone who was desirous could readily check the amount of steam it was possible to generate. The author had spent the major portion of his life in pioneering and development work, and from past experience fully appreciated that the low steam consumption figures being obtained with the engines would be questioned.

The auxiliary water consumption would naturally appear high in relation to the main engine water consumption, as the consumption of the main engines was itself so low; the lower the main engine consumption the greater the ratio, as a low main engine steam consumption did not have the effect of reducing in a relative degree the steam consumption of the auxiliaries. In the table referred to, it was approximately 18 per cent. of the total water. This figure the author felt sure Mr. Christianson would agree was not high when it was considered that all the auxiliaries were independent of the main engine, were steam driven, and included a feed pump, air pump, circulating pump, steering engine, oil fuel pump, sanitary and bilge pump (part time), electric light engine (part time), evaporator (part time) and oil transfer pump (part time), and he might even concede that the inaccuracy, if any, had been in crediting too little to the auxiliaries.

As a matter of fact on several of the later vessels, where careful measurement both by meter and by weight of the auxiliary steam consumption had been made, it had been found that the proportion ranged between 22 per cent. and 25 per cent. of the total water evaporated. Hence the author's remarks during the discussion that, while considerable thought had been given and a lot of work done to reduce the consumption of steam in the main engines, very little appeared to have been done by the auxiliary engine makers to reduce the consumption of their auxiliaries.

With regard to Mr. McAfee's remarks, the author decided on using piston valves after having run a series of experiments with an engine fitted with drop valves. These experiments and water consumption tests indicated that there was no gain or economy obtained from using drop valves in a short stroke engine, and as reduced clearance volumes were obtained with piston valves in the short-stroke high-speed engine, the author adopted this type of valve as standard. He agreed, however, with Mr. McAfee that they increased the overall length of the engine, and he would have no objection to using any of the many suitable types of valves if requested to do so. Up to the present, however, he had had no cause to regret his decision, as no trouble had been experienced with the piston valves in any of the vessels in service.

With regard to the point raised by Mr. McAfee on the precautions taken to prevent cylinder lubricating oil getting into the boilers, the author had fitted two filters in all his jobs, and in his standard design used two gravity filters. A strainer or grease extractor and trap were also fitted in the auxiliary steam line. On a recent examination of the "Adderstone" boilers after three years' running no indication of oil whatever was found.

With regard to out of balance forces, the author in the several vessels fitted had not found any increased tendency of the engine to vibrate when running in the ship; all the larger vessels' engines had been amidship, but the trawler engine was well aft. A coin could be balanced on edge on the top of the cylinder escape valves on any of the engines when running.

The author had not yet had the opportunity of obtaining reliable comparative data to ascertain with any degree of accuracy the economy due to the fitting of reheaters, but was satisfied that if any was obtained it would be quite small when the increased consumption of fuel necessary to raise the steam to the higher superheat temperature was also taken into account. In the White engine the factor governing the use of interstage reheat when using a low initial steam pressure was to obtain the steam to the turbine as dry as possible and free from water, not only to ensure maximum efficiency in the turbine but also to ensure long life to the turbine blades by eliminating as far as possible erosion of the blades by the scoring effect of water particles.

The vessels of which Capt. Tostevin attended the trials were the first installations and the disposition of the piping and fittings did undoubtedly give the appearance of a complexity of piping. In the later jobs a better arrangement had been arrived at and the engine room now had the appearance of a normal engine room with independent auxiliaries. Of course, owing to the small size of the main engine, the steam and exhaust pipes and fittings appeared large in proportion. The proved gains, however, well warranted the expenditure on pipes and fittings.

While the main auxiliaries were arranged to take superheated steam, the actual temperature used was only in the region of 420° to 430° F., the major portion of the superheat having been taken out in the turbine steam reheater and given to the turbine steam, and while the auxiliaries arranged for superheated steam were fitted with lubricators, it was found in service that they did not require to be used, the swabbing of the rods being found sufficient.

The cylinder lubricating oil used for the main engines of the later vessels was averaging 2 to 2·25 pints per 24 hours.

The improved turning effort indicated in Fig. 1a was not an accident, but arranged for in dimensioning the l.p. cylinders.

The entropy diagram mentioned was given only to indicate the effect of reheat.

The comparison figures before conversion of the "Adderstone" were taken from a service voyage with the machinery in general normal good condition, and were found to be in line with her early records.

The total repairs and replacements to the main engines of the "Adderstone" over the three years she had been in service had been the replacement of one set of h.p. piston rings.

The gears for a White combined engine required no more overload margin than turbine gears.

The consumption of lubricating oil for the complete combination, i.e. reciprocating engine, turbine and gears, ran out to from $2\frac{1}{2}$ to $2\frac{3}{4}$ gallons per 24 hours.

In reply to Mr. Reid, the boilers of the "Llanashe" were 13ft. 3in. diameter × 11ft. 6in. long. They had three furnaces 3ft. 4in. diameter, and were good for over 20 per cent. more steam than the machinery required at its maximum power of 1,800 i.h.p.

As regards obtaining proper combustion when burning coal, from gas analysis and the temperature of the outgoing gases, which in the case of the "Llanashe" was under 400° F. for the complete voyage, the author considered the combustion to be satisfactory; also, he had not encountered any difficulties in obtaining the required evaporation when running with 700/720° of superheat. No trouble had been experienced with these superheaters.

The trawler was a natural draught job and was fitted with an uptake type of superheater which when first fitted interfered with the draught, but this was immediately remedied by a re-arrangement of the superheater tubes.

The ship Mr. Reid had in mind must, of course, have been using less horse power than the "Llanashe" (with which he compared her) when she was only burning 16 to 17 tons per day, as under the conditions of superheat he gave of 600° F., and allowing a boiler pressure of say 220lb., she would not be burning less than 1·3lb. of coal per i.h.p., and when developing the same horse power as the "Llanashe" would be burning 19 to 20 tons of coal per day.

The insertion of feed heating tubes in the uptakes would require very little looking after other than periodical steam blowing, and as any vessel so fitted would naturally be fitted with bye-pass arrangements, even in the event of their breaking down, this would not hold up the vessel.

With reference to Mr. Gibson's remarks, the figures given by the author applied to the "Llanashe". This vessel fully loaded and with a total displacement of 12,240 tons maintained, in a rough sea with a beam wind of Beaufort scale 7, a steady 10 knots for 6 hours, and burned over the full period 7,380lb. of coal, equal to 1,230lb. per hour, all of which was carefully weighed by the owners' representative.

The fires in all furnaces at the end of the run were agreed to be not smaller than at the commencement, and the water in the boilers was slightly more at the end of the run.

Coal consumed = 1,230lb. per hour.

S.h.p. = 1,230, allowing a mechanical efficiency of 90 per cent.

I.h.p. = 1,366.

As the coal burned per hour came out to the same figure as the s.h.p., the coal per i.h.p. per hour must therefore be the same figure as was used for the mechanical efficiency. Mr. Gibson unfortunately had used a figure which it was evident was a misprint. Had he used the consumption given in the trial data table on the same page of 13¼ tons per day, he would have arrived at 1,236·6lb. of coal per hour, and a mechanical efficiency of 89·5 per cent. or an increase of 0·0056 per cent. in place of his 5½ per cent.

In the author's opinion all comparisons of *coal* per *horse power* should be based on *s.h.p.*

The author agreed that still better results could be expected from larger power installations. A definite saving would be effected owing to the clearance water being reduced.

Regarding the final paragraph and its spirit of optimism, the author visualised :—

(1) All sea auxiliaries being driven direct from the main engine.

(2) Adopting mechanical stoking.

(3) Heating the feed water from the heat contained in the waste gases.

These three items would effect a further saving of approximately 20 per cent., bringing the 0·9lb. per i.h.p. already being obtained down to 0·75lb. and under.

The author was aware that several economisers were on the market, and of this method of feed heating having been applied in the U.S.A. In fact he was very anxious to install one on his own vessel, but his findings were that the possible saving in consumption on a vessel burning 15 tons of coal per day did not warrant the expenditure entailed, based on a saving of 1 ton of coal per day being considered as warranting a capital expenditure of £1,000.

INSTITUTE NOTES.

ELECTION OF MEMBERS.

List of those elected at Council Meeting held on Monday, May 3rd, 1937.

Members.

Robert Turnbull Campbell, Vacuum Oil Co., Ltd., Marine Department, Canute Road, Southampton.

William Arthur Peters Jackson, 36, Sandringham Gardens, North Shields.

Archibald Martin, 31, Giffnock Park Avenue, Giffnock, Renfrewshire.

George Turnbull Golightly, 140, Howdon Road, North Shields.

James David Barron Mundie, 31, Alma Street, Falkirk.

William George Plews, 21, Luath Street, Glasgow, S.W.1.

Alfred Frazier Shortt, Sociedad Petrolifera Espanola, Apartado 120, Las Palmas.

William Brown Stevenson, 24, Titchfield Road, Troon, Ayrshire.

Walter Thompson, 2, Leven Street, Middlesbrough.

Frederick Hector McDonald Wilson, Vizagapatam, India.

George Robert Freeman, 822, Baldwin Avenue, Norfolk, Virginia, U.S.A.

Associates.

Joseph Bowman, 74, Northumberland Terrace, Willington Quay-on-Tyne.

William Brandwood, 17, Regents Way, Higher Bebington, Cheshire.

William Victor George Duffin, 6, Crescent Road, Melksham, Wilts.

John James Dwyer, 47, Hatcham Park Road, New Cross, S.E.14.

Gilbert Foggin, 34, Ramsden Dock Road, Barrow-in-Furness.

Thomas William Froggatt, 27, Skellingthorpe Road, Lincoln.

David Morgan Job, Glen View, Tyrfran, Llanelly.

Gilbert Armstrong Kerr, 78, Wellesley Road, Ilford, Essex.

Geoffrey Newhouse, 5, Havelock Terrace, Kowloon Dry Docks, Hong Kong, China.

David James Paterson, c/o Wilson, 75, Titchfield Road, Troon, Ayrshire.

James Smith, Irrawaddy Flotilla, Mandalay West, Upper Burma.

Student.

Sydney Cook, 161, Hawthorn Avenue, Anlaby Road, Hull.

Transfer from Student to Associate.

David James Collins, 171, Green Lane, Chislehurst, Kent.

ADDITIONS TO THE LIBRARY.

Purchased.

List of the Principal Acts of Parliament, Regulations, Orders, Instructions, Notices, etc., relating to Merchant Shipping issued prior to the 1st January, 1937. H.M. Stationery Office, 6d. net.

Instructions to Surveyors as to the Survey of Ships for Load Line Certificates and as to the Loading of Ships. H.M. Stationery Office, 9d. net.

K.R. and A.I. Amendments (K.R. 2/37). H.M. Stationery Office, 3d. net.

The Mercantile Navy List and Maritime Directory for 1937. H.M. Stationery Office, 25s. net.

"The Shipping World Year Book, 1937". "The Shipping World", 25s. net.

Presented by the Publishers.

"The Cunard White Star Quadruple-screw Liner 'Queen Mary'." Reprint of articles which appeared in "Engineering". Published by "Engineering" at 6s. net.

"Patents and Trade Marks", by B. T. King. King's Patent Agency, Ltd., Wardrobe Chambers, 146A, Queen Victoria Street, London, E.C.4.

"Improvements in the Economy and Output of Heavy Oil Engines", by J. Calderwood, M.Sc. Diesel Engine Users Association, 7s. net.

"A History of the Merchant Navy", by H. Moyse-Bartlett, B.A. George G. Harrap & Co., Ltd., 84 illus., 303pp. 7s. 6d. net.

The author was very ambitious in endeavouring to crowd into so small a compass the history of British shipping extending over so vast a period. It must, however, be conceded that he has not been unsuccessful in his effort, and this book includes an amount of general information which should be acceptable to most readers. The work also provides a handy book of reference, and with the accompanying bibliography a reader who wishes for more detailed information on any particular section should be able to pursue easily his search from writers who have made a more minute study of individual subjects.

It will perhaps be enlightening to many readers to learn that England owes a great deal to the Portuguese, who were pioneers in European navigation.

On page 29 the author tells us that the Church of the period, 1399, issued indulgences to those who would contribute to the repair of the harbour of Bridport. It would be interesting to know what sins the contributors for these licences committed, but this is a detail that we are left to imagine.

On page 268 we find a very succinct remark regarding the marking of ships with the Plimsoll line and disc which this enthusiast was trying to have made compulsory, in the face of considerable opposition in Parliament. He says, "But the Government were unable to prevent it".

This book is well worth the price for which it is sold.

"'Verbal' Notes and Sketches" (13th edn.) by J. W. M. Sothern. Published in two volumes by The Technical Press, Ltd., 1,500pp., 1,100 illus., 50s. net. (two volumes).

Recent editions of this book having attained dimensions which made its use possible only at considerable inconvenience, the author has wisely divided this new issue into two compact and handy volumes. This innovation will certainly meet with approval.

Primarily intended for candidates for Board of Trade certificates, almost all marine engineers are familiar with the wide scope of this manual of marine engineering practice, and it is unnecessary to give a detailed summary of its contents. Features of the book are the illustrated answers and page reference numbers to all Board of Trade new "engineering knowledge" questions, while many new illustrations and descriptions of the latest practice in steam engineering are included.

Students will appreciate that sound fundamental knowledge is now required in answering the "Engineering Knowledge" questions in the Board of Trade Examinations, and that a mere superficial knowledge or memorised answer will in most cases not be sufficient to ensure success. The author's aim therefore has been to describe and illustrate most of the subjects in full detail, so that the student has reliable and extensive information available. This refers, for example, to the Bauer-Wach and Götaverken auxiliary drive systems, Andrews & Cameron cam-operated valves, the Grinnell system of fire extinguishing, etc.

The two volumes are intended to be read in sequence, and for convenience a complete index is included in each volume. A work which has been issued in thirteen editions hardly needs recommendation. It is self-evident that those for whom the book is intended recognise its merit.

"Elements of Electricity", by W. H. Timbie. Chapman & Hall, Ltd., 3rd edn. (123 thousand), 569pp., 469 illus., 15s. net.

The object of this book is "to afford a sure foundation on which to build more advanced courses and yet to present a practical and well-rounded course for the man who is to receive no further electrical education". The title of the book is far too modest as an indication of its contents. The matter in the text is in reality limited to facts and principles which a technical student needs to know, but to know well, and if these are learned we believe the object of the book will be attained. The information given covers electrical laws in a form directly applicable to real engineering practice and not to abstract theory and conditions.

In each chapter are given worked examples, usually accompanied by explanatory diagrams, providing the reader with precise and clean-cut conceptions of the conditions referred to in the problem. We consider this method of presenting problem and answer particularly worthy of praise and emulation. In addition to the worked-out practical examples are numerous allied problems which illustrate useful applications of the subject matter and afford excellent mental training for the student.

The ground covered is extremely wide and includes Ohm's law and its applications, electrical power and energy, measurement of resistance, magnets and the magnetic circuit, principles and operation of d.c. generators and motors, batteries, a.c. circuits, distribution systems, power measurement, etc., a.c. motors, converters, etc., vacuum tubes and gaseous conduction, the electrostatic circuit and various tables.

The enormous sale of the book in America testifies to its popularity amongst teachers and students in that country, and we sincerely agree with the opinion thus implied. We consider the book eminently suited to the needs of students of electrical engineering and especially those, such as marine engineers, whose knowledge of the subject is limited to that acquired by private study. For some reason a book of answers to the numerous problems appears to be published separately.

"The Modern Diesel". Iliffe & Sons, Ltd., 4th edn., 223pp., illus., 3s. 6d. net.

This new edition has been thoroughly revised and considerably enlarged to cover the continued development and progress in the use of Diesel engines in transport of all kinds.

The book deals in detail with the application of high-speed compression-ignition oil engines to road transport, aircraft and marine work, and gives tables alphabetically arranged showing the characteristics and leading dimensions of engines available. A new chapter is included dealing at length with developments in connection with the use of Diesel engines for railcar and locomotive use. The application of this type of power unit to rail transport in this country, the Continent, and the United States is discussed in detail and fully illustrated.

"The 'Secant Stroke' Engine", by Engineer Captain W. A. Williams, C.I.E., R.I.N. (ret.). Published by the author at "Elderbank", Castletown, Isle of Man.

This pamphlet describes what is known as the "Net Power Crosshead, Infra Centrifugal, Non Harmonic, Double Acting, Crank Shaft Enveloping, Wedge Driving, Non Connecting Rod, Secant Stroke Engine". The claim is that of "an improved driving gear for eight, or multiple of eight, cylinder (four stationary cylinders opposed to four stationary cylinders) fuel injection or fuel suction internal combustion reciprocatory engines working either on the two or four stroke cycle, such driving gear for each unit of eight cylinders consisting of eight pistons, rigidly attached to a crosshead, four on one side and four on the opposite side, such crosshead embodying one crank pin driving device in the form of a slot, so that the driving gear, when actuated by burning gases in certain cylinders, shall deliver to the crank pin only the net power remaining after all resistance to reciprocation of the crosshead and its eight pistons, not counting the resistance of the crank shaft, has been subtracted".

The pamphlet is available in the Library for the information of members.

"Electric Wiring Diagrams", by W. Perren Maycock, revised by H. C. Fabian. Sir Isaac Pitman & Sons, 2nd edn., 155 pp., 258 illus., 5s. net.

Many of us remember the popularity of Perren Maycock's books years ago when electrical engineering was really in its infancy and literature on the subject very scarce. The present volume is a revised edition of one of this author's publications, which was mainly composed of material from one of his larger books.

It has been brought up-to-date by alteration of the original diagrams where necessary and the addition of new diagrams and explanatory notes to bring the book in line with modern practice. The book forms a useful reference work of connections for distribution, lighting, heater, cooker, and motor circuits. It also deals extensively with circuits for bells, alarms, telephones and transformer lay-outs.

It will be found useful by mains' engineers, wiremen, and others as a reference book of standard circuits. A great deal of space is devoted to various lighting and switching systems, signs, high-pressure discharge lamps, etc.—a section which is thoroughly up-to-date and will be of great service to wiremen generally.

The diagrams are clear and well produced and the descriptive matter enhances their value considerably.

JUNIOR SECTION.

Exhaust Steam Engineering.

On Thursday, April 15th, 1937, the last of this Session's series of joint meetings of the Junior Section with students of various technical colleges in the London area was held at West Ham Municipal College.

Mr. Edward W. Green, O.B.E. (Member) delivered an instructive lecture on Exhaust Steam Engineering to a large and appreciative audience. A feature of the lecture was the exceptionally valuable series of slides with which Mr. Green illustrated his remarks. The lecture will be published in full in the June TRANSACTIONS.

On the proposal of Mr. A. E. Gladwyn, B.Sc. (Head of the Mechanical Engineering Department) a vote of thanks was heartily accorded to Mr. Green for his lecture, and on the proposal of Mr. B. C. Curling (Secretary of The Institute), Dr. H. Baker (Principal of the College) was cordially thanked for his able Chairmanship.

Coronation of Their Majesties King George VI and Queen Elizabeth.

On the occasion of the Coronation a Loyal Address has been sent to Their Majesties in the name of the leading Engineering Institutions of Great Britain. We reproduce a photograph of the text of the Address, which was followed by pages bearing the seals of the institutions and signatures of the Presidents and Officers.

TO HIS MOST EXCELLENT MAJESTY
KING GEORGE VI.

May it please Your Majesty,

WE, on behalf of the members of the principal Engineering Institutions and Societies in the United Kingdom of Great Britain and Northern Ireland, desire to tender on the occasion of your Coronation our heartfelt congratulations and to offer our earnest wish that your Reign may be long, prosperous and peaceful.

YOUR MAJESTY has always evinced a deep interest in the Institutions that exist for the advancement of engineering science and for the better utilization of the forces of nature for the benefit of mankind. In the course of the services which you rendered as Duke of York during the lifetime of your much beloved father, His late Majesty King George V, you have been brought into direct contact with workshops, with mines and with works of engineering construction, wherein the art of the Engineer and the manual skill of the workman have combined to supply many of the needs of modern civilization.

IN the assurance of your knowledge of and close interest in the work on which we are engaged, we beg leave humbly to express to Your Majesty and to Her Majesty Queen Elizabeth our loyalty and support in the onerous duties consequent upon your accession to the Throne, and we pray that Almighty God may bestow upon Your Majesties health and strength to bear these responsibilities, and happiness in the knowledge of the devotion of your people.

ABSTRACTS OF THE TECHNICAL PRESS.

Characteristics of Screw Propellers.

The author gives the results of a systematic series of open water tests on model four-bladed screws. The geometrical particulars are: pitch ratio range, from 0·6 to 1·4; developed blade area ratio 0·4 and 0·55; thickness-diameter ratio 0·045 (at axis); boss-diameter ratio 0·167; rake angle 15 deg. The blade outline is approximately elliptical, but there is a certain amount of sweep-back. The sections near the boss are of aerofoil type, but the tip sections are of the segmental type. The face pitch is constant and equal to the nominal value over the outer half of the radius, but is reduced by 20 per cent. at the root. The screws were 240mm. dia. and were tested at an immersion of one diameter and at constant r.p.m. so as to give a practically constant Reynolds number (at 0·7 of the radius from the boss) of 200,000. The results are plotted for the slip range from 0 to 100 per cent. in the form of the usual thrust constant, torque constant and efficiency curves on a base of advance constant. Taylor diagrams are also plotted to facilitate the use of the results in selecting a screw for any given power, speed, and r.p.m. The maximum efficiencies range from 0·61 to 0·78 for the narrower blade group, and from 0·55 to 0·77 for the wider blades. In a numerical example the author demonstrates that the efficiency of a screw designed on the basis of this series of tests is very nearly the same, in a particular case, as that of one designed from an earlier series of results for models having aerofoil sections throughout.—*W. P. A. Van Lammeren, "Het Schip", 16th April, 1937, pp. 84-90.*

Vehicular Ferry "Stedingen".

The Diesel-engined double-ended vehicular ferry-boat "Stedingen" which was completed in 1935 by Messrs. Abeking & Rasmussen of Lemwerder, Germany, was designed to convey passengers as well as vehicles of all types between Lemwerder and Vegesack on the river Wesser below Bremen. Owing to local conditions a high degree of manœuvrability is required on this run and the heavy gales which blow across the course in winter demand a low free board and low erections. The principal dimensions are as follows:—Length o.a., 60ft. 8in.; breadth moulded, 22ft. 11½in.; depth moulded, 8ft. 1½in.; draught, 5ft. 7in. All deck erections are placed on one side of the vessel. This arrangement was adopted because it was found that a greater open deck area than with a symmetrical arrangement could thus be made available. At the same time the formation of a draughty centre passage could thus be avoided. As designed the vessel, being double-ended, can always be navigated in such a manner as to place the erections to windward, thus forming a lee for passengers who, owing to the small size of the vessel, cannot be accommodated in the cabins. The machinery of the vessel is of particular interest. The author states that the Diesel-electric drive was considered but rejected on account of its higher cost and smaller efficiency. Instead of this two Benz RH 224 V motors each developing 80 b.h.p. at 600 r.p.m. geared down to 400 r.p.m. were fitted. The motors, which are placed athwartships, are bridge-controlled by means of an oil pressure operated Renk-reversing gear. They are arranged in such a manner that each motor is capable of turning either of the two propellers. If both motors are to be run simultaneously the coupling between the two shafts is disconnected so that each motor then operates one shaft. A 5 b.h.p. solid injection four-stroke Benz motor driving an auxiliary compressor-lighting and pumping set completes the installation.—*Kurt Oehlmann, "Schiffbau", Vol. 38, No. 8; pp. 123-126.*

Recent Researches on Sea-water Corrosion.

The author reproduces the results of a series of tests designed to show what pairs of metals or alloys may be exposed to the electrolytic action of sea-water without the risk of corrosion, and to indicate the protective measures to be adopted in cases where this risk is unavoidable. In tests made with synthetic North Sea water, pairs of metals were exposed to the electrolyte in such a manner that in each series one metal was tested in turn against a number of other metals. It was found that the intensity of the corrosive attack could not be conclusively accounted for on the basis of the potential difference recorded at the beginning of each test, as this difference was reduced by resistance-increasing secondary processes of a purely chemical nature during the further course of such tests. The loss of weight per unit of time per unit of superficial area was therefore taken as being the measure of the intensity of attack. Materials tested in a state of tension or compression within the yield limit did not corrode more heavily than in the unstressed condition. The magnitude of the difference between electro-chemical and purely chemical attack was demonstrated by tests on cast-iron and cast-iron in combination with special brass, the loss of weight in the second case being only 12·5 per cent. of that obtained in the first. The potential difference was found to depend among other things on the temperature of the electrolyte, the maximum occurring between 122° F. and 196° F. To simulate natural conditions in an exaggerated manner, compressed air was blown into the glass vessel containing the test pieces and the contents stirred. The results obtained indicate that gases, notably oxygen, occluded in cold eddying sea-water promote corrosion in a striking manner. Tests made in a container of 3ft. length did not indicate any variation of the potential difference with the distance apart of the test pieces. The author therefore concludes that electro-chemical action will proceed at a distance if only a connection through sea-water is established. Subsequent tests in natural North Sea water confirmed the general conclusions given, the intensity of attack proving more severe than that recorded in the synthetic water.—*Dipl. Ing. Mangold, "Schiffbau", Vol. 38, No. 8; pp. 119-123.*

EXTRACTS.

The Council are indebted to the respective Journals for permission to reprint the following extracts and for the loan of the various blocks.

Geared Diesel Ships in Service.

Results from the Operation of Two-stroke Trunk-piston Single-acting High-speed Machinery in Conjunction with Mechanical Gearing. The Advantages Claimed.

"The Motor Ship", February, 1937.

There are now about a dozen German geared Diesel ships in service and at least 20 under construction. It is evidently the view of German shipowners that the indirect drive should invariably be employed, utilizing high-speed Diesel engines, either with electric transmission or gearing, and ships up to liners of 25,000 tons gross are being built with such machinery.

Some particulars of the performance of ships so equipped are, therefore, of importance, and will be of interest, not only to shipowners who have not adopted the arrangement, but to engineers, many of whom admittedly question the desirability of utilizing propelling engines running at well over 200 r.p.m.

The first ship to be equipped with the new type of M.A.N. two-stroke trunk-piston engines with geared drive was the "Sofia", placed in service at the end of July, 1934. The performance of this vessel and that of the "Athen", completed last year, and equipped with similar machinery, was recently recorded by Mr. Ernst Ofterdinger in a paper before the Schiffbau Technische Gesellschaft, of which the following article is a summary.

In the "Athen", the first Demag gearbox and coupling of a new type specially designed for ready withdrawal was utilised. It is stated that this coupling can be withdrawn in one minute by means of a handwheel. The advantage claimed, compared with the Vulcan coupling, is that the engine can be uncoupled from the propeller shaft whilst the other motor continues to run (Fig. 3).

In the installations in question, two 2,000 b.h.p. 225 r.p.m. six-cylinder engines, with cylinders 520 mm. in diameter and a piston stroke of 700 mm., drive a single propeller through gearing at 94 r.p.m. The mean indicated pressure of the engines is 5·44 atmospheres at the rating mentioned; in service it does not exceed 5·3 atmospheres. The mean piston speed is 5·25 metres per second, and the combustion pressure is normally 42 atmospheres, with a maximum of 50 atmospheres. The flywheels are provided with Blohm and Voss brakes. The "Sofia" is a ship of 4,600 tons d.w.c., with a speed of 15 knots, whilst the "Athen" caries about 7,100 tons at 14 knots.

A few notes on the type of engine (Fig. 1) may be of interest as it is similar in cylinder dimensions to that installed in many large vessels now building in Germany, including 25,000-ton passenger liners.

The reversible rotary displacement blower is arranged between cylinders Nos. 1 and 2, and the speed of the rotors is 900 r.p.m. The weight and size of the blower are small, and there is little noise, a suction silencer being fitted. The fuel pump drive for the engine is unusual. The cams are carried on the crankshaft and operate the vertical plungers of the fuel pumps through rollers and levers, the pumps being close to the fuel valves.

The piston design (Fig. 2) is an important feature of this engine. Experience with earlier trunk-piston engines has been unsatisfactory, and piston and cylinder liner cracks have been common. The employment of guide rings in the new piston has led to a remarkable improvement in its

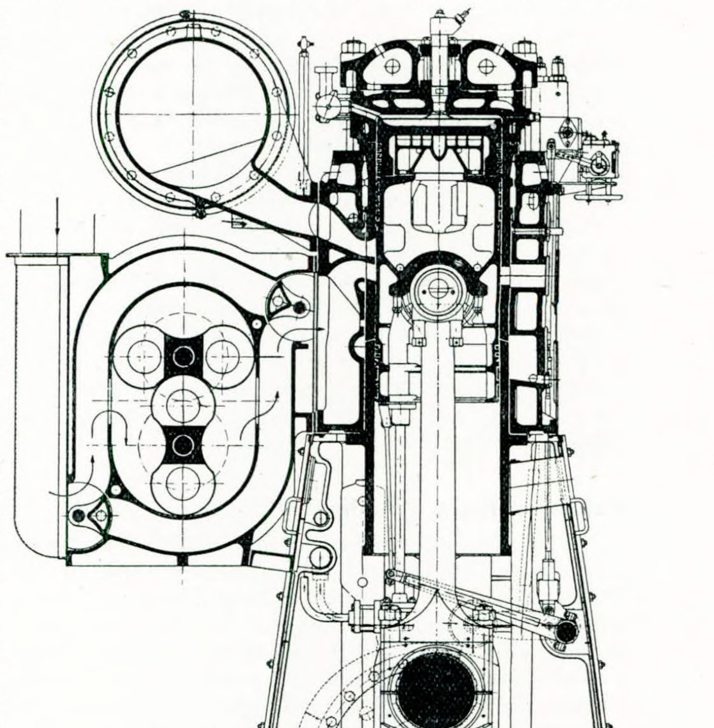


FIG. 1.—Sectional elevation of the M.A.N. standard two-stroke single-acting trunk-piston engine.

durability. These are made of material with good wear-resisting characteristics, but with a certain degree of elasticity. The piston has two upper and either two or three lower guide rings as shown, and in the case of the "Sofia" and "Athen" the rings are made of lead bronze. The fear of cracks in the cast-iron piston or cylinder liner, present with the original trunk-piston design, is now entirely abolished, and the opposition to the trunk piston has disappeared. The wear on the guide rings is small, amounting in the "Sofia" to less than 0·03 mm. per 1,000 hours. The wear on the cylinder liner caused by the guide rings is negligible.

The liner wear due to the friction of the piston rings is approximately the same as is found in normal Diesel experience, and in the case of the "Sofia" is about 0·08 mm. per 1,000 hours. It varies in different liners between 0·06 mm. and 0·13 mm. The most satisfactory figure should be capable of attainment in all cylinders, but that, however, is a question that concerns Diesel engines in general and not merely the trunk-piston type.

As piston rings in the engines being described, there are two double and, below, four working rings. The wear of the double rings in the combustion area is about 0·11 mm. per 1,000 hours, whilst the wear on the working rings is about one-third of this figure. In two years' operation of the "Sofia" no piston ring has been changed.

The trunk-piston engine uses a relatively large amount of lubricating oil. The cylinder oil consumption in the installations in question is between 0·4 gr. and 0·7 gr. per b.h.p.-hour, or 0·5 gr. in the case of the "Sofia". In normal Diesel installations there is a consumption of 1·2 gr. to 1·5 gr. per b.h.p.-hour of the oil in circulation, and in special cases it is as low as 0·6 gr. In the "Sofia" it is 2·2 gr., but it is estimated that in new installations the figure will be reduced to 1·2 gr. per b.h.p.-hour, making a total of 1·7 gr.

Noise measurement.

The M.A.N. have succeeded in these latest installations of trunk-piston engines in diminishing much of the noise. Special measurement trials were carried out by the Deutsche Werft. In the m.s. "Athen", the degree of noise between the two main engines and the cylinder grating at full speed was 102 phons. This is the same as is attained with a ship propelled by a double-acting two-stroke 4,100 b.h.p. six-cylinder engine running at 116 r.p.m. On the measured-mile run, with increased power, the noise measurements recorded were 111 phons and 113 phons respectively.

According to exact measurements, the noise made by two engines is only three phons greater than when a single motor is running. Measurements were made by Demag on a voyage of the "Athen" on the auxiliary Diesel engines, and these provided confirmation. Two engines are, therefore, only slightly noisier than one; the degree of noise to be anticipated is, therefore, dependent not on

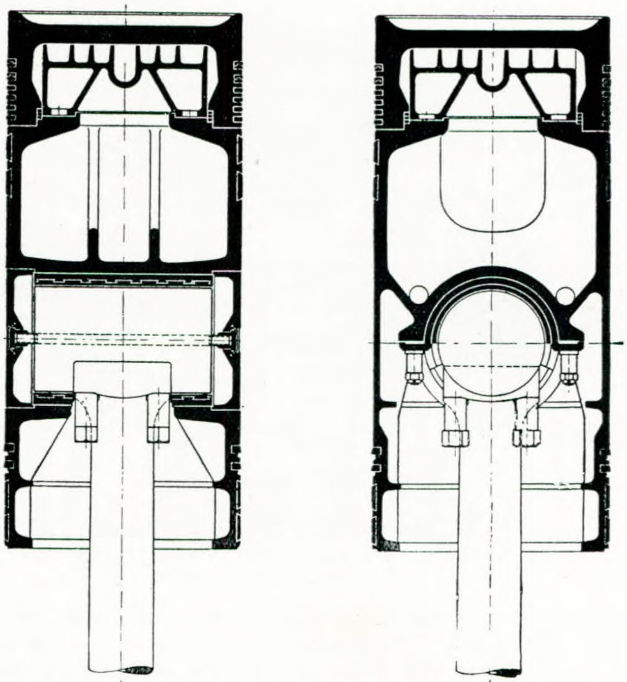


FIG. 2.—The piston of the M.A.N. trunk-piston two-stroke engine.

the number of cylinders or of engines, but on the noise generated from the noisiest cylinder and its gear.

Particularly noteworthy is the fact that two four-stroke auxiliary engines totalling 400 b.h.p. at 400 r.p.m. caused more noise than one two-stroke main engine of 1,500 b.h.p. running at 170 r.p.m. The actual measurements were 95 phons and 93 phons.

Table I shows the weight of the parts and the total weights of Diesel engines of various types, and explains why it is possible to handle these parts more easily in the case of high-speed engines with direct drive. The weights of piston rods, cylinder covers, crossheads and guides are all reduced. The total weight is also lower, allowing increased deadweight carrying capacity.

The engine costs are not in proportion to the weight, but probably the trunk-piston machinery is cheaper than the other types. Double-acting two-stroke high-speed engines with indirect drive are not brought into comparison as they are mainly suitable for higher powers than those under consideration.

There is little variation in the fuel consumption with the various arrangements. On the test bed at full load it is 164 gr. per b.h.p.-hour, or 0·36lb. per b.h.p.-hour; but when comparison is made with direct-coupled engines, the loss in the gearing is, to some extent, counterbalanced by the more efficient propeller at lower speed.

The lowest speed at which the propeller in the "Athen" will run is 18-20 r.p.m., corresponding to

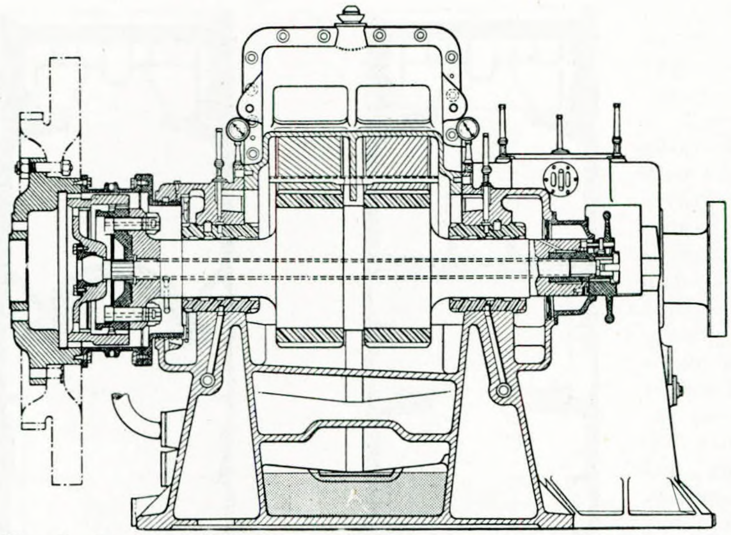


Fig. 3.—Sectional elevation of the Demag gearbox and coupling used in the M.S. "Sofia".

2 knots. At starting, the propeller speed rises only to 22 r.p.m. The engine stops rotating in about 6 sec. from full speed at sea, on account of the employment of the flywheel brake. Without the brake in the "Athen" the time required is 2½ min.

The properties of the machinery arrangement described may be summarized. It is well designed for marine service. The low height is favourable from the standpoint of attendance and it improves the ship's stability. Owing to the high rate of revolution, the dimensions and weight of the whole installation, also individual parts, are relatively small. This is helpful for the periodic overhauling. There is a small increase in deadweight capacity and hold space and, possibly, a slight reduction in building costs. From the aspect of noise, the geared arrangement is not unfavourable. Fuel consumption is satisfactory, and when only one engine is employed, for a voyage at reduced speed, it is very economical. The ship's personnel approves of the geared arrangement. The only disadvantage is the high lubricating oil consumption, the increased cost of which, however, is relatively small in view of the advantages gained. It is to be remarked that in two years' service of the m.s. "Sofia" there has been no case of cracks in the cylinder covers, pistons or liners, and there is every prospect of the machinery having a long life.

An American Geared Turbine for Cargo Ships.

The latest double-reduction cross-compound installation of the De Laval Steam Turbine Company is discussed.

"The Marine Engineer", April, 1937.

On numerous occasions we have discussed at some length the geared turbine tank steamships which American owners have developed to such a high pitch of efficiency in recent years. These articles have usually included general arrangement plans of the machinery spaces, but we have not hitherto published an arrangement drawing—such as we now give—of the particular type of geared turbine which has been used in many of these

TWO-STROKE 4,000 B.H.P. INSTALLATION.

Engine Part which has Normally to be Occasionally Dismantled.	Indirect Drive.		Direct Drive.			
	Trunk Piston.		Single-acting with Crosshead.		Double-acting	
	2×6 cylinder 520mm.× 700mm. 225 r.p.m.		8 cylinders 680mm.× 1,200mm. 110 r.p.m.		6 cylinders 600mm.× 1,100mm. 115 r.p.m.	
	Wt. in kg.		Wt. in kg.		Wt. in kg.	
	1 cyl.	12 cyls.	1 cyl.	8 cyls.	1 cyl.	6 cyls.
Upper cylinder cover, upper and lower parts ...	700	8,400	1,165	9,320	920	5,520
Lower cylinder cover with glands	—	—	—	—	1,000	6,000
Piston with piston rods	—	—	1,800	14,400	2,030	12,180
Piston with connecting rods	1,275	15,300	—	—	—	—
Piston rods	—	—	(545)	(4,360)	(1,050)	(6,300)
Conecting rods	(485)	(5,820)	1,940	15,520	1,835	11,010
Crosshead with guide	—	—	763	6,104	635	3,810
Crankshaft bearing with shell and keep	157	1,884	450	3,600	335	2,010
Crosshead bearing or gudgeon pin bearing	58	696	157	1,256	160	960
Main bearing cover	70	840	130	1,040	136	816
Main bearing shell	70	840	183	1,464	190	1,140
	Each		Each		Each	
Fuel pumps	12		8		12	
Fuel valves	12		8		12	
Starting valves	12		8		6	
Safety valve	12		8		12	
Total (exclusive of liners)		28,000		53,000		43,500
Comparative weight of the complete machinery, excluding flywheel coupling drive and bearings	kg. 220,000		kg. 315,000		kg. 250,000	

vessels, including the "G. Harrison Smith", "W. S. Farrish", "Socony Vacuum", "Magnolia", and others. The turbine design in question is that of the De Laval Steam Turbine Co., of Trenton, New Jersey, and similar machinery embodying detailed improvements has been adopted for the four new large tankers recently ordered from the Federal Shipbuilding and Dry Dock Co., of Kearney, New Jersey, by the Standard Oil Co., of New Jersey.

High superheat.

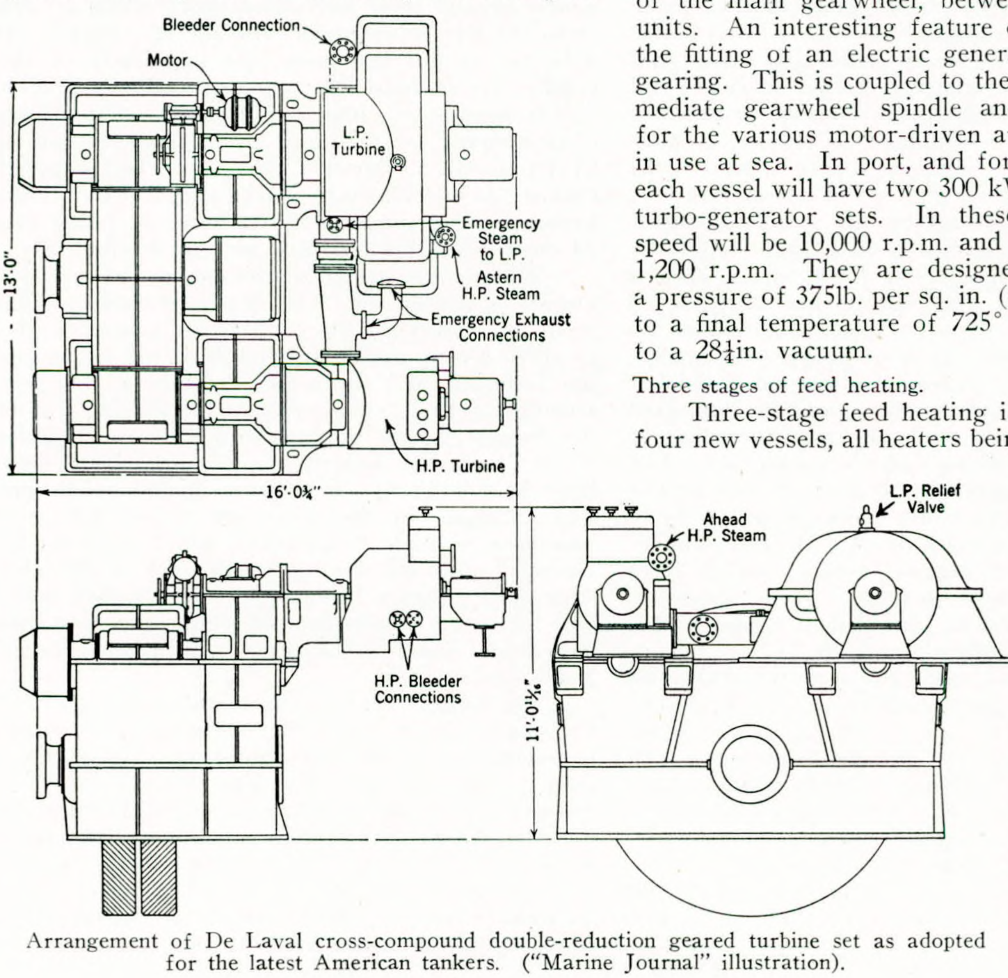
These vessels are of somewhat lower power than the pioneer installation of this class in the "G. Harrison Smith". The normal rated output of that vessel is 4,400 s.h.p. on a single screw, while the power in the case of the four new ships is to be 3,000 s.h.p. at 90 r.p.m. of the propeller; the maximum rating is 3,300 s.h.p. at 93 r.p.m. The accompanying drawings indicate that the turbines are of the cross-compound type and are designed for a working steam pressure of 400lb. per sq. in. (gauge) at the superheater outlet, while the temperature of the superheated steam at normal full load is 750° F. It is interesting to note that the specification stipulates that the superheat temperature must not vary more than 15° F. at any load between 25 per cent. of the normal full load and 50 per cent. overload. Steam is to be supplied in each vessel by two oil-fired water-tube boilers of the Foster Wheeler type similar to those which have been described and illustrated on several occasions in "The Marine Engineer". These boilers have no air pre-heaters, but are provided with economisers, according to "Marine Journal".

High turbine speeds.

With a view to achieving lightness, compactness, and an even greater efficiency, the speed of the turbines in these new vessels is to be higher than has hitherto been employed by the De Laval Co. in similar installations. The high-pressure turbine spindles are to have a normal speed of 6,000 r.p.m., while the low-pressure pinions have a normal speed of 5,000 r.p.m. The two sets of speed-reduction gearing are to be incorporated in a single casing so as to economise space and weight. The Foster Wheeler condenser, which is designed to maintain a vacuum of 28¼in. in ordinary service, is underslung. The thrust block is of the Kingsbury single-collar type and is located at the forward side of the main gearwheel, between the two turbine units. An interesting feature of the equipment is the fitting of an electric generator, driven off the gearing. This is coupled to the low-pressure intermediate gearwheel spindle and provides current for the various motor-driven auxiliaries which are in use at sea. In port, and for stand-by purposes, each vessel will have two 300 kW. De Laval geared turbo-generator sets. In these units the turbine speed will be 10,000 r.p.m. and the generator speed 1,200 r.p.m. They are designed to take steam at a pressure of 375lb. per sq. in. (gauge), superheated to a final temperature of 725° F., and exhausting to a 28¼in. vacuum.

Three stages of feed heating.

Three-stage feed heating is to be used in the four new vessels, all heaters being of the bled-steam type. The first stage takes steam at 100lb. per sq. in. (absolute) from the h.p. turbine, the second at 35lb. per sq. in. (absolute) from the h.p. turbine, and the third receives steam at 6lb. per sq. in. (absolute) from the low-pressure casing. The feed pumps are to be of the two-stage centrifugal type, each driven by a De Laval steam tur-



Arrangement of De Laval cross-compound double-reduction geared turbine set as adopted for the latest American tankers. ("Marine Journal" illustration).

bine and designed to deliver 65 gallons per minute against a head of 1,160ft. with a pressure on the suction side of 70lb. per sq. in. These turbo feed pumps have a speed of 6,000 r.p.m. and their turbines exhaust against back pressure to one of the feed heaters already mentioned. De Laval double-reduction worm gear electric-motor-driven turning gear is provided and each vessel will have a De Laval centrifugal purifier for treating the lubricating oil for the main and auxiliary turbines.

It is anticipated that despite the smaller size of the turbines these new vessels will show an appreciable improvement on the fine performance of the "G. Harrison Smith" and "W. S. Farrish", which were built a few years ago. Each of these vessels has steamed about 250,000 miles without serious trouble and regularly returns an average all-purposes fuel consumption of about 0·613lb. to 0·619lb. of bunker oil per s.h.p. per hour. In the case of the new vessels, however, it is anticipated that an all-purposes fuel consumption of ordinary bunker oil of 0·58lb. per s.h.p. per hour will be obtained in service.

Interesting service performances.

It has frequently been asserted by those engineers who are opposed to the use of these modern high efficiency steam plants that water-tube boilers in ordinary general cargo carriers and tankers that the excellent performances obtained on trial will never be maintained in service, neither will such machinery, with its severe steam conditions, prove reliable in service. This opinion has, of course, been exploded by the results obtained with numerous vessels, but in view of the length of service obtained with some De Laval turbine-driven American vessels it may not be out of place to give a few details of these performances in the present article as showing what progressive marine steam engineers are achieving on the other side of the Atlantic. The two tankers of the Standard Shipping Co. to which we have just referred are a case in point and others could be mentioned. For example, the well-known Luckenbach Steamship Co. has four vessels which are each propelled by a 2,500 s.h.p. installation of De Laval double-reduction compound geared turbines which have proved very reliable in service. These ships are the "Harry Luckenbach", the "Lilian Luckenbach", the "Dorothy Luckenbach", and the "J. L. Luckenbach". Each has covered more than 600,000 miles without turbine or gear troubles and the fuel consumption figures obtained are in every way satisfactory. Equally good are the results obtained with the two Sinclair 9,300 tons deadweight tankers "Harry F. Sinclair, Junr." and "Virginia Sinclair". These vessels are driven by De Laval double-reduction compound geared turbines and have an average service speed of 12½ to 13 knots on a daily consumption of only 160 barrels of bunker oil. The Baltimore Mail Steamship Co.'s vessels "City of Baltimore", "City of Norfolk", "City of Havre", "City of Hamburg", and "City of Newport News" are passenger and cargo vessels of considerably higher power than those referred to above, each being provided with De Laval double-reduction compound geared turbines of 9,500 s.h.p. They have logged more than one million sea miles without any interruptions which might be attributed to the propelling equipment. Here again the consumption is low.

Large Semi-uniflow Engines with Modified Valve Gear.

A double-eccentric Klug-Spranz gear has been standardised for the larger Christiansen & Meyer engines.

"The Marine Engineer", April, 1937.

One of the most successful of the improved forms of marine steam engine developed in recent years and, incidentally, one of the simplest, is the Christiansen & Meyer semi-uniflow double-compound engine, which, it will be recalled, operates on the Wolff compound principle with a single piston valve for each half of the four-crank engine. The bulk of the exhaust steam from each l.p. cylinder is handled through ports at the middle of the liner, while the balance is dealt with by the piston valve, which also is used for steam distribution to and from the h.p. cylinder and into the l.p. engine. As indicated in our last issue, the popularity of this engine has increased to an appreciable degree in recent months, for there are no fewer than 20 sets of machinery of this type under construction by Christiansen & Meyer and their licensees, 10 building in the Harburg works and 10 by different licensees in Germany and elsewhere; no fewer than 54 engines of this type are now in service.

The general design of the engine is fairly well known to readers of "The Marine Engineer", and so it is interesting to be able to announce that recently a new and improved form of valve gear has been adopted for larger engines of this type. During the first few years that this engine was on the market it was not made in sizes much above 1,500 i.h.p., but latterly some fairly large units have been built to Christiansen & Meyer designs. For example, in the news section of this issue reference is made to an order which the firm has recently obtained for an engine of 2,500 i.h.p. This is to propel a Finnish passenger vessel, which will have the distinction of being the first new mercantile vessel to be fitted with Brown Boveri Velox boilers.

The larger* illustration on the next page is of the main engine for the steamship "Nerissa", belonging to the Kirsten Shipping Co., of Hamburg. This vessel, London readers may be interested to know, regularly visits the Thames; she operates on the service between London and Hamburg and other near Continental ports, and we have no doubt that Mr. W. A. Christanson, of 34, Victoria Street, London, S.W.1, the British representative of Christiansen & Meyer, would be very

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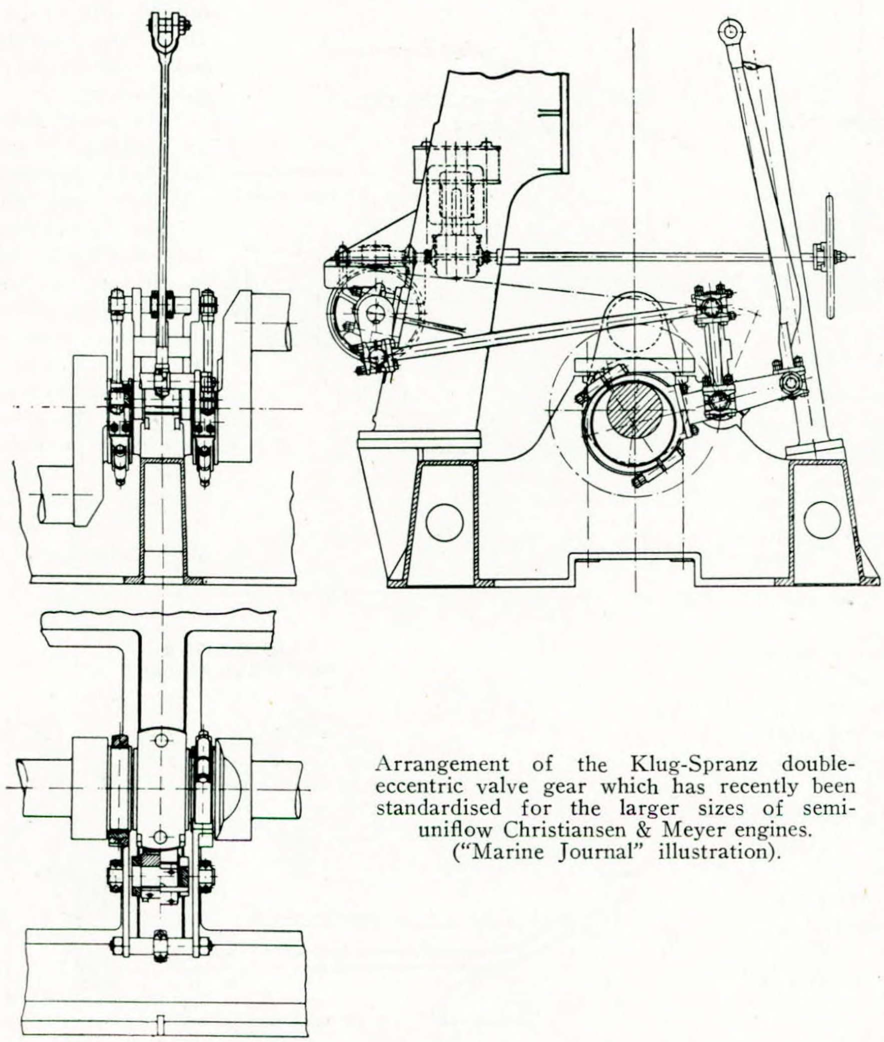


Arrangement of the Klug-Spranz double-eccentric valve gear which has recently been standardised for the larger sizes of semi-uniflow Christiansen & Meyer engines. ("Marine Journal" illustration).

gear is the symmetrical nature of the driving mechanism, there now being an eccentric on each side of the crankshaft main bearing between each pair of cylinders comprising a half unit of the engine. Formerly, it will be recalled, only one eccentric was employed and the drive was consequently unsymmetrical. The two eccentric straps and short rods are connected at the front through a short stiff cross piece, to the middle of which a single valve connecting rod passes to the piston valve rod as shown. Two swinging links connect the eccentric rods to the special reversing frame, which, in turn, is coupled to the samson rod along the back of the engine in the manner shown. Reversal is, of course, effected by the usual steam reversing engine, which can be seen in the illustration of the "Nerissa's" engine; in smaller engines of this type, it will be remembered, reversing can be effected by hand. The new valve gear has been standardised for all engines having a stroke in excess of 800 mm.

pleased to show interested engineers over the engine-room of this vessel. The "Nerissa's" engine employs the usual double-compound principle, with four cranks. The high-pressure cylinders are 355 mm. in diameter, the low-pressure units are 760 mm., and the stroke is 800 mm. The "Nerissa" has been on this run for some months and since her service performance indicated that the economy was of the high order claimed for the engine, a repeat of the vessel and engine were ordered by the same company.

New valve gear.

This illustration shows the new valve gear to which reference has just been made. This is of the Klug-Spranz type and is an improvement on the earlier modified Klug gear which many Christiansen & Meyer engines use. (In one C. & M. installation, it may be recalled, ordinary Stephenson gear is employed.) Furthermore, a drawing showing this improved valve gear is given with this article.

The principal feature of this recently-adopted

Gotaverken Steam Turbo-compressor System with Electrically-driven Auxiliaries.

By Hugo Heyman, Gothenburg.

"The Marine Engineer", April, 1937.

For a reciprocating steam engine installation the steam consumption of the auxiliaries varies generally between 10 and 15 per cent. of the steam consumption of the main engine. It is therefore obvious that the application of a more economical way of driving the auxiliaries will result in an appreciable reduction in the fuel consumption of the installation. The exhaust steam from the auxiliaries is used for heating the feed water, but even so this feed water heating can be done in a more economical manner by using steam bled from the reciprocating engine exhaust and the l.p. receiver. The steam which is bled from the reciprocating engine exhaust (i.e., before the exhaust turbine) has worked in all the engine cylinders and the steam which is bled from the l.p. receiver has been working in the h.p. and i.p.

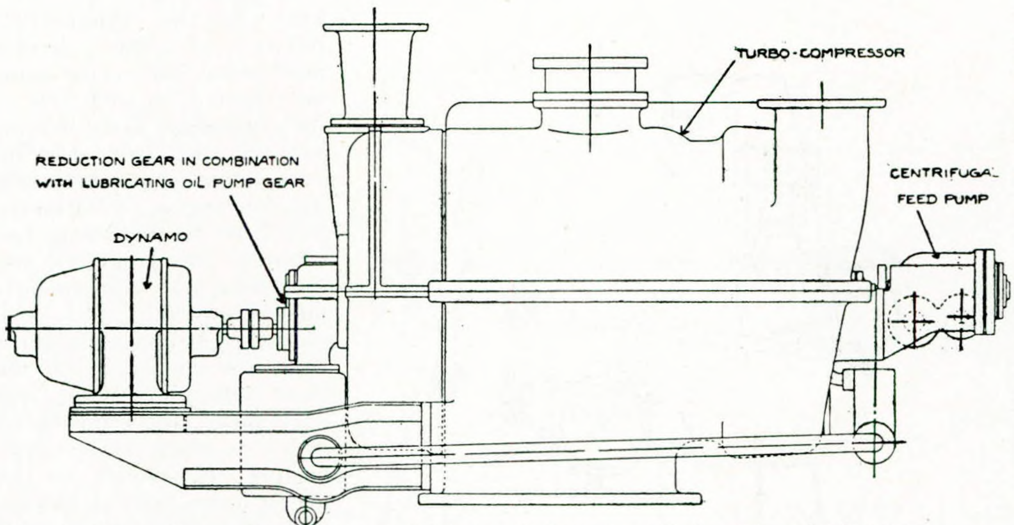


FIG. 1.—Gotaverken turbo-compressor driving a feed-pump, and, through reduction-gear, a dynamo.

cylinders before it is used for feed water heating purposes.

Electrically-driven auxiliaries have, with the Götaverken system, been arranged in two different manners :—

(1) Dynamo driven from the shaft of the turbo-compressor by means of a reduction gear, or

(2) Dynamo driven by a separate steam turbine, supplied with exhaust steam from the main engine.

In most cases where turbo-compressors have been installed a centrifugal feed pump, directly driven off the turbo-compressor, has been arranged. In certain cases a dynamo has also been arranged which is driven off the other end of the turbo-compressor by means of a reduction gear. This dynamo supplies electrical power for some auxiliaries, such as the forced draught fan, the refrigerating machinery, the electric lighting equipment, etc. Fig. 1 shows a turbo-compressor arranged for driving a feed pump and dynamo. A number of installations have been carried out according to this system and the results have been very satisfactory.

In some ships an auxiliary unit has been installed together with the turbo-compressor. The auxiliary unit is driven by exhaust steam from the main engine in the same manner as the turbo-compressor. In addition it is provided with a nozzle through which live steam is automatically admitted if

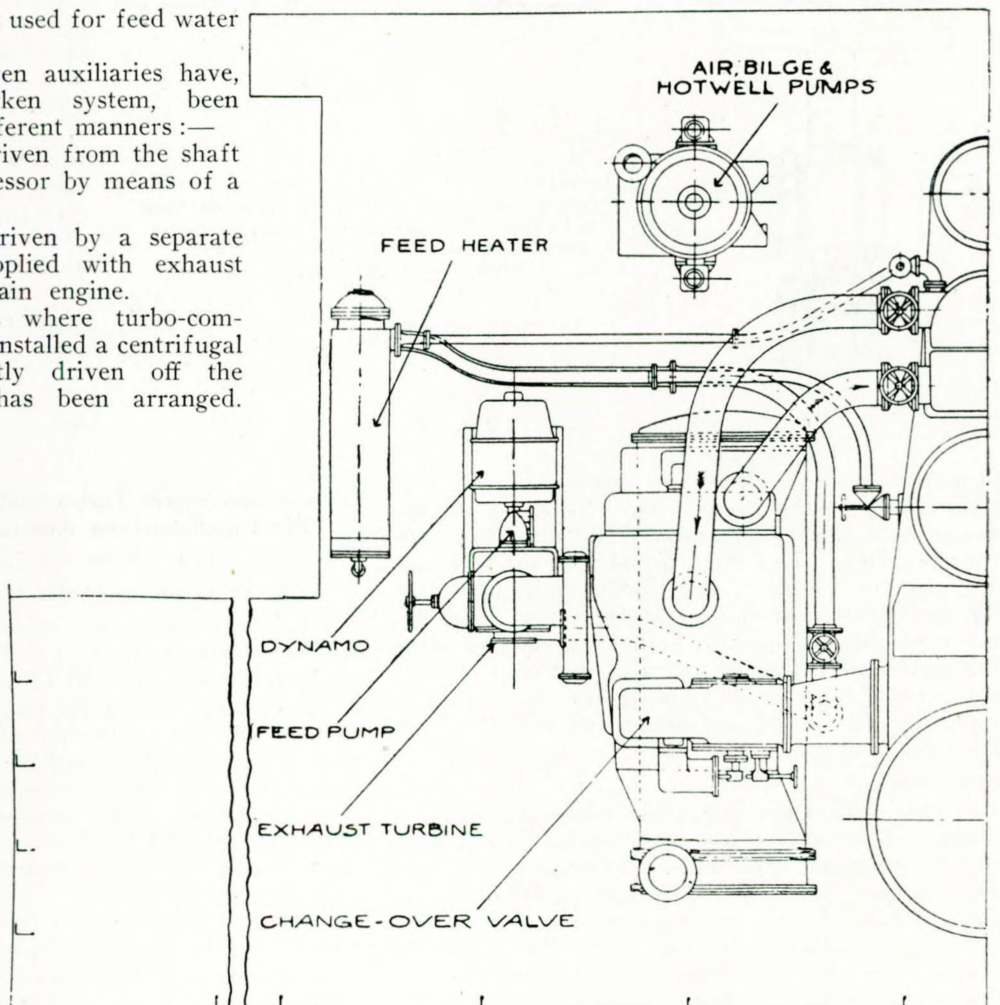


FIG. 2.—In the alternative new development of the Gotaverken turbo-compressor system, as shown, a separate small turbine using exhaust steam drives the generator and feed pump.

the main engine is running at low speed or stopped.

The auxiliary unit consists of an exhaust turbine with a directly-connected centrifugal feed pump and a dynamo driven by the turbine through a reduction gear, and when the auxiliary unit is used, no feed pump is fitted on the turbo-compressor. The auxiliary unit is given sufficient capacity for supplying electric power to all the auxiliaries which are working continuously. For some installations the steering engine has been steam-driven and for other installations it has been electrically-driven. The total coal consumption for an installation, with electric steering engine, has been recorded at 1·06lb. per i.h.p. per hour, figured on the fuel consumption during one whole voyage, with coal having a calorific value of 11,700 B.Th.U.'s per lb., i.e., a heat consumption of 12,400 B.Th.U.'s per i.h.p. per hour. With best Welsh coal, having a calorific value of 14,500 B.T.U.'s per lb., this corresponds to a total fuel consumption of 0·855lb. per i.h.p. per hour.

Figs. 2 and 3 show an installation with turbo-compressor and auxiliary unit.

Another New Steam Reciprocator.

The Moss three-cylinder compound engine is described. A special valve gear and interesting arrangement of valves are features of a design which is combined with an Elsinore exhaust turbine.

"The Marine Engineer", April, 1937.

A new steam reciprocator has been developed by the A/S Moss Vaerft & Dokk, of Moss, Norway. The first engine of this new type is now being installed in a 4,000 tons deadweight cargo vessel which the firm is completing for Th. Brovig, Farsund, Norway; a sister ship is to follow in the near future. The new engine is largely the design of Dr. J. L. Mansa, Professor of Engineering at the Norges Teckniske Hoiskole. The basis of the design, as the accompanying drawings and illustrations show, is efficiency with simplicity and compactness.

In order to obtain a well-balanced engine with good manœvring qualities the three-cylinder double-compound arrangement with cranks at 120° has been chosen. There are two high-pressure cylinders and one low-pressure cylinder, steam distribution being effected by slide valves. A single valve performs both steam admission and exhaust functions for the two h.p. cylinders, but for the l.p. cylinder there are separate inlet and exhaust slide valves. All valves, with the exception

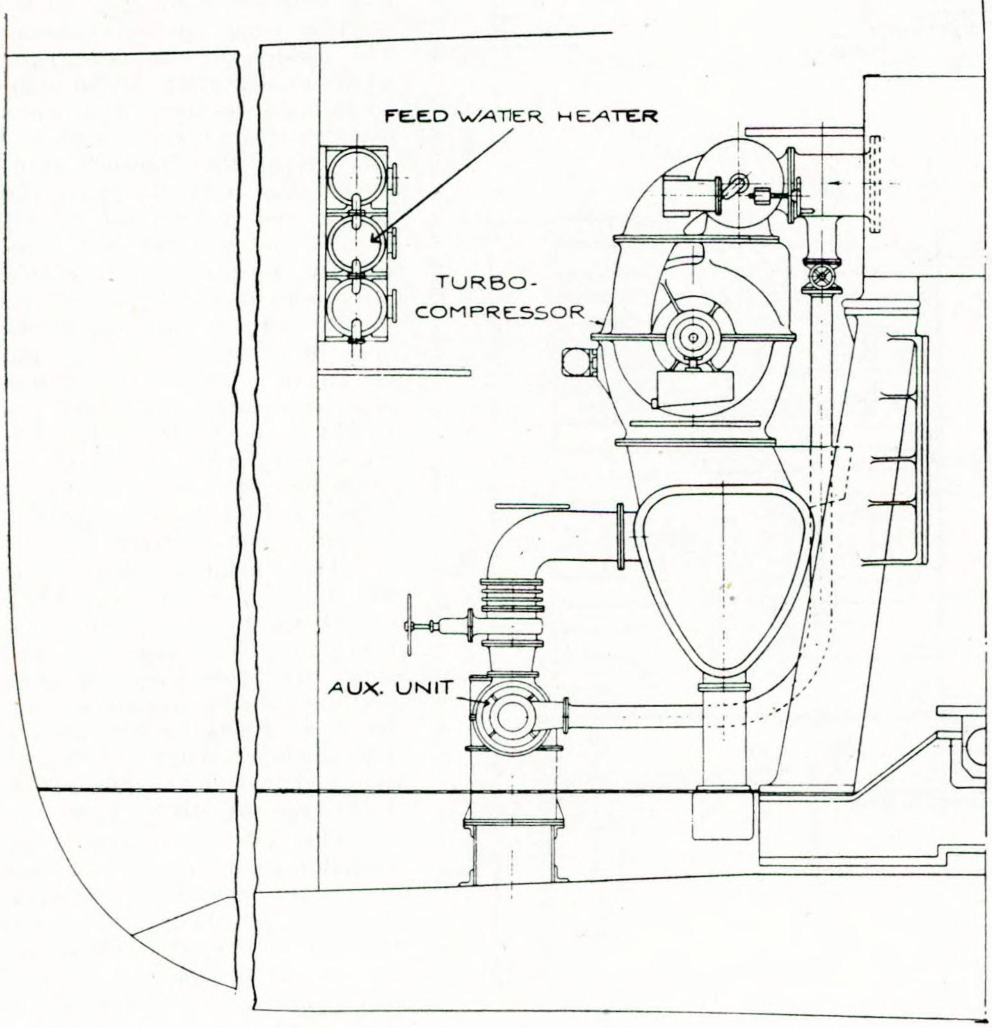
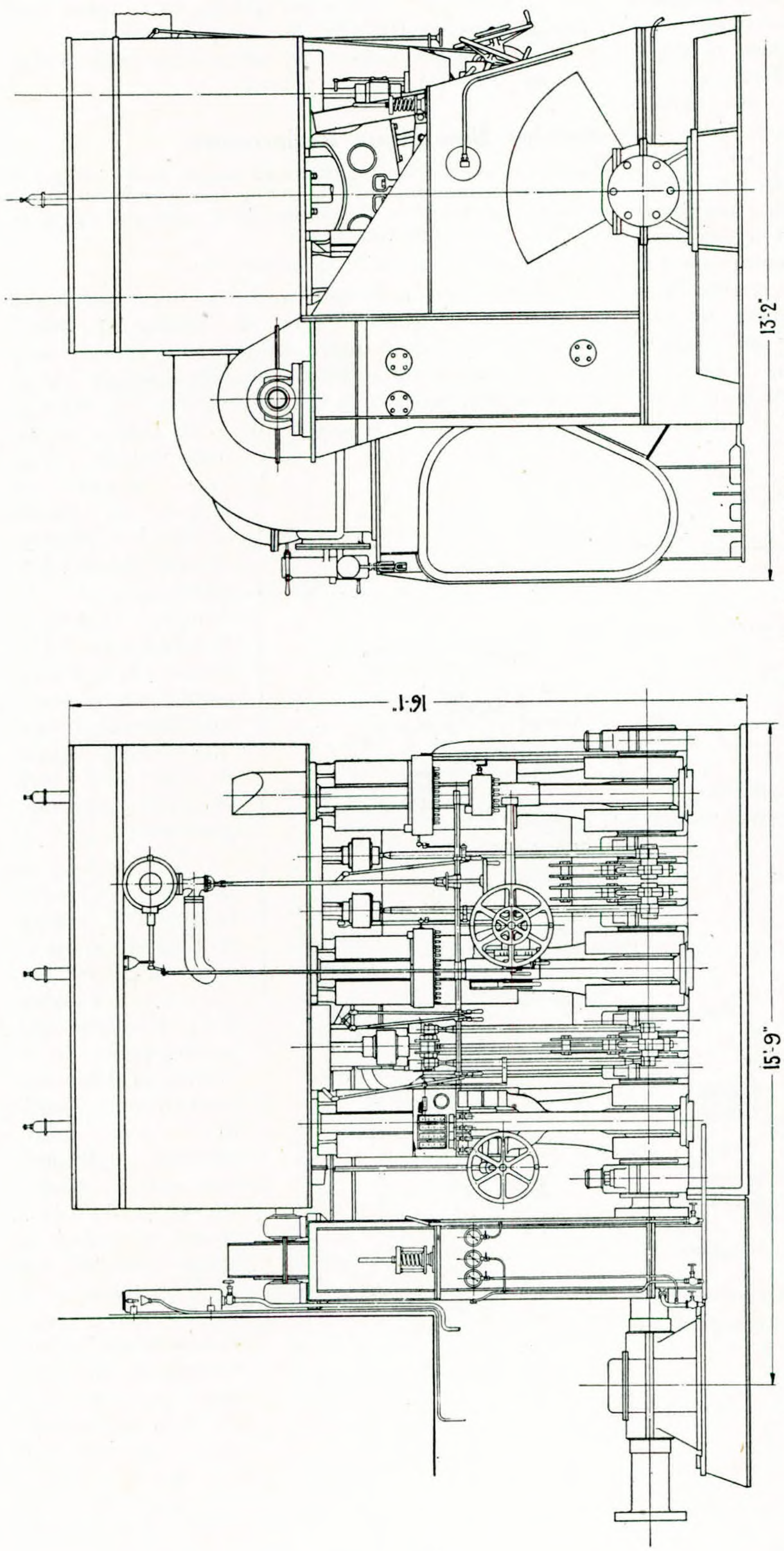


Fig. 3.—The auxiliary unit as shown in Fig. 2 is located at floor level, and the steam and and exhaust piping are neatly arranged.



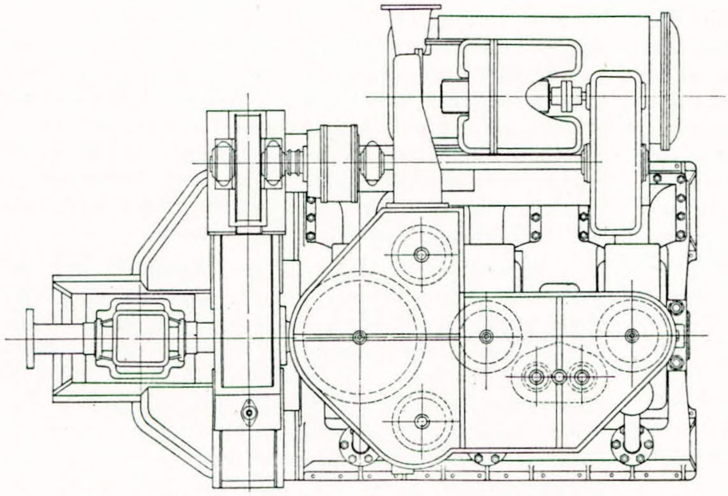
Side and end elevation of the new engine, which has a modified form of Klug valve gear.

of the l.p. exhausts, are at the front of the engine, where the main controls are located, while the l.p. exhaust valves are at the back, an arrangement which minimises the length of passage for the steam on its way to the exhaust turbine which is provided. Between the h.p. and the l.p. cylinders there is a receiver of relatively large volume but small surface. This is so designed that it is well protected from radiation and condensation losses and at the same time it does not add appreciably to the length of the engine.

The valve gear is of a new type developed and patented by the Moss Vaerft & Dokk Co. and is a modified Klug type with a single eccentric for each cylinder. The builders of this new engine advise us that after careful calculations and analyses of indicator diagrams from various engines of other types they satisfied themselves that with this particular engine and valve gear it was possible to increase the mean indicated pressure for the particular conditions to about 58lb. per sq. in. without any appreciable drop in efficiency. In this way the engine dimensions have been kept reasonably small and the weight is moderate. The high-pressure cylinders are $15\frac{3}{4}$in. in diameter, the l.p. cylinder is $39\frac{3}{8}$in., and the stroke, which is relatively short, is $31\frac{1}{2}$in.

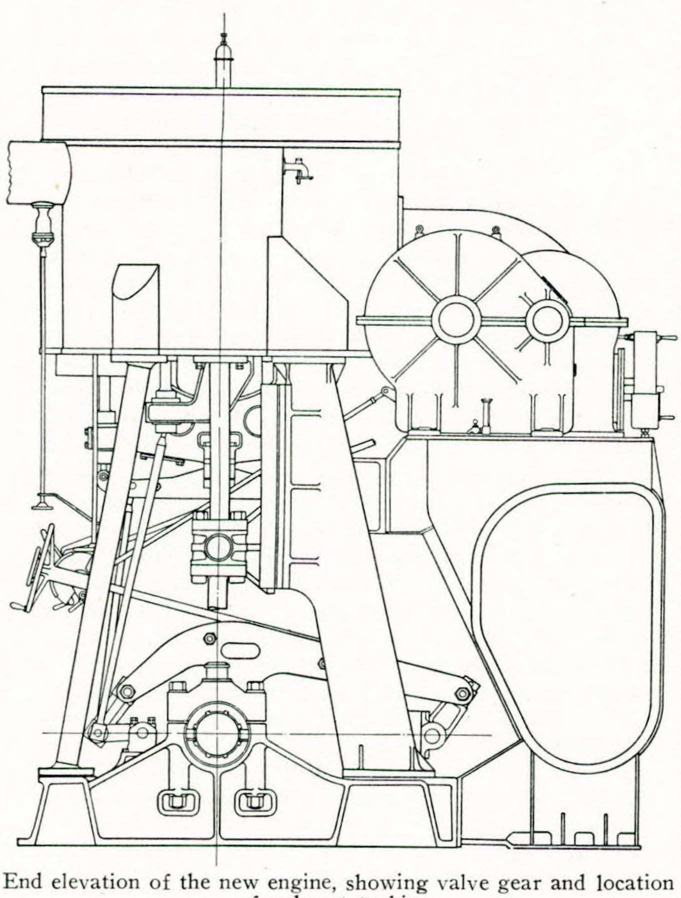
The exhaust steam is led into an Elsinore exhaust turbine similar to the device which we have already described in this paper, the Moss Vaerft & Dokk Co. being the Norwegian licensees for this exhaust turbine system. The combined output of the engine and turbine is equivalent to 1,700 i.h.p. at 108·5 r.p.m.

The calculated steam consumption at the above-mentioned output is about 8·9lb. of steam per i.h.p. per hour when using steam at a pressure of 235lb. per sq. in. and superheated to a final temperature of about 620° F. This performance is obtained with a cut-off in the high-pressure cylinders of 42 per cent.



Plan of the new Moss three-cylinder engine and Elsinore exhaust steam turbine, showing the short length and interesting valve locations.

The vessel in which this engine is now being installed will run her trials during the present month and as she is intended for service in tropical waters the condensing plant is large for the size of engine, a high vacuum being necessary in the interests of economy. The four-pass condenser has a cooling surface of approximately 2,000 sq. ft., while the extra large circulating pump can deal with 650 tons of water per hour. With such equipment a high vacuum should be assured, even if the sea temperature is appreciably above what is generally regarded as normal. At a later date we hope to give performance results for this interesting new machinery installation.



End elevation of the new engine, showing valve gear and location of exhaust turbine.

Coal as a Marine Fuel.

"The Shipbuilder and Marine Engine-Builder", March, 1937.

During the past 20 years the relative merits of coal and liquid fuel for the propulsion of merchant ships have been exhaustively discussed and analysed; and the desirability of fostering by every possible means the use of coal for British shipping, with the concomitant curtailment of our dependence — as a coal-producing nation—upon imported oil fuel, has been fully realised and unceasingly emphasised, in many cases by influential and disinterested individuals and associations. An excellent case has been made out from time to time for one or the other of these alternative fuels under carefully selected and specific conditions, but there are so many potent and varied geographical and economic factors surrounding the problem that no hard and fast lines of demarcation are really capable of being drawn.

Meanwhile there is no lack of statistical data pointing clearly to the relentless progress of oil fuel at the expense of coal. According to Lloyd's Register, coal-fired steamships declined from 45,338,327 tons gross in 1922 to 31,947,618 tons in 1936—a reduction of nearly 30 per cent.—while in the same period the tonnage relying upon oil fuel rose from 16,004,625 tons gross to 32,057,267 tons—an increase of over 100 per cent. For the first time in history more than 50 per cent. of the total world tonnage was in 1936 using oil fuel, either as Diesel oil in motorships or as boiler fuel in steamships. Since by common consent it would be difficult, if not impossible, to find any other form of business in which sentiment plays a more insignificant rôle than in shipping, or one which is so much exposed to ruthless international competition, the statistical picture painted above would certainly seem to suggest that in the eyes of the world's shipowners oil fuel is possessed of certain operating and economic advantages not enjoyed by coal as at present utilised and marketed.

It is to examine the possibility of arresting this decline in the use of coal as a marine fuel that an important conference was held in London last month under the chairmanship of Captain H. F. C. Crookshank, M.P., the Secretary for Mines. The conference was fully representative of shipping, coal and marine engineering interests, and the national aspect of the problem as it might affect British shipping in the event of war or world crisis was rightly stressed. The results of investigations recently carried out by the Marine Sub-Committee of the Combustion Appliance Makers' Association (Solid Fuel) were ably presented by Mr. Herbert Williams, M.P., in the form of a five-point memorandum. In that memorandum the almost unassailable advantages of oil for small coasting vessels of 600 S.H.P. and under and for large express passenger liners were frankly recognised and conceded. For a variety of ships between these two extremes, the potential economic advantages of coal-fired steamships were, it was suggested, worthy of more careful consideration than they sometimes received.

With regard to the final conclusion that "the reliability and efficiency of modern steam machinery are insufficiently appreciated, so that to some extent the preference for Diesel machinery represents a 'fashion' rather than the result of economic facts", this avenue of attack upon the strongly entrenched position of oil is, we fear, likely to be unproductive. When hard-headed British shipowners, after generations of experience with coal-fired steamships, are building and operating their latest vessels as motorships, the reason for such a change in policy must, we suggest, be traced to something less flippant and more solid than mere fashion. The factors at work here are wholly economic. If a shipowner finds, as clearly some do, that for a particular service a motorship, despite its admittedly greater initial cost, shows at the end of a year a greater margin of revenue over operating and standing charges than can be secured with a corresponding steamship, then clearly a more attractive standard of efficiency and economy of the steam engine, as well as cheaper-priced coal, will be required to pursuade him to return to the coal-burning and usually slower-speed steamship.

On whatever grounds the preference for Diesel machinery may have become an established vogue or fashion, it hardly appears to be strictly correct —of this country at least—to say that the reliability and efficiency of modern steam machinery are insufficiently appreciated. For a number of years past we have been building in this country a larger proportion of coal-burning steam-driven tonnage than have the majority of Continental countries and Japan, and many progressive British owners have spent large sums of money in rejuvenating existing steamships by applying the latest steam-machinery improvements. Despite these efforts, the advance of the motorship and the oil-burning steamship has in no way been checked, for the very obvious reason that the physical properties of oil are such that its storage and mechanical handling on board ship are so immeasurably superior to those of coal as at present utilised. If the volume of the world's coal-burning tonnage is to be increased—and there are a number of excellent arguments in favour of such an increase—the first essential is that the present almost universal and wasteful system of hand-firing should be superseded by mechanical processes. Further improvements in the efficiency of the steam engine will doubtless take place, but the scope in this direction is infinitesimal in comparison with that which is inherent in the firing problem.

According to the case as presented by Mr. Williams, there are only some 63 ships at sea to-day with mechanically-fired coal-burning boilers. The latest ship to be completed in this country with mechanically-fired coal-burning boilers is stated to be consuming only 0·9 lb. of coal per S.H.P. per hour; and if this excellent performance could be assured in general, there should be no lack of inducement to persevere with this promising line of development. Some of the Southern Railway Company's latest cross-Channel train-ferry steamships fitted with mechanical stokers have recorded the very high boiler efficiency of over 88 per cent.—a performance which is probably almost 20 per cent. better than the average of hand-fired practice. Where plentiful supplies of coal are available at moderate cost, performances such as these most certainly strengthen the case for the coal-fired steamship, and stamp the mechanical stoker as being the one factor most likely to increase the use of coal at sea. An alternative method of mechanical firing by the use of pulverised coal has been successfully developed for shore power plants on a large scale, but its progress for ships has been less marked. Some of the pioneer marine installations of a few years ago were gravely prejudiced by unduly flattering publicity before their development was sufficiently mature, and the system as a whole has never really recovered from that inauspicious start. Further research and development with pulverised-coal systems, with their associated problem of uniform distribution of the fuel to multiple-furnace installations, might well be undertaken now when the question of the security of oil sources of supply is rightly exercising some concern.

It is precisely in respect of the national aspect of the question as it affects British shipping and its fuel supplies in the event of war that this matter of the use of coal has now assumed an importance more vital than ever before. A very large proportion of what we have become accustomed to regard as neutral shipping, *i.e.,* Scandinavian and Dutch, is now dependent upon oil. In any large scale European war, oil and petroleum products must inevitably be at a premium. That country or consortium of nations which can ensure continuity of oil supplies for its sea, land, and air operations, while cutting off the corresponding flow of supplies to the enemy, must clearly triumph, and triumph quickly. The services of neutral shipping to this country will be contingent—as in the last war—upon our being able to provide fuel supplies, oil largely having taken the former place of coal. Our own shipping will have a prior claim on available oil supplies, and here it is obvious that every ton of shipping capable of using native coal will be doubly a source of national strength. Admiralty requirements of oil will be enormous, and fuel for mercantile motorships will take second place. Oil-burning steamships will, in such emergency, be compelled wherever practicable to revert to coal; and, fortunately, this change-over presents few mechanical difficulties, provided the necessary equipment is available in advance.

There are thus clearly the strongest national grounds for encouraging by every possible means the greater use of coal in the British merchant service. The Government, shipowners, marine engineers, and the coal interests have each and all a common concern in this objective. The great coal-owning and mining interests of this country, which for several years have encountered depressed conditions, stand to benefit greatly from the conditions now being created, if they will but bend to the task their utmost effort and goodwill.

Coal and Steam.

By H. L. Pirie, M.I.Mar.E., etc., Chief Engineer, Coal Utilisation Council.

An expert discusses the future of solid fuel afloat.

"The Marine Engineer", April, 1937.

Reference is commonly made in the Press and on public platforms to the competition between oil and coal for marine purposes. Actually, there are two questions involved:

(1) whether the vessel shall have steam engines or diesel engines; and

(2) whether the steam shall be produced by oil or coal firing.

In 1914, 96·6 per cent. of the world's mechanically-propelled tonnage of shipping used coal and 3·4 per cent. oil, but by 1936 this had changed to 50 per cent. using coal and 50 per cent. oil. In 1914 only 220,000 tons (0·45 per cent. of the world's tonnage) used internal-combustion engines; to-day the figure is over 12 million tons (18·8 per cent.).

Decline in the use of coal.

These changes have had a profound effect on the market for bunker coal. According to the annual reports issued by the Mines Department 21,030,000 tons of British coal were consumed as bunkers in ocean-going vessels in 1913. In 1936 the figure was 11,947,525 tons. Moreover, it must be remembered that much of our coal which is classed as "export" is eventually used for bunkers at foreign ports, and that the substantial decline in the tonnage of this "export" coal has adversely affected the volume of freight available to our ships on their outward voyages.

The Mining Association, The Coal Utilisation Council, and the Combustion Appliance Makers Association have addressed themselves to the problem, and the last-named organisation recently prepared a memorandum setting forth steps which might be taken to arrest the decline in the building of coal-fired ships. As a result, the Secretary for Mines, with the approval of the President of the Board of Trade, recently invited representatives of the shipping, shipbuilding, marine engineering, and coal industries to meet him to discuss what might be done to recover some of the bunker coal trade which has been lost. The conference led to a useful exchange of views and a committee, under the chairmanship of Sir Alfred Faulkner of the Mines Department, will be set up with the object of examining the matter in further detail.

The C.A.M.A. memorandum states that in the case of small vessels of low power—except in the British fishing fleet where coal retains a dominating position—and high-speed passenger liners where special considerations apply, diesel engines and oil-fired boilers, respectively, are probably superior to coal unless the relative prices of oil and coal are strongly in favour of the latter.

Where the steamship scores.

As power rises the steam-cycle becomes increasingly favourable as compared with that of the internal-combustion engine, and for vessels of from 1,500 to 8,000 S.H.P. steam engines, the writer considers, are to be preferred to diesel engines. In order to get the best results out of coal at sea the following conditions must be fulfilled:

(1) The boiler plant must be thoroughly up-to-date in all respects, and efficiency can be obtained in smaller vessels by the use of modern furnace equipment and in the larger vessels by employing mechanical stokers.

(2) Full advantage must also be taken of modern steam propelling machinery.

Three steamship groups.

Modern boilers and propelling machinery for coal-fired steam ships can be classified in three main groups:

(1) That in which steam is raised in Scotch boilers, but superheaters and air-heaters are added. Reciprocating engines are retained, but exhaust steam feed-water heating increases the overall efficiency. Improved types of Scotch boilers and forced draught or steam-jet furnaces will increase the combustion efficiency as compared with the simple hand-fired Scotch boiler.

(2) Vessels with mechanically-fired Scotch boilers of improved types which are usually fitted with high efficiency furnaces. Exhaust steam turbines are used in conjunction with the main engines, the auxiliaries, with the exception of feed and circulating pumps, being normally driven by electricity. Two-stage feed water heating is generally employed.

(3) Vessels in which the increased overall efficiency is obtained by means of higher temperatures and pressures. They are fitted with mechanically-fired water-tube boilers operating at steam pressures up to 400lb. per sq. in. and superheat up to 700° F. The prime movers consist of geared turbines; all auxiliaries are electrically-driven and multi-stage feed-water heating is employed.

Whereas the average consumption of the older type of coal-fired steamships afloat is at the present day about 1·44lb. of coal per shaft horse-power per hour, that of the more modern vessels fitted with up-to-date installations is less than 1lb.

Within the last few years 63 merchant ships have been fitted with mechanical stokers of various types, but all these ships are running on well defined runs and operate on "standard" coals.

Buying bunker coal.

In some quarters the view has been expressed that those responsible for the supply of bunker coals have not always done all they might to ensure their suitability and uniformity, nor advised the shipowner as to the best way to use the coal. On the other hand price rather than suitability appears to be the shipowner's main consideration when purchasing bunker fuel, and it is felt that they would be better advised to pursue a policy of buying coal on a value for money basis. Much work remains to be done in determining the suitability of coals for use under different conditions. A coal which may be perfectly satisfactory when burned under natural draught may not prove suitable when burned under forced-draught conditions. Investigations might be conducted to ascertain the most suitable specifications relating to grading, ash-fusion temperature, moisture and caking power so as to produce greater uniformity of heat-producing power measured by the evaporation of water per lb. of coal.

Price.

One further word in conclusion about price. Shipowners are apt to complain that the price of bunker coal is rising. They seem to forget, however, that the price of oil is also showing a tendency to rise, although that tendency may be more noticeable at present in the case of petrol than of diesel oil. But a diesel-engined vessel cannot be converted to use steam except at great expense, and in a few years time the shipowners who are now busily ordering diesel-engined ships may possibly be wishing that they had not put all their money on oil, and had paid more attention to the claims and possibilities of Britain's national fuel.

Copper-Nickel Condenser Tubes.

"Shipbuilding and Shipping Record", 11th March, 1937.

Until the advent of the turbine-driven steamship, with its high-vacuum condenser involving the use of high velocities for the circulating water, tubes made of Admiralty mixture (70 Cu, 29 Zn, 1 Sn) were usually found to give completely reliable service. The incidence of corrosion and pitting, often very severe in its nature, which was a feature of many of the condenser tubes of the earlier turbine ships, led to the trying out of various corrosion-resisting alloys and of these, the copper-nickel alloys of approximately 70-30 composition were ultimately found to yield the most satisfactory results. As a consequence, their adoption has become almost standard practice for all high-class steamships both in the merchant fleets and the navies of the world. Their use, however, is not entirely unaccompanied by troubles, particularly in condensers where at either one or both ends, the tubes are expanded into the tube plates instead of being fitted by means of packing and screwed ferrules. In particular, the U.S. Navy Department found that with a certain consignment of copper-nickel alloy condenser tubes manufactured in accordance with the most rigid specification, a number of failures (representing about 0·5 per cent.) were experienced, due to splitting at the ends while the tubes were being belled into the tube plates. An investigation was therefore undertaken to determine what factor or factors were responsible for the poor cold-forming characteristics of the material, details of which, together with the results obtained, were communicated to the *Journal of the American Society of Naval Engineers* by Mr. Joseph A. Duma, of the U.S. Navy Yard, Norfolk, Va., who was responsible for the investigation.

After describing the circumstances which led to the investigation being undertaken, the author gives a summary of the tests carried out. These included cold-working tests by means of drift pin expansion, rolling and belling, and by flattening; mechanical tests in order to determine the tensile strength, elongation and hardness of the material; corrosion tests in mercurous nitrate and boiling salt water, as well as the determination of the chemical composition of the material, microscopic examination of etched surfaces and heat treatment experiments. In all, thirteen tubes, each ⅝in. external diameter and about 10ft. long, were selected for the tests, these comprising six defective tubes from the consignment with which trouble was being experienced, one satisfactory tube from this batch and a further six taken at random from a new lot of unused tubes. With all the tubes examined, it was found that except in one instance, the alloy consisted only of copper and nickel, and as far as the tests showed, zinc, iron, phosphorous, lead and tin were entirely absent as also was manganese, except in the case referred to above, where it was present to the extent of 0·14 per cent. In this case, it may be noted, the tubes were of an unsatisfactory nature. The relationship between the copper and the nickel varied, however, to a considerable extent in the different tubes tested, the highest copper content being 81·57 per cent. with

18·43 per cent. of nickel, and the lowest having 73·89 per cent. copper with 26·11 per cent. nickel.

Broadly speaking, the ability to withstand the effects of cold working is shown to depend upon the copper content. Thus, tubes which have a copper content in excess of 80 per cent., despite their greater softness and superior ductility, are not able to be cold formed to the same degree as those containing from 65 to 75 per cent. of copper, while the highest combination of strength and ductility—65,000lb. per sq. in. and 25 per cent. elongation in 2 in.—is obtained when the copper content is at about 70 per cent. Like so many other investigations, this one yielded much unexpected information concerning the more fundamental properties of the alloys of copper and nickel. Thus, experiments were undertaken to determine the correct annealing temperature, specimens being heated to temperatures ranging from 1,000° F. to 1,650° F., held for 30 minutes at the desired temperature and then quenched either in cold water or in still air. It was found, as a result of these, that optimum physical properties are obtained from a 1,550° F. water-quenching treatment although as regards the flattening test, the 1,650° F. water-quenched specimens are better. The author therefore concludes that 1,600° F. is probably the best temperature for annealing. Subsequent experiments to show the effect of age hardening for a period of six hours at different temperatures ranging from 200° F. to 1,200° F. showed that the material is age-hardenable to a very considerable degree, the Vickers Brinell hardness number increasing from 50 after treatment at 200° F. to 90 after treatment at 1,000° F., after which it decreases. It should be noted, however, that all the alloys are not susceptible to age hardening to the same extent, and it appears as though the extent to which the material has been previously cold worked affects the extent to which it can be age hardened. Generally, however, it may be concluded that the best cold-forming properties are to be found in tubes of pure copper-nickel alloy containing not more than 70 per cent. copper, although if small percentages of other elements are present such as carbon, silicon, manganese, iron, aluminium, and zinc, greater amounts of copper can be employed, but never in excess of 80 per cent.

Corrosion in Sea Water.

"Shipbuilding and Shipping Record", 11th March, 1937.

The work which has been carried out by a committee of the Institution of Civil Engineers during the past 20 years into the deterioration of structures of timber, metal and concrete in sea water, still continues, and the sixteenth (interim) report which has just been issued* continues the account of the work which is still in progress. It is of interest to note that timber specimens exposed at Plymouth in 1924 have now been recalled and examined so that the oldest series extant is that exposed in 1925 at Singapore, Wellington and Auckland. These were creosoted by the Bethell process with the addition of various quantities of D.M. (chlorodihydrophenarsazine) up to 0·3 per cent., and of carbazole up to 5·3 per cent., and are still showing in general a good resistance to toredo. The specimens from Plymouth were also in good condition, but those protected with less than 0·3 per cent. of D.M. contained toredo channels. Of particular interest among the other matters reported upon, is the effect of the weight of zinc coating upon the resistance to corrosion of galvanised steel plates exposed at half-tide and below low-water levels at Southampton. It was found that the heavier the coating of zinc the better the protection, and that the specially-prepared plates employed by the committee had a coating of 2·33 oz. per sq. ft.

*Deterioration of Structures in Sea Water; Sixteenth (Interim) Report, Department of Scientific & Industrial Research. London: H.M. Stationery Office, Adastral House, Kingsway, W.C.2.

The Case for Propulsion Amidships.

By John De Meo.

"Shipbuilding and Shipping Record", 29th April, 1937.

The interesting series of photographs which were published in "Shipbuilding and Shipping Record" last week showing the various changes which had been made to the propellers of the "Normandie", serve as a reminder that, notwithstanding all that has been done experimentally, the problem of propulsion has not yet been solved. This encourages me to outline the case for the consideration of central propulsion, which I have advocated for many years and which has been the subject of considerable research.

It is indisputable that there are many disturbing factors which prevent a proper functioning of the screw propeller which, in the orthodox position, has to work in the zone of wake turbulence. There is considerable loss of efficiency, together with the loss in the thrust power of over 10 per cent. of the engine output, the latter loss being identified from research work as the thrust deduction and is mainly due to the screw suction behind the hull.

Pitching is another source of loss, whereas if the propellers were suitably placed amidships, these impediments and disturbances would be eliminated. Extensive investigations and experiment tank researches have confirmed that screw efficiency can be greatly enhanced where the propellers work in the calmer water amidships.

I submit, further, that the problem of vibration, which originates mainly from the combined action of the machinery, the propellers working in the turbulent race of the wake at the stern, and the whipping of the shafting, induces vibration, which, I contend, would be eliminated if the propellers were suitably placed amidships.

There are many practical advantages and economies to be derived from my suggested treatment of propulsion, and they may be outlined as follows:—

(1) Reduction of first cost of hull construction through a simplified design and a lower cost of the propelling machinery through dispensing with the shafting, bearings and supports, shaft tunnels, and connected hull structures under present conditions.

(2) Economy through a substantial saving of internal space in the lower-deck after-holds, abaft the machinery space, resulting from the elimination of the shaft tunnels, with a corresponding increase of deadweight capacity.

(3) The extra available space for cargo or other accommodation leaves undisturbed the quarters for passenger cabins, free from the nuisance of hull and propeller vibration.

(4) Economy arising from improved propeller efficiency and from the betterment of hull efficiency, together with the elimination of the losses from actual thrust deduction or wasted power (altogether due to the exclusive characteristics of the screws working in the undisturbed water amidships). This will increase the vessel's speed compared with the actual rating of the engine power necessary for the working of propellers at the stern.

(5) Economy of engine power now wasted by friction of shafting and tunnel bearings.

(6) The saving of fuel, oil, and provisions for passengers and crew from the mileage now lost through pitching, which will not affect the screws as immersed amidships. Moreover, the smooth running of the propellers is ensured in calm water whatever the state of the seas.

(7) Appreciable reduction of first cost of hull construction due to a simplified design with equal and symmetrically disposed transverse sections, fore and aft of amidships, and to the saving in constructional costs, time, and calculations.

(8) Dispensing with the existing cumbersome rudders and steering engines, as turning the vessel is facilitated by the increased moment of the engine power upon the propellers placed amidships. This offers a powerful and quick means for turning without the assistance of tugs for manœuvring in harbours or approaches. Moreover, risks of collisions would be minimised.

(9) Economy of running costs for ships at sea arising from the elimination of possible damage to the propellers from striking submerged obstacles, or of steel hawsers often involved in the screw-race turbulence at the stern; also, the elimination of the risk of breaking propeller blades and tail-shafts in consequence of straining or metal fatigue from the continuance of pitching.

(10) Economy by the effective increase in speed because of the utilisation of the excess engine power now needed for propulsion; the consistent running of amidships propellers, with an absence of racing, as they are undisturbed by pitching, would effect economy of fuel and improve the daily mileage runs.

This new system of ship propulsion is the outcome of prolonged study of the major problems of the subject, particularly in connection with seeking a solution of the problem of vibration and of damage which arises from wear and tear such as is familiar to all who have had experience of surveys and repairs.

Sixty Years Ago.

"The Engineer", 9th April, 1937.

"The Story of H.M.S. Tourmaline" detailed in a leading article in our issue of April 13th, 1877, provided, so we maintained, a striking additional instance of the inefficient conditions prevailing in the engine-rooms of her Majesty's ships. Mishap had followed mishap and in some cases, as in that of the "Thunderer", mishaps had become catastrophes so much so that the First Lord of the Admiralty, Mr. Ward Hunt, had apparently given up as vain the task of explaining away the successive failures of engines in men-of-war. The "Tourmaline" was a composite screw corvette of 1864 tons burthen and 1972 indicated horse-power. She was commissioned in October, 1876, and was ordered to the West African Coast station. On trial off Sheerness there was trouble with a hot bearing. The bearing was patched up with white metal but off Deal it again gave trouble. Next the tank in connection with the hotwell gave way, distilled water could not be obtained for the boilers and sea water had to be used instead. Very soon the boilers started to prime and the engines stopped dead. They were re-started with difficulty but the trunk of the after cylinder gave way. The spare trunk was fitted but the vacuum failed. On examination it was found that several hundred of the condenser tubes had started out of the tube plates. Five days were spent in the Bay of Biscay making the necessary repairs, the ship in the meantime being kept going on the high-pressure cylinder only. Next the low-pressure slide valve gave trouble, six sets of liners being used up, each set lasting only about a day. The pistons gave constant trouble. New springs were fitted in the low-pressure cylinder but they had to be taken out because when they were in place it was found that the engine would not turn round. The screw was of the lifting type but it was so badly fitted that before it could be raised or lowered the thrust block had to be slackened. Four hours were required to get the screw up or down. Next the helical pump employed for circulating water through the condenser developed a defect as a result of absence of lubricating means and was responsible for a noise which could be heard throughout the whole ship. On arrival at her destination it was found that the cross-head guides had lost a quarter of an inch by wear, that the back trunk guide was cracked and that the piston-rod glands once they had been removed could not be replaced in their boxes. Finally worry, fatigue and want of rest drove the chief engineer to attempt suicide. He was saved for a time from the consequences of his rash action but, deprived of rest by the noise in the engine-room, he died in a few days of exhaustion and anxiety. We withheld the names of the makers of the ship's engines and boilers, but we made it clear that they were not necessarily to be blamed for this series of mishaps. We recognised that in all probability they had been compelled to comply with almost impossible conditions and in particular had been forced to fit their engines into a space totally inadequate to accommodate the machinery.

Improvements in Safety-valve Clacks.

By G. H. Pearson, A.M.I.Mech.E.

"The Steam Engineer", May, 1937.

The prevailing method of effecting guidance of safety valve clacks, namely, the provision of webs on their undersides, free to slide within the seating as shown in Fig. 1, is open to severe criticism, and many instances could be recalled of serious explosions—in some cases attended by serious mishap and even loss of life—due to the imperfect functioning of the safety valve by reason of such webs becoming gummed fast by incrustation and other causes.

The provision of a testing device, whereby the valve may be eased off its seating periodically, and

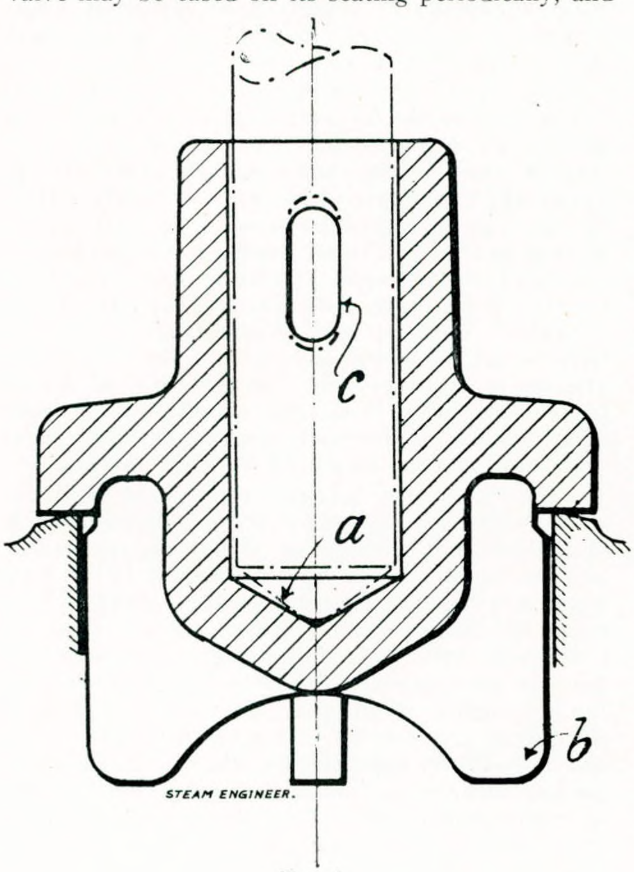


Fig. 1.

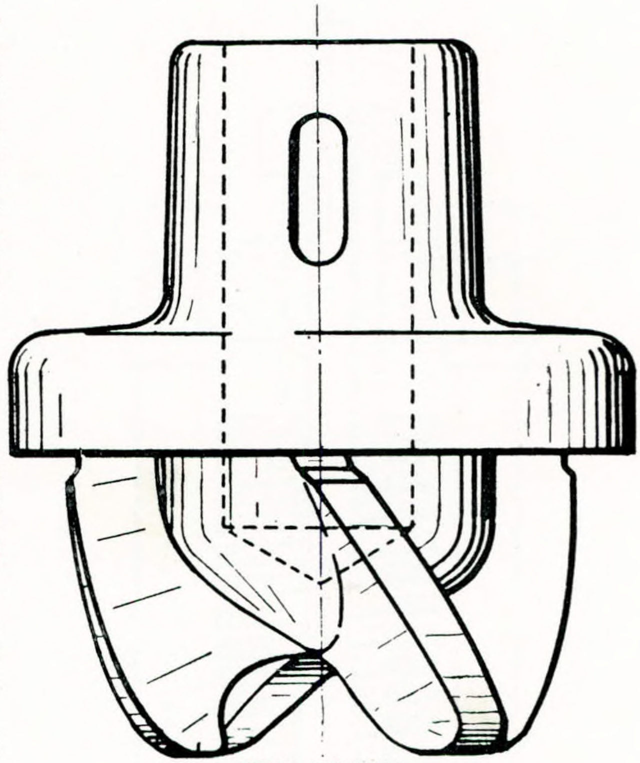


Fig. 2.

the framing of rules to ensure that this operation is carried out, would do much to lessen the chances of this state of affairs arising, and certain authorities, notably the Board of Trade, insist on some form of lifting gear being incorporated on all safety valves coming under their jurisdiction.

In the choice of a suitable clearance allowance between the webs and the seatings, opinions appear to be divided; some authorities favour small clearances of the order of a few thousandths of an inch, whilst others go to the other end of the scale and demand allowances which are not only extravagant, but definitely useless, since their extent is such as to render the valve liable to the possibility of rocking over on its seating face, resulting in the bearing surfaces becoming rapidly impaired by erosion due to the high velocity of the steam in passing through such a restricted opening as is presented by the relatively microscopical lift of any safety or reducing valve clack in normal operation.

The above are but two outstanding disadvantages accruing from the adoption of webs on the underside of the clack; others may suggest themselves to those whose duties bring them into intimate contact with such fittings, but sufficient has been already stated to deplore the afore-mentioned practice, and the following suggestions for improvements are advanced in the hope that these may be helpful to the designer of such equipment.

Let it be clearly stated at the outset, however,

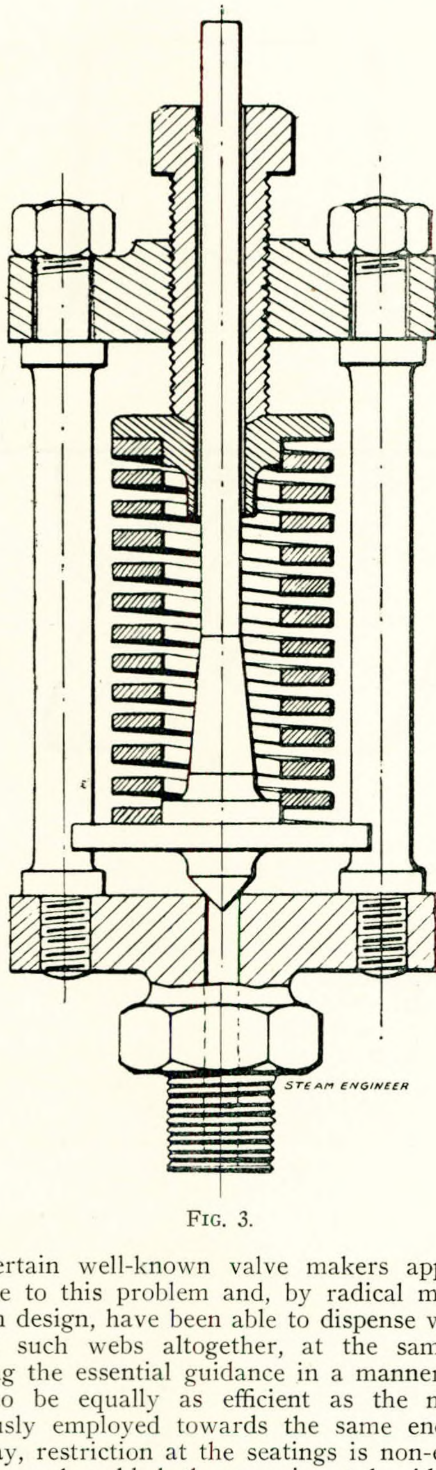


FIG. 3.

that certain well-known valve makers appear to be alive to this problem and, by radical modifications in design, have been able to dispense with the use of such webs altogether, at the same time effecting the essential guidance in a manner calculated to be equally as efficient as the methods previously employed towards the same ends. In this way, restriction at the seatings is non-existent and, it may be added, the experimental evidence of certain investigators has shown that not only do webs situated in the exhaust thoroughfares restrict the discharge capacity of a safety valve, but, what is equally disadvantageous, cause the accumulation figures to rise to an alarming degree, a most undesirable feature especially on boilers of the quick steaming type. This is quite a serious matter, and the aim of all designers should be to produce a valve that will open promptly at the set pressure, discharge its quota of steam with as little accumulation as possible and reseat without "feathering", within a few lb. per sq. in. of the set pressure. This ideal is very nearly realised in well-designed valves of the "pop" type now being almost exclusively fitted on modern locomotive boilers, although their use, of course, is not confined to this particular type of boiler.

Reverting to the material points of these notes, the type of clack to which the foregoing criticisms are directed is generally found to be of the form shown in Fig. 1, and if this form is to be employed at all the following points should be rigorously observed. In the first place, one of the chief essentials of a clack of this form is that the end abutment (*a*) of the valve spindle should be situated at some distance below the level of the seating face in order to minimise the tendency of the clack to tilt under the influence of the axial load on the valve. This tendency is more pronounced in the case of valves of the lever and weight type since the load is not purely axial, as is the case in a direct loaded spring safety valve.

The lower edges of the guide webs should be well rounded as at (*b*) with a radius equal to their thickness (or slightly less), as *sharp edges* would assist in such wedging should the clack become tilted from any cause. It is the writer's opinion that the weakening effect of the cotter-hole (*c*) is responsible for many valve failures of this nature, and instances have been encountered from time to time of the valve spindle being twisted (especially in the case of marine type safety valves) by too rigorous application of a spanner to the turning gear which is provided for the purpose of rotating the clack upon its seating, and some other means of attaining this objective appears to be desirable. As the present Board of Trade Regulations stand, it would appear that the only other alternative is to provide for a spindle of liberal proportions and of a suitable material; a cast gun-metal spindle is practically useless, yet this seems to be the material employed by some makers.

The clearance between the spindle and the clack, whilst providing for some amount of side play, should not be excessive, $\frac{1}{32}$in. being reasonable, and the depth of the housing should be at least equal to twice the diameter of the spindle. It is usual on clacks of this type to include three or four guide webs providing straight line contact in the seat thoroughfare, but it is better practice by far to employ spiral webs, as shown in Fig. 2, since these give contact over a much greater length and more nearly approach the ideal of a cylindrical guiding member, although this latter ideal would be impracticable since it would restrict the area of escape. Such a clack naturally calls for some little extra skill both in the making of the actual pattern and also in the moulding process, but this should not be a deterrent having regard to the many advantages it possesses over the prevailing type already described.

In valves of exceptionally small bore, such as are employed on high-pressure air-receivers and the like, the use of webs becomes almost impossible since these would be of such diminutive proportions as to have no well-defined strength, and even so would restrict the flow of escaping fluid to a marked degree.

The drawing illustrated in Fig. 3 shows how this difficulty was overcome by the writer in a ¼-in. bore relief valve working at 2,000lb. per sq. in. on an air line supply to a compressed air-testing plant, and which has given every satisfaction. It will be seen that the valve and spindle are in one piece, and guidance is effected by two lugs thrown off the spring plate and arranged to slide freely along the pillars. Thus it will be seen that this arrangement of guiding the valve permits of occasional lubrication of the sliding surfaces, a feature which is impossible of attainment in a valve whose means of guidance is within the working fluid itself. The valve shown, whilst possessing many commendable features, suffers at least one disadvantage in that it is impossible to impart any rotation to the valve member, but this could easily be overcome by a modification to the spring plate, which could be arranged as a loose member resting on a collar formed on the spindle, and the latter provided with a square termination for engaging a spanner.

In the case of controlled discharge spring safety valves, that is, valves in which the discharge is piped off instead of being allowed to exhaust to atmosphere, the arrangement shown in Fig. 4 has been found to give every satisfaction. The upper portion of the clack is in the form of a piston, with labyrinth grooves, and is free to slide in a cylinder situated in the neck of the valve body. In addition to providing an ideal arrangement for the effective guidance of the clack, somewhat increased lift of this member at blow-off is obtained since the back pressure—often amounting to something like 30lb. per sq. in.—exerts a force on the underside of the piston which, being of greater area than that of the seat opening, results in a somewhat increased lift. Objections might be raised to this arrangement on the grounds that it is possible for the piston and cylinder to seize up, especially when the valve is subjected to high temperature as in a superheated steam range, but this is a matter calling for some amount of careful thought in assigning suitable working clearances having regard to the materials employed for the parts in contact. Ordinarily the clack is of nickel alloy, of which there are many excellent proprietary brands on the market, and the cylinder of gunmetal (for low temperatures) or Monel metal where the conditions are more severe. Similar metals for both clack and cylinder should not be tolerated since, under conditions of high temperature, these are more prone to seizing up than would be the case with dissimilar materials.

The excellent results obtained by the adoption of this particular arrangement would appear to belie the various criticisms levelled at its adoption by certain authorities. It is surprising how anything in the nature of a definite departure from hitherto generally accepted standards is met with such a volume of disapproval as to make designers think twice before proffering anything likely to appear arrestingly novel. It is as well, therefore,

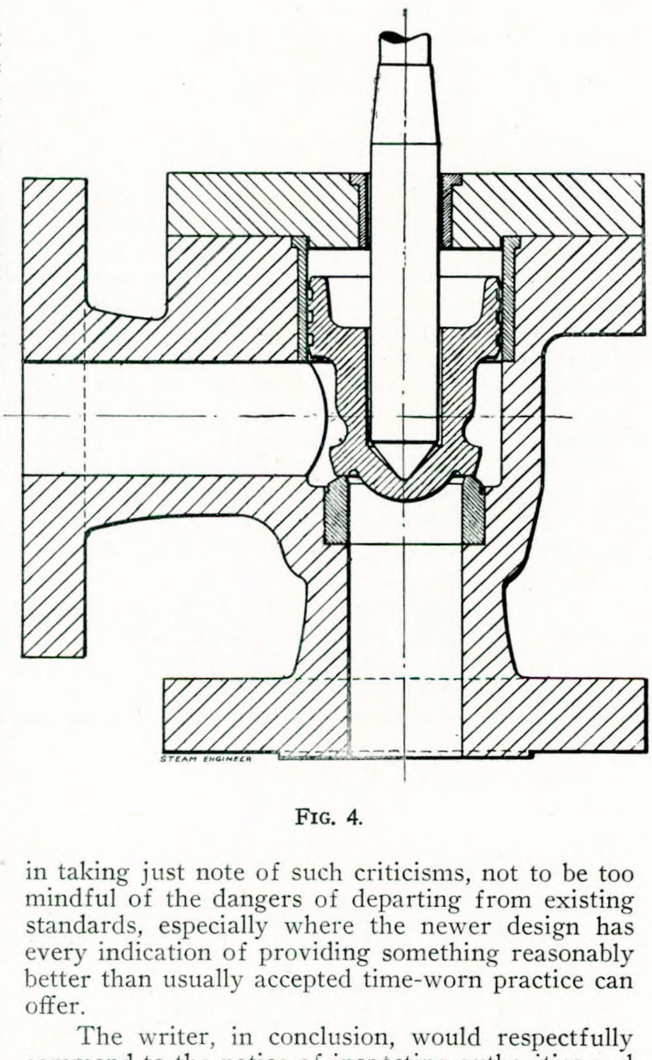


FIG. 4.

in taking just note of such criticisms, not to be too mindful of the dangers of departing from existing standards, especially where the newer design has every indication of providing something reasonably better than usually accepted time-worn practice can offer.

The writer, in conclusion, would respectfully commend to the notice of inspecting authorities and others the necessity for an all-round improvement in the design of the particular apparatus outlined in these notes, improvements which—if length of satisfactory service and improved performance are any criteria—leave no shadow of doubt as to their superiority over the more prevalent methods adopted hitherto.

Considerations Affecting Design, Manufacture and Erection of Steam Mains for Modern Plants.

By A. G. Bugden, B.Sc., A.M.Inst.C.E., etc.

Abstracts from a paper read by the author before The South Wales Institute of Engineers.

"The Steam Engineer", May, 1937.

Whilst considerable attention has been focused upon the design of machinery for the generation of steam and of power, it is extaordinary how little consideration is given to the means of transferring the working fluid from the boiler to the prime mover, i.e. the steam pipework.

The design and layout of the piping for a modern plant requires careful and early attention in order that a satisfactory arrangement may be provided. The layout of the steam mains can so seriously affect the cost as well as the satisfactory operation of the system that careful planning is of paramount importance.

Modern temperatures practically preclude the use of cast iron due to the change in the graphitic carbon at about 500° F., and the methods of manufacture of cast-iron pipes are such that considerable risk of failure in operation is incurred. It is, in the author's view, very regrettable that cast-iron pipes for feed-water delivery to boilers are still found in some plants, having been installed in preference to steel pipes on account of their greater resistance to corrosion. The fact is that the corrosion is merely transferred to some other (and probably less convenient) point in the system, such as the boilers, and this has been substantiated in practice, particularly where corroded steel mains have been replaced by cast iron. A recent fatal accident in an industrial plant operating at only moderate pressure has caused engineers in charge of plants with similar systems to give serious consideration to this matter.

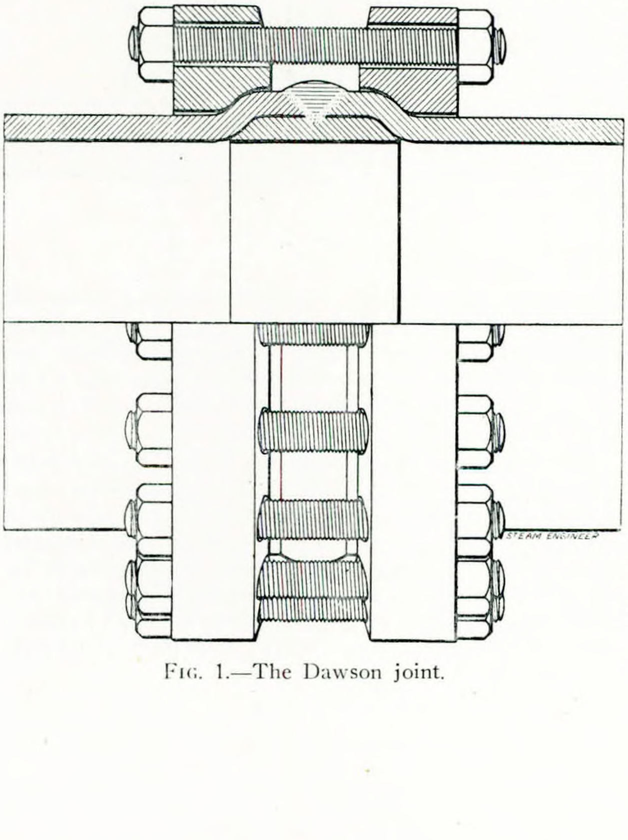


Fig. 1.—The Dawson joint.

Cast steel is used to an ever-decreasing extent due to the advantages and reliability of welding in the fabrication of such details as bends and tees, although the use of the latter is frequently obviated by the welding on of branches.

Most of the items comprising a high pressure pipework system can, therefore, be in mild steel, and it is only when temperatures above 900° F. are encountered that the necessity arises for tubing of special alloy.

Steam pipes in modern practice are of mild steel having an ultimate tensile strength of 23 to 30 tons sq. in. with about 20 per cent. elongation in 8in., and tubes of this material may be either welded or (in certain ranges of size) seamless.

The increase in steam conditions has drawn particular attention to the methods of connecting together the lengths of pipes comprising a system. The obvious ideal would be continuous pipes without joints, but due to limitations of manufacture, transport and erection, this ideal cannot be attained, and if it were, doubtless some engineers would raise the objection that they could not readily effect alterations in the layout of their mains. Many designs of special pipe-joints have been devised and patented, but very few are in common use, and for steam pipes these do not number more than half a dozen. The commonest method of joining pipes is the flange method, flanges being secured to the pipes, faced and drilled for bolts, gaskets of various jointing materials being used between the faces. Such joints gave reasonable satisfaction in the past for moderate pressures and temperatures, but where high temperatures are encountered the effects of creep in the bolts may be serious, ultimately resulting in leaky joints. Such defects will be aggravated by unsuitable arrangements for expansion and supports, and there are many cases where these leaky joints are looked upon as an inherent feature of any steam main.

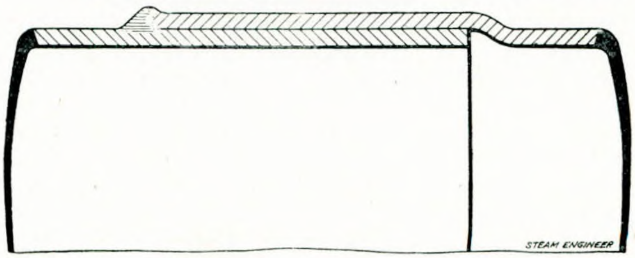
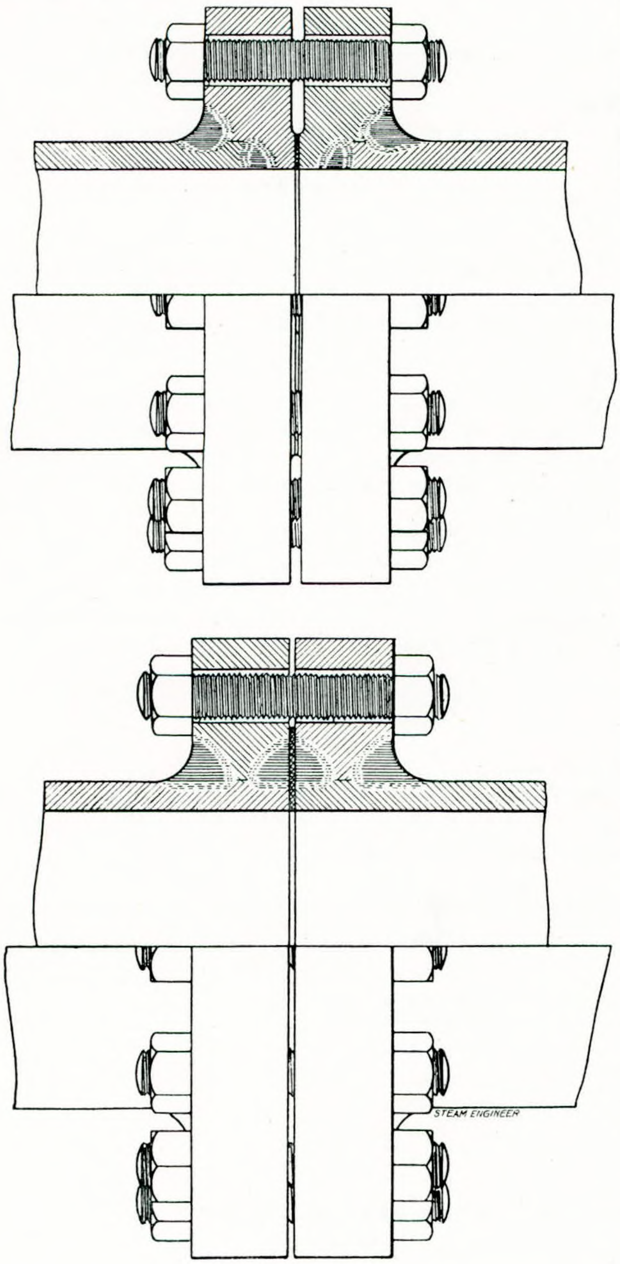


Fig. 2.—The Stewarts' long sleeve welded joint.

Now that the welding processes appear to have become established, and with the considerable development of the technique of welding in recent years, structures connected by these processes can be accepted with a fair degree of confidence. In pipework particularly considerable developments have occurred, and the installations in which welded constructions are adopted have proved entirely satisfactory over a number of years of service, often under high pressure and temperature conditions. Welded joints have removed much of the annoyance of maintenance work inherent in steam mains having bolted joints and gaskets.

A most excellent welded joint known as the Dawson Joint is illustrated in Fig. 1, from which it will be noted that it is an all-welded joint, almost fulfilling the ideal condition mentioned above in that the main can be completely welded up, including, if necessary, the valves and other fittings. The



FIGS. 3 and 4.

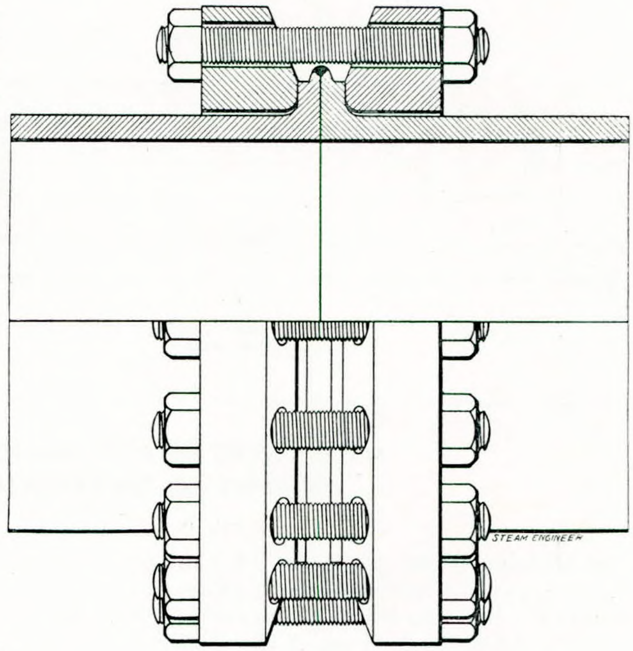


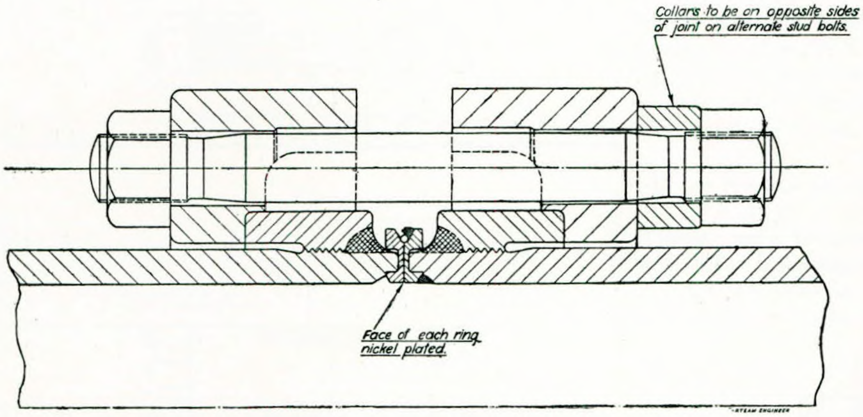
FIG. 5.—The seal-weld joint.

figure shows the joint in part section, and clearly indicates details of construction.

Where pipes having the Dawson Joint have to connect to a valve, an extended spigot is provided on the valve flange, and an internal sleeve is furnished in the ordinary way, the only difference being, of course, that the valve flange is fixed instead of loose. The Dawson Joint has been supplied in many power plants for steam pressures and temperatures up to 675lb. sq. in. and 900° F., and is entirely suitable for all steam conditions for which mild steel tubes can be used.

Another very satisfactory joint is the sleeve weld joint indicated in Fig. 2, from which it will be noted that a sleeve is formed in one pipe, and the end of the adjacent pipe is forced into this sleeve, a seal weld being afterwards added. This joint is suitable for moderate steam pressures, and particularly for pass-out steam systems.

Where ordinary bolted flange joints must be adopted, the flanges may be welded or screwed on to the pipes, but the former method must be adopted where temperatures are in the region of 800° F. or above, as at these high temperatures, due to the temperature gradient between pipes and flanges, there is a possibility of the screwed flange becoming slack on the pipe, and where any vibration occurs in the steam main this effect is greatly aggravated. For temperatures below 750° F., however, the screwing method is satisfactory, but the pipes must be afterwards expanded into the flanges by means of a roller expander. It is inadvisable to add any welding to the back of the flange after this process, as this welding destroys the effect of the expanding, and the only excuse for specifying



M-V PIPE JOINT FOR EXTRA HIGH STEAM PRESSURES AND TEMPERATURES.

FIG. 6.

this addition is the possibility of torsion occurring in the pipework, but even here, unless the torsional forces are likely to be of a high order, this addition is to be deprecated. Figs. 3 and 4 show two types of flanged joints.

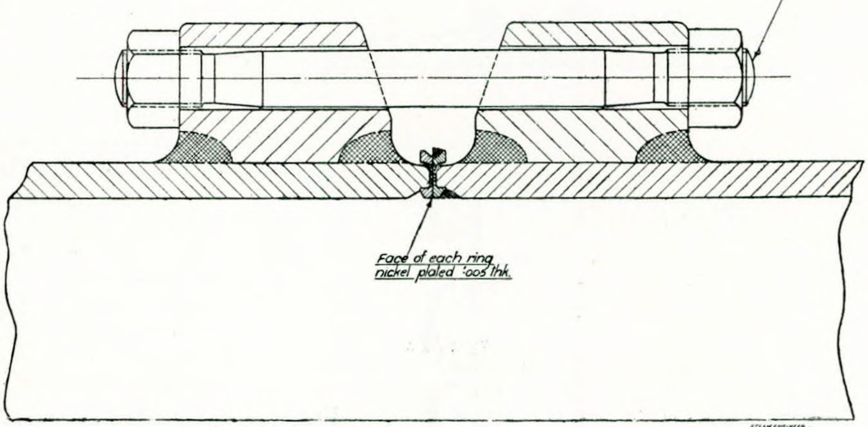
For large pipework, and sometimes in marine installations, riveted-on flanges are still used, although to an ever decreasing extent, and this method is not suitable for high temperatures, or for tubes under 7in. bore, it being generally more expensive than the more common methods of attachment, e.g. screwing or welding. Where riveting is specified, the flange should be shrunk on to the pipe, then secured by rivets of the raised countersunk type (the heads being inside the pipe), and the back of the flange then thoroughly fullered, the tube ends being hammered up on the front to provide a slight bell-mouth. For low pressures riveted-on flanges were formerly almost universally employed, but they have now been largely superseded by welded-on flanges either of the angle or the plate type.

In steampipe flanges where the bolts are subjected to the full steam pressure and temperature—and this applies to seal-weld joints equally with ordinary bolted flanges—it is necessary to give careful consideration to the question of the material of the bolts. For pressures not exceeding 450lb. and temperatures not above 750° F. the bolts may be standard headed bolts of carbon steel of 28 tons minimum tensile strength; for pressures exceeding 450lb. but not exceeding 900lb. and temperatures not over 750° F. the bolts should be of carbon steel of tensile strength not less than 35 tons; where pressures are above 450lb. and temperatures exceed 750° F. (and for all pressures above 900lb. whatever the temperature) the bolts should be in the form of studs and of special alloy steel having a tensile strength of not less than 65 tons and screwed for the whole of their length British Standard Fine Thread, nuts being in all cases of carbon steel. The alloy for the bolts may have an approximate analysis of

C 0·5%   Mn 0·6%   Cr 1%   Mo 0·6%

The material should be heat-treated, no subsequent forging being permissible. In modern high-duty steam joints, and particularly in the seal-weld types, the bolts are very long—for a 10in. joint they may be as much as 14in. in length—and under starting-up conditions serious stress may occur. The resilience of the bolts is therefore important and hence studs continuously threaded (or preferably with the threads between nuts turned off) should be fitted.

The ordinary seal-weld joint, one form of which is illustrated in Fig. 5, has found favour with some engineers. It will be seen that this type consists essentially of a loose flange construction, in which the ends of the pipes are staved up and



M-V PIPE JOINT FOR EXTRA HIGH STEAM PRESSURES AND TEMPERATURES.

FIG. 7.

afterwards faced. In erecting pipe-work having such joints, it is necessary to ensure that the faces are in contact over the whole surface, and to harden up the bolts to a sufficient extent to ensure pressure tightness before the weld is added. After being bolted up the welding is proceeded with, the customary method being to remove one bolt at a time and after welding a sector of the circumference to re-insert the bolt, hardening up the nuts as each section of welding is completed. It will be seen that, due to the high initial tightening stresses in the bolts, the effects of creep can be serious, especially during the starting up of the plant when the interior of the pipe is raised to a much higher temperature than the flanges and bolts, thereby causing the pipe to expand and impose an even greater load upon the bolts, producing excessive stress, and resulting in the slackening of the bolts.

An improved form of seal-weld joint is the Metropolitan-Vickers joint, shown in Figs. 6 and 7, which embodies special rings welded to the pipe ends, the flanges, secured to the pipes by approved means, being bolted together, the nuts being hardened up to resist the internal pressure. A light seal weld is added in the manner described above for the ordinary seal-weld type. It will be noted that as creep occurs in the bolts, axial movement in the pipes can be accommodated by the flexibility of the joint rings, the faces of which part slightly, and to obviate corrosion these faces are nickel-plated.

The Velox Steam Generator for Merchant and Naval Vessels.

By Dr. Adolphe Meyer.

(American Society of Naval Architects and Marine Engineers; November, 1936).

"The Shipbuilder and Marine Engine-Builder", April, 1937.

In an article on "Light Boilers for Fast Ships" Dr. Münzinger states that boilers cannot be accepted as a final solution of the problem so long as they do not :—

(*a*) Adapt themselves automatically to varying loads as rapidly as internal-combustion motors do and without considerable loss in efficiency;

(*b*) Carry without harm very high specific loads for very long periods or continuously;

(*c*) Work over big ranges of loads with high efficiencies.

Let us see what a Velox can do in this respect :—

(*a*) *Varying Loads.*—The boiler can be started from cold to full pressure in four minutes and to full capacity in five minutes, according to Fig. 1, which shows the starting characteristics of a Velox of 40,000lb. per hour capacity.

Tests with a Velox steam generator on board a small vessel have shown that it is perfectly feasible to manœuvre the boat entirely by the control gear of its turbines without touching anything whatsoever on the boiler side.

(*b*) *High Continuous Load.*—Table I shows a comparison between the characteristics of different types of boilers used on board ships. The very high loads per cubic foot of combustion chamber as well as per square foot of heating surface obtained in Velox steam generators are made possible by supercharging on the gas side and by forced circulation on the gas as well as on the water side.

The possibility of carrying such high heat transfers continuously has been questioned many times, but calculations and tests show that, in spite of the unusually high heat transfer on the gas side of the tube, the heat transfer on the water side is still so much higher that the wall temperature of

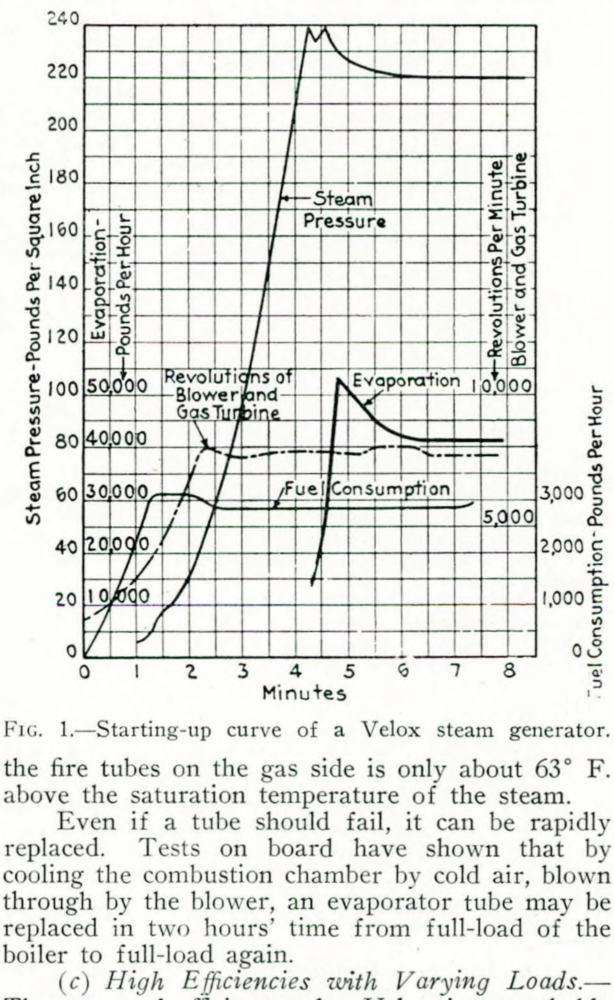


FIG. 1.—Starting-up curve of a Velox steam generator.

the fire tubes on the gas side is only about 63° F. above the saturation temperature of the steam.

Even if a tube should fail, it can be rapidly replaced. Tests on board have shown that by cooling the combustion chamber by cold air, blown through by the blower, an evaporator tube may be replaced in two hours' time from full-load of the boiler to full-load again.

(*c*) *High Efficiencies with Varying Loads.*—The curve of efficiency of a Velox is remarkably flat so that the efficiency may be termed practically constant from one-quarter to full-load.

From the foregoing it is seen that the Velox steam generator can claim to fulfil the three conditions which Dr. Münzinger asks for, for light boilers for fast ships.

In most of the new boilers recently introduced for use on board ship nothing has been done to improve materially the combustion itself or the rate of heat transmission, whereas these two points are the main characteristics of the Velox boiler, which has the following special features:—

(1) Combustion takes place under a pressure far in excess of that generally employed.

(2) Partial transformation of the above pressure into velocity, in order to obtain high flue-gas speed. This speed may reach the velocity of sound.

(3) The use of a turbo-blower, worked by an exhaust-gas turbine, in order to produce the pressure in the combustion chamber referred to in (1).

(4) Disposition of the gas turbine between two heat-absorbing parts of the boiler, in order to reduce the temperature of the flue gases so that cooling of the turbine is unnecessary, but also permitting of sufficient heat being left in the gas to provide the necessary energy for driving the turbo-blower. The heat which remains is passed on to the feed water by an economiser following the gas turbine.

(5) A special device for separating steam from water.

(6) The very small weight of steel and water involved in the heat transmission which, together with the entire absence of fire bricks and the small amount of water necessary, permits rapid starting of the boiler and gives flexible operation under variations of load.

(7) Forced circulation in the evaporating part of the boiler.

(8) Entirely automatic governing.

(9) Small weight.

(10) Small space requirement.

(11) High efficiencies over large variations in load.

Fig. 2 represents a steam generator of the Velox type, and indicates the temperatures and pressures prevailing in its different parts. In modern Velox boilers there is no separate superheater as shown in Fig. 2. The superheater elements are placed inside the evaporator tubes.

The boilers are fitted with turbo-blower sets consisting of a reaction turbine with four rows and an axial blower especially developed for this purpose. The overall efficiency of this set is of the order of 83 (turbine) × 73 (blower) = 60 per cent.

In principle the automatic governing system is similar to that employed by Brown, Boveri on their turbines for the last 20 years.

Table I.—Comparison Between the Characteristics of Different Types of Boilers Used on Board Ship.

	Merchant Vessels.			Naval Vessels.	
Type of boiler	Standard water-tube boiler.	Radiant-type boiler and boiler with forced water circulation.	Velox.	Forced water-tube boiler.	Velox.
Combustion chamber heat release, B.Th.U. per cu. ft. per hour	34,000—67,000	67,000—135,000	500,000—800,000	135,000—280,000	670,000—1,000,000
Evaporation per unit of heating surface, lb. per sq. ft. per hour	5—7	8—11	90—115	15—20	115—135
Heat tranmission, B.Th.U. per sq. ft. per hour.					
Evaporator	5,000—7,000	7,000—10,000	85,000—120,000	11,000—18,500	90,000—130,000
Superheater	3,000—4,000	4,000—7,000	18,000—35,000	5,500—10,000	22,000—37,000
Economiser	900—1,800	900—1,800	5,500—9,000	1,500—(3,000)	5,500—10,000
Boiler-room: Floor area required, sq. ft. per ton of steam per hour ...	43—27	27—16	14—10	14—10	9—6·5
Space required for boiler including uptakes, cu. ft. per ton of steam per hour	1,400—900	900—550	550—300	230—160	160—100
Weight of boiler, including all boiler-room auxiliaries, economiser, pre-heater, piping, fittings, brickwork, insulation, uptakes and water, lb. per lb. steam	7·5—6	4—2·5	1·8—1·1	1·8—1·3	1·2—0·8

The burners had to be especially developed for the conditions prevailing in Velox steam generators, as neither burners for atmospherical pressure nor nozzles as used in Diesel engines could be used. Up to the highest output for which Velox steam generators have been built, i.e., 80 tons of steam per hour, only one burner is applied. These burners give perfect and smokeless combustion from no-load to full-load with an amount of excess air down to 6 per cent. The advantage of having only one burner lies in the fact that the nozzle section is so big that no choking or clogging of the nozzle occurs.

As the ship would seem to be the ideal field of application for the Velox, because here its many unique features can be used to the greatest advantage, it is strange and regrettable that, besides 34

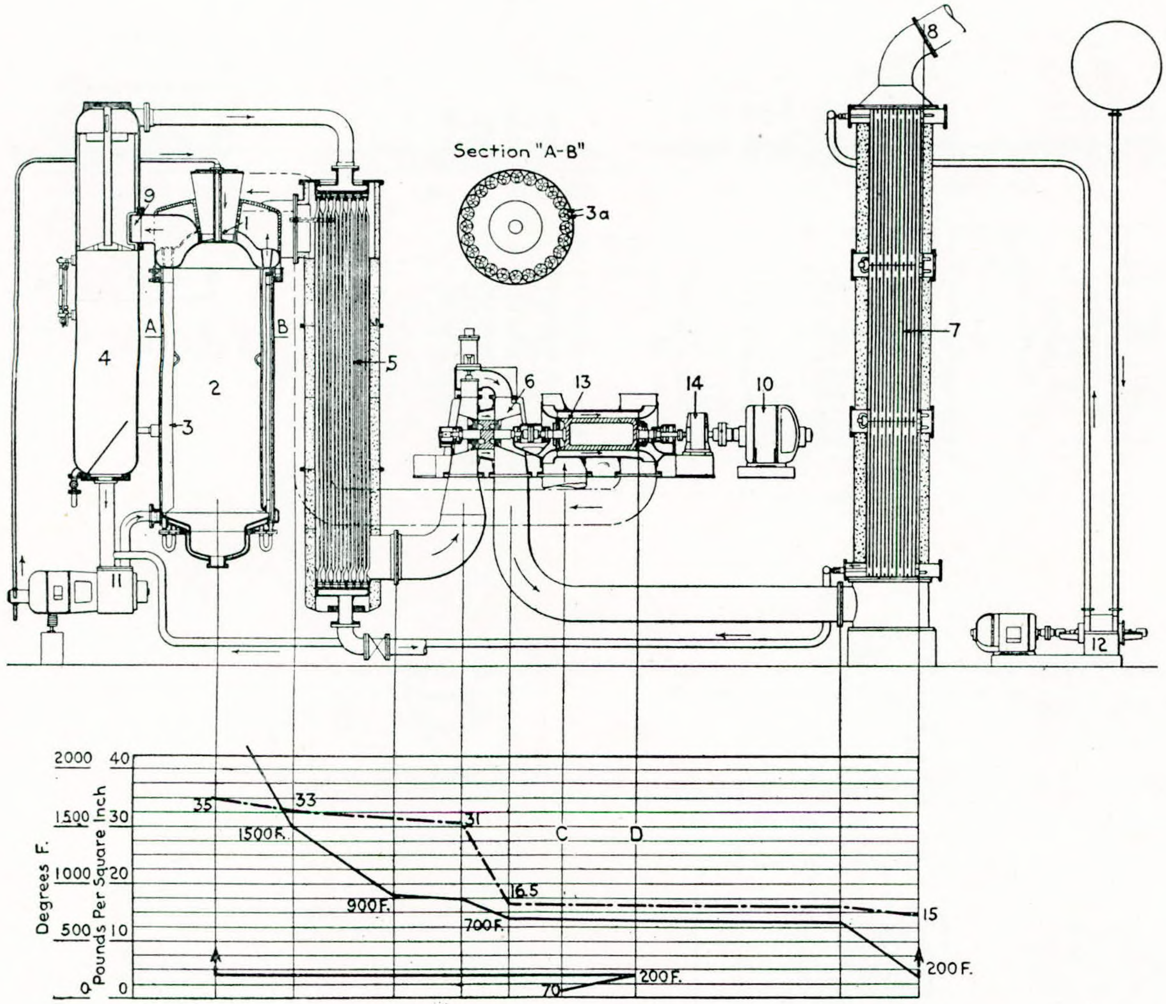


FIG. 2.—Diagram of Velox steam generator, showing pressures and temperatures.

Velox steam generators built or on order as yet for stationary plants and locomotives, only one has been ordered for the mercantile marine and a certain number for a few progressive navies for trial purposes on shore and on board.

The small space requirements of a Velox, its low weight and high efficiency are of paramount importance for the transformation of existing plants in old ships. It is generally possible to replace one existing boiler with a Velox which equals the output of all the other boilers. By means of such a substitution the power range and economy of old ships can be considerably increased, and even more so if at the same time the steam pressure and superheat are increased and the additional heat drop is used in a supplementary high-pressure turbine which may drive through the existing gear.

In new vessels the effect of substituting Velox for the normal Scotch or water-tube boiler may be still greater than in a transformation job, because this substitution may justify the building of an entirely different hull, which may have less displacement and still the same or improved carrying capacity and speed.

A comparison is shown in Fig. 3 for the "Queen Mary" where 12 Velox replace 24 water-tube boilers with a saving in weight and space as shown in Table II and a reduction in stacks from 3 to 2. One may object that the "Queen Mary", if built to-day, would have a more modern boiler

TABLE II.

Main boiler plant.	"Queen Mary".	Velox.
Steam output, tons per hour	800*	840
Boiler-room: Length, ft.	265	108
Floor area, sq. ft. ...	21,100	7,450
Floor area, sq. ft. per ton of steam per hour	26·3	8·87
Space required up to B-deck, cu. ft. ...	915,000	330,000
Space, cu. ft. per ton of steam per hour	1,145	393
Weight, tons†	5,200*	1,050

*Estimated.

†Weight of complete main boiler plant, including all boiler-room auxiliaries, uptakes to B-deck and water.

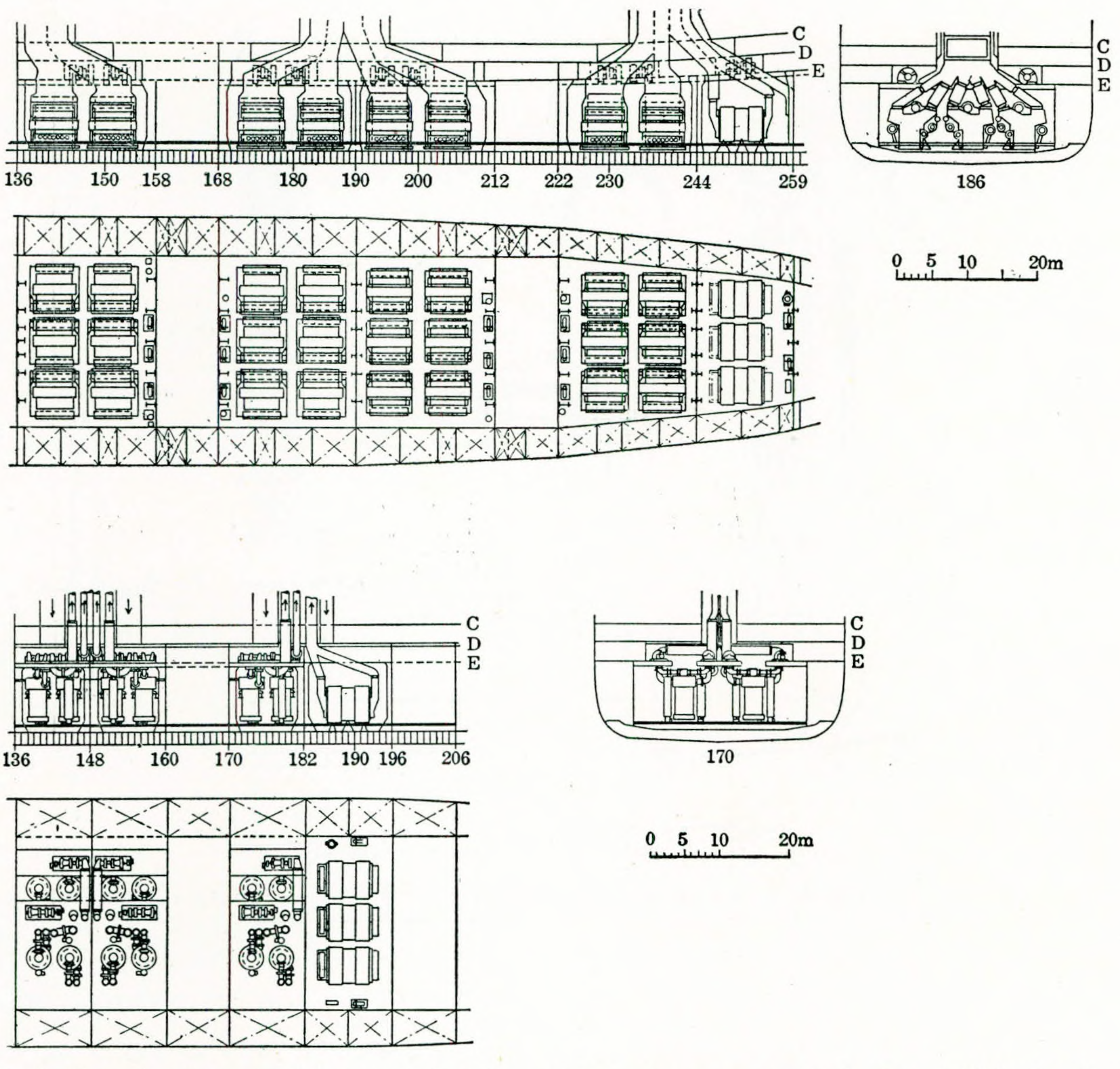


FIG. 3.—Comparison between existing water-tube boilers and Velox steam generators in the Cunard White Star liner "Queen Mary".

plant; but even comparisons made with most modern boilers, such as the Johnson in the "Asturias" or the Benson in the "Potsdam", show a reduction in space requirement of 2 to 1 in favour of the Velox.

In smaller vessels much can be gained by assembling the Velox boiler and turbine into a compact unit, thus dispensing with long pipe-lines, eliminating various fittings, and having the same attendance for turbine and boiler.

The Velox has the advantage over the Diesel engine that every kind of oil can be used, and there is no restriction as to the use of the more expensive gas and Diesel oils. The Velox steam-turbine plant is also more favourable in regard to weight, except for long voyages where the excess fuel oil equalises the difference in weight in about 40 days' steaming.

In warships of ordinary design the full steam output is obtained by forcing the boilers to about three to four times the amount which such a boiler would normally give if otherwise applied. The efficiency of such boilers, therefore, decreases when the load increases with increasing exhaust temperature of the flue gas. The differences in weight and space requirements of the Velox boiler, when compared with such forced boilers, are thus not so considerable as is the case when compared with boilers which have to be operated at highest efficiency. However, the weight and space requirements of the Velox boiler, as developd for warships, remain considerably below those of the lightest type of marine boiler.

Space requirements and weight have already been referred to. The gain in these is, however, not restricted to the boiler proper. Because the flue gases are under pressure it is possible to reduce the stack dimensions considerably without appreciable loss of efficiency.

A fundamental difference between the Velox boiler and naval boilers is that full-load can be maintained continuously for any length of time with the same high efficiency as on part load. The efficiency amounts to between 88 and 90 per cent., compared with about 75 per cent. obtained with a forced water-tube boiler. When steam is available from one boiler only, supplementary boilers can be brought from cold up to full-load in less than five minutes. These two facts represent a further reduction in weight according to the amount of fuel saved under otherwise identical steaming conditions.

The Velox boiler operates entirely automatically. This may be of great importance in case of a gas attack. The exhaust gas is completely invisible even at maximum output and has a low temperature.

Progress of the National Tank.

"The Marine Engineer", May, 1937.

The Report of the National Physical Laboratory for 1936 has recently been issued by the Department of Scientific and Industrial Research* and the section on the William Froude National Tank makes interesting reading, especially as Dr. Baker's report gives the cheery news that during 1936 no fewer than 88 ship designs were submitted to the tank for test and modification. The growth of work carried out at the tank for private firms during the past nine years is shown in a table, which we reproduce.

1928.	1929.	1930.	1931.	1932.	1933.	1934.	1935.	1936.
59 ...	51 ...	30 ...	28 ...	28 ...	45 ...	60 ...	73 ...	88

An encouraging feature of the work carried out last year was that no fewer than 14 of the designs tested were for foreign owners and builders. Of the 88 designs submitted, 18 hulls were redesigned by the laboratory officials before commencing tests in order to save the time and work of the experimenters and to reduce the cost to the client. Of the various hulls submitted 30 of them were improved by more than 3 per cent. and four by more than 10 per cent. These are excellent results of the utmost commercial value, having regard to the useful life of the average vessel and the high cost of its propulsion through the water.

A considerable amount of propeller work of various kinds was also completed by the tank during the year under review, this including the testing of three sets of new patent screws. Work in the new propeller tunnel was commenced last autumn, investigation into scale effect being combined with a series of tests on screws of varying blade area for a high-speed vessel. The screw propeller research carried out for single screws has shown that:—

(1) The aerofoil type of propeller has a definite advantage in efficiency over the circular backed type, which increases as the pitch ratio diminishes.

(2) Propellers with pitch ratio down to 0·8 can be used with circular backed blades, and down to 0·6 with aerofoil blades, without material loss of efficiency.

(3) It is better to reduce the diameter as necessary to maintain a pitch ratio near 0·6, rather than to use a screw of large diameter and very low pitch. This is particularly the case with lightly loaded screws.

(4) The rate of revolution of the screw had very little effect upon the wake and thrust for propulsion.

We are interested to see in the report that work has been carried out on the comparison of model and ship data, this being a side of the tank's activities which should be extended, we feel, in order to bring the benefits of the organisation to a wider circle of practical-minded men who are at the present time not entirely convinced as to the practical utility of such establishments. During the year the trials of six vessels were attended by members of the tank staff, three of these being twin-screw ships. The log data of three ships have been analysed and, in the case of the single-screw vessels, it is interesting to note, reasonably good agreement has been obtained between the practical full-scale results and the predictions from the model tests. In the case of the twin-screw vessels, however, the difference between the actual and the estimated powers was greater, rising in one instance to 10 per cent. at the vessel's top speed. This discrepancy was confirmed in a sister ship and the report frankly states that no explanation of this variation can be given, but that further experimental work is in hand with a view to finding an explanation.

We are rather disappointed to see that the report contains no reference to wind resistance reduction by the adoption of a measure of streamlining of the deck erections such as is now being done by certain builders, particularly abroad. We suggest that work of this character could form a useful addition to the researches undertaken by the laboratory by seeking the co-operation of the aerodynamics department of the establishment, which has an unrivalled wind tunnel equipment. Streamlining, even for slow-speed cargo vessels, is of greater value than is appreciated in certain quarters, especially in a headwind, and it would be very valuable to have the results of experiments on this subject in the form of a paper by one of the members of the N.P.L. staff.

* Published by His Majesty's Stationery Office, Adastral House, Kingsway, London, W.C.2. Price 2s. 6d. net.

BOARD OF TRADE EXAMINATIONS.

List of Candidates who are reported as having passed examinations for certificates of competency as Sea-Going Engineers under the provisions of the Merchant Shipping Acts.

Name.	Grade.	Port of Examination.
For week ended 15th April, 1937:—		
Fenton, Cecil L.	2.C.	Liverpool
Fisher, John	2.C.	,,
Jones, Thomas L.	2.C.	,,
Tracey, James J.	2.C.	,,
Fairlem, Walter H.	2.C.M.	,,
Thompson, George F. ...	2.C.M.	,,
Brain, Henry	2.C.	London
Ralfs, Cyril A. G.	2.C.	,,
Anderson, Norman W. G. ...	2.C.M.	,,
Boater, Jack A.	2.C.M.	,,
Deverell, Eric G.	2.C.M.	,,
Humphrys, Lionel J. ...	2.C.M.	,,
McCarlie, James R.	2.C.M.	,,
Davies, Hugh J.	2.C.	Cardiff
Freeth, Joseph H.	2.C.	,,
Cowan, Norman R.	2.C.	Glasgow
Fletcher, Donald C.	2.C.	,,
Higgins, Edward T.... ...	2.C.	,,
Johnston, Adam	2.C.	,,
Johnston, Peter	2.C.	,,
McKeown, John	2.C.	,,
Sinclair, Thomas B.... ...	2.C.	,,
Brodie, William McI. ...	2.C.M.	,,
Gulstad, John C.	2.C.M.	,,
Sinclair, Ronald M.... ...	2.C.M.	,,
Dawson, George B.	2.C.M.	Newcastle
Howey, Reed S.	2.C.M.	,,
Mitchell, John W.	2.C.M.	,,
Slater, Michael S.	2.C.M.	,,
For week ended 22nd April, 1937:—		
Lovely, William	1.C.	Newcastle
Pinkney, Cyril	1.C.	,,
Tait, Andrew J.	1.C.	,,
Shaw, Thomas	1.C.M.	,,
Jones, Clifford E.	1.C.S.E.	London
McTaggart, Peter McL. ...	1.C.M.E.	,,
Charles, Arthur J.	1.C.M.E.	Cardiff
Rees, Robert L.	1.C.M.E.	,,
Watkins, Herbert L. ...	1.C.M.E.	,,
Entwistle, Leslie	1.C.M.E.	Liverpool
Mills, James	1.C.M.E.	London

Name.	Grade.	Port of Examination.
Griffiths, Walter I.	1.C.M.E.	Liverpool
Smith, William J.	1.C.M.E.	London
McIntyre, Alexander B. ...	1.C.M.E.	,,
Webb, Frederick W. J. ...	1.C.M.E.	,,
Morrison, John R.	1.C.S.E.	Glasgow
Harley, Douglas	1.C.M.E.	,,
Bates, Hubert O.	1.C.	Cardiff
Bridges, Russell E.	1.C.	,,
Williams, David S.	1.C.	,,
Rundle, John E.	1.C.M.	,,
Clare, George	1.C.	Liverpool
Dickinson, Ronald H. ...	1.C.	,,
Hinson, Norman J.	1.C.	,,
Radcliffe, Francis A. ...	1.C.	London
Rushton, John M.	1.C.	,,
Barclay, William T. ...	1.C.	Glasgow
Faulds, Joseph S.	1.C.	,,
Mackie, John	1.C.	,,
Murray, James C.	1.C.	,,
Neilson, Matthew	1.C.	,,
Robertson, Duncan J. E. ...	1.C.	,,
Shaw, Thomas	1.C.	,,
Lloyd, William E.	1.C.	Newcastle
Hunter, James	1.C.M.E.	Glasgow
Atkinson, Thomas	1.C.M.E.	Newcastle
Greenhorn, Frank	1.C.M.E.	,,
Coates, Kenneth S.	1.C.M.E.	,,
Griffin, Frederick	1.C.M.E.	,,
Robson, Richard	1.C.M.E.	,,
Kirby, William E.	1.C.M.E.	,,
Spoors, Charles H.	1.C.M.E.	,,
Grandison, James	1.C.S.E.	,,
For week ended 29th April, 1937:—		
Black, Harry	2.C.	London
Bridgwater, Arthur D. ...	2.C.	Liverpool
Goldsmith, Leslie C. ...	2.C.	,,
Sutherland, Donald W. ...	2.C.M.	,,
Cree, James H.	2.C.	Glasgow
Johnston, James	2.C.	,,
MacKay, John S. G. ...	2.C.	,,
Purdie, Donald G.	2.C.	,,
Sinclair, Robert D.	2.C.	,,
Bell, Harry	2.C.	Newcastle
Carney, William H.... ...	2.C.	,,
Hogg, Thomas E.	2.C.	,,
Lee, William T.	2.C.	,,
Howden, John	2.C.M.	,,
Lilley, Bertie	2.C.M.	,,
Patterson, James H. ...	2.C.M.	,,

The Late Rt. Hon. Lord Kylsant of Carmarthen, G.C.M.G. (Past President).

Portrait by H. J. Whitlock & Sons, Ltd.

OBITUARY.

The Rt. Hon. Lord Kylsant of Carmarthen, G.C.M.G. (Past President).

We record with regret the death of Lord Kylsant, at the age of 74, which occurred on Saturday, June 5th, 1937, at Coomb, Carmarthenshire.

The Right Hon. Owen Cosby, first Baron Kylsant, of Carmarthen and of Amroth, was born on March 25th, 1863, the third son of the late Canon Sir James Erasmus Philipps, twelfth baronet, and the Hon. Lady Philipps, sister of the fifth Baron Wynford.

He was educated at Newton College, South Devon, and commenced his business career in a shipowners' office in Newcastle-on-Tyne, later going to Glasgow, where, in 1888, he formed the King Line Ltd., of which he became the chairman. He subsequently moved the head office to London, and in 1902 he was elected a member of the court of directors, and was shortly afterwards appointed chairman of the Royal Mail Steam Packet Co., the oldest British shipping company incorporated by Royal Charter, having been granted a Royal Charter by Queen Victoria in 1839, two years after she ascended the Throne.

During the years he was chairman of the Royal Mail Steam Packet Co. (1903 to 1931) it gradually increased its shipping interests, and Lord Kylsant became chairman of the Royal Mail Steam Packet Co., White Star Line, Union-Castle Line, Pacific Steam Navigation Co., Lamport and Holt, Elder Dempster and Co., and associated shipping companies. The tonnage operated by the group totalled 2,800,000 tons. He was also chairman of Harland and Wolff, and a director of the Southern Railway and of the Midland Bank.

Lord Kylsant was a member of the Royal Commission on Shipping Rings, 1906-1909; chairman of the Departmental Committee on Distressed Indian and Colonial Subjects, 1909, and vice-chairman of the Port of London Authority, 1909-1913. He was an ex-president of the Chamber of Shipping, The Institute of Marine Engineers, the Federation of Chambers of Commerce of the British Empire, and the London Chamber of Commerce.

During the war Lord Kylsant served on a number of Government and other committees, and was a member of the Admiralty Board of Arbitration 1914-1917; Edible and Oil-Bearing Nuts Committee, 1915-1916; Empire Settlement Committee, 1917, and the Company Law, Amendment Committee, 1918.

Lord Kylsant was owner of 6,000 acres in Carmarthen and Pembrokeshire, and of the historic castles of Llanstephan and Amroth. Politics and the Church in Wales were two of his great interests outside business. Before his elevation to the peerage, Lord Kylsant sat in Parliament as a Liberal for Pembroke and Haverfordwest from 1906 to 1910, and as Unionist member for Chester from 1916 to 1918 and 1918 to 1920. For a brief period in 1933 Lord Kylsant returned to public life, and when he attended the House of Lords was cordially welcomed.

In March, 1902, he married Mai Alice Magdalen, C.B.E., a Lady of Grace of the Order of St. John of Jerusalem in England, co-heiress of the late Mr. Thomas Morris, J.P., D.L., of Coomb, Carmarthenshire. She survives him with three daughters — the Countess of Coventry, Lady Suffield, and the Hon. Mrs. Charles Pilkington. There is no heir to the peerage, which, therefore, becomes extinct.

The funeral took place at Llangunnock Church, on Thursday, June 10th.

Sir Vernon Thomson, ex-president of the Chamber of Shipping, has said of Lord Kylsant: "In his prime he rendered many and great services to British shipping, and he had been one of the outstanding personalities of his generation in world shipping".