New Designs of Large Two-Stroke Marine Diesel Engines

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The paper describes at the outset the present-day tendencies in the use of the ship Diesel engine and presents the new Sulzer design, referring to its possible applications. Heavy fuel oil problems are discussed, together with results obtained from various tests. The prototype engine is then described with special reference to the use of heavy fuel and the application of welding, together with the overhauling and maintenance aspect. The shop trial results are given and these are followed by a few notes on future development with special reference to supercharging.

INTRODUCTION

The inherent fuel economy of the internal combustion engine, further enhanced by the ever increasing use of cheap boiler fuels, makes the Diesel engine a very suitable choice for marine propulsion. This feature, which is its chief advantage over the steam turbine is, however, in certain cases outweighed by the limited output attainable with a single engine. During the past few years specifications for new cargo vessels and tankers have called for the design of larger and faster ships, requiring higher outputs of the propelling machinery. As the highest propulsion efficiency in such cases is obtained with a single propeller turning at a little over 100 r.p.m., a high concentration of power on the propeller shaft is required and it is difficult, with existing engine types, to meet the competition of the steam turbine.

The well proven Sulzer line of single-acting two-stroke SD72 engines of 720 mm. bore, 1,250 mm. stroke and a rated speed of 125 r.p.m.[1] is ideal for many classes of vessels requiring up to 8,000 b.h.p. where the full load can be used by a reasonably efficient propeller, but for higher outputs twin screw installations with their smaller propulsion efficiency would be necessary. There is thus a definite need for a new engine type, allowing higher outputs to be transmitted to the propeller shaft.

Increasing the output of an individual engine may be accomplished in various ways. The mean effective pressure may be increased without materially higher thermal stresses by supercharging, which will be mentioned later in the paper. Double-acting engines may also be adopted. But most shipowners are reluctant to use this design because of the greatly increased height, its complicated nature and difficulties associated with overhauling. Statistics show that there is a decreasing percentage of double-acting engines being built, which indicates that his company were correct in favouring the much simpler single-acting type from the outset.

Complicated schemes such as multiple Diesel-electric drives or even free-piston gas generator installations may be disregarded for the applications in question, in the interests of low cost and simplicity. This leaves as a practical choice for high-output machinery the largest feasible long-stroke engine for direct propeller drive or a number of smaller and faster engines, connected by couplings and gearing to a common shaft.

To meet both these special requirements, his company have recently developed two new engines designated RS, one of which will be described in this paper. They will supplement and not replace the existing range.

In the larger of the new engines, the RS76 intended for direct drive, the bore has been increased to the limit set by the satisfactory handling of heat stresses. That is 760 mm. (approximately 30 inches), a size which has given good results in service, for instance in the Dutch motor liner *Oranje,* which, after thirteen years is still the fastest and most powerful passenger motor vessel afloat[2]. The stroke has been lengthened to 1,550 mm. (61 inches) in order to arrive at a reasonable piston velocity in spite of the very low number of revolutions required. This design delivers 900 to 1,000 b.h.p. per cylinder at shaft speeds of 110 to 115 r.p.m., and the engine can be built in units of four to twelve cylinders, giving corresponding maximum outputs of between 4,000 and 12,000 b.h.p. (Fig. 1). Even higher outputs of up to 15,000 b.h.p. will be made possible by the introduction of moderate supercharging.

Owing to its length of stroke the RS76 engine is comparatively high, which is not of great importance in cargo vessels or tankers. However, for many uses, as for instance in passenger ships, its height may be troublesome, and in this case the geared drive with its many other advantages—as well

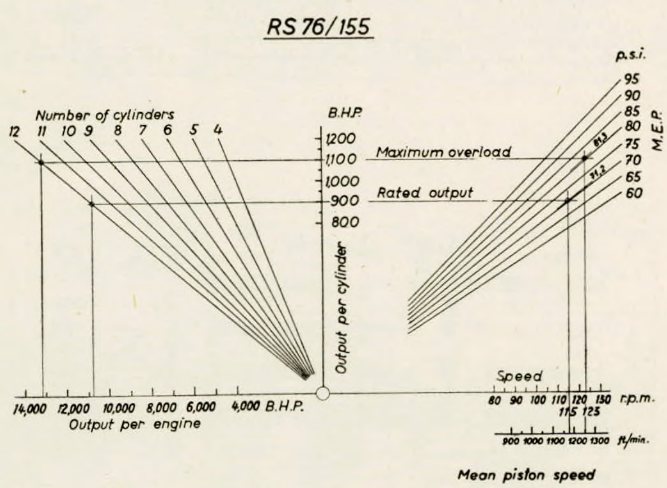


FIG. 1—*Graph showing relationship between output, speed and m.e.p. for varying numbers of cylinders on the Sulzer RS76/155 marine Diesel engine*

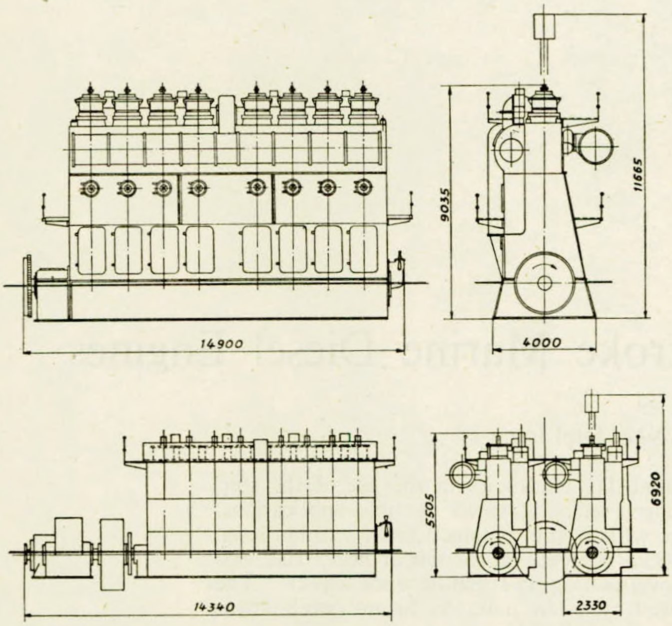


FIG. 2—*Comparison between overall heights of RS engines used for direct and indirect drive*

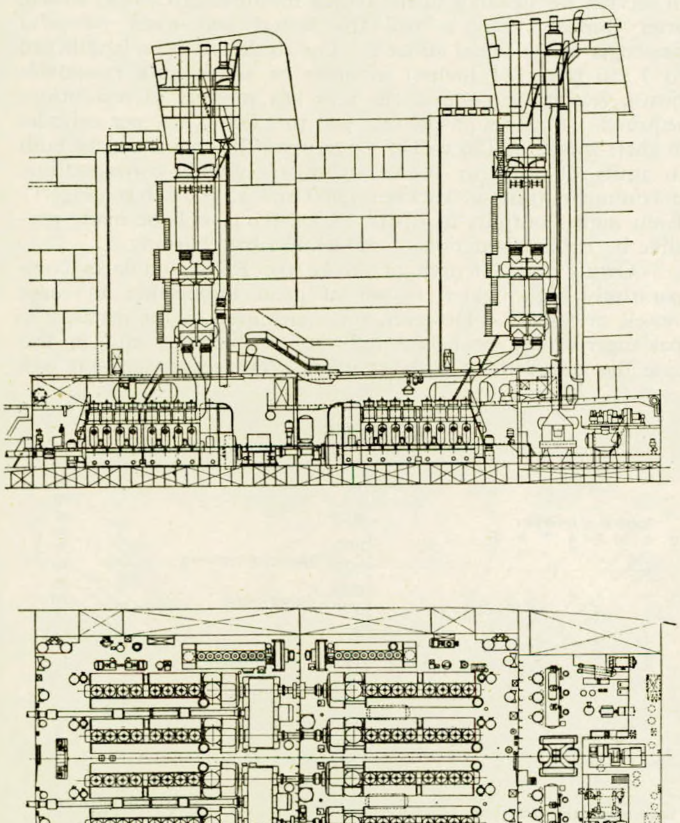


FIG. 3—*Views showing the low engine room height required for a twin screw installation comprising eight engines of 580 mm. bore in the Dutch motor liner* Willem Ruys

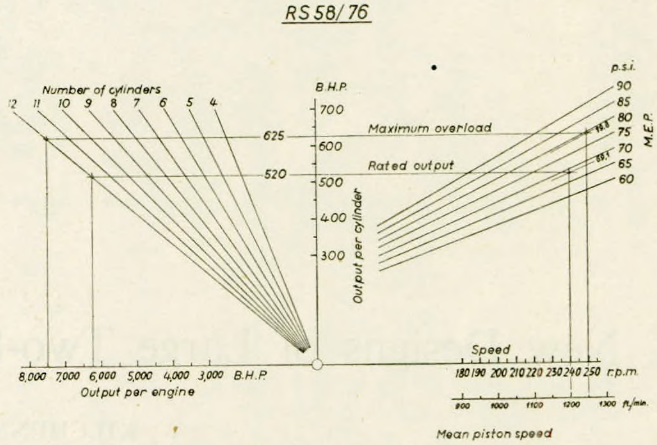


FIG. 4—*Graph showing relationship between output, speed and m.e.p. for varying numbers of cylinders of the Sulzer RS58/76 marine Diesel engine*

as disadvantages—enters into consideration. The difference in height for direct and indirect drive is clearly shown in Fig. 2. An excellent example of the low engine room possible with an indirect drive is given in Fig. 3, showing the installation of the Dutch passenger vessel *Willem Ruys*. Here it was possible to carry two additional decks over the engine room, thanks to the adoption of the geared drive. In some instances, special considerations may lead to the use of the indirect drive for cargo vessels also. This is the case with the new motor vessels *Surrey* and *Cornwall* of the New Zealand Shipping Co., Ltd. An article dealing with the various aspects of direct and indirect drive has appeared recently in the technical press[(3)].

The smaller of the new engines, the RS58, has been designed chiefly for geared installations. Its bore is 580 mm. (approximately 23 inches), the same as that of the engines installed in the ships mentioned above. In order to increase the engine speed at the usual piston velocity and thus obtain a more satisfactory ratio for the reduction gear, the stroke has been reduced as far as possible, i.e. from 840 to 760 mm. (30 inches), still, however, allowing the utilization of a semi-built-up crankshaft. The rated cylinder output is 520 b.h.p. at 240 r.p.m. The engine may be built in units of four to twelve cylinders, and one to four engines may be geared to one shaft. These variations give a choice of from 4 to 48 cylinders and corresponding outputs of 2,000 to 25,000 b.h.p. per shaft (Fig. 4).

In special cases, as for instance in small, fast vessels such as cross-Channel boats, this type may be used for direct propeller drive. Its comparatively high speed also makes it very suitable for generator drive in power stations.

HEAVY FUEL CONSIDERATIONS

Having decided to build the RS type engines, the company tried to meet all the requirements and wishes expressed by engine users in the new design. These were, above all, the operation on heavy boiler oil—which should be rendered as satisfactory as when using fuels of a lighter quality—and easy overhauling.

As may already be known, the problem of heavy fuel combustion is not new to the company. A considerable number of stationary engines have been supplied, developing approximately 125,000 b.h.p. in all, that are equipped for the combustion of heavy oil with a viscosity of up to 4,000 seconds Redwood I at 100 deg. F. Some of these, of the two-stroke loop-scavenged type, installed as long ago as 1926, have run for over 120,000 hours on Bunker C fuel of Central American origin. However, in most of these installations, sufficient attention had not been paid to the care and preparation of the fuel. The conviction has gradually grown that the combustion of heavy fuel is not so much a problem of the Diesel engine itself—that is, as far as its troublefree functioning is concerned

TABLE I.—FUEL ANALYSES

Analysis	Method of testing*	Type	Fuel oil, 1,400 secs.		Fuel oil, 3,000 secs.		Fuel oil, 2,600 secs.		Fuel oil, 6,000 secs.		Fuel oil, 5,000 secs.		Fuel oil, 6,200 secs.	
		Test group No.	I		II		III		IV		V		VI	
		Unit	Centrifuging Before	After	Centrifuging Before	After	Centrifuging Before	After	Centrifuging Before	After	Centrifuging Before	After	Centrifuging Before	After
Specific gravity at 68 deg. F.	SNV 81109	—	0·928	0·935	0·967	0·965	0·966	0·965	0·968	0·968	0·973	0·973	0·990	0·990
Flash point (open cup)	SNV 81110 (Marcusson)	Deg. F.	400	398	359	370	302	292	376	379	273	262	284	291
Flash point (closed cup)	SNV 81120 (Pensky-Martens)	Deg. F.	356	351	315	342	246	250	—	—	—	—	—	—
Pour point	SNV 81107	Deg. F.	0	66	37	45	25	25	50	48	21	25	14	12
Gross calorific value	Bomb calorimeter (Peters, Berlin)	B.Th.U. per lb.	18,837	18,949	18,065	18,589	18,364	18,616	18,518	18,380	18,502	18,574	17,996	18,248
Net calorific value	$Hi = Hs - 1,080\ W_{H_2O}$	B.Th.U. per lb.	17,717	17,842	16,929	17,478	17,208	17,451	17,280	17,154	17,325	17,361	16,790	16,997
Viscosity at 68 deg. F.	Hoeppler's viscosimeter	C.St.	—	—	6,446	6,355	3,439	3,546	—	—	6,380	6,680	—	—
100 deg. F.		C.St.	300	343	728	781	654	700	1,428	1,518	1,190	1,280	1,556	1,522
100 deg. F.	(Converted according to Ubbelohde's tables)	Sec. Redwood I	1,240	1,400	2,950	3,150	2,648	2,834	5,800	6,000	4,818	5,182	6,352	6,212
122 deg. F.		C.St.	148	156	300	306	273	285	547	555	516	536	628	597
176 deg. F.		C.St.	—	—	—	—	—	—	95·1	98·5	92·8	96·5	110	112
212 deg. F.		C.St.	20·5	19·5	29·6	27·8	24	25	—	—	—	—	—	—
Distillation test: initial boilng point		Deg. F.	595	579	515	545	322	304	401	536	329	356	284	302
5 per cent vaporized at		Deg. F.	644	637	621	631	477	518	542	624	491	500	491	486
15 per cent		Deg. F.	667	666	657	662	612	624	615	694	610	603	576	581
25 per cent		Deg. F.	676	678	667	676	657	648	631	714	649	658	612	619
35 per cent		Deg. F.	685	685	673	684	671	653	640	727	671	685	644	651
45 per cent		Deg. F.	693	694	676	689	682	658	644	732	685	703	673	673
50 per cent	SNV 81113	Deg. F.	694	696	678	691	687	660	648	734	691	711	680	—
55 per cent		Deg. F.	700	700	679	694	693	664	649	736	694	718	—	—
65 per cent		Deg. F.	705	705	681	698	707	671	652	738	700	727	—	—
75 per cent		Deg. F.	711	—	682	702	721	682	656	741	703	734	—	—
85 per cent		Deg. F.	—	—	—	—	729	693	—	—	—	—	—	—
Final boiling point		Deg. F.	712	707	682	703	729	693	662	745	703	738	680	662
Recovery		Per cent.	81	70	79·5	81	85	85	84	82	76	80	55	47
Residue		Per cent.	11·4	—	20	18·5	14	13	15	13	23·6	19·5	41·5	50
Loss		Per cent.	7·6	—	0·5	0·5	1	2	1	5	0·4	0·5	3·5	3
Carbon residue	SNV 81104 (Conradson)	Per cent.	8·18	7·74	7·52	3·91	8·32	8·42	10·43	9·95	11·57	12·22	13·7	13·8
Sulphur content	Quartz tube method	Per cent.	2·14	2·35	3·25	3·13	2·50	2·73	3·95	3·94	2·64	2·67	3·81	3·64
Water content	SNV 81111 (xylol)	Per cent.	Trace	Trace	0·17	Trace	0·5	Trace	Trace	Trace	Trace	Trace	0·5	Trace
Ash	SNV 81101 (600 deg. C.)	Per cent.	0·06	0·05	0·04	0·03	0·09	0·10	0·03	0·01	0·08	0·06	0·08	0·05
Hydrogen content		Per cent.	—	11·5	—	11·5	—	—	—	—	12·2	12·6	12·4	12·8
Aniline point	SNV 81100	Deg. F.	176-185	198	189	169	158-167	140-149	167-176	176-185	158-176	158-176	172-181	174-183
Sediments	SNV 81112/10	Per cent.	0·32	0·20	0·04	0·035	0·013	—	0·02	0·02	0·01	0·03	0·04	0·03
Diesel index	$D.I. = \frac{1}{100}$ (deg. A.P.I. × aniline point)	—	36·7	—	27·4	24·8	23·6	21·3	24	25·3	22·2	22·2	20·3	20·5

* SNV=Swiss Standard Association. The methods used correspond in general to those laid down by I.P. and A.S.T.M.

—but of the fuel and of its preheating and cleaning. It must be clearly understood that the properties of the injected fuels have a direct influence on engine wear. In most cases, even with careful handling of the fuel, it is to be expected that greater wear of the cylinder, piston and piston rings will take place. It is also necessary to employ additional personnel to attend to the preparation of the fuel and to provide the installation with the necessary apparatus, such as fuel heaters, centrifuges, circulating pumps and heated piping which, of course, will complicate the plant and increase its cost.

The heat for warming up the fuel is provided on stationary installations by direct utilization of the exhaust gases; however, this method is not acceptable in a ship because of the fire risk.

In order to find the most advantageous methods of preparing various fuels with viscosities of 1,500 to 6,000 seconds Redwood I at 100 deg. F., a 720 mm. bore single-cylinder engine was used to carry out extensive tests of some 2,000 hours' duration, spread over a period of one year (Fig. 5, Plate 1). The tests included investigation of the best centrifuging method, the number of centrifuges necessary to produce the required cleaning effect, together with the correct operating temperature for these, and, further, the injection temperature necessary for troublefree injection and combustion. Experience was also obtained of the wear to be expected with engines burning heavy fuel of various grades. Furthermore, several new features of the new engine design, such as the scavenge system including the exhaust valve and the injection system, were given a thorough try-out.

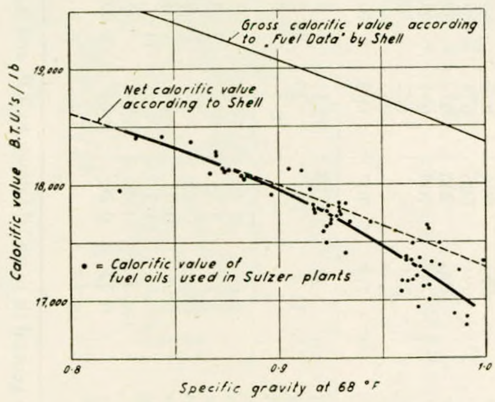


FIG. 6—*Net calorific value of fuel as supplied as a function of the specific gravity*

The range of fuels used is shown in Table I. It represents the heavy fuels of 1,400 to 6,200 seconds Redwood I at 100 deg. F. that were available on the European market in 1952. From these analyses, together with many other figures obtained with heavy fuels burnt in Sulzer installations, a curve was drawn showing the relationship between the specific gravity and the calorific value of these fuels. As indicated in Fig. 6, the curve seems to deviate somewhat from that mostly used in practical service, as it shows lower calorific values at high specific gravities.

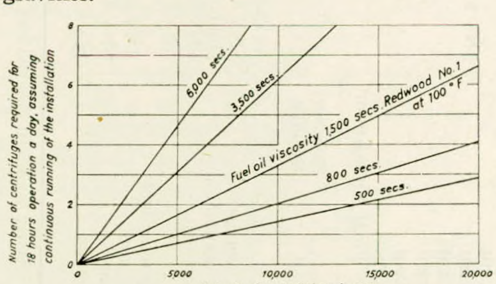


FIG. 7—*Number of centrifuges recommended for a given output*

It is well known that the centrifuge throughput has to be reduced below the nominal rating, especially when treating the more viscous grades of boiler oil. Fig. 7 shows—in this case for the De Laval VIB 1929-C centrifuge—that, for treating fuels of 6,000 seconds, a very large centrifuging installation is required for even moderate engine outputs. This indicates again that the economic advantage of using the heaviest possible fuel has to be considered carefully. Usually the price reduction is not very great for fuels above 1,500 seconds viscosity, an addition of only some 10 per cent of gas oil to fuel of 6,000 seconds reducing its viscosity to 1,500 seconds, as shown in Fig. 8.

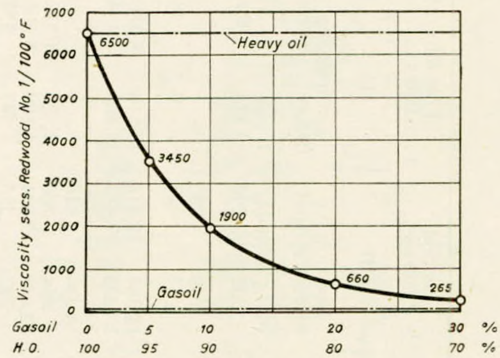


FIG. 8—*Mixing of heavy oil with gas oil*

The results of the centrifuging tests will be published in detail shortly, and only a few indications will be given here. Conclusions drawn from the tests indicate that centrifuging is most efficiently carried out at the highest possible temperature, thus producing the lowest fuel viscosity and the highest difference in specific gravity between the fuel and water. This temperature is at present limited by vaporization of the water and the lighter fuel constituents, resulting in foaming and fire risks. Thus, it would appear to be an advantage if a sealed centrifuge could be used, allowing cleaning to be carried out under pressure at still higher temperatures.

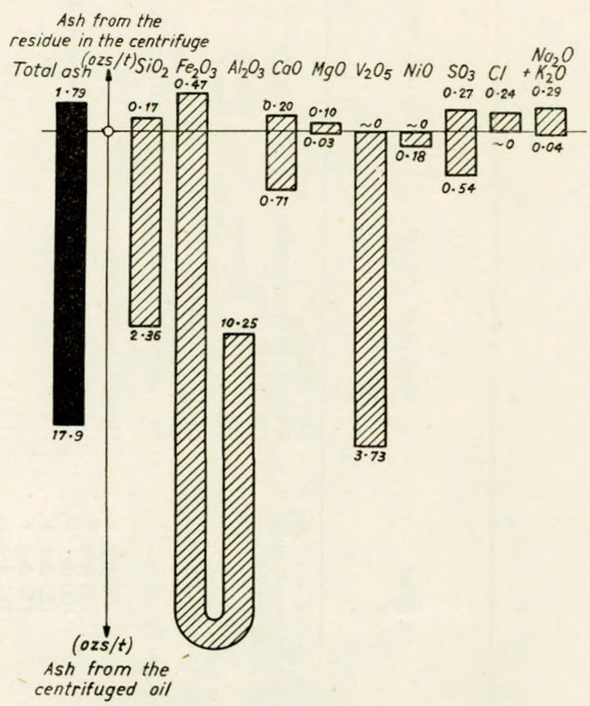


FIG. 9—*Oil ash analysis*

TABLE II.—RESULTS OF THE CENTRIFUGING EXPERIMENTS

Test group No.	Viscosity at 100 deg. F.	Specific gravity at 20 deg. C.	Quantity of centrifugal fuel oil	Centrifuging conditions			First centrifuge*						Second centrifuge*			Ratio			Total sediment		
							Throat			Wall and cover			Wall and cover			o/l	p/m	q/n	$h+l+o$	$i+m+p$	$k+n+q$
				Diameter of gravity disc	Through-put	Temp-erature	Sedi-ment	Insol-uble in ben-zene	Ash	Sedi-ment	Insol-uble in ben-zene	Ash	Sedi-ment	Insol-uble in ben-zene	Ash	Sedi-ment	Insol-uble in ben-zene	Ash	Sedi-ment	Insol-uble in ben-zene	Ash
—	secs. Red-wood I	—	tons (m)	mm.	tons/h	deg. F.	oz./tons	oz./tons	oz./tons	oz./tons	oz./tons	oz./tons	oz./tons	oz./tons	oz./tons	per cent.	per cent.	per cent.	oz./tons	oz./tons	oz./tons
a	*b*	*c*	*d*	*e*	*f*	*g*	*h*	*i*	*k*	*l*	*m*	*n*	*o*	*p*	*q*	*r*	*s*	*t*	*u*	*v*	*w*
I	1,400	0·928/0·935	68·8	113/116	0·25-1·20	109-203	0·29	0·21	—	3·63	1·50	—	0·59	0·20	—	16·2	13·6	—	4·5	1·91	approx. 1·50
II	3,000	0·967/0·965	119·8	103·7 103·7/110 110	0·640 0·348 0·674	187 187 187	0·65 0·39 0·45	0·49 0·30 0·30	0·41 0·27 0·26	4·18 6·38 7·38	1·19 2·32 2·15	0·54 1·19 1·17	3·18 1·25 1·46	1·12 0·46 0·57	0·73 0·24 0·28	76·0 19·5 19·8	93·8 19·8 26·3	135·3 17·0 24·6	8·01 8·02 9·28	2·80 3·08 3·03	1·68 1·70 1·72
III	2,600	0·966/0·965	21·9	110	0·674	189	0·22	—	0·15	2·51	—	0·63	0·14	—	0·03	5·57	—	5·33	2·87	—	0·82
IV	6,000	0·968/0·968	35·6	107/110	0·358	185	0·28	—	0·16	2·34	—	0·63	0·73	—	0·42	31·2	—	66·8	3·35	—	1·20
V	5,000	0·977/0·973	8·6	110	0·350	187	—	—	—	5·77	—	1·65	0·49	—	0·12	8·4	—	7·7	6·25	—	1·77
VI	6,200	0·990/0·990	17·8	102·4 103·7	0·350	185	0·73	—	0·46	7·78	—	1·29	2·77	—	1·62	35·7	—	125	11·30	—	3·37
Total weight			272·5																		

* In the majority of tests the first centrifuge was used as purifier and the second as clarifier, both being De Laval type VIB 1929-C

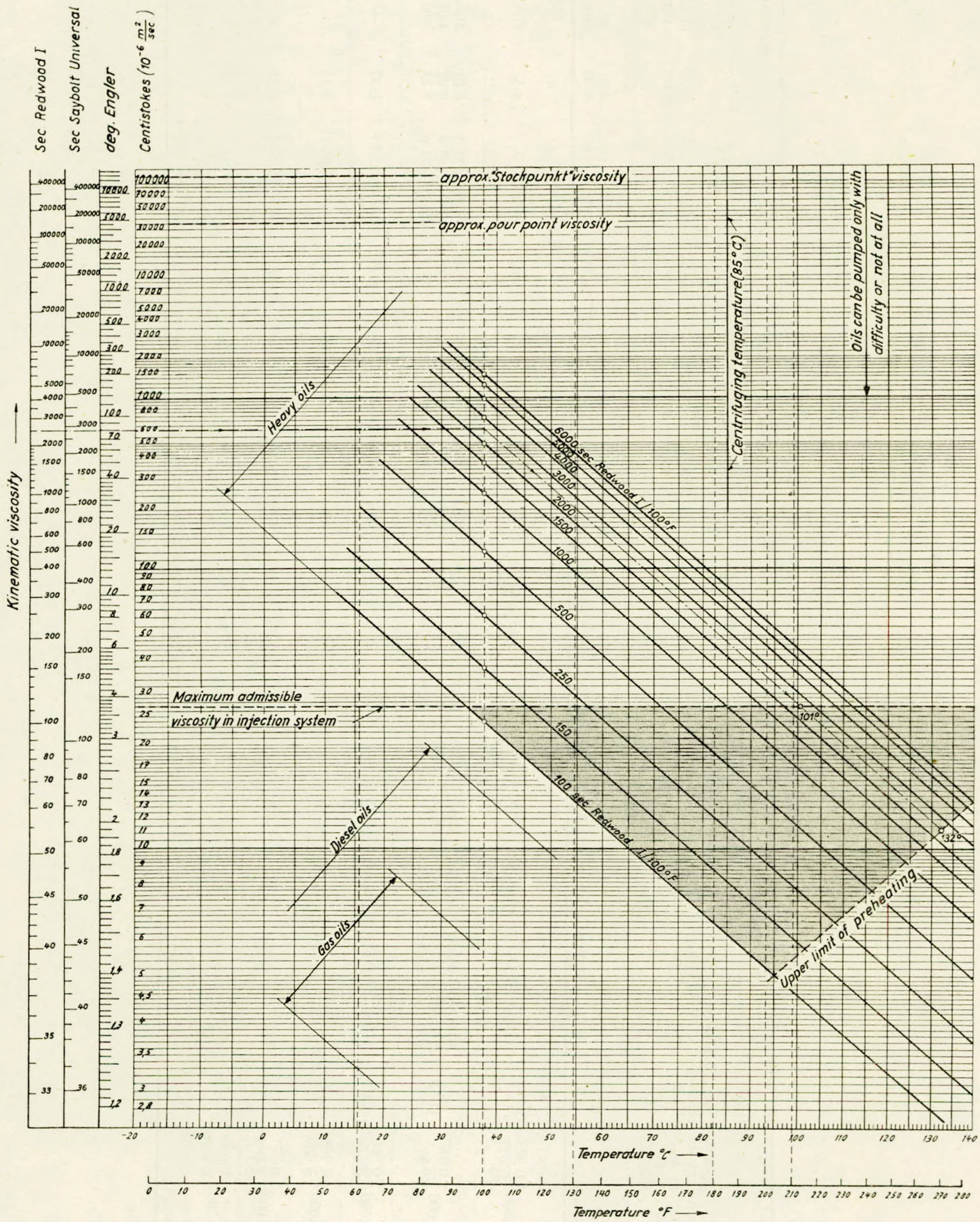


FIG. 10—*Viscosity temperature diagram* for determining the preheating temperature of heavy oils*
Example: a heavy oil with a viscosity of 2,500 seconds Redwood I at 100 deg. F. (see chain-dotted line) must be preheated to at least 101 deg. C. if it is to be above the highest admissible viscosity of 27 cSt. However, it should not be heated above about 132 deg. C.

* Raster of the diagram after Prof. L. Ubbelohde.

Table II shows the quantities and proportions of impurities extracted from purifier and clarifier. Fig. 9 reveals that, of the total ash content of a fuel, only a minor part is present in a state in which it can be removed by normal centrifuging, the remainder, including, for instance, vanadium ash and large amounts of iron, probably being present in the oil in the form of soluble organic compounds. The cleaning treatment of the oil may also affect its physical properties; as an example of this, the pour point of a certain fuel consignment was altered by heating, centrifuging and storage for several weeks from 0 deg. F. to 66 deg. F. (see Table I).

As was to be expected, in view of the long Sulzer experience with heavy fuel, there were no surprising conclusions to be drawn from the results of the engine tests. The fuel heating and feeding system was modified somewhat, in order to improve the manœuvring characteristics and to allow preheating of the fuel to the desired temperatures shown in Fig. 10.

For starting and manœuvring, it is not necessary to change over to Diesel oil; when suitable injection parts are fitted, all manœuvres may be carried out whilst using heavy oil. Attention, however, must be paid to ensure that the fuel heating is not interrupted, and further that the circulation of the fuel during stops is maintained, at least up to the fuel pumps, in order to keep the latter and the piping warm. On engines fitted with long fuel pressure pipes between the pump and the injection valves, for example, the normal SD crosshead type engine, these must be provided with a means of heating. This heating is most important during the stop periods between individual manœuvres and at very low loads. The heated fuel should be kept under pressure in order to prevent liberation of light volatile constituents and gases. Liberation of gas in the fuel piping may lead to injection stoppage and therefore interrupt the operation of the engine. The arrangement shown in Fig. 11, now used with slight modifications on most Sulzer installations, includes all these desirable features.

Whilst there is a definite need for manœuvring with heavy fuel, the economic aspect of doing so in all cases requires special consideration. Some cases have been encountered where it would have been cheaper to stand-by on Diesel fuel during long waiting periods than to keep the whole system heated with the aid of the donkey boiler or the auxiliary burner of the exhaust boiler. No free steam from the exhaust energy of the main engines is available in such a case and the additional steam required has to be paid for by expenditure of boiler oil. It may be possible to install a small exhaust boiler on the auxiliary

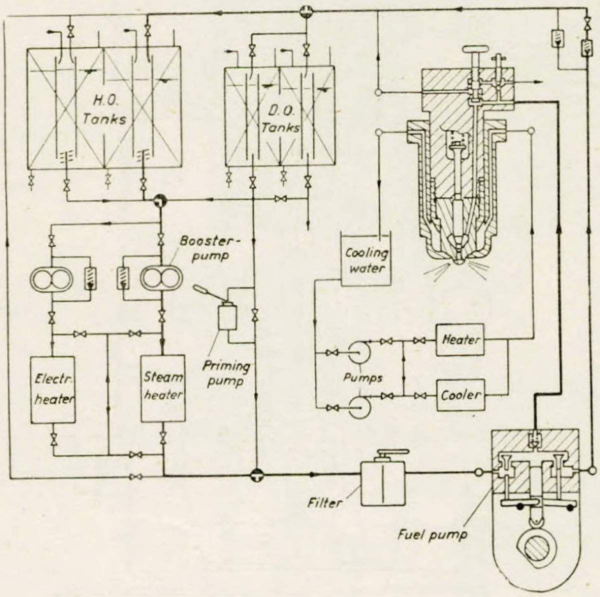


FIG 11—*Heavy fuel installation diagram*

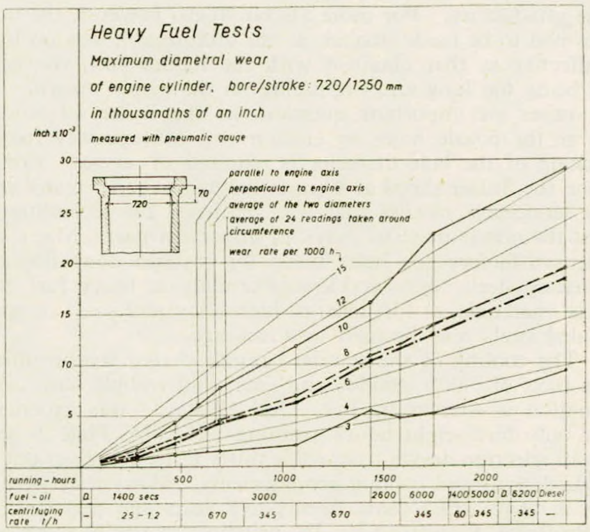


FIG. 12—*Maximum cylinder wear on 720 mm. bore engine*

engines or to preheat the system by electricity, but the cheapest solution to cope with such special cases would be to switch over to light fuel on the main engines.

Fig. 12 shows the cylinder liner wear, which is approximately doubled compared with the average service results on Diesel fuel. During the tests there was difficulty, of course, in determining very small changes of cylinder diameter between the comparatively short runs on the various fuels. The ordinary micrometer readings are not accurate enough, owing to temperature differences, to allow less than a thousandth of an inch to be measured on a diameter of 28 inches, and give reliable figures only over periods of at least 1,000 running hours. The micrometers have been supplemented by a special wear measuring device, based on the pneumatic principle developed some ten years ago and allowing contour maps of wear to be drawn on the basis of 600 measuring points per cylinder in a comparatively short time. Another method based on the change of shape of a small diamond impression was also tried out. It showed that a large part of the wear was due to corrosion of the liner metal beneath a brownish layer of varnish found on the liner surface.

The kink in the transverse wear curve in Fig. 12 between 1,100 and 1,700 hours can be explained by a difference of varnish thickness present at the time when the dimensions were taken. The piston-ring wear was measured very carefully, over 630 individual readings being recorded. Here the wear also amounted to approximately twice the figure for average Diesel fuel. An attempt to reduce wear and liner varnish by the use of a strongly detergent lubricating oil for the cylinders gave no conclusive results, possibly due to the shortness of the test run of only 142 hours.

The above wear figures are, of course, only representative in the case of the fuels used. The results of wear in service show considerable scatter, as they depend very much on the quality of the fuel burnt, viscosity alone not being the decisive factor. On the one hand there are cases showing figures higher than those obtained in the tests and on the other hand wear rates of only two to three thousandths of an inch per 1,000 hours have been published, these having been obtained over more than 10,000 running hours with fuels of up to 8,000 seconds viscosity on Sulzer two-stroke loop-scavenged trunk-piston engines[4].

In the injection system the clearances for fuel-pump plunger and fuel needle had to be increased to approximately 10 microns (0·0004 inch) to provide adequate lubrication of the parts when using the more viscous fuel, and to prevent binding due to differential expansion with changing fuel temperature. With fuels up to 1,500 seconds, the normal nozzle type proved to be satisfactory. For more viscous fuels, however, the nozzle holes had to be made smaller, as the atomization was no longer so effective as that obtained with the lighter fuel, the jets of fuel being too long and impinging on the piston crown. This fact raises an important question, as the boiler oil tends to enlarge the nozzle holes by erosion and consequently frequent checking of the hole diameter is required in service. For this reason the Sulzer shops at Winterthur had to scrap many otherwise serviceable nozzles that were returned for reconditioning. Here, the nozzle material plays an important part. Many steels quite satisfactory for light fuels, for instance low-alloy case-hardening steels, show accelerated erosion on heavy fuel, while special materials, as for instance high-alloy tool steels or special nitrided steels may be used to advantage.

The cooling of the standard nozzle design was insufficient with fuels of 3,000 seconds and over, and trouble with carbon formation as shown in Fig. 13(a) (Plate 1) was experienced after only forty-eight hours' running. Fig. 14 (Plate 2) shows a small selection drawn from some thirty tests, using various fuel nozzle designs and cooling arrangements. Many tests were run with a device tried some twenty-five years ago and described by Professor Eichelberg[5]. Its principle is the blowing away of any fuel droplets from the nozzle tip after each injection before they have time to carbonize. This is done by an air jet from a small chamber around the nozzle, which is charged during the compression stroke. The results are very promising, but the design requires still more development work, teething troubles with heat cracks and differential expansion having been encountered with the first few designs tried out on the test bed. Finally a special nozzle design, the main features of which are shown in Fig. 15, incorporating more intensive cooling, was developed and gave satisfactory results with all fuels used. It remained clean, as indicated in Fig. 13(b) (Plate 1), after 210 hours of running on the same fuel as used with the nozzle shown in Fig. 13(a) (Plate 1). Even with the fuel of 6,200 seconds, this design developed no carbon deposits, as shown in Fig. 13(c) (Plate 1) after fifty-one running hours. The chief features incorporated in this design have already been satisfactorily proved by the fitting of some 5,000 similar nozzles on engines of a smaller bore.

No trouble was experienced with liner deposits. The liner surface was usually covered by a brownish-black varnish as shown in Figs. 16 and 17 (Plate 2). The exhaust ports showed only small deposits, and the scavenge ports remained practically clean over the whole duration of the tests. Fig. 18 (Plate 3) shows the usual appearance of the piston after an endurance test.

Besides the extensive investigations that have been carried out in the laboratories in recent times and the knowledge acquired over many years with stationary engines, some useful experience of operation on heavy fuels in ship installations has been accumulated during the last few years.

Some information on this is given in two papers read recently before the International Internal Combustion-Engine Congress at Milan by Mr. D. Ruys of the Royal Rotterdam Lloyd and Mr. Sozonoff of the Compagnie Maritime Belge.

Large numbers of the SD type marine engine have already been adapted for operation on heavy oil. Clients have expressed complete satisfaction with the behaviour of the SD72 under these more severe working conditions. In spite of these good results, additional features to facilitate operation on boiler oil have been incorporated in the new design.

During the course of last summer the prototype RSG58/76 was given its shop trials on the test bed (Fig. 19, Plate 3). The chief sections through this engine are shown in Figs. 20(a), 20(b) and 20(c).

FEATURES OF THE RS ENGINES ESPECIALLY SUITED TO HEAVY OIL COMBUSTION

Poor quality heavy fuels generally have a large ash and sulphur content, their combustion residues therefore containing

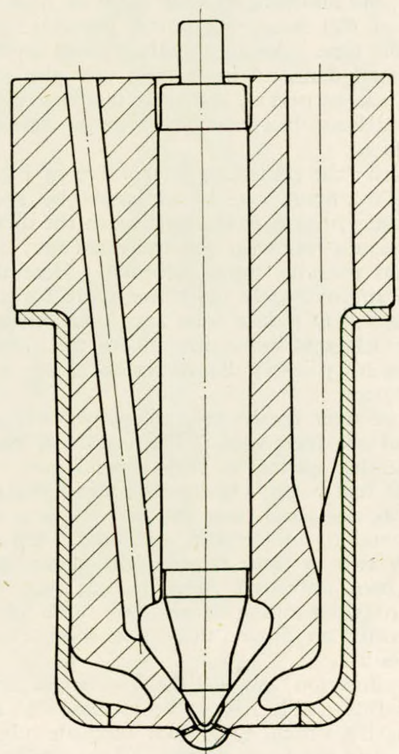


FIG. 15—*Drawing of special nozzle*

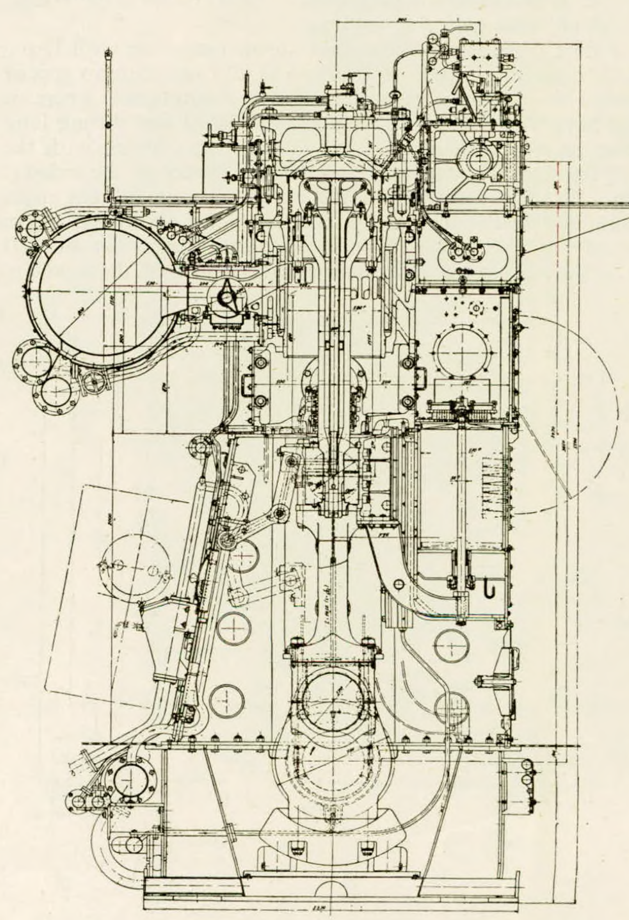


FIG. 20(a)—*10RSG58: cross section through cylinders*

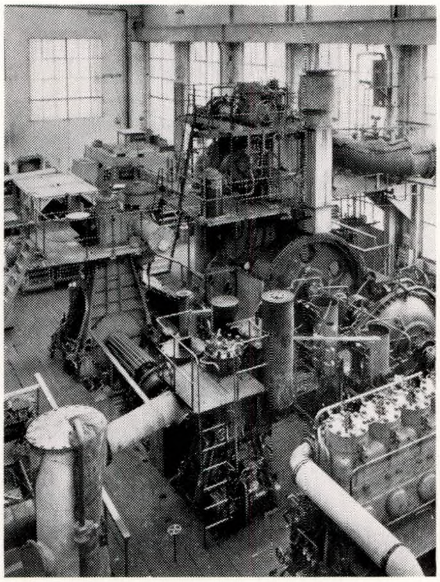


FIG. 5—*720 mm. bore 1,250 mm. stroke, single cylinder, 2-cycle, heavy fuel test installation*

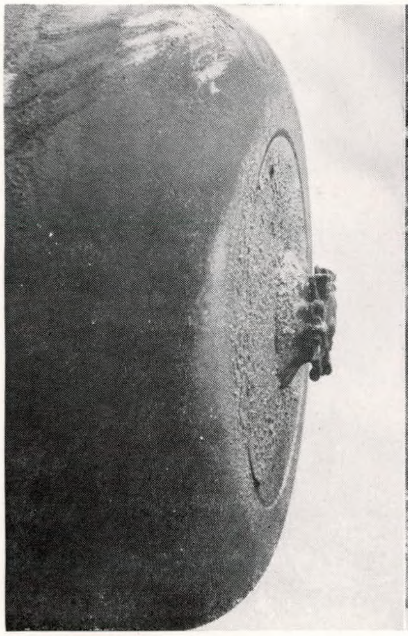


FIG. 13(a)—*Standard fuel nozzle after forty-eight hours' service with 3,000 seconds Redwood I fuel, showing formation of carbon trumpets*

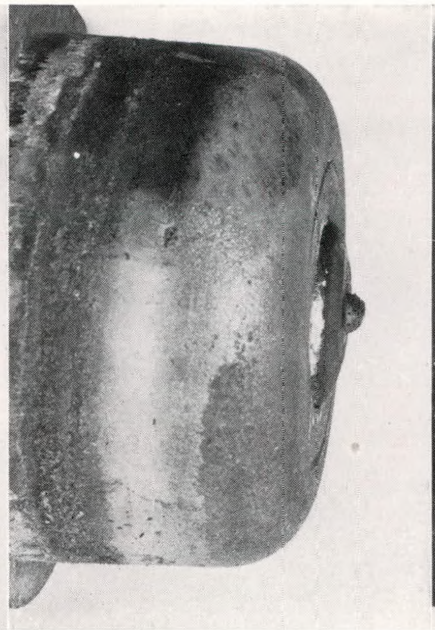


FIG. 13(b)—*Special fuel nozzle, fitted with cooling jacket, after 210 hours' service with 3,000 seconds fuel*

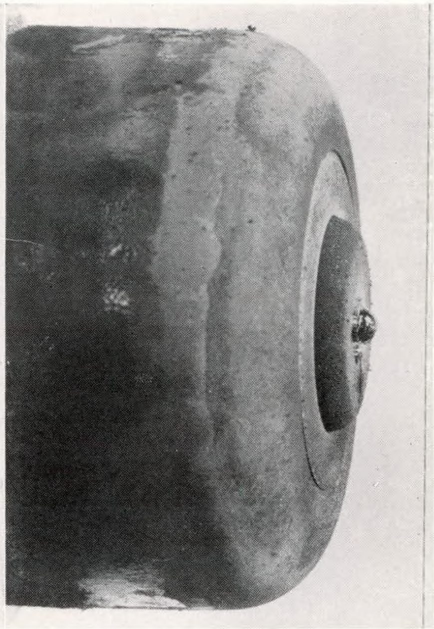


FIG. 13(c)—*Special fuel nozzle, fitted with cooling jacket, after fifty-one hours' service with 6,200 seconds fuel*

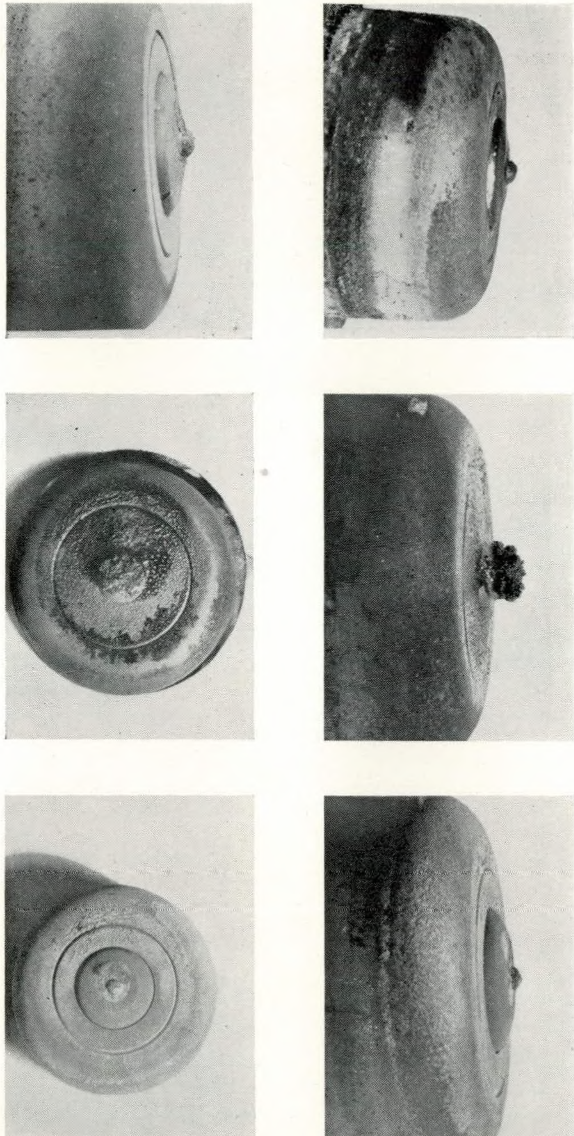


FIG. 14—*Selection of results from nozzles tested*

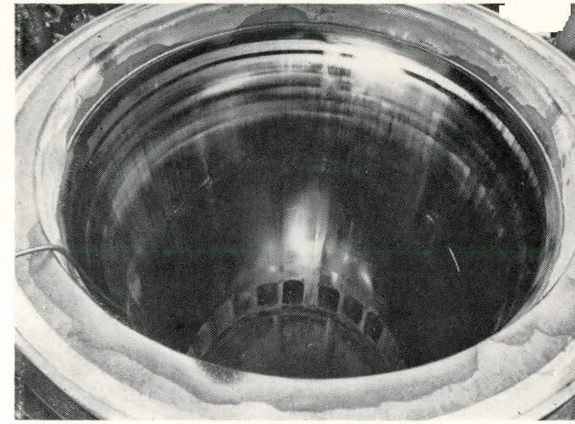


FIG. 16—*Cylinder liner (scavenge side) after 1,550 hours' running on heavy fuel, the last 148 hours in continuous service with 2,600 seconds Redwood I fuel*

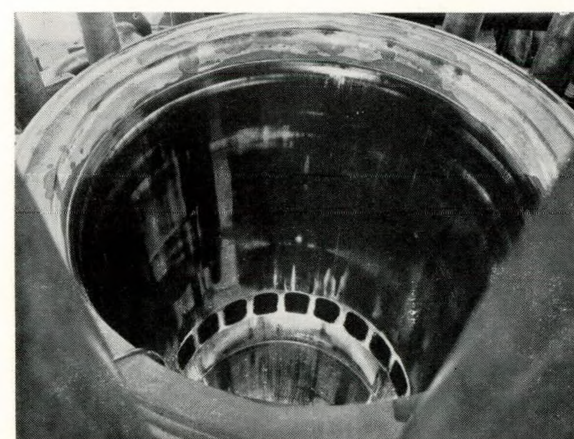


FIG. 17—*As Fig. 11, but showing view of exhaust side*

FIG. 18—*View on scavenge side of piston after 220 hours' continuous service with 3,000 seconds Redwood I fuel*

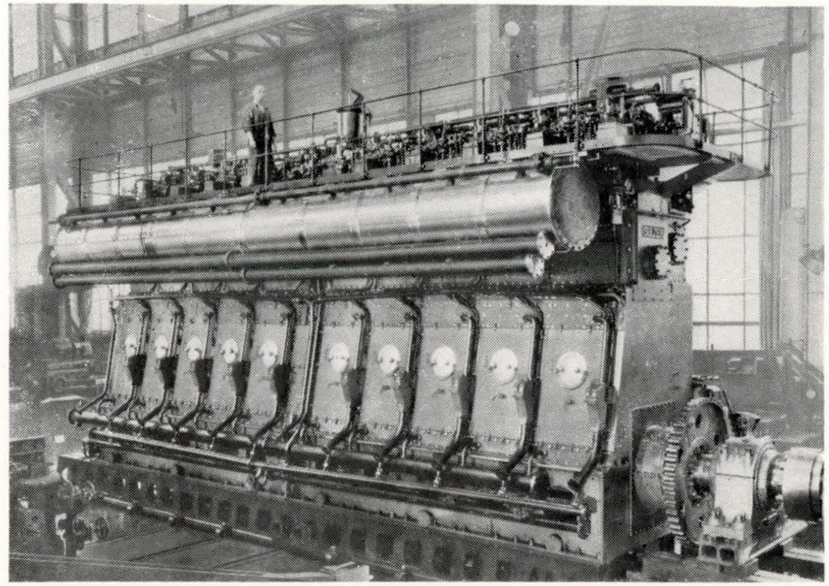


FIG. 19—*10-cylinder marine Diesel engine: 580 mm. bore, 760 mm. stroke, 5,200 b.h.p. developed at 240 r.p.m.*

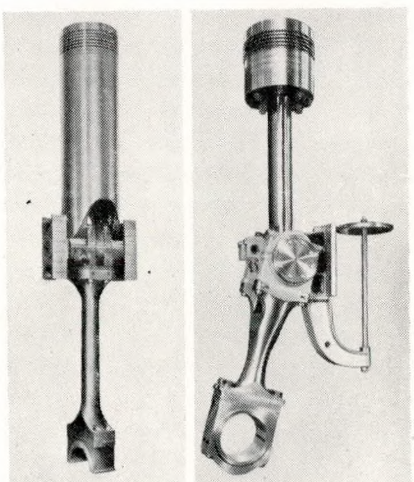


FIG. 22—*Comparison between running gear of SD and RS engines*

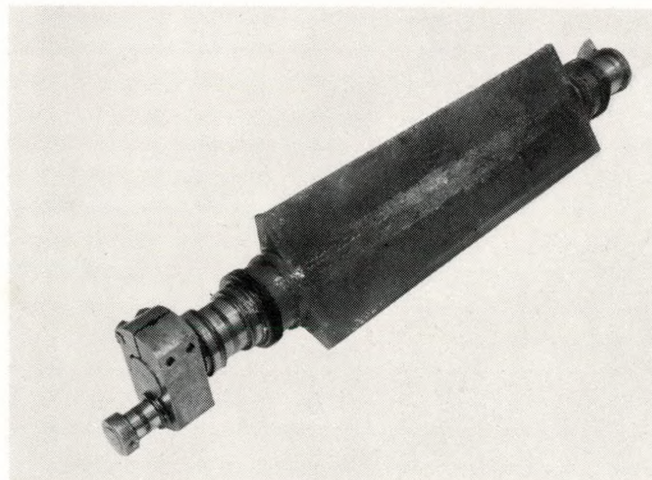


FIG. 23—*Exhaust valve from 720 mm. bore test engine after 1,720 hours' running*

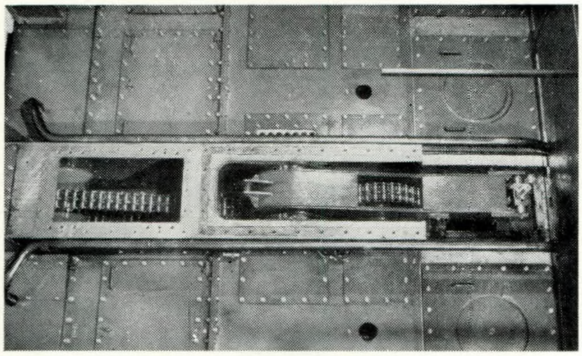


FIG. 25—*Camshaft chain drive with tensioning device*

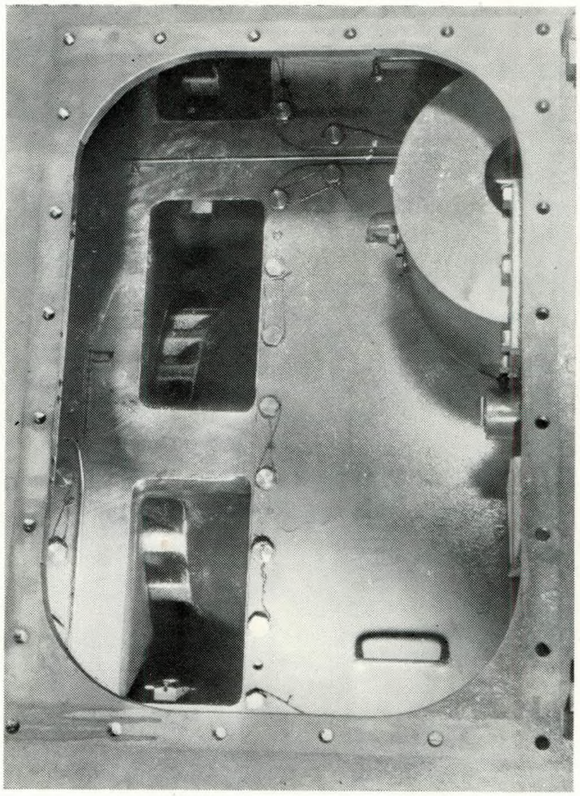


FIG. 27—*Accessibility of scavenge ports through air receiver*

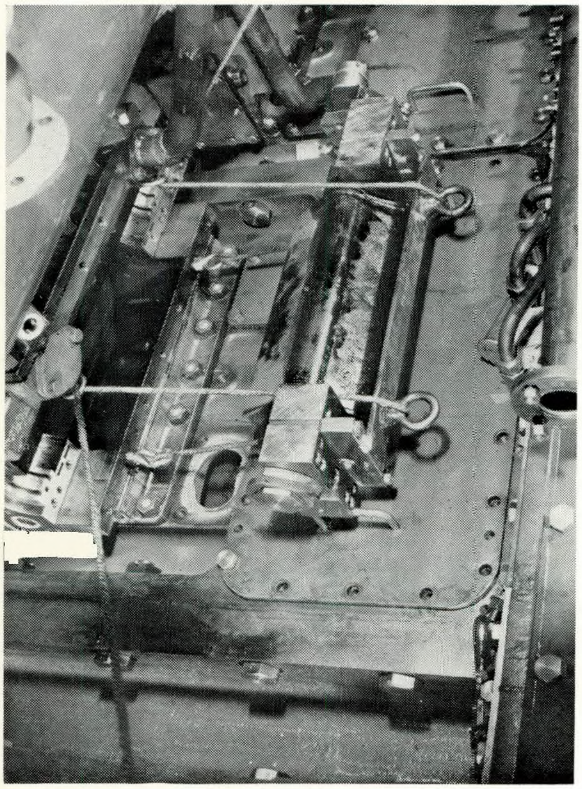


FIG. 24—*Dismantling oscillating exhaust valve*

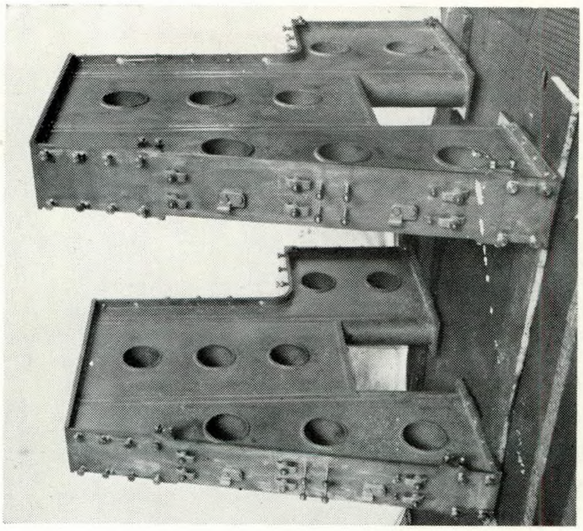


FIG. 26—*A-frame construction showing recess at back for scavenge-pump block*

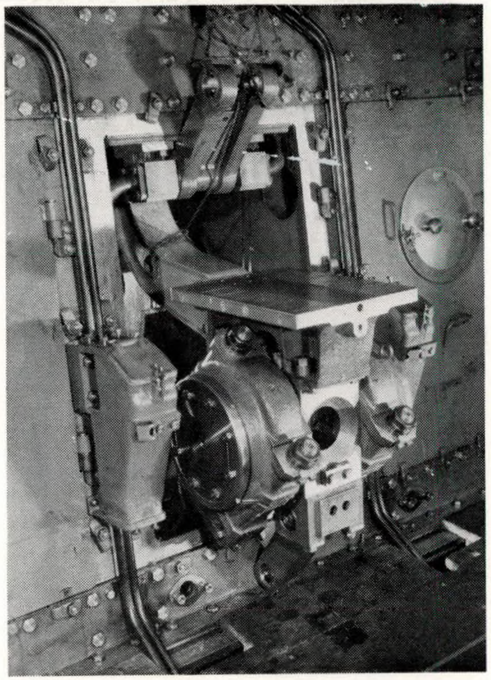


FIG. 28—*Crosshead assembly resting on the side of the crankcase*

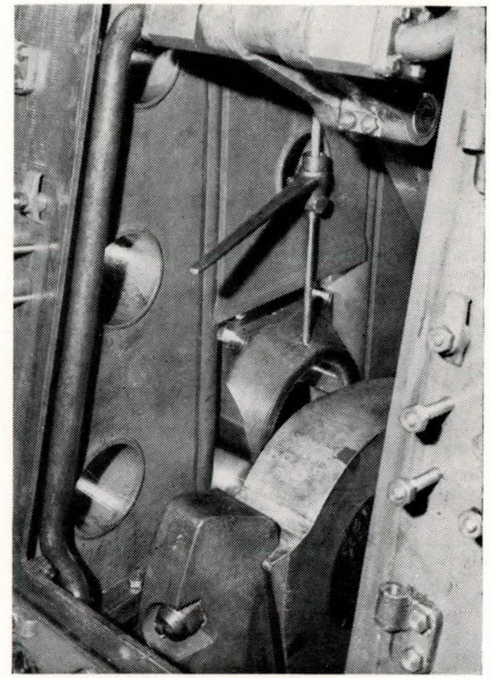


FIG. 29—*View of crankshaft bearing cover in raised position*

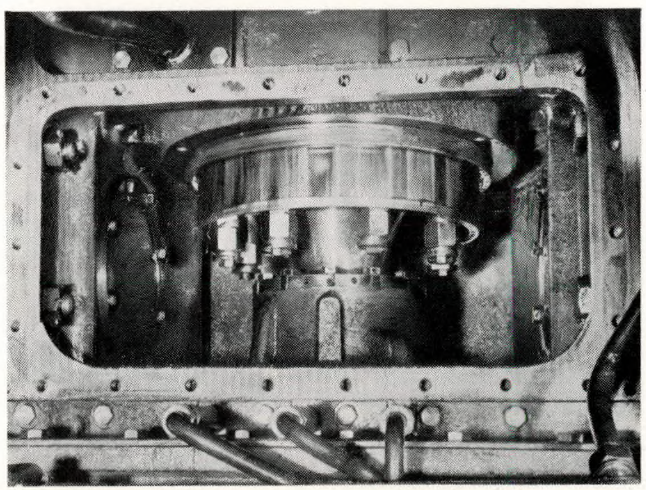


FIG. 30—*View of easily accessible piston securing nuts*

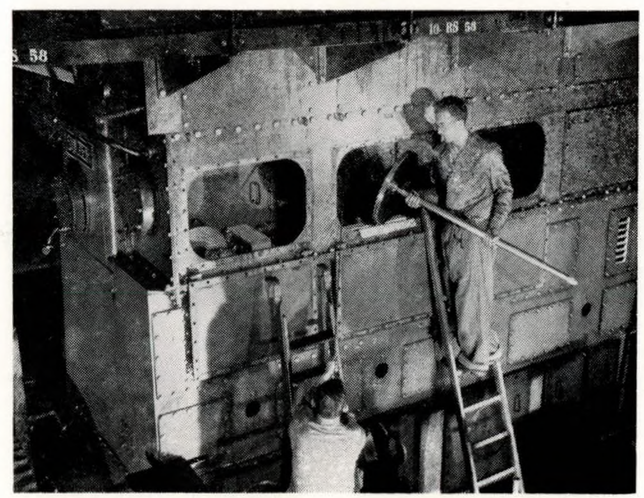
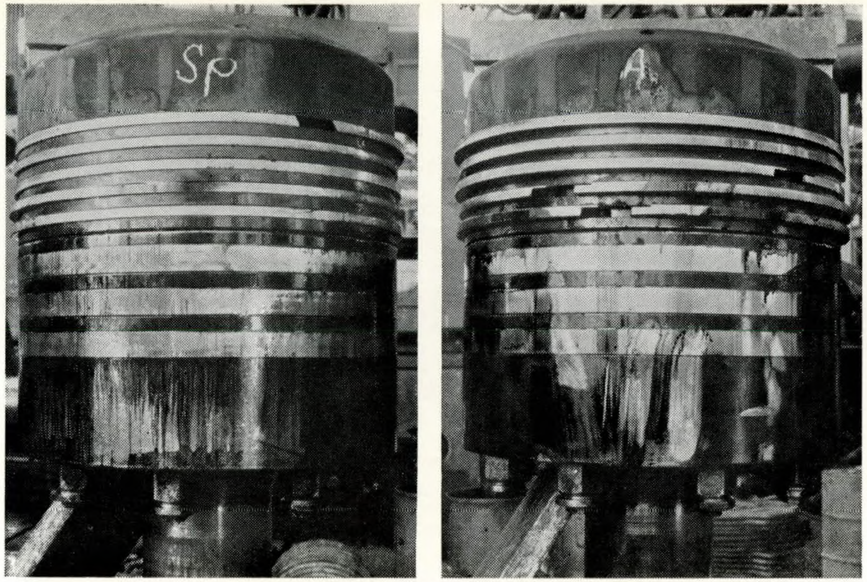
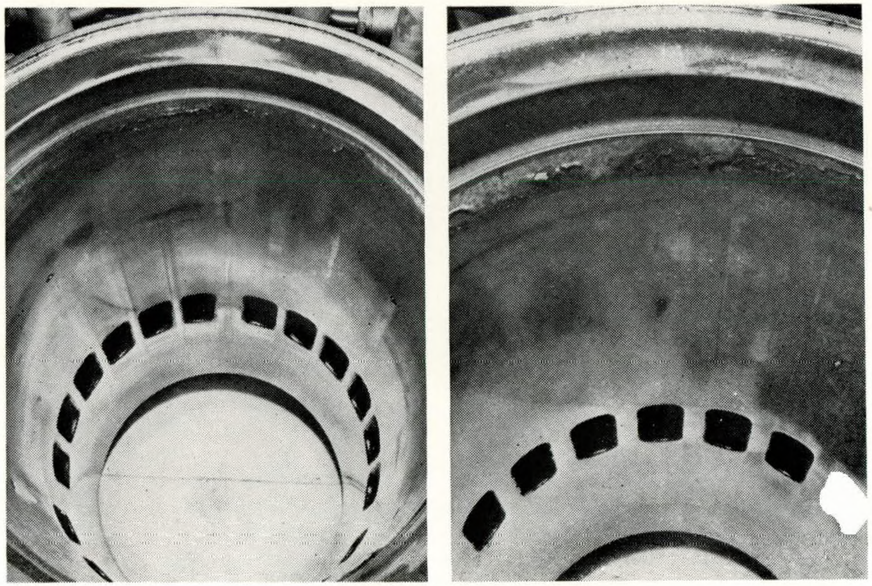


FIG. 31—*Removal of scavenge-air pump piston*





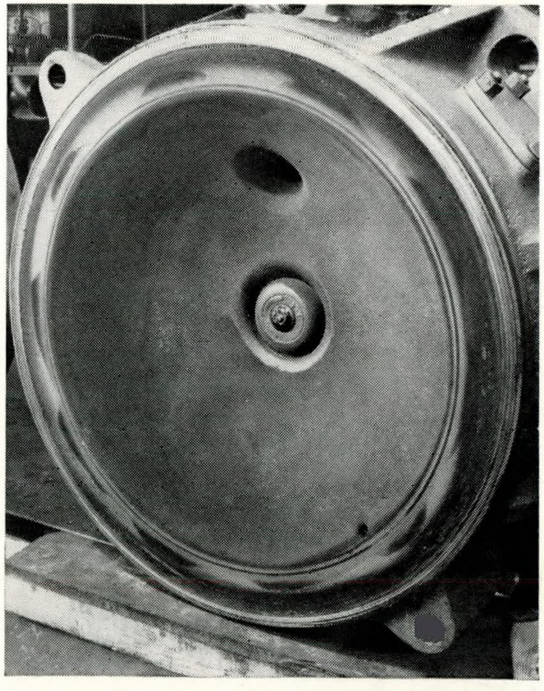


FIG. 33 *(top left)*—*View of scavenge side (left) and exhaust side (right) of main piston after 235 hours' running*

FIG. 34 *(above)*—*Views of cylinder liner ports, scavenge (left) and exhaust (right), after 235 hours' running*

FIG. 35 *(left)*—*View of cylinder head and injector nozzle after 235 hours' running*

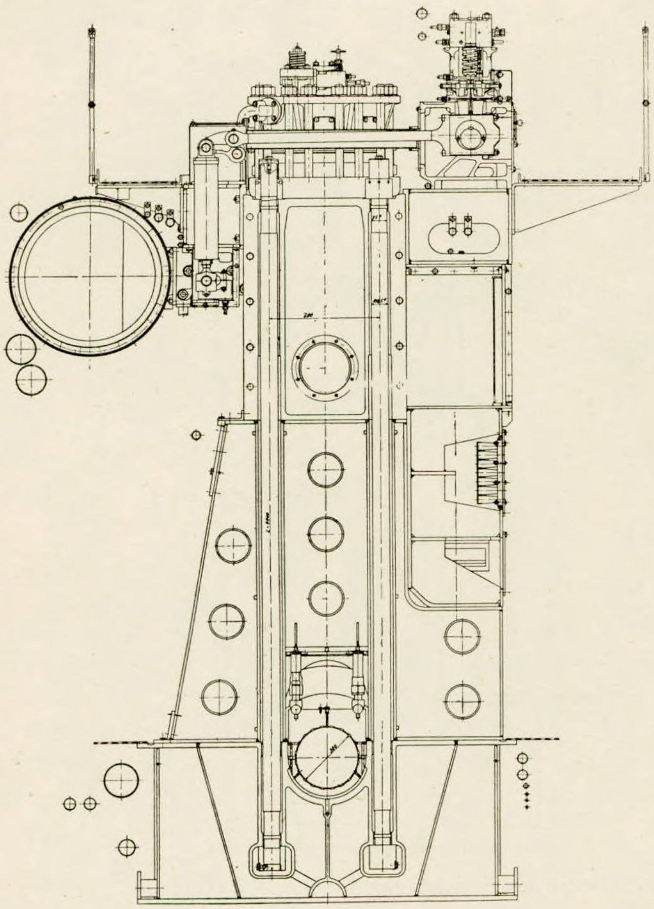


FIG. 20(b)—*10RSG58: cross section through tie rods between the cylinders*

sulphuric acid and abrasive matter. To prevent these injurious products of combustion entering the crankcase, where they may cause corrosion and wear of the crankshaft, connecting rod, crosshead and bearings, the cylinder is separated from the lower part of the engine by a compartment containing a stuffing box (Fig. 21).

This method of sealing is not new on Sulzer engines, a similar stuffing box having been used very successfully on smaller two-stroke crosshead type engines, thousands of which were built some twenty years ago. On the SD range the lantern space allows any gases leaking past the piston rings to escape to the atmosphere, and the double-sided scraper ring device working on the circumference of the pistons is nothing else but a stuffing box. However, this is not so effective as in the new engine, where, as shown in Fig. 21, the stuffing box is sealing on the much smaller diameter of the piston rod.

The clean combustion of heavy oil requires very close control of the injection. To facilitate this the fuel pumps are fitted at cylinder head level (Fig. 20(b)), thus making the length of the high pressure piping between pump and injection valve as short as possible. The flow resistance to high viscosity fuels is thereby reduced and the maximum pump pressures need not be so great. Further, with the shortening of the high pressure pipes, it is unnecessary to heat them.

To improve precise control of the injection, the beginning and end of the fuel pump stroke have both been made adjustable. The injection timing can also be altered quite simply during operation. This will be a distinct advantage, as the ignition qualities of heavy fuel may vary from one bunkering station to another.

The introduction of a neutral air space between cylinder and crankcase, effectively isolating the piston from the latter, reduces the possibility of a crankcase explosion. A piston which is scored and running hot no longer comes into contact with the vapour mixture of oil and air in the crankcase and therefore cannot ignite the mixture. Hence, the main cause of crankcase explosions is removed. However, in conformity with recent tendencies and the recommendations of Lloyd's Register of Shipping, explosion relief valves are provided on the crankcase doors. These latter consist of spring-loaded steel plates that are forced open when the pressure in the crankcase rises above a certain value and close again when this pressure returns to normal. The immediate closing of the valves is very important, preventing a stream of fresh air entering the crankcase after a primary explosion which would assist an incipient fire and promote further explosions.

The provision of a stuffing box normally entails increasing the height of the engine, which is most undesirable. This can be avoided by utilizing a shorter piston (Fig. 22, Plate 3). Against this, however, is the necessity, in an engine provided with transverse scavenging, to have a piston long enough to ensure that at top dead centre the piston skirt covers the exhaust and scavenge ports, preventing the scavenge air escaping into the exhaust manifold. The provision of an oscillating valve takes

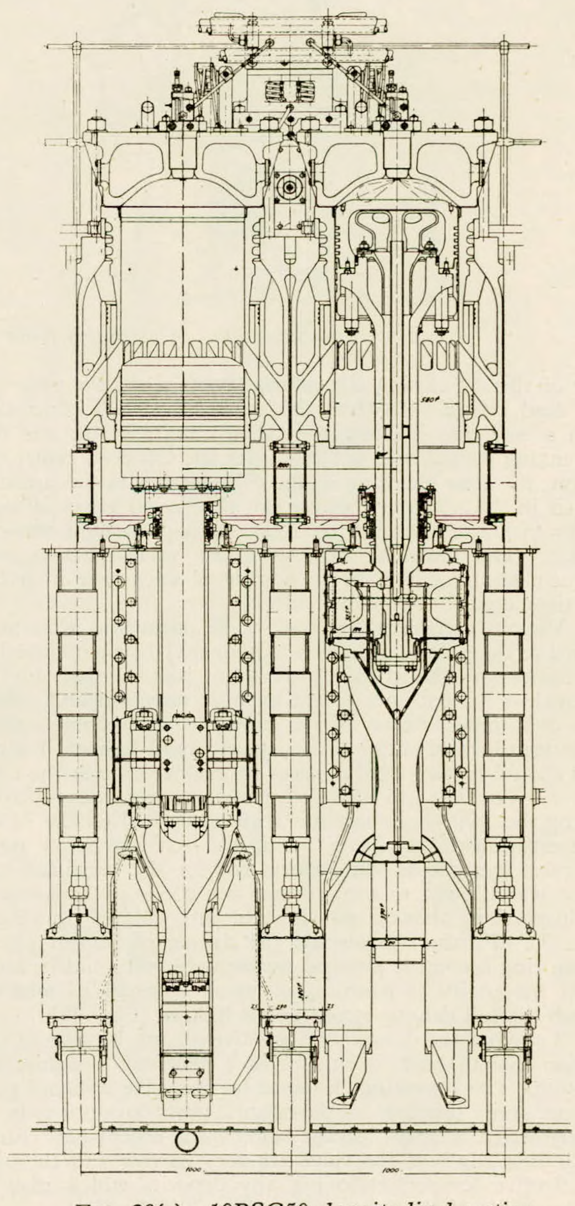


FIG. 20(c)—*10RSG58: longitudinal section*

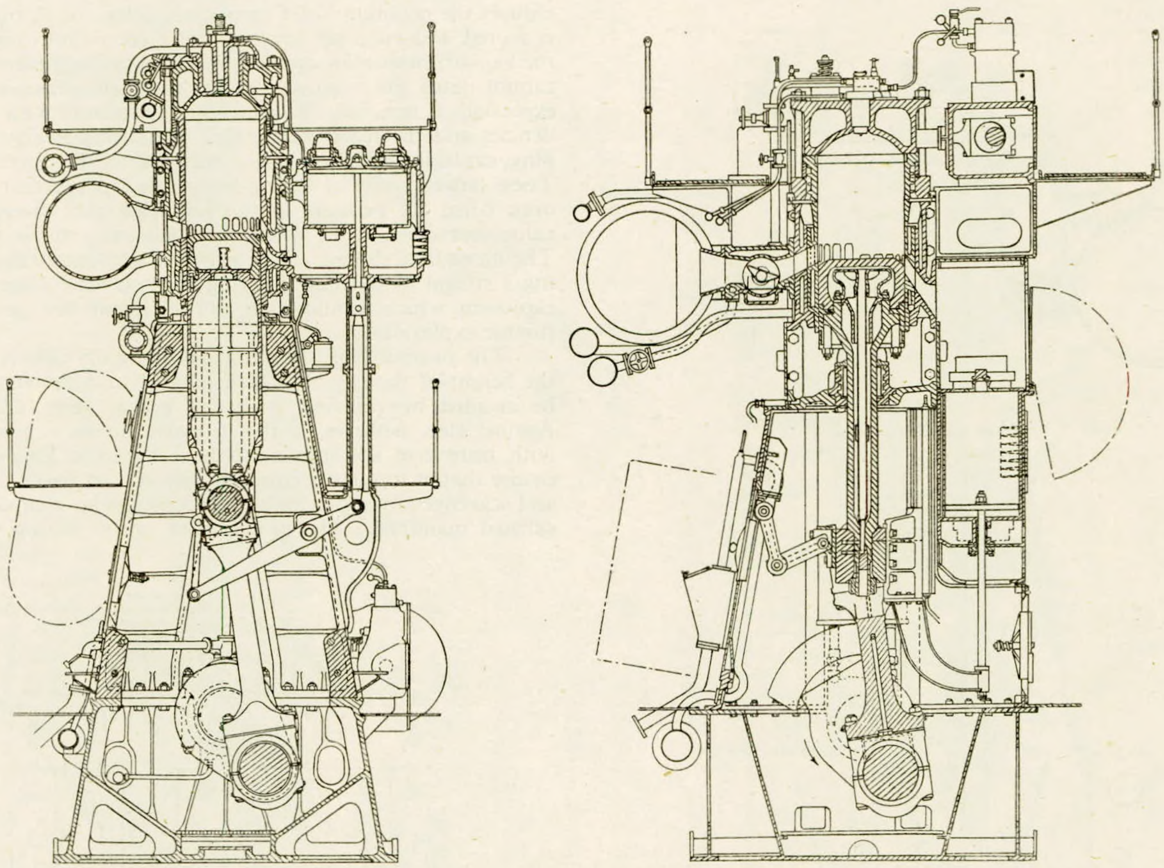


FIG. 21—*Separation of cylinders from crankcase: comparison between the SD72 and RSG58*

care of this by closing the exhaust ports when the piston is at top dead centre. Further, this exhaust valve is controlled in such a way that it closes the exhaust ports at the end of the scavenging period, and not only over the top dead centre of the piston; the after charging of the cylinders up to the pressure of the air in the scavenge receiver can thereby be achieved without the provision of a second row of scavenge ports with non-return valves. The omission of non-return valves, which require frequent cleaning, will be an additional advantage in reducing the time needed for maintenance.

Various considerations have to be taken into account with regard to the potential drawbacks that may be encountered with the use of an exhaust valve. Is it not possible that this valve, situated at the entrance to the exhaust manifold and subjected to a continuous stream of hot gases, would become encrusted with deposits and stick? Also, what would happen if a piston ring should break and the pieces be blown through the exhaust ports? To clear up the first point, such a valve was used during the 2,000-hour test run with heavy fuel on the 720-mm. bore experimental engine. During the first few hours' running the valve was cooled with water, but for the remainder of the test it was allowed to run without any cooling whatsoever and, on inspection, showed no signs of any coking up (Fig. 23, Plate 3). Further, to obviate the danger of broken pieces of piston ring becoming jammed between the valve and its housing when the engine is running, a spring is provided which can absorb the full driving stroke in the linkage (Fig. 20(b)).

To avoid the danger of the valve seizing because of deformation due to heat, it runs with a generous clearance in its housing. This operating clearance for the valve does not prevent it functioning properly, as the exhaust ports have not to be completely sealed, a small leakage being quite acceptable. Further, the sealing edges of the valve are very narrow and they act as an effective scraper, removing any deposits which may form on the inside of the valve casing (Fig. 20(a)).

The valves are driven individually by an oscillating linkage from the camshaft, as shown in Fig. 20(b). This individual drive to each cylinder lends itself to easy dismantling (Fig. 24, Plate 4). The first oscillating valve controlled in this way was used for the aftercharging of a small Sulzer engine as long as twenty-six years ago, but was not developed further at that time.

Not only does the short piston, made possible by the use of the exhaust valve described above, allow a reduction in the overall height of the engine, but it also possesses better running characteristics compared with a long-skirt piston. Thus, the possibility of trouble caused by deformation of the cylinder liner or piston skirt is reduced, and the exact lining-up of the crosshead guides with the cylinder axis is not of such great importance. Also the lubricating-oil consumption will tend to be lower, as, on the one hand, no oil can be pumped from the crankcase because of the stuffing box, and, on the other hand, the smaller piston surface area will consume less oil. Oil scraper rings are no longer necessary, as the oil supplied by the cylinder lubricators should remain to fulfil its purpose in the cylinders.

The camshaft, fitted just under the fuel pumps, is situated some distance from the crankshaft, and the nature of its drive thus raised several problems. The drive for the prototype RS58 and the engines at present building in Britain is effected by means of a Renold roller chain (Fig. 25, Plate 4). In the case of the still bigger distance involved between the camshaft and crankshaft on the long stroke engine with 760 mm. bore, a chain drive is the technically correct, and certainly the cheapest, solution. However, both drive cases are designed in such a way that a gear train may be utilized in place of the roller chain.

Special attention has been paid to the design of the chain tensioning device, which is constructed in such a way that an alteration of the chain tension does not change the phase relationship between crankshaft and camshaft. Adjustment of the chain may be easily effected by tightening or slackening a single nut.

NEW WELDED ENGINE COMPONENTS

The change from heavy cast-iron components to a lighter welded construction was mainly influenced by the fact that during the recent war a considerable amount of experience had been gained—particularly in British shipyards—in the welded construction of ships and prime movers. Coupled with this, after the war there was great difficulty in obtaining castings of the size required for large marine engines. It was decided, therefore, to change radically the design of the new engines in order to suit welding techniques.

In welded crankcases, a great deal of attention must be paid to the distribution of forces throughout the frame. The large and changing internal forces arising from the ignition of the fuel in the combustion chamber must be transmitted directly from the cylinder head to the crankshaft bearings, as far as possible avoiding transverse welded seams. These conditions are fulfilled in the simplest way by incorporating tie-rods (Fig. 20(b)), which have always been used on Sulzer stationary engines. Here the forces are transmitted from the top of the cast cylinder block directly down through the tie-rods to the transverse bearing members in the bedplate. Pretensioning of the tie-rods places cylinder blocks, A-frames and bedplate under a compression such that, even with the action of the ignition forces, they are not subjected to tensile stress. For this reason it is not necessary to stress-relieve the welded components, with the exception of the transverse crankcase members which carry the crankshaft main bearings and to which the tie-rods are secured.

The tie-rods are also used to advantage in the design of the bedplate, as the large crankshaft bearing forces are transmitted directly to the bottom tie-rod nuts. In this way minimum tensile and bending stresses occur. As the tie-rods have been brought as near as possible to the crankshaft bearings in order not to distort the bedplate unnecessarily, there is no room left for securing the crankshaft bearing covers by means of the normal stud arrangement. This is achieved with special jack bolts which are supported by the underside of the A-frame (Fig. 20(b)). The same method has already been used on several other Sulzer engines, including some of double-acting design which have large forces acting upwards away from the crankshaft.

In order to make the fabrication of the A-frames as easy as possible, a departure was made from the earlier double-sided crosshead guides in favour of the single-sided type. The A-frames now consist of a very simple box-form structure, the main wall plates of which are braced to one another by welded-in pipes of ample dimeter, as shown in Fig. 26 (Plate 4).

Normally, on engines designed for operation with single-sided crosshead guides, the individual A-frames are bolted together with heavy plates, these serving the double purpose of supporting the guides and ensuring the longitudinal rigidity of the engine as a whole. However, by suitable arrangement of the side-mounted scavenge-air pump housings in two rigid blocks, these have been utilized to act both as crosshead guide carriers and to provide longitudinal rigidity. The scavenge-pumps are individual, following the practice employed on the SD60 and SD72, and discharge the scavenge air into a common receiver.

Naturally, the forms used for bedplate and A-frame, which are specially suited to welding conditions, can also be made in cast iron, but in place of a box form the open, ribbed type of construction would be employed. However, all the other parts remain unaltered for the engine supplied with a cast crankcase. An approximate saving in weight of 12 to 15 per cent in favour of the welded engine can prove, under certain conditions, to be of considerable advantage.

OTHER NEW FEATURES

The scavenge-air pump piston is driven in the simplest possible manner. The drive on the SD72 engine is effected with a rocking lever, requiring several small bearings and a small crosshead on the scavenge-pump piston (Fig. 21). On the new engines, this has been very simply overcome by driving the scavenge piston with an arm bolted on directly to the bottom of the crosshead. With this solution, the scavenge pump piston has the same stroke and therefore the same speed as the main piston. This piston speed is fairly high for an air pump. However, as scavenge-pump valves of special Sulzer design are available which possess a large enough free cross-sectional area in relation to the space available for fitting them, the air velocity through the valves is retained at its normal low value, in spite of the increased scavenge-pump piston speed.

The piston cooling oil, and that for the lubrication of the running parts, is no longer carried by telescopic tubes but through swing links as shown in Fig. 20(a). This alteration removes the cause of the pressure fluctuations that occur in a piston cooling-oil system through the pumping action of the telescopic tubes. The cooling oil is guided up through the bored-out piston rod to the piston, where it is forced over the inner surface of the hot piston crown and then returned downwards through a pipe placed in the centre of the piston rod. The oil for the lubrication of crosshead bearing, crosshead guides and big-end bearing is tapped off the piston cooling-oil circuit on the inlet side. Supplying the oil necessary for lubricating the big-end bearing through a hole bored out in the connecting rod has the advantage that no oil passages are necessary in the crankshaft. In this way the fatigue strength of the crankshaft is increased. This method of crankpin lubrication has been adopted on all large Sulzer engines for more than fifteen years. The lubrication of the crankshaft main bearings is effected in the normal manner through oil pipes situated in the bedplate.

OVERHAULING AND MAINTENANCE

When the overhauling aspect is considered, it will be seen that there are many advantages in the new construction. For example, the cylinder cover has not to be removed in order that the ports may be cleaned. For this the piston is brought round into the top dead centre position, and, by opening a cover in the scavenge receiver (Fig. 27, Plate 4), the deposits in the scavenge ports may be quickly knocked out with the aid of a bar. These deposits then fall on to the dividing wall between the cylinder and crankcase, where they may remain until a convenient time for their removal, which may be easily effected by taking off a further cover on the exhaust side of the engine. On this side also, by removing the small cover on the exhaust valve housing and by placing the valve in a horizontal position, the deposits may be knocked out from the exhaust ports in the same manner as those on the scavenge side.

The crosshead can be removed after dismantling the steel retaining bars, which may be done with the removal of a few bolts, and uncoupling of the piston rod. It is now possible for the crosshead still attached to the connecting rod, to be swung out from its position, pivoting on the big-end bearing and to come into the position shown in Fig. 28 (Plate 5). It may then be easily lifted out of the engine with the assistance of a crane after disconnexion of the small end bearing covers.

Each main bearing may be dismantled for inspection as usual without removal or raising of the crankshaft. This is accomplished by first drawing the bearing cover upwards by means of two threaded rods (Fig. 29, Plate 5) and removing it. The crankshaft is then rotated and carries the bottom bearing shell round with the aid of a small fixture. The shell may then be lifted out.

There are two methods that may be used for dismantling the piston: either it is removed in the normal manner together with the piston rod, in which case the rod connexion to the crosshead is uncoupled through one of the crankcase doors, whereby the piston and rod may be lifted out from above, or the crankcase is left closed and the cover on the chamber under the piston is opened on the exhaust side (Fig. 30, Plate 5), and the nuts securing the piston to the piston rod are removed with a special spanner, the piston then being drawn without the piston rod. This latter dismantling method has the advantage that, whenever the piston is drawn, the inside may be inspected and any deposits left by the cooling oil can be removed.

The camshaft may be rolled out sideways on to special supports without dismantling the fuel pumps. However, it is not necessary to remove the camshaft in order to adjust or replace the fuel cams. The latter are made in two pieces and are provided with narrow serrations which correspond to similar ones on a bush keyed to the shaft, thus allowing a fine adjustment of the cam angle. The scavenge-pump piston and piston rod are extremely light and may be removed through the scavenge-air receiver without the help of an overhead crane, as shown in Fig. 31 (Plate 5).

SHOP TRIALS OF 10RSG58

General

The test bed trials of the 580 mm. bore 10-cylinder prototype RSG58 were carried out in great detail and included endurance runs on heavy oil. The results proved to be extremely satisfactory. The long test run of 235 hours' duration, 100 hours of which was run on heavy fuel, would have shown up any weaknesses; it also gave a good indication of how the engine would behave when running on heavy oil over a lengthy period. No signs of trouble were experienced at all with the new features, such as the welded A-frames, bedplate and scavenge pumps, the short piston, the exhaust valve, the division between cylinder and crankcase and also the piston rod and stuffing box. The new double-controlled fuel pumps, situated at cylinder cover level, ran satisfactorily, and, together with the possibility of adjusting the injection timing during running, they may be considered as a definite step forward in the correct

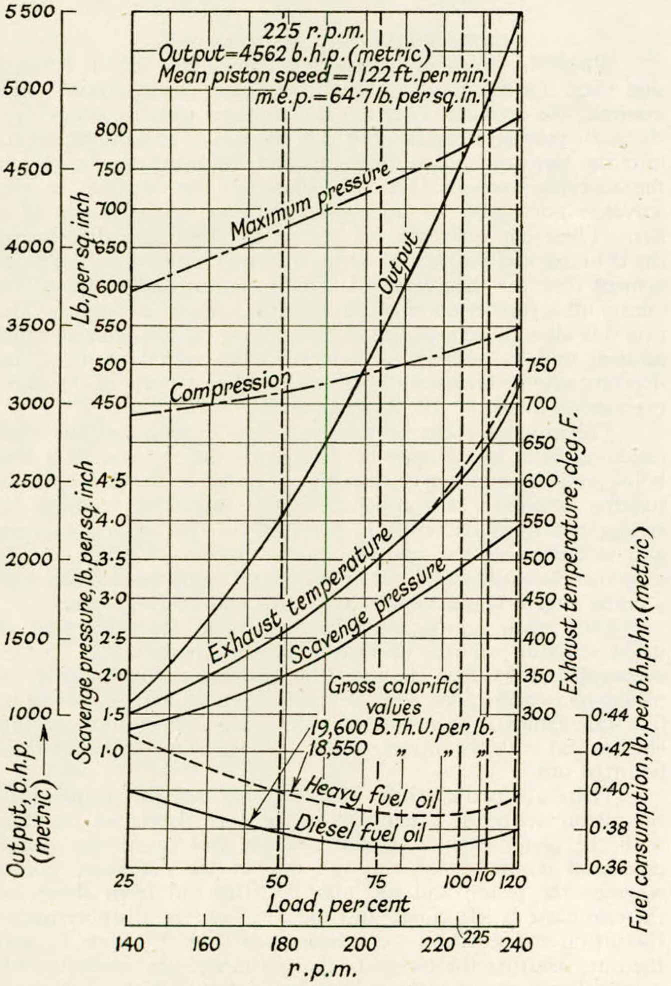


FIG. 32—*Test results*

burning of heavy fuels. A few minor teething troubles such as oil leaks, vibration of one or two components, etc., which are normally encountered on a new design, were located and rectified at the time of the tests.

During running, the engine remained strikingly still on the test stand, no abnormal vibrations being noted. Although the engine noise was slightly greater than that found on engines of a lower speed, it was by no means in excess of that for units of a comparable speed. A further reduction in the noise level has now been achieved by shrouding the suction openings to the scavenge-air pumps.

During its 235 hours on the test bed the engine clocked over three million revolutions in all, which, for the estimation of the fatigue strength of the welded bedplate and A-frames as well as the connecting rod and the bearings, piston rod and general engine components, allows a reasonable although not final assessment to be made.

Results

Running on Diesel oil, the fuel consumption at full load lay between 0·370 and 0·375lb. per b.h.p.-hr. The heat consumption for Diesel and heavy oil was exactly the same; owing to the smaller calorific value the consumption of fuel in the latter case was, of course, correspondingly greater. The fuel consumption curve was strikingly flat; in the region between half-load and approximately 10 per cent overload, the consumption varied by only some 0·005lb. per b.h.p.-hr. (Fig. 32). On overload, the rise of the fuel curve was very moderate, which indicates that a large load reserve was still available. The maximum load attained on the test bed was 6,000 b.h.p., which corresponds to a mean effective pressure of 80lb. per sq. in.; at this load the exhaust was still absolutely clear. The volume of scavenge air was 1·3 × piston displacement at a scavenge-air pressure of 3·4lb. per sq. in. The exhaust temperature at full load equalled 320 deg. C. (610 deg. F.) at 225 r.p.m. with an m.e.p. of 64·6lb. per sq. in.

The pistons and cylinder liners were in perfect condition when they were dismantled after 235 hours' operation (Figs. 33 and 34, Plate 6), 100 hours of which was run on heavy oil of 1,500 seconds Redwood I at 100 deg. F. The scavenge and exhaust ports were clean, giving no indication of fouling, and the injection nozzles were without a carbon deposit, as shown in Fig. 35 (Plate 6). The exhaust valves were quite free of any deposits corresponding to the one shown in Fig. 23 (Plate 3), and, further, all engine bearings were in absolutely perfect condition.

The welded parts were carefully scrutinized for cracks, especially bedplate and A-frames, as the latter had not been stress-relieved. No cracks whatsoever could be found.

During the trials the normal tests were supplemented by investigations with strain gauges, electrical pressure pick-ups, oscillographs, sampling valves, etc., in order to measure the stresses set up in various components, pressure fluctuations, scavenging efficiencies and so on. The results of these tests confirmed the calculated figures.

FUTURE DEVELOPMENT

For the further development of the RS-type engines, a single-cylinder RS test engine will be used. With this engine, a number of questions can be investigated. Extensive supercharging tests will be made, together with more investigations concerning heavy oil, especially with a view to finding a means of further reducing cylinder wear.

The possibility of raising the mean effective pressure with supercharging has been mentioned earlier. For more than fifteen years, a high percentage of Sulzer four-stroke engines have been supercharged on the Büchi principle. Several engines are in service running with a mean effective pressure of 166lb. per sq. in. Notes on development work carried out on two-stroke supercharging have already been published[6]. In this field, not only have mean effective pressures of 256lb. per sq. in. been reached and over 20,000 hours been run at mean effective pressures of 140 to 170lb. per sq. in. on smaller engines, but endurance tests of several hundred hours' duration at 149lb.

per sq. in. have also been carried out with a 4,000-b.h.p. engine.

Immediately after the completion of heavy-oil tests, lasting over 2,000 hours, on the two-stroke single-cylinder 720 mm. bore test engine, supercharging trials were made. A test with low supercharging was run at an m.e.p. of 100lb. per sq. in. for 72 hours, followed immediately by an overload test in which an m.e.p. of 128lb. per sq. in. was attained, the exhaust remaining quite clear. However, this does not mean that it is expected to deliver the large bore engines with a mean effective pressure of 128lb. per sq. in. in the near future. The conclusion of the tests on the single-cylinder RS58 may lead to the use of RS engines of 580 and 760 mm. bore for ship propulsion and stationary installations with moderate supercharging and a reasonably increased mean effective pressure.

CONCLUSION

The first RS58 engine built in Winterthur is to be installed in the 10,780 d.w. cargo ship *Middlesex* for the New Zealand Shipping Co., Ltd. The second engine for this installation is being built in Britain, together with four twelve-cylinder and two ten-cylinder engines of the same bore. Further, two seven-cylinder engines are also being built in Holland.

For the 760 mm. bore machine, two engines are being built at the Sulzer works in Switzerland, two in France and three in Japan.

The individual components of the above engines are in no way radically new, the chief aim having been to incorporate the best possible arrangement of well tried features in the light of past experience, and this, it is hoped, has been substantially achieved.

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Discussion

MR. A. G. ARNOLD (Member) congratulated the author on an excellent and extremely interesting paper and on the way it had been presented. It was fair to say that if most of those present were required to speak in one of the many languages Mr. Kilchenmann was known to possess, they would hardly have made such a good job of it.

There were a few points which he would like to take up. First of all, Mr. Kilchenmann had a great many friends in Denmark. On page 137 of the paper, he had said that the cylinder liner limit was 760 mm. (approximately 30 inches). He himself had before him a pamphlet by the late Dr. H. H. Blache written in 1933 referring to an engine built in 1931. It was a two-stroke double-acting engine. The bore was 840 mm., approximately 33·5 inches, and the stroke was 1,500 mm. plus 400. The engine was still running. There might have been some trouble, but certainly not with the cylinder liners. He believed it was the largest Diesel engine in the world. He felt sure that in fairness to his Danish friends, Mr. Kilchenmann would like to correct his statement.

Those who had been fortunate enough to visit Winterthur were familiar with the high standard of the work put into the engines and he felt sure the care taken in recording the results of the tests of the engines was such that they could not be more accurate.

The author pointed out that in many plants sufficient attention had not been paid to the preparation of the fuel system for separating and purifying fuel oil. He could not more readily agree with that. Fig. 7 was very much in accordance with general practice and certainly conformed to his own experience. He had had a lot of good experience with two separators dealing with 30 tons, and he hoped shortly to have similar experience with three separators dealing with 50 tons, and Mr. Kilchenmann's graph was a great comfort to him.

Mr. Kilchenmann would like to use a centrifuge under pressure at a very much higher temperature. He would very much appreciate it if the author would enlarge upon this. Had he investigated the possibility of serious accidents, because some oils were less stable than others. The Ministry of Transport and Lloyd's Register of Shipping would no doubt require very thorough tests to be carried out on this system before it could be adopted. Any remarks on that matter would be of great interest.

In his arrangement for separating fuel, the author had not shown how to get rid of the sludge, etc. If the burning of boiler oil was to be successful, it was essential to consider the personnel who would be asked to clean and maintain these machines. The author said that additional personnel would have to be employed for the cleaning of the separators, but it would be very much better to concentrate upon machines of the self-cleaning type, and not engage additional labour.

Fig. 8 was a very interesting graph, showing how viscosity could be reduced by mixing heavy oil with gas oil. Could the author say how he proposed to do this in a ship normally carrying bunkers in the double bottom tanks? The economics would have to be studied carefully, as gas oil today cost 270s. against 236s. for Diesel oil. Would it not be better, as the author had suggested in another paragraph, to limit the viscosity of the bunkers to, say, 1,500-1,750 seconds Redwood I?

On page 143 the author said that for starting and manœuvring it was not necessary to change over to Diesel oil. If this meant that the engine would start and burn the fuel and manœuvring would be carried out successfully, he readily agreed. This had been well proved. But he would like to know whether the author still considered that the disadvantages of changing over to Diesel oil outweighed the advantages of remaining on boiler fuel.

The question of heating the fuel was of considerable importance, but there were ships with a boiler that was not exhaust

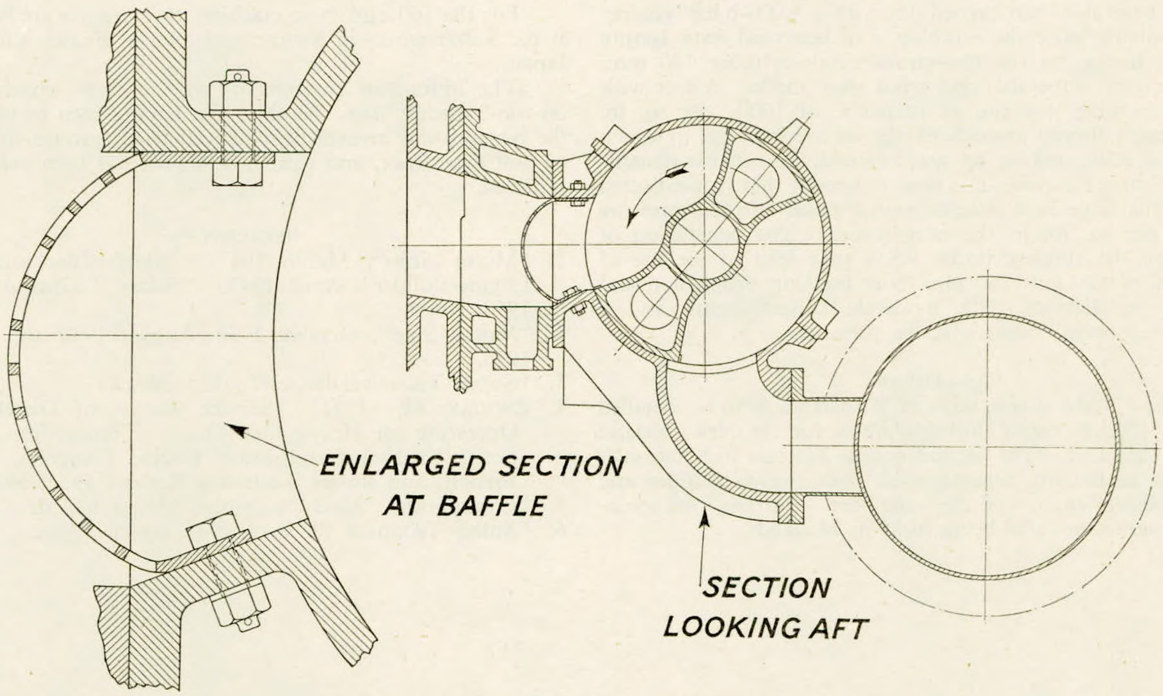

FIG. 36—*M.V.* Eurybates: *baffle for main engine exhaust valve*
Area through cylinder exhaust port at end of release = 93·5 sq. in.
Area through baffle = 126·5 sq. in.
Area through rotary valve = 186·2 sq. in.

gas fired, so that one could have heat whilst manœuvring at slow speed and so on. He had proved to his satisfaction that the change was well worth while.

On page 144 the author remarked that for some viscous fuels the fuel valve nozzle holes had to be made smaller; others had expressed this view, but he had certainly not found it necessary in the engines in which he was interested. In one or two cases where it had been done, it had been found to be a mistake. He was at a loss to understand how this theory could have come about. He readily agreed that if the nozzles were not made of the correct material or with the correct care, fuel (Diesel oil as well as boiler oil) would enlarge the holes and sometimes make them irregular, misdirecting the sprays. But these matters could be controlled and he would be interested to know why this point had been stressed.

He agreed that the manufacturers of fuel valves and fuel valve nozzles could often pay greater attention to the material from which these very important parts were made.

Reference was made to the introduction of high pressure air to blow away any droplets at the nozzle tips. It was also said that this had caused the nozzles to crack and he felt that this was bound to happen.

On page 145 reference was made to a special arrangement for adjusting at the beginning and end of the fuel pump stroke in order to improve precise control of the injection. This seemed to be a novel idea, and it would be greatly appreciated if the author would enlarge on this point.

On page 146 reference was made to the exhaust controlling valve, and the author had described this very well in presenting his paper. He had had considerable experience of this type of valve, and would like to explain what was done to stop the troublesome feature of broken piston rings affecting the valves. The valve shown in Fig. 36 was fitted into a two-stroke single-acting engine, similar in every way to the valve referred to by the author. With the original arrangement a number of rings were broken, but the valve was so strong that it spoilt the bearings in which the spindle was running. This was stopped by fitting a baffle as shown and the results were extremely satisfactory. There was no restriction to the gas flow.

It would appear from the author's illustration that labyrinth packing was used to seal the glands. Most people believed this to be correct, but in his experience it was wrong, as the glands could not be kept tight and the atmosphere became polluted. Something had to be done, so the glands were packed with air, taken from the scavenge pump, with extremely satisfactory results. It cooled the spindles and the engine room atmosphere became normal. The valve shown in the drawing was fitted in 1925 and the baffle had been fitted for some five years.

On page 147 the author referred to the fabricated bedplates, stating that they were not stress relieved or annealed. Most people believed that these parts should be so treated and he wondered whether the reason for this non-treatment was because of the lack of a furnace large enough to take these parts.

The screw jack arrangement for securing the main bearing keep was not new. M.A.N. adopted this many years ago, particularly with their smaller engine, but the experience in service was not satisfactory. Could the author state what provided the elasticity with this arrangement that was usual with the long mild steel studs.

The piston cooling system had been described. The cold oil passed up between pipe and rod and returned down the pipe. He wondered whether the author could say why the reverse was not the case. In most cases everything possible was done to reduce the dew point. If such excellent results were being obtained in cylinder liner wear, would there not be even better results with the lower dew point? The answer might be that the cylinder liner wear was not affected, because the rod was so far away. But then, of course, there was the packing through which the rod was running, and this was similarly affected. It was the dew point which caused corrosion and set up wear, and so it went on! He would like to hear a little more about this.

Under "Overhauling and Maintenance", the author said the deposits in the scavenge ports could be knocked out with the aid of a bar. If the exhaust ports had not been mentioned, he would have concluded that this was a printer's error. He had never seen more than a dirty oily film on the sides of scavenge ports. Consequently, nothing could be knocked out of any port with which he was familiar. He could not understand why there should be any difference in the author's engine. Perhaps he would be good enough to enlighten them.

On page 148 it was said that the volume of scavenge air was 1·3 × piston displacement, which would seem to be less than normal. He would like to know how this compared with the SD series of engines. The engine seemed to be a very suitable one for supercharging, particularly with the rotary valve, but something had been done away with that he had always envied. The patent was so thoroughly well guarded. He referred to what he termed the "mouth organ" valve. He had always thought it an extremely interesting fitting to the Sulzer engines. He had never believed that it could offer much resistance to air flow but perhaps the author would say a little about it. It was an excellent valve and he could see many virtues in it. It would stop many ailments, scavenge belt fires being the one uppermost in his mind.

There were many other points which he would have liked to mention. The author had not mentioned the weight per h.p. of his engine. It was true that with fabrication over cast iron there would be a reduction. Furthermore, he could not remember reading of the weight of the bigger engines. Mr. Calderwood had given a paper* in Liverpool on the smaller ones, and the weight was very good indeed. For the bigger ones, he did not seem to have seen any weight per h.p. or any figure for total weight. Then there was cylinder liner wear. With supercharging, the cylinder liner wear would be reduced. It was certainly reduced in the case of the four-stroke cycle, and he could see no reason why it should not be reduced in the two-stroke cycle, always providing the combustion was good.

The author referred to fuel oil pumps on the top of the engine and that was very good. He had shortened the fuel discharge pipes. Some people, including himself would ask why there should be any pipes at all with this type of engine. A gas-operated pump with no pipes could very easily be applied.

The author said that the chain drive of the camshaft, which was a very considerable length of chain, was provided with a special tightening arrangement so that it was not necessary to alter the position of the cams when it was tightened up. He would like a little enlightenment about that. He had been trying to achieve such an arrangement, and he had never succeeded. Chain drives had been used since 1926 but neither he nor any member of his staff had ever succeeded in tightening these chains without making a slight alteration to the setting. This, of course, could only be detected with the compression diagrams which should always be taken preceding the power diagrams.

MR. G. R. GRANGE, D.S.O., M.C., B.Sc. (Vice-President) said that in speaking so early in the proceedings he was in rather a difficult position. Obviously if he started to criticize the Sulzer engine, Mr. Kilchenmann would at once say, "Why didn't you say that when you started to build it?" For his company had built the second engine of the *Middlesex*. However, there were one or two points on which he would like guidance.

First of all, in his diagram in Fig. 11, the author showed the hot fuel returning to the service tank. This must be a mistake; the temperature of the service tank would go up above 125 deg. F. which was, he thought, the figure allowed by the Ministry of Transport. He agreed that there were some baffles in the tank, showing it going down to the suction, but he did not think that would stop the tank from overheating. The return should, he thought, come back into the suction of the booster pump.

The exhaust valve was a great improvement on the old valves in the scavenge main. They used to soot up and cleaning them was a long slow job. This rotary valve in the exhaust was completely trouble-free so far. It would be a great improvement and lent itself to supercharging and to the blanking up of a cylinder if anything happened to it, as the valve could easily be disconnected and put in the closed position.

There was one disadvantage about the exhaust valve; it meant having a very large overhung exhaust pipe. In the diagram of the engine the exhaust pipe stood out a long way from the engine. In a geared engine this meant putting the engines further apart in order to house the exhaust pipe, with the consequence that the gear-wheels were larger than they need be. It was a difficult problem to overcome. One had to have the exhaust valve close to the exhaust ports, and had to have a Venturi from it to the exhaust pipe. He did not see how it could be avoided. But it meant spacing the gear case and engines further apart than they need be, and the gear case was consequently more expensive.

He noticed from the experiments made on the 760 mm. engine that the cylinder wear athwartships and fore and aft showed a very big difference. The fore and aft wear was almost double the athwartships wear. He did not know whether this was general experience; and he would like to know the reason for it.

As a matter of interest, torsional vibration dampers of the viscous fluid type were fitted in the engines in the *Middlesex*. The maximum stress as shown by strain gauges was about 1,600lb. per sq. in., so there were no criticals at all in the engine beyond Lloyd's Register of Shipping's limit. The engine went through the shop without any alterations and manufacturing difficulties were few and far between. On the test bed the only trouble was one hot crosshead. This was due to a bad surface finish, and once it was rectified everything went well. Incidentally, a good surface finish was essential, and a small portable instrument which would measure surface finish would be a great advantage. Instruments of the Taylor Hobson type were too large and expensive for shop use. Good surface finish was fundamental with a two-stroke Diesel engine where the crossheads were so sensitive.

MR. G. M. SELLAR (Member) said he would like briefly to refer to the section of the paper dealing with the burning of heavy oil.

At Lloyd's Register of Shipping one obtained a broad view of the effect of burning boiler oil in motor ships but the picture was at present far from clear. In some ships complete success had been experienced. In others maintenance had been increased to various degrees. In a few cases failure had occurred.

Some of the reasons for abandoning the use of boiler oil had been increased liner wear, piston ring wear and breakage, the sticking of valves, contamination of the crankcase oil, and pitting of the crankshaft journals and pins. It was not known whether these troubles resulted from lack of knowledge on the part of the personnel, or from the type of oil available on the particular trade routes, or from the type of engine, but it was evident that the economies obtainable from the use of boiler oil were not easy to achieve and called for much effort and resourcefulness from those responsible for operation and maintenance.

It would add to the value of the paper if the author would state in more detail the results of his firm's experiments in the burning of boiler oil and would give some indication of the alterations to injection pressure, temperature, timing and valve nozzles required with different grades of boiler oil. Guidance of this nature was very much needed by the tramp motor ship owner.

On page 146 the author referred to the potential drawbacks to be encountered with the use of the oscillating exhaust valve situated at the entrance to the exhaust manifold. He would suggest that they were very real drawbacks. In the past similar deflector valves were fitted in four-stroke engines—to enable one poppet valve or group of valves to be used alternately for inlet and exhaust—and proved troublesome in service.

* Calderwood, J. 1947. "Recent Developments in the Design of the Sulzer Marine Engine". Trans. Liverpool Engineering Society, Vol. LXVIII, p. 121.

Running clearances became filled with deposits and with an uncooled valve in a water cooled casing any sudden drop in temperature of the cooling water tended to produce sticking.

The author referred to crankcase explosions and suggested that for a crosshead engine hot pistons were the main cause of explosions. This was not typical, and he would ask if the crankcase explosions in crosshead engines of Sulzer type had been generally associated with hot pistons.

MR. J. F. R. ELLISON (Member) said that Mr. Kilchenmann had given a masterly description of the events leading to the development of the latest series of RS type engines and of the design features of the engine itself.

The paper contained many useful facts which had been thoroughly proved by prolonged tests at Winterthur and there was, therefore, little to dispute. Nevertheless, there were one or two points which he would like to raise.

In Table II, giving the results of the centrifuging tests with De Laval type VIB 1929-C machines, the rates of throughput were very low indeed. In his own experience, there was no increase in efficiency below 1·25 tons per hour and up to 2 tons per hour was quite permissible. Taking a maximum rate of approximately 1 ton per hour, as given in Table II, with 1,400 seconds oil, six machines would be required for a 12,000 h.p. installation, assuming a consumption of 0·39lb. per s.h.p.-hr., as against four machines, according to Fig. 7.

In five of the vessels with which he was associated, two-stroke Diesel machinery of 14,200 s.h.p. was operating successfully with four machines of the type quoted.

Turning to liner wear, the author's experience agreed with that of most two-stroke engine users in this country. The rate was approximately double that when using Diesel oil. With high speed engines for geared installations, this became about 2·25 times as rapid, compared with direct propulsion engines. And although the liners were smaller and cheaper, there was definitely a case for further experiment in an effort to reduce it. In this connexion, he would like the author's opinion on the chromium plating of liners or piston rings.

He noticed that there was no illustration of the cylinder tops showing the camshaft housing, fuel pumps and governors. A great deal of ingenuity had been displayed in the design of the governor and fuel pump control gear, but it could with advantage be simplified. For example, experience in service would show whether power-operated spill valves on the fuel pumps were really necessary.

In conclusion, his company had now had experience extending over three voyages with the SG58 type engine operating on 1,500 seconds boiler oil. Although this type of engine had not been designed specially for using this quality of fuel, the combustion at all times had been excellent.

They had now taken delivery of the first RS58 engine installation in m.v. *Middlesex,* and the performance on both shop and sea trials had left nothing to be desired, both with regard to combustion and manœuvring. Their confidence in this new engine was such that two 12-cylinder engines of the same type had already been shop-tested and were now in the fitting-out stage.

They looked forward with interest to the experiments which were to be carried out with regard to supercharging.

MR. T. W. BUNYAN, B.Sc. (Member) said that he would like to raise one point in connexion with the design of the new engine. He noticed that in order to achieve a considerable lowering in the overall height of the engine, the connecting rod had been shortened very considerably. It was also noted that a forked rod was now employed with two separate crosshead bearings. The combination of these two factors would tend to make the crosshead bearings much more sensitive to crankshaft errors and deflexion.

It appeared from Mr. Grange's remarks that a very high quality surface finish was required with the crosshead pins. It was to be hoped that the abandonment of the previous design with the single central crosshead bearing had not introduced new difficulties. It was early days yet to draw any such conclusions. It would be of value if the author would indicate the order of surface finish which his company specified for this somewhat critical part of the engine.

MR. J. CALDERWOOD, M.Sc. (Vice-President) said he hoped the author would forgive him if his remarks were somewhat critical. He agreed with most of what was said in the paper, and he intended to raise only points of criticism.

He would like first to ask whether there was a printing or typing error in Table I. In the columns relating to 6,000, 5,000 and 6,200 seconds oil there was more sediment in the oil after centrifuging than before. He presumed the two figures had been reversed.

There was one point on which he could not agree with the author. He said that the short piston reduced the need for careful alignment. That might be true up to a point, but it was a dangerous statement to make. There was already enough difficulty in getting people who overhauled engines (he was not talking about superintendent engineers but about the men on the job in the ship repair yard) to fully understand the importance of alignment. One did not want to encourage them to think any engine did not need to be properly aligned.

He supported Mr. Arnold's point about annealing. He felt worried about any large welded machinery structure not being subjected to annealing before it went into a ship. It might be all right, or it might not be.

On page 137 the author dismissed a little lightly what he called "complicated schemes such as multiple Diesel-electric drives or even free-piston gas generator installations". All those who were connected with Diesel engines had to pay very serious attention to the future of gas generator installations. They had not met with great success yet, but there was no doubt that if some day they were successful they would become most serious competitors to the large Diesel engine. They allowed of smaller reciprocating parts for the same h.p. output.

Fig. 1 contained a line, as did other figures, showing maximum overload, as it was called. He thought what was meant was the maximum recommended in service. But many people would take it to be the maximum load that could be carried, and the author was, therefore, being unfair to himself in stating the rating in those terms.

The paper did not make clear whether the general design of the 760 mm. 1,550 engine followed the same lines as the 580 mm. This should be explained for the sake of clarity, though it was true the diagrams referred to it.

In referring to the short-stroke engine, the author said the stroke had been reduced as far as possible. He would have thought that as this engine was designed primarily for geared drives, where every revolution was helpful, it would be an advantage still further to reduce the stroke. He imagined it could be cut down by another 80 to 100 mm. He would like to hear the author's reasons for saying it had been reduced as far as possible.

Fig. 9 was most illuminating. It showed the amount of the various ashes that was removed by the centrifuge and the amount that was left in the oil. It suggested that there was still much room for improvement in methods of cleaning the oil. The two components most likely to be serious from a Diesel engine point of view, namely iron and vanadium oxides or salts, were not removed from the oil by centrifuging. He did not know whether the analysis was chemically correct or whether the iron and vanadium were simply stated as equivalent to Fe_2O_3 and V_2O_5. Vanadium pentoxide was a bad thing to have in fuel oil, because it was a good catalyst for the formation of sulphur trioxides. Iron in the form of a mixture of Fe_2O_3 and Fe_3O_4 could also react with SO_2 and water to form sulphuric acid. From that point of view alone, this was one of the most valuable diagrams he had seen published on the cleaning of fuel oil.

On page 144 the author referred to cooling. He said that the cooling of the standard nozzle design was insufficient with fuels of 3,000 seconds and over, and that a special nozzle design incorporating more intensive cooling was developed. He would like to know whether the need for more intensive cooling was related to viscosity or some other characteristic of the fuel. It was his own impression that it was not viscosity but some other characteristics which made cooling necessary. In the past some quite light fuels would not burn satisfactorily without very intensively cooled nozzles.

On page 145 the author gave the reasons for dividing the cylinder and crankcase. While he was in complete agreement as to the advantages of dividing by a stuffing box, he did not think crankcase explosions would be reduced. Analysis of serious explosions that had occurred suggested that a higher proportion, comparing the number of engines in service, had occurred where engines were divided than where they were not. Furthermore, Mr. Arnold had suggested that these explosions might have had causes other than a hot piston.

Finally, he was afraid he could not agree that 235 hours' testing was long enough to settle many things. It was a very short period on which to base conclusive results of the performance of an engine, particularly a large marine engine. Even the smallest engines were usually run several thousand hours before being put into production.

REAR-ADMIRAL(E) W. G. COWLAND, C.B.(ret.) (Member) associated himself with Mr. Arnold's opening remarks: this was a most interesting paper, giving very full information about the two new Sulzer engines.

He regretted that there was very little reference to turbo-charging, however. In the last year or so, various engine manufacturers had shown most startling results from their engines. It looked to him as though this turbo-charging of marine two-stroke engines was practically the biggest advance of the century. The economics alone were remarkable. The additional power from a turbo-charged engine cost, to give the very broadest possible figures, something like £6 per h.p., together with an improved fuel consumption and a reduction in weight and volume of the engine. It seemed to him that concentration on proving in practice the results obtained and published by these firms was of prime importance. He was sorry the author had not said anything about what his company were doing—if they were doing anything—in that line.

He presumed the illustrations were more or less small-scale photographs, but the volume of the scavenge pump appeared to be smaller than that of the cylinder. Could the author say what was the scavenge pump ratio?

MR. H. N. PEMBERTON (Member of Council) said that several speakers had implied that the author had shown courage in the fabrication of this welded engine. Quite frankly, he would like to refer to the author's timidity in the design of this welded structure.

It was suggested that he had been courageous to put a welded engine structure into service without stress-relieving. There was a considerable amount of experience of fabricated engine structures in service without stress-relieving. Indeed, one should be quite clear as to why one would or would not stress-relieve a fabricated engine structure. It was for one primary reason, namely, to rid the structure of residual stress which might otherwise lead to distortion when the structure was machined.

There was less reason for stress-relieving if by suitable welding sequence and either by providing adequate machining allowances or by jig welding the effects of distortion during machining could be counteracted; or, on the other hand, if distortion did not matter.

The author himself, in his concluding remarks, said that this particular engine had had so many hours test-bed running and when it was examined no cracks were found in the welded structure. He (Mr. Pemberton) would have been surprised had any cracks been induced in this engine due to the running tests.

Cracks could be attributed to the formation of brittle constituents during welding, and the inhibition of yield which prevented the redistribution of residual stresses in a component when external loads were applied. In a welded engine structure it was more important to avoid yield inhibitory conditions such as stress concentrations, multi-axial loading, and weld defects, than to stress-relieve the structural components. In the absence of yield inhibitors residual weld stresses were of little significance.

On page 147 the author said that the A-frames and bed-plates were under compression and even with the action of the ignition forces they were not subjected to tensile stress. For this reason the author claimed it was not necessary to stress-relieve the welded components. In Mr. Pemberton's opinion the author had adopted the right policy for the wrong reason.

The fact that there were tensile stresses in a component was no reason for stress-relieving it, and the fact that this part of the engine had been put into compression should not be the reason for omitting stress-relieving.

He would like to know why the author had taken the trouble to fit long steel through bolts to take the firing loads. He suggested they were unnecessary and were not fitted in other types of fabricated engines which had given many years' satisfactory service. Indeed, one of the great advantages of a steel structure as compared with a cast iron structure was that it could stand tensile loads.

Correspondence

MR. A. M. BENNETT (Member) considered that the new Sulzer design described by the author, and the account of the preliminary tests, showed the commonsense approach and thorough attention to essentials which were characteristic of the author's firm.

Detailed results of centrifuging tests were of the widest interest and the promised publication of those obtained by the author would be a useful contribution to the information available. He was pleased to see that in Fig. 10 the author had given a categorical opinion on the maximum admissible viscosity in the injection system, a subject on which other authorities had been somewhat reticent. The indications in this figure, and others, agreed broadly with the results of experience of heavy oils in his own company.

It had been their practice to take blended oil, except where a straight residual oil of less than 1,500 seconds viscosity had been available. Provided that the grades blended were compatible and neither stratified nor threw down asphaltic deposits, it would appear from Fig. 8 that a relatively small proportion of distillate would bring a very viscous fuel into a suitable condition for storage and treatment. This reduced viscosity might be misleading, however, if applied to Fig. 7. The bulk of the oil in such a case might be low grade residuum which would require to be centrifuged very slowly. He understood that the author had burned all the grades of oil listed in Table I in the test engine with complete success, and it would be interesting to know if he had formed any opinion on the limiting specification beyond which oil was unsuitable for burning in a two-cycle engine.

The "Number of Centrifuges recommended for a given output" shown in Fig. 7, agreed with practical experience in his company's fleet under what were the worst conditions likely to be experienced. On this subject, he was rather puzzled by what appeared to be an anomaly in Table II, where the 3,000 seconds test group gave up less impurities in the centrifuges at throughput rates of 0·348 tons per hour than it did at 0·674 tons per hour. Did this in fact indicate that no useful purpose was served by a throughput rate of less than 0·6 tons per hour? In his company, the throughput rate had not been reduced below 1 ton per hour so far. They had not experienced anything so startling as the change in pour point shown for Test Group I in Table I and found this phenomenon hard to understand. The centrifuged oil seemed to be more volatile than the original, as shown by the distillation tests, and there was nothing in the quantity or type of deposits listed in Table II for this oil to explain the very large increase in viscosity. Could the author suggest any reason for the change?

The cylinder liner wear experienced would not appear excessive in view of the different oils tried in the engine. The difference between wear on the diameter parallel to the engine axis, and wear on the diameter perpendicular to it, shown in Fig. 12, did seem to be larger than was found in normal practice.

On the subject of liner wear in general, the statement that a large part of the wear was due to corrosion of the liner metal beneath a brownish layer of varnish was very interesting. In one ship burning heavy oil, he remembered the appearance of varnishing being very marked on the first voyage, particularly on piston skirts. This caused some disquiet, although no pitting was observed. On the next voyage, the varnish disappeared, and it had not come back. Would the author kindly expand his remarks on the effect of the varnish in liner wear?

MR. J. E. CHURCH (Member of Council) had read the paper with much interest and considered that the designs put forward were very good indeed.

His interest was in the RS58 medium speed short stroke engine specially designed for use in geared installations and those now put forward would seem to be ideal for the purpose and a very great step in the right direction.

His experience of Sulzer Diesel engines went back some twenty years and when the *Prince Badouin* type engines for cross-Channel boats were first developed and went to sea it seemed to him that there was the basis for the ideal engine for geared installations of medium and higher powers. He was, therefore, ready to accept that Sulzer's very great experience gained with these and other engines which had gone into the new designs would result in a sound job of a type which many had been waiting for.

Modification of the design so that the short piston skirt did not enter the crankcase was very important but it was felt that for the average cargo liner where headroom was of little or no consequence, matters could be simplified by lengthening the skirt just a little to enable it to keep the ports covered at top centre in order that the complication of the rotary exhaust valve could be avoided and, as the stroke was relatively a short one, this would not increase the engine height very much and would not interfere with the present good arrangement of diaphragm plate and piston rod stuffing box.

If supercharging were required, he thought it should be high enough to make it worth while and the most satisfactory way to do this would surely be by means of a poppet exhaust valve in each cylinder head, giving at the same time the advantage of uniflow scavenging.

However, this was not meant to be a serious criticism because, provided the semi-rotary exhaust valve described in the paper, proved as satisfactory in service as the designers expected, the engine was a very attractive one.

Although the paper only described the actual prime movers, the shipowner was interested in a complete set of engines; that was to say, the twin set of Diesel engines, together with the elastic couplings and gearing ready for attachment to the propeller shaft, and he not unnaturally felt that the whole installation should be the concern of the engine builder, even though he might sub-contract the gearing, etc., elsewhere, thereby ensuring that all these were designed as a unit and the satisfactory operation of the whole made all the more certain. Whilst this might be somewhat beyond the scope of the present paper, there were some points he would like cleared up since his interest in the engines was only maintained if the whole geared unit could be satisfactorily operated.

Firstly, he was more in favour of hydraulic couplings than any other type, since these were purely a mechanical device, and would enquire whether, in the event of complete immobilization of one engine at sea, the remaining engine could continue indefinitely without any ill-effects due to windage and possibly overheating of the disconnected coupling running in that condition for long periods. This was considered important because one of the major attractions of the multiple geared installation would be the ability to disconnect and stop one engine at sea for adjustment or repairs. On this subject it seemed that the question of safely interlocking the three turning gears with the coupling and starting controls would be necessary and this would be complicated by the fact that, whilst all was well, neither engine should be capable of being started with any one of the three turning gears "in", yet there would be times when this interlocking would have to be such that, with any engine stopped for repairs whilst the other was running, it would have to ensure that under no circumstances could the coupling of the engine under repair be filled, before it would be safe to put a man into the crankcase at sea to attend to bearings, etc.

So far as the gearing itself was concerned, there seemed to be a tendency to dismiss this as something which would never give trouble, whereas if the medium speed engine was to be adopted, no matter how good the Diesels might be, the unit as a whole would be a failure if gearing troubles were to arise. Could the author say how long geared installations of the powers covered by his proposals had been in satisfactory operation at sea and whether a set of gears could be expected to operate without serious renewals during the normal life of the ship and, if not, how often recutting or renewals would be necessary?

It was understood that generally speaking a separate lubricating oil system was advisable for the gearing independent of the Diesel engine system and this resulted in an added complication due to the duplication of pumps and coolers which the shipowner would like to avoid if possible, since his aim was always towards simplification. Could the author state whether the two independent systems were really necessary and, if the answer to this was in the affirmative, it was suggested that there would be need for special pumps to be developed comprising two independent pumps driven from a common motor and also combined oil coolers with two separate lubricating oil spaces for the common cooling water passage.

Personally, he would prefer to see a single lubricating oil system for the whole unit but with very much increased centrifuge purifying capacity so that, rather than allow the gearing to run on dirty oil, the engines would be supplied with a much cleaner lubricant which would also be capable of taking care of the gearing without trouble.

If, as seemed probable, these points could be satisfactorily dealt with, it seemed that the geared installation using these new specially designed engines would prove a very attractive proposition and could offer to the shipowner tremendous advantages which he would be very ready to adopt and the progress of those now building would be watched with very great interest.

MR. N. E. THOMPSON (Member) thanked the author for his very interesting paper, and as one who came through the hard school of the early Diesel engine, it was interesting to note the many improvements which had been effected, though a number were somewhat belated.

The abolition of the jet pipe for piston cooling in this particular type of engine had taken more than twenty years to accomplish.

The rate of liner wear given when using boiler fuel oil was alarming. Of all that had been written and said about the use of this type of fuel in Diesel engines, in particular the types where the piston skirt was common to the crankcase, in his opinion, boiled down to the fact that this type of fuel should only be used if one were compelled by force of circumstances to do so.

He would welcome the author's opinion as to whether boiler fuel oil, such as Bunker "C", could be used with advantage on a Diesel driven vessel for all purposes, i.e. on the auxiliary engines running at from 450-1,000 r.p.m.

Would the author be good enough to furnish figures of lubricating oil consumption of the new engines?

Author's Reply

In answer to Mr. Arnold's remark concerning the maximum cylinder diameter, the author said that he had made no statement concerning the cylinder liners but only that a bore of 760 mm. was in the neighbourhood of the limit imposed by the satisfactory handling of the heat stresses. These latter affected the cylinder cover and piston design rather more than that of the liner. He would like to state that 760 mm. did not represent an absolute maximum. Test engines of much greater cylinder bore had been built by various manufacturers; for example, his firm had built one with a bore of 1,000 mm. as far back as 1910 and another double-acting single cylinder of 900 mm. bore, developing 2,400 h.p., in 1924. Service engines of such large cylinder diameters were also built; in 1926 six five-cylinder engines of 900 mm. bore were manufactured by a British licensee but gave many difficulties in operation. About 1930, nine Sulzer engines of 820 mm. cylinder bore were installed in single engined freighters, which were still in satisfactory service, but experience showed that around the 800 mm. mark there seemed to be a limit where difficulties due to heat stresses, etc., suddenly increased. The author was aware that other manufacturers had built very large bore engines, such as the one mentioned by Mr. Arnold, but it was interesting to note that these now seem to have disappeared from the market. The largest size manufactured today by the various engine builders was around the 760 mm. diameter chosen for the large RS type engine, e.g. Doxford and Fiat 750 mm., M.A.N. 780 mm., B. and W. 740 mm., etc.

Centrifuging under pressure at higher temperatures had been under consideration by his firm for several years. The problem had been discussed with various centrifuge manufacturers and a solution seemed to be quite possible. In the author's opinion the danger of centrifuging at increased temperature should not be overrated. As long as the oil was maintained at a suitable pressure, no ignitable vapour could be generated by exceeding the flash point. Certain safety precautions must nevertheless be observed when opening the centrifuge for cleaning. In order to reduce these danger periods it would be as well to have centrifuges of the self-cleaning type. During operation, properly designed pressure centrifuges should at least be less dangerous than the type in general use today. In the latter an ignitable air-fuel vapour mixture might possibly be generated in the space surrounding the separator bowl, as soon as the flash point of the oil was lower than the operating temperature (generally about 185 deg. F.). According to British Standard 742:1947, Fuel Oils for Burners, this flash point might be as low as 150 deg. F.

The author thought that it had yet to be proved whether preference would be given to the self-cleaning or ordinary type of centrifuge.

Fig. 8 was not intended to be a recommendation in favour of lowering the heavy fuel viscosity by mixing the oils on board. The ship's engineer had no control over the characteristics of the heavy residual fuel and diluting stock and might, therefore, run into trouble with sludge formation due to incompatibility of the fuels. Blending was apparently carried out in some refineries, as indicated by the low flash point, in order to make the product more manageable and of a greater market value. The cost of mixing did not seem to affect the price very much, this being reflected by the small price differential between 1,500 and 6,000 seconds fuel.

With regard to use in the engine, 6,000 seconds fuel might be utilized just as easily as 1,500 seconds fuel, the viscosity alone not being a criterion of the oil quality. On the other hand, for a given size of fuel cleaning installation the former oil was not so well cleaned as the latter. For this reason, therefore, it might be advantageous to limit the viscosity to about 1,500 seconds Redwood I at 100 deg. F.

Manœuvring would normally be carried out on heavy fuel and the installation had been designed to allow for this. The question of whether or not one should switch over to Diesel oil was one of economics and was a matter which only the shipowner could decide.

Experience had shown that a burning jet of heavy fuel penetrated further into the combustion chamber than one of light fuel. This was probably due to the slower evaporation of the individual droplets and—in spite of preheating—poorer atomization due to a higher viscosity. By reducing the injector nozzle hole diameter, the penetration of the jet was decreased and atomization improved. The latter was further improved by the higher pressures generated in the fuel system.

The idea of a double controlled fuel pump was certainly not new, but had been abandoned by most manufacturers. This more complicated type of pump was reintroduced because it was felt that in view of the still poorer residual fuels to be expected in the future the injection control could never be close enough.

The suction and spill valves were each controlled by an eccentric shaft actuated from the governor. By rotating the shafts in opposite directions the suction valve was closed earlier (or later), according to the governor movement, and the spill valve opened later (or earlier), thus increasing (or decreasing) the effective stroke of the plunger and thereby the quantity of fuel injected. A hand adjustment, which could be carried out whilst the engine was running, allowed both eccentrics to be moved in the same direction, thus altering the injection timing without affecting the quantity of fuel injected.

On page 146 of the paper the elastic feature of the exhaust valve drive had been described. This drive eliminated any danger of a damaged valve assembly, due to jamming of a broken piston ring, without the necessity of installing the baffle described by Mr. Arnold. The elastic drive had the further advantage that, should any of the moving parts in the assembly stick or seize under abnormal circumstances, then no harm would occur. The rotary valve shown in Mr. Arnold's drawing had a large area in close proximity to the inside surface of the valve casing. The author felt that deposits on the casing would tend to jam the valve. He had provided for this by making narrow edges on the valve which served the primary purpose of sealing and at the same time removed any deposits that built up on the inside of the casing.

The sealing of the exhaust valves against leakage of exhaust gases into the engine room was in fact provided by a labyrinth sealed with air bled from the scavenge manifold in a way similar to the one used by Mr. Arnold.

Much could be said for and against stress-relieving welded components. It was certainly an advantage with a view to avoiding distortion and relieving additional stresses, but it increased the manufacturing costs considerably and required large furnaces; these were available at Winterthur, but it was possible that potential licencees might not possess them.

Experience on Sulzer engines with the screw jack arrangement for securing the bearing cap had been very satisfactory even in double-acting engines with large forces acting on the jack bolts, showing that apparently no additional elasticity was necessary with a correct design. On his firm's four-stroke locomotive engines, bearing caps had even been secured to the frame very satisfactorily with a tapered keying arrangement.

The cooling oil entering the piston had a temperature of about 35 to 40 deg. C. (approximately 100 deg. F.). Thus the rod, even with the inlet oil carried in the outside channel, would certainly be above the dew point of the pure scavenging air contained in the space above the stuffing box.

Usually no heavy deposits were found on the type of ports used in the new engine, as shown in Fig. 34 for the RS58 and Figs. 16 and 17 for the experimental 720 mm. engine after a much longer running period. But the author had encountered cases of severe fouling in service, due probably to an unsuitable combination of fuel and lubricating oil and possibly poor maintenance. The accessibility provided on the new engine was intended to deal with such exceptional cases, which might be expected more often in the future in view of the probable decrease in the quality of residual fuels.

The ratio of the displacement volume of the double-acting scavenge pump to the displacement volume of the main cylinder was 1·55 : 1 on the new engine compared with 1·42 : 1 on the SD type. The figure 1·3 mentioned on page 148 referred to the actual air volume passing through the cylinder during scavenging, the difference being accounted for by the volumetric efficiency of the pump, losses due to exhaust valve leakage, sealing air, etc. The actual scavenge ratio was thus very much the same as that on the SD type.

The stream valve or "mouth organ" valve, as Mr. Arnold termed it, was a most effective design, as it offered minimum resistance to air flow, sealed completely in the return direction and possessed the lowest possible inertia. It was retained on the scavenge pumps of the new engine; but as there was always a danger of fouling on the scavenge valves in the main cylinder, the author felt that however good an automatic valve might be it was still better to dispense with it altogether.

The weight of the new welded 10RSG58 engine for an output of 500 b.h.p. per cylinder was 92lb. per b.h.p.; for the RSD76, the corresponding figure at 1,000 b.h.p. per cylinder was 136lb. per b.h.p.

The gas operated type of pump was certainly the most straightforward solution of the injection problem, as the energy for pumping had to be derived somehow from the pressure of the cylinder combustion gases. Many Sulzer pumps of this type were in satisfactory operation, mostly on engines converted from air injection. But it was felt—again in view of the varying properties of residual fuels—that a still closer control of injection than the one possible with the gas type pump was necessary.

The manner in which the chain for driving the camshaft was tightened without changing the angular relationship between cam and crankshaft could be seen in Figs. 37 and 38.

In answer to Mr. Grange's query, the author did not understand why the bypass in the day tank shown in Fig. 11 should not correspond with the Ministry of Transport specifications, but he would investigate the legal side of this question further.

Mr. Grange mentioned the baffles shown in Fig. 11; these were in reality a pipe of about 10 inches diameter which communicated at the top and the bottom with the main volume of the tank, and into which entered at the top the return line and at the bottom the suction line. Its function was to separate the air from the hot returning fuel and to lead the latter to the suction line without any mixing with the bulk of the fuel by preventing it from spreading over the top surface of the tank, the pipe opening being above the level of the fuel. Air and fuel vapour left from the upper venting holes into the main volume of the tank, where the vapour condensed on coming into contact with the cooler fuel. Owing to the lower specific gravity of the hot returning fuel and the fact that the entrance of the suction line was at a higher level than the lower openings of the central pipe, the hot fuel could not leave the pipe. On the other hand the bulk of the fuel in the tank was maintained at a temperature of some 100 deg. F. by heat transmission through the pipe wall. This temperature resulted from an equilibrium of heat losses from the tank to the ambient air on one side, and the heat conducted through the wall of the central pipe on the other side. It was practically independent of the fuel level in the tank, thus the danger of overheating was eliminated. The author was convinced that the installation described, which had been patented, being simple and foolproof, gave the maximum possible safety.

Liner wear fore and aft was often higher than athwartships. One reason for that might be the arrangement of the

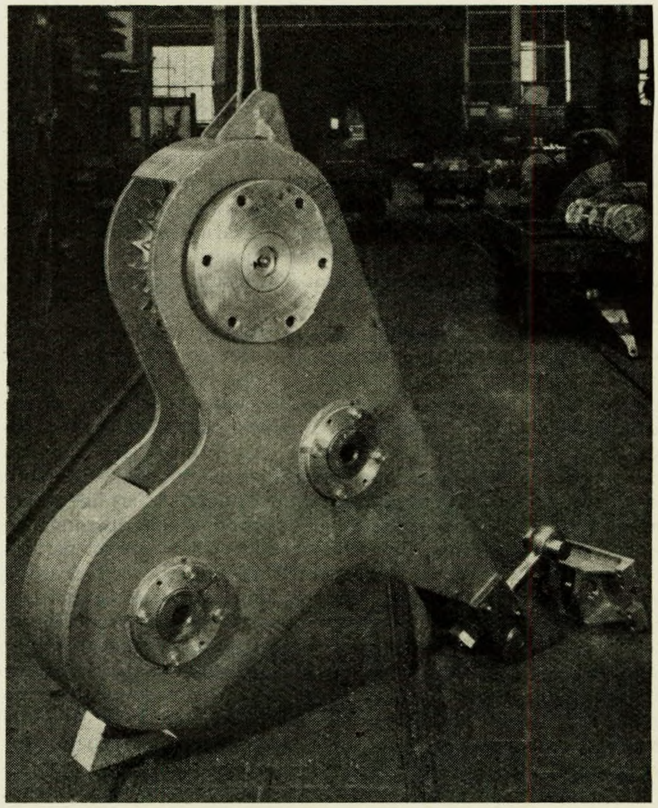

FIG. 37

cylinder lubrication feed points. The holes for the lubricating studs were usually drilled perpendicular to the engine axis, resulting in a somewhat greater arc between the points on the fore and aft side of the liner. This would be modified on the new RS76 engine, where the holes were drilled radially, with an equal spacing of the lubricating points.

The surface finish specified for the crosshead pin was checked in the shops by comparison with a set of standards and a small home-made adaptation of a Johannson indicator. This was supplemented by a Brush "Surface Analyser" in the laboratory and a portable American "Profilometer".

The picture drawn by Mr. Sellar of the results of operation on heavy fuel corresponded more or less with the author's views, and also to the consensus of opinion expressed at the International Combustion Engine Congress held in Milan during April. The papers and discussions held at this Congress contained, incidentally, much valuable information concerning operation with heavy fuels in Diesel engines.

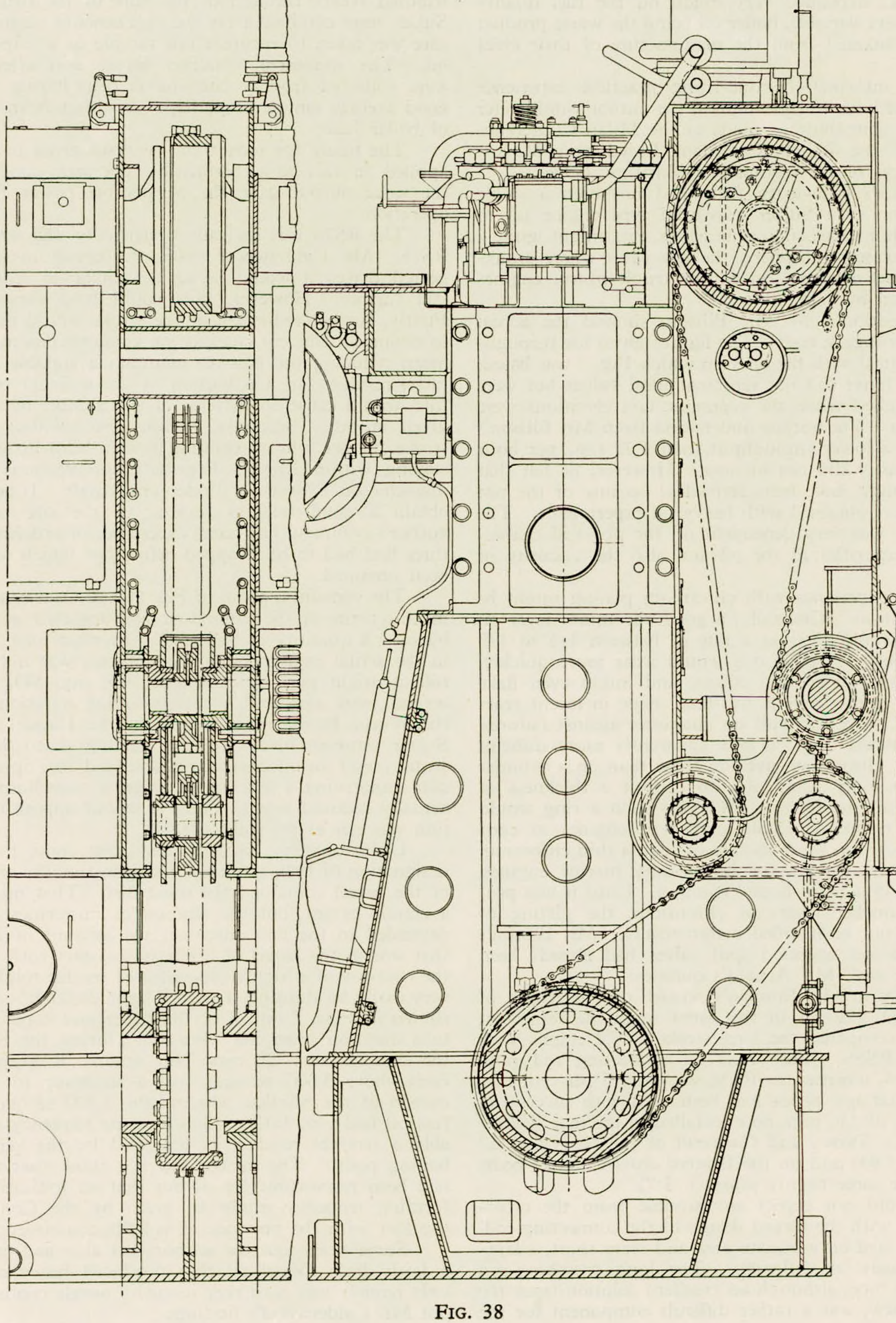

FIG. 38

There was certainly a great divergency of results in practical service; whilst Sulzer Brothers had cases where wear and trouble had hardly been increased at all by changing over to heavy fuel, in other cases the wear had increased considerably. The factor of 2 given on page 143 in the paper might be considered as a fair overall average. It was certain that operation on heavy fuel without adequate installation and proper maintenance only invited trouble. On the other hand burning of heavy fuel in an efficiently laid out and properly maintained plant was no problem; but even so the results in respect of fouling, wear, etc., depended very much on the fuel quality used, which was very variable, boiler oil being the waste product of the refineries obtained from the manufacture of their chief product, petrol.

For further information concerning practical experience obtained in service with Sulzer engines, the author would refer Mr. Sellar to the contributions made at the Milan Conference by Mr. Ruys of the Royal Rotterdam Lloyd and Messrs. Sozonoff and Pluys of the Compagnie Maritime Belge.

To the author's knowledge there had not been a single crankcase explosion in a Sulzer crosshead type engine up to date. While any hot parts, such as bearings, etc., might ignite a combustible crankcase atmosphere, the few cases where crankcase explosions had occurred on Sulzer trunk piston engines had all been caused by a hot piston.

Table II, mentioned by Mr. Ellison, showed the actual results of the centrifuging tests. The figures given for throughput were not identical with the ones on which Fig. 7 was based. Furthermore, the latter did not represent ideal values but were a compromise reached when the economic considerations were taken into account. The author understood from Mr. Ellison's remark that with a lower throughput than 1·25 tons per hour the cleaning efficiency did not increase. However, he felt that this conclusion might have been arrived at because of the test result scatter always obtained with heavy oil experiments. The cleaning efficiency was very dependent on the physical characteristics of the impurities in the oil and also the viscosity of the latter.

The author's experience with chromium plating might be summed up as follows. Generally, a good chromium layer on the liner surface would wear at a rate of between 1/5 to 1/3 of that of cast iron. A poor one would wear more quickly, especially as a result of etching effects, and might even flake off. Whilst great improvements had been made in recent years with liner plating, there was still no guarantee against failures. With regard to plated rings, it was apparently more difficult to deposit a heavy chromium layer on these than on a cylinder liner, most platers only being able to deposit a thickness of 0·004-0·006 inch on the former. Whereas such a ring would show very good results in small lorry type engines in conjunction with a cast iron liner, the life of such a thin chromium layer was wholly inadequate for heavy duty marine engines. It would wear off in a 1,000-hours' running. Until it was possible to deposit thicker layers of chromium, the plating of large rings could not be justified commercially. Mr. Ellison's comment about power operated spill valves had already been answered together with Mr. Arnold's question.

With reference to Mr. Bunyan's remark on the length of the connecting rod, Fig. 21 in the paper might be somewhat misleading, as it compared the long stroke SD72 engine with the short stroke RS58. The crank to connecting rod ratio of the SD was 1:4, whereas the RS58 was only 5 per cent less, i.e. 1:3·82. Similar low ratios had been used with success on the S-type engines of 580 mm. bore installed in the m.s. *Willem Ruys* and the m.ss. *Surrey* and *Cornwall* of the New Zealand Shipping Co. (1:3·93) and on the Belgian cross-Channel boats now in service for some twenty years (1:3·7).

The author did not expect any trouble from the crosshead-pin bearings with the forked design of the connecting-rod. The two bearings had an adequate area and were short, adapting themselves easily to deflexion. The long crosshead-pin bearing of the SD type, although an excellent solution from the design point of view, was a rather difficult component for the shop and it had to be scraped in a special manner to relieve the centre part where it was difficult to carry the heat away. The surface finish specified for the crosshead pins was a good ground finish of about 8 micro-inches.

The figures mentioned by Mr. Calderwood in Table II, showing discrepancies in sludge content, were not printing errors but the figures actually obtained by analysis, and were a demonstration of the scatter in results often obtained with such analyses of heavy fuel. It was a fact that owing to the high viscosity of the boiler oil the impurities were not distributed evenly throughout the bulk of the fuel. During the Sulzer tests conducted on the experimental engine, the utmost care was taken to ensure a fair sample of a shipment of heavy oil. The specimens obtained before and after centrifuging were collected from a continuous drip bypass which gave a good average sample of perhaps one quart from about one ton of boiler fuel.

The figure for maximum overload given in Figs. 1 and 4 applied in fact to a permissible overload condition, whereas the rated output gave the permissible power for continuous operation.

The RS76 was basically designed on the same lines as the RS58. Mr. Calderwood was quite correct in saying that for geared drives it would be an advantage to reduce the stroke still further. However, this would have various drawbacks. Firstly, further reduction of the stroke would make it difficult to obtain an efficient combustion chamber design. This would mean reducing the bore to maintain a suitable bore to stroke ratio, resulting in a reduction of the cylinder output; this in turn would cause an increase of the number of cylinders for a given output. Secondly, it was not possible to reduce the stroke further whilst retaining a semi-built-up crankshaft and staying within Lloyd's Register's specifications for crankpin diameter on a twelve cylinder crankshaft. It was difficult to obtain a good integral forging of the size required. The author's company had some experience of ordering such shafts; three had had to be scrapped before one which was correct had been obtained.

The vanadium given in Fig. 9 as V_2O_5 was actually present in this form in the sample of ash analysed and obtained by burning a quantity of boiler oil. In what form it was present in the actual engine combustion process was not known. The rôle it might play by catalysing SO_2 into SO_3 was discussed several years ago, and a corresponding question presented, at the World Petroleum Congress at the Hague in 1951, by a Sulzer representative. In the ensuing discussion the representatives of an oil company discredited this opinion by giving data concerning a test engine where a vanadium addition had actually reduced wear. Thus, it would appear that this question was not as yet fully settled.

Unfortunately, viscosity had now come to be a quality designation of fuel, probably based on the idea that the bottom of the barrel contained the most dirt. That might be true to a certain extent, but the amount of impurities in the residue depended in the first place on the amount of foreign matter that was in the barrel of crude oil to start with and, secondly, the viscosity of a highly concentrated residue might be decreased very easily by diluting it with a light distillate. This had been shown in Figs. 7 and 8 of the paper and explained in answer to a question from Mr. Arnold. During the Sulzer tests on the experimental 720 mm. bore engine, it happened that the fuels above 3,000 seconds had a tendency to produce more carbon at the nozzles, whereas the 1,500 seconds Redwood I fuel oil had very little carbon-forming tendency and was probably a straight residue, as indicated by the very high initial boiling point. The author did not think that viscosity alone had been responsible for it, but that an indication of carbon-forming tendency might be given by the Conradson figure, together with the presence of volatile constituents.

Some years ago the author had also had experience with a fairly light Diesel oil that produced heavy carboning; the only remedy had been very intensive nozzle cooling. This bore out Mr. Calderwood's findings.

The main feature of the nozzle shown in Fig. 15 was not only intensive cooling of the nozzle body but the fact that the cooling medium was brought down close to the place which required cooling, that is, the nozzle tip. In this way a close control of the tip temperature was obtained and it was no longer necessary to use very cold water.

A period of 235 hours did not mean much compared to the many thousands of hours the engine would have to run afterwards, but on the other hand it was much more than the normal run of a few hours' duration on the test bed. The three million load reversals represented a reasonable amount of endurance testing; furthermore, several of the principal innovations in the RS design, such as the double controlled fuel pump with a long chain drive and the scavenge arrangement with the exhaust valve and its elastic linkage, had undergone a very thorough preliminary test of more than 2,000 hours' duration on the 720 mm. bore experimental engine.

The author was sorry not to be able to give Admiral Cowland more information about supercharging. Some data concerning work in this direction was published in 1941 in a special number of the Sulzer Technical Review. Additional work was now going on, but it was too early to publish details on it. The question concerning the scavenge pump ratio had already been answered for Mr. Arnold.

Mr. Pemberton's remarks had been noted with interest, as had his recent paper "Welding in Marine Engineering". The author was aware that many large welded structures were manufactured in Britain without being stress-relieved. Those for opposed piston engines operated under the same conditions as in the new Sulzer design, i.e. they did not carry any tensile stresses. In the former case the combustion forces were absorbed by the moving side rods, whereas in the latter they were led through the stationary tie rods.

Referring to Mr. Bennett's comments concerning the question of diluting boiler oils in order to reduce their viscosity, the author did not recommend such a procedure to be carried out on board ship owing to possible incompatibility of the components. On the other hand, if this mixing had been done already by the refinery, he did not see any reason why additional precautions or specially low centrifuging rates would be necessary for treating such an oil. As he had mentioned before, it was very difficult to judge a fuel oil from the analysis, therefore specifications giving admissible limits were extremely hard to draw up. Sulzer Brothers were about to issue new specifications for heavy fuel oil selection and the figures contained therein were given below. They represented values which he thought it was better not to exceed. In special emergency cases, or when the origin of the residues was known, then the bracketed figures were also acceptable.

Specific gravity	0·98 (0·995)
Viscosity	
3,000 (10,000) seconds Redwood I at 100 deg. F.	
Sulphur, per cent	3 (4)
Conradson, per cent	10 (15)
Ash, per cent	0·06 (0·1)
Flash point ... according to legal requirements	

The anomaly in Table II was rather puzzling, and the author thought that for the lower values the influence of a small throughput on the separating effect was already so slight that it had been overshadowed by the discrepancies in the fuel quality.

The alterations of the pour point, Table I, Test Group I, had at one time also surprised the author. At first it was thought to be a mistake in measurement. However, it was soon apparent that the measurement was correct. In the meantime the same phenomenon had been noted with other heavy oils, although perhaps to a less pronounced extent. The phenomenon had been described more than once in literature on this subject, and might also be a reason for the readings obtained*.

The question of the difference between liner wear on the fore and aft axis and that athwartships had already been answered in conjunction with a point raised by Mr. Grange. In the particular case of the experimental 720 mm. engine, very careful readings, taken with an air gauge from a fixed axis in the cylinder, showed that the wear was concentrated on the flywheel side of the liner and no specific reason for this phenomenon could be found. However, at this particular spot the liner surface was bright, whereas the remainder was covered by a brownish varnish underneath which the metal was attacked by corrosion. It would seem, therefore, that at the places where the varnish was present, it protected the weakened metal structure underneath it from the rubbing effect of the piston rings and resulted in reduced wear. When this varnish was removed, however, the bare metal was rapidly worn away by the combined corrosive and abrasive action.

Mr. Church asked for an interlocking device preventing hydraulic or electric couplings from cutting in while engines were being inspected. This was fitted on all Sulzer installations of this kind, preventing any wrong manœuvre if the turning gear was engaged. Electrical couplings had given no more trouble when disconnected than those of the hydraulic type, and the author felt that there was not much to choose between them.

With geared Diesel installations, experience showed that gear life was not a limiting factor. Some data had been published on this in Sulzer Technical Review No. 1, 1951. As an example, geared drives on Dutch freighters had been in service for twenty years without giving trouble and the geared installation on m.s. *Willem Ruys*, with eight main engines totalling 32,000 h.p., did not show any indication of gear wear after five years.

Mr. Church proposed a common lubricating system for gears and engine. The author was against such simplification. Firstly, two separate systems allowed the most suitable oil to be selected for each purpose, and, secondly, there were cases where, in spite of all precautions, unexplained corrosion had occurred in Diesel engine crankcases. Whilst it was certainly disagreeable to have a slightly pitted crank pin and consequently a run-out bearing, the same amount of corrosion on the teeth of a gear wheel would mean a complete failure.

Mr. Thompson's question concerning the possibility of boiler oil operation for auxiliaries could not be answered from a purely technical standpoint. It was certainly possible to burn fuel oil in auxiliary engines, but usually wear and fouling increased in such a manner that operation became uneconomical due to high maintenance, spare part costs and low availability of the engines on one hand, while on the other hand the saving in the fuel bill was much smaller than on the main engines.

The service lubricating oil consumption of the new RS58 engine was approximately 1 gram per b.h.p. hr. for the cylinders. On the test bed the total lubricating oil consumption from the sump amounted to about 0·25 gram per b.h.p. hr., including the various leakages which were reported in the paper.

In conclusion, the author said that he would like to thank all the contributors for the excellent points that they had raised. He felt that the only way to make practical progress towards better engine design and operation was by open and frank discussion. The presentation of a paper and the ensuing discussion was a great help towards achieving that end.

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* McNab, J. G., Rogers, D. T., Michaels, A. E., Hodges, C. E. 1948. "Pour-point-stability Characteristics of Winter-grade Motor Oils". S.A.E. Quarterly Transactions, Vol. 2, No. 1, p. 34.

INSTITUTE ACTIVITIES

Minutes of Proceedings of the Ordinary Meeting Held at the Institute on Tuesday, 14th April 1953

An Ordinary Meeting was held at the Institute on Tuesday, 14th April 1953, at 5.30 p.m., when a paper by Mr. W. A. Kilchenmann, Dipl.Ing., entitled "New Designs of Large Two-Stroke Marine Diesel Engines" was presented and discussed. Mr. S. Hogg (Chairman of Council) was in the Chair. Seventy-six members and visitors were present and eight speakers took part in the discussion. The vote of thanks to the author proposed by the Chairman was heartily endorsed. The meeting ended at 7.20 p.m.

Summer Golf Meeting 1953

The Summer Golf Meeting took place on the 11th June 1953 at Sundridge Park Golf Club exactly nineteen years to the day since it was held there on the 11th June 1934. Twenty-four members took part, and, in spite of the rain which commenced before lunch and continued during the whole of the afternoon, both competitions were carried through.

Mr. F. P. Bell won the Cup in the morning with a net score of 72; Mr. W. Sampson was second with a net score of 74.

In the afternoon Bogey Greensome Competition, Messrs. W. Donaldson and W. J. D. Middleton came in first with the record score of 1 up, while Messrs. J. White and H. E. Upton, R. K. Craig and R. Ward tied for second place with 4 down. As they also tied on the best net score for the last nine, twelve and fifteen holes, it was decided to cut the cards, and the second prizes went to Messrs. Craig and Ward.

Mr. F. P. Bell

The prizes were presented after tea by Mr. A. Robertson (Vice-Chairman of Council). Mr. Bell received a brief case as winner of the Cup, and Mr. Sampson a flask. Messrs. Donaldson and Middleton received travelling clocks, and Messrs. Craig and Ward cigarette boxes.

Mr. Robertson proposed that a hearty vote of thanks be accorded to the Committee of Sundridge Park Golf Club, the Secretary, and the catering staff, for the arrangements which had been made for the meeting. This was carried unanimously. He also proposed a vote of thanks to the donors of the prizes—Messrs. J. A. Goddard, J. M. Mees, A. Robertson, C.C., P. C. Speechly, W. Tennant, H. E. Upton and R. Ward.

Junior Section

85, The Minories, E.C.3

On Thursday, 9th April 1953, at 7 p.m., the Junior Section held a discussion at the Institute entitled "So You Want to Build a Ship!" The Chairman of the meeting was Mr. F. D. Clark (Convener of the Junior Section Committee) and members of the panel comprised Messrs. J. Calderwood, M.Sc. (Vice-President), F. A. J. Everard (Member), H. C. Gibson (Associate Member), R. Munton, B.Sc. (Member), J. Turnbull, O.B.E. (Vice-President), and Captain J. C. Taylor, C.B.E.

There was a relatively small attendance at the meeting but a spirited discussion followed the reading by the Chairman of a paper* entitled "How New Ships Are Ordered", contributed by Mr. A. C. Hardy, B.Sc. (Associate Member of Council), who was unavoidably absent.

Visit to H.M.T.S Monarch *at Greenwich*

A party of Junior Section members, in the charge of Mr. G. F. Gatward (Associate Member), visited H.M.T.S. *Monarch* at Greenwich on Saturday afternoon, 27th June 1953.

On arrival, the party was welcomed by Fourth Officer Patterson, who conducted the visitors over the bridge, chart room, etc. He also gave a very interesting talk on the procedure for retrieving and repairing broken cables from the sea bed.

Both the radar equipment and Decca navigational aid were operated and the method of plotting readings from the latter demonstrated with the aid of a chart.

The party next proceeded to the engine room, where, under the able guidance of Engineer Makins, the party was shown the generator flat, main boiler room, tunnel, etc. As steam was on the donkey boiler, the opportunity was taken to make a particular study of the oil-firing equipment. Engineer Makins was called upon to answer numerous questions and went to a great deal of trouble to make quite sure that his appreciative audience fully understood the explanations given.

An interesting item, which was open for inspection, was a fractured casting from the turning gear of the starboard engine. This mishap was caused by a barge hitting the propeller while the vessel was loading.

A hearty vote of thanks was expressed to both Fourth Officer Patterson and Engineer Makins for their kindness.

The party numbered thirteen. They arrived aboard the *Monarch* at 2.20 p.m. and left at 5 p.m. having enjoyed a very interesting and instructive afternoon.

* This paper, and the discussion which followed, are recorded in the Supplement to this issue, pages 1-5.

Second visit to H.M.T.S. Monarch

On Saturday, 4th July 1953, at 2 p.m., some twenty members of the Junior Section, in the charge of Mr. H. C. Gibson (Associate Member), visited H.M.T.S. *Monarch,* by permission of her master, Captain Betson.

On their arrival at the ship, which was moored off Greenwich, Mr. Flynn, Fourth Officer, conducted the party to the bridge and chart room, where modern navigational aids, such as the Decca Navigator, Pitometer, Echo sounder and Gyro recorder, were displayed and explained. With a gross tonnage of 8,056, the *Monarch* is the largest cable-laying vessel afloat, and is equipped with many devices to enable her master to locate and repair cables efficiently.

From the bridge, the tour continued with a visit to the main deck to watch the stowing of cable in two of the tank-like holds. These have a total capacity of approximately 3,500 miles of cable, varying according to the size of the cable stowed.

After this, the bow and stern sheaves were inspected and it was interesting to hear that, when laying long runs of cable over the stern, a speed of about 7 knots was possible.

To complete the visit, the party went below to the engine room, where Mr. Jones, chief engineer officer, had arranged for Mr. Waddle, junior engineer officer, to show to the party the main engines (twin triple expansion), the four single ended Scotch boilers and the auxiliary Cochran boiler; the generator flat, with its two turbo and two Diesel generators, was also visited.

The visit ended at 4.30 p.m. when the members of the party were ferried ashore, having had a most interesting and valuable visit.

Local Sections

Sydney

A meeting of the Sydney Local Section was held on Thursday, 30th April 1953, at 8 p.m., at Science House, Gloucester Street, Sydney. Mr. H. A. Garnett (Local Vice-President) was in the Chair and seventy-eight members and guests attended the meeting.

Mr. W. A. Harrington delivered a lecture entitled "Piston Rings, Their Functions and Manufacture"; he dealt with the subject most comprehensively and the paper was well illustrated by lantern slides. Some very interesting points were raised in the discussion which followed and to which Messrs. R. S. Renfrew, E. L. Buls, J. R. Robertson, H. W. Lees and several visitors contributed.

A vote of thanks proposed by Mr. Renfrew and seconded by Mr. A. S. H. Spain was carried with enthusiasm.

South Wales: Third Annual Golf Competition

A field of forty-four golfers competed in the 18-hole medal competition of the South Wales Local Section held at Glamorganshire Golf Club, Penarth, on Friday, 5th June 1953. This entry, consisting of members and their guests, included many well known local shipping personalities and was the highest since the event was instituted in 1951.

Prizes for the best net scores (Members) were awarded to J. D. Buchanan (71) and David Skae (77); the prize for the best net score for a member with a handicap of 18 or over was won by T. Grieve (78). L. Howard Emery (69) and Peter D. Jones (73) received the first and second prizes respectively for the best net scores among the guests.

The sealed holes (second nine) competition for members was won by R. Reid with a net score of 38 and for the guests by W. Henderson and H. Johnson, each of whom scored 36½.

The "Hard Luck" prizes were awarded to J. O'Shea and J. Hickey.

Later, a gathering of fifty-five members and guests were served with supper in the club house. Mr. J. H. Evans, M.B.E., Vice-Chairman of the Section, presided, and proposed the Loyal Toast; then, at his request, the Glamorganshire Golf Club captain, Mr. K. H. McIlroy, presented the prizes. Mr. F. F. Richardson thanked the club for placing the course and club house at the disposal of the meeting for the third successive year and Mr. McIlroy, in responding, expressed his personal pleasure in being invited to join the company of marine engineers and, on behalf of his committee and members, extended a hearty welcome to the Section for future golf meetings. Mr. L. Howard Emery, speaking for the non-members present, expressed appreciation of the hospitality which had been shown to them and thanks for a most enjoyable evening.

Membership Elections

Elected 23rd June 1953

MEMBERS

Harold Noel Wykeham Barnes
Stanley Hewitson Drew
Edward Bernard Dunn
Archibald Weir Gardiner
Edward Joseph Healy
William Ewart Honey, Lieut.(E), R.N.(ret.)
Wilfred Harry Locke, Lieut.(E), D.S.C., R.N.
John Mather
Russell F. Mercer
Martin John Noon
The Hon. Geoffry Lawrence Parsons
Gilbert David Francis Pringle
George Launcelot Meyrick Salter, Capt.(E), R.N.(ret.)
George Milne Shewan
Hugh Morrison Stewart
Iain Maxwell Stewart
Alexander Henry Stirling
Sidney Matthew Thorpe
Walter Tinker
William Morton Valentine
Harold Johnston Whitgreave
Joseph Gardiner Wood

ASSOCIATE MEMBERS

Peter Gordon Magnus Goudie, Lieut.(E), R.N.
John West

ASSOCIATES

Harvey Addison
Purnendu Kumar Banerji
Arthur Conquest Barker
Roy William Beames
George Samuel Blair-Smith
Suvendu Bose, Lieut.(E), I.N.
Frank Brooks
Walter Brown
Archibald Cockburn
Peter Crooks
Amal Ranjan Ghose Dastidar, Lieut.(E), I N
William George Elliott Douglass
Thomas Drysdale
Edward Derek Evans
Edward Arthur Hardcastle
John Allan McEwan Hartin
Robert Hopps
Peter Hughes
Henry Ingle
Robert Ingram
Robert Kavanagh
William David Alexander Kirk
Ronald Herbert Macintosh
William Archibald McKimm
John Hanson May
Alan Arthur James Mills
Subramanya Muthuswamy, Lieut.(E), I N.
Henry Arthur Newman
Herbert Neilsen
Colin Penman
Edwin Hughes Pritchard
Huibert Victor Quispel
Thomas Russell

John Cheesman Stuart
Donald Bayne Sutherland
Alfred Roland Swann
Ronald James Sinclair
D. T. Thadani
Frederick John Craythorne Timmins
Keith Gilbert Ward
John Embleton Waugh
Arthur Hermon White
Stanley Cleveland Whitpaine
Reginald Anthony Xavier
Robert Coxon Young
John Galt Yuill

GRADUATE
David Glynn Owen

STUDENTS
Gareth Frederick Beale
Robert Walker Findlay
Peter Richard Addison Gillard
David Keith
Richard George Lukes
Roger Edward Pullin
Derek John White

PROBATIONER STUDENT
John Michael Charles Hockridge

TRANSFER FROM ASSOCIATE TO MEMBER
William George Attwell, Lieut.(E), R.C.N.
George Stewart Ramsay Gordon
Samuel Douglas Lomas
Kenneth John Pover
Nathaniel John Mason
Arthur Leonard Sage

TRANSFER FROM ASSOCIATE TO ASSOCIATE MEMBER
Walter Ronald Guerin

TRANSFER FROM GRADUATE TO ASSOCIATE MEMBER
Thomas Sproull Heatly
Paul Faulconer Morgan, M.A.

TRANSFER FROM STUDENT TO ASSOCIATE MEMBER
Aidan Freear Lade, Lieut.(E), R.A.N.

TRANSFER FROM STUDENT TO ASSOCIATE
Ishwar Singh Rawat

TRANSFER FROM STUDENT TO GRADUATE
John Olav Tollefsen

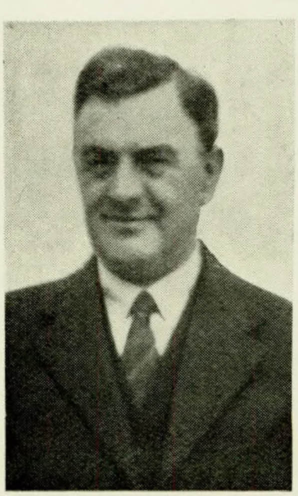
TRANSFER FROM PROBATIONER STUDENT TO STUDENT
Frederick Charles Cousins
William McPherson
Brian Rex Sheppard

OBITUARY

William Powrie Hunter

An appreciation by Dr. A. C. West (Member)

Mr. W. P. Hunter was born on the 26th February 1886. He was educated in Arbroath and served an engineering apprenticeship there at the beginning of the century with Douglas Fraser and Sons. Thereafter he joined the Royal Mail Steam Packet Co., Ltd., as a seagoing engineer. At the beginning of hostilities in 1914, he was commissioned engineer lieutenant in the Royal Naval Reserve and served, throughout the first World War, in various cruisers, taking part in many major operations. While on monitor service he took part in the tracking down and sinking of the German cruiser *Konigsberg* on the East Coast of Africa.



When the war ended he reverted to the Merchant Service and, later, obtained an Extra First Class Board of Trade Certificate of Competency under the late Mr. Peter Youngson at Liverpool. His association with Mr. Youngson led him in 1921 to take up teaching and to the post of lecturer in marine engineering at Robert Gordon's Technical College in Aberdeen, to which work he devoted the remaining and major portion of his active life, retiring thirty years later in January 1952. During his brief retirement, until his death on 24th January 1953, his chief interests were his garden and the game of golf.

Mr. Hunter was elected a Member of the Institute in 1921 and was appointed Vice-President in Aberdeen in 1922, a position he held until his death. For a considerable period he was particularly helpful in arranging for candidates in Aberdeen to take the Institute Student Graduate, and later the Student Examinations at the college. Throughout the years of his Vice-Presidency, he supported the applications of large numbers of candidates for election to membership of the Institute.

Mr. Hunter exercised a wide influence as a teacher of marine engineering and it was at Robert Gordon's Technical College that his greatest work was done. His single hearted enthusiasm for his job was an inspiration to all. Among the several outstanding men who contributed towards the building-up of the marine engineering department at Aberdeen, he played a most important part. Many students who studied under him and who are now scattered over the world will remember him with esteem and affection and the College is faced with the loss of a greatly valued and trusted colleague.

R. B. Baseley (Member 2483) was born in 1878. He served an apprenticeship with J. U. Harper in Jersey for two years and Cochran and Company, Annan, for three years. He served with the Royal Navy for a short period and spent some years at sea in the Merchant Navy, obtaining a First Class Board of Trade Certificate. In 1911, when he was elected a Member of the Institute, he was engineer-in-charge for the

Falkland Island Company, Port Stanley, and he retired in 1928. He was an Associate Member of the Institution of Mechanical Engineers. The date of his death is not known.

CHARLES VICTOR ALBERT ELEY (Member 3320), who has recently died, was elected a Member in 1917. He was apprenticed to Spencelayh and Company of Rochester for three years and the Medway S.P. Co., Ltd., for a further four years. He had two years' experience in sea and river vessels and in 1917 he was a consulting engineer in Birmingham.

JOHN DAVID HAMILTON (Member 10660) was born in Christchurch, N.Z., on 21st September 1898. He received his secondary education at Christchurch District High School and his technical education at Canterbury College, N.Z. He served an apprenticeship at the Islington works of the New Zealand Refrigerating Company and worked there until January 1926, when he joined the sea staff of the Union Steam Ship Company of New Zealand as junior engineer in t.s.s. *Marama*. He served in this vessel in the company's Pacific service until February 1931, when he went to Barrow-in-Furness to take the position of third engineer in t.e.v. *Rangatira,* then under construction at the yard of Vickers-Armstrongs, Ltd. Thus started a long association with the company's Wellington-Lyttelton steamer express service, as he remained in this vessel until May 1946, having been appointed chief engineer in 1944. His special knowledge and experience of the service was utilized by the company when he stood by the t.e.v. *Hinemoa* at Barrow, the first British passenger vessel to be built subsequent to the second World War.

In March 1949 his services were employed as acting assistant superintending engineer at the head office of the company in Wellington and his success in this position, especially his ability to handle post war staff problems, resulted in his permanent appointment to the head office staff.

On 26th April 1953, while on holiday leave, he was involved in a motor accident on the eastern shore road of Lake Taupo, his death being instantaneous. He is survived by Mrs. Hamilton, who was driving the car at the time and who is in Rotorua Hospital recovering from multiple injuries. Mr. Hamilton had many friends on the staff of Vickers-Armstrongs, Ltd., who built the *Rangatira* and *Hinemoa,* and he was well known to the staff of the marine department of the British Thomson-Houston Co., Ltd., of Rugby, who supplied the main and auxiliary machinery for both vessels.

Mr. Hamilton was a member of the New Zealand Institute of Marine Engineers and was elected to membership of the Institute of Marine Engineers in 1946.

JAMES HENDERSON (Member 2811) died in June 1952 at the age of eighty-six. He was born and educated in South Brisbane and started an apprenticeship with Harry Sargeant and Company at Alice Street, Brisbane, which was completed with the Howard Smith Shipping Company. He went to sea in vessels owned by the Colonial Sugar Refining Company and remained with them for seven years, obtaining a First Class Board of Trade Certificate. He left the sea to enter government service in the Harbours and Rivers Department. In 1908 he joined the staff of the machinery department and for twenty-two years was chief inspector of machinery, scaffolding, weights and measures. In 1933, at the age of seventy, Mr. Henderson retired from the government service but was appointed engineer to the Queensland Insurance Company, where he remained for nearly ten years, finally retiring from business at the age of eighty. He had been a Member of the Institute since 1913.

JOHN MCKINLAY (Member 7655) was born in 1871. He served an apprenticeship with David Rollo and Sons, Liverpool, from 1887-92 and then spent six years at sea with the Red Star Line and Houston Lines. He came ashore in 1898 obtained an Extra First Class Board of Trade Certificate, and was engaged as marine draughtsman with Fawcett, Preston and Co., Ltd., of Liverpool, until 1906. From 1906-17 he was employed abroad in work unconnected with marine engineering but from 1918-20 he was yard superintendent of the Tebo Yacht Basin, Todd Shipyard. Mr. McKinlay was engineer superintendent for the Chile Steamship Company from 1920-26 and was then promoted manager and vice-president, positions he held at the time of his election to membership of the Institute in 1934. During his residence in America he became an American citizen; he was a member of the American Society of Naval Architects and Marine Engineers. Mr. McKinlay died on 30th March 1952.

GEOFFREY MAUGHAN (Associate 13751) was born in 1927 and died as the result of an accident at sea on 22nd April 1953. He served an apprenticeship with the Liverpool and Glasgow Salvage Association from 1944-48 and then sailed as fourth to second engineer with Coast Lines, Ltd., until 1951. After a six months' course of study at the College of Marine Engineering, Liverpool, he obtained a Second Class M.o.T. Motor Certificate and in February 1952 was appointed third engineer with United Whalers, Ltd.; it was while serving in that company's m.v. *Powell* that Mr. Maughan met with the accident which caused his death. He had been elected an Associate of the Institute in April 1952.

THOMAS RIDLEY HETHERINGTON MORRISON (Member 4782) was born in 1889. He served an apprenticeship with D. and W. Henderson and Co., Ltd., in Glasgow; he worked for the same company subsequently as a draughtsman, also for Burmeister and Wain and David Rowan and Co., Ltd. He went to sea for four years in steamers owned by the Hogarth Shipping Company, until the ship in which he was serving was lost through torpedo action in the 1914-18 war. For two years afterwards he served the Admiralty at sea, before his appointment by them as a technical officer ashore. During this time Mr. Morrison had obtained a First Class Board of Trade Certificate.

In 1919 he was appointed a ship and engineer surveyor to Lloyd's Register of Shipping and, after six months at Southampton, he was sent to Buenos Aires, where he remained for five years. The greater part of his work there was the survey of refrigerating machinery and appliances, which enabled him to obtain considerable experience in this branch of the society's work. In March 1925 he was transferred to Genoa, where much of his time was given up to new construction work in connexion with some of the large Italian passenger vessels then building. In November 1927 he came into the society's head office to serve on the staff of the chief engineer surveyor, being promoted a senior ship and engineer surveyor in January 1935; he continued in this appointment until January 1953, when he became seriously ill, and his death followed on 17th May.

Mr. Morrison leaves a widow and a son aged nineteen, who recently left Epsom College with high honours and is now reading classics at Trinity College, Oxford.

He was elected to membership of the Institute in 1923.

M. D. NEIL (Member 2124), elected to membership in 1909, died on 11th December 1950.

DAVID BLAIR PATERSON (Member 13700) was born in 1915. From 1929 to 1934 he served an apprenticeship with the Forth and Clyde Roperie Company, Kirkaldy, and continued in their employment thereafter until 1937 as a journeyman. For the next five years he was engaged as junior and fourth engineer with the Peninsular and Oriental Steam Navigation Company and obtained a First Class Steam Ministry of Transport Certificate in 1943. He returned to sea as second engineer with the Currie Line, Ltd., and was promoted chief engineer in 1946, continuing in that capacity until he left the company in 1949 to gain experience in motor ships. For the next six months he sailed as third engineer with the Lyle Shipping Co., Ltd., and obtained a First Class Motor Endorsement to his M.o.T. certificate early in 1951. In April 1951 he joined

Buries Markes, Ltd., as second engineer and was promoted chief engineer three months later; it was while engaged as chief engineer of the *La Estancia* that he died suddenly at Djibouti, French Somaliland, on 19th May 1953, at the early age of thirty-eight years.

Mr. Paterson had been elected a Member of the Institute in March 1952.

WILLIAM ROBERT GEORGE THOMAS (Member 7725) was born in 1893. He served an engineering apprenticeship with White Brothers Engineering and Machinery Co., Ltd., Stratford, London, from 1908-13; during this period he attended classes at the West Ham Technical College. From 1913-22 he was seagoing, with the Aberdeen and Commonwealth and Dominion Lines in the first place and, when the 1914-18 war broke out, in vessels as directed by the Royal Naval Transport Service, including tankers. Mr. Thomas obtained a First Class Board of Trade Certificate in 1918.

From 1922-25 he was employed by R. and H. Green and Silley Weir, Ltd., as chargehand fitter on all classes of Diesel, petrol, turbine and reciprocating engines. In 1925, however, he was appointed temporary engineer at Woolwich Ferry, thus commencing the service with the London County Council which was to continue until his death on 22nd February 1953. In 1927 he was promoted second engineer, in 1929 engineer, and in 1931 he was appointed assistant superintendent. In 1940 he was transferred to the chief engineer's department at County Hall as higher assistant (First Class A) and in 1947 he was appointed senior assistant, the position he held when he died.

Mr. Thomas had been a member of the Institute since 1934.