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Development of merchant ship propulsion machinery over the past 25 years

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INTRODUCTION

It is of course a privilege to be asked to deliver the second Presidential Address in my 2 year term of office although I have to say that the selection of subject does not come easier the second time around. At the same time you as an audience face the problem of listening to me a second time and as to that I can only undertake to use my best endeavours to make it as painless as possible.

This is in fact the last Presidential Address to be delivered in the first 100 years of our Institute and it occurred to me therefore that it was time for certain reflection. Your Presidents are in fact allowed a great deal of latitude in the choice of subject for their dissertation and looking back it will be seen that a very wide range of subjects have been chosen reflecting the broad range of interests of marine engineers.

For my part, I have decided that the time was appropriate for an address in engineering development as we can see it over the last 25 years in the marine field and in deference to the ladies present I will seek to develop my theme in terms which are technologically comprehensible.

I have chosen a period of 25 years because it is conveniently a quarter of a century and earlier periods have already been well documented.

Additionally however the era I have chosen begins in 1963 and covers the working lives of most of us – it is interesting therefore to look back from our perspective of today to consider the very remarkable changes that have taken place, few of which would have been predicted 25 years ago.

In 1963 there was everything to play for – a wide choice of propulsion machinery types and designs – and the possibility emerging of completely new machinery forms being developed. Some of these have not materialized, others have disappeared into obscurity, and after years of intense competition and extensive development a dominant pattern of choice has emerged in recent years. It is the fascinating pattern of these years that I will seek to reflect in my address.

THE MARINE STEAM TURBINE – APOGEE, DECLINE AND FALL

Immediately before the period we are considering Daniel Ludwig astonished the shipping world in 1960 by building the *Universe Daphne* – a tanker of 115,000 tons deadweight and 25,000 hp which was almost twice the size of the largest tankers in general service at the time, thus inaugurating the era of very large bulk carriers.

The growth in size of oil tankers over the ensuing 20 years has been tabulated by Professor Jung[1] and was followed closely by combination dry bulk carriers, ore/bulk/oil and ore/oil carriers as the economies of scale led to larger and larger ships.

Alex Harrold served an apprenticeship with North Eastern Marine Engineering Co. Ltd., Wallsend, from 1941 to 1945, during which time he was awarded the Superintendent's Cup for best apprentice and the 1943 Scholarship of the North East Coast Institution of Engineers and Shipbuilders, and obtained his B.Sc. in Marine Engineering at King's College, University of Durham. In 1946 he joined the sea-going staff of Anglo-Saxon Petroleum Co. and obtained his First Class Certificate of Competency (Steam and Motor). In 1952 he was appointed ashore as Superintendent Engineer. Following the creation of Shell Tankers U.K. Ltd. he was appointed Fleet Manager in 1959, Technical Manager (Group New Construction), Shell International Marine Ltd. in 1965. In 1969 he joined the Hill Samuel Group as a Director of Lambert Brothers Shipping Ltd., leaving in 1976 to set up the partnership of Vine, Able & Harrold Ltd., consulting marine engineers. In 1984 he established a practice in his own name as a consulting marine engineer and divides his time between London and his home in North Wales. Mr. Harrold was appointed Deputy President of the Institute of Marine Engineers in 1983, after three periods of office on its Council totalling 11 years, and serving on or chairing some of its key committees. His 2 year term as President commenced on 12 March 1987.

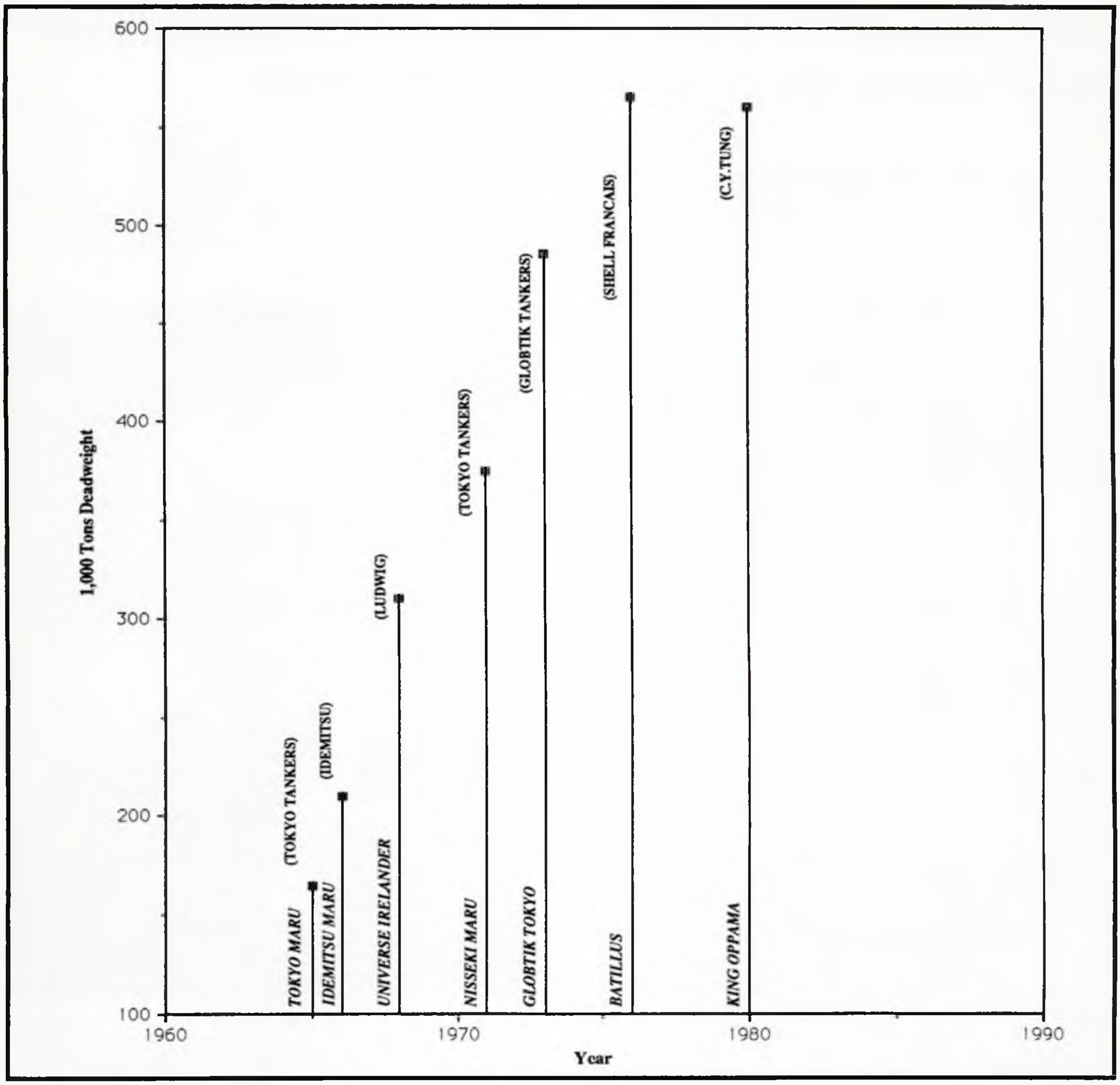

Fig. 1. Tanker growth 1965–1980

Fig. 1 shows the progressive growth in tanker size over a period of 15 years between 1965 and 1980 from 150,000 tons deadweight to 560,000 tons deadweight. By 1980 as a result of the second OPEC oil crisis the party was over. In response to the trebling of crude oil prices in 1973 the oil market pressed on with the development of oil fields closer to the centres of consumption, notably the North Sea, and consumers concentrated on economies in oil consumption, both factors producing a drastic reduction in the requirement for large tankers.

1980 saw the end of the Ultra Large Crude Carrier (ULCC) phenomenon and by 1985 one-third of the world tanker fleet was in lay-up.

While this pattern persisted the marine steam turbine enjoyed it's heyday. The increasing power demand from 30,000 to 50,000 hp on one propeller frequently over-reached the capacity of diesel engines then available (see Fig. 2). Power availability was therefore the fundamental controlling factor. However, at least one well-known Norwegian shipowner timed his progress in the large tanker field to the availability of the diesel engine and he not only survived the ensuing shipping crisis but remains a successful shipowner to this day.

This remarkable era for the steam turbine spanned no more than 15 years and as Fig. 3 shows significant ascendence was effectively limited to a period of 10 years, the rate of decline after 1974 was much more precipitous than the rate of growth prior to that fateful year.

Of course application of the steam turbine had been widely adopted in smaller tankers prior to the period we are considering, particularly by some of the international oil majors. However that was largely attributable to the influence of the U.S.A. where the ubiquitous and highly successful T2 tanker was developed during World War II in the absence of a low-speed diesel manufacturing industry and added to powerful prejudices against the less attractive environmental features of

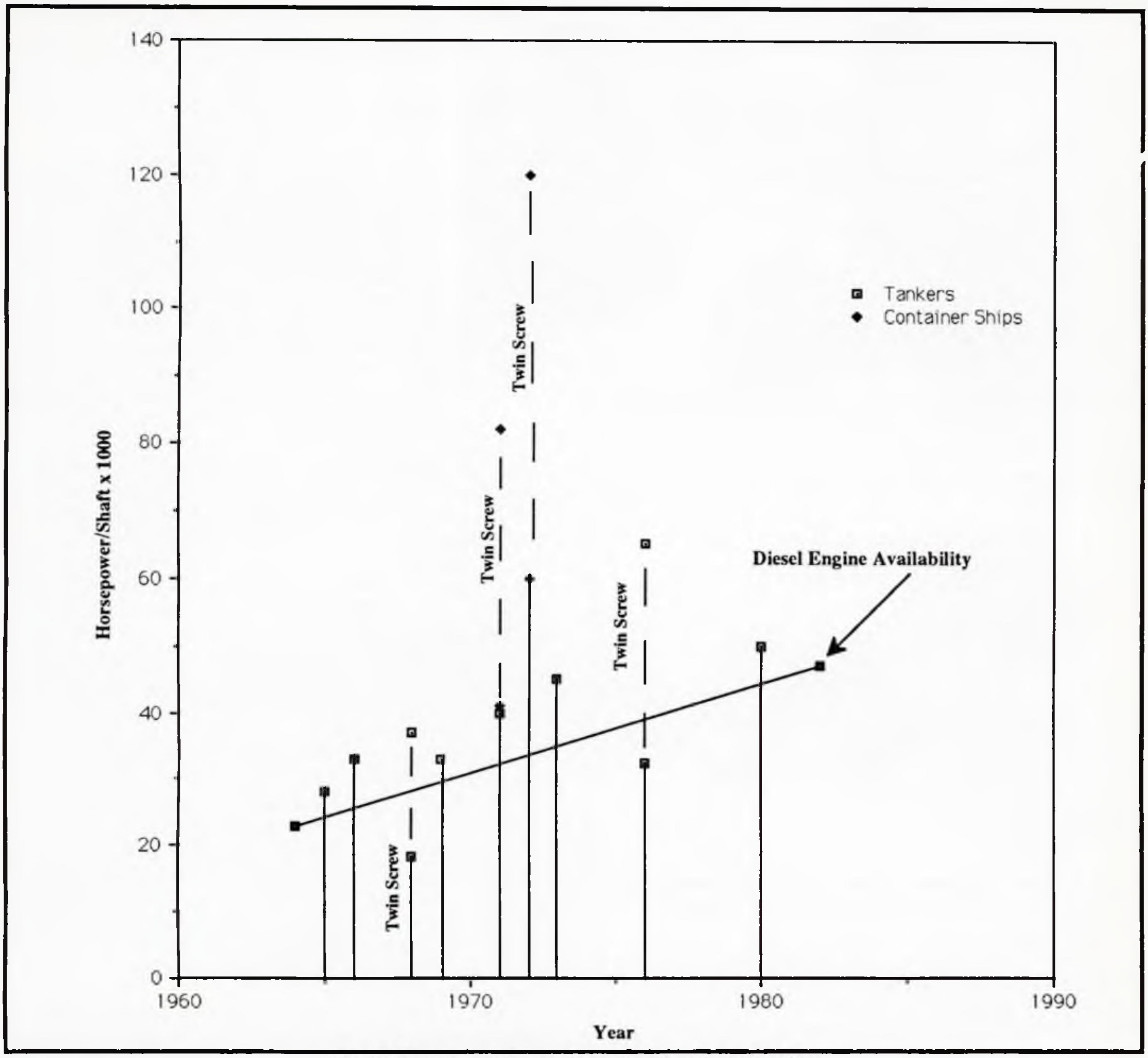

Fig. 2. Power requirement versus diesel availability

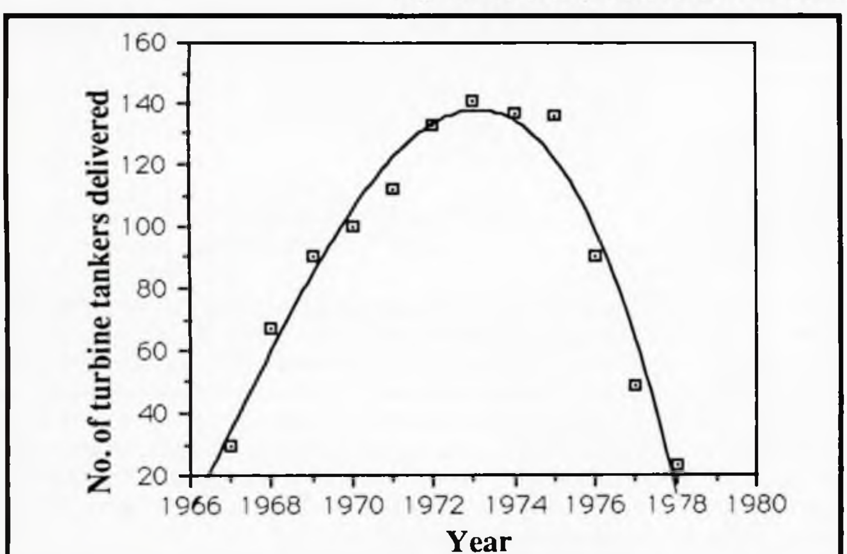

Fig. 3. 10 year era of steam turbine

earlier diesel engines – high noise, gas and oil leakages and a high manual maintenance load.

Nevertheless during that earlier period most independent tanker owners had consistently built motor tankers to their great commercial advantage. However there was a short 4 year period between 1968 and 1972 when the number of steamships on order actually exceeded the number of motor ships in ships of all classes exceeding 20,000 tons deadweight.

Although the steam turbine had an open field for a period of some 15 years it did not rest upon its laurels. Professor Jung[1] has recorded a fascinating account of the developments in steam turbine installations over that period – compelled by the unrelenting gap of more than 20% between the fuel economy of a steam turbine installation and a diesel engine.

Of course the steam turbine had one advantage in that fuel economy improved with

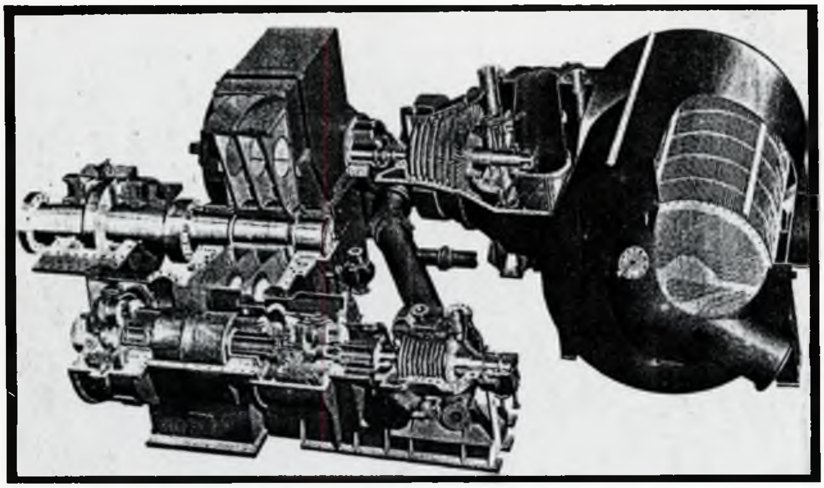

Fig. 4. Arrangement of Stal-Laval AP1V engine

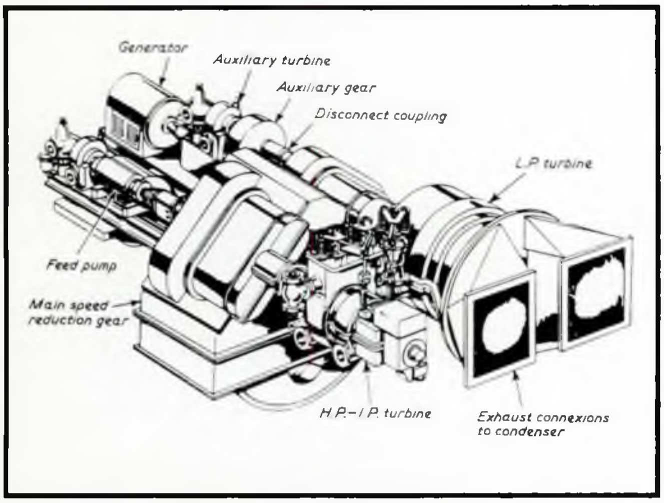

Fig. 5. Arrangement of GE MST 14 engine

increased power, mainly due to the fact that the blade clearances remained the same as engine size increased. The fuel economy of the diesel engine remains more or less constant over a broad range of power requirement.

For this reason alone the French Shell ship *Batillus* and the Ludwig *Universe Irelander* were ill-conceived aberrations - there could be no justification or requirement to design these vessels for twin-screw propulsion. *Universe Irelander*'s length/beam ratio was comprised to suit the IHI building dock and her canted twin rudders gave problems. *Batillus* was built when her possible market had already disappeared and the low propulsion and fuel efficiency arising from her twin screws did not help. Both ships traded disappointingly and demonstrated that twin screws for this class of ship was the wrong route to take.

The battle between the steam turbine and the diesel engine was nevertheless enjoined and for the steam turbine it was waged in the form of higher steam conditions leading ultimately to the adoption of reheat cycles and of improved gearing arrangements permitting higher reduction ratios and hence lower propeller speeds and giving increased propulsion efficiency.

Increases in steam temperatures were modest – ranging from 450° to 525°C. The incremental advantage of increased superheat was a law of diminishing returns and eventually came up against the barrier of using austenitic steel which was inhibitive due to high cost and the adverse corrosive effects of low-grade marine fuel.

Steam pressures, on the other hand, more than doubled from 42 bar to 104 bar leading to a remarkable reduction in the physical size of the HP turbine, shorter blade lengths and the adoption of supercritical rotors. HP rotor speeds increased to 6700 rev./min – a far cry from the 1500 rev./min typical of the Parsons reaction turbine.

The adoption of single-plane gearing, including the use of planetary gears in the first reduction, improved compactness and facilitated the adoption of axial flow to the condenser, thereby reducing kinetic exhaust losses. Generator and feed pump drives were incorporated in the gearbox. A remarkable 30% reduction in machinery weight was also achieved (Figs. 4 and 5).

Finally the reheat cycle which had been pioneered by Canadian Pacific and Fairfields in 1956 with the Empress ships, and had for long been standard practice in the land power generating industry, was successfully adapted to marine use. The problem of reheat on board ship had always been associated with the requirements of manoeuvring and astern operation, when the reheater was starved of steam and therefore liable to burn out. The increased steam inertia in the system also presented problems of turbine steam control in the event of sudden loss of load. These problems were however overcome by the adoption of gas by-pass reheaters in the main boiler and additional emergency overspeed trip valves immediately after the reheater.

Notwithstanding all these efforts there remained in 1980 a gap of almost 20% in thermal efficiency between the marine steam turbine and the diesel engine – a gap which would be even greater today and is unlikely to be closed.

Other types of ship had little impact on the scene. The only steam passenger ship of note built during the period was *QE2* in 1969 – that was not an engineering success and has recently been re-engined. The fast container ships of the Bay class and Sealand are shown in Fig. 2. They were high powered but commercially ill-conceived and have mostly been re-engined with diesel engines.

This brilliant era of marine steam turbine propulsion in merchant vessels was short lived, spanning an effective period of only 15 years. It did however rise to some remarkable and inspired design and production technology. Inevitably however when the diesel engine was developed to cover the higher-power ranges and had demonstrated its ability to operate on low-grade fuel, predictably the laws of thermodynamics were not to be defied and steam propulsion came to an abrupt end.

THE GAS TURBINE – BRIEF APPEARANCE

Application of the gas turbine to merchant ship propulsion had been ushered in during the 1950s with the Shell ship *Auris* (350 g/hp/h) and Marad's *John Sergeant* (230 g/hp/h). Both these installations incorporated gas/air regenerators and were based on derivatives of industrial gas turbines. They were not economical, the problems with burning residual fuels were not resolved, and neither ship traded continuously. This appeared to be the end of the line for the maritime application of industrial-type gas turbines.

The navies of the world, in the meantime, proceeded with

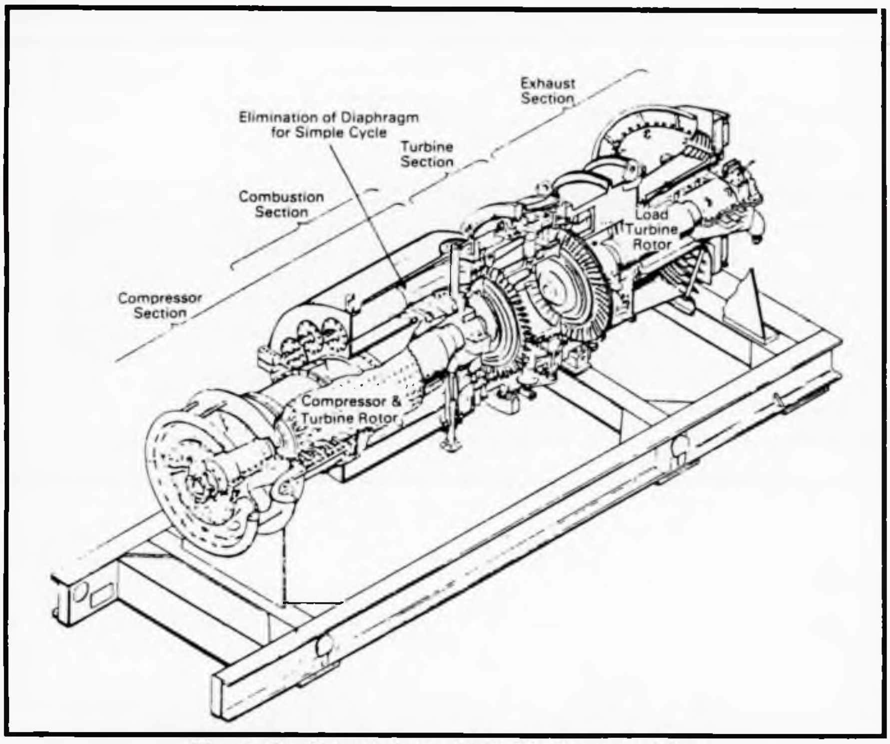

Fig. 6. Sectional view of MS 5000 gas turbine

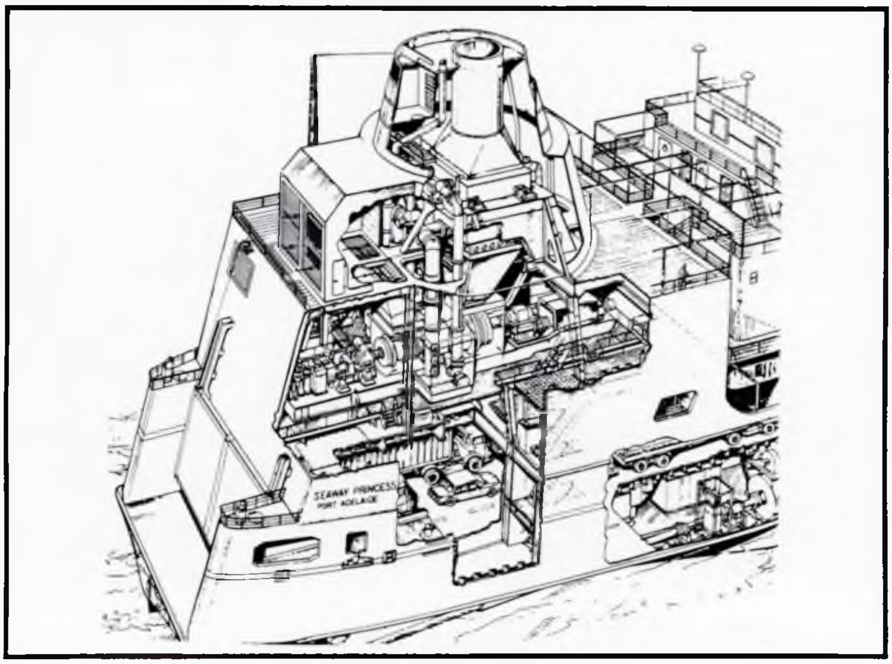

Fig. 7. Cutaway of *Seaway Princess*

the marinization of aircraft-type gas turbines running on distillate fuel and in various combinations these have now become standard in naval surface vessels.

In 1967 the *Admiral Callaghan* ran trials in the U.S.A. fitted with two Pratt & Whitney aircraft-type gas turbines developing 20,000 hp and giving a fuel rate of 230 g/hp/h, running on distillate fuel and without a gas/air regenerator. This performance, however, was no better than the *John Sergeant*, 12 years earlier. The Pratt & Whitney machinery was later replaced by General Electric gas turbines which could operate on vanadium-free heavy distillate fuel or carefully controlled specification at a price some 50% above that for heavy fuel oil.

In 1970 Pratt & Whitney returned to the fray with their FT4-A units for the 30 knots 60,000 hp Euroliner container ships. These units were similar to those fitted in the *Admiral Callaghan*, and again without gas/air regenerator, although the fuel rate had now been reduced to 220 g/hp/h and some further progress had been made in fuel treatment. However, the ships were hopelessly uneconomic and all four vessels were subsequently re-engined with diesel engines.

The scene shifted to the Antipodes when the Broken Hill Proprietary Co. built to 15,000 tons deadweight ro-ro vessels *Iron Monarch* and *Iron Duke* in 1973. Now the adapted industrial-type gas turbine made a reappearance with the installation of two General Electric MS5000 gas turbines having a relatively modest power of 17,500 shp (Fig. 6). The power turbines of these two shaft units drove CP propellers (CPP) through single input, single output locked train double reduction gears, and a large gas/air regenerator was fitted between the compressor and power turbines.

In fact the gas/air regenerators (which incidentally weighed 100 tons) proved to be the most troublesome features of these installations due to repeated crack formation in the tube bank assembly caused by frequent thermal recycling. Repair of these cracks *in situ* was immensely difficult and the regenerators were eventually by-passed resulting in a 30% deterioration in fuel economy.

Residual fuel was used in these ships by a system of dual washing treatment with Tretolite but the treatment varied on the basis of fuel analysis of each batch of fuel both on shore and on board to determine the sodium, potassium and vanadium content. The process was therefore onerous and precluded the admixture of different batches of bunkers.

In 1974 the Union Steamship Company entered the scene with two 5000 ton ro-ro vessels on 10,500 shp powered by General Electric MM3112R gas turbines driving twin fixed-pitch propellers through ac/dc electric drive (Fig. 7).

Then in 1976 Union Steamship Co built two trans-Tasman vessel of 27,500 shp powered by General Electric MM5262RB gas turbines driving twin screw CPP through ac electric drive.

Finally in 1977 two 45,000 ton deadweight bulk carriers were built by BHP with similar machinery but with epicyclic drive to twin screw CPP (Table 1).

None of these ships were a commercial success. *Seaway Prince* and *Seaway Princess* operated on distillate fuel on which the Australian Federal Government imposed a tax after the vessels entered service, adding $38M annually to the fuel bill. Both vessels were withdrawn from service in 1983 and scrapped in 1986.

One of the trans-Tasman vessels, *Union Rotorua*, is apparently still operating on distillate fuel but the other vessels were eventually re-engined with diesel propulsion

Table 1. Specifications of GT-powered vessels

Year	Vessel	Owner	Type	Power	Engine	Drive	Propeller
1973	*Iron Monarch*	BHP	15,000 tons	17,500	GEMS5000	Geared	CP
	Iron Duke	BHP	Ro-ro	"	"	"	"
1974	*Seaway Prince*	Union	5000 tons	10,500	GEMS53002R	Geared	Twin
	Seaway Princess	Union	Ro-ro	"	"	ac/dc electric	Fixed pitch
1976	*Union Rotorua*	Union	Trans-	27,500	GEMS5002RB	ac	Twin
	Union Rotoiti	Union	Tasman	"	"	Electric	CP
1977	*Iron Carpentaria*	BHP	45,000	10,500	GEM3002R	Epicyclic	Twin
	Iron Curtis	BHP	tons bulk	"	"	Epicyclic	CP

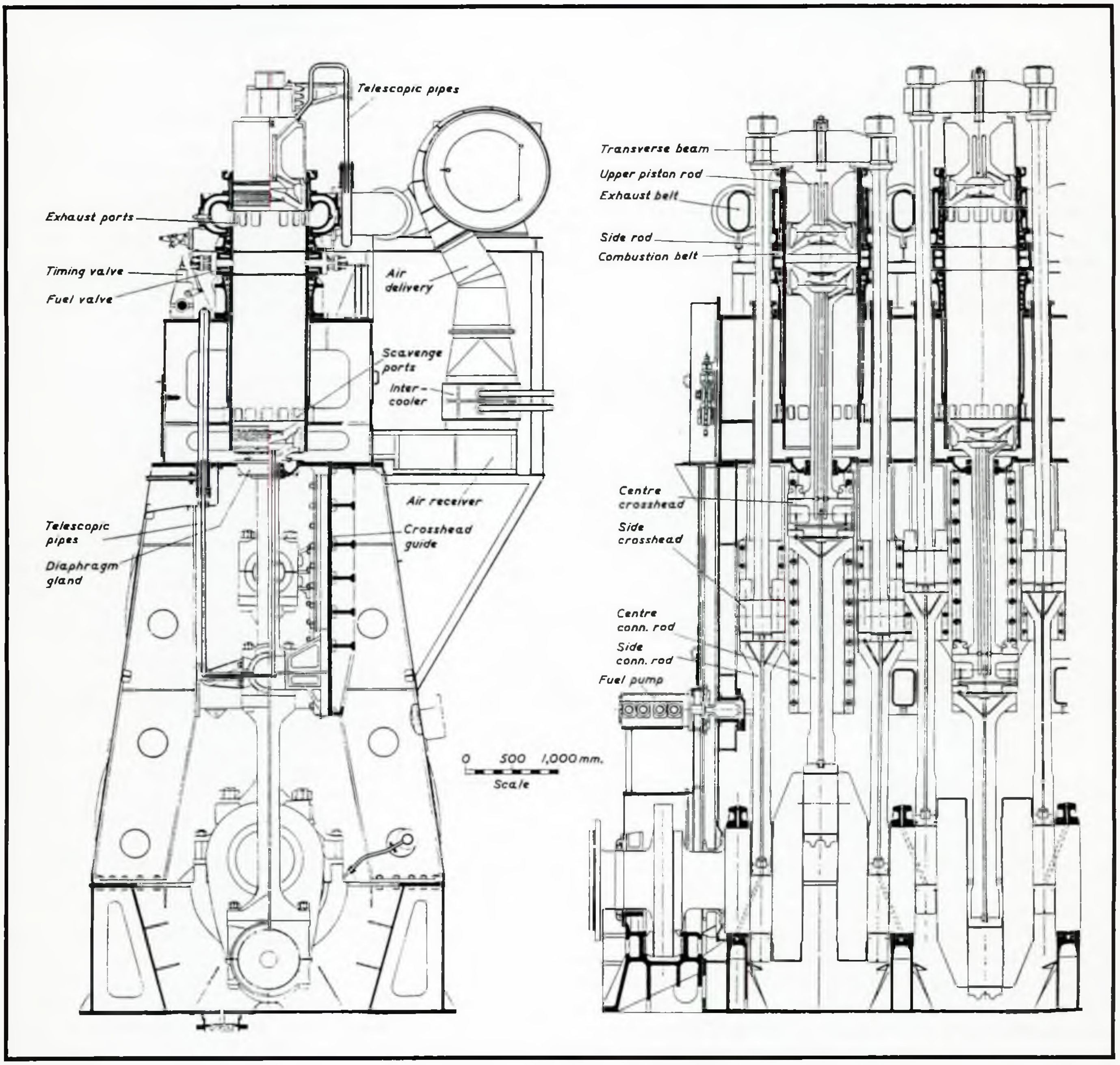

Fig. 8. Sections of a 'J' series engine

– an expensive misjudgment indeed because if error is made in the choice of propulsion machinery requiring re-engining then it follows that a high proportion of the original ship investment has to be written off and re-financed.

The reasons for the misjudgment are of course clear. Restrictive crew practices militating against on-going maintenance and repair, and restricted trades allied to the false allure of controlled fuel specifications at reasonable cost. The apparent simplicity of purely rotating machinery seemed irresistible.

But the theory of Carnot and past experience were not to be denied.

Troubles with gas/air regenerators are reflective of similar endless troubles with gas/air heaters in boilers in the 1950s which led to the adoption of steam/air heaters. Gas-cooled nuclear reactors have experienced similar problems.

Even with trouble-free regenerators the efficiency of the gas turbine cycle cannot be brought to acceptable limits until the maximum combustion temperature can be raised to levels which are still unattainable with metal blades when burning residual fuel oil.

Ash deposition on nozzles and blading inevitably reduces the output and efficiency of a gas turbine. Sodium sulphate attacks silicate and alumina in refractories. Vanadium, particularly when sodium is present, attacks high-temperature-resistant nickel alloys.

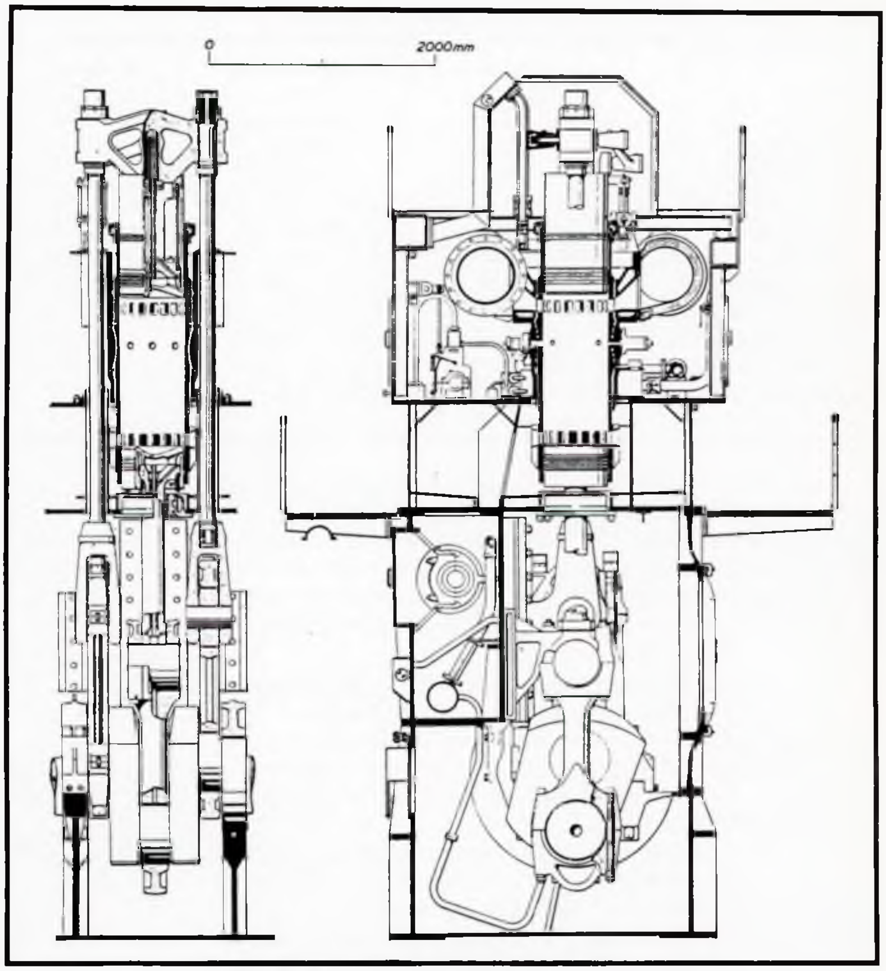

Fig. 9. Sections of a Seahorse engine

None of these phenomena are new – they have been known throughout my working lifetime and predictably the gas turbines installed incurred tip erosion and hot corrosion in the first-stage blades and nozzle cracking, reducing the life of these components to 10,000 h or less depending on fuel quality.

It is true that the price of bunkers changed dramatically between the inception and the service life of these installations. But that alone does not account for the failure – fuel has always been a predominant cost in ship operation. Failure was in fact endemic due to technical considerations which inhibited high efficiency and sustained reliability.

The experiment was brave. However the conclusion must be that gas turbines will not play a significant role in merchant ships as prime movers until either:

1. a whole new technology develops based on the application of ceramics and/or sintered materials or;
2. clean fuels become available at acceptable cost.

Neither of these developments is presently in sight.

THE 2-STROKE OPPOSED-PISTON ENGINE – END OF A STORY

The remarkable development of the Harland & Wolff B & W opposed-piston engine under the guidance of C. C. Pounder has been well documented by Professor Crossland[2]. This development peaked in the early years of the period we are considering with a turbocharged engine of 750 mm bore and 2300 mm total stroke, delivering 2500 hp/cylinder up to a maximum of 25,000 hp. C. C. Pounder retired in 1964 and thereafter development of the B & W engine was exclusively to the poppet valve uniflow design.

In 1965 P. Jackson unveiled the Doxford 'J' engine having a bore of 760 mm and 2180 mm total stroke delivering 2360 hp/cylinder up to a maximum of 21,000 hp at 115 rev./min (Fig. 8).

In an earlier period of development of the Doxford engine as powers increased serious problems had been experienced with crankshaft fractures. These had been overcome by re-design and in the new engine a determined attempt was made to stiffen and shorten the crankshaft by utilizing the side crank webs as bearing journals, thus dispensing with the spherical bearings which had been a distinguishing feature of Doxford engines. As a result the cylinder centres were reduced to 2.28 m. Nevertheless the 9-cylinder engine was still some 12 m in length and the crankshaft weighed in at 132 tons.

The engine was pulse-charged with oil-cooled lower piston and water-cooled upper piston and retained the common rail fuel system at an injection pressure of 460 kg/cm^2. The scavenge pressure was 0.6 kg/cm^2 and at an MIP of 9 kg/cm^2 the fuel consumption was 155 g/hp/h.

The Doxford engine was always a low-compression engine with a compression pressure of 42 kg/cm^2 and P_{max} was in this case limited to 62 kg/cm^2, values significantly lower than

Table 2. Engine particulars

	'J' engine	Contemporary engine	Modern engine
Bore (mm)	760	900	800
Stroke (mm)	2180	1500	2300
No. of cylinders	9	9	5
Horsepower	21,000	20,700	22,140
Rev./min	115	115	83
P_{max} (kg/cm^2)	62	75	125
P_{comp} (kg/cm^2)	42	60	115
mep (kg/cm^2)	9.1	9.7	16.2
Scavenge ata.	0.6	0.8	3.0
Fuel consumption (g/hp/h)	166	155	121

contemporary engines at the time. Subsequently the rating of the 76J engine was increased to 2500 bhp/cylinder and 1975 and 1976 a number of 67J4 engines fitted with pulse converters were delivered having a specific fuel consumption of between 145 and 150 g/hp/h (Table 2).

In fact the engine configuration did not permit development to even higher ratings – imagine the scantlings for the 'J' type crankshaft and side rods if the maximum cylinder pressure was to be increased to today's levels. The Doxford engine was therefore excluded from the higher powers which were beginning to dominate the market.

Concentration was thereafter directed towards the development of lower power units with the concept of the Seahorse engine emerging in 1972. The engine was designed to operate at 300 rev./min, a choice apparently not unconnected with its unique suitability to develop electric power at both 50 and 60 cycles (a characteristic incidentally shared with the medium-speed engine running at 600 rev./min). It was also decided to limit the design to a bore of 580 mm for all powers to minimize production cost (Fig. 9).

For propulsion purposes the concept required geared drive and the engine was envisaged with 4, 5, 6 or 7 cylinders developing from 10,000 to 17,500 hp. The challenge was therefore to the direct-drive low-speed engine, at that time still limited to propeller speeds in excess of 110 rev./min and to the medium-speed 4-stroke engine running at 600 rev./min with its greater multiplicity of cylinders and working parts. History has shown of course that this apparent gap was only transitory – the long-stroke direct-coupled engine now covers propeller speeds down to 60 rev./min while the medium-speed 4-stroke engine has been progressively uprated and its ability to burn heavy fuel has been established.

The development of the Seahorse engine was never completed due primarily to unsolved difficulties with the piston ring pack.

Nevertheless many of the Seahorse design features were incorporated in a 3-cylinder version of the 'J' engine developed in 1976 and intended to be offered in bore sizes of 760, 670 and 580 mm (Fig. 10).

A 580 mm bore 3-cylinder engine (same bore as the Seahorse) was installed in the container vessel *City of Plymouth* in 1978. This engine incorporated constant-pressure supercharging, assisted by an auxiliary fan, and developed 5500 bhp and 220 rev./min at a bmep of 11.42 kg/cm^2 and $P_{max.}$ of 85 kg/cm^2 and a specific fuel consumption of 150 g/hp/h – values representing a considerable advance on the original 'J' engine.

Nevertheless the competition from both low-speed 2-stroke and medium-speed 4-stroke engines was relentless – today a 4-cylinder 500 mm bore low-speed 2-stroke engine can develop 6500 hp at exactly half the rev./min of the *City of Plymouth* engine with 20% lower fuel consumption (122 g/hp/h). Such an engine would be 4.94 m in length – 25% shorter than the 58JS3 engine.

Regrettably the 'J' engine did not attract enough sales. The Doxford engine works ceased engine production in 1980 and the works were closed in 1985, thus ending the era of opposed piston engines for ship propulsion.

THE 4-STROKE DIESEL ENGINE – PERSISTENT CONTENDER

Already in 1963 when our story begins the medium-speed 4-stroke diesel engine running at speeds between 450 and 600 rev./min was making a impact in merchant vessels. It had of course always dominated the scene in smaller vessels such as tugs, coasters and ferries when limiting headroom was a governing factor.

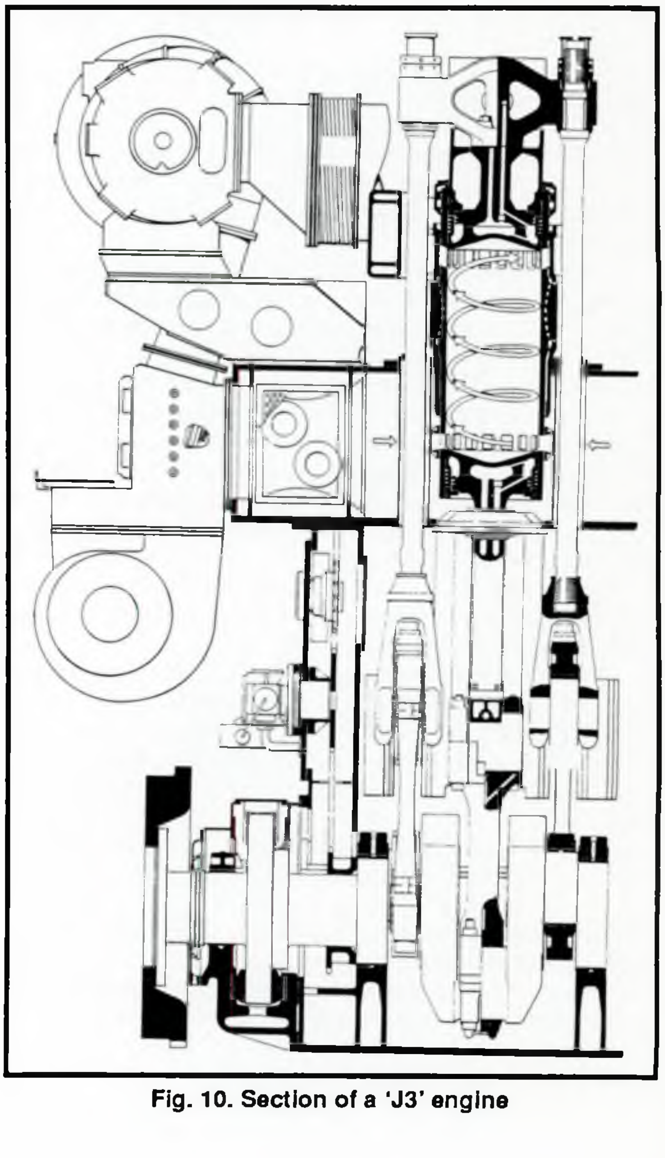

Fig. 10. Section of a 'J3' engine

Table 3. Leading engine particulars in 1967

	R&H AO	Mirrlees OP	Sulzer Z40/80	Pielstick PC2	MAN V40/54	Werkspoor TM410
Bore (mm)	362	381	400	400	400	410
Stroke (mm)	470	2 x 381	480	460	540	470
Stroke/bore ratio	1.3	1.0	1.2	1.15	1.35	1.12
hp/cylinder	500	1000	500	445	500	500
Rev./min	450	600	445	500	400	500
Piston speed (m/s)	7.06	7.62	7.11	7.62	7.21	7.32
bmep (kg/cm^2)	10.2	10.6	9.7	14.7	17.3	14.0
$P_{max.}$ (kg/cm^2)	102	102	–	88.4	111.2	116.0
Specific weight (kg/hp)	7.4	7.64	8.72	9.4	11.23	12.27
Fuel consumption (g/hp/h)	154	162	162	155	153	155

Table 4. Configurations of current medium-speed engines

	Sulzer ZA40S	Pielstick PC-4-2E	MAN 58/64	Werkspoor TM620	Wartsila Vasa 46
Bore (mm)	440	570	580	620	460
Stroke (mm)	560	620	640	660	580
Stroke/bore ratio	1.4	1.09	1.1	1.07	1.26
hp/cylinder	885	1481	1776	1897	1213
hp/litre	12.46	9.34	10.48	9.43	12.55
Rev./min	510	400	428	425	514
Piston speed (m/s)	9.52	8.3	9.1	9.4	9.9
bmep (kg/cm^2)	22.07	21.0	21.9	20.0	25.0
P_{max} (kg/cm^2)	–	–	150	–	180
Fuel consumption (g/hp/h)	128	129	129	128	126

The application of larger vessels offered the following advantages:

free choice of propeller speed through gearing;
suitability for electrical power take-off;
multi-engine flexibility;
saving in weight and engine-room length;
ease of installation;
lower cost per hp.

The limiting factors were:

high lubricating oil consumption;
limited ability to burn heavy fuel oil;
multiple maintenance.

A comparison by Neumann & Carr[3] in 1967 of leading engine types is shown in Table 3.

The AO and OP engines were 2-stroke engines under development at the time with assistance from the then Ministry of Technology. The AO engine subsequently had a disastrous entry to the market while development of the OP engine was never completed. It may be noted that these were the two lightest engines tabulated.

The remaining 4-stroke engines are still market leaders in improved and uprated forms some 20 years later and larger versions of these engines have been developed to reduce the number of cylinders required for a given horsepower.

It may be noted that the 1967 comparison shows:

1. all engines were of the 400 mm bore class;
2. the 40/54 engine had a significantly higher stroke/bore ratio than the other engines (1.35 against 1.12);
3. the 40/54 engine had significantly higher bmep (17.3 kg/cm^2);
4. the specific weight of the engines varied from 8.72 to 12.27 kg/hp.

Over the ensuing 20 years great progress was made by all these engine builders to increase the power output/cylinder, to improve performance on heavy fuel and enhance reliability and overhauling intervals.

These improvements were achieved by increasing the maximum cylinder pressure by 50% to 150 kg/cm^2, partly by redesign of combustion chamber components with improved cooling and further development of constant pressure turbocharging, higher fuel injection pressures up to 1300 kg/cm^2 giving smaller droplet sizes to 20 μm or less, and improved lubrication with high TBN lubricants and generally improved piston running conditions.

By 1980 all engine builders were claiming an ability to run on fuel of 3500 s. Redwood viscosity, albeit with some difficulty at part-load and the power/cylinder with larger-bore engines had trebled to 1500 hp/cylinder – a most remarkable development! Market penetration by the medium-speed engine reflected these achievements.

As so often happens in these matters the goal posts were moved! The low-speed 2-stroke engine builders had by now developed their long-stroke engines, thus robbing the medium-speed engine of its exclusive claim to offer low propeller revolutions. Other advantages in weight and space saving by the medium-speed engine were also now seen to be less significant and the development of constant-speed power take-off from the low-speed engine also robbed the medium-speed engine of this advantage. All that was left in favour of the medium-speed engine was therefore multi-engine flexibility and since single-engine drive has traditionally been found to provide acceptable reliability the use of multi-engines with gearing has limited attraction. As a result during the 1980s the market penetration of the medium-speed engine has probably halved. Nevertheless, in applications involving very high electrical load at sea and/or in port the medium-speed engine offers great advantages and has become more or less standard in passenger ships, the re-engining of the *QE2* being a prominent example.

In recognition of the progress made by the medium-speed 4-stroke engine over 25 years it is interesting to compare a tabulation of engines currently on offer with those tabulated in 1967 (Tables 3 and 4).

The comparison shows that:

1. the popular stroke/bore ratio has remained in the range of 1.1–1.25;
2. output per cylinder has generally trebled;
3. piston speeds have increased by 30%;
4. maximum cylinder pressures have increased by 50%;
5. fuel consumption has reduced by almost 20%.

The Vasa engine is interesting being the latest engine on the scene and although smaller in bore is developing the highest power/swept volume. It is claimed to be the first engine designed *ab initio*, for operation on the heaviest fuels, and incorporates 'inverse cooling', pilot injection, 'Swirlex exhaust' and reversion to thick pad bearings.

Much more will be heard of medium-speed development but the prospect now of it displacing the low-speed engine for the majority of merchant ships seems unlikely in the foreseeable future.

THE 2-STROKE DIESEL ENGINE – AGE OF SUPREMACY

There were seven leading European designers of low-speed 2-stroke diesel engines 25 years ago – each with their own distinctive features. Today the names of Doxford, Gotaverken, Stork, Fiat and MAN have gone into the history books as far as this type of engine is concerned and the world market is today dominated by what is now MAN–B&W based in Copenhagen and Sulzer based in Wintenthur, Switzerland.

The reasons for this remarkable rationalization are to be found in the weak shipbuilding market which has persisted over the past 15 years, the intense competition in manufacturing costs which has led designers to become almost entirely dependent on license income, and the inordinate cost of development and testing the modern low-speed engine.

Coincidentally the past 15 years has seen one of the most remarkable periods of engine development and improvement in performance by the two remaining world market leaders. Competition in thin markets has been the spur for this development but technically it has been made possible by application of sophisticated methods of theoretical analysis and detailed design, both of which have been dependent on modern computer software. The application of such technology has greatly reduced design lead times and has eliminated much of the trial-and-error testing on which advanced engine design formerly depended.

The Sulzer story

Our survey starts with the RD engine which had been introduced in 1957 with a rating of 1600 hp/cylinder and was to continue in production for the first 5 years of our period up to 1968 (Fig. 11).

The engine was loop-scavenged and pulse-supercharged with under-piston pumping into a divided scavenge trunk to enable operation over the full load range without independent scavenge assistance. The design was characterized by the rotary exhaust valve which performed three functions:

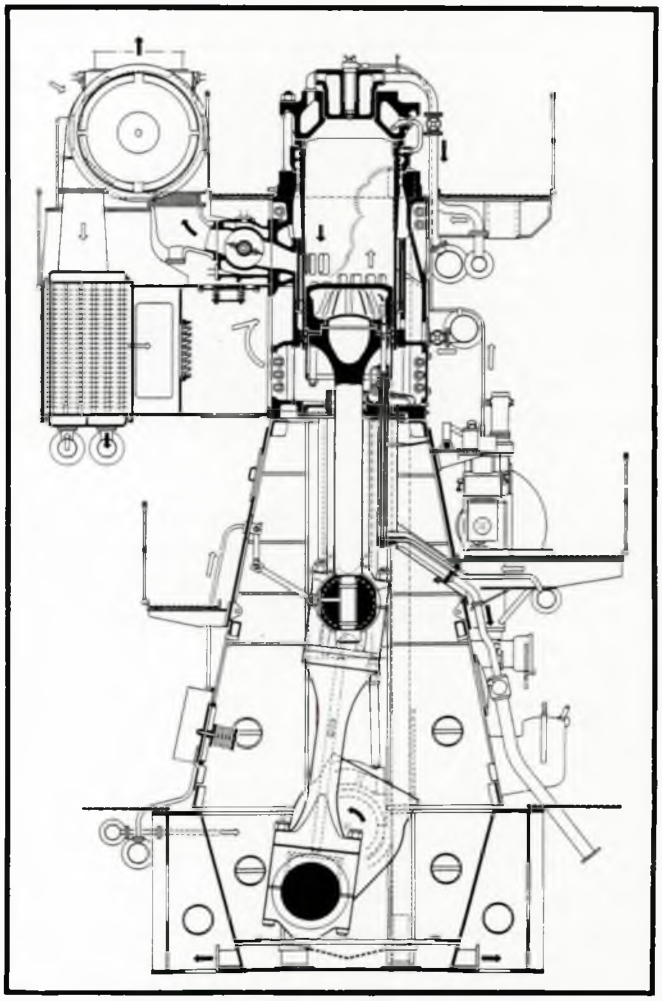

Fig. 11. RD engine section

1. closing of the exhaust ports immediately after scavenge to permit full supercharging of the cylinder prior to compression;
2. preventing blowback from other cylinders during scavenging;
3. keeping the exhaust closed throughout the compression stroke thus preventing exhaust flow into the below-piston space.

The cylinder head was of the familiar two-part form with cast-steel outer ring and cast-iron centre-piece. The piston crown was of ribbed design in cast-steel and water-cooled by telescopic pipes arranged within an enclosure which was entirely separated from the crankcase.

The RD engine continued in production over a period of 12 years and was very successful albeit not trouble-free. Early engines experienced cracks in the main bearing saddles, rotary exhaust valves were prone to seizure and fracture of their drive, cracked cylinder heads were not unknown and fuel pump cams and rollers were in some cases subject to cracking and spalling.

In 1968 it was realized that with improved turbocharger performance available constant pressure supercharging would provide the next step forward in engine ratings. The RND engine was then introduced with a maximum cylinder bore of 1050 mm, i.e. in excess of 1 m – this was in fact not quite historic because Sulzer had actually built a single-cylinder test engine with a bore of 1 m in 1911!

Nevertheless in all other respects the RND engine represented a major step forward. The rotary exhaust valve disap-

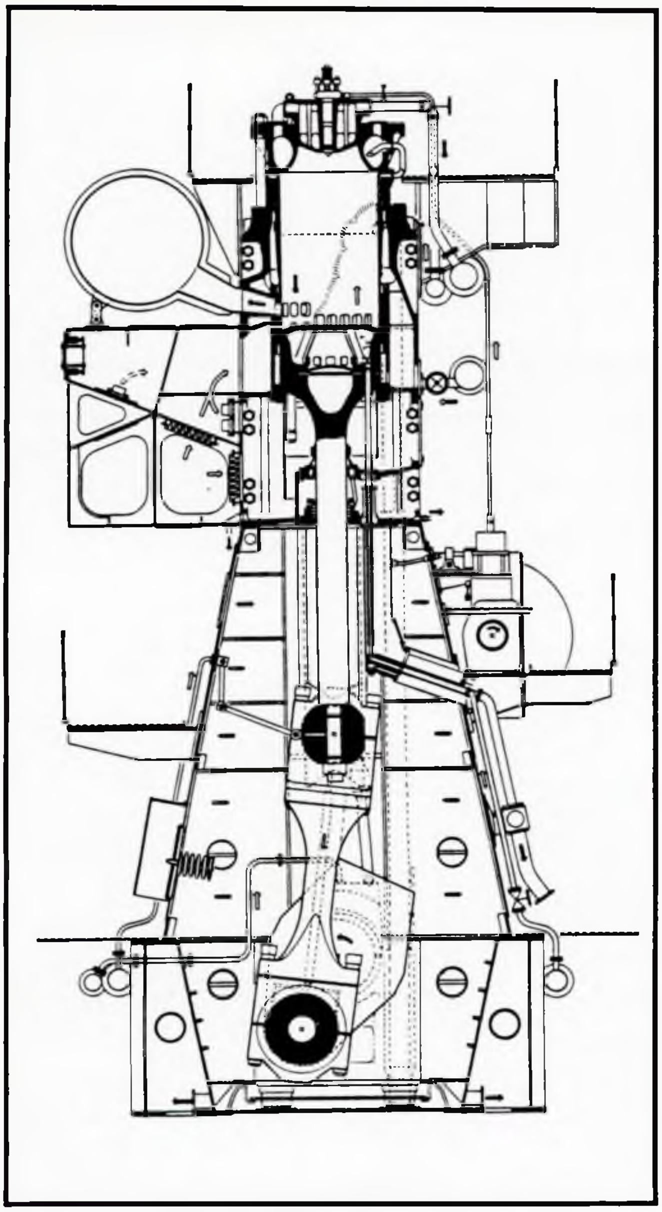

Fig. 12. RND engine section

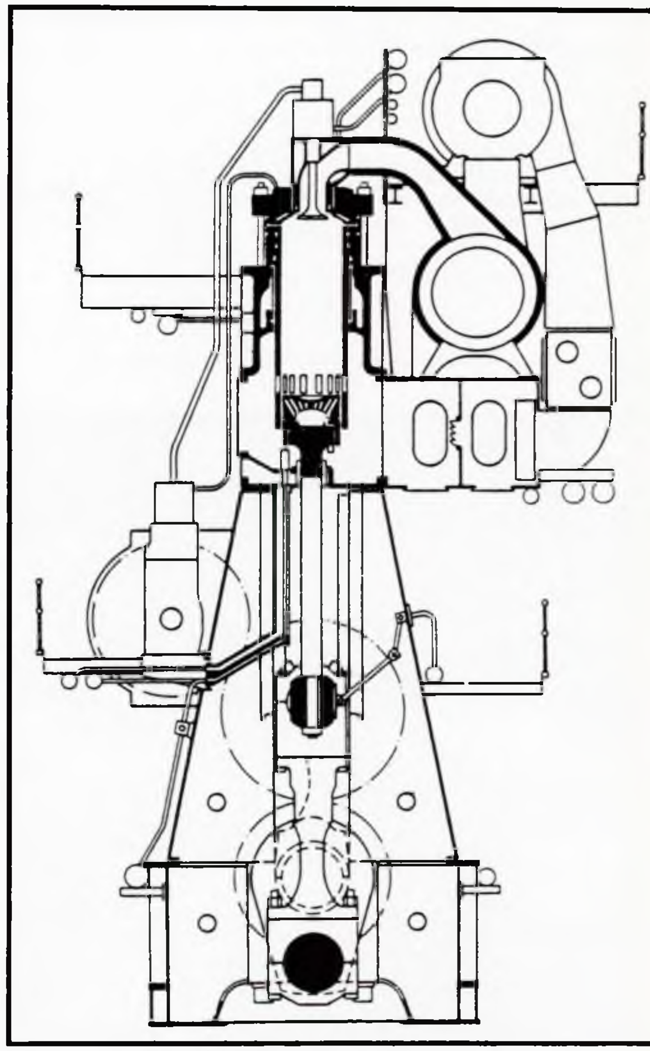

Fig. 13. RTA engine cross-section

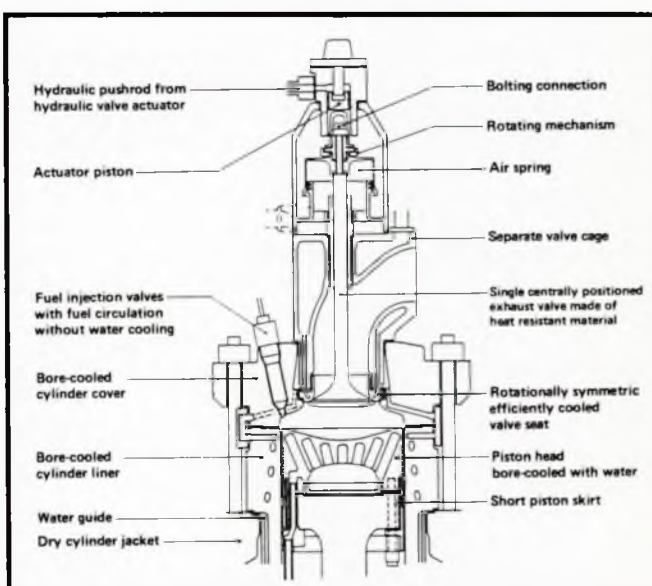

Fig. 14. RTA cylinder head, liner and piston

peared, the covering of the exhaust ports during the compression stroke being achieved by a slightly lengthened piston skirt. Under-piston supercharge and loop scavenging were retained but with double separation between the under-piston space and crankcase thus minimizing cross-contamination and also permitting egress to atmosphere for any piston cooling water leakage (Fig. 12).

The original RND engine retained the 2-part cylinder head but when the RND-M engine was introduced in 1976 a solid, forged steel, fully bore-cooled cylinder head was adopted. The upper part of the cylinder liner, from which the flame ring was now removed, was also fully bore-cooled.

In 1979 long-stroke RL engines were introduced with a stroke/bore ratio increased from 1.67 to 2.1 to permit lower propeller speeds and further improvements in specific consumption.

Finally in 1982 it became apparent that the limits of development of the loop-scavenged engine had been reached and in a dramatic move achieved in a remarkably short design time Sulzer introduced the RTA engine with uniflow scavenge through a single exhaust valve in the head (Fig. 13). The stroke/bore ratio was further increased to 3.45, the largest engine having a stroke of no less than 2.9 m. The piston itself was now bore-cooled so that the entire combustion space operated under the most favourable thermal stress conditions and oil cooling was reverted to (Fig. 14). Even higher turbocharger efficien-

Table 5. Engine specifications

	RD *1963*	*RND* *1968*	*RND–M* *1976*	*RLB* *1979*	*RTA* *1982*
Max. bore (mm)	900	1050	900	900	840
Stroke/bore ratio	1.72	1.52	1.67	2.11	3.45
Max. piston speed (m/s)	6.3	6.48	6.48	6.4	7.45
Minimum rev./min	120	122	122	90	56
mep (kg/cm²)	8.65	10.6	12.3	14.3	16.6
Max. pressure (kg/cm²)	76	84	94	118	125
hp/cylinder	2300	3000	3400	3940	4750
Specific consumption (g/hp/h)	155	155	144	133	116

Fig. 15. The world's largest diesel engine

cies have permitted elimination of under-piston pumping and more recently have led to the so-called efficiency booster system whereby power generated in a second-stage turbocharger is fed back to the engine crankcase, so reducing specific consumption by a further 3%.

This remarkable development story over a period of 20 years has been made possible by continuous and determined research to obtain improved performance. Research to optimize the scavenge process has been most thorough, as well as research into the injection and combustion process both to improve fuel economy and to permit the burning of steadily worsening fuels. Variable injection timing was introduced in 1981.

Recent published information also includes research undertaken to improve piston running conditions by means of:

1. boron alloyed die cast liners to minimize wear and improve fatigue strength and ductility;
2. control of cylinder wall temperatures at all loads;
3. multi-level cylinder lubrication;
4. improved surface finish of liners and ring surfaces and profiles.

Such developments are now pointing to 2 year overhaul periods for all main components of the engine.

The culmination of this development was the construction recently of the world's largest diesel engine, a 12RTA84 engine developing 57,000 hp – no less than 4750 hp/cylinder – built in Korea (Fig. 15).

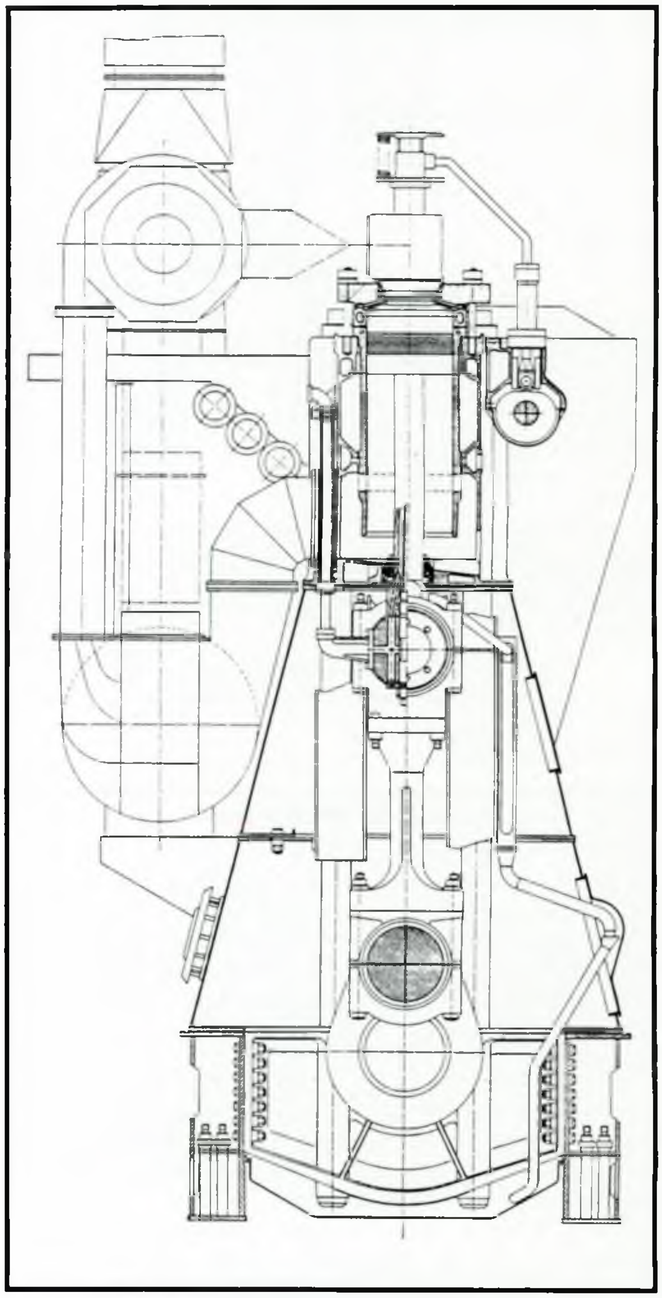

Fig. 16. Sectional view of KGF engine

Power output apart, however, the performance improvement over all can be seen to have been continuous and dramatic over the entire period of our review (Table 5).

The MAN–B&W story

The performance development of the B&W engine over 35 years has been no less dramatic than the story just reviewed, although it can be related in rather shorter form since it did not involve a fundamental change in engine type but rather continuous and systematic improvement of the uniflow engine which has been the standard B&W form throughout this period.

In 1963 the production engine was the VT2BF built with bores of 620, 740 and 840 mm, a stroke/bore ratio of 2.1–2.3 and operating at a bmep of 8.6 kg/cm². In 1968 the design was upgraded to form the KEF range with the same bore and stroke but operating at a bmep of 9.6 kg/cm².

In 1973 the KGF engine was introduced with many design modifications (Fig. 16). The crankshaft was now semi-built with cast-steel crank throws, a deep section welded bedplate incorporating cast-iron cross girders; steel-backed shell bearings were introduced in the crosshead, a forged steel cylinder head with radial cooling bores was mounted on a forged annular cooled ring and the hydraulically operated exhaust valve was introduced with welded stellite seats. The engine was pulse-charged with an auxiliary fan when required.

By 1975 a 12KGF90 engine was operating to the parameters shown in Table 6.

Table 6. 12KGF90 engine operating parameters

Bore	900 mm
Stroke	1800 mm
Stroke/bore ratio	2.0
Minimum rev./min	110
hp/cylinder	3100
Piston speed	6.6 m/s
mep	11.6 kg/cm²
Specific consumption	152 g/hp/h
Weight	1135 t (30.5 kg/hp)
Length	22.52 m

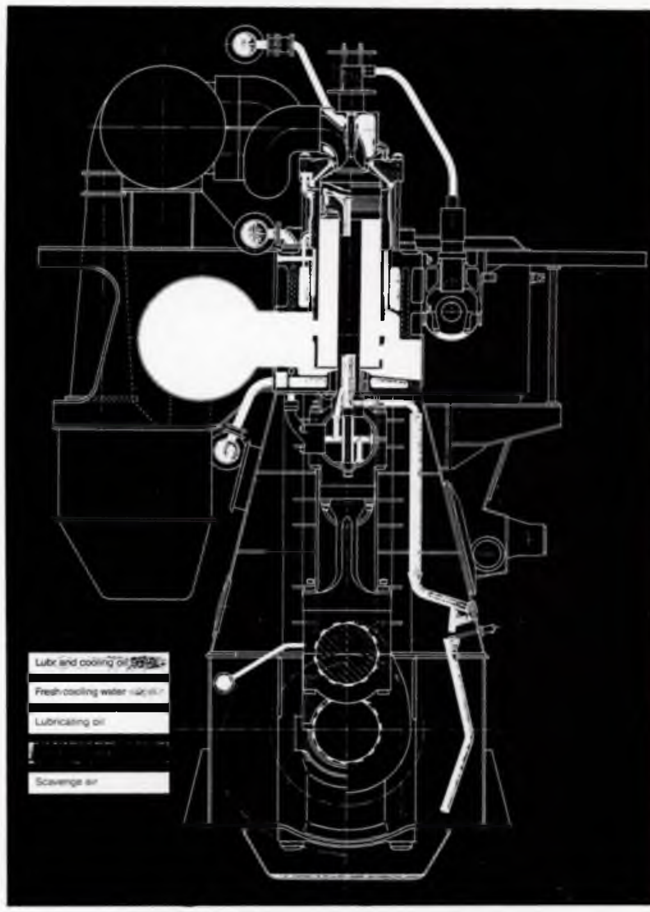

Fig. 17. LGF engine cross-section

A very different engine was produced in 1978 designated LGF with 22% increased stroke to give a stroke/bore ratio of 2.4 (Fig. 17). This engine was constant-pressure supercharged, again with an auxiliary electric blower, and a supercharge pressure of 3.1 kg/cm². Steel-backed white metal-lined bearing shells were fitted in the main and bottom end bearings and steel-backed 4% tin alloy-lined bearing shells were fitted in the crosshead bearing. The power per cylinder increased by almost 50% to 4575 hp, bmep increased by 35% to 15 kg/cm² and specific consumption reduced by 16% to 128 g/hp/h – certainly a much higher rating and lower fuel consumption than had been achieved up to that time.

Finally the super-long-stroke engine emerged in 1982 with the LMC engine having a stroke/bore ratio of 3.82 and a piston speed of 7.6–7.8 m/s (Fig. 18). Now the results of continuous research and development were very evident in many features of the engine.

Bedplate –	Welded cross-girders with cast-steel bearing supports.
Crankshaft –	Narrow gap submerged arc welded.
Piston –	Chrome molybdenum cast-steel head with hard chrome-plated ring grooves. Ring pack reduced from 5 to 4 rings. Phosphatized surface of skirt.

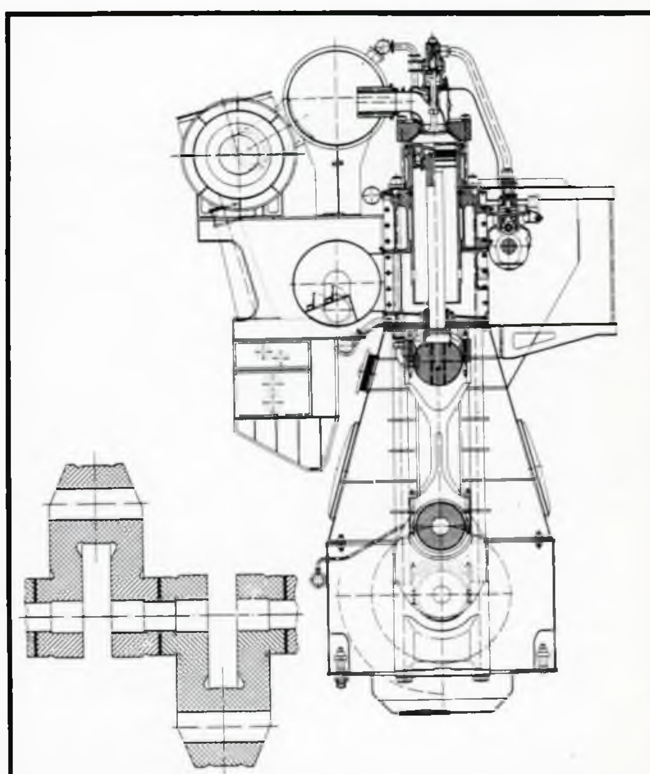

Fig. 18. LMC engine cross-section

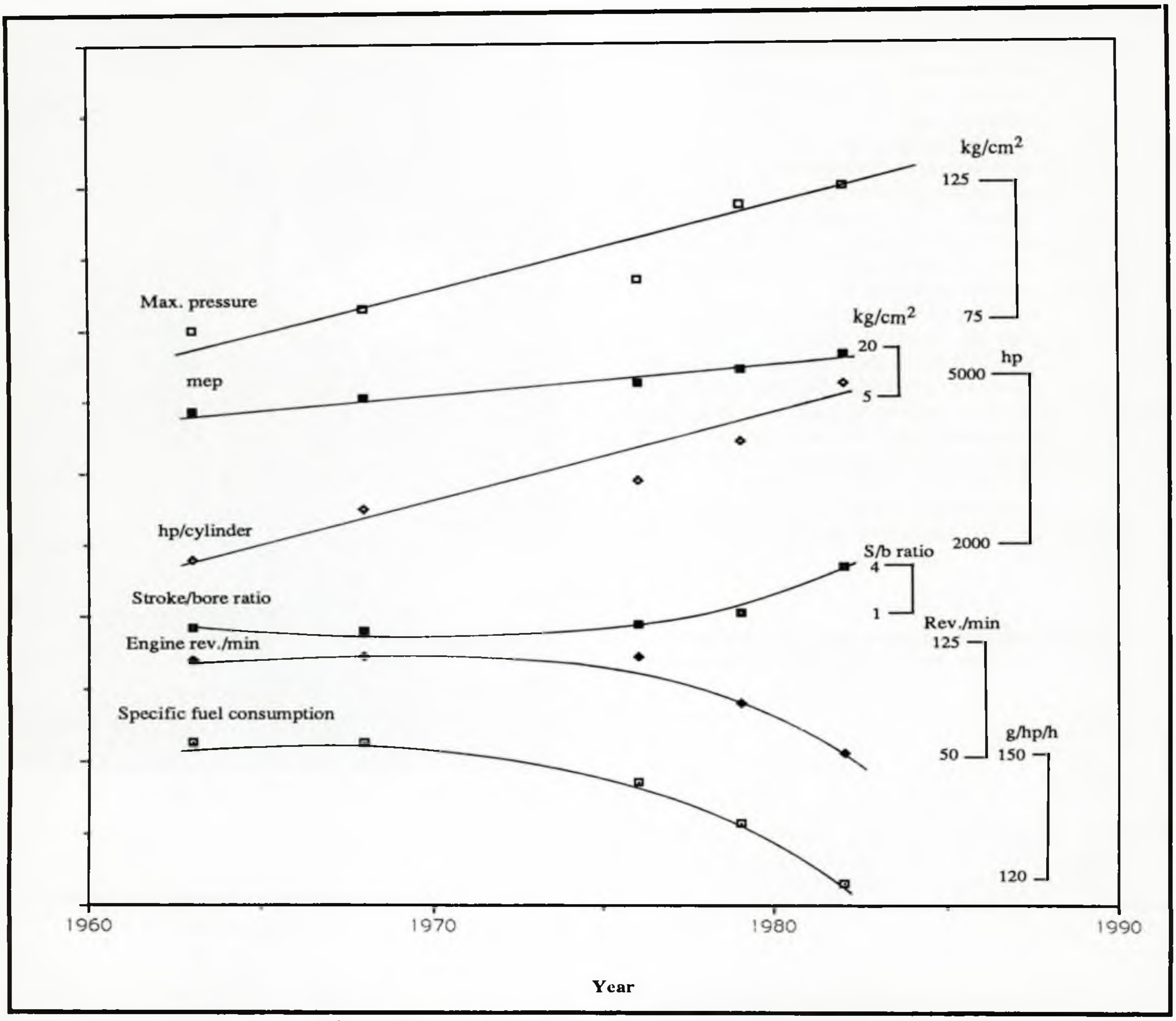

Fig. 19. Development of 2-stroke engines over 20 years

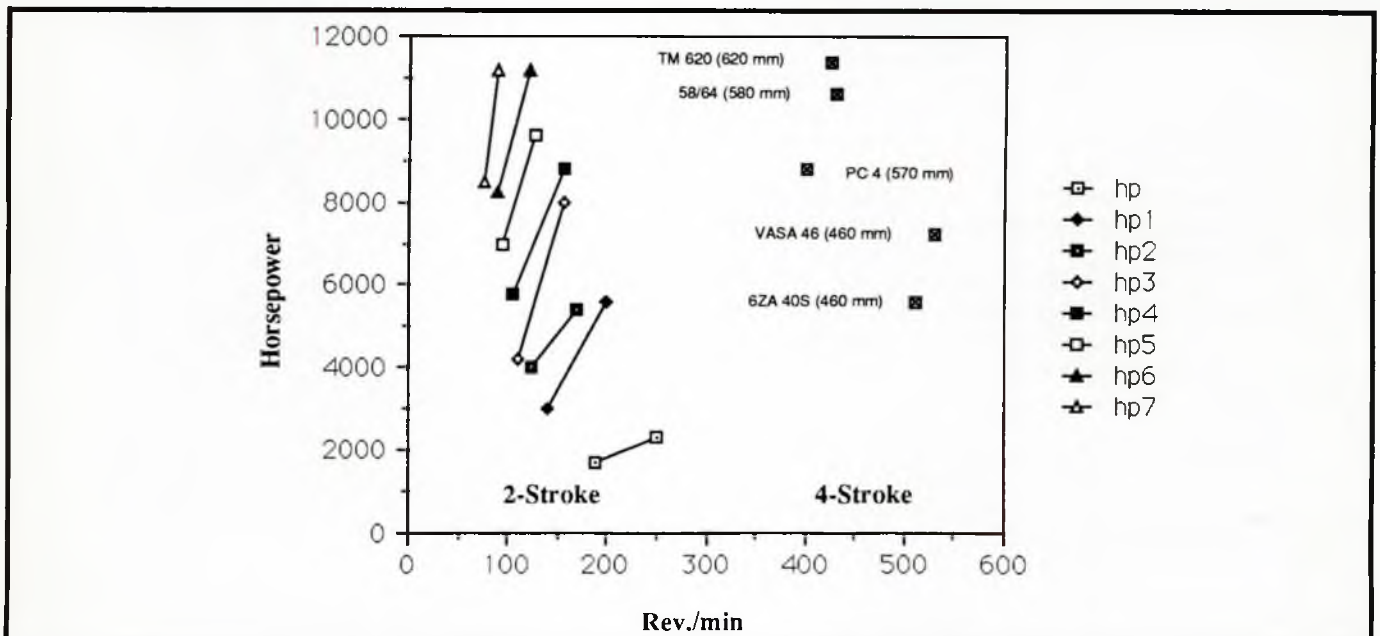

Fig. 20. A comparison of 2- and 4-stroke engines

Table 7. Characteristics of the 12L90MC engine

Bore	900 mm
Stroke	2916 mm
Stroke/bore ratio	3.24
Minimum rev./min	58
hp/cylinder	5241
Piston speed	7.58 m/s
Max. pressure	112 kg/cm^2
bmep	16.2 kg/cm^2
Specific consumption	119 g/hp/h
Weight	1835 t (32.76 kg/hp)
Length	21.09 m

Piston rod – Re-designed piston attachment. Hardened surface of rod. Scraper ring pressure increased to 4 kg/cm^2.
Crosshead – One-piece bottom bearing shell. Floating guide shoes.
Fuel pumps – Variable injection timing.
Exhaust valve – Cooling of valve seat insert.
Reversing – Self-locking fuel cam rollers.

The remarkable improvement in performance which really took place over a period of 10 years between 1973 and 1983 is mainly attributable to the development of sophisticated computerized models which have embraced the basic thermodynamics of the engine, heat release and combustion processes, system dynamics and the fundamental design of engine components. These techniques have given rise to increased power output, improved fuel consumption, reduced manufacturing cost and greater reliability, while the specific weight of the largest engine remains in the 33 kg/hp range, notwithstanding the adoption of the very long stroke. Latest published characteristics of the 12L90MC engine are given in Table 7.

The general development of the low-speed 2-stroke engine can be seen in Fig. 19 and adds up to 75% increase in power per cylinder and 25% reduction in specific fuel consumption.

In our review of this progress we have tended to concentrate on the ever-increasing horsepower availability. However an equally fascinating development of the low-speed 2-stroke engine has been in the smaller engine sizes and power ranges. Fig. 20 shows a comparison of currently available 6-cylinder 2-stroke and 4-stroke engines in the 3000–11,000 hp range which shows the spread of the low-speed 2-stroke engine into the lower-power territory which was hitherto exclusively occupied by 4-stroke engines. It really is now a question of whether a designer can accommodate 1–2 m more headroom – otherwise a medium-speed engine with gearbox will have to be fitted.

The most recent aspect of this trend was the introduction in 1987 of the so-called mini-bore S26MC engine offered over a speed of 188–250 rev./min (Fig. 21). This 260 mm (10.24 inch) bore engine is designed in 4 to 8 cylinders developing 490 hp/cylinder at the same specific consumption as the latest medium-speed engines but with conceivably a wider range of fuel digestability and certainly lower lubricating oil consumption.

The end of the story has not of course yet been written – but already the shipowner enjoys a full freedom of choice.

Electrical propulsion – a revised role

Electrical propulsion of merchant vessels has a long pedigree but with few exceptions, of which the wartime-built T2 tanker was the most remarkable, application has been restricted to particular types of vessels where the genre offered advan-

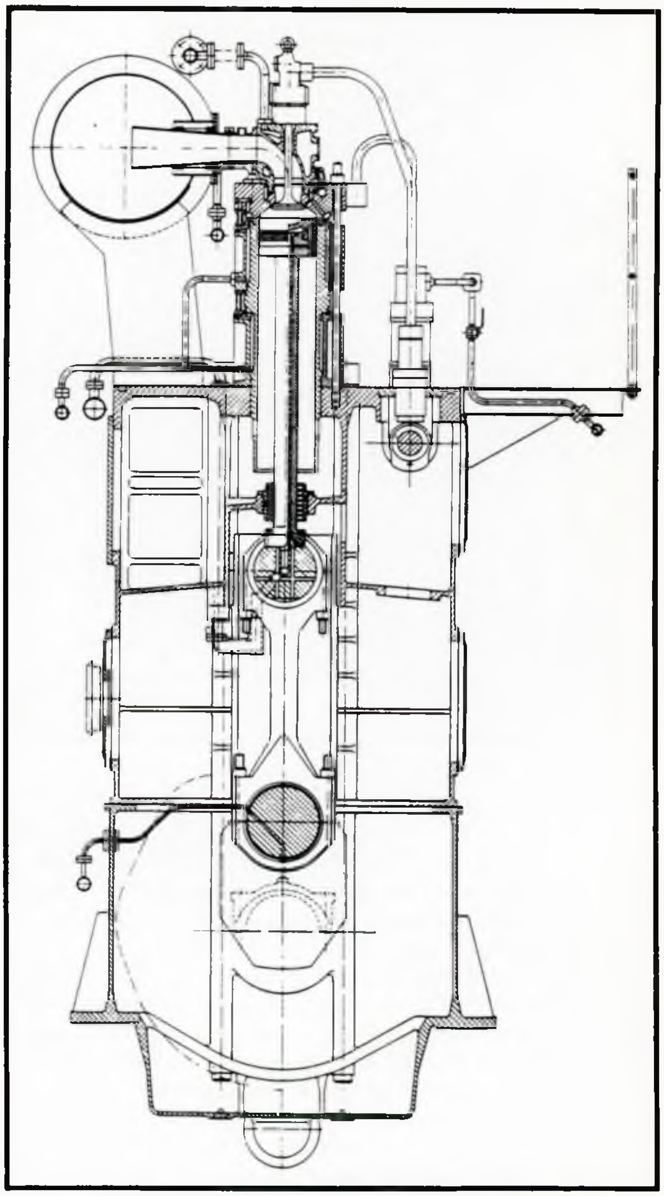

Fig. 21. S26MC engine cross-section

tages which could override the penalties of higher installed cost and weight and lower overall efficiency which otherwise applied.

Thus vessels requiring high torque or low revolutions such as icebreakers and cable layers have been natural candidates. Quietness is an attractive feature in research vessels and passenger ships and ships having a high power requirement when stopped such as self-positioning support and heavy lift crane ships can utilize electrically generated power for both propulsion and working loads.

Passenger cruise ships with a very high hotel load and requirements of passenger comfort, free of noise and vibration, derive perhaps the most obvious benefits from electrical propulsion and, not surprisingly with present day fuel costs, it is the diesel electric installation which attracts attention, special installation precautions being taken to minimize noise and machinery-induced vibration by means of resilient mountings.

Two recent designs are of particular interest and have been widely discussed.

Fig. 22. *QE2* propulsion motor

The recent re-engining of Cunard's *QE2*[4] with nine 9-cylinder medium-speed engines developing 120,000 hp on two controllable-pitch propellers represents the highest power installed for electrical propulsion to date with a 10 kV, 44 MW synchronous motor driving each shaft (Fig. 22). Two propeller speeds of 144 and 72 rev./min are achieved by means of a synchro-converter, the engines running at a constant speed of 400 rev./min to provide 60 Hz at the busbars and ship speed and reversal being effected by variation of propeller pitch. The large ship service load is provided through 11 MVA transformers. The performance of this propulsion system will be followed with considerable interest.

A quite different solution has been adopted for the lower-powered Sitmar cruise vessels soon to join the P&O fleet. Fixed-pitch propellers have been preferred, each driven by a 6.6 kV, 12 MW motor, the speed and direction of which are controlled by thyristor synchro-converters. Four 8-cylinder medium-speed diesel engines run at a constant speed of 400 rev./min to provide 60 Hz at the busbars. Although very much smaller than the *QE2* installation the flexibility, reliability and overall efficiency of this propulsion system will also excite interest.

The cruise ship and large ferry markets are the most buoyant section of the shipping business today and week by week we hear of dramatic proposals for newbuildings. Electrical propulsion will be a strong contender for many of these ships and further variations on the method of speed control and reversal will no doubt emerge.

A recent design of unusual form was adopted for the ill-fated contract for so-called Superflex car/passenger ferries. These very simple double-ended ferries of 1300 tons deadweight are fitted with ten containerized generating sets of 275 kW each to supply the four thrusters which propel the ship in either direction. High-speed engines burning diesel oil are used and it is difficult to see how such an approach could be attractive to other than small short-range vessels of this type.

For the future superconductive electrical machines are now undergoing intensive development and may succeed in bridging the efficiency gap which is currently inherent in electrical propulsion. From Japan also we hear excited noises about the application of linear motor technology to ship propulsion. Electrical propulsion may indeed still have some surprises in store.

A GLIMPSE AT THE FUTURE

It is not the purpose of this paper to look specifically at the future – that task must be left to others.

However, a review of the past 25 years does appear to provide some insight into the direction of future development.

Certainly the availability, quality and cost of fuel will continue to be a controlling factor as indeed it always has been. Availability of fuel oil does now seem to be assured for the next 25 years. Quality seems guaranteed to get steadily worse. As for price, it will always seem expensive and the overriding need for economy will not disappear.

It is, I suppose, just possible that exploitation of the enormous reserves of natural gas worldwide may lead to its further deployment in liquefied form in which case the gas turbine may indeed make a re-appearance.

Coal is plentiful enough but highly inconvenient and the worldwide infrastructure for coal bunkering no longer exists and is unlikely to reappear in the foreseeable future.

Nuclear power has made no impact on the merchant scene in the past 25 years and seems unlikely to do so in the next 25 years. It was the conventional wisdom that high oil cost would enable nuclear power to overcome its high installed cost and weight penalty; however two oil crises have not borne that out. Environmental resistance to nuclear-propelled merchant ships is a formidable barrier and economic uncertainties seem to militate against commercial decisions having a long time-scale which the high cost of a nuclear installation imposes.

The advantage of the diesel engine as by far the most efficient thermodynamic machine known would seem to ensure its supremacy far into the future. We cannot expect to see a repetition of the amazing thermal efficiency gains of the past 10 years, but step by step improvements will emerge particularly in conjunction with compound turbocharging.

The other field for development will undoubtedly lie in systems design. Space has not permitted treatment of this subject in this paper which has concentrated on prime movers but great scope still remains for rationalizing the approach to auxiliary power generation and utilization on board. The accurate matching and regulation of auxiliary services on board can contribute greatly to simplification, improved reliability, reduced maintenance and overall efficiency.

Members of this Institute have played a leading role in all the fields of development which we have been able to review over the past 25 years. A new generation of marine engineers will enjoy a similarly exciting and fruitful field for their endeavours into the next century and our Institute will undoubtedly be there to offer them full support. We wish them well in the years ahead.

REFERENCES

1. Professor I. Jung, The Marine Turbine (3 volumes), National Maritime Museum Trustees (1982–1986).
2. Professor B. Crossland, *Trans.I.Mar.E.*, vol. 98, paper no. 19 (1986).
3. J. Neumann & J. Carr, *Trans.I.Mar.E.*, vol. 79, paper no. 5 (1967).
4. IMAS 88, *Trans.I.Mar.E.(C)*, vol. 100, conf. 1, papers 16–20 (1988).