Jørn Dragsted

THE FIRST 50 YEARS OF TURBOCHARGED 2-STROKE, CROSSHEAD, MARINE DIESEL ENGINES

Fig. 2 Scavenging systems in turbocharged, crosshead diesel engines (Author)

Jørn Dragsted

THE FIRST 50 YEARS OF TURBOCHARGED 2-STROKE, CROSSHEAD, MARINE DIESEL ENGINES

CONTENTS

INTRODUCTION

This story of diesel engine development is, apart from personal experience, very much based on information found in CIMAC congress papers, papers read in Engineering institutions and articles in the trade magazines, mainly The Motor Ship and not least the many publications from the engine designers in question. It hurts a little to cut the story down to the present extent, as so many valuable engineering ideas and in depth research results cannot be given credit.

It may be considered easy to judge the importance of ideas and the designs introduced in the past, but when going through this story it should be remembered **"that any technical solution is only a compromise of certain advantages and disadvantages, the original standard and tradition being of great influence"** as said by MAN's Professor Sörensen and Dr. Smidt.

The selection of the content is made to cover both interested engine designers and manufacturers but also the engine users, who in great numbers, have served in the commercial fleets, down below.

SETTING THE SCENE

In October 1952 the M/T "Dorthe Mærsk" went to sea propelled by the first turbo-charged 2-stroke, crosshead diesel engine. For years, and even during the war engine designers had been calculating and testing the possibility of turbocharging 2-stroke engines to obtain increased power intensity, but Burmeister and Wain (B&W) came first with their uniflow scavenged crosshead engine, the 74VTBF160.

At that time no less than 9 designers of large 2-stroke engines were fiercely competing in the shipbuilding market to supply the yards with powerful and reliable engines, dominated by B&W in Copenhagen, Denmark, MAN in Augsburg, Germany and Sulzer in Winterthur, Switzerland. The other players were: Doxford in Sunderland, Great Britain, Stork in Hengelo and Werkspoor in Amsterdam, Holland, Götaverken in Gothenburg, Sweden, Fiat in Turin, Italy and far from the European scene Mitsubishi in Nagasaki, Japan.

Apart from production taking place at the company's own premises a substantial number of engines were produced under licence by big, mainly shipbuilding companies all over Europe and in Japan. This meant, that the companies that had strong allies serving the shipbuilding industry were well situated to gain from an increased shipping activity. The demand for ocean transport after the war was rising by about 10% per year, and so was demand for ships and engines rising. Vessel's dwt. was growing as well, and consequently the need for higher power for propulsion.

Turbocharging of 4-stroke engines was already well accepted by the market in the mid 1930's, but it is more complicated to turbocharge the 2-stroke engines, as there is no separate exhaust and air intake stroke, but merely a short period of time around the bottom dead centre to empty/ push out the exhaust gas and charge the cylinder with fresh air.
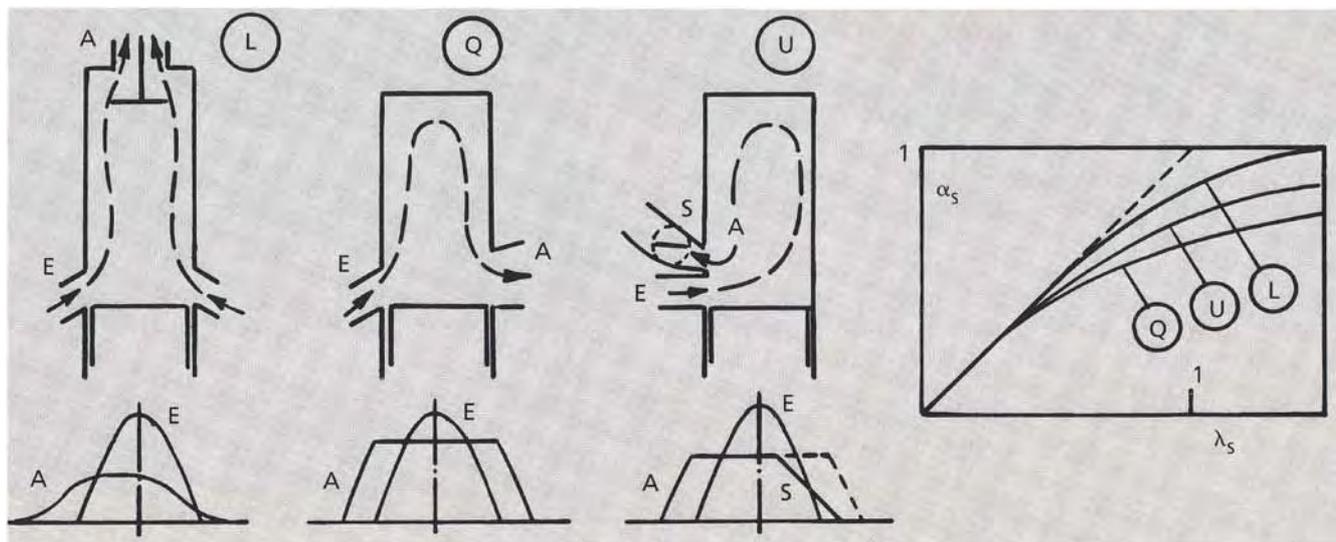
Fig. 1   The three scavenging systems: L Uniflow scavenging, Q Cross scavenging, U Loop scavenging. Time cross-sections (below) for inlet E, outlet A and possible slide valves S. Right attainable degree of purity in the cylinder plotted against scavenging air excess λ_s. (ABB [6])

The scavenging process of the engines designed by the mentioned companies were quite different, B&W, Doxford, Götaverken, Mitsubishi, Stork and Werkspoor used the uniflow principle, Fiat and Sulzer used cross scavenging and MAN preferred loop scavenging, Fig. 1, so it was not an easy common solution, that could be found.

This story is just about 2-stroke low speed crosshead diesel engines. Trunk engines made in parallel by the same designers are not included, as they disappear from the market early in the period considered, i.e. in the 1960's.

The history of turbocharger development has already been published by two highly professional writers and is consequently not included either.

The BBC/ABB turbocharger development is described in a comprehensive book "The BBC Turbocharger" by Dr. Ernst Jenny [6]. A more widely covering book including and comparing most of the used chargers for 2-stroke engines is written by Mr. K. Imakire of Mitsubishi [86].

In Fig. 2 (inside cover page) an overview is shown of the development of the scavenging systems used in the 2-stroke crosshead engines from the first engine types at sea and the following 50 years. Further the figure shows, when the various engines types developed by the designers were introduced (given by the position of the first letter), and when the companies for various reasons decided to cease operation. So, here is given the answer to "what" and "when" – hopefully the following will enlighten the reader in respect to "why".

I POWER & RELIABILITY

1. The first turbocharged engines and some later generations

Burmeister and Wain

Supercharging of B&W engines was previously done by rotating blowers according to the Roots principle, and the 74VTF160 long stroke engine was the first engine to be turbocharged. These single acting machines were designed to ease maintenance, that was rather time consuming on previous double acting engines, and had a diaphragm between crankcase and scavenging air belt to avoid contamination of the lubrication oil, which was important for the use of heavy fuel oil instead of diesel oil.

It was B&W tradition to apply uniflow scavenging, as the requirement to air supply was less demanding than in other systems. The crosshead engine with a poppet valve in the cylinder cover was thus a relatively easy starting point for a conversion from chain driven Roots blowers to turbocharging. The difficulty was the part load performance, or so it was generally believed. Actually B&W engineer's calculations as well as BBC's showed, that utilisation of the exhaust pulse when opening the valve would make the system work even at part load.

The 6 cylinder 74VTF160 shown in Fig. 3 was thus able to run just with the turbo blowers supplying air at all loads, a considerable advantage compared to many competing systems, as will be demonstrated later. The relative simplicity of the conversion is demonstrated in Fig. 4, however missing the increase of exhaust valves and scavenging air ports as well as the capacity of the fuel pumps. BBC turbochargers were applied to a start, but later Rateau, Napier chargers and chargers of B&W's own design were available. It was unacceptable for B&W to rely on just one supplier for such an important engine component, and furthermore to risk that the knowhow gained by sub suppliers would be spread to their other costumers i.e. B&W's competitors.

The engine structure shown on fig. 3 is all welded without staybolts. Alternatively the engines were designed in cast versions with staybolts – and this more traditional design was chosen for subsequent engine types having higher ratings and as a consequence of crack formation in the welded structures.

Engine power was increased from 920 to 1250 BHP/cyl by an increase of the mean effective pressure, (mep). And with the few modifications this was a win win situation for manufacturers, shipyards and engine users, as the fuel consumption was reduced – from 162 g/BHPh to 158 g/BHPh. B&W earned a fortune on the turbocharged engines, but sadly invested a lot of it in the shipyard rather than in the engine facilities.
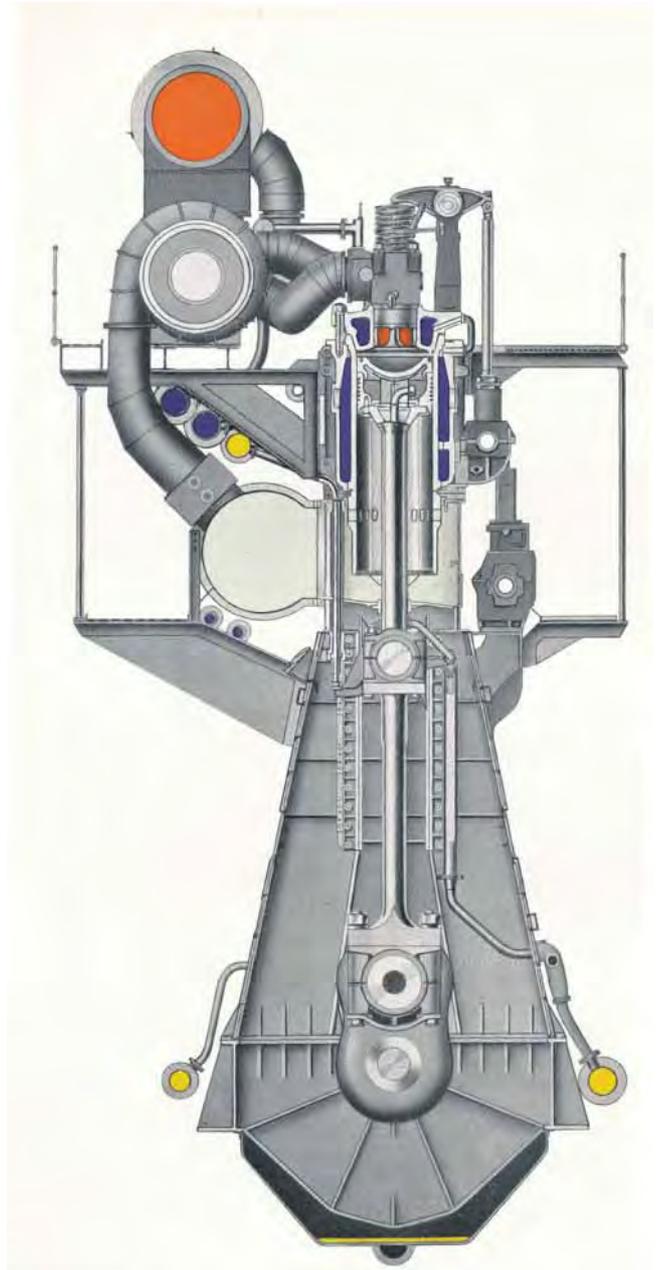

Fig. 3   Cross section of the first turbo-charged 2-stroke engine 74VTBF160, bore 740 mm, stroke 1600 mm, yielding 1250 BHP/cyl (MCR) at 115 rpm and Mep = 7 bar. (B&W)
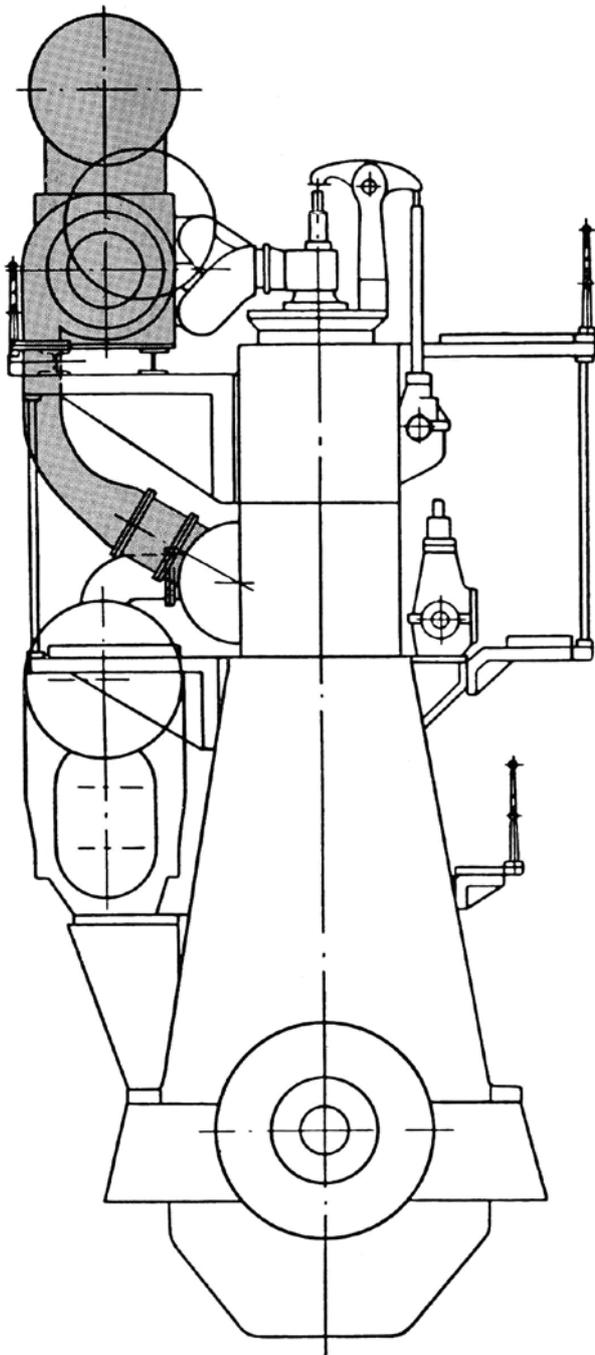
Fig. 4   The required modification of the standard unturbocharged 74VTF160-engine to provide turbocharging. (B&W)

The engine had two camshafts to simplify the reversing arrangement, which adds to cost and the exhaust valve and valve gear was also relatively costly to manufacture, but who cared knowing the overall gain. It was only with the following engine series, that the design was changed to encompass just one common camshaft. Of the other components the piston design with a screwed on piston crown was causing trouble by leaking oil and was later replaced with a shell type design with support of the crown by the ring belt, however this piston, which was also used in the following VT2BF engines, showed cracks in the transition between crown and the top land, and

only at the introduction of the later engine series K-EF and K-FF was a reliable solution found (see K98FF).

Reliability and maintenance are closely related, but especially the maintenance "burden" has undergone considerable development. Søren Hansen, B&W chief development engineer, reported at the CIMAC congress in 1955 [7], that the cylinder wear was just as good as on the non turbocharged engines, i.e. 0.2 to 0.4 mm/1000h. In those days seagoing engineers were used to frequent maintenance, so the result was acceptable. Today's acceptance level would be less than 1/5!

The user's comment of that time was generally very positive – it was a good engine.

B&W could, in 1959, present a high pressure turbocharged 84-bore engine as a start to a general uprating of the VTBF-engines to high pressure turbocharging – the VT2BF-engine series. These engines followed the same concept as the 74VTBF160 engine that definitely was B&W's "bread and butter" design. It was easy in those days for the sales department of B&W, so maybe they fell asleep – and at the end of 1964 Sulzer managed to beat B&W on market share. It hurt the pride of B&W engineers, but more seriously it hurt the top management, that was well aware of the advantage of being number one. So, in 1965 it was decided to launch uprated versions of the VT2BF series of engines – the K-EF types. This decision was an economical necessity, but B&W were technically unprepared, so the design was nothing to be especially proud of.

For turbocharged engines cooling of the compressed air is a necessity meaning that air coolers are required after the turbocharger. However, the ever increasing scavenging pressure means that condensed water need to be removed after the air cooler as otherwise water droplets acting as projectiles when entering the cylinders through the ports can lead to scuffing. In the first turbocharged engines condensation is avoided by increasing the scavenging temperature, later highly rated engines are equipped with so-called water mist catchers.

OPPOSED PISTON ENGINES

B&W had, in parallel with their poppet valve engines, also a series of opposed piston (or coverless) engines in the portfolio. During the war Harland & Wolf in Belfast had independently developed the opposed piston type under C.C.Pounder's leadership, and it obtained just as Doxford's opposed piston engine success in the UK. Consequently H&W put heavy pressure on B&W to design opposed piston engines.

The first turbocharged opposed piston engine to be delivered from B&W in 1955 was a 5 cylinder 50VTBF110/40.

A cross section of the lager sister 75VTBF150/50 is shown in Fig. 5. The total stroke is 2000 mm, but the upper exhaust stroke only 500 mm. A later execution had a combined stroke of 2300 mm. The forces from the upper piston are transferred to eccentrics on the crank webs. This resulted in a very stiff crankshaft allowing B&W to build the engines with 5 to 10 cylinders, whereas the competing opposed piston design

could only build their engines up to 6 cylinders (see later). The engine is pulse pressure charged and has no requirement for air-blower or pump to assist at low loads.

One different aspect of the Harland and Wolf built engines was that they used an Archaouloff-type fuel injection pump. In this pump compression pressure from the engine cylinder was connected to the top of the fuel pump where there was a piston, approximately 10 times larger in cross-sectional area than the fuel pump attached to it. Although this meant that a separate supply of cylinder oil had to be fed to the upper part of the pump, it negated the need for a camshaft so saving on building costs.

The B&W opposed piston engine is gradually phased out during the 60's as it was more costly to produce than the poppet valve engine. A longer upper stroke might have justified the expensive running gear of the exhaust piston to a certain degree.

Burmeister & Wain's competitors were taken by surprise and Fig.2 really illustrates this, as it takes more than two years for most of the competitors, before they launch their new turbocharged engines. At B&W there are speculations on the sort of solution that they sooner or later will meet in the market place. Technical director Haakon Andresen goes as far as to present the expected moves by all the competitors at meeting in the Norwegian Engineering Society in Oslo in April 1953, it could be wishful thinking, but he did not mention anybody "copying" the B&W system: **Uniflow, pulse pressure charging without additional air supply**. He forgets to include Mitsubishi, and he is mistaken as far as Stork is concerned [8].
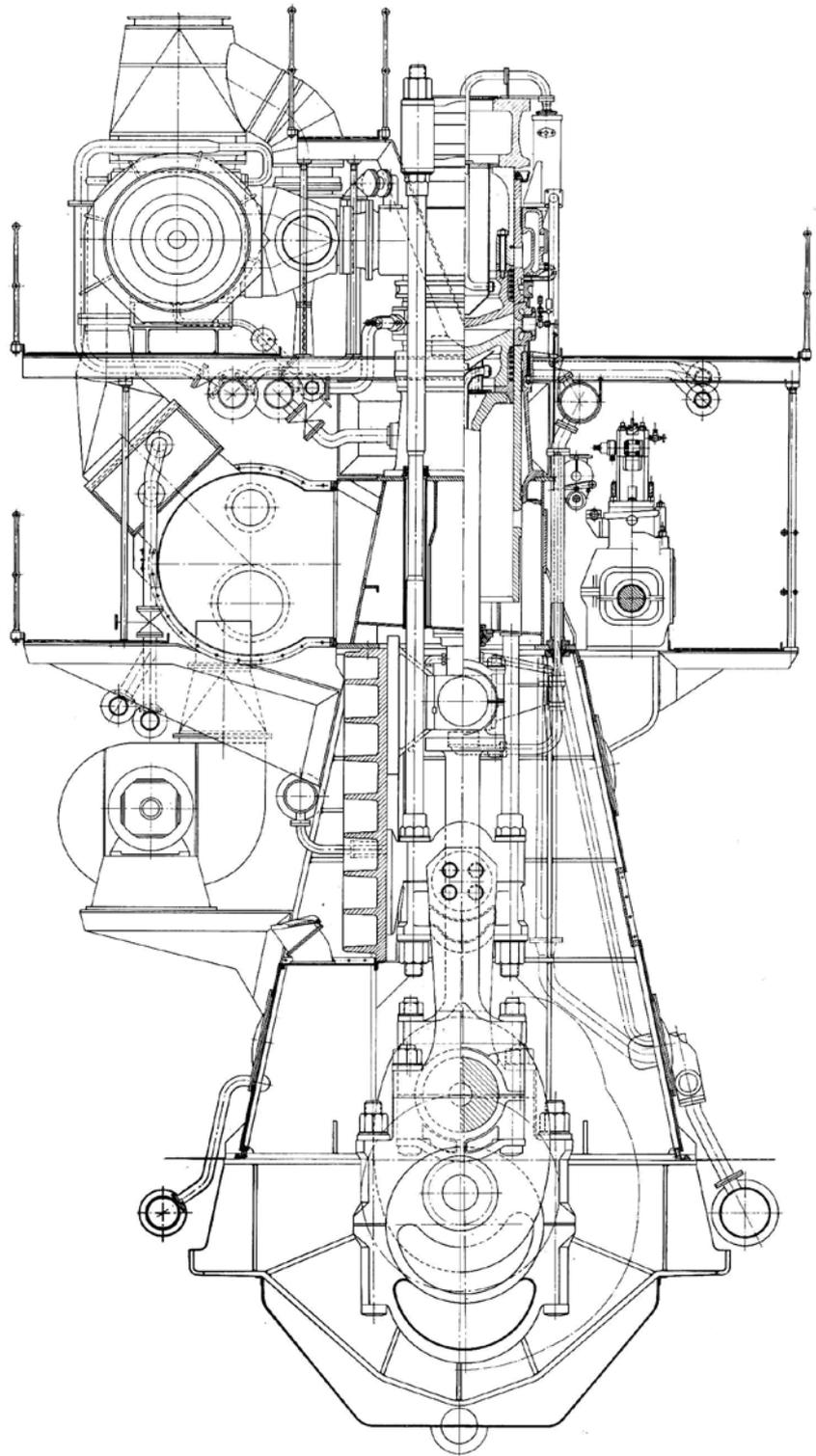
Fig. 5   B&W opposed piston engine 75VTBF150/50, bore 750 mm, combined stroke 2000 mm, yielding 1650 BHP/cyl (MCR) at 120 rpm and Mep = 7 bar. (B&W)

Werkspoor

Werkspoor are the first to follow B&W. The Werkspoor-Lugt 12-cylinder KEBS68/125 is not their first turbocharged engine, but the first real crosshead engine with turbocharging [9]. The engine was installed in the passenger-cargo liner "Prins Willem van Oranje" in 1953. The "inventor" of the basic engine concept Mr. G.J.Lugt had passed away before the turbocharging was introduced, but many of his ideas are still maintained in the turbocharged version of the Lugt engine, a cross section of which is shown in Fig.6, with the turbocharger arrangement shown in Fig. 7.

Fig. 6   Cross section of the first turbocharged, crosshead Werkspoor-Lugt engine, KEBS68/125, bore 680 mm, stroke 1250 mm, yielding 800 BHP/cyl at 125 rpm. Constant pressure charged with engine driven double-acting pumps and a special exhaust valve gear. (MS [10]) $\longrightarrow$
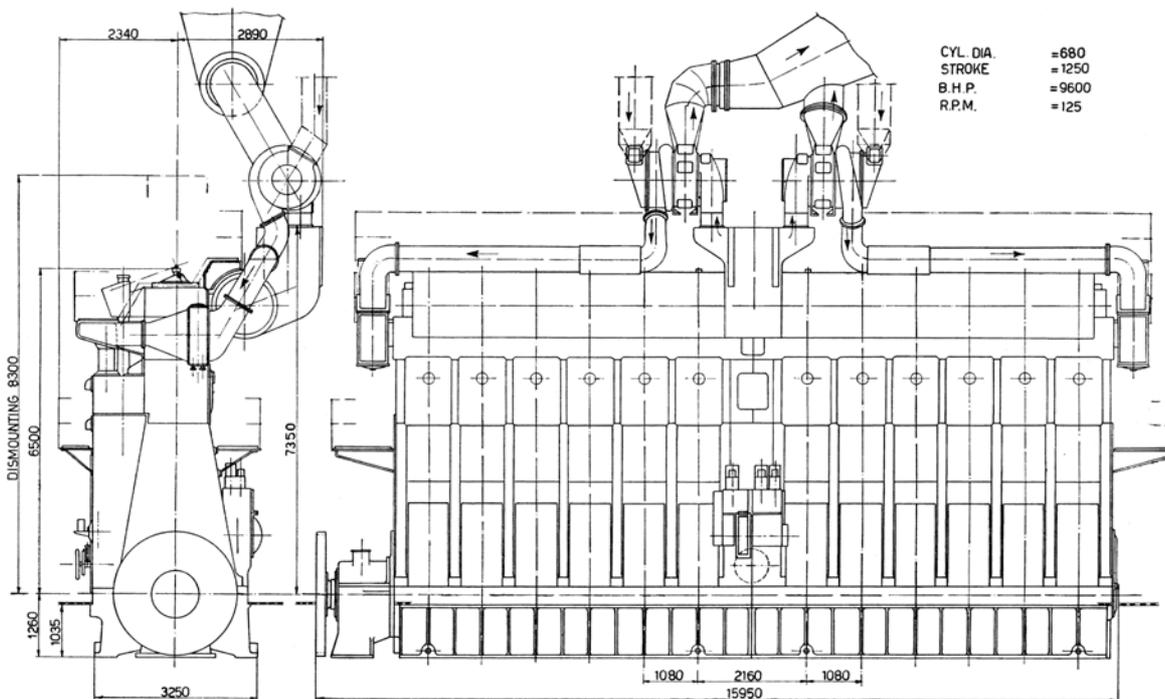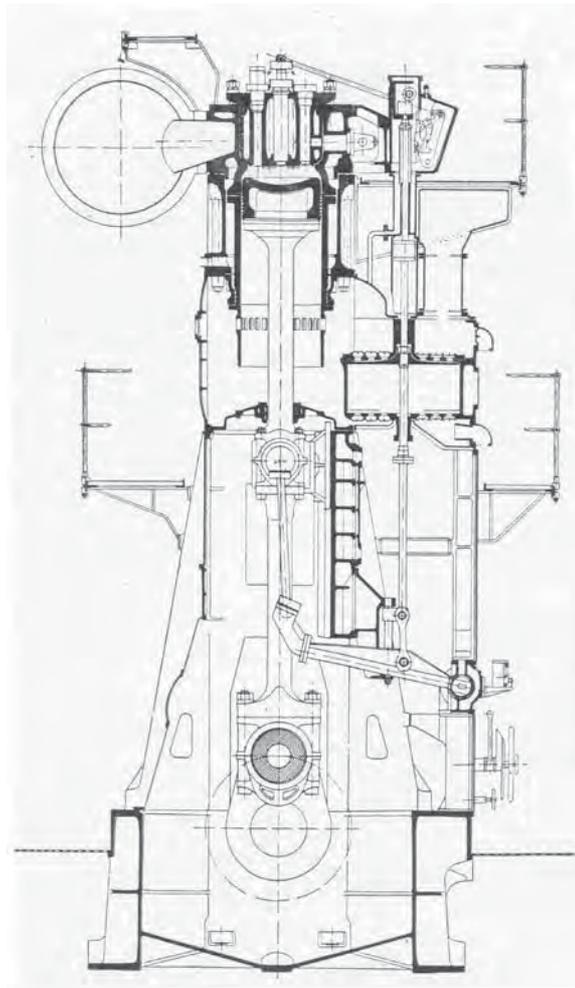
Fig. 7   12 cylinder KEBS68/125, main dimensions and turbocharging arrangement. (MS [10])

The 12-cylinder engine developed 9600 BHP at 125 rpm and had a bore of 680 mm and a stroke of 1250 mm. The supercharging arrangement is a constant pressure system and consist of two BBC VTR630 chargers arranged in series with the upper sections of the 12 engine-driven reciprocating double-acting scavenge pumps, but at full load the chargers will deliver directly to the scavenging receiver, the lower sides of the pumps deliver directly to the scavenging receiver and draw air from the engine room, see Fig. 8 [10].
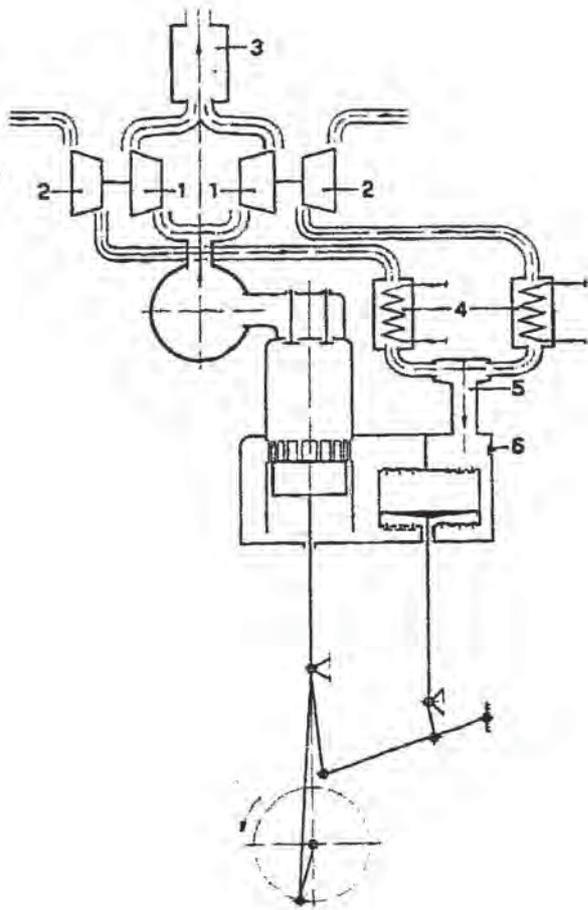
Fig. 8 Diagram of the Werkspoor-Lugt constant pressure scavenging principle. (CIMAC [9])

The engine has 4 exhaust valves in cages and a very special actuating gear operated from an extension of the scavenging pump piston rod by means of cams and rollers and pull rods, each rod operating a pair of exhaust valves, as seen in Fig. 6 and 9. A reporter from "The Engineer" found: "the working of this gear to be very smooth and practically silent, not unlike the valve gear of a modern horizontal steam engine" [11]. There is no need for a camshaft, the reason for this is also attributed to the fact that instead of having one fuel pump for each cylinder, the fuel pump is contained in a completely separate unit driven from the crankshaft through gearing between cylinder 6 and 7.
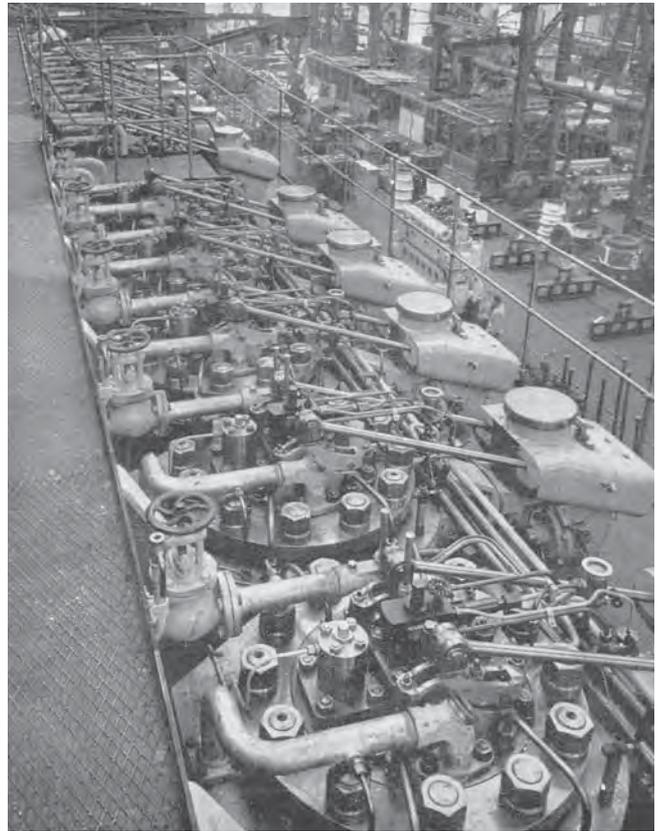
Fig. 9 KEBS68/125 engine top. Note the special exhaust valve gear. (MS [10])

The engine structure is of cast iron and the bedplate carries short columns supporting the scavenging pump beam which in turn carry the cylinder frame. The cylinders are in two parts to facilitate exchange of piston rings without pulling the piston. This feature adds to the height and weight of the engine, but with the wear rates of the piston rings normally ten times as high as the cylinder liner wear, it was perhaps a sensible solution.

Like B&W Werkspoor focus in their first published service records on cylinder liner wear. The average liner wear running on heavy fuel is between 0.3 and 0.45 mm/1000h and of corrosive nature, very much the same as found in the B&W 74VTBF160 engine. Werkspoor also found some occasional burnings of the valve faces after 1.200 running hours, but were able to improve the lifetime by the introduction of stellited valve faces.

In 1954 Werkspoor and Stork merge and new Stork-Werkspoor engines are now developed based mainly on the Stork HOTLo concept [12] (see next chapter). The époque of the turbocharged Werkspoor-Lugt cross-head engine was just about one year – and the Werkspoor-Lugt engine design gradually disappeared.

Stork

The Stork engine is in scavenging respect like the B&W engine, but as Werkspoor, it also applies 4 exhaust valves in the cylinder cover. Indeed the obvious system choice for Stork's turbocharged engines is also Uniflow, pulse pressure charging without additional air supply at part load. Stork presents in 1954 their 8 cylinder HOTLo75/150, Fig. 10, where they chose to apply one BBC VTR500-charger per two cylinders in order to obtain as short exhaust pipe connections as possible and in this way to get the most out of the kinetic energy in the exhaust pulses Fig. 11. This system is supported by arranging a crankshaft with adjacent pairs of opposed cranks so that the pulses for one turbocharger follow at equal time intervals. Mr. Wieberdink mentioned in his CIMAC paper at the congress in den Haag 1955 [13] – that they are aware that the intervals could be minimised to 120º as wanted for a 6 cylinder engine, and further the fact that they have two exhaust ducts per cylinder has been a helpful contribution for the 5 and 7 cylinder engines, Fig. 12. The mean effective pressure of the engine was 6 bar at 100% load.
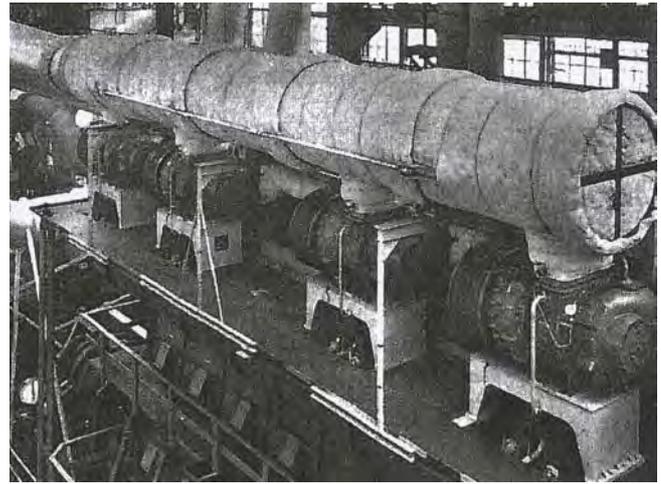
Fig. 11   8 cylinder HOTLo75/150 engine with 4 turbochargers. (CIMAC [13])

Fig. 12   Cylinder Cover with two outlets per cylinder, HOTLo75/150. (CIMAC [13])

Fig. 10   Cross section of the first Stork turbocharged crosshead engine, the HOTLo75/150, bore 750 mm, stroke 1500 mm, 1100 BHP/cyl at115 rpm and Mep = 6 bar. The engine is impulse pressure charged. (CIMAC [13])

Why the choice of 4 exhaust valves? Today one valve is "standard", but at the time there are four expected advantages that promote this solution: 1) a large time cross-area, facilitating scavenging 2) only one fuel valve is required, and this valve can be placed in the cylinder cover in the centre of the cylinder, 3) the thermal load of the components of the piston crown and cylinder liner becomes more symmetrical and easier to handle and 4) the bosses for the exhaust valves supports the cover bottom which then can be made relatively thin. The clear disadvantage is the more complicated (costly) and main-

tenance requiring valve gear, Fig. 13. In fact the valve spindles themselves becomes an extra burden because these are not uniformly heated, however the valve seats are water cooled in cages relatively easy to handle, Fig. 14, a feature that more than 15 years later is seen on other engine makes. Mitsubishi chooses, as it will be shown, three exhaust valves, and only much later, in 1981 they solve their reliability problem with the exhaust valves by changing to just one.

Fig. 13   Cylinder coover with exhaust valve gear, HOTLo75/150. (CIMAC [13])
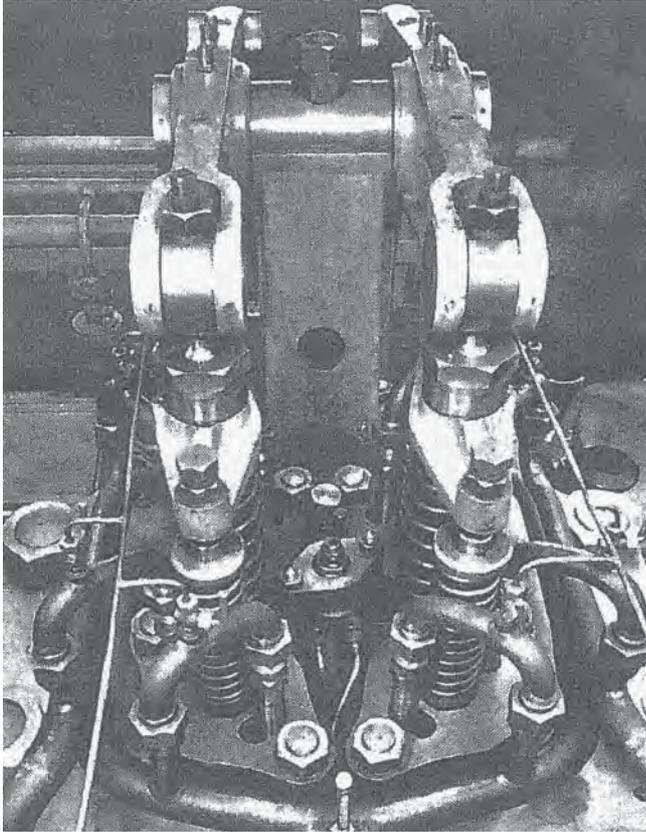
Fig. 14   Exhaust valves with water cooled valve seats, HOTLo75/150. (CIMAC [13])

The engine structure is with welded bedplate, welded columns and cast iron cylinder blocks, all held together with through-going staybolts. Thermal loaded components are simple, Fig. 15 and the cylinder liner is protected from excessive heat load by the dish shaped cylinder cover allowing a low position of the liner collar.

The firing order of the 8 cylinder engine is different from the practise used by others, but Stork also comes back to usual practise, when they later introduce their large bore engines SW85/180, Fig. 16 or SW85/170 [14] and apply the same firing order as used in B&W engines with 8 cylinders. Now Stork also realize that the efficiency on 6 cylinder engines becomes higher if instead of using one charger per two cylinders one use the 3-group with 120º between firing. So, the engine mean effective pressure becomes 0.7 bar higher for 6, 9 and 12 cylinder engines than for the 8 cylinder engines (B&W differentiate in the same manner when it comes to their K-EF-engines), whereas the 7 cylinder engines is missing in the programme.

Fig. 15 Piston mounted on rod connected to the crosshead, HOTLo75/125. (CIMAC [13])

Fig. 16 Cross section of SW85/180, bore 850 mm, stroke 1800 mm, 2100 BHP/cyl at 115 rpm and Mep = 8.1 bar. (MS [14])

Götaverken

In the late thirties Götaverken (GV) started design and production of their own engines. Previously from 1915 to 1948 GV have had a license agreement with B&W, but due to B&W in 1937 not being willing to design a single acting 2-stroke crosshead engine for them, they took on the job themselves. GV were indeed so tired of their client's complaints about the workload of the B&W double acting engines (trouble acting as the engineers said) that they got the push to realise some of their own ideas. Ironically B&W designed the crosshead engine that GV was asking for in 1939, and this engine was the one converted to impulse turbocharging. The reason for this strange behaviour of B&W is not known, it was certainly mismanagement, a lost licensee and future loss of market share.

So, GV were actually introducing a single acting uniflow crosshead engine ahead of B&W, whereas most people usually thought, that it was the other way round. The first turbocharged engine was a small 6 cylinder 520/900VGS-U, Fig. 17, which went to sea in 1955 installed in M/S "Axel Gorthon", 3 years later than B&W's 74VTBF160. It is not possible to explain this hesitation.

The 520/900VGS-engine is the first all welded engine (the "S" stands for welding), and it was tested as early as in 1944. When turbocharged it was named VGS-U (where "U" stands for Turbocharged). The experience gained from the welded type was so satisfactory, that new engines with one exception, the 850/1700VGA-U, were all of welded design without staybolts. VG-engines without further letter attachments are just engines with cast bedplate and -entablature.

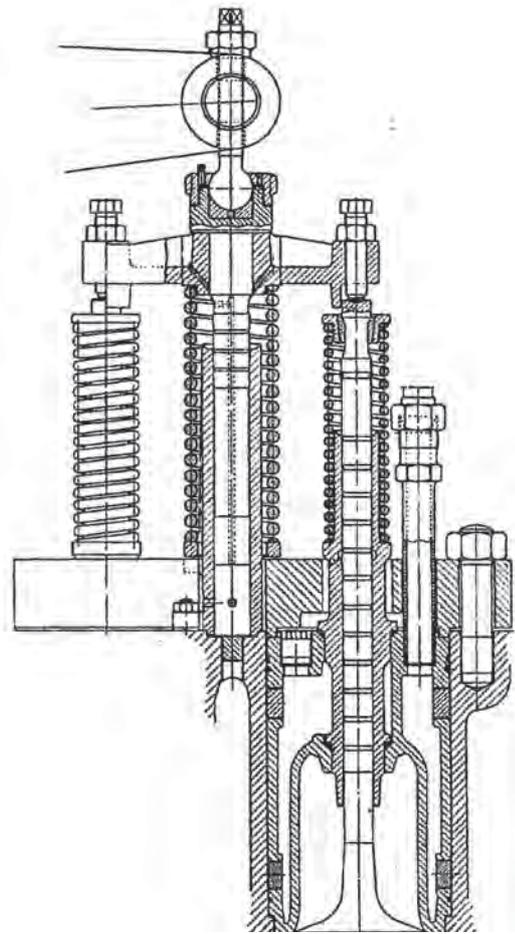The GV engine was constant pressure charged with assistance of one or two crosshead driven, double acting scavenge air pumps for each cylinder and maintained this feature throughout GV's existence. All other uniflow designs at that time chose pulse pressure charging. The reason was to ensure a high degree of operational reliability in service at sea with complete combustion at all loads, while the pumps would provide ample air for acceleration and manoeuvring.

Professor Collin tries in 1962 at the CIMAC congress in Copenhagen to justify the constant pressure system, Fig. 18, versus the impulse system [15], but realistically Götaverken were choosing a too complicated system at the time – later with increased turbocharger efficiency available the priority changes.

The GV engines have various design features that deviate from other brands and several will be mentioned later. However, regarding the mechanism driving the exhaust valve mention should be made. It was a unique characteristic of Götaverken engines retained since the birth of the first engine in 1939.

Each exhaust valve is operated by cam segments mounted on the respective crank webs through rollers, levers and pull rods to the yoke into which the valve stem is connected, clearly seen in Fig. 19, and the valve motion is retarded by a nest

of four helical springs surrounding each pull rod being exerted between the cover and the yoke. The advantage of this design is clear, the valve stem is actuated vertically without horizontal side forces, and troubles with worn out steering bushes and leaks of exhaust gas through these worn bushes as in engines with rocker arm actuation has been avoided. The disadvantage is that it conflicts with an optimal design of the crankshaft and make overhaul of main bearings more difficult. Further, and with reference to the discussion following a paper delivered by Professor Collin to the North East Coast of Shipbuilders and Engineers in 1962, there was apparently another disadvantage associated with exhaust valve gear: It was noisy. The opponent said: "I would prefer to see a poppet valve operated by cam and push rod and as there is a camshaft already on this engine in the right position that would represent a considerable improvement".

In 1960 a large bore engine is ready, the 850/1700VGA-U, Fig. 19 and front cover. It was to a start moderately rated with a mean effective pressure of 7.3 bar, but dimensioned for a higher rating after sufficient service experience had been accumulated, and in 1964 the mean effective pressure is increased to 8.7 bar, which is more on line with competition (see table later). This large bore engine is a sort of a bastard engine in the sense that it does not maintain the fully welded features of the entablature. It has quite conventional A-frames and staybolts. The reason was, that classi-fication societies at the time of designing the engine were issuing new rules for welded structures. The bedplate, however, remain welded as is has been proved trouble free, and not a single crack has been detected in the bedplate construction [3]. The engine does otherwise maintain typical GV features:

- the pioneering full width cross head bearing, but now with oil channels milled in the pin instead of in the bearing metal
- Bosch type fuel pumps supplying two valves per cylinder,
- two part cylinder cover with a cast iron section facing the combustion chamber and a steel cast part as the strong back
- the unique exhaust valve actuation system previously mentioned and
- scavenging system with constant pressure turbocharging

Regarding the latter it has been mentioned by former GV experts, that the most delicate parts of the system were the many non return valves in scavenge air system. These valves should be exchanged when damaged in order not to spoil the scavenging efficiency. This maintenance was often neglected and caused increased thermal load on the engine components, especially hurting the large bore engines.

Referring to maintenance; Götaverken took pride in designing for easy main-tenance and quick overhaul operations – on the test bed GV demonstrated that a piston overhaul could be done in 58 minutes from stopping the engine to starting up again with 4 men working on the top and two inside. This was quite remarkable and is not done faster on the following engines to be described.
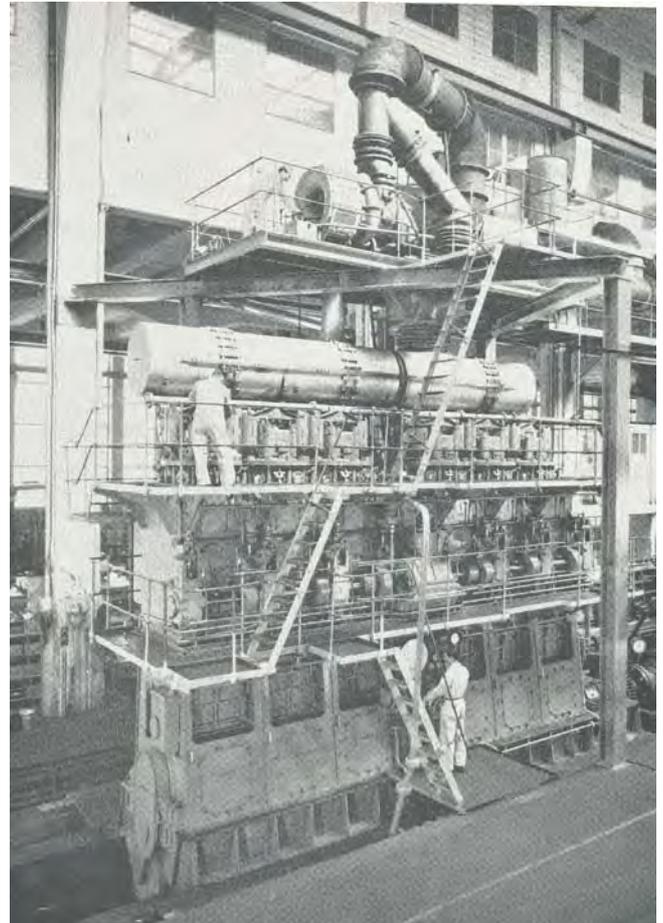
Fig. 17 First turbocharged Götaverken 2-stroke engine on test bed, 520/900VGS-U, bore 520 mm, stroke 900 mm, constant pressure charged. (GV [3])
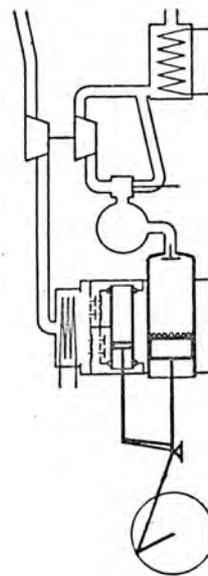
Fig. 18 Diagram of the Götaverken constant pressure charging system. (CIMAC [15])
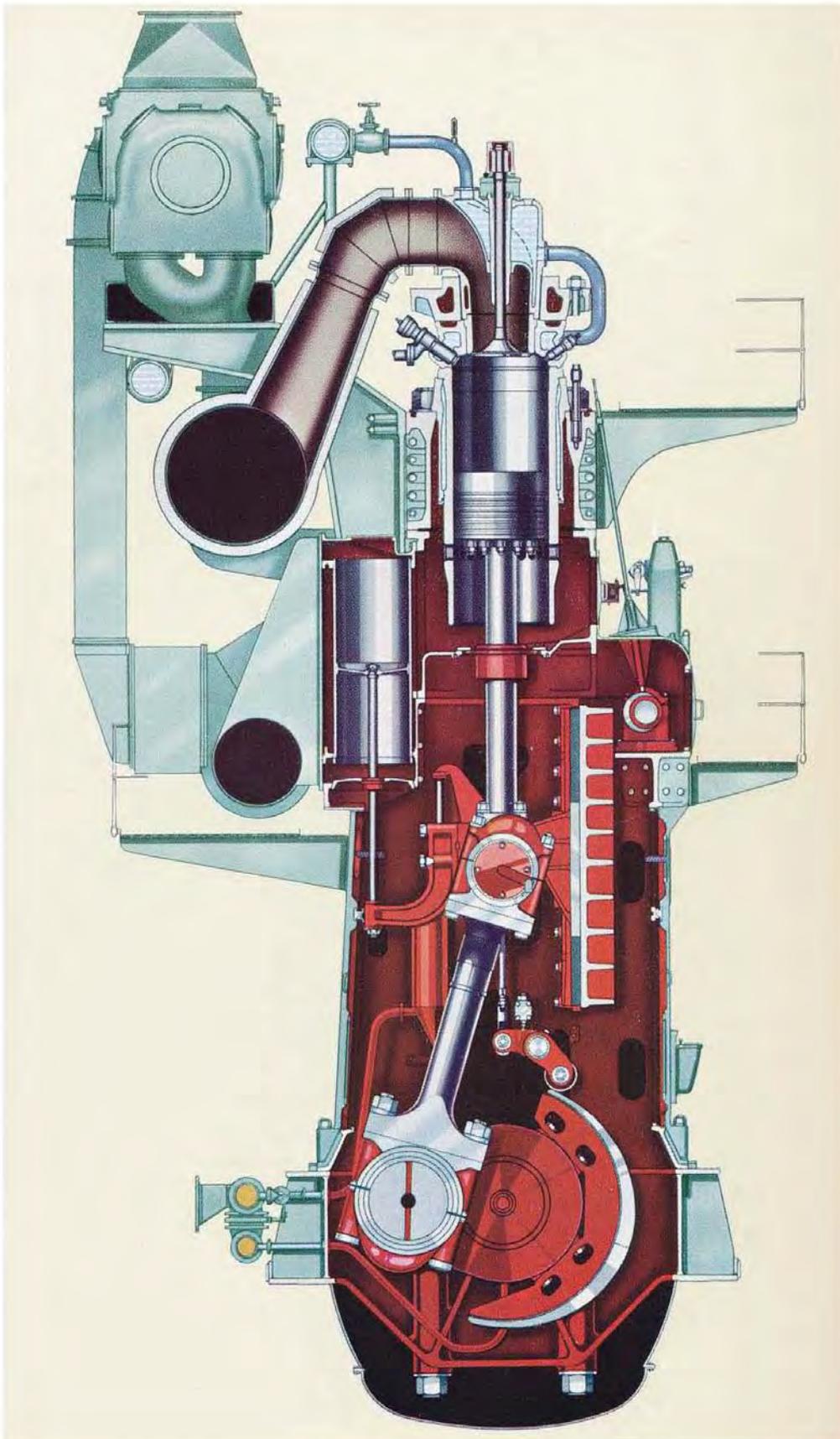
Fig. 19   Cross section 850/1700VGA-U, bore 850 mm, stroke 1700 mm, yielding 1850 BHP/cyl at 115 rpm and Mep = 7.3 bar. Note the special cam segment for the exhaust valve drive on the crankshaft. (GV [3])

Mitsubishi, just like Stork resembles the B&W engine in respect of scavenging, but has 3 exhaust valves.

Mr. Fujita reports at the CIMAC congress in 1955 [16], that MHI after many basic tests decided first to build a small, turbocharged 220 mm bore experimental engine and later in November 1952 a large size 3 cylinder, turbocharged experimental engine, 3UEC72/150. The first full size turbocharged crosshead engine, 9UEC75/150 was manufactured in September 1954 and after the confirmation tests installed in M/S "Sanuki Maru" of Nippon Yusen Co, that went in service in May 1955.

In contrast to **all** other engine designers, that have involved BBC when launching their first turbocharged 2-stroke engines, MHI courageously develops their own water cooled chargers for the UEC-engines. They could not benefit from BBC's growing experience with similar scavenged engines obtained at B&W and Stork. Not that MHI were unknown with BBC chargers, also being a Sulzer licensee they should soon test the first turbocharged Sulzer RSA76-engine in Kobe in 1955/56 equipped with BBC VTR630 chargers [6].

The 75-engine is shown in Fig.20 in a side view as well as a cross section. It features a cast bedplate of several pieces bolted together, A-frames, or columns connected to a box section and the upper part consisting of the cylinder block cast in several pieces bolted together and separate cylinder jackets. Tie rods secure that it all functions as one block.

The hot components are all water cooled. Only the piston crown is made of forged steel all other components are of cast iron. With the 3 exhaust valves MHI obtain the same "advantage" as Stork with the four valves: The fuel valve gets an "optimal" position in the centre of the cylinder. The valve gear is with Mr. Fujita's words "ingenious" and "attractive" and is shown in Fig. 21, but maintenance of the many bearings and bushings guiding the spindles is as in the Stork engines labour demanding. 25 years has to pass before MHI with the introduction of the UEC-H-engines settle on a single exhaust valve.
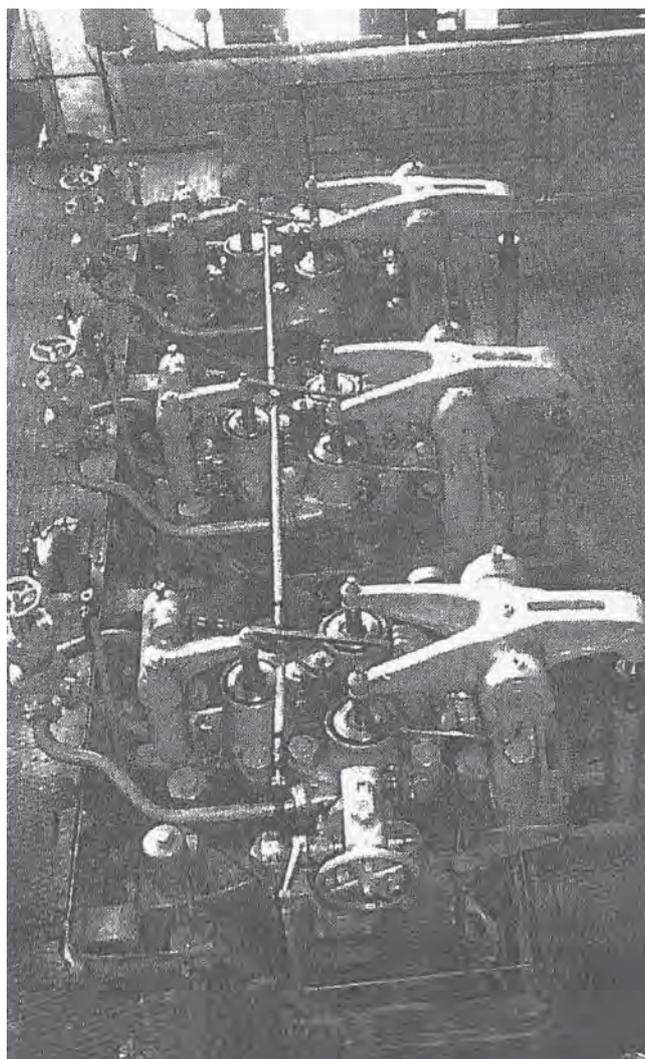
Fig. 21   Engine top with exhaust valve gear, 3UEC72/150. (CIMAC [16])
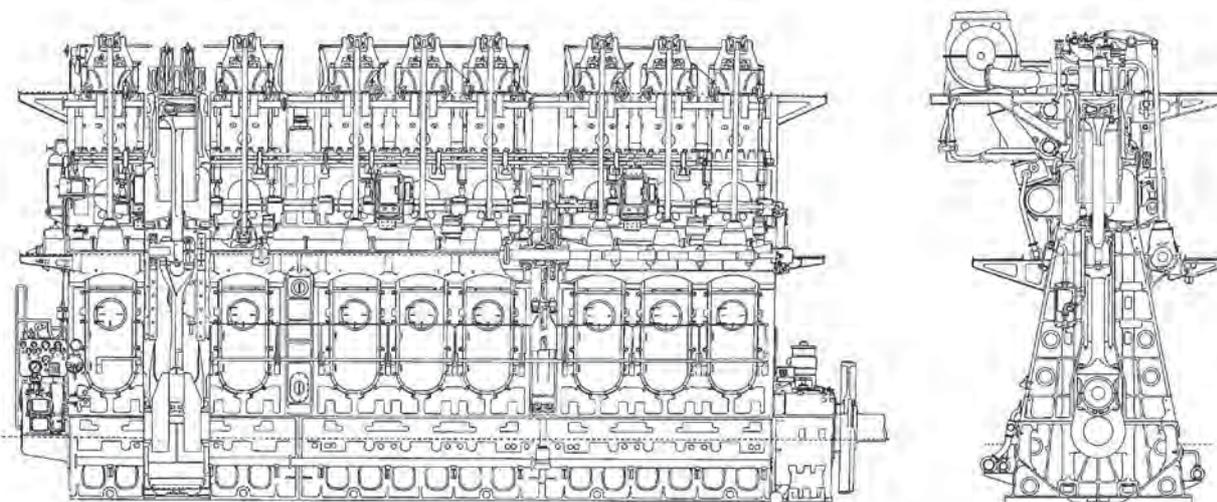
Fig. 20   The first Mitsubishi turbocharged 2-stroke crosshead engine, 9UEC75/150, bore 750 mm, stroke 1500 mm, 1333 BHP/cyl at 120 rpm and Mep 7.4 bar. (MHI [2])

The camshaft to drive the exhaust valves is located along the side of the engine crankcase near its top. Another camshaft for fuel injection and starting air control is placed above the exhaust cam shaft. Both cam shafts are driven by a train of gear wheels from the crankshaft. As in B&W's case the functions of the two shafts are soon combined in one.

Until 1960 the engine type is developed with 7 bore sizes, and the series is finalized with the UEC85/160. The first UEC engines have hereafter an A attached, and the following engines of higher rating or of new design will have the next letter in the alphabet attached.

A major improvement of the MHI engines is when in 1964 the MET uncooled turbocharger is introduced. Hereby expensive replacements of corroded gas inlet and outlet casings, caused by sulphuric acids of the exhaust gasses, are done away with. Whereas the MAN turbochargers from the outset were uncooled, the other competitors introduced this feature at a much later stage.

The UEC85/160 is in 1970 converted to a longer stroke as UEC85/180D, Fig. 22, an easy design change with a uniflow engine, but with an output of 2900 BHP/cyl at 118 rpm the mean effective pressure is increased to 10.7 bar. This requires in turn an increased turbocharger efficiency and compressor pressure ratio and leads to the introduction of the "High Pressure MET" turbocharger. The increased thermal load also necessitates introduction of water cooled exhaust valve seats.

Mitsubishi did not invest in development of a super large bore en-gine (see later), but for their own demands they had the Sulzer license and actually also a B&W license, that by the way only was used to build a single 8K98FF engine at the engine factory in Nagasaki.
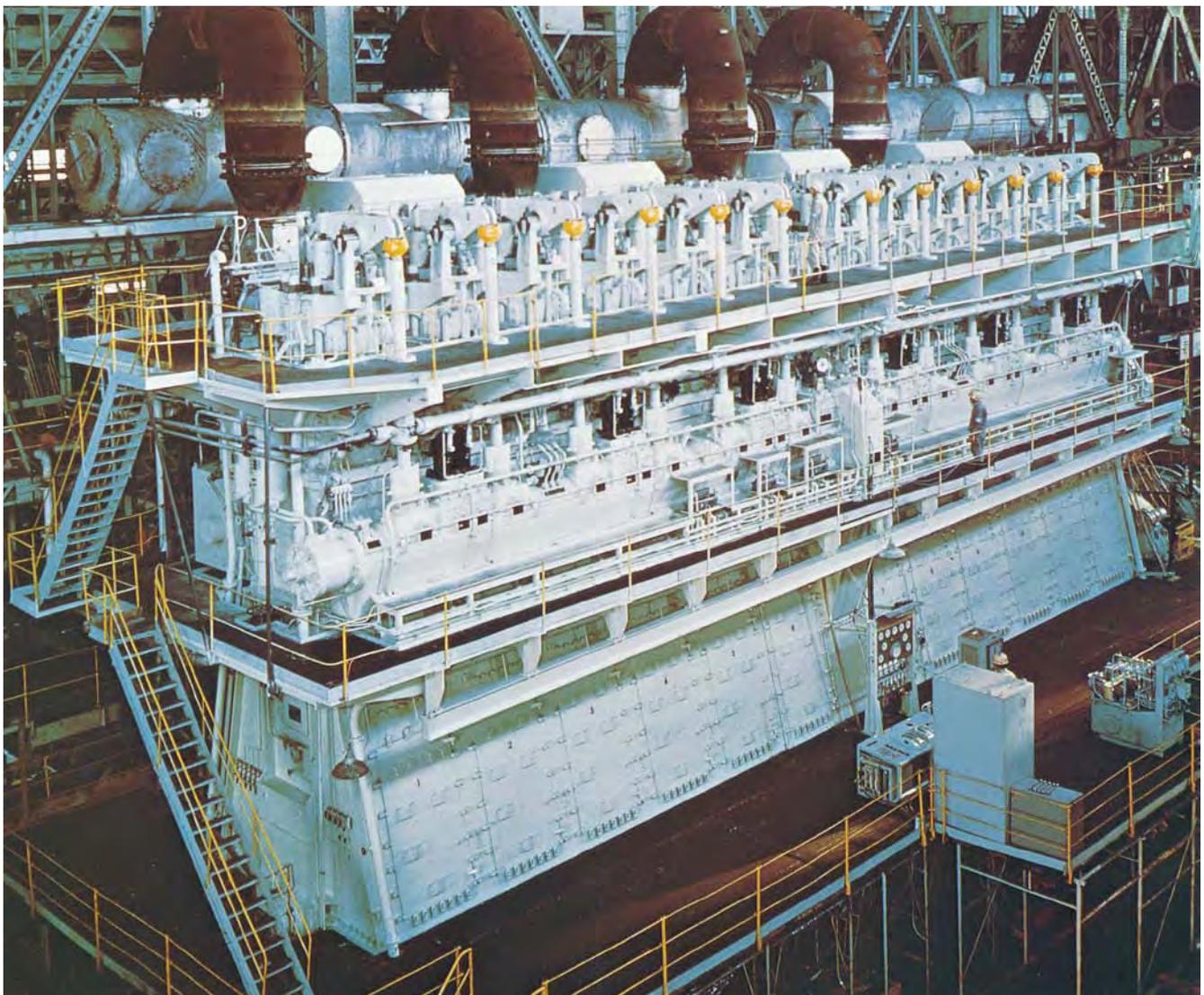
Fig. 22   12UEC85/180D on testbed, bore 850 mm, stroke 1800 mm, 2900 BHP/cyl at 118 rpm and Mep 10.7 bar. (MHI)

Doxford

Doxford is synominous with opposed piston engines. They designed their first engine in 1913 and continued the manufacture of such engines ever since. Doxford made significant contribution to Britain's survival during the war and the post war rebuilding of the fleet. Even though someone considered it a somewhat antediluvian engine at the time when Doxford were contemplating turbocharging, it was certainly in high favour by British shipyards and ship-owners, and the engines were produced by a handful of British workshops on license.

The opposed piston concept have the inherent advantage of being very well balanced and that the force flow due to combustion pressure is transferred to the crankshaft as pure torque and not transmitted to the engine frame, Fig. 23 [17]. The all-welded structure must, however, accept the torque reaction of the engine in the form of the horizontal thrust from the guide shoes.

Actually Doxford were the first to order a large turbocharger from Brown Boveri for their experimental 600LBD-engine, but as mentioned they were not the first to get the design ready for sale and being sent to sea. However, in 1954 the 3 cylinder 600LBD-test-engine with a total stroke of 2000 mm was installed in M/V "British Escort", where it could produce 3580 BHP at 120 rpm.

At the CIMAC congress in Den Haag in 1955 Mr. P. Jackson [18] is very optimistic about the power to be achieved by turbocharging and concludes that powers up to 15.000 BHP in the largest six cylinder engine will (in future) be undertaken on a single shaft. The question about additional air supply during low load and starting is more difficult than for the poppet valve engines, as the lead of the exhaust crank is limited for reversible engines. Doxford therefore decides to arrange the turbo blower in series with engine driven reciprocating pumps rather than reducing the port heights and thus the air amount and power.

The LBD engine design has one demerit that prevents Doxford in obtaining more power for a given bore and that is the crankshaft, it is long and heavy and very flexible. Torsional vibration conditions will not allow more than 6 cylinders and bearings have to be mounted on spherical seats to prevent edge contact. Experience with the LBD-engine and further development activities resulted in the appearance of the BDS range, Fig. 24 which incorporated many improvements and dispensed with lever-driven scavenge air pumps, yet still maintaining the same crankshaft design.

The P-type in 1960 does not solve the problem either, even though the crankshaft has been shortened, lighter and stiffer, a six cylinder engine is still the maximum.

Doxford are at this time accused of not producing an engine of large power, but they defend themselves by saying, that from the commencement of the 2. World War to 1958 the companies' engine works were fully occupied in producing engines for cargo ships and tankers and in this period of
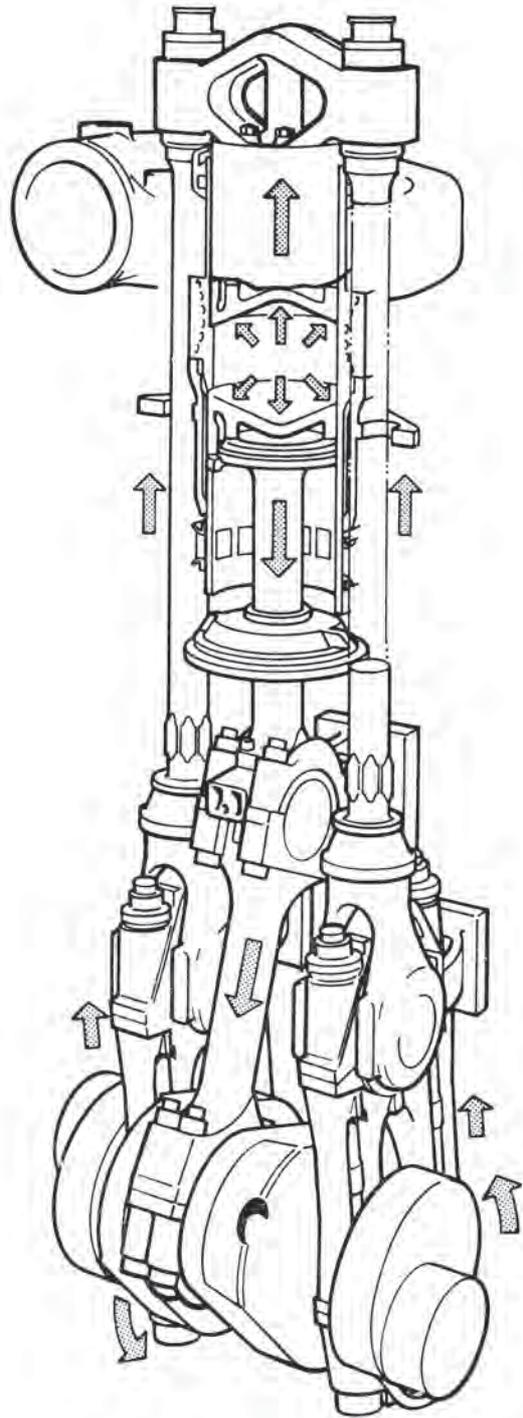
Fig. 23   Pictorial arrangement of running gear and gas forces. (Doxford [17])

intense production activity it was not possible to do much development. Doxford's trade is in ships up to 18.000 tons requiring about 7.000 BHP. But British shipbuilding is seeing mergers between yards, and bigger ships are planned just at Doxford's doorsteps in Sunderland.

It became apparent that a design had to be found that would permit more than six cylinders. This leads to the J-series of
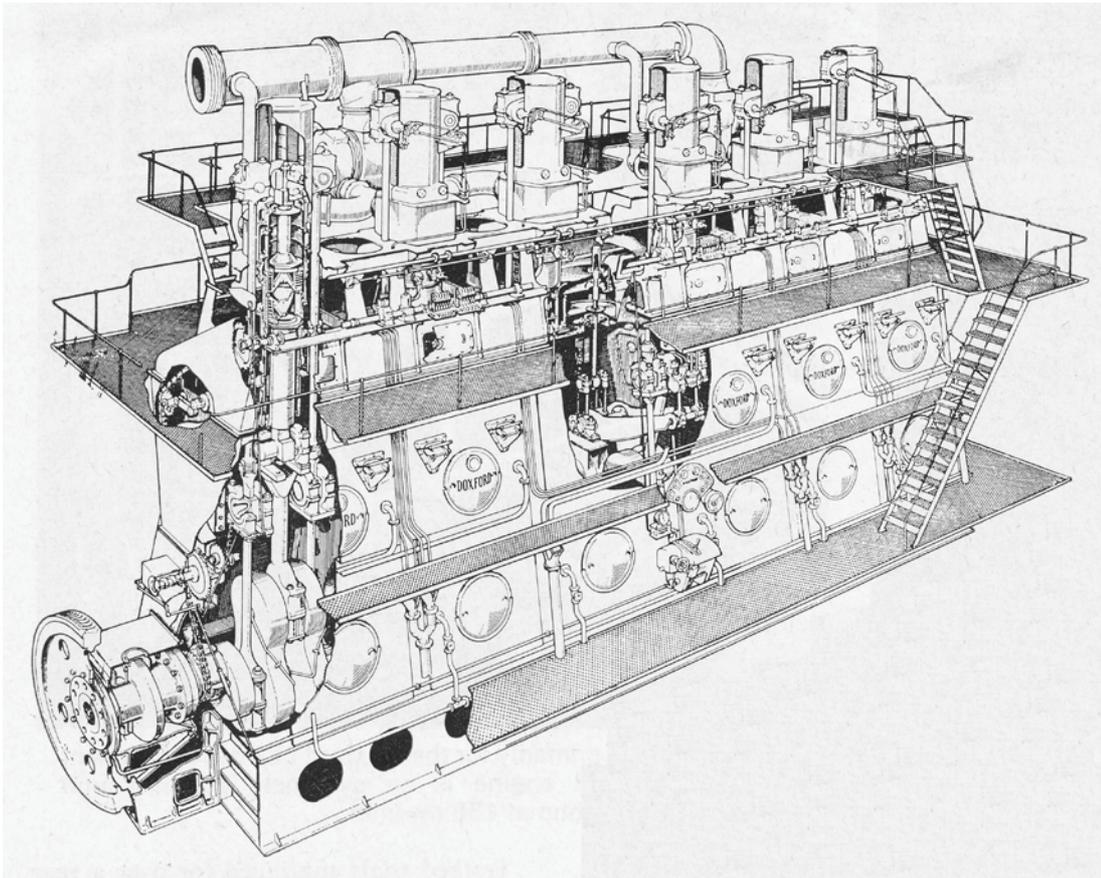
Fig. 24   Cut away view of the 6 cylinder BDS engine with 1833 BHP/cyl. (MS [19])
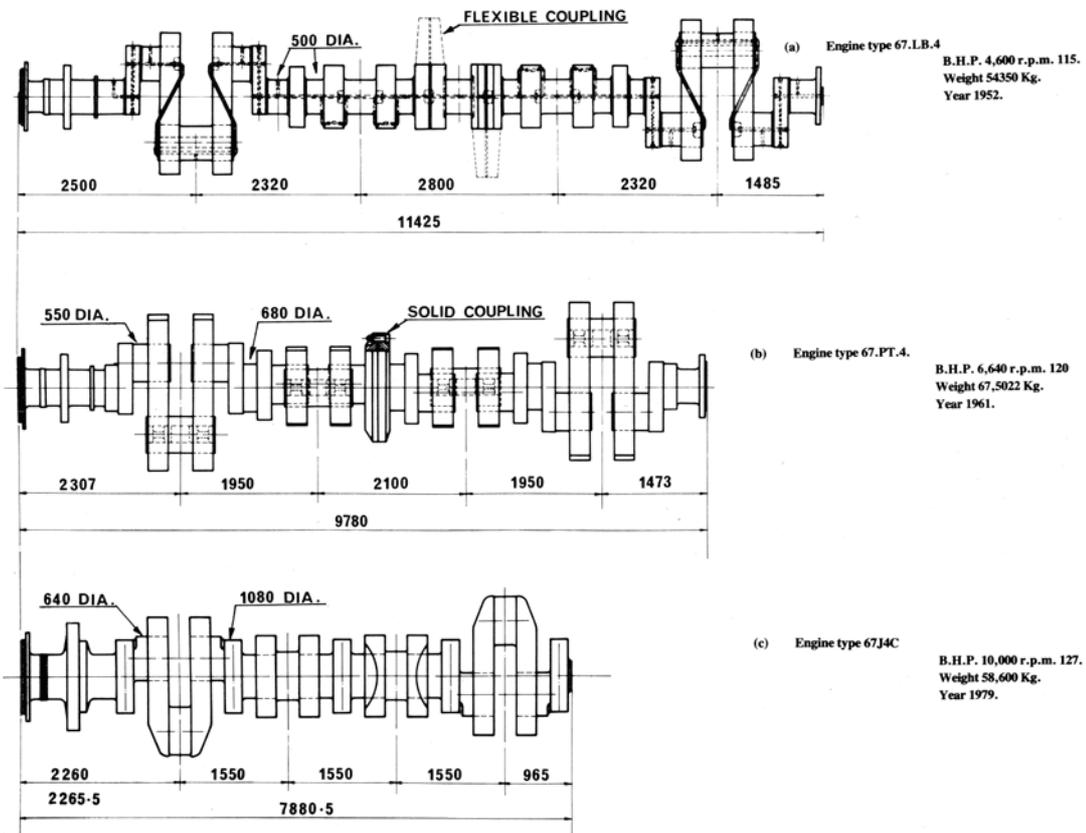
Fig. 25   Comparison of crankshafts for 4 cylinder engines. (Doxford [17])

engines, not exceeding present bore sizes, but with a crank-shaft design that allows up to 10 cylinders.

The development of the crankshaft design is shown in Fig. 25, where the J-type is on line with principles used in the B&W opposed piston engine design promoted by Harland & Wolff. Here eccentrics on the main crank webs are connected to the side rods and the side crank webs form the main journals. Somewhat strange that Doxford not readily had adopted this concept, as it was used for Harland & Wolff double acting engines since the war, and at a time where Doxford in the beginning of 1950's had tremendous problems with cracked crankshafts. The broken crankshafts turned the attention of many of Doxford's former licensees and customers to other makes of engines [19].

The 76J-engine is tested in 1963 and an elevation is shown in Fig. 26. Now Doxford find themselves in the same league as "the continental" companies and publish a table of comparison with all existing 20.000 BHP engines showing that the J-type has a specific weight of 29 kg/BHP compared to 36 to 45 kg/BHP for the continental competitors, and that the length of the engine is about 3 m shorter [20]. The combined stroke of the engine is 2180 mm, so it is indeed a long stroke engine with accompanying, attractive, low revolutions – 120 rpm. On the elevation it is not possible to study the cylinder liner design, but in Fig. 27 it is seen that it consist of 3 parts, the upper and lower liner of cast iron and a steel cast combustion belt with bosses for two fuel valves, starting air valve and safety valve. On the figure it is also shown, that drillings for water cooling of the upper liner above the ports have been introduced – this was a necessity to prevent the synthetic rubber ring to harden due to high temperatures. Further – cooling has been intensified in the combustion zone of the upper and lower liner with a large number of axial ribs that also serves to transmit the combustion load to the steel jackets [21].
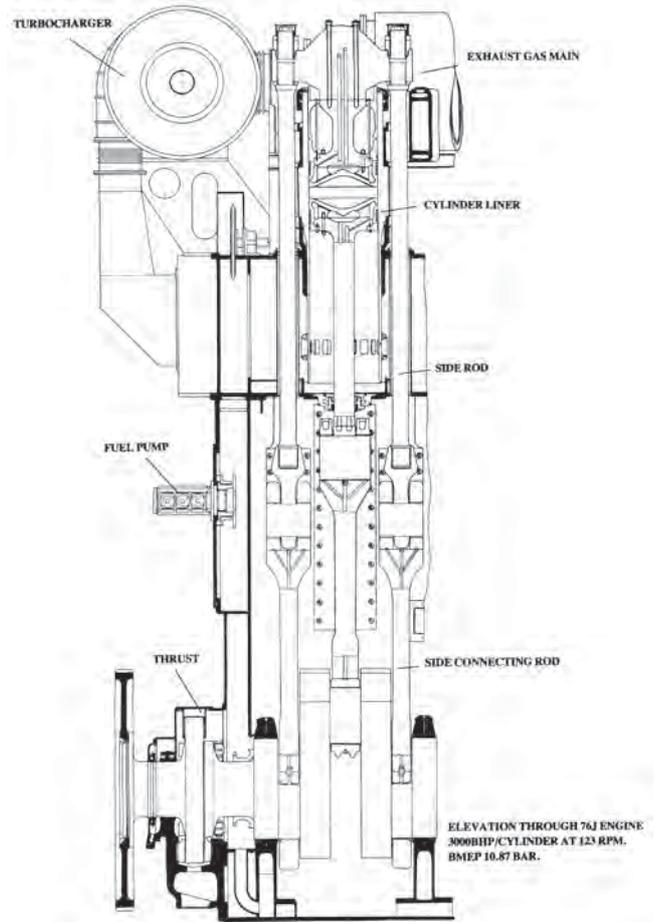
Fig. 26   Elevation through 76J, 760mm, stroke 2184 mm, 3000 BHP/cyl, at 123 rpm and Mep 10.9 bar. (Doxford [17])

This engine is pioneering the application of loose bearing shells for the crosshead bearings – more than 10 years ahead of any competitor.
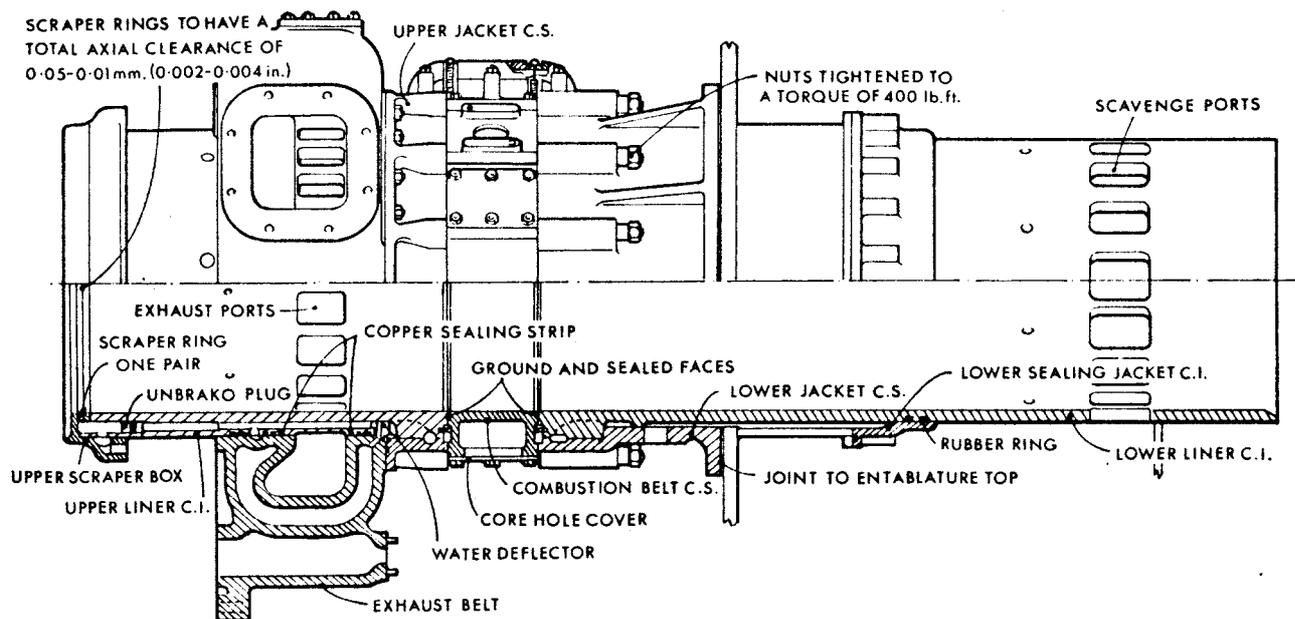
Fig. 27   Cylinder liner in 3 parts for 76J engines (MS [21])
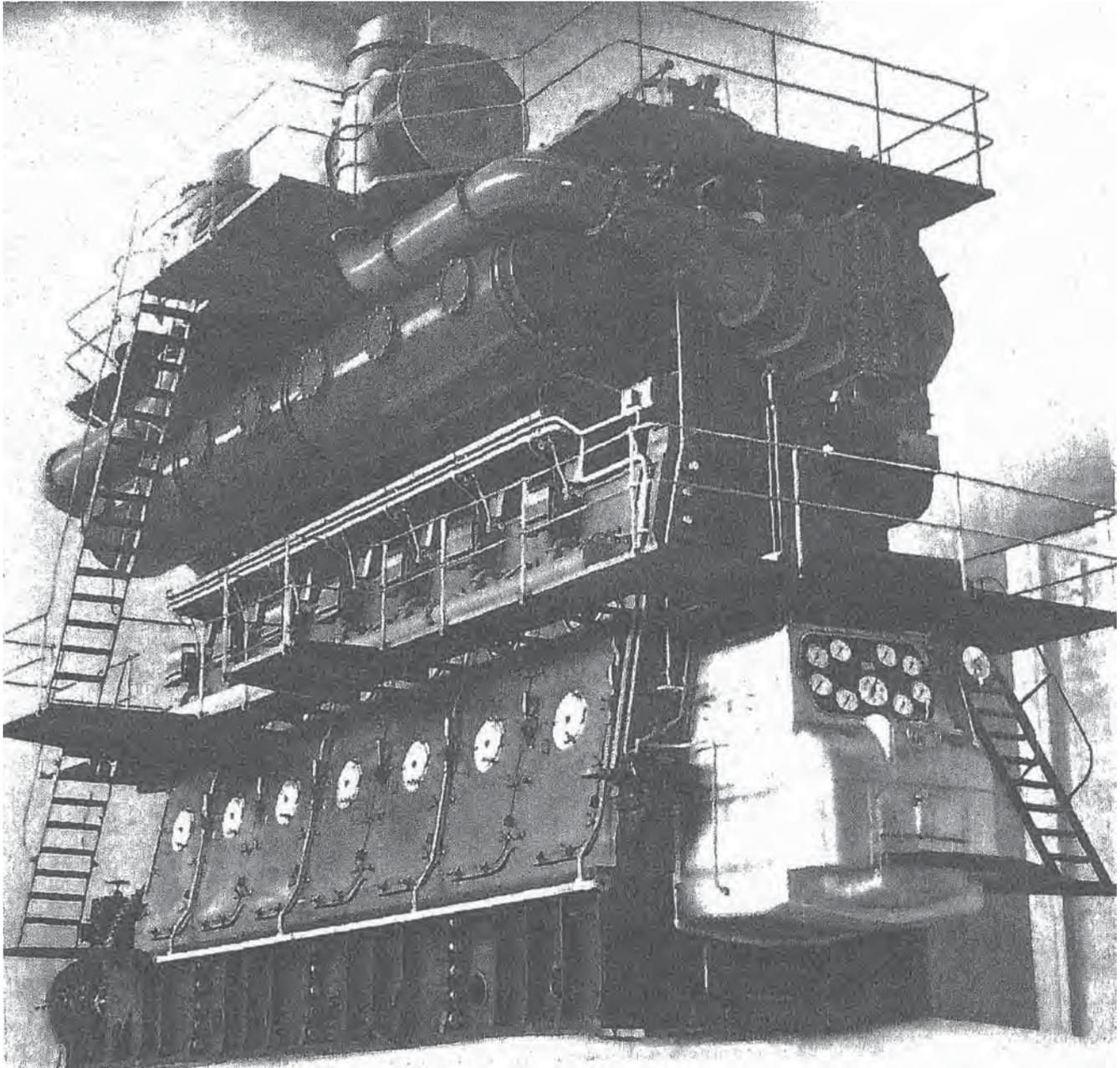
Fig. 28   750S on test bed in Turin. (CIMAC [22])

Dr. Gregoretti mention at the CIMAC congress in Zürich in 1957 [22] that Fiat started testing turbocharged engines in 1952 and in December 1955 the first marine application, a converted 8 cylinder 680S engine installed in M/S "Sestrière", was sent to sea. It is a cross scavenged, constant pressure charged engine with scavenging pumps working in series with the turbochargers, and this system was maintained in all Fiat designed turbocharged, 2-stroke cross head engines.

However, the first engine specifically designed for turbocharging is the 750S with a bore of 750 mm and a stroke of 1320 mm developing 1100 BHP/cyl. at 125 rpm with a mean pressure of 6.7 bar, Fig. 28 and 29.

The scavenging principle is illustrated in Fig. 30 with its two stage air compression system; the first stage is obtained from the turbo blower and the second through the engine driven air pumps, each with its corresponding cooling stage. To cope with the increased thermal load and to maintain the same time between overhauls as the non turbocharged engines i.e. 6000 to 7000 hours great care is invested in the design of the hot components. The piston is a new design having material temperatures matching the non turbocharged engines, but with a 40 % higher mean effective pressure. Meticulous thermocouple measurements on the piston crown were carried out and the outcome is illustrated in Fig.31, indeed a very acceptable temperature level.
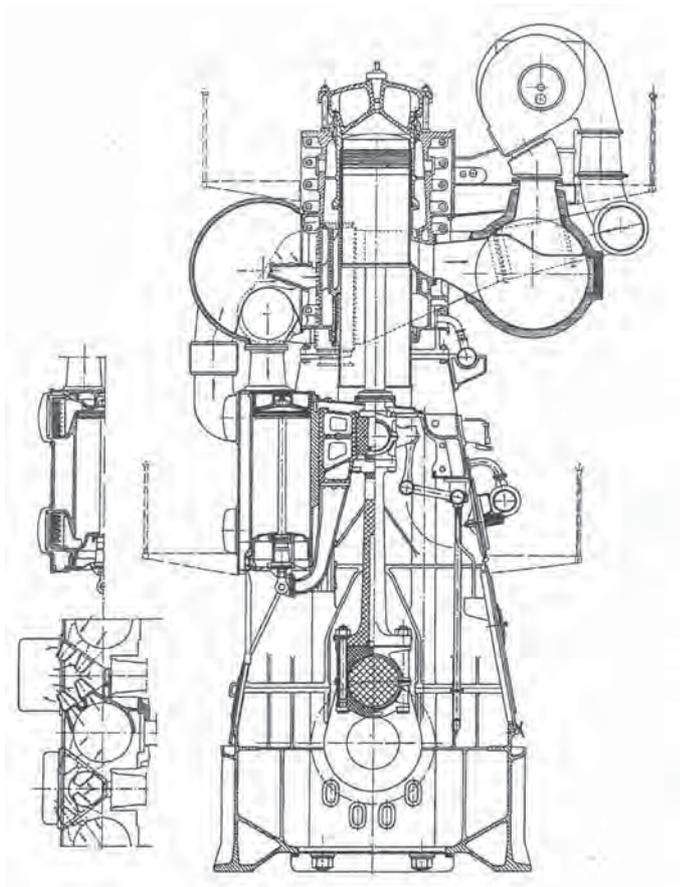
Fig. 29   Cross section of the first purpose built turbocharged Fiat engine, the 750S. Cross
scavenged concept with constant pressure charging and cylinder driven assisting
punps. Bore 750 mm, stroke 1320 mm, 1100 BHP/cyl at 125 rpm and Mep = 6.7
bar. (CIMAC [22])
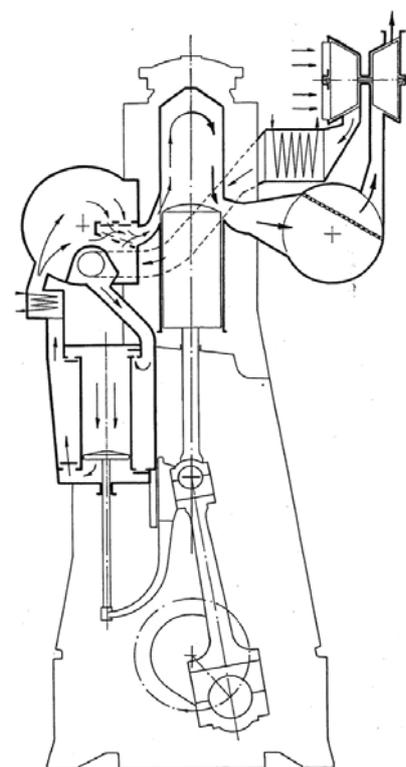
Fig. 30   Diagram of the Fiat cross
scavenging system with
engine driven scavenge pumps
(CIMAC [22])
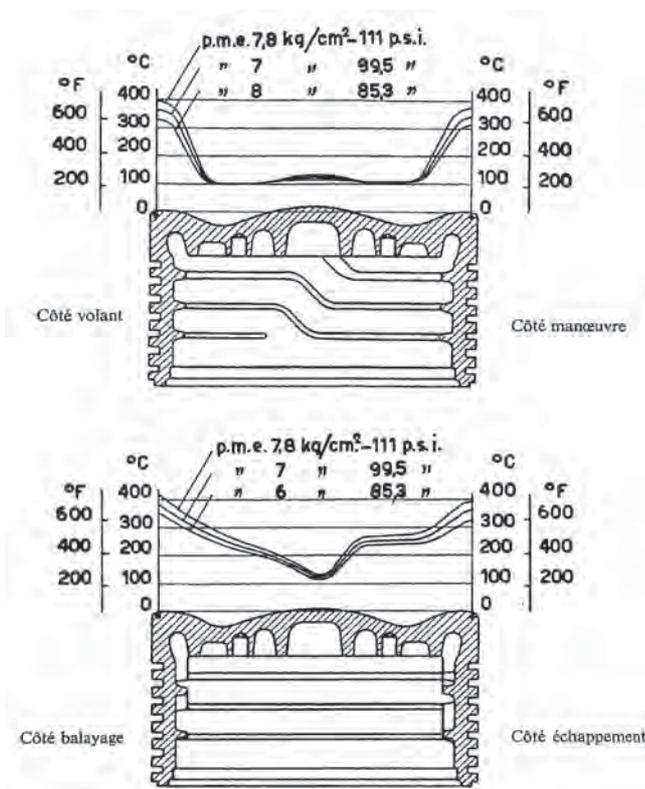
Fig. 31   Thermal load of the oil cooled piston, S750S. (CIMAC [22])

This piston design does not look like any of the competitors design – it is oil cooled, yet the heat transfer is just accomplished by circulation of the oil through oil channels without any splash effect – this design won't stand for higher mean effective pressure as pointed out later. Quite unique is also the number of piston rings – 7, hereof 3 with chrome plating of the running surface and 4 made of spheroid graphite – absolutely avant-garde. The cylinder liner design is also very special, as will be illustrated later.

In 1959 Fiat launch a large bore 900S engine [23] after having produced a 2-cylinder prototype for tests. The concept remain the same as for the 750S, but dimensions require some modifications, e.g. of the cylinder cover, now made in two parts as seen on a cross section of the combustion chamber, Fig.32. The liner design consists of an upper part with a cast iron bushing in a steel cylinder bolted to a lower part in cast iron containing the exhaust and scavenging ports. The intention of the designers were to allow cheap retrofitting of the cast iron bush when worn out or cracked and save the costly lower part, but the price paid for this feature must have been considerable. The two parted liner remains a design feature of the Fiat engines as will be reported in later sections.

Fiat presents in 1964 an extremely short stroke engine, the B600S with a bore of 600 mm and a stroke of 800 mm i.e.
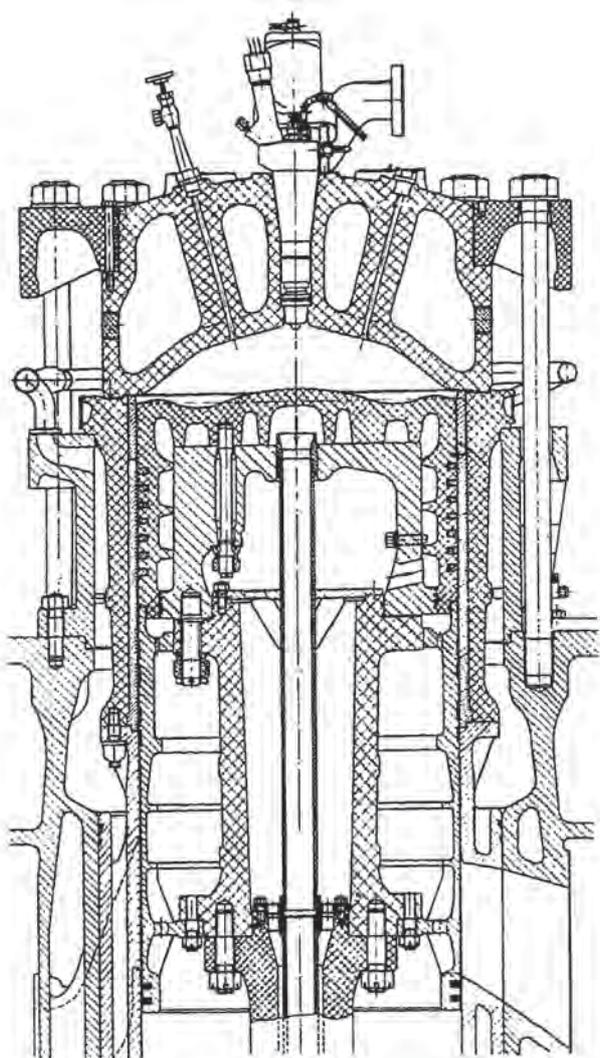
Fig. 32   Combustion chamber 900S-engine. (CIMAC [23])

a stroke/bore ratio of 1.3 in comparison to the ratio 1.8 on the larger engines – the effective mean pressure is 7.7 bar. This engine is mainly intended, without reduction gear, for use in ferries.

In the late 60's Fiat presents a 780S engine with 2000BHP/ cyl and a stroke of 1800 mm [24] and a really new and interesting engine the 1060S superlarge bore engine, mentioned in chapter 3.

MAN

MAN's first turbocharged engine ready for sales was a KZ78/140 [25]. MAN were late in introducing the turbocharged 2-stroke engine, but they were of course quite familiar with the challenge, as they during the war had made various tests on double acting 2-stroke engines and even produced some. However, the war put an end to that, but it took until 1949 before MAN could manage to provide a new engine programme, and with quite an effort they were in 1953/54 ready to board turbo charging. They felt forced by B&W's success in 1952 having the M/S "Dorthe Mærsk" at sea with the 74-engine to get started – and many test were accomplished, mainly on the K6Z70/120-engine, with nearly all possible combinations of impulse/constant pressure exhaust systems combined with under piston pumps in parallel, in series or in series-parallel with the turbocharger compressor.

MAN were sure that the loop scavenged engine was suitable for turbocharging, whereas their licensees would have preferred a change to uniflow scavenging, but MAN saw, as Sulzer did, a lot of mechanical complications in production and service of poppet valve engines – the loop scavenged engine was in their eyes preferable, because of its "extraordinary simplicity" as Professor Sörensen and Dr. Schmidt

argued in a paper read in The Institute of Marine Engineers in 1964 [26]. They did however also say that: "On the other hand it should, however, be pointed out, that any technical solution is only a compromise of certain advantages and disadvantages, the original standard and tradition being of great influence." They refer to facts, it is hard to leave the known tracks and do something completely different.

The MAN turbo charging concept was also very pragmatic, they did not go for a uniform solution. As a start they select the impulse-parallel system for 6, 9 & 12 cylinder engines, whereas they preferred impulse-series-parallel system for the other cylinder numbers. Later, with a higher degree of turbo charging they chose inpulse charging were it was preferable, i.e. for 6, 9 & 12 cylinder engines and constant pressure charging for the other cylinder numbers and until the introduction of the KSZ series of engines (starting with the KSZ105/180) they preferred the series-parallel supercharging system.

The turbocharging systems are schematically shown in Fig.33.

A cross section of the KZ78/140 is shown in Fig. 34. The mean effective pressure is 6.4 bar compared to the 5.1 bar for the normally aspirated engine – a power increase of 25 %. The rigidity of the engine is accomplished through the welded bedplate and the line of cylinder blocks bolted together. Between bedplate and cylinder blocks A-frames and a two part entablature is arranged. Tierods hold the components together. The bottom of the entablature is designed as diaphragms with stuffing boxes for the piston rods, which permits the use of the piston undersides as scavenging air pumps. The piston is fresh water cooled through telescopic pipes arranged outside the actual crankcase – in the smaller engine versions the pistons are oil cooled. The cylinder cover is a strong back design with fresh water cooling
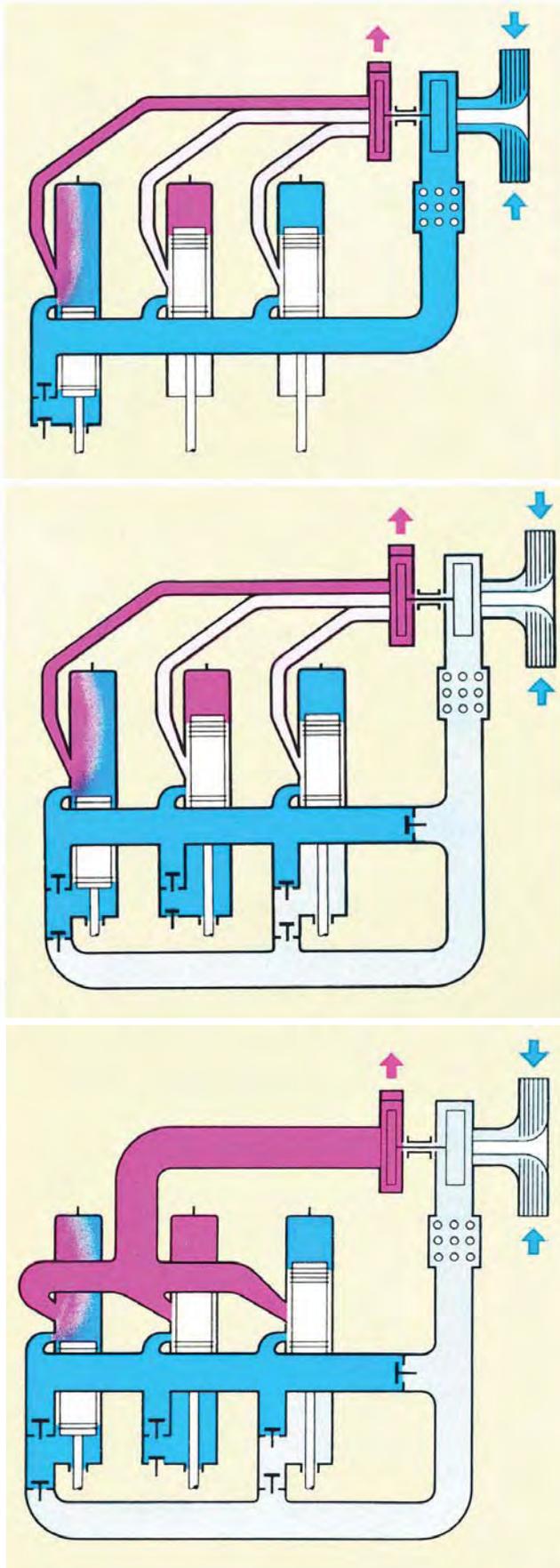
Fig. 33 Impulse-parallel, Impulse-series parallel and constant pressure-series-parallel turbocharging system (MAN)

of the bottom part. Fuel injection pumps have plungers of the Bosch type and are driven by a camshaft which in turn is driven by a train of gears from the crankshaft. The crosshead bearing is in two parts, and lubrication is secured by an attached high pressure pump on each cross head.

The race for power continues, in Augsburg they are successful and obtain already in 1956 a bigger market share than Sulzer. MAN increases the power range and designs the 74-type with an increased stroke of 1550 mm. To increase the stroke of the loop scavenged engine is not so easy, and actually MAN made 64 tests of possible port configurations to find the optimal solution. Soon however the competition, and in particular the demands from the Japanese licensees and shipyards, forces MAN to develop a large bore engine, the KZ84/160, but within a year this engine is replaced with KZ86/160 – in fact the old engine with 20 mm bigger bore.

Until 1970 MAN launches 8 different engines of the KZ-type, that concept wise are very similar to the KZ74/140C, with a mean effective pressure ranging from 6.5 to 9.5 bar, each class of engines are according to their design stage named C, D, E and F.
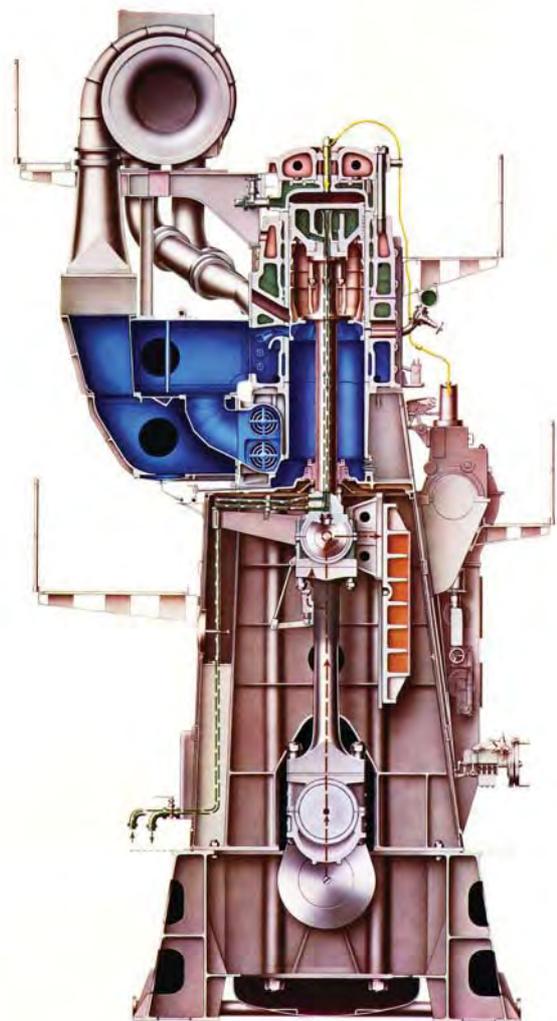
Fig.34 Cross section of the KZ78/140, the first turbocharged MAN crosshead engine to be sent to sea, bore 780 mm, stroke 1400 mm, 1140BHP/cyl at 118 rpm. Loop scavenged with Mep = 6.4 bar. (MAN)

Sulzer

Sulzer's start on turbocharging was based on tests with the 2-stroke 6TA48 trunk engine already in 1941, and with the RS58 crosshead engine from 1953 [4]. In the 50's Sulzer have made trials with cross-scavenged engines applying pulse as well as constant pressure charging and various under-piston arrangements. They also compared performance with a uniflow scavenged engine featuring 4 exhaust valves in the cylinder cover – just as in the Stork engines. This concept was however discarded presumably because of the complicated cylinder cover and valve gear, and probably the safe route applying the SD-engine with small modifications, now time was running out, was quite attractive. The first turbocharged engine, the SAD72, Fig. 35 has its turboblower charging in series with the scavenge pumps in a constant pressure system as in the case of Fiat and Götaverken.

As it can be seen in Fig. 2 the SAD72 engine is after one year followed by the RSAD76, which also is a sort of fast adaptation to the existing cross scavenged RSD-type. The constant pressure system used in the SAD-engines is now replaced with a pulse pressure system, and the separate scavenging pumps are discarded. This is possible because the engine has a diaphragm so that piston under-sides could be used to supply the scavenge air. A series-bypass system was applied where only a part of the air supplied by the turbochargers was led to the piston uder-side pumps.
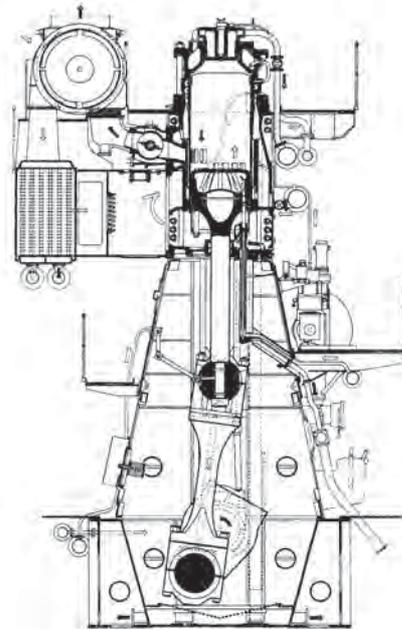
Fig. 36   RD90 first loop scavenged engine design from 1959, bore 900 mm, stroke 1550 mm, 2100 BHP at 115 rpm and Mep 8.1 bar. (Sulzer [27])

In 1957 the RD-series was introduced [27]. It was the first engine types, designed for turbocharging from the outset. A concept was established that characterised Sulzer engines the coming nearly 25 years, Fig. 36. The engines were loop scavenged, impulse charged with exhaust rotary valves to omit the long piston skirts, and the uderpiston pumps were working in series with the blower. The cylinder porting geometry of the loop system cause a reduction of flow resistance as the air flow don't need to be deflected as shown in Fig, 37.

Fig. 35   First turbocharged Sulzer crosshead engine, SAD72. Cross scavenged constant pressure charged with scavenge pumps, Mep = 6.2 bar. (Sulzer [27])

What would have happened if Sulzer had chosen differently? – A contra factual story with great possibilities, as it will appear later. Be it as it may – it was certainly a wise decision to stick to a cross scavenged engine, soon converted to the loop scavenging concept, and keep production costs down. Later in 1964 the Sulzer loop scavenged engine should conquer the leading position in the market place.

Fig. 37   Scavenging efficiency with different porting arrangements. (Sulzer [27])

The basic outcome is that the RD engine has an output of 150 % to 170 % compared to previous unturbocharged engines of the similar size.

A marked design change is the water cooled pistons, where the water is led into and out from the pistons through telescopic pipes placed inside a watertight enclosure to avoid water contamination of the lubrication oil and consequent corrosion of bearing journals. Sulzer claims that the increased heat transfer to the combustion chamber components, a consequence of the increased mean pressure, necessitates water cooling with superior heat transfer, and water cooling on the other hand makes it possible to design the piston with radial ribs supporting the piston crown. The new design is shown in Fig. 38, where it is compared to the not very well designed, oil cooled piston used previously.

The other thermal loaded components follow basically the previous concepts.

The engine has welded A-frames, and the bedplate is welded as well. Despite the application of pretensioned tie rods the bedplate caused troubles as cracks developed in imperfect welds under the main bearing saddle. The design was consequently modified with insertion of a forged or cast steel part as bearing saddle instead of the 100 % welded design. It should however be mentioned that this precaution was not a guaranty against cracks – poor castings has in several cases been the origin of cracks in modern engines.
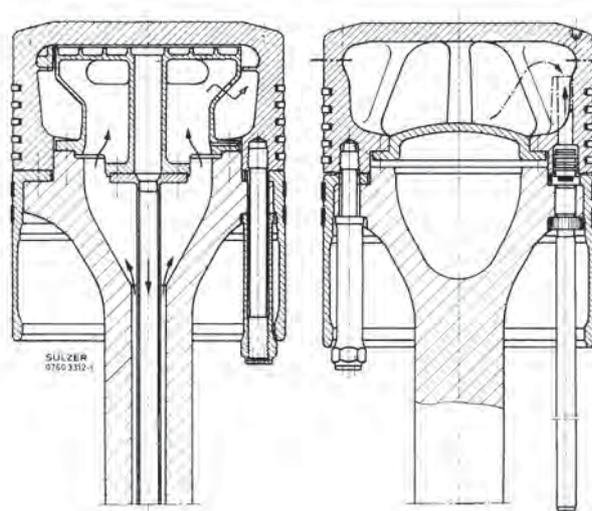
Fig. 38   Piston development for RD-engines, from oil cooling to water cooling. (Sulzer [27])

Sulzer strengthen in 1967 their market position by the introduction of the RND series of engines [28]. These are now constant pressure charged engines with under piston pumps in series with the turbo blower. Hereby it is possible to get rid of the rotary exhaust valve (a very welcome simplification), as the exhaust pressure in the receiver prevents the cylinder charge to escape. The first engine type is the RND105 (see later), and this engine is rapidly followed by 90, 76 and 68 bores, all of which gives 25 % more output than their equivalent predecessors, which improves the market position. The engine programme for the two arch rivals, B&W and Sulzer, is compared in the table below and shows, that Sulzer has chosen their bore sizes smartly, always being able to offer more power with a comparable engine size.

	RND105	K98FF	RND90	K84EF	RND76	K74EF	RND68	K62EF
BHP/cyl	4000	3800	2900	2540	2000	1900	1650	1360

The simplicity of the RND design is attractive for the engine users and the popularity of this engine type, also by the yards because of costs, makes it possible for Sulzer to consolidate their market leadership, conquered from B&W in 1964.

2. Large Bore Engines

The technical competition between the eight engine designers is interesting to follow and will perused throughout this history, but at the time when turbocharging was introduced, diesel propulsion had a total market share below 50 %, the majority was still in the hands of the steam turbines. The development of sea-going transportation also favoured steam turbine plants, as the ship types became bigger, among others triggered by the first closing of the Suez Canal in 1956 – 1957 – and even more so when the Canal in 1967 was closed for 8 years as a consequence of the Arab/Israel conflict. For the ships of 23.000 tons deadweight and upwards the market share of diesel ships were only around 20 % in 1954, but due to introduction of larger engine bores and high-pressure turbocharging the market share increased to 60% already in 1960 [29].

It was obvious, that the engine designers were keen to develop even more powerful engines, as they all would like to grab a big share of the steam market. So in the beginning of the 1960s the engine designers had their engines uprated to higher mean effective pressure, mep and bore, as set out in a table of large bore engines below:

The challenge of the diesel engineer is always to design the larger bores. This is primarily because the thermal load of the combustion room becomes more and more delicate to handle, as the wall thickness to cater for the increased combustion forces has to be increased to maintain unchanged safety against fatigue cracks in the design. The increased wall thicknesses mean on the other hand, that the surface temperatures of the combustion chamber components will raise everything equal. So, new designs are the only way to preserve reliability, however, this was being learned the hard way.

Most of the components in the combustion chamber were more or less just scaled to the bigger dimensions, and consequently there were generally many failures in service due to:
- Too hot surfaces, leading to material "burn down" of piston crowns and exhaust valve bottoms and especially valve seats
- Fatigue cracks in pistons, cylinder covers and cylinder liners
- Thermal cracks of cylinder liner surfaces and around ports (loop & cross scavenged engines)

Designer	Engine type	Bore mm	Stroke mm	RPM	Mep, bar	BHP/cyl CSR	Year in service
B&W	84VT2BF180	840	1800	110	8.5	2100	1960
Doxford	76J	760	2180	115	8.6	2222	1965
Götaverken	850/1700VGA-U	850	1700	115	7.3	1850	1961
Fiat/GMT	900S	900	1600	110	8.3	2100	1961
MAN	KZ84/160C	840	1600	115	7.7	1800	1961
Mitsubishi	UEC85/160A	850	1600	120	8.1	2000	1961
Stork/SWD	SW85/180	850	1800	110	7.3	1875	1962
Sulzer	RD90	900	1550	115	8.1	2100	1960

The shipping society was not too happy with the engine development and in particular the Norwegian shipping circles – they organised a so called "Large Bore Project" with DNV and NTH involvement, where cylinder component temperatures were to be measured at sea in selected engines running on heavy fuel, which by the way was believed to be a major influential factor for the high temperatures. Several of the engine designers were more or less forced to participate in the project not to lose face.

3. Super Large Bore Engines

Despite the mentioned difficulties the power race was continued. 4 super large bore designs were presented in the late 60's and in the beginning of the 70's, and are shown in the following table:

Designer	Engine type	Bore, mm	Stroke,mm	RPM	Mep, bar	BHP/cyl	Year in service
B&W	K98FF	980	2000	103	10.8	3800	1968
Fiat/GMT	1060S	1060	1900	106	10.0	4000	1971
MAN	KSZ105/180	1050	1800	106	10.7	4000	1969
Sulzer	RND105	1050	1800	108	10.,5	4000	1969

Cross sections of these engines are shown in Fig. 41, in the same scale.

At the first look the loop and cross scavenged engines looks somewhat simpler than the uniflow engine with its complex exhaust valve drive, push rods and rocker arms. It is cheaper built, but it is not so simple, the complexity is hidden in the scavenging system with its many small valves and a somewhat complex cylinder liner, and as an outstanding example the 1060S liner is shown in Fig. 39 – a cross section as well as an exploded view [30].

Sulzer decides to change their scavenging system, and the RND105 is the first engine to have constant pressure turbocharging [31].

MAN also, finally decides for all their engines irrespective of cylinder number to use a constant pressure parallel system with a limited number of under-piston pumps in action plus injectors, that only are used within the lower power range, Fig. 40, [32].
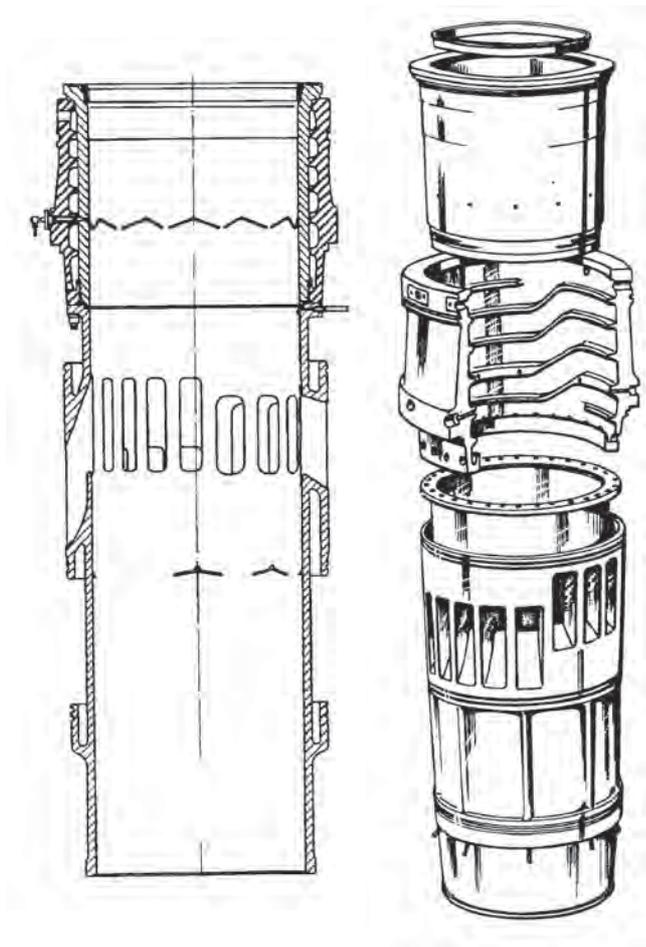

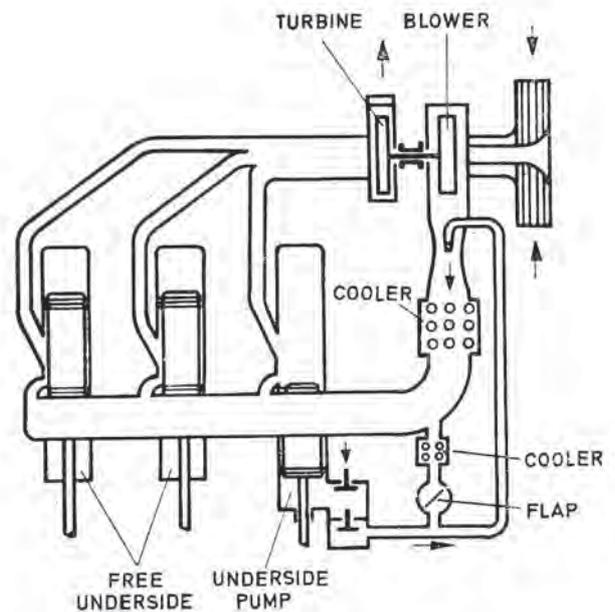Fig. 39 FIAT 1060S cylinder liner design, cross section and exploded view. (CIMAC [30])

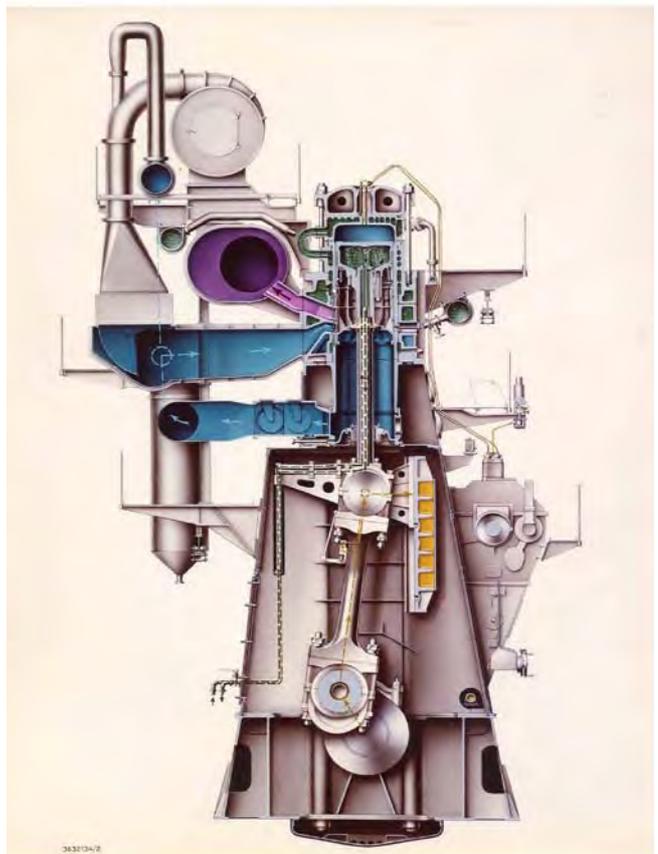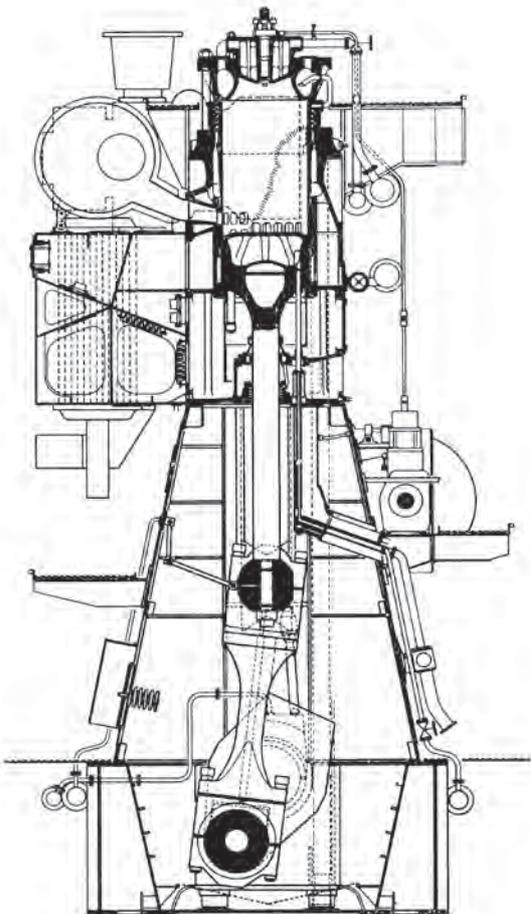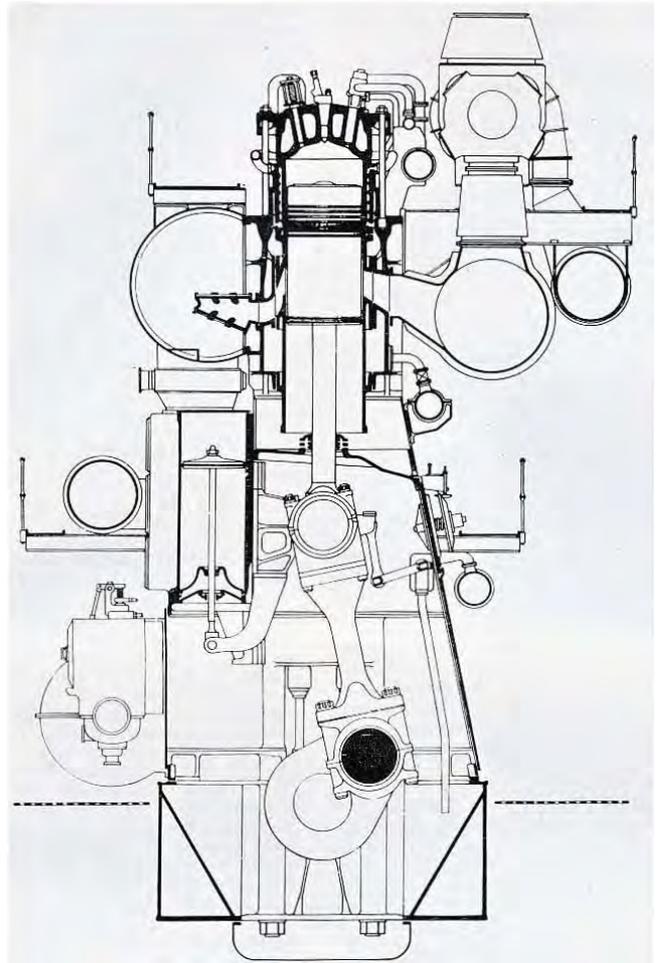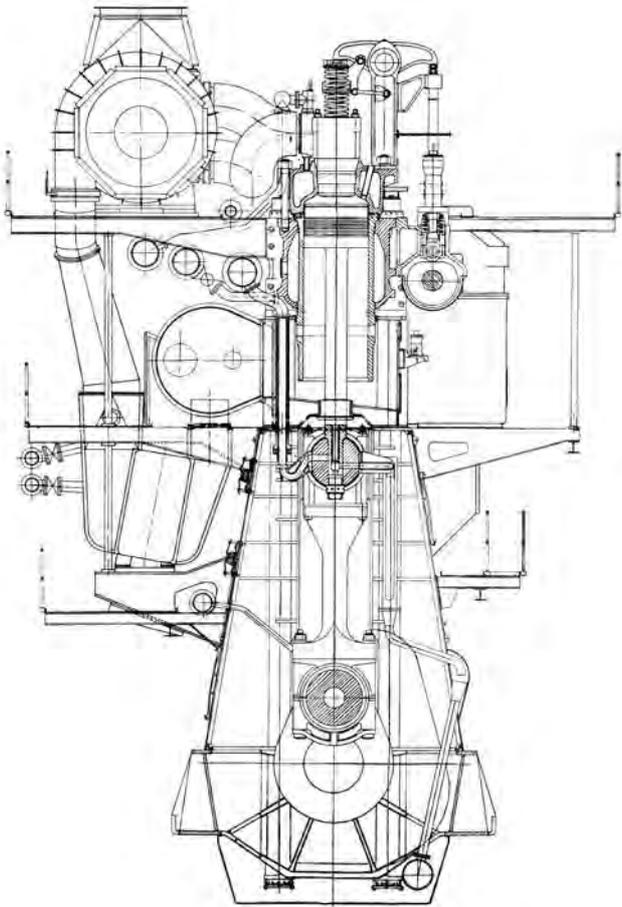Fig. 40 MAN, Constant-pressure parallel system. (MS [32])

Fig. 41   K98FF, 1060S, RND105, KSZ105 (B&W [33], MS [24],
          Wärtsilä [31], MAN)

However, something had to be done to improve the reliability of the engines in general, but especially of the hot components.

The K98FF is the first large 2-stroke engine where bore cooling is applied – in the collar of the cylinder liner, and the RND105 follows although the design differs.

Three of the engines get new piston designs:

The oil cooled piston of K98FF, Fig. 42 is a shell design with inner support and the top clamped to the piston rod through the piston skirt by means of a belleville spring – this design is used in B&W engines for the next 15 years. It replaces the VT2BF-design with outer support of the crown, also shown in Fig. 42 [34].

MAN also introduces a new piston Fig. 43, water cooled as usual, but this time with a double backing of the piston crown, by elastic pillars supporting at two diameters – a clever way to avoid too high thermal stresses and temperatures of the crown surface. Even the best concepts may, however suffer from some children's diseases and MAN's piston is no exception – slight modifications of the supports were required to eliminate fatigue cracks.

Fiat, who like most of the others, have had difficulties with their piston design, made a new support of the piston crown, where the channels for cooling oil were machined in the support element rather than in the crown behind the piston ring belt. This reduced stresses in the area concerned. This measure should however soon become obsolete. Test were carried out using water cooling instead of oil and despite previous conclusions that the oil cooled wall temperatures were very acceptable, Fiat decides to introduce water cooling even on engines with 4000 BHP/cyl.
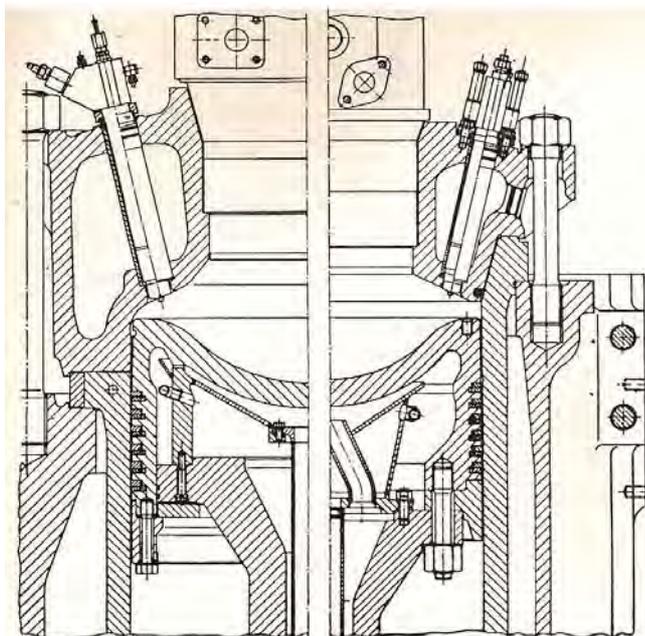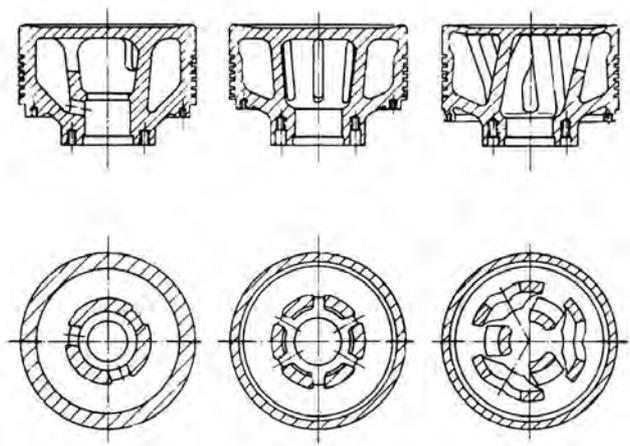
Fig. 43   Piston design development for the KSZ-engines. (MS [32])

Fiat argues that this is done to be prepared for a higher rating of the engine due to normal technical progress.

Sulzer continues with water cooling. They are pretty happy with their design that after modifications of the ribs, function well, without cracks. Actually they are so happy that they announce in trade magazines: "Sooner or later everyone will apply water cooled pistons" and they are exposing pictures of the different temperature levels ref. Fig. 38. It was an unusual announcement that indirectly told the readers that B&W's and Fiat's piston designs were obsolete – later in this report it will be clear that Sulzer proved themselves wrong.

All the other thermally loaded components were more traditional and some created trouble in service. The B&W cylinder cover was a disaster-skilled welders had to be flown around the world to repair cracked covers and spare covers were in high demand. A new design was prepared and the cover was split into two cast steel parts. Sulzer were also having difficulties with cracked covers.

The wall temperatures of these super large bore engines were naturally a case of great interest to the users, and the temperatures was published in various technical bulletins as shown below, Fig. 44, 45, 46, 47.

Looking at the temperature levels there seems to be a certain agreement as to the desired maximum temperature of the pistons, and it is a bit difficult to understand why Fiat changes to water cooling instead of a well-designed oil cooled piston. Cylinder liner temperatures at the point of the upper piston ring in top dead centre is between 140 and 170 ˚C, low by today's standard (see later), but adequate considering the firing pressure and fuel quality. B&W had a problem in keeping the exhaust valve temperatures low enough to avoid thermal corrosion, "burning" of the valve bottom.

Fig. 42   Combustion Chamber and fuel injection arrangements
comparison between K98FF (left) and the 84VT2BF180
engine (right). (MS [34])
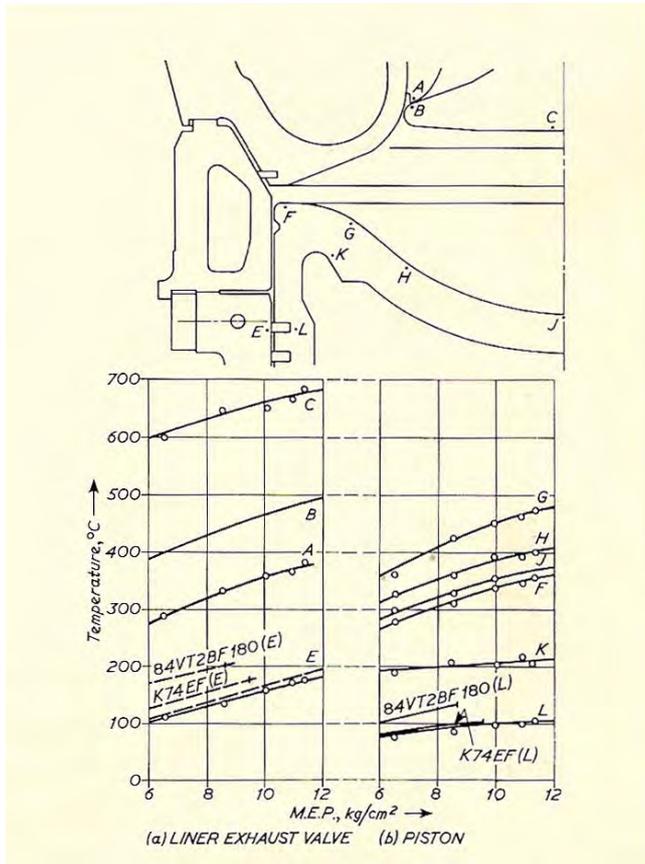
Fig. 44 Combustion Chamber temperature measurement, 7K98FF and comparison of liner and ring groove temperatures of 84VTBF180 and K74EF engines. (B&W)
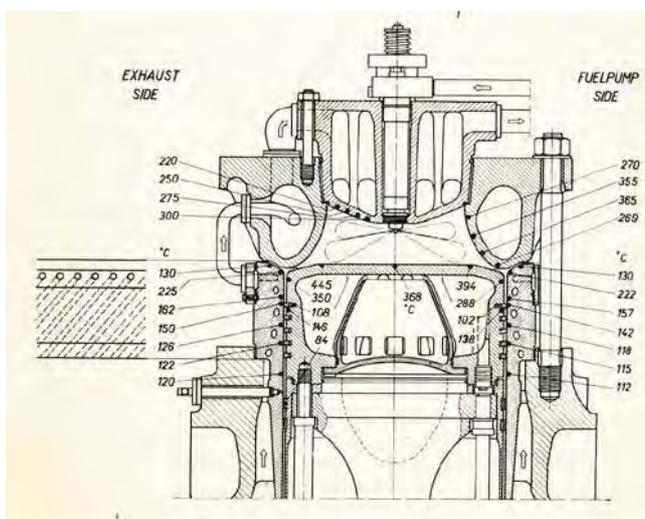
Fig. 46 Combustion chamber temperatures, 1060S with an oil cooled (left) as well as a water cooled piston (CIMAC [30])
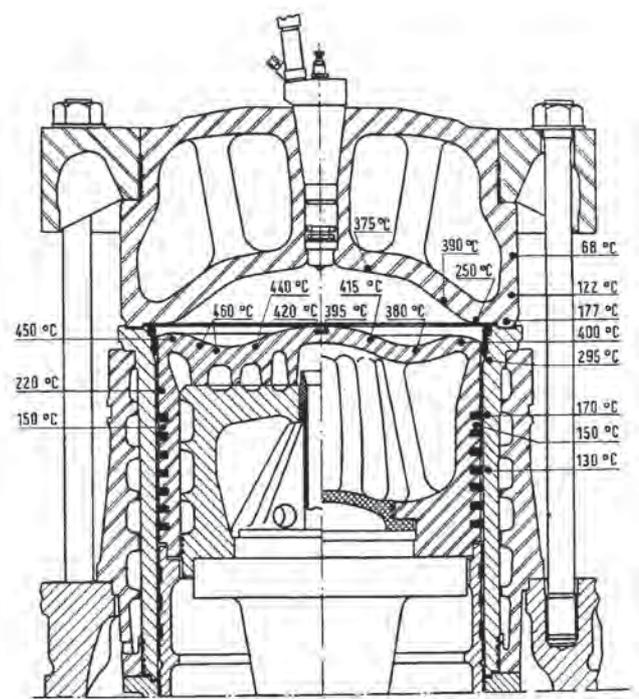
Fig. 47 Combustion chamber temperatures RND105 at 4000 BHP/cyl. (MS [28])
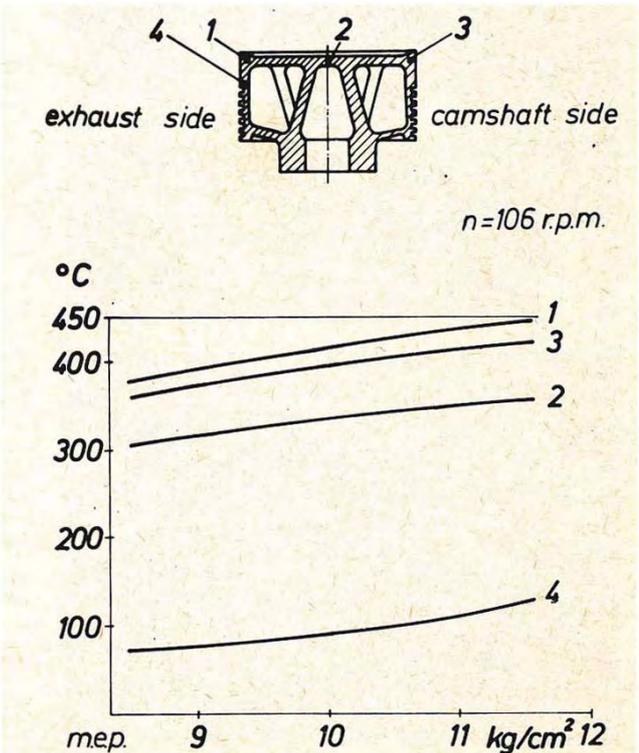
Fig. 45 Piston crown temperatures KSZ105. (MS [32])

4. The last Stork-Werkspoor 2-stroke crosshead engines

Stork did not present a super large bore engine, but they preferred a different way to obtain high power by modifying their SW 85/170 (the successor of the SW85/180). The new engine, the SW90/170 was presented in 1966 [36,37] and was in fact the engine with the highest cylinder output at the time – 2800 BHP per cylinder at 115 RPM. The high power was a consequence of Stork being able to increase the bore of the 84 engines without altering the cylinder distance and increase the mean pres effective mean pressure to 10.2 bar. After some turbocharger and fuel injection system modifications some of the first engines were indeed tested with up to 3333 BHP/cyl at a mep of 12.1 bar, a 6-cylinder engine is here shown on the test bed at Hengelo, Fig. 48. Following this success Chief Designer van der Molen declared that 3166 BHP/cyl should be regarded as the "day in, day out" rating.

Like the SW84/170 the 90-bore engine series did not contain a 7 cylinder engine, and the 8 cylinder version had a lower rating than the others. It was the intention that the 6 and 9 cylinder engine of the 84/170 series should fill the gap in the power range. Stork had hoped that two stage turbocharging would provide a means of giving equal cylinder outputs irrespective of the cylinder groupings. Experiments were in 1965 done in co-operation with BBC on their 3 cylinder 75/160 test-engine; however, this was of course a too costly way of solving the equal cylinder rating question.

In general the engine characteristics were following the previous uniflow scavenged Stork engine, but with further emphasis on easy maintenance and prolonged time between overhaul. To that end the 4 exhaust valves were provided with rotators (just as on the medium speed engines), a feature that is generally introduced in the 80's for all other designs.

Furthermore Stork-Werkspoor made use of a full width cross head bearing, pioneered by Götaverken, which made the previous cross head driven high pressure lube oil pumps superfluous. Turbocharging was affected by BBC VTR750 chargers with non standard, plain bearings, the same that Götaverken previously had "forced upon" BBC, and not the usual ball bearings.

Stork-Werkspoor was a relatively small company and could not financially support the development that was required to follow the leaders in the market place. Consequently they decided to stop the development of low speed marine diesel engines in 1969 and instead use their resources on the groving market for production of medium speed engines – the market was right at their doorsteps.
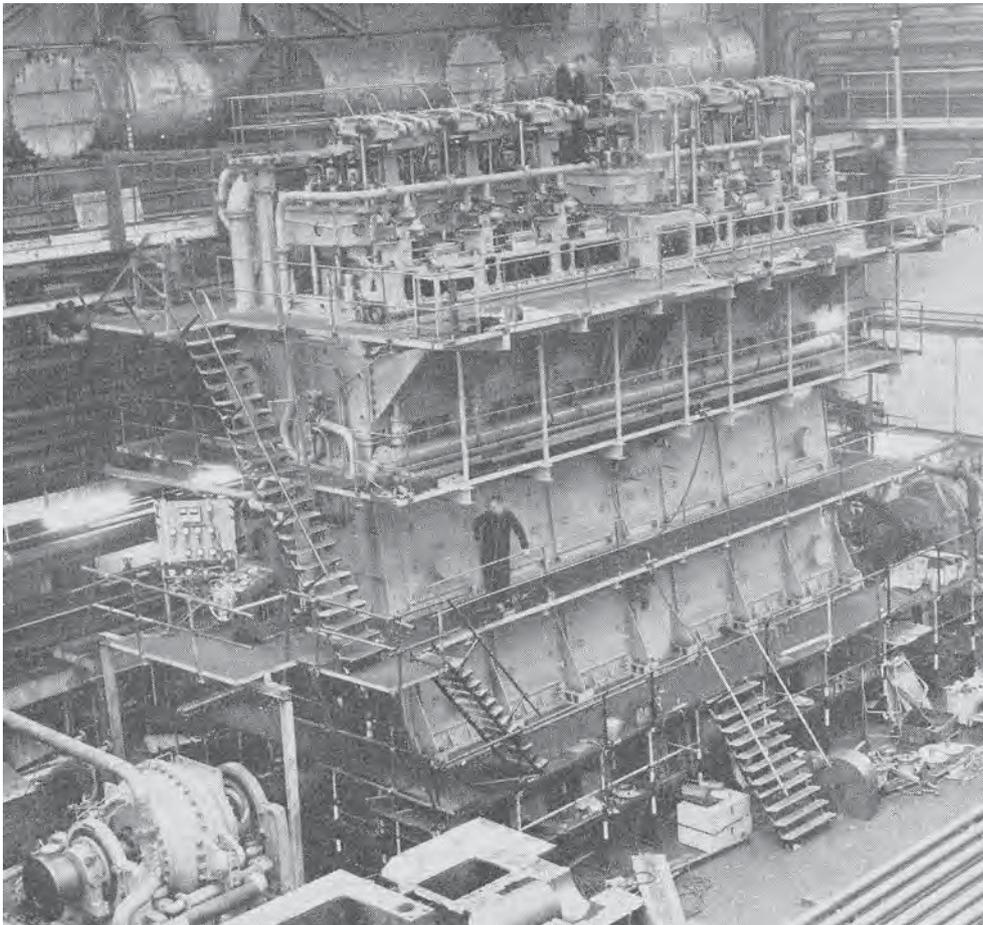
Fig. 48   SW90/170 at testbed in Hengelo, bore 900 mm, stroke 1700 mm, 3300 BHP/cyl at 115 rpm and
          Mep = 12 bar. (MS [36])

5. Götaverken stops in 1972

In 1964 Götaverken launch a new series of VGS-U-engines with bores of 850, 750, 630 mm, later to be followed by the 520 mm bore engine. This is indeed a modernisation and up-rating programme. Now lessons learned in the past and new possibilities are combined in this all welded design shown in Fig. 49.
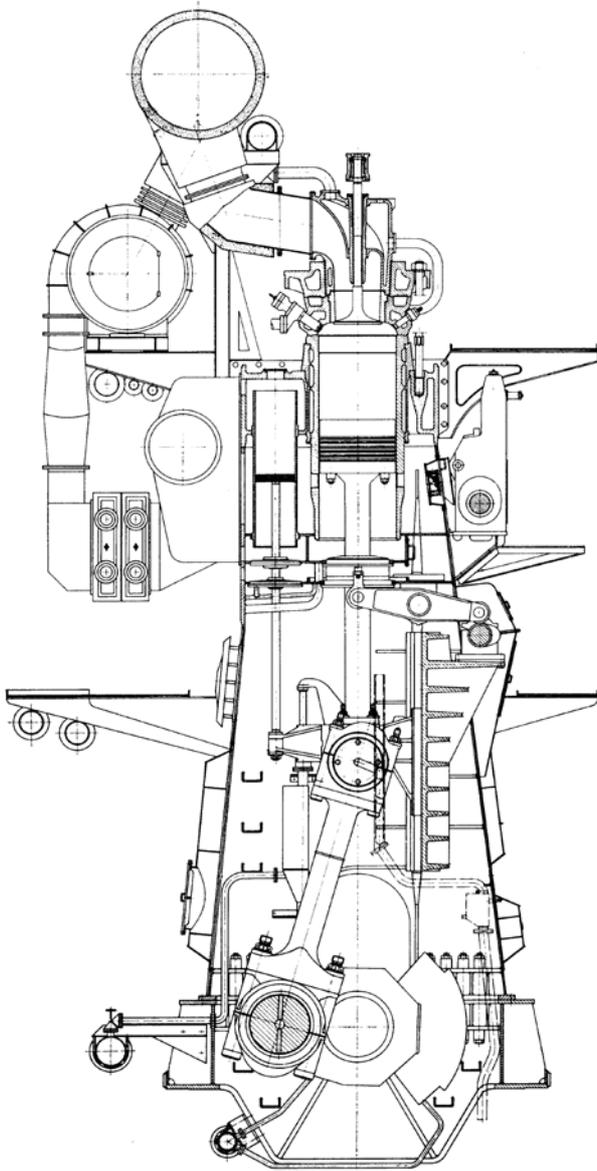
Fig. 49   Cross section 850/1700VGS-U, all welded structure and separate camshaft, bore 850 mm, stroke 1700 mm, yielding 2400 BHP/cyl at 119 rpm and Mep = 10.8 bar. (GV)

The engines are as a start rated to a mean effective pressure of 9.2 bar, but to get any further thermal loaded components had to be scrutinized to reduce temperatures and stresses.

In Fig.50 the modification done to the combustion chamber is shown, most remarkable is the reduction of the exhaust valve disk from 58 to 51 per cent of the cylinder bore, but the higher position of the valve spindle will together with the reduced material thickness contribute to a reduction of the spindle temperatures. Further better water circulation is arranged in both the cylinder cover and liner, where the temperature on the liner surface at position of the upper piston ring in top dead centre is as low as 170°C. GV contemplate water cooling of the pistons, but refrain because of the risks of contaminating the crankcase due to water leaks. Instead GV decides to do work on modification of the present piston design. The result is a change of the contour of the inner part of the piston as seen in Fig. 51, which reduced the maximum temperature facing the combustion chamber with 120 °C to 455 °C, a very acceptable level to avoid hot corrosion (burning) [38].
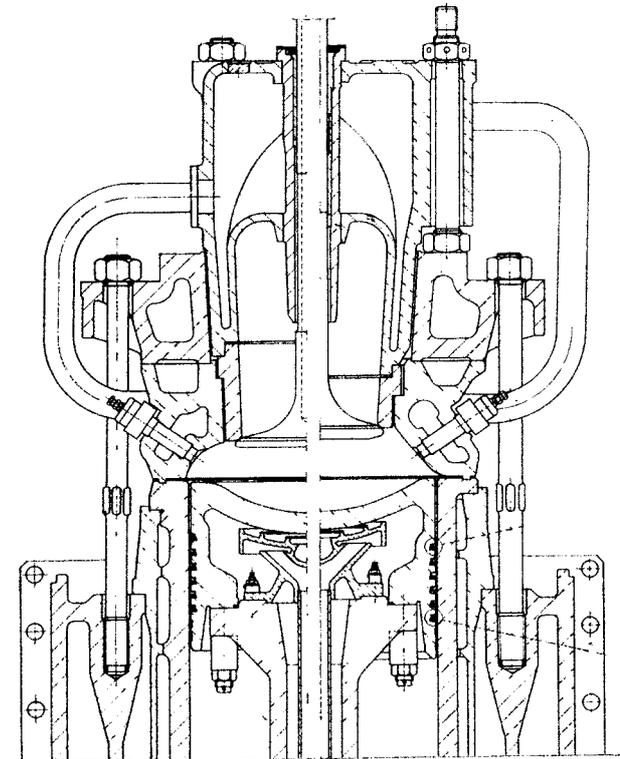
Fig. 50   Combustion chamber development for 850/1700VGS-U. (GV)

Most remarkable is the modification of the characteristic GV exhaust valve actuating gear. The pull rods are no longer driven from a cam segment on the crankshaft web, but are now connected to a fork shaped lever actuated by cams on a separate camshaft. The actuating gear is contained entirely in the crankcase which at least has some lubrication merits; however the vertical actuation of the exhaust valve spindle, which is the key parameter of the design, is still maintained.

Much less modification has been done to the crosshead design, Fig.52. The pin diameter has been increased to cope with the increased firing pressure a consequence of the increased mean pressure, and the connecting rod ration has been reduced from 1:4 to 1:3.6, which provides a higher periphery velocity, hence increased oil film thickness.
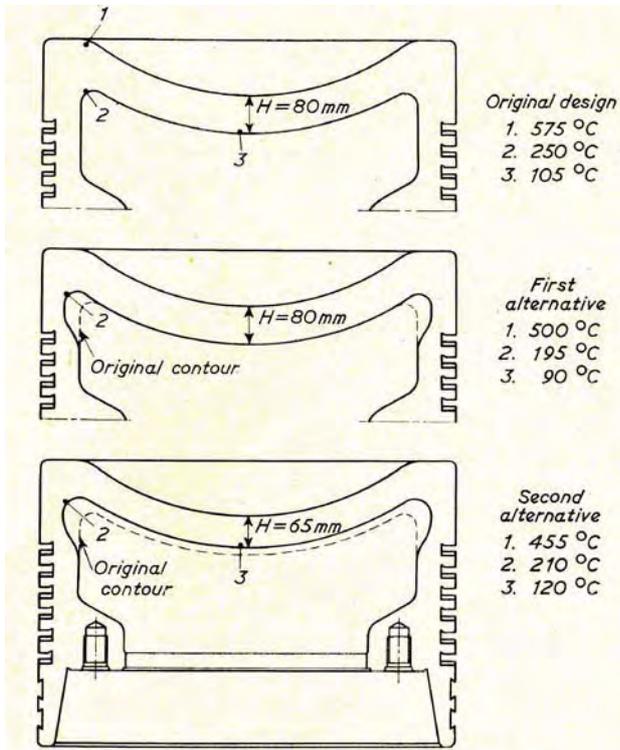
Fig. 51 Piston crown development for the 850/1700VGS-U engine. (GV)
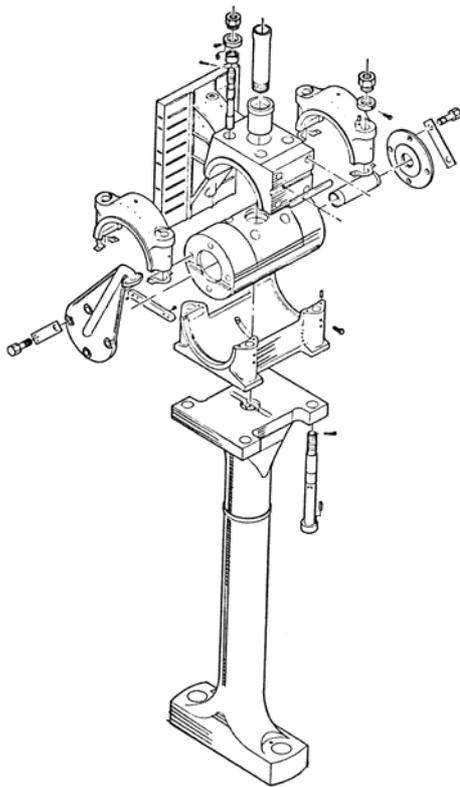
Fig. 52 Crosshead design for VGS-U engines. (MS [39])

The reduced connecting rod ratio serves of course to reduce the engine building height as a main purpose [39].

The welded engines are now Götaverken standard. The concept is interesting and different from what is used by all other designers in the field. The entire entablature comprise one cylinder only and staybolts are not used, Fig. 53. Instead a set of bolts are securing the connection to the bedplate.

Fig. 53 Engine columns ready for assembly. (GV [3])

The units require minimum machined surfaces and further the cylinder units can be completely assembled before being placed on the welded bedplate. In the same way the units are used when shipping the engine to the shipyards. GV's design witness that they have faith in their welding procedures and quality, and not to forget, they have gained a lot of experience since the first welded 520/900VGS engine was produced in 1944. All other engine designers apply through-going stay bolts to compress the welded areas of their bedplate and entablature and thus reduce tensile stresses, but this is not necessary on the GV engine – as the design has a very modest varying stress level.

This new engine series will more or less entirely be built at Götaverken's own premises; unfortunately the licensees only build a couple and no 850 bore engine. Of the total 539 turbocharged GV engines to be built only one quarter is made by the licensees and in the late 60's the market share of GV-engines is pendling around 3 %. No engine development can be paid by so few produced engines. To make things worse Götaverken have in co-operation with other Swedish engine builders embarked on the development of a big medium speed engine – the UDAB project that never was commercialised.

The most logical thing to do for Götaverken was to continue engine production in their factory based on a license, and in 1971 they re-entered in an agreement with Burmeister & Wain. The technical development staff and access to test facilities in Gothenburg was taken over by B&W for a period of time, which actually helped B&W in their modernisation programme – the K-GF series of engines as mentioned in the next chapter.

6. Engine modernisation

B&W

Several of the engine designers had in the beginning of the 70's engine series that needed a brush up for one reason or another.

As mentioned previously B&W's K-EF engines were just an uprating of the VT2BF types. A thorough revision of many components was required. With Carsten Andersen as new technical director the design process was carefully planned and followed up. The basic scavenging principles were maintained, but compared to previous designs new emphasis were put on optimising production and mounting work. The engines were thus equipped with fully welded frame boxes with mounted on guide planes. This was indeed also complying with the user's wishes – for long time the B&W engines had been compared with robust Danish agricultural machines, whereas the Sulzer engines were compared to Swiss clockwork – to say it, B&W engines were not tight, and oil was seeping out in many places.
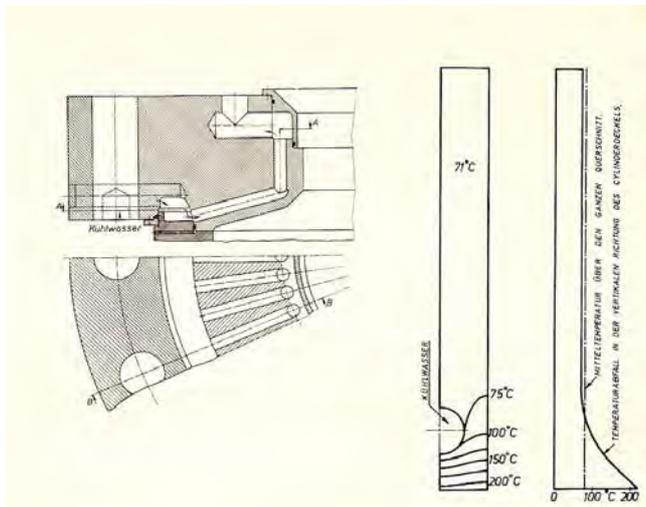
Fig. 54   Bore cooled Cylinder Cover, principal calculation of temperature distribution at Mep = 12 bar (below section A-A, above section B-B) (MTZ [40])

The K98FF engine experience made it also quite clear that some of the thermal loaded components needed upgrading.

The previously mentioned cylinder cover difficulties were solved by the pioneering design of a bore cooled cylinder cover, now used in all modern engines irrespective of make. The idea behind this design is briefly to control varying stresses due to ignition forces by the height of the cover more or less independent of the thermal stresses that in principle are proportional to differences between the mean tempe-rature of the block and actual temperature [40], Fig 54. Even though the basic concept was sound, there were once again some children's diseases to cure such as cracks in welded-on cooling jackets etc.
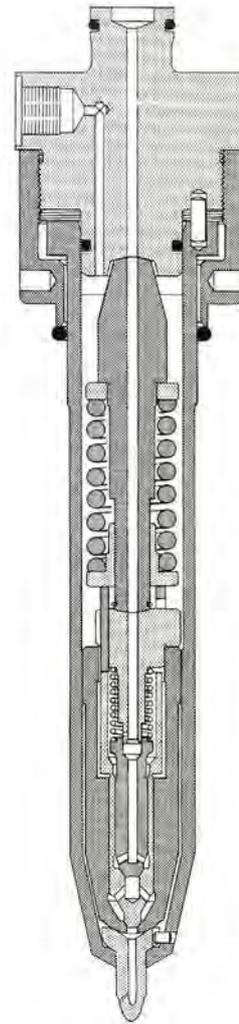
Fig. 55   Slim Fuel Valve for all B&W 2-stroke engines (B&W)

Slim, elegant, light, uncooled fuel valves were designed to fit into these covers Fig, 55, a design really appreciated by sea going engineers.

Further the engines were provided with a hydraulic actuation system for the exhaust valve, Fig. 56, never seen before on a two-stroke crosshead engine, however previously tested at sea on a K84EF engine [41]. A major advantage apart from lower first costs is the elimination of the transverse force acting on the valve spindle. This is an unavoidable consequence of the rocker arm movement that causes high wear rates of the guiding spindle-bushing – something that Götaverken, as mentioned previously, solved in a different way. Today all 2-stroke engines use hydraulic activation.

The new engines series was named K-GF and consisted of 4 bores, a K90GF as the first followed by K80, K67 and K45GF, where the old Götaverken staff designed the 80 bore engine. All engines with a stroke to bore ratio ~2 and a mep of ~ 11.5 bar.
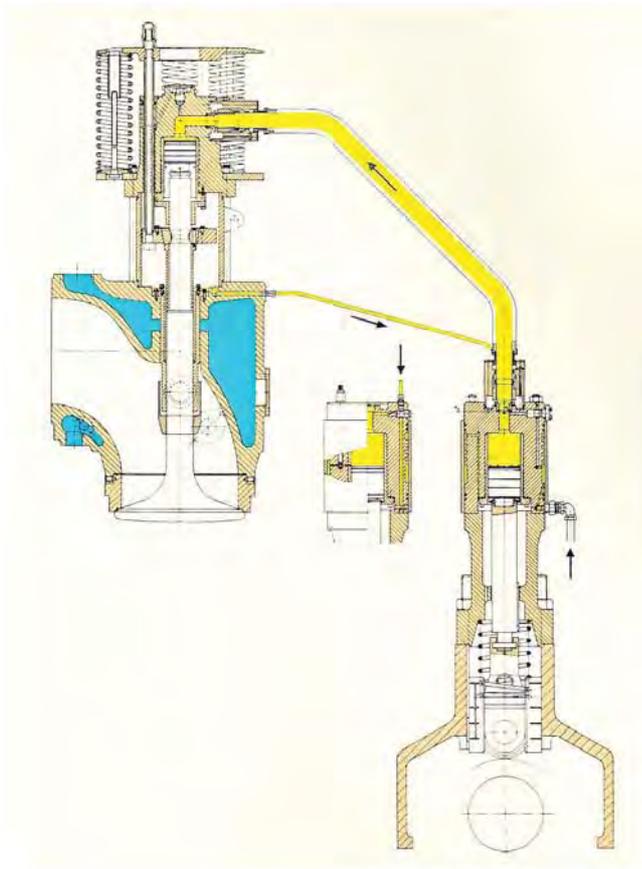
Fig. 56 Hydraulically activated exhaust valve for the K90GF engine series with the gear consisting of a piston pump driven from the roller guide by a cam disc on the camshaft and delivering pressure oil to the working cylinder on top of the valve spindle. (B&W)

GMT

In the beginning of the 70's Fiat Grandi Motori in Turin and state owned Ansaldo S.P.A. in Genova, previously building B&W engines on license, are combined in "Grandi Motori Trieste", GMT and move to a big, brand new diesel engine factory in Trieste. Five years after the beginning of the GMT production, modernisation and uprating of the previously mentioned engines is prepared, and a new B600 long stroke engine with a stroke to bore ratio of 2.08 is developed. The B-series of engines then consists of: B600, B780, B900 and B1060 [42].

The new engine series offers 15 % higher power at lower specific fuel consumption, about 150 g/BHP. The mean effective pressure is close to 12 bar and the corresponding maximum pressure 95 bar. The most important changes required to accommodate the increased thermal load and maximum pressure are:
1. Improved efficiency of the scavenging air coolers
2. Increased capacity of the injection system and reduced injection duration
3. Reinforcement of the engine structure and improvement of oil tightness
4. Improved oil film in cross head bearings
5. New cylinder covers with reduced varying stresses

Most interesting is the new crosshead design – the so called eccentric-type bearing where the pin and the matching bearing shells are divided into two zones eccentric to each other. The design is shown in an exploded view in Fig. 57. The movement of the connecting rod will cause a lift of the pin relative to the bearing counterpart and allow oil to fill the clearance while the other part of the bearing is carrying the load. The two bearing zones are not having the same size and the wider, inner part takes the load when the forces are biggest from just before the top dead centre to somewhat after the bottom dead centre. This concept is applied in all following GMT-engines even in engines with a full width bearing.
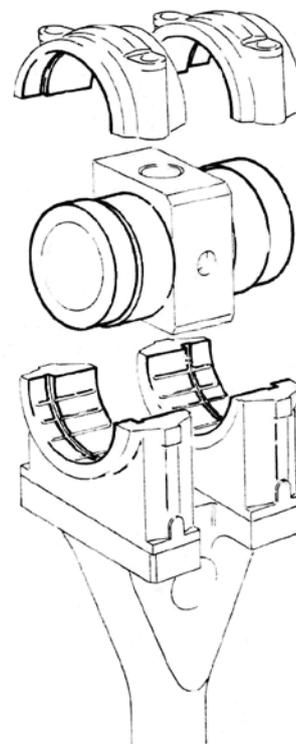
The cylinder cover is now made of a forged block and bore cooled following the same concept as in the B&W and Sulzer engines.

Fig. 57 Crosshead design for the GMT B-type engines with eccentric journals. (MTZ [42])

MAN

In 1968 MAN are, after discussions with licensees, ready with a development program for modernized KZ machines to be named KSZ [43] where the S stands for "simplified Service". The engine series consist of the 105-engine (to be updated) and new KSZ 900/160, 78/155 and 70/125 engines. The effective mean pressure lies between 10.3 and 10.7 bar with a firing pressure about 80 bar. These engines are all constant pressure charged as mentioned in the "Super large Bore" section.

The essential design upgrades are:
1. Crosshead with full width bearing
2. Piston as for the KSZ105/180 for the two biggest bores but only with a single annular support for the small engines.
3. Reduced connecting rod ratio (1:3.6) for the 3 smallest engines, to reduce engine building height.

In 1973 the KSZ engines are replaced with KSZ-A types where the mean effective pressure of the engines is increased to 11.2 to 11.8 bar. Now a higher degree of turbocharging is required and an exhaust diffuser is introduced after each cylinder as also seen later. The supercharging system is further simplified; the under-piston pumps are deleted and replaced with electric blowers in series with the turbocharger, but only for use at part load and during acceleration, Fig 58. Pistons are once again modified and now a build-up type is preferred to get control of the quality of the inner surfaces and thus increase the fatigue strength of the crown. So in 1973 MAN are well prepared for fighting their main competitors.
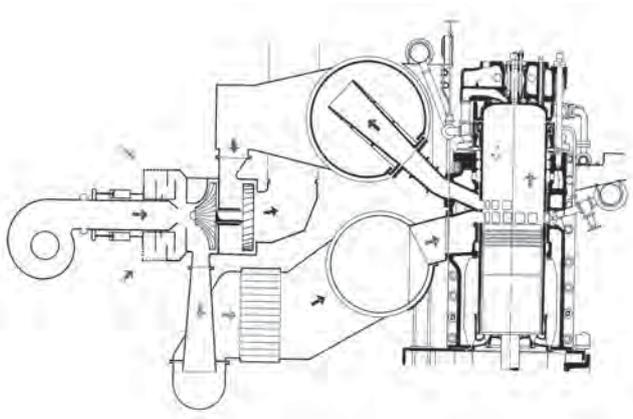
Fig. 58   Constant pressure system with electric blower in series for use during low load. (MAN)

90-bore engines, 1973	BHP/cyl, MCR	RPM	Mep, bar	Pmax, bar
MAN KSZ90/160A	3330	122	11.8	90
B&W K90GF	3410	114	11.6	90
Sulzer RND90	2900	122	10.4	84

However, the low number of revolutions of the uniflow scavenged B&W engine is attractive for the shipbuilder, as it means higher propulsion efficiency.

Sulzer

As just shown in the previous table Sulzer need to catch up in order not to lose their leading market position, and in 1975 they are ready with an all metric version of the RND – the RNDM series of engines.

Their 90-bore engine is now on line with competition; however, something happens in parallel to this introduction, the long stroke engines are marching in, as reported later.

	BHP/cyl, MCR	RPM	Mep, bar	Pmax, bar
Sulzer RND90M	3350	122	12.3	92

The RNDM-types have the same bore, stroke and rpm as the RND counterpart, however the higher overall efficiency of turbochargers have facilitated an 15 % uprating with but few design modifications: The cylinder cover is now as on the B&W engines bore cooled and also with a lower split between cover and liner to protect the liner from the highest thermal load. Actually the only really new modification is found in the crosshead bearings that now are equipped with thin tin-aluminium alloy lined steel shells. The fatigue limit of this bearing type and its load carrying capacity is considerably higher than that of a white metal bearing – the question is if the bearing sensitivity to dirt is adequate – white metal has an unsurpassed imbedability to dirt particles. Sulzer make use of well proved 4-stroke technology, but many 2-stroke protagonists believe that it is essential to preserve 2-stroke low speed features, where robustness is the keyword. However, no serious accidents happened caused by this new bearing design.

Additionally the RNDM had a new accumulator-type of cylinder lubrication system – the CLU-1.
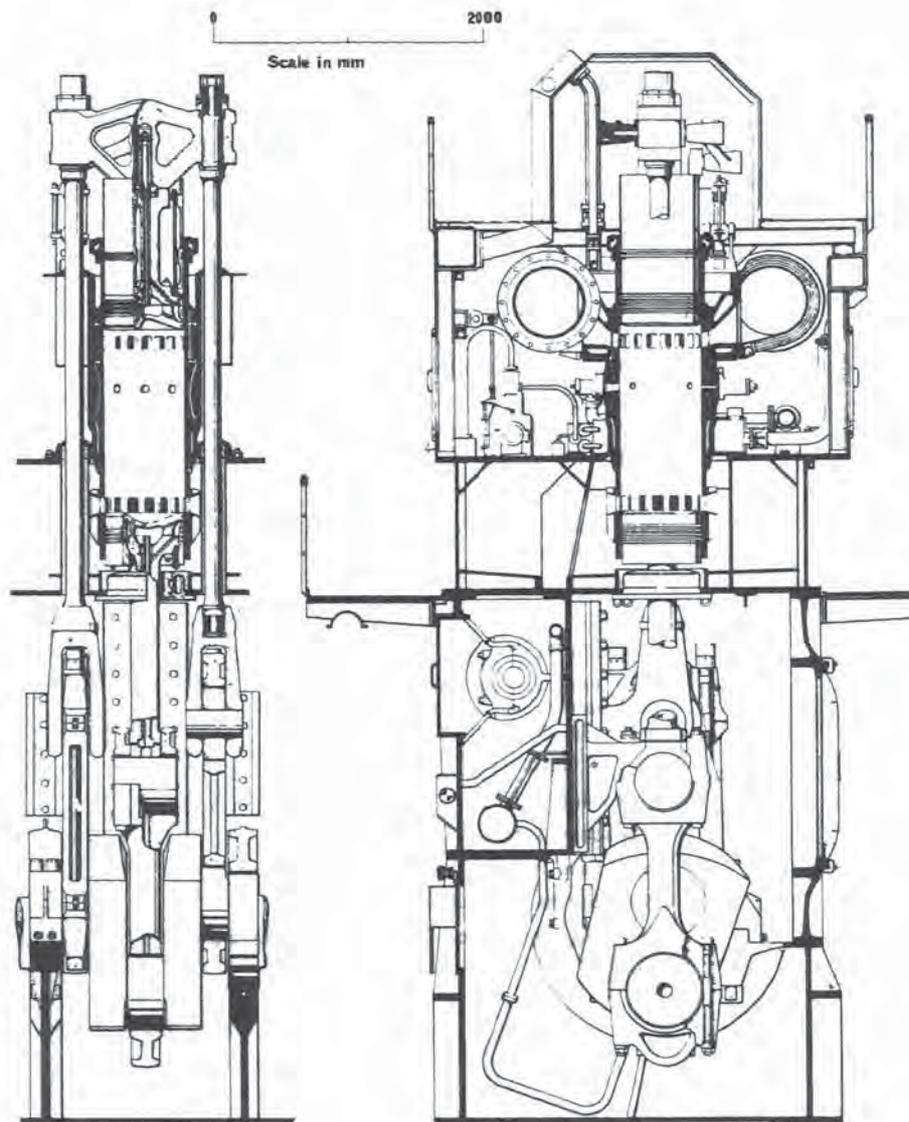
Fig. 59 The "Seahorse" engine. Projected and tested to 2500BHP/cyl at 300 rpm and Mep 10.7 bar. (CIMAC [45])

In 1969, under the inspiration of Mr. E.P.Crowdy technical director of Hawthorn Leslie & Co (a licensee of Sulzer and Doxford), it was announced that Doxford and Hawthorn had joined forces in a research programme on an engine to augment the Doxford J-series and provide powers up to 70.000 BHP. The concept was based on the view that: "the optimum marine engine must be able to burn the heaviest fuels with complete reliability, have only one cylinder size to cover the complete range, thus making possible world-wide provision of interchangeable spares and must permit propeller speed to suit vessels of all sizes" [44]. Of course they were choosing an opposed piston crosshead engine, and not surprisingly they choose a 580 mm bore engine with a combined stroke of 1300 mm, developing 2500 BHP/cyl. at 300 rpm with a mean effective pressure of 10.7 bar. A cross section of the "Seahorse", as the new engine was called, is shown in Fig. 59 [45]. The development work including testing of a prototype engine was finalised in 1976 after many interruptions due to financial difficulties.

The opposed piston engine is in principle very suited as a medium speed engine, but the seahorse concept was actually positioned between two chairs – slow speed and medium speed, and the latter engine type was very cost competitive with its higher number of revolutions. Further and unfortunately the "Seahorse" became a marketable product at a time where shipbuilding was hit by depression, and the engine was never sold.

Mitsubishi

Mitsubishi modernise their engine programme in a way that many have talked about for quite some time, as reported in the next chapter.

7. Two-stage Turbocharging

I the first half of the 70's engine designers sees an end to further exploring the power potential of the engines as the turbochargers not have the efficiency to yield a higher pressure ratio in a single stage. Two-stage turbocharging becomes more realistic, than it was for Stork in the 60's.

Sulzer first undertook tests with two-stage turbocharging in the mid 1960's and continued in 1975 on three cylinders of an 8RND90, with a mean effective pressure up to 16 to 17 bar. However, the added complications and costs prevents realisation of new 2-stage turbocharged designs.

B&W have as a consequence of the increased demand for high powered super tankers where a 300.000–400.000 tdw tanker would require 30.000 to 45.000 BHP at 80 to 70 rpm drafted a project, the L94NF with 5500 BHP/cyl at 83 rpm and a mean effective pressure of 17 bar. 2-stage turbocharging was the only way to reach this high rating [1]. The new license agreement with Götaverken included as mentioned the possibility to utilise GV's test facilities, and 2-stage trials were run on the 3 cylinder 750VGS-U in Gothenburg, Fig.60, where even Mitsubishi were invited to the testbed. The total collapse of the tanker market during 1973–74 made B&W burry the project.

Mitsubishi finds in 1975, that the best way to improve cost performance of their engines is to increase the power rate and introduce 2-stage turbocharging – their E-type engines [46]. The E-type obtain abt. 30 % higher power output within the same frame size as the predecessor D-type with single stage turbocharging, meaning also a substantial saving in engine room space requirements.

The impressive potential in increasing the mean effective pressure without compromising thermal load was demonstrated by Mitsubishi, who so far are the only company to commercialise 2-stage turbocharging.

The first production unit is an 8UEC52/105E, and the arrangement of the 2-stage turbocharging system is shown in Fig. 61, and the engine cross section in Fig. 62. To accommodate the increased firing pressure in the cylinders the engine frame, bearings and combustion chamber components are all reinforced relatively to the D-type and the liner is bore cooled in the upper part as in the B&W KG-F-engines.

Fig. 60   Two-stage turbocharging test at Götaverken's premises in Gothenburg performed on a 3-cylinder 750VGS-U engine (B&W)
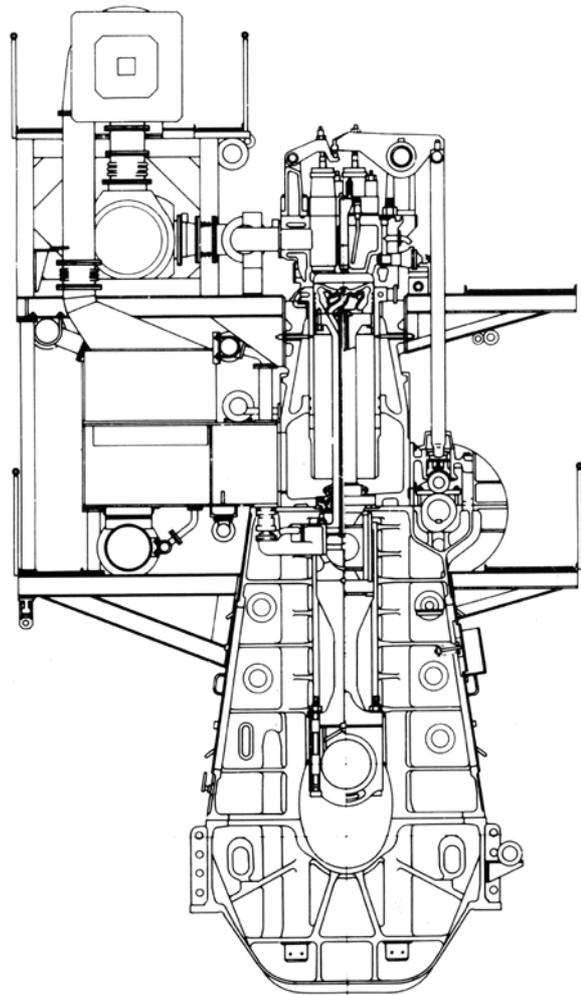
Fig. 61   Two-stage turbocharging system applied to 8 cylinder 52-engine (MS [46])
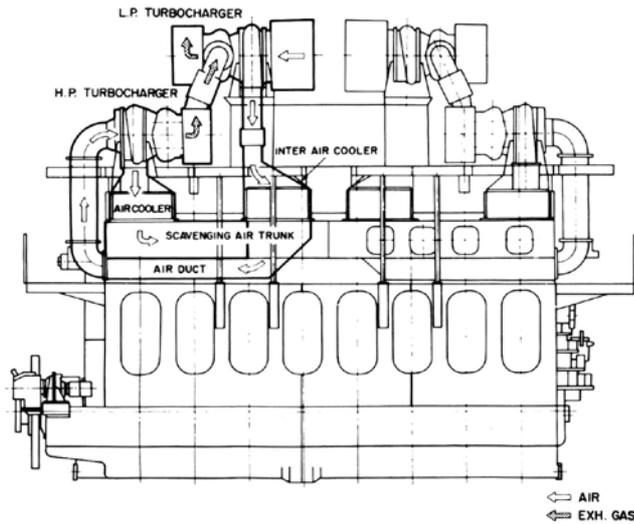
Fig. 62   The world's first 2-stage turbocharged 2-stroke engine, MHI's 8UEC52/105E, bore 520 mm, stroke 1050 mm, 1330 BHP/cyl (980 kW/cyl) at 175 rpm and Mep 15.1 bar. (MS [46])
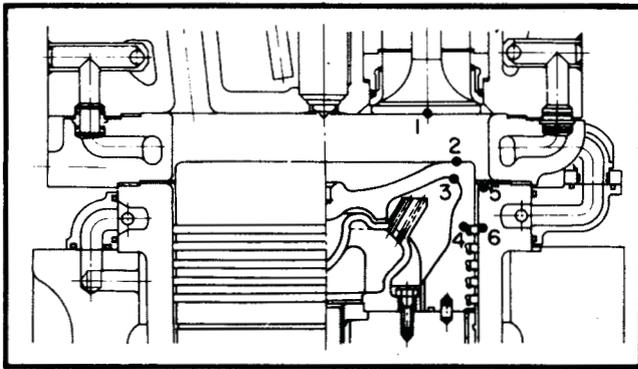
Model Point	UEC52/105D (M.e.p 11·9 kg/cm²) :	UEC52/105E (M.e.p. 15·4 kg/cm²) :
1	570	560
2	480	430
3	250	205
4	150	140
5	320	240
6	220	175

Fig. 63   Comparison of engine performance between the single stage and the two stage turbocharged UEC52 –engine. (MS [46])

It is interesting to compare the performance of the D and E-type and Fig. 63 shows clearly that the increased power is not bought for free; the maximum pressure is now 110 bar in order to maintain the specific fuel consumption.

The comparison on thermal load is shown in Fig. 64, and the temperatures are all moderate – maybe not surprising as the cylinder diameter relatively small, but the remarkable thing is, that the highly rated 2-stage turbocharged engine actually comes out much better than the predecessor. In addition to the "normal" stroke to bore ratio of ~2 the engine series is

supplemented with a 52/125E and a 60/150E long stroke engine with a stroke to bore ratio of ~2.5, see later.

The service result of these engines in respect of cylinder liner wear is 0,01 to 0,04 mm/1000hours, which must be considered very satisfactory and the ring wear is about 10 times higher, which must be considered normal.

MAN announces [47] in February 1980 that they have adopted 2-stage turbocharging in their KSZC/CL-type of engines (see later). They have developed turbochargers where both stages are accommodated in one casing. In the 1980 production programme these engines are indentified with the suffix "H" which stands for high efficiency turbocharging.
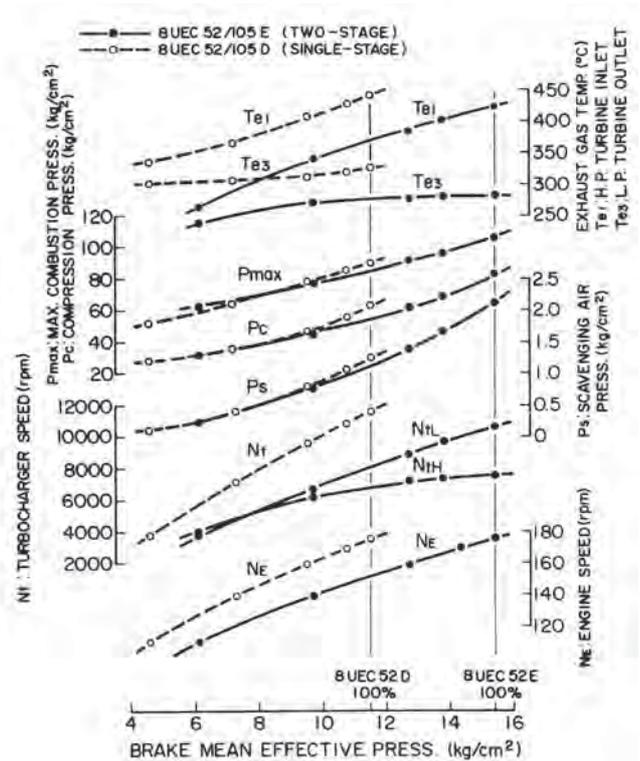
Fig. 64   Comparison of combustion chamber wall temperatures between the single stage and the two stage turbocharged UEC52-engine. (MS [46])

The engine power remains the same, but the fuel consumption between 80 and 85 % MCR is reduced by 3 g/BHPh, however, the engine price will be 4 to 5 % higher. To quote Mr. Schiff, the technical director: " Every ship-owner can work for himself whether, at the present-day and future prices of heavy fuel oil, the saving in fuel will offset this higher capital expenditure". The H-engines never went to sea.

8. Fuel quality versus engine reliability

It was an opportunity as well as an associated problem that all the just mentioned crosshead engines were running on some kind of heavy fuel.

The opportunity is related to the fact that the trunk engines at that time did not run safely on heavy fuel, but the competition in form of steam plants could make use of these cheap bunkers – now the crosshead engines could do that as well.

It was, however, not an easy task for the engineers to make it happen, mainly because heavy fuel is just a common name for what in fact is a variety of products with very, very wide specifications and also containing a lot of unspecified material. Normally the bunkers are classified according to the fuel viscosity in order to handle it, but additionally there are several handling parameters [48], such as specific gravity, pour point and flash point. As far as the engine condition is concerned, the conradson carbon and asphaltenes are influencing fouling of gas-ways and compatibility with other oils, sulphur, ash, cat-fines and water has an impact on corrosive wear, mechanical wear, thermal load and fouling, and lastly sodium and vanadium could be harmful to exhaust valve corrosion and turbocharger deposits.

Burning of heavy fuels has been and is still a subject of great interest to designers as well as engine users, but to get into a discussion of this subject is beyond the scope of this report. One can, however, notice that many, many discussions between designers and users has been related to the fuel influence – if this or that component failed because of the fuel used or, if it was a question of poor design.

The early use of heavy fuel showed that something had to done to save the cylinder condition. Something beyond the capability of the designers. The helping hand came from the oil industry with their "invention" of highly alkaline cylinder oils to combat the corrosive cylinder liner wear.

II FUEL CONSUMPTION, RELIABILITY & MULTIPLICITY

9. Long stroke and slow running engines, engines for geared application

B&W

The oil crisis 1973-74 increased the interest in fuel economy. At B&W the development department had for some time been considering to design a long stroke version of the K-GF series of engines. The mean piston speed and effective pressure should be retained to maintain engine reliability, but the reduced engine rpm would provide a substantial improve-ment of the propeller efficiency provided the yards also agreed to modify the hull design. In 1976 the first L67GF engine was ready for test at Mitsui – it was a relatively simple job to design the new series, as the majority of the components and/or concepts were reused. A comparison of the K and L-series of engines is shown in the table below and in Fig. 65.

Engine	K90GF	L90GF	K80GF	L80GF	K67GF	L67GF	L55GF	K45GF	L45GF
Stroke,mm	1800	2180	1600	1950	1400	1700	1380	900	1200
BHP/cyl	3410	3410	2640	2640	1870	1870	1340	880	850
rpm	114	94	126	103	145	119	150	228	164
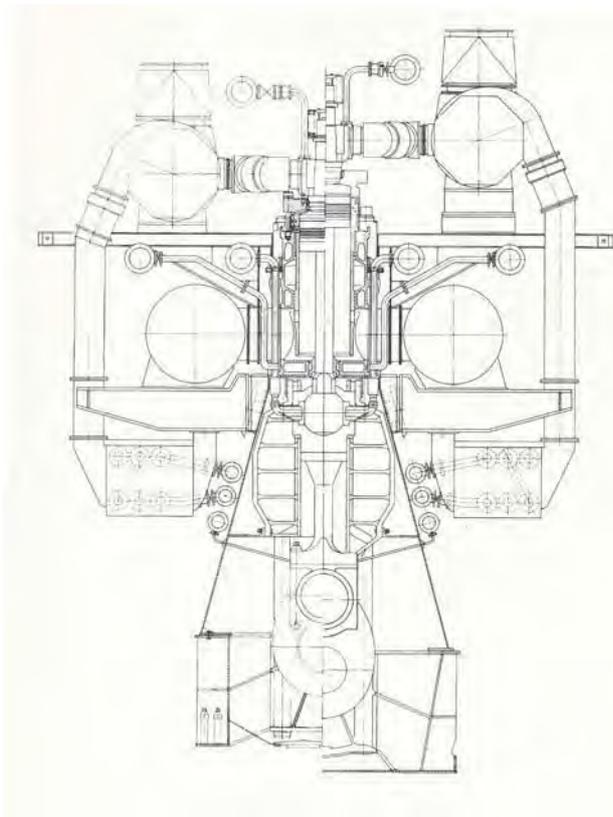
Fig. 65   Comparison of short and long stroke engines, K/L67GF (B&W)

B&W claim that the 18 percent rpm reduction will result in approximately 5 % reduction of fuel consumption, actually 3 to 5 % would have been more realistic, and on top of this about 2 g/BHPh reduction in fuel consumption because of the more favourable stroke to bore ratio.

It was obvious, that the long stroke engine would put the loop and cross scavenged engines under heavy pressure as these concepts require a relatively short stroke because of the scavenging process in order to obtain high air purity in the cylinders and low fuel consumption. B&W had hoped that the long stroke engines would bring back some of the lost market-share, but they were surprised of the ingenuity of their main competitors to cope with the new situation, as will be seen in the following.

Mitsubishi

At the same time as B&W Mitsubishi had, as previously mentioned, included two long stroke engines in their 2-stage turbocharged engine programme; the UEC52/125 and UEC60/150. These engines are, as the B&W engines hav-ing a stroke to bore ratio of ~2.5 and retain by and large the mean piston speed and pressure as on the standard engines, from which they design-wise are hard to distinguish. The long stroke programme is later completed with the addition of two smaller bore, constant pressure charged engines. This so called H-type will be dealt with later.

MAN

MAN has from 1975 replaced the KSZ-A with an uprated KSZ-B-type. The B engines are more or less a copy of the A-type having the same stroke to bore ratio, only the mean effective pressure is increased and so the power – in the range between 9 and 15 % dependent on the engine type.

More interesting is the "addition", the BL-type that is found on the sales list in 1977. The BL version is basically of the same design as the B version, however being slow running

yet maintaining identical maximum power. MAN has simply reduced engine rpm by 10 % and at the same time increased mep by the same percentage, and claim that the thermal load of combustion chamber components is proportional to the product of mean piston speed and effective pressure [49]. This is based on an empirical formula, however, the measured temperatures as shown in Fig. 66 clearly shows that wall temperatures are inclined to follow mep. Furthermore the fuel consumption is increased with ~2 g/BHPh relatively to the B-version. The BL-engine is an uprated B-type in disguise and has no added value to offer in terms of fuel consumption at full load. However, at part load a new variable injection timing system provides for a consumption reduction of ~ 3 g/BHPh at the most, around 75 % load.

Fig. 66   Comparison of combustion chamber temperatures between KSZ70/125B and KSZ70/125BL. (MAN)

The B-Engine design is much more attractive, as shown in Fig. 67. The construction of the engine frame now consists of a welded box (two for the 90-bore engine) instead of the A-frames. The smaller engines have got bore cooled pistons, with finger bores as in the Sulzer 56RLA design (see later), but with a quite different way of directing the coolant into the bores as illustrated in Fig. 68. This cooling method is also applied in the outer cylindrical region of the cylinder cover, Fig. 68, where it elegantly is combined with the traditional design. The 52-engine has even a bore cooled cylinder liner collar, and all engines are provided with cast-in cooling pipes between the exhaust ports.

For all the KSZ-B-type engines the cylinder cover is still a strong back design, where the bottom is clamped to the top by several studs thus prestressing the bottom plate. It is however difficult to avoid relative micro movements between the two parts and many covers of the big engines suffer from cracks in the bottom, steel part. It is difficult to understand why MAN not have adopted a fully bore cooled design, as most of the competitors, and as they have done in the piston design.
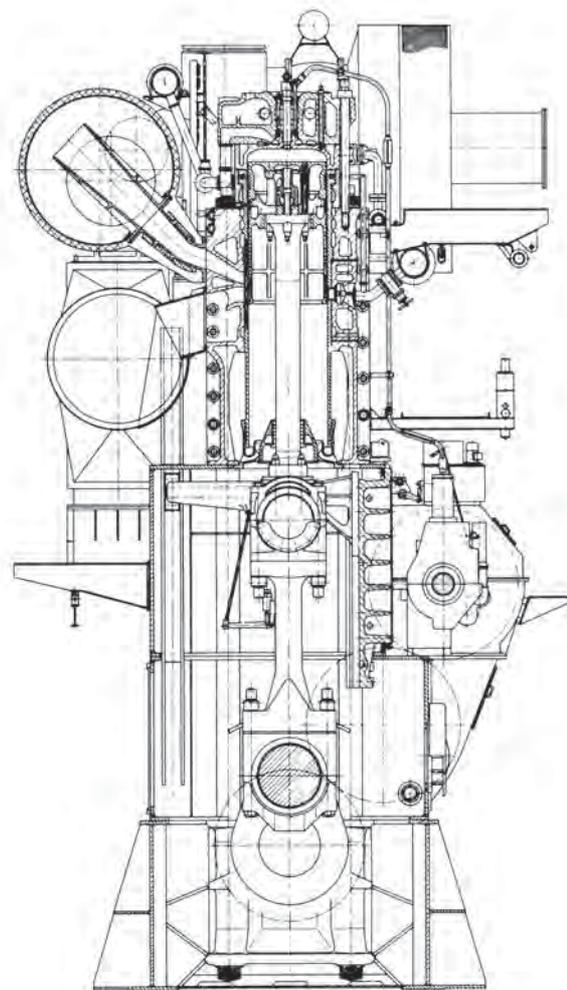

Fig. 67   Cross section of KSZ90BL, bore 900 mm, stroke 1600 mm, 3670 BHP/cyl (2700 kW/cyl) at 110 rpm and Mep 14.5bar. (MAN)
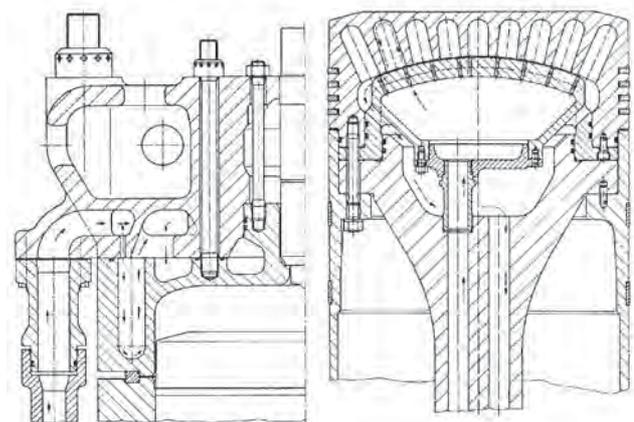
Fig. 68   Cylinder Cover and Piston, KSZ52/105. (MTZ [50])

Sulzer

Sulzer has just finalised their RND-M series of engines, but they have prepared themselves for the new era of low revolutions, as they in 1976 announce a 56 bore engine with a stroke to bore ratio of 2.05 – the RLA56, to be followed in 1978 by RLA90, and in 1979 the RLA66 with a stroke to bore ratio 2.10–2.12. This stroke to bore ratio seems to be the limit for the loop scavenged engines, and MAN reach the same conclusion.

GMT

In 1979 after 3 years of intensive study and development GMT introduces the CC600-engine as their answer to the increasing demand for low fuel costs. Whereas B&W have chosen to design a range of engines with long strokes and low revolutions in addition to their standard programme GMT have, as Doxford with their "Seahorse" engine, opted for a short stroke (stroke/bore = 1.3) engine ideal for geared installations. This gives of course the freedom to select optimum propeller speed, with a limited number of engine designs and provides also ideal conditions for use in vessels with limited head room. It is indeed a competitor to 4-stroke medium speed engines, as the 2-stroke characteristics of simplicity and robustness is maintained, but costs are relatively high.

The introduction rating is 1650 BHP/cyl at 250 RPM and 12.9 bar mean effective pressure and the fuel consumption is 146 g/BHPh.

The concept retains all the traditional GMT features, cross scavenging, constant pressure turbocharging, scavenge air pumps and the open under-piston space, but some new features are of interest.

All the components surrounding the combustion room are now planned to be of bore cooled design and water cooled, Fig. 69. The first engines will however have the normal strong back liner design, but when it comes to ratings exceeding the introduction figures it is all bore cooling. The liner design is especially interesting because the bores ends blindly which means that high stress concentrations are avoided in the top of the liner collar. This concept is later copied by MAN-B&W in their MC-engines.

The crankshaft is now fully forged, and the crosshead is having a full width bearing, however still maintaining the eccentric pin feature introduced in the B-type engines.
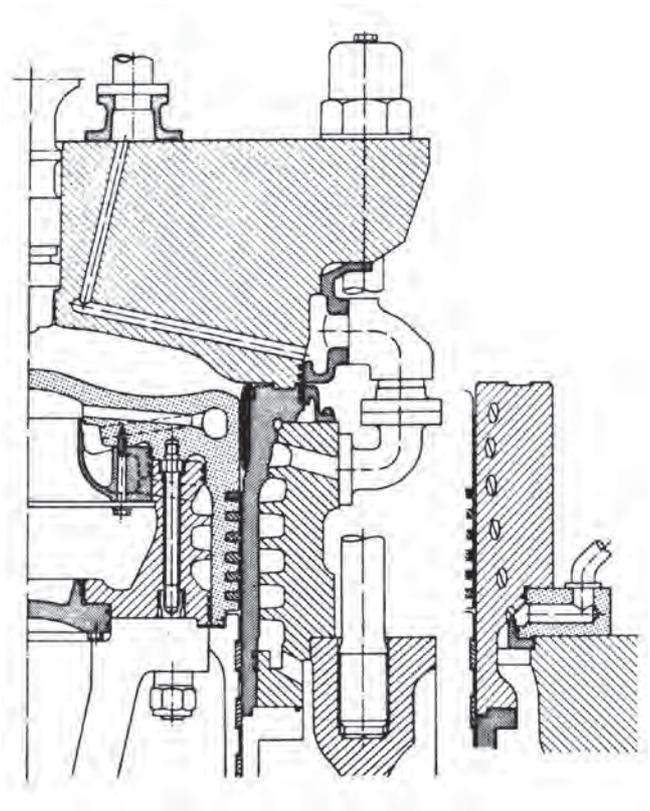
Fig. 69   Combustion chamber with finger bore cooled liners for the CC600 engine. (GMT)

The final 2-stroke crosshead engine programme consists in 1980 of:

Engine type	Bore, mm	Stroke,mm	Power, BHP	RPM	Mep, bar
B1060	1060	1900	4600	106	11.5
C900	900	1600	3700	110	13.1
C780	780	1600	2800	122	13.2
C600	600	1250	1650	160	12.9
CC600	600	800	1650	250	12.9

In mid 80's GMT decides to take a license from New Sulzer Diesel and that stops the development of some of the most interesting 2-stroke crosshead engines, unique in their concept and in many components .

Doxford

The J-engine was until 1975 designed with a 4 cylinder configuration as the smallest in the engine series having cylinder diameters of 580, 670 and 760 mm. After the experiments with the "Seahorse" engine it was found that constant pressure turbocharging instead of impulse charging would provide several advantages; one being that the side cranks which are connected to the upper piston, could be given a crank lead of 180° relative to the centre crank, which is driven by the lower piston whereas the impulse charged engines would require 188°. This in turn means that it now was possible to design engines with just 3 cylinders and obtain an even more

excellent mechanical balancing. As compactness is one of the competitive edges of the opposed piston engines the addition of the three cylinder configuration was an advantage ad par with the reduced number of components to overhaul and the increased reliability. An elevation of the 76J-C engine, as the constant pressure charged engine is called, is shown in Fig. 26, and the engine is seen on the test bed in Fig. 70.

In 1978 Doxford introduce short stroke versions of the J-C series of engines named JS-C, and the total engine programme in 1978 is seen below:

Engine type	BHP/cyl	RPM	Stroke mm	MEP, bar	No of Cyl.
58J-C	1750	150	1850	11.52	4
58JS-C	1833	220	1220	11.42	3–6
67J-C	2500	127	2140	11.52	3–6
67JS-C	2500	160	1700	11.51	3–6
76J-C	3000	123	2184	10.87	3–9
76JS-C	3000	155	1870	10.07	3–7
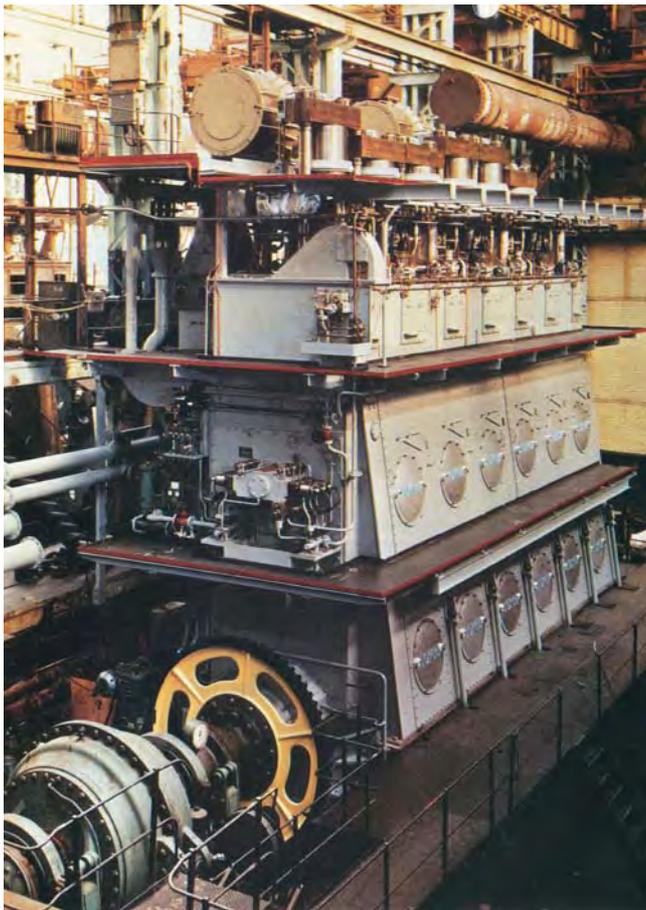
Fig. 70   J-type on testbed at the Pallion works in Sunderland. (Doxford)

Especially the short stroke version of the 58 bore engine, Fig. 71 is seen as a competitor to medium speed engines, but just as the "Seahorse" engine this engine was not a success, maybe because British shipbuilding at that time was shrinking. The fuel consumption of the listed engines was quoted to 145.5 g/BHPh which was too much compared to contemporary engines of other makes, but Doxford have apparently not the tools at hand to make use of the inherent advantages of their super long stroke engines.

Comparing the 58J-C to B&W's L55GFCA (see next section) as in the table above it is seen that the specific fuel consumption of the 58J-C engine is 6 g/BHPh higher than the B&W L55GFCA's. The influential parameter S/D is in favour of Doxford, whereas the Pmax/MEP relation is in favour of B&W – in this case they more or less outweigh each other. The question then remains, why is the 58J-C engine having such a high consumption? The main reason is that the maximum pressure is too low, but it can't be higher because the pressure influenced components are not designed to the higher values – maybe as a consequence of Doxford's limited experience with the use of modern calculation tools. Looking at the design of the cylinder liners, cast in one piece and bore cooled, Fig. 72, it is clear that the potential for higher cylinder pressure is present.

Other areas could influence the consumption, and looking at the piston design, Fig.73 suggests that the very low positioned piston rings will lead to big pressure losses in the scavenging ports and the strange wave cut top land would require a clearance that is higher than with a straight cut surface. The temperature level in the piston crown of this small bore size is low enough to allow a higher position of the piston rings.
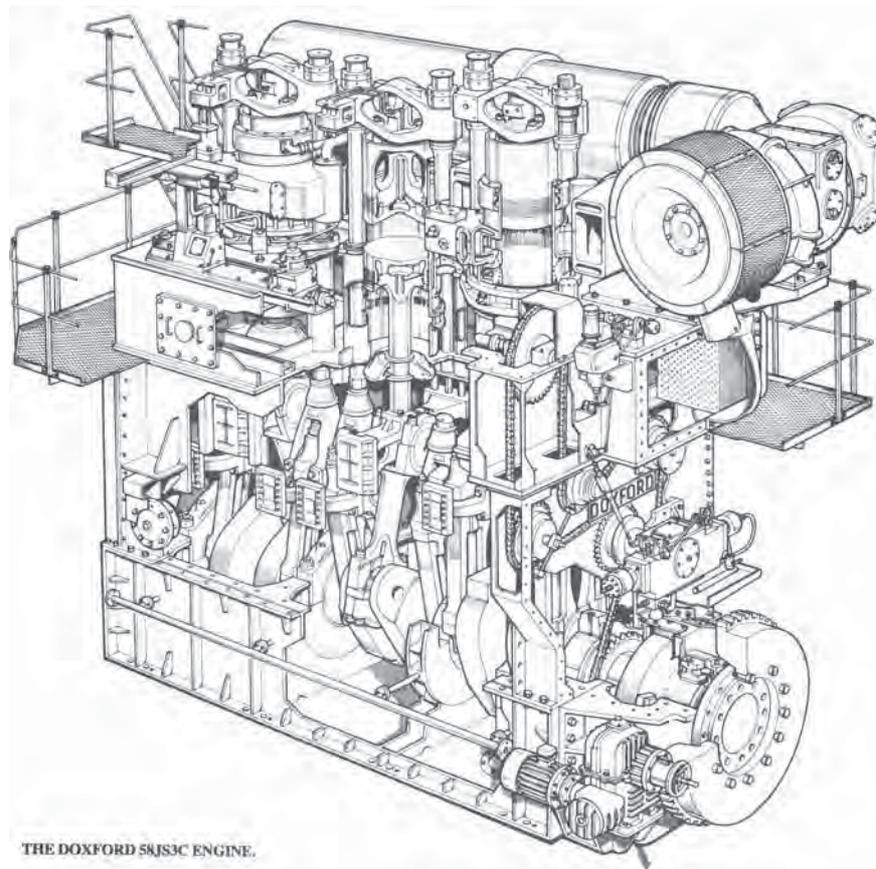
Fig. 71   Doxford 58JS3C engine, bore 580 mm, stroke 880/340 mm, 1833 BHP/cyl (1349kW/cyl) at 220 rpm and Mep 11.4 bar. (Doxford [17])

Engine type	BHP/cyl	RPM	Bore mm	Stroke mm	S/D	MEP bar	Pmax bar	Pm /MEP	SFOC* g/BHPh
58J-C	1750	150	580	1850	3.2	11.52	85	7.4	145.5
L55GFCA	1495	155	550	1380	2.5	13.00	100	7.7	139.5

* ISO ambient conditions, and fuel oil calorific value: 10.200 kcal/kg
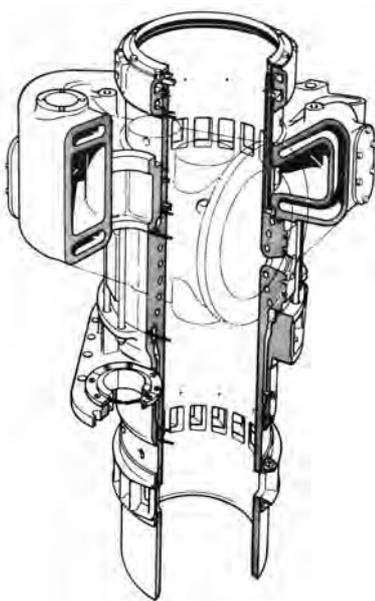
Fig. 72   Cylinder liner for J-C type engines. (Doxford [17])
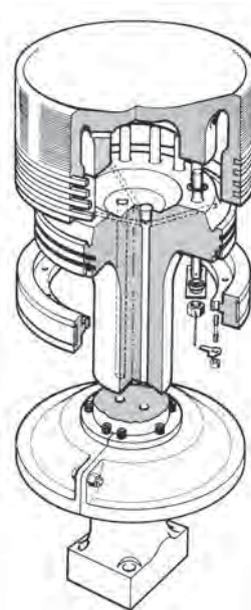
Fig. 73   Piston for 58JS3C. (Doxford [17])

A pity that Doxford chose to modernise their engine programme with the "Seahorse" engine instead of modernising the design of the super long stroke engines. Doxford is closed down in beginning of the 80's despite Sir Robert Atkinson's (Chairman of British Shipbuilders) desperate attempts to give the Sunderland Works a new chance.

10. The fuel consumption race

B&W

It took the maritime world by surprise when B&W in the summer 1978 suddenly, but well planned introduced a completely new engine series with 7% lower specific fuel consumption. During many years B&W had made their propaganda in relation to the benefits of Uniflow scavenging and inpulse turbocharging without assistance of part load blowers. The new engines were constant pressure charged! The gradual increase of turbocharger efficiency had made it possible to apply this system with just small auxiliary blower capacity at part load. B&W may have become a victim of their own propaganda, as it definitely would have been possible to introduce constant pressure earlier. The gain in engine efficiency is illustrated in Fig. 74, the P-V diagram of the engine cylinders.

The new charging system made it possible to offer engines with identical rating irrespective of cylinder number, what Stork so desperately wished in 1965, and even with better reliability as the exhaust temperature during the opening of the exhaust valves were lower which reduced the material temperatures of the valve spindles and the valve seat – a sore point for B&W in the competition. In November 1978 the new concept were tested on a 7L67GFC-engine in B&W's workshop in Copenhagen and the expectations were fulfilled [52].

Apart from the scavenging system the constant pressure engine is a copy of the impulse engine, the K/LGF, meaning that it is a fairly simple and cheap new development.

At the CIMAC congress in the spring 1979 Mr. A. Østergaard (B&W's chief design engineer) presented a paper [53], that by technical and economical argumentation should clear the way for a further power increase of the constant charged engines and document the large layout flexibility, of course with great emphasis on the fine, low specific fuel consumption. So, from spring 1979 the engines were rebaptised and offered with a 15 % increased power and unchanged specific fuel consumption – the CFCA-type was born. A cross section of the 12L90GFCA-engine is shown in Fig. 75; this engine has a specific fuel consumption of 142 g/BHPh at 95 bar maximum pressure.

Fig. 75   Cross section of the L90GFCA engine, bore 900 mm, stroke 2180 mm, yielding 3940 BHP/cyl (2900 kW/cyl) (MCR) at 97 rpm and Mep 12.9 bar. (B&W)

Fig. 74   Arrangement of exhaust pipes for 6 cylinder impluse- and constant pressure charged engines, and comparison of the lower part of the engine working diagram for impulse- and constant pressure charged engines (MS [52])

Sulzer

The B&W constant pressure engines put pressure on Sulzer to make a counterattack. They have already in September 1977 released that the RLA90 was under development. In the autumn 1978 at the SMM exhibition in Hamburg Sulzer announce the new RLA90. Peter Sulzer, vice president, had previously told the press [54], that it was Sulzer's aim to maintaining their position as market leader by providing "total economy" – it was obvious that the threat from B&W was recognized.

Except from the increased stroke to bore ratio and following reduced revolutions the RLA engines differ from the RNDM engines with a 3 g/BHPh lower consumption i.e. 146 g/BHPh.

Already a year after the introduction of the RLA engines, Fig.76 a RLB series of engines are offered with two ratings. A full rated engine (10 % higher rating than the RLA) as usual and a derated engine, a so called economy rated engine, with lower BHP and rpm, which provides for a fuel consumption reduction of 3 g/BHPh, Fig 77.

Simultaneously an optimisation of combustion and scavenging combined with use of the new VTR4-series turbo-chargers, with higher efficiency and pressure ratios, Fig. 78 & 79, is respon-sible for further reductions in fuel consumption. The development of turbochargers comes in very conveniently for Sulzer, but is of course generally a prerequisite for the continuous uprating of the engines.

In the performance diagram, Fig. 80, one will note rather low fuel consumption over the normal service range. This is due to a new feature – a simple mechanism that permits automatic control of the permissible maximum combustion pressure. The flat maximum pressure curve from 110 % down to about 85 % engine load is a consequence of this variable injection timing, the VIT. Unfortunately it is not possible to maintain the high maximum pressure further down in the low load range due to bearing loads being influenced both by the forces from combustion and the oscillating masses, and piston rings influenced by the pressure rise.

The combustion optimisation is also a consequence of an increase in maximum pressure, and the RLB version of 1980 has now reached 120 bar versus the RLA56 of 1976 with 98 bar.
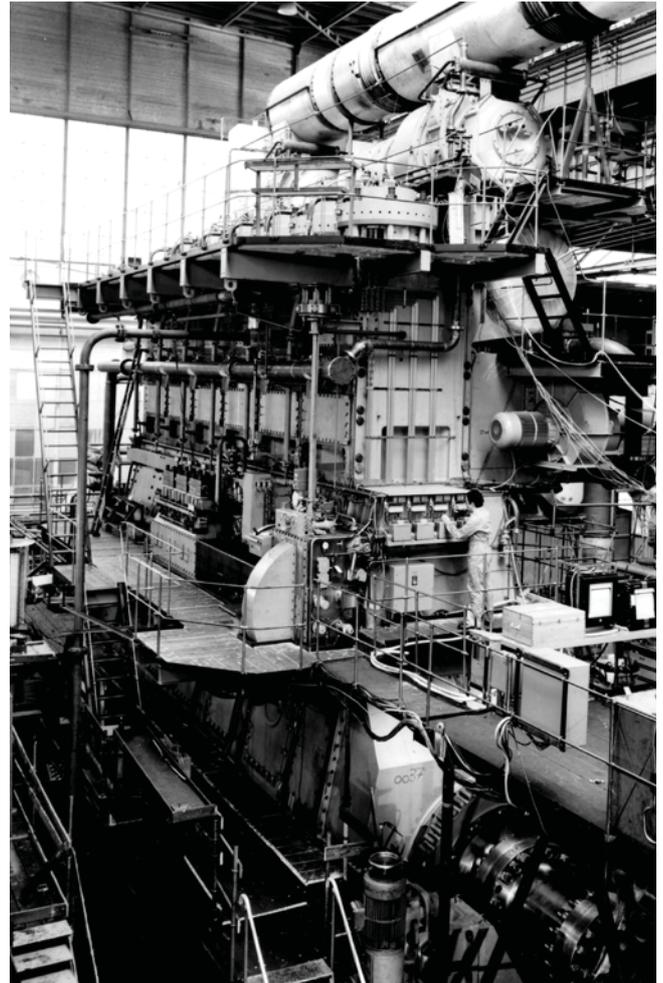
Fig. 76   Sulzer 6RLA90 on testbed. (Wärtsilä)
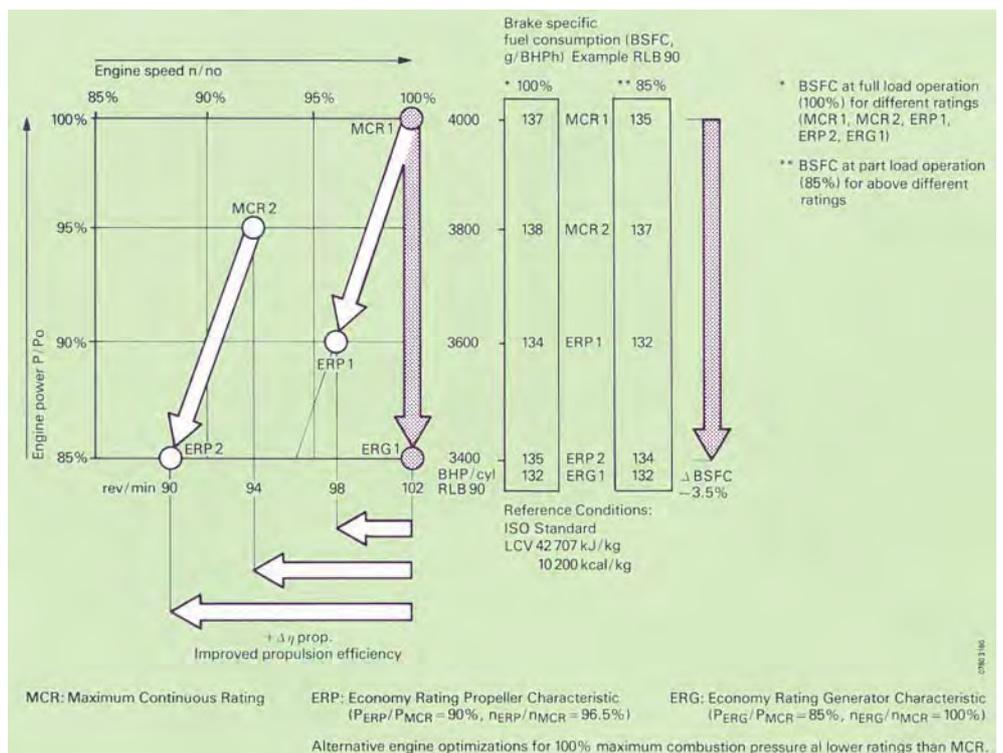
Fig. 77   Engine rating principle, RLB-engines and Fuel consumption RLB90. (SULZER)
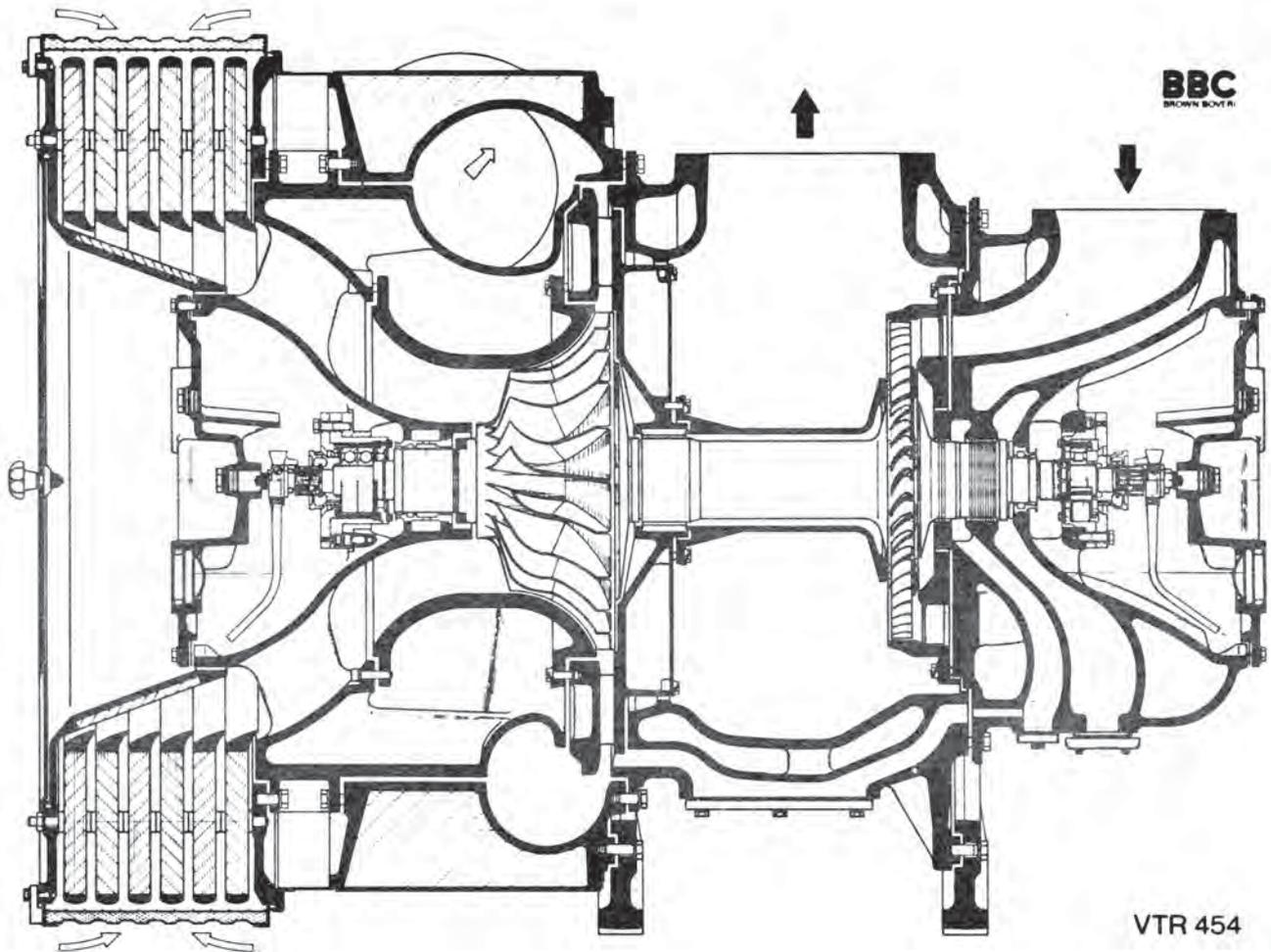
Fig. 78   ABB's turbocharger, VTR454. (ABB)
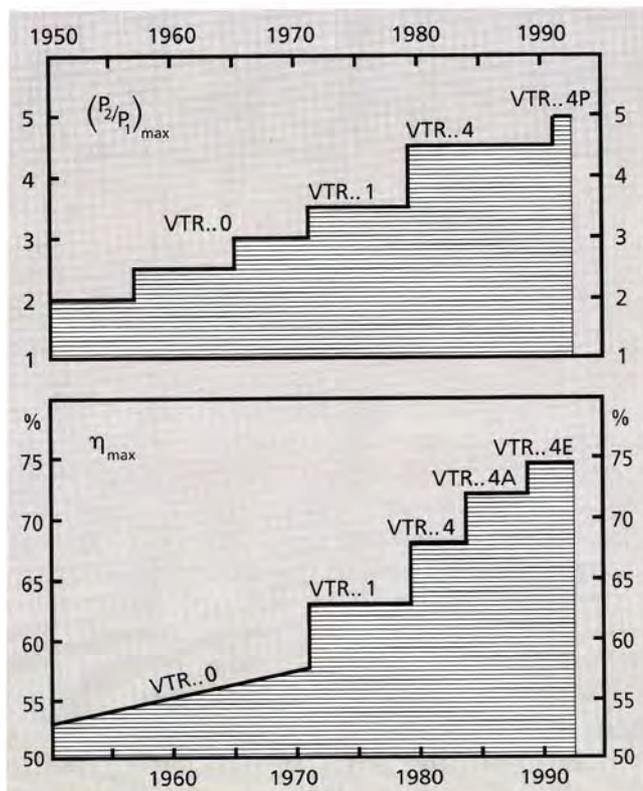
Fig. 79   Turbocharger development at ABB in pressure ratio and
           maximum overall efficiency (BBC). (ABB)
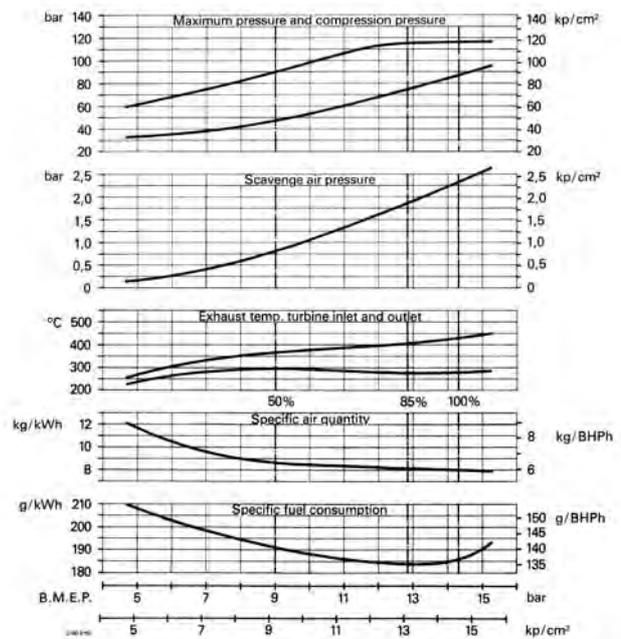
Fig. 80   RLB90 engine performance with the VIT mechanism.
           (Sulzer)

The fuel consumption race   49

This require great skill if reliability shall be maintained and is among others seen in the combustion chamber design, Fig. 81 – a masterpiece with bore cooled components. Especially the piston design is outstanding, the RLB version is water cooled, as the Sulzer tradition dictates. No wonder that Sulzer feel content, when the in trade magazine advertisments they present a simplified version of the combustion chamber with the text: "Why bother with exhaust valves, burn heavy fuel in our simple furnace", Fig. 82. Indeed a daring text as it will appear in the following.
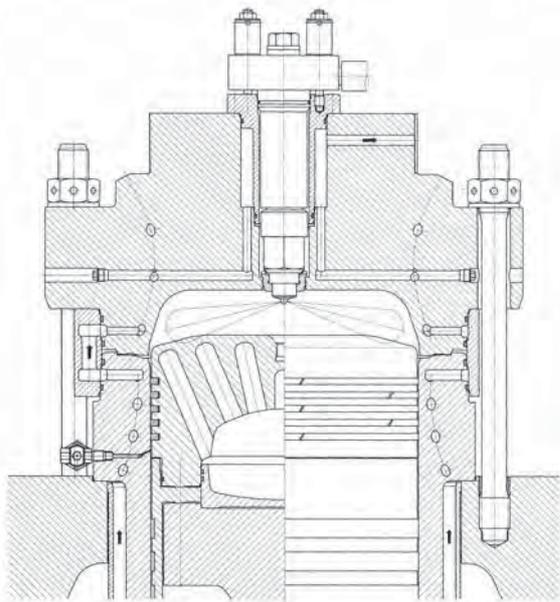
Fig. 81   Combustion chamber RLA56. (Sulzer [55])
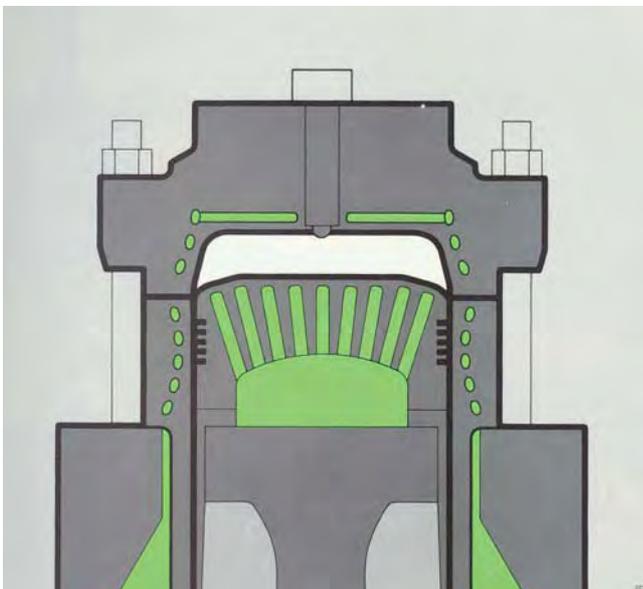
Fig.82   Simplified drawing of the combustion chamber. (Sulzer)

MAN

Mr. A. Schiff, of MAN announces on a press conference in Tokyo in May 1979 [56] new long stroke diesel engines with a stroke to bore ratio between 2 and 2,2, the C/CL range, but only covering the following three types: 90/190, 70/150 and 52/105. The design principles are identical to the B/BL series and the design features have only been modified in cases where the increase in stroke made this necessary, the moderate increase in maximum pressure of 5 bar is apparently not considered important.

Mr. Schiff compares openly the consumption figures with B&W and express his discontent, as B&W are using different ambient conditions in their published figures and the comparison shows: "that the loop scavenging system is not inferior to the uniflow scavenging system" he says and continues: "With the same specific fuel consumption, the valveless engine is simply better than the uniflow scavenged unit with its high expenditure on the exhaust valves or valves per cylinder and the connected maintenance etc." That remains to be seen.

At the same occasion MAN explains that they since April have been testing an electronically controlled injection system developed jointly with Bosch on a 3 cylinder KEZ52/105 C/CL test engine. The system, Fig. 83 has many interesting features:

- It operates with constant pressure injection based on feed pumps driven off the camshaft as in the Doxford common rail system.
- Regardless of pump delivery it is possible to vary timing, duration and pressure.
- Due to the load independent pressure an optimal atomization is secured at part load with markedly lower consumption rates and smoke free combustion.

The E-engine is publically demonstrated in October 1979, where it is reported, that MAN hope to get one or two plants to sea in 1980. Unfortunately the E-type never got a break through at sea.
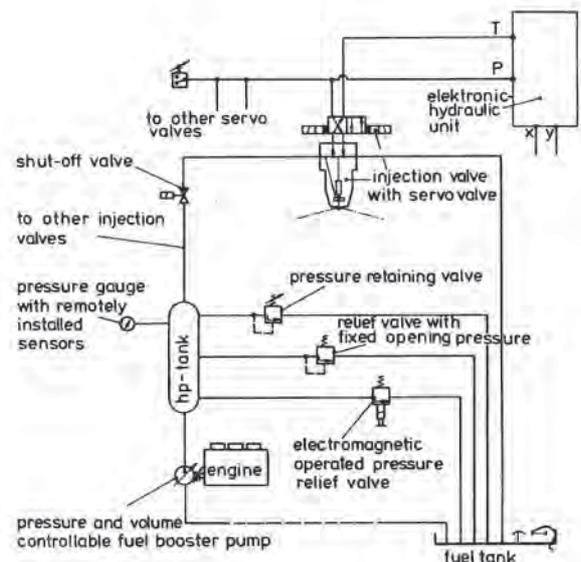
Fig. 83   Electronically controlled fuel system for KEZ engines. (MAN [57])

The three main competitors

The very tense competition between the just mentioned designers is summed up in the table below. New engines are introduced at a rate never experienced before, and before anybody could gain sufficient service experience, even though they would always claim the opposite.

From the table it is clearly seen that the two loop scavenged designs are squeezed on maximum pressure compared to the uniflow engines with the longer stroke. B&W have here a possibility to do something relatively fast with limited effort. An overview of the development in SFOC will be demonstrated later in greater detail.

Engines with 900 mm bore from B&W, SULZER and MAN

Date of announcement	Engine type	Stroke, mm	Power/cyl BHP	RPM	Pmax bar	MEP, bar	SFOC, g/BHPh *
Oct. 1977	RLA90	1900	3600	98	100	13.4	149
June 1978	K90GFC	1800	3410	114	84	11.6	141.5
June 1978	L90GFC	2180	3410	94	84	11.6	140.5
Aug. 1978	RLA 90	1900	3600	98	100	13.4	146
May 1979	KSZ90C	1900	3740	105	110	13.0	139.5
May 1979	KSZ90CL	1900	3740	95	110	14.4	142.5
June 1979	K90GFCA	1800	3945	117	95	13.0	142.5
June 1979	L90GFCA	2180	3945	97	95	12.9	139.5
Dec 1979	RLB90	1900	4000	102	118	14.3	142
Mar. 1980	KSZ90CH	1900	3945	105	115	13.7	139
Mar. 1980	KSZ90CLH	1900	3945	95	115	15.2	142
Mar. 1980	KEZ90C	1900	3740	105	110	13.0	140
Mar. 1980	KEZ90CL	1900	3740	95	110	14.4	142.6
Sep. 1980	RLB90	1900	4000	102	?	14.3	137

* At MCR, ISO conditions and fuel oil calorific value: 10.200 kcal/kg

Mitsubishi

Already in 1979 MHI are ready with a series of constant pressure charged engines following the same concept as the B&W engines [58]. The UEC52H, Fig. 84 and UEC60H engines are more or less a copy of their counterparts with 2-stage turbocharging and have identical power and revolutions, however the specific fuel consumption is reduced ~9 g/BHPh and the maximum pressure is now 108 bar. After just 4 years Mitsubishi have abandoned the two-stage turbocharged engines, and they can do that, because the development of their MET turbocharger now has reached the required efficiency level to safeguard scavenging – providing high air purity and low
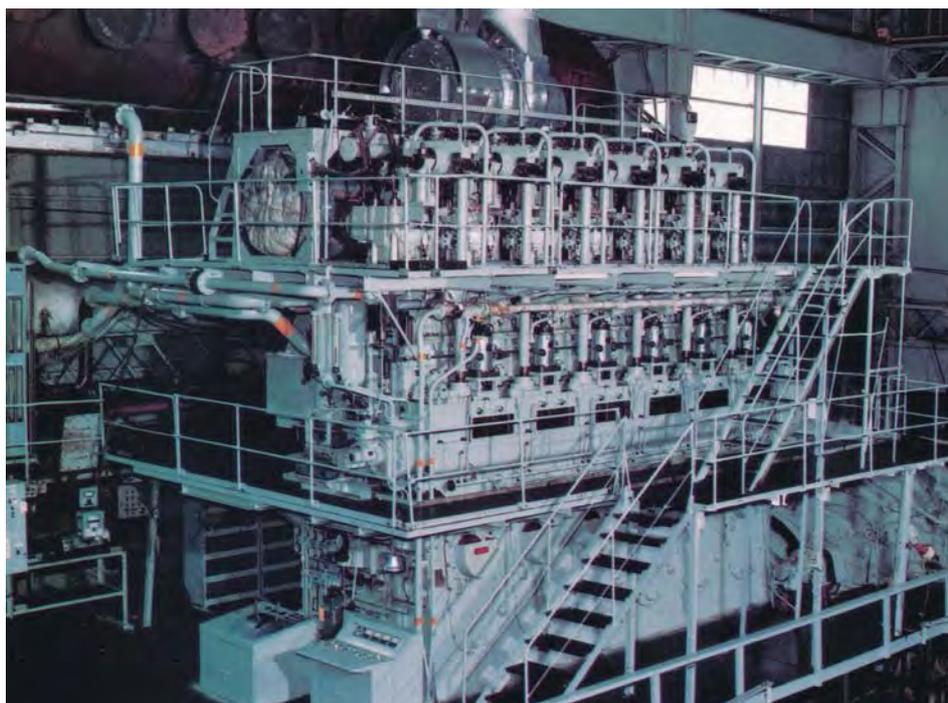
Fig. 84   6UEC52H on test bed at Kobe Diesel. (Kobe Diesel)

thermal load. The "Super MET", that is used, has the features that also later on will prove successful: Non water cooled casing, plain bearings and sturdy, wide cord turbine blades without damping wire, Fig. 85.
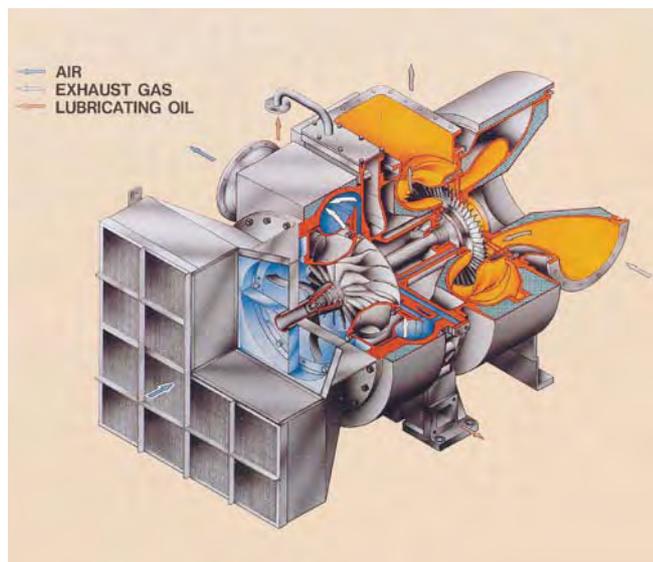
Fig. 85   Mitsubishi, MET SC turbocharger, pressure ratio 3.5. (MHI)

Instead of following their big European competitors MHI chose to concentrate on the lower power range, and they introduce two new engines; the UEC37H & UEC45H [59]. No other company can offer direct drive low speed crosshead engines in this power range, but MHI will soon get company by B&W and Sulzer, as the Japanese market is quite attractive in this power range, Fig. 86. The design of the small engines differ from the big engines and has just one central exhaust valve, and now it is also possible to introduce a bore cooled cylinder cover, something that would have been very complicated with the three exhaust valves. Previously the fuel valve was positioned in the centre of the cylinder, now two fuel valves are applied and mounted in the periphery of the cylinder. So, two features so far synonymous with Mitsubishi engines have disappeared. Gradually the new concept is transferred to the other engines in the series.

The specific fuel consumption is very low compared to the B&W uniflow and constant pressure charged engine of similar size, as shown in the table below.

Engine type	Bore, mm	Stroke, mm	Power, BHP/cyl	RPM	MEP, bar	Pmax, bar	SFOC, g/BHPh*
UEC45H	450	1150	1000	165	14.6	108	139
L45GFCA	450	1200	985	175	13.0	95	142.5
UEC37H	370	880	650	210	14.4	108	141

* ISO ambient conditions, and fuel oil calorific value: 10.200 kcal/kg

11.Three survivors
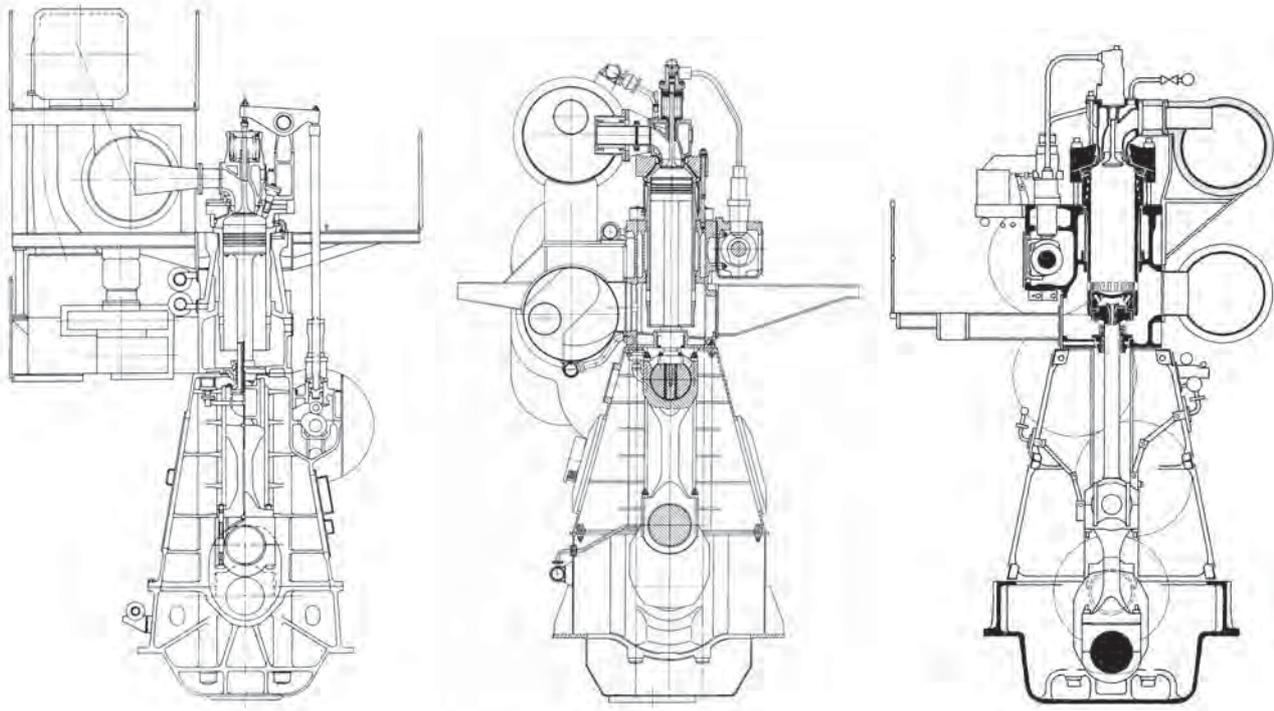
Fig. 86   UEC37H (MHI), L35GB (B&W), RTA38 (Sulzer).

B&W

In 1979 the majority owner of Burmeister & Wain is facing bankruptcy, and in an attempt to overcome this threat he decides to sell the engine divisions of the company to MAN. So, on the verge of B&W's greatest triumph since the introduction of the turbocharged engines, at a time where the probability of re-establishing market leadership is almost certain the competitor, MAN takes over. How does this influence the development in Copenhagen and Augsburg? At the time there were many speculations, but in 1980 MAN have their plan ready: The designers in Copenhagen get the responsibility for 2-stroke engines while the people in Augsburg get the responsibility for 4-stroke design. The B&W name disappear and is transformed to MAN-B&W, and so it will remain during the rest of this story. Further development of the loop-scavenged engine is stopped, but previous MAN licensees sees an advantage in getting a MAN-B&W license, and B&W licensees accept the new condition, which provides a good starting point for the new MAN-B&W.

MAN-B&W are quick in improving their market position and already in March 1981 they are presenting a new engine programme – the LGB/LGBE series of engines. The new series is basically uprated L-GFCA-engines, the mean effective pressure of the L-GB-engine is 15 bar as against 13 bar in the L-GFCA whereas the L-GBE has the same mean pressure as the L-GFCA, 13 bar. The maximum pressure has been increased from 89 bar to 105 bar on both models. The fuel consumption is respectively 3 and 6 g/BHPh lower for the two models than for the L-GFCA engines i.e. down to 133 and 130 g/BHP for the 90-bore engines. This is somewhat strange as the ratio Pmax to mep for the L-GB engine is practically the same as for the L-GFCA, however increased turbocharger efficiency is referred to as the decisive parameter. The fuel system is modernised in the sense that variable injection timing now also is a part of the MAN-B&W concept, as it was on the MAN engines in the past. On the hydraulic actuated exhaust valve the set of springs arranged around the spindle to close the valve are now replaced with an air spring, which remains standard on future engines. Interesting is also that MAN-B&W now are offering an economy version, the E-version as an alternative to the full rated engines – fuel prices will decide what is the most economical selection.

New to the programme is a small bore, the L35GB/GBE engine, the first competitor to Mitsubishi's UEC37-engine – also to be produced in Japan at Mitsui's sublicense, Makita. The comparison is shown below:

Engine type	Bore, mm	Stroke, mm	Power, BHP/cyl	RPM	MEP, bar	Pmax, bar	SFOC, g/BHPh*
UEC37H	370	880	650	210	14.4	108	141
L35GB	350	1050	680	200	14.8	120	136

* ISO ambient conditions, and fuel oil calorific value: 10.200 kcal/kg

This engine is a real long stroke engine with a stroke to bore ratio of 3. The design is not just a down scaling of the bigger engines in the series – it is really designed as a small engine. The entablature and cylinder block is made in one cast iron part and with the chain drive at the front end. The crankshaft is fully forged in one piece and the crosshead bearings are for the first time in MAN-B&W engines of full width, a neat little engine as seen in Fig. 86 – also showing the competitors engines in the same scale.

The previously mentioned race on fuel consumption continues mainly between Sulzer and MAN-B&W but also with Mitsubishi on the small and medium size bores, especially in the Japanese market.

Sulzer

Sulzer have realistically seen that their loop scavenged concept could not be developed further in direction of longer stroke, and they also realise, that this is the way the demand for en-

gine design go. If they want to stay in business there is only one way to go – to change the concept and design a uniflow 2-stroke engine series. To a certain extent they are familiar with the uniflow concept, as they once have used this system in their 2-stroke Z40 medium-speed engines (from 1963).

In December 1981 Sulzer announced a brand new engine series, the RTA-engines. It takes the maritime world by surprise, but nearly nobody doubts that Sulzer also will master this concept, and the decision is much respected by influential customers.

The new RTA's are announced as the Superlongstroke engines, and indeed – they have a stroke to bore ratio of 2.9 on line with the small MAN-B&W's

L35GB-engine and what certainly must be expected from the competition in the future. The engine programme consists of the following, here shown with data related to the highest rating (R1):

Engine type	Bore, mm	Stroke, mm	Power, BHP/cyl	RPM	MEP, bar	Pmax, bar	SFOC, g/BHPh*
RTA84	840	2400	4030	87	15.35	125	124
RTA76	760	2200	3290	95	15.31	125	127
RTA68	680	2000	2650	105	15.34	125	128
RTA58	580	1700	1920	123	15.31	125	129
RTA48	480	1400	1320	150	13.62	125	131
RTA38	380	1100	830	190	13.61	125	133

* ISO ambient conditions, and fuel oil calorific value: 10.200 kcal/kg

Sulzer are offering the new engines with four different standard ratings R1, R2, R3 and R4, where the minimum rating R4 represents abt. 70 % full power at 80 % speed. By retaining maximum and mean pressure, the R4 rating offers abt. 4 g/BHPh reduction in fuel consumption compared to the R1 level.

Mr. G. Wolf, in charge of the RTA development, has decided to split the design between large bore and small bore engines. Mr. M. Briner to be responsible for the big engines and Dr. G. Lustgarten to be responsible for the small bore engines with Dr. M. Eberle in charge of general research and thermodynamics/engine performance [60].

The large bore engines are where most of the "old" Sulzer design features are carried along. Most markedly is the retaining of the water cooled piston and as a consequence the split crosshead bearings. Also the tin-aluminium lined bearing shells are kept as in the loop scavenged engines for the crosshead bearings – a consequence of the water cooled piston concept. The oil supply to the bearings is delivered from a special high pressure system as usual in Sulzer engines.

The engine structure is by and large as in the RL-series of engines. Crankshafts are semibuilt and main bearings and connecting rod bottom end bearings are shells lined with white

metal. The Sulzer valve controlled fuel injection pumps are also retained in the RTA series as is the variable injection timing concept.
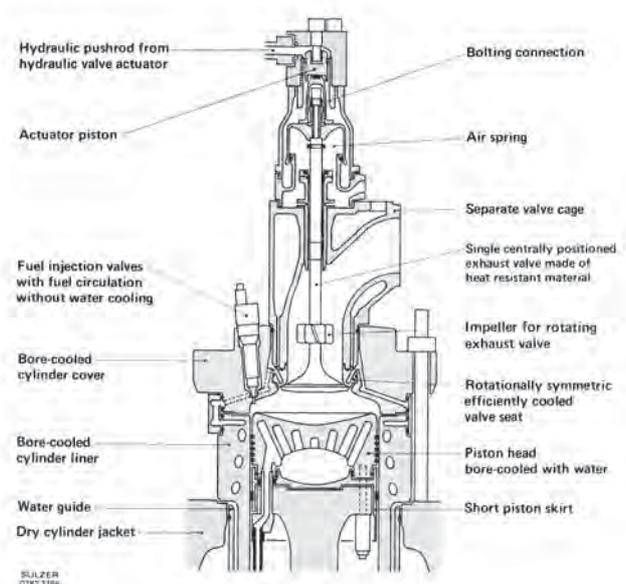

Fig. 87   RTA combustion chamber and hydraulically activated exhaust valve. (MS [60])
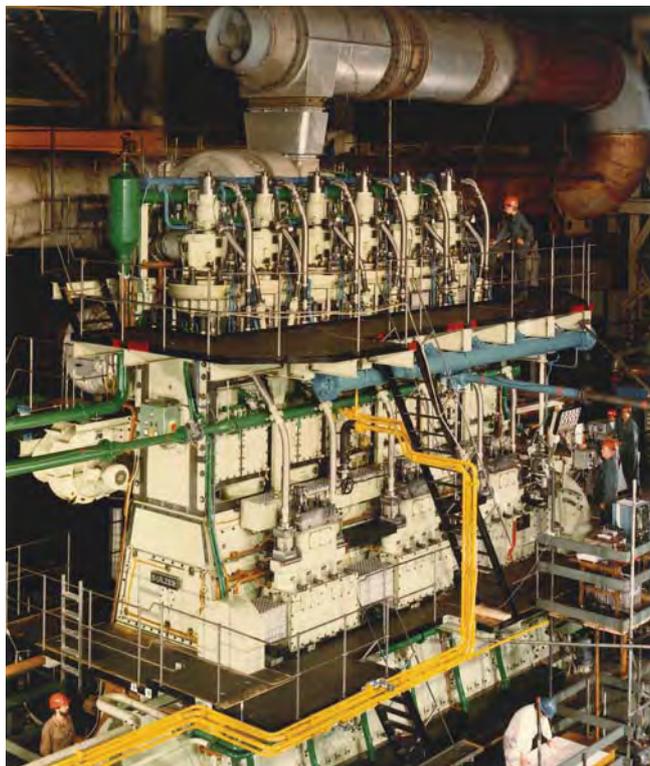
Fig. 88 The first RTA to be built in Europe, the 6 cylinder RTA58, 11520 BHP (8460 kW) at 123 rpm at testbed at Sulzer's Winterthur works. (Wärtsilä)

A consequence hereof is that full width crosshead bearings are introduced, however still with use of tin-aluminium lined shell bearings. The crosshead assembly is provided with two oil supplies through swinging link connections. One to supply the oil for piston cooling and guide lubrication another with high pressure to supply the crosshead bearings in accordance with Sulzer design practice. Another new feature is that the water transfer from cylinder liner to cylinder cover is taking place "inside" the engine, Fig. 89. The camshaft is positioned near the engine top and the multi cylinder jacket enable optimal accommodation in an integral camshaft box – as on 4-stroke engines.

New is also the fuel pump. The traditional Sulzer valve controlled pump is replaced with a helix type as on the 4-stroke engines and given a VIT-like shape derived from the variable injection timing concept on previous engines.

Of course the great question is related to the new Sulzer feature: Design and reliability of the exhaust valve. Sulzer have to make the exhaust valve a success – because "why bother with exhaust valves ..." So the message is clear: Here is no room for mistakes. And Sulzer make no mistakes, the valve is hydraulically operated with a rotating spindle in Nimonic 80A material, will in the future set records, and be unsurpassed as far as time between overhauls is concerned. The exhaust valve assembly and the combustion space are illustrated in Fig. 87. The camshaft is positioned at engine mid height and is driven by a gear train from the crankshaft with just one intermediate gear wheel. The other new components are the fuel valves, and here Sulzer are very cautious and introduce 4 uncooled valves in the 84-engine i.e. one more than MAN-B&W in order to obtain as even a temperature distribution in the combustion chamber as possible. This was maybe a bit overdone, and later the number is reduced to three as in MAN-B&W engines of similar size. The first RTA-engine was tested in 1983. Fig. 88 show the engine on test bed in Winterthur.

The small bore engines are of a different breed. The low costs and compactness required for installation in smaller ships dictates different design initiatives compared to the bigger brothers, see Fig. 86. The engine structure is thus cast instead of fabricated. On the RTA38 the crankshaft is forged in one piece, whereas the 48-engine maintains the semibuilt design because of weight considerations.

The specific requirements of the small bore engines make it possible to provide a fair compromise between low temperatures and low mechanical stresses in the piston crown with application of oil cooling, the system to become standard in later RTA engines (page 66).
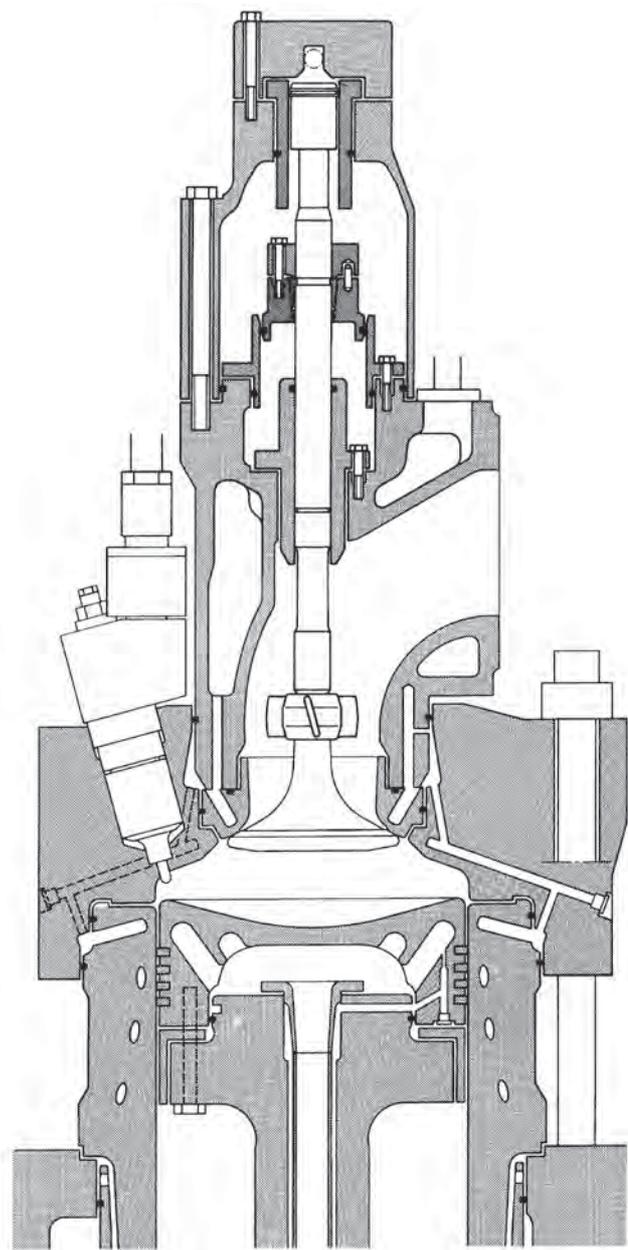

Fig. 89 RTA38 combustion chamber. (Wärtsilä)

Mitsubishi

Following the introduction of the GB and RTA engines Mitsubishi decides to uprate the H-series, now named HA. The power is 11 to 14 % higher than the corresponding models of the H series, and the engines are now offered with an economy rating as well. The fuel consumption is reduced by 8 % by adoption of the latest MET-SB turbocharger, improvements of the fuel system and a substantial increase of the cylinder maximum pressure.

MAN-B&W

The superlongstroke engines from Sulzer will soon get company, as MAN-B&W already in February 1982 announces their ultra longstroke engine, the L-MC/MCE line of engines. In December the same year all details are published [61] in The Motor Ship. The new line consists of 6 engines, as Sulzer's, this time however always with some cm "oversize" in bore compared to the competitor and now with an even higher stroke to bore ratio 3.24:1. In addition MAN-B&W maintain the E-versions with lower rating and specific fuel consumption. The L35GB, previously mentioned, is renamed to L35MC.

The maximum pressure is now 125 bar, as in the Sulzer RTA engines and the fuel consumption is consequently on line with that of the RTA's. More about fuel consumption later.

Structure wise the MC-engines are like the GB type, but there are several interesting new components. Most remarkable is the welded crankshaft produced by using narrow gap submerged arc techniques, Fig. 90. The crankshaft is introduced as an alternative to the semibuilt shafts, but is announced to become the only standard for the MC/MCE engines. The idea behind this production method was to get the freedom to allow overlap of the journals which would be required in short stroke engines, but with the present long strokes the purpose of introducing the new technology was merely a matter of costs, or so it was thought.

Long last MAN-B&W got a full width crosshead bearing, as in the MAN engines.

The high maximum pressure required reinforcement of the piston crown to keep wall thicknesses reasonably thin and temperatures low and with a similar purpose the exhaust valve is now provided with water cooled seats.

Also from MAN a new technique in casting cylinder liners was adopted; instead of drilling the cooling passages in the collar of the liner for the large engines, cooling was now provided with cast-in cooling pipes. MAN used this technique for cooling between the exhaust ports, ref. section III.
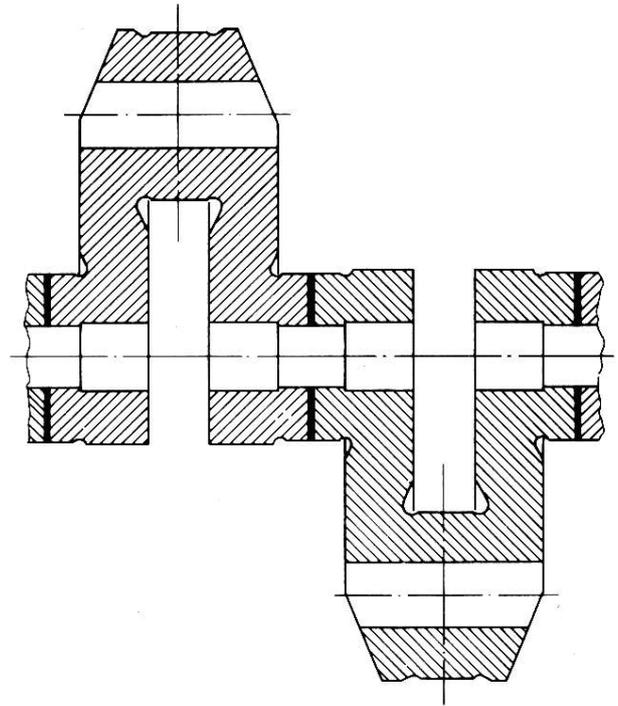
Fig. 90   Welded Crankshaft, using narrow gap submerged arc techniques (MS [61])

Mitsubishi

Already a year after the HA engines MHI are ready to challenge the competition with a new engine series, the L-type, super long stroke engines [62], however still based on the concepts of the H and HA types. Apart from the longer stroke the L-type has increased maximum pressure , 130 bar, but to get to a fuel consumption as low as 125 g/BHPh for the largest UEC60L engine Mitsubishi have invented an entirely new scavenging principle, the so called controlled swirl scavenging system, CSS. It is all about changes of the scavenging port design as illustrated in Fig. 91, and thereby achieving a cleaner and thus colder air charge in the cylinder at compression start, hence reduced fuel consumption. In addition the exhaust valve temperature is decreased.
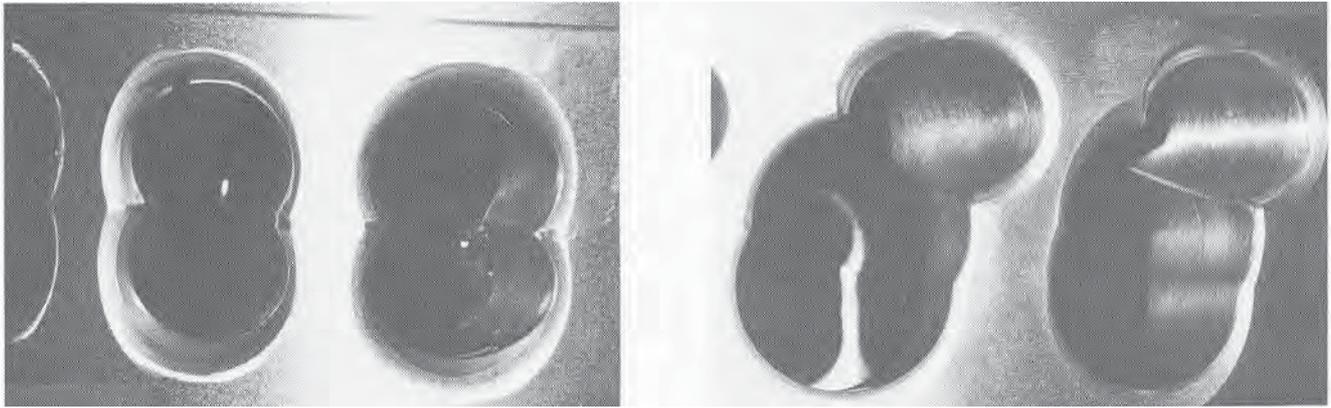
Fig. 91   CSS scavenging air porting (right) compared to previous conventional porting (left). (MHI)

MCR performance data of the UEC60LS is compared to MAN-B&W and Sulzer engines of similar size in the following table:

Engine type	Bore, mm	Stroke, mm	Power, BHP/cyl	RPM	MEP, bar	Pmax, bar	SFOC, g/BHPh*
UEC60L	600	1900	2100	110	15.6	127	125
L60MC	600	1944	2080	111	13.0	125	128
RTA58	580	1700	1920	123	15.31	125	129

* ISO ambient conditions, and fuel oil calorific value: 10.200 kcal/kg

Because of the increased maximum pressure the piston crown is redesigned with an inner annular ring support, and previous radial ribs have been done away with. Further the cylinder liner collar is increased in height and thickness and is now more intensely cooled. The exhaust valve housing is now just cooled around the valve spindle and insulation has been applied around the duct in order to increase the exhaust temperature for more effective use of the exhaust energy.

The fuel injection system is with Bosch-type pumps and VIT, but an interesting fuel valve feature with automatic control of the opening pressure of the valve based on load improves low load fuel consumption and reduces smoke emissions, Fig. 92.
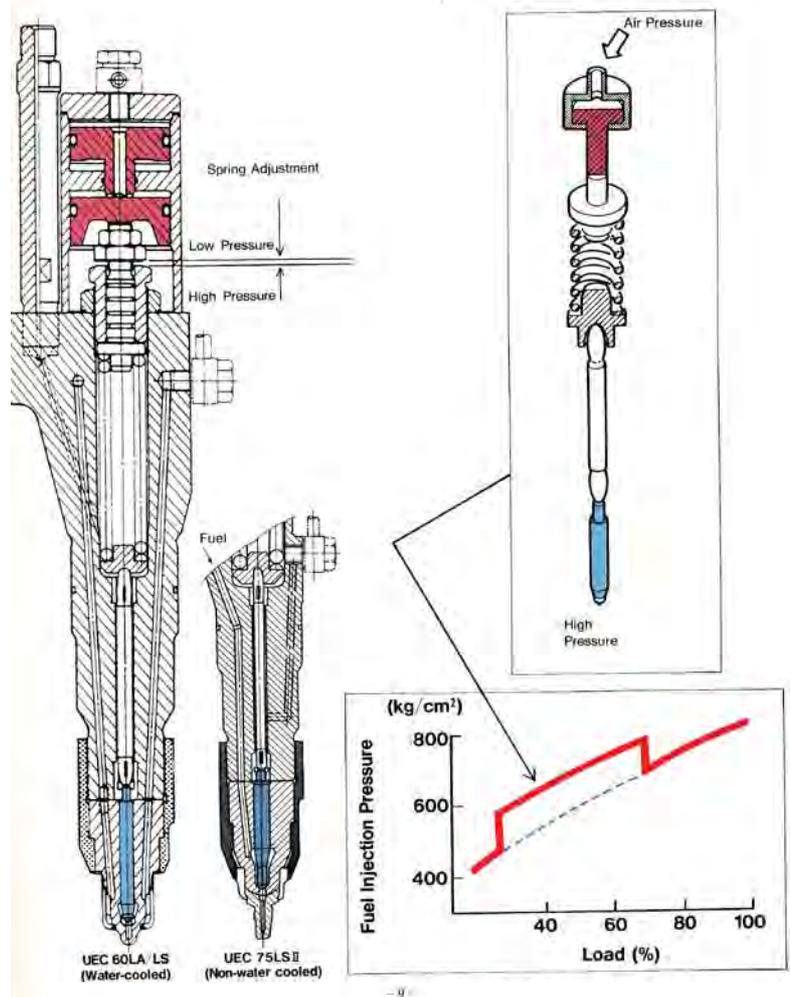
Fig. 92   Fuel valve with automatic control of opening pressure dependent on engine load shown in water cooled as well as non cooled versions. (MHI)

The L-types are developed into a LA-type and further a LS-type for the 52 and 60 bore engines to be introduced in 1985 and 1987 respectively. A comparison of the engines with 600 mm bore is shown below:

Engine type	Year	Stroke, mm	Power, BHP/cyl	RPM	MEP, bar	Pmax, bar	SFOC, g/BHPh*
UEC60L	1983	1900	2100	110	15.6	127	125
UEC60LA	1985	1900	2100	110	15.6	127	122
UEC60LS	1987	2200	2400	100	17.0	130	122

* ISO ambient conditions, and fuel oil calorific value: 10.200 kcal/kg

The reduced consumption is claimed to be a consequence of a modification of the scavenging timing and hereby extending the effective stroke, which is possible due to the higher efficiency of the MET-SC turbocharger, Fig. 85 now applied on these engines.

In addition to the LS programme of 1987 a LSII engine is developed, as will be mentioned later.

Sulzer

To compete with MAN-B&W's L-MC engines Sulzer introduces in 1984 the so called 2-engines, RTA52, RTA62 and later (1986) RTA72 – all 2 cm larger in bore than the corresponding MAN-B&W engines and furthermore with a

stroke to bore ratio of 3.47 against the 3.24 of MAN-B&W and 3.16 for the Mitsubishi L and LA engines. Furthermore a RTA84M tanker-engine based to a great extent on the RTA84 is added to the programme [63]. There is no end to the attraction of low revolutions and higher propeller efficiency.

The new engine design follow the previous except for one important feature: The pistons are now oil cooled – not exactly the prophecy of Sulzer in the late sixties: "Sooner or later everybody will apply water cooled pistons", but maybe the prophecy can become right sometime. The oil supply to the crosshead is arranged as in the two small engine types. As a conse-quence of oil cooled pistons the cross-heads are now designed with full-width lower bearings.

Fig. 93   Development of the specific fuel consumption shown for large bore B&W and Sulzer engines. (Author)

Fuel consumption development and comparison

The very intensive competition on fuel consumption between the two left main competitors is illustrated in Fig. 93, where the red line shows the dramatic development over 10 years. However, the heading "official" fuel consumption refers to the fact that during this period of time, the major engine designers are introducing rather unfair methods, when they present the consumption figures and that goes for Mitsubishi too.
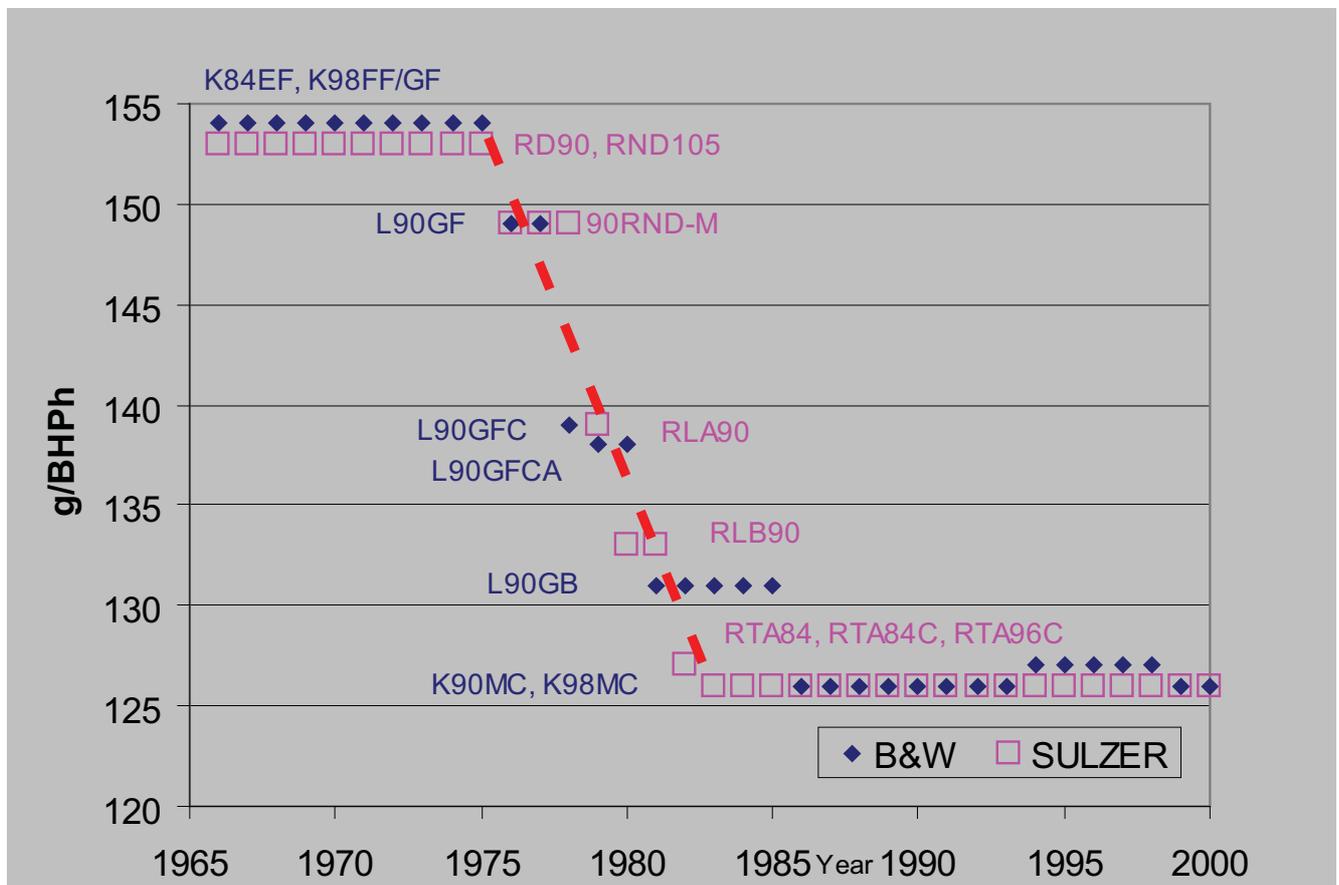
Before 1980 the tolerance on consumption was ± 3%. Anders Østergaard, Head of research and development at B&W publish [64] in "Schiff & Hafen" the outcome of measurements from delivery tests with many L-GF engines, Fig. 94, and it is noteworthy that the tolerance margin of 3 % is ample to cover the spread in the measured values. Gradually the consumption is given with + 3 % margin and in the late 90's in connection with IMO NOx certification it becomes + 5 %. What happens? There is no more any tolerance and with modern measurement techniques at the time one don't need ± 3 %. As however, very effective calculation tool have been developed for estimation of the fuel consumption at the design stage, the tolerance is "eaten up", and actual measurements on test bed will show values very close to 3 to 5 g/BHP higher than shown on the figure. A closer analysis of the engine performance data also clearly reveals that the ratio between maximum and mean pressure, being the most decisive parameter in relation to fuel consumption, not has been increased corresponding to the dramatic reduction in fuel consumption.

The IMO NOx regulations [65] are more or less forcing the engine designers, by engine internal measures, to reduce the NO_x emission as the cheapest and fastest solution. This means that the beginning of the combustion in the cylinder is retarded (the maximum pressure reduced) so, that the combustion is taking place at lower temperatures, but unfortu-

A = EXPECTED AVERAGE

B = GUARANTEED UPPER LIMIT

C = MEASURED AVERAGE
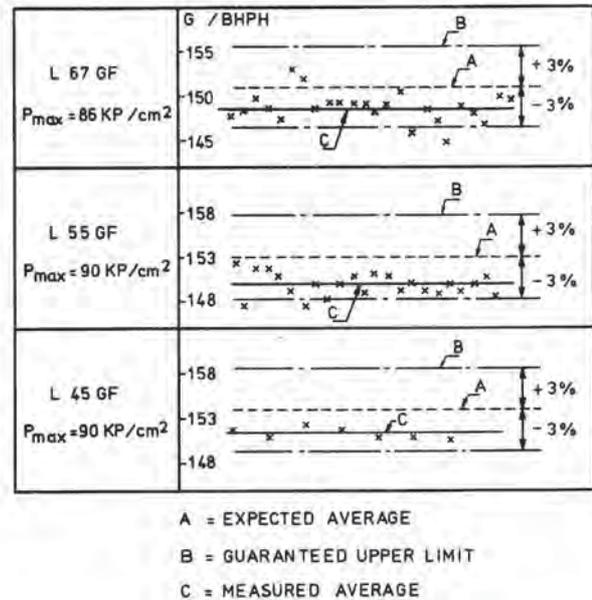
Fig. 94   Actual fuel consumption measurements on B&W's testbed compared to guaranteed figures (S&H [64])

nately also that the fuel consumption is deteriorated. Instead of increasing the quoted SFOC figures the designers just add extra 2 % to the margin. Not too smart, and certainly not smart either that IMO regulations concentrate so one-sided on NO_x emissions and ignore, that the CO_2 emissions are increased as a consequence of the regulation.

Comparing all 3 survivor's fuel consumption on engines of similar size as done in Fig. 95, clearly shows that MHI typically are ~3g/BHP lower than the others, so the mentioned special features of the Mitsubishi engines as the CSS ports and others must certainly be very important. The competitors have tried to imitate the CSS ports without success.
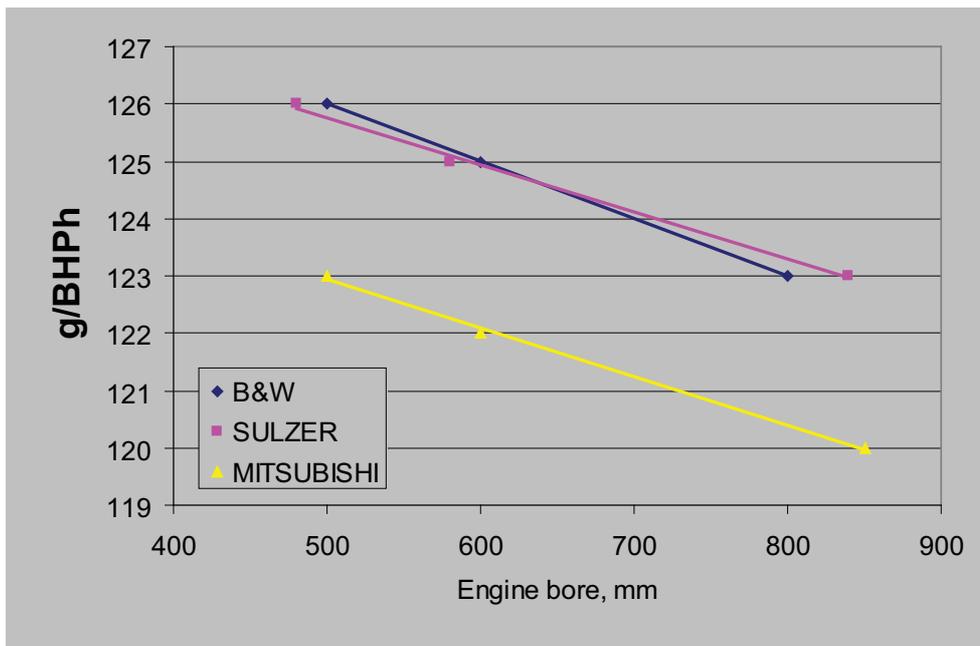
Fig. 95   Typical fuel consumption figures from MAN-B&W, Sulzer and Mitsubishi for engines of similar bore sizes. (Author)

The market share development reflects the importance of being first in offering low consumption and low engine Revolutions. This can be seen in Fig. 96, where MAN-B&W after introduction of the long stroke, constant pressure turbocharged engines recapture the position as market leader lost to Sulzer in 1964.

Fig. 93 also illustrates the rapidness of engines being introduced. Compared to the past, at the start of turbocharging, there were 6 to 7 years between new engine types. Here in the 70-80's there are only 2 to 3 years. There are many good explanations, how this was

possible, but one thing for sure is missing; the implementation of service experience of the previous design to new designs. It is simply not possible. Engine users are often a bit annoyed and the industry also see through their co-operation in CIMAC, that it would be appropriate to include the users more actively, so in 1993 the CIMAC "Engine Users" working group is established. This working group starts to collect experience data from their fleet and provides documentation that will be use in the following.
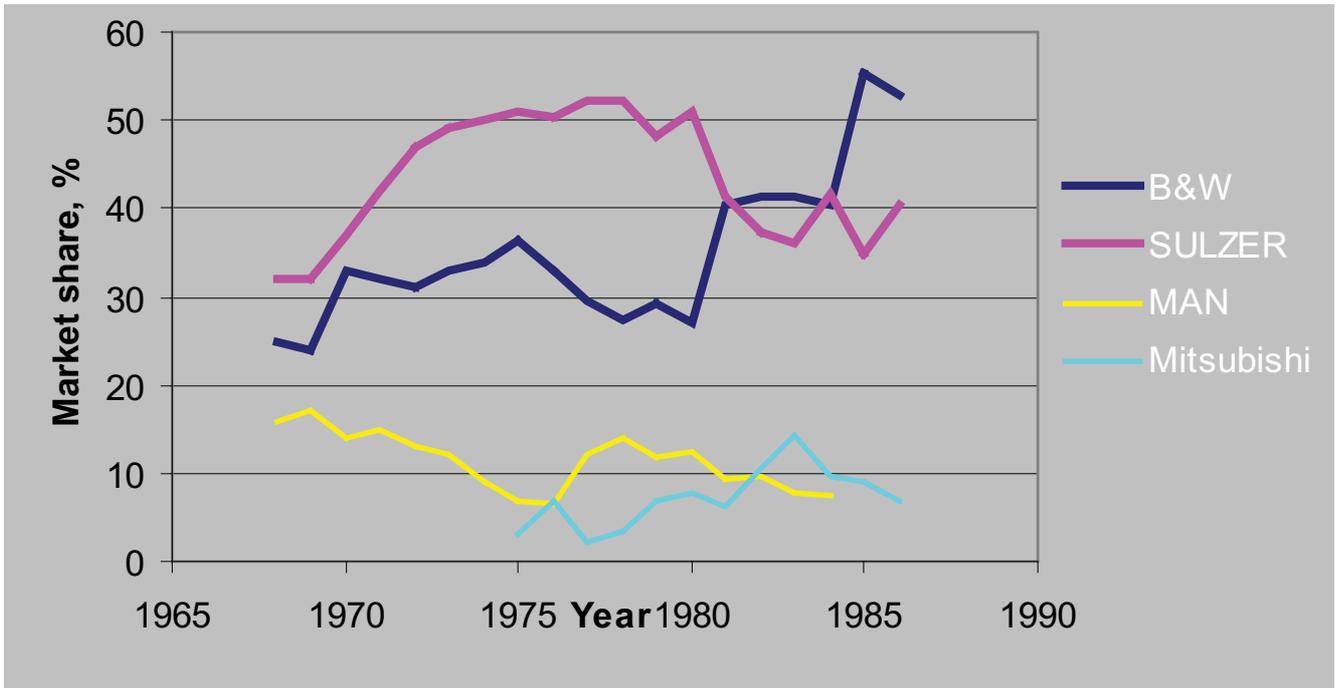
Fig. 96   Market share development. (Author)

Engine failures

Due to the very size of the 2-stroke engines it is not possible to run endurance tests and sort out failures at the test bed. The usual practice is therefore to carry out tests on new components on existing engines in service. However, even the most well designed new engine will have some failures that need to be solved, and sometimes the time factor is important, as off service of the vessels could be a costly affair. Many ship-owners have as a consequence of this acted as the test field for the engine designers.

The engine users experience concerning the period from 1975 to 1999 has been reported at several CIMAC congresses [66,67,68], and the following general observations are:
- About 50 % of all engine failures are related to the thermal loaded components, hence the focus in this report on this specific item.
- Poor cylinder condition including cylinder liner cracks represents nearly half of the failures on the thermal loaded components, meaning that overhauls of cylinders were much more frequent than foreseen by the designers, Fig. 97.
- Turbocharger failures are frequent Fig. 98 and complete break downs sometimes disastrous, Fig. 99.
- Main bearing failures are significant and found in most engine types.

More specifically the engine users analysed the distribution of failure cases related to all the engines out of guarantee of the three designers. The failures are categorised in main sections of the engine as shown in the following table, Fig. 100. It is clearly seen that hot components as mentioned is a "hot issue", and dominating for the two major makers. The reason why so few failure cases on MHI-engines are found in "Hot Components" section could be, apart from a well done design, because the engines generally are designed for relatively slow ships with low revolutions and consequently have a high stroke to bore ratio and an amply sized combustion chamber.

The economical consequences of failures are included in engine maintenance cost. For the years 1980–2000 the CIMAC WG found among others that the specific operation costs (excluding fuel, lubrication oil, crew and financial costs) of 14 different engine types and makes was larger for the small bore engines than for the large bore engines, Fig. 101, and just as surprising that the cost/kWh were fairly low. The small figures of the specific costs could be misleading – but the actual yearly cost for a big large bore engine could be in the region of 200.000 US$ per year, which is more tangible.
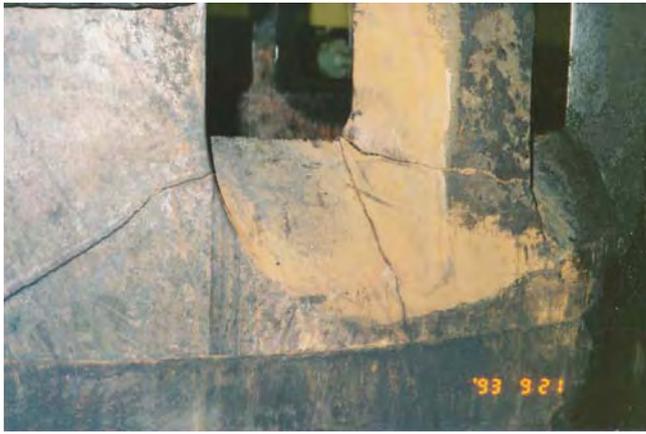
Fig. 97   Cracked cylinder liner, KSZ90BL. (Author)

Fig. 99   Blower wheel of complete wrecked turbocharger (Author)

Fig. 98   Fatigue crack in turbine blade (Author)
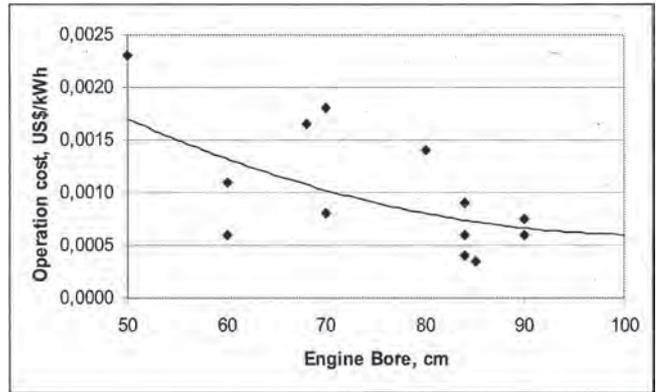
Fig. 101   Engine operational costs versus engine bore. (CIMAC [68])

Engine Make	MAN-B&W	SULZER	MHI
Engine Section	% of cases		
Hot Components	46.9	53.0	17.6
Running Gear	10.5	10.3	19.6
Mechanical Control Gear	7.5	0.9	11.8
Injection System	3.7	9.4	23.5
Camshaft Drive	2.0	0	0
Frame and Entablature	6.5	0	11.8
Charge Air System	5.0	8.5	5.9
Turbocharger	12.0	15.4	7.8
Engine Performance	0.2	0	0
Miscellaneous	5.7	2.6	2.0

Fig. 100   Distribution of failure cases on main engine sections. (CIMAC [67])

12. Engines for the big Container Ships

MAN-B&W

It must be noted that contrary to previous MAN-B&W practice all new engines will maintain the name "MC" or "MCE", meaning that is becomes difficult to distinguish between the different designs. Later in time MAN-B&W add a mark number, e.g. "MC6" which make identification easy. Where needed a mark number will be used in this text.

In 1986 MAN-B&W are ready to extend their portfolio of engines and update the existing L-types. The programme include two new classes, the S-type (see later) and a K-class intended for rather big and fast container vessels and the entire programme now totals 14 different engines with a cylinder power from 395 to 5360 BHP to cover the entire propulsion requirements. The multiplicity of this 2-stroke crosshead engine programme is impressive and of course decisive for maintaining a large market share.

The two K-engines, K90MC3 and K80MC3 are having a relatively short stroke (stroke to bore ratio 2.8:1) and consequently the combustion space is relatively flat making it more difficult to avoid high thermal load on the components, now the mean pressure is increased to 16.2 bar. Most of the design features introduced in the L-MC series are maintained in the K-class – also the cast-in cooling pipes in the cylinder liners mentioned earlier. This turned out to be a disaster as most of the liners cracked in the cast-in tubes because of loss of fatigue strength of the tubes, as these were decarbonised during the casting process and the steel transformed to ferrite having low fatigue strength. A design change was required and that was a return to bore cooling, but this time MAN-B&W used the GMT principle applied in the CC600 with bores that ends blindly, as shown in Fig.69, a very effective and good solution. However, a solution had to be found for the vessels in service, and here the licensee Mitsui provided an exciting "dual cast liner" that could be mounted without any major modifications of the existing engine [69]. The "dual cast liner" is produced by centrifugal casting where the outer layer that first is poured into the machine is plain carbon steel, while the inner layer, the running surface is the standard cast iron for cylinder liners. In this way high tensile strength is achieved at the cooling surface, where the tensile stresses

are highest, while the running conditions for the piston rings are maintained as before.

Years later improved production technology made room for the use of cast-in cooling pipes.

The thermal load of the piston crowns also became a problem, as burn down of the material and large thermal cracks reduced the lifetime considerably. At the same time the exhaust valve spindles had an unsatisfactory time between overhaul and lifetime. Some users made a change to Nimonic material, as the Sulzer standard solution, and finally in 1994 MAN-B&W also introduced Nimonic material as standard for valve spindles of large bore engines.

In 1989 a K90MC-C is added to the programme and the K80MC is converted to K80MC-C (the additional "C" to indicate the suitability for Containerships). The K90MC-C has a shorter stroke and higher revolutions than the MC counterpart and is consequently cheaper to produce.

When B&W in 1978 decided to stop their turbocharger production they made an agreement with BBC to use their chargers. So BBC became more or less absolute on B&W engines. The BBC chargers had the efficiency that was required, but were facing a draw back because of expensive maintenance; water-cooled cases were corroded and the lifetime of the roller bearings too short. So with MAN as the owner it became possible to have a competitive product specified. MAN's
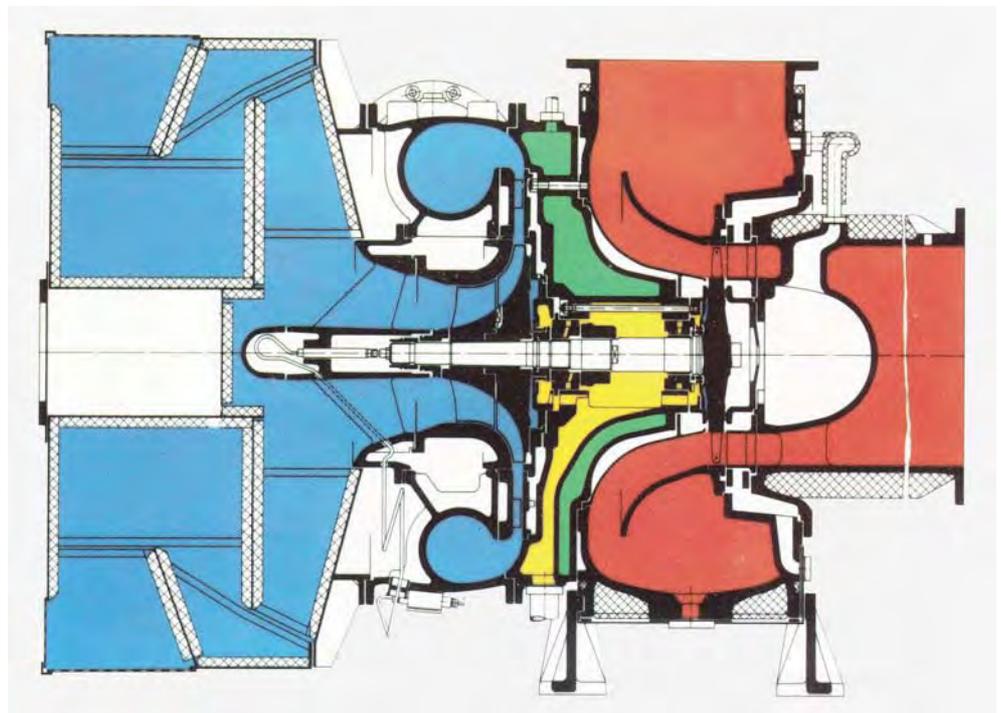

Fig. 102   MAN, NA-TO turbocharger (MAN-B&W)

turbochargers were uncooled and had plain bearings that functioned well, Fig. 102.

However, it is difficult to make things right, and on the large K90MC3 engines the MAN NA-turbochargers with long, slim turbine blades suffered in several cases from breakage because of resonance problems between the turbine nozzles and blades with accompanying fatigue cracks, Fig. 98. MAN-B&W were consequently pressed by users to also make the Mitsubishi MET turbocharger standard on their engines, as this charger had short wide chord blades with a very good service record. Now there was full competition.

Solutions to the problems were found, and in 1991 MAN-B&W are ready with the first uprating, the K90MC5, where the mean pressure is increased from 16.2 to 17 bar and the revolutions from 90 to 94 rpm.

The demand for large containerships were however growing as the transportation market growth was about 10 % per year and that meant also an increased demand for higher engine power for larger vessels offering lower transportation costs. Post panmax container ships were being built at the Odense Shipyard, and that forces MAN-B&W already in 1993 to offer a new K90MC6, now with 18 bar mean effective pressure. A cross section of K90MC6 is shown in Fig. 103.

Fig. 103 Cross section of K90MC (6), bore 900 mm, stroke 2550 mm, yielding 6220 BHP/cyl (4570 kW/cyl) at 94 rpm and Mep = 18 bar. (MAN-B&W)

As mentioned before the time between introductions of new designs is too short to accumulate service experience, and that is also the case of the mark 6 engine. The combustion in the uprated engine was not satisfactory, actually it was not too clean either in the old mark 3 engine, and coke formation was building up on the piston top land and grew to a size, where it wiped off the lubrication oil on the cylinder running surface. The consequence was heavy scuffing of piston ring and cylinder liner surfaces. After introduction of several patch-work solutions, among others introduction of a so called "piston cleaning ring" integrated in the top of the liner where it scrapes off deposits built up on the piston topland, the right solution was found i.e. to eliminate the soot formation and secure clean combustion. An old idea of Dr. Sunn Pedersen (MAN-B&W's technical director) to eliminate the oil sack in the fuel valve was introduced. This so called "slide valve", Fig. 104 had a decisive effect on the cleanliness in the engine cylinders as seen in Fig. 105, where the condition of the piston rings is seen with and without "slide valve". In addition the fuel consumption was reduced as the oil left in the atomizer tip now was burned effectively.
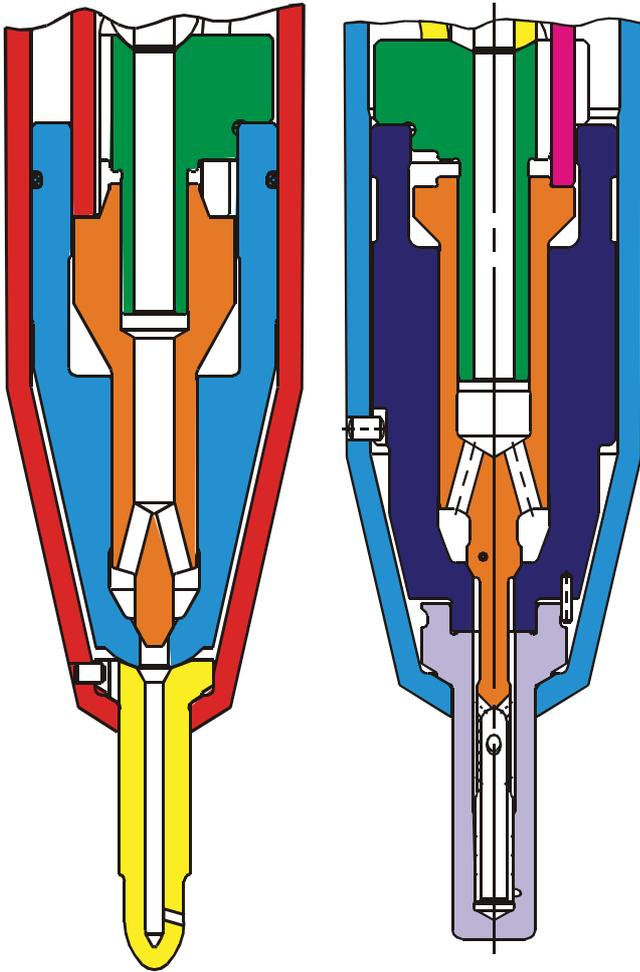
Fig. 104   Standard and Slide Valve (MAN-B&W)

Fig. 105   Engine cylinder condition showing a vast improvement when using slide fuel valves instead of standard fuel valves (Author)

For some time there had been focus on the lubrication oil consumption as the cost of lube oil were of the same of magnitude as the maintenance cost of the entire engine. The cleaner ring pack was a good starting point for introduction of more a modern lubrication system that made lube oil consumption reduction possible in a controlled way.

It was also necessary to find a better piston design, as the temperatures in this highly rated, "short stroke" engine were too high with risk of burning and crack formation of the piston crown. By use of advanced computer simulation tools now available a completely new design of the piston crown with a small mountain in the centre instead of the traditional bowl reduced heat transmission so much, that temperatures could be reduced about 100 ˚C to 409 ˚C, Fig. 106 and Fig. 107 [70]. A high margin for future use in uprated engines were hereby established, and a further development potential in applying an entirely bore cooled design were at hand.

In Fig. 106 it is also demonstrated how technology is transferred from MAN to MAN-B&W in the adaptation of the finger bore cooling system for the cylinder cover, introduced for the KSZB/BL-engines, Fig. 68.

Max 418 ˚C Mean 413 ˚C		Max 373 ˚C Mean 335 ˚C
Max 611 ˚C Mean 602 ˚C		Max 409 ˚C Mean 392 ˚C
Max 205 ˚C Mean 199 ˚C		Max 209 ˚C Mean 189 ˚C

Fig. 106   Combustion chamber temperatures K90MC (6), Mep = 18 bar. (MAN-B&W)

Fig. 107 Comparison of the effect on piston and exhaust valve temperatures when using a standard piston crown versus a design having a small top (Oros) at the centre of the crown. 100% load, Mep = 18 bar. (CIMAC [70])

In 1994 MAN-B&W introduced again a 98-bore engine, the K98MC-C intended for use in the large post panmax container vessel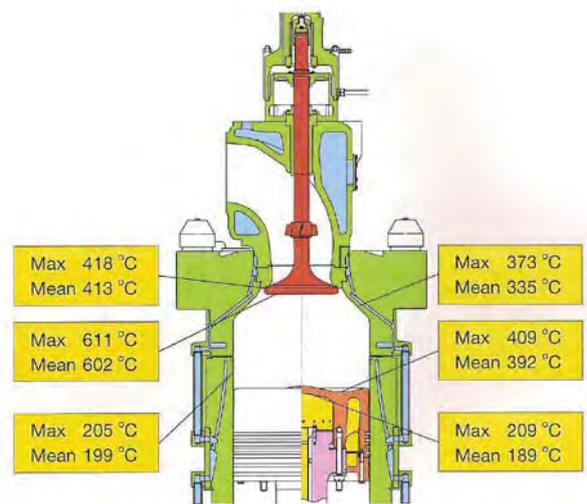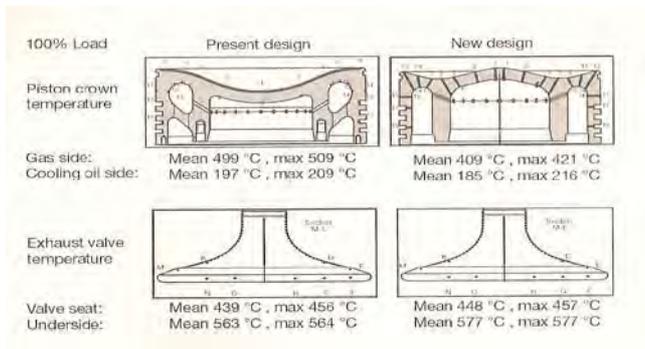s, that were built in a drastic increasing number all over the world. The specific power of the engines was 7760 BHP/cyl at 104 rpm and with a mep of 18.2 bar as the K90MC6. The engine did not become popular at the yards because of the high number of revolutions, and in 1998 a K98MC with increased stroke and lower revolutions was designed and immediately sold. The considerable differences between engine parameters are illustrated in the table below.

Engine type	Charging principle	Year	Stroke mm	Power, BHP/cyl	rpm	MEP bar	Pmax, bar	SFOC, g/BHPh*
K98FF	impulse	1967	2000	3800	103	10.8	80	157
K98MC6	constant p.	1998	2660	7780	94	18.2	130	126

* ISO ambient conditions, and fuel oil calorific value: 10.200 kcal/kg

Sulzer

The development of the prestigious, high output engines for the large, fast containerships is the most challenging engineering task. Sulzer, although losing market share generally, were matching MAN-B&W's share on the large engines for containerships. Particularly the RTA84 and to some degree RTA84M were ordered for many containerships. However, in late 80's there were increasing indications towards the need for higher speed and power, so, on the basis of the RTA84 the new RTA84C was ready in 1988 with a cylinder power of 5200 BHP at 100 rpm and a mean effective pressure of 17.2 bar [71].

The RTA84C is shown in a cut-away view in Fig. 108. It has basically the same design characteristics as the RTA84M and the 2-series, which means that engine structure, combustion chamber components, running gear with white metal crosshead bearings are having known track records. The most recent improvement of these engines are introduction of a so called "Core Package" to improve piston running conditions and this feature is also a part of the RTA84C concept.

The engine cylinder condition is a multi disciplinary engineering task and as such involving the entire design team to find a reliable solu-
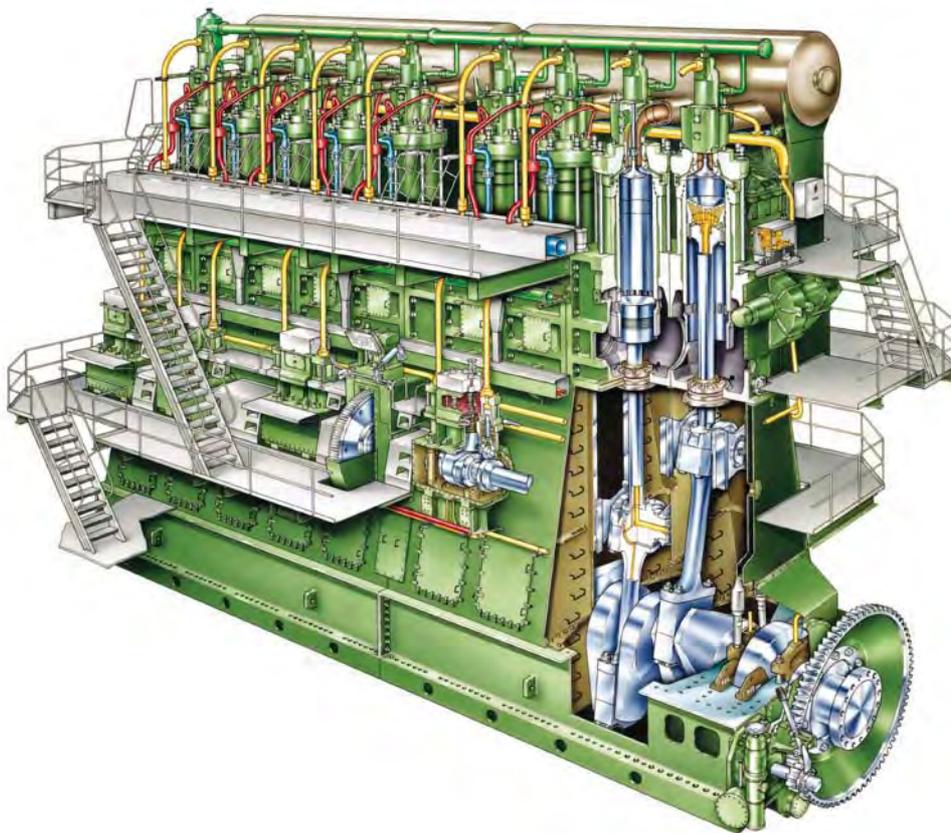

Fig. 108 Cut away view of the RTA84C, bore 840 mm, stroke 2400 mm, 5200 BHP/cyl at 100 rpm and Mep 17.2 bar. (Wärtsilä)

tion. Sulzer's "Core Package" [72, 73] consisted of three major improvements:

1. Better cylinder liner material with higher hard phase content
2. Increased temperatures of the cylinder liner running surface
3. Introduction of multilevel lubrication (seen on Fig. 108)

This "Core Package" was based on a lot of research and experiments, and is exemplary described in the references. However, one thing is theory another is practice, and the cylinder condition was not in all cases acceptable and reliable, as reported later.

A disturbing element of company politics at this time must be reported: Sulzer Brothers in Winterthur had little interest in maintaining the heavy industry in their portfolio of enterprises and in 1989 they discuss with MAN in Augsburg to sell out the diesel division. Apparently the two parties reach a con-

clusion, but the acquisition was vetoed by the West Germany Economics Ministry in January 1990 on the grounds that it would create a monopoly. This action was indeed a Sulzer top management blunder, because who would buy engines designed by somebody who's destiny was in the hands of the biggest competitor, so Sulzer's market share was hurt. Now as it was common news, Sulzer had to find other solutions fast. In July 1990 it was reported that the majority of shares in Sulzer diesel was sold to Fincantieri and a consortium of West Germany's Bremer Vulcan and Deutsche Maschinen und Schiffbau in East Germany and a small management stake. On the 15 November 1990 "New Sulzer Diesel" was born, still producing and designing engines bearing the Sulzer name.

In 1993 New Sulzer Diesel uprates the RTA84C on both revolutions and mean effective pressure and in December 1994 a super large bore engine the RTA96C is added to the programme of container ship engines [74], ref. the table below:

Engine type	Year	Bore mm	Stroke, mm	Power, BHP/cyl	RPM	MEP, bar	Pmax, bar	SFOC, g/BHPh*
RTA84C	1988	840	2400	5200	100	17.2	135	126
RTA84C	1994	840	2400	5510	102	17.9	140	126
RTA96C	1994	960	2500	7470	100	18.2	142	126

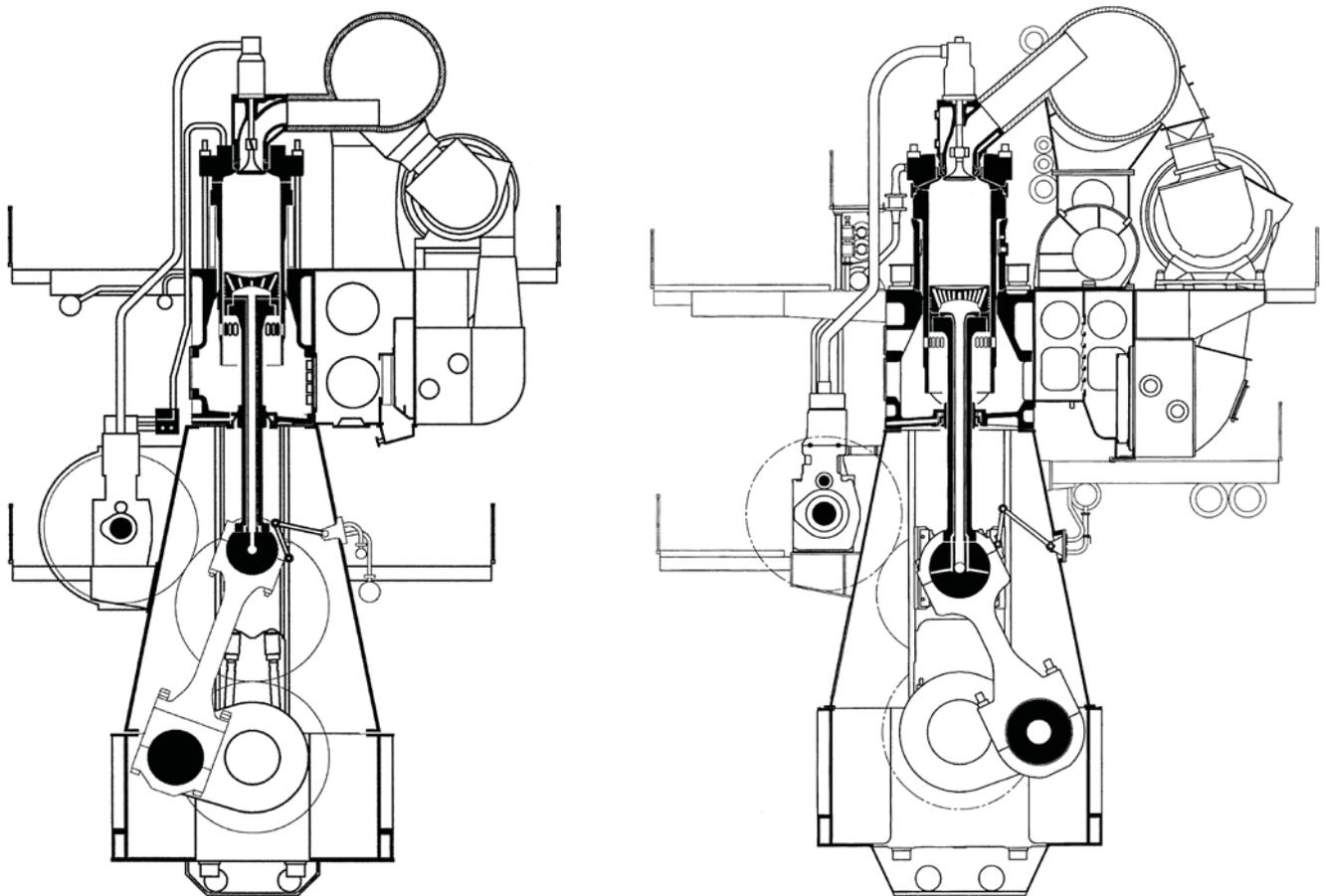* ISO ambient conditions, and fuel oil calorific value: 10.200 kcal/kg

Fig. 109   Cross sections of the RTA84C (left) and the RTA96C (right) to the same scale. (Wärtsilä [74])

A cross section of the two engines to the same scale is shown in Fig. 109.

The first RTA96C engine completed tests in May 1997, Fig. 110. The general performance and the temperatures of the combustion chamber components, Fig. 111 were measured and found in line with precalculations.
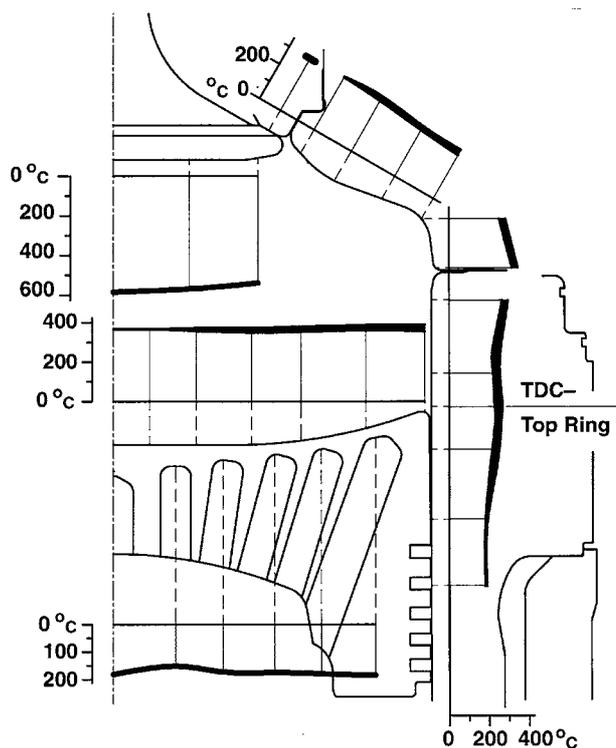
Fig. 111   Combustion chamber surface temperatures, 11RTA96C at Mep = 18.2 bar. (Wärtsilä [74])

The wall temperatures were expected to be the main hurdle to pass, as this engine has a very short stroke to bore ratio of about 2.6 and consequently a rather shallow combustion chamber. The difficulty in overcoming high thermal load of the components were overcome by selecting a relatively high scavenging air pressure, which allows to lower the piston position somewhat and further to improve the cooling effect by researching the opportunities of the combined effect of shaker and oil jet measures, Fig. 112 [75]. The cooling system inside the piston crown is shown in Fig. 113.
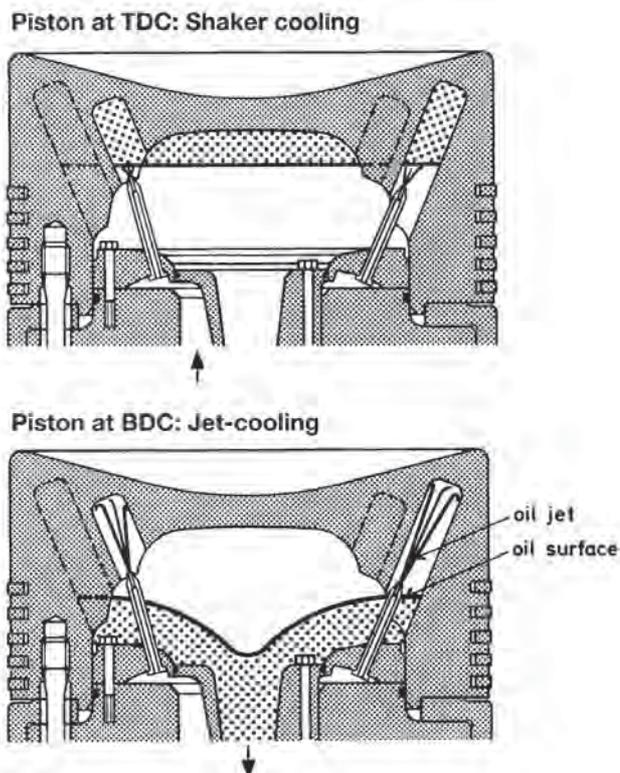
Fig. 112   Operating principle of jet-shaker piston cooling system (drawing based on RTA-U engine). (Wärtsilä [74])

Fig. 113   Cooling inserts for pistons, RTA96C. (Wärtsilä [74])

The service experience with these container ship engines was generally acceptable. The "Core Package", was now replaced with a more comprehensive "Tribo Pack" [76] containing among others an Anti-Polishing Ring as in the MAN-B&W engines and ceramic chrome plating of the upper piston ring, as has been used for quite some years in medium speed engines. After some years in service the cylinder condition was still not satisfactory – there were too many scuffing incidents.

Much was done to find the root cause of these incidents, and some of the conclusions were interesting: The liner temperature designed to prevent acid corrosion of the liner surface was so high that there not were margins left to allow for deterioration of engine conditions (aging), and the hard phase content of the liner material already introduced in the "Core Package" had to be reduced.

Another dominant contribution to higher reliability proved to be the introduction of so-called zig-zag oil grooves. These groves replaced the former cylinder standard with grooves only slightly inclined (Fig. 108).

Strangely enough the prescribed remedies of the "Core Package" or "Tribo Pack" worked well on most other engine types, even the big tanker engines.

Mitsubishi

Mitsubishi had not been very active to provide solutions for the large container ships, and it took until 1993 before they had a design, the UEC85LSC, to challenge the virtual duopoly in this sector MAN-B&W and New Sulzer Diesel. Size wise it matches the RTA84C as seen in the table:

The engine is developed based on the UECLSII concept with the 85 bore as the largest, as will be reported later.

Mitsubishi were also keen to introduce measures to reduce the expensive lubrication oil. The Japanese Shipping companies had traditionally used a rather high lube oil dosage in their engines, but with MHI's introduction of the Danish SIP lubrication system in the UEC85LSC engine it was possible to reduce the consumption by 40 % and even maintain a low wear rate of piston rings and cylinder liners [77]. In the late 90's lube oil consumption had become an issue, as the price was 10 times that of fuel oil and MHI were very active to promote their new system as retrofit on existing engines. Also MAN-B&W and NSD were keen to introduce new systems, the Alpha system and the CLU-3 load dependent accumulator system respectively.

Engine type	Bore, mm	Stroke, mm	Power, BHP/cyl	RPM	MEP, bar	Pmax, bar	SFOC, g/BHPh*
UEC85LSC	850	2360	5300	102	17.1	135	121
RTA84C	840	2400	5510	102	17.9	140	126

* ISO ambient conditions, and fuel oil calorific value: 10.200 kcal/kg

13. Engines for Very Large Crude Carriers, VLCC's

The term Superlongstroke engine was used, when Sulzer introduced their RTA series of engines. Since then the stroke to bore ratio has been stepwise increased to what now may be termed "Mega long stroke" engines. These mega long stroke engines are intended for big tankers having relatively low speed and slow running propellers.

MAN-B&W

In 1986 the S-series is introduced and consists of 5 bores, the S80MC3 being the largest and most powerful. It develops 4560 BHP/cyl at 77 rpm and has a mean effective pressure of 17 bar. The stroke is no less than 3056 mm which gives a stroke to bore ratio of 3.82 – so far a new record. The engine is gradually uprated and in year 2000 it develops 5220 BHP/cyl at a mep of 19 bar.

The design concept follows the K90MC3 already mentioned and the S80MC3 also shares some of the same reliability difficulties among others cracks in cylinder liners due to the cast-in cooling pipes. Additionally the cylinder condition was extremely poor with high liner and piston ring wear rates, so the time between piston overhauls were short, around one year. The means to remove the difficulties were also the same as for the big containership engines.

In 1998 a S80MC-C, a S90MC-C, and a L90MC-C is added to the programme (the "C" here stands for Compact, but more about this in the following section) being more cost effective than the S80MC. A comparison between the engine types offered in year 2000 and the old K98FF primarily used in supertankers in the 70's is shown in the table below.

3200 mm stroke is now the record and is expected to stand for some years – still quite formidable, just as the stroke to bore ratio of 4, nearly the double of the K98FF engine, and the specific weight is nearly reduced by 50 %.

The extremely long cylinder liner and the relatively high mean piston speed, C_m compared to the old K98FF engine is a challenge in terms of lubrication, but with the new remedies at hand, firstly introduced on the containership engines, the cylinder condition has been very good.

Engine type	Bore, mm	Stroke,mm	Stroke/Bore	Mep,bar	Power,BHP	rpm	Cm,m/s	BHP/m	Kg/BHP
6S90MC-C	900	3188	3,54	19	39900	76	8.1	3301	27.7
6L90MC-C	900	2916	3,24	19	39840	83	8.1	3213	27.0
6S80MC-C	800	3200	4,00	19	31680	76	8.1	2907	25.4
6S80MC	800	3056	3,82	19	31320	79	8.0	2874	27.6
6K98FF	980	2000	2,04	10.8	22800	103	6.9	1351	45.2

Mitsubishi

Mitsubishi are ready with their super-tanker engine in 1988, the UEC75LSII.

The cut away view of the engine is shown in Fig. 114, from where it is seen that the engine construction now features monoblock, welded bedplate and entablature, but the other major engine parts are following previous MHI concepts adapted as needed to the increased engine size. Here is also seen that the cylinder lubrication takes place at two levels in the liner as in the Sulzer RTA engines. The combustion chamber is well cooled and component temperatures are demonstrated in Fig 115. New, but not seen, is the system that uses part of the energy of the hydraulic oil to rotate the exhaust valve spindle [78].

The most interesting feature is the option to install MET66SR-VG turbochargers with a radial-flow turbine and VG (**V**ariable **G**eometry) nozzles instead of the standard MET SC-charger with an axial flow turbine [79]. The first 8-cylinder engine was actually equipped with the VG unit, and when

inspected after two years at the ships guarantee docking the actuating device and the nozzles were free and operational – much to the surprise of users often seeing turbines fouled as a consequence of the heavy fuel burning. Basically it was the intention that VG should serve to

- reduce the fuel consumption at part load,
- increase the engine overload ability up to 120 %,
- maintain the surge margin in rough sea.

The idea was not adopted by customers, as it was considered too expensive, but there is no doubt that VG is an interesting feature, when total economy of the engine plant, making use of the waste heat, is a must.

The UEC75LSII had one sore point – the main bearings. For some ships in service white metal separation was observed on the side edge of the lower bearing shell. This phenomenon was analysed and the results were reported at the CIMAC congress in Interlagen in 1995 [80] by Mr. S. Yoshihara. It appeared that due to vertical forces in some of the bearings
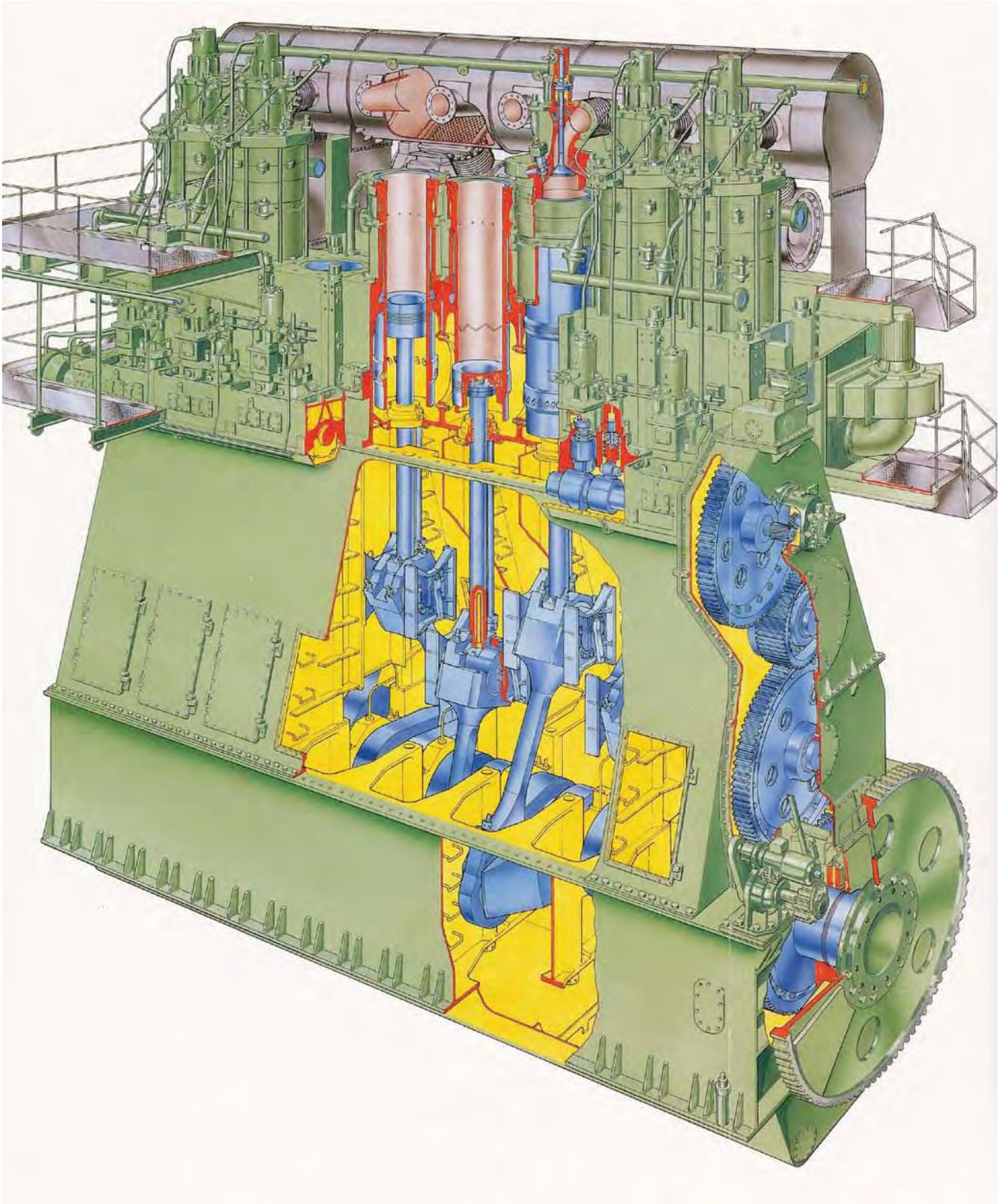
Fig. 114   Cut away view of the tanker engine 8UEC75LSII, bore 750 mm, stroke 2800 mm, 4000 BHP/cyl at 84 rpm and Mep 17 bar. (MHI)

it was necessary to reinforce the bearing caps in question to solve the problem. The research required and the effectiveness of MHI to find a fast solution was appreciated by the Users of the engine type.

In 1991 a larger brother, the UEC85LSII, is launched with very much the same characteristics as the smaller engine.

This engine has more or less the same performance data as the Sulzer RTA84T introduced at the same time. The funny thing is that while MHI have given up selling the low load, VG feature of the 75LSII, Sulzer have included features to the same effect in their new engine, as will be told later. Another funny thing is that MHI with their Sulzer license actually have the possibility of offering both engines to their clients.

Fig. 115 Combustion Chamber temperatures for UEC75LSII at 4000 BHP/cyl and Mep = 17 bar. (MHI [78])

Sulzer

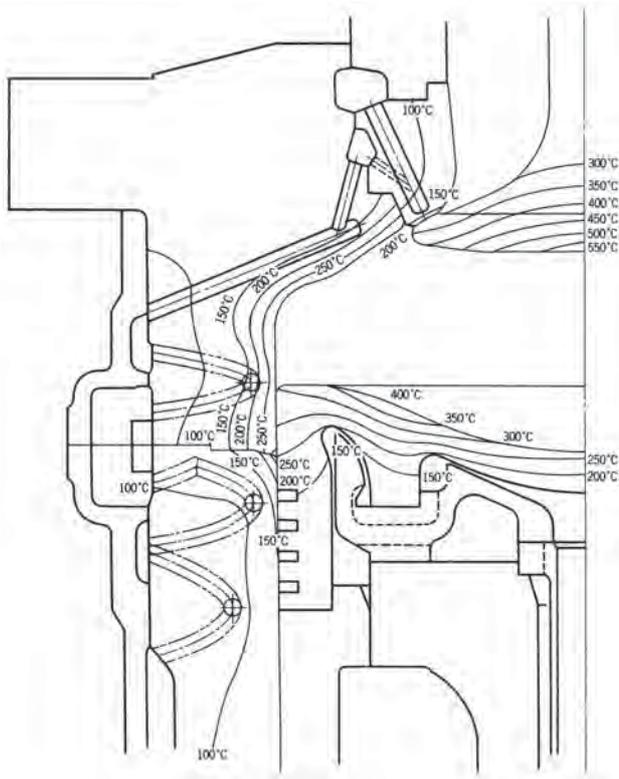The RTA84M introduced in 1984 has been the preferred Sulzer engine for super tankers.

In the beginning of the 90's there is a strong competition from the MAN-B&W's S80MC and Mitsubishi's UEC75LSII and the new UEC85LSII. Especially the Mitsubishi engines are both more compact and not so heavy i.e. cheaper to produce than the M-type. So, a new engine is required to be competitive in this important market segment, and the RTA84T is introduced in 1991. The performance data of the RTA84T and M engine is shown in the table below and compared with 1991 versions of MAN-B&W and Mitsubishi engines.

Even though the stroke is higher than on the M-type the total engine height is lower as the connection rod ratio (half the stroke/the length of the connecting rod) has been reduced from 1:2.4 to 1:2.1, as also has been the case in some previous RTA engines.

The RTA84T is introduced with some very interesting and brand new features not seen before in 2-stroke engine design [81]. Sulzer anticipate that VLCC's will slow steam for quite some time, and that it consequently would be an advantage to provide the engine with features that makes this possible without sacrificing specific fuel consumption nor putting cylinder running conditions at risk.

To that end the well known VIT (variable injection timing) is combined with VEC (variable exhaust closing). The VEC works between 100 % and 80% of engine maximum load, and its function is to keep the compression pressure and therefore the maximum pressure constant by early closing of the exhaust valve at part load. The VIT takes over from 80 % to 65 % engine maximum power, meaning that the maximum pressure of 140 bar is constant down to 65 % load – indeed requiring some considerations regarding the bearing design; however the bearing pressure of the RTA84T engine is even slightly lower than on previous engines.

The other new feature was a load dependent cylinder liner cooling. This was introduced, as it is important to avoid liner temperatures below the dew point in the cylinder in order to avoid cold corrosion over the whole load range. The idea is logical and has furthermore the additional advantage that the increased liner temperatures at low load will lead to increased mechanical efficiency, as the viscous friction losses are reduced. It is strange that this feature disappears from later new designs.

Despite the new cooling system the parameters were not set quite right from the outset, and the consequence was heavy corrosive liner wear, and soon after the engine came into service it became necessary to introduce an improvement package [82], where water ingress from condensation after the air coolers were reduced and/or drained, and the liner temperature level was increased.

In 1996 the engine was upgraded or modified on some points to improve the competitiveness in design for manufacturing and named RT84T-B; however the performance parameters remained unchanged. The cylinder condition of the B-type was very good, as it was on the next following version the D-type announced in 1998, where the previous mentioned "Tribo Pack" cylinder running solution was implemented. However, the cylinder oil consumption was pretty high.

Engine type	Bore mm	Stroke, mm	Stroke/Bore	Mep, bar	Power, BHP	rpm	BHP/m	Kg/BHP
6RTA84M	840	2900	3.45	17,2	30240	81	2535	32.2
6RTA84T	840	3150	3.75	18	31680	74	2830	27.1
6S80MC	800	3056	3.82	18	29700	79	2521	29.6
6UEC85LSII	850	3150	3.70	18	31500	76	2876	27.2

14. Medium and small bore engines up to year 2000

The medium and small bore engines has to cover a broad variation of ships applications above 2000 tdw, such as Bulk carriers, Product tankers, Container feeders, Gas carriers, Car carriers, and many specialised ships. This area constitutes the biggest market for diesel power, but to be preferred as a 2-stroke low speed designer you need a good coverage in power and engine revolutions to cope with the competition from the medium speed engines, where the required propeller revolutions always can be matched.

Multiplicity is the key word to success in this power market, and comparing engine programmes of the three designers' one really sees who is the market leader:

MAN-B&W have 15, Wärtsilä (Sulzer) have 6 and MHI have 6 engines with different bore or stroke to cover the market (bores above 800 mm not included).

Most of these engines are design-wise very close to the larger bores and from that aspect not so interesting, but they are generally characterised with a very high degree of reliability. Large engines – big problems, small engines – small problems. In comparison with medium speed engines the medium size slow speed engines are very robust and even big casualties will rather seldom result in total disasters. An example may explain what this could mean: A fatigue crack in a exhaust valve spindle caused that the valve to fall into the cylinder, but apart from some damages to the piston top and cylinder cover bottom there were no other damages – on a medium speed engine the result would most probably have been, that the connecting rod was passing through the side of the crank casing. Two such failure cases are shown in Fig. 116 & 117.

Fig. 116 Broken valve spindle forged through piston crown. (Author)

Fig. 117 Broken valve spindle curled-up in the compression space. (Author)

MAN-B&W

MAN-B&W were as mentioned having a vast engine programme. One small engine deserves specific mention, the S26MC included in the programme in 1986. It has a cylinder diameter of 260 mm and a stroke of 980 mm yielding 495 BHP/cyl at 250 rpm and at a mep of 16.8 bar. The maximum pressure was higher than in any other engine at the time – 160 bar. A cross section of the small engine is shown in Fig. 118. The small engine was demanding component solutions with different design than the larger engines, as the camshaft integrated in the engine structure as in medium speed engines, and the running gear with round guideshoes Fig. 119, as in some of the Werkspoor-Lugt engines. Crosshead bearing shells are lined with tin aluminium.

The most important development at MAN-B&W was however the new MC-C series of engines (C for compact) with bores of 46, 50, 60 and 70 cm and of very different design compared to the predecessors of the same bore [83]. Concentrating on the 50-bore a com-parison between the MC and the MC-C is shown in Fig. 120.

As the main development target has been to provide lighter and shorter engines the designers have focussed on removal of material, where it was unnecessary and adding material where required by strength and stiffness considerations.

The new engine features "twin staybolts" secured in threads in the top of the bedplate compared to the traditional through going single staybolt. This is the most radical design change in B&W structural design practise. The through going staybolt was intended to provide compression in the welding seams of the bedplate, now welding quality is considered so perfect, that a higher degree of tensile stress can be accepted – actually this was already done in the Götaverken VGS-engines more than 50 years ago. Details of the frame box and the bedplate are shown in Fig. 121. There are several advantages of this new design compared to the traditional design: 1) Deformation of the main bearing bore is much less than previously, 2) The crosshead guides are less deformed, 3) The welding process has been much simplified, and the casting of the main bearing saddle has been simplified too. The MC-C engine has now a cylinder frame with integrated camshaft housing as on the small bore engines.
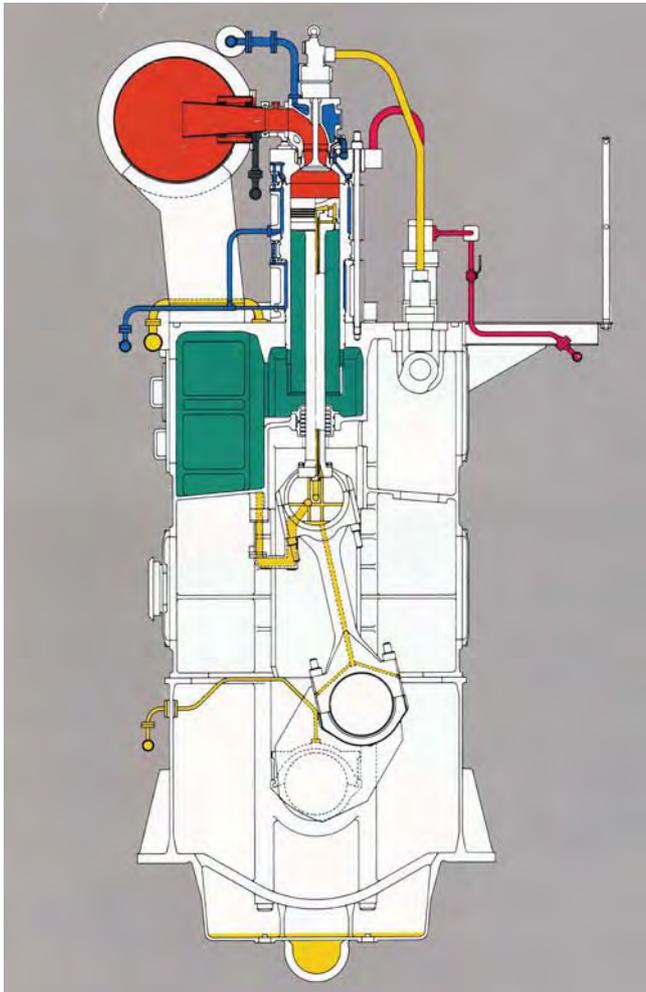
Fig. 118 Cross section of S26MC, bore 260 mm, stroke 980mm, yielding 495 BHP/cyl (365 kW/cyl) at250 rpm and Mep = 16.8 bar. Maximum pressure 160 bar. (MAN-B&W)
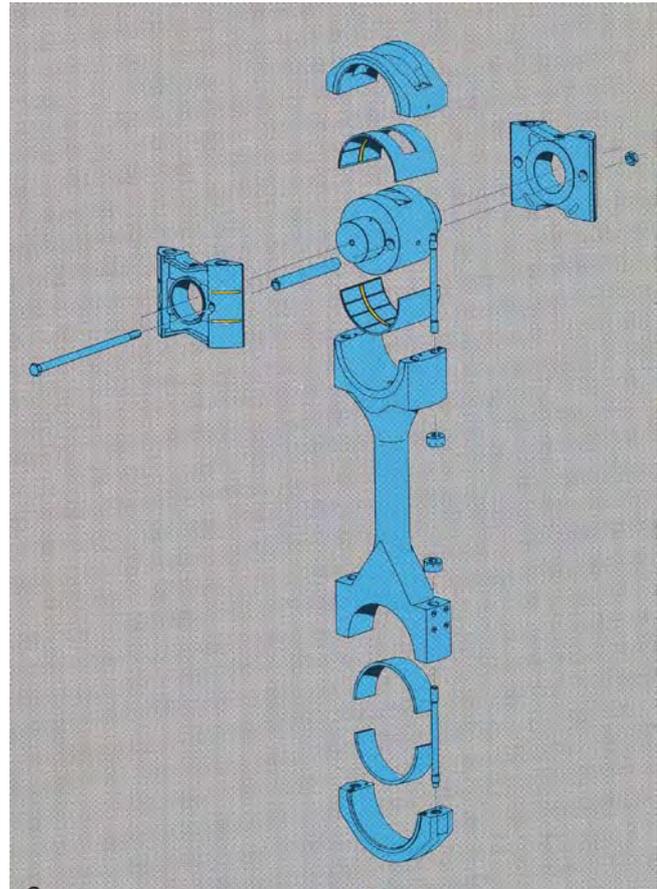

Fig. 119 S26MC running gear with a relatively short connecting rod. (MAN-B&W)
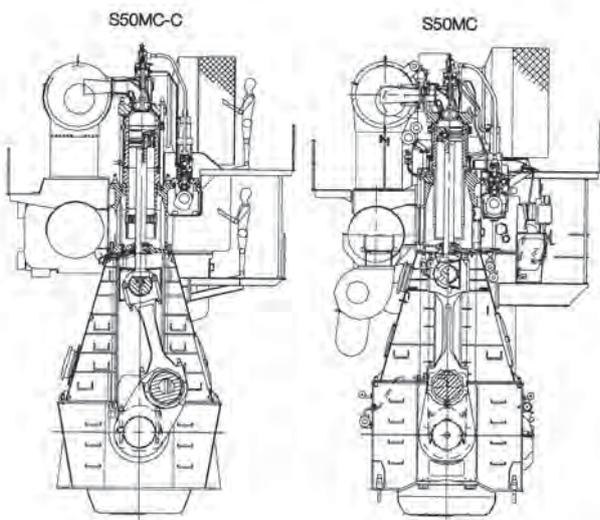

Fig. 120 Comparison of the compact S50MC-C and the standard S50MC engine. The MC-C with 10 % more power per cylinder than the MC but with about 16 tons less dry mass irrespective of cylinder number. (CIMAC [83])
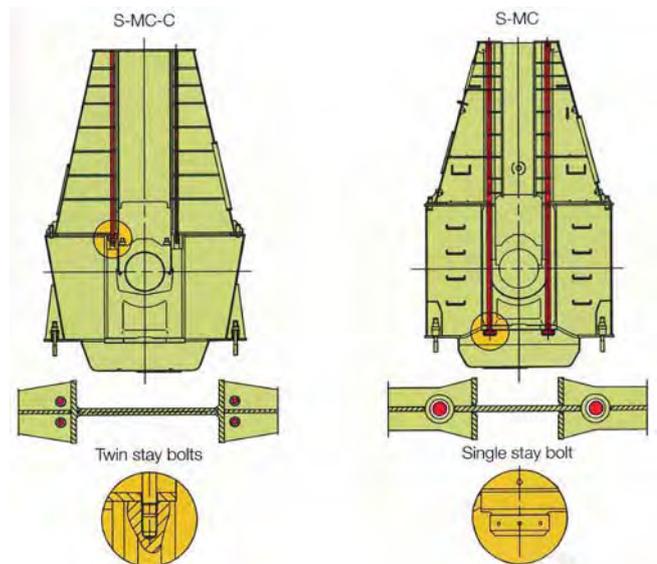

Fig. 121 Comparison on engine frame design between S50MC-C and S50MC engines. (CIMAC [83])

The weight of the 6S50MC-C is 210 tons; a reduction of 28 tons compared to MC and has now reached a specific weight of only **16, 3 kg/BHP**.

The length of a 6S50MC-C-engine has been reduced by more than 700mm compared to the MC counterpart due to a shorter cylinder distance and a shorter thrust bearing/chain drive section. This means also that the width of bearing is reduced and the diameter increased to maintain reliability, however with an additional reinforcement of bearing shell design using Aluminium-tin to increase material fatigue strength – another break in B&W bearing design practise.

Sulzer

The RTA-2 type engines launched in 1984 was in 1992 upgraded in view of adaptation for simplified production and assembly, and the power output was increased by 3 %, now designated RTA-2U. Further in 1997 another 3% power increase was added together with a modernisation package, based on the latest engine development of the T line of engines launched the year before. The designation was now RTA-2U-B [82].

The mentioned T line of engines has bores of 48, Fig 122, 58 and 68 cm, and are collectively termed RTA-8T [84]. These engines are introduced to cover the propulsion demand for the greater standardisation of ship designs for bulkers and tankers, the so called Handysize, Handymax and Panamax types. The 8T-line is characterised with the very high stroke to bore ratio ranging from 4 to 4.17 (as the MAN-B&W MC-C engines) to accommodate low revolutions required by the large propellers of the mentioned vessels with sailing speed typically between 14 and 16 knots. Furthermore the engine length is extremely short.

The introduction rating is on level with the other RTA engines, but in 1997 an updated version; RTA-8TB was introduced with a mean effective pressure of 19 bar and a mean piston speed of 8.5 m/s. Even so, the thermal load of the combustion chamber is very moderate compared to the large bore engines, as shown in Fig. 123.

In 1997 New Sulzer Diesel is taken over by Finland's big engine builder Wärtsilä and the company is now Wärtsilä NSD Corporation. The 2-stroke engine development is maintained in Winterthur, whereas New Sulzer Diesel's 4-stroke medium speed engines are phased out and replaced with a Wärtsilä developed 4-stroke programme.

The first brand new engine to be developed under the new ownership is the RTA60C, which is ready mid 1999. This engine is oriented towards faster vessels such as container ships, container feeders, car carriers and so on. The trend towards designing the engines to be more economical to manufacture is continued in this engine, and the specific engine weight is now down to 17.1 kg/BHP, fairly low for that size of an engine.

The engine is designed in two versions, 1) the standard mechanically controlled version and 2) a RT-flex version which has a fully electronically control-led common-rail systems for fuel injection and exhaust valve ope-ration. Actually the first engines to be sold are of the RT-flex version; how-ever this is outside the scope of this story.
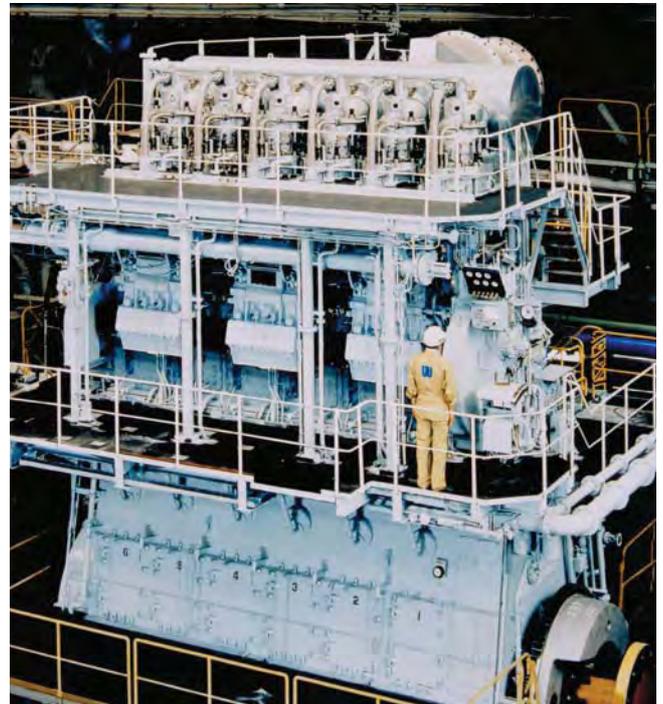

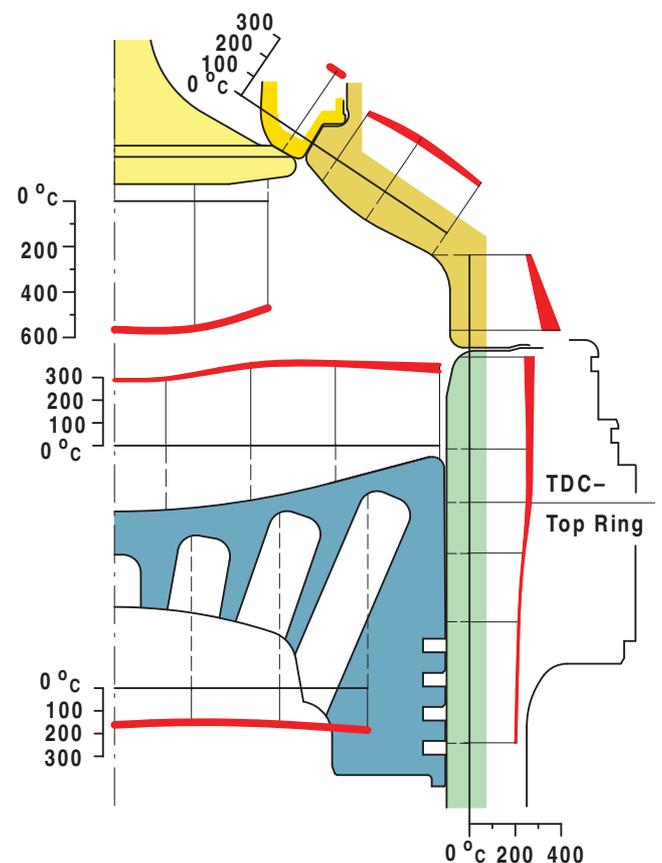Fig. 122   The RTA48T on testbed in Aioi. (WNSD [84])

Fig. 123   Combustion chamber RTA58T, surface temperatures at full load i.e. 2720 BHP/cyl (2000kW/cyl). (Wärtsilä [84])

Mitsubishi

Following the development of the big engines for supertankers the UEC-LSII series is completed during 1993 to 1997 with 5 smaller engines and these are rapidly followed by 4 LSE-engines as the next generation is named. The LSII engines are very much like the bigger brothers apart from the engine structure, which for the smallest engine still is made of cast iron assembled monoblocks. The LSE-engines are getting a power boost, and the engine performance parameters are now very much on line with competition as seen in the table below. The only novelty is application of the newly developed A-ECL cylinder lubrication system being modified from the SIP system mentio-ned previously and the application of Ni-monic material for the exhaust valve spindles, just like the competition.

Engine Type	Bore mm	Stroke, mm	Stroke/ Bore	Mep, bar	Power, BHP	rpm	BHP/m	Kg/BHP
6UEC50LSII	500	1950	3.90	17.4	11220	124	1736	16.9
6UEC52LSE	520	2000	3.85	19.0	13920	127	2034	18.9
6RTA48T-B	480	2000	4.17	19.0	11880	127	1936	16.5
6S50MC-C	500	2000	4.00	19.0	12870	127	1998	16.1

15. The future of 2-stroke crosshead engines

In the 1970's there were many that thought that diesel technology was outdated, and that the large engines would be replaced with advanced steam turbine plants and later on with gas turbines. It did not happen that way, at contrary, what was believed to be a threat for the diesel engines is today nearly expelled from the market, because of the diesel engine's unsurpassed thermal efficiency.

It is indeed an impressive development that has taken place during the 50 years that this report covers. The specific power of the largest engines at any time, have been increased from about 1000 BHP/cylinder (735 kW/cylinder) to 8000 BHP/cylinder (6000 kW/cylinder) on the basis of an increase in mean effective pressure from about 6 bar to 19 bar, a maximum pressure from 50 bar to 150 bar and an increase in mean piston speed from 5.5 m/s to 8.5 m/s, as shown in Fig. 124.

The future will also see the diesel engine as the preferred propulsion machinery. Apart from the superior fuel economy this is due to the great possibilities at hand to make use of the exhaust gas waste heat, where consumption reductions from 10 to 15 % are possible. This will of course require considerable additional investments in the ship machinery plant, but the pay pack time will only be 3 to 4 years under the prevailing economical conditions, and the inherent reductions in CO_2, NO_x, SO_x and particulates will be an attractive extra "free" bonus.

The engine development, as demonstrated in this report, has brought along concepts that now are so close to each other that it is nearly impossible to distinguish the three present makes. The new development will take place on electronic control systems and hydraulic activation systems. Sulzer, in 1981, was the first to make experiments with an engine having a single cylinder activating concept. This was abandoned, but MAN-B&W have taken over the idea and could in 2000 as the fist designer install the new concept on a 6L60MC engine in service. Mitsubishi has adopted a similar principle. Sulzer have instead chosen an electronically controlled common rail system, the RT-flex system mentioned previously, and the first RT-flex common rail engine entered service in 2001. So, now again there will be a marked difference between the makes, however, it is outside the intentions of this report to get into the details of this development.

The low-speed two-stroke engine has proved to be the most efficient prime mover for ship propulsion and, with turbocharging, has been developed to even greater efficiency with longer times between overhauls. The power output has been developed by an order of magnitude from the first turbocharged engines in the 1950s to the most powerful engines today. It has also been developed to have the flexibility of burning really poor quality fuel oils and even natural gas. Besides the drastic reductions in fuel consumption, there have also been dramatic reductions in exhaust emissions. With such a past, today's turbocharged two-stroke crosshead engines have an exciting future.
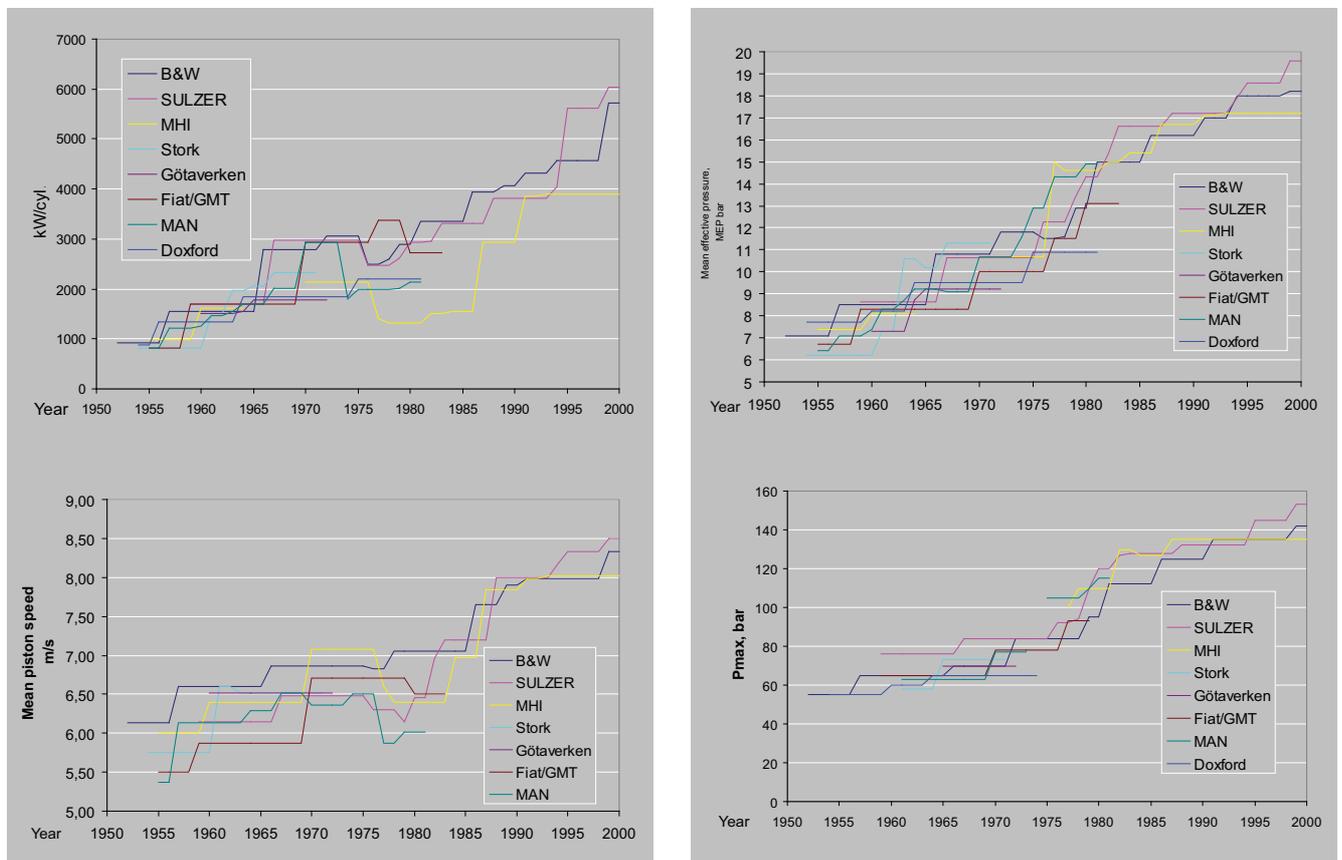

Fig. 124   development in performance parameters over 50 years (Author)

III Design timelines

In this section the development of some of the most important engine component characteristics are shown for all the designers during the 50 years. The somewhat simplified characterisation is clarified by examples of actual designs and by references to figures presented in the text of the previous two sections.

Components for B&W engines from VTBF to K/LGF are very clearly shown in "Half Century of Diesel Engine Progress at Mitsui" [85].

The systems or components listed are:
• Scavenging systems (inside cover)
• Crossheads
• Cylinder covers
• Cylinder liners
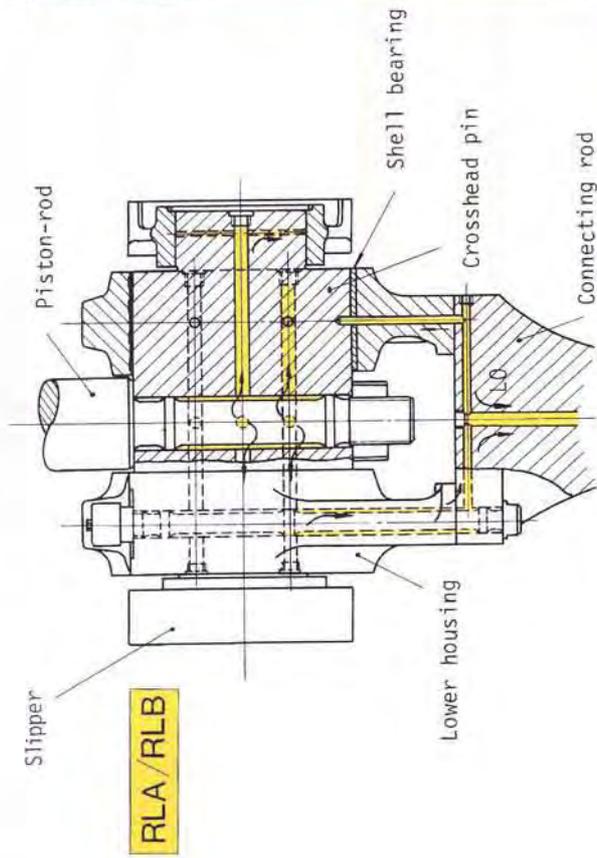• Pistons
• Exhaust valves
• Engine frames
• Crankshafts.

The survey makes it possible to follow where and when new ideas were introduced and by whom, and how they possibly were adopted by competition later on. The design characteristics are in certain cases not identical for small and large bore engines and differences have to a certain degree been noted by additional text.

In the horizontal lines the engine types of the different designs are included with the first name letter positioned at the year of introduction. The name position on one of the three horizontal lines belonging to each designer has no specific meaning – it is just a matter of finding space.
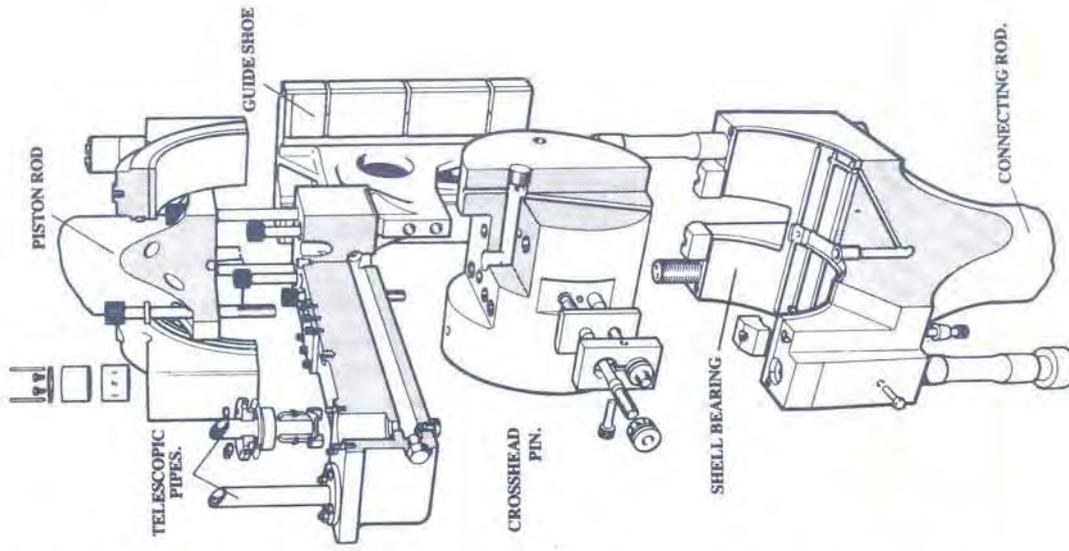
Crosshead

Legend (bearing material): cast-in white metal | thin steel shells, white metal lined | steel shells with Al Sn lining

Legend (design): 2 seperate bearings | full width bearing

Legend (bearing lubrication): from common lube oil system | from high pressure system | from attached high pressure pumps

	bearing material	design	bearing lubrication
	cast-in white metal	thin steel shells, white metal lined	steel shells with Al Sn lining
	2 seperate bearings	full width bearing	
	from common lube oil system	from high pressure system	from attached high pressure pumps

Burmeister & Wain:
VTBF · 74VTBF160 · VTBF opposed piston · VT2BF · K-EF/FF · K98FF · K-GF · K/L-GFC · L-GF · K/L-GFCA · L-GB/GBE · K/L-S-MC/MCE ......mark 3,4,5,... · L-MC/MCE · K-MC-C · K98MC · S-MC-C

Doxford:
LBD · BDS · -P · 600LBD · §§ §§§§§§ · -J · -J-C · -JS-C · SEAHORSE project (D) · 58JS3C
§ with seperated central pad support

Götaverken:
VGU / VGSU · 520/900VGS-U · 850/1700VGA-U · 750/1600 & 630/1400 VGS-U* · 520/900VGS-U* (B) · 850/1700VGS-U*
*with seperate exhaust camshaft

Fiat /GMT:
...S · 750S · B...S · B... (C) · C... · 1060S · CC600
* eccentric X-head pin

MAN:
KZ-C · KZ78/140C (A) · KZ-D · KZ-E · KZ-F · KSZ · KSZ-A · KSZ-C/CL · KSZ105/180 · KSZ-B · KSZ-BL

Mitsubishi:
UEC-A · UEC75A · UEC-B · UEC-C · UEC-D · UEC-E · UEC-H · UEC-L · UEC-LS · UECLSII · UEC85LSC · -LSE · UEC52E · UEC-HA · UEC-LA · UEC75LSII · UEC85LSII

Werkspoor / Stork-Werkspoor / Stork:
KEBS68/125 · HOTLo75 · SW85/180 · SW90/170

Sulzer:
SAD · RD · SAD72 · RSAD · RND · RND105 · RNDM · RTA · RTA-M · RTA-C · RTA84T · RTA84TD · RLA · RLB · RTA-2 · RTA-2U · RTA-2UB · RTA-8T · -TB · RTA96C · RTA60C · (E) (F)

Year: 1950 · 1960 · 1970 · 1980 · 1990

(D) *Doxford JC, centre running gear* (Doxford)

GUIDE SHOE

PISTON ROD

TELESCOPIC PIPES.

CROSSHEAD PIN.

SHELL BEARING

CONNECTING ROD.

(A) *X-head oil pump KZ-engines* (MAN)

(E) *Sulzer Crosshead, RLA/RLB engines* (Hitachi)

Piston-rod

Shell bearing

Crosshead pin

Connecting rod

Slipper

RLA/RLB

Lower housing

(F) *Crosshead pin and guide shoes, RTA52* (Sulzer)

(B) *Fig. 52, (C) Fig. 39*

Cylinder liner

80

design

Legend:
- simpel with collar
- bore cooled (collar)
- strong back
- cast-in cooling tubes
- 2 piece liner with strong back
- 3 piece liner

*bore/cast-in tubes cooling for large bore engines only

Burmeister & Wain
- VTBF
- 74VTBF160
- VTBF opposed piston
- VT2BF (A)
- K-EF/FF
- K98FF (A)
- K-GF
- L-GF
- K/L-GFC
- K/L-GFCA
- K/L-GF
- L-GB/GBE *
- L-MC/MCE
- K-MC-C
- K/L/S-MC/MCE......mark 3,4,5,....
- S-MC-C * (K)
- K98MC (K)

Doxford
- LBD   BDS   -P
- -J
- 600LBD (B)
- -J-C   -JS-C
- (F)
- SEAHORSE project
- 58JS3C

Götaverken
- VGU / VGSU
- 520/900VGS-U
- 750/1600 & 630/1400 VGS-U*
- 520/900VGS-U*
- 850/1700VGA-U
- 850/1700VGS-U*

Fiat /GMT
- ...S
- 750S
- B...S
- B...
- C....
- (D)
- 1060S
- # # # (J)
- CC600
- #liner in two pieces
- *with seperate exhaust camshaft

MAN
- KZ-C
- KZ78/140C
- KZ-D   KZ-E
- KZ-F   KSZ   KSZ-A
- KSZ-B (E)
- KSZ-BL
- KSZ-C/CL
- KSZ105/180

Mitsubishi
- UEC-A
- UEC75A
- UEC-B
- UEC-C
- UEC-D
- UEC-E
- UEC52E
- UEC-H
- UEC-HA   UEC-LA   UEC75LSII
- UEC-L   UEC-LS   UECLSII   UEC85LSC
- UEC75LSII (I)
- -LSE

Werkspoor Stork-Werkspoor Stork
- KEBS68/125
- STORK HOTLo75
- SW85/180
- SW90/170
- •2 part liner

Sulzer
- SAD   RD
- SAD72 (C)
- RSAD
- RND
- RND105
- RNDM
- RLA (G)
- RLB
- RTA-2
- RTA (G)
- RTA-M
- RTA-C
- RTA84T
- RTA84TD
- RTA-2U
- RTA-2UB
- RTA-8T   -TB (H)
- RTA96C
- RTA60C

Year: 1950   1960   1970   1980   1990
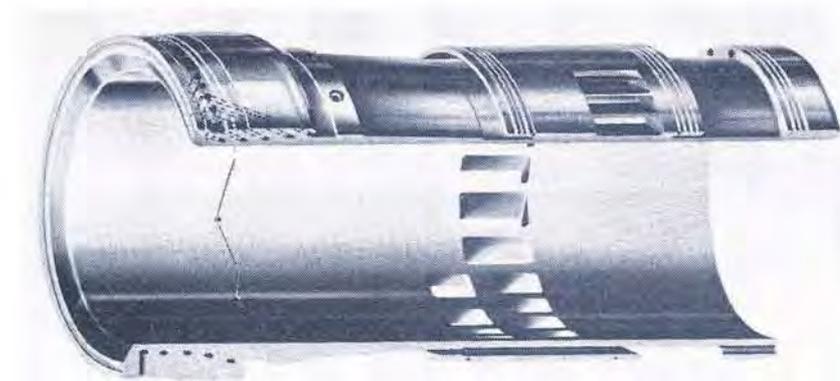
(I) *UEC60L, Cylinder Liner with scewed ports* (MHI)

(H) *Bore cooled RTA cylinder liner* (Sulzer)
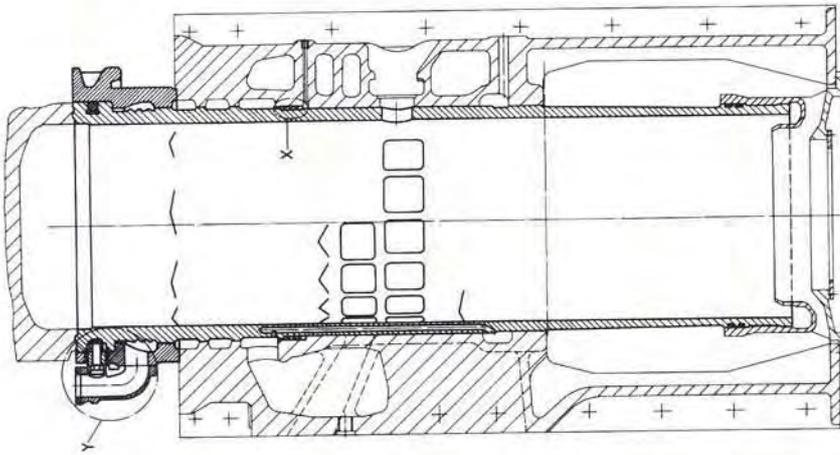
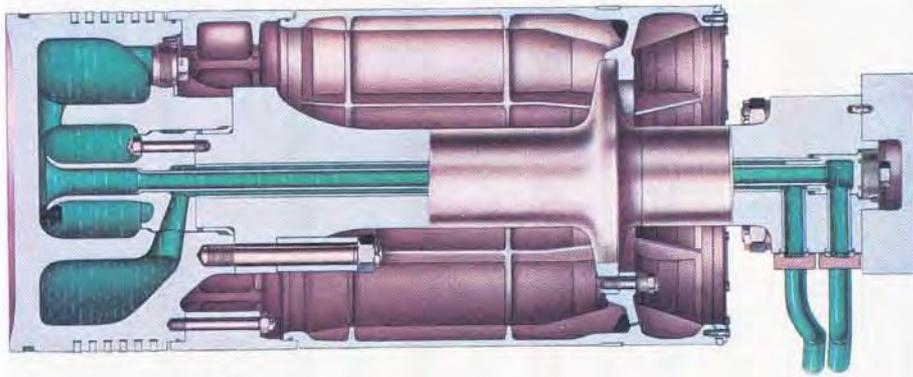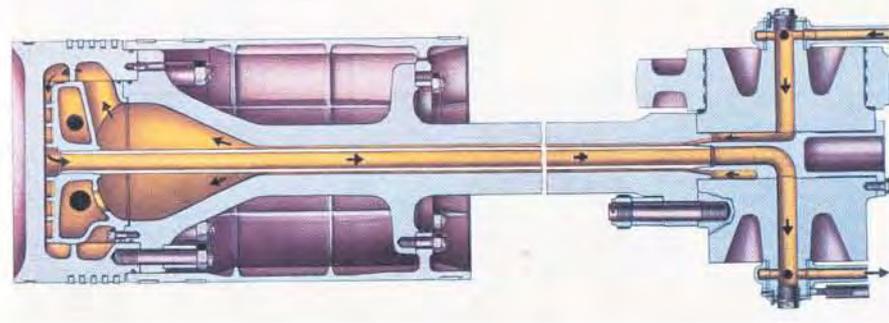(E) *Cylinder liner with water cooled exhaust port bars, KSZ90B* (MAN)

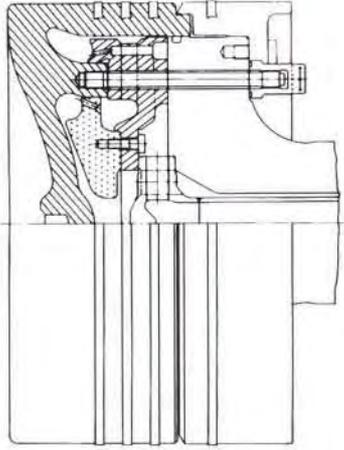(C) *Borecooled cylinder liner RND105* (Sulzer)

(A) *Fig. 42,* (B) *Fig. 27,* (D) *Fig. 39,* (F) *Fig. 72,* (G) *Fig. 81,* (J) *69,* (K) *Fig. 106*

Piston

Legend

piston cooling: water cooling | oil cooling

piston crown: shell, outer support | inner support | outer & inner support | cast, radially ribbed | strong back | bore cooled | semi bore cooled

Burmeister & Wain
- VTBF | 74VTBF160 | VTBF opposed piston (C)
- VT2BF (C)
- K-EF/FF (C) | K98FF
- K-GF | L-GF
- K/L-GFC | K/L-GFCA | L-GB/GBE
- K/L-MC/MCE | L-MC/MCE (L)
- K/L/S-MC/MCE......mark 3,4,5.... (L)
- ¤ with inconel cladding, large bore.
- K-MC-C | S-MC-C

Doxford
- lower piston: LBD | BDS | -P | 600LBD
- upper piston: -J | -J-C | -JS-C (I)
- SEAHORSE project | 58JS3C

Götaverken
- VGU / VGSU | 520/900VGS-U
- 850/1700VGA-U
- 750/1600 & 630/1400 VGS-U* (F)
- 520/900VGS-U*
- 850/1700VGS-U*
- *with seperate exhaust camshaft

Fiat /GMT
- ...S | 750S (D)
- B...S | B... | C....
- # # # # # #
- 1060S | CC600
- # with crown stainless steel coated
- ¤ 1060 engine

MAN
- KZ-C (A) | KZ78/140C
- KZ-D | KZ-E | KZ-F (G) | KZ105/180
- KSZ | KSZ-A | KSZ-B | KSZ-C/CL
- KSZ-BL *
- 90/160B/BL only
- ¤ 57 & 60 bore only

Mitsubishi
- UEC-A | UEC75A
- UEC-B | UEC-C
- UEC-D | UEC-E | UEC-H (J)
- UEC52E
- UEC-HA | UEC-LA | UEC75LSII
- UEC-L | UEC-LS | UECLSII | UEC85LSC | -LSE

Werkspoor / Stork-Werkspoor / Stork
- KEBS68/125 (B)
- STORK HOTLo75
- SW85/180 | SW90/170

Sulzer
- RSAD | SAD | SAD72 | RD
- RLB | RLA (H) | RND | RNDM | RND105
- RTA (H) | RTA-M | RTA-C | RTA-2 | RTA-2U | RTA-8T (K)
- RTA84T | RTA84TD | RTA-2UB | -TB
- RTA96C | RTA60C (M)
- 38 & 48 bore only

Year: 1950 | 1960 | 1970 | 1980 | 1990

(J) *Piston design UEC52L* (MS)

(K) *RTA84C piston withdrawal.* (Wärtsilä)

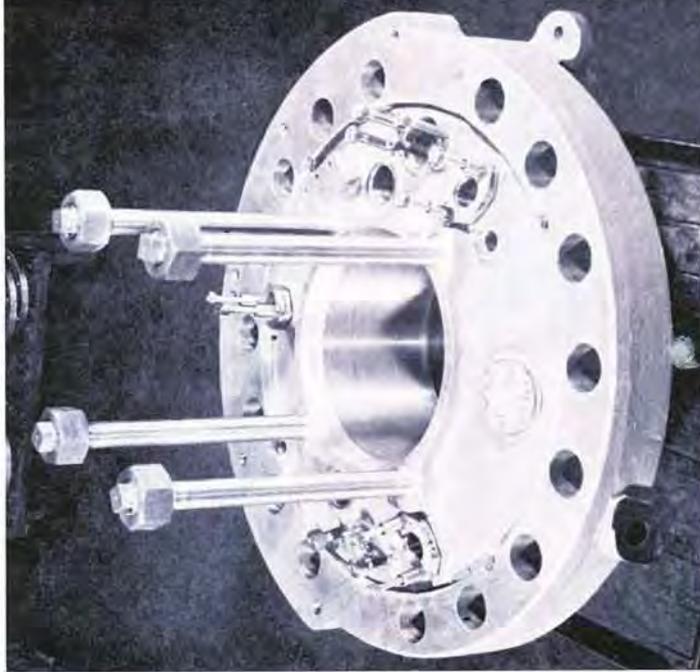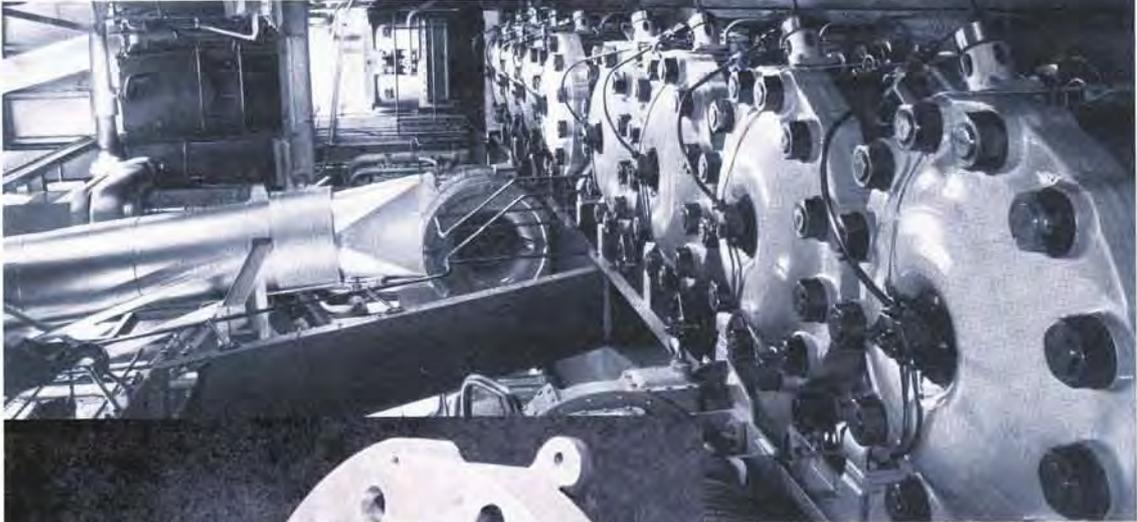(A) *Pistons for small (oil cooled) and larger (water cooled) KZ-engines* (MAN)

(B) *Fig. 15,* (C) *Fig. 42,* (D) *Fig. 32,* (E) *Fig. 38,* (F) *Fig. 51,* (G) *Fig 43,* (H) *Fig. 81,* (I) *Fig. 73,* (L) *Fig. 106,* (M) *Fig. 113*

Cylindercover

design
- strong back, semi bore cooled
- strong back, steel/cast iron
- single piece, steel cast, shell type

design
- forged, bore cooled
- single piece, cast iron
- steel cast, 2 concentric pieces

*with seperate exhaust camshaft

\# large bore, 850 & 750 only

\# small bore, 60 and below

KSZ52 only

Burmeister & Wain

VTBF · 74VTBF160 (B) · VT2BF (B) · K-EF/FF (B) · K-GF (H) · K/L-GFC · K/L-GFCA · L-MC/MCE · K/L/S-MC/MCE......mark 3,4,5,.... · K98MC
VTBF opposed piston · K98FF · L-GF · L-GB/GBE · K-MC-C · S-MC-C

Götaverken

VGU / VGSU · 750/1600 & 630/1400 VGS-U* · 520/900VGS-U* \#
520/900VGS-U · 850/1700VGA-U · 850/1700VGS-U*
(E)

Fiat /GMT

...S · 750S (D) · B...S (G) · B... · C...
(G) · 1060S · CC600

MAN

KZ-C · KZ78/140C (C) · KZ-D · KZ-E · KZ-F · KSZ · KSZ-A · KSZ-B (C) · KSZ-BL · KSZ-C/CL
KSZ105/180 (I) (J)

Mitsubishi

UEC-A · UEC75A · UEC-B · UEC-C \# · UEC-D · UEC-E · UEC-H · UEC-L · UEC-LS · UEC75LSII · UEC85LSC · -LSE
UEC52E · UEC-HA · UEC-LA · UECLSII

Werkspoor
KEBS68/125 (A)

Stork-Werkspoor
STORK HOTLo75 · SW85/180 · SW90/170

Stork

Sulzer

SAD · RD · RSAD · RND · RNDM · RTA · RTA-M · RTA-C · RTA84T · RTA84TD
SAD72 (F) · RLA (K) · RLB · RTA-2 (L) · RTA-2U · RTA-2UB
RND105 · RTA-8T · -TB · RTA96C · RTA60C

Year: 1950 · 1960 · 1970 · 1980 · 1990

84

(J) Cover for KSZ90B/BL (MAN)
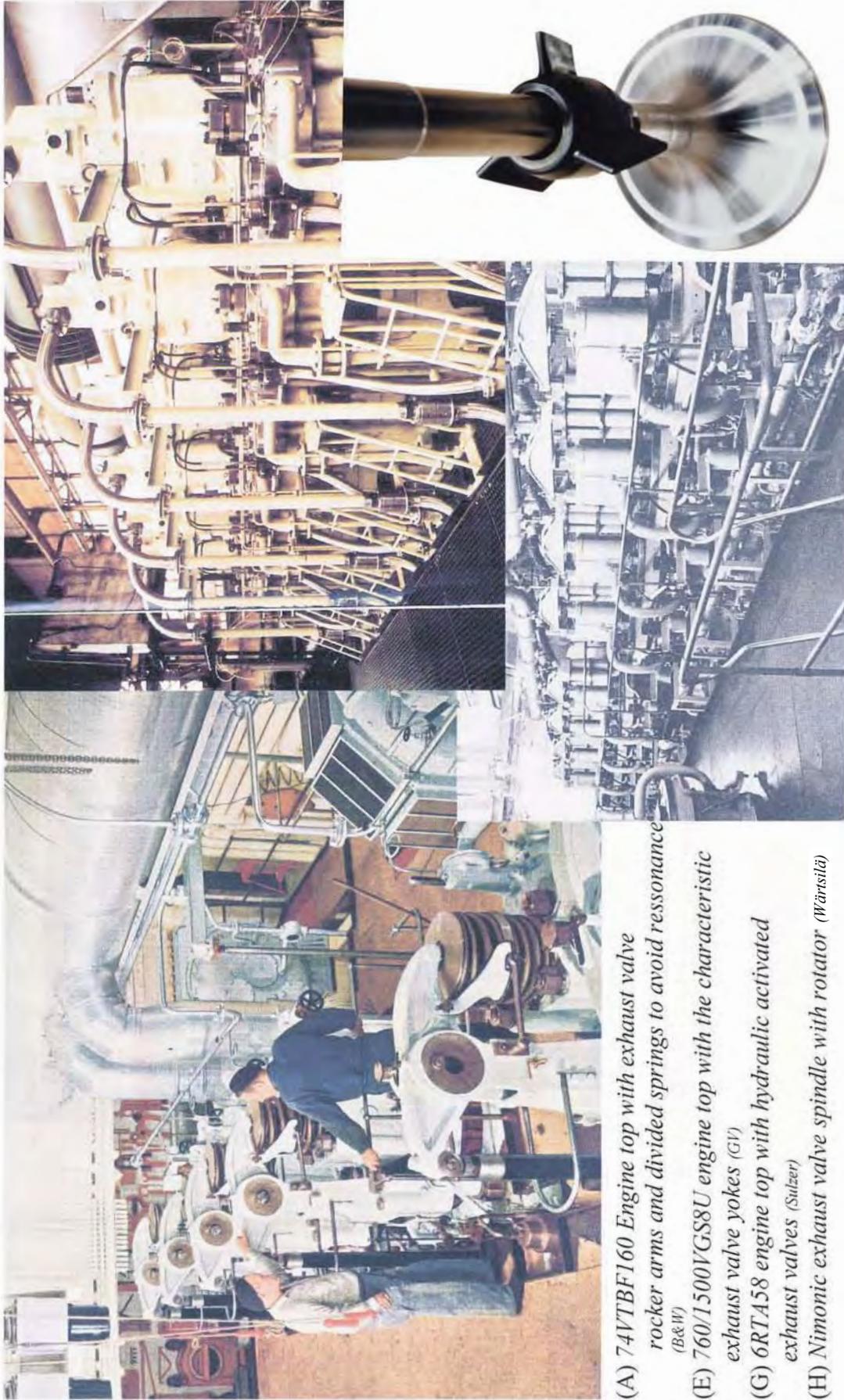
(E) Fig. 50, (F) Fig. 47,
(K) Fig81, (L) Fig. 89,

(B) Cylinder cover 84VT2BF180 (B&W)

(C) Engine top with cylinder covers
KZ70/120 (MAN)

(A) Fig. 13, (B) Fig42, (D) Fig. 32,
(G) Fig. 46, (H) Fig. 54, (I) Fig. 68,

Exhaust valves

Exh. valve(s), general	single valve, rocker arm	3 valves, rocker arm	4 valves, rocker arm/pullrod	1 valve, pullrods-yoke	1 valve, hydraulic actuation
Spindles	austenitic steel, stellite	nimonic, with rotator	austenitic steel no facing	chrome steel, no facing	
Seat	uncooled	water cooled			

Burmeister & Wain
VTBF (A) · VT2BF · K-EF/FF · K-GF (F) · K/L-GFC · K/L-GFCA · K/L/S-MC/MCE……mark 3,4,5,…… (I)
74VTBF160 · K98FF · L-MC/MCE · ¤ with rotator…… · K98MC
L-GF · L-GB/GBE · K-MC-C · S-MC-C

Götaverken
VGU / VGSU · 750/1600 & 630/1400 VGS-U* · *with seperate exhaust camshaft
520/900VGS-U · 520/900VGS-U*
850/1700VGA-U · 850/1700VGS-U*

Mitsubishi
UEC-A (D) · UEC-B · UEC-C · UEC-D · UEC-E · UEC-H · UEC-L · UEC-LS · UECLSII · UEC85LSC · -LSE
UEC75A · ¤ with hydraulic rotator………
UEC52E · UEC-HA · UEC-LA · UEC75LSII

Werkspoor / Stork-Werkspoor / Stork
KEBS68/125 (C) · SW85/180 · SW90/170
(B) STORK HOTLo75

Sulzer
SAD · RD · RND · RNDM · RTA · RTA-M · RTA-C · RTA84T · RTA84TD
SAD72 · RLA · (G) (H) · RTA-2U · RTA-2UB
RSAD · RLB · RND105 · RTA-2 · RTA-8T · -TB
RTA96C
RTA60C

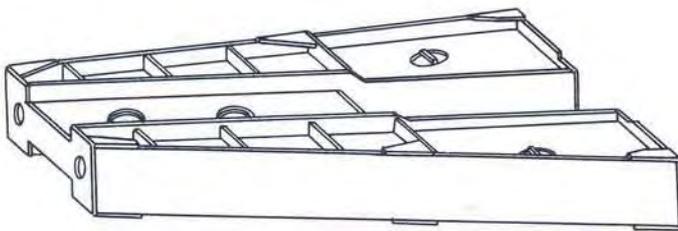| Year	1950	1960	1970	1980	1990

(A) 74VTBF160 Engine top with exhaust valve rocker arms and divided springs to avoid ressonance (B&W)

(E) 760/1500VGS8U engine top with the characteristic exhaust valve yokes (GV)

(G) 6RTA58 engine top with hydraulic activated exhaust valves (Sulzer)

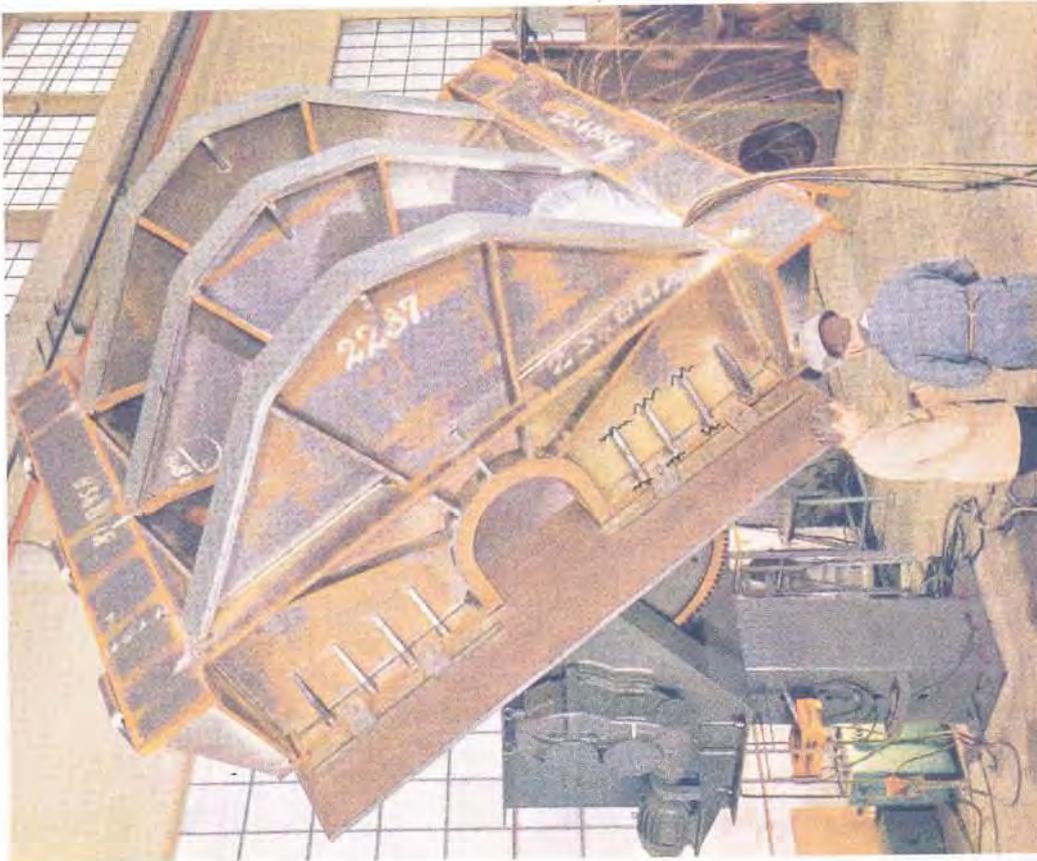(H) Nimonic exhaust valve spindle with rotator (Wärtsilä)

(B) Fig. 9, (C) Fig. 12 & 14, (D) Fig. 21, (F) Fig. 56, (G) Fig. 87, (I) Fig. 106.
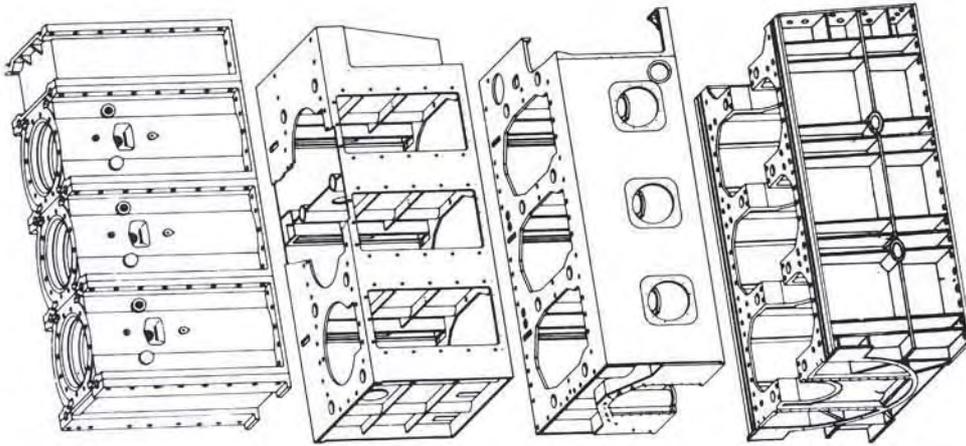
Engine frame

Legend:

cylinder frame	cast per 1 or more cylinders (boltet together)	welded single cylinder units	welded multi cyl box
crank case frame	A frames, cast or welded	welded single cylinder units	welded multi cyl box
bedplate	cast iron, one or more parts (bolted together)	welded, one or two parts (bolted together)	

cast iron multi cylinder box

Burmeister & Wain
VTBF · 74VTBF160 · (A) · VTBF opposed piston · VT2BF · K-EF/FF · K98FF · K-GF · K/L-GFC · K/L-GFCA · L-GF · L-MC/MCE · K/L-MC/MCE......mark 3,4,5.... (E) · L-GB/GBE · K-MC-C · S-MC-C · small bores

Doxford
LBD · BDS · -P · 600LBD · -J · -J-C · -JS-C · SEAHORSE project · 58JS3C

Götaverken
VGU / VGSU · 520/900VGS-U · # · # 850/1700VGA-U · # 750/1600 & 630/1400 VGS-U* · (B) · # 520/900VGS-U* · 850/1700VGS-U*
*with seperate exhaust camshaft
VGU/VGA-engines only

Fiat /GMT
...S · 750S · B...S · B... · C... · 1060S · CC600

MAN
KZ-C · KZ78/140C · KZ-D · KZ-E · KZ-F · KSZ · KSZ-A · KSZ-B · (D) · KSZ-C/CL · KSZ105/180 · KSZ-BL

Mitsubishi
UEC-A · UEC75A · UEC-B · UEC-C · UEC-D · UEC52E · UEC-E · UEC-H · UEC-HA · UEC-LA · UEC-L · UEC-LS · UEC75LSII · UECLSII · UEC85LSC · -LSE

Werkspoor Stork-Werkspoor Stork
KEBS68/125 · STORK HOTLo75 · SW85/180 · SW90/170

Sulzer
SAD · RD · SAD72 · RSAD · RND · RND105 · RNDM · RLA · (C) · RLB · RTA · RTA-2 · RTA-M · RTA-C · RTA84T · RTA-2U · RTA-8T · -TB · RTA84TD · RTA-2UB · RTA96C · RTA60C · 38 type only · small bores

Year: 1950 · 1960 · 1970 · 1980 · 1990

(D) *Bedplate & Frames*
KSZ90B/BL (MAN)

(C) A-frame, RND90M
(Hitachi)

(A) *Welded Bedplate,*
74VTBF160 (B&W)

(B) *Fig.53, (E) Fig. 121*

Crankshaft

Legend: fully built · semi-built cast or forged cranks, forged pins · single piece, forged · welded shaft

Burmeister & Wain

VTBF · 74VTBF160 · VT2BF (A) · VTBF opposed piston · up to 9 cyl · K-EF/FF · K98FF · K-GF · K/L-GF · K/L-GFC · K/L-GFCA · K98MC · L-GF · L-GB/GBE · L-MC/MCE · (E) K/L/S-MC/MCE......mark 3,4,5,.... · K-MC-C · S-MC-C · small engines

Doxford

LBD · BDS · -P · 600LBD (B) · -J · -J-C · -JS-C · (B) · SEAHORSE project · 58JS3C

Götaverken

VGU / VGSU · 520/900VGS-U · 750/1600 & 630/1400 VGS-U* · 520/900VGS-U* · 850/1700VGA-U · 850/1700VGS-U*
*with seperate exhaust camshaft

Fiat /GMT

...S · 750S · B...S · B... · C...

MAN

KZ-C · KZ78/140C · KZ-D · KZ-E · KZ-F · KSZ · KSZ-A · KSZ105/180 · KSZ-B · KSZ-C/CL · * · *52/105 only · KSZ-BL

Mitsubishi

UEC-A · UEC75A · UEC-B · UEC-C · UEC-D · UEC-E · UEC-H · UEC-L · UEC-LS · UEC52E · UEC-HA · UEC-LA · UEC75LSII · UEC85LSC · -LSE · UECLSII · 52 and smaller bores

Werkspoor / Stork-Werkspoor / Stork

KEBS68/125 · STORK HOTLo75 · SW85/181 · SW90/170

Sulzer

SAD · RD · SAD72 · RSAD · RND · RND105 · RNDM · RLA · RTA · RTA-M · RTA-C · RTA-2 · RTA84T · RTA84TD · RTA-2U · RTA-8T · RTA-2UB · -TB · RTA96C · RTA60C · RLB · (C) · (D) · 38-type only

Year: 1950 · 1960 · 1970 · 1980 · 1990

(E) *Welded crankshaft 6L60MC* (B&W)

(C) *Semi built crankshaft 6RTA58* (sulzer).
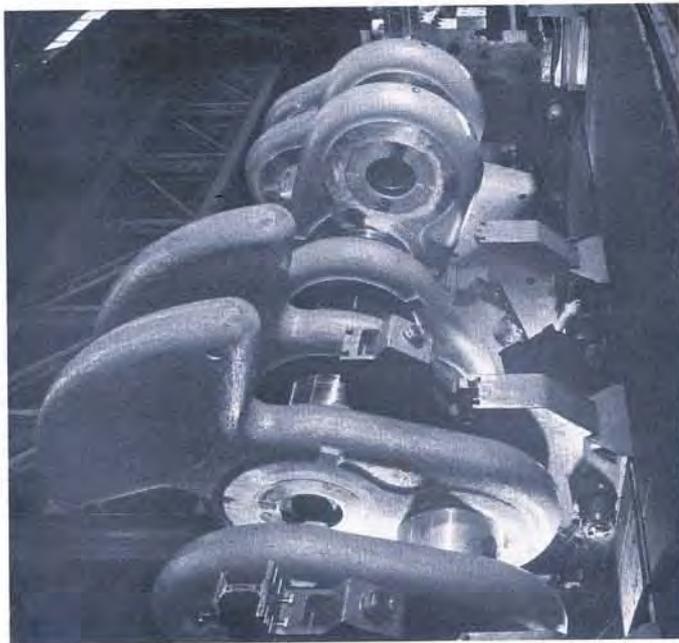
(A) *Fully built crankshaft 84VT2BF180* (B&W)
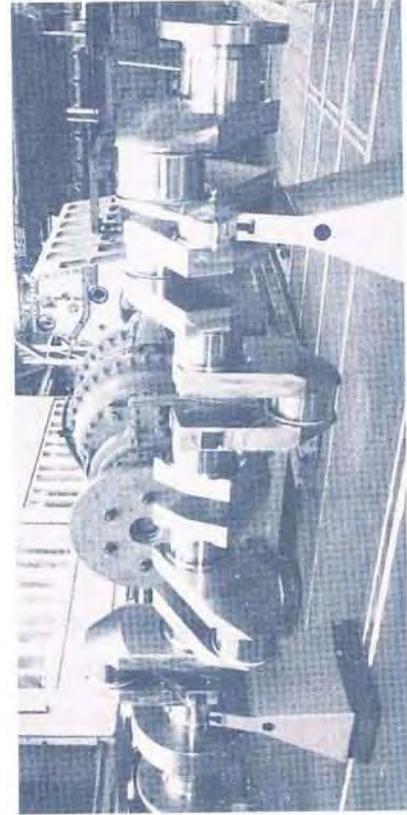
(D) *Forged single piece crankshaft 6RTA38* (ms)

(B) *Fig. 25,* (E) *Fig. 90.*

REFERENCES

Reference to Motor Ship in figure texts are in agreement with engine builder/designer in question.

GENERAL REFERENCES:

1. MAN Diesel, Ole Sørensen: B&W Dieselmotorens Historie 1898 – 2008 (in Danish).

2. Keijiro Tayama: Historical Development of Two Stroke Slow Speed Marine Diesel Engine, National Science Museum Japan, vol. 8 March 2007. (in Japanese).

3. Ture Rinman: The story of an engine, Götaverken publication, 1964

4. David T. Brown: A History of the Sulzer low-speed marine diesel engine, Sulzer publication 1984

5. MAN internal documents from 1941 to 1984, MAN-Museum and Historical Archive of MAN Augburg (in German)

6. Ernst Jenny: The BBC Turbocharger, Birkhäuser Verlag AG 1993

SPECIFIC REFERENCES:

7. Søren Hansen: 2-stroke turbocharged engines for outputs above 10.000 BHP per shaft – experience with the first plants of this type. CIMAC Den Hague 1955

8. H. Andresen: The Present Day Marine Engine (Paper read at Norsk Ingeniørforening 1953), B&W publication

9. F.G. van Asperen: Development and service results of a high powered turbocharged two cycle marine diesel engine, CIMAC Den Haag 1955

10. The Motor Ship: A 9600-BHP Werkspoor-Lugt Engine, July 1953

11. The Engineer: 1800 b.h.p. Werkspoor – Lugt Marine Oil Engine, February 1951

12. K.K.van Hoffen: Werkspoor's Stake in Marine Diesel Engine Development, The Motor Ship April 1970

13. G. Wieberdink & A. Hootsen: Supercharging by means of turboblowers applied to two-stroke diesel engines of large output, e.g. More than 10.000 BHP, for ships propulsion, CIMAC Den Haag 1955

14. E. A. Van der Molen: Test-bed Research on the first Stork Large Bore Engine, The Motor Ship January 1963

15. L-T Collin: The constant pressure charged two stroke engine, its present state and development possibilities, CIMAC Copenhagen 1962

16. Hideo Fujita: High Powered "UEC" Marine Diesel Engine, CIMAC Den Haag 1955

17. G.G.Jackson: Progress on the Development of the Doxford Oil Engine, Doxford Engines 1979

18. P. Jackson: Development of the Opposed Piston Engine to 10.000 H.P. and over, CIMAC Den Haag 1955

19. T.A.Greenwell: 60 Years of Doxford Opposed-piston Engine Development, The Motor Ship, April 1970

20. The Motor Ship June 1962: A Doxford Opposed-piston Engine of 30.000 b.h.p. Envisaged

21. The Motor Ship October 1966: Design changes in the Doxford J Engine

22. Dr. Ing Antonio Gregoretti: La suralimentation des moteurs à 2 temps à balayage transversal ayant particulièrement égard aux moteurs Fiat, CIMAC Zürich 1957

23. Dr. Ing. Antonio Gregoretti: Considerations sur le Projet d'un Moteur Diesel de la Puissance maximum de 3.000 CH/CYL., CIMAC Copenhagen 1962

24. P L Scioldo: A marine unit to give more than 40.000 bhp at 102 rev/min, The Motor Ship, Special Survey, August 1967

25. Internal MAN document by Dr. Ing. D. Von Lassberg 1974/75: Die entwicklung der Mittel-Gross-Dieselmotoren im Werk Augsburg der MAN 1941–1971

26. Prof. Dr. Ing. E. Sörensen & Dr. Ing. F. Schmidt: Recent Development of the MAN Marine Diesel Engine, The Institute of Marine Engineers 1964

27. W. Kilchenmann: 75 Years of SULZER Diesel Engines, Sulzer Technical Review 4/1974

28. G. Wolf: The design and development of the RND engine, The Motor Ship, Special Survey, October 1969

29. Henry Klintorp: Burmeister & Wain engine installations in tankers, B&W 1962

30. Dr. Ing. G. Ciliberto & Dr. Ing. L. Mariani: Considerations sur le projet et les essais de mise au point du moteur Fiat 1060. CIMAC Stockholm 1971

31. G. Wolf: Konstruktion und Entwicklung der neuen Sulzer RND-Motoren, MTZ November 1970, Courtesy of Wärtsilä

32. Dr. Hugo Schobel: Development and operating results of MAN KZ105/180 engine, The Motor Ship, August 1967

33. Carsten Andersen: Development of a Super Large Bore Engine, North East Coast Institution of Engineers and Shipbuilders, 1968

34. The Motor Ship December 1967: Burmeister & Wain's New K98FF Engine

35. J. Schmidt-Sorensen: Development of B&W KGF-Engines

36. The Motor Ship April 1970: New Stork 900 mm-bore Engine Capable of 3.500 bhp/cylinder

37. E. A. Van der Molen: The Development of Stork Engines in 1966, The Motor Ship February 1967

38. L.G. Thulin & G. Dahlbring: Götaverken turbo-charged crosshead diesel engines, GV, 1969

39. Sven Landberg: A Review of Götaverken Diesel Engine Progress, The Motor Ship, Special Survey, November 1964

40. J. Dragsted: Untersuchung von Zylinderdeckeln für Schiffs-Zweitakt-Dieselmotoren, Motortechnische Zeitschrift, November 1970

41. J. Dragsted: Considerations on hydraulic operation of exhaust valves, The Motor Ship, Special Survey, November 1972

42. Dr. Ing. Ruggro Maciotta: Uber den gegenwärtigen Entwicklungsstand der GMT-Dieselmotoren – 5 Jahre nach Beginn ihrer Produktion, Motortechnische Zeitschrift 38, 1977

43. Internal MAN document by K Luther: Die Viertakt- und Zweitaktmotoren der M.A.N.Augsburg 1970 – 1975

44. J.F.Clarke: Power on Land & Sea, Hawthorn Leslie Ltd.

45. J.F. Butler & E-P Crowdy: Development of a high output medium speed two cycle opposed piston engine, CIMAC Washington 1973

46. R. Tsnuneya, The two-stage turbocharged E-type engine, The Motor Ship, special survey , May 1979

47. A. Schiff: Hand-out at a press conference in Tokyo 20. February 1980

48. Ole Grøne & Per Lindhardt: New problems in burning today's residual fuel, B&W 1979.

49. Folker Böhm: Die MAN –Zweitaktmotoren des konstruktionsstandes B/BL, Motortechnische Zeitschrift 39 , 1978

50. R. Häfner & H. Lindner: Strahlgekühlte Waben – ein neues MAN-konstruktionselement für Brennraumwende, MTZ 1977

51. GMT, March 1979

52. A. Østergaard & R Tjønneland: Turbocharging of two-stroke diesel engines, The Motor Ship November 1978, Special survey.

53. A. Oestergaard & R. Tjoenneland: Further development of constant pressure turbocharged, poppet valve type low-speed diesel engines. CIMAC Vienna 1979.

54. Schiff & Hafen/ komandobrücke, Heft 9 1979

55. G. Wolf, A. Bitterli & A. Marti: Bore-cooled Combustion Chamber Components on Sulzer Diesel Engines, Sulzer Technical Review 1/1979.

56. A. Schiff: Hand-out at a press conference in Tokyo 29. May 1979

57. Folker Böhm: MAN's Two-Stroke Programme with particular reference to C/CL long-stroke engines and electronic injection, MAN publication 1979. (Historical Archive of MAN Augsburg)

58. K. Ujita: The constant-pressure turbocharged H-type engine, The Motor Ship, May 1979

59. The Motor Ship July 1980: Mini-bore low speed crosshead engines launched by Mitsubishi.

60. The Motor Ship, Special Supplement, August 1983

61. The Motor Ship, Special Supplement, December 1982

62. Y. Hirayama: Design and development of the superlongstroke L series engines, The Motor Ship, supplement November 1983

63. M. Briner, N. Mikulicic and P. Baumgartner: Sulzer RTA engines for economical ship propulsion, Sulzer paper April 1986

64. A. Østergaard: L-GF exceeding expectations, Schiff & Hafen, Heft 9 1978

65. Protocol of 1997 to amend MARPOL73/78, Annex VI of Marpol 73/78 Regulations for the Prevention of Air Pollution from Ships. International Maritime Organisation 1997

66. Y. Okumura: Comprehensive countermeasures to enhance reliability and durability of marine diesel plant, CIMAC Interlagen 1995

67. K. Wilson, N. Hansen, P Storegaard Jensen & J. Dragsted: Service and operational experience as discussed and compiled by CIMAC WG "Engine Users", CIMAC Copenhagen 1998

68. J. Dragsted, N. Hansen, & K. Wilson: Service experience with diesel engines as compiled by the CIMAC Users WG, CIMAC Hanburg 2001

69. T. Biwa, T. Fukuda, T. Uyama & T. Tanaka: Development of double layered cylinder liner, CIMAC Florence 1991.

70. C-E Egeberg & A. Østergaard: The MC Engine and its future development, CIMAC Hamburg 2001.

71. G.A. Lustgarten & K Aeberli: Sulzer RTA84C – The Modern Containership Engine, Motor Ship Conference London 1990

72. K. Aeberli & G.A. Lustgarten: Improved running performance of low-speed diesel engines, New Sulzer publication 1987

73. K. Aeberli: Results of the application of "High technology research methods" and their transformation into the design of simple and reliable two stroke engines, CIMAC Florence 1991.

74. R. Demmerle: Sulzer RTA84C and RTA96C engines: The reliable driving forces for large, fast containerships, Wartsilä NSD October 1997

75. K Aeberli & M Geist: The Recent Development in the Sulzer RTA Low Speed Engine Programme; Highlights and Results of Operation, CIMAC Copenhagen 1998.

76. R. Demmerle et al.: New insights into the Piston running behaviour of Sulzer large bore Diesel Engines, CIMAC Hamburg 2001

77. H. Sakabe & K. Sakaguchi: The Uec engine program and its latest development, CIMAC Kyoto 2004

78. Y. Hirayama et al: Development of the Mitsubishi UEC75LSII Type Diesel Engine, MHI July 1988.

79. Imakire et al: Development of MET-SR-VG Turbocharger Driven by Radial-Flow Turbine with Variable Geometry Nozzle; CIMAC Tianjin 1989

80. S. Yoshihara et al: The main bearing behaviour of UEC type large bore diesels, CIMAC Interlagen 1995

81. H Brunner: The Sulzer RTA84T engine design, New Sulzer Diesel publication, August 1992

82. K Aeberli & M. Geist: The Recent Development in the Sulzer RTA Low Speed Engine Programme; Highlights and Results of Operation, CIMAC Copenhagen 1998

83. C. E. Egeberg; The MAN B&W two stroke diesel engine and its prospects, CIMAC Copenhagen 1998

84. Markus Geist: Sulzer RTA-8T engines: Compact two strokes for tankers and bulk carriers, Wärtsilä NSD April 1998

85. Half Century of Diesel Engine Progress at Mitsui, Mitsui publication 1976

86. Koichiro Imakire: Historic Development of Exhaust Gas Turbocharger for Low Speed Engine in Japan, National Museum of Nature and Science 2011