MAN B&W Low Speed Large Bore Engines

Now with higher Power Concentratio

Engineering the Future – since 1758.

MAN Diesel & Turbo

Contents

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Introduction

For a number of years our engine programme has been rather stable, with relatively long time between major modifications. This state of affairs has been a result of the hectic market conditions prevailing as well as little need for modifications improving the competitiveness.

For many years, the MAN B&W K80MC-C and MAN B&W K90MC/MC-C engines have been the primary main engine chosen in container vessels of medium to large teu capacity. Nowadays, the largest container vessels are typically powered by the K98 engines, in mechanically or electronically controlled versions.

In the VLCC market, as well as the market for very large bulk carriers, our traditional S80MC has been selected for vessels with limited speed requirements, while for faster and larger vessels the S80MC-C or S90MC-C engines have typically been chosen, the latter also in the electronically controlled version.

A comprehensive updating of our S/K80ME-C and K90ME/ME-C engines is now introduced and designated Mk.9, by boosting the power and adopting a number of the latest design features, also introduced on the S65ME-C engine. Photos of this engine are shown in Figs. 1, 2 and 3. New design features will be applied with reference to the higher rating and a strong emphasis on low production costs.

This paper will describe the updating of our S/K80ME-C and K90ME/ME-C engines from a technical and marketing perspective. Owing to the fact that the K types of these engine types were originally designed as offsprings of the L80MC and L90MC engines back in the 1980s, there is substantial cost down potential in updating as well as uprating the engines, also utilising the design features of today. The design features to be applied are: cast or welded cylinder frame, framebox with triangular stay bolt tube design plates, twin stay bolts and 360° degree thrust bearing, depending on the number of cylinders. It is the intention to launch the updated engines in the electronically controlled ME version only. The main reason for this is the general change over in the market from mechanically to electronically controlled engines.

By now some 353 ME engines have been sold out of which some 114 are in

Fig. 1: Shop test of 7S65ME-C engine at Hitachi in March 2006

Fig. 2: Upper platform on 7S65ME-C

Fig. 3: Welded cylinder frame for a 7S65ME-C

service. The ME engines are characterised by a number of benefits, such as:

Controlled benefits

The ME engine is characterised by:

- Low SFOC and superior performance parameters thanks to variable, electronically controlled timing of fuel injection and exhaust valves at any engine speed and load
- Appropiate fuel injection pressure and rate shaping at any engine speed load
- Flexible emission characteristics with low NO_x and smokeless operation
- Perfect engine balance with equalised thermal load in and between cylinders
- Better acceleration in ahead and astern operation and crash stop situations
- Wider operating margins in terms of speed and power combustions
- Longer time between overhauls
- Very low speed possible even for extended duration and Super Dead Slow operation manoeuvring
- Individually tailored operating modes during operation
- Fully integrated Alpha Cylinder Lubricators, with lower cylinder oil consumption
- The ME engine design is lighter than its mechanical counterpart

Target Market

When the MAN B&W K90MC/MC-C engines were launched in the 1980s, they were targeting the very largest container vessels, at that time with a teu capacity of up to 4,000 teu and a speed of up to 24 kts. The K80MC-C was targeting the step just below, say up to 3,000 teu with the same ship speed.

During the process of designing the K90MC/MC-C engines, many discussions were undertaken, both internally at MAN Diesel as well as externally with hydrodynamic specialists, to define the optimum propeller speed for the targeted container vessels. Thereby, the K90 engines could be tailor-made to match these vessels. The conclusion from the market was that certain ship yards preferred a relatively low propeller speed, slightly above 90 rpm, whereas other ship yards preferred a propeller speed slightly above 100 rpm.

To meet this demand, we made a K90 engine for each preference, hence the K90MC having an MCR output speed of 94 rpm, and the K90MC-C having an MCR speed of 104 rpm. As for the K80MC-C, only the 104 rpm model was made, as a lower-speed model was, and still is, not considered necessary. In the consideration of the updated K80/90 engines also the super long stroke engines, the S80/90MC-C engines, were scrutinised; and we have concluded that the S80ME-C should also be updated.

Mk. 9 engines

	S80ME-C	K80ME-C	K90ME	K90ME-C
Power kW per cyl.	4,510	4,530	5,720	5,730
Speed (rpm)	78	104	94	104
Bore (mm)	800	800	900	900
Stroke (mm)	3,450	2,600	2,870	2,600
Mep (bar)	20	20	20	20
P_{max} (bar)	160	160	160	160
Mps (m/s)	9	9	9	9
Length (7 cyl.) (mm)	12,034	12,034	13,395	13,395

Table I

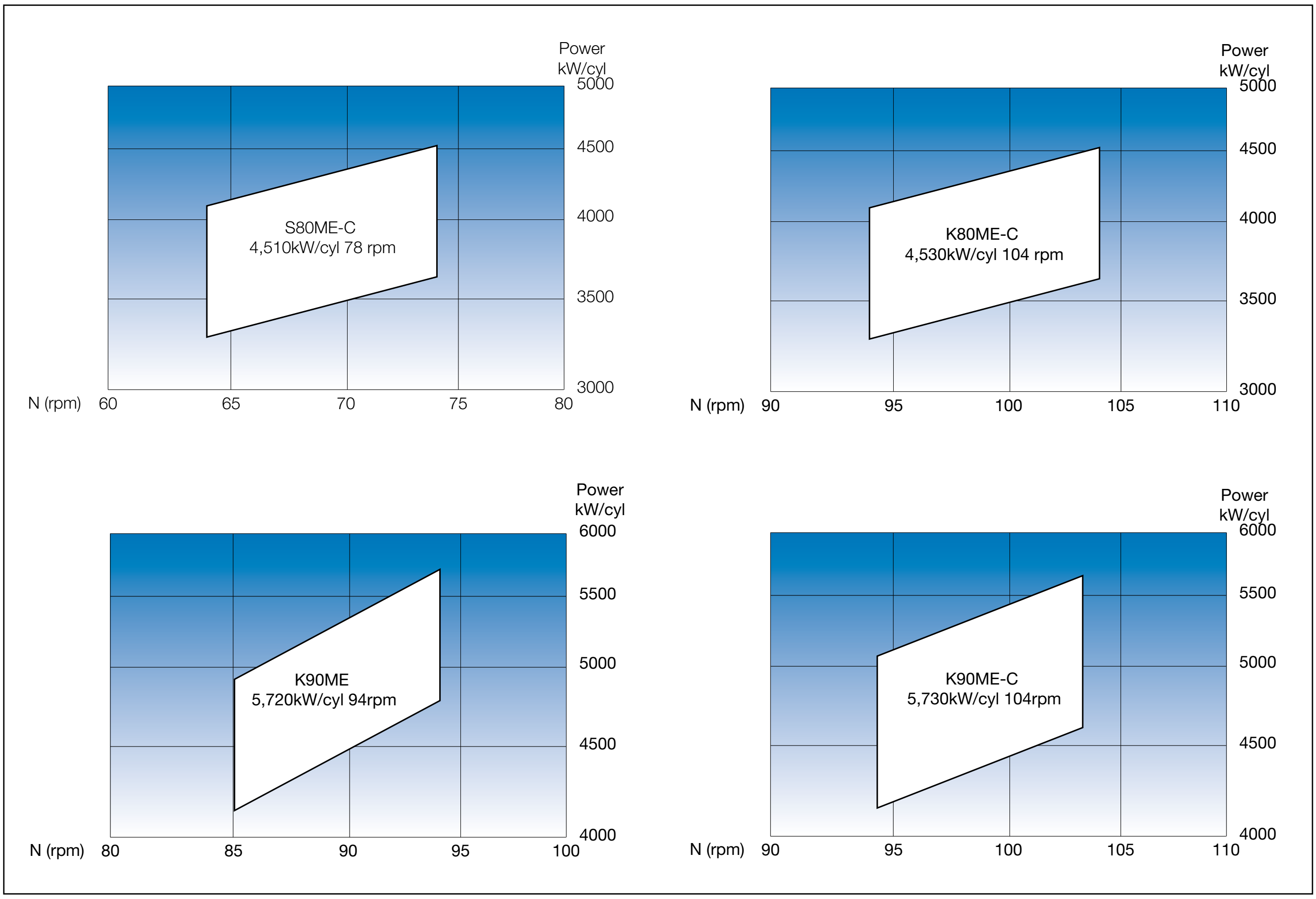


Fig. 4: Layout diagrams for the updated 80 and 90 bore ME/ME-C engines

Performance Data

The data for the modernised K80/90 and S80 engines make them extremely competitive.

The mean effective pressure (MEP) of 20 bar has been chosen and, in order to offer a really attractive specific fuel oil consumption, the maximum firing pressure is 160 bar. These performance data together with a mean piston speed of 9 m/s yield a power per cylinder of 5,720 kW/cyl for the K90ME, 5,730 kW/cyl for the K90ME-C, 4,530 kW/cyl for the K80ME-C and 4,510 kW/cyl for the S80ME-C. The MCR revolution number for the K80ME-C and the K90ME-C will stay at 104 rpm, and for the K90ME the MCR revolution number will stay at 94 rpm, whereas for the S80ME-C the MCR revolution number is 78 rpm.

The combination of mean effective pressure and maximum firing pressure ensures an unchanged specific fuel oil consumption, despite the significant increase in output. In fact, the output of the three modified K type engines is increased by up to 25%.

Table I outlines the situation. It must be noted that the reason why such a large power boost is possible, is that these engines have not been uprated for a very long time.

Layout diagrams for the updated engines are shown in Fig. 4. It will be seen that the increase in power of the K types is so large that they will be able to replace contemporary engines of different design with larger cylinder diameter as well as brand new engines of a different design with slightly larger cylinder diameter.

It has been decided to launch the uprated engines in relevant cylinder numbers only, and this means that the K80/K90 engines are available with 6 to 12 cylinders, whereas the S80ME-C are available with 6 to 9 cylinders.

Design Features

Power is also important, because a higher specific output means that engines with fewer cylinders can be applied, thereby significantly reducing costs. Besides the power aspect, we have selected a number of design features, to cater for the increased performance parameters while at the same time further increase the reliability.

The design features are already known from other engines, thus we are talking about well-proven technology. These are described in the following. There is also one major design change which yields a large weight saving, and this means introduction of differentiated cylinder distances. Typically, it is possible to utilise a shorter cylinder distance in the fore cylinder units because the torque to be transferred through the crank throws is smaller compared to the aft end of the engine. This means for a 12K90ME/ME-C Mark 9 engine that the 6 foremost cylinder units can have a cylinder distance of 1,480 mm compared to 1,588 mm for the 6 aft most cylinder units. The weight saving from utilising this for a 12K90ME/ME-C is more than 25 t, and the reduction in total engine length more than half a meter.

For an 8K90ME/ME-C engine these figures are even more impressive, in that the shorter cylinder distance can be applied on all cylinders, giving rise to a weight saving of more than 34 t and a reduction of total engine length of nearly one meter.

For K80ME-C a similar situation applies.

For the S80ME-C engines, the cylinder numbers in which this engine is offered, do not justify differentiated cylinder distances, but compared with the Mark 8 engine the cylinder distance is reduced from 1,424 mm to 1,334 mm. These figures are valid for both S and K versions. Again, the reductions mean that 7-cylinder engines achieve a length reduction of half a meter, and a weight saving of more than 22 t for the 7K80ME-C.

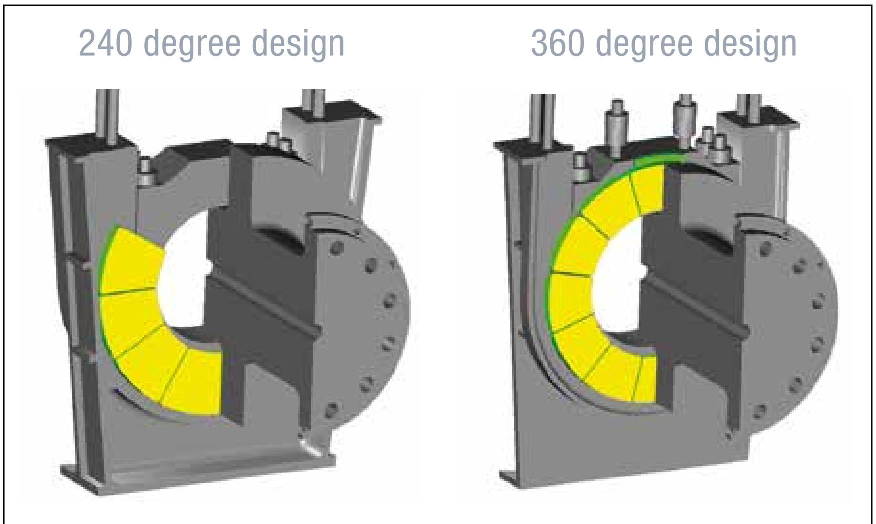


Fig. 5: Thrust bearing design

Bedplate

In order to reduce production costs, no machining of bearing girder side walls is applied after welding.

Main bearings are of the well-proven thin shell design using white metal as bearing material. With a view to reducing production costs, the building-in of the bearing housing is modified in that the 25°/65° mating face between cap and bearing support in the present design, is changed to a horizontal assembly. This simplification necessitates a pair of high friction plates between the cap and bearing support in order for the assembly to be able to handle the shear forces.

For the updated engines with 9–12 cylinders, we are making use of the 360° degree thrust bearing, which gives a weight saving compared to the traditional 240° degree thrust bearing, while at the same time achieving a stronger bedplate structure in the aft end, see Fig. 5. Also the diameter of the thrust cam is reduced, and so is the thrust bearing length. The thrust bearing housing forms a vessel and contains an oil bath which gives additional lubrication of the thrust bearing segments. Further benefits include only one type of segments, and optimal utilisation of the segments according to the magnitude of the thrust at the fore and aft ends by using 12 segments fore (360 deg.) and 8 segments aft (240 deg.). A further advantage with this design is that the thrust acts on the centreline of the crank shaft compared with the traditional design where the force is somewhat offset from the centreline.

For engines with fewer cylinders than mentioned above, the 240-deg. thrust bearing will be maintained to cater for the lower thrust force to be transferred.

In the thrust bearing design, our goal is to benefit from our latest design improvements. A common feature to all of them is that performance and reliability will be improved without increasing the cost.

Twin Staybolts

As it has been our practice for a number of years the bedplate is designed for twin staybolts attached in the top of the bedplate.

This well known design feature is very important to obtain good main bearing performance, as tightening of the staybolts with this design does not give rise to geometrical distortion of the main bearing housing.

Flexible thrust cam

This design, see Fig. 6, has already been introduced on our S65ME-C. The machined grooves in the thrust cam optimise the bearing load in the thrust bearing, which means:

- Reduced oil film pressure whereby the max. bearing pressure is reduced
- Increased oil film thickness
- Improved load distribution on the thrust segments, thereby reducing the area and, thus, the thrust cam dimensions.

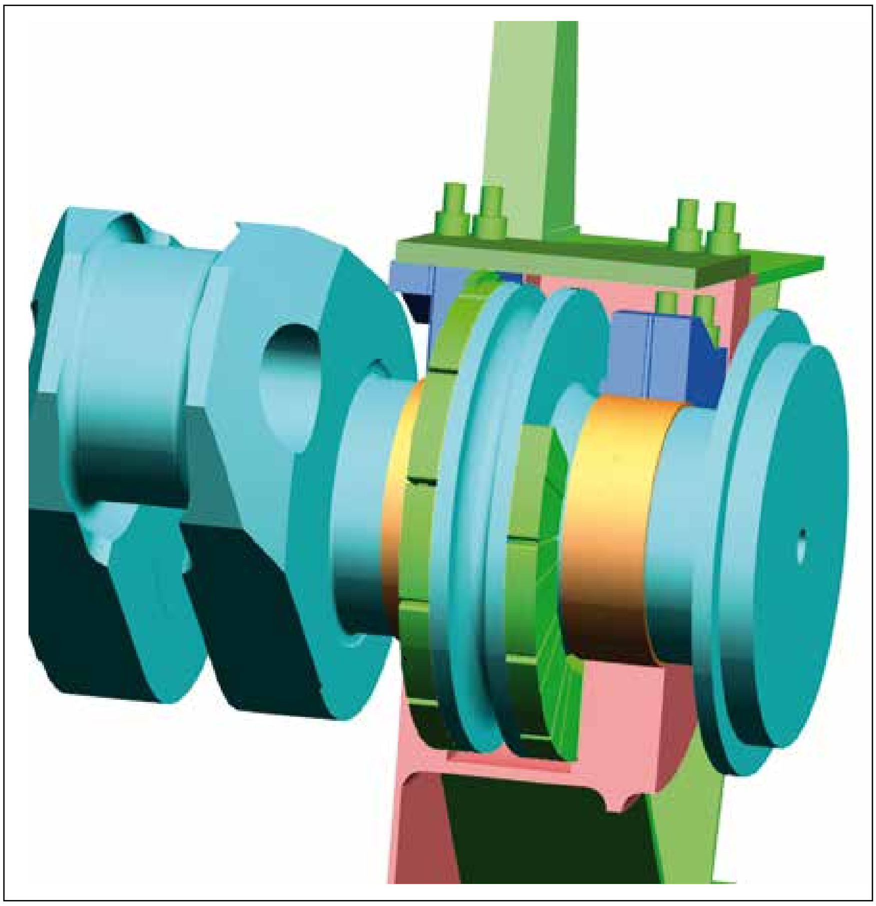


Fig. 6: Flexible thrust bearing cam

Cylinder Frame

As standard, the cylinder frame of the updated engines is of the cast design. A welded cylinder frame is optional, see Fig. 7. This was decided in order to reflect the preference of the licensees as experienced over the later years. We see a number of inherent advantages in the welded design, such as increased rigidity as well as reduced weight. Furthermore, the welded design makes it possible to integrate the scavenge air receiver in the cylinder frame, see Fig. 8, giving rise to a total weight saving of 30%. In addition to this important feature, dimensions are smaller.

The reason why the welded design is made optional reflects the general preference among engine builders and this is because the foundry capacity in general has been increased, thus the foundry capacity shortage recently experienced is generally no longer a problem. The welded cylinder frame will also be available with bolted on scavenge air receiver as an option in order not to take up too much time on large plano millers. Parts are interchangeable between the welded and cast cylinder frame design.

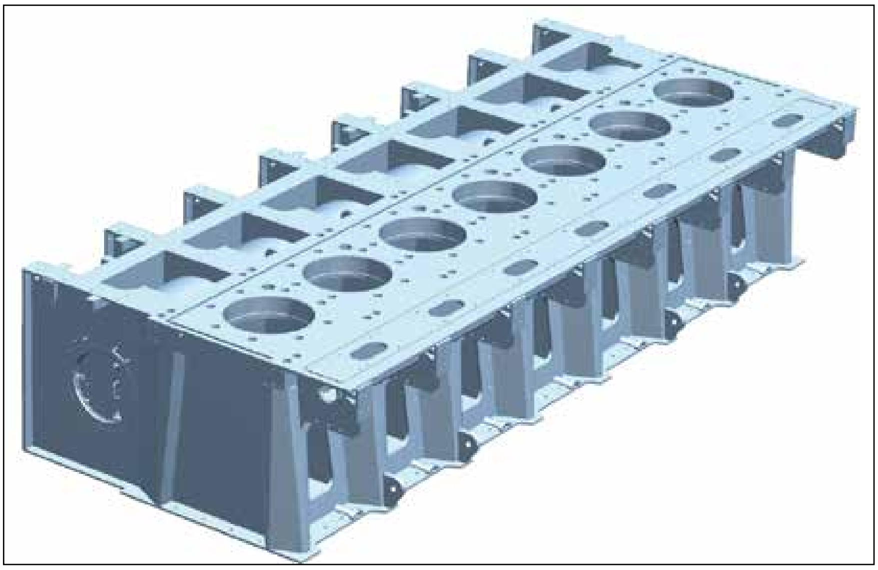
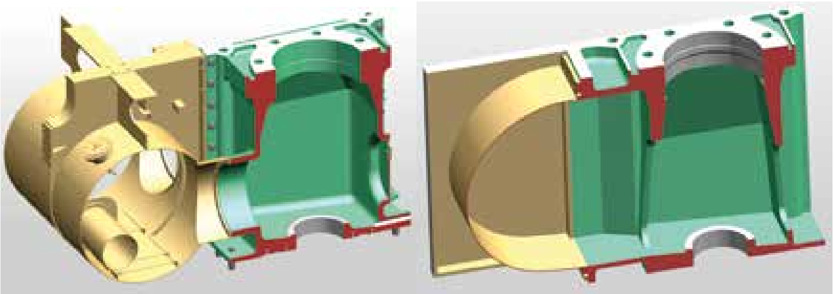
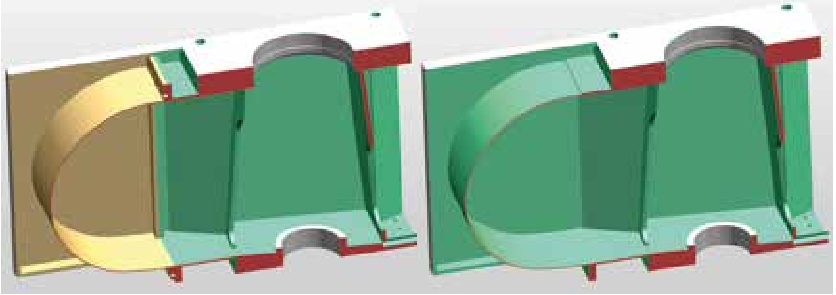


Fig. 7: Welded cylinder frame



Conventional cast cylinder frame design with bolted on scavenge air receiver

New Cast cyl. frame design with bolted on D-shape scavenge air receiver



New welded cyl. frame design with bolted on D-shape scavenge air receiver. Same D-shape scavenge air receiver for cast and weld design

Welded cyl. Frame design with integrated D-shape scavenge air receiver

Fig. 8: Cylinder frame design

Framebox

Like other newer engines among the ME/ME-C/MC-C types, the updated engines have a triangular plate framebox with twin stay bolts, see Figs. 9 and 10.

During the final development of the design for the S90MC-C, the K98MC/MC-C and all other S-MC-C engines, we formulated the following main design criteria:

1. The design should match or be superior to any existing MAN Diesel design in terms of functionality, both the MC stay bolt tube design as well as the twin stay bolt rib design.
2. Production costs should be reduced.
3. There should be an uprating potential for the future, without losing production benefits.
4. It should be possible to introduce a design prepared for omission of PW-HT (Post Weld - Heat Treatment).

Basically, the design is like the S-MC-C framebox with ribs. The main difference is the substitution of horizontal ribs with vertical plates, forming a continuing triangle profile from top to bottom. In this way, we obtain an even stiffness along the guide bar, which improves the conditions for the guide shoes' sliding surface. Contact simulations have verified this improvement, and service experience has also shown an excellent bearing condition.

In general, production costs are reduced – especially welding costs. With the lower stress level in the structure, we could use weld joints with reduced requirements to the quality, ideally suited for welding robot techniques.

After large-scale laboratory testing and two years of service with the 'triangular plate' design in our large bore engines, S90MC-C and K98MC/MC-C, we initiated tests on omission of PW-HT.

The decision was based on positive results from the laboratory test and service inspections of the guide shoe slide surface conditions as well as general inspections of the framebox.

Two frameboxes for the 6S90MC-C were selected for service test in agreement with the owner and classification society involved. On one framebox, PW-HT was made as specified, on the other PW-HT was omitted. Dimension measurements were taken at selected production stages from 'as welded' to 'after shop trial'. Furthermore, measurements of the guide bar geometry and inspection of guide shoe conditions were made after approximately 5,000 service hours. These measurements all encouraged the omission of PW-HT.

Based on the above positive test results, we have decided to omit the specification of PW-HT for frameboxes with triangular plate guide bar design.

In essence the benefits of the design are:

- Uniform higher rigidity
- Lower stress level
- Easier production

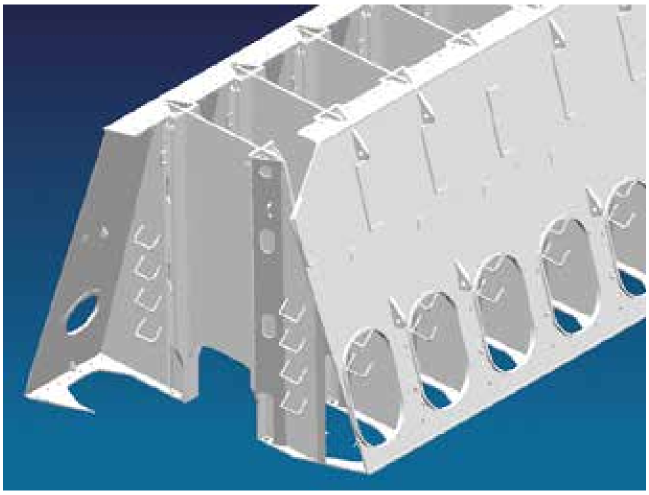


Fig. 9: Triangular plate design of frame box

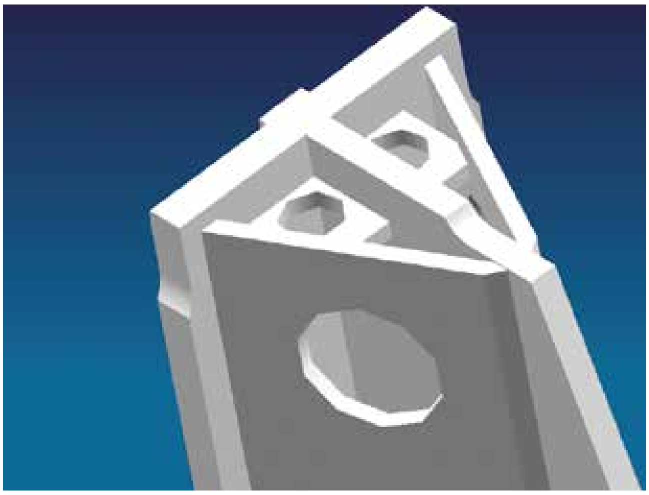


Fig. 10: Triangular profile

Combustion Chamber

Other than the above-mentioned design features the S/K80ME-C and K90ME/ME-C engines will of course have Oros combustion chamber. Our large bore engines have utilised the Oros combustion chamber since the late 1990s, and this design is now also used on medium bore engines. In terms of cylinder condition, the Oros design has been very successful, actually to an extent where we have found it possible to extend the overhauling intervals for combustion chamber related parts. This was communicated in a service letter in 2003 (SL03-423).

The backbone of the Oros design is the piston and cylinder cover geometry, which concentrates the hot combustion gases around the fuel nozzles, and the high top land piston, which allows a low position of the mating surface between cylinder liner and cylinder cover, thereby reducing liner wall temperatures and giving controlled cold corrosion, which is good for refreshing the liner surface.

Besides these basic Oros design features, the piston ring pack is of the latest design. This means a change of the ring pack, as shown in Table II, which gives a schematic comparison between the former and the present standard for K type engines.

The change in design was recently introduced to increase the resistance against scuffing. Hard coating to the first and the fourth piston rings on K90 and K98 has been introduced, and the position of the CL grooves (controlled leakage grooves) has also been changed to secure a more even heat distribution. The CL groove depth has been reduced by 20%.

Earlier standard for large bore engines in general

Piston ring	Coating Liner running side	Coating Ring groove running side	Lock Type	Ring material	CL-groove depth	CL-groove Position from lock	Running surface profile
1	Alu-coat	Chrome	GT	CV1	5.0 – 5.5	30° & 90° *)	Straight
2	Alu-coat	Chrome	Oblique L	CF5	- - -	- - -	Straight
3	Alu-coat	- - -	Oblique R	CF5	- - -	- - -	Straight
4	Alu-coat	- - -	Oblique L	CF5	- - -	- - -	Straight

Present standard for large bore engines in general

Piston ring	Coating Liner running side	Coating Ring groove running side	Lock Type	Ring material	CL-groove depth	CL-groove Position from lock z	Running surface profile
1	Cermet Coating + running in layer	Chrome	GT	CV1	4.0 – 4.5	30° & 90° *)	Taper
2	Alu-coat	Chrome	Oblique L	CF5	- - -	- - -	Straight
3	Alu-coat	- - -	Oblique R	CF5	- - -	- - -	Straight
4	Cermet Coating + running in layer	- - -	Oblique L	CF5	- - -	- - -	Taper

Table II

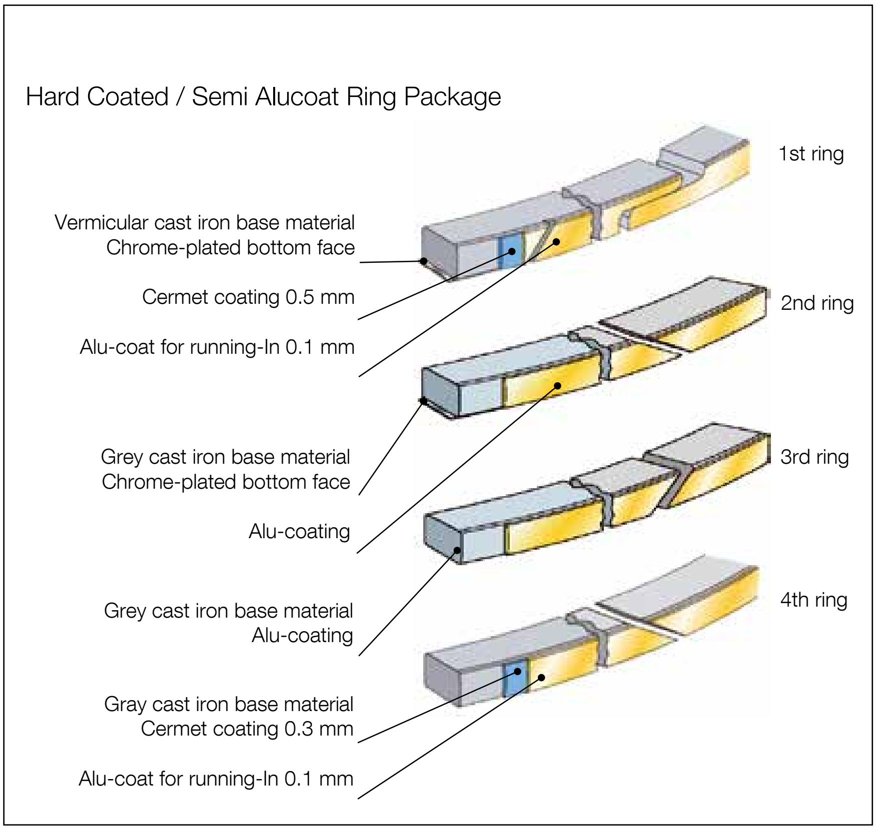


Fig. 11: New piston ring pack

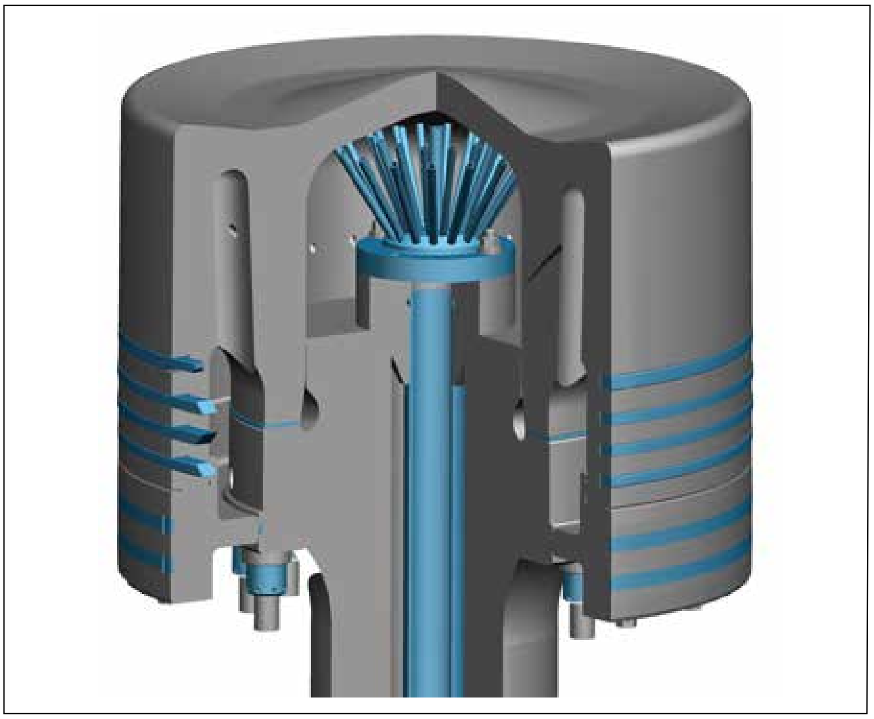


Fig. 12: Piston cooling insert

The hard coating introduced is a Cermet (ceramic-metal) hard coating with a 0.1 mm Alu-coat running-in layer

The Cermet coating consists of 50% chrome carbides, in a matrix of NiCr with some molybdenum, see Fig. 11.

Since the hard coating takes longer time to adapt to the liner surface, there is a higher risk of getting a poor gas seal during the running-in. To minimise this risk, the running surface of the piston ring is chamfered on approximately half of the surface.

A well-running ring has a ballistic shape, starting at the top of the ring and ending at the bottom of the ring; the chamfering is therefore to help the ring to run-in fast and safe as well as to obtain a good gas sealing.

PC rings are also applied to avoid the building-up of carbon deposits on the piston top land. Such deposits could otherwise harm the cylinder condition by scraping off the cylinder lube oil on the liner walls.

The result of the combustion chamber design is low wear rates at low cylinder oil feed rates, long time between overhaul and good scuffing resistance.

Piston cooling insert

In order to obtain a comfortably low piston temperature in service, the so-called spray cooling insert, see Fig. 12, is applied.

To further protect the piston, inconel is applied on the piston top.

The improved cooling has been confirmed by piston crown temperature measurements taken on test bed at several occasions as well as tested on engines in service. Comparative temperature measurements with and without spray cooling insert have revealed a temperature reduction at the central part of the crown, both on the combustion chamber side and on the cooling side, i.e. the spray cooling insert increases both the margin against hot corrosion and underside coke formation.

Exhaust Valve

The exhaust valve unit is of a modified type, implying the use of a low grade cast iron housing with water cooling around the spindle guide and bore cooled twin stud bosses. The upper part of the housing forms the bottom of the air spring and the lower part forms part of the cooling chamber in the cylinder cover. The exhaust valve is shown in Fig. 13.

Two engines with this design of exhaust valve housing have been in service for more than 13,000 hours with excellent results, and besides eliminating cold corrosion of the exhaust valve duct, the lower cooling intensity slightly increases the exhaust gas energy, thus possibilities for waste heat recovery are improved.

A W-seat of low height is used and a high spindle guide lubricated by the controlled oil level system (COL), presently being introduced. This system implies a minimum oil level inside the air cylinder, which is secured by a high outlet (overflow) to the safety valve. This ensures a high oil level, high enough to have oil on the top of the sealing ring.

Furthermore, the safety valve adjustment corresponds to this high oil level. The oil to the top of the sealing ring is passing through three bores in the flange, see Fig. 14.

This design neither requires a sealing air arrangement nor a special type of sealing ring, this means that the old type of sealing ring can be used regardless of the type of valve spindle.

Fig. 13: Exhaust valve

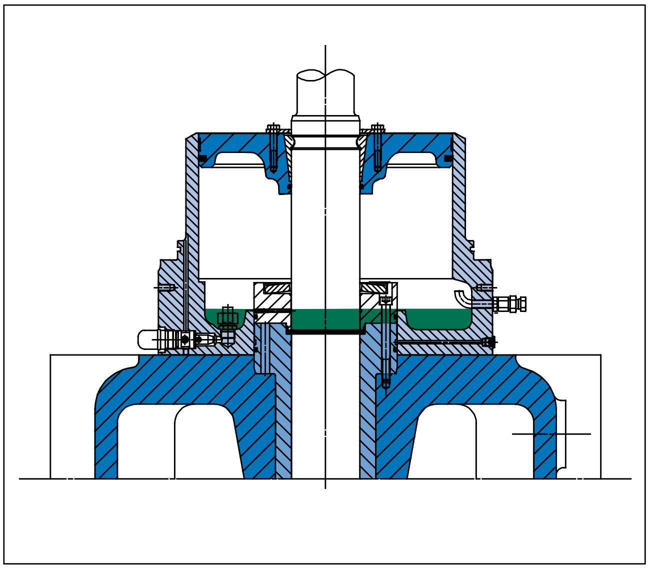


Fig. 14: Controlled oil level (COL)

Design Improvements on the Exhaust Side

During the recent years, the scavenge air receiver and the exhaust gas receiver have undergone a series of design improvements. An increased focus on welding details, smoother design and rigidity have made this possible. Modifications have been carried out on the auxiliary blower inlet, scavenge air cooler housing, turbocharger foundation, scavenge air receiver and exhaust gas receiver.

The improved rigidity has reduced the vibration level on the turbocharger, scavenge air cooler housing and exhaust gas receiver. The safety factor against cracks in the related parts has been increased while at the same time achieving a simplification. An example of a simplification is the reduction of exhaust receiver bottles on the K98ME/C engines, see Fig. 15.

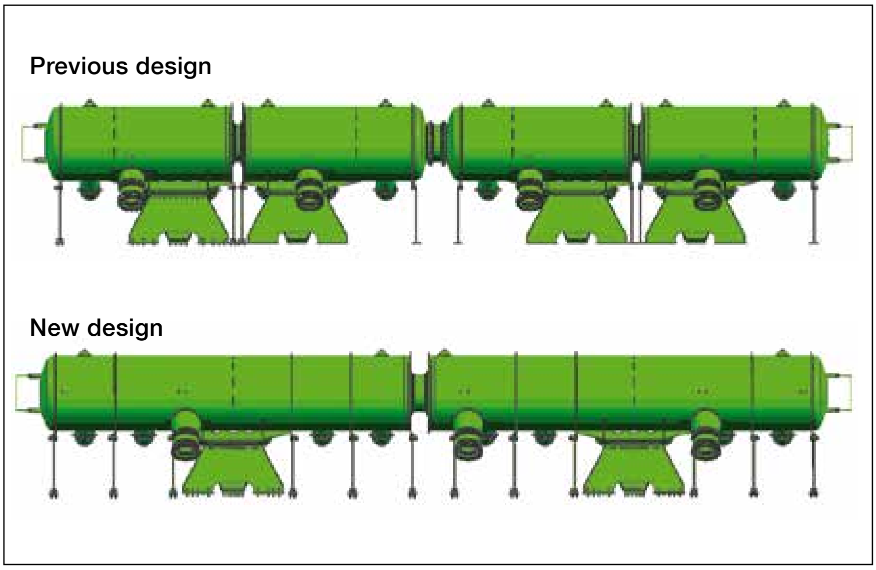


Fig. 15: New exhaust receiver design

Water mist catcher

Over the years, we have seen cases of breakdown of water mist catchers due to poor mechanical quality and poor welding quality. Also cases of inadequate installation with poor sealing, leaving areas open for water mist to bypass the water mist catcher elements have been seen, and in addition wrong profiles have been applied in a few cases. In certain cases, the mentioned deficiencies gave rise to poor cylinder condition, i.e. scuffing. To change this situation, we decided recently to improve the quality by upgrading the design. This implies a more rigid frame to ensure structural integrity, optimising the lamella profiles so as to further improve the water mist catching efficiency, and improve the sealing around the water mist catcher to avoid by-pass of water mist.

These design features relate to all engines, not only the large bore engines descriped in this paper.

ME Technology

Serveral modifications have been carried out, and they reflect partly the desire for cost down as well as the experience gained from engines in service. Besides modifications to electronics (not described in this paper), modifications to hydraulic power supply systems as well as hydraulic piping systems have been carried out.

Servo oil supply

There are two standards for hydraulic servo oil supply also on the updated engines:

- Engine-driven oil supply
- Electrically driven oil supply.

Engine-driven oil supply

The well-proven design with a chain-driven gearbox, which drives three to five high-pressure oil pumps, has also been chosen for the new engines. Especially on large bore engines, the chain is preferred in the first step of the step-up gear, mainly because of its easy installation and high tolerance against relative movements of the crankshaft. Furthermore, the chain is used for driving a second-order moment compensator on six-cylinder engines ordered with such compensators.

Compared with the initial ME/ME-C large bore design, see Fig. 16, with a large combined accumulator and safety block positioned on the gearbox, the new design has been simplified. It consists of small accumulator blocks, one for each pump, and a safety block positioned on the aftmost hydraulic cylinder unit, see Fig. 17. The accumulator is a part of the pump regulating system, and the position on the pump is thus furthermore the most efficient position for the accumulators.

The large double-walled supply pipe with its very large forged connection pipes has been replaced by standardised high-pressure pipes. In order to ease installation and avoid vibrations from the pumps, high-pressure hoses are installed between the pumps and the high-pressure pipes. With regard to assembly, the new design is simpler compared to the double-walled pipe, which requires manual erection on test bed.

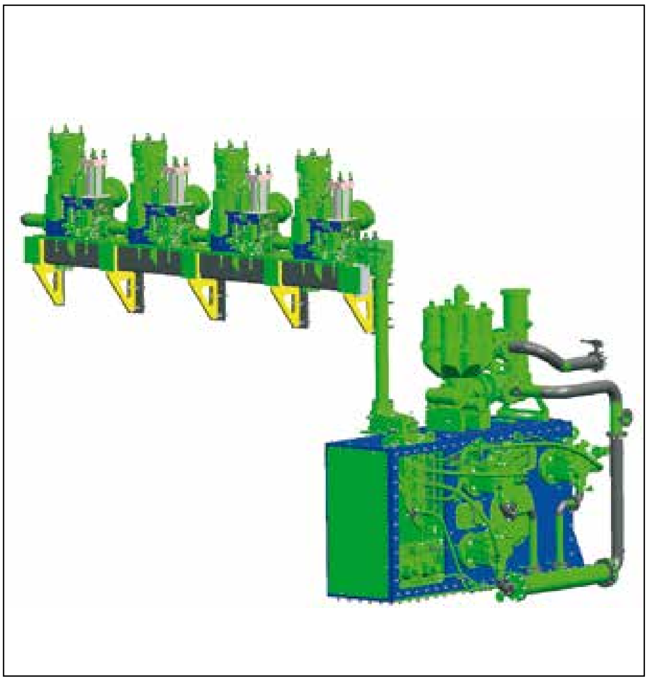


Fig. 16: Previous design with safety and accumulator block and startup pumps in the gearbox and double-walled pipes

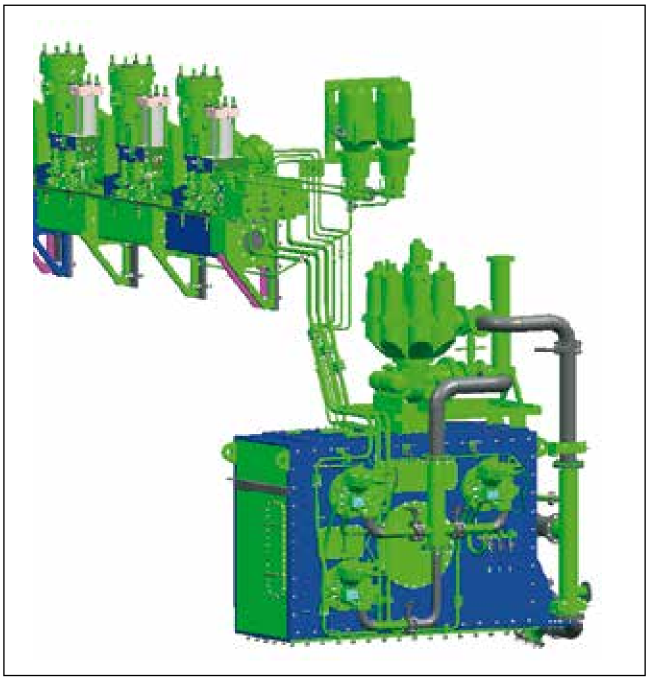


Fig. 17: New design with standardised high-pressure pipes, small accululator blocks, one for each pump, and a safety block positioned on the aft most hydraulic cylinder unit

The start-up pumps are positioned below the upper gallery, thus ensuring good accessibility to the pumps.

The production costs for the new accumulator and safety block design have been reduced, mainly because the mass has been reduced. Furthermore, the safety and control valves have been reduced in size.

The high-pressure pipes and hoses which have replaced the double-walled pipe design with its special dynamic sealing, are naturally a cost down.

On the pictures, the fine filter is positioned on the engine. However, based on yard, shipowner and/or licensee requirements it can be positioned on the hull side as an alternative standard.

Electrically driven oil supply

If electrically driven oil supply is preferred, the complete gearbox and chain drive are replaced by electrically powered high-pressure pumps. The pumps are similar to the mechanically driven pumps, and the pump principle is the same, see Fig. 18.

The chain drive section of the framebox is omitted, and the engine is thus structurally simpler compared with the mechanically driven servo oil supply type engine.

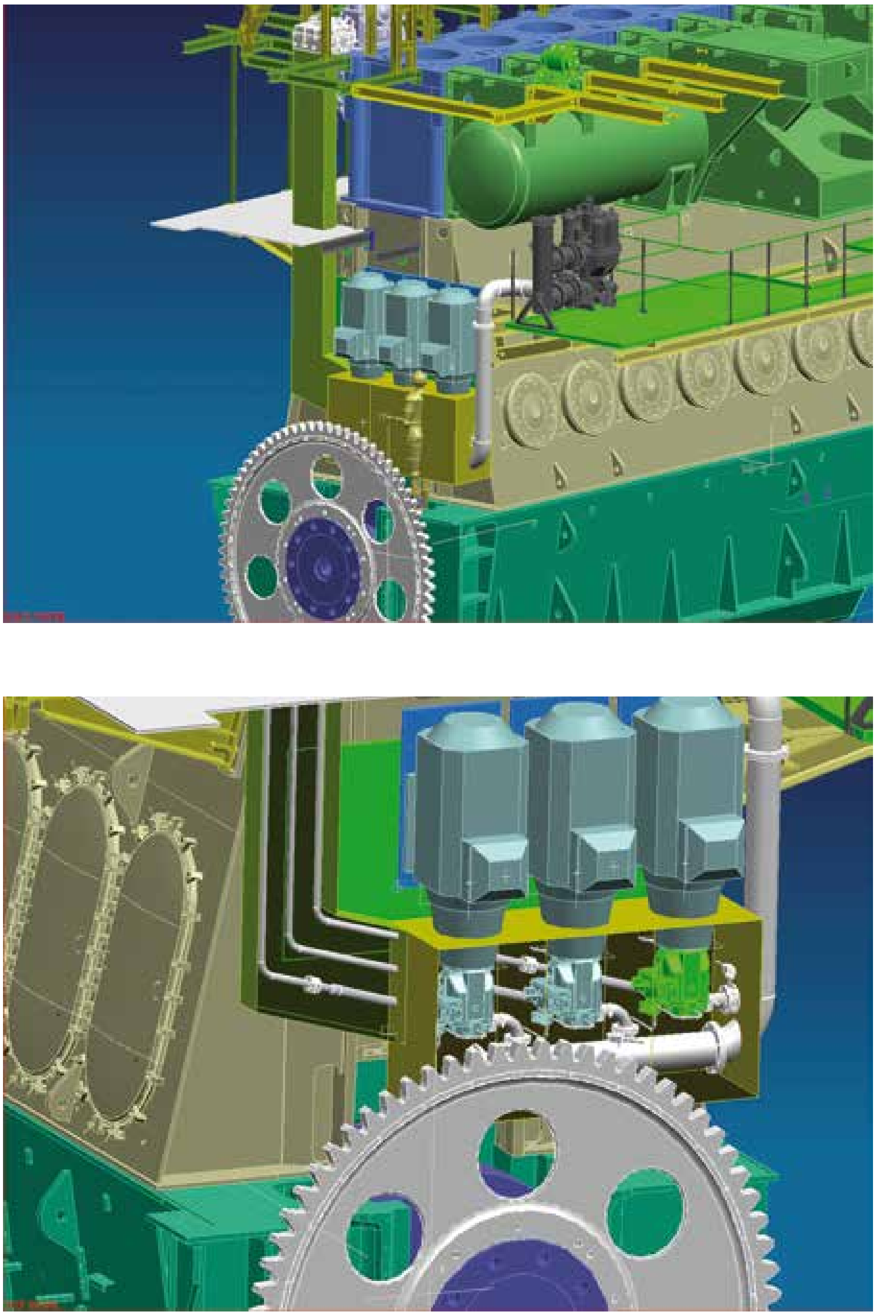


Fig. 18: Arrangement of electrically driven pumps and filter

In order to reduce the length of the high pressure pipe lines between the electrically driven pumps and the hydraulic cylinder units, the pumps are placed on the aft end of the framebox, depending on engine type.

Alternatively the pumps can be located in the engine room.

With the objective of maintaining the high reliability of the ME/ME-C oil loop, the same safety block is used as on the engine with mechanically driven oil supply. Also on engines with electrically driven oil supply, the filter, which is of the same size on both types, can be arranged on either engine or hull side.

Concluding Remarks

The modernised MAN B&W S/K80ME-C and K90ME/ME-C engines are extremely competitive, both with a view to production costs and cost per kW. At the same time, modern and yet well-proven design features are applied with the object of facilitating the production as well as ensuring a high reliability and long service intervals.

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