Service Experience 2007, MAN B&W Engines
The ME/ME-C and MC/MC-C Series

Contents:

MAN Diesel A/S • Copenhagen, Denmark

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Introduction

The introduction of the electronically controlled camshaft-less low speed diesel engines is proceeding rapidly, with many ME engines ordered and, consequently, many ME engines entering service. At the time of writing, more than 300 ME engines are on order or have been delivered. This number proves the market's acceptance of this technology. Of the ME engines, 82 are in service as of January 2007, and they range from the L42ME engine up to the K98ME/ ME-C engines.

Fig. 1a : Prototype 7S65ME-C, is installed in the tanker M/T Ice Explorer, October 2006

Fig. 1b: Prototype 7S65ME-C on testbed

The vessel *M/T Ice Explorer*, see Fig. 1a, is equipped with the 7S65ME-C prototype engine, Fig. 1b. The S65ME-C engine is a "pre-runner" of the S/K80ME-C and K90ME/ME-C Mk 9 engines.

Although the ME technology may seem brand new to many in the industry, MAN Diesel has been devoted to the development of electronically controlled low speed diesels for a long time, actually since the early 1990s. The first engine featuring the ME technology was a 6L60MC/ME, the name indicating that it was originally built as a conventional MC engine with camshaft, and then later rebuilt into an ME engine. The ME version of this engine has now logged about 30,000 running hours and it has, throughout this period, been used to fine tune the ME technology.

The main objectives for the ME technology are:

1. Improved fuel economy at all load points, Fig. 2

2. Flexibility with respect to present as well as future emission requirements Fig. 3

3. Easy engine balancing/adjustability, Fig. 4

4. System integration (Fig. 5: Alpha Lubricator fully integrated in the ME system)

5. Smokeless operation

6. Stable running at very low load.

All these objectives have been accomplished to a very satisfactory level on the first ME engines in service, Fig. 6.

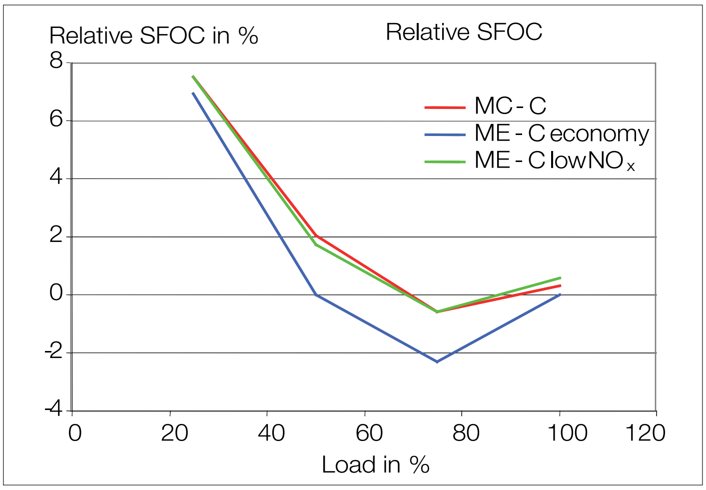


Fig. 2: Improved fuel economy at all load points

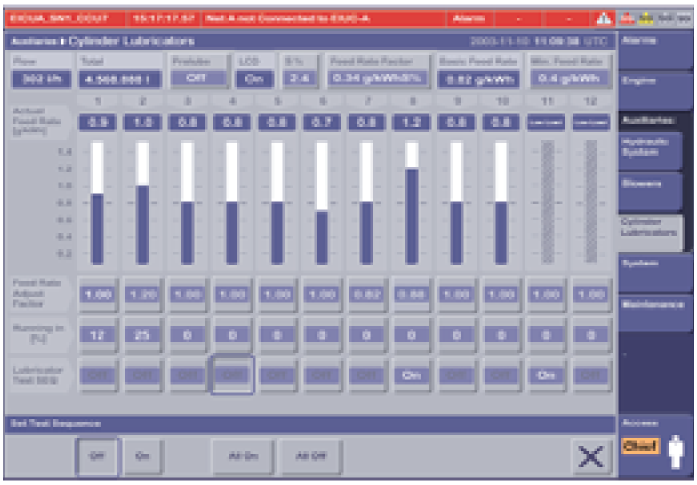


Fig. 5a: System integration, example: Alpha Lubricator

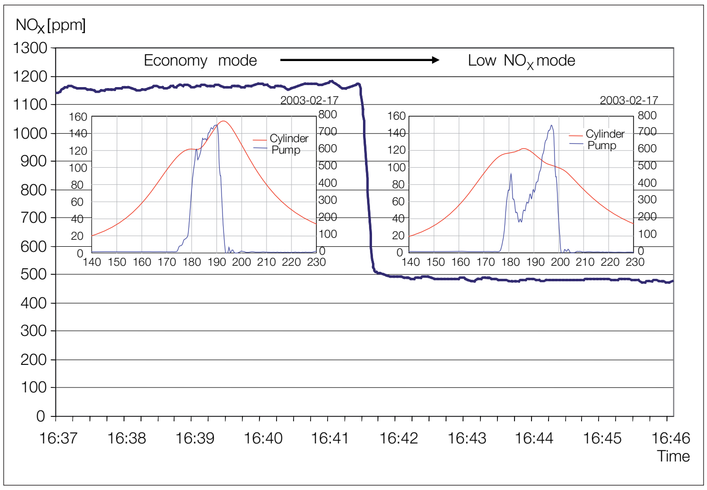


Fig. 3: Flexibility with respect to emission

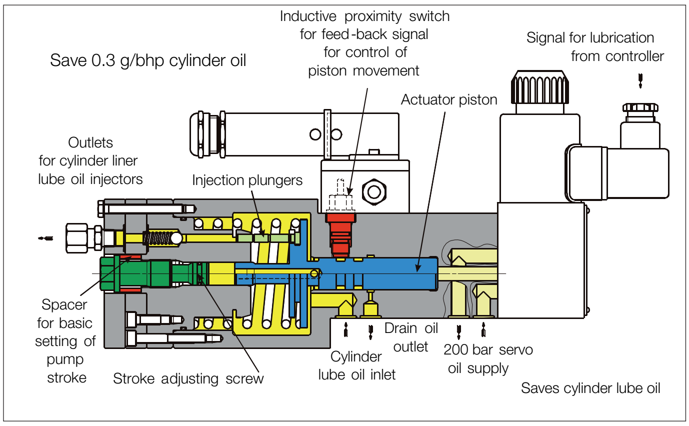


Fig. 5b: System integration, example: Alpha Lubricator

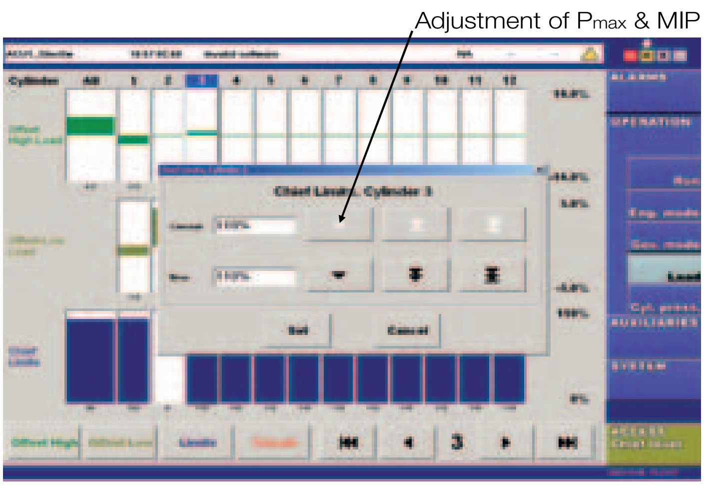


Fig. 4: Easy engine balancing/adjustability

Type	In Service
K98ME	11
K98ME-C	11
S90ME-C	2
K80ME-C	1
S70ME-C	16
L70ME-C	5
S65ME-C	3
S60ME-C	23
S50ME-C	9
L42ME/MC	1
Total	82

Fig. 6: ME/ME-C list of references, January 2007

Future updates and fine tuning with new releases of the control software further enhance the advantages of the ME concept.

Other ME features realised are:

- Real camless fuel injection with either engine driven or electrically driven standard industrial pumps for the hydraulic power supply
- Standard flexibility with respect to changing between HFO and MDO operation
- Simplicity is achieved with a low number of components (e.g. only one control valve per cylinder)
- The number of assembly points is kept low by having only one high pressure oil system
- Cylinder control computers are located away from areas with a high heat exposure in order to limit thermal heating

Apart from the ME specific features, other mechanical designs such as the Oros combustion chamber with slide fuel valves, Nimonic exhaust valves with W-seat will secure long times between overhaul and a very satisfactory cylinder condition.

This paper will describe the service experience obtained with the commercial ME and ME-C engines in service.

For the MC/MC-C engine series, the feedback from service has over the last 4-5 years resulted in an extension of the Time Between Overhauls (TBO). We have not yet fully experienced the benefits of this development. However, the latest feedback from service indicates that five years between major overhauls are looking to become realistic. It will be discussed how this development can benefit different operators. Also the development in relation to the cylinder condition, with focus on cylinder oil consumption, will be touched on.

The ME/ME-C Engine Series

The ME concept

The ME engine concept consists of a servo-hydraulic system for activation of the fuel oil injection and the exhaust valves. The actuators are electronically controlled by a number of control units forming the ‘Engine Control System’, see Fig. 7.

Fuel injection is accomplished by pressure boosters, which are mechanically simpler than the fuel pumps on conventional MC engines. The fuel plunger on the ME engine is driven by a piston actuated with pressurised control oil from an electronically controlled proportional valve as the power source. Also the exhaust valve is opened hydraulically, and closed by an air spring as on the MC engine. Similar to the fuel injection pressure booster, the electronically controlled exhaust valve actuator is driven by the pressurised control oil which, for the exhaust valve, is fed through an on/off type control valve or a proportional type control valve.
In the hydraulic loop, see Fig. 8, lubricating oil is used as the medium. It is filtered through a fine filter and pressurised by a hydraulic power supply unit mounted on the engine. A separate hydraulic oil system is optional. Furthermore, separate electrically driven main pumps are optional.

From the hydraulic power supply unit, the generated servo oil is fed through shielded pipes to the hydraulic cylinder units, see Fig. 9. There is one such unit per cylinder. Each unit consists of a fuel oil pressure booster and an exhaust valve actuator. A Fuel Injection and Valve Actuation (FIVA) control valve is mounted on the HCU. On early ME engines, ELectronic valve Fuel Injection (ELFI) and ELectronic Valve Actuation (ELVA) control valves are mounted on the HCU. Also the Alpha Lubricator is mounted on the HCU.

It should be realised that even though an ME engine is simple to operate, training of crews in the ME technology is important to ease the understanding

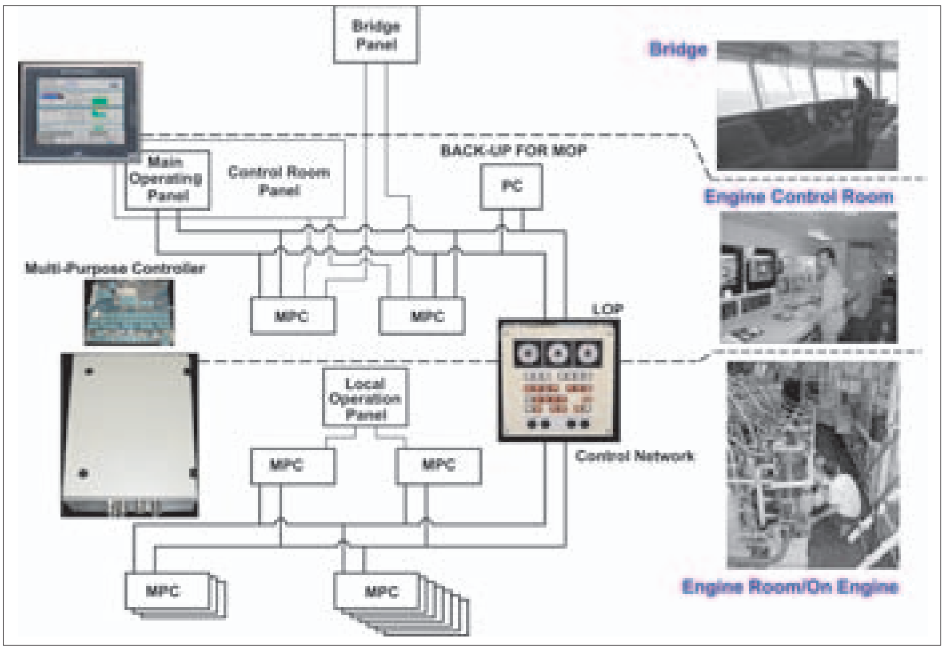


Fig. 7: Engine Control System (ECS)

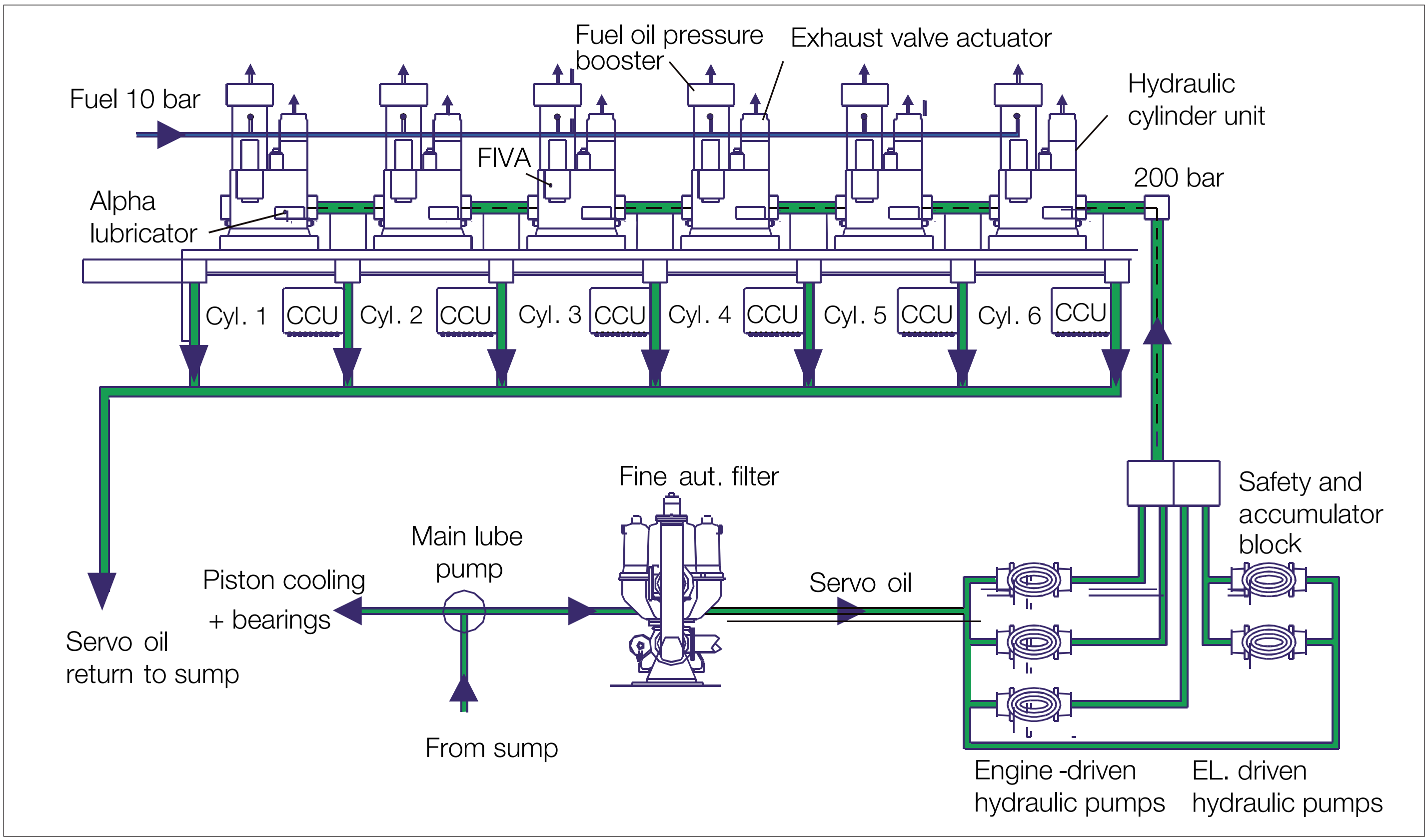


Fig. 8: Hydraulic loop of the ME engines

and avoid any confusion and anxiety that could otherwise occur. To facilitate this, MAN Diesel has set up an ME training centre including a complete ME simulator, so that crews can get hands-on training at the MAN Diesel works in Copenhagen. Mobile versions of such ME simulators have been built in order to be able to train crews and customers at other locations than Copenhagen. Presently, three such portable simulators, Fig. 10, are in operation in China, Korea and Japan.

In general, operators have reported, and thus confirmed, the expected benefits of the ME technology, such as lower FOC (Fuel Oil Consumption), better balance between cylinders, better acceleration characteristics and improved dead-slow performance. Also, the detailed monitoring and diagnostics of the ME engine provide easier

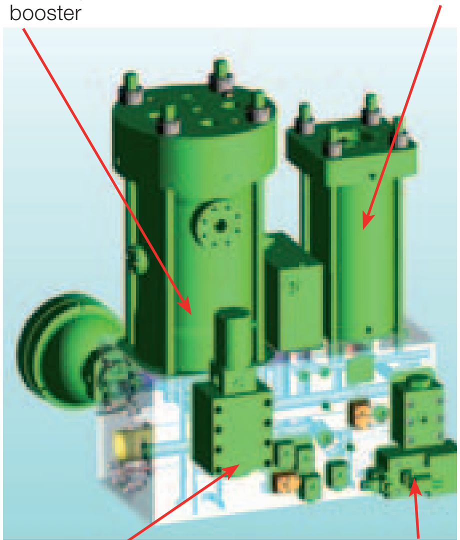


Fig. 9: Hydraulic Cylinder Unit (HCU)

operation and longer times between overhauls, and indeed the ME technology makes it much easier to adjust the Mean Indicated Pressure (MIP) and $p_{max.}$ This is carried out via the Main Operating Panel (MOP) in the control room, see. Fig. 7.

Operators have also found that when operating in rough weather, there is less fluctuation in engine rpm compared to an engine with camshaft-driven fuel injection. Importantly, owners of ME engines in service for a longer period of time report savings in fuel oil consumption in the range of up to 4%, when comparing with a series of sister vessels having the camshaft equipped counterpart type of engines. Apart from the inherent better part-load fuel oil consumption of an ME engine, one reason for the reported improved fuel consumption figures is that the ME engine makes it very easy to always maintain correct performance parameters.

Fig. 11 demonstrates the cylinder condition on the first 12K98ME after 10,762 running hours, and the condition is perfect, as illustrated by the very clean condition of the ring package. As regards the cylinder condition in particular, observations so far indicate that we can expect an improved cylinder condition in general, probably owing to the fact that the fuel injection at low load is significantly improved, compared to conventional engines.

Fig. 10: One of three portable ME simulators in operation

ELFI valves

On the Print Circuit Board (PCB) components have come loose due to vibrations. Fig. 12 shows a DC-DC converter which has come off. Improvements by means of resilient mountings have been introduced on all vessels in service with ELFI valves, and performance has been good hereafter, see Fig. 13.

ELVA valves

We have, at an early stage, seen the same or similar vibration related issues for the ELVA valve PCB as described above for the ELFI valve. These issues have been fully clarified.

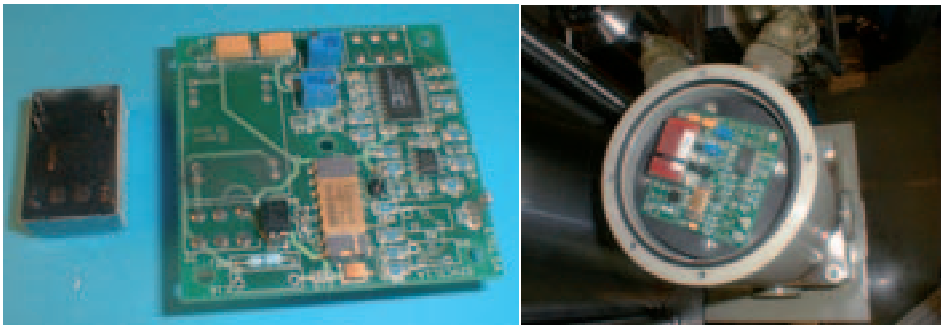


Fig. 12 : ELFI valve, DC-DC converter coming off due to vibration of PCB

Fig. 11: 12K98ME prototype engine, piston inspected after 10,762 running hours

Hydraulic cylinder unit

The hydraulic cylinder unit (HCU), of which there is one per cylinder, consists of a hydraulic oil distributor block with pressure accumulators, an exhaust valve actuator with ELVA control valve and a fuel oil pressure booster with ELFI control valve. Each individual HCU is interconnected by shielded piping leading the hydraulic oil. After delivery of the first 20 ME engines, the ELVA and ELFI valves were substituted by one common FIVA valve controlling both the exhaust valve actuation and the fuel oil injection.

Early service experience proved that low ambient temperatures, as often experienced during shop tests in the winter season, gave rise to sticking high-response valve spools in the ELVA valve due to low hydraulic oil temperatures. The diameter of the spool was reduced in order to obtain correct functioning of the high-response valve as shown in Fig. 14.

However, a production quality problem with too small clearances between the high-response valve spool and the housing has been encountered in service. This has led to a sticking spool, Fig. 15. In such a case the control system encounters improper lifting of the exhaust valve and, as a consequence,

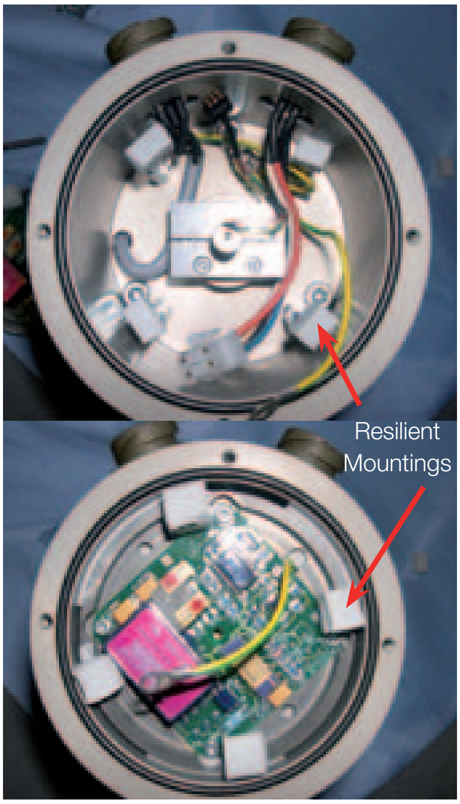


Fig. 13: ELFI valve, PCB secured by resilient mountings

fuel is stopped to the unit in question. Engine running with one cylinder misfiring is then experienced. A change of ELVA valves to 100% quality controlled units has taken place.

Apart from mechanical modifications (including the vibration related issues), we have continued to investigate reasons for premature failures of the ELVA control valve.

In a number of cases, the connector between the ELVA electronic and the high-response valve has failed. Fig. 16 shows a faulty connector. Furthermore, we are testing a new high-response valve in which a larger force can be applied to move the pilot spool. In cooperation with the sub-supplier, we will decide what needs to be done to achieve satisfactory reliability of the ELVA valves.

For the 20 vessels equipped with ELVA/ELFI control valves, an exchange service will have to be arranged. This demonstrates MAN Diesel's commitment to update products, even products that are no longer produced.

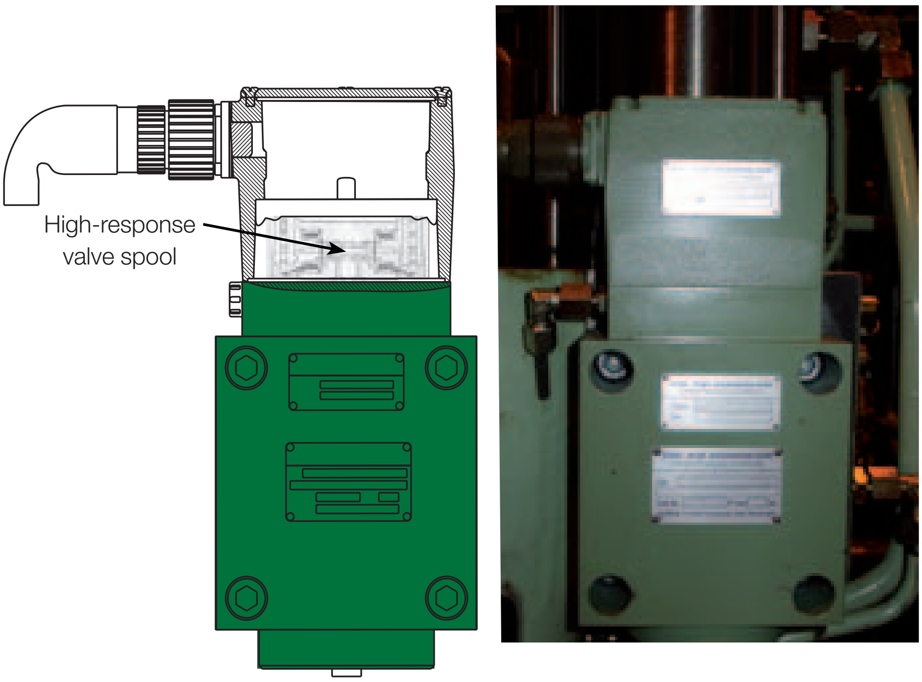


Fig. 14: ELVA valve, location and high-response valve spool

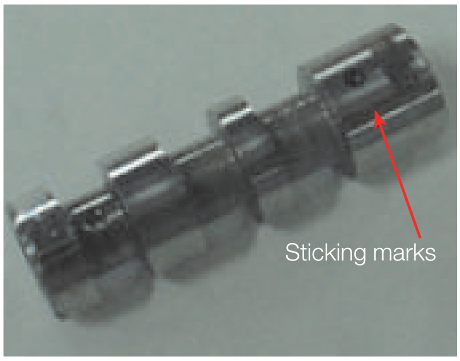


Fig. 15: ELVA valve, sticking high-response valve spool

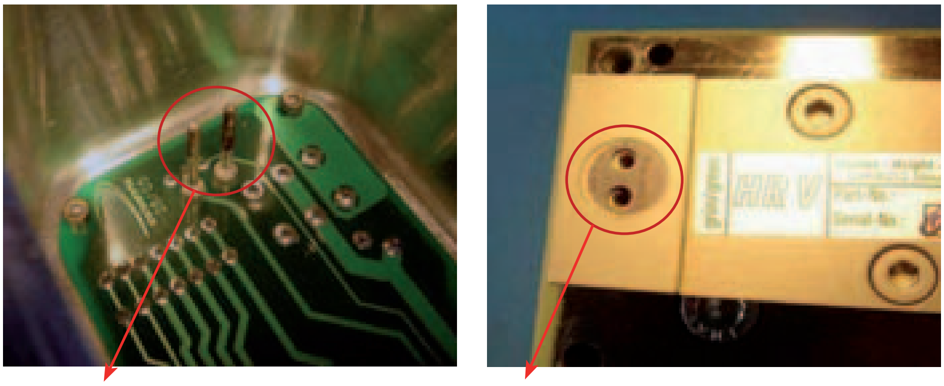
FIVA valves

At year-end 2006, the combined Fuel Injection and exhaust Valve Actuation (FIVA) control valve were in service on approx. 70 engines. Most of the FIVA valves on these engines are produced by a European subsupplier. However, in autumn 2006, an in-house designed version of the FIVA valve was launched in full scale service on a series of 12K98ME engines. Before being launched, the MAN B&W FIVA valve, Fig. 17, has undergone substantial tests both on our research engine and in service on individual cylinder units on a K98 engine.

For the subsupplied FIVA valve version (based on the first 70 engines in operation) experience can be outlined as follows:

- In general, the FIVA valves have seen much fewer vibration-related troubles than the ELFI and ELVA valves. A resilient mounting design, Fig. 18, has been applied from the beginning, and extended vibration testing (up to 1 kHz) has been used, Fig. 19.

- In a few cases, we have seen growth of the main spool in the FIVA valve. Fig. 20 shows a main spool where the diameter has grown 6 micron. The reason has been put down to improper heat treatment of the main spool and the process has been corrected.



Sign from sparks due to bad connection and high current

The spring force in the connector has dissapeared due to sparks/high temperature

Fig. 16: Faulty connector between high-response valve and ELVA PCB

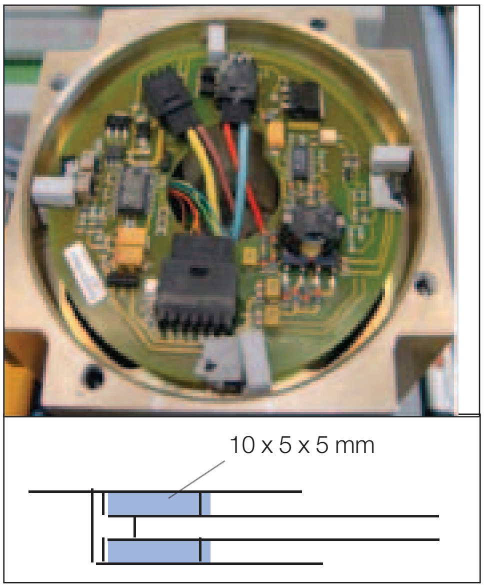


Fig. 18: FIVA valve, resilient mounting of PCB

- We have experienced untimed injection and exhaust valve actuation and, in a few cases, the untimed injection has caused cylinder cover lift. With stopped engine, two observations have been encountered. Either the exhaust valve moves by itself, or both fuel injection and exhaust valve movement occurs. The above phenomena have been identified to have been caused by a component on the FIVA PCB which shuts down or freezes the FIVA feedback signal caused by overtemperature shutdown of the component (self protection). In Fig 21, correct functioning as well as malfunctioning due to faulty controller feedback are illustrated. It has been found that a high ambient temperature of the FIVA PCB will result in the above described malfunction. Therefore, we have screened all FIVA valves in service as well as on new deliveries at an ambient temperature of 70°C, Fig. 22. If the function is not correct, the FIVA valve is returned to the sub-supplier. A redesign of the FIVA PCB reducing the internal heat production is presently being carried out. This will further increase margins temperature-wise.

Fig. 17: MAN B&W FIVA valve

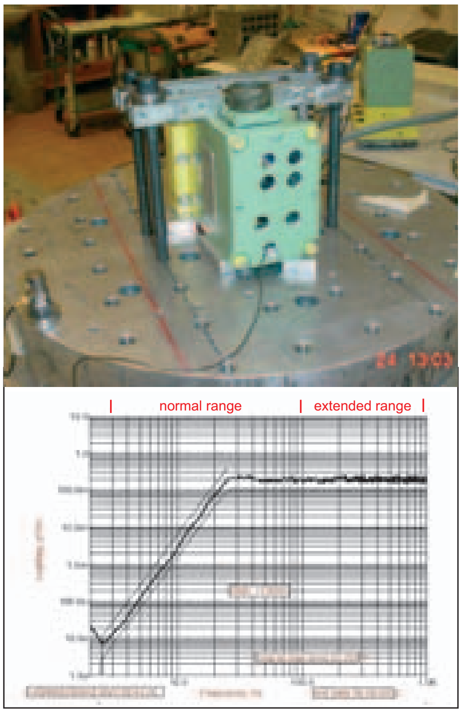


Fig.19: Control valve, extended vibration testing

Feedback sensors for exhaust valve and fuel oil booster

On certain engines the sensor signal has been found to be out of range in one end of the exhaust valve stroke, which resulted in a signal failure alarm. The cause was incompatibility between the sensitivity of the sensor and the material of the cone on the exhaust valve. The calibration of the sensor has been changed.

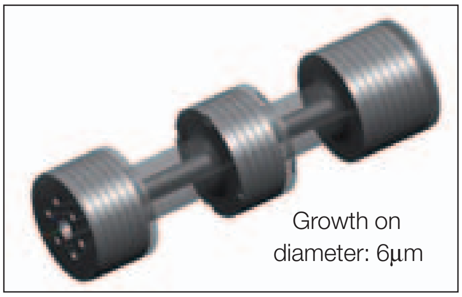


Fig. 20: FIVA valve, growth of main spool

Fig. 22: Temperature test of FIVA valve with mobile equipment, the valve is connected to a CCU during the test

The plastic sensor tip has broken loose or it has been pressed in, Fig. 23. The tip has been reinforced and the internal moulding in the tip has been improved by process improvements.

A number of malfunctioning feedback sensors have been returned from the vessels with ME engines in service. However, a large part of these sensors functions satisfactorily when they are later tested at the sub-supplier. In a number of cases, visual inspection of the sensors has revealed “scratch marks” on the connector, see Fig. 24. We believe that a new type of connector is needed. A new type of connector is currently being tested, and it is expected that this will soon be introduced as the new standard on the inductive feedback sensors.

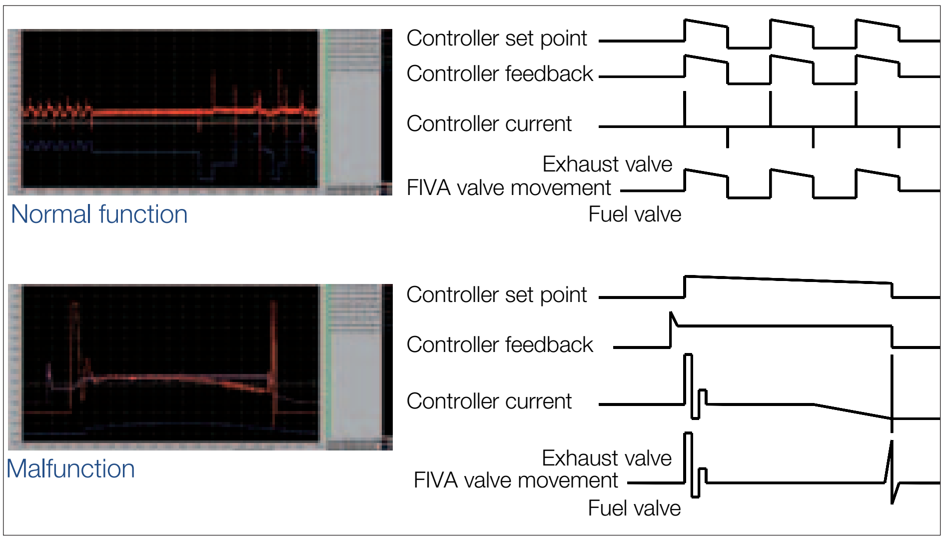


Fig. 21: Uncontrolled feedback signal due thermal overload of FIVA electronics FIVA valve, growth of main spool

Accumulator service experience

Regarding accumulators, we have seen a number of cases of damage to the diaphragms inside the accumulators. These failures have occurred primarily before/during/after shop test and during sea trial. So far, very few cases of diaphragm damage have been experienced in service. The above pattern has led us to do the following revisions of procedures and specifications:

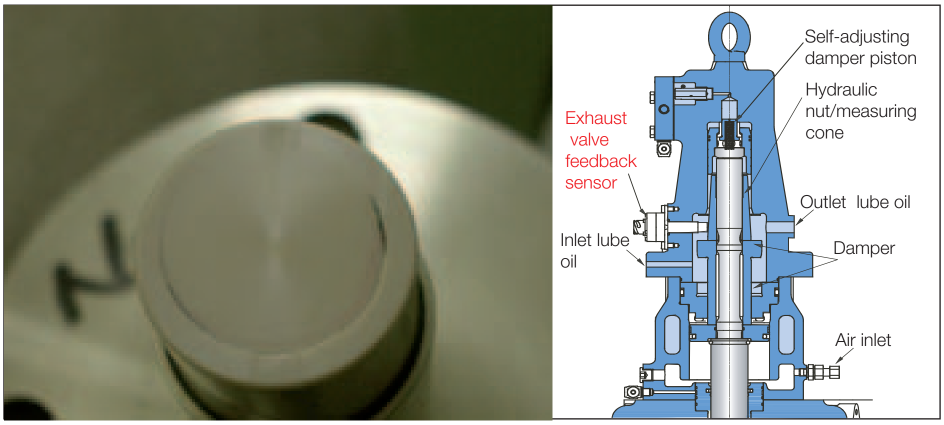


Fig. 23: Exhaust valve feedback sensor

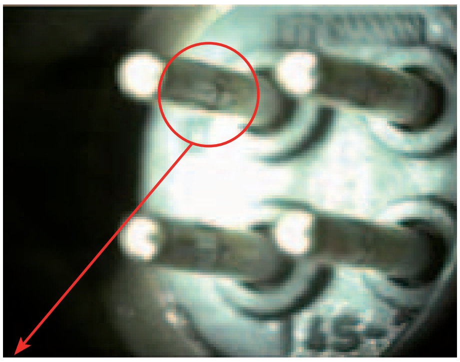


Fig. 24: Scratch marks on connector for exhaust valve feed back sensor

- New flushing instruction using a lower pressure than the full hydraulic start-up pressure
- N2 charge pressure lowered from 105 to 95 bar
- New instruction on how to pressurise an HCU after maintenance work
- New software for prevention of exhaust valve actuation at too low hydraulic oil pressure during wind willing after engine shutdowns
- Instruction on how to prevent leakage at the MiniMess
- Check of N2 charge pressure at six months intervals

Maintenance of accumulators was the subject of the first dedicated service letter on ME engines. Fig. 25 summarises our recommendations, which are to adjust the nitrogen pressure to 95 bar, check the MiniMess for leakages and apply the MiniMess cap after check and charging. Furthermore, it is recommended to check the nitrogen pressure at six months intervals. Also, instructions regarding pressurising an HCU after maintenance work are included in the service letter.

Fuel injector non-return valve

The much higher pressure rate of an ME pressure booster than on a conventional fuel pump means a much higher impact on the fuel injectors on ME engines. This has led to cracking/ breakage at the cut-off shaft, Fig. 26, on the non-return valve.

In order to control/minimise this high impact, internal damping has been applied on the non-return valve, Fig. 27. Service feedback has confirmed the applied solution.

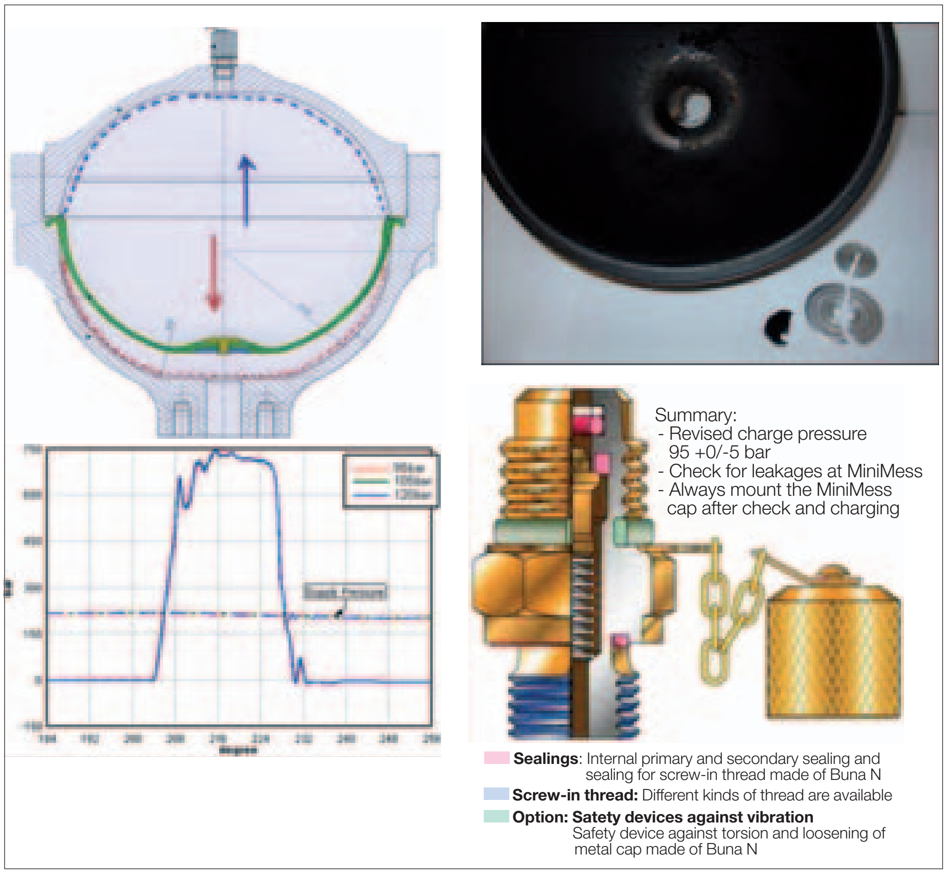


Fig. 25: Maintenance of accumulators

Exhaust valve high-pressure pipe

For the exhaust valve high-pressure pipes we have experienced heavy wear marks on the actual high-pressure pipe, Fig. 28. The reason is that the protection tube touches on the high-pressure pipe and, due to a certain vibration level, wear marks have developed.

A solution by inserting plastic distance pieces (so-called chafing guards, Fig. 29) between the high-pressure pipe and the protecting tube has been tested successfully in service. Chafing guards are thus introduced as the new standard.

Hydraulic power supply

The hydraulic power supply (HPS) unit produces the hydraulic power for the hydraulic cylinder units (HCU). The HPS unit includes both the engine-driven pumps, which supply oil during engine running, and the electrically driven pumps, which maintain the system pressure when the engine is at a standstill. The engine-driven pumps are coupled through a gear drive or a chain drive to the crankshaft, and are of the electronically controlled variable displacement type.

The hydraulic power supply system features, as standard, a number of engine-driven pumps and electrically driven startup pumps. The engine-driven pumps are axial piston pumps (swash plate types), and the flow is controlled by a proportional valve. On some K98 engines, we have initially seen problems with noise from these pumps during astern operation. As a preliminary countermeasure, this has effectively been cured by installing booster pumps securing that cavitations on the suction side of the swash plate pumps will not occur during astern running. A permanent countermeasure has been to equip the largest engines (e.g. 12K98ME/ME-C) with more swash plate pumps of a smaller size. These smaller-size pumps do not have problems with astern operation.

For certain engine types, startup pump capacities have been increased to be able to deliver sufficient startup pressure on one startup pump within 90 seconds.

Fig. 26: ME fuel injector: breakage of cut-off shaft and non-return valve

Fig. 28: Exhaust valve high pressure pipe: Serious dent marks has been observed

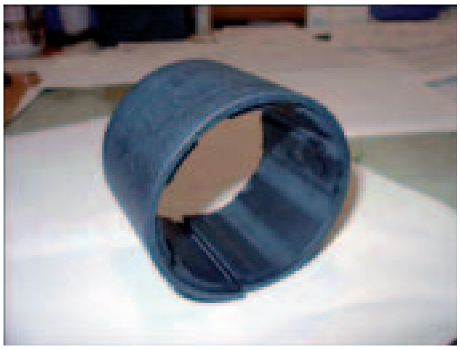


Fig. 29 Exhaust valve high pressure pipe: Chafing guard installed between high-pressure pipe and protective tube

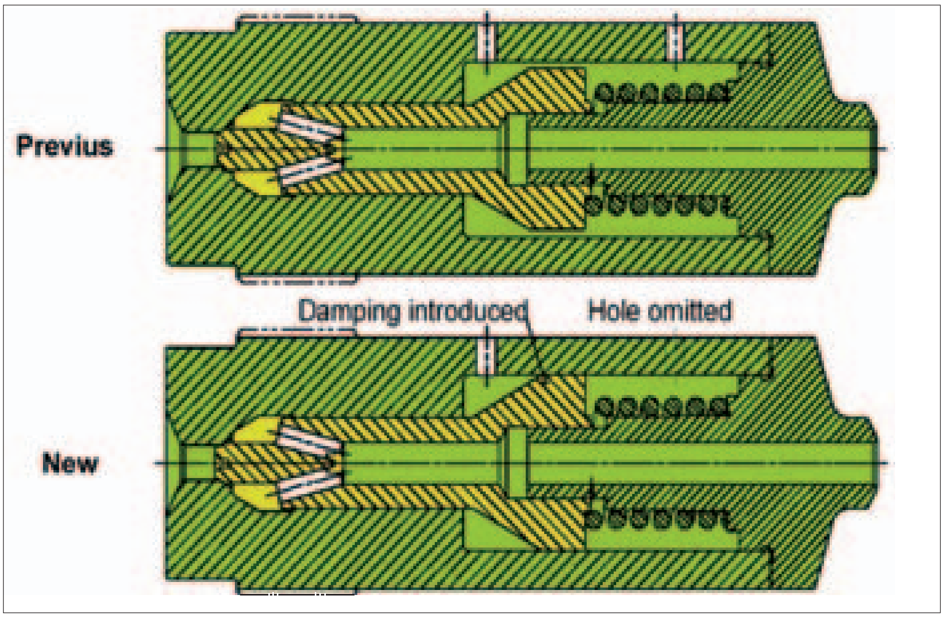


Fig. 27: ME-fuel injector: internal damping applied at non-return valve

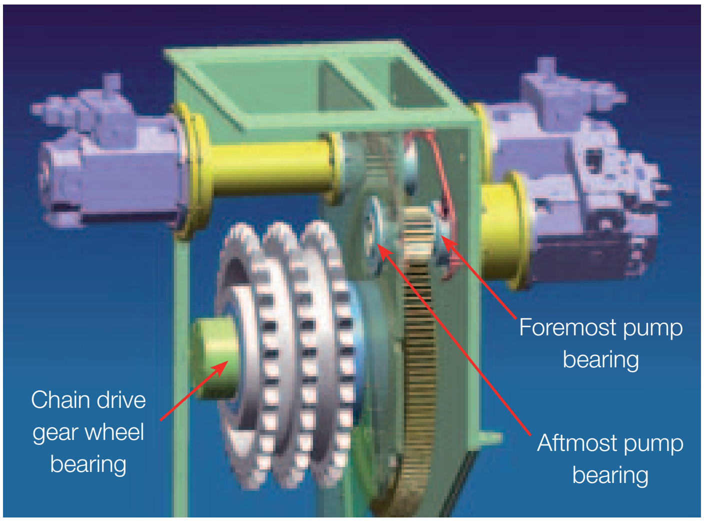


Fig. 30: HPS gearbox: Position of bushings

Fig. 31: HPS gearbox: Failed bushings/bearings

On new engines, the hydraulic oil filter has been re-specified from mesh size 10 micron to mesh size 6 micron. The reason for this was to prolong the lifetime of various components subjected to wear.

In July 2006, a HPS gearbox breakdown was experienced on a 12K98ME engine. The failure contemplated a major breakdown of the large bushing carrying the chain wheel and the large gear wheel, Fig. 30.

The failed bushing/bearing is seen in Fig, 31. In addition to the failure of the major bushing, also wiping of the white metal on the pump shaft flange bushings was seen.

An investigation showed that the clearance was at the lower end of the tolerance range, both for the large bushing and for the pump shaft flange bushings. To ensure sufficient clearance in any case, the tolerances were changed, see Fig. 32.

In order to get early warnings for such breakdowns in the future, temperature monitoring of the large hydraulic power supply gearbox bushing has been introduced on K98ME/ME-C engines in service.

Hydraulic pipes

Cases of cracked hydraulic pipes for the servo oil to the swash plate pumps have been seen, and investigations have proved these cracks to occur due to vibrations. To avoid this, the pipe dimension has been changed, and flexible hoses have been introduced as an extra precaution, see Fig. 33.

Shafts for engine-driven hydraulic pumps

Initially, teething problems have included breakage of the shafts for the engine-driven hydraulic pumps.

The purpose of the shaft design is to set an upper limit to the torque transferred, so as to safeguard the common gear in the event of damage to a pump. However, the shafts broke due to a too low torque capability.

The design of the shafts has been changed in order to increase the margin against breakage. The initial design, shown in Fig. 34, featured six studs and a frictional connection, and the bolts were sheared at too low a torque.

The new design shown in Fig. 35 has a centre bolt, which tightens together a frictional connection. No problems have been experienced with this design.

Besides this, we have introduced forced lubrication of the shaft assembly to counteract cases of wear of the splines for the shaft and gear wheel. Splines are now also hardened. We have had good experience with this designs.

Gearbox

Fig. 36 shows an example of an inspection of the gearbox for the pump drive after 10,762 hours. The condition of the gearbox was found to be excellent.

Bearing for chain drive gear wheel	
Hydraulic pump configuration	4x 750 cm3
Bearing diameter	Ø 342 mm
Updated min. clearance	0.18 to 0.26 mm
Updated max. clearance	0.27 to 0.35 mm
To be updated before sea trial	Yes
Bearings drive shaft	
Hydraulic pump configuration	4x 750 cm3
Bearing diameter	Ø 150 mm
Updated min. clearance	0.09 to 0,16 mm
Updated max. clearance	0.15 to 0,23 mm
To be updated before sea trial	Yes

Fig. 32: HPS gearbox: revised clearances for the bushings/bearings

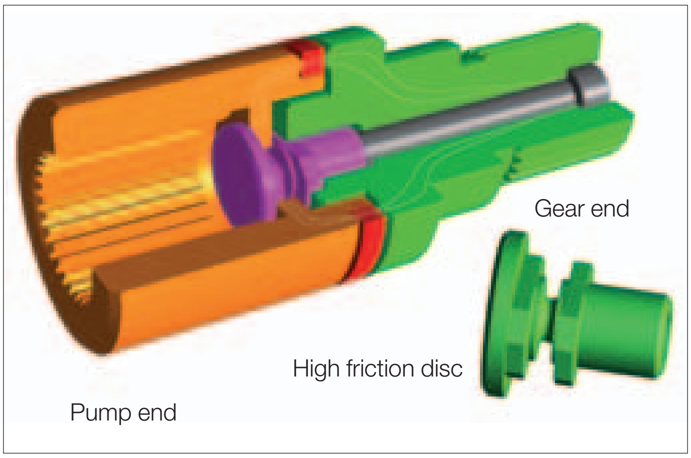


Fig.35: Pump safety shaft, new shaft assembly with central bolt

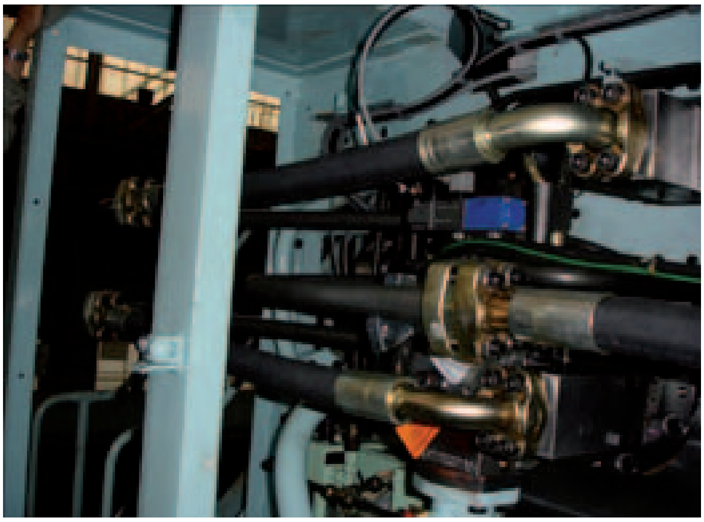


Fig. 33: Flexible hoses on K98

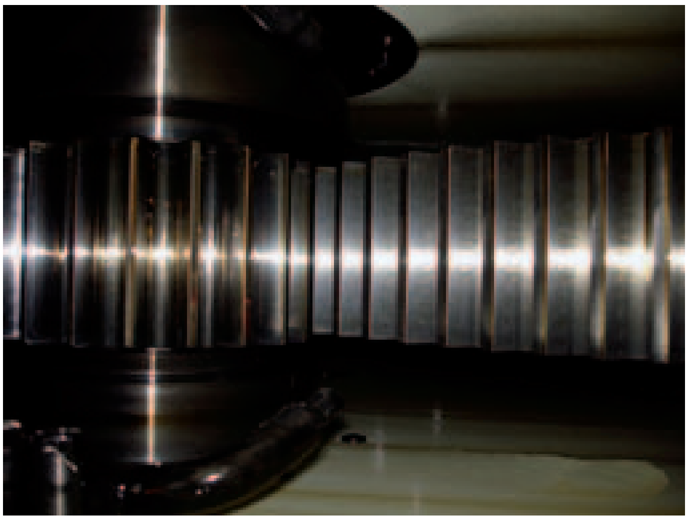


Fig. 36: 12K98ME, gearbox for engine driven hydraulic pumps, excellent condition of gears after 10,762 hours

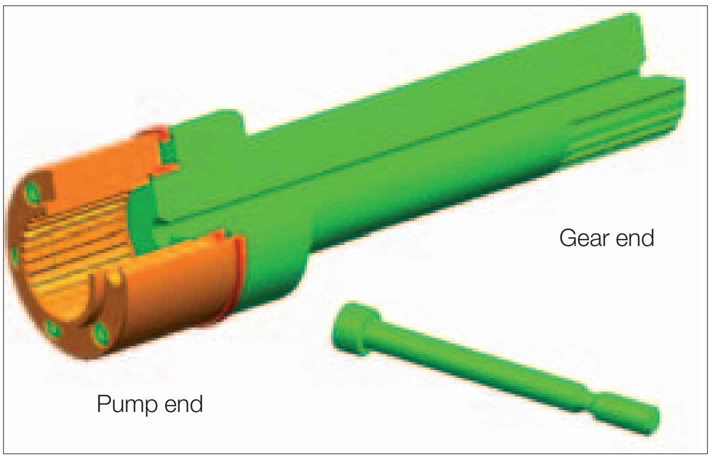


Fig. 34: Pump safety shaft, initial design

Engine control system

The ME Engine Control System (ECS) consists of a set of Multi Purpose Controllers (MPCs). These are generally used in Auxiliary Control Units (ACU), Cylinder Control Units (CCU), Engine Control Units (ECU) and Engine Interface Control Units (EICU), and they are identical from a hardware point of view. Once connected in the individual application (CCU, ACU, ECU or EICU), the MPC will load software according to the functionality required.

On the MPC, channels 70 and 71 have been damaged in some cases. This was caused by wrong or fluctuating signals at the outputs. Consequently, a breakdown of a capacitor in the DC-DC converters (Fig. 37) occurred. This was determined to happen when 24V or higher voltages (noise, etc.) were applied backwards into the terminals of the channels. Initially, the AO-DO daughter boards of the MPCs in production were improved by applying a transorber across the output terminals. Later, the board has been redesigned.

Production failures in the Printed Circuit Boards (PCBs) have caused broken connections in the inner layers, Fig. 38. The PCB base material has been changed to a type with a lower thermal expansion coefficient in the cross-sectional direction. Furthermore, the copper layer thickness in plated-through holes has been increased to fulfil the specification.

We have experienced bend pins in the PCB to PCB connectors. This is a production failure, and additional production tests have been added on the fully assembled units to sort out erroneous units for repair.

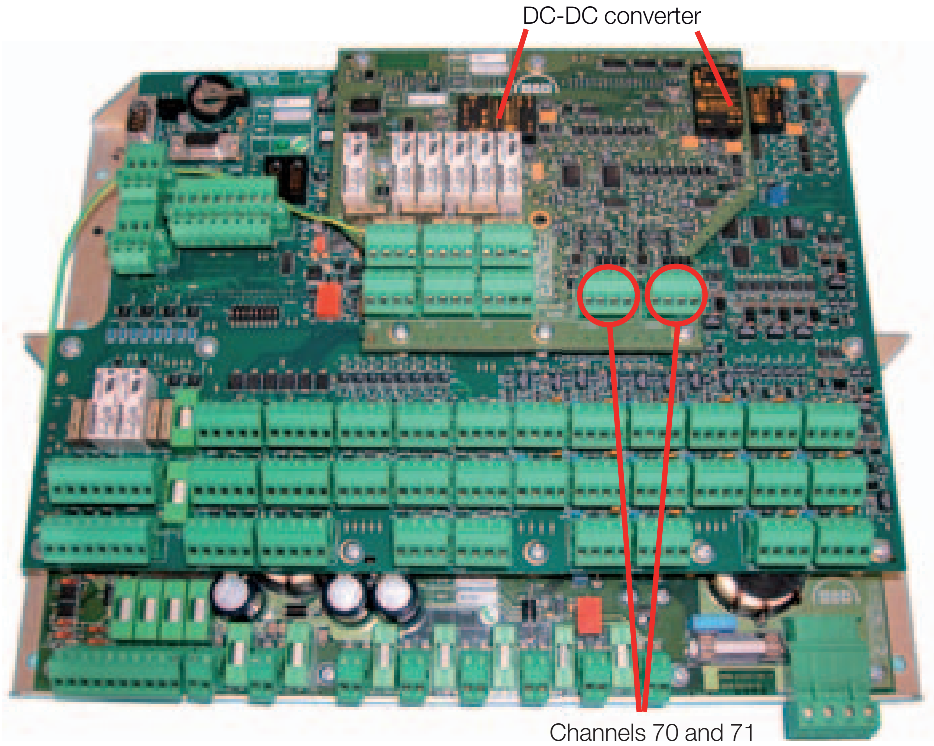


Fig. 37: MPC board, failure of channels 70 & 71

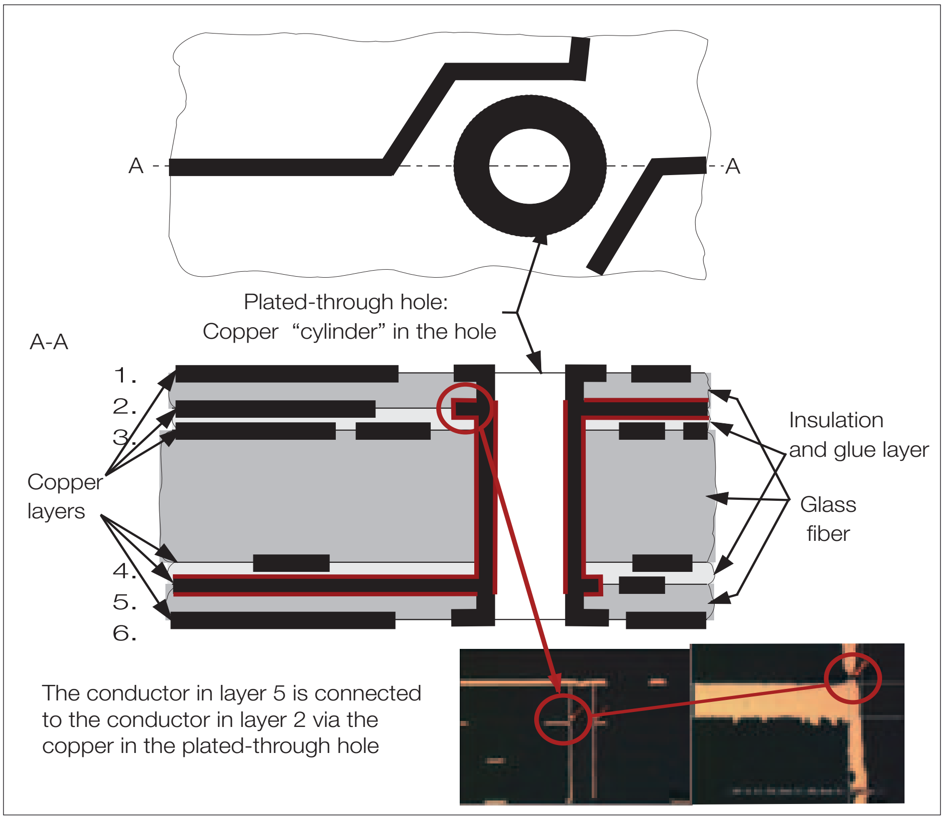


Fig. 38: PCB, broken copper layers

Main Operating Panels (MOPs)

The present execution of the ME engine control system comprises one Main Operating Panel (MOP), which is an industrial type PC with an integrated touch screen from where the engineer can carry out engine commands, adjust engine parameters, select the engine running modes and observe the status of the control system. In addition to this, the system comprises also a conventional marine approved PC serving as a back-up unit for the MOP.

Both PCs are delivered with their own customised PC type specific operation system image software pre-installed. At the time of installation, and prior to test and commissioning, both PCs are loaded with the same application software and the same plant specific parameter software.

Because of the use of conventional PC types for the back-up unit, we experience very frequent model changes to this unit. In connection with the introduction of a new model, it is necessary to prepare new software images together with updating of documentation. This creates a lot of logistical issues.

In order to ease the handling, installation and support on plants in service for the licensor, licensee, shipyard and owner, we will introduce the same hardware for the back-up unit as for the Main Operating Panel. The PC type will remain unchanged for a longer period as it comprises a chipset with an extended product life support (Intel Industrial PC platform solutions). The solution will comprise a separate PC with a separate touch screen display. This solution will be more flexible and meet various specific requirements faced in relation to arrangement and installation. For instance, the display can either be mounted in the control room console or, alternatively, in an optional cabinet (bracket) for use as a desk top type.

The new configuration will, as a consequence, only use the 24V supply. In this way, the 110V Uninterrupted Power Supply (UPS) can be omitted.

Software updates

Since the introduction of the ME engine, the Engine Control System software has been updated a number of times.

These updates have been introduced for a number of different reasons listed below:

- Updates because of software defects
- Updates because of extension in Human Machine Interface (HMI)
- Updates because of inconveniency in the way the HMI was working
- Updated because of changes in the hydraulic/mechanical system
- Change of operating system

Examples of changes and corrected defects between from version 0510-5 to 0510-6 are given below.

Changes:

- Combined HPS is supported. Combined in this context is when the HPS is electrical and engine-driven in any combination in normal running condition
- Double pipe pressure is displayed on MOP
- An ‘Emission Functionality Version Number’ (EFVN) has been introduced
- Timed actuation og cylinder lubrication at low engine speed has been added
- Crash Stop Detection has been improved
- Engine Speed Fine Tune from MOP added
- Both tacho positions are shown on MOP
- Handling of shutdown has been revised to safeguard hydraulic accumulators
- On combined HPS only one start-up pump is started in normal conditions
- Handling of failures on FIVA/ELFI position feedback has been improved.

Corrected defects:

- MOP display freeze problem has been minimised
- Blower starting failure has been corrected
- Tacho “self-curing” ability has been improved
- HPS operation has been improved.
- Exhaust by-pass standby delay time parameter unit changed from sec. to min.
- Improved display of alarm descriptions
- On/Off exhaust bypass function corrected
- “Open stroke low alarm” and missing ignition on one cylinder during reversing could occur in some cases.

A: Angular Encoder Design

B: Trigger Ring Design

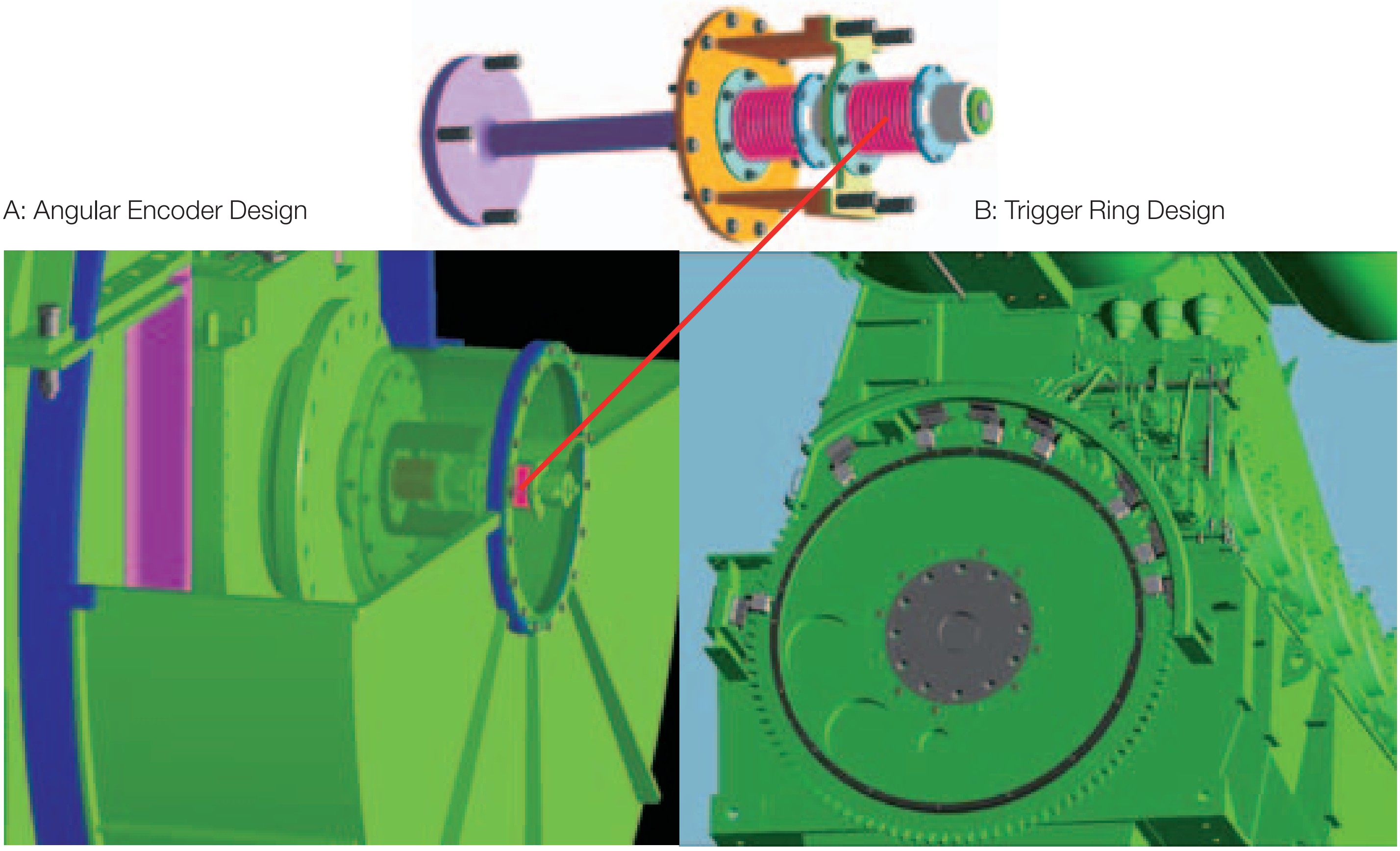


Fig. 39: Tacho systems

At the time of writing this paper, the official ME ECS software version is the 0510-6.2. This version will be used for updating all engines in service to ensure that we have the best-tested control software on the ME engines, and the best foundation for further development of control software for the ME engines.

Tacho system

Initially, the ME tacho system was designed on the basis of trigger segments with a sine-curved tooth profile mounted on the turning wheel. The total trigger ring was built from eight equal segments.

Two redundant sets of sensors were applied. This initial tacho system is relatively expensive, and the system is also rather time consuming to commission on testbed/sea trials. Today, this system is only specified if the free end of the crankshaft is occupied by other equipment like power take-offs.

The new tacho system is based on optical angular encoders installed on the free end of the crankshaft. This system, consisting of two redundant encoders, is easier to install and adjust. Fig. 39 shows the two systems.

When properly adjusted, both tacho systems have, in general, given rise to only minor concern. However, one event where an incorrectly installed (tightened) Geislinger damper fell off the crankshafts has been experienced. This caused damage to both angular encoders, and at the same time resulting in 'loss of manoeuvrability'.

ME system documentation

In one incident, 'loss of manoeuvrability' was partly caused by a lack of precise documentation/information. This has been rectified both by updating our instruction book and by introducing two additional alarms.

In order to be able to understand the incident it is necessary to know the principle of redundancy applied in the ME system. This principle of redundancy dictates that no single failure must stop the engine or prevent further propulsion. However, the consequence of more failures is undefined. This principle is fully accepted by the classification societies.

The incident occurred on an ME engine with four (4) engine-driven hydraulic

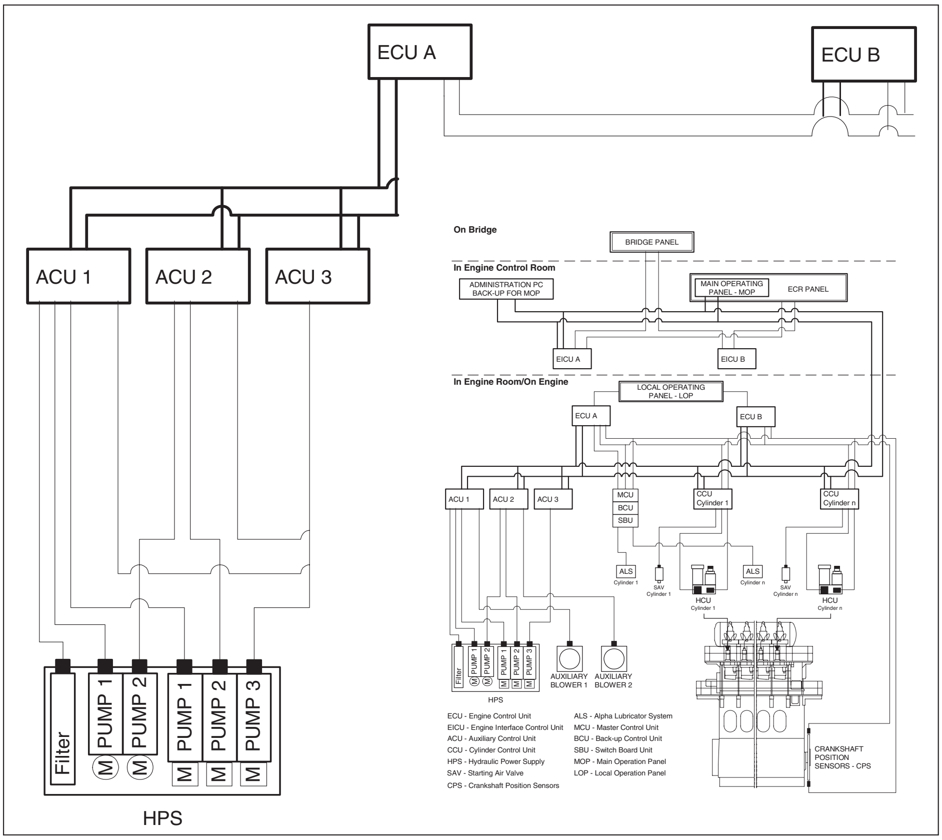


Fig. 40: ME-system, control diagram (initial version)

pumps. Control of one of these pumps was lost. When this happened, the swash plate for the uncontrolled pump went to ‘full ahead’.

In the original version of the instruction book, a system consisting of only three (3) engine-driven pumps is shown, Fig. 40. Each of these pumps is controlled by an ACU (Auxiliary Control Unit). However, the control of a system with four or more engine-driven pumps is not described.

In the updated instruction book, a system with up to five engine-driven pumps is shown, Fig. 41. It can be seen that, if installed, pump Nos. 4 and 5 are controlled by ECU A and ECU B, respectively.

The incident described above developed further as the crew took the decision to shift the engine control from ECU A to ECU B and dismantle ECU A.

According to the updated instructions, the pump control for pump No. 4 is thus also lost. Pump No. 4 then goes to ‘full ahead’ and astern operation is no longer possible.

On the basis of the above incident, in addition to updating the instruction book, we have added the following two alarms:

1. Alarm for ‘Pump Failure’ if an ACU or a pump controlling ECU fails.

2. Alarm for ‘Lost Manoeuvrability’ if two or more pumps fail.

Having informed the crews of the above improvements, similar incidents will be avoided in future.

Alpha lubrication system

The ME engine has the advantage of an integrated Alpha lubrication system, which utilises the hydraulic oil as the medium for actuation of the main piston in the lubricators. Thus, a separate pump station and control are not needed, compared with the MC counterpart.

Most of the ME engines in service feature this system and, in general, the service experience has been good. Cylinder liner and piston ring wear rates have been low, giving promising expectations of long intervals between overhauls.

On certain engines of the S50ME-C type, we have experienced a number of teething troubles in the form of broken lubricator plungers as well as damage to the main activator piston.

In order to alleviate these problems, a revised design of the plungers and main pistons has been introduced on the Alpha Lubricators.

A new actuator piston with a reinforced disc without holes and damper has been introduced, together with a new stroke limiter. The solenoid valve has also been modified by introducing a damping orifice to reduce the hydraulic impact, which previously influenced the problems observed.

Additionally, steel spacers have been fitted below the return spring to remove the turning effect created from compression of the spring, and hereby affecting the alignment of the small plungers.

In the event of a low engine room temperature, it may be difficult to keep the cylinder oil temperature at 45°C in the ME Alpha Lubricator mounted on the hydraulic cylinder unit.

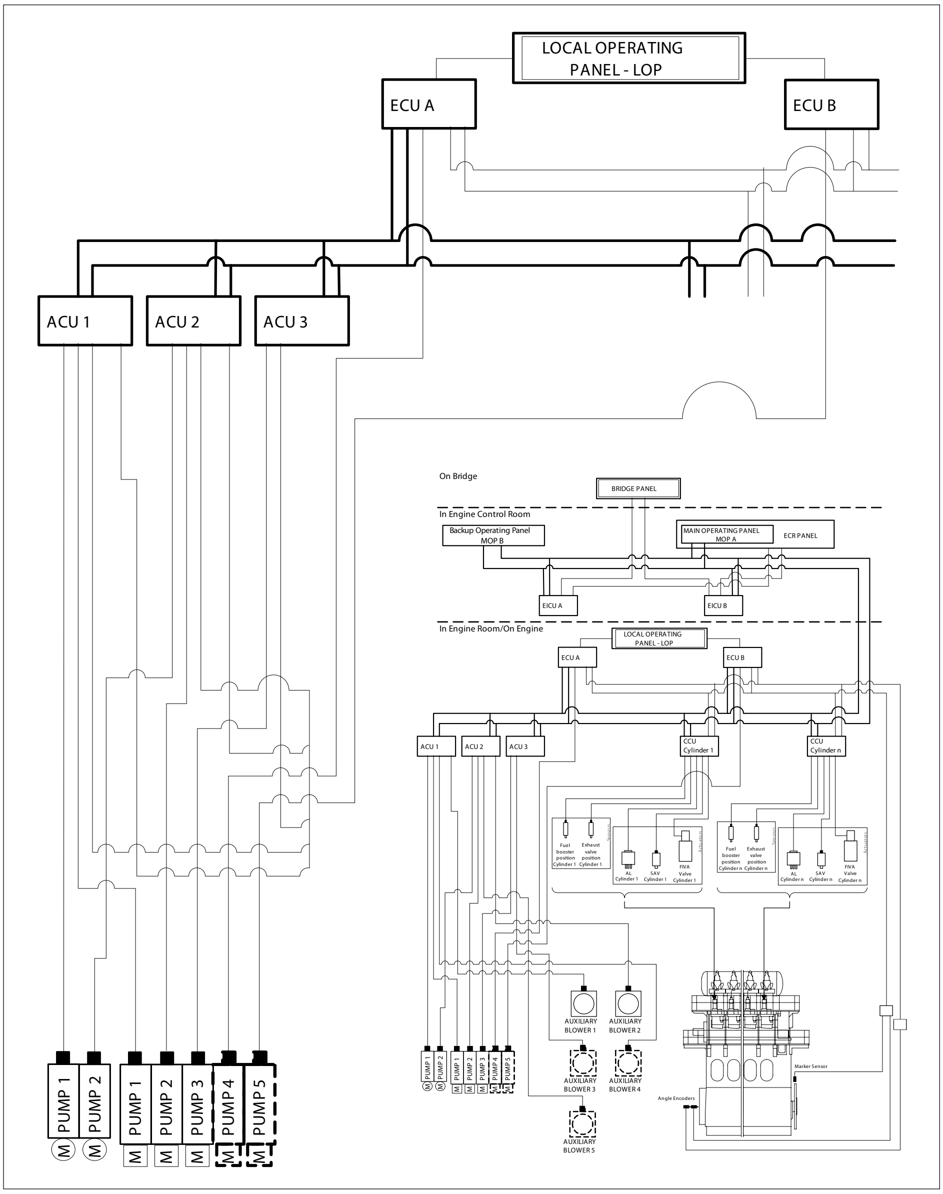


Fig. 41: ME-system, control diagram (updated version)

Therefore, we have introduced insulation and electrical heating of the cylinder oil pipe from the small tank in the vessel and of the main cylinder oil pipe on the engine.

ME engine service experience – summary

The comments presented in this paper are all based on actual feedback experience from owners and ship crews.

All issues are addressed continuously as they occur, so as to control and eliminate teething troubles immediately.

Some of the very positive feedback that we have received, by way of statements received from operating crews, are summarised below in bullet points:

- Engines of this type allow a considerable saving of fuel and cylinder oil.
- The electronic control system of the engine allows supervising of practically all operating processes, such as: lubricator management, cylinder oil consumption control, load distribution on cylinders, cylinder cut-off in the event of a malfunction without stoppage of the main engine.
- A considerably smaller amount of fuel deposits from combustion in the scavenge air boxes and the exhaust gas economiser is observed.
- The system provides wider options for adjustment of the engine.
- In spite of its complexity, the system is divided by several standard modules, thereby, allowing the crew to quickly locate a faulty module.
- No special periodic maintenance is required for the electronic parts.
- The modules' design allows easy and rapid replacement.
- The modules and control units of the system have a built-in central processing unit (CPU) that ensures continuous self-monitoring of the technical condition, and an alarm is given to the crew in the event of any abnormalities.
- The communication between the operators at the three remote control stations, i.e. the bridge, the starboard wing, and the engine control room, and the control units of the system is effected by means of a special industrial network that reduces the number of wires needed for data transferring, i.e. reliability is improved.

We take this as a proof of the ME engines gaining momentum in the market, and most certainly presenting operating advantages to owners and crews.

The MC/MC-C Engine Series

Time Between Overhaul for the latest generation of MC engines

Over the last 4-5 years, the Time Between Overhauls (TBO) has been gradually extended in our written material describing typical obtainable TBOs, Fig. 42.

This development has triggered the wish to extend TBOs further, and for certain ship types (e.g. VLCCs), it has prompted investigation into whether 32,000 hours (or 5 years) between overhauls is realistic.

As the basis for the investigation, we have chosen the S90MC-C/ME-C engine series as a representative for the newest generation of MC engines. This engine series has been designed and delivered with the newest features available for the MC/ME engines:

- Oros combustion chamber with high topland piston
- Cylinder liner with optimised wall temperature
- Alu-coated piston rings, Controlled Pressure Relieve (CPR) top ring
- Alpha Lubricator in ACC mode (0.19 g/bhphXS%)
- Exhaust valve: Nimonic spindles and W-seat bottom piece
- Slide fuel valves.

Approximately 40 vessels, Fig. 43, with 6S90MC-C/ME-C engines have been used to back up the claim that TBOs of 32,000 hours (or 5 years) is a realistic option.

TBO S90MC-C/ME-C				
Overhaul guiding interval (Hours)				
Component	Old MC-C	New MC-C	ME-C	Realistic potential
Piston rings	12-16,000	16,000	24,000	32,000
Piston crown	12-16,000	16,000	24,000	32,000
Piston crown, rechroming	24,000	24,000	24,000	32,000
Exhaust valve, spindle and bottom piece	16,000	16,000	16,000	32,000
Fuel valve	8,000 (nozzle) 8,000 (spindle guide)	8,000 (nozzle) 16,000 (spindle guide)	8,000 (nozzle) 16,000 (spindle guide)	8,000 (nozzle) 16,000 (spindle guide)
Fuel pump	16,000	32,000	-	32,000
Fuel pressure booster	-	-	48,000	48,000

Fig. 42: Time Between Overhaul (TBO), guiding intervals

On the vessel *M/T Maria Angelicoussis* (equipped with a Hyundai-built 6S90MC-C engine), piston overhauls have been carried out successively from 8,000 hours and upward, see Fig. 44. The piston ring wear is extremely low, and from this point of view indicates ‘infinite lifetime’.

The vessels *M/T Kos and M/T Astro Cygnus* are also both equipped with Hyundai-built 6S90MC-C engines. On these engines, the pistons were pulled between 20,000-21,000 hours and 22,000-24,000 hours, respectively. The pulling of pistons on both these engines was caused by ‘internal coking’ of the pistons. The reason for this was fuel oil contamination of the system oil, in both cases caused by leaking fuel pumps. Apart from this specific problem, both engines have shown excellent cylinder condition with low piston ring wear rates, Fig. 45.

The engine onboard *M/T Astro Cygnus* has been a ‘test vehicle’ for the further cylinder oil consumption testing according to the so-called Alpha ACC principle (ACC = Adaptive Cylinder oil Control. As can be seen in Fig. 46, this test has been extremely successful and it indicates further potential for reduction in the cylinder oil consumption.

Below is a summary of the cylinder condition based on all observations on the S90MC-C/ME-C engine:

1. Cylinder liner wear rates: 0.02-0.07 mm/1,000 hours (Fig. 47)
2. Piston ring wear rates: Predicted lifetime: 50,000 hours (Fig. 48)
3. Piston ring groove wear rates: Predicted time between reconditioning: 40,000 hours (Fig. 49).

The exhaust valve condition also gives rise to optimism with respect to the increase of TBOs. Fig. 50 shows a bottom piece of the W-seat design in combination with a Nimonic spindle on a K90MC engine inspected after 36,400 hours without overhaul.

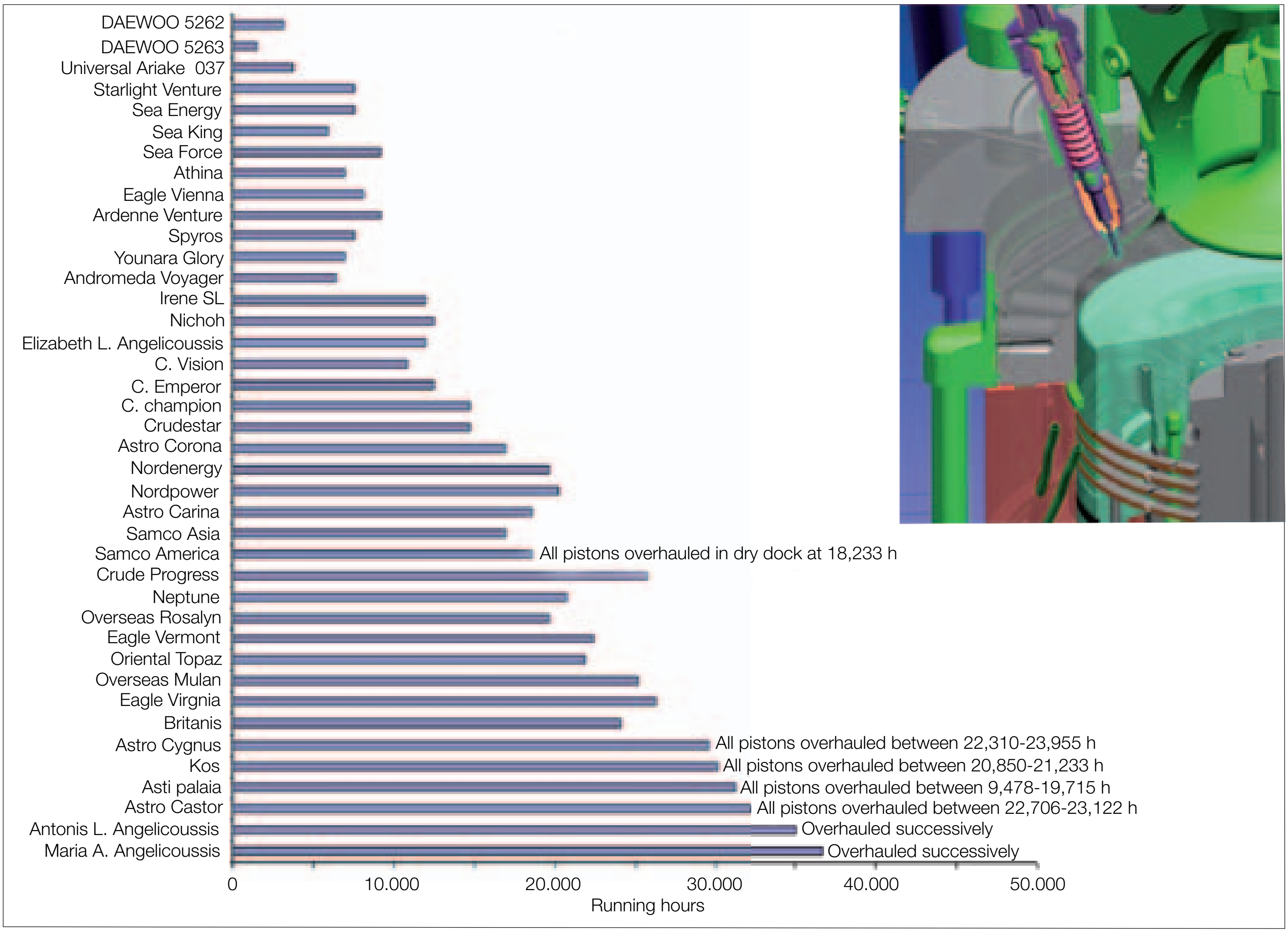


Fig. 43: Fleet of VLCCs equipped with 6S90ME-C/MC-C

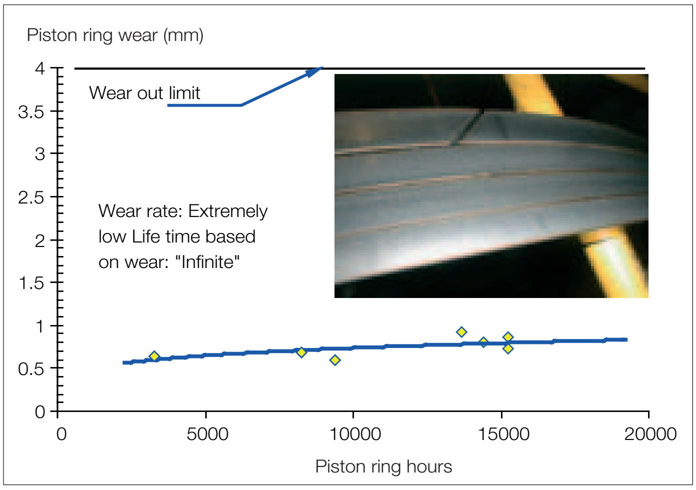


Fig. 44: Piston ring wear measurements, prototype 6S90MC-C (M/T Maria A. Angelicoussis)

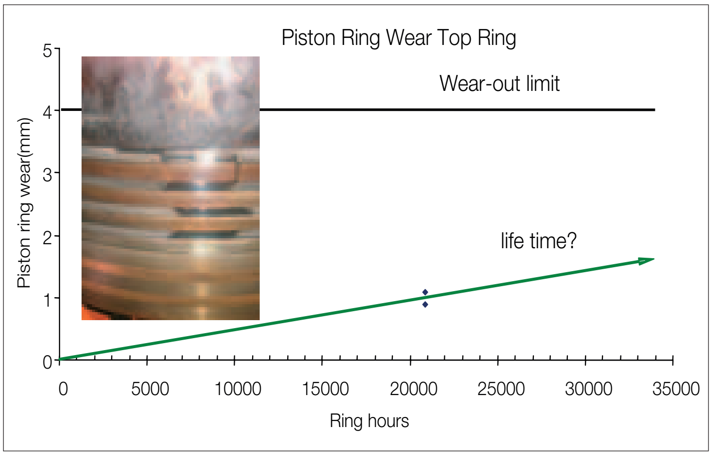


Fig. 45: Piston ring wear measurements, 6S90MC-C (M/T Kos)

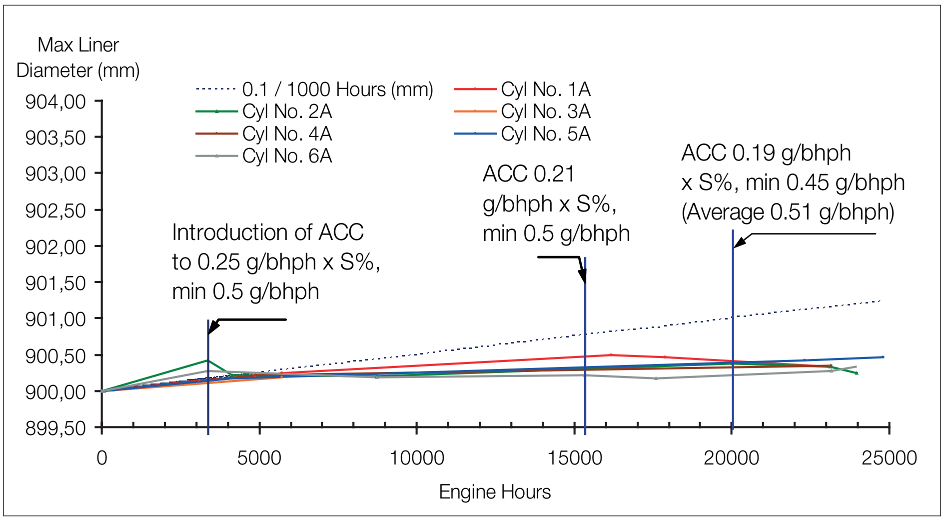


Fig. 46: Cylinder liner wear, cylinder oil reduction test, 6S90MC-C (M/T Astro Cygnus)

K90MC, W-seat and nimonic spindle at 36,400 hours without overhaul

Fig. 50: Nimonic exhaust spindle and W-seat bottom piece

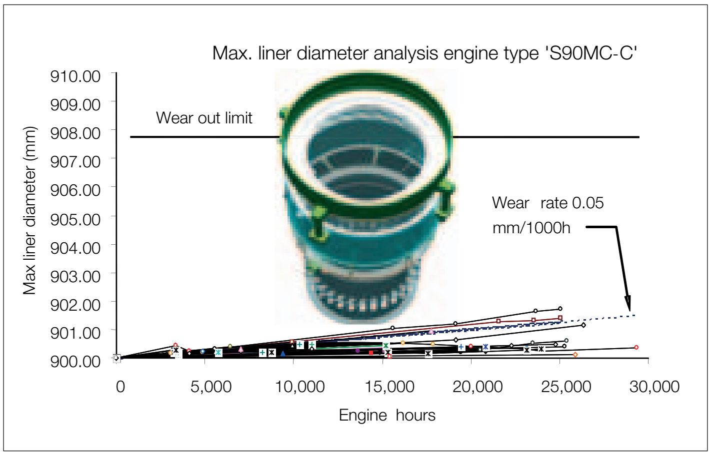


Fig. 47: 6S90MC-C, cylinder liner wear

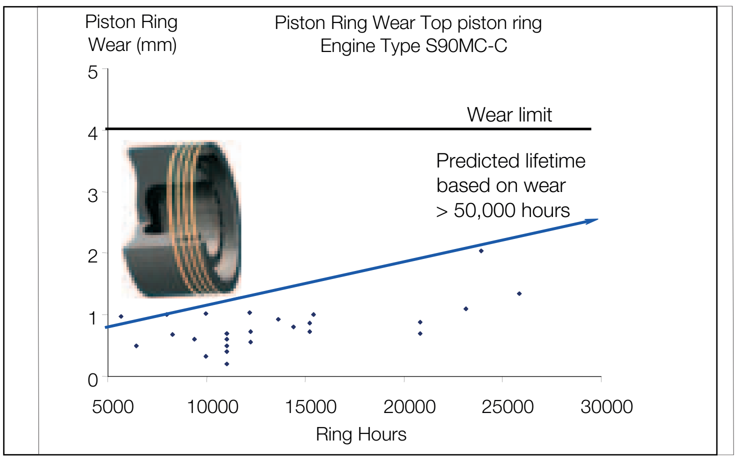


Fig. 48: 6S90MC-C, Piston ring wear

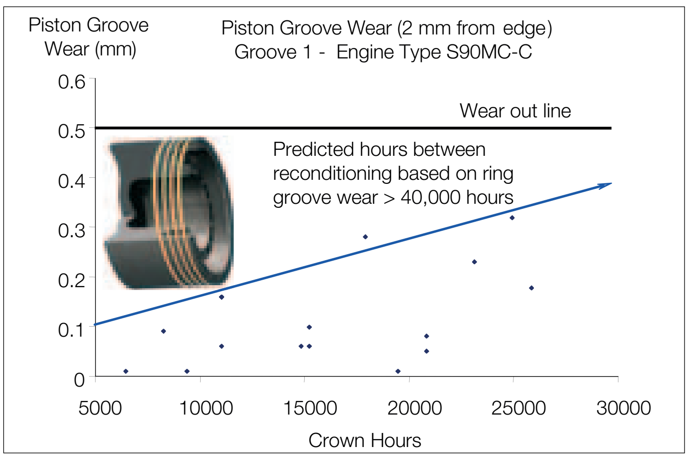


Fig. 49: 6S90MC-C, Piston ring groove wear

Fig. 51 outlines the features of the present exhaust valve design.

With respect to the fuel equipment, 32,000 hours seem to be realistic for the fuel pump itself. The latest experience with the fuel valves confirms overhaul intervals of 8,000/16,000 hours, at which point both the fuel nozzle and the spindle guide should be exchanged. This experience is based on fuel valves of the slide valve type equipped with nozzles of the compound type.

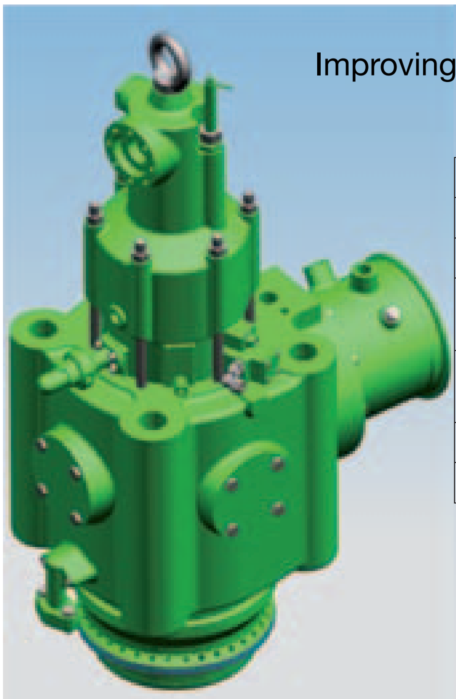
Based on service experience in general, we can conclude that the time between major overhauls of 32,000 hours (or 5 years) is within reach, Fig. 42.

To increase margins further in this respect, we will introduce the following design improvements which are not present on the 6S90MC-C engines described in this section:

- Increased scuffing margin: modified piston ring package, Fig. 52
- Anti internal coking device: piston cooling insert, Fig. 53
- Ring groove wear reduction: underside chrome plating on ring Nos. 1 and 2, Fig. 52.

For tanker operators, these higher TBOs mean that major overhauls can be done in connection with the scheduled dry dockings of the vessels.

For container carrier operators, another more condition based philosophy will pay off. Such a philosophy is practised on the K98MC prototype engine on board *M/V Antwerpen Express*. Fig. 54 shows that on this engine, unit No. 1



Improving the overhauling intervals:

Items	Improved Items
Bottom piece seat	Hardened steel/W-seat
Spindle seat	DuraSpinde/Nimonic
Spindle guide	Cast iron/high spindle guide
Sealing system	Sealing oil Controlled Oil Level (COL)
Viton sealing	U-seal
Spindle stern	HVOF

Fig. 51: Features of present exh. valve standard

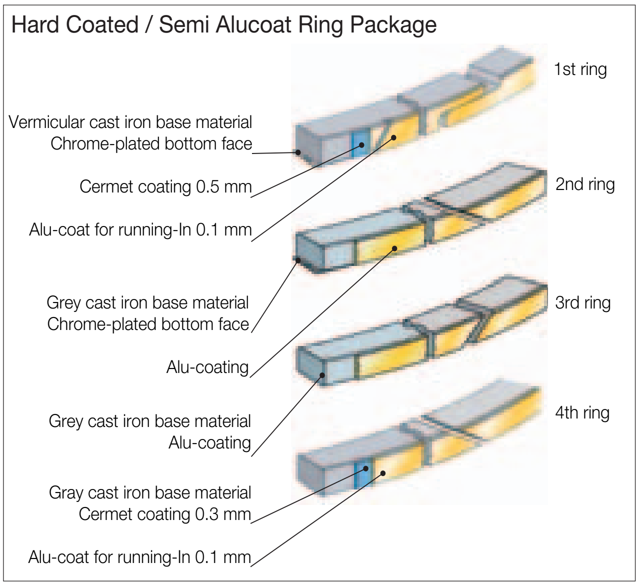


Fig. 52: Updated piston ring package

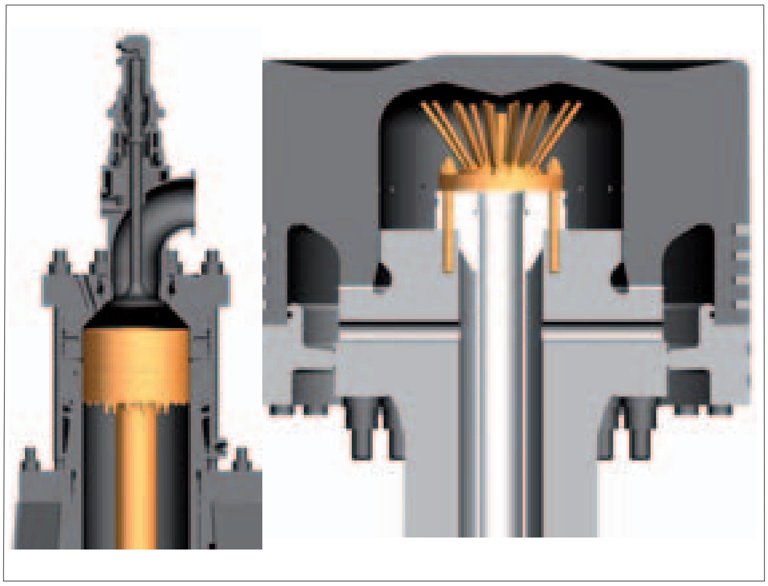


Fig. 53: Piston cooling insert

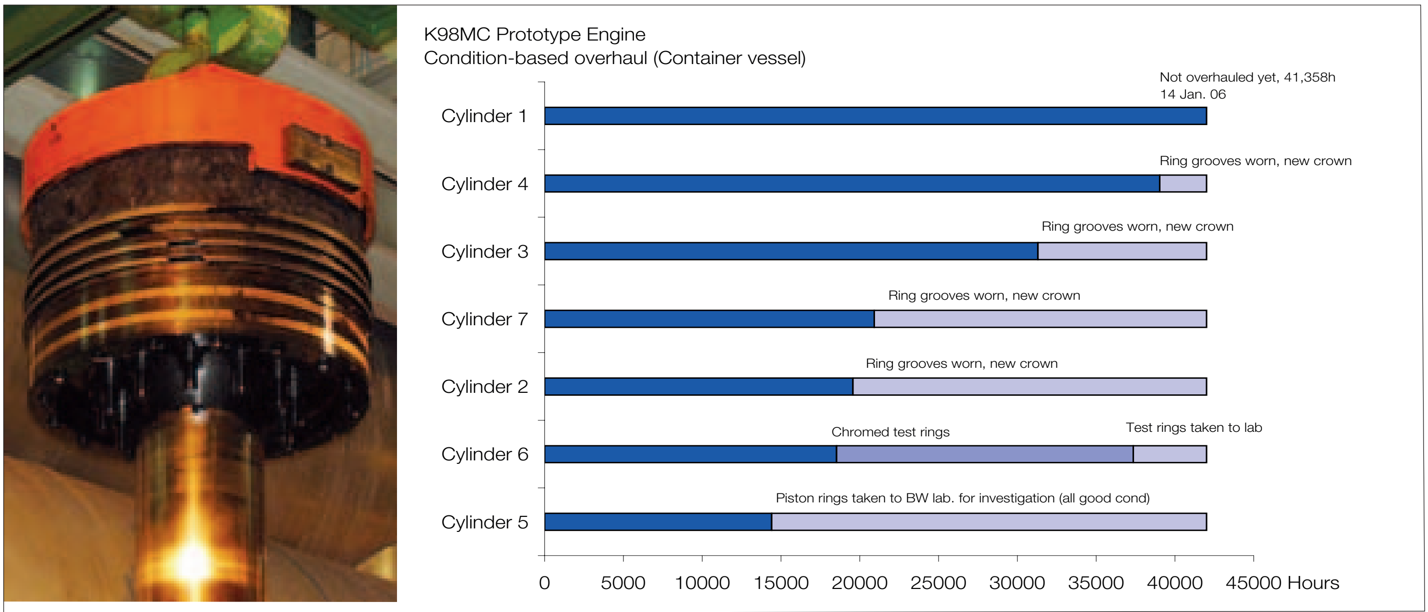


Fig. 54: 7K98MC, condition-based overhaul piston No. 1 overhauled first time after 42,500 hrs

was overhauled after 42,250 hours of operation, seen Fig. 54.

As a conclusion, we can support the wish to extend TBOs further, and for certain ship types (e.g. VLCCs) up to 32,000 hours (or 5 years) between overhauls is realistic, Fig. 42.

Increased scuffing margin

Scuffing of cylinder liners has become a recurring incident on some K98 and K90 engines. Other engine types have also been affected, but to a much lesser degree. Some of the cases have been related to traditional service disturbances like production mistakes and poor fuel cleaning. However, other cases remain unexplained.

The above described possible increase in time between overhauls becomes illusive if scuffing incidents occurs at too high a frequency. Therefore, countermeasures to establish larger margins for scuffing to occur are constantly searched for.

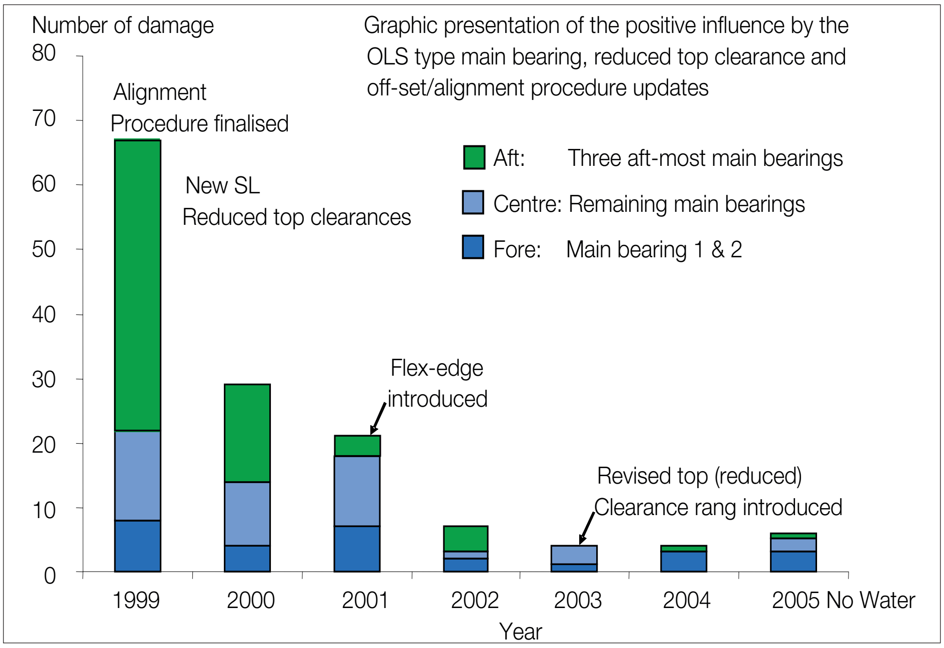
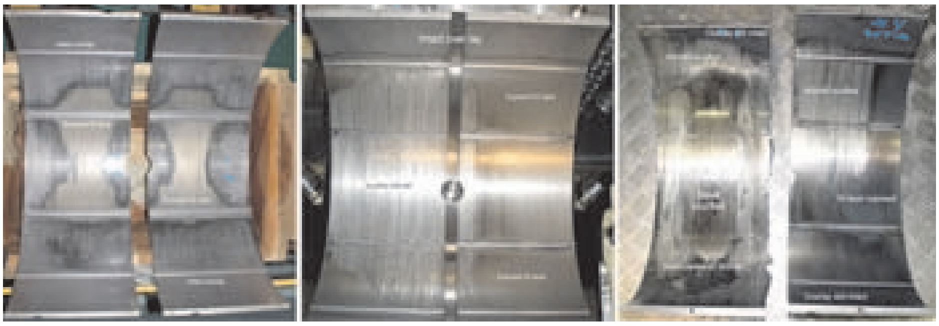


Fig. 55: Main bearing damage statistic

To increase margins against scuffing for K90 and K98 engines, we have introduced cermet coated rings, Nos. 1 and 4, Fig. 52. On the initial version of this new ring package a nickel-graphite running-in layer was applied. However, this running-in layer did not show sufficient stability. Therefore, we reverted to apply an Alu-coat as running-in layer on all 4 piston rings, as also shown on Fig. 52. The technology to apply an Alu-coat on top of a cermet coating is now available from all major piston ring makers.



Partially corroded overlay, not yet scuffed

Overlay completely corroded, away, Ni 100% exposed, partial scuffing between Ni-layer and pin

Overlay completely corroded, away, partly scuffing between Ni-layerand pin partly steel-to steel contact

Fig. 56: Crosshead bearing overlay corrosion

Bearings

Since the late 1990s a positive development with respect to main bearing damage has been seen. Despite the heavy increase in the number of main bearings on MC/MC-C engines, the number of reported damage remains at a constant low level, Fig. 55.

For AlSn40 crosshead bearings, we have had a number of reports (nine altogether) where overlay corrosion has been found. In most cases, this has ocurred on bearings where an interlayer of nickel has been exposed. It is well-known that nickel has bad tribological properties, and that there is a risk of scuffing between the bearing shell and crosshead pin, Fig. 56.

In all cases of overlay corrosion, excessive water in the system oil has been detected. If the oil system becomes contaminated with an amount of water exceeding our limit of 0.2% (0.5% for short periods), corrosion may start. A water content higher than 1% could lead to critical damage within few days of operation. A service letter has been sent out to inform (reinform) about this phenomenon. In this service letter, the lead content level in the system oil has also been devised as an early method of detecting overlay corrosion of crosshead bearings, Fig. 57. Also water in oil monitoring of the system oil are described in the service letter. Water in oil monitoring equipment is available from several sources for onboard use.

The following values for the lead content in the oil system can be used as a guideline:

0-4 ppm lead:
normal

5-10 ppm lead:
Inspect filters & crankcase for bearing debris, prepare inspection of crosshead bearings when convenient

>10 ppm lead:
Inspect filters & crankcase for bearing debris, prepare inspection of crosshead bearings as soon as possible

Fig. 57: System oil lead content guideline

Service tests for crosshead bearings with new synthetic coatings based on polymer, molybdenum disulphide/graphite have been concluded with good results, Fig. 58. This technology can be spread to other bearings than crosshead bearings where static friction is a limiting factor.

Bearing wear monitoring systems

In the past, the majority of bearing monitoring systems was temperature based. Even the compulsory oil mist detection system reacts to changes in temperature, although it is in a crude way and often at a late stage of the development of damage.

For all MAN B&W two-stroke engines, now and in the past, tribologically forgiving bearing material is/was used for the principal crank-train bearings. The materials are tin-based white metals (e.g. HMO7) and tin rich tin-aluminium (e.g. AlSn40). These materials excel in not developing destructive temperatures if a bearing fault develops within the lining material. Even if the oil supply is cut, the bearing element temperatures will not reach above the melting point of tin, which is far below the critical temperature for steel. That means for our standard bearing lining materials, severe damage to journal and housing is not expected to occur unless the lining is worn through and steel-to-steel contact occurs, Fig. 59.

All the temperature-based systems suffer from one basic shortcoming: late response to damage. In some cases, the response is too late to avoid severe mechanical damage, and in the case of oil mist detection only, severe damage will inevitably already have occurred when an oil mist alarm occurs.

Furthermore, with the normal oil splash temperature monitoring of crank pin and crosshead bearings, a major shortcoming shows if the oil supply is cut off. In such a case, the system may not react at all. If a bearing deteriorates slowly by a fatigue or slow abrasive process, the bearing temperature is very unlikely to be affected until the point where steel-to-steel contact occurs. From that point on, the major

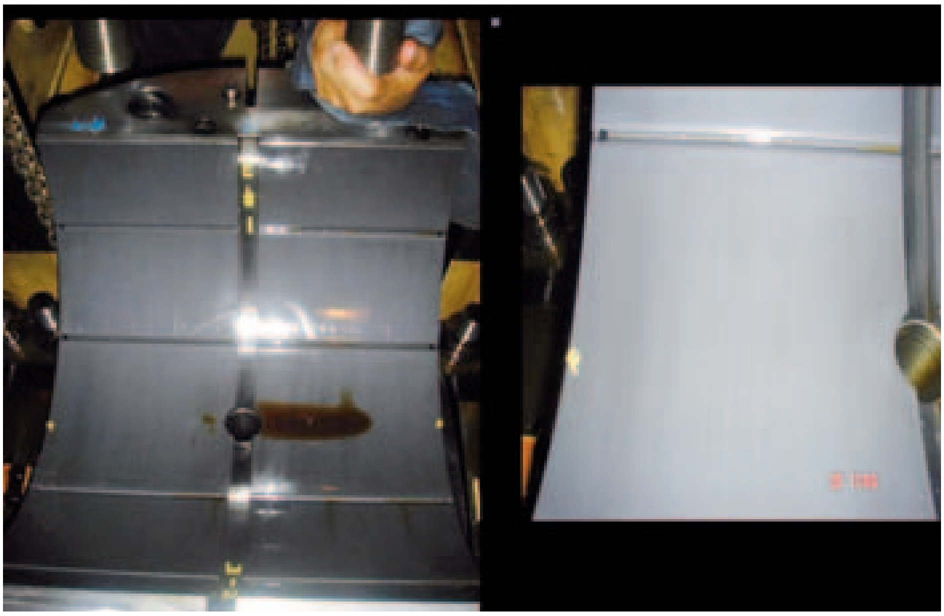


Fig. 58: Synthetic overlay on AlSn40 crosshead bearing shell after 10,000 hours

parts, such as the crankshaft, bedplate and connecting rod, are liable to suffer from severe damage, Fig. 60.

Based on the facts above, we have been working together with external partners on developing alternative bearing monitoring systems. The outcome of this development work is now entering the market as ‘Bearing Wear Monitoring’ (BWM) systems.

Present state of BWM

Together with our partners, we have developed BWM to be the logical bearing monitoring choice for two-stroke engines.

The advances in technology are mainly from the use of proximity sensor technology providing signals intelligently computed and digitally presented to computer hardware, from which a useable and easily interpretable interface is presented to the user.

The initial goal was to ensure an alarm if wear of 0.5 mm had taken place in any of our principal bearings. Therefore, we would have been satisfied to get a repeatable resolution of better than 0.2 mm, but our initial testing showed a far higher precision potential. The present precision is approx. 0.05 mm with excellent long-term stability. Therefore, we consider BWM to be more than just an alarm system, but a system also capable of providing long-term wear data at far better precision and repeatability than the manual vertical clearance measurements normally performed by the crew.

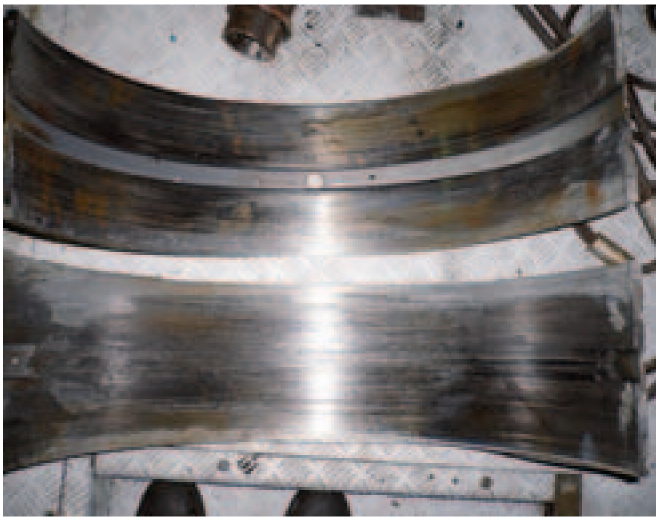


Fig. 59: Steel-to-steel contact in thin shell main bearing shells

Fig. 60: Main bearing journal as result of steel-to-steel contact

Advantages of BWM systems:

- Will in all cases alarm prior to steel-to-steel contact
- Easy and inexpensive to install from new or as retrofit (one bracket per cylinder unit)
- Provides active condition monitoring important for Condition Based Maintenance (CBM)
- MAN Diesel will omit scheduled open-up inspections of all bearings in instruction material if BWM is applied
- MAN Diesel will cut down on external inspections (deflection, clearance measurements, etc.) if BWM is applied.

Installation aspects

Installation is simple and quick, involving an absolute minimum of machining (drilling) in the engine. The BWM system monitors all three principal crank-train bearings using two sensors fwd/aft per cylinder unit placed onto the frame box, targeting the guide shoe bottom ends, see Fig. 61.

Scheduled open-up inspection of crank-train bearings

On a modern large bore two-stroke diesel engine, the reliability, particularly for critical components, has been very much improved compared to the past. Nevertheless, MAN Diesel wishes to maintain and improve reliability for the next generation of machines in spite of a higher specific output. We consider reliability a most important competitive parameter.

Traditionally, safe and reliable performance has been obtained through:

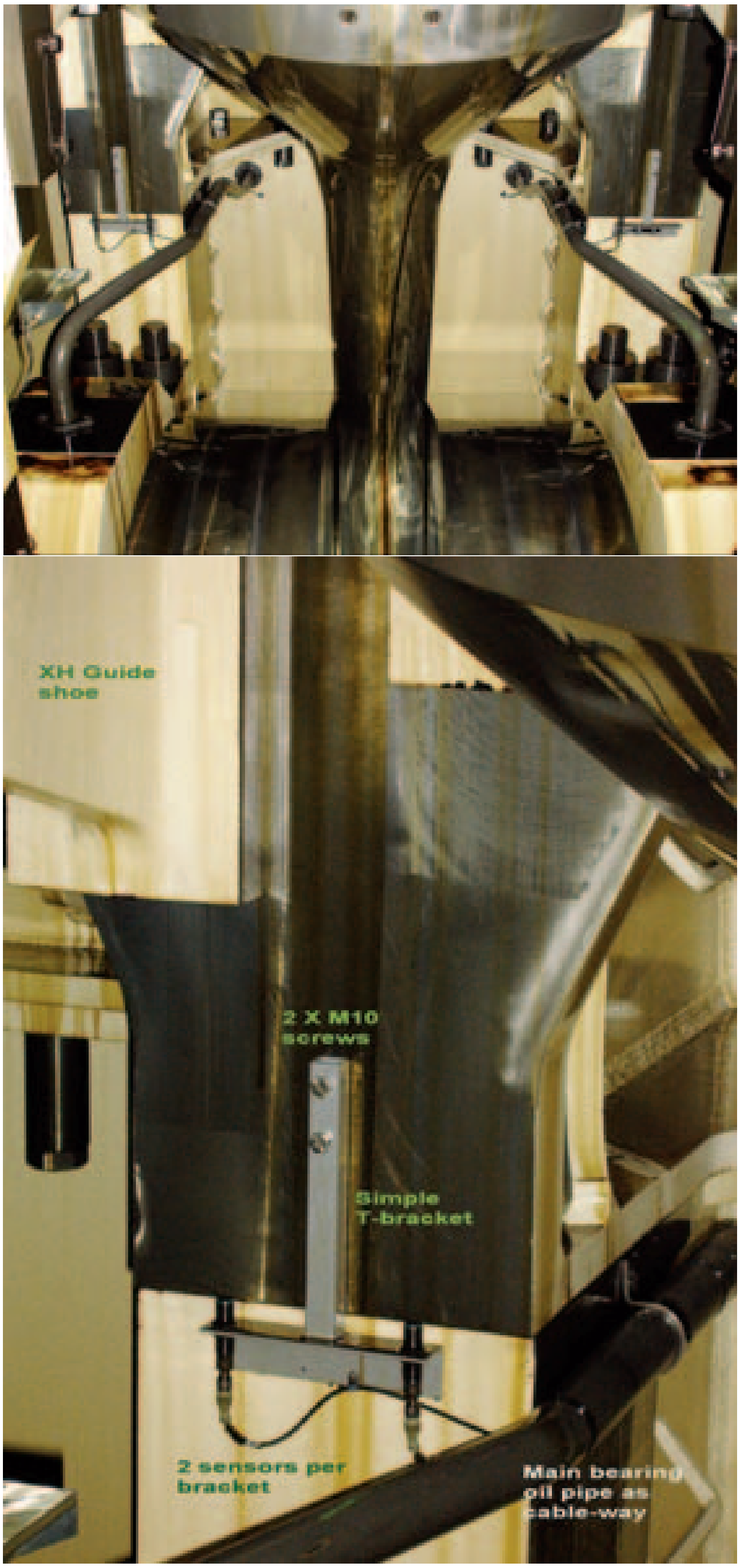


Fig. 61: BWM installation on a K98MC-C

Fig. 62: K98MC-C, crankpin bearing upper shell damaged by foreign material entered through the oil way

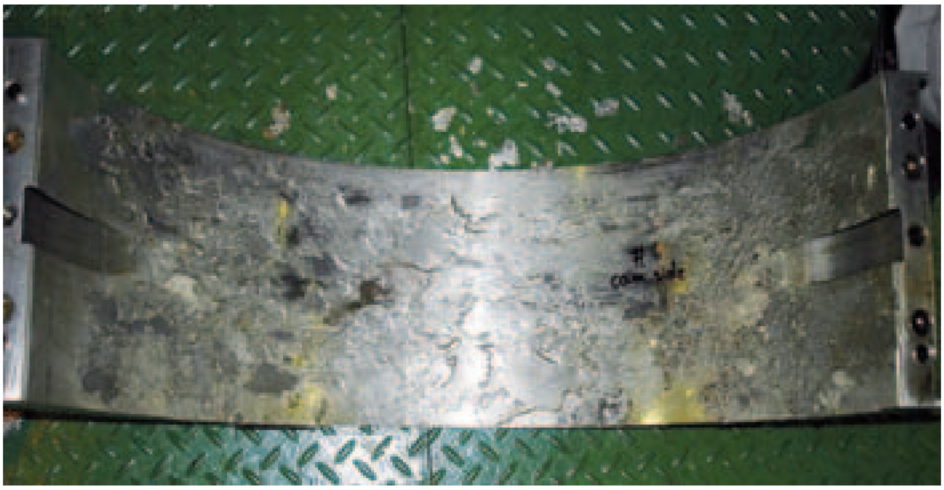


Fig. 63: Thick shell main bearing severely, but not yet critically, damaged. Bearing lining worn, but not yet worn away

1. Careful and conservative design

2. Precision-finished machined components, as opposed to past hand-finished components

3. Modern precise manufacturing methods in general

4. Effective quality control during manufacture

5. Clear instructions to owners to follow appropriate part replacement and overhaul/inspection schedules.

This five-legged strategy will continue, but in the future condition monitoring systems as basis for Condition Based Maintenance (CBM) may allow us to greatly increase the time between overhaul/ inspection of certain parts, which are basically designed to last the entire lifetime of the engine. With the introduction of effective bearing wear monitoring, we consider scheduled open-up inspections as obsolete.

First of all, constant monitoring of operating conditions and performance increases the chance of detecting a developing problem at an early stage. Secondly, experience shows us that some components most frequently fail shortly after an overhaul, due to incorrect reassembly, foreign particles being introduced, Fig. 62. Finally, servicing a part only when necessary, reduces the owner's maintenance costs.

For several years, we have been working on optimising maintenance schedules for the crank-train bearings. We are working in cooperation with owners and the classification societies towards a less open-up oriented maintenance schedule, and our future proposal for the maintenance schedule will reflect our intentions for this development. Several factors contribute in making further development possible.

The most important of course is the improvement in reliability over the years for the crank-train bearings, but on top of long standing indirect temperature-based systems, BWM takes condition monitoring to a higher order of predicting bearing damage before it becomes critical, Fig. 63.

In summary, MAN Diesel recommends the installation of the systems described above in new engines.

Conclusions

The reference list of ME engines, see Fig. 6, comprises engines right from the L42ME to the K98ME/ME-C, in a fairly even distribution among about 35 owners of tankers, bulkers and container ships.

The ME/ME-C engines have had a successful introduction in the market, and they are well accepted.

As with other products containing new technology, there has been some teething troubles, most of which have been eliminated quickly.

The service experience for the traditional range of MC/MC-C engines is characterised by a stable cylinder condition, a stable bearing performance and a general extension of the realistic time between overhauls.