Service Experience

Two-stroke Engines

Engineering the Future – since 1758.

MAN Diesel & Turbo

Contents

Service Experience Two-stroke Engines

Introduction

This paper gives an update on the latest service experience of MAN B&W two-stroke engines in both the electronically as well as the mechanically controlled versions.

Updates on achievements in relation to ME engines are given, and new challenges for the so-called Dot 2 version of the Tier II engines are touched on.

Low-load operation is still a hot topic and an update in this relation is included.

Fuel injection equipment and combustion chamber issues are also dealt with.

Furthermore, the new standard equipment, PMI autotuning, for automatic balancing and tuning of ME engines is introduced.

Early service experience with the S90ME-C9.2 and the G-type engines are not available at the time of writing. However, these items will be included in a later paper.

ME Related Issues

Since the introduction of the ME engines, we have given updates at regular intervals on the service experience related to the various parts of the ME electronic and hydraulic system. The system consists of controllers, actuators, sensors and operating panels. Furthermore, it is relevant to describe changes to ME-ECS software, hydraulic pipes and other hydraulic parts as well as aspects related to installation quality.

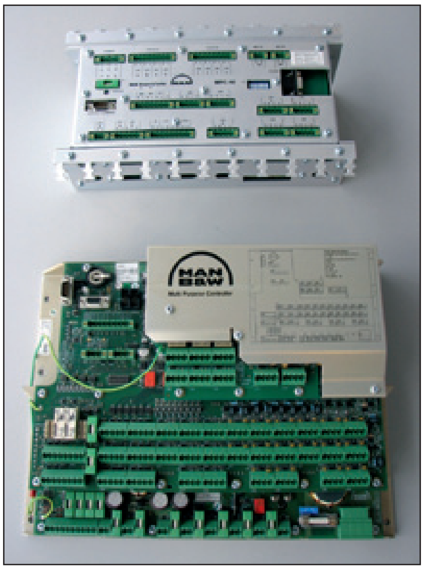

Fig. 1: MPC10 (upper) and MPC (lower)

In this paper, we focus on the following items/issues:

- MPC10 – new dedicated cylinder control unit (CCU)
- update of ELVA actuator
- position sensor on Eaton hydraulic pump
- new main operating panel: EC-MOP
- new production specification, example: electrical wiring in junction boxes
- New ME-ECS software for Dot 2 engine types
- new injection profile for 300-bar actuated engines, impact on high-pressure pipes.

MPC10 – new dedicated cylinder control unit (CCU)

A new MPC, the MPC10, has been developed as a dedicated CCU. The reason for this development is cost reduction and reliability improvements through fewer components and stronger mechanical design. Also, quicker screen termination omitting EMC cable glands makes the installation easier and safer. Fig. 1 shows the MPC10 and the classical MPC, which is still used as ECU, EICU and ACU.

Since the MPC10 was introduced in January 2012, more than 1,200 units have been produced until November 2012. Very few claims have been received for the MPC10 until now.

Update of ELVA actuator

The first 18 vessels of the ME type were equipped with two actuators, one for fuel injection (ELFI) and one for exhaust valve actuation (ELVA). All later ME engines have been equipped with the combined actuator, the FIVA valve.

Initially, the ELVA valve did not demonstrate an acceptable level of reliability and extensive modification was required. However, the troubles continued and were always related to the on/off control valve.

MAN Diesel & Turbo (MDT) therefore decided to introduce an alternative control valve (make: Moog) on all 18 vessels, after successful service testing of the Moog-ELVA valve. Fig. 2 shows the original ELVA valve and the new ELVA valve version.

All modified valves are delivered from MDT in Copenhagen, complete, tested and inspected, without any cost for the owners.

This example demonstrates MDT's commitment to solve problems and live up to high standards in relation to support to owners long time after expiration of the guarantee period.

Position sensor on Eaton hydraulic pump

Over the years, various feedback sensors have given rise to concerns about stable production quality and general reliability. However, these concerns have been overcome by a combination of increased focus on a constant high production quality and the introduction of design changes aimed at achieving more durable versions of the sensors.

Fig. 2: ELVA valves (left: original - right: new)

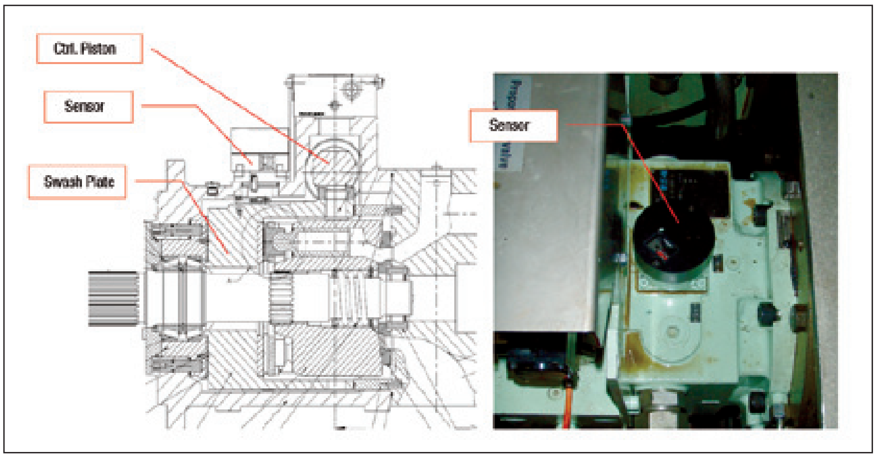

Fig. 3: Hydraulic pump sensor failure

The latest example of a sensor that has created troubles in relation to ME components is the position feedback sensor on the Eaton version of the main hydraulic swash-plate pumps. Fig. 3 shows the arrangement of the position feedback sensor on the Eaton pump. As illustrated in Fig. 3, the sensor connects mechanically via a coupling disc directly to the control arm for the swash plate.

In a number of cases, the coupling disc has broken, see Fig. 4, causing lack of control of the hydraulic pump and, thereby, reduced hydraulic power and reduced engine output.

New touchless position feedback sensors sensing on the control piston of the pump, Fig. 5, instead of on the swash plate control arm are being developed and tested in order to have a reliable feedback sensor system for the Eaton hydraulic pumps.

New main operating panel: EC-MOP

Over the years, we have used various makes and designs of the main operating panels (MOPs) for the ME engines.

Recently, we have introduced a new in-house designed MOP, the EC-MOP, see Fig. 6. The main advantages are:

- robust and reliable design
- low cost
- design owned by MDT, which makes localisation easy
- only one software image
- computer and screen in one unit.

Before releasing the EC-MOP for full-scale production in early 2012, it had undergone a long testing programme, including highly accelerated life testing (HALT), service testing and test for type approval. Furthermore, results from pre-production series have been carefully scrutinised.

Since April 2012, the EC-MOP has been the standard specified for all new engine orders, and several hundred EC-MOP units have been delivered from various electronics manufacturing services (EMS) providers. At the time of writing, no guarantee claims have been seen.

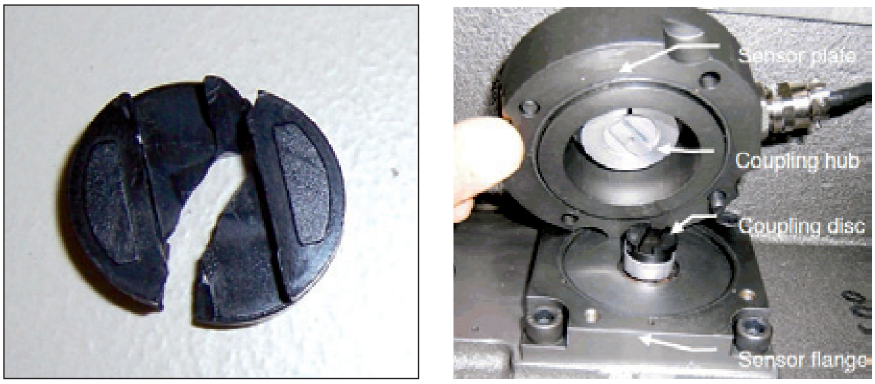

Fig. 4: Feedback sensor coupling failure

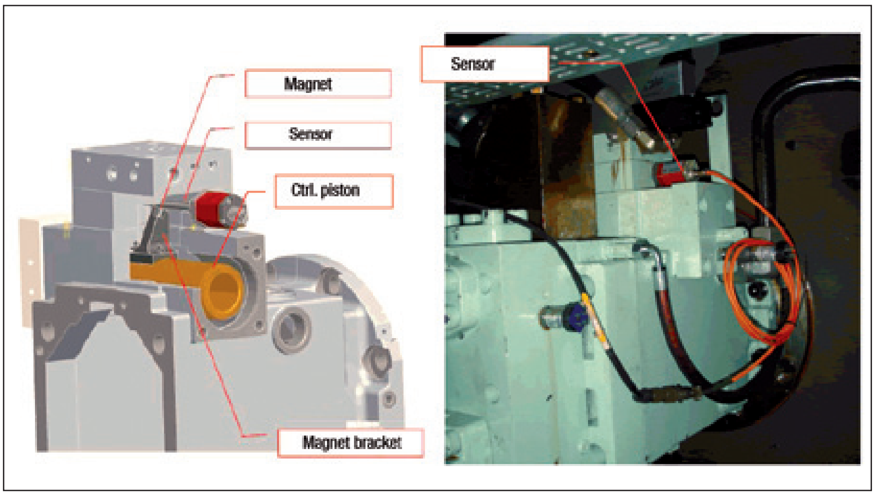

Fig. 5: New touchless feedback sensor

Fig. 6: EC-MOP

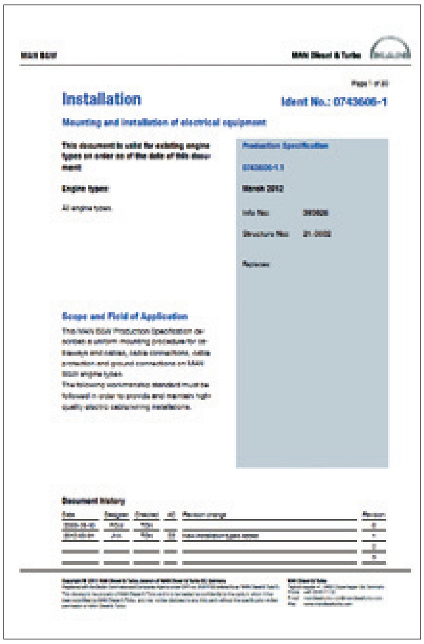

Fig. 7: Updated production specification

New production specification, example: electrical wiring in junction boxes

Cabling quality and general installation quality of cables and junctions of cables have been and still are a cause of failures related to the ME engines. In order to improve this situation, we issued an updated production specification in March 2012, see Fig. 7, focusing on updated installation types, new screen terminations and wiring in junction boxes.

One of the most common causes for electrical failures on the ME engines is isolation errors. Fig. 8 shows examples of installations in junction boxes that can/will cause isolation errors. The examples show wires touching interior surfaces of the junction box, causing rubbing between the box and wire due to vibrations, or wires getting wedged between the junction box and lid.

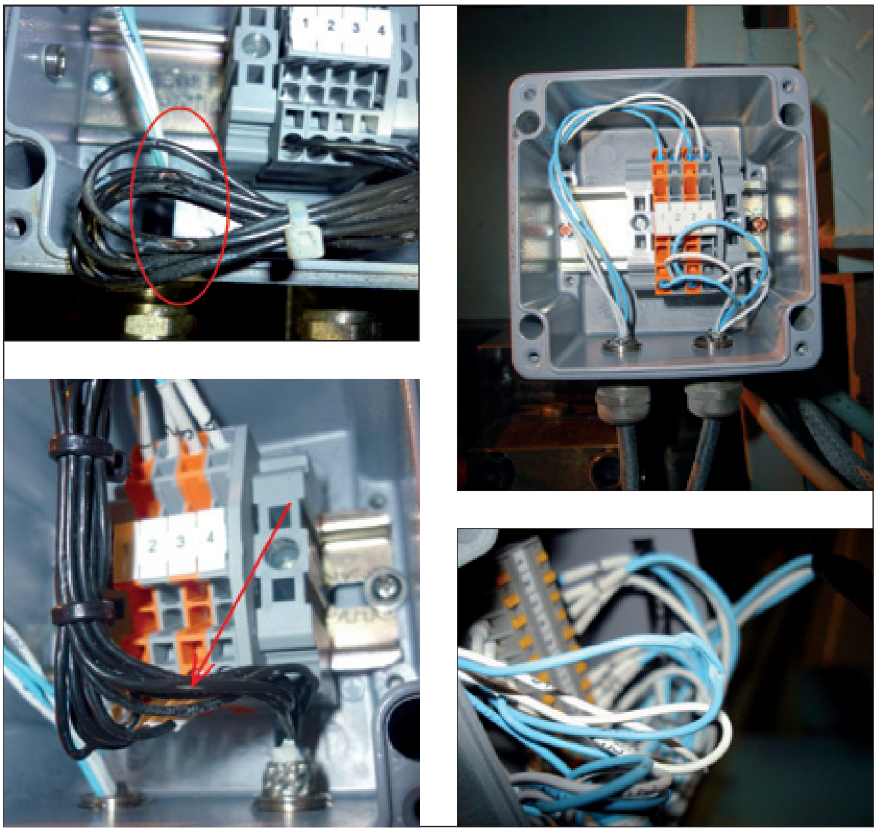

Fig. 8: Not optimal installations

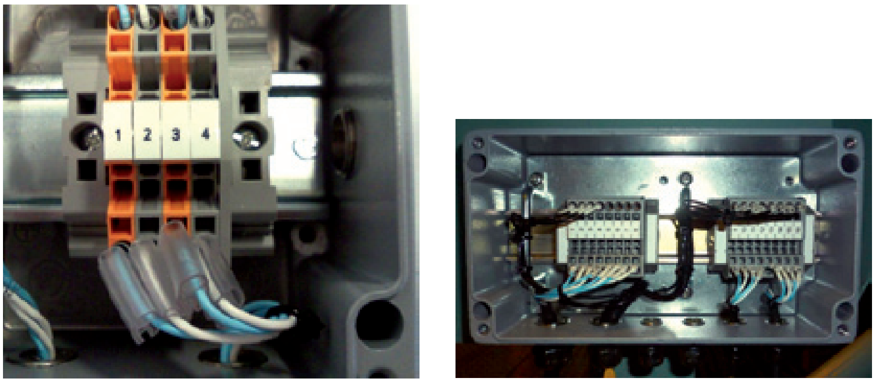

Fig. 9: Correct junction box installations

Fig. 9 shows a correct cable/wire installation in junction boxes. The characteristics of a correct installation are mentioned below:

- wires must be cut to correct length – excessive wire length is not acceptable
- wires must be tied securely, preferably to support bars
- wires must under no circumstances touch the interior parts of the junction box or the junction box itself.

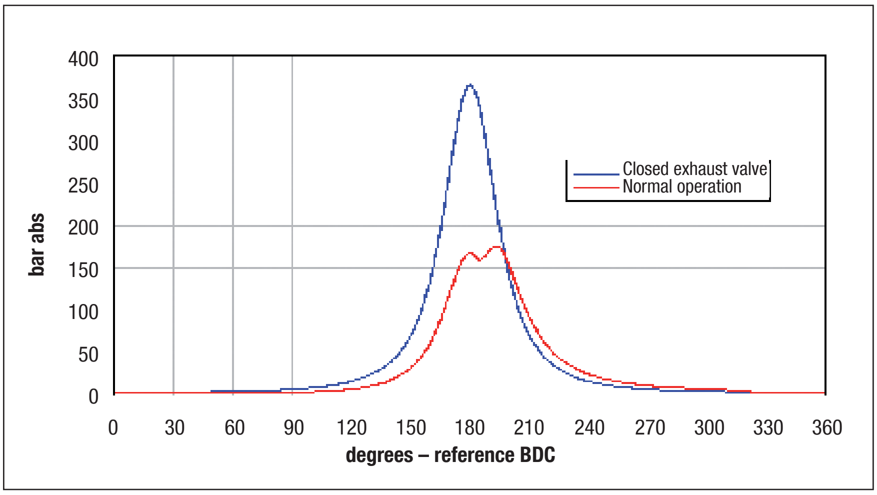

Fig. 10: 12S90ME-C9.2 running at 110% load

We are confident that with the new Production Specification in place we will see fewer failures on the ME engines caused by improper cabling installation.

New ME-ECS software to handle Dot 2 engines

Lately, we have experienced one case of cylinder cover lift on our S90ME-C9.2 engine series. The cover lift was caused by untimed exhaust valve closing, see Fig. 10. On Dot 2 engines with two-stroke Miller timing, this situation may cause excessive compression pressures at high loads (in this case 110% load on the test bed) leading to cylinder cover lift. Two-stroke Miller timing consists of high scavenge air pressure, late closing of exhaust valve and small compression volume.

The engines (MC/MC-C as well as ME/ME-C/ME-B) are designed in such way that gasses will escape in forward or afterward direction. Gas escapes towards the manoeuvring side or exhaust side are avoided by the design of a "guiding edge" on the cylinder cover.

In 2007, we designed a protection shield to protect against escaping gasses from the foremost and the aftmost cylinders. This protection shield is now the standard on all ME/ME-C/ME-B engines. On engines with divided crankshafts, protection shields are also the standard on cylinder covers adjacent to the crankshaft division.

Previously, cylinder cover lifts could only take place as a result of a large untimed fuel injection. In order to limit the occurrence of this extremely rare event even further, a so-called window function was introduced in the ME/ME-C control system in 2007.

The additional reason for cylinder cover lifts (excessive compression pressure due to malfunction of an exhaust valve) has resulted in work on a modified control strategy in case of malfunction of an exhaust valve. This strategy involves modified software as well as minor modified exhaust valve hardware. These modifications were ready in April 2013 and have solved the problem.

New injection profile for 300-bar actuated engines, impact on high-pressure pipes

On our 300-bar hydraulic pressure ME engines, we have experienced cases of malfunction of the fuel injectors. This matter has been cured by introducing softer closing profiles for fuel injection, see Fig. 11. Needle landing speeds have typically been reduced from above 4 m/s to below 2.5 m/s. The range of needle landing speeds below 2.5 m/s is a well-known range on our MC/MC-C engine types.

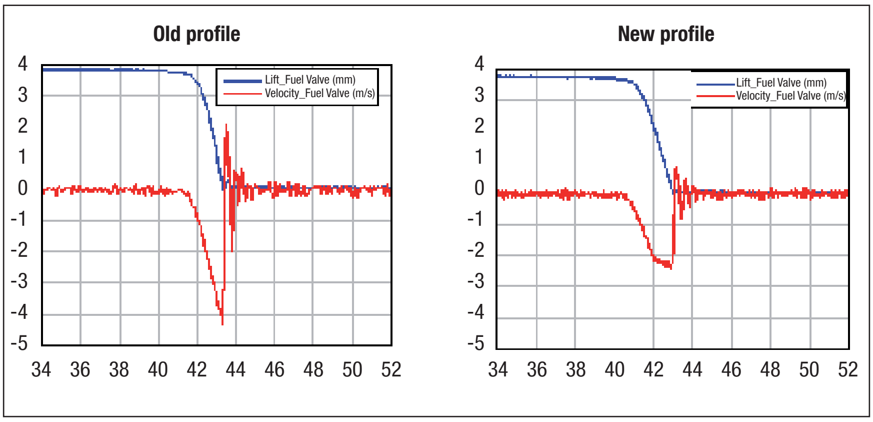

Fig. 11: Fuel valve needle landing speeds

Fuel injection closing profiles also have an impact on the structural integrity of the high-pressure fuel pipes. This is illustrated in the following with an S80ME-C9.2 engine design as the example. Fig. 12 shows the first design of high-pressure fuel pipes on the left-hand side and the second design with the so-called sigma pipe between the fuel booster and distributor block on the right-hand side.

Strain gauge measurements on the high-pressure pipe ends have shown the improvements of dynamic stresses as a function of the fuel injection closing profile. Two sets of strain gauges have been mounted on the upper end of the sigma pipe, one on the upper side and one on the underside. Measurements have then been made to compare dynamic stresses (strains) with the original injection closing profile and the softer injection closing profile.

Fig. 13 shows strains measured at 100% load with the original injection closing profile. It can be noted that large dynamic bending stresses are introduced when injection is closing.

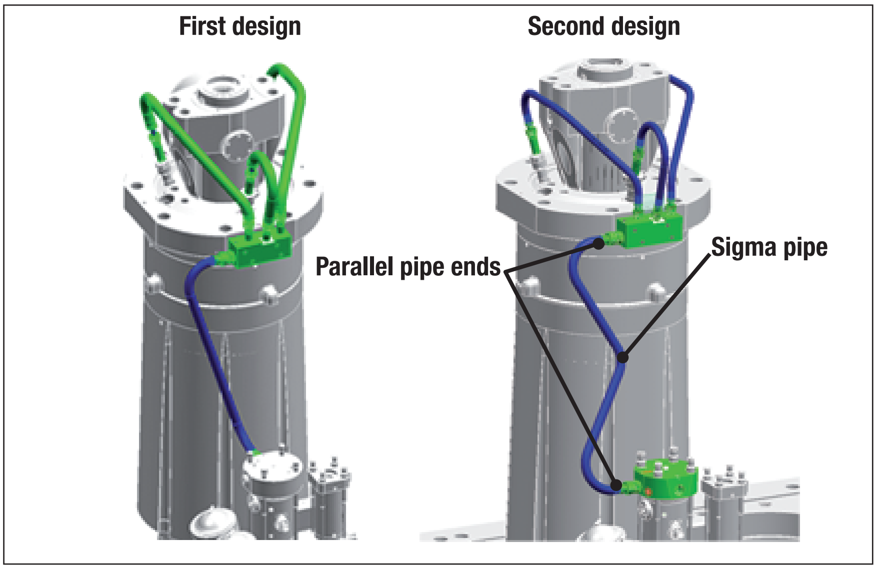

Fig. 12: 6S80ME-C9 high-pressure pipe

Fig. 14 illustrates similar strain measurements with the softer injection closing profile. And even if dynamic high frequency bending stresses are still introduced when injection is closing, these stresses are clearly seen to be reduced significantly. Fig. 15 shows a summary of various measurements done on the high-pressure fuel pipes on the S80ME-C9.2 engine. Dynamic stresses are seen to be reduced to a level less than 60% with the sigma pipe design and the softer injection closing profile.

A third design of the common high-pressure fuel pipe from the fuel buster to the distributor block has been introduced on coming S80ME-C9.2 engines. Fig. 16 shows the future so-called stick pipe design, which has

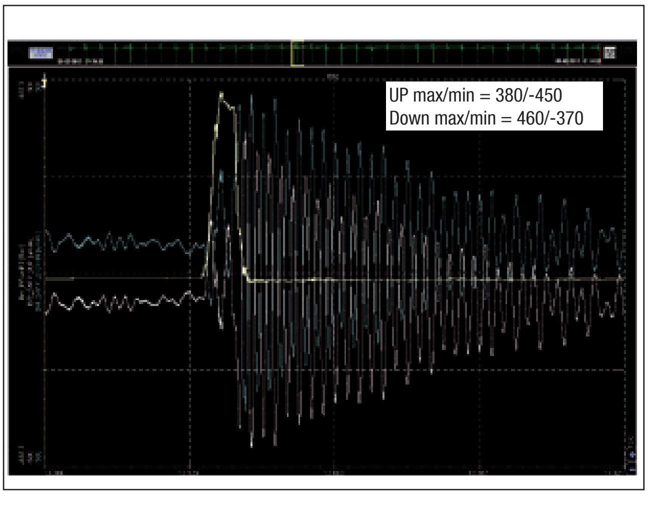

Fig. 13: 6S80ME-C9 300-bar 100% load – original injection closing profile

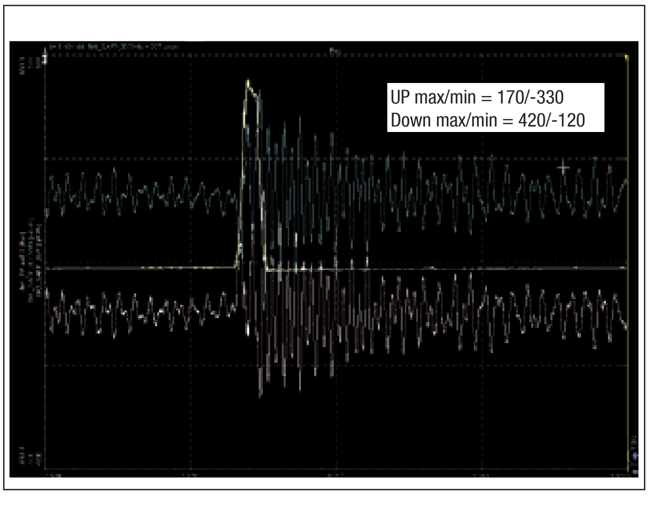

Fig. 14: 6S80ME-C9 300-bar 100% load – softer injection closing profile

Pipe	Injection profile	Dynamic strain micro-strain	Relative Dynamic Stress %
Original pipe – no support	Old	+/- 440	100
Original pipe – with support	Old	+/- 435	99
Original pipe	New	+/- 325	74
”Sigma” pipe	Old	+/- 415	94
”Sigma” pipe	New	+/- 250	56

Fig. 15: Measurements from 100% load

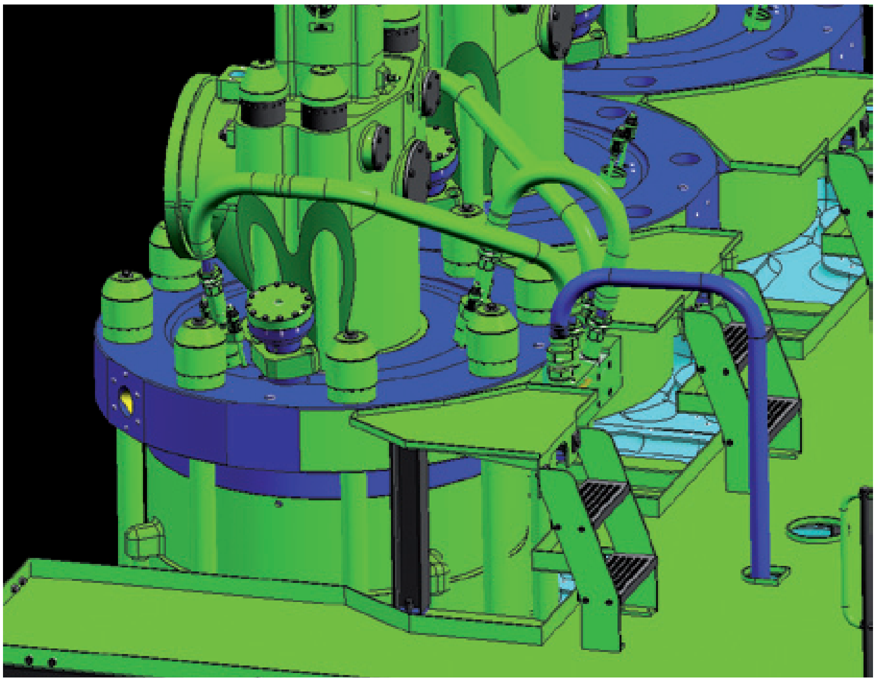

Fig. 16: New stick-type pipe design

a dynamic stress level reduced by a further 30% compared with the sigma pipe design.

Low-load Operation – A 2013 Update

Since the start of the worldwide financial crisis in the autumn 2008, low-load operation, or slow steaming, has become the standard operation mode for many owners operating MAN B&W two-stroke engines. In the early days of slow steaming, mainly container vessel operators requested to operate at low loads. Today, also operators of tankers, bulkers, etc., are beginning to operate continuously at low load.

Presently, we are collecting data to support operation as far down as 5-6% continuous load for VLCCs having a barred speed range at around 10% load.

In late 2008, we issued a service letter dealing with continuous low-load operation down to 40% load, and in May 2009, we were ready to support continuous low-load operation down to 10% load. Nearly all service experience with continuous low-load operation since then has been positive. The application of fuel injection valves of the slide valve type is very important for this success. Slide-type fuel valves significantly reduce fouling of exhaust gas ways, especially when operating at low load.

Soon after it became normal to operate at extremely low loads, the request to optimise running at low load surfaced. This can be supported in two ways on MAN B&W two-stroke engines:

1. increase the scavenge air pressure at low and part loads
2. reduce the cylinder oil feed rate at low load

Most elegantly, this is supported on the electro-hydraulically controlled ME-engines. The ME engine control system (ECS) is designed to control variable turbine area (VTA) turbochargers, exhaust gas bypasses (EGBs) and flexible turbocharger cut-out systems.

For engines in service, the flexible turbocharger cut-out system with control of so-called swing gate valves has become a retrofit solution applied in many cases on engines with 2, 3 or 4 turbochargers.

Today, optimisation of the cylinder oil feed rate is also requested for extreme low-load operation in the load range from 25% and down to 10%. Service experience on this issue is described in a service letter on low-load operation from June 2011. We have tested several engines' running load proportional lubrication reduction of the cylinder oil feed rate all the way down to 10% load with very good results. In this way, it is possible to save half the cylinder oil at loads in the range of 10-15%. This is illustrated in Fig. 17.

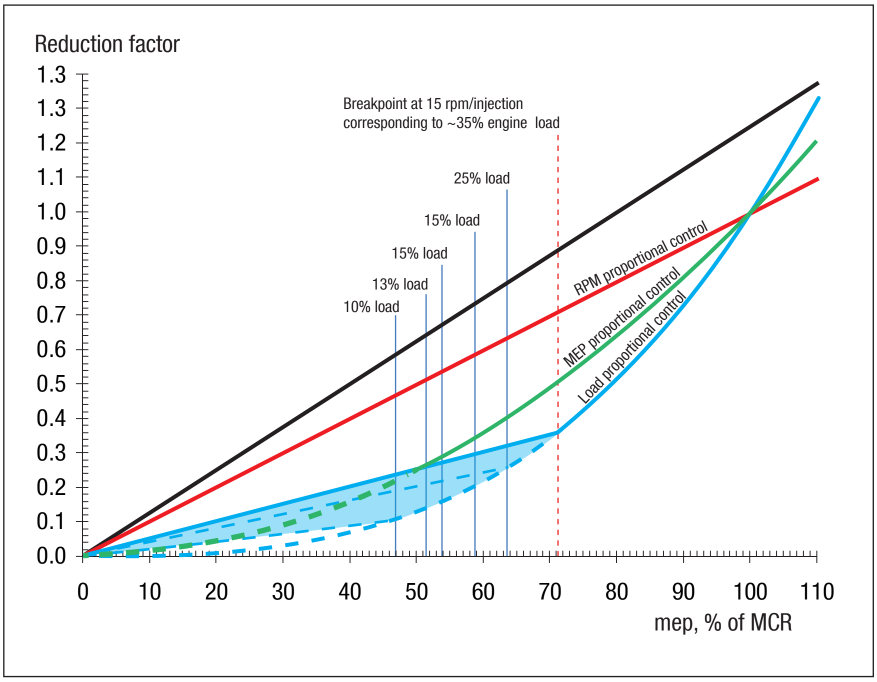

Fig. 17: Alpha Lube low-load optimisation

For ME engines, this changed adjustment can be accomplished simply by making parameter changes in the ME ECS software. For mechanical engines with Alpha Lubricators, a rebuild of the lubricators is required.

As for low load operation in general, a few issues listed below should be taken into consideration:

- K98 crosshead bearings when running in T/C cut-out mode
- cylinder liner cold corrosion when running in T/C cut-out mode, especially when one out of two T/Cs cuts out
- exhaust valve burn away on spindle lower face.

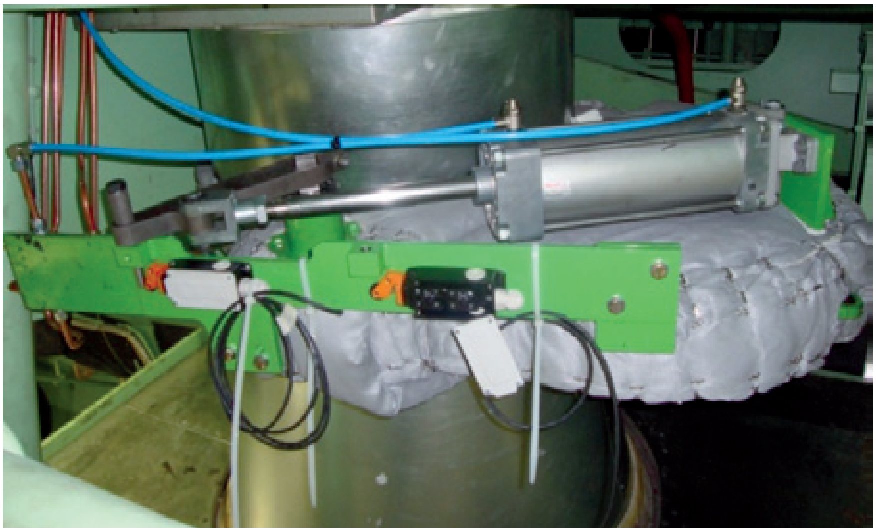

Fig. 18: T/C cut-out valve (compressor side)

K98 crosshead bearings in T/C cut-out mode

By now, we have gained experience from around three years of operation on K98 engines optimised for low-load by means of turbocharger cut-out, either by permanent installation of blind flanges to cut-out one turbocharger or by installation of flexible swing-gate cut-out valves, see Fig. 18.

Due to the changed balance between inertia forces and gas forces, we will get an increased load on the crosshead bearing shells in T/C cut-out mode with the increased gas pressures at low rpm. On the K98 engine, this has resulted in minor slow-developing fatigue damage on the central pad in the lower crosshead bearing shell, see Fig. 19.

Modified designs of crosshead bearing shells are currently being tested in service. However, these will not be concluded quickly as the development of fatigue damage takes two years typically. Fig. 20 shows elasto-hydro dynamic (EHD) bearing calculations of one of the designs being tested, i.e. the design with increased circumferential distance between the axial oil grooves. Larger oil

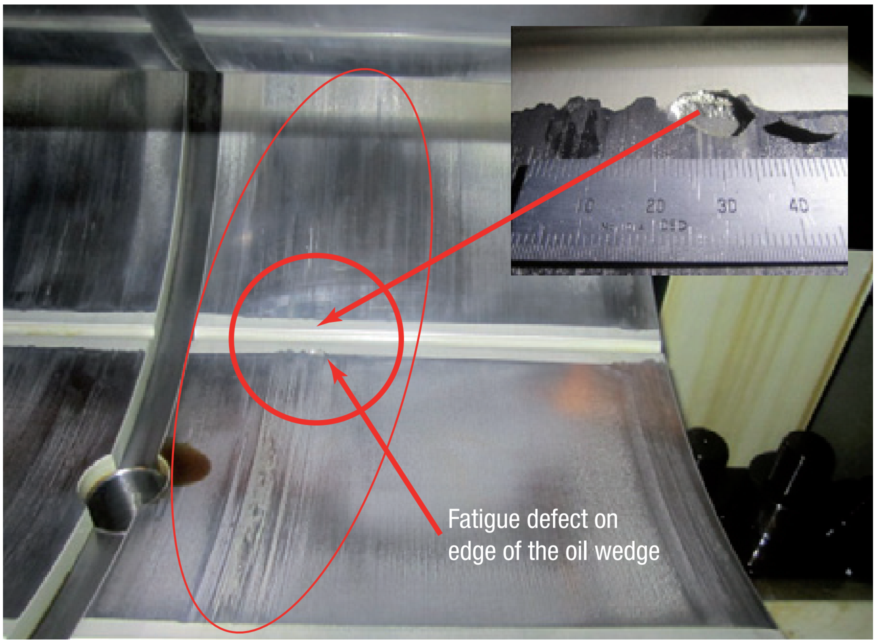

Fig. 19: K98 crosshead bearings and T/C cut-out

film thicknesses as well as lower oil film pressures have been calculated.

We have issued a Circular Letter to owners and operators about K98 engines with instructions on how to inspect and assess the crosshead bearing condition. We also assist owners in assessing bearing damage on a case-to-case basis.

We definitely advise owners to continue low-load operation with T/C cut-out despite of the cases of minor fatigue damage on the crosshead bearing shells. A 12K98 engine operating, on average, at 40% load will save approx. 1,000,000 USD/year by using T/C cut-out. This saving is so significant that we continue recommending the use of the T/C cut-out mode.

However, it is important to note that normal inspection for white metal in the crankcase should still be carefully observed. Also, we underline that extra open-up inspections of crosshead bearings are not recommendable. In general, crosshead bearings should only be opened if external signs of damage are found. To supplement normal inspections, we have instead developed a method to do additional inspections by endoscopic method. We recommend using this method, for example before a scheduled dry docking of a vessel with K98 and T/C cut-out.

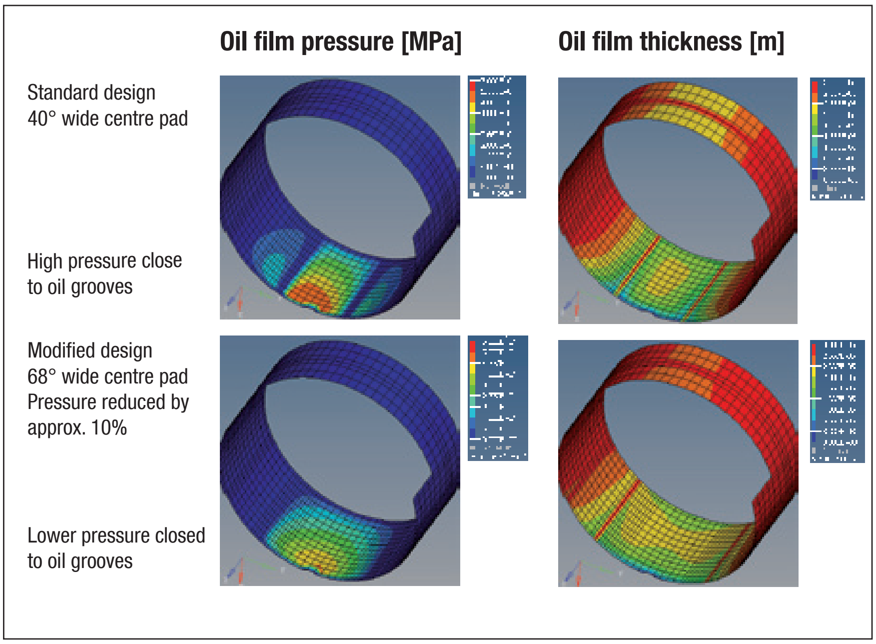

Fig. 20: K98 crosshead bearings and T/C cut-out: modified bearing shell design

Cylinder liner cold corrosion in T/C cut-out mode

We have, in a few cases, seen excessive cold corrosion in the top of cylinder liners when operating at low load in T/C cut-out mode. This has been seen especially in cases where one of two T/Cs has been cut out. In such a case, a very large scavenge air pressure increase is obtained at low load, resulting in rather high pressures and low temperature exposure for the cylinder liner top.

As a countermeasure, the jacket cooling water bypass was introduced, as described in the section of this paper related to cold corrosion in general. On K98 engines, the cooling water bypass can be arranged as shown in Fig. 21.

With this system, normally up to 75% of the jacket cooling water will be by-passed the cylinder liner cooling bores.

If cold corrosion is observed at low load, we do not recommend to lower the cylinder oil feed rate by changing breakpoint from rpm-dependent lubrication (Fig. 17). On the contrary, in such cases new cylinder oil feed rate guidelines must follow an increased breakpoint for rpm-dependent lubrication at 50% load (see also the section on cylinder oil guidelines).

Exhaust valve burn away at low load

For some engine types, low load operation means an increase in the exhaust valve temperature in the 30-45% load range. In this load range, just above cutting-in of the auxiliary blowers, exhaust valve spindle temperatures are known to be rather high. This increase in the exhaust valve spindle temperature may reduce the time between overhaul for the exhaust valve spindle and, since overhaul (rewelding) is recommended up to two times only, this will also reduce the lifetime of the exhaust valve spindles.

The above was mentioned in our first low-load operation service letter (SL08-501/SBE, October 2008), and it will require somewhat more frequent inspection intervals to be able to judge the so-called "burn-away" of the exhaust valve spindle disk.

Fig. 22 shows an exhaust valve spindle disk from an S60MC-C8 operating at low load for a long time. The burn-away level after 26,000 hours is 11 mm. This is above the maximum of 9 mm burn-away for reconditioning of the spindle.

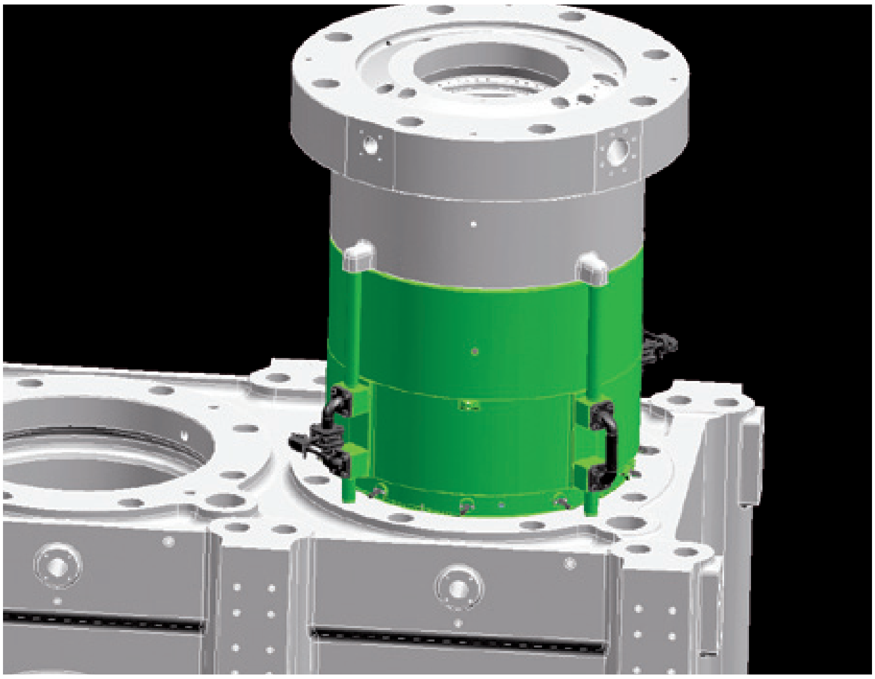

Fig. 21: Jacket cooling water bypass on K98 engine

S60MC-C8: Running hrs. 26,000
Burn away: > 11 mm (max. 9 mm)

Fig. 22: Burn-away of spindle disc during low-load operation

Engine type	Measured burn away	Running hrs.	Measured burn away rate	Reduction in lifetime of spindle
K98ME	9 mm	14,000	0.64 mm/1,000 hrs.	35%
S60MC-C	11 mm	26,000	0.43 mm/1,000 hrs.	35%
K98MC-C	7.5 mm	15,000	0.50 mm/1,000 hrs.	15%
S90MC-C8	14 mm	15,000	0.93 mm/1,000 hrs.	61%

Fig. 23: Burn-away on exhaust valve during low-load operation

So in this case, earlier inspection of the exhaust valve could have made reconditioning possible.

Fig. 23 shows examples of burn-away on exhaust valve spindles from various engine types. Based on these measurements, the burn-away rates can be calculated and the reduction of the exhaust valve lifetime can be estimated. However, MAN Diesel & Turbo recommends maintaining low-load operation, as the huge fuel oil savings can easily pay for the extra wear and tear of the exhaust valves.

We are preparing a service letter on exaust valve condition during low-load operation. Examples showing the influence of T/C cut-out, see Fig. 24, any operation down to 10% load as well as the cut-in point for the auxilliary blowers, see Fig. 25, indicate that strict guidelines cannot be given when considering low-load operation in general. The exhaust valve condition must be evaluated on the basis of inspections.

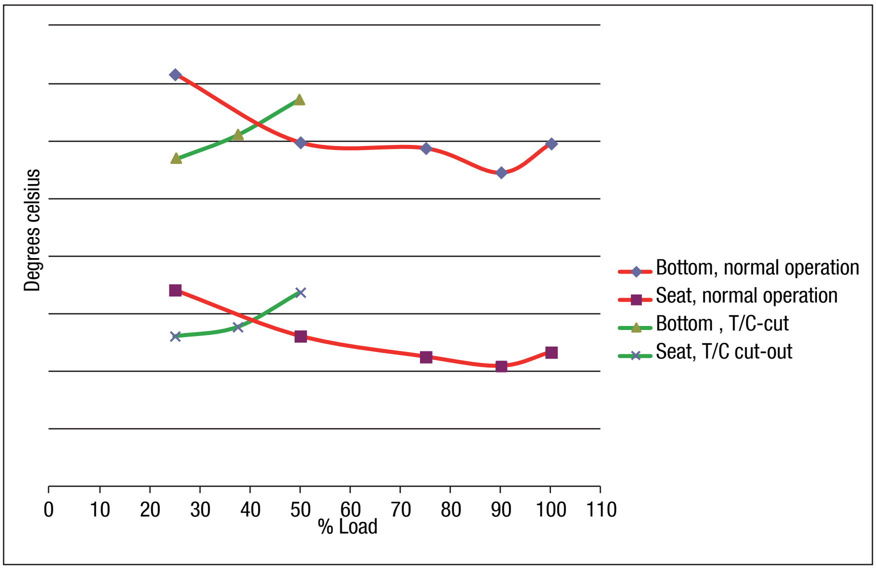

Fig. 24: Low-load operation with and without turbocharger cut-out on an 8K90MC-C

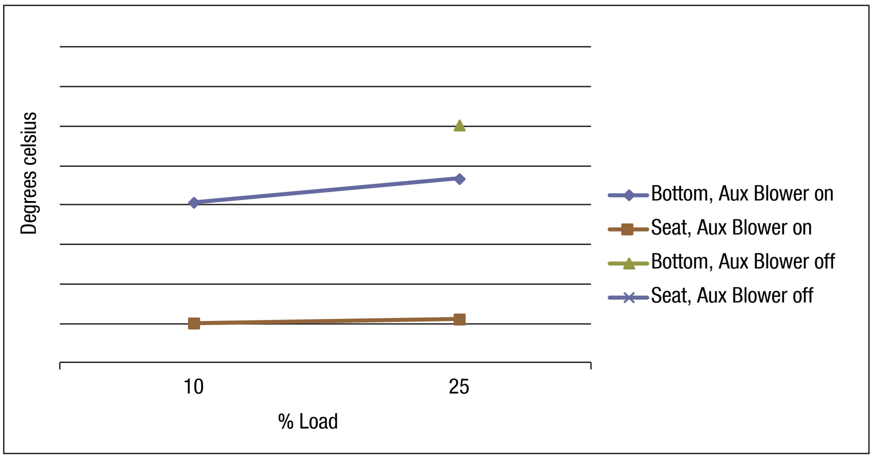

Fig. 25: Influence of auxiliary blower cut-in during low-load operation on a 10S90MC-C9.2

Cylinder Liner Cold Corrosion

Recently, we have seen an increased number of cases of cold corrosion attacks in cylinder liners on the newer generation of engines. The reason for these cases varies from uneven temperature distribution in circumferential direction in the top of the liner to various combinations of low-load operation and new engine tuning methods increasing the pressures in the combustion chamber at low load.

In the following, three examples of cylinder liner cold corrosion are described as well as the solutions to the problems.

Case one: S70ME-C7 with semi-high topland (SHTL)

In a number of cases, high oval cylinder liner wear has been reported after relatively few operating hours. One example can be summarised as follows where liner calibration after approx. 10,000 hours revealed:

- max. wear in forward-aftward direction: 0.7 mm
- max. wear in exh.-manoeuvring direction: 2.2 mm

From the above, it can be seen that cylinder liner wear is normal in one direction (forward-aftward) and excessive in the other direction (exhaust-manoeuvring). The oval wear leads to too early wear-out of the cylinder liner and it also shortens the time between piston overhauls due to a high piston ring wear and poor gas-sealing condition.

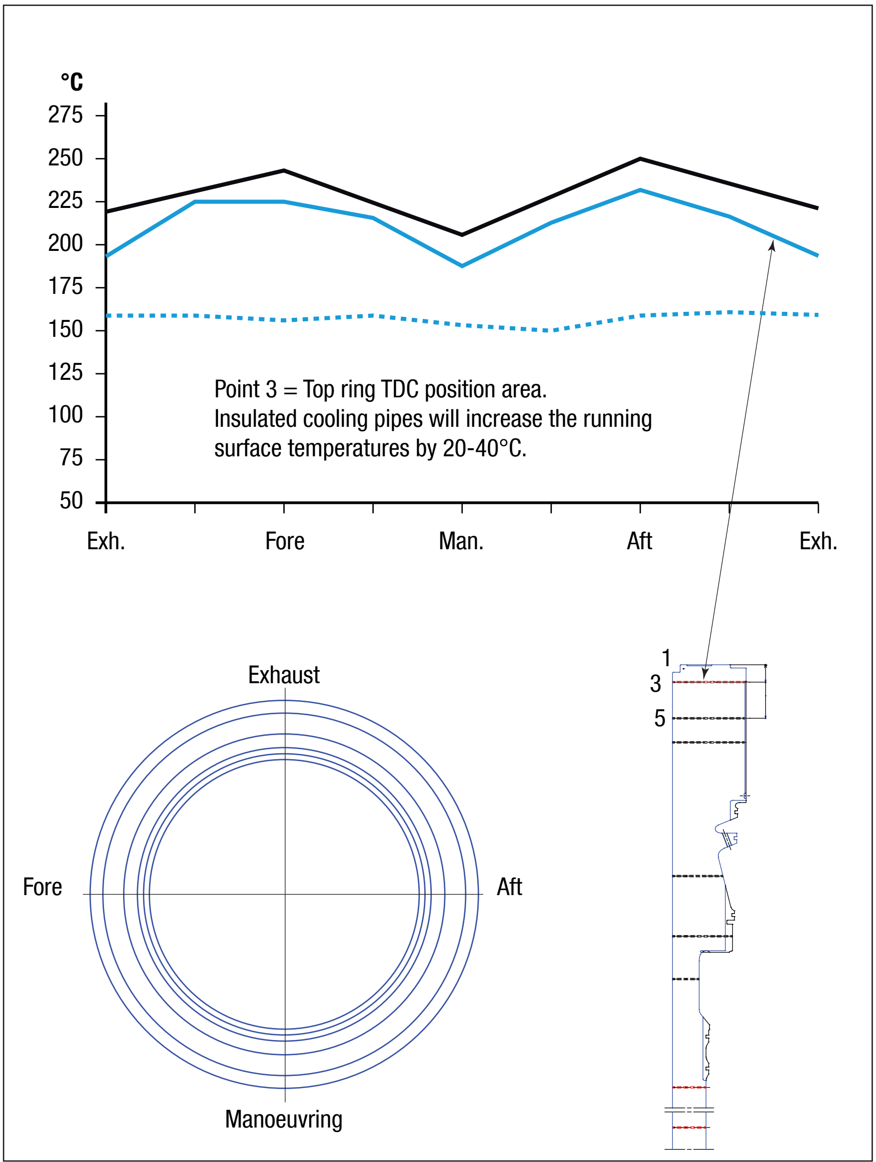

Fig. 26: S70 Semi-high topland (SHTL): circumferential temperature distribution

The reason for the oval wear is found when looking at the circumferential temperature distribution in the top of the cylinder liner measured on the test bed on the prototype engine. Fig. 26 shows what was measured around the circumference of the cylinder liner. In forward and afward direction, 225ºC was measured, whereas in exhaust and manoeuvring direction only 190ºC was recorded. This gives sufficient protection against cold corrosion in one direction, but not in the other direction.

The cylinder liner design of the bore-cooled type is illustrated in Fig. 25. Cooling bores in the top of the liner are evenly distributed around the circumference. Pipes are inserted in the cooling bores in order to distribute water to the very top of the cooling bores. A version of these pipes is insulated towards the liner surface. This will restrict cooling water closest to the liner surface, and it will also limit the total amount of water flowing through each cooling bore.

By inserting insulated cooling pipes in the cooling bores in exhaust-manoeuvring direction, see Fig. 28, we can increase the temperature on the liner surface in this direction without changing the situation in forward-aftward direction.

Cylinder liner wear measurements have confirmed the above-described change to be effective against excessive (oval) liner wear, see Fig. 29. Trend curves on cylinders with 2 x 5 insulated cooling pipes are well within acceptable limits.

Case two: S70MC-C7 with high topland (HTL)

On a series of 6S70MC-C7 engines, cylinder liners with 30 cooling bores were supplied instead of, as specified, cylinder liners with 24 cooling bores. This led to excessive cylinder liner wear caused by cold corrosion attacks in the upper part of the cylinder liners.

One, in theory, easy countermeasure would be to exchange the cooling pipes with insulated ones. However, in practice this is a rather time consuming operation. Therefore, an alternative more-easy-to-retrofit solution has been developed for this series of engines in order to save as much as possible of the cylinder liner lifetime.

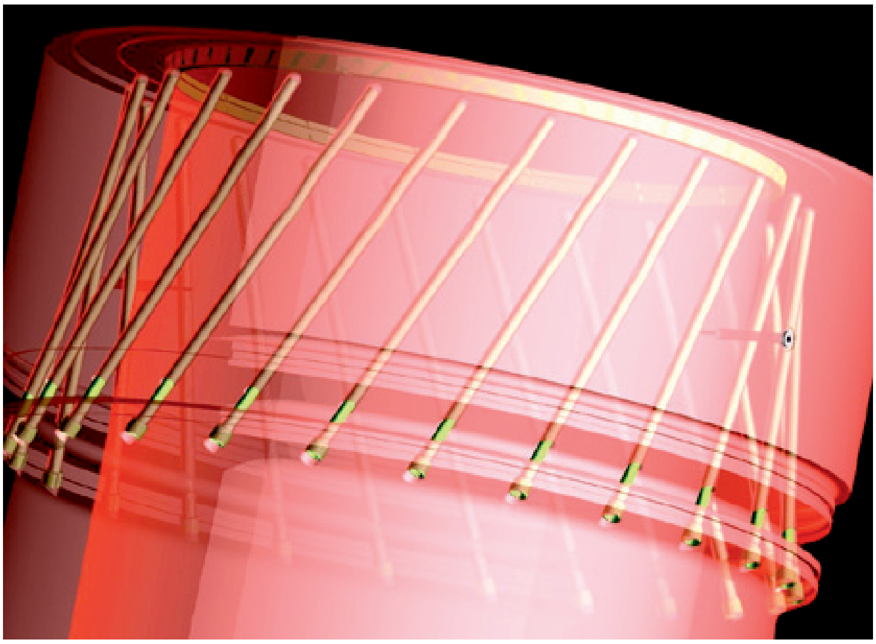

Fig. 27: S70 semi-high topland (SHTL): cooling bores

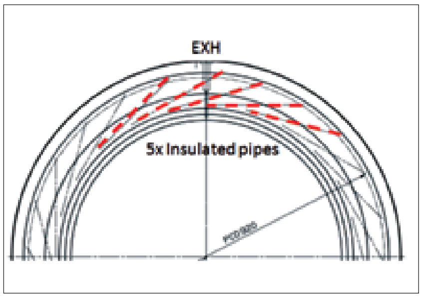

Fig. 28: S70 semi-high topland: insulated cooling pipes

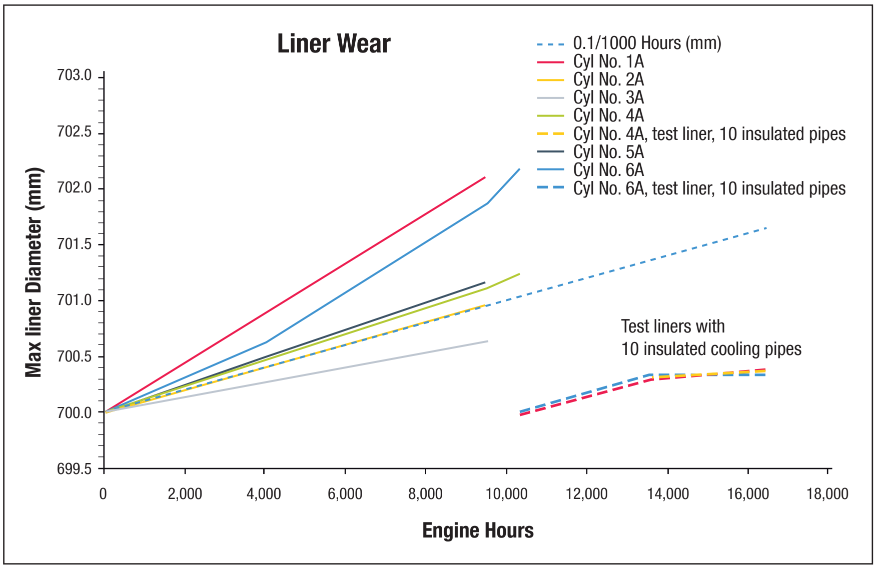

Fig. 29: S70 semi-high topland (SHTL): cylinder liner wear

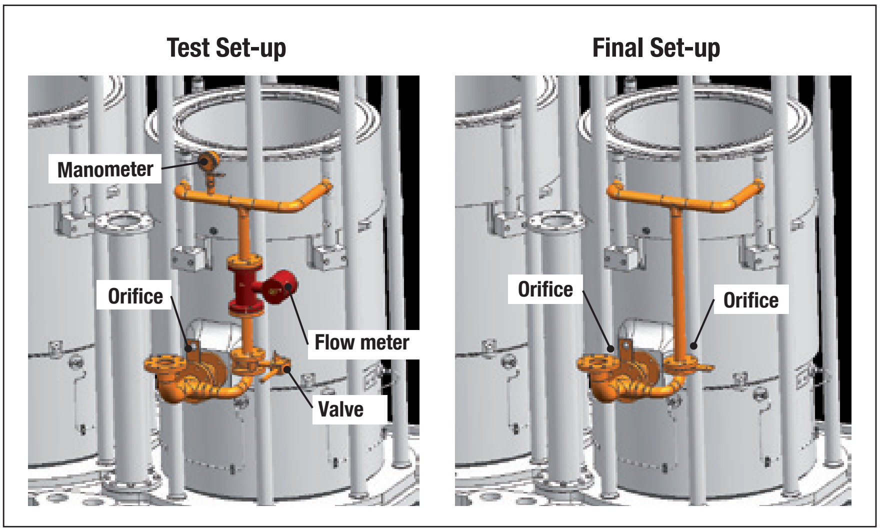

Fig. 30: S70M7/8 jacket cooling water bypass

Fig. 30 shows a jacket cooling water bypass that has been shop tested to determine the correct amount of cooling water to be bypassed. The test setup is shown on the left-hand side of Fig. 30. The final setup for service is shown on right-hand side of Fig. 30. The amount of bypassed water is determined by an orifice.

Bypassing 75% of the jacket cooling water will increase the liner wall temperature by 12ºC. In addition, the jacket cooling water outlet temperature is increased to 90ºC. All together, a 16ºC increase of the liner wall temperature is achieved, and the first indications from online drain oil analyses show that the cold corrosion issue has been solved.

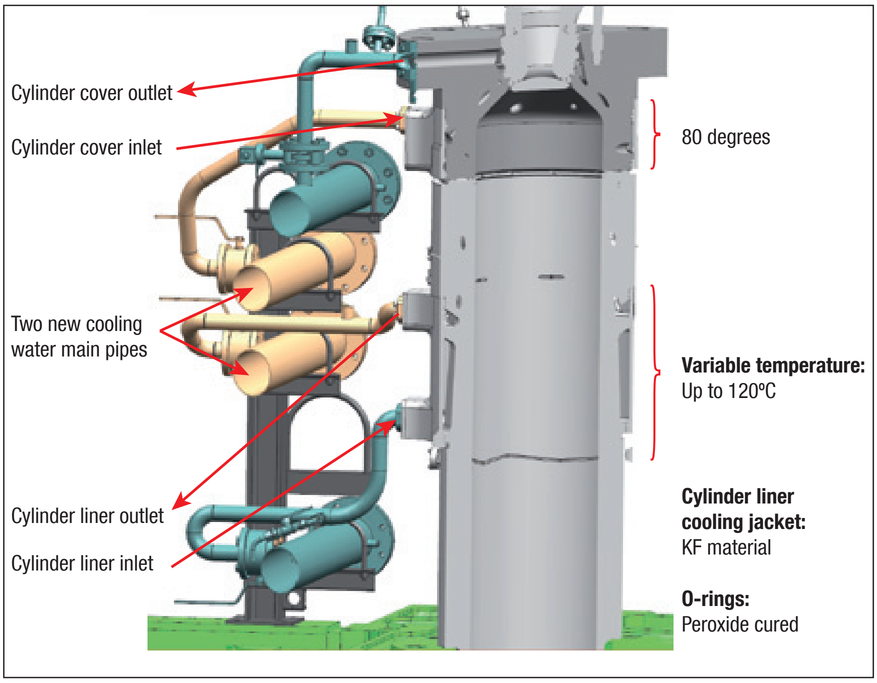

Fig. 31: Controlling corrosive wear: 4-pipe jacket cooling water system

A more active bypass system is now being prepared on our new generation of G-type engines and future other Mark 9 engines. The system shown in Fig. 31 consists of two extra cooling water pipes along the engine. An extra pump and an extra control valve ensure up to 120°C on the cooling water for the cylinder liners while maintaining 80-90°C on the cover and exhaust valve. A high temperature on the cylinder liner is only maintained in the low and part load range.

Case three: S50ME-C9.2

For the new S50ME-C9.2 engine, cold corrosion in the upper part of the cylinder liner was discovered already after sea trial, see Fig. 32. Such an early sign of cold corrosion called for the immediate action of introducing a design change to counteract any further development towards high corrosive liner wear values.

An inspection on a sister vessel after approx. 1,000 hours of operation, see Fig. 33, confirmed the signs of corrosion in the upper part of the cylinder liner. Liner calibration confirmed too high wear levels.

Liner calibration after approx. 1,000 hours:

- max. wear in forward-aftward direction: 0.26 mm
- max. wear in exhaust-manoeuvring direction: 0.22 mm.

The countermeasures introduced on the S50ME-C9.2 can be summarised as follows:

- jacket cooling water outlet temperature increased to 90ºC
- insulated cooling pipes in all cooling bores on cylinder liners already machined
- new cylinder liner designed with less cooling intensity. This is obtained by decreasing the number of cooling bores and increasing the distance between liner running surface and cooling bores.

A liner inspection on an engine where countermeasures had been introduced after sea trial, see Fig. 34, showed that

Fig. 32: S50ME-B9.2: cold corrosion, after sea trial

Fig. 33: S50ME-B9.2: cold corrosion, sister vessel inspected after approx. 1,000 hours

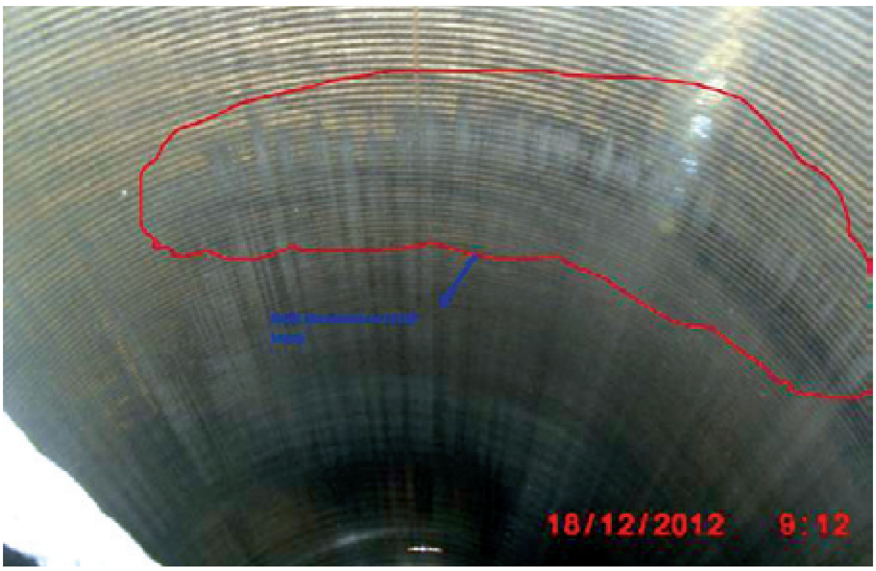

Fig. 34: Solution provided to cold corrosion on S50ME-B9.2

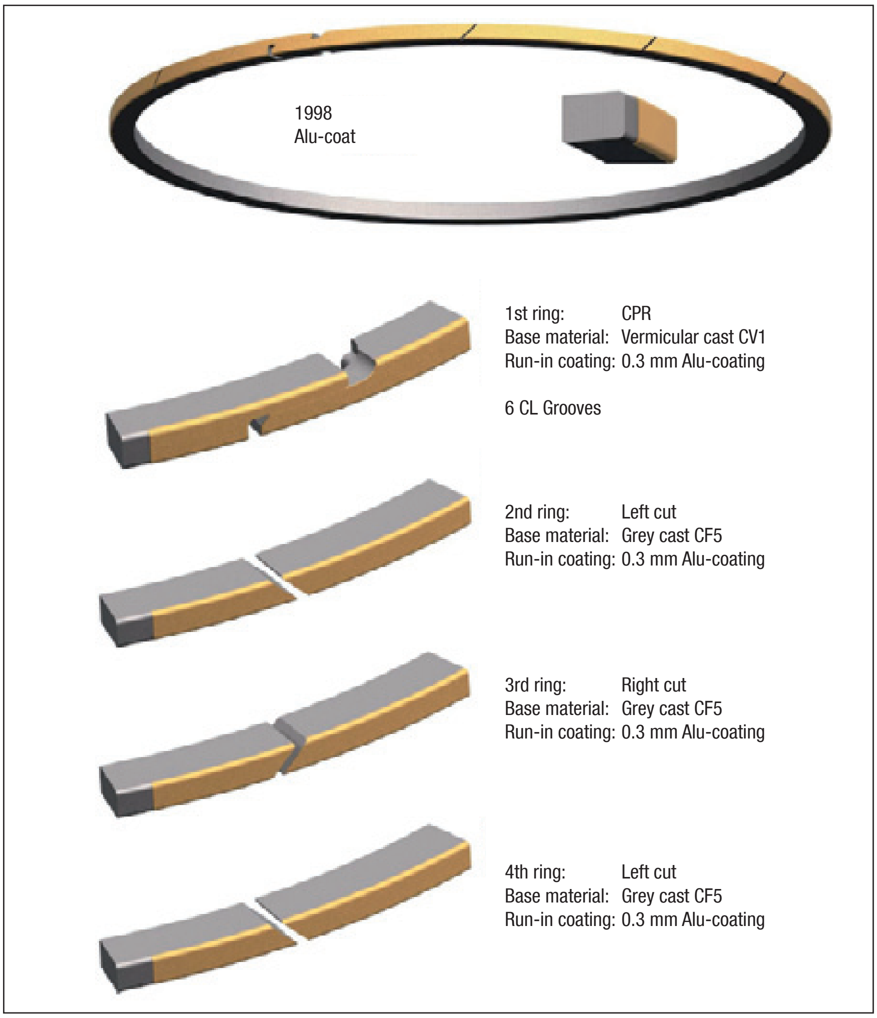

Fig. 35: CPR piston rings on small bore engines

cold corrosion in the liner top had disappeared. This means that the countermeasures introduced to increase the liner wall temperature was successful.

Cylinder Condition Update 26 to 50 Small Bore Engines

For some years, small bore engines have been specified with our well-proven alu-coated CPR piston ring pack, see Fig. 35. However, in some cases, on small bore engines, this has resulted in shorter overhaul intervals because of wearing out the CL-grooves of the top piston ring.

If complete wear out of CL-grooves on the top piston ring occurs, the pressure drop will often be so large that adhesive wear between liner and ring may be the result and, eventually, leading to damage on the liner surface.

The port-on-plane (POP) top piston ring, see Fig. 36, will have the opposite characteristic. When the top ring wears, the relief ports will increase the leakage area and, thereby, the pressure drop across the top ring will decrease. This will reduce the contact pressure be-

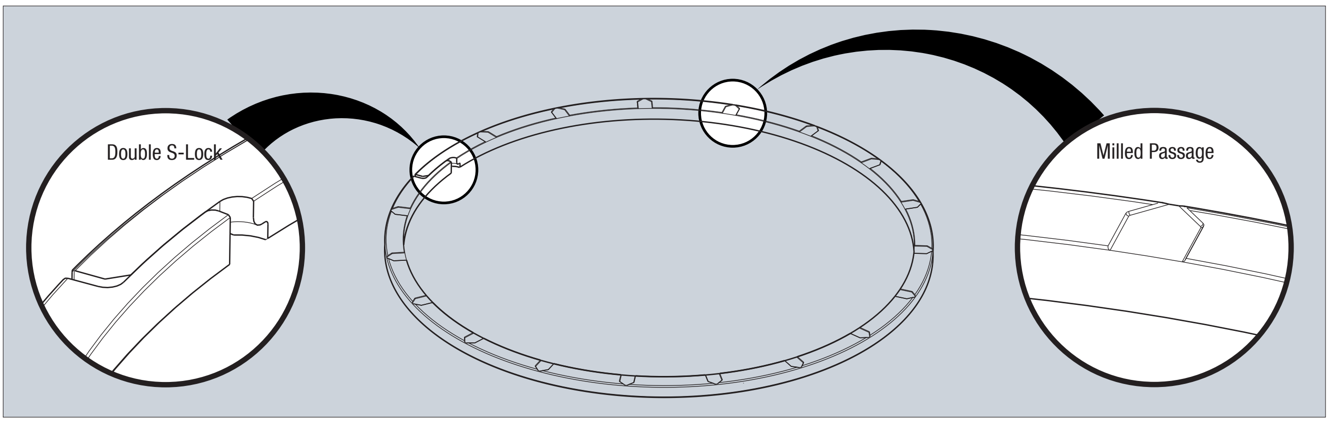

Fig. 36: Port-on-plane (POP) top piston ring design

tween the top piston ring and cylinder liner and reduce the piston ring wear. The POP top piston ring has a stabilising influence on piston ring wear. This is opposite to the conventional CPR ring. For small bore engines this is an important difference that gives the piston ring pack a longer lifetime.

The background for introducing the POP ring pack can be seen, e.g., on the 6S40ME-B prototype engine. At first glance, the cylinder condition looks very satisfactory after nearly 10,000 hours of operation, see Fig. 37. Also the cylinder liner wear is very satisfactory, see Fig. 38. All liners inspected have wear rates below 0.1 mm per 1,000 hours.

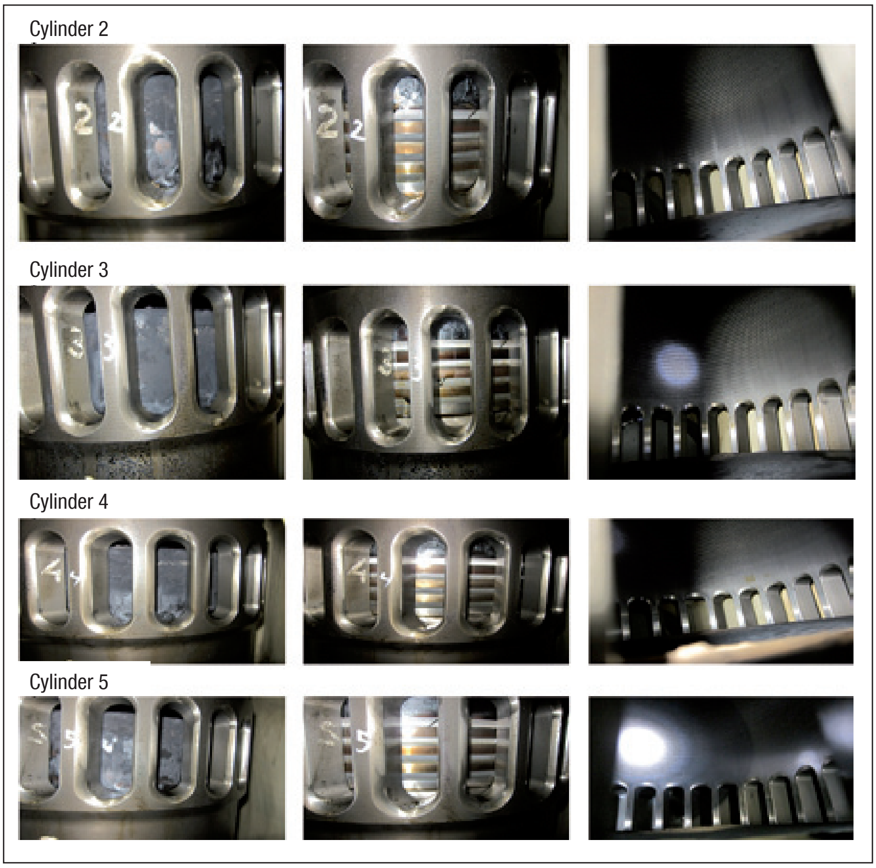

Fig. 37: Scavenge port inspection after approx. 10,000 hours with conventional CPR CL rings

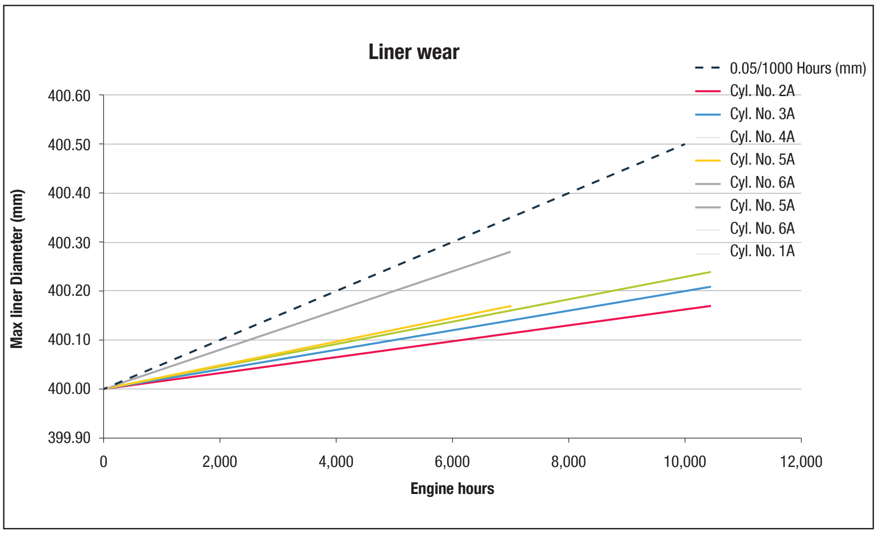

Fig. 38: Cylinder liner wear: 6S40ME-B at 10,000 hours

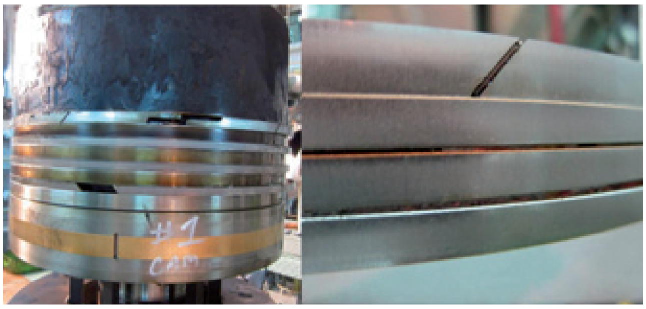

Fig. 39: Piston rings indicating rather high wear rates

Fig. 40: 6S46MC-C unit with POP ring pack

However, when looking more closely on the wear condition of the top piston ring it can be seen that unit No. 1 is close to the wear limit due to the CL groove reduction, see Fig. 39.

We now specify POP ring packs for this kind of engine. We have accumulated more than 230,000 service hours on various engine types with the POP rings. One of the service tests has been carried out on all cylinders on a 6S46MC-C engine. After 6,735 hours, the width of the „port-gap" has increased by 1.1 mm, corresponding to a top piston ring wear of 0.55 mm, see Fig. 40. With a wear potential of 2.7 mm, wear-out is estimated to happen after approx. 33,000 hours. Due to the self-stabilising nature of the POP ring, as described above, the estimated lifetime of 33,000 hours is conservative. We can expect even more running hours before wear-out.

Fig. 41 confirms the self-stabilising nature of the POP piston ring pack both on cylinder liner wear and on piston ring wear based on measurements on an S46MC-C engine after approx. 15,000 hours.

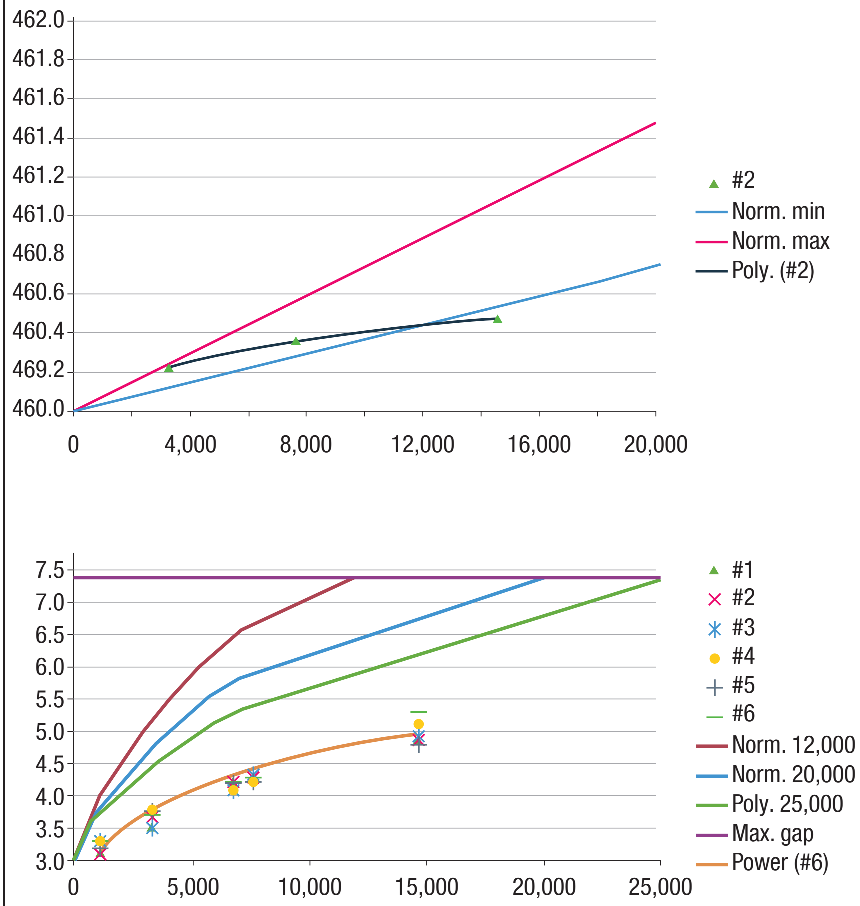

Fig. 41: Cylinder liner wear (upper) and POP piston ring wear (lower) after 14,554 h on 6S46MC-C

New Cylinder Lubrication Guidelines

Our guidelines on cylinder lubrication of MAN B&W low speed engines have called for an update in response to the below development, see Service Letter SL2013-571:

- Recent changes in operational patterns towards lower-load operation
- Development of new cylinder oils aiming at being better able to cater for a larger variation of fuel oil sulphur contents
- The general development of engines towards larger stroke-to-bore ratios and changed process parameters triggered by environmental compliance rules.

Based on the above, MAN Diesel & Turbo recommends the following:

- Lubrication with cylinder oils with at least the same acid neutralisation ability as the traditional BN 70 cylinder lube oils, i.e. BN 70-100 and SAE 50, on our newest engine designs (Mk 9 type and newer engines) when operating on high sulphur heavy fuel oil
- Increased lube oil feed rate or lubrication with higher-BN oils on part-load and low-load fuel-optimised engines requiring increased neutralisation ability
- Lubricator part load control break point set at 50% for Mk 8 and newer.

Lately, MAN Diesel & Turbo has concentrated on further enhancing the fuel efficiency while at the same time fulfilling Tier II. In order to improve the specific fuel oil consumption, the pressure in the combustion chamber has been increased on the newest engine designs, especially at low load. This pressure increase, together with the increased operating time at low load, has led to increased water and acid condensation on the cylinder walls, which leads to cold corrosion.

Also the most recently developed part-load and low-load tuning options utilise increased combustion chamber pressure as the main tool to ensure a low SFOC (specific fuel oil consumption).

Appropriate cylinder oil feed rates and ACC (Adaptable Cylinder oil Control) values must be obtained on the basis of service inspections, measurements and wear data from combustion chamber parts (piston rings, liner and crown), and can with benefit be supplemented with scavenge drain oil analyses.

Cylinder oil is essential for the two-stroke engine. Today's cylinder oils are made with a complex chemistry, and the individual feed rate must therefore be assessed for each oil brand, viscosity class and BN level.

A cylinder oil is mixed to achieve the necessary level of detergency and dispersancy to keep the piston rings and piston crown clean, and the necessary base number (BN) to neutralise the acids formed during combustion.

The cylinder oil not only serves to lubricate the moving parts, but is also designed to control the degree of corrosion on the liner surface.

This is illustrated by our feed rate guide, which sets the minimum feed rate to the level needed to keep the parts moving within a safe margin. However, so as to ensure the necessary lubrication effect, an increased formation of acid would call for a higher BN level than specified at the minimum feed rate. This is compensated for by calculating a feed rate on the basis of an ACC factor within the guide shown in Fig. 42.

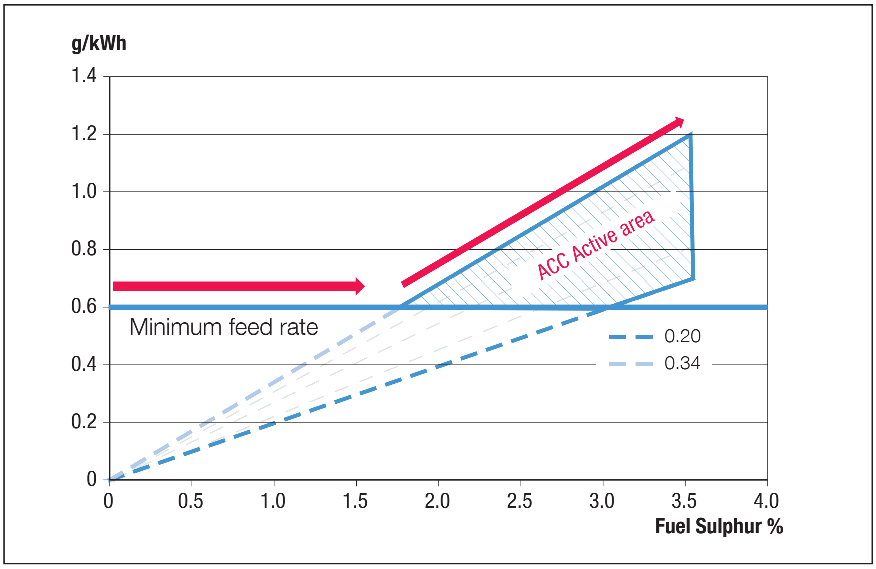

Fig. 42: BN 70 ACC range

In order to simplify the lubrication process on board the ships, as well the logistics of supply, the oil companies have started the process of developing a single-cylinder lube oil that can lubricate the cylinders regardless of the sulphur content in the fuel:

- Such oils have BN levels that are lower than the traditional BN 70 cylinder lube oils
- Such oils have performed acceptably in the service tests carried out
- Such oils can very well be used on the vast majority of earlier-type MAN B&W engines that are not affected by cold corrosion, but may not be applicable on newer engine designs with higher levels of cold corrosion.

MAN Diesel & Turbo recommends using cylinder lube oils characterised primarily by its BN number and SAE viscosity and to use a feed rate according to the BN in the cylinder oil and sulphur content of the fuel. MAN Diesel & Turbo is aware that some engines may be operated satisfactorily at even lower feed rates. Hence, feed rates are, just as before, based on practical experience rather than pre-calculated figures.

The above mirrors the importance of the fact that the crew should challenge the cylinder oil feed rate ACC factor, so as to find the correct ACC value that suits the actual engine configuration and engine load.

The best way to establish the optimum ACC factor is to measure the engine wear. If the wear rate of the liner and piston rings is too high, because of corrosion, the ACC factor must be increased to reduce the wear.

However, the ACC factor can only be assessed when the fuel sulphur level has been high enough to ensure that the lubrication has been in the ACC active area (the blue area marked in Fig. 42). At lower fuel sulphur levels the engine is excessively protected against corrosion because of the active minimum feed rate.

The acceptable wear rates must be in line with our recommendations on overhaul intervals and expected lifetime of the components. Liner wear rates are normally below 0.1 mm/1,000 running hours.

High ovality in the liner wear could be a sign of corrosive wear. As the liner surface temperature is not necessarily uniform, more corrosion will occur in the colder areas.

The piston ring wear must also be kept under observation, and it must be assured that the controlled leakage (CL) groove on the piston rings is not worn below the acceptable minimum and that the POP-ring groove does not exceed its maximum allowable wear.

Drain oil analysis is also a strong tool for judging the engine wear condition. Drain oil samples taken in active ACC operation will show if the oil feed rate can be optimised while keeping the BN between 10-25 mgKOH/kg and the iron (Fe) content below 200-300 mg/kg in the drain oil, see Fig. 43.

Used oil taken from the engine through the scavenge bottom drain can be used for cylinder condition evaluation.

On board sets exist, but it is important to get a valid test result that shows the total content of iron (Fe). Laboratory testing according to ASTM D5185-09 is the only certain measuring method. The BN must be tested in accordance with ISO 3771:2011(E).

A cylinder oil can be degraded to a certain level where the corrosion level begins to increase. The level of depletion is different among oil brands as well as among engines, and an individual evaluation of each engine is therefore recommended.

One possibility is to perform a stress test called "feed rate sweep". This will shorten the ACC familiarisation period considerably. The sweep test is based on a fast six-day test at steady load and, preferably, running on fuel in the high-sulphur range of 2.8-3.5% sulphur content. The feed rate is adjusted to set values, i.e. 1.4, 1.2, 1.0, 0.8 and 0.6 g/kWh. Each feed rate must be applied for 24 running hours before taking a sample and switching to the next feed rate. A detailed feed rate sweep pro-

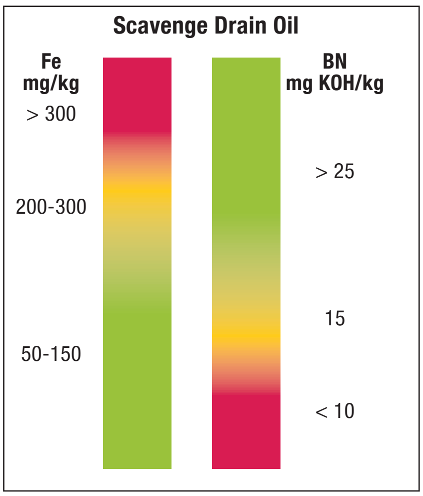

Fig. 43: Scavenge drain oil result

tocol is enclosed in our Service Letter SL2013-571.

The various oil suppliers offer cylinder oils with a broad range of BN levels. Our MAN B&W engine design is based on the 70 BN oil traditionally used, however, as new oil products have been introduced, BN levels have changed.

When switching to a different BN level, we recommend to start out with scaling the ACC factor from 70 to the new BN level by multiplying the ACC factor with the fraction of 70/BN oil.

Example:
Using a BN 45 and ACC (BN 70) = 0.26
ACC (BN 45) = 0.26 × 70/45 = 0.40

When changing to a new oil brand or type, the ACC factor may need to be reassessed as described above, starting with an ACC factor in the upper range. After this, a gradual reduction can be carried out based on actual observed conditions or the sweep test.

When running on low-sulphur residual fuel (HFO), the feed rate will be set at the minimum feed rate. High-BN cylinder oils will lead to over-additivation in the aspect of controlling the corrosion as well as lead to increased build-up of piston crown deposits.

We therefore recommend switching to a low-BN cylinder oil at the same time as switching to a low-sulphur heavy fuel. Continuous running on high-BN cylinder oils can only be recommended in special cases, and not for more than 1 to 2 weeks.

Application	BN
Distillate and LNG	< = 40
Low-sulphur residual fuel	40-60
High-sulphur residual fuel (70-100 for Mk 9 and newer)	55-100

Fig. 44: Cylinder oil guide

Also when switching to distillate fuels (MGO/MDO), we recommend switching to a low-BN cylinder oil at the same time as the switching of the fuel. We do not recommend the use of a high-BN cylinder oil when running on distillate fuels. For further information see Fig. 44.

When operating the engine at part load, the cold corrosion behaviour may deviate from operation at normal load. When the vessel is slow steaming, the engine is operated at low load, and the liner surface will become colder and, therefore, increase the risk of corrosion. Waste heat recovery and various part-load optimisation possibilities, e.g. T/C cut-out, variable turbine area (VTA) turbocharger and exhaust gas bypass (EGB), may call for a re-assessment of the ACC factor to accommodate the new corrosion level.

PMI Autotuning

PMI systems for the MC/ME engines have been developed over a number of years. So-called offline versions where pressure sensors are shifted from cylinder to cylinder have been developed and applied on many engines over the years. Online versions with pressure sensors permanently installed in each cylinder cover have also been widely applied.

In 2012, MAN Diesel & Turbo decided to make the newest online system versions standard on all large bore ME/ME-C engine types from 80 bore and up. The newest online version is called PMI autotuning and, in addition to continuously taking online pressure measurements for each cylinder, the system also offers the possibility of automatically tuning the engine to optimal combustion parameters.

As shown in Fig. 45, the present PMI autotuning system consists of pressure sensors in each cylinder cover. The signals from the sensors are transmitted to the data acquisition unit (DAU), which is a dedicated computer for signal recording and conditioning. Data are shown on the CoCos-EDS PC, and proposals for adjustments are presented on the main operating panel B (MOP-B). Changes to combustion parameters are then introduced via the MPCs on the control of the individual cylinders.

The benefit of applying PMI autotuning is illustrated in Fig. 46. An average increase of the max. pressure of 1 bar will result in a saving in the specific fuel oil consumption (SFOC) of 0.20-0.25 g/kWh. Furthermore, balancing of the compression pressures and the mean pressure will be beneficial to both fuel oil consumption and overhaul intervals for various components.

Fig. 47 shows an example of a PMI autotuning application on a 10K98ME-C engine. Before applying continuous autotuning, the average maximum pressure is approx. 4 bar below the reference value. This is changed by applying the autotuning, and within 1 to 1.5 minute, the average maximum pressure is increased by approx. 4 bar, thereby reducing SFOC by approx. 1 g/kWh.

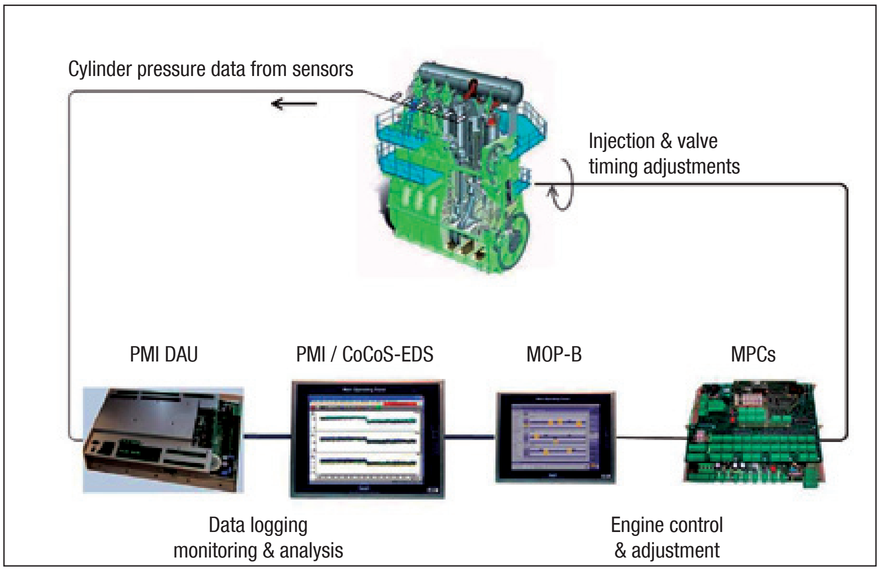

Fig. 45: PMI autotuning

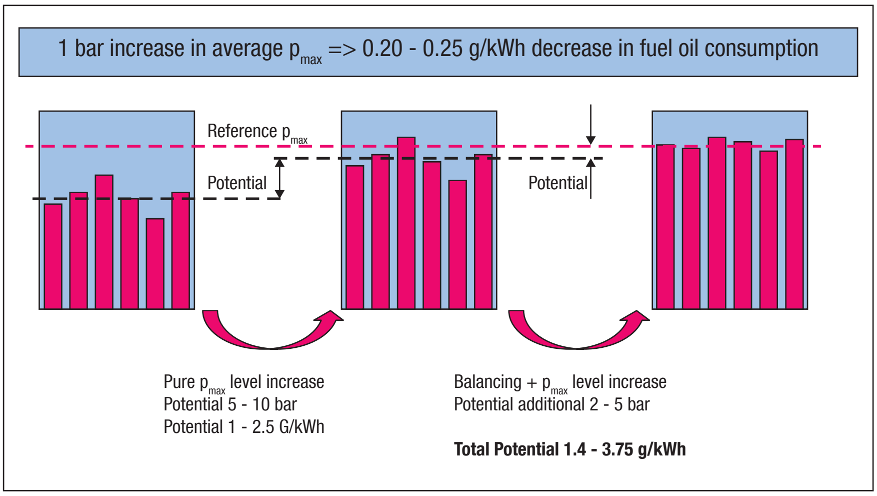

Fig. 46: PMI autotuning: potential fuel reduction

Fig. 47: PMI autotuning: 10K98ME-C

Conclusion

An update on service experience has been given. Today, MAN B&W two-stroke engines in the electronically controlled versions, the ME engines, account for a very large part of the engines ordered. We can conclude that this coincides, time-wise, with the fact that the reliability both electronic and hydraulic components has improved significantly.

However, it is also important to mention that the classical topics for two-stroke engines, such as cylinder condition, bearing performance, fuel equipment reliability, still needs attention in order to safeguard the continued success of the MAN B&W two-stroke brand.

Also challenges coming from the new methods for optimising the Dot 2 versions have been touched on, and these challenges will also need to have some attention in the future.

The worldwide situation in shipping continues to put focus on extreme low-load operation. Service experience from low-load operation with MAN B&W two-stroke engines has generally been very positive.

The future focus must remain on maintaining a high reliability and on EEDI (energy efficiency design index) and, not at least, Tier III technology.

5510-0143-00ppr May 2013 Printed in Denmark

MAN Diesel & Turbo
Teglholmsgade 41
2450 Copenhagen SV, Denmark
Phone +45 33 85 11 00
Fax +45 33 85 10 30
info-cph@mandieselturbo.com
www.mandieselturbo.com