Service Experience 2014

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MAN Diesel & Turbo

Service Experience
MAN B&W Two-Stroke Engines

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Introduction

This paper describes in detail the service experience of the new generation of super-long stroke S and G Mk 9 type engines. Focus will be on the cylinder condition in general and cold corrosion control in particular. The service experience with new jacket cooling water systems, new cylinder oils (BN 100 types), modified combustion chamber design and new versions of the Alpha Lubricators (main focus: Alpha Mk II) will be outlined.

An update on low-load operation is given and, furthermore, the initial service experience with EGR systems will be touched on.

Early version of S90ME-C9 at 830 hours and lubrication 0.90 g/kWh

Early version of S80ME-C9 at 557 hours and lubrication 0.90 g/kWh

Fig. 1: Poor cylinder condition – recent examples

Other two-stroke issues (case stories) will be addressed, including fuel injector development, cavitation in the ME hydraulic exhaust valve actuation system and ME-B9.3 updates, especially related to the timing unit of the dot 3 design. Furthermore, emergency running of the 6G70ME-C9.2 engine type without turbocharger, acceleration issues for ME-C/ME-B in dot 2 versions, and service experience for the main hydraulic pumps on the ME engines will be mentioned.

Cold Corrosion Control

Recently, cold corrosion of cylinder liners has grown to become a major issue for the latest generation of MAN B&W two-stroke engines, see Fig. 1. This has called for measures to control/suppress the cold corrosion, leading MAN Diesel & Turbo to take the following initiatives:

1. Introduction of BN 100 cylinder oils
2. Increased jacket cooling water temperatures – various systems
3. Redesigned cylinder liner
4. New cylinder lubricators: Alpha Lubricator Mk II

Introduction of BN 100 cylinder oils

Since autumn 2013, we have requested oil companies to focus on the development of BN 100 cylinder oils for the newest generation of engines. We have stated that the BN 100 oil is the “design basis” for our new engine generations, and we have received a positive response to our request from all the major oil companies, which now have BN 100 oils available in all important ports.

Our guidelines on cylinder lubrication of MAN B&W low speed engines have called for an update in response to the following development:

- Recent changes in operational patterns towards optimising low/part-load operation
- Development of new cylinder oils that are even better to cater for a large variation in fuel oil sulphur content levels
- 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 on our newest engine designs (Mk 8-8.1 and newer) with cylinder oils with higher acid neutralisation ability than the traditional BN 70 cylinder lube oils, i.e. BN 100 and SAE 50, 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.

Lately, MAN Diesel & Turbo has concentrated on further enhancing the fuel efficiency while 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/part load. This pressure increase, together with the increased operating time at low/part 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 by service inspections, measurements and wear data from combustion chamber parts (piston rings, liners and crowns), and can with benefit be supplemented with scavenge drain oil analyses.

Cylinder oil is essential for a two-stroke engine. Today, 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 based on an ACC factor within the guide shown in Fig. 2.

In order to simplify the lubrication process on board the ships, as well as the logistics of supply, the oil companies have developed cylinder lube oils 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 should not be applied on newer engine designs with higher levels of cold corrosion.

MAN Diesel & Turbo recommends use of cylinder lube oils that are 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.

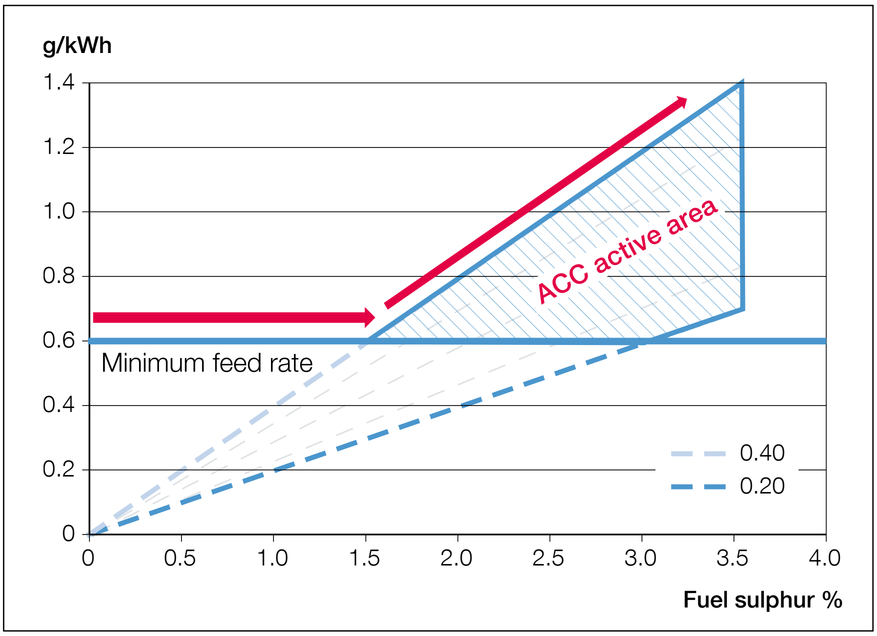


Fig. 2: BN 100 ACC range for Mark 8-8.1 and newer engines

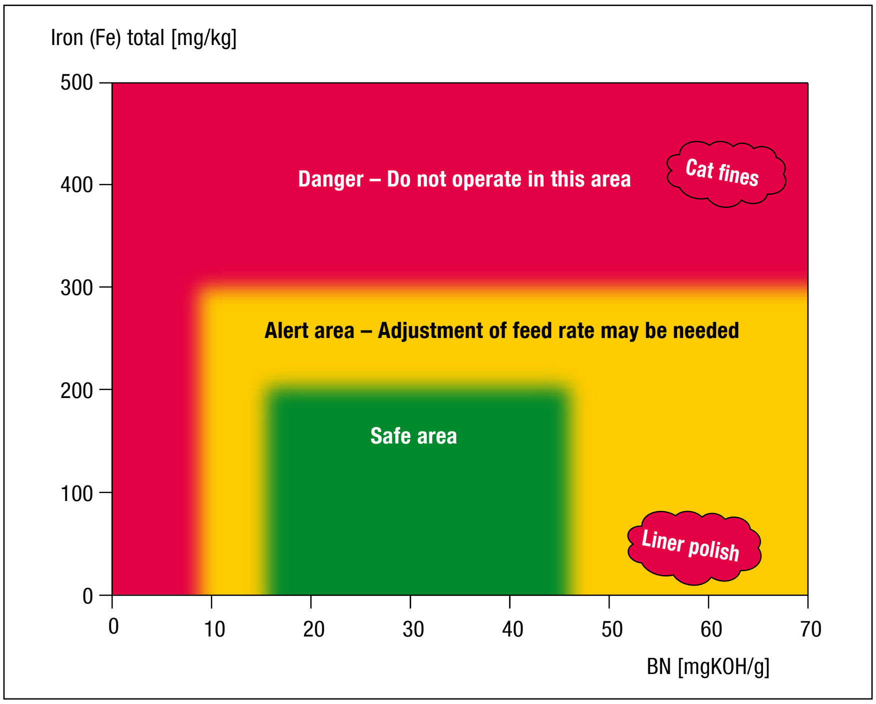


Fig. 3: Scavenge drain oil result

The above mirrors the importance of the fact that the crew challenges the cylinder oil feed rate ACC factor 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 cylinder liner and piston ring 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. 2). 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 occurs 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. POP-rings are designed with gas leakage grooves on the bottom surface of the piston ring.

A 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. 3.

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

On-board sampling 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).

Cylinder oils 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 option is to perform a stress test called “feed rate sweep”. The sweep test is based on a fast six-day test at steady load and, preferably, while running on a fuel in the high-sulphur range of a 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 feTed rate sweep protocol is enclosed with our Service Letter SL2014-587.

Fig. 4 shows the result of a cylinder oil feed rate sweep test for a 9S90ME-C8.2 performed at 25% load using BN 70 cylinder oil operating on 2.7% HFO.

The influence of the use of higher BN cylinder oil has also been validated in a number of test cases. In general, it can be said that, based on these test cases, neutralising the efficiency is proportional to the BN number. Fig. 5 shows the result of a cylinder oil feed rate sweep test for a 9S90ME-C8.2 performed at 25% load using BN 85 test cylinder oil operating on 2.7% HFO.

The various oil suppliers offer cylinder oils with a broad range of BN levels. Our MAN B&W engine design is based on the BN 100 oil.

When switching to a different BN level, we recommend starting out with scaling the ACC factor from 100 to the new BN level by multiplying the ACC factor with the fraction of 100/BN oil.

Example:
Using a BN 85 and ACC (BN 100) = 0.26
ACC (BN 85) = 0.26 × 100/85 = 0.31

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. Next, 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 must 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 1-2 weeks.

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 Table 1.

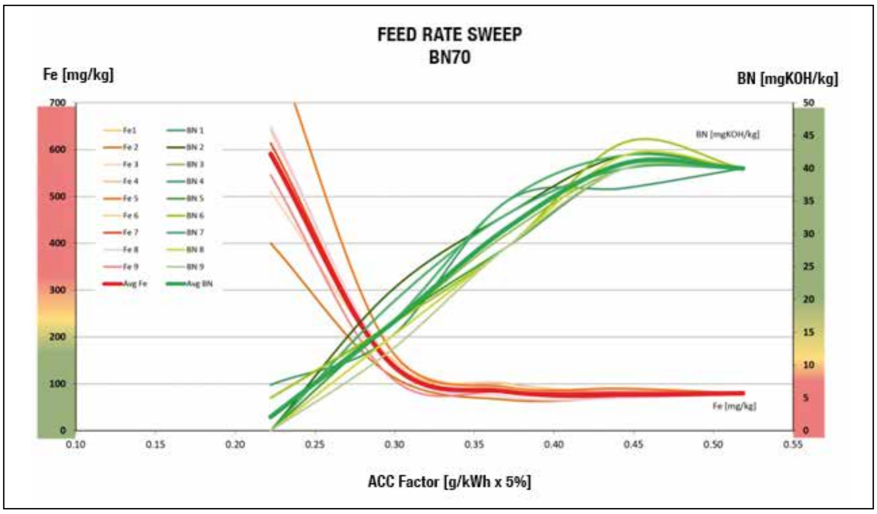


Fig. 4: 9S90ME-C8.2 cylinder oil feed rate sweep test, 2.7% HFO, BN 70 cylinder oil (Source: ExxonMobil)

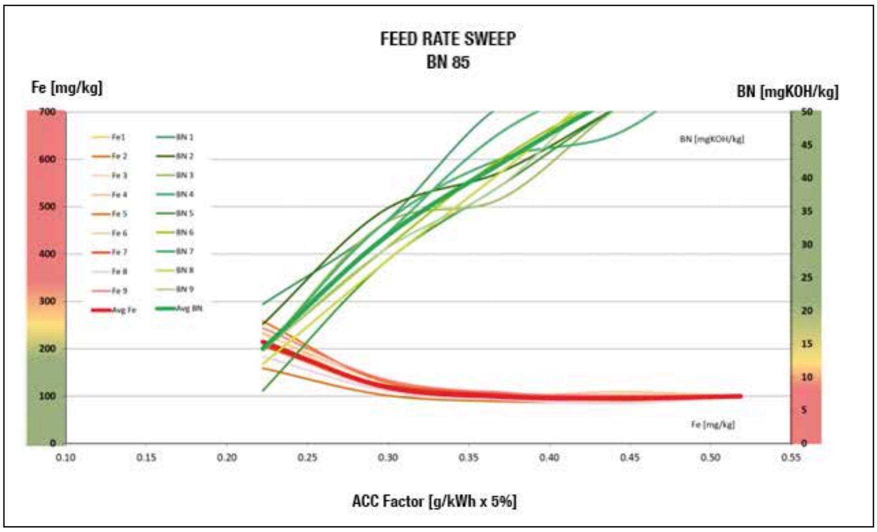


Fig. 5: 9S90ME-C8.2 cylinder oil feed rate sweep test, 2.7% HFO, BN 85 test cylinder oil (Source: ExxonMobil)

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, for example, T/C cut-out, variable turbine area (VTA) turbocharger, retrofit ECO-cam for MC/MCC engines and exhaust gas bypass (EGB), may call for a reassessment of the ACC factor to accommodate the new corrosion level.

Updated design for Cylinder Lube Oil (CLO)
For engines operating on destillates and LNG ≤ 40 BN CLO, SAE 50
For previous engine types operating on heavy fuel (Mk 7 and older) 70-100 BN CLO SAE 50
For newer engine types operating on heavy fuel (Mk 8-8.1 and newer) 100 BN CLO, SAE 50

Table 1: Cylinder oil guide

Increased jacket cooling water temperatures – various systems

In order to suppress cylinder liner cold corrosion, we have introduced various systems to increase the cylinder liner wall temperature.

Fig. 6 shows a jacket cooling water bypass that has been shop tested to determine the correct amount of cooling water to be bypassed. For the final setup for service, the amount of bypassed water is determined by orifices in the cylinder outlet pipes.

Bypassing 85% of the jacket cooling water will increase the liner wall temperature by approximately 15ºC. In addition, the jacket cooling water outlet temperature is increased to 90ºC. All together, an approximately 20ºC increase of the liner wall temperature is achieved. Tests have revealed that bypassing an even larger amount is possible.

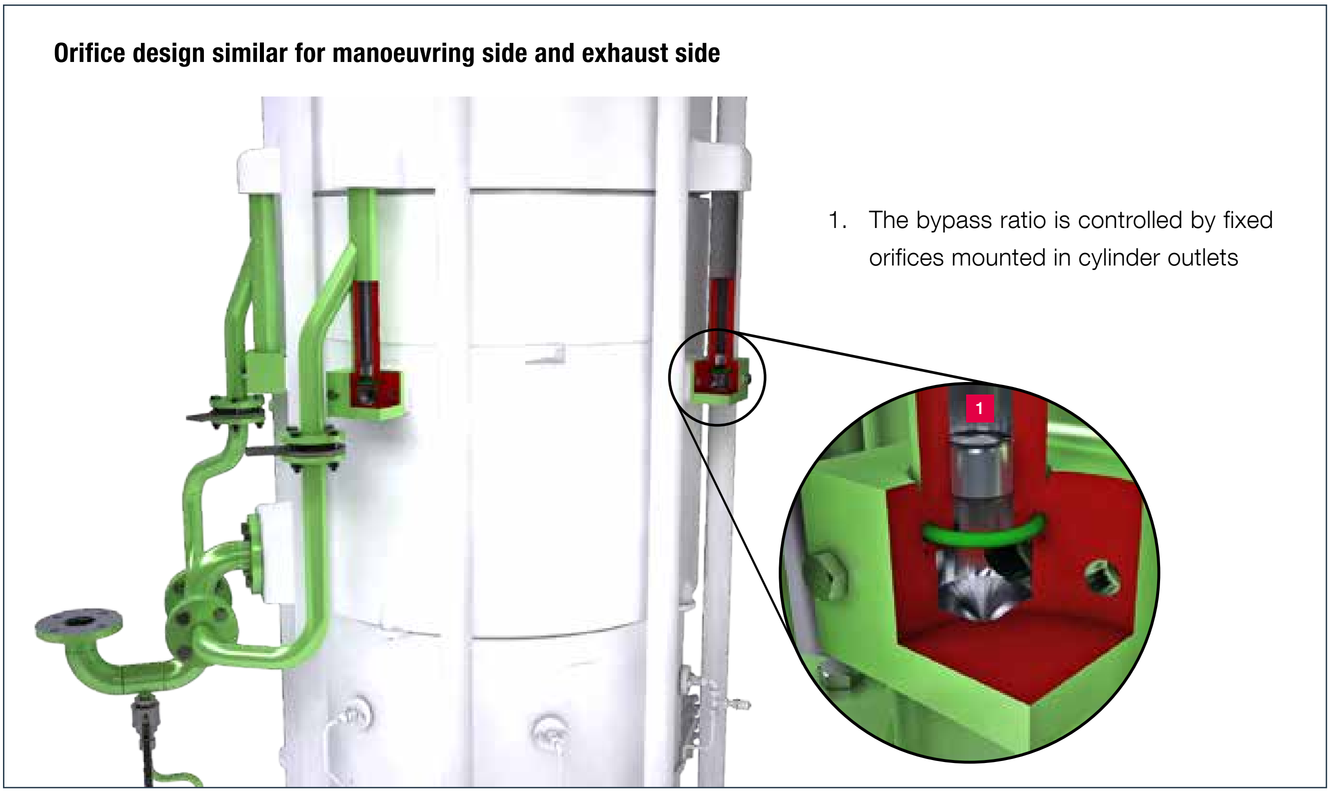


Fig. 6: Jacket cooling water Bypass Basic (JBB) system

Furthermore, a controlled version of the JBB system, called JBC, see Fig. 7, is being developed. In this system, a thermostatic valve controls the amount of bypass in such a way that a large amount of water is bypassed at low load and less is bypassed at higher loads.

The JBB and JBC systems can both be easily fitted on engines already in service.

A more active bypass system is being prepared for future engines. At the time of writing, this system is the standard on new S and G 80, 90 and 95-bore engines. Furthermore, it has been decided to introduce the system on future G50, G60 and G70 engine types. The system shown in Fig. 8 consists of two extra cooling water pipes along the engine. An extra pump and an extra control valve ensure up to 130°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 maintained up to 90% load.

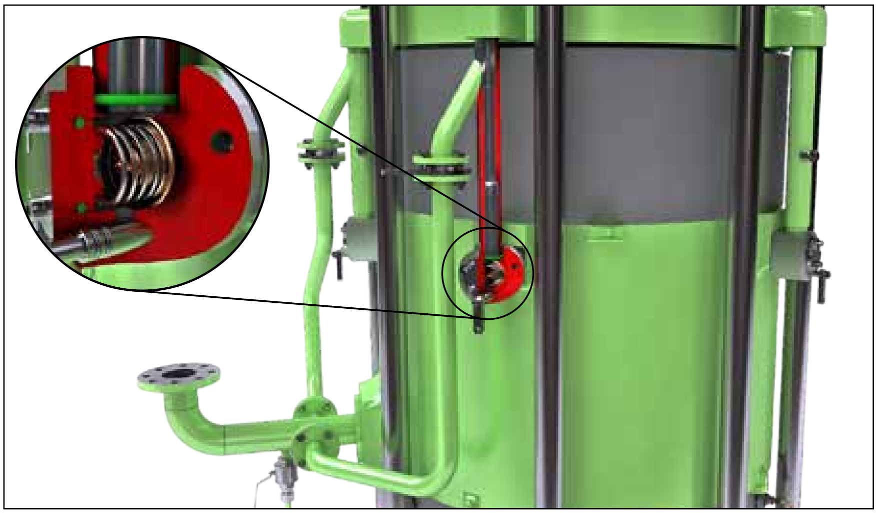


Fig. 7: Controlling corrosive wear by internal Jacket cooling water Bypass Controlled (JBC) system

The LDCL system does not mean changing the connections to the vessel's cooling water system. The LDCL system is designed with an extra mixing circuit on the engine comprising a pump, a three-way valve and a control system. The jacket water temperature out of the cylinder liner can then be controlled according to the diagram in Fig. 9.

Preliminary results of service tests have shown that a significant reduction of the specific cylinder oil consumption is obtained with this system.

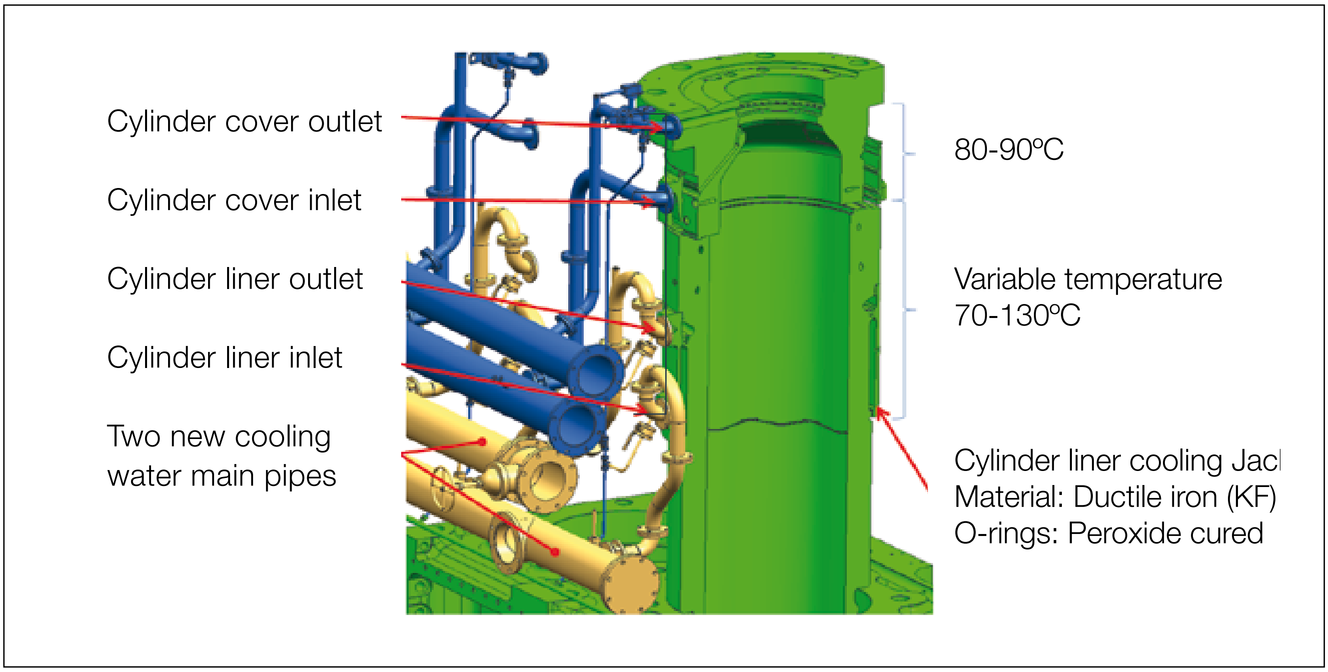


Fig. 8: Controlling corrosive wear: four-pipe jacket cooling water system – load-dependent cylinder liner (LDCL) jacket cooling water system

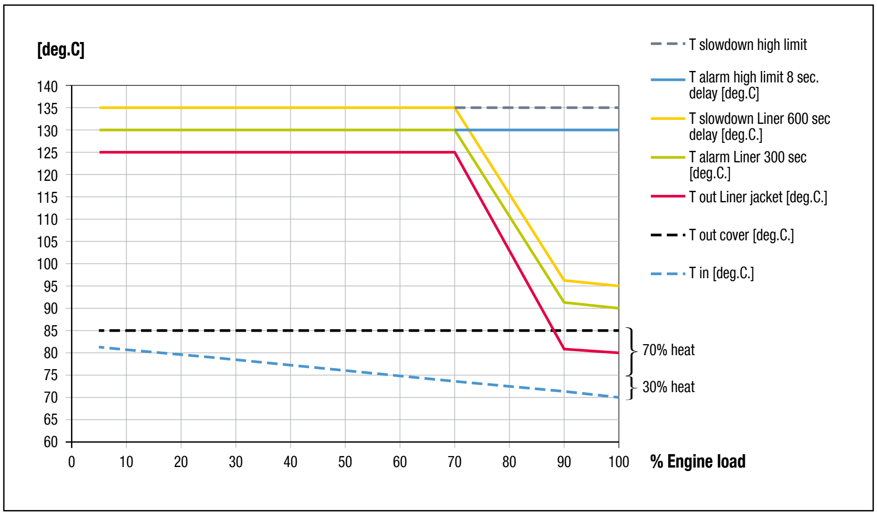


Fig. 9: LDCL setpoint temperatures vs. engine load, S80ME-C9.2

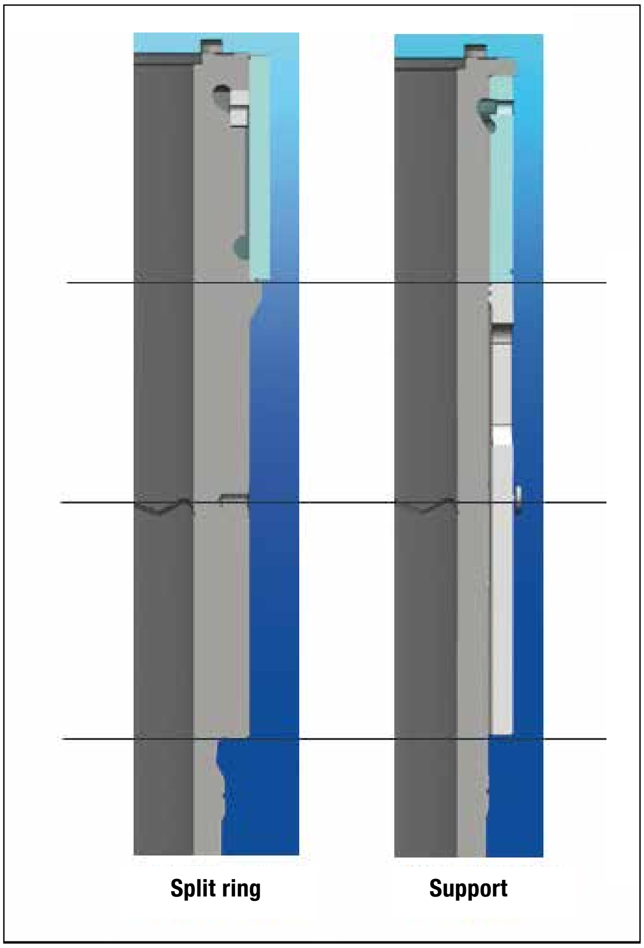


Fig. 10: Hot replacement cylinder liner concepts

New cylinder liner designs

The design of a cylinder liner with a higher cylinder wall temperature has been initiated for a number of engine types. A modified cooling design is being considered. Fig. 10 shows the two candidate designs, the split ring design and the design with a support cylinder between the liner and the cylinder frame. Also, a redesign of the cooling water jackets with a significantly reduced cooled area is under consideration, see Fig. 11. The aim of these measures is to increase the liner wall temperature over a rather large area in the top part of the liner at all loads. By doing so, the “wear-down” of the cylinder oil BN-reserve is reduced and a lower required cylinder oil feed rate can be established.

In order to improve the lifetime of the wear component (the cylinder liner) in an economical manner, the hot replacement cylinder liner project is aimed at existing engines.

Furthermore, we have recently introduced cylinder liners with varying designs based on the rating of

Fig. 11: Hot replacement cylinder liner concepts: shorter cooling jacket

the engines. Derated engines can be equipped with cylinder liners with lower cooling intensity without exceeding the maximum cylinder liner temperature allowed when running at the maximum specified rating.

We are currently designing rating-dependent cylinder liners for the G-engine series. Three different liner designs will be introduced, which will be based on three MEP ranges, see Fig. 12. The liner design is modified by changing the position and length of the cooling bores according to the engine rating (MEP), see Fig. 13. The resulting temperature profiles on the liner suface are illustrated in Fig. 14, using a G70ME-C9 as an example.

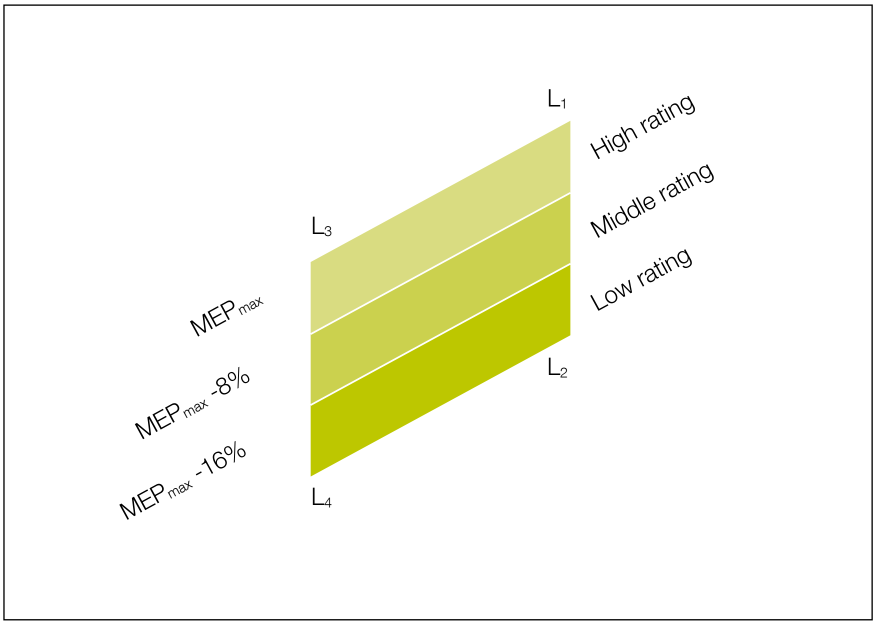


Fig. 12: Three rating ranges for cylinder liner design

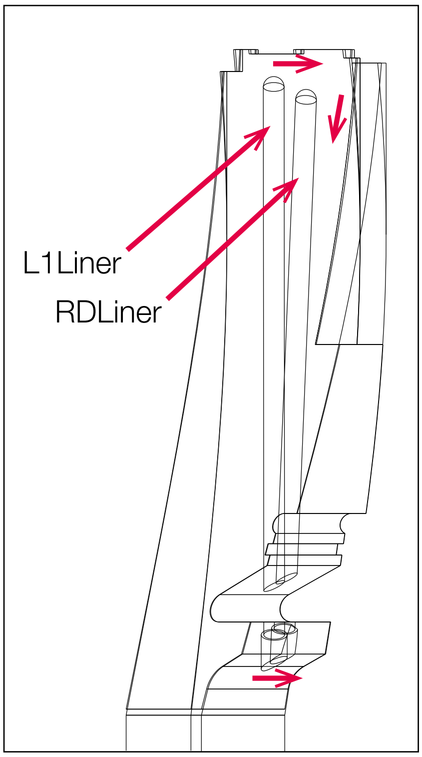


Fig. 13: Liner cooling bore variation (length and position of bores)

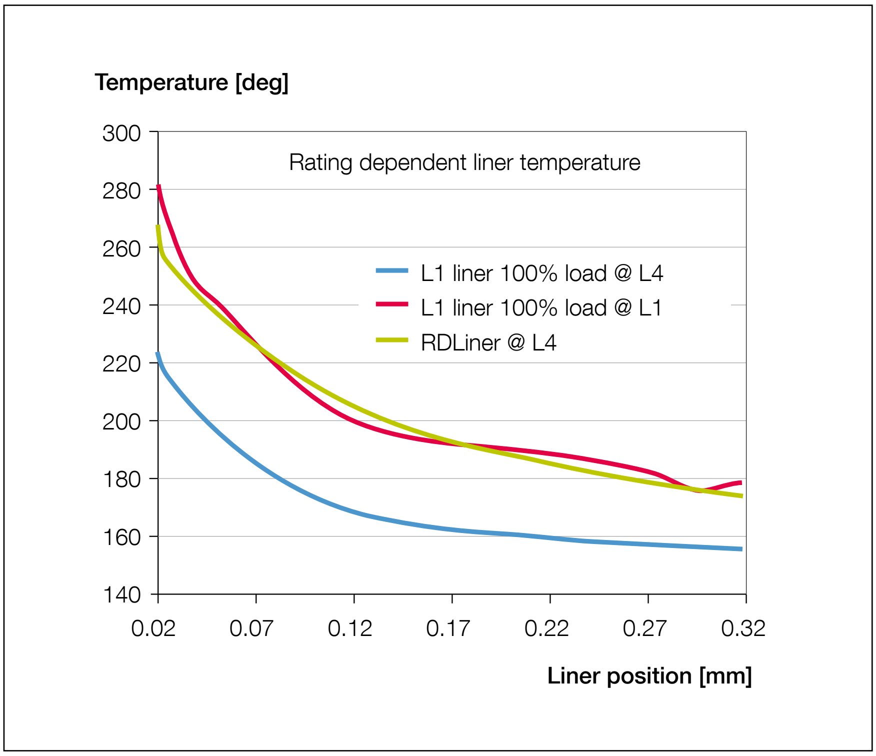


Fig. 14: Temperature profile in upper part of liner with RDL design

New Alpha Mk II cylinder lubricator

A new version of the well-proven Alpha Lubricator is presently being introduced. The first engines to be equipped with the Alpha Mk II cylinder lubricator as standard are the S90ME-C10.2 and G95ME-C9.2 type engines. An outline of the application on S90ME-C10.2 is illustrated in Fig. 15.

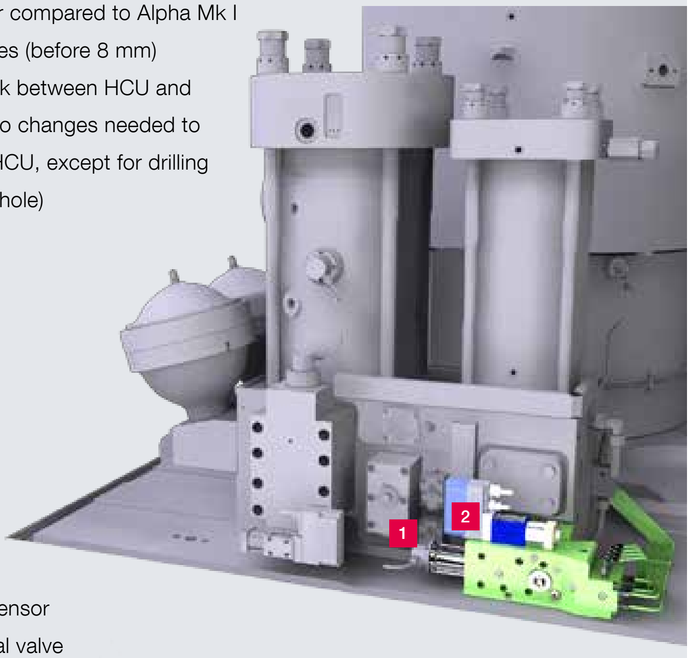
The new Alpha Mk II cylinder lubricator can inject cylinder oil with great flexibility:

- in one or multiple portions per revolution
- all injections are timed according to the crank angle
- multiple injection during each revolution is possible
- injection intensity can be varied
- plunger can deliver oil in the requested portion until full stroke is reached and then return, see Fig. 16.

At the time of writing, the Alpha Mk II cylinder lubricator is being tested in service on an S80ME-C9.2 engine. These tests follow the intensive rig-testing, see Fig. 17, completed at our Diesel Research Centre in Copenhagen in 2013.

10.2 Standard:
Alpha Lubricator Mk II

- slightly larger compared to Alpha Mk I
- 6 mm oil pipes (before 8 mm)
- adapter block between HCU and Lubricator (no changes needed to the current HCU, except for drilling of one extra hole)



1. Inductive sensor
2. Proportional valve

Fig. 15: Controlling corrosive wear with Alpha Lube Mk II

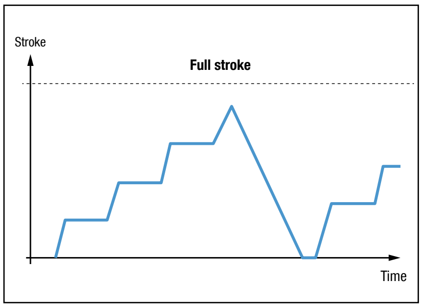


Fig. 16: Alpha Lube Mk II; the plunger moves in various portions of full stroke

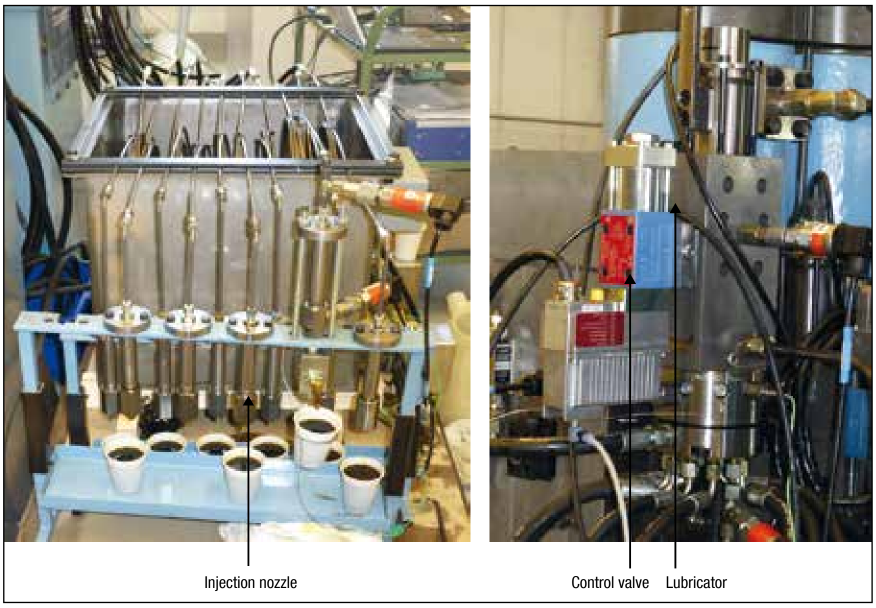


Fig. 17: Alpha Lube Mk II; new cylinder oil lubricator, test rig

Acceleration Issues for ME-C/ME-B Mk 9 – S and G Type Engines

Lately, low acceleration capability has been experienced on some S50ME-B8.2/9.2 engines installed on ships with high-efficiency propellers operating at a relatively low rotational speed.

The root cause has been identified to be a low excess air ratio (air for combustion) at low load and heavy running especially during the acceleration phase. This is due to the application of two-stroke Miller timing with late closing of the exhaust valve.

The countermeasure introduced by MDT has been the installation of a controllable orifice in the exhaust valve oil push rod, see Fig. 18, to facilitate earlier closing of the exhaust valve during low engine speed running. This solution will increase the air excess ratio in the cylinder.

Fig. 19 shows the influence of the controllable orifice on the opening/closing timing of the exhaust valve.

On ME-B dot 3 and ME-C type engines, the exhaust valve closing and, accordingly, p_{comp}/p_{scav} ratio can be varied by means of the ECS parameter settings. This means that no hardware changes are needed.

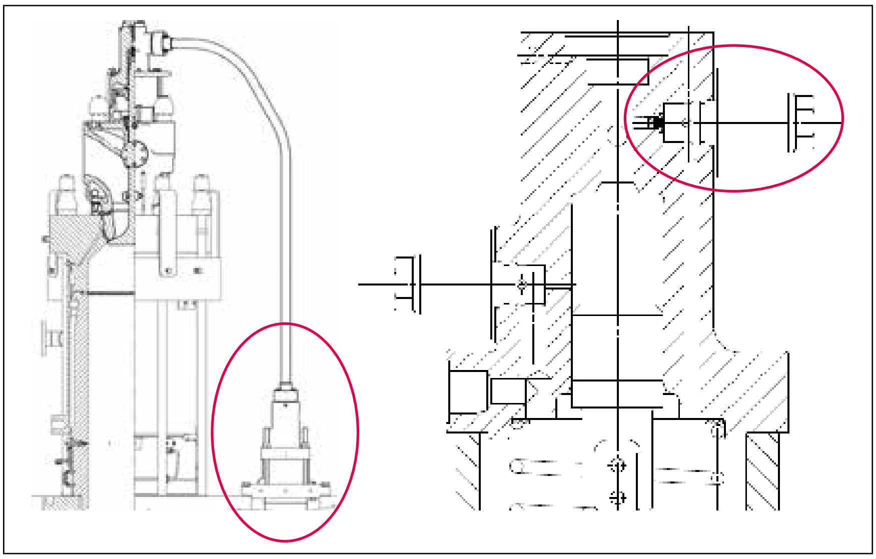


Fig. 18: ME-B/ME-C engine acceleration problems

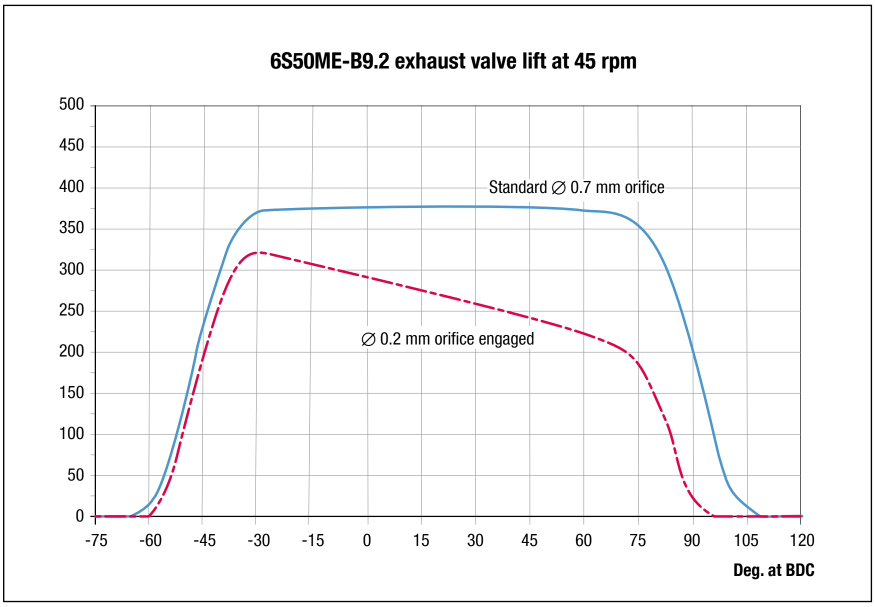


Fig. 19: ME-B9.2 engine acceleration problems

Astern Start and Running Issues – ME-B Types Engines

Long exhaust cams with negative cam lead were introduced on the dot 2 ME-B engines.

For ahead start, the standard starting air interval of 5-95 deg does not conflict with the opening of the exhaust valve at 115 deg ahead of TDC. The exhaust valve closes at 80 deg before TDC, resulting in a compression pressure of approximately 50 bar. Fig. 20 indicates the timing during ahead start.

As a result, at astern start, the exhaust valve opens already at 60 deg ahead of TDC, which leads to loss of starting air when applying the standard starting air timing (5-95 deg). Furthermore, the exhaust valve closes at 135 deg before TDC resulting in a compression pressure of approximately 95 bar. Fig. 21 illustrates the timing during astern start. The power of the "air engine" is decreased and the work needed has increased.

In order to change this situation, making the astern starting procedure more efficient, two countermeasures have been introduced:

- In astern direction, the starting air timing is changed to 5-82 deg for five-cylinder engines and to 5-75 deg for six to nine-cylinder engines. This limits the loss of starting air.
- A throttle valve delays the exhaust valve closing to 110 deg before TDC, see Fig. 22, resulting in a considerably reduced compression pressure.

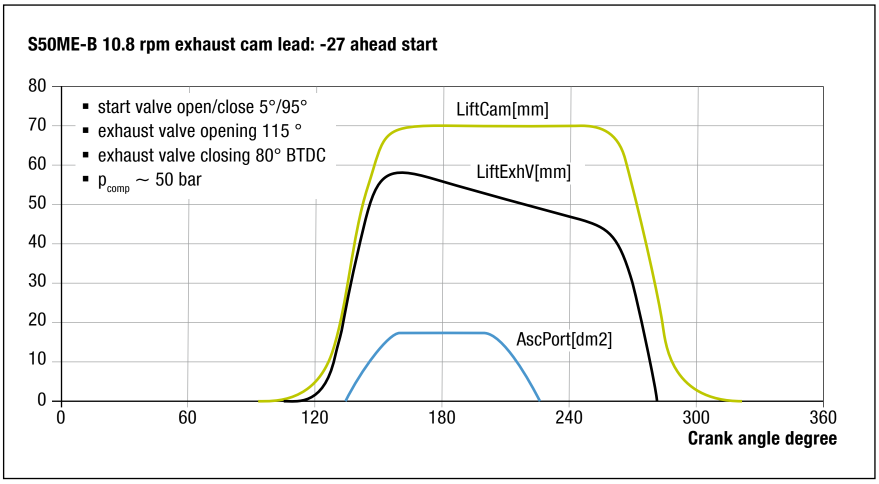


Fig. 20: ME-B9.2 astern start problems

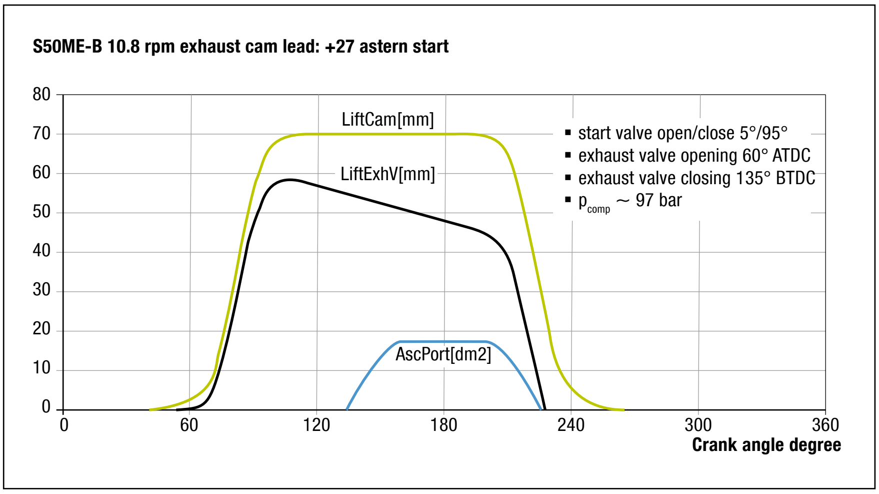


Fig. 21: ME-B9.2 astern start problems

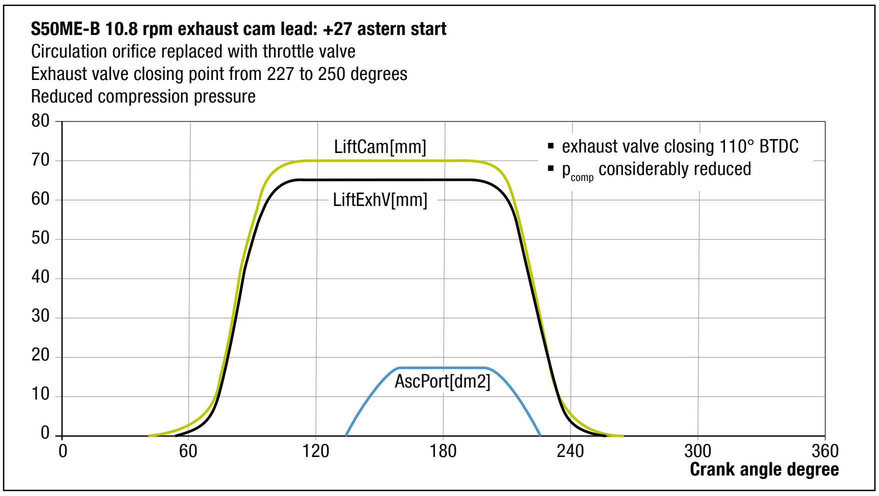


Fig. 22: ME-B9.2 astern start problems

For dot 3 ME-B engines, astern start is facilitated by the use of shorter exhaust cams, which will, in itself, reduce the compression pressure. However, we have decided to reduce the starting air interval also on these engines, as described in Item 1 above.

Sea trials performed lately on S50ME-B9.2 engines have revealed severe difficulties with obtaining 80% rpm in astern direction (as stated in manuals). Accordingly, we have evaluated this operating condition using tests and calculations and have concluded the following:

- The test results from sea trials indicate that the engine, ship and propeller are capable of absorbing a higher power than expected in astern direction, with high scavenge air pressure as a consequence.
- The astern high scavenge pressure in combination with the Miller timing layout of the ME-B engines with highly asymmetric exhaust cam timing and small compression volumes, result in significant excessive compression and maximum pressures when the engine is operated at 80% of the specified MCR rpm in astern.

In order to avoid this situation, we have limited the max. acceptable rpm in astern to 70% of the specified MCR rpm with immediate effect on all ME-B9.2 and ME-B9.3 engines. This is in accordance with the requirements for astern running as specified by IACS (International Association of Classification Societies).

ME-B9.3 Updates – Timing Unit

When the dot 3 technology was introduced, we experienced some difficulties with the timing unit system on a number of engines running on test bed, see Fig. 23. High-pressure peaks occurred in the timing unit and in the oil cylinder of the exhaust valve actuator. The high pressure resulted in a number of incidents with broken bolts on the puncture valve and a few cases of broken bolts on the timing unit end cover, see Fig. 24.

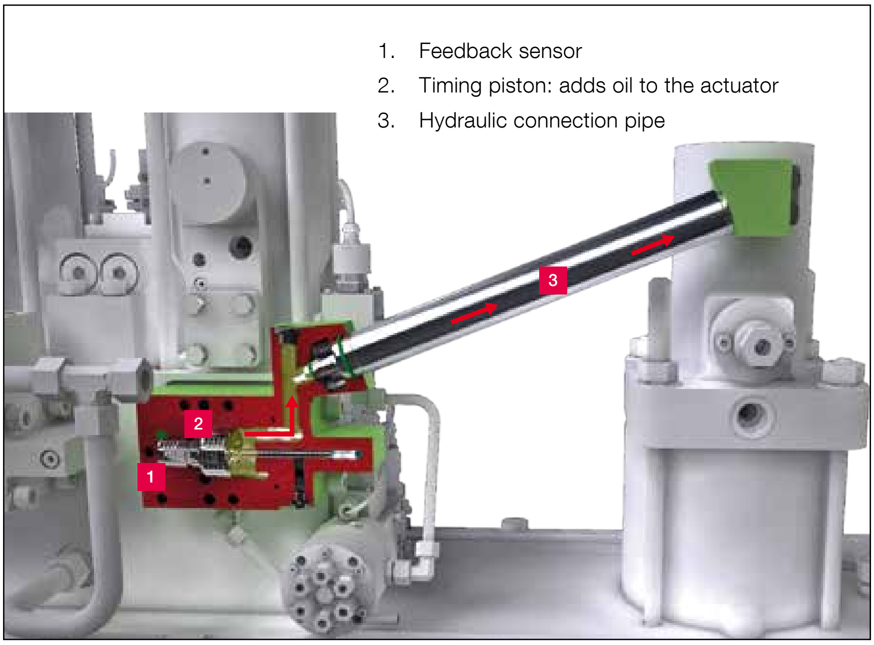


Fig. 23: Variable exhaust valve timing for ME-B

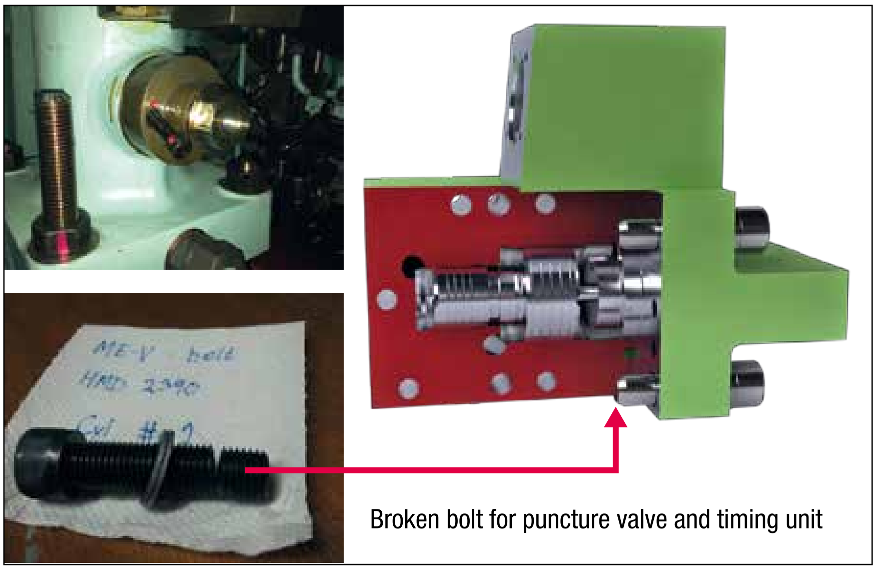


Fig. 24: ME-B dot 3 feedback from test bed

Measurements and simulations revealed that the timing piston had a high landing velocity on the return stroke when the timing unit is deactivated. A damper for the timing piston has been introduced as a countermeasure and the landing speed has been considerably reduced, see Fig. 25. Furthermore, this countermeasure reduces the maximum pressure to an acceptable level.

Fretting has been found on the double-walled pipe connecting the timing unit and exhaust valve actuator. The cause of the fretting is relative movements between the HCU with the timing unit and the exhaust valve actuator.

To counteract the fretting and ensure good reliability for the sealing rings, the pipe system has been upgraded as described in the following. HVOF-coating, using the same spray powder as for the exhaust valve spindle, is added on the ends of the pipes together with a PTFE sliding ring and a PTFE U-cup sealing ring. Furthermore, a wiper ring is added on the outer pipe to minimise the risk of particles, or the like, entering the sealing area. Modifications are shown in Fig. 26.

For new ME-B9.3 engines, the timing unit has been redesigned and the double-wall pipes have been replaced with single-wall pipes connecting the dual HCU and the exhaust actuator through the camshaft housing, see Fig. 27.

Low-Load Operation Update

Since the start of the worldwide financial crisis in 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 decided to operate at low loads. Today, also operators of tankers, bulkers, etc. are beginning to operate continuously at low load.

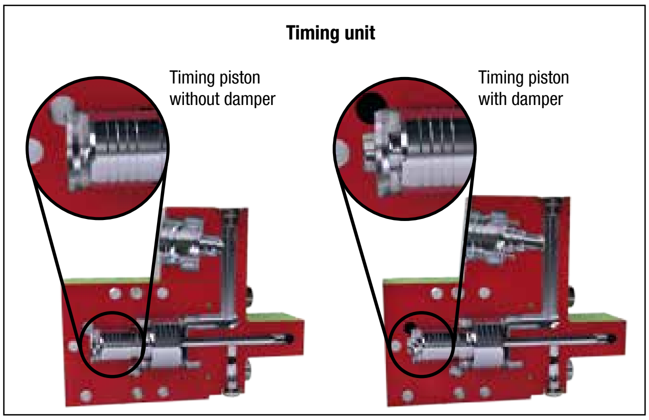


Fig. 25: ME-B dot 3 feedback from test bed

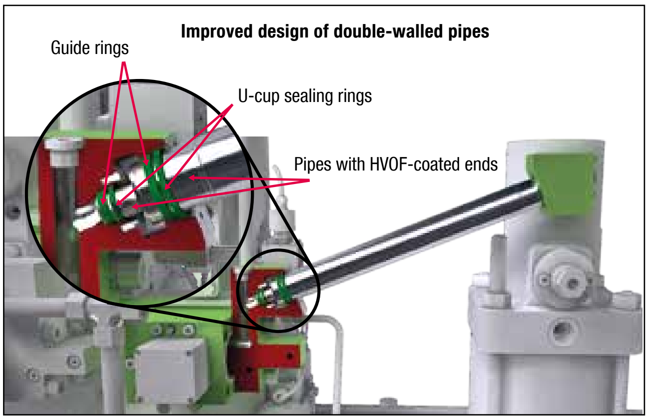


Fig. 26: ME-B dot 3 feedback from test bed

Fig. 27: ME-B dot 3 updated design: timing unit on the exhaust valve actuator

We are currently collecting data to support operation as far down as 5-6% continuous load for VLCCs with 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. Since then, nearly all service experience with continuous low-load operation 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, and the request to optimise running at low load surfaced. This can be supported on MAN B&W two-stroke engines by increasing the scavenge air pressure at low and part loads, thereby reducing the fuel oil consumption at these loads.

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.

As for low-load operation in general, a few issues listed below should be considered:

- 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 is cut out
- exhaust valve burn away on spindle lower face.

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 with 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. 28.

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. 29.

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 typically takes two years. Fig. 30 shows elasto-hydro dynamic (EHD) bearing calculations of one of the

Fig. 28: T/C cut-out valve (compressor side)

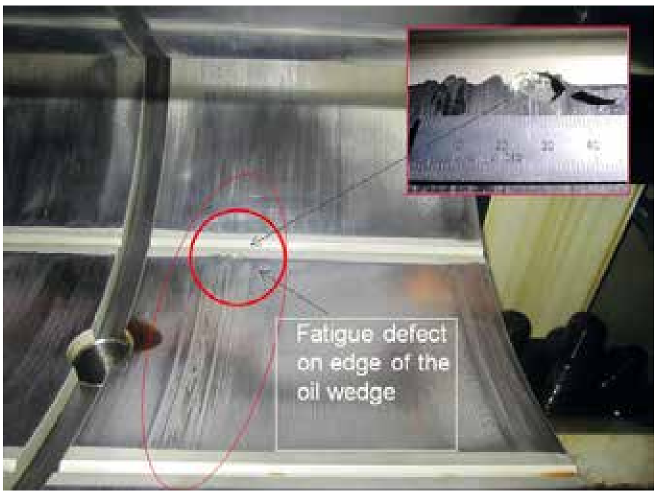


Fig. 29: K98 crosshead bearings and T/C cut-out, minor fatigue damage

designs being tested, i.e. the design with increased circumferential distance between the axial oil grooves. Larger oil 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 an endoscopic method. We recommend using this method, for example before a scheduled dry docking of a vessel with K98 and T/C cut-out.

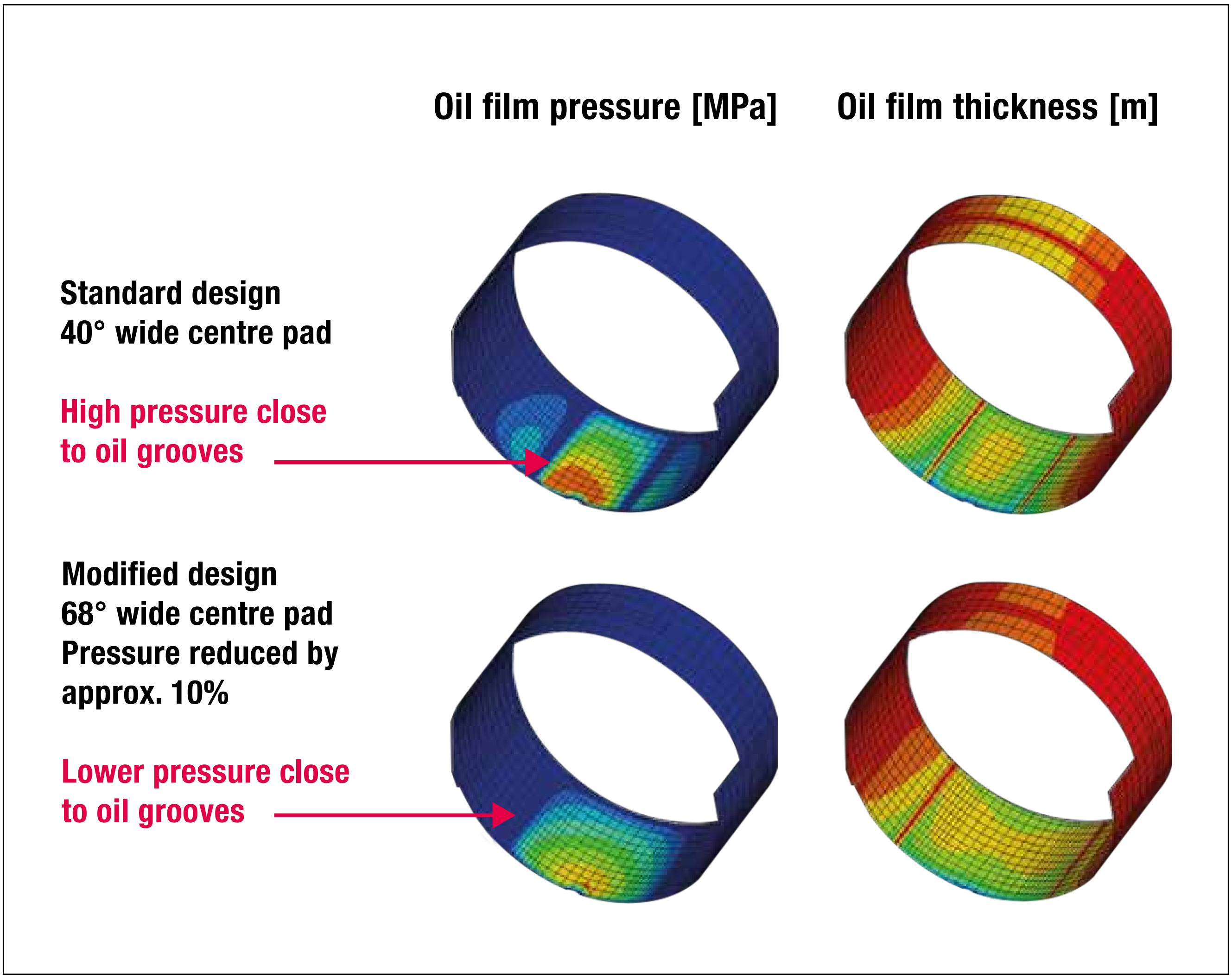


Fig. 30: K98 crosshead bearings and T/C cut-out: modified bearing shell design

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 turbochargers have been cut out. In such a case, a 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. 31.

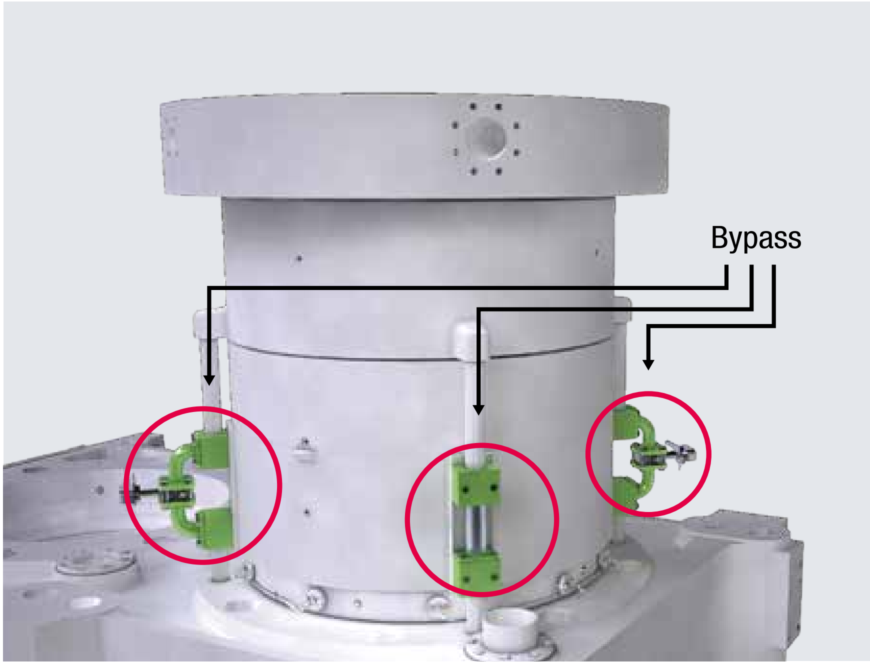


Fig. 31: Jacket cooling water bypass on a K98 engine

With this system, normally up to 85% of the jacket cooling water is bypassed the cylinder liner cooling bores.

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 more frequent inspection intervals to be able to judge the so-called "burn-away" of the exhaust valve spindle disk.

Fig. 32 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. So in this case, earlier inspection of the exhaust valve could have made reconditioning possible.

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

Fig. 32: Burn-away of spindle disc during low-load operation

Table 2 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 have issued a Service Letter (SL2013-573) on exhaust valve condition during low-load operation. Examples showing the influence of T/C cut-out, see Fig. 33, operation down to 10% load as well as the cut-in point for the auxiliary blowers, see Fig. 34, 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.

We have changed our max. burn-aways, see Table 3.

With these new limits, three times (instead of two times) reconditioning and the new limit for burn-away, the spindle lifetimes illustrated in Table 2 will be as shown in Table 4.

As can be seen, three times reconditioning combined with new burn-away limits will, to a large extent, mitigate the shortening influence of low-load operation on the exhaust valve spindle lifetime.

Engine type	Burn away	Running (hrs.)	Burn away rate	Spindle lifetime
K98ME	9 mm	14,000	0.64	61,000 hrs.
S60MC-C	11 mm	26,000	0.43	64,000 hrs.
K98MC-C	7.5 mm	15,000	0.50	78,000 hrs.
S90MC-C8	14 mm	15,000	0.93	39,000 hrs.

Burn away rate = mm / 1,000 hrs.

Normal lifetime = 100,000 hrs. including reconditioning of spindle

Table 2: Previous spindle lifetimes (examples)

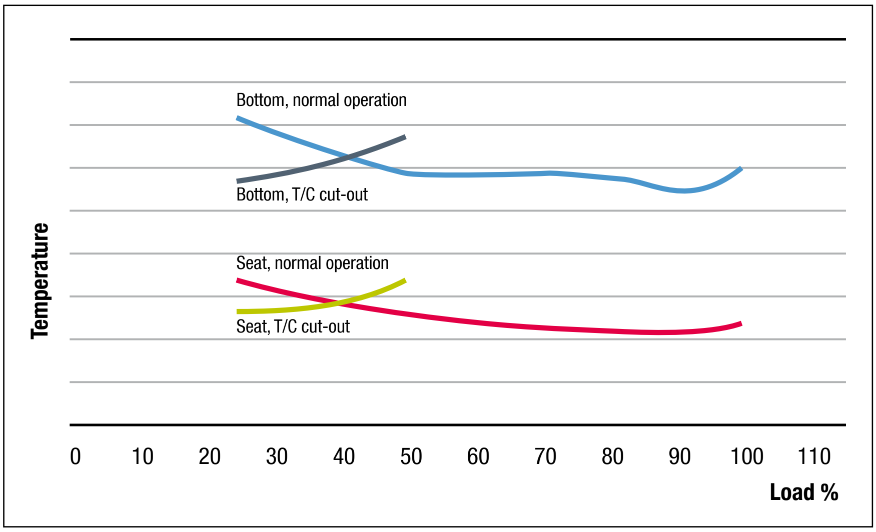


Fig. 33: Exhaust valve temperatures, 8K90MC-C

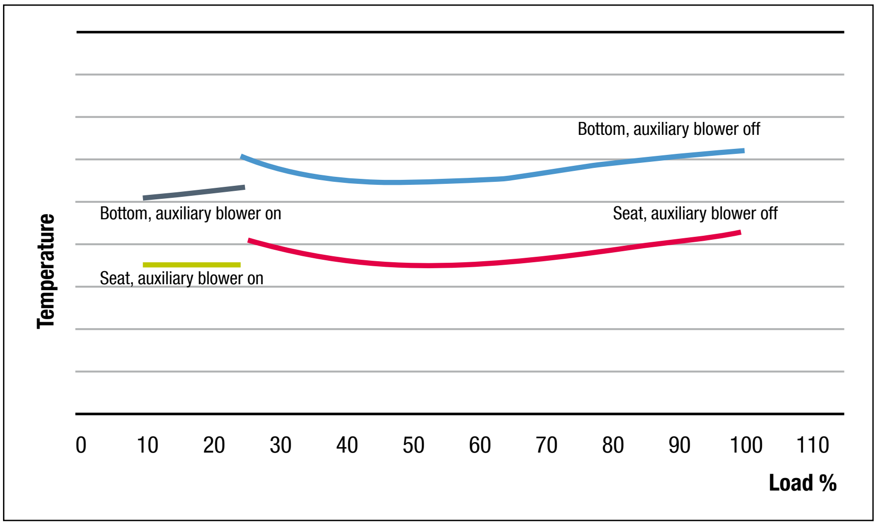


Fig. 34: Exhaust valve temperatures, 10S90ME-C9.2

Engine type	Previous	New
60	9 mm	Unchanged
70	10 mm	Unchanged
80	11 mm	14 mm
90	12 mm	17 mm
98	13 mm	20 mm

Table 3: Maximum burn-aways

Engines	Burn-away	Running hours	Spindle lifetime
K98ME	9 mm	14,000	124,000 hrs.
S60MC-C	11 mm	26,000	85,000 hrs.
K98MC-C	7.5 mm	15,000	160,000 hrs.
S90MC-C8	14 mm	15,000	73,000 hrs.

Table 4: Updated spindle lifetimes (examples)

Exhaust Gas Recirculation Service Experience

Exhaust gas recirculation (EGR) systems have been service tested for the last couple of years on MAN B&W two-stroke engines. It started with the retrofitted EGR system on the feeder container vessel *Alexander Maersk,* and continued early in 2013 with the first fully engine-integrated EGR system on the 4,200 teu container vessel *Maersk Cardiff*. In both cases, the EGR systems were tested while the engines were running on high-sulphur heavy fuel.

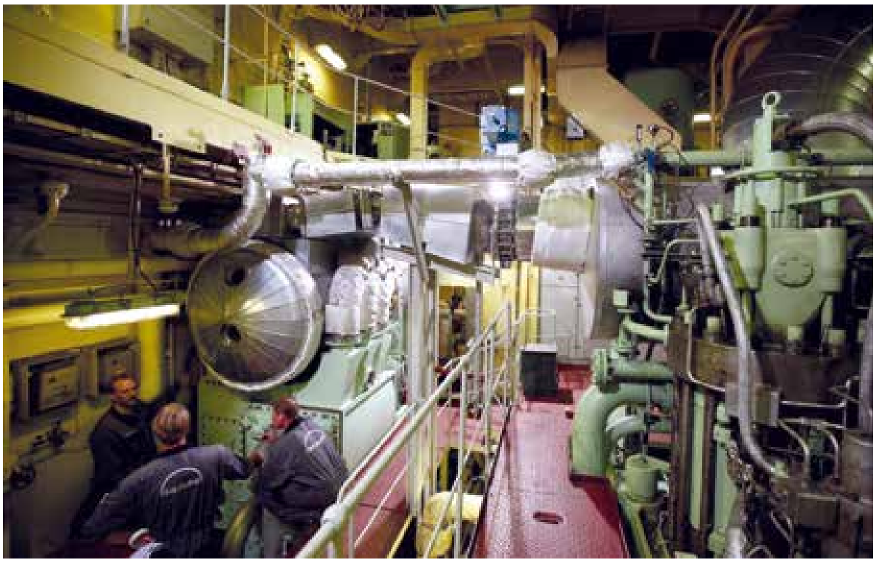


Fig. 35: EGR service test on Alexander Maersk with 3% sulphur HFO

The test on *Alexander Maersk* was completed at the end of January 2014. By that time, the EGR system (Fig. 35) had clocked about 2,700 hours of operation. In the beginning, the tests were interrupted by various technical challenges both on the EGR system itself and on the water treatment system (WTS). As a curiosity, it is worth mentioning that the tests were actually interrupted for a three-month period during which *Alexander Maersk* was used as the main scene for the Hollywood movie "Captain Phillips" starring Tom Hanks.

During the testing period on *Alexander Maersk*, we confirmed the expected NO_x reduction of 60%, see Fig. 36. Furthermore, after a number of modifications, good and stable operation was confirmed for the various subsystems of the EGR and the WTS. Fig. 37 illustrates some of the areas where satisfactory condition was obtained:

- Good (unchanged) cylinder condition
- Good performance of the main engine cooler
- Good condition for the EGR housing
- Well-performing water mist catcher
- EGR control system verified
- Water treatment system delivering clean water for bleed off.

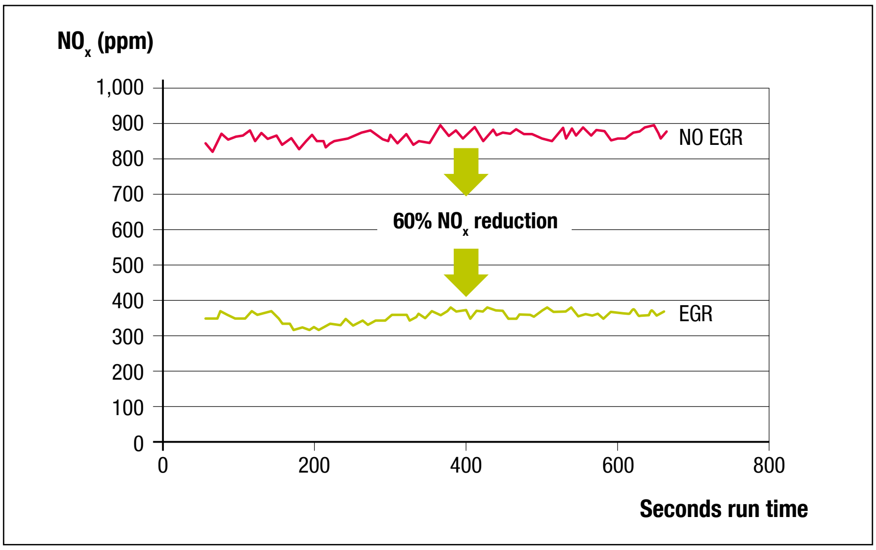


Fig. 36: EGR service test on Alexander Maersk with 3% sulphur HFO

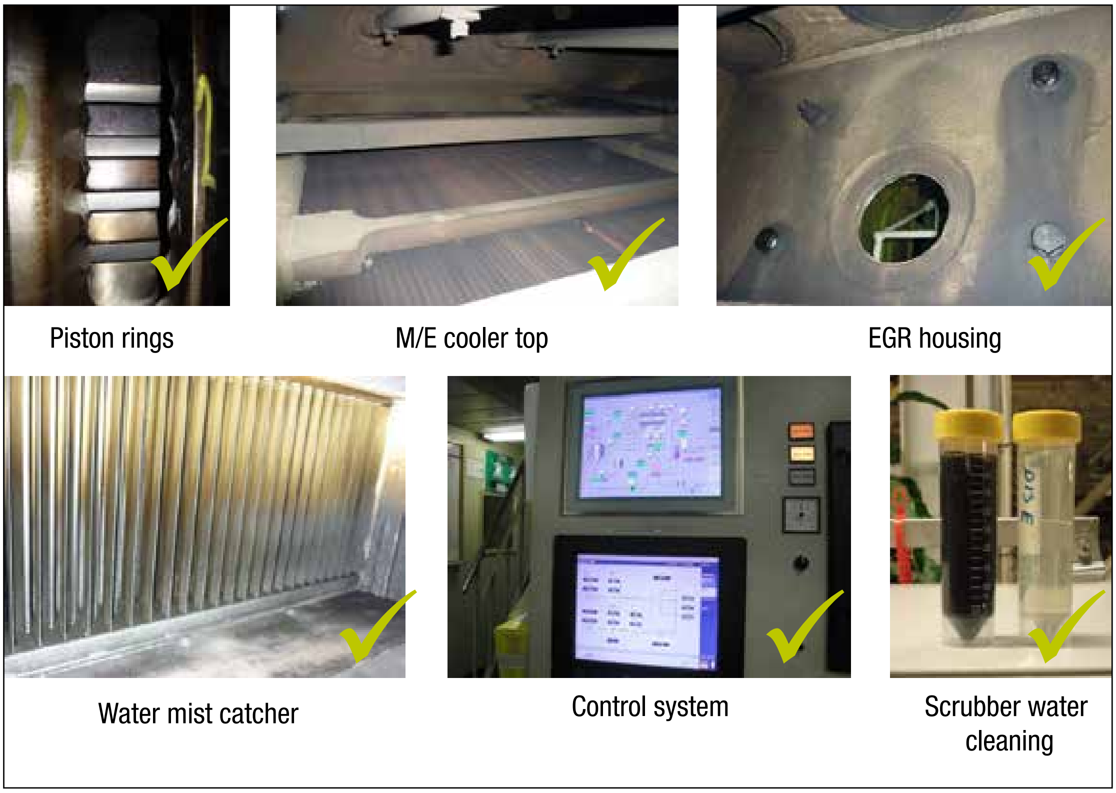


Fig. 37: EGR service test on Alexander Maersk with 3% sulphur HFO

The world's first fully engine-integrated EGR system was shop tested on a 6S80ME-C9.2 engine at HHI-EMD in Ulsan, Korea, in October 2012, see Fig. 38.

EGR test runs were completed in January 2013 during sea trial on the vessel *Maersk Cardiff* which, in the meantime, had had the 6S80ME-C9.2 EGR-engine installed. The EGR and WTS systems were fully commissioned during trials in March 2013 and, at the time of writing (mid-January 2014), the system has clocked more than 1,000 hours in operation. From the start, the system has been operated by the crew via the control system integrated in the standard ME engine control system. Only minor issues needed to be addressed during the first stage of operation.

During EGR operation, we have confirmed Tier III compliance in the so-called "high-EGR" mode, where approx. 40% of the exhaust gas is recirculated. Furthermore, the fuel benefit has been confirmed in the Tier II (non-ECA) "low-EGR" mode, where approx. 20% of the exhaust gas is recirculated.

One item which has been observed is "turning" of the upper pre-scrubber nozzles, see Fig. 39. A new welded nozzle-flange design has been introduced to prevent the nozzles from "turning" during operation.

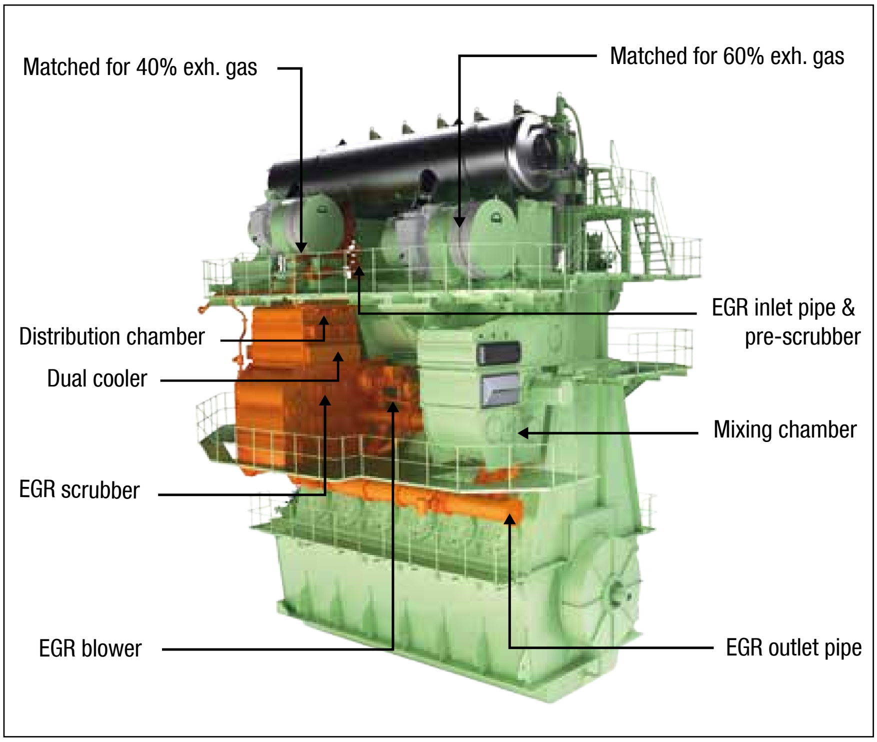


Fig. 38: MAN B&W two-stroke diesel EGR engine

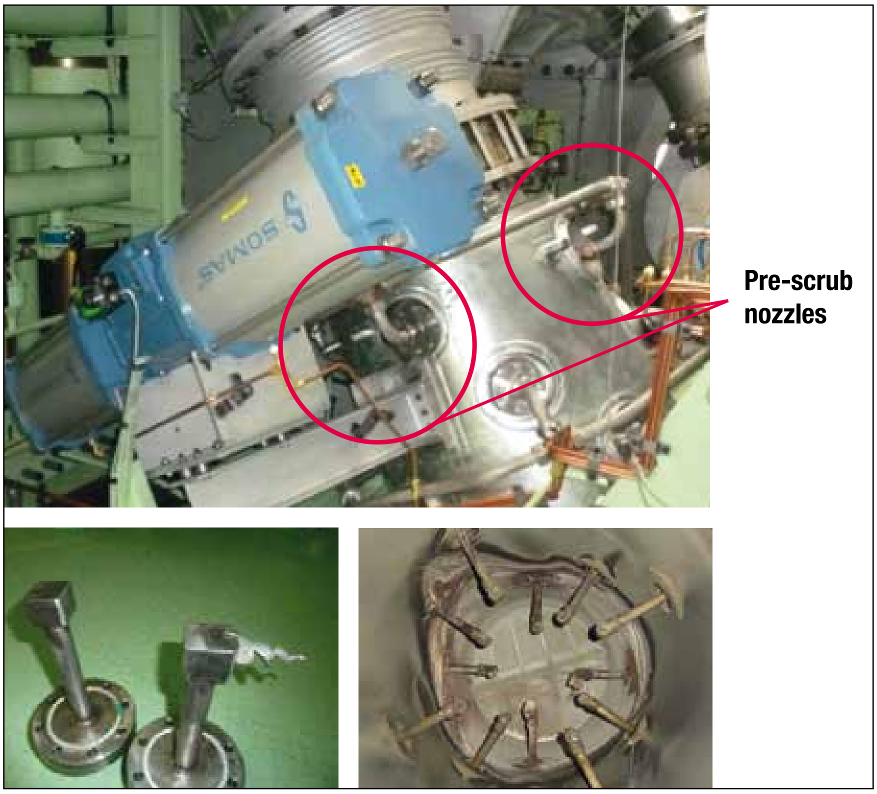
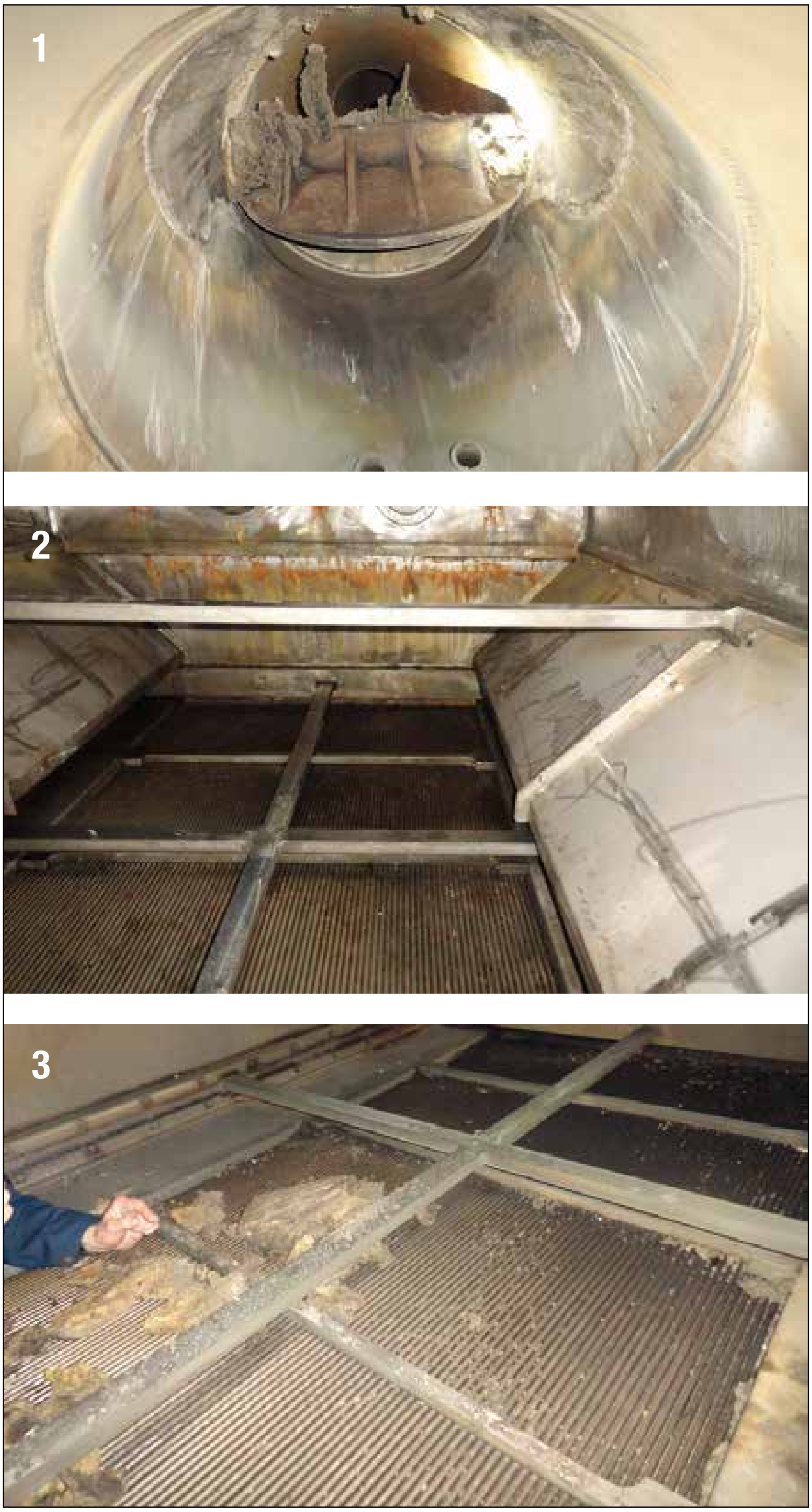


Fig. 39: Pre-scrubber nozzles

Another observation is the salty deposits (Na_2So_4) created on the turbocharger cut-out valve during EGR operation. When starting non-EGR operation with open T/C cut-out valve, these deposits end on the top of the EGR cooler. When switching back to EGR operation, the deposits on the EGR cooler are dissolved by water. This sequence is illustrated in Fig. 40. In order to eliminate the formation of these salty deposits, we are presently working to change the flow at the T/C cut-out valve.



1: During EGR running, scrubber water hits the T/C cut out valve and forms salty deposits.

2: During non-EGR running, salty deposits end on top of EGR cooler.

3: During the next EGR running period, the salty deposits will be dissolved by water.

Fig. 40: EGR cooler and T/C cut-out valve

In the initial phase of EGR operation, greasy oily lumps have accumulated in the EGR blower suction chamber and at the blower inlet, see Fig. 41. No operational troubles were observed due to this occurrence. However, in order to stop this formation we blocked the non-return valves to the scavenge air receiver in way of the EGR unit, see Fig. 42. After this modification, a good condition has been confirmed for the EGR blower suction chamber and blower inlet, see Fig. 43.

In the meantime, we have shop tested the first commercial Tier III EGR two-stroke engine, the 6G70ME-C Mk 9.2 for Chevron in December 2013 at Doosan in Changwon, Korea. Further optimisation and EGR-testing took place in the shop on the second engine for Chevron in February 2014. These engines will enter into service later in 2014, and be in operation mainly on the US west coast for lightering service in these waters.

Emergency Running of 6G70ME-C9.2 without Turbocharger

During testing of the first 6G70ME-C9.2 engine, the emergency running demonstration with the turbocharger cut out turned out to be difficult. A demonstration of emergency running is required at loads of, typically, up to 15% on single turbocharger engines.

The development of the high-rated 9.2 engines with the so-called two-stroke Miller timing has resulted in smaller turbochargers with more narrow passages especially on the turbine side, see Fig. 44. This means that when the dem-

Fig. 41: EGR-blower suction chamber greasy oily sooth lumps accumulated

Fig. 42: Blanking of non-return-valves

Fig. 43: Suction chamber and blower inlet found in good condition

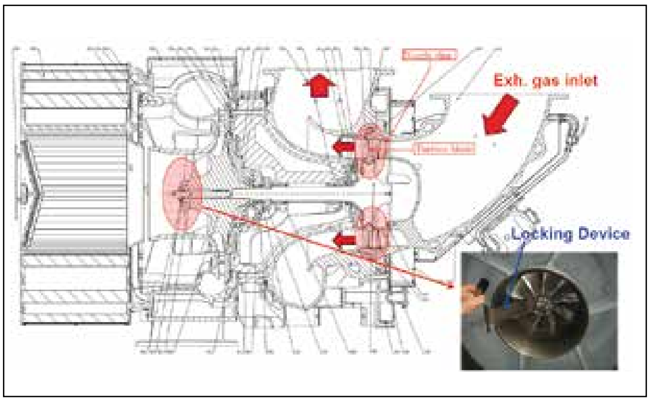


Fig. 44: Narrow passages on the turbine side

Fig. 45: T/C cut-out test and black carbon deposits

onstration of the emergency running mode was carried out, the amount of air available was insufficient to achieve a satisfactory combustion of up to 15% load. The air amount available is shown in Table 5 for various recent versions of the 70-bore engines. Critically low values for air amount and oxygen content are seen for the various version layouts for the 6G70ME-C9.2 engine.

The attempt to run the 6G70ME-C9.2 engine in the emergency running mode without turbocharger resulted in a lot of black smoke and a large amount of black carbon deposits in the combustion chamber, both as a result of the engine trying to reach the required power by “over-fuelling”, see Fig. 45.

A solution of the problem has been to establish a Ø444-mm emergency bypass, see Fig. 46, to prevent exhaust gasses from passing the turbocharger turbine in the emergency running mode. The air amount is then increased to 7.9 kg/kWh at 15% load. This is in line with what has previously been found acceptable for other 70 bore engines, see Table 5. Tests have confirmed that the emergency bypass is working well, and

Engine	Rating	Emission	Remark	Turbine area / SMCR cm^2 / MW	T/C cut-out, 15% SMCR Air amount kg / kWh	O^2 in exhaust gas Volume % in wet gas
6S70ME-C7.1	L1	Tier I		50	6.2	10.4
6S70ME-C8.1	L1	Tier I		49	6.1	10.1
6S70ME-C8.1	L1	Tier II		44	5.6	9.3
6S70ME-C8.2	L1	Tier II		37	4.8	7.5
6G70ME-C9.2	L1	Tier II		34	3.9	4.5
6G70ME-C9.2 With EGB PL tuning	15,536kW @ 73.9 rpm	Tier II	Closed EGB	31	3.7	3.9
6G70ME-C9.2 With EGB PL tuning	15,536kW @ 73.9 rpm	Tier II	Open EGB	31	3.9	4.8

Critically low values / too low values

Table 5: Air amount when running without T/C at 15% load for various 70-bore engines

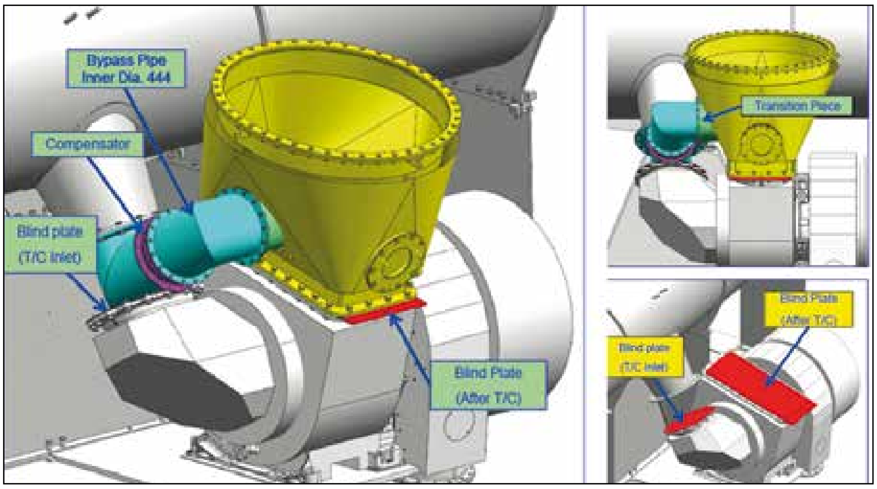


Fig. 46: Bypass pipe is installed between gas receiver outlet and transition piece

that black carbon deposits are avoided in the combustion chamber, see Fig. 47.

At the time of writing, we are testing another emergency procedure for running without turbocharger. To reduce the pressure-drop on the turbine side, this procedure involves removal of the nozzle ring on the turbine side.

Fig. 47: T/C cut-out test with emergency bypass installed. No black carbon deposits visible.

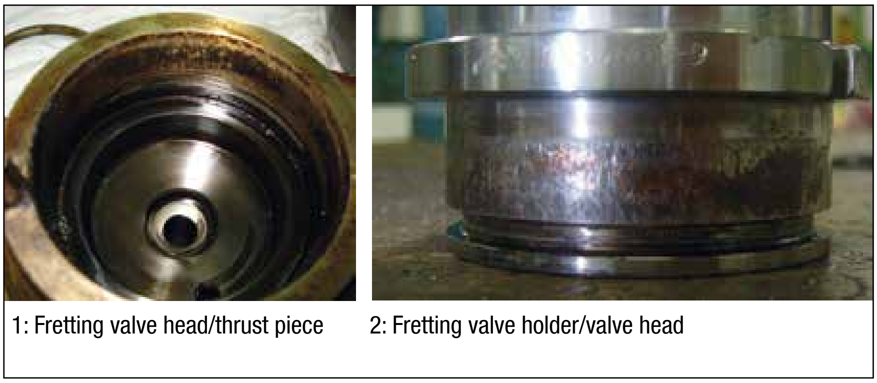


Fig. 48: Fuel valve service experience ME engines

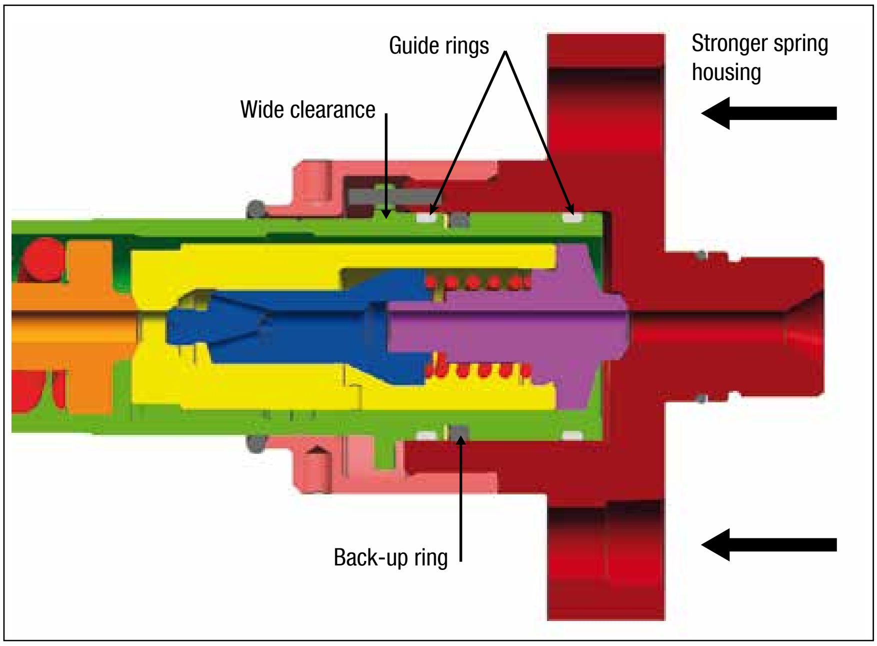


Fig. 49: Test Design 1: fuel valve with guide rings

Fuel Equipment

Fuel injectors for the 300-bar hydraulic pressure ME engines have caused some difficulties in relation to achieving satisfactory overhaul intervals. Initially, the rapid injection rate possible with the 300-bar system caused breakdown of various components inside the fuel injector such as shim breakage, cavitation of thrust spindle, spring breakage, thrust piece breakage and wear of the thrust foot. These issues have been overcome by various design changes of the components. Furthermore, the fuel injection profile has been modified to a softer profile limiting the force on the various components, which has contributed to reducing/eliminating some of the issues.

However, especially on some large bore engines, we still see fretting occurring in the top of the fuel injectors, see Fig. 48. We are therefore testing various design modifications to eliminate this fretting. At the time of writing, the most promising test result comes from a fuel injector with guide rings in the top part of the injector, see Fig. 49. This design also involves application of stronger

Fig. 50: Fuel valve with guide rings. Inspection after 2,000 hours in service on an S90ME-C9.2

spring packages for tightening down the fuel injector. Results from service testing on an S90ME-C9.2 engine indicate that this new design eliminates the fretting problem. Fig. 50 shows an injector disassembled after some 2,000 hours – no fretting can be seen. If confirmed by further inspections during 2014, we will introduce the new injector design not only on the S90ME-C, but also on other large bore ME/ME-C engines, as example the K98ME/ME-C type engines, suffering from fretting in the upper part of the fuel injectors.

Cavitation in Hydraulic Exhaust Valve Actuation System

We have received reports of cavitation damage found in various areas in the standard 300-bar low force exhaust valve actuation system. At the time of writing, we are conducting tests to find a solution to this issue. Fig. 51 indicates the areas where cavitation has been found. The extent of cavitation on the thrust piece can be significantly reduced by using a new material. Cavitation has also been found around inlet holes, cooling bores and relief grooves in the actuator top cover, see Fig. 52. We are currently testing some modified actuator top covers, and Fig. 53 shows two alternative test executions where the inlet holes have been redesigned.

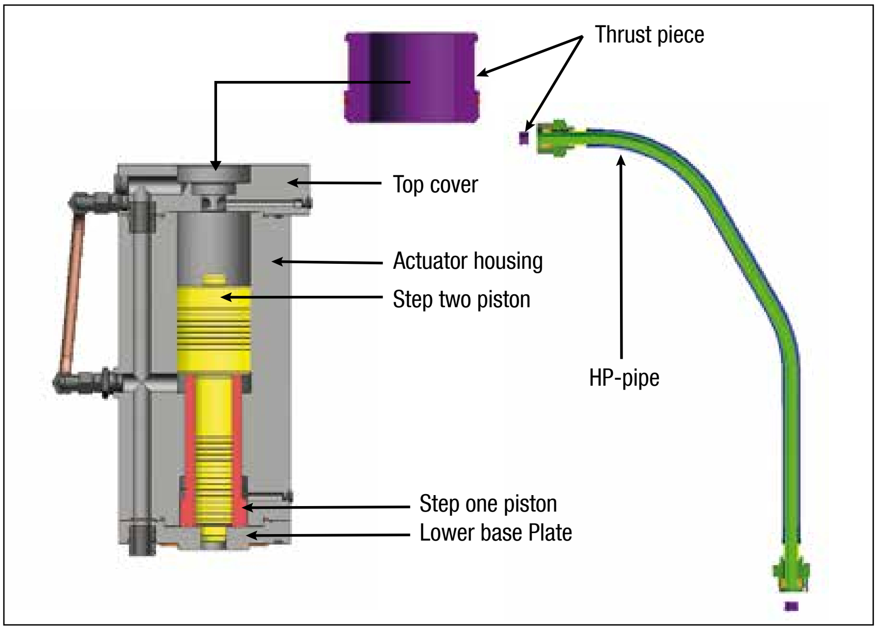


Fig. 51: Points of interest on exhaust actuator – low-force 300-bar

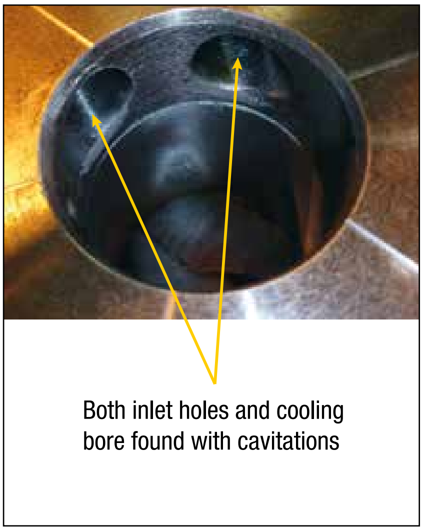


Fig. 52: Exhaust actuator – low-force top cover

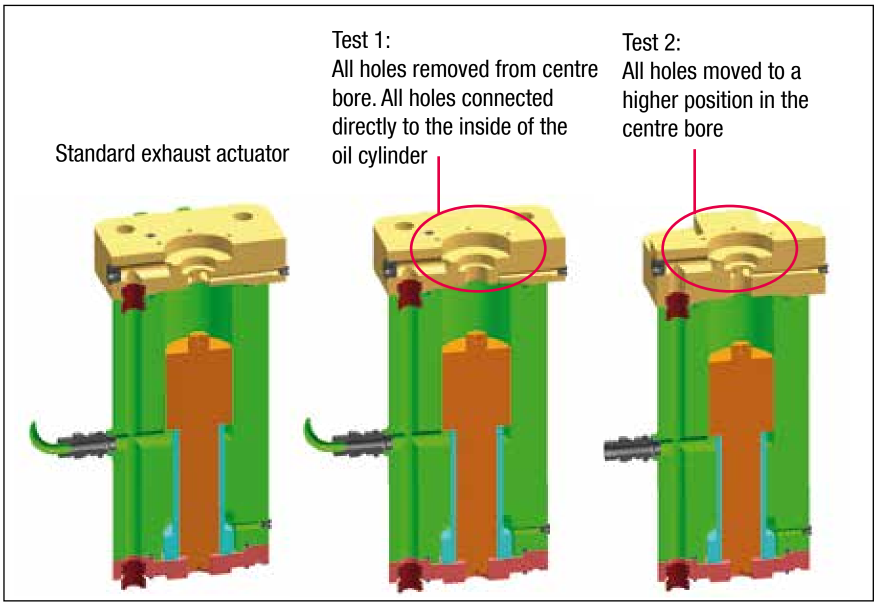


Fig. 53: Tests against cavitations, modified top covers

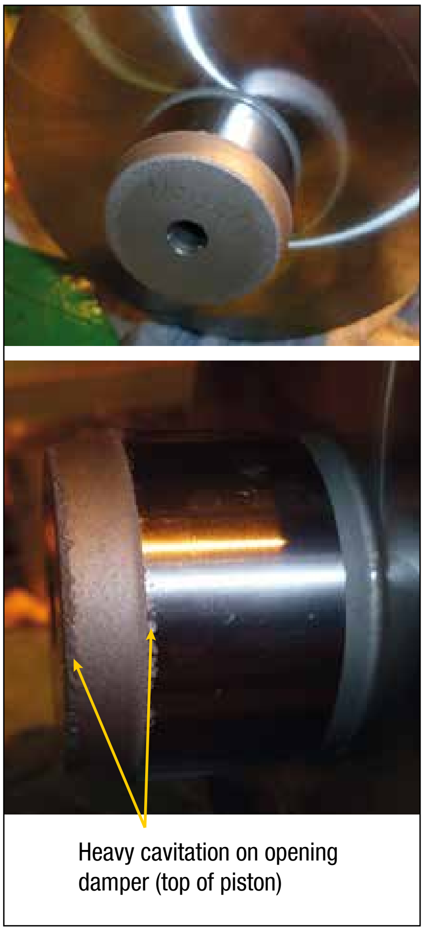


Fig. 54: Exhaust actuator – low-force step-two piston

The actuator housing is in perfect running condition, and only minor cavitation has been found on the step-one piston landing surface at the lower base. On the other hand, on the step-two piston we have found rather heavy cavitation damage on the opening damper at the top of the piston, see Fig. 54. We have tested a piston with a bolted-on "damper-nose" consisting of a harder alternative damper material (S85W6Mo). As can be seen in Fig. 55, this solution has eliminated the cavitation on the damper. Both actuator pistons show excellent condition on the running surface. Fig. 56 shows a typical condition for the running surface of the step-two piston.

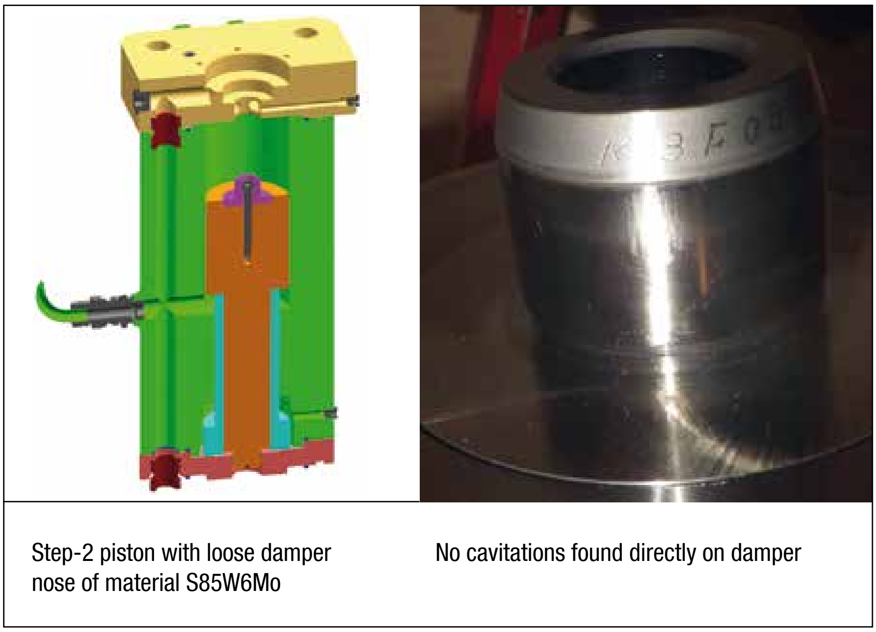


Fig. 55: Tests against cavitations, damper nose of different material

We have also found cavitation damage inside the high-pressure pipe, especially in the lower pipe bend. We are presently testing various design changes to overcome this problem.

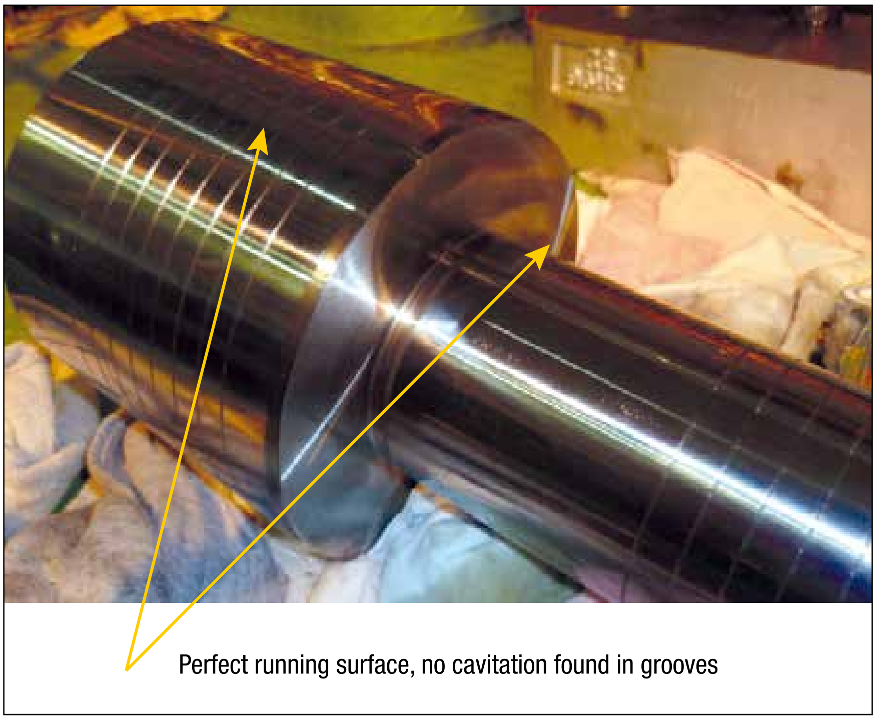


Fig. 56: Exhaust actuator – low-force step-two piston

Service Experience for Main Hydraulic Pumps on ME Engines

The main hydraulic pump is a central component of the ME engine system. It has generally demonstrated very satisfactory performance results achieving overhaul intervals in the vicinity of 32,000 hours. The main hydraulic pump has a lifetime similar to the engine lifetime.

Good service experience has been obtained with the first brand of main hydraulic pump introduced – the Bosch Rexroth brand.

A few years ago, we introduced an alternative brand – the Eaton Hydrokraft brand. This move was taken in order to have two suppliers of main hydraulic pumps to the ME engines.

However, the Eaton pumps showed service-related troubles mainly related to two areas:

- The feedback sensor for swash plate position
- The swash plate bearing shells.

The feedback sensor problem has been addressed by introducing a new touchless sensor principle.

For the bearing shells of the original design, the use of steel shells with a thin layer of polymer has not resulted in sufficiently reliable service experience with satisfactory overhaul intervals. Fig. 57 shows an example of a damaged pump – the root cause of the damage is the steel bearing shells.

Fig. 57: ME engine Eaton main hydraulic pumps. Pump swash plate/piston assembly, bearings and holder damaged

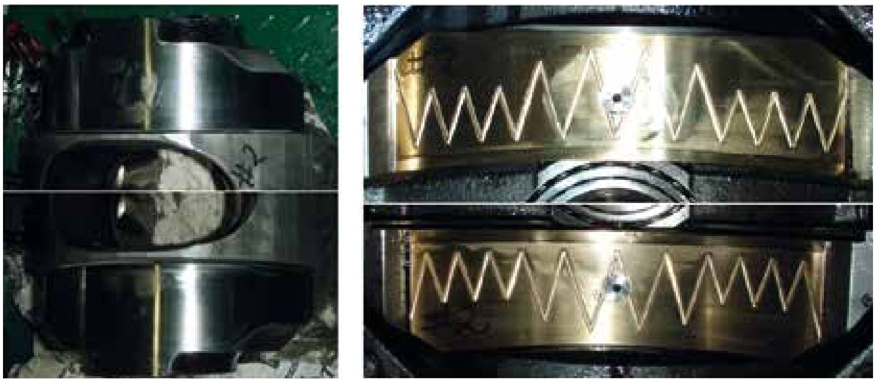


Fig. 58: ME engine Eaton main hydraulic pumps

Based on the poor service experience logged, we decided to stop specifying Eaton pumps as an alternative supplier. However, in parallel we have initiated service tests, together with Eaton, of bearing shells of a new design where steel is substituted by brass. At the time of writing, the initial inspection results look promising, see Fig. 58.

If later inspections continue to show positive results, it is the intention to begin specifying the Eaton pump as an alternative brand in 2014.

Conclusion

This paper gives an insight into many of the concurrent operational issues that we have experienced and investigated on our MAN B&W two-stroke low speed engines. Obviously, focus is on issues with the new successful ME engine generation of the S and G types.

Especially in relation to cold corrosion, significant progress has been made in order to suppress this phenomenon. The introduction of BN 100 cylinder oils and modified jacket water cooling systems have successfully counteracted the influence of cold corrosion on cylinder liners on the latest generation of engines. As described, further work to optimise the cylinder liner wall temperature by cylinder liner design is ongoing at time of writing.

We firmly believe that we can demonstrate efficient solutions to even the most challenging operational issues on engines in service. It is important to note that MAN Diesel & Turbo is geared and fully ready to cope with the next series of challenges related to Tier III, SO_x control, ME-GI and ME-LGI.

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