

Service Experience

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Service Experience

Introduction

As a service to our customers, MAN B&W Diesel have for many years, as part of our open information policy, at regular intervals published papers under the title 'Service Experience'.

The present paper gives an update on how the latest generation of MC and MC-C low speed engines, i.e. those having entered service after 1997, are performing. The major development steps are shown in Fig.1, which includes the engines of the recent new breed of large container vessels that have attracted special attention in the marine market, and therefore the large bore main engines of our range in these vessels will be dealt with in this paper.

From the start of the service periods of these large engines, we have made intensive follow-ups, including many visits on board the vessels. As a result, a number of modifications have been introduced.

It is important to underline the fact that, irrespective of the great attention being given to large engines, we are also carefully following up on all other sizes of MC/MC-C engines.

Hence, the full range of S-MC-C engines have been subject to the same intensive follow-up, primarily in order to detect and cure any teething troubles as early as possible.

Cylinder Condition

The cylinder condition of the MC/MC-C engines has developed in a most positive direction in recent years. Contributing to this are the Oros combustion chamber, high topland pistons, piston cleaning (PC) ring, controlled pressure relief (CPR) piston ring, alucoat, etc. See Fig. 2.

With the Oros configuration, the combustion air is concentrated around the fuel nozzles and the distance from the fuel nozzles to the piston top is increased. This has resulted in lower heat load on the piston top and basically unchanged heat load on the cylinder cover and exhaust valve.

The higher topland and the introduction of the piston cleaning ring have proved to be very beneficial for avoiding a build-up of lube oil derived deposits on the

piston topland. Such deposits would, by hard face sponge effect, scrape off and absorb the oil film, leaving the naked liner wall vulnerable to extensive wear and/or scuffing.

The use of the high topland piston also means that the mating surfaces between the cylinder liner and the cylinder cover are lowered, thus reducing the thermal load on the cylinder liner and improving the conditions for lubricating the liner. This was taken into account before introducing the Oros configuration.

Years back a number of cracked cylinder liners were reported and, consequently, countermeasures were introduced. Today, we can state that these countermeasures, in terms of bore-cooled liners, and for smaller engines, slim liners, have been successful, in that cracked liners are very rarely reported. For engines originally specified with cast-in cooling pipes in the liners, the design with oval pipes has stopped the crack occurrence.

Alu-coating of piston rings and wave-cut in cylinder liners

Controlled breaking-in and subsequent running-in of piston rings and cylinder liners are considered very important in achieving good performance of a liner later on in service.

New piston rings and a fresh liner surface have to adapt to each other in the breaking-in period to optimise the surface for running-in. An ample amount of cylinder lube oil is therefore needed initially.

However, besides providing the necessary lubrication, this high cylinder lube oil amount will create deposits and thus a more difficult condition for piston ring performance.

With the introduction of the alu-coated piston rings, a thin layer of alu-coating will be worn off the rings during the first

	Mk	mep (bar)	Cm (m/s)	P _{max}
1982 Full L-MC programme	1	15.0	7.2	125
1986 K-MC introduced	3	16.2	7.6	130
S-MC introduced	3	17.0	7.8	130
L-MC upgraded	3	16.2	7.6	130
1988 K-MC-C introduced	3	16.2	8.0	130
1991 MC programme upgraded	5	18.0	8.0	140
1994 K98MC-C introduced	6	18.2	8.3	140
1996 S70MC-C, S60MC-C, S50MC-C and S46MC-C introduced	7	19.0	8.5	150
	7	19.0	8.5	150
1998 K98MC introduced	6	18.2	8.3	140
S80MC-C, S90MC-C, L90MC-C introduced	7	19.0	8.1	150
S35MC upgraded	7	19.1	8.1	145
2001 L70MC-C introduced	7	19.0	8.5	150
L60MC-C introduced	7	19.0	8.3	150

mep = mean effective pressure • Cm = mean piston speed • P_{max} = maximum firing pressure

Fig. 1: Major development steps

<p>Features:</p> <ul style="list-style-type: none"> • High topland • Oros shape of piston top • CPR top piston ring • Alu-coat piston rings • Bore cooled, forged piston of heat resistant steel • Piston cleaning (PC) ring <p>Improvements:</p> <ul style="list-style-type: none"> • Approx. 100 °C lower temperature on top compared with former type piston • Elimination of Inconel coating on piston top • Increased chrome layer thickness in bottom of ring grooves • Anti-erosion bushing in oil outlet in piston rod foot <p>Verification:</p> <ul style="list-style-type: none"> • Extensive calculations • Comprehensive tests on K90MC, K90MC-C and K98MC 	
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Alpha Cylinder Lubricator, cylinder oil savings

One of the tools for optimising the cylinder condition is optimising the cylinder lube oil dosage.

Examples show that over-lubrication (above 1.6 g/bhph) can be damaging to the cylinder condition, and by analysing the correlation between the fuel sulphur level, the lube oil BN and the wear rate as measured from the composition of the drain oil from the bottom of liners, we are able to recommend the optimal feed rate for the engines.

In addition to being potentially detrimental, excessive cylinder lube oil consumption represents a large expense for engine operation. Therefore, the aim is to reduce the cylinder lube oil dosage while, at the same time, retaining satisfactory piston ring and liner wear rates and

Fig. 2: Oros combustion chamber geometry

1,000 - 2,000 hours. This makes it possible to reduce the breaking-in and running-in time, as well as the cylinder oil feed rate during most of the breaking-in period. This is beneficial not only to the engine manufacturer, who can reduce the time at engine delivery, but also to the operator who will receive an engine with improved running-in conditions.

The surface of MAN B&W cylinder liners is specified as semi-honed. This semi-honed surface, together with alu-coated piston rings, ensures safe and stable running-in. The semi-honing process cuts off the tops of the wave-cut, thus reducing the necessary breaking-in between piston rings and liner surface, while still having circumferential pockets for lube oil. We apply semi-honing only, in that the alu coated piston rings, see Fig. 3, do the remaining removal of broken or damaged cementite from the cylinder liner surface during the initial wear period, thereby performing a “free of charge” full honing.

<p>The combination of:</p> <ul style="list-style-type: none"> • Wave-cut, semi-honed liner • Alu-coated rings <p>Result in:</p> <ul style="list-style-type: none"> • Unproblematic production of cylinder liners • Safe and stable running-in 	
	<p>Alu-coated rings</p>

Fig. 3: Alu-coated piston rings + wave-cut

maintaining, or improving, the time between overhauls. This can be done by means of the Alpha Lubricator system which, by way of being computer controlled, provides means of feed rate control based on e.g. the fuel oil sulphur content, referred to as "Alpha Adaptive Cylinder oil Control" (Alpha ACC).

The principle of the Alpha ACC is shown in Fig. 4. The cylinder oil amount is controlled so that it is proportional to the amount of sulphur entering the cylinder with the fuel (the "Sulphur throughput").

Two basic needs have to be catered for:

1. The cylinder oil dosage must not be lower than the minimum needed for lubrication.
2. The additive amount supplied through the BN (the "Alkalinity throughput") must be only sufficient for neutralisation and for keeping the piston ring pack clean.

As the second criterion usually overrides the first one, the following two criteria determine the control:

- The cylinder oil dosage shall be proportional to the engine load (i.e. amount of fuel entering the cylinders)
- The cylinder oil dosage shall be proportional to the sulphur percentage in the fuel.

The implementation of the above two criteria will lead to an optimal cylinder oil dosage, proportional to the amount of sulphur entering the cylinders, but with a low minimum setting to ensure lubrication.

This principle is founded on the observation that the main part of cylinder liner wear is of a corrosive nature. Therefore, the amount of neutralising alkaline components needed in the cylinder should be proportional to the amount of sulphur (generating sulphu-

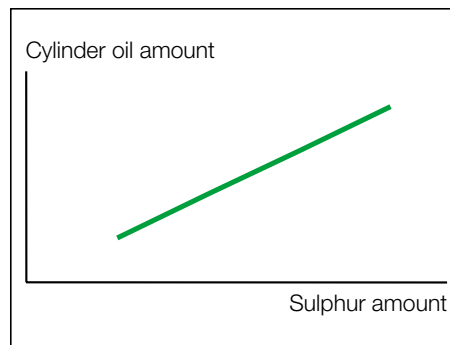


Fig. 4: Cylinder oil amount proportional to sulphur amount entering the cylinders (basic principle of Alpha ACC)

rous acids) entering the cylinders. A minimum cylinder oil dosage is set in order to account for other duties of the cylinder oil (securing sufficient oil film detergency, etc.). Thanks to its accuracy, the Alpha Lubricator does just that.

Fig. 5 shows control of cylinder oil dosage proportional to the sulphur percentage in the fuel. A minimum dosage of 0.5 g/bhph is indicated, based on experience so far. This minimum value is preliminary and, given the efficient lubrication achievable with the Alpha Lubricator system, we expect to be able to further reduce this minimum value in the future.

Alpha Lubricators can be retrofitted, and have a relatively short payback

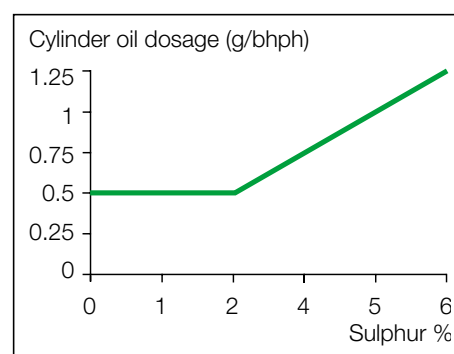


Fig. 5: Cylinder oil dosage proportional to sulphur percentage in the fuel

time. Thus, Alpha Lubricators are currently being retrofitted on a large number of plants in service.

Cylinder wear

Wear of the liners is monitored on a large number of engines through visits by our engineers, because liner wear is traditionally considered an engine quality criterion.

As can be seen in Fig. 6 for cylinder liner wear, both K98MC/MC-C and S90MC-C show very convincing low wear performance data and good liner condition.

While initial wear is naturally higher, the wear rate is reduced to less than 0.05 mm/1,000 hours after about 1,500 hours, which is considered very satisfactory.

Initial wear is part of the running-in of cylinder liners and piston rings, and high wear is expected in this period. The wear study is based on cylinder feed rates between 0.9 and 1.2 g/bhph. Experience with the Alpha Lubricator indicates that there is a significant potential for cylinder lube oil reduction while still having a fully acceptable cylinder wear rate and mean time between overhauls.

Fig. 7 shows extracts from our database regarding cylinder wear rates.

S-MC engines, S-MC-C engines and K-MC-C engines all experienced low and uniform wear rates.

On the K98 engines, as mentioned above, the cylinder condition has been good, with very low wear rates and low cylinder oil feed rates. However, on these engines the good condition has been temporarily overshadowed by a number of failures of the top piston ring. Though there are strong indications of this being a production-related issue, the top ring design has been upgraded to increase the safety margin against

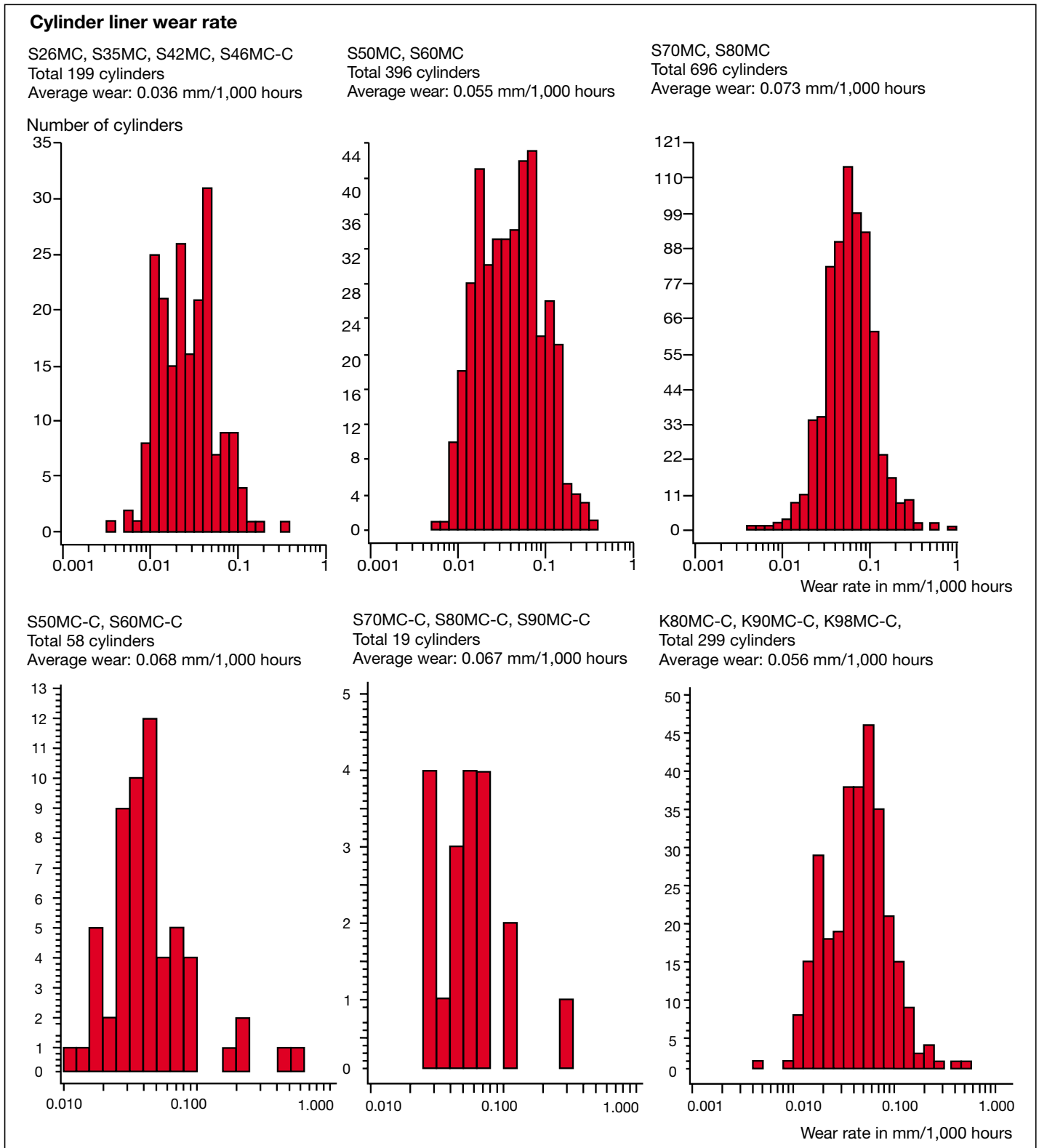


Fig. 7: Extracts from our database regarding cylinder wear rates

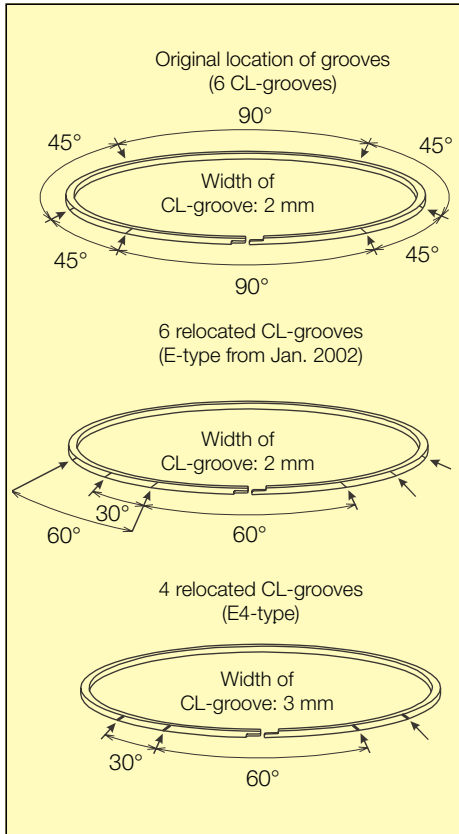


Fig. 8: K98 CPR piston ring development

Bearings

Main bearing

Thick shell bearing

Since 1998, we have seen a decrease in the number of reported main bearing failures. In 1998 a number of features were introduced in the design of the bearings, the adjustment of the bearings, and to the installation of the engine/shaftline.

The major updates can be summarised as follows:

As an evolution of the Mark 5 bearing type, the “Optimum Lemon Shape” type main bearing was introduced, see Fig. 9. This type features optimised (reduced) top and side clearances. Service experience has confirmed the efficiency of this bearing type simply by the significantly reduced number of reported damage incidents.

A revised engine installation recommendation (available upon request), including an updated shaftline alignment procedure together with differentiated bearing height in the aft end of the engine, has added to the safety margin.

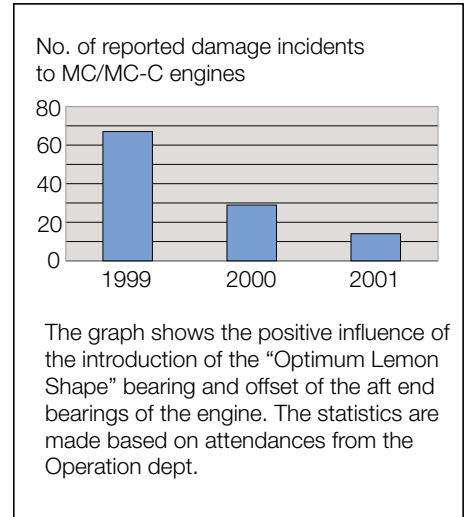


Fig. 10: Bearings – statistics

Optimum Lemon Shape (OLS) bearing with flexible edges

There has been a significant reduction in the number of main bearing failures, as will appear from Fig. 10. However, it must be acknowledged that main bearing damage sometimes still does occur, e.g. due to poor bonding of the bearing metal.

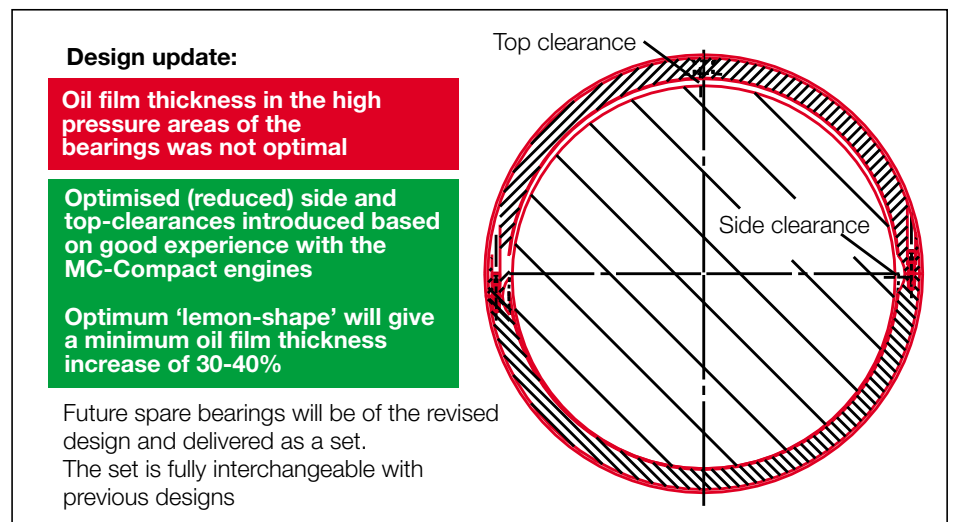


Fig. 9: Main bearing, thick shell design

S50MC-C main bearing lower shell –
18,000 hours



No remarks – excellent condition

Thickness measurement of shell did not reveal indication of significant wear rate – less than 0.01 mm

Fig. 11: Thin shell AlSn40 main bearing

In nearly all cases, main bearing damage is initiated from a fatigue crack at the edge of the bearing, the aft edge/ manoeuvring side being the most common point of initiation. Often, geometrical non-conformities are involved in these cases. Such non-conformities further increase the damage frequency as margins established during the design phase are reduced.

Calculations, combining the dynamics of the complete crankshaft with the hydrodynamic and elastic properties of

the bearing, have provided detailed information about the mechanisms leading to local loading of the main bearing edges. The calculations have indicated that a slight radial flexibility of the bearing edge will increase the overall minimum oil film thickness significantly. At the same time, the maximum oil film pressure will be reduced.

A bearing design with flexible forward and aft edges of the bearing shell has been successfully tested.

The flexibility has been achieved by removing the contact between the shell and bedplate at the end portions of the shell. The unsupported width of the shell is equal to the shell thickness. Apart from the flexible edges, the properties of the bearing are similar to those of the Optimum Lemon Shape type. This design provides a larger safety margin in the event of geometrical non-conformities.

Thin shell bearing

The thin shell bearing design has been introduced on our latest engine types. For the small and medium bore engines (S46MC-C to S70MC-C), the main bearings are lined with AlSn40 and provided with a running-in coating

consisting of PTFE as standard. 9,000 shells are now in service, many of which have been in service for more than a decade.

Service experience with this configuration has been excellent, see Fig. 11.

On the large bore engines, the bearings are lined with white metal, see Fig. 12.

In general, few damage incidents to our thin shell main bearings have been reported:

- Following Q/A inspections, weld rectification of bearing saddles has occasionally been carried out during the production of the engine. In a few cases, these welds have been introduced at a late stage (after final machining). This has provided either high spots on the running surface of the bearing shell or areas with missing contact. Both could disturb the oil film, leading to high local load, and thus fatigue damage, see Fig. 13. Countermeasures introduced have been either adaptation of the running surface of the bearing shell by means of scraping or the introduction of bearings with AlSn40 lining, offering an increased margin against fatigue
- Deformed bearing housings of the thin shell design have been seen due to the improper use of the triple and double hydraulic tightening jacks. This has occurred either prior to machining of the bearing housing or during mounting of the bearing shells. Such deformation changes the ratio between the side and the top clearances. New instructions and Service Letter SL02-400/HMH highlighting the features and the proper use of the jacks have been issued. In addition, a Quality Specification instruction has been issued, intended for use at the production facilities, thus assisting the licensees in preventing similar incidents in the future.



7K98MC main bearing No. 4 after 6,000 hours
No remarks – except for minor traces of dirt

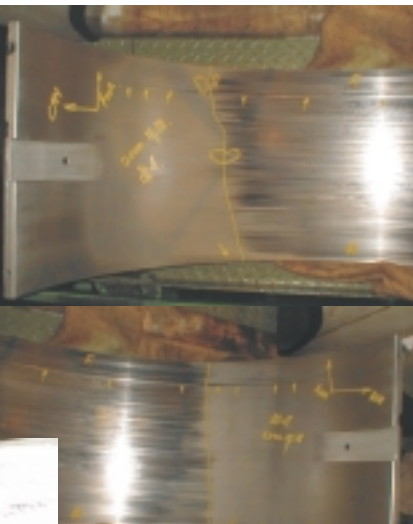
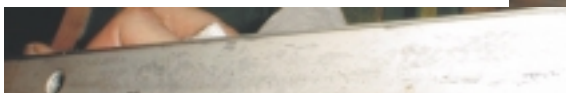


Fig. 12: K98MC-C white metal main bearing

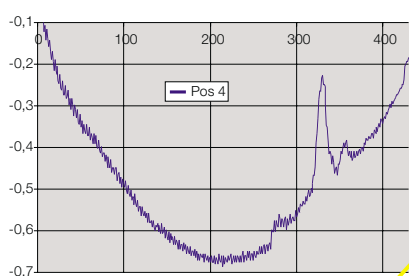
Alignment and vertical offset of aft-end bearings

In addition to other initiatives, attention has been paid to some cases of repeated damage to the aft-end bearings in the engine. This is, presumably, caused by missing static load, particularly in the

second aftmost main bearing during normal operating conditions.

On the basis of comprehensive investigations made together with a number of licensees, shipyards and classification societies, a new alignment procedure

has been introduced. The new procedure includes precalculated bedplate sagging as well as vertical offsets to the main bearing saddles. This has resulted in a significant drop in the number of reported damage incidents to the aft-end bearings in the engine.



Large rectified area – missing contact between shell and support. Maximum depth of rectified area 0.20 mm

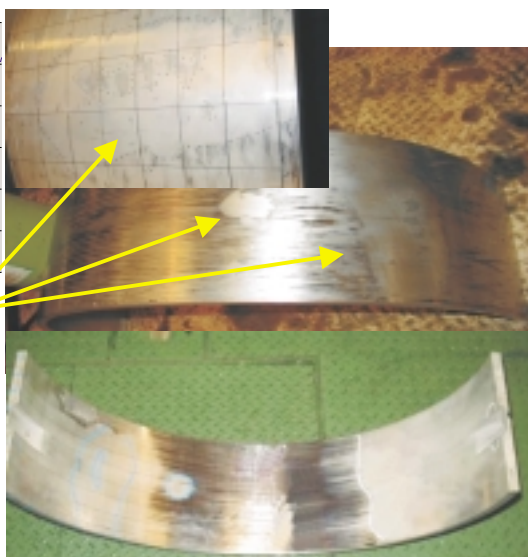


Fig. 13: Weld-rectified bearing support



S90MC-C crosshead # 6 (3,500 hours) before dressing up. Note that the lead overlay apparently has a tendency of non-conformity. In this case the problem is only cosmetic



S90MC-C crosshead # 6 (3,500 hours) after dressing up

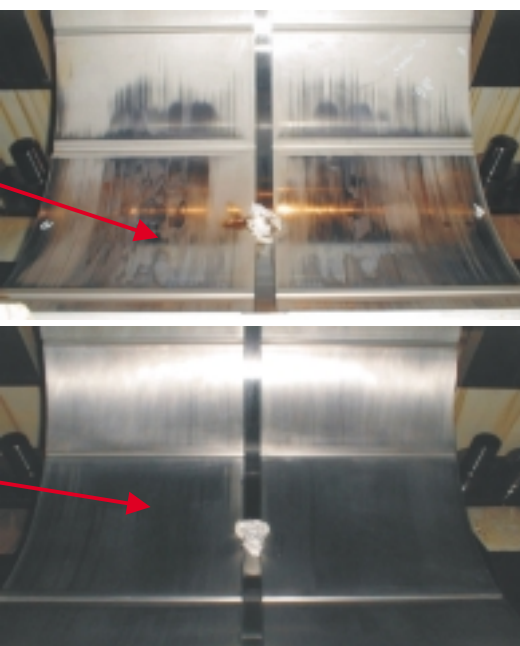


Fig. 14: S90MC-C crosshead bearing

Crosshead bearing

In general, the crosshead bearings in both MC and MC-C engines perform very satisfactorily, but cases of wiping have been observed. In itself, this wiping is of a cosmetic nature, but it can sometimes cause blockage of the oil-wedges that normally build up the oil film to the “pads” inside the bearing, see Fig. 14. Disturbance of the oil film build-up inside the bearing could result in slight fatigue damage just behind the blocked area of the oil-wedge. If the phenomenon is observed at an early stage during inspections, the problem is solved by removing the wiped lead from the oil-wedge.

Exhaust Valves

Nimonic spindles are well accepted now that the operators have become acquainted with the long-lasting seat performance despite dent marks, see Fig. 15. Nimonic spindles are standard for 50MC and for 60MC/MC-C upwards, and Stellite spindles are standard for smaller engines.

Corrosion in the valve housing is effectively minimised by the introduction of the optimised cooling water system, which raises the wall temperature in the housing above the critical level for the formation of acid on the gas side of the duct. With the high temperature level, a cast iron spindle guide bushing is necessary.

Wear of the previously chrome-plated spindle stem has been effectively reduced by the introduction of the HVOF-



Fig. 15: Nimonic valve normal condition, after 11,500 running hours

based cermet coating. All spindles from our Copenhagen works have been provided with this coating since 1997, and approximately 6,000 spindles have been produced. The very few claims received were due to initial production problems. The licensees are gradually also using the HVOF process.

Furthermore, wear and corrosion problems at the spindle guide/spindle/seal area, caused by combustion products, have been minimised over the years by design changes on the sealing air system. However, the stem seal has difficulties in reaching lifetimes similar to those of the valve seats. Tests have shown that improvement can be obtained by lubrication. It has, therefore, been decided to replace the sealing air system by a lubricating device, which has been long-term service tested with good results.

Sealing oil dosage unit

Good results have been obtained regarding the reduction of wear of the spindle stem (HVOF coating) and of the

long spindle guide (grey cast iron). However, the lifetime of the stem seal itself is still sometimes too short.

Tests with oil as the sealing medium, instead of air, have shown very low wear rates on the seals. At the same time, a high cleanliness level is obtained on the surfaces of the spindle stem and spindle guide.

A system delivering the necessary dosage of only approximately 1 kg/cyl./day has been developed for medium and large bore engines. It is located in the top of the exhaust valve and is fed by oil from the hydraulic system of the exhaust valve. The oil is fed to the spindle guide via a small pipe. The sealing oil is taken from the circulating oil and is thereby part of the necessary minimum oil consumption for keeping the system oil viscosity and BN-level at the prescribed equilibrium.

Cooling water leaks/U-seal

The latest design of exhaust valve on small and medium bore engines occasionally suffers from cooling water leaks at the lowermost O-ring between bottom piece and cylinder cover. Investigations resulted in replacing the lowermost O-ring with a special Teflon seal with spring back-up (U-seal). This was communicated by Service Letter SL01-389/PGK.

Seat geometry/W-seat

During our test work, results have shown that the best way to increase the seat

lifetime was by altering the seat geometry of the bottom piece to the patented W-seat, see Fig. 16. On a few S50MC-C engines with Al50 spindles, the results were remarkably better using the W-seat, which is now standard on all MC/MC-C engines. When using the new type of slide fuel valves as well, the result is even better.

Seat material standards for MC/MC-C Engines are summarised in Table 1.

Engines with Oros combustion chamber, hydraulic safety valve

Engines equipped with the so-called Oros combustion chamber leave little distance between the piston top and the underside of the exhaust valve spindle. Therefore, the usual extended lift system for release of high hydraulic pressure cannot be utilised. A safety valve located in the actuator is used instead.

Unfortunately, a few cases of damaged exhaust valves and camshaft sections have been reported, due to different external factors, including insufficient release action of the safety valve. Consequently, the safety valve has been re-designed. Once activated, it implements a special function to keep it open for about 20 seconds. Furthermore, a disc spring has been introduced in the exhaust valve on top of the spindle guide to avoid damage to the air piston in the event of 'over-shoot'/extended lift of the valve spindle. This has eliminated the problem. Fig. 17 shows the various design details mentioned.

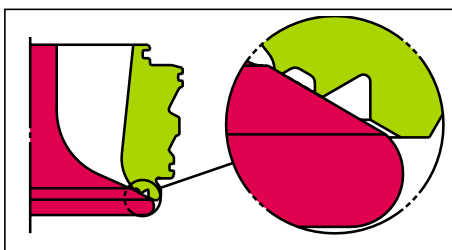


Fig. 16: W-seat, exhaust valve

Engine type	Spindle	Bottom piece
26-42MC	SNCrW/Al50 *	Cooled/hardened steel/W-shape
46-50MC-C	SNCrW/Al50 *	Semi-cooled/hardened steel/W-shape
50-98MC	Fully forged Nimonic 80A	Semi-cooled/hardened steel/W-shape
60-98MC-C		

* Nimonic can be delivered on request

Table 1: Seat material standards

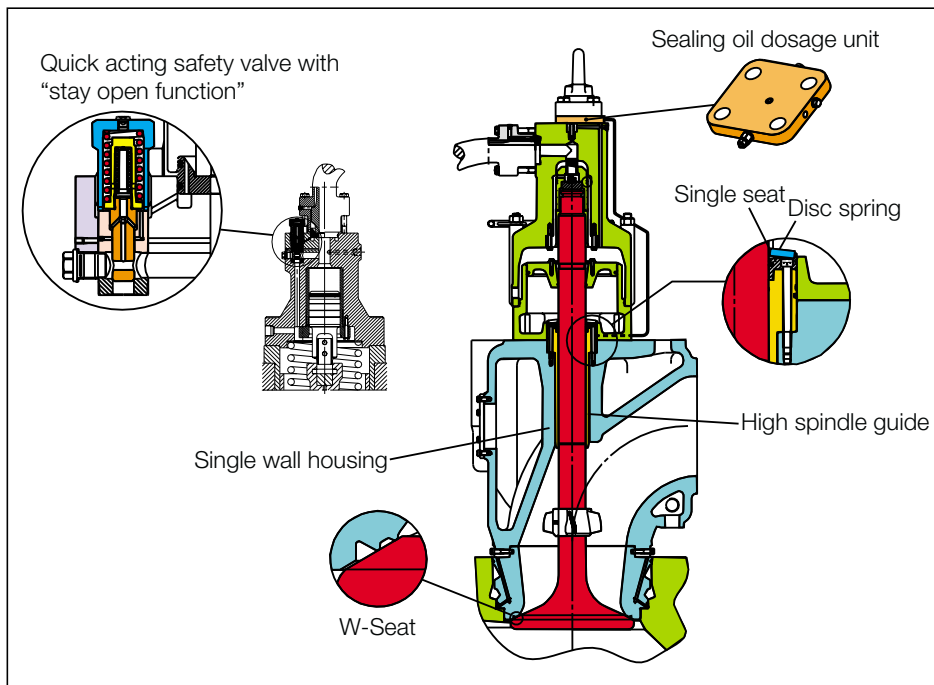


Fig. 17: Exhaust valve, 80-90MC

Fuel Injection Gear

Fuel pump

In general, the fuel pumps work well and without difficulties. However, a few incidents have been experienced.

On the S60MC-C, S70MC-C, S90MC-C and K98MC/MC-C engines, a combined puncture and suction valve is used. This design originally involved a bellow as substitute for the conventional sealing rings, in order to have a component needing no or very little maintenance. However, the reliability of the bellow was not satisfactory, and a new design without the bellow has been introduced on the above-mentioned engine types.

As a cost-down measure, fuel pumps without shock absorbers were introduced on the S60MC-C, S70MC-C, S90MC-C and K98MC/MC-C, but this

resulted in annoying, but not damaging, pressure fluctuations in the fuel supply system.

Even though the pressure fluctuations measured were within our design limits, as well as within the classification societies' limits, shipowners experienced problems with shipyard-installed equipment such as pumps, filters and preheaters. In order to avoid such problems, a shock absorber has been re-introduced on each fuel pump on the above-mentioned engines.

On K98MC/MC-C and S90MC-C, leakage has been seen between fuel pump housing and top cover. One problem was related to production tolerances, as a gap of up to 0.5 mm was found between the pump housing and the top cover, see Fig. 18. However, leakage has been experienced even in cases where the machining was correct. Therefore, a new type of gasket has been introduced. The primary sealing is

obtained with a viton ring, protected against erosion attack by a steel bushing. A soft iron plate of the same shape as the original seal forms the "groove" for the square viton ring.

Fuel pump top cover fractures have been experienced on small and medium bore engines. The fractures were initiated at the position where the inclined drillings for the high-pressure pipes intersect with the central bore, see Fig. 19. The cause of the failure has in all cases been related to roundings which did not fulfil our specification. However, to improve the safety margin against failures we have changed the design as per Fig. 19, which has made the design easier to manufacture and less sensitive to minor tolerance deviations.

Fuel valve and atomisation

Fractured fuel valve nozzles have been found on large bore engines where slide type fuel valves have been standard for some years. The main reason for the cracks was residual stress from machining. However, the high temperature of the valve nozzle itself contributed to the fractures because of the consequential high mean stress. This has been cured by optimising the production parameters.

As for pressure testing of conventional fuel valves, it must be noted, however, that the testing device is only capable of supplying 1-2 % of the normal fuel flow rate on the engine, which is not sufficient to ensure proper atomisation. If the remaining test items in the procedure are fulfilled, the fuel valve nozzle will work perfectly. As the atomisation test can be omitted, it is not described in new testing procedures, so verification of a humming sound as earlier is no longer possible, nor necessary.

Pressure testing procedures for slide fuel valves are quite different from those for conventional valves and have,

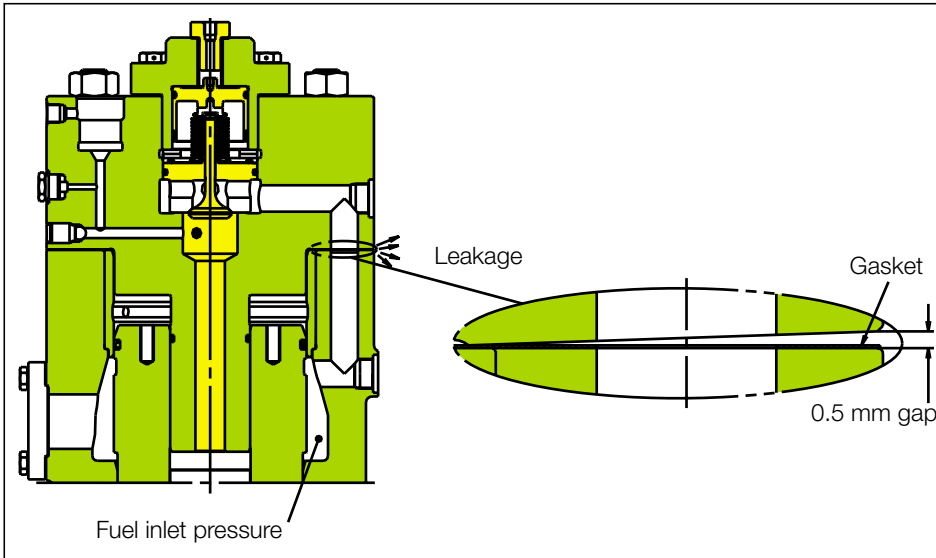


Fig. 18: Fuel pump top cover leakage

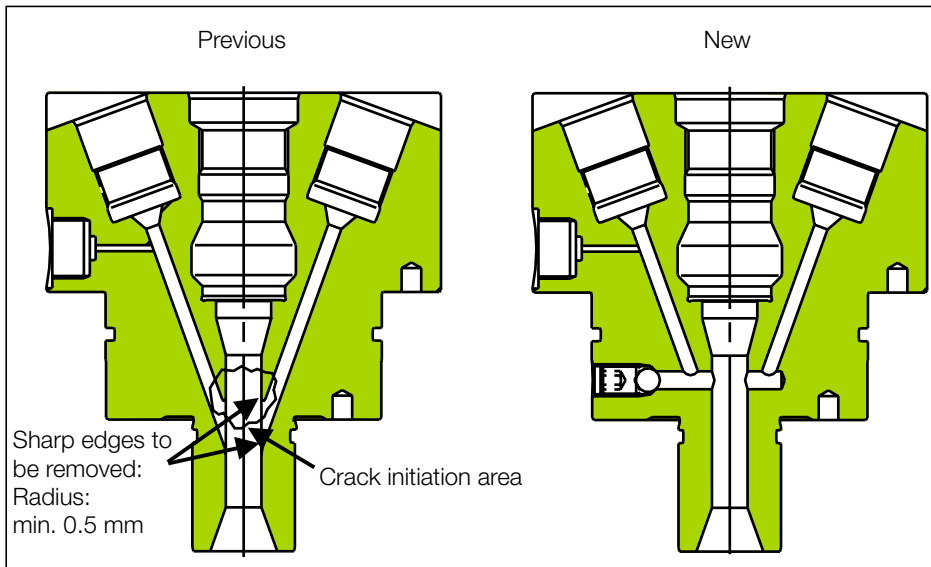


Fig. 19: Drillings in fuel pump top cover

probably because of the difference, in some cases been misinterpreted by the operators. Slide fuel valves must be disassembled and cleaned before pressure testing, and an atomisation test must not be performed on slide fuel valves. The cleaning is necessary because the cold and sticky heavy fuel oil, in combination with the very small clearance between the cut-off shaft

and the fuel valve nozzle, would significantly restrict the movement of the spindle. An atomisation test is not acceptable, because the very small needle lift obtained during such a test would result in an unequal pressure distribution on the cut-off shaft, resulting in a relatively hard contact in a small area, see Fig. 20. This, together with the high frequency oscillations during an

atomisation test, and the low lubricity of the test oil, would increase the risk of seizure significantly. The full lift of the needle, and the very good lubricity of heavy fuel oil, completely eliminates this risk during normal operation of the engine.

During scavenge port inspections on engines with slide fuel valves, we often find wet spots under each fuel valve. In the conventional valve, the sac volume is emptied after each injection. In the slide valve, the fuel stays in the fuel nozzle until the next injection. However, when turning the engine and after some hours of engine standstill, the nozzle will empty and fuel will drip down on the relatively cold piston crown, shown as wet spots of up to 300 mm diameter. After some hours the drops will more or less evaporate, depending on the actual position of the piston in relation to the scavenge ports, leaving only some ash on the piston top. This is not normally an indication of malfunction.

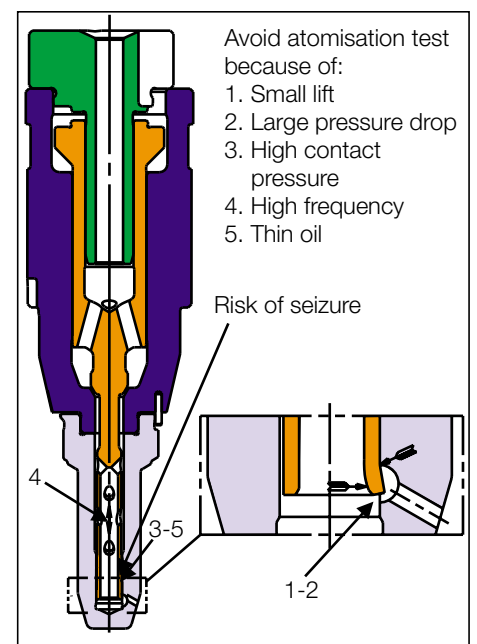


Fig. 20: Slide valve pressure testing

Engine Structure

Crankshaft thrust bearing

The thrust bearing introduced on Mark 5 engines has completely solved the previous problem of cracks in the horizontal support plates. No cracks have been reported on engines with the so-called Calliper design of the thrust bearing.

The bearing saddles have remained totally free of cracks, in compliance with precalculated stress levels.

Safety Precautions

Tools and lifting equipment

During recent years, there have been examples of lifting tools that have not been manufactured according to our specifications.

Wire ropes and wire locks

In some cases, experience has shown that the length of wire ropes differs from the specified length. Often, this is acceptable, because the tool is designed to adapt to these minor variations. However, in some situations (e.g. lifting tools for cylinder liners), the tool **must** have the specified wire rope length, as the lift would be dangerous if not performed straight.

The use of wire rope types other than those specified has been seen, for example, wire ropes with fibre-core instead of steel-core, which results in a significant reduction in the safety margin of the lifting gear.

In connection with aluminium wire-locks, there have been examples where wire ropes have not been cut to the correct length before clamping the wire-lock around the wire rope, thereby creating a risk of injury to the hands of the crew,

service personnel, etc. An example is shown in Fig. 21.

Regarding the new lifting tool for main bearing cap for the 60 - 98MC-C, experience has shown, in some cases, that the wire-lock used was not in accordance with the specification.

As seen in Fig. 22a, the wire-lock to the right does not have the correct inside shape. The specified lock (which is shown to the left and as No. 2 in Fig. 22b) is wave-shaped inside to further secure the wire rope. In the wire-lock to the right, there is a risk that the wire rope will be pulled straight through the lock.

Furthermore, the centre hole in the lock to the right is oversized, which makes it possible to tighten the two halves of the lock completely together, giving too low a clamping force on the wire rope. The result was that the wire rope could slide inside the wire-lock. The correct lock, on the left, leaves a gap between the two halves after tightening.

Other tools related to personal safety

Hydraulic tools: The hydraulic hoses and couplings are sometimes not of an adequate quality which, according to the operators, means that they can only be used a few times before a replacement is necessary, and others are leaky from the start.

Chain tackles: There have been cases where the chain tackles do not have sufficient dimensions for the carrying out of normal as well as special overhauling tasks on board, e.g. when removing a connecting rod from the engine.

Oil mist detection

Besides applying an oil mist detector, our efforts to prevent crankcase explosions have, so far, been concentrated on designing the engine with ample margins in order to prevent overheating of the components. However, we have



Fig. 21: Wire ends with strands of wire sticking out

realised that when rare explosions occur, they originate from other causes, such as factors related to production, installation, or incorrect maintenance of components or the lubricating systems.

Thus, it is recommended that:

- The oil mist detector is connected to the slow down function
- The oil mist detector has remote monitoring from the engine control room
- The relief valves are made with approved flame arrester function.

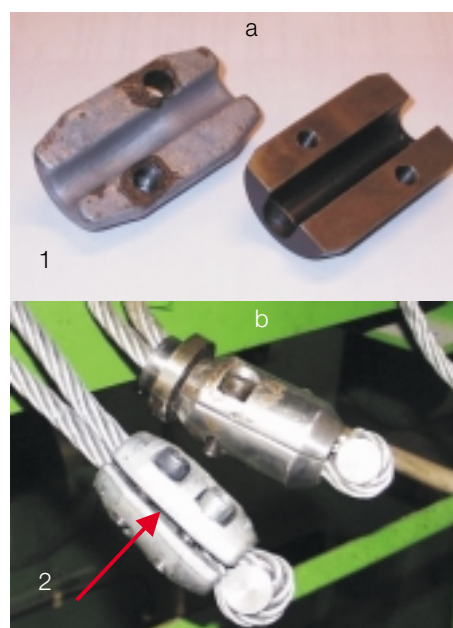


Fig. 22: Wire-locks



Fig. 23: Testing of a non-functioning crankcase relief valve



Fig. 24: Testing of an approved type of crankcase relief valve

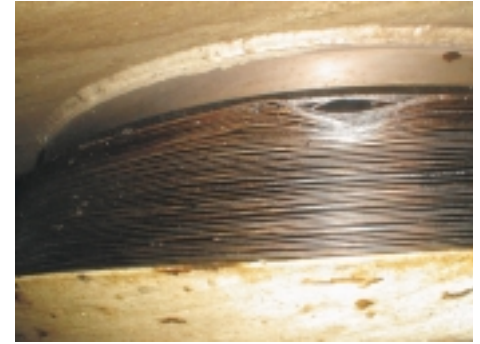


Fig. 25: Deformation of flame arrester

In the past, reliance was placed on specifying safety equipment that had been type approved by the classification societies. However, type approval is not sufficient as it only comprises the checking of the opening pressure of the relief valve.

In the event of a crankcase explosion, the pressure wave will send a large amount of oil mist out of the crankcase and into the engine room, where it will be moved around by the ventilation. A major part of this could be sucked up into the turbocharger air inlet.

If the oil mist meets a hot spot, it can be ignited. Therefore, it is important to keep the insulation around the exhaust pipes in good condition. Of course, the largest risk of igniting the oil mist would be if the flame arrester on the relief valve does not function properly.

Therefore, updates to the relief valve specification have been made. Only relief valves approved according to this specification are accepted on engines ordered after 1st May 1999. A draft of this specification was sent to IACS in November 1998. Regrettably, so far, no new rules have been introduced.

Fig. 23 shows the result when testing a non-functioning relief valve, and Fig. 24

shows an approved type. So far, we have approved two types of crankcase relief valves:

- Hoerbiger type EVN
- Mt. Halla type HCSG-N.

There have been cases of fire after an explosion in crankcases equipped with approved relief valves where the functioning of the flame arrester had been prevented by a local deformation, see Fig. 25. Thus, in the event of deformations, the flame arrester must be renewed.

Concluding Remarks

On the basis of service experience, it can be concluded that the continuous development and updating of the design and production of the MC and MC-C engines have clearly resulted in a significant increase in reliability and longer time intervals between overhauls. At the same time, the engines' unit outputs have been increased while the physical dimensions have been decreased, thus demonstrating the independence of the three parameters, reliability, power and size.