

Service Experience

Small Bore Four-stroke Engines

Engineering the Future – since 1758.

MAN Diesel & Turbo



Contents

Introduction	5
Cylinder incidents – L16/24 units.....	5
Exhaust gas temperature – L16/24	6
Valve adjustment procedure – L16/24, L21/31, L27/38	7
Excessive wear on valve – L21/31 and L27/38.....	7
Cavitation – L21/31 and L27/38	7
Pressure fluctuations in the fuel system – L21/31 and L27/38	8
Exhaust gas sensors – L21/31	8
Cracked piston – L23/30	9
Oval/cracked connecting rod – L23/30 and L28/32	10
Broken gear wheel – 9L27/38.....	11
Liner and ring groove wear – L27/38	12
Bush for cam follower – V28/32S	12
Prolonged Service Interval L23/30	14
New roller on valve cam – L28/32H	15
HFO service experience.....	16
Leak oil	17
Conclusion.....	17

Service Experience

Small Bore Four-stroke Engines

Introduction

For the small bore four-stroke engines, considerable know-how is available in Denmark within the four-stroke organisation at Holeby. The MAN Diesel & Turbo Holeby organisation is the day-to-day cooperation partner when it comes to component sales, technical support, GenSet and engine design, etc.

In the following, we will report about some of the important findings regarding service experience of small bore four-stroke engines. Furthermore, we describe how we have identified and rectified a number of technical problems on the small bore four-stroke engines in service.

Some engines in service are in need of updating to be able to perform satisfactorily, and it is necessary to improve their condition in order to regain the trust of the operators.

Cylinder incidents – L16/24 units

It is well-known that the L16/24 engine has experienced cylinder incidents over the last years. At the same time, similar engines have logged 50-80,000 running hours in service and never experienced such problems. The investigation of the incidents revealed that most cases were related to extremely low load operation in combination with fuel of poor ignition properties. Because the fuel fulfilled the standard, even with the poor ignition properties, we had to make some modifications. From the beginning, the modifications were concentrated on preventing sticking valves, better charge air preheating, better turbocharger and adjustment of valve stem clearance, see Fig. 1.

With the low-lift cam, stronger spring valve, etc., the risk of collision with the piston has been eliminated. After the

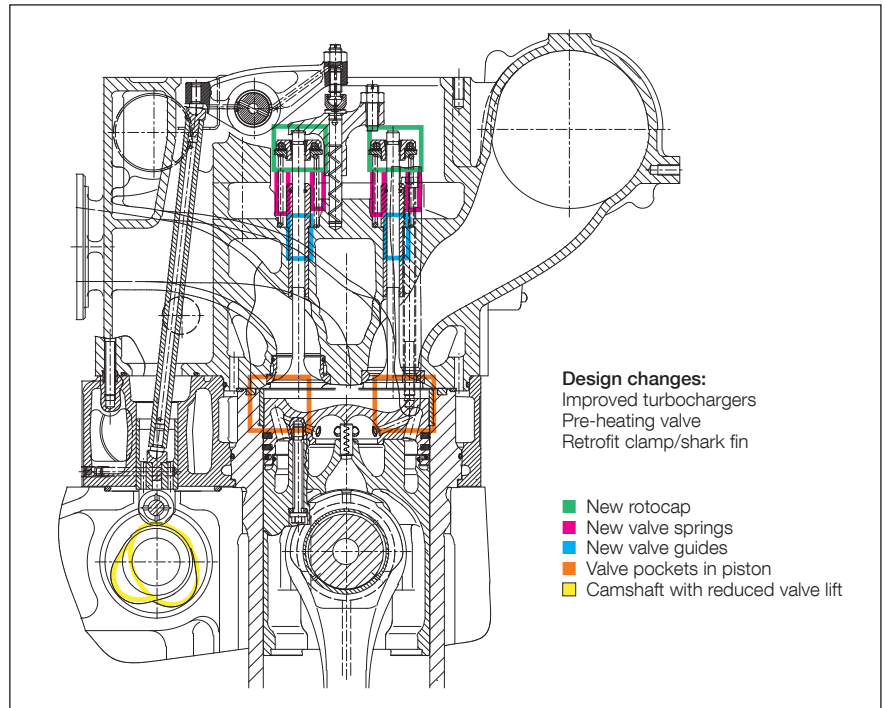


Fig. 1: L16/24: higher safety against sticking valves, i.e. sticking valves will not cause engine breakdown

introduction of the low-lift cam (Mk 2), we have not experienced valve damage related to sticking valves in service.

Unfortunately, incorrect valve adjustment still resulted in valve bridges getting out of place during operation of engines. Therefore, we recently introduced a valve bridge guide to align the rocker arm and valve bridge as a retrofit, see Fig. 2.

For new engines, the solution to prevent the valve bridge from turning is a so-called “shark fin”, see Fig. 3.

A new valve clearance procedure has also been introduced to minimise the possibility of incorrect adjustment. This new valve adjusting procedure is common for L16/24, L21/31 and L27/38, and it was introduced to the operators with the service letter: SL12-559/MIKA. To summarise, the Mk 2 package launched in the beginning of 2012 comprises:

- low-lift cam
- insulation of the front-end box top
- improved turbochargers
- pre-heating valve
- retrofit clamp/shark fin
- new rotocap
- new valve spring
- new valve guide
- valve pocket in piston.

Exhaust gas temperature – L16/24

On some 5L16/24 engines, the exhaust gas temperature is too high. Generally, the temperature was acceptable when the engines were tested at the shop trial and after cleaning of the turbine side on board, but the time between necessary cleaning is unacceptably short, and many operators are not cleaning efficiently.

The problem has been solved on many engines by introducing a valve cam with a longer overlap. The turbocharger has proven to have better long-term performance with the longer valve overlap.

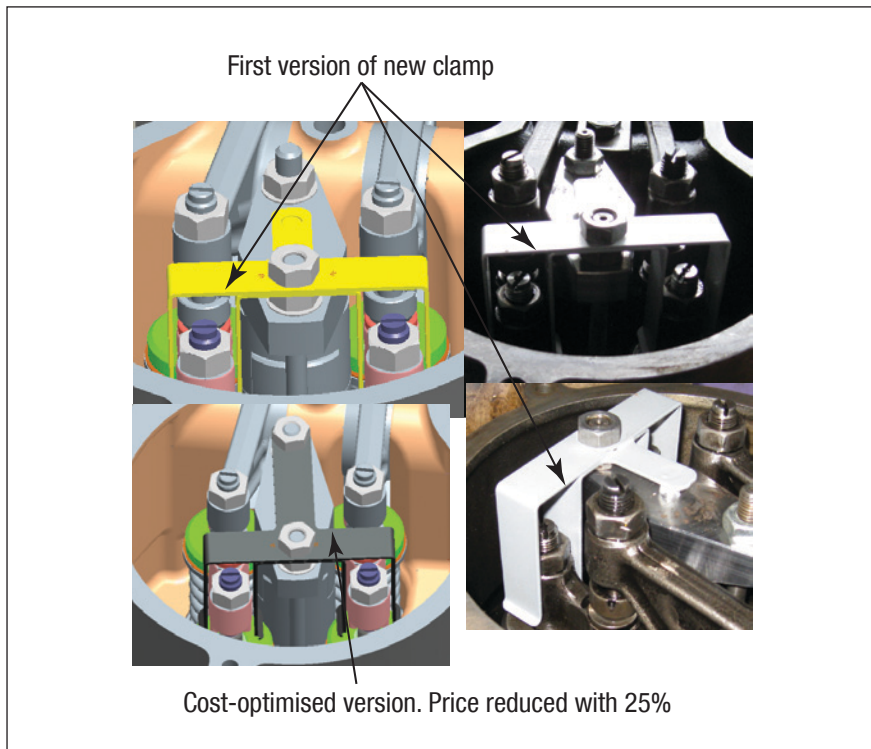


Fig. 2: L16/24: new rocker arm and self-aligning valve bridge introduced

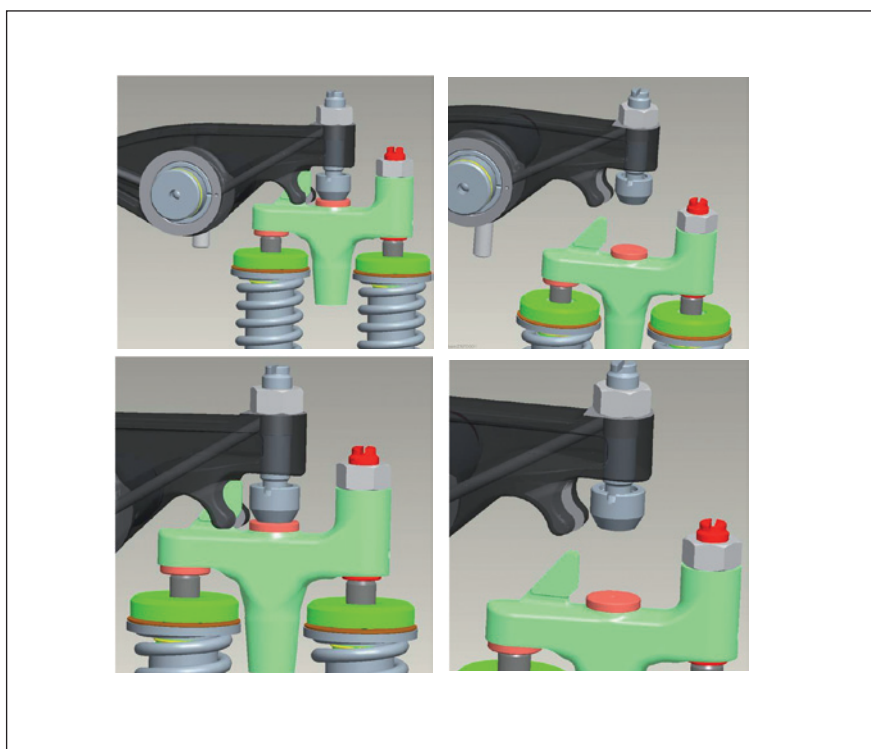


Fig. 3: L16/24: valve arrangement, new serial solution

Our next step is to test different turbo-charger characteristics for better part load performance. This may include different valve timing and different turbochargers. We expect this will result in satisfactory cleaning intervals without increased exhaust gas temperature.

Valve adjustment procedure – L16/24, L21/31, L27/38

Service experience has revealed that, from time to time, valve clearance adjustment was not done properly or with the required accuracy. This has largely been the case for the L16/24 type engine, but only to a very limited extent for the L27/38, and very seldom for the L21/31 engine.

For this reason, we simplified the adjustment procedure and safeguarded a correct tightening torque by means of a new special tool, see Fig. 4.

Excessive wear on valve – L21/31 and L27/38

We have experienced heavy wear on the inlet valve seat for engines running on low-sulphur diesel, see Fig. 5.

This observation has been made on a very small number of engines, but still we have decided to introduce counter-

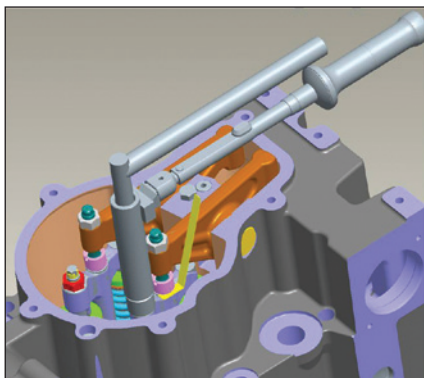


Fig. 4: Tool for correct tightening torque (L16/24, L21/31 and L27/38)

measures.

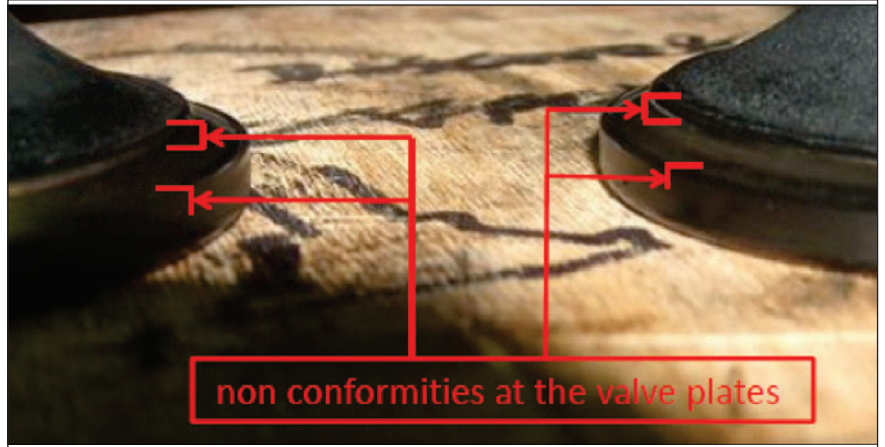


Fig. 5: Wear on valve seat (L21/31 and L27/38)

This kind of wear has never been seen on HFO-running engines or on most common diesels. To overcome the problem, we are currently testing a new armouring of the valve seats.

Cavitation – L21/31 and L27/38

Ever stricter requirements on exhaust gas emissions imply that more marine engines and power plant engines operate on gas oil. Operationally, this can have a negative influence on the fuel injection equipment. Marine gas oil (MGO) has a lower vapour pressure than HFO, and the system pressure is also lower. This may lead to cavitation, and cavitation on the plunger leads to lower capacity of the fuel pumps, with a lower power output as a possible consequence. Cavitation of plugs and valves in the pump may result in larger particles blocking the nozzle holes. Erosion of the delivery and constant-pressure valves can result in loss of residual pressure in the high-pressure system. This may result in cavitation of the high-pressure pipes and nozzle parts, see Fig. 6.

Actions taken to minimise cavitation:

- supply system pressure increased
- circulation flow increased
- temperature kept as low as possible.

We have implemented these actions on several plants, and subsequent service experience looks promising. However, we still need more running hours to close the topic.

Pressure fluctuations in the fuel system – L21/31 and L27/38

MGO operation may also lead to pressure fluctuations in the low-pressure fuel system. At the end of each injection, the high pressure is released to the suction chamber in the fuel pump. On HFO, the viscosity of the fuel absorbs most of the pressure peak, whereas on MDO/MGO, the pressure peak is transmitted to the fuel piping.

This has been experienced on marine engines and power plant engines with low-volume fuel systems, and has

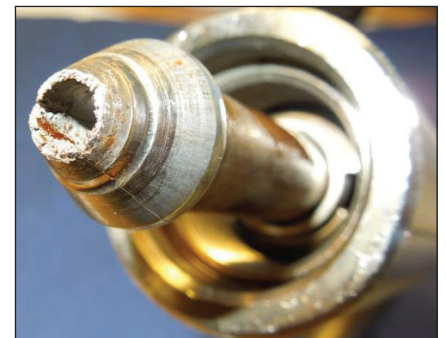


Fig. 6: Cavitation of high-pressure pipe (L21/31 and L27/38)

caused damage on pressure transducers and other components in the system.

Actions taken to decrease pressure fluctuations:

- installation of gas dampers (Fig. 7)
- matching of fuel pump inlet and outlet nozzles.

We are testing the gas damper on several plants, see Fig. 7. Service experience looks promising, but we need more running hours to close the topic. Matching on different nozzles is still on-going.

Exhaust gas sensors – L21/31

In connection with the L21/31 Tier II release and introduction of the SaCoS_{one} system, the exhaust gas temperature PT1000 sensor type was released.

On the L21/31 engines, licensees and operators have often reported a failing temperature signal. The problem was claimed to MAN Diesel & Turbo, and an investigation of the root cause was initiated.

The problems with the exhaust gas measurement turned out to be complex:

1. Sensors were not sufficiently reliable from the manufacturer
2. Junction box connections suffered from loose wires and failures
3. Longer sleeves had not been fitted correctly
4. Teflon cables had not been introduced correctly
5. Insufficient exhaust gas pipe insulation resulted in sensor cable failure.

To solve the problems, the following countermeasures were introduced:

1. Junction boxes removed (see Fig. 8)
2. Sensors with full cable length applied
3. Washer applied on engines in service, and long sleeves applied on new engines
4. Service Letter with photo documentation of the exchange procedure issued
5. New assembly drawings and parts lists introduced (new cable route, no

junction box and full cable length on sensors).



Fig. 7: Gas dampers being tested in service (L21/31 and L27/38)

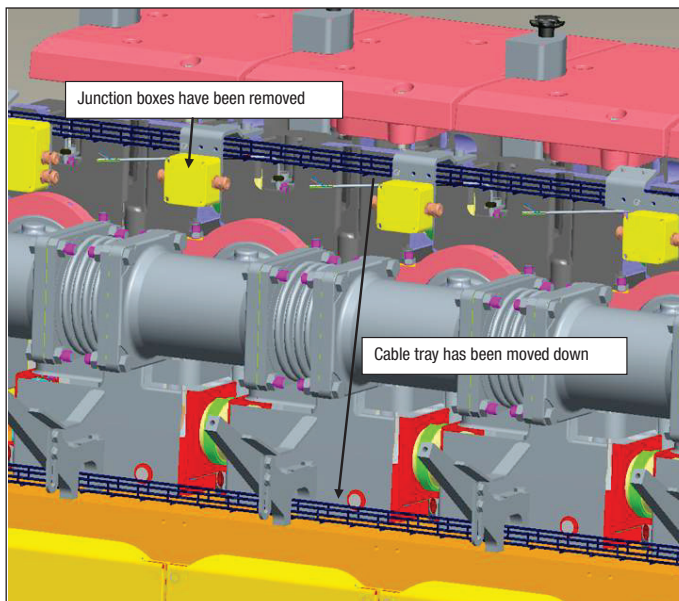


Fig. 8: Exhaust gas sensors on L21/31



Fig. 9: Cracked pistons on L23/30

Cracked piston – L23/30

In mid-2010, we received a report from a vessel claiming cracks in the L23/30 pistons. After further inspections, cracks were found in all pistons on all three genset engines on board. During the second half of 2010, the same type of failure was reported to MAN Diesel & Turbo (MDT) from more than ten different vessels and ship operators. In some cases, fatigue cracks caused total piston failure with consequential damage to the engine. The cracks were found in pistons that had been in service from 20,000, but also up to 50,000 hours. The pistons were sent for material examination, which revealed that the pistons always had a primary initial crack, and one or two typical consequential cracks. Primary cracks start from the inner connecting bar towards the piston bottom.

Consequential cracks run from the lateral piston bolt accommodation in circumferential direction to the piston skirt or through the pin boss. Investigations into the origin of the problem proved that a too small rounding and poor casting quality of the surface roughness had led to an excessively high stress level. All engines affected were equipped with pistons from the same maker. Nearly 21,000 pistons were in service worldwide. According to an agreement between the licensees involved and MDT, service letters were issued from all parties involved. One general service letter from MDT, and one from each licensee to the specific operator with pistons installed from their engine production. The service letters strongly recommended to replace the pistons due to the potential risk of engine damage and

serious personal injury for people who are near the engine.

After issuing the service letters regarding the pistons, we received reports from operators with pistons from other licensees, using other sub-suppliers, stating that they had also found cracks in the pistons.

Thorough investigations of the pistons from the involved makers are still in progress to eliminate any future risk of engine damage and injury of people around the engines. During the investigations, all involved licensees will be approached/involved to eliminate the problem in cooperation before issuing new service letters. According to our knowledge, there are around 15,000 high-risk pistons in service from different makers.

Oval/cracked connecting rod – L23/30

and L28/32

In 2011, we received notice from operators claiming that their connecting rods had an ovality at the big end bore exceeding the limit of 0.08 mm. When the connecting rods were sent for reconditioning, they were often found with cracks in the serrations, and reconditioning was therefore not possible.

After thorough investigation of connecting rods from various engines, we found the following two root causes, which depend on the running hours:

1. If the connecting rod is found with high ovality and/or cracks in the serration the first time the connecting rod is disassembled, then this is related to incorrect machining of the serration by the maker of the connecting rod. The geometry of the serrations on both sides of the big end was measured, and there was not correct correspondence between the upper part and the lower part. The incorrect geometry of the serrations will have a negative influence on the ovality and the occurrence of cracks.

The connecting rods are from engines built by different licensees, but the connecting rods are mainly from one sub-supplier. The serrations are, however, not made by this sub-supplier. Fig. 10 shows an example where the requirement of 80% contact between the mating surfaces between the teeth is not fulfilled.

We always recommend full measurement and reporting when a connecting rod is disassembled to ensure the correct quality of the connecting rod.

2. If the ovality or cracks are found after the second disassembly or later, this is related to failure during the last assembly of the connecting rod. The

failures are often related to inappropriate tightening of the connecting rod and/or lubrication of the serration before assembly.

As another important issue related to connecting rods, see Fig. 11, we have seen broken connecting rods causing serious damage to the engine and the risk of personal injury. These failures are due to machining errors of the thread in the connecting rod resulting in a fatigue crack initiated in the bottom of the thread hole.

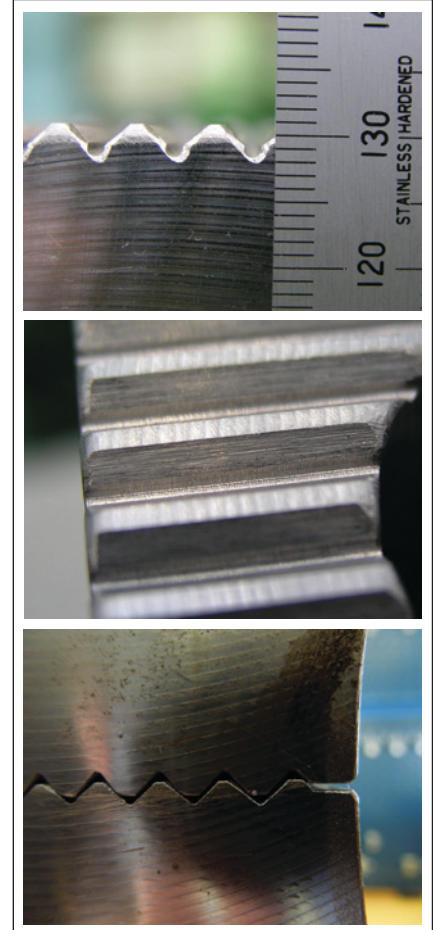


Fig. 10: Connecting rod serration (L23/30 and L28/32)

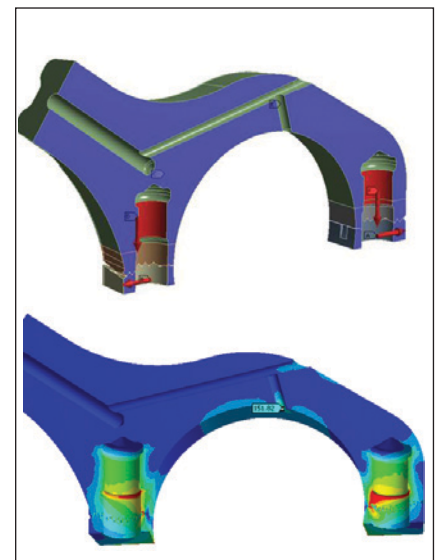


Fig. 11: L23/30 connecting rod

Broken gear wheel – 9L27/38

The problem with broken gear wheels is cracks between the teeth of the intermediate wheel driven by the crankshaft. See Fig. 12.

The conclusion is that the crankshaft of a 9-cylinder engine has a small up-and-down movement during rotation, thereby pressing the teeth of the gear wheels so close to each other that they get into full contact. Thus, the stress in the bottom of the teeth becomes high and the resulting fatigue may initiate a crack. Statistics show that only 9-cylinder engines with single bearing alternators are affected.

New two-part gear rings for installation on the crankshaft are in production. The new gear rings will increase the clearance, thereby overcoming the problem. Today, two-bearing alternators are the standard for new 8 and 9L27/38 engines and, thereby, the problem has been eliminated.

A circular letter (CL001-2012) alerting the operator about the issue has been sent out, and new gear wheels will be delivered to the relevant operators.



Fig. 12: Broken gear wheel from L27/38

Liner and ring groove wear – L27/38

A high liner and ring groove wear on the L27/38 engine was a problem earlier, and various solutions have been tested in the field. As a result, the liner wear is now kept below 10 $\mu\text{m}/1,000$ hours, and the ring groove wear rate can be kept below the target limit, i.e. less than 5 $\mu\text{m}/1,000$ hours. The entire solution package consists of:

1. Changed cylinder liner support
2. Changed piston top land clearance
3. Improved scraper ring groove drain
4. Changed compression ring shape and material
5. Cooling jacket with controlled water flow
6. Low-sac fuel nozzle.

All the changes were tested and introduced in 2006, see Figs. 13 and 14. Nevertheless, some operators were still claiming high wear rates in the ring grooves, i.e. more than 0.25 mm within 12,000 hours.

Investigation of these cases showed that not all engines built after 2006 were equipped with the mentioned package, which effectively will cure the problem.

Bush for cam follower – V28/32S

Due to cracks in the engine frame in the area of the roller guide's 12-o'clock stud (see Fig. 15), as well as cost-down considerations, the original roller guide design was simplified and integrated in the engine frame.

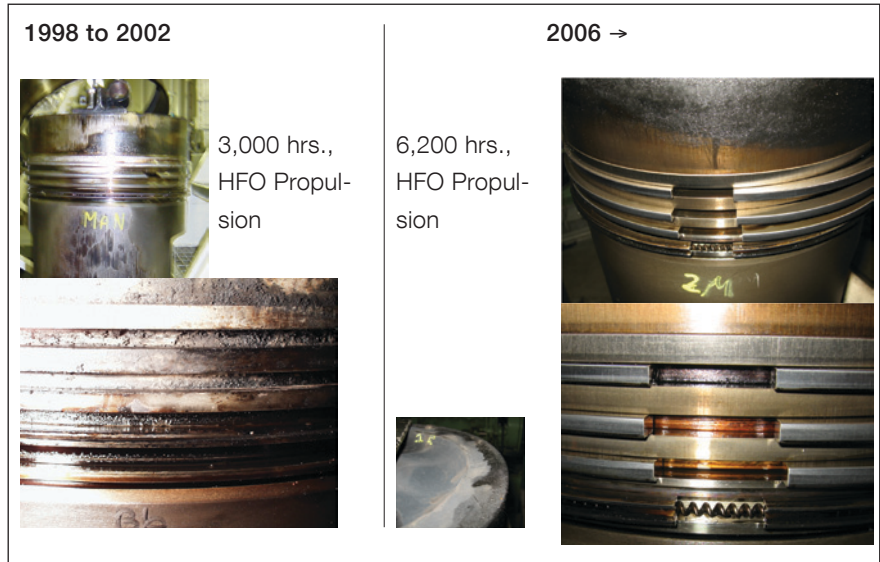


Fig. 13: L27/38: improvement on hardware

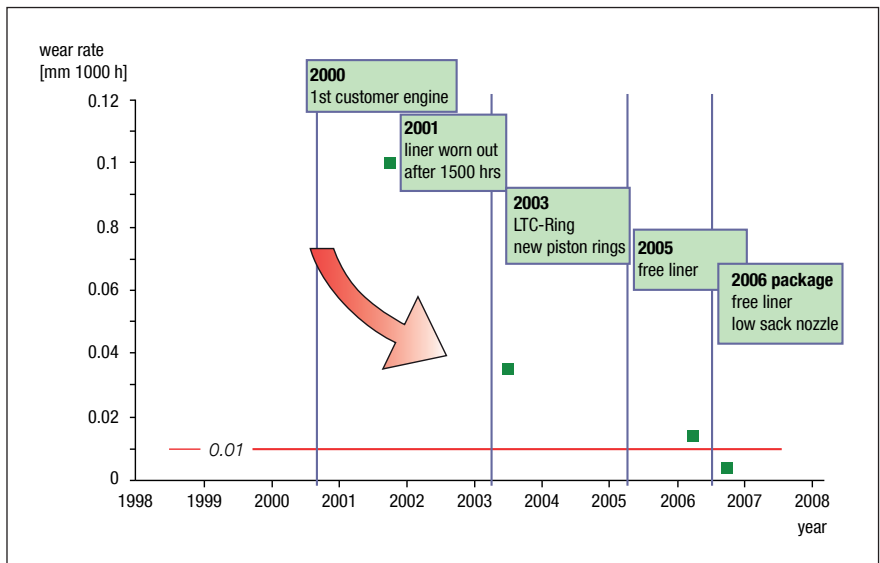


Fig. 14: 27/38 liner wear rates

Original roller guide design:

- mounted in the roller guide housing, which is mounted in the engine frame
- steered by a pin on the roller guide stud and a groove in the housing.

The original roller guide design was working very well, but the frame was

weakened by the cut-out for the housing, and cracks developed around the lower cylinder cover stud. Especially the gas engine version was affected.

Service experience revealed afterwards that, with an improved design of the roller guide, the wear rates of roller guides and bushings were not as low as

expected and the required lifetime was not achieved.

A re-introduction of the guiding bush in combination with enforcement of relevant wall thicknesses was introduced as the new standard and retrofit solution, see Fig. 16.

New roller guide design (2006)

- mounted in bushes that are shrink-fitted directly into in the engine frame
- steered by a pin on the roller guide stud and a groove in the bushes.

The new roller guide design was based on the old one from the V28/32H, L28/32H and L23/30H, etc. Now, a bush is mounted in the engine frame, with a steering groove similar to the housing on the above-mentioned types. In this way, the frame strength is maintained.

The new design requires correct positioning of the bush and the roller guide stud. Otherwise, the roller guide, cam, etc., will fail.

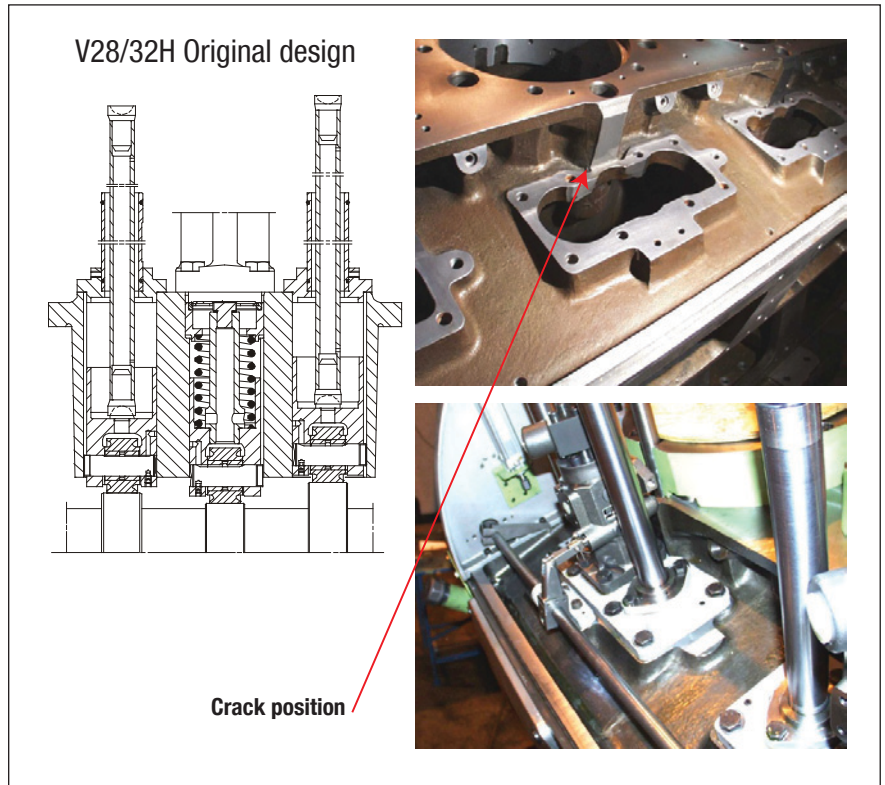


Fig. 15: Original roller guide design on V28/32H

Prolonged Service Interval L23/30

A recent inspection of exchanged engine parts after main overhaul of a 7L23/30H auxiliary engine, following an interval of 21,199 hrs. in operation, confirmed that the L23/30H can oper-

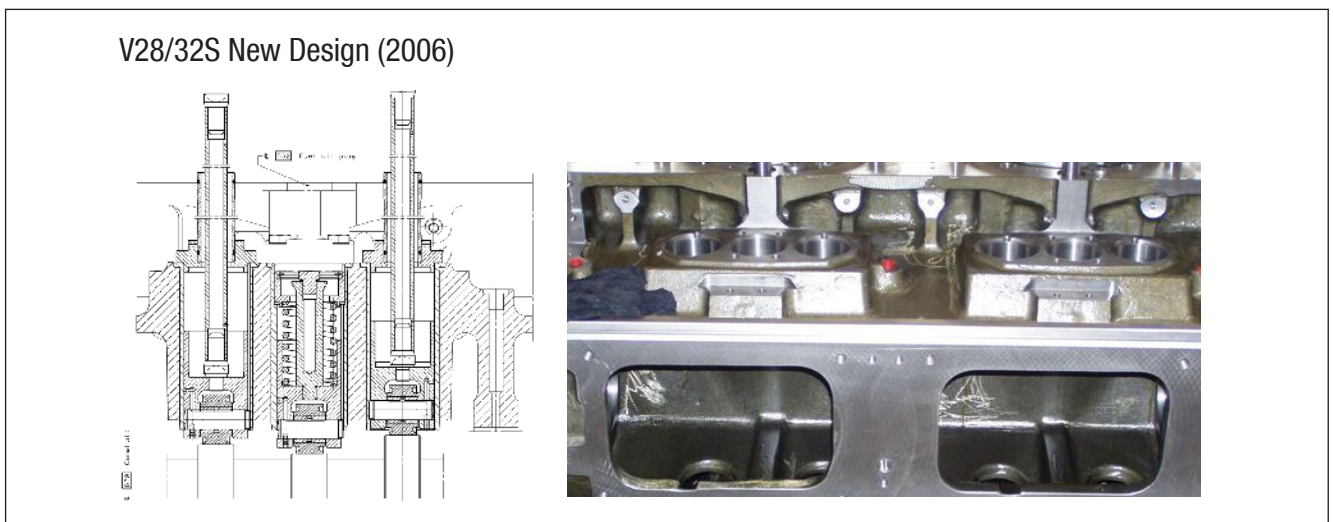


Fig. 16: V28/32S: present roller guide design

ate for 20,000 hours on HFO between main overhaul. The specific wear rates recorded for the cylinder components is listed below:

- Cylinder liners, TDC 1st ring: 5 $\mu\text{m}/1,000$ hrs.
- Piston, 1st ring grove: 2 $\mu\text{m}/1,000$ hrs.
- Piston rings: all with intact Cr-layer and no side face corrosion.

Fig. 17 shows the good condition of the piston rings from the 7L23/30H auxiliary engine after 21,199 hrs. in service.

To achieve this prolonged service interval, regular maintenance cleaning of the turbocharger, lube oil, and fuel oil is required. Furthermore, the main overhaul must be carried out according to the instructions and with correct spare parts, and the engine performance must be brought back to “shop test level”. The inspection revealed that the exhaust valves and seat rings seemed to have reached their limit, while bearings, pistons and connecting rods had even more hours left, see Fig. 18.



Fig. 17: Piston ring condition after more than 20,000 hrs. in service (7L23/30H)

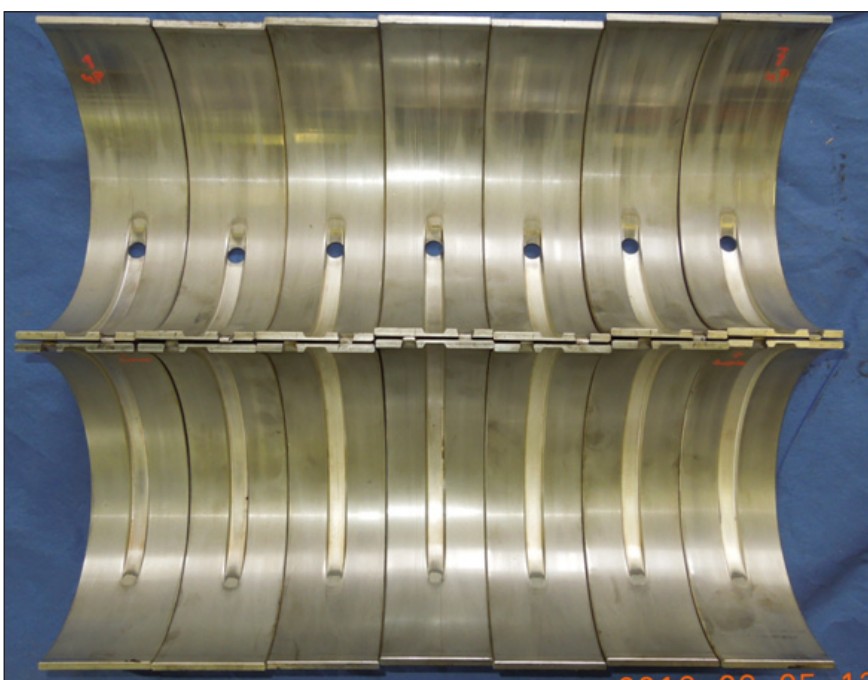


Fig. 18: Main bearing condition after more than 20,000 hrs. in operation (7L23/30H)

New roller on valve cam – L28/32H

A number of damaged valve cams (ballistic type) were claimed from the operators, and the root cause has been identified to be insufficient material properties and/or an incorrect hardening process, see Fig. 19.

A solution involving the release of “soft shoulder” cams (see Fig. 20) has been introduced, similar to the ones used on newer engine types, as this design is easier to handle production-wise.

The below production procedure must be followed to obtain the best soft shoulders:

1. Roller is pre-machined.
2. Roller is hardened to a specified hardness and, more importantly, to a specified hardening depth
3. Roller is grinded to specified dimensions
4. Area to create soft shoulders to be machined away. This machining ensures that hardened material is removed and that the remaining non-hardened material has the wanted flexibility.

This design variant will also mean a cost reduction for some engine producers.

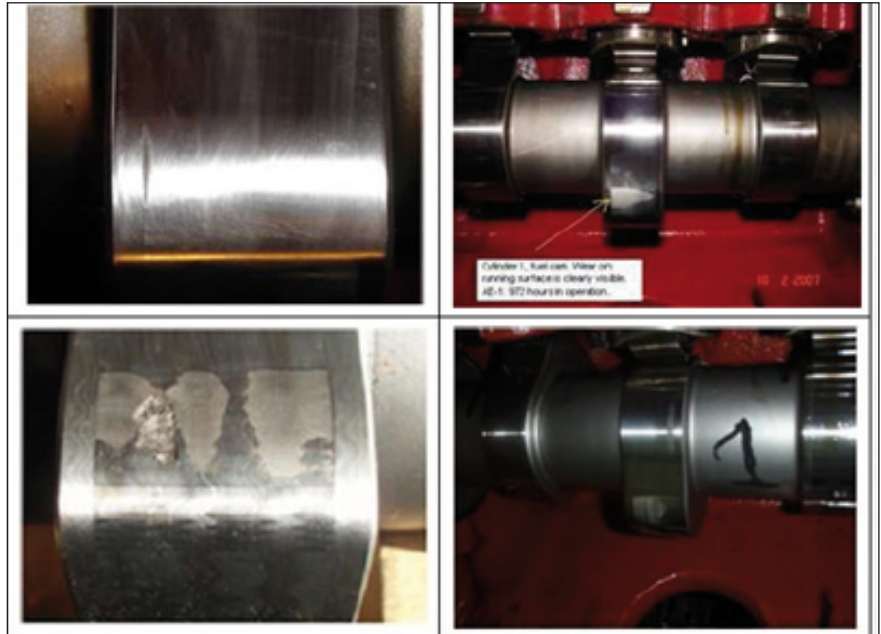


Fig. 19: Camshaft pitting – L28/32 fuel cam with 5,000 operating hours



Fig. 20: L28/32 soft shoulder cam design

HFO service experience

Sticking fuel pumps have always been an annoying problem for engines operating on heavy fuel. Lacquer deposits on the lower part of the plunger reduce the clearance and can lead to sticking of the plunger, especially after standby periods and change to diesel fuel.

High viscosity and low-quality HFO leads to a higher temperature to achieve the correct injection viscosity. The higher temperature increases the tendency to lacquering in the lower drain grooves. Furthermore, when the drain bores are filled/blocked by coke, the leak oil will be trapped there, see Figs. 21 and 22.

The L16/24 does not have this problem, and it seems that a special sealing oil system with controlled oil flow in the lower part of the pump element minimises the problem.

In response to the above, investigations are ongoing to eliminate the problem also on engines larger than the L16/24. A similar system for the other small four-stroke engine pumps has been function-tested and will go into field test. Another approach to lower the risk of lacquering is to keep the fuel temperature in the pumps as low as possible within the limits for viscosity and to have a high-circulating fuel flow to maintain a sufficient standby temperature.

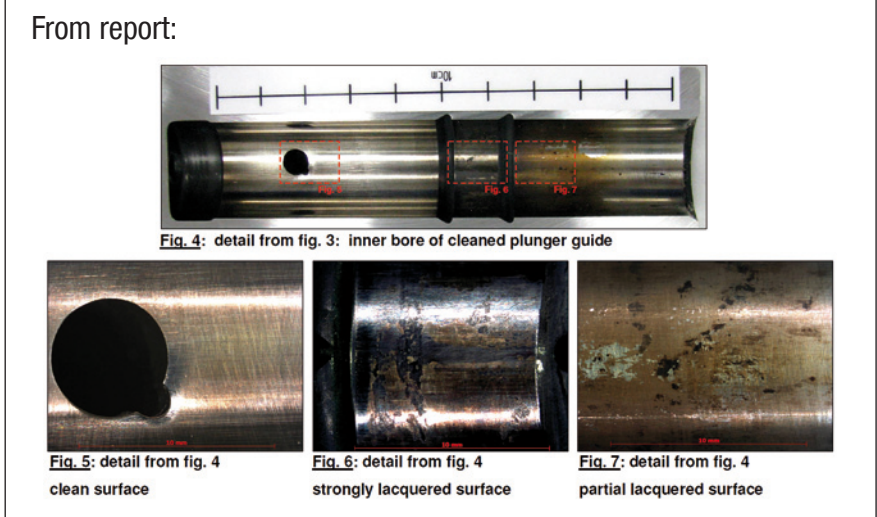


Fig. 21: L21/31 pump

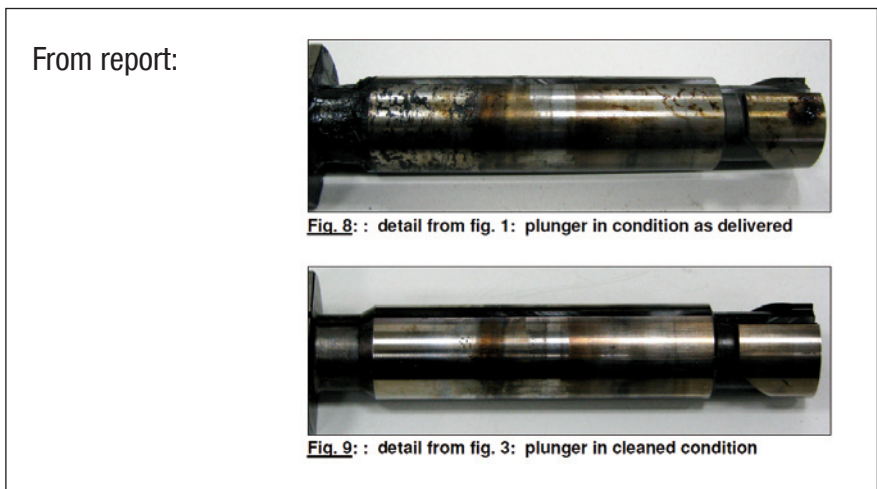


Fig. 22: L21/31 pump plunger

Leak oil

When operating engines on MGO, the leak oil amount increases to nearly 1% of the full load fuel consumption. The increased leak oil flow is caused by the lower viscosity of MGO compared with HFO. The main part of the leak oil comes from the fuel nozzle, and a minor part comes from the fuel pump drain. All this oil can be reused directly. A new system is in the design phase where leak oil from the fuel nozzle and pump will be separated from leaks in the low-pressure system and engine frame top, see Fig. 23.

Conclusion

MAN Diesel & Turbo works hard to ensure the best service to engines in operation and when it comes to continuous development of the engine programme. It goes without saying that we will do our utmost to make the necessary changes and improvements. As is already the case, we will continue to offer our strong support to both our engine licensees and the owners, so as to ensure that we all achieve our goals.

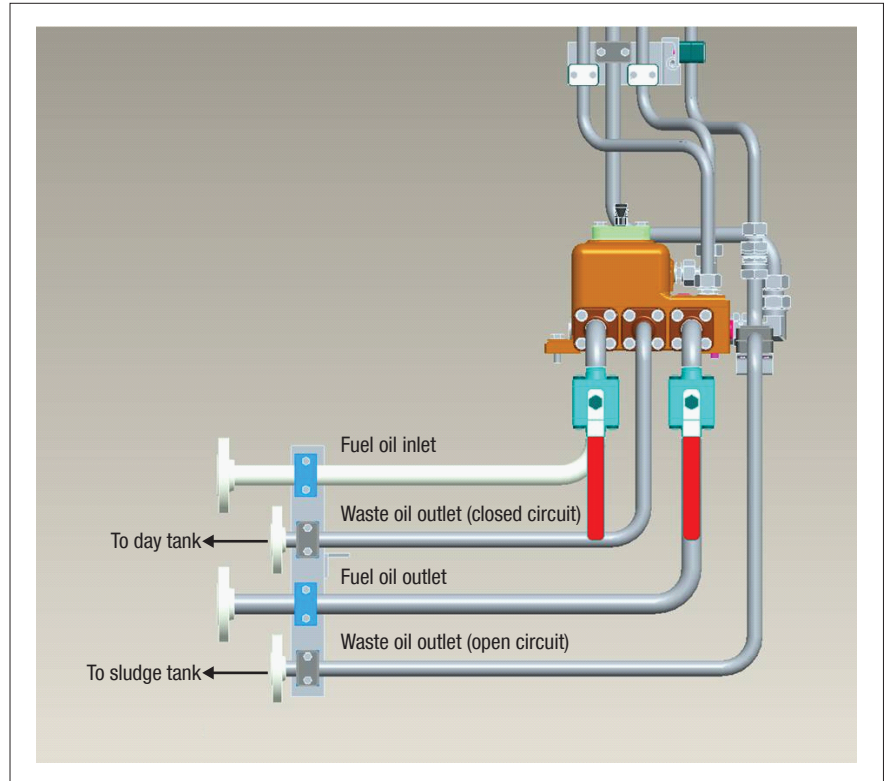


Fig. 23: L21/31 split drain

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