Service Experience with Sulzer A-Type Engines on Heavy Fuel Oils

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SYNOPSIS

A-type engines have been built for over 17 years as in-line and V-versions. They have been fitted in numerous marine and stationary installations and their design and output were adapted to the changing requirements, including operation on heavy fuel oil. This report surveys the experience obtained with this universal type of engine (Fig. 1) on heavy fuel oil, and related design aspects. Some reference is also made to engines operating on marine diesel oil.

B. Eckert graduated in Mechanical Engineering at the Technical College of Winterthur and joined the Diesel Department of Sulzer, Switzerland in 1961 as design engineer for slow-speed engines. Following a two years' assignment to Sulzer (UK) in London, he returned to Switzerland to join the Commissioning and Service Group for diesel engines. He was stationed in Bombay as representative responsible for technical services for Sulzer diesel engines in India and Pakistan and, in Winterthur, he was appointed Head of Service for four-stroke and small RTA engines in 1974.

DEVELOPMENT AND APPLICATION AREA

In its original version as a two-valve type, the A-engine was developed between 1964 and 1968. The objective was a robust, simple and inexpensive engine with universal capabilities including marine and stationary applications, to supersede the successful Sulzer B-engines. Already in 1970, the first ships were commissioned with a 6AL25 engine as main propulsion unit operating at a nominal speed of 750 rev/min. Soon, there was also increasing demand for this engine type as marine auxiliary units and for stationary duties. Some of these engines were operating on intermediate fuel oil (Fig. 2).

In 1973, the A25 series was supplemented by the A20 and AS25 versions, four-valve types with 750 and 1000 rev/min nominal speed. These all-purpose engines too have since found wide application as marine main and auxiliary, as well as stationary and traction, units.

As early as 1976, a heavy-fuel oil version—AS25H—was introduced to cover the demand in this market segment (Fig. 3).

In 1982 the A25-programme was further extended by the AT25 version with 750 and 1000 rev/min and up to 220 kW (300 bhp) per cylinder (Figs. 4 and 5). The first AT25 engines—already in a heavy fuel-oil version—were commissioned during 1983.

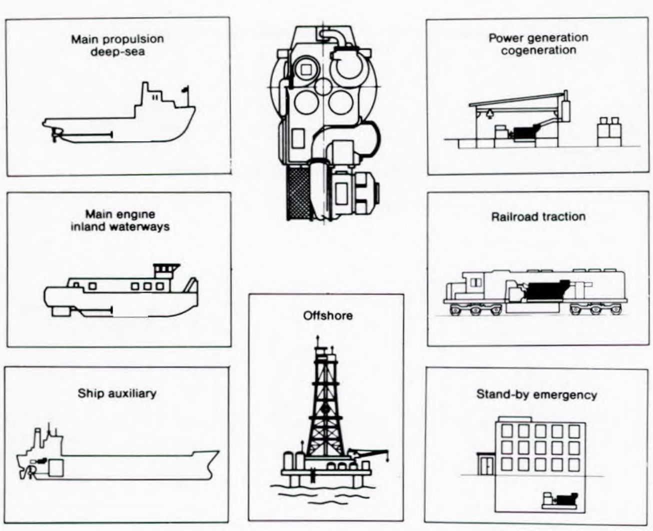


FIG. 1: Applications of A-type engines

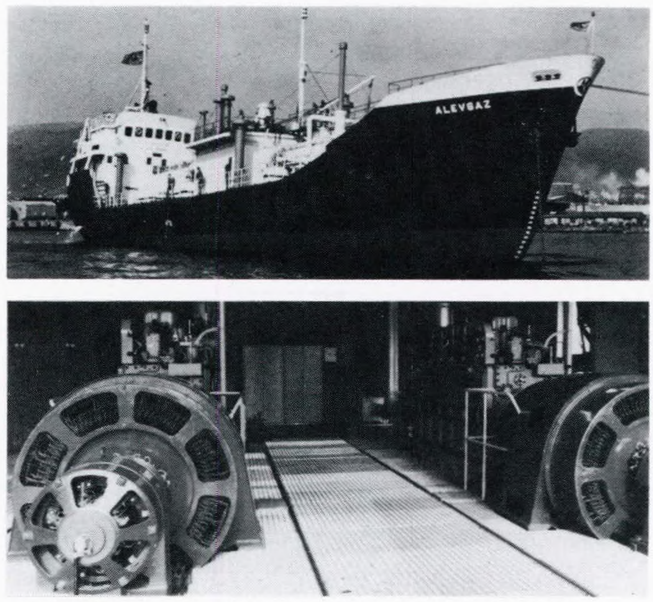


FIG. 2: First installations of 6AL25 operating on IFO (1972)

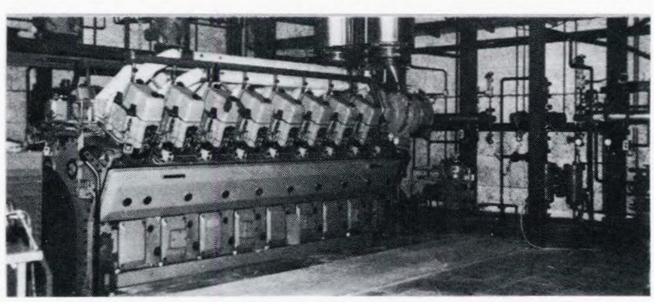


FIG. 3: First installation of 16ASV25H operating on HFO (1976)

Engines for heavy fuel oil

With the AS25H and AT25H versions, there are two choices of Sulzer type 'high speed' engines available, specifically designed to cope successfully with the implications of operation on heavy fuel oil (Fig. 6). For example, their design incorporates:

- Bore-cooled cylinder heads with water-cooled exhaust valve seat inserts
- Two-part pistons
- Nimonic exhaust valves with Rotocap rotators
- Oil cooled injection nozzles

The heavy fuel oil specifications (as bunkered) have now been standardardised according to the engine's application, e.g., marine auxiliary (including single-fuel ships) or marine propulsion and stationary applications (Fig 7).

The difference in fuel specifications reflects the different engine loadings usual in the two duties. The 500cSt limit set for auxiliary duties caters specifically for single-fuel ships in which the fuel oil for both main and auxiliary engines is handled by one treatment system. For the smaller vessels, which could typically have A-type main engines, 380cSt is a more realistic limit in view of general fuel availability and the limit on complexity of fuel treatment plant to be expected on board.

In every case, however, adequate fuel and lubricating oil treatment is essential for good overall economy. Proper on-board fuel treatment must be used to remove water and abrasive particles (such as catalytic fines, indicated by the aluminium content, etc.) prior to oil entering the engine. Adequate heating is also necessary to ensure the prescribed viscosity at the fuel injection pumps.

Charge-air heating at low engine loads is advantageous, enabling prolonged low-load operation on heavy fuel without the need of change-over to diesel oil. It shortens the ignition delay at low loads.

As it is generally agreed that, for heavy fuel oil operation, a high service reliability has greater priority than a high output, a 10% derating was introduced for such applications. The engine reliability is increased, not only due to the lower load but also because the turbocharger can be specially matched to provide increased excess air for combustion.

In the example of Fig. 27 the exhaust valve temperature was thus lowered by some 35°C in comparison to the same engine running on marine diesel oil.

A20—now in HFO version

In this context we should also mention the smallest engine type of the Sulzer programme, the AL20, introduced in 1973. The overall total of about 3020 A-type engines delivered by end 1984 includes some 520 A20s. The longest-running examples exceed 37,000 h operation (Fig. 8).

The new heavy fuel-oil version of the A20 makes it particularly attractive as a really heavy-duty ship's auxiliary. The 500cSt fuel specification limit covers the great majority of

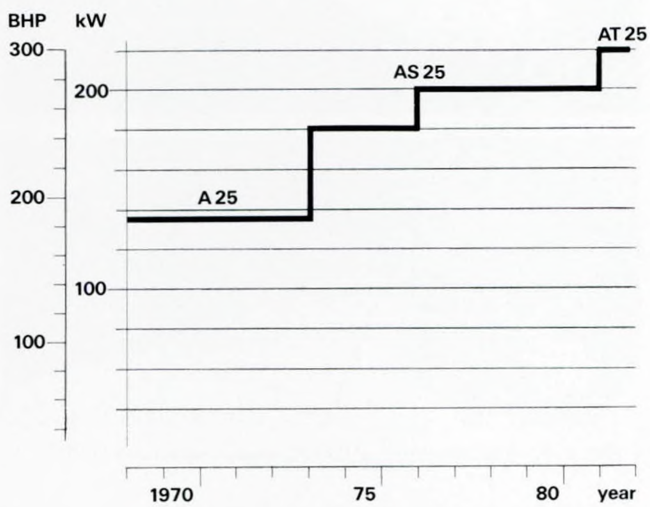


FIG. 4: Cylinder outputs

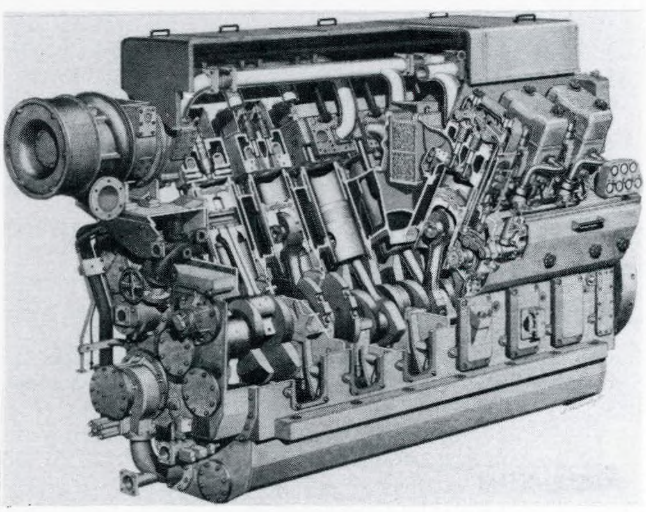


FIG. 5: 12AT25 engine

Viscosity	Pumping Poor atomization
Density	Separation
Sulphur	Low temperature corrosion
Conradson carbon residue	Deposits Wear Fouling Smoke
Asphaltenes	Slow burning Smoke
Vanadium Sodium	High temperature corrosion Fouling
Ash	Abrasive wear
Water	Corrosive wear Fouling
Cetane Number	Starting Rough running Fouling
Fuel compatibility	Incompatibility of Mixture

FIG. 6: Implications of heavy fuel oil operation

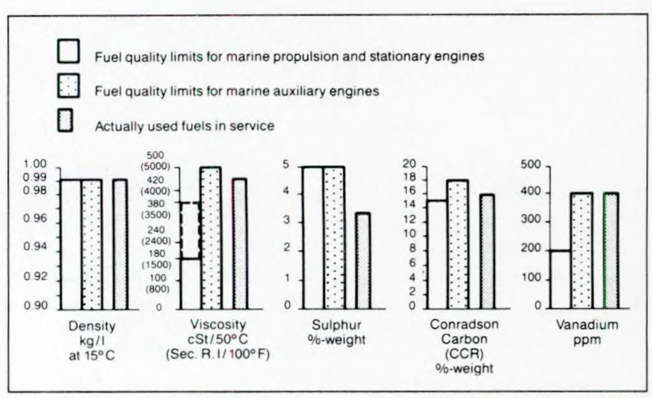
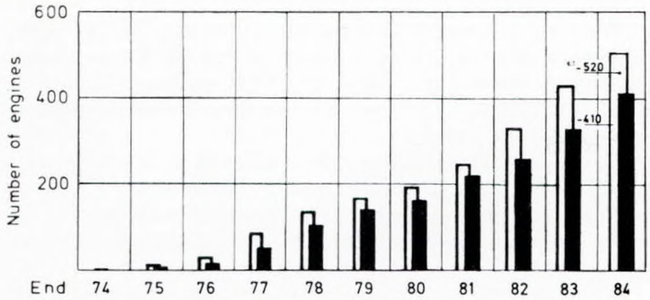


FIG. 7: Heavy fuel oil specifications and fuel oils actually used on the testbed and in service



1000 / 750 r/min	Propulsion	Stationary	Marine Aux.
Number of engines	Total -410		
	-175	-15	-220
First commissionings	June 75	April 75	Jan. 76
Longest operating times	over 37000 hrs.	over 10000 hrs.	over 36000 hrs.
Fuel oil burned	M.D.O.	M.D.O.	M.D.O.
Actual lub oil consumptions	Average $1.0 g/BHP_{nom} \times hr$		

FIG. 8: A20 engines delivered and in service

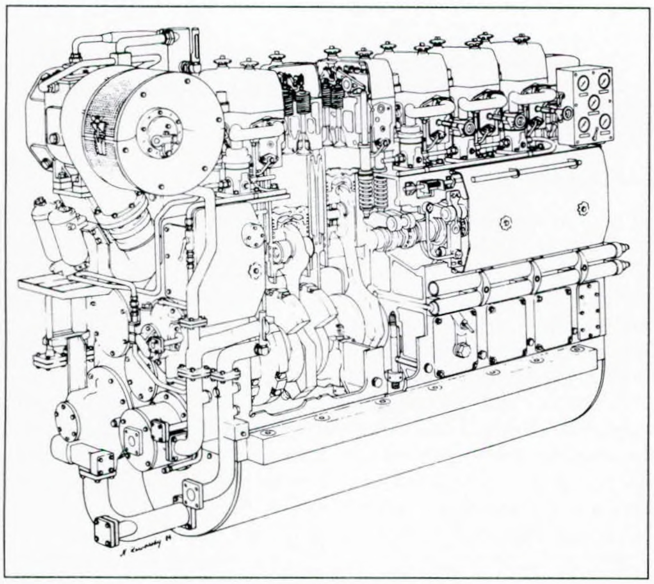
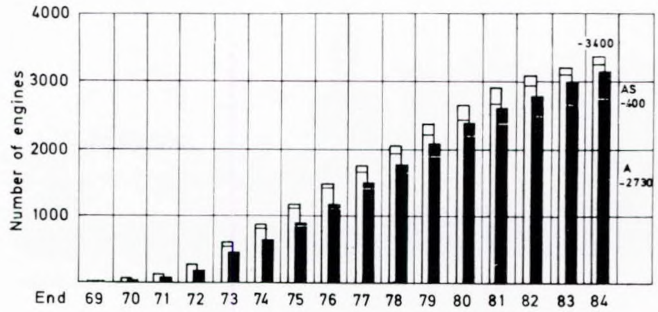


FIG. 9: 6AL20 engine

FIG. 10: A25 engines delivered and in service

bunkers being lifted and this allows the full economic benefits to be gained by single-fuel operation.

Two A20 engine models are available, with six and eight cylinders in-line, respectively, and maximum continuous outputs for heavy fuel-oil operation ranging from 525 to 760 kW (715–1035 bhp) at 900–1000 rev/min. They thus cover a generator output range of 500–720 kWe at 50/60 Hz, which is suitable for most ship types. It is a particularly compact engine design, well suited to incorporation in complete generating-set packages (Fig. 9).



A 25 750 r/min	Propulsion	Stationary	Marine Aux.
Number of engines in service	Total -2730		
	-180	-30	-2520
First commissionings	Oct. 70	Febr. 71	Dez. 70
Longest operating times	over 50000 hrs.	over 38000 hrs.	over 70000 hrs.
Engines operating on M.D.O.	-177	-28	-2450
I.F.O.1)	3	2	-70

AS/AT 25 720-1000 r/min	Propulsion		Stationary		Marine Aux.	
Number of engines in service	Total -400					
	-100		-70 3)		-230 4)	
First commissionings	June 77		Oct. 74		Oct. 76	
Longest operating times	over 28000 hrs		over 20000 hrs		over 39000 hrs	
Engines operating on M.D.O.	-76		-69		-210	
H.F.O.2)	-24		1		-20	
Actual H.F.O. properties	Visc SR I	S %	Va ppm	Na ppm	Carb %	Ash %
	200-4000	0.9-3.15	14-400	18-130	5.0-13.2	0.03-0.09
Actual lub. oil consumptions	average $0.7 g/BHP_{nom} \times hr$					

1) Engines modified accordingly 2) H.F.O.- Version
3) Incl. 15 engines for traction 4) Incl. 3x 6 AT 25H engines

FIG. 11: Single fuel ship with 4 × 8AL25 converted from MDO to HFO operation

Upgrading existing A25 engines for heavy fuel oil

Existing A25 and AS25 auxiliary engines, originally equipped to burn only marine diesel oil, can be readily adapted to operate on heavier grades of fuel oil and residual fuels with considerable economic benefits.

Correct handling and treatment of the fuel prior to entering the engine is naturally of the utmost importance. Adequate preheating is necessary to ensure that the prescribed viscosity is maintained at the engine. Additional separators, heaters, filters, pumps, etc may be needed.

Two levels of engine modifications are possible, depending upon the shipowner's requirements and the anticipated fuel limits. For intermediate fuels, up to 30 cSt/50°C viscosity, the conversion involves mainly new fuel nozzles; nimonic exhaust valves with Rotocaps; new sets of piston and scraper rings; smaller turbocharger nozzle ring (where suitable); and turbocharger washing equipment.

For heavier fuels, up to 180 or 380 cSt/50°C depending upon the application, these modifications need to be supplemented by bore-cooled cylinder heads; two-part pistons; and alterations to the fuel injection pumps.

Already some 85 A25/AS25 engines have been modified by Sulzer, Winterthur, to burn heavier fuels. Although the conversion for IF30 fuels is the most popular, a number of auxiliary engines have been converted to the full 380cSt standard to achieve a single-fuel ship (Fig. 10).

SERVICE SUMMARY

By the end of 1984, a total of over 3400 engines of the A25, AS25 and AT25 types had been delivered for marine propulsion, marine auxiliary and stationary power generation.

FIG. 12: One of five fishery ships with original main engines replaced by 4 × 6ASL25H operating on HFO

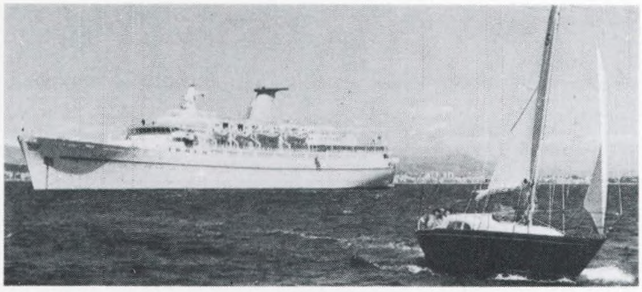


FIG. 13: Single fuel cruise ship with base load auxiliary engine 6ASL25H

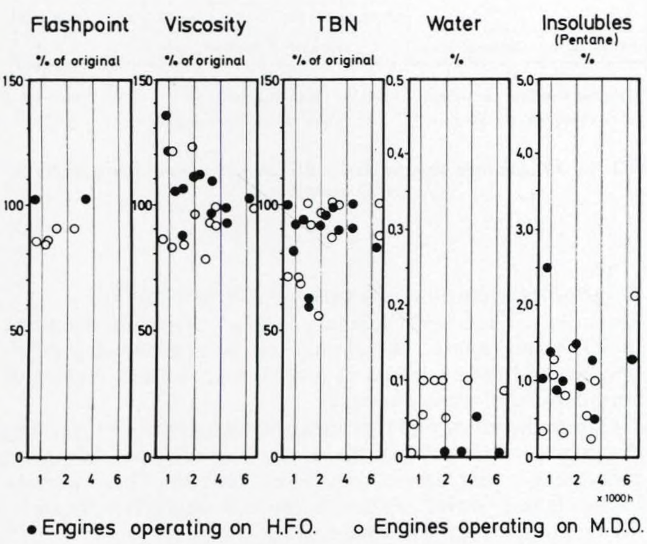


FIG. 14: Some lube oil analyses

The longest-running of these are exceeding 70,000 hours' operation (Fig. 11). In this total, there are 120 heavy-fuel-oil burning A-type engines. Many are AS25 versions but the first three of the latest AT25 type and some 85 conversions (Figs. 12 and 13) are also included.

The heavy-fuel service with the AS25 engines has been most encouraging. The following summary can be made.

Heavy fuel oils in use extend to 420 cSt (4000 SR1) viscosity, 3.4% sulphur, 400 ppM vanadium, 13.4% Conradson carbon residue and 0.09% asphaltenes (Figs. 7 and 10).

Lubricating oil consumptions have averaged 1.3–2.0 g/kWh (1–1.5 g/bhph). Cylinder wear rates have remained small. The mean value of maximum liner wear generally settles down around 0.01 mm/1000 after running-in, only slightly more than on marine diesel oil operation.

Piston-running behaviour has been good. Piston overhauls have been required at intervals of only some 6000–9000 hours' service. Exhaust valve performance has also been without failures. Valve overhauls could be combined with piston overhaul.

Nominal outputs per cylinder were as follows:

A25H: 110 kw (150 bhp)
AS25H: 180 kW (245 bhp)
AT25H: 200 kW (270 bhp)

The longest-running example on heavy fuel oil, a 6ASL25H auxiliary engine installed in the cruise ship *Daphne* in January 1981, has now accumulated more than 30,000 running hours.

Lubricating oil

As per our specification, SAE 40 lubricating oils with a fresh-oil alkalinity of between TBN 22–30 and in accordance with MIL-L-2104C, CD or Series 3 classification, are normally applied. The actual lubricating oil characteristics remain remarkably stable in heavy fuel-oil service (Fig. 14). Thus it is feasible to arrange oil changes according to analyses without limiting the life time of an oil beforehand.

However, the amount of oil contamination by water and insolubles depends on the degree of separation (preferably in the clarifier mode, either by-pass or batch-wise). This has not always been fully realised by engineers previously more accustomed to marine diesel oil operation.

The same applies in regard of heavy fuel-oil separation.

Main and bottom-end bearing

For some years now, the main and bottom-end bearings have been of aluminium-alloy with an overlay as running surface (Figs. 15 and 16). Although, these bearings are, in general,

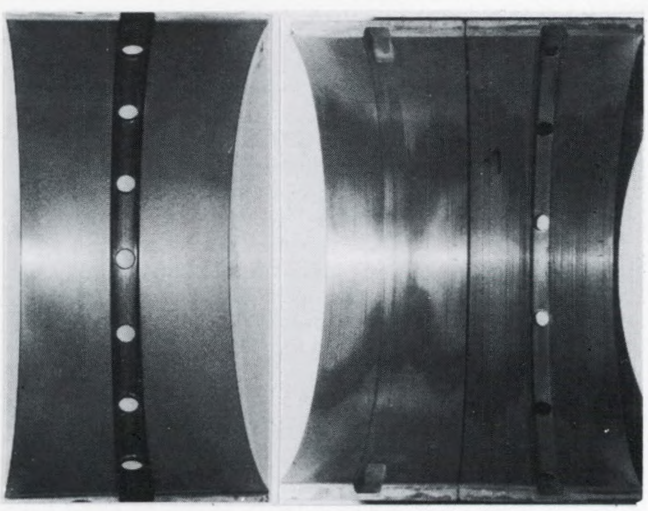


FIG. 15: Lower main and upper/lower bottom end bearing shells at 14 900 running hours on MDO

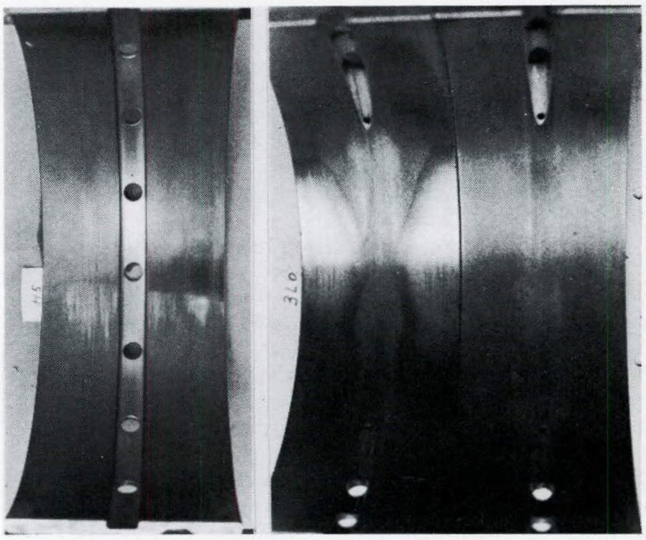


FIG. 16: Lower main and upper/lower bottom-end bearing shells at 11 200 running hours on HFO

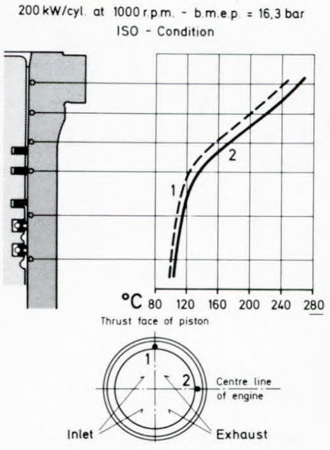


FIG. 17: Cylinder liner wall temperature

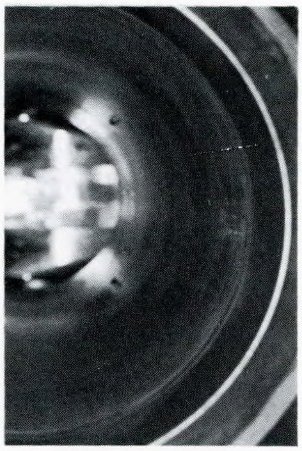


FIG. 18: Cylinder liner operating on HFO

giving satisfactory service, their practical life became limited in certain cases on account of faster overlay wear, mainly caused by increased lubricating oil contamination.

In this context I would again stress the importance of good fuel and oil care, with appropriate separators and filters etc.

Following extensive field tests, these trimetal bearings will be succeeded by aluminium-alloy bearings of increased wear resistance but without overlay. For the main bearings of the in-line engines, a modern whitemetal bearing will be re-introduced. The life now expected for main- and bottom-end bearings is 12 000–18 000 h on HFO and 16 000–24 000 h on MDO.

Except for random checks, bearings should only be opened up for surveys by the classification societies.

Cylinder liner

The wear rate of the cylinder liners of the engines operating on marine diesel oil has constantly been low, below 0.01 mm/1000 h.

For the engines operating on heavy fuel oil the wet type of cylinder liner has been retained, together with an intermediate, relatively elastic sleeve. The absence of any noticeable low-temperature corrosion and the stability of the lube-oils confirm the matched temperature level of the cylinder liner wall and the uniform and small distortion (Figs. 17 and 18). Thus, low long-term cylinder-liner wear rates are recorded (about 0.01 mm/1000 h), only slightly higher than obtained when operating with marine diesel oils (Fig. 19).

Based on the wear rates so far reported, we now expect a cylinder liner life of 24 000–36 000 h on HFO and 32 000–48 000 h on MDO.

Piston

Depending on the engine type and revolutions, oil-cooled cast iron or aluminium-alloy pistons of monoblock type are generally used on A and AS25 engines operating on marine diesel oil. With over 18 500 of these pistons in service, no noteworthy irregularities have been reported and piston seizures are almost unknown.

For the engines operating on heavy fuel oil, a two-part piston has been introduced with intensive oil cooling, resulting in temperature levels of piston head, grooves and rings (Fig. 20) which are adequate for HFO. This two-part piston is also specified for AT25 and higher-output AS25 engines operating on marine diesel oil.

The pistons are equipped with three piston rings and two spring-loaded oil-scraper rings. Two chromium-plated piston rings with a barrel-shaped face promote initial running-in and obviate re-honing of cylinder liners at overhauls. One stepped ring promotes stability so as to reduce lubricating oil consumption

The good running behaviour of the many pistons in opera-

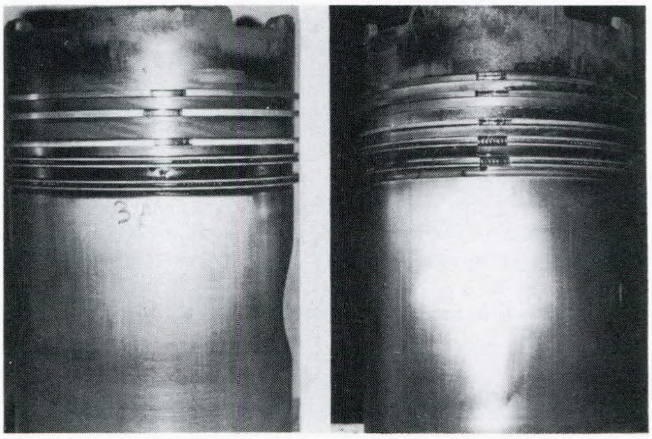


FIG. 21: Comparison between piston operating on MDO/HFO at 14 900 and 7500 running hours respectively

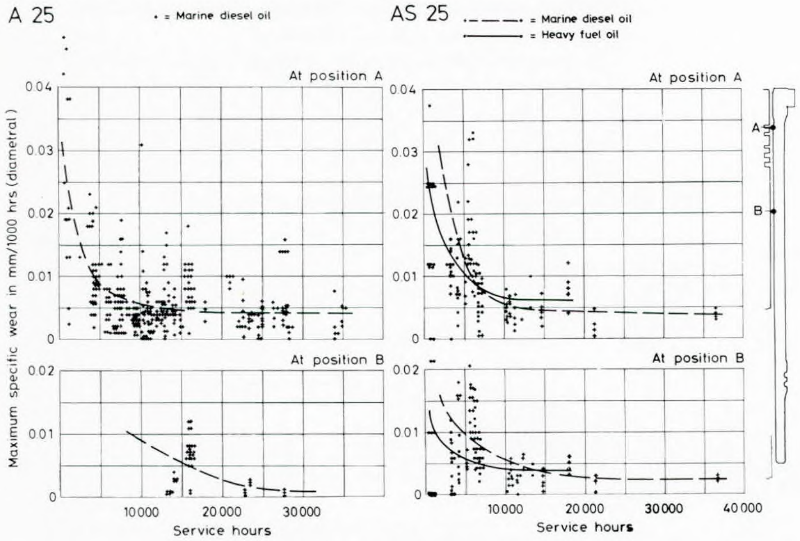


FIG. 19: Comparison of cylinder liner wear

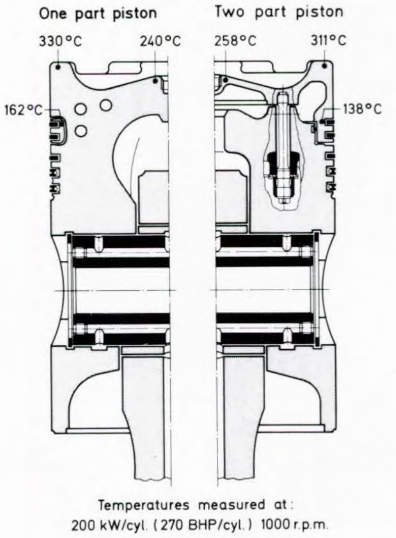


FIG. 20: Piston temperatures

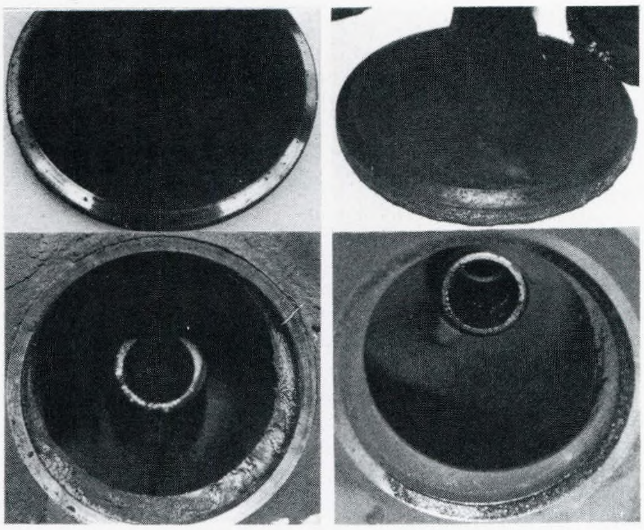


FIG. 22: Inlet and exhaust valves and seats after 14 900 running hours on MDO without overhaul

tion, with small deposit formation also in the ring grooves, confirm the appropriate temperature level and good stability and sealing properties of the ring package (Fig. 21).

Today, piston and scraper ring renewal are usual after operating 6000–9000 h on HFO and 8000–12 000 h on MDO. These replacement intervals can be regarded as satisfactory and economical, especially with the valve overhauls coinciding.

Piston ring groove

Generally the abrasive components of the heavy fuel oil and its combustion products increased the groove wear only slightly. However, higher groove wear was experienced in certain installations, mainly where fuel and oil separation was not up to expectation. In the future, the owners will have the option of ordering pistons with the flanks of grooves and rings chromium plated.

The re-conditioning intervals for piston ring grooves can be planned at 18 000–27 000 h (chromium plated grooves) on HFO and 24 000–36 000 h on MDO.

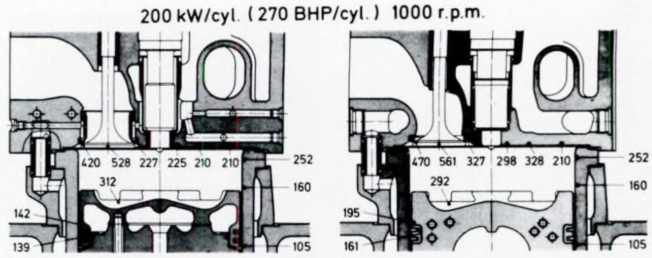


FIG. 23: Bore-cooled and original cylinder head

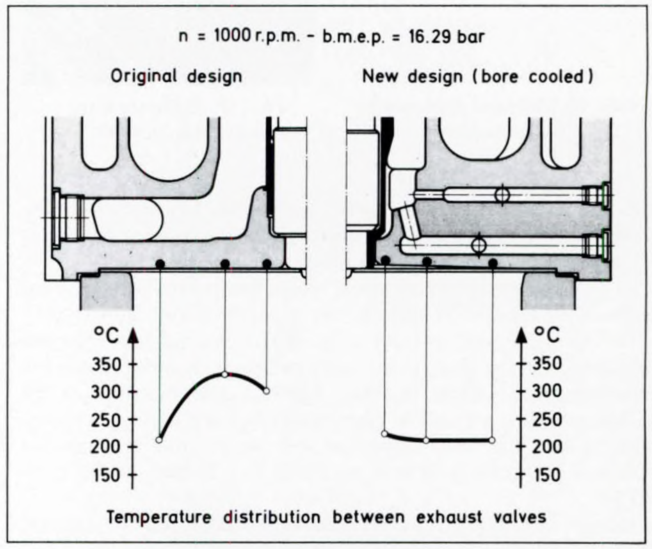


FIG. 24: Uniform and low temperatures

Cylinder head and valves

The double bottom design of the cylinder head, equipped with stellite-faced valves (Fig. 22) for the engines operating on marine diesel oil has been replaced by a bore-cooled cylinder head for operation on heavy fuel oil, already a proven feature of the Sulzer slow-speed R and medium speed ZA-engines (Fig. 23).

This bore-cooled cylinder head is also specified for the AT25 engines operating on marine diesel oil.

The bore-cooled head with a single thick bottom of high

FIG. 25: Even and low deformations

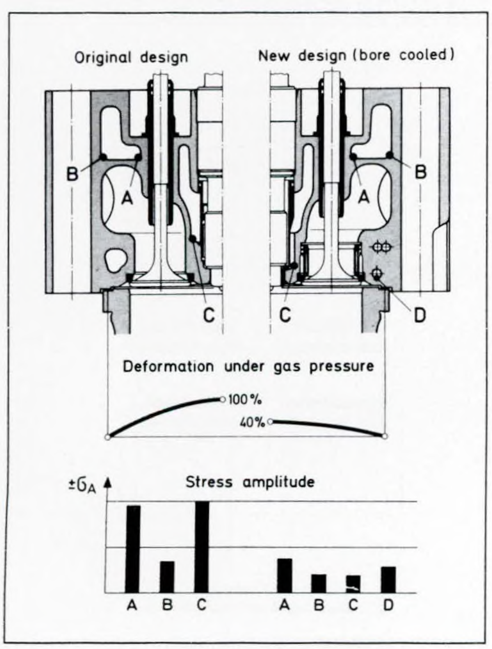
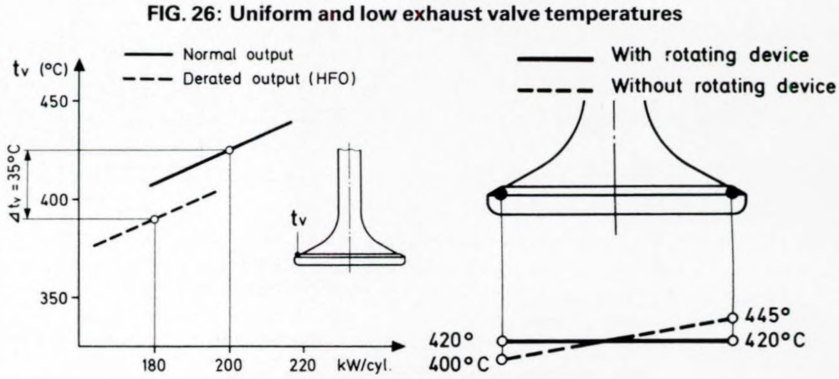


FIG. 26: Uniform and low exhaust valve temperatures



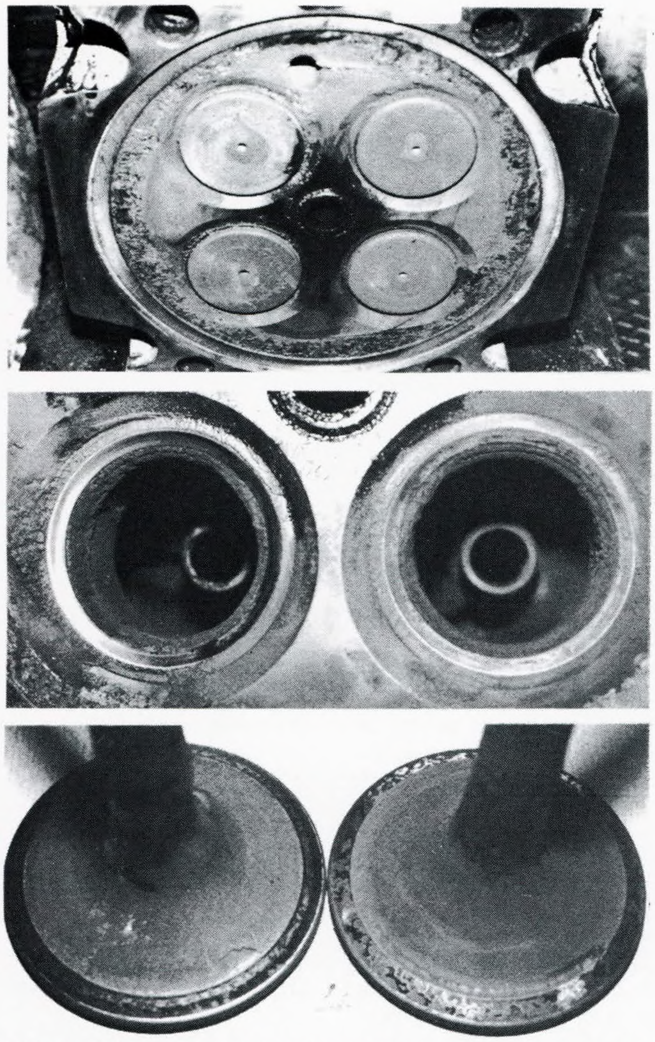


FIG. 27: Cylinder head, exhaust valves and seats at 6290 running hours on HFO without overhaul.

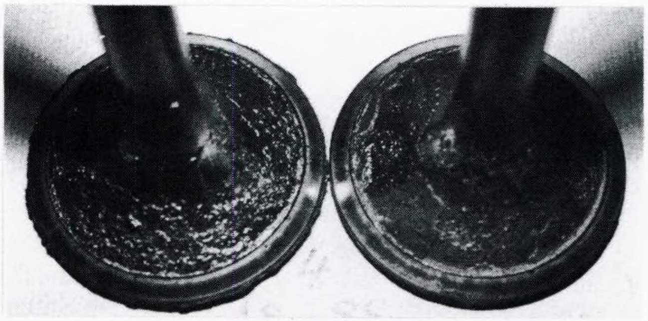


FIG. 28. Inlet valves of cylinder head Fig. 27

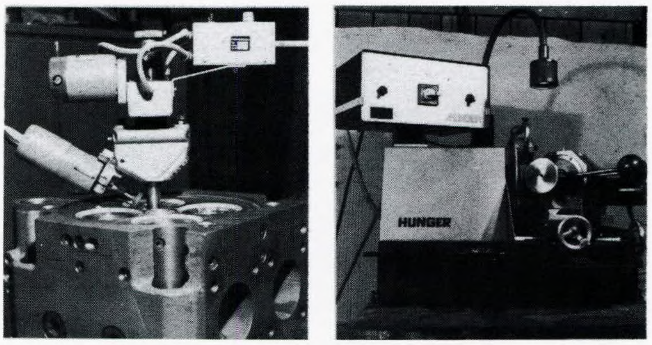


FIG. 29: Grinding devices for cylinder head seats and valves

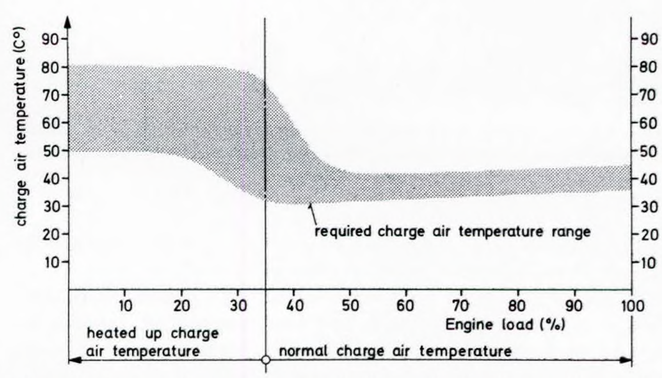


FIG. 30: Charge air temperature for heavy fuel oil operation with example of a central cooling system, permitting operation over full load range on HFO

stiffness and water-cooled exhaust valve seats offers considerable advantages:

- Optimum surface temperature with even and small thermal and mechanical deformation of the cylinder head and correspondingly low stress level (Figs. 24 and 25).
- Very low and uniform temperature at the exhaust valves and seat inserts, also due to the valve rotators (Fig. 26).
- Good valve sealing as a result of the valve rotator and the small and even cylinder head deformation.
- Exhaust valves of Nimonic with optimum corrosion resistance.
- The insulating air gap of the valve seat insert preventing low-temperature corrosion.

The service results obtained so far fully confirm the advantages of the basic design principles applied (Fig. 27 and 28). They include operational experience on heavy fuel oils with the vanadium/sodium ratio at its worst. Not a single valve failure has been reported.

Taking all the various operational influences into consideration, we can recommend the following inspection and overhaul intervals for inlet and exhaust valves: 6000–9000 h on HFO and 8000–12 000 h on MDO.

Valve overhaul procedure

Since the valve overhaul interval generally coincides with the piston ring renewal, this work can be combined. Furthermore, the absence of any valve age is not only a time-saving factor but makes the work straight-forward too.

The work is limited to refacing the valve seat in the cover to the specified angle using a grinding device and to refit a separately refaced or new valve (Fig. 29). Lapping-in of the valve in the seat is no longer required or specified.

Fuel injection system

Fuel pumps

The behaviour of the fuel pump of the engines operating on marine diesel or heavy fuel oil can be described as entirely satisfactory. Rare plunger seizures were mainly caused by initial impurities in the systems or in the fuel or lubricating oil itself.

With certain installations operating on heavy fuel oil, starting difficulties were experienced after some time. Investigations revealed an increased plunger clearance, mainly as a result of unsatisfactory fuel oil separation.

These early incidents have led to a new matching of plunger and barrel material (both are now nitrided), which has already proved its margin for punishment in actual service.

Fuel injection nozzle

Nozzles with rounded-off inner edges of the injecting holes have been in use for many years. The behaviour of such nozzles

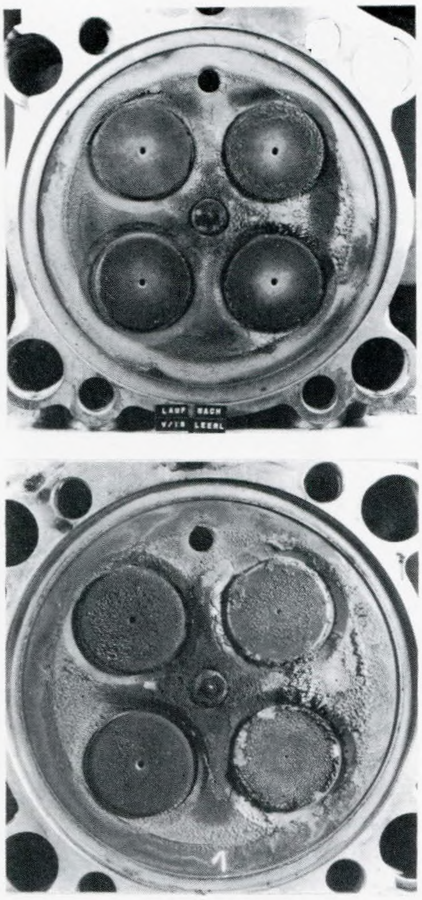


FIG. 31: Comparison of cylinder heads after 50 hours heavy fuel oil operation at very low engine load, with and without charge air heating

Component	Inspection or Overhaul: Interval (hrs)[1)] MDO[2)] A 20 D A/AS/AT 25 D	Inspection or Overhaul: Interval (hrs)[1)] HFO A 20 H AS/AT 25 H	Inspection or Overhaul: Time (min.)[3)]	Lifetime[1)] (hrs) MDO[2)] A 20 D A/AS/AT 25 D	Lifetime[1)] (hrs) HFO A 20 H AS/AT 25 H
Fuel nozzle	2000– 3000	1500–2500	15	4000– 6000	3000– 5000
Inlet valve			5 [5)]	16000–24000	12000–18000
Exhaust valve			5		9000–12000
Piston	8000–12000	6000–9000	40	48000–72000	36000–54000
Piston rings				8000–12000	6000– 9000
Scraper rings					
Piston ring groove				24000–36000[6)]	18000–27000[7)]
Gudgeon pin			10	48000–72000	36000–54000
Gudgeon pin bearing					
Bottom end bearing			60	16000–24000	12000–18000
Main bearing	[4)]	[4)]	60		
Fuel pump			35		
Valve seat			D:30 H:20	16000–24000	12000–18000
Cylinder liner			30	32000–48000	24000–36000

1) Expected intervals and lifetimes may vary and subject to:
- Environmental and operating conditions
- Fuel and lube-oil qualities within Sulzer specifications
- Engine load factor up to 0.9
- Fuel and lube-oil care according to Sulzer specifications
- Forgoing overhauls and genuine spares

2) Destillates

3) Total time for withdrawing and refitting under good conditions with ready tools, devises and lifting arrangements assumed (time for cleaning and overhaul should be added).

4) Checks at random (all within 4 years)

5) Cylinder cover 75 min.

6) Cast iron piston: oversized grooves

7) Steel crowns: oversized grooves or rechroming (piston skirt dismantling: 30 min.)

Replaces all previous issues
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FIG. 32: A20 and A25 inspection and overhaul intervals, man-hours and lifetimes

is more stable over a longer service period, with a narrower scatter of the injection rates on individual cylinders.

Engines, operating on heavy fuel oil, fully benefit from this design which, together with the introduction of a nitrided nozzle body, results in satisfactory life times with hardly any wear of the injection holes, even after 5000 or more running hours.

Low-load operation and shut-downs on heavy fuel

As is generally acknowledged, the rather poor ignition property of heavy fuel oil may lead to fouling of the engine in low-load operation. Two solutions can be offered: either switching over to marine diesel oil below about 30% load or—the more practical way—heating the charge air (Fig. 30). According to our tests, the heating up of the air shortens the ignition delay and provides cleaner combustion which, in turn, reduces the deposits in the combustion chamber and the exhaust gas turbines (Fig. 31).

However, experience has also shown that the borderlines are not strictly defined. Recently one of our service engineers visited a single-fuel ship and found our 6AS25H auxiliary engine operating satisfactorily on heavy fuel oil at about 28% load, with a charge-air temperature of only 32°C. This had been going on for 2–3 days.

Furthermore, many of our heavy-fuel A-type engines are not switched over to marine diesel oil for shut-downs. With preheating of the heavy fuel oil and the engine itself, restarting was possible.

Figure 32 summarizes the maintenance intervals and lifetimes of various engine components for the two sites.

BIBLIOGRAPHY

1. G. A. Lustgarten, 'Recent developments on the Sulzer A25 engine'.

Discussion

F. C. BOWN (Hamworthy Engineering Ltd): Could Mr Lowe tell us the result of the tests on the difficult fuel he obtained from an oil company? What would the effect have been if the fuel had found its way on to a ship? Would merely excessive wear have been caused or would the engines have failed, so hazarding the safety of the ship?

I next have a comment on Mr Nissen's paper regarding the LO purifying system. It would appear that, as drawn (Fig. 3), this could be a recipe for disaster, unless, of course, some valves had been omitted from the sketch.

S. G. DEXTER (Ricardo Consulting Engineers): We were interested to see Mr Eckert's comments on page 19 of his paper where he states that aluminium-alloy bearings of increased wear-resistance will be used in future to replace the traditional tri-metal bearing. Would he please clarify this and say whether he was referring to the Rillenlagern bearing as introduced by MIBA?

R. M. HOBSON (Lloyd's Register of Shipping): The three papers make interesting reading and add to the continuous build-up of experience in the operation of relatively small diesel engines on bunker oil.

All three authors refer to the importance of comprehensive fuel-oil treatment and my contribution concerns their recommended throughput of bunker fuel to the centrifugal separators.

Mr Nissen suggests on page 9 under 'Single-fuel system design' that low flow rates (less than 25%) be aimed for to ensure high cleaning efficiency. But he also suggests that separators be arranged for both parallel and series operation. If an owner is contracting for an installation on a new ship, then this matter of fuel-oil separator capacity is very important.

First I would ask what is meant by 25%. Separator capacity can mean all things to all men, particularly their manufacturers. Do they all use the same nomenclature in their brochures and is there a temptation for the competing manufacturers to overstate the capacity of their models?

If one follows the argument to its logical conclusion then the temptation might be for the manufacturer to understate the capacity and ensure a higher purity of fuel at the separator discharge than obtainable with competing models.

Secondly I would ask the authors how the alternative of series, as opposed to parallel, operation affects the choice of separator capacity. The difference in required separator size would, at first sight, appear to be considerable. Can Mr Nissen please elaborate more fully on this aspect of his paper?

Finally, may I express my appreciation to all three authors for the wealth of information imparted by their efforts.

Authors' replies

W. LOWE (Paper 13):

In reply to Mr Bown, the tests for which ignition delay is shown in Fig. 6 were carried out on a 600 rev/min engine and on two designs of 1000 rev/min engine, using a special fuel supplied by a major oil refinery. This fuel was not commercially available and was intended to provide information on the behaviour of fuels which might possibly result from future refinery marketing policies.

In fact, the fuel behaved much better than was expected and on the larger 600 rev/min engine gave quite satisfactory performance both at constant speed and at propeller law variable speed. On the 1000 rev/min engines performance was also quite acceptable at constant speed but on a propeller law characteristic the engine exhaust became smoky at low speed and torque. This would certainly not result in engine failure but would have been a cause for complaint if the engine had been run in this condition in port. As mentioned in the paper, the condition was improved by increasing the air-manifold temperature to recover some of the ignition delay.

After completion of the combustion tests, the heavy-fuel ring main in the laboratory was scavenged with distillate fuel and subsequently the main storage tank was topped up with a commercially available residual fuel. This mixture of fuels was later used for a different development project during which some of the fuel was held at about 130°C for several days. Thickening of this fuel occurred, the viscosity increasing from around 50 to 80 cSt at 80°C. By that time it was not possible to establish whether similar thickening of the original special fuel would have occurred by storage at high temperature or whether the situation had been complicated by a compatibility problem. Although there was no engine stoppage in this case, the experience suggests that it would be possible for engine failure to occur if thickening of the fuel from thermal degradation was severe enough. The effect would be to produce high fuel-injection pressures with possible seizure of the fuel-pump plunger or failure of the fuel cams or followers.

The only instances of wear attributed to fuel characteristics are described in the section of the paper entitled 'Fuel treatment' and illustrated in Figs 1 and 2. The piston-ring groove wear was initially found on a routine overhaul of one ship and the discovery of it caused examination of the sister ship, where it was also found.

Replying to Mr Hobson, the practice of my company has always been to run the purifier and clarifier in series, as this gives the most secure protection from water contamination of the fuel. In recent years this system has become the one most generally preferred by other manufacturers.

The choice of separator capacity arises from experience and, of course, the experience includes the manufacturer of the equipment as well as the model size. An understatement of the capacity by the equipment manufacturer would result in a price disadvantage, whereas an overstatement would result in poorer performance. The proof of the pudding is, as always, in the eating.

P. C. NISSEN (Paper 14):

In response to Mr Bown's comment on the lubricating oil purifying system, common treatment of lub oil from several diesel engines with one separator is due to technical and economical reasons:

- Small separators of corresponding size are not available;
- One separator per engine exceeds the acceptable price limit.

In 1981, my company introduced the common system, and so far 67 auxiliary engines are successfully running on it, with a maximum of four engines being supplied in common. The outlet for the lub oil from the engine's circulating tank to the collecting tank is at the maximum oil level, by means of a direct connection (without tube bend as in Fig. 3). The tube bend (with a ventilation bore at the highest point) functions well, but vibration damage has occurred. With pipe cross-sections of the correct dimensions, cooler oil will circulate around a switched-off engine.

The feed and return pipes of the circulating tank must be provided with a shut-off valve.

An alternative to the above system is to clean the lub oil of several engines one after the other with one separator. This method has the following disadvantages:

- Faults can occur when switching over;

- It is not always possible to clean the lub oil of a running engine (immediate removal of dirt particles is important);
- The separator will usually be oversized as regards capacity.

In reply to Mr Hobson, this definition applies to Alva-Laval and Westfalia equipment. The flow rate depends on the viscosity of the heavy fuel. The following values must not be exceeded:

up to 180 cSt at 50°C, 30%
up to 380 cSt at 50°C, 25%
up to 460 cSt at 50°C, 20%
up to 700 cSt at 50°C, 15%.

As to the question of parallel or series arrangement, the separator size is identical in both cases. With series arrangement of the purifier and clarifier the flow rate (referred to 380 cSt at 50°C) is 25%, whereas it is 12.5% for parallel arrangement. For parallel arrangement a common separate feed is required. This is also valid for modern separators made by Alfa-Laval and Westfalia, the ALCAP system in general being designed for parallel arrangement (main and standby separator) or for individual operation. In this case individual operation means a flow rate of 25%.

B. ECKERT (Paper 15):

In reply to Mr Ricardo, aluminium-alloy bearings with increased wear resistance do include Rillenlager-type bearings.

Answering Mr Hobson's comment on series and parallel operation, Alfa-Laval recommend that for heavy-fuel-oil cleaning plants dimensioned specifically for operation in series, the maximum throughput capacities are increased by 35% (purifier and clarifier plus standby).