SULZER **RTA-T**

# Technology Review

Sulzer RTA48T-B, RTA58T-B and RTA68T-B types

WÄRTSILÄ NSD
CORPORATION

# Sulzer RTA-8T engines:
# Compact two-strokes for tankers and bulk carriers

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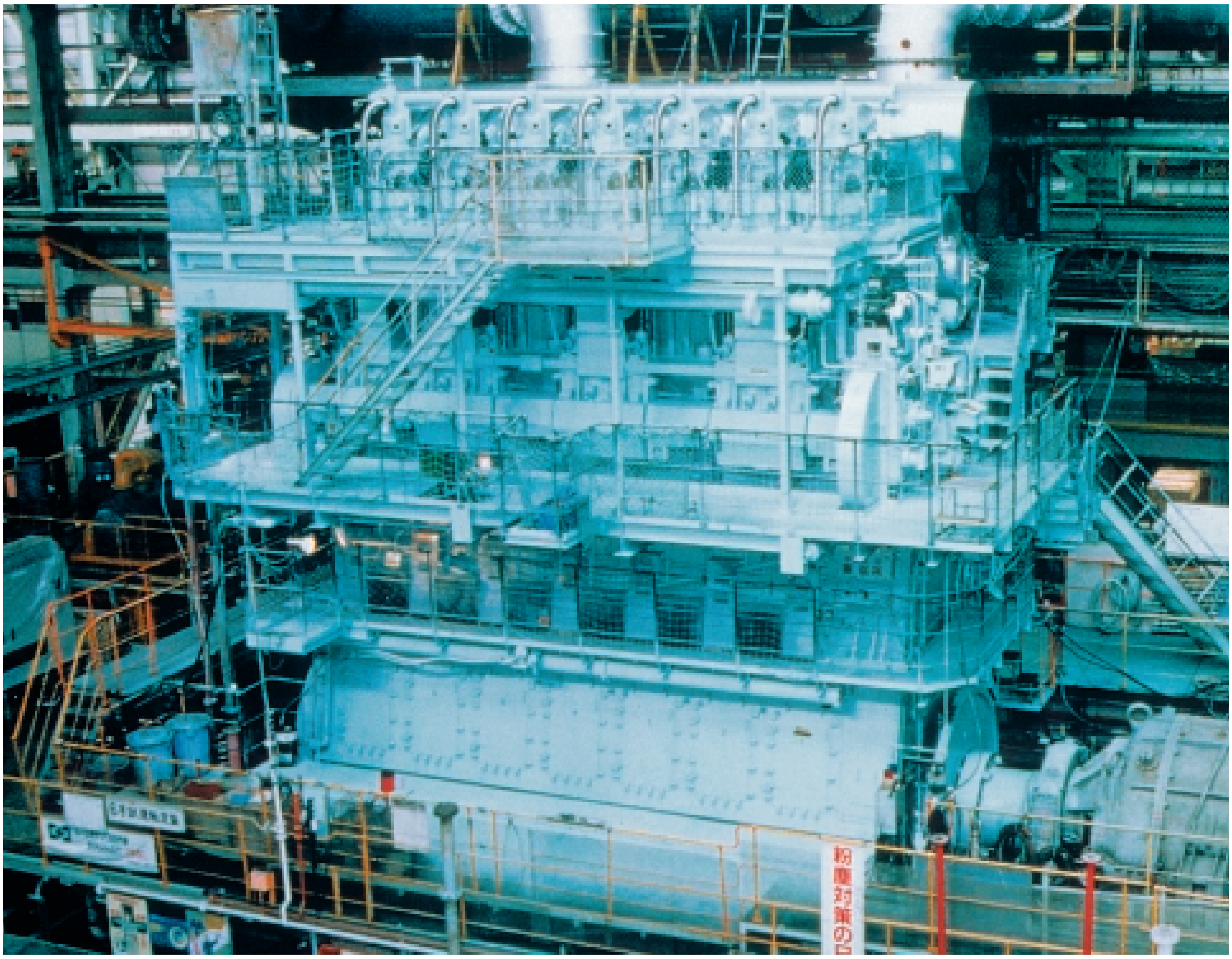


Fig. 1
Seven-cylinder RTA58T engine on test in the Kobe works of Mitsubishi H.I. in December 1997
[7798-3005-1]


# Summary

The introduction of the Sulzer RTA48T, RTA58T and RTA68T types of low-speed marine two-stroke engines, which are collectively designated the RTA-8T line, is extremely successful with more than 116 engines ordered by the end of 1997. The engines meet perfectly the needs of Handysize, Handymax, Panamax and Capesize bulk carriers and their equivalent tankers, including Aframax and Suezmax. To ensure the safe introduction of the new engine types, the parameters which determine the power output were selected conservatively similar to those of other engines in Wärtsilä NSD's low-speed engine programme, although the majority of the engine components already incorporate the potential for a greater power output.

After the first RTA-8T engines had accumulated 3000 running hours, the new 'Version B' design was announced at the beginning of October 1997. The Version B provides a power increase (uprating) of seven per cent, together with a limited number of design changes first to comply with the requirements of the increased power and second to make the engines better for manufacturing. The development step is based on the experience with many Sulzer RTA engines in service, particularly with the 20 RTA-8T engines in service.

This paper presents the design of the RTA-8T engine types, together with accounts of their testing, early service experience and development for the Version B.


**Key points**
Summaries of key points are given in boxes throughout the paper on pages 5, 13, 30, 37 and 39.

# Introduction

Since its introduction some 16 years ago, the well-known Sulzer RTA series of low-speed marine two-stroke crosshead diesel engines has been continuously updated to adapt it to the latest market requirements and to incorporate the benefits of technical developments. It now comprises engines from 380 to 960 mm bore covering a power range of 1480 to 65 880 kW (2000–89 640 bhp). Within this series, there are three main lines of engines – the RTA-C, RTA-U and RTA-T engines – characterised by their respective stroke/bore ratios of less than 3.0, about 3.5 and around 4.0.

Of these three lines, the RTA-T is purpose designed for the economical propulsion of tankers and bulk carriers from 20 000 to 300 000 tdw, or more. The RTA-T engines offer clear, substantial benefits in ship operation, as well as for shipbuilders and engine builders. Particular mention can made of their unparalleled compact dimensions, ease of installation, ideal matching for power and speed, and manufacturing-friendly design, together with their objective of long times between overhauls based on the solid background of the service experience from the many RTA-series engines in service. The engine design is based on the well-proven RTA-U engines as well as on the outcome of close discussions with shipyards and shipowners about their requirements.

The RTA-T engines are thus today the most modern, manufacturing friendly and reliable 'workhorses' for bulk carriers and tankers.

The RTA-T line comprises four engine types, the RTA48T, RTA58T, RTA68T and RTA84T, which cover a power range of 4375–34 920 kW (5950–47 520 bhp) at speeds of 127–54 rev/min (Fig. 2). The first RTA84T entered service operation in 1994 and had accumulated some 28 000 running hours at the end of 1997. The RTA84T type, however, is not included in the coverage of this paper because it has a design concept that is distinctively different to the other three types. The three smaller-bore RTA-T engines, namely the RTA48T, RTA58T and RTA68T, are collectively termed the RTA-8T engines.

The first two RTA48T engines went into service at the beginning of 1997 (only about one and a half years after the engine's announcement), and each had accumulated more than 5000 running hours at the end of January 1998. Inspections carried out so far confirmed that the RTA48T engines are running to our full satisfaction, with maximum diametrical liner wear rates of only 0.012 mm/1000 hours.

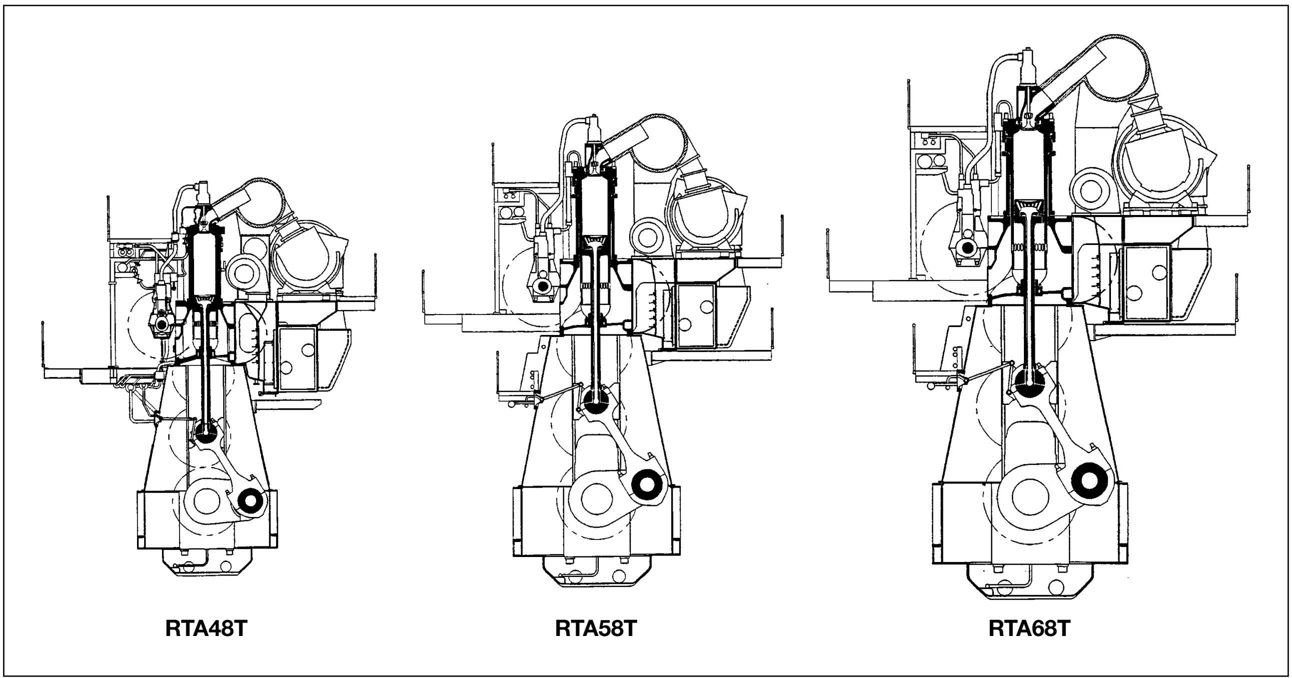



Fig. 2
Cross sections of the three engine types in the Sulzer RTA-8T engine family, the RTA48T, RTA58T and RTA68T to approximately similar scales
[97#222-1]

The engine parameters of the three RTA-8T types were selected to offer optimum power and speed ranges with six- and seven-cylinder engines for the respective specific ship types. However, recent market investigations clearly showed that there is a strong demand for slightly higher power outputs per cylinder for the RTA-8T types. By using the inherent potential of these three engine types, a power increase of seven per cent was introduced in October 1997. The power output is achieved by simultaneously raising the brake mean effective pressure by 4.5 per cent to 19.0 bar and the mean piston speed by 2.5 per cent to 8.5 m/s. Nevertheless these new parameters are still within the experience gained with the 4RTX54 'Technology Demonstrator', the former research engine of Wärtsilä NSD Switzerland which was in operation from 1990 to 1995 for totally more than 3000 running hours. The parameters of that engine were a stroke/bore ratio of 3.98, brake mean effective pressure of 20 bar, mean piston speed of 8.5 m/s, and maximum cylinder pressure of 180 bar.

## Market requirements

Before the development of the RTA-8T types was initiated in the early 1990s, an intensive market investigation was carried out among shipowners, shipyards and engine builders in East Asia and Europe.

A clear trend could be recognised towards a greater standardisation of ship designs (Fig. 3), such as Handysize (20 000–40 000 tdw), Handymax (40 000–50 000 tdw) and Panamax bulk carriers (60 000–75 000 tdw), as well as product tankers (40 000–45 000 tdw) and Aframax tankers (85 000–100 000 tdw). A slight increase in required engine power but also a rather low engine speed compared with the past could also be identified, especially for the smaller ships.

As a consequence of recognising this standardisation, it was possible to design engines that fit optimally to the ship's requirements with regard to power, speed and cylinder number. An 'optimum fit' further reduces engine first cost, because 'heavy derating' need no longer be requested. Moreover, it was also possible to reduce the size of the rating fields, thereby giving more freedom for optimising the engine design itself.

An even more important result of the discussions with shipowners, shipbuilders and engine builders was that a clear priority list could be formulated for their individual requirements regarding propulsion engines. Apart from optimum engine power and propeller speed requirements, the shipowners' number one requirement is still reliability, whereas the shipyards are asking for low engine first cost as first priority. Furthermore, the engine builders expect a manufacturing friendly, cost effective engine design, together with the common reliability potential.

**Key points**

The trends in bulk carrier and tanker newbuildings during 1986–1994 were:

- Slight increase in required engine power;
- Lower engine speeds than in the past;
- Greater concentration on the standard ship sizes with narrower ranges of deadweight and a shift to greater deadweight.

The market needs of all parties for a propulsion engine in bulk carriers and tankers were found to be, but without any implied priority, as follows:

- Power and speed to be optimum for the standard bulkers and tankers;
- Low engine first cost;
- Manufacturing friendliness;
- Well adapted to installation on board ship;
- Reliability and durability;
- Economical in operation;
- Simplicity in operation and maintenance;
- Compliance with forthcoming exhaust gas emissions regulations.

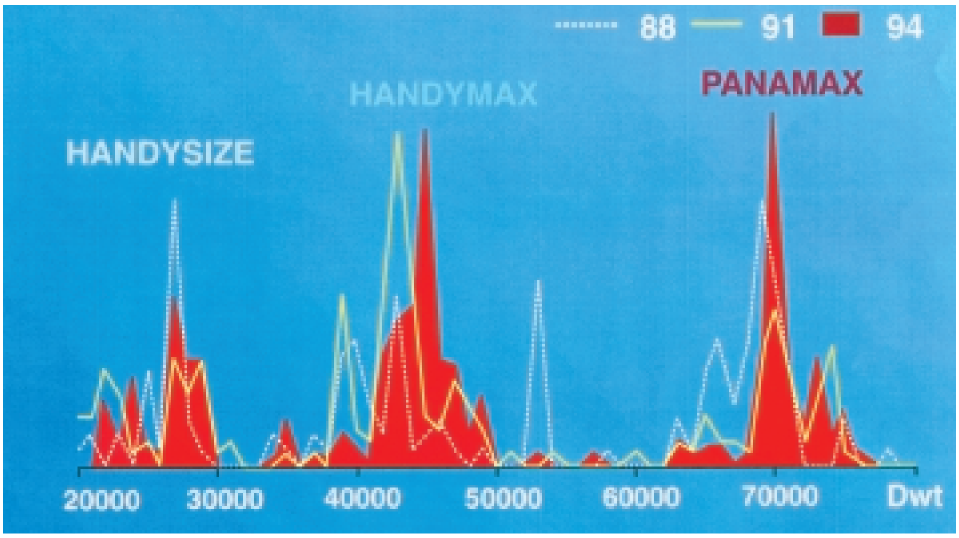



Fig. 3
The number of bulk carrier newbuildings showed during 1986–1994 an increasing concentration of ships on standard sizes. For the three standard sizes, Handysize, Handymax and Panamax, there was a narrowing of the tonnes deadweight range and a shift to greater tdw for each three-year period
[98#100]

# Market success of RTA-8T engines

The introduction of the Sulzer RTA48T and RTA58T in June 1995 was sensationally successful for completely new marine engine designs. Only one year after their announcement, more than 50 engines could be registered in the order book. By the end of December 1997, more than 116 RTA-8T engines had been ordered, with an aggregate power output of 1.15 million kW (1.56 million bhp). These include 84 RTA48T and 32 RTA58T engines (see Table 1), of which some are already of the Version B, and a significant number is currently under discussion for further projects. Many more repeat orders can be expected since these engine types have been selected for standard ship designs.

This market success is based on the fact that the RTA-T engines fully comply with the above-mentioned market requirements. In brief, the engines are distinguished by the following attributes:

- Power/speed combination as required by today's ships, but without derating.
- The engines are designed for manufacturing. Moreover, as less derating is required compared with previous engine generations, additional cost savings are obtained in the first cost.
- The engines are fully optimised for easy installation in the engine room, as for example by their minimum number of side stoppers, fewer holding-down bolts, etc.
- The engines offer unparalleled compact dimensions.

The advantages of RTA-T engines were soon appreciated by the market and the reaction can be seen in the very satisfactory number of RTA-T engines ordered so far. The first engine of the RTA-8T series, a 6RTA48T, was ordered from Diesel United Ltd in Japan in July 1996 for a 42 000 tdw bulker contracted with Ishikawajima Harima Heavy Industries (IHI).

The centre of gravity of the market for the RTA-8T engines lies, as expected, in East Asia, yet it was very encouraging when also Szczecin Shipyard in Poland decided to install the 6RTA58T into their standard 1800 TEU open-hatch container/bulk carrier. Twelve of these ships have already been ordered and their engines are currently under construction by H. Cegielski in Poland. By the end of 1997, the first two Polish-built 6RTA58T engines had already successfully passed their sea trials. The second of these engines served in October 1997 for a type test to Classification Society requirements.

**Table 1: Numbers of Sulzer RTA-8T engines ordered and in service (at the end of December 1997)**

| Type | Cylinders | Engines ordered | Engines in service |
|---|---|---|---|
| RTA48T | 5 cyl | 4 | – |
| | 6 cyl | 35 | 6 |
| | 7 cyl | 43 | 10 |
| | 8 cyl | 2 | – |
| | Total | 84 | 16 |
| RTA58T | 4 cyl | – | research engine only |
| | 5 cyl | 2 | 1 |
| | 6 cyl | 18 | 2 |
| | 7 cyl | 12 | 1 |
| | Total | 32 | 4 |
| Total RTA-8T | – | 116 | 20 |

# Development targets

The development goals for a low-speed marine engine are many, often interact and in some cases even conflict with each other. The RTA-8T engines are no exceptions. Apart from the goals coming from the market and application requirements given above under the heading 'Market requirements', several goals were defined for these engines.

As the overall goal, it was agreed at an early stage that the RTA-8T engines should be significantly more attractive than the competitors' corresponding latest designs! This included all aspects, such as reliability, first cost and operational cost. This primary goal was commonly regarded as the most important because it was our target not just to replace RTA-U engines in existing standard designs but mainly to gain new markets for Sulzer engines. From this overall goal, a number of targets for the RTA-8T types follow as a consequence, namely:

- To be as similar as possible to the existing modern types of RTA engines to maintain superior reliability, but incorporating the latest developments in engine design;
- Highest possible degree of reliability envisaging two years' time between overhauls (TBO);
- Low thermal load in the combustion chamber components, with three fuel injection valves per cylinder for bore sizes greater than 520 mm;
- Low fuel consumption by the application of the latest design high-efficiency turbochargers in combination with 'low-port' liners, while not penalising the shipyard with exhaust gas temperatures that are too low;
- Make use of the potential available from the deep combustion chamber enabled by the large stroke/bore ratio to obtain greater freedom for fuel spray optimisation towards lowest fuel consumption without penalising engine reliability;
- Lowest possible wear rates for cylinder liners and piston rings;
- Reasonably low cylinder lubricating oil consumption for good overall costs;
- To comply with the IMO exhaust emissions regulations to be implemented in the year 2000;
- Improved piston rod gland design and performance in terms of:
  - Less drainage from the neutral space,
  - Low crankcase oil contamination,
  - Retarded rise in base number (BN) of system oil;
- Ease of manufacture;
- Increased structural safety through simplified welding procedures for the columns and bedplate;
- Ease of maintenance in service;
- Ease of installation for the shipbuilder;
- Ease of access for monitoring the engine in service.

Of course, it is normal practice that the levels of mechanical stresses and thermal strains in all the relevant components are kept well within known limits.

Special note must be made of exhaust gas emissions. Today every engine development must also take into account the IMO regulations for the control of $NO_X$ emissions that will be introduced for new ships (with reference to the laying of their keels) on 1 January 2000. All Sulzer diesel engines can be delivered so as to comply with the speed-dependent $NO_X$ limit. In the great majority of cases, the limit will be met simply by adapting the engine tuning. However, as fuel consumption and $NO_X$ emissions are interrelated, engines that are to be offered with both a guaranteed fuel consumption and to comply with the IMO limit for $NO_X$ emissions are subject to a wider tolerance in fuel consumption.

Reaching the above development goals is accomplished mainly on three technological pillars:

- The accumulated service experience from more than 1500 RTA engines already in operation assures proper feedback.
- New design concepts, the feedback from service experience and the results from engine testing are combined in the latest computer-based analytical and design tools, including extensive use of three-dimensional finite-element techniques.
- Fresh knowledge is collected from tests on the research engines in Winterthur, from a great number of engines running under field-testing supervision, and from production engines at our licensees. For the RTA-8T engines, this fresh knowledge mainly came from the 4RTX54 'Technology Demonstrator' which was running at high parameter levels for more than 3000 hours in the early 1990s.

Together, these three pillars form a sound foundation for the design of new engines, with the objective of achieving good reliability in service right from the beginning.

# The new Version B compared with original version

In recent years, Wärtsilä NSD Switzerland has pursued a clear strategy of consolidating and upgrading the competitiveness of its engine programme (Fig. 4). The RTA series was inaugurated in 1981 with the introduction of the RTA38, RTA48, RTA58, RTA68, RTA76 and RTA84 engines which are collectively termed the RTA-8 types. In 1984 they were joined by the RTA-2 types, namely the RTA52, RTA62 and RTA84M, with the RTA72 being added in 1986. The stroke/bore ratio was increased from 2.86 in the RTA-8 engines to about 3.47 in the RTA-2 types.

The ratings of the RTA-2 engines were consequently increased in 1987. Various design improvements, together with higher ratings, were also introduced for the RTA-2 engines in 1988. Also in 1988, the RTA84C was introduced to match the specific needs of large container ships. In 1991, the RTA84T engine with stroke/bore ratio of 3.75 was added to the engine programme, covering the needs of VLCCs.

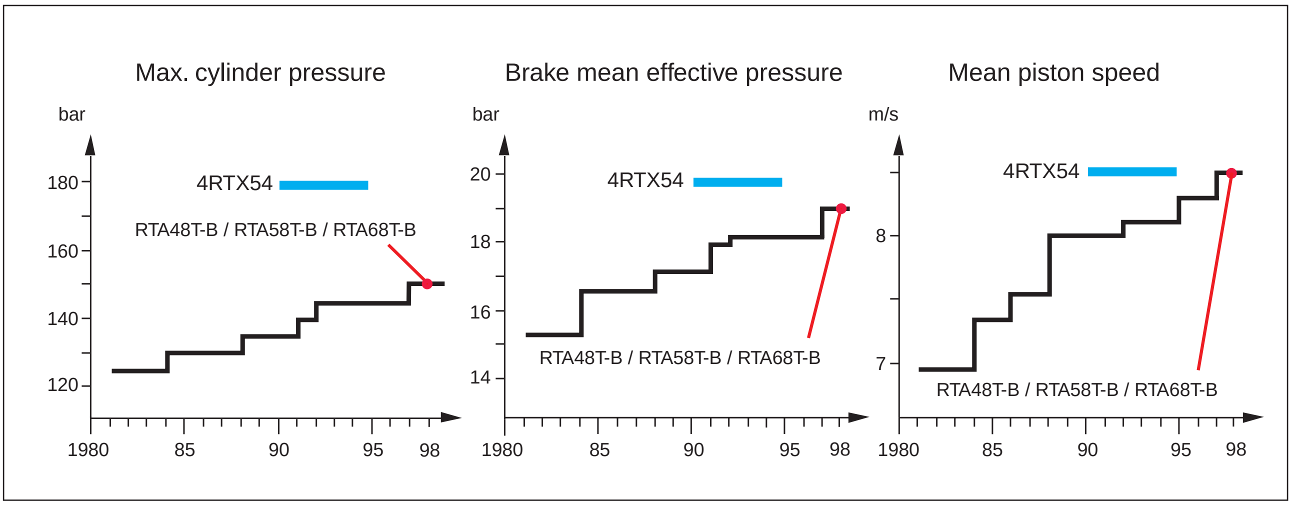
The next step in this development progress was the upgrading of the RTA-2 types in 1993 to give the RTA52U, RTA62U and RTA72U engine designs, the RTA-U types. As a result, their power outputs were increased by some nine per cent. Shortly afterwards, the rating of the RTA84C was increased to correspond with that of the RTA-U types. In 1994, the RTA96C with a stroke to bore ratio of 2.6 was introduced, covering the power needs of the future container ship market, soaring further upwards. In 1995, the first two RTA-8T types, the RTA48T and RTA58T with a stroke/bore ratio of 4.17, were introduced, completed by the RTA68T in 1996, covering the needs of tankers and bulkers.

It is notable that the RTA96C as well as the RTA-8T types were introduced at the same thermodynamic parameters as the RTA-U types, namely at a brake mean effective pressure of 18.2 bar and a mean piston speed of 8.2 m/s. It has always been the policy of Wärtsilä NSD Switzerland to introduce new engines at the same parameters as the existing engines. This is to minimise the specific risks of introducing a new product. The merit of this policy was confirmed by the inspections of the first RTA48T engines in service, when each had accumulating more than 2000 running hours. The inspections proved that the engines are running to our full satisfaction, for example with maximum diametrical liner wear rates measured after 2000 hours of only 0.025 mm/1000 hours, levelling out to be 0.012 mm/1000 hours measured after 5135 running hours.

Consequently, owing to the inherent engine potential and the satisfactory service experience accumulated up to now, a small increase of the power output of seven per cent was announced in October 1997 (Fig. 5). To allow adequate differentiation between the existing and the uprated engine types, the uprated engines are identified by adding ‘B’, as in ‘RTA68T-B’.

This increased rating provides the power required for the modern tendency in ship design towards slightly increased propulsive power, yet keeps the same compact engine dimensions. The new rating is also provided at the same specific fuel consumption as at the previous rating. So, as an alternative, re-

Fig. 4
Progress achieved in some key parameters of Sulzer RTA-series engines
[98#014-1]



duced fuel consumption levels are possible at the previous power outputs. In certain cases, the increased output also helps to reduce the number of cylinders required.

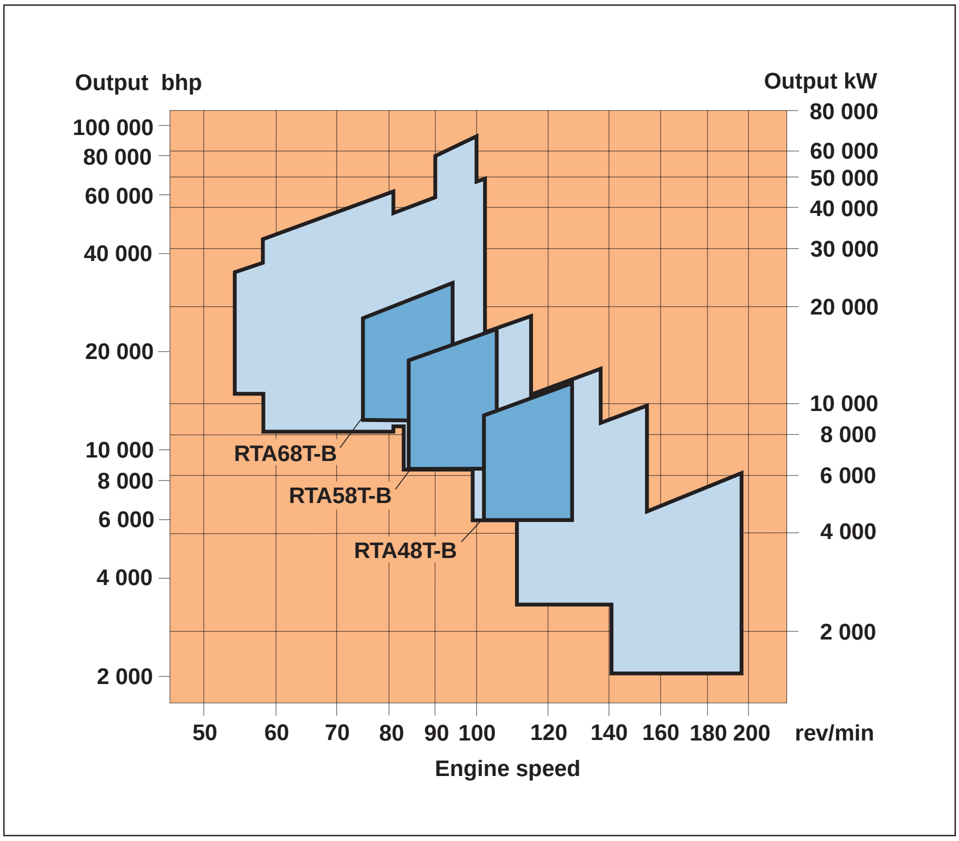
To comply with the requirements of the higher power outputs, some components have been slightly modified. The opportunity was also taken to incorporate some design modifications to further improve the engines' competitiveness in terms of 'design for manufacturing', allowing the engine manufacturing cost to be kept at the same level, or even slightly lower, compared with the previous version. In this way, the engine 'price per horse-power' could be reduced by some seven to eight per cent.

The increased power output of the Version B of the RTA-8T engines is based on three key sets of knowledge, namely:

- Service experience from 332 Sulzer RTA-series engines in service with an MCR brake mean effective pressure of 18 bar or higher, and mean piston speed of 8 m/s or higher, including the first few RTA-8T types;
- The experience gained with the 4RTA58T research engine in the Diesel Technology Center in Winterthur, operated at brake mean effective pressures up 19.5 bar and mean piston speeds up to 8.6 m/s;
- The research and development work with the 4RTX54 'Technology Demonstrator' at a mean piston speed of 8.5 m/s, brake mean effective pressure of 20 bar and maximum cylinder pressure of 180 bar.

The total of more than 1500 RTA-series engines now in operation world-wide provides a solid basis for successful engine development. The first RTA-series engines began operation some 15 years ago in November 1983. After initial teething problems, mainly in piston-running behaviour during the 1980s, the RTA engine acquired a clear competitive edge in terms of reliability and durability. Two years' operation, or 12 000 running hours, time between major overhauls have been achieved as standard (with the best engines being even higher!). Exhaust valves are now routinely running more than 30 000 hours between overhauls, while piston rings and cylinder liners are attaining unparalleled low wear rates.

Fig. 5
Rating fields of Sulzer RTA-8T Version B engines superimposed on the overall field of the complete RTA series
[98#025-1]



## Engine parameters

To ensure the best basis for engine reliability and durability in operation, the principal parameters of the RTA-8T types were selected without exceeding those already employed in the RTA-U engines introduced in 1993. Thus, for the six-cylinder models, the power of 8160 kW (11 100 bhp) was offered for the RTA48T, 12 000 kW (16 320 bhp) for the RTA58T and 16 500 kW (22 440 bhp) for the RTA68T.

The power increase in the resulting Version B combined both an increase of the brake mean effective pressure by 4.5 per cent to 19.0 bar and the mean piston speed by 2.5 per cent to 8.5 m/s (see Table 2). Thus, for the six-cylinder models in the Version B, the new power of 8730 kW (11 880 bhp) could be offered for the RTA48T, of 12 750 kW (17 340 bhp) for the RTA58T and of 17 640 kW (24 000 bhp) for the RTA68T.

Moreover, the brake specific fuel consumption (BSFC) of the uprated engines remains unchanged at the nominal MCR values (R1) so that the previous power outputs can be obtained with about 2 g/kWh lower BSFC. This has been achieved by increasing the maximum cylinder pressure of the RTA-8T engines from 142 bar to 150 bar in the Version B.

The Version B engines offer their greater power concentration within exactly the same outline dimensions as the previous versions of the RTA-8T engines. The RTA-8T engines are distinguished by their unparalleled compact dimensions. The length of a 6RTA58T engine, for example, is about one metre shorter than a 6RTA62U engine but, more significantly, the engine is also some 600 mm shorter than the latest corresponding model of the competitors' engines (Fig. 6). This gives shipbuilders the advantage of shortening the engine room accordingly (Table 3). In addition, despite their long stroke, the engine height is also kept within the same dimensions as RTA-U engines, owing to the shorter connecting rods and other improvements.

**Table 2: Principal parameters of the Sulzer RTA-8T engines**

| Type | | RTA48T | | RTA58T | | RTA68T | |
|---|---|---|---|---|---|---|---|
| Version | | original | Version B | original | Version B | original | Version B |
| Bore | mm | 480 | | 580 | | 680 | |
| Stroke | mm | 2000 | | 2416 | | 2720 | |
| Output, R1 | kW/cyl | 1360 | 1455 | 2000 | 2125 | 2750 | 2940 |
| | bhp/cyl | 1850 | 1980 | 2720 | 2890 | 3740 | 4000 |
| Speed Range | rpm | 124–99 | 127–102 | 103–82 | 105–84 | 92–74 | 94–75 |
| BMEP at R1 | bar | 18.2 | 19.0 | 18.3 | 19.0 | 18.2 | 19.0 |
| Pmax | bar | 142 | 150 | 142 | 150 | 142 | 150 |
| Mean piston speed | m/s | 8.3 | 8.5 | 8.3 | 8.5 | 8.3 | 8.5 |
| Number of cylinders | | 5–8 | | 5–8 | | 5–8 | |
| BSFC at full load, R1 | | | | | | | |
| | g/kWh | 171 | | 170 | | 169 | |
| | g/bhph | 126 | | 125 | | 124 | |

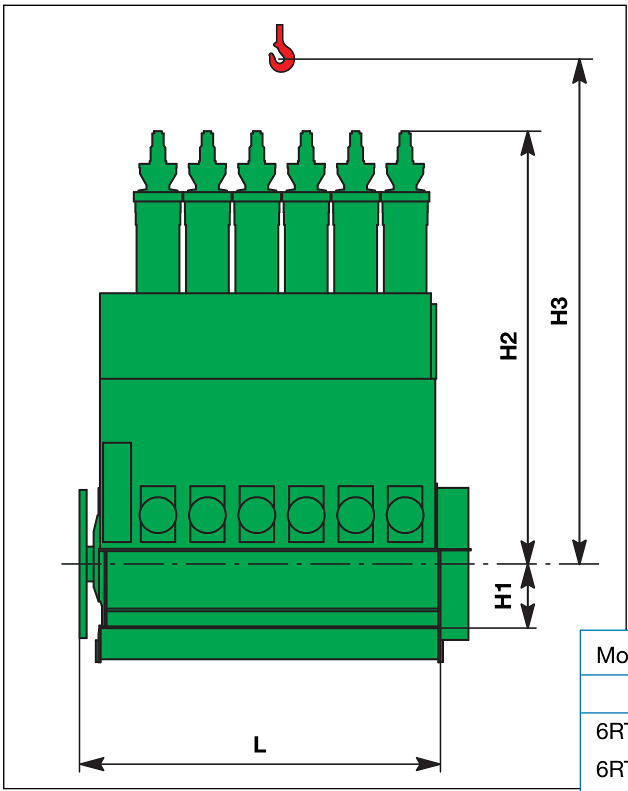



Fig. 6
Dimensions and masses for six-cylinder RTA-8T engines
[98#104]

| Model | Length | Height | Height | Seating | Mass |
|---|---|---|---|---|---|
| | L, mm | $H_2$, mm | $H_3$, mm | $H_1$, mm | tonnes |
| 6RTA48T-B | 6134 | 7334 | 9030 | 1085 | 195 |
| 6RTA58T-B | 7336 | 8810 | 10 880 | 1300 | 322 |
| 6RTA68T-B | 8620 | 10 250 | 12 000 | 1520 | 472 |

**Table 3: Comparison of main dimensions for six-cylinder engines ('L' is indicated in Fig. 6)**

| Marque | Engine | Mass | Power | Power | | Length | BSFC |
|---|---|---|---|---|---|---|---|
| | | tonnes | [kW] | [bhp] | [kg/kW] | L, [mm] | [g/bhph] |
| Sulzer | RTA68T-B | 472 | 17640 | 24000 | 26.8 | 8620 | 124 |
| MAN-B&W | S70MC-C | 525 | 18630 | 25320 | 28.2 | 9388 | 124 |
| Mitsubishi | UEC75LSII | 610 | 17650 | 24000 | 34.6 | 9640 | 121 |
| Sulzer | RTA58T-B | 322 | 12750 | 17340 | 25.3 | 7364 | 125 |
| Sulzer | RTA58T | 320 | 12000 | 16320 | 26.7 | 7364 | 125 |
| MAN-B&W | S60MC-C | 345 | 13530 | 18420 | 25.5 | 8047 | 125 |
| Mitsubishi | UEC60LSII | 310 | 11918 | 16200 | 26.0 | 7712 | 122 |
| Sulzer | RTA48T-B | 196 | 8730 | 11880 | 22.5 | 6148 | 126 |
| Sulzer | RTA48T | 195 | 8160 | 11100 | 23.9 | 6148 | 126 |
| MAN-B&W | S50MC-C | 210 | 9480 | 12870 | 22.2 | 6765 | 125 |
| MAN-B&W | S46MC-C | 185 | 7860 | 10710 | 23.5 | 6223 | 128 |
| Mitsubishi | UEC50LSII | 190 | 8253 | 11200 | 23.0 | 6462 | 123 |

## Engine modifications for the Version B

The detailed component modifications that have been necessary for the new ratings of the RTA-8T Version B engines are listed in table 4.

**Table 4: Modified components for RTA-8T Version B engines**

| Item | Modification | Reason |
|---|---|---|
| Crankshaft | enlarged diameter journal/pin | dynamic stresses due to increased Pmax |
| Bedplate | thicker lateral plates | dynamic stresses due to increased Pmax |
| Main bearing | adapted to crankshaft | – |
| Bottom end bearing | adapted to crankshaft | – |
| Column | thicker lateral plates | dynamic stresses due to increased Pmax |
| Cylinder block | reinforced | dynamic stresses due to increased Pmax |
| Camshaft damper | adapted to new rating (only single cases) | new vibration modes due to increased Pmax and modified crankshaft |
| Air receiver, T/C support, air duct | welded in one piece, no separate housing for cooler | design for manufacturing |
| Cylinder lubrication | integral lub. oil pump, support eliminated, no tacho, less assembling | design for manufacturing |
| Camshaft coupling | flange type | design for manufacturing |

# Design features and performance

The design features of the RTA-8T engines (Fig. 7) are presented here in the Version B form with particular emphasis on their influence on engine performance in terms of reliability, fuel consumption, durability, times between overhauls, exhaust emissions, etc.

## Fuel consumption and $NO_X$ emissions

To achieve low operational costs, particular emphasis during the development was put on the fuel consumption. Owing to their large stroke-to-bore ratio, the RTA-8T engines reach impressively low fuel consumption figures; for example, giving 169 g/kWh (124 g/bhph) at the R1 rating for the RTA68T, corresponding to a thermal efficiency of effectively 50 per cent.

Achieving low fuel consumption figures seemed to be a dilemma for many years. At the beginning of the 1990s, the turbocharger manufacturers introduced their 'high-efficiency' turbochargers, for example the ABB VTR-4E and VTR-4D, and the Mitsubishi

**Key Points**
The RTA-8T engines are very similar to the thoroughly well-established RTA-U types. The key features of the designs are given below with significant differences noted.

**Structure:**
- Rigid;
- Gondola-type bedplate;
- Stiff thin-wall box-type columns;
- Cast-iron cylinder blocks;
- Main bearing caps secured by elastic holding down bolts.

**Running gear:**
- Semi-built crankshaft;
- Main bearing: thin-walled white-metal shells;
- Crosshead with full-width lower half bearing;
- Crosshead bearing: thin-walled white-metal shells;
- Separate high-pressure lubricating oil supply to the crosshead for hydrostatic lift off;
- Gear drive for camshaft.

**Fuel injection equipment:**
- Three fuel-injection valves per cylinder in the RTA58T and RTA68T, but two valves per cylinder in the RTA48T;
- Two-piece uncooled injection nozzles with Stellite 6 tips;
- Double-valve controlled fuel injection pumps;
- Electronically-regulated VIT (variable injection timing) system.

**Combustion chamber components:**
- Full bore cooling for all combustion space components;
- Cylinder covers of higher grade material for greater margin against corrosion fatigue;
- Piston crowns with combined jet-shaker oil cooling for low surface temperatures;
- Cladding of cylinder covers near fuel injection valves as standard.

**Piston-running behaviour:**
The proven features for the good running behaviour necessary for long TBO are:
- Cylinder liner material with sufficient hard phase and ductility;
- Smooth machining of the liner surface, maintaining a precise geometry;
- Full honing of the running surface;
- Bore cooling of all combustion chamber components;
- The careful matching of the liner wall temperature distribution to eliminate corrosion attack;
- Multi-level cylinder lubrication for optimum distribution of the precious cylinder lubricating oil;
- Highly-efficient water separator and drain after scavenge air cooler;
- Plasma-coated top piston rings.

**Supercharging system:**
- High-efficiency turbochargers, such as ABB VTR-4E and VTR-4D types, and the Mitsubishi, MET-SD and MET-SE types. In addition, MAN NA/S and NA/T9 turbochargers may be chosen for selected projects;
- Efficient exhaust gas flow, with the manifold having a tangential inlet and outlet for maximum conservation of gas energy;
- Scavenge air ports with reduced height.

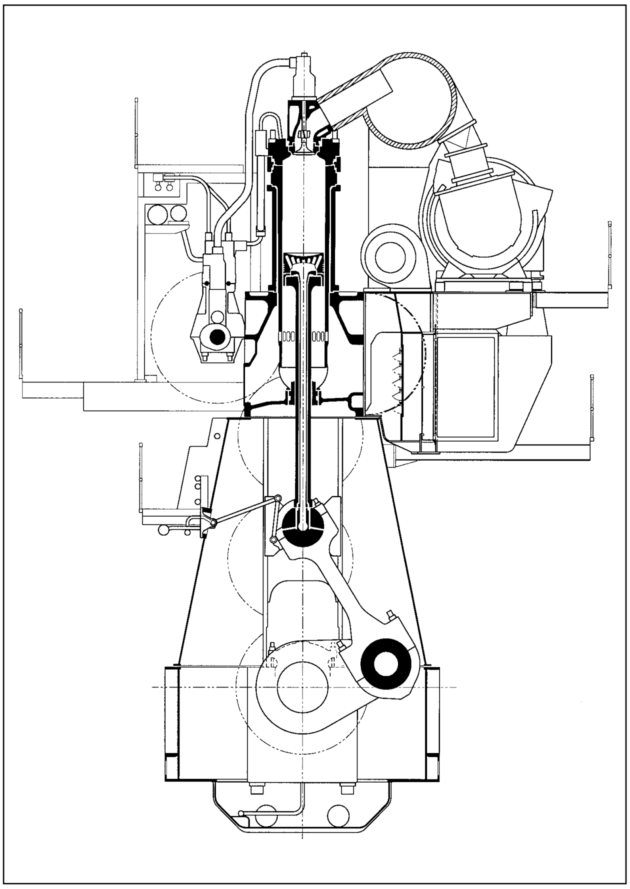


Fig. 7
Cross section of the RTA58T-B
[7798-3001]

MET-SE, with efficiencies being some three to four per cent higher than their previous versions. Owing to their great efficiency, the new turbochargers allowed engine designers to select an engine tuning that is optimised on maximum scavenging efficiency or, in other words, the purity in the cylinder is further improved. This meant that the amount of exhaust gas remaining in the cylinder was further reduced. However, the purity in the cylinder is today already at a very high level, even when applying conventional turbochargers, which do not have the highest efficiency available today. Consequently, the application of high-efficiency turbochargers on 'standard-design' engines can only improve the fuel consumption by some 1 g/kWh but, in parallel, the exhaust gas temperatures drop by some 25 to 30 °C, as confirmed by measured results on the RTX54 research engine.

In the RTA-8T engines, this dilemma could be surmounted by the combination of state-of-the-art 'high-efficiency' turbochargers in combination with scavenging air inlet ports having a lower height thereby making it possible to achieve low fuel consumption figures without penalising the shipyards with exhaust gas temperatures that are too low (Figs. 8 and 9). As a result of the extended effective expansion stroke of the piston, the specific fuel consumption is reduced by some 3 g/kWh compared with the RTA-U engines using the standard turbochargers, and the exhaust gas temperature is only reduced by some 12 °C. This achievement was possible because of the highly effective Sulzer design of piston cooling and the bore-cooled combustion chamber (see Fig. 23). Owing to these features, it was not necessary to use a large air flow to overcome a less-effective cooling of the combustion chamber.

Today approved turbochargers for the RTA-8T engines are the ABB VTR-4E and VTR-4D types, and the Mitsubishi MET-SD and MET-SE types. In addition, MAN NA/S and NA/T9 turbochargers may be chosen for selected projects.

Additional reasons for the low fuel oil consumption of the RTA-8T engines are various. One point is the high mechanical efficiency of engines with low rotational speeds, another is that the large stroke-to-bore ratio of 4.0 to 4.17 allows a deep combustion chamber which gives more freedom for optimisation of the fuel spray pattern thereby resulting in an almost ideal rate of heat release, and another is the short connecting rod length (compared with the stroke) which also has a beneficial effect.

Moreover, part-load fuel consumption is minimised in the RTA series by the standard application of the load-dependent, variable fuel injection timing (VIT) which has been employed for many years in Sulzer low-speed engines (Fig. 10). In today's engines, however, the VIT system is actuated by a pneumatic positioning cylinder with electronic control from the engine control system, according to the DENIS interface specification. This arrangement gives very reliable and precise regulation of the proper load-dependent injection timing. Furthermore, the use of electronic control gives smaller actuating forces in the regulating linkages thereby leading to much improved lifetimes compared with the previous mechanical VIT system.

With the VIT, the best fuel consumption is achieved at approximately 85 per cent load, being 4 g/kWh lower than the full load fuel consumption for engines rated at MCR.

The RTA series engines are prepared for the prospective IMO regulation concerning $NO_X$ emissions to be implemented in the year 2000 (Fig. 11). To com-

Fig. 8
Comparison of fuel savings achieved by different supercharging philosophies. The use of lower air ports in the cylinder liners with high-efficiency turbochargers gives greater fuel savings than simply by applying high-efficiency turbochargers with standard ports
[98#105]

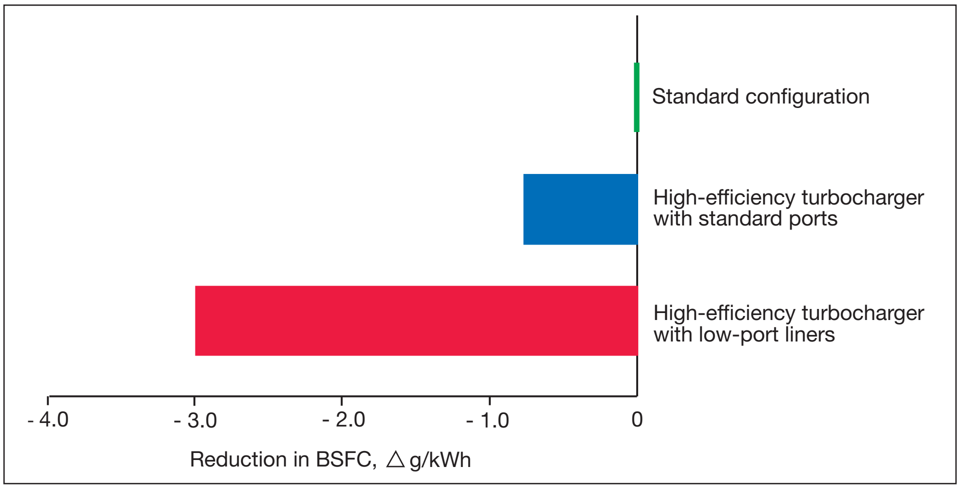



Fig. 9
The fuel savings shown in figure 8 for high-efficiency turbochargers in conjunction with lower air ports also afford higher exhaust gas temperatures than simply applying high-efficiency turbochargers alone. The exhaust valve temperations are no higher than in the standard solution
[98#103]

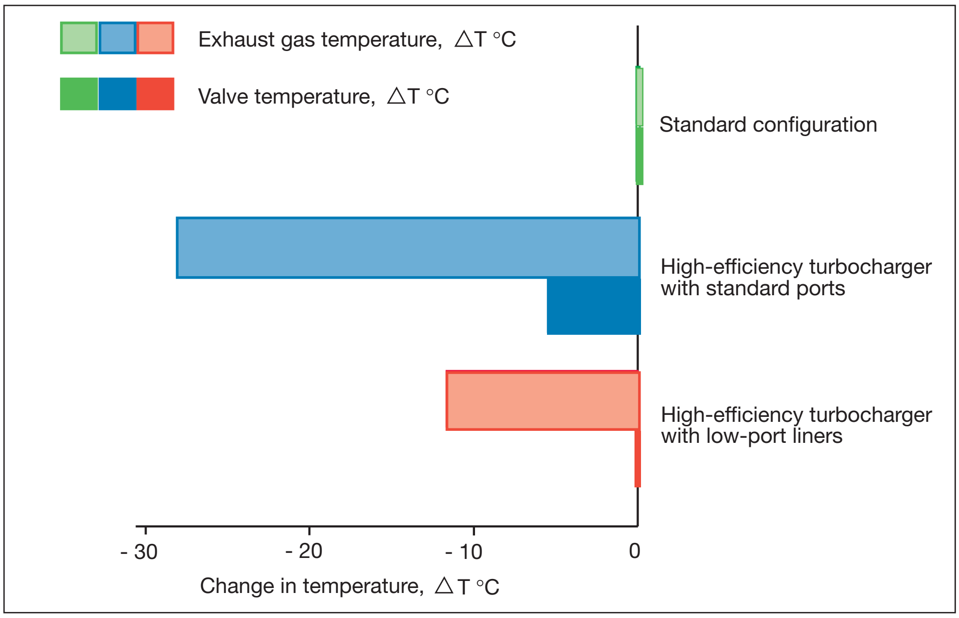
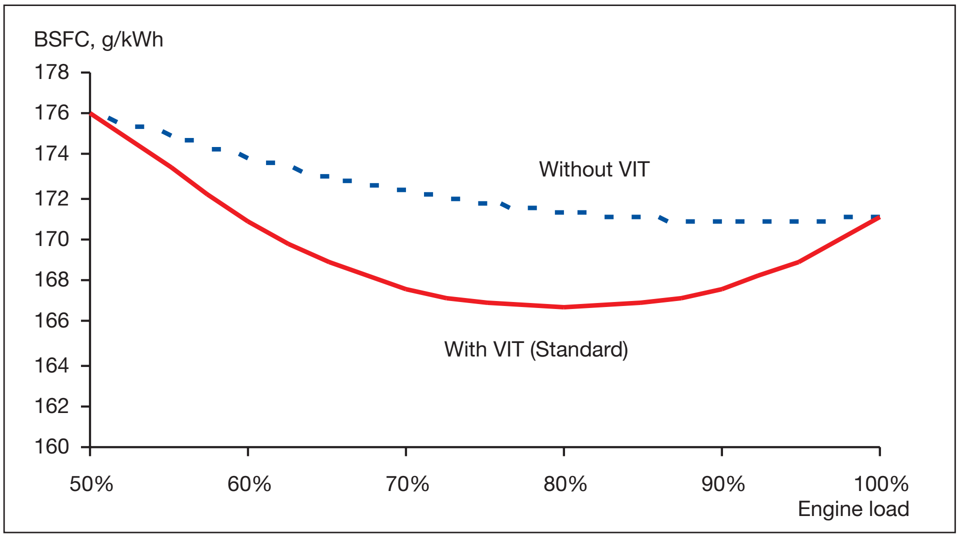



Fig. 10
Influence of variable fuel injection timing (VIT) on the part-load specific fuel consumption. The example shown is from the RTA48T at the rating R1
[98#106]



ply with the IMO regulation, the RTA-8T engines only require the application of certain engine tuning measures, as the $NO_X$ level of these engines is already very low.

The measures to be applied have been tested on several production engines and on the 4RTA58T prototype in the Diesel Technology Center. The selected measures – high compression ratio, adjusted VIT and adapted rate of fuel injection – allow three targets to be achieved: IMO compliance, conservation of high reliability, and low fuel consumption.

In addition, their DENIS 6 remote control specification incorporates provisions to run the engine in the so-called E3 mode, modifying the engine operation in a way such that both the $NO_X$ emissions measured according to the IMO E3 cycle and the fuel consumption rise are minimised.

Moreover, there is also integrated in the DENIS specifications, the possibility of electronically retarding the fuel injection temporarily by up to five degrees crank angle thereby reducing $NO_X$ emissions by some 20 to 25 per cent for use in certain situations, such as to comply with local emissions regulations when sailing in coastal areas.

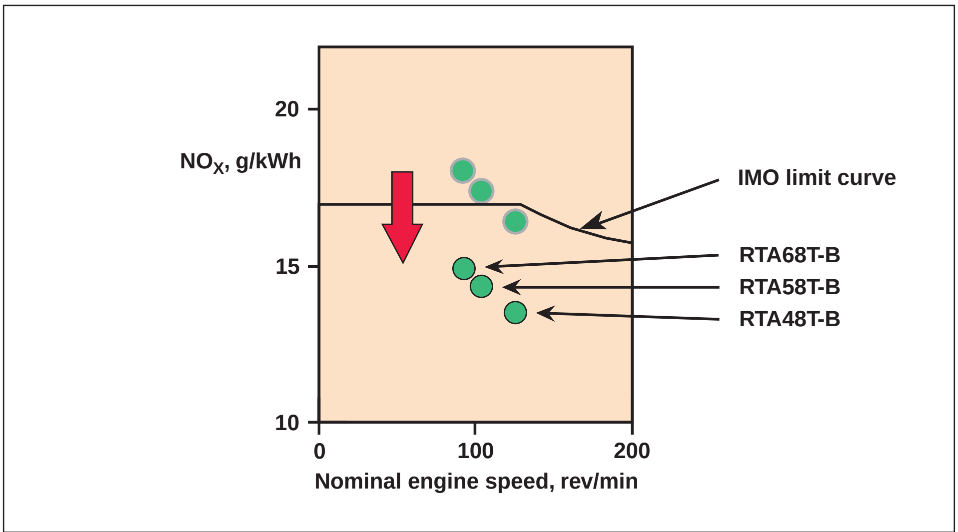



Fig. 11
Sulzer RTA-8T engines will comply with the prospective IMO regulations. As indicated here, the $NO_X$ emissions of RTA-8T Version B engines will be reduced to below the IMO limit by adaptation of tuning, particularly with the aid of the standard VIT system
[98#107]

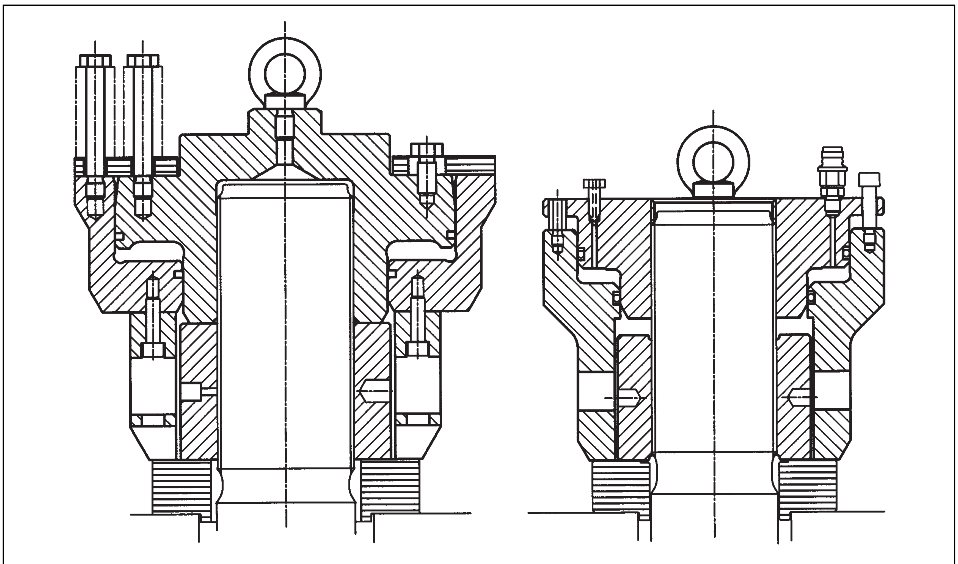


Fig. 12
Hydraulic jacks for the RTA58T (right) are much smaller than those for the RTA62U (left)
[97#242]

## Design optimisation

For the design of the RTA-8T engines, great effort was put into the engine design to optimise it for:

- Engine reliability;
- Easy manufacturing;
- Easy maintenance.

Many examples of this design optimisation are given below.

Beforehand, however, it can be mentioned that, apart from the compact engine dimensions, emphasis in the design was put on using the minimum number of parts which, where possible, are welded together, machining only where really required, standardising manufacturing processes (for example, using the same bore diameters wherever possible), using standard through bolts instead of bolts and threaded holes, round nuts instead of hexagonal nuts for main studs, etc.

The effort to achieve compact dimensions also resulted in lighter components which, together with lighter tools (Fig. 12), make for easier maintenance.

## Engine structure

The structure of the RTA-8T engines is similar in concept to those of the RTA-2 and RTA-U engines. For the development of the RTA-8T engines, stress and deformation calculations were carried out by using a full three-dimensional finite-element model (Fig. 13). For the Version B, the calculations were checked and, wherever necessary, the required measures were taken to reduce the mechanical loads down to the previous level.

The very compact engine dimensions could therefore be maintained. Much design work was especially necessary to keep the short cylinder distance because the crankshaft had to be slightly modified for the new power output (see below).

Once again it has to be underlined that, owing to the use of gear wheels for the camshaft drive, the thrust bearing can be very short and very stiff (Fig. 14). As gear wheels, unlike a chain, cannot snap, it is not necessary to provide a chain trap for a potential failure. It is therefore possible to install a closed and rigid thrust bearing housing, so that no ribs, always a potential source for cracks, are required. The closed thrust bearing also helps to reach a total engine length that is extremely competitive.

The bedplate and especially the columns are designed for minimum machining requirements. Owing to the higher position of the camshaft, the monoblock columns could be very much simplified so that inclined machining is no longer necessary. For the Version B, the lateral plates of the column and of the bedplate had to be slightly reinforced (by 1 mm).

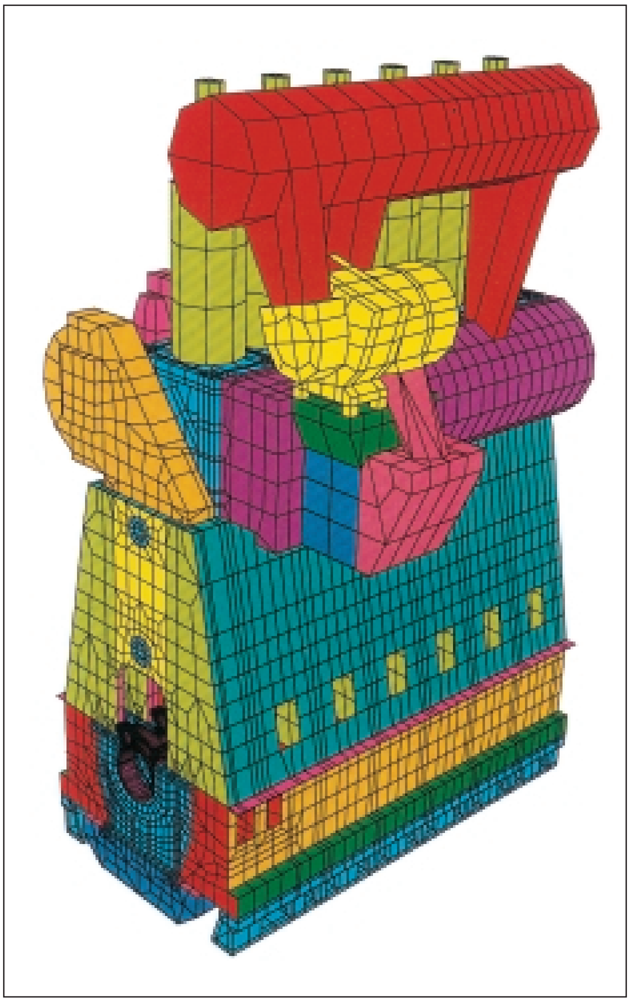


Fig. 13
Finite-element model of the six-cylinder RTA58T
[97#227]

## Crankshaft

The crankshafts of the RTA-8T engines have to cater for the longest stroke/bore ratio ever built. They thus required careful optimisation by three-dimensional finite-element analysis. Great effort was placed on the fillet areas and the shrink fits to cope with the significantly reduced cylinder distances compared with RTA-U engines. Although the stroke of the RTA58T is longer (2416 mm) than the RTA62U

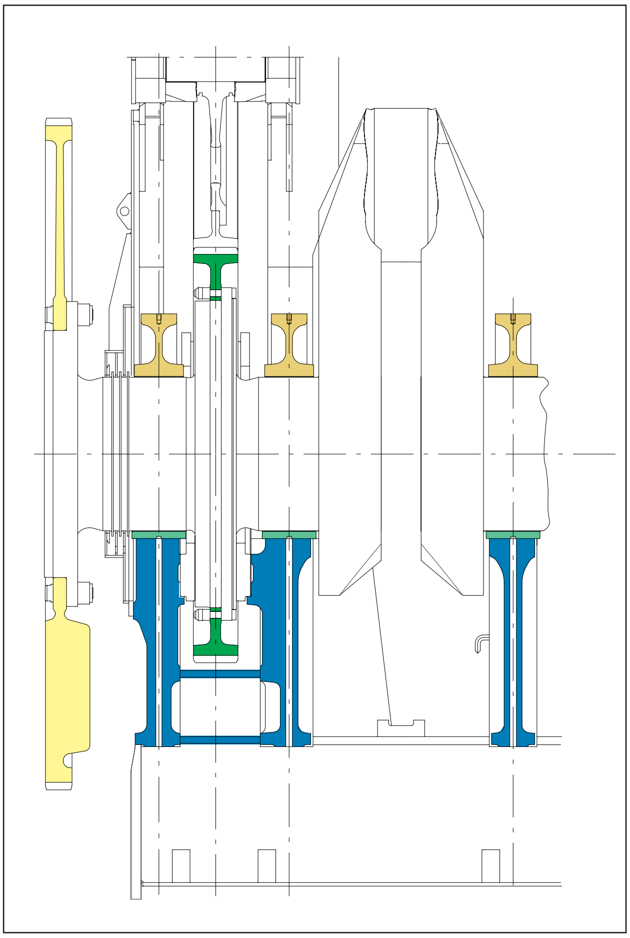


Fig. 14
Thrust bearing
[98#102]

(2150 mm), the cylinder distance could be reduced from 1100 mm in the RT62U to 1006 mm in the RTA58T.

To confirm the calculated stresses especially around the fillet area, intensive stress measurements were carried out on the 4RTA58T research engine in Winterthur as well as on the first 6RTA48T engine on the test bed at Diesel United Ltd in Aioi.

For the Version B, both the crank pin as well as the main journal were slightly increased in diameter to accommodate the increased torque and to reduce the bearing load down to the original level.

## Main, bottom-end and crosshead bearings

As a result of closely following up the first RTA-8T engines in service, small modifications had to be introduced for the main bearings and the crosshead guides:

- To increase the safety margins, the maximum tolerable bearing load capacity was increased by the application of shorter oil groves, thereby increasing the effective area of the bearing in the crucial region. Owing to the short connecting rods in today's two-stroke engines, the horizontal loads on main bearings are more significant than in the past. In addition, a better understanding of the horizontal loads is gained by today's finite-

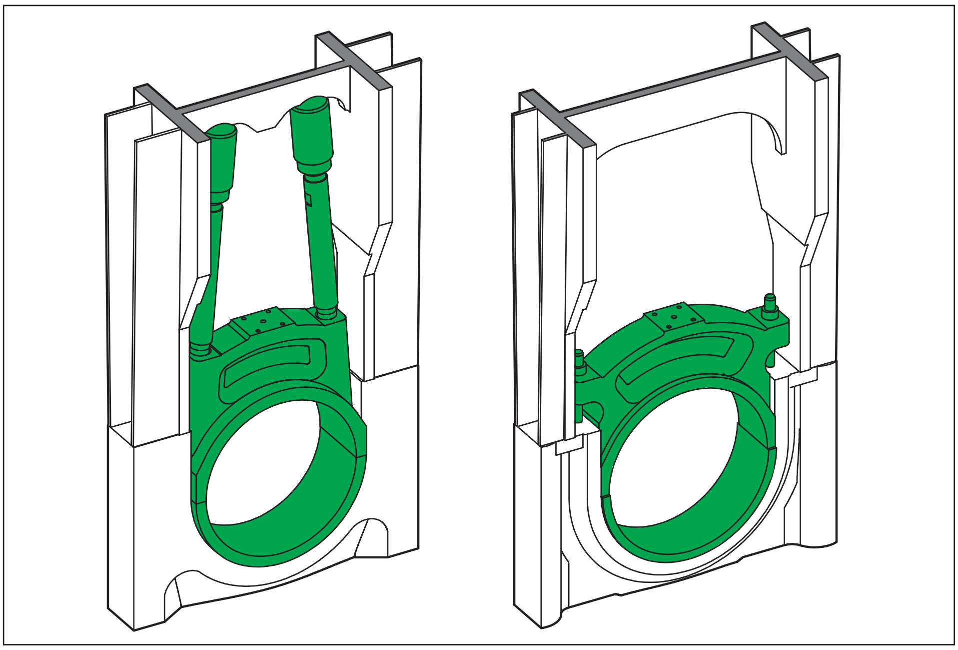


Fig. 15
Comparison of main bearings of RTA62U (left) and RTA58T (right) engines. There is a weight saving of 40 per cent with the RTA-8T design. The Version B engines have four holding-down bolts for each main bearing cap.
[97#228]

element analysis techniques that take into account the structure around the bearing and also vibration of the shaft (see below). Even if the horizontal load is still smaller than the vertical one, it might cause bearing problems if the bearing shells are not absolutely correctly aligned.

- Some failures occurred on the crosshead guide shoes after only a few thousand running hours. In today's compact engines, the crosshead guide shoes protrude from the guides at the top and bottom of the stroke, but mostly at the bottom where the guides are cut away to accommodate the crank webs at top dead centre. In addition, the ends of the guides originally had a small chamfer to prevent oil being scraped off the guide shoes. However, as it proved to be very difficult to manufacture the crosshead guides within the specified tolerances, the crosshead did not perform as expected. The crosshead guides thus had to be lengthened at the bottom to improve the support for the crosshead.

For manufacturing and maintenance reasons, the main bearing covers are each secured by four elastic holding-down bolts, instead of the pair of jack bolts used in earlier RTA designs (Fig. 15).

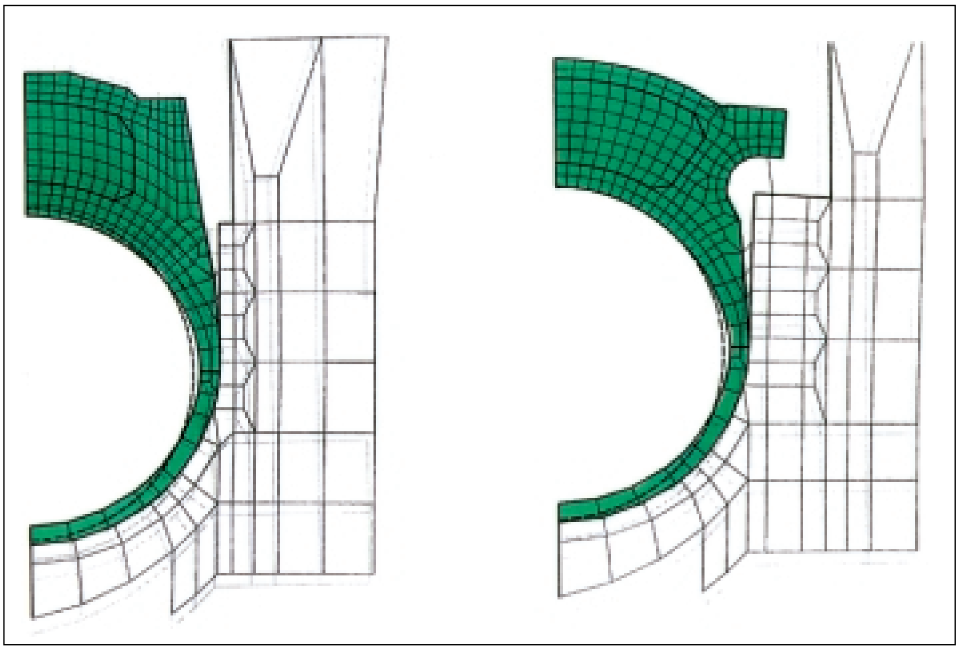
For the Version B, the diameters of the main and bottom end bearings had to be adjusted to the new diameters of the crankshaft pins and journals.

The main bearings have thin-shells of white metal as, unlike aluminium-tin, white metal offers a great safety and tolerance against misalignment, particles, seizure, etc., and moreover, it also has excellent emergency-running behaviour.

The crosshead bearings for RTA-8T engines are designed according to the same principles as used for all other RTA engines and also feature a full-width lower half bearing. The crosshead bearings have thin-walled shells of white metal for high load-bearing capacity. Sulzer low-speed engines retain the use of a separate high-pressure lubricating oil supply to the crosshead. It provides hydrostatic lubrication to give a higher bearing lift. This has proved crucial to long-term bearing security. The hydrostatic force created by oil pressure lifts the crosshead pin off the shell during each revolution to leave enough oil film thickness under the gas load.

Today's finite-element analysis techniques give very accurate calculated results for main bearing loading. They use a full model of the shaft and its bearings together with the corresponding structure (Fig. 16), and enable the boundary conditions to be taken into consideration by incorporating the crankshaft stiffness into the calculation of the bearing loading. The long strokes of the RTA-8T engines make it is necessary to take the crankshaft stiffness into consideration when calculating the main bearing loading (Fig. 17). These calculations are important

Fig. 16
Finite-element model of main bearings of the RTA58T (right) and RTA62U (left). The dotted lines show deformation from the static forces
[97#229]



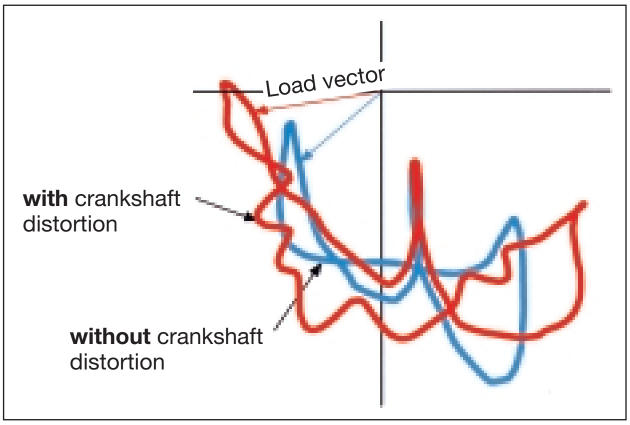
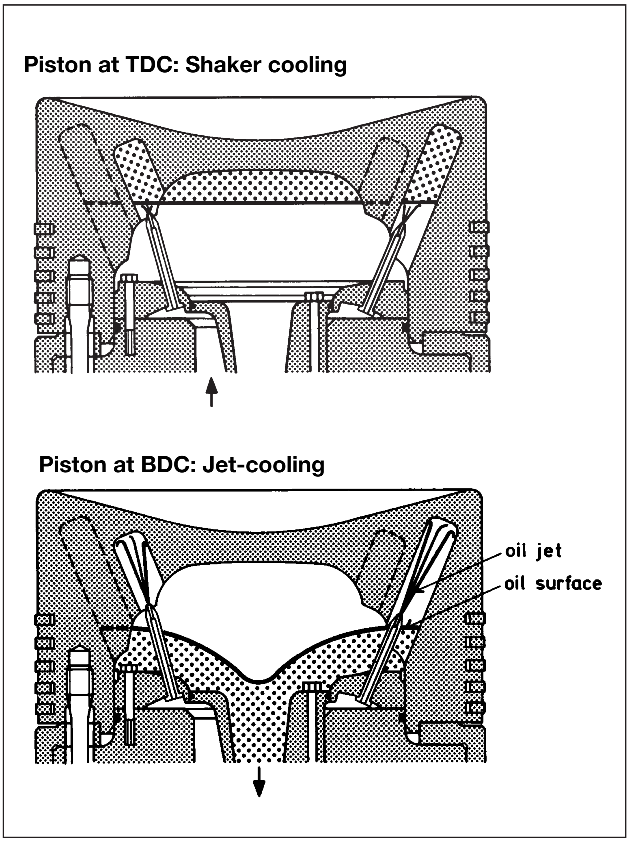


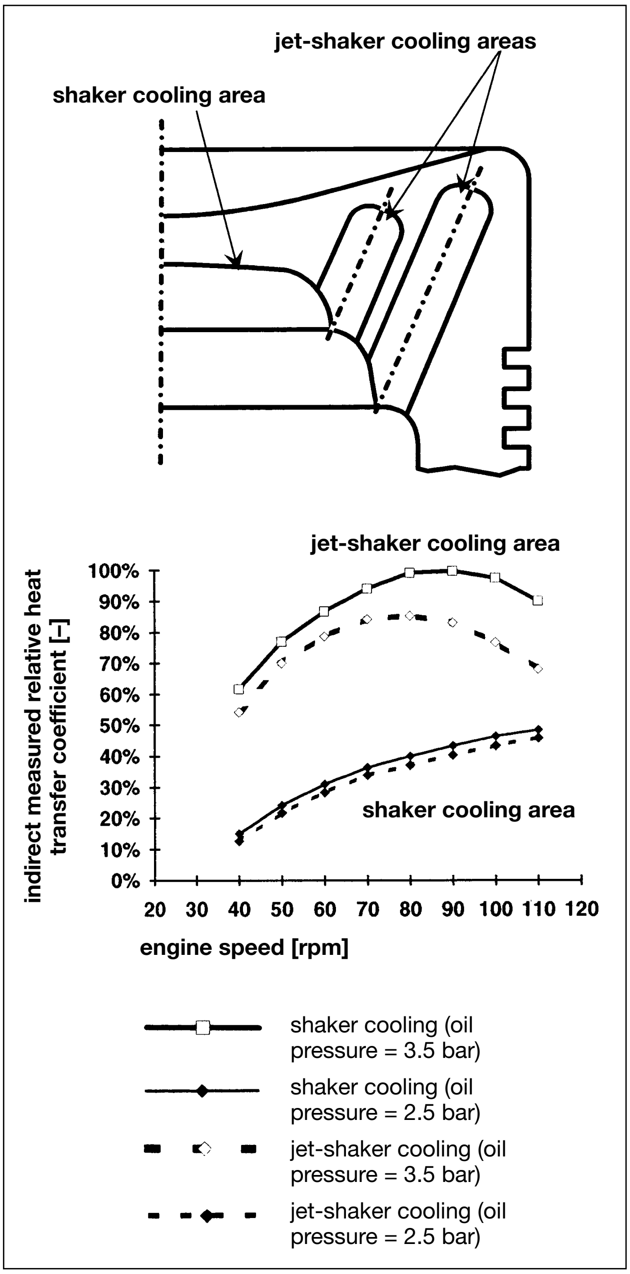

Fig. 17
Load diagrams for the main bearing with and without crankshaft distortion taken into consideration during the calculations
[97#232]

Fig. 18
Operating principles of jet-shaker cooling of the piston (drawing based on RTA-U engine). At top dead centre (upper diagram), shaker cooling is predominant, while jet cooling applies more at bottom dead centre (lower diagram). The jet cooling has the advantages of increased effectiveness and the prevention of carbon deposits on the oil side
[0789-3039-1/-2]




because the load vector can vary considerably depending on the crank angle between two neighbouring cylinders but mainly on the firing order sequence.

Fig. 19
Relative heat transfer coefficient measured from an oil-cooled piston crown based on an RTA-U engine at different oil pressures, and in the shaker and the jet-shaker cooling areas
[95#113-1/-2]



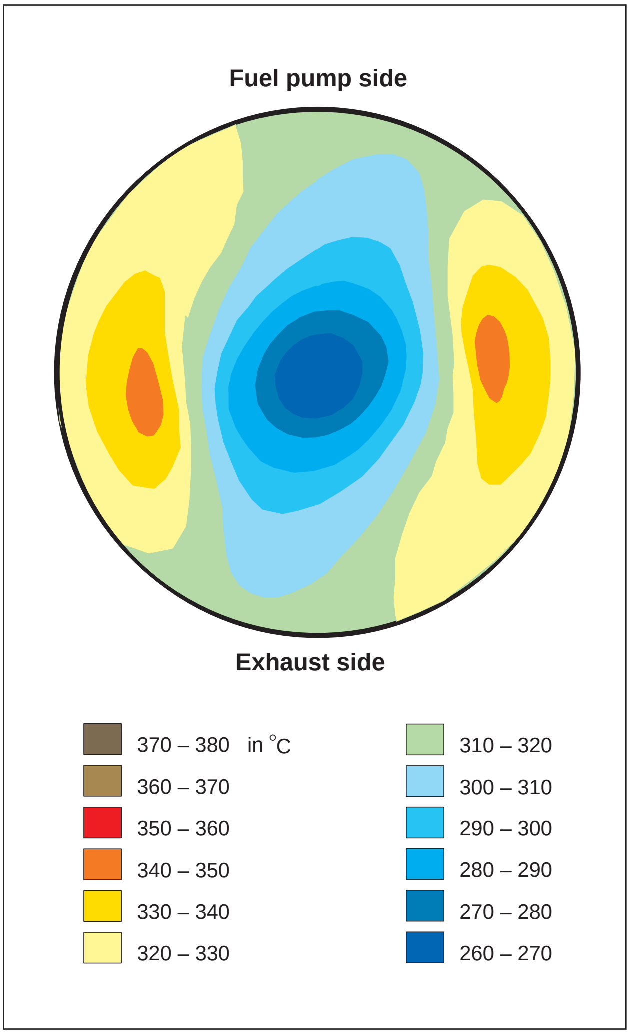



Fig. 20
Piston crown surface temperatures measured on the first 6RTA48T at Diesel United running at R1 full load, 8160 kW output at 124 rev/min
[98#108]

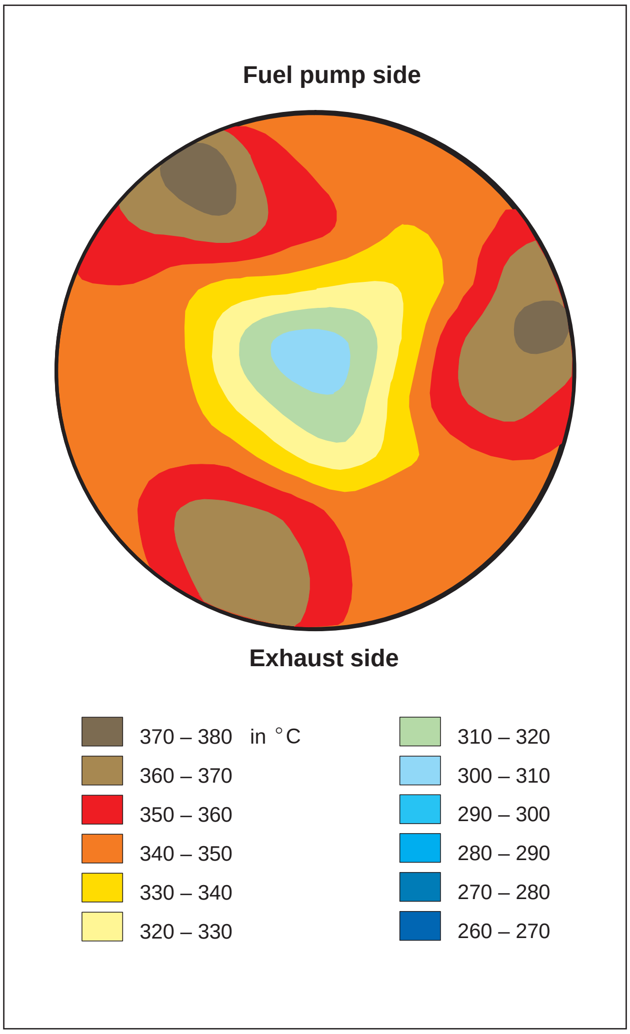



Fig. 21
Piston crown surface temperatures measured on the 4RTA58T at the Diesel Technology Center running on diesel oil at full load, 8000 kW output at 103 rev/min
[98#109]

## Piston

The piston design of the RTA-8T engines is similar to that of the RTA-U engines and no modifications had to be introduced for the Version B. The jet-shaker oil cooling principle provides optimum cooling performance (Figs. 18 and 19). It gives very moderate temperatures on the piston crown and no coatings are necessary (Figs. 20, 21 and 22).

Four piston rings were chosen owing to the good RTA-U service experience. The top rings are coated with a thin plasma layer.

## Combustion chamber

In the combustion chamber components, the well-proven bore-cooling principle is used to control temperatures, as well as thermal strains and mechanical stresses (Fig. 23). No modifications had to be introduced for the Version B.

The combustion chamber in today's diesel engine is recognised as the most important design area because of its major influence on the reliability of the engine. However, for the RTA-8T engines with their large stroke to bore ratio of 4 and above, this fact is less important owing to the deep combustion chamber which gives more freedom in the layout of the fuel injection spray pattern. Nevertheless, in engines with large stroke to bore ratios, a relatively large amount of fuel is injected per stroke and thus emphasis still had to be put on optimising the fuel injection. This was carried out first by the use of modern calculation tools as for example CFD (computerised fluid dynamics) analysis. Second, the calculated results had to be confirmed on the first test engines which, in fact, proved to be highly successful. As a result, very moderate surface temperatures could be achieved in the combustion-space components.

The cylinder cover is also bore cooled. It is secured by eight elastic studs. Three fuel injection valves are basically standard for Sulzer low-speed engines. However, it could be confirmed that for the smaller bore engines (less than 580 mm bore) at today's mean effective pressures the results achieved with two injection valves are still satisfactory. Therefore, on the RTA48T, two injectors are being used and three on the RTA58T and RTA68T. The fuel injection valves are symmetrically distributed in the cylinder cover. Anti-corrosion cladding is applied to the cylinder covers downstream of the injection nozzles to protect the cylinder covers from hot corrosive or erosive attack.

Fig. 22
Piston crown surface temperatures measured on the 4RTA58T at the Diesel Technology Center running on heavy fuel oil at full load, 8000 kW output at 103 rev/min
[98#110]

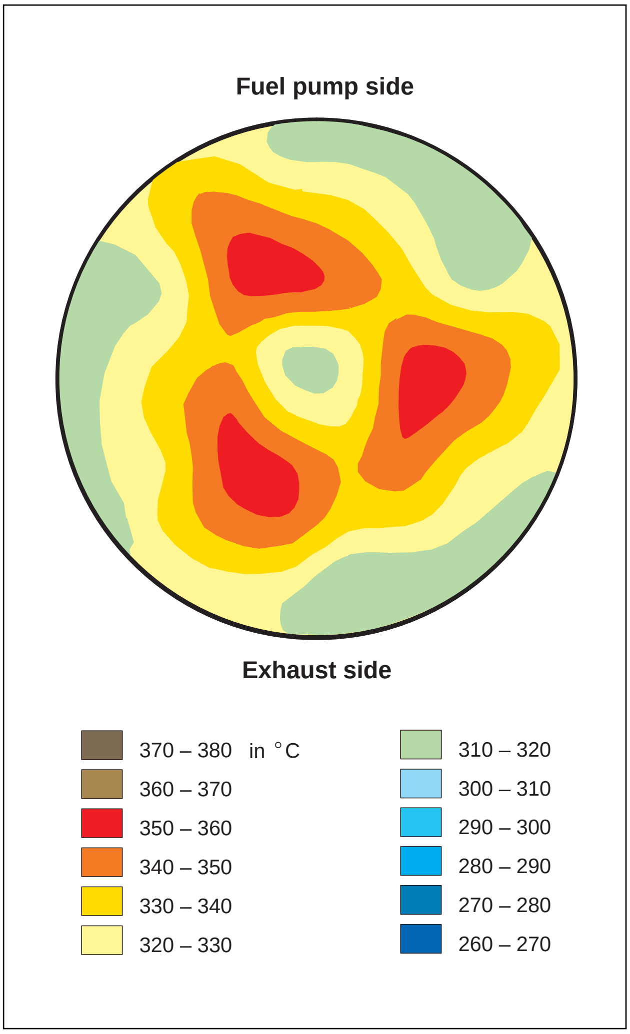
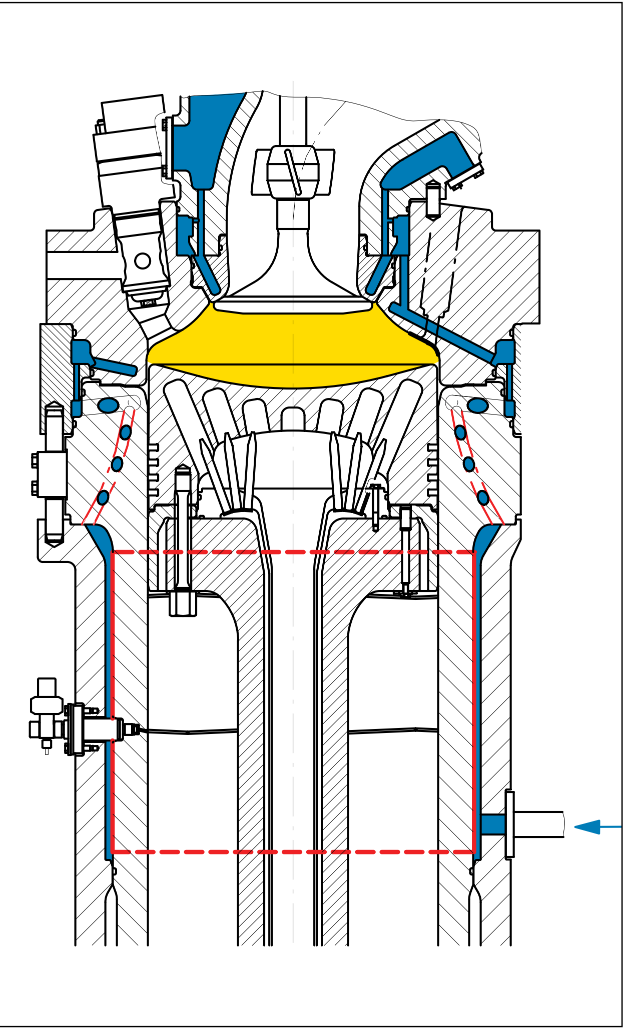



Fig. 23
The combustion chamber of the RTA-8T engines are fully bore cooled. The red dashed lines across the cylinder mark the extent of the mid-stroke insulation applied to the outside of the liner
[98#101]

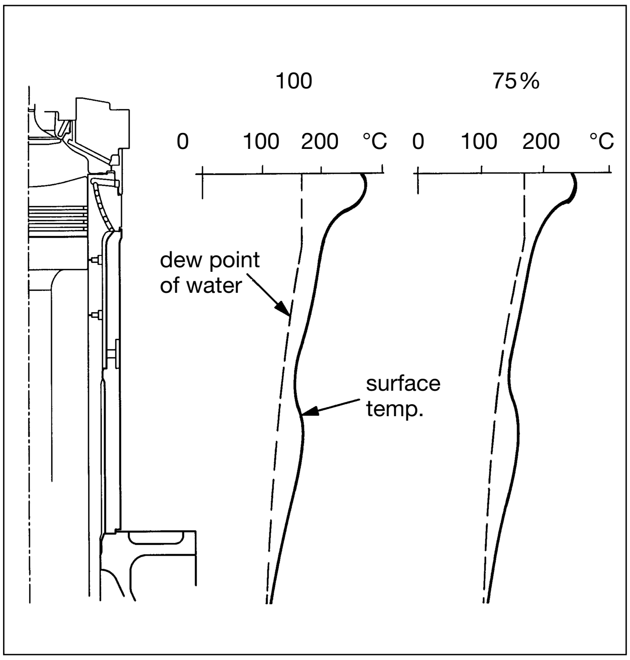


The cylinder liner is bore cooled as well. Great effort was made on the layout of the surface temperatures of the liners (Fig. 24) and on the stress and strain calculations.

Service experience of the RTA84T engines has shown that it is essential to obtain correct surface temperatures over the whole liner stroke in order to have satisfactory piston ring and liner wear rates. Therefore a solution was chosen for the RTA-8T engines with a rather high position for the sealing of the water space (Fig. 23).

All the service experience of the RTA84T as well as from RTA-U engines was taken into consideration when developing the combustion chamber components of the RTA-8T engines to reach the goal of two years' operation between major overhauls. The efforts were then confirmed by the first inspections made on the first RTA48T whose combustion chamber components were found to have no loss of material and with only very little amounts of carbon deposits (see section 'Test results and service experience' below).

Fig. 24
Calculated cylinder liner surface temperatures for the RTA58T at 100 per cent and 75 per cent load
[97#121]



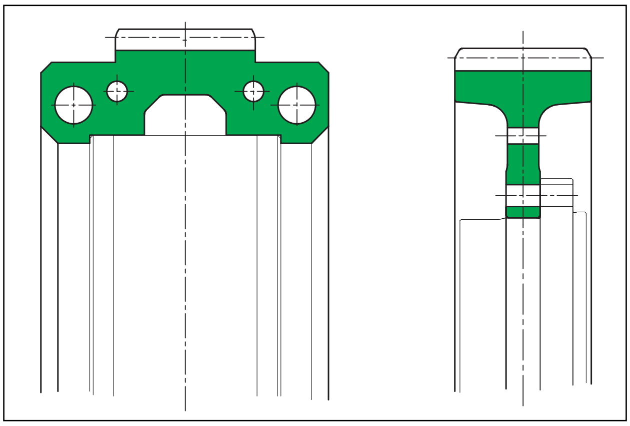



Fig. 25
Simplified one-part flange type gear wheel for the RTA-8T engines (right) compared with that for the RTA-2 engines (left)
[98#121]

## Camshaft and camshaft drive

The camshaft drive uses the well-proven Sulzer gear arrangement. A gear drive allows the shortest possible engines. A moment's thought, however, will confirm that a gear wheel drive for the camshaft is technically a much more elegant solution than a chain drive. Moreover, unlike a chain drive with its thousands of moving parts, a gear drive cannot lose tension over the years of operation, so that readjustment of the exhaust valve and fuel injection timing is not required. Gears are not sensitive to axial vibration and, as gears do not tear to pieces, it is not necessary to foresee a chain trap for a potential failure, enabling the design of a very stiff thrust bearing. Additionally, the relevant parts of a chain drive are not usually visible, and would thus be a great risk in case of a damage.

The camshaft itself of the RTA-8T engines is located at the top of the cylinder jacket, which significantly reduces the pressure losses between the fuel injection pumps and the injection nozzles. Owing to this, the cost of an additional intermediate wheel could be compensated in various ways, by using a smaller camshaft diameter, omitting inclined machining of the columns, etc. Moreover, through bolts are used on the gear rims instead of fitted bolts, and especially a much simplified single-piece flange-type gear wheel is fitted on the crankshaft (Fig. 25). For replacement, a flange-type two-part wheel was also developed.

The camshaft gear wheel with its plain bearings is assembled in a welded bearing housing and it can easily be aligned by adjustable bearing supports.

The camshaft and reversing servomotor are of similar design to those in the RTA-U engines but are more compact and smaller in dimension. No modifications had to be introduced for the Version B. For cost reasons the cams are simplified without side collars and with straight radial bores for the oil connections of the hydraulic tools.

The flange-type couplings without sleeves are now used as standard, in parallel with the option of the normally more expensive SKF sleeve coupling (Fig. 26).

## Fuel injection pump

The fuel injection pump is of the well-proven double-valve controlled type. No modifications had to be made to the fuel injection equipment for the Version B, apart from the adjustments for the larger amount of fuel to be injected. These include an enlarged plunger diameter and enlarged bores at the fuel injection nozzles. Additionally, the cam length of the RTA58T Version B was extended by some five degrees crank angle to give more freedom in optimising for low $NO_X$ emissions.

In a double-valve controlled injection pump – *nomen est omen* – the fuel injection timing is controlled by valves regulated through eccentrics on hydraulically-actuated lay shafts. Consequently, there is great flexibility in selecting the timing without any special adaptation. In today's engines, this flexibility is employed in the variable fuel injection timing (VIT) system for improved part-load fuel consumption, and for the fuel quality setting (FQS) lever to adjust the injection timing according to the fuel oil quality.

However, even greater variation is conceivable, and would offer a competitive advantage in low-$NO_X$ technology. Electronic control of the pump control valves by solenoids would theoretically even be feasible but this is not required today. In a helix-controlled fuel injection pump, variable injecting timing can be obtained by an appropriate shape of the helix. However, a change of the selected timing with load is only possible by changing the plunger. Other ways to achieve flexible timing in a helix-controlled fuel pump would be possible but tend to be highly expensive and difficult to achieve. Moreover, adjustment for fuel injection quality has to be made by shifting the cams whereas, in a valve-controlled injection pump, timing adjustments can easily be made by operating the fuel quality setting (FQS) lever.

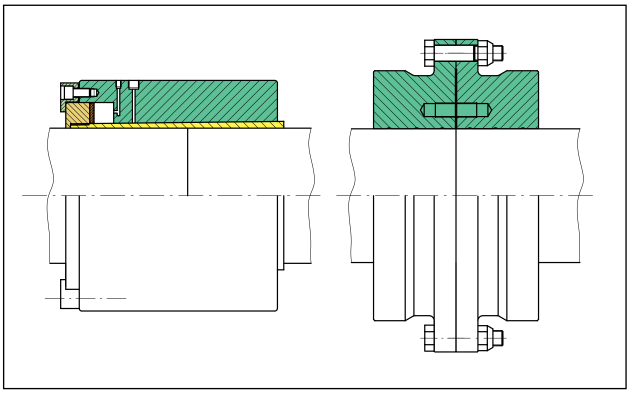


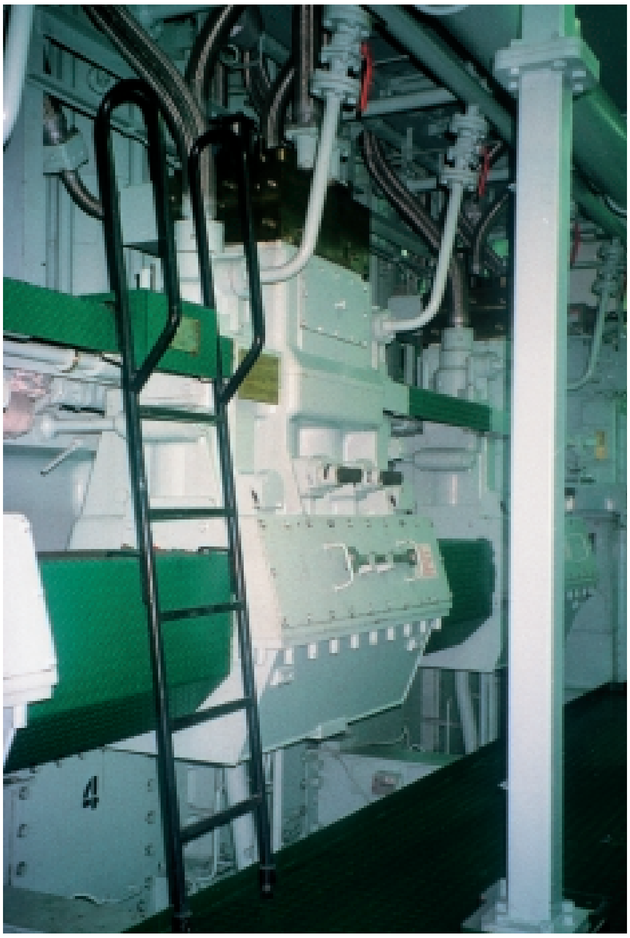
Fig. 26
Flange-type couplings on the camshaft (right) are now standard instead of the sleeve coupling (left)
[97#233]

As the timing in the valve-controlled fuel injection pump is not controlled by a helix, the plunger has a significantly greater sealing length, thereby resulting in a higher volumetric efficiency. Owing to this, the torque in the camshaft is some 15 per cent lower than in a helix-controlled pump, for the same fuel injection pressure. Additionally, fuel injection from a valve-controlled pump is far more stable at very low loads so that no special measures are necessary to operate the engine at low loads. With an electronic governor, rotational shaft speeds down to 15 per cent of the rated speed are be achieved. This was confirmed recently on the first 6RTA48T where the engine could be run steadily at 20 rev/min which, according to the propeller characteristic, corresponds to about 0.5 per cent engine power! This can hardly be achieved by any type of helix-controlled injection pump. Moreover, unlike the helix, the control valves do not wear over the years so that injection timing with the valve-controlled pump does not change over the years.

For the operator, one major aspect of fuel injection pumps is their reliability. Valve-controlled injection pumps are fully cavitation free. In helix-controlled pumps, cavitation has always been a problem and will remain a problem in the future, according to major fuel injection equipment manufacturers.

The housing of the injection pumps incorporates the fuel injection and exhaust actuator pumps for two cylinders and is directly bolted to the cylinder jacket. In the RTA-8T engines, the pump housing is significantly simpler with two camshaft bearing halves bolted from beneath (Fig. 27). The bottom of the housing is sealed with a welded casing.

Additional improvements for easy manufacturing compared with the RTA-U engines are the smaller size and weight of the fuel pump block, with smaller



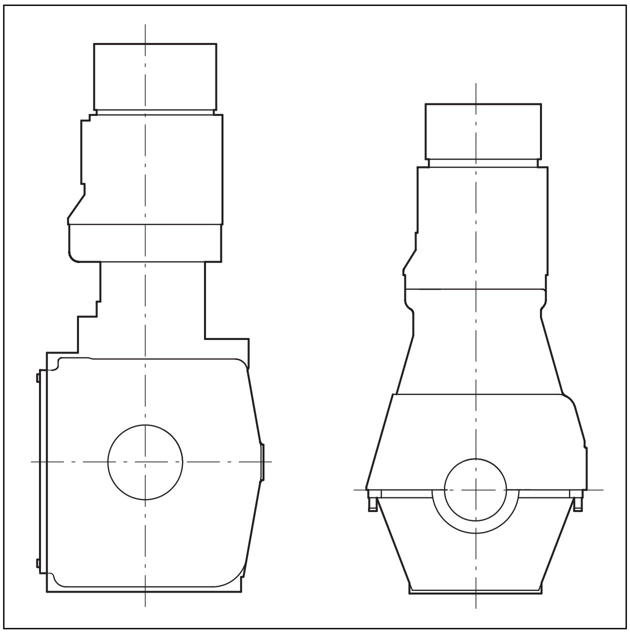
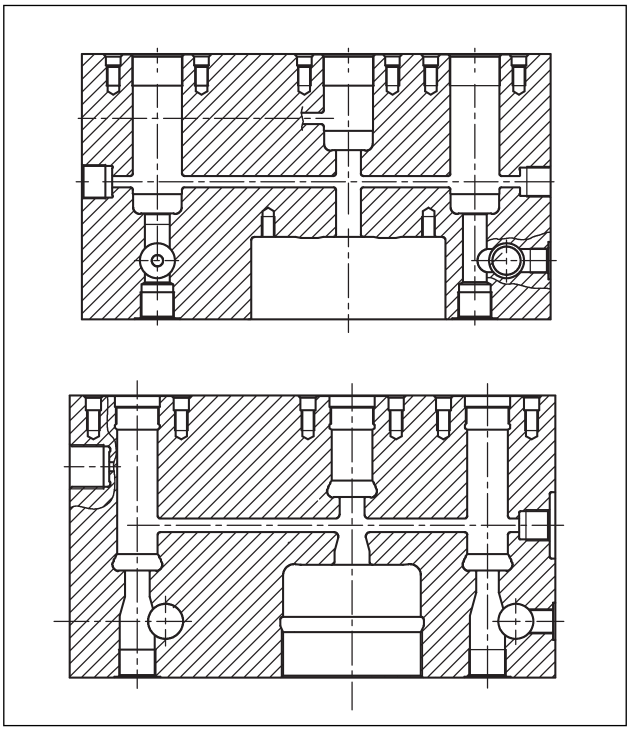


Fig. 27
Fuel/actuator pump housing for the RTA-8T engines (above right) compared with that for the RTA-2 engines (above left). The pump housing (left) was photographed on an RTA58T engine
[97#248]

Fig. 28
Fuel pump blocks for the RTA-8T engines (below) compared with that for the RTA-2 engines (bottom)
[97#247]



suction and spill valves, omission of recesses, same bore diameters wherever possible, etc. (Fig. 28). The new design has thereby significantly reduced the sensitivity of the housing to poor machining, and further increased the margin against cracks in the housing.

The camshaft segments can easily be dismantled downwards after dismantling the bottom casing and the lower bearing halves.

## Cylinder lubrication system

The RTA-8T engines are equipped with the well-proven Sulzer multi-level accumulator system. As in other more modern RTA-series engines, the cylinder lubricating pumps are driven by a frequency-controlled electric motor. However, the lubricating pumps are of a new design and are bolted together to form a compact module.

The cylinder lubricating system of a six-cylinder engine, for example, consists of an electric motor, a small step-up gear and three lubricating oil pumps including sight glasses. This module can be readily

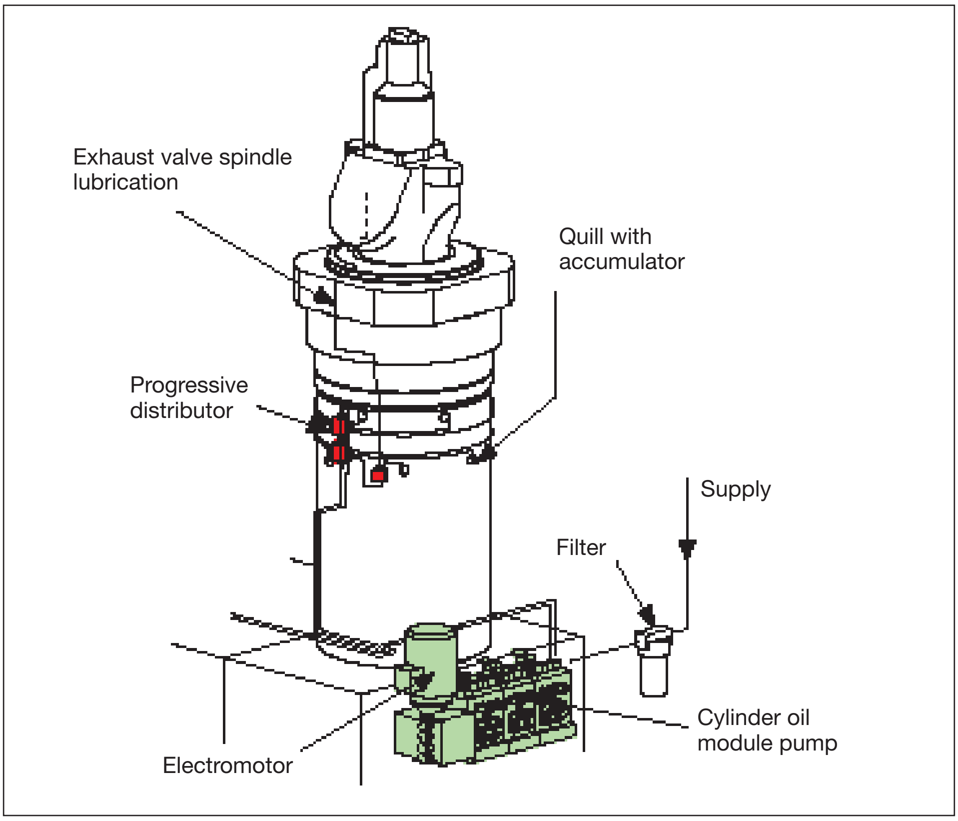



Fig. 29
Cylinder lubricating system
[98#120]

pre-assembled for easy mounting on the engine (Fig. 29).

One lubricating pump serves two cylinders. It has only two outlet pipes per cylinder, with one each for the upper and lower lubricating levels. On the cylinder liner, the oil is distributed to the different oil accumulators by oil distributors. With this solution, much less piping is required. The lubricating oil feed rate is controlled according to the engine load. Adjustments depending on the engine condition can easily be made by using software in the DENIS-based engine control system.

This cylinder lubricating system has been running with very satisfactory results on the 4RTA58T research engine in Winterthur for about 1000 hours, as well as the first RTA48T and RTA58T engines that have entered service.

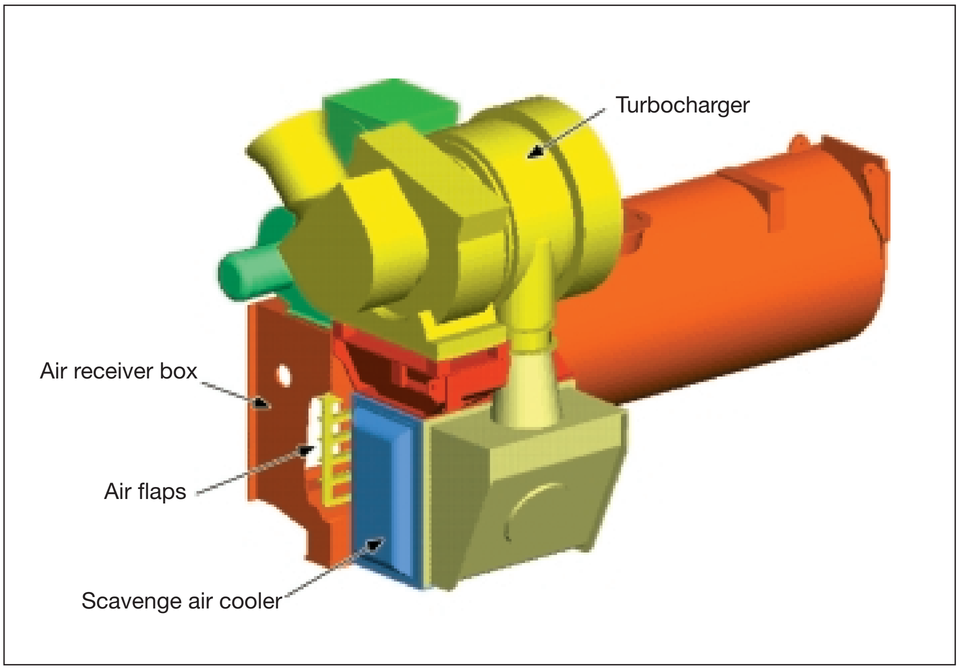
## Scavenge air system

The scavenge air system of the RTA-8T engines is based on that of the RTA84T-B design. By using an inclined exhaust gas outlet from the valve cage and a tangential inlet to the exhaust manifold a positive (clockwise) gas swirl is created. In contrast to the previous design with horizontal exhaust gas outlet, this inclined arrangement gives more freedom for the arrangement of the turbochargers alongside the engine. The position of the outlet of the exhaust gas receiver can freely be selected because it does not intersect with the inlet. The created swirl in the exhaust gas receiver follows the Sulzer philosophy in thermodynamics, to conserve the velocity of gases wherever possible and not to convert the speed to pressure before re-converting the pressure again to speed, as is normally done by diffusers in competitors' engines.

The scavenge air receiver consists of a so-called T/C module and a very simple half-round manifold. The T/C module incorporates the turbocharger, its support, the air cooler, the water separator, the air flaps and the fixed foot for the exhaust manifold including the auxiliary blowers (Fig. 30).

To further optimise the manufacturing process for the RTA-8T Version B engines, the scavenging air inlet receiver, the turbocharger support, the air duct and the scavenging air cooler housing are now welded in one piece. This principle was first applied on a RTA72U-B engine built at Hanjung, and it significantly reduced the time required for assembly.

Fig. 30
Scavenge air system with turbocharger/cooler (T/C) module and half-round receiver
[97#235]




## Water separator

For the RTA-8T engines, as also for all other Sulzer RTA-series engines, the water separators are made in polypropylene. No modifications had to be introduced for the Version B.

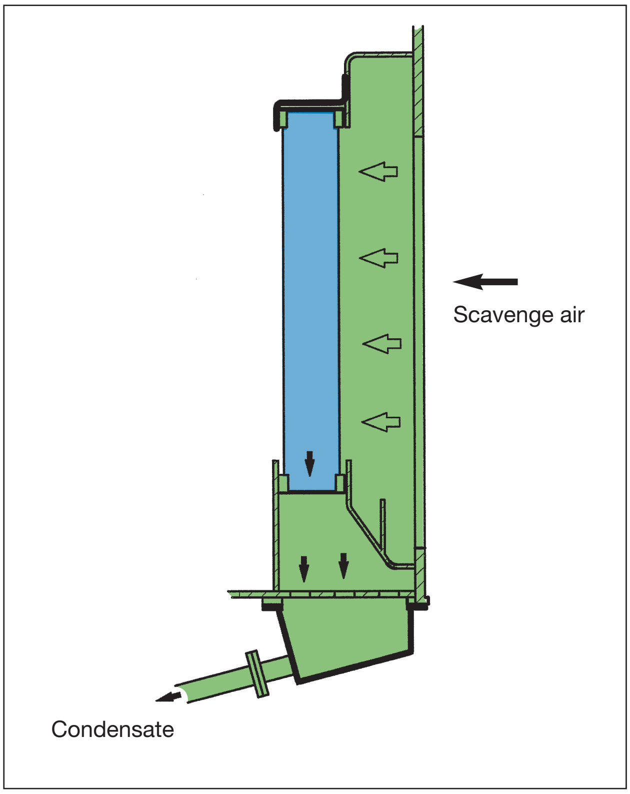
The water separators are located at a sufficiently high position to leave enough room beneath for the drainage arrangements for the condensed water that is separated from the scavenging air after the coolers (Fig. 31).

Effective separation of condensed water, as it is achieved by the selected design, is imperative for proper piston-running behaviour. Wärtsilä NSD Switzerland has been co-operating for many years with major suppliers of water separators to create an optimum design of both the separators and their surroundings with optimum airflow (appropriate speed, lamellosity, etc.).

## Installation simplifications

During the development of the RTA-8T engines, emphasis was given to simplifying the installation in general and to reduce the time for installing these engines in the ship, which is finally a cost saving for the yard.

Fig. 31
Water separator arrangement
[97#246]



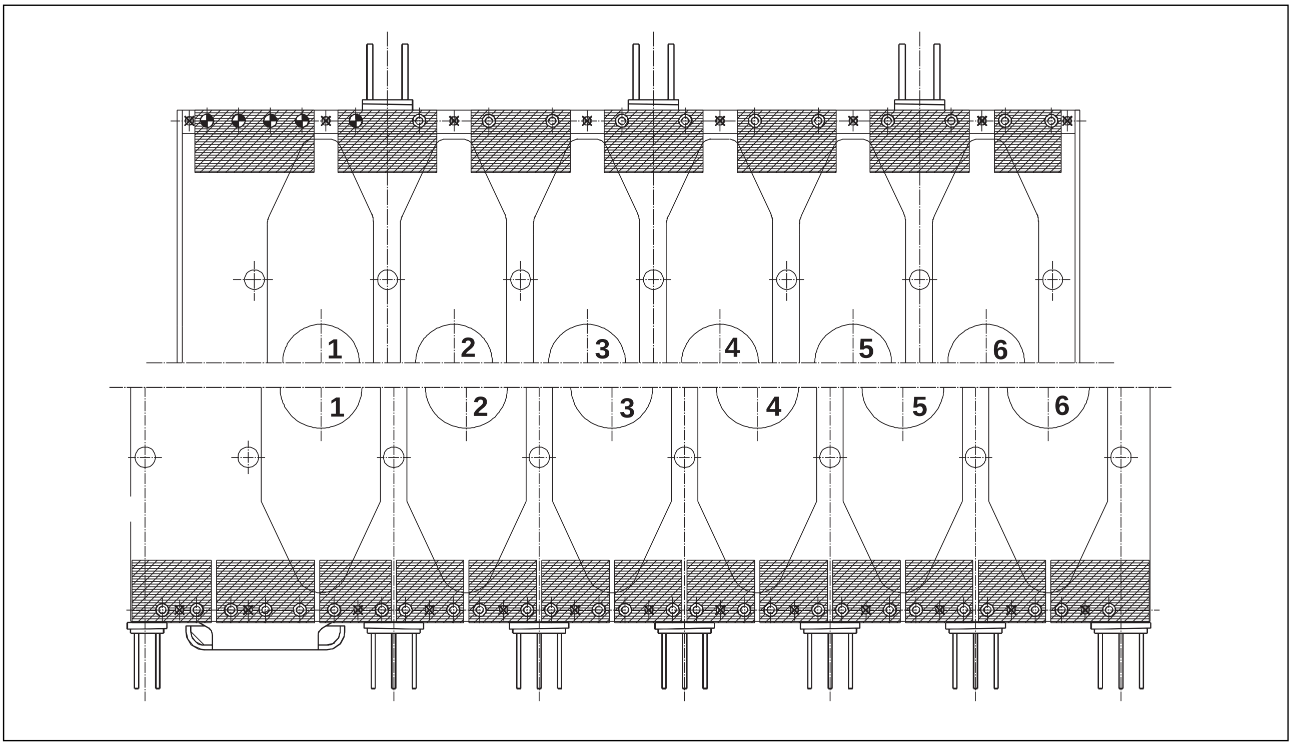
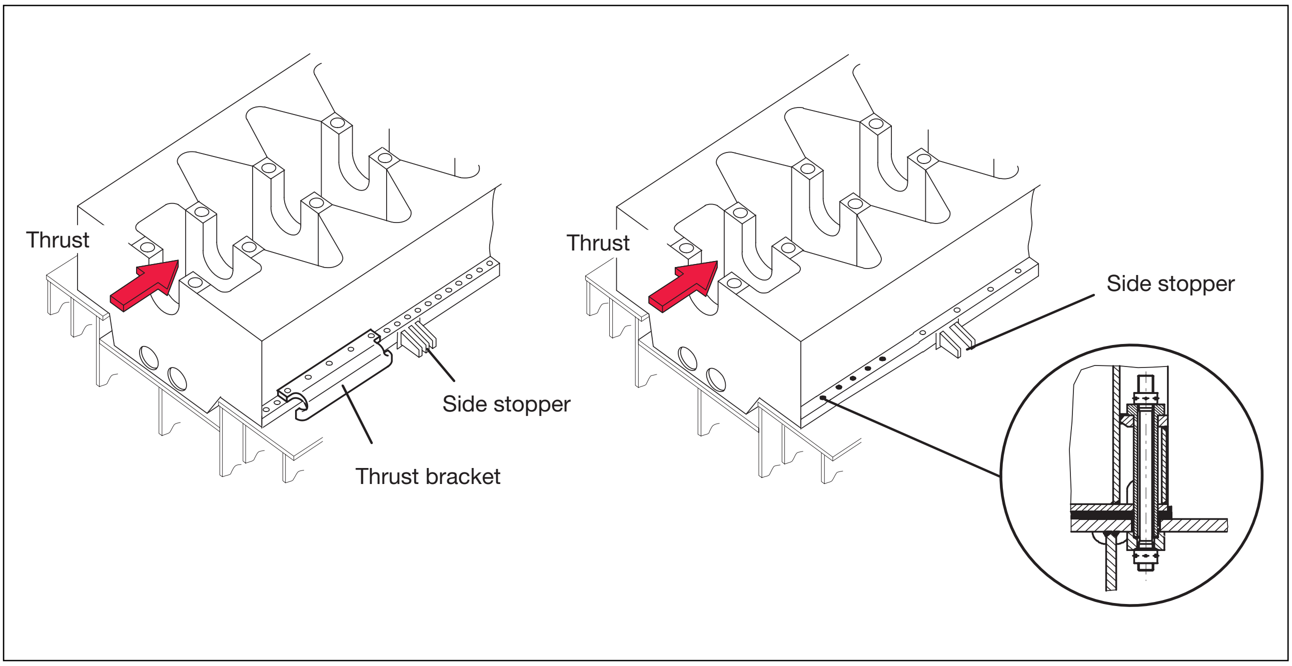



Fig. 32
Engine seating plan for the RTA-8T engines (top) and previous designs (above)
[97#091]

Fig. 33
Arrangements for transmitting propeller thrust to the engine seatings, for the RTA-8T engines (right) and previous designs (left).
The thrust sleeve for the thrust bolts in the RTA-8T design is illustrated in figure 34
[97#085]



One important point which has been confirmed by shipyards, is the good access to all foundation bolts. The number of foundation bolts could reduced by approximately 40 per cent compared with the RTA-U engines which also helps to save time during engine erection (Fig. 32). This advantage could also be retained in the RTA-8T Version B engines.

Furthermore, the machining underneath the tank top plate (spot face) is no longer needed. The number of side stoppers could be reduced by 50 per cent and the side stoppers themselves are simplified compared with the RTA-U engines. For example, on a 6RTA58T, only six stoppers are necessary instead of the twelve used with previous engine types (Fig. 33). A new solution was also developed to take up the propeller thrust. To eliminate thrust brackets welded on the tank top plate and attached by fitted bolts to the engine, a so-called thrust sleeve solution (patent pending) together with resin chocking was designed which does not require any welding or machining work after the engine is aligned. All holes can be drilled, or even burned out on the tank top plate with reasonable tolerances. This thrust-sleeve solution is now approved by all relevant Classification Societies.

After alignment of the bedplate, the thrust sleeves are simply put into the bores and the usual resin chocking material is poured in to place (Fig. 34).

Additionally, the number of pipe connections on the engine that have to be made by the yard could be significantly reduced and the drain tank arrangement is notable simplified. The oil drains from the engine to the oil drain tank have been modified for easier installation.

All ancillaries, with respect to their capacities, arrangement, etc., have been thoroughly optimised to reduce the installation cost.

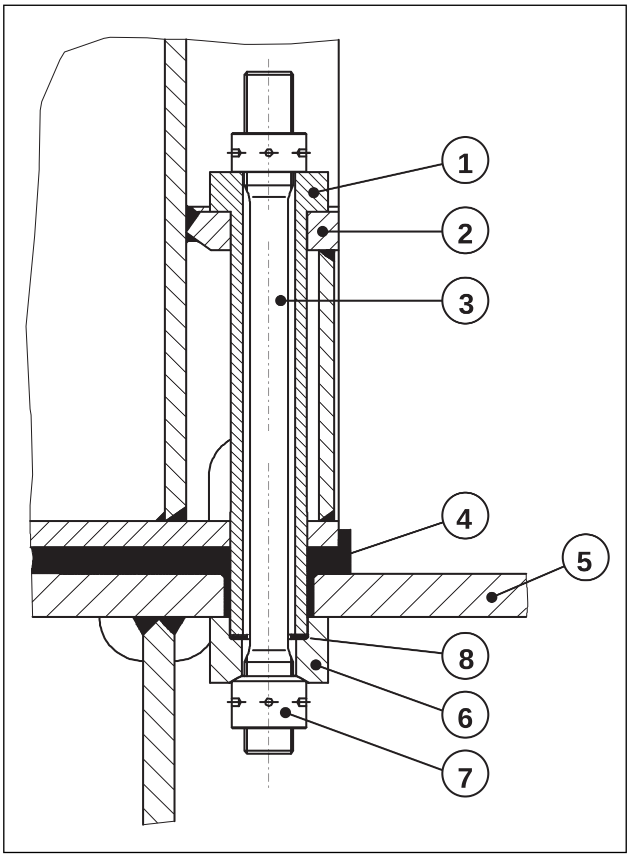



Fig. 34
Thrust sleeve with resin chock
1. Thrust sleeve
2. Engine foot
3. Holding-down stud
4. Epoxy resin chock
5. Tank top plate
6. Washer
7. Nut
8. Dam
[97#084]

# Vibration aspects

**Key points**
Vibration analysis of marine installations with RTA-8T engines involves:
- Torsional vibration
- Axial vibration
- Engine and ship vibrations

A good vibration analysis is a must to achieve an optimum plant arrangement.
The measuring campaigns have shown that, today, vibration analysis is well under control and that measurements are in good accordance with calculations.
Wärtsilä NSD Switzerland offers assistance regarding vibration calculations and measurements to all licensees. Moreover, close team work with shipyards and other designers allows the best plant arrangement to be found for each particular project.

As a leading designer and licensor of large diesel engines, Wärtsilä NSD Switzerland is concerned that satisfactory vibration levels are obtained with our engine installations. The assessment and reduction of vibration are subject to continuing research and we have developed extensive computer software, analytical procedures and measuring equipment to deal with the subject.

The dynamic analysis of marine installations propelled by an RTA-8T engine usually involves the steady-state operation. In this mode, the engine with the shaft line runs at constant speed and load. Only small perturbations act on the installation. They are of periodic nature and originate in the diesel engine and the propeller. The dynamic behaviour of the installation can thus be described by vibrations of various types. The vibration characteristics can then be analysed in two steps: for free vibration (determination of natural frequencies and critical speeds) and forced vibrations (level of vibration arising from the engine and propeller excitation).

Dynamic analysis of the concerned marine installations necessarily involves the torsional vibration calculation requested by the Classification Societies, together with the axial vibration. Basic countermeasures against engine and ship vibrations are also mentioned in the technical documentation for specific engine types.

## Torsional vibration

Torsional vibration involves the whole shaft system of the ship propulsion machinery and is caused by the torsional excitation from the engine (gas pressure and rotating/oscillating masses) and the propeller.

Limitation of torsional vibration is vitally important to avoid damage or even fracture of the crankshaft or other propulsion system components. For this reason, the Classification Societies require detailed calculations for torsional vibration together with verification by shipboard measurement during sea trials.

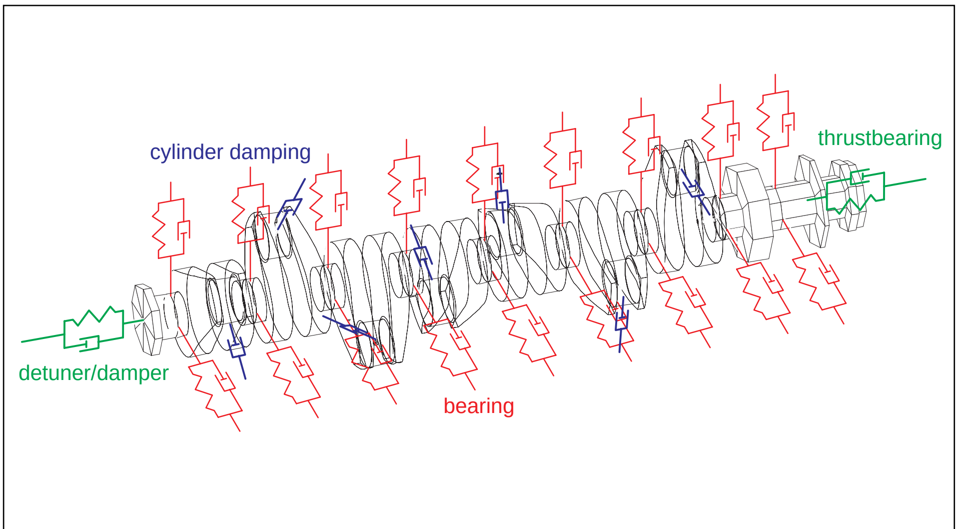



Fig. 35
The calculation model used for dynamic analysis of the crankshaft and bearing force determination consists of super-elements representing stiffness, damping, mass and inertia. It takes into account the stiffness and damping of the radial and axial bearing structures, including damping effects from engine cylinders [98#124]

These calculations have to be carried out in the case where the engine is running in the normal condition of operation and when one cylinder is not firing. This last condition is important to determine if a barred speed range has to be respected or not if it should occur.

Depending on the number of cylinders, the length of the intermediate and propeller shafts, the propeller inertia and the nominal engine speed and power, it may be necessary to consider fitting a torsional damper to reduce the level of stresses at the main critical speed (resonance). Table 5 shows the installations where a damper is definitely needed, or tends not to be required.

For each new type of engine, all the engine data needed in the torsional vibration calculations (namely, inertia, stiffness of the crankshaft, damping and engine excitation) are estimated using very efficient analytical methods and computer programs. Figure 35 shows the finite-element model for a Sulzer crankshaft suitable for the calculation of inertia and stiffness (torsional and axial vibrations).

A sophisticated computer program developed at Wärtsilä NSD Switzerland is used for analysis of torsional vibrations. Owing to the large experience with this program, the comparison between calculated and measured torsional vibration is very good.

More information about this topic is given in [1].

## Axial vibration

Axial or longitudinal vibration involves the crankshaft with the thrust bearing and propeller shafting. It is excited by the radial components of the gas and mass forces from the engine cylinders, together with the axial forces from the propeller. An additional axial excitation source comes from the torsional vibration, as twisting the crankshaft induces an axial deflection.

Axial vibration induces additional stresses in the crankshaft. Moreover, the axial forces acting on the thrust bearing also act on the engine and therefore may be a source of additional ship superstructure vibration.

For these reasons, axial vibrations must remain below admissible limits. The axial amplitude at the free end of the crankshaft can easily be measured, therefore the admissible limit for Wärtsilä NSD is given in terms of a maximum admissible axial amplitude at this location.

Each Sulzer RTA-8T engine is equipped with an integral axial detuner fitted close to the free end of the crankshaft (Fig. 36). The detuner shifts the natural frequency of the first mode of axial vibration to above the nominal speed. The required damping effect is achieved by adjustment of the throttle valve position which limits the transfer of oil between the two oil chambers of the detuner. The efficiency of

**Table 5: Influence of propeller shafting on requirements for countermeasures to limit torsional vibrations (tendency only)**

| Cylinders | Length of propeller shafting (total) | | | |
|---|---|---|---|---|
| | very short | short | medium | long |
| 5 | * / x | o / x (*) | o | – / o–x (*) |
| 6 | x | – / o | – / o | – |
| 7 | – | – / o | – | – |
| 8 | – | – | – | – |

where:

| | | |
|---|---|---|
| – | : | no countermeasure needed |
| * | : | reinforced shafting diameter |
| o | : | free-end disc |
| x | : | damper needed |
| (*) | : | depending on the propeller inertia |

Condition: Nominal engine working condition R1, intermediate and propeller shaft 600 N/mm$^2$ UTS

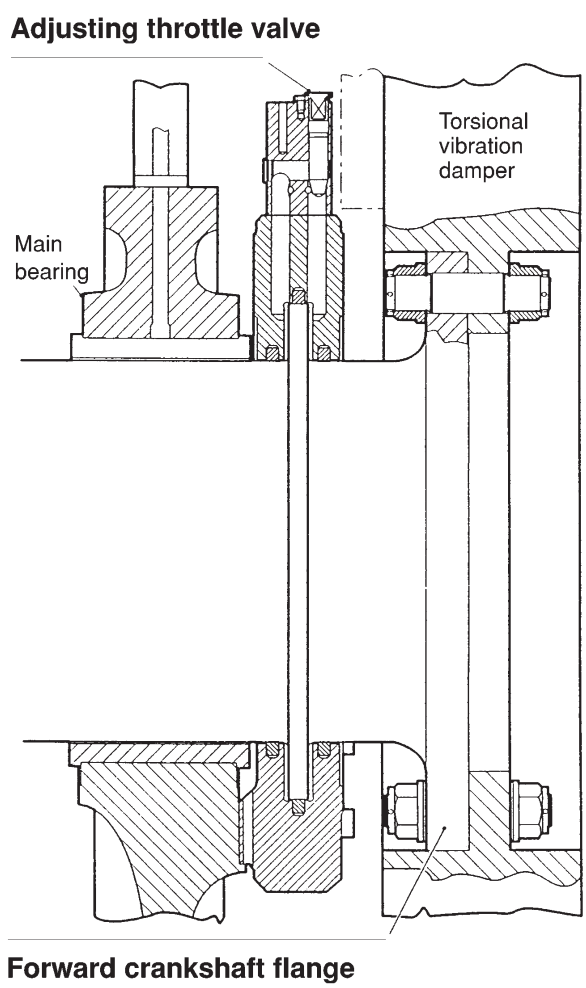



Fig. 36
Integrated axial detuner in RTA-series engines
[96#066]

this equipment has been checked on several installations. At nominal speed, the axial amplitude is largely reduced as well as the corresponding axial forces on the thrust bearing. This latter point should ensure that the axial forces acting on the ship structure are too small to induce unacceptable engine and ship vibrations.

The axial vibration problem, which mainly concerns the crankshaft, is properly solved with the integral detuner. Basically, there is no longer a necessity to calculate axial vibration because Wärtsilä NSD experience shows that dynamic problems with excessive axial vibration never occur if the integral axial detuner is correctly adjusted (more than 250 engines are equipped with this unit, without any axial vibration problem). At the request of some Classification Societies, an axial detuner monitoring system has also been developed by Wärtsilä NSD Switzerland. This new equipment is standard on RTA-8T engines.

## Engine and ship vibration

The entire ship can be regarded as a flexible structure that is apt to vibrate. The particular form of vibration experienced, however, depends on the type of excitation present, the form of ship construction, and the region of the ship considered. The hull girder can experience cyclic horizontal and vertical bending, and torsion while a tall deckhouse would be particularly susceptible to longitudinal bending vibration. The sources of excitation are the propeller, main and auxiliary engines, and even the waves of the sea.

The amplitudes of hull vibration depend on the cyclic excitation forces and moments, the dynamic relationship between harmonic excitation frequencies and the structure's natural frequencies, the engine location in the ship and the ship's cargo loading condition. For the safety of the ship's structure and comfort of the passengers and crew, there are standards laid down for the upper admissible limits for the amplitudes of hull vibration.

The risk of hull vibration is ever present because hulls always tend to become lighter in construction and thus more flexible. To minimise vibration, it is necessary to try to avoid resonance while keeping down the influence of the various excitation sources.

Controlling hull vibration is, therefore, very much a co-operative effort. Many shipyards can calculate the natural frequencies of their ship structures using finite-element analysis, which greatly helps to avoid resonance. Propeller manufacturers, in conjunction with the shipyards, can help by minimising excitation forces from propellers. The contribution of engine builders is covered in the following paragraphs.

Because of the rather low natural frequencies of many ship structures, slow-running RTA-8T engines may be more liable to excite hull vibration than faster-running engines.

Excitation from the engine can originate in the free (external) forces and moments generated by the cyclic inertia forces of unbalanced reciprocating and rotating masses as well as the lateral guide forces and moments. More rarely, axial and torsional vibration of the shaft line may also have an influence.

## Free forces and moments

The engine's reciprocating and rotating masses produce cyclic forces of first and second orders at each cylinder, which generate free moments (Fig. 37); principally the first-order vertical moment $M_{1V}$, the first-order horizontal moment $M_{1H}$, and the second-order vertical moment $M_{2V}$.

For Sulzer RTA-8T engines with five and six cylinders, the second-order vertical free moment is the most significant. To reduce the influence of the free moments, compensation of these excitations might be necessary:

$M_{1H}$, $M_{1V}$: counterweight fitted at the ends of the crankshaft (standard);

$M_{2V}$: Lanchester second-order balancer for the five- and six-cylinder engines (Fig. 38), or the new electrical balancer ELBA (Fig. 39).

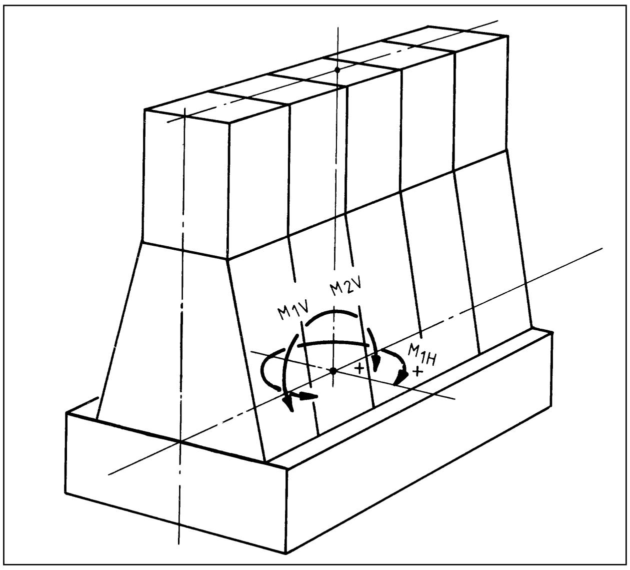



Fig. 37
Free (external) moments from the engine
[96#072]

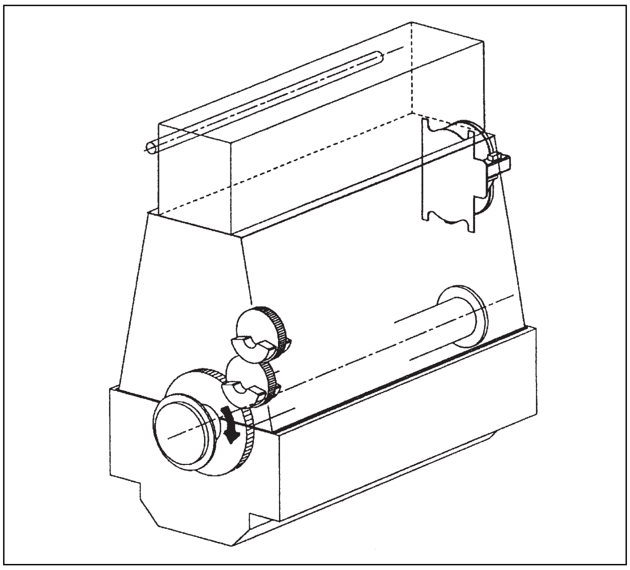
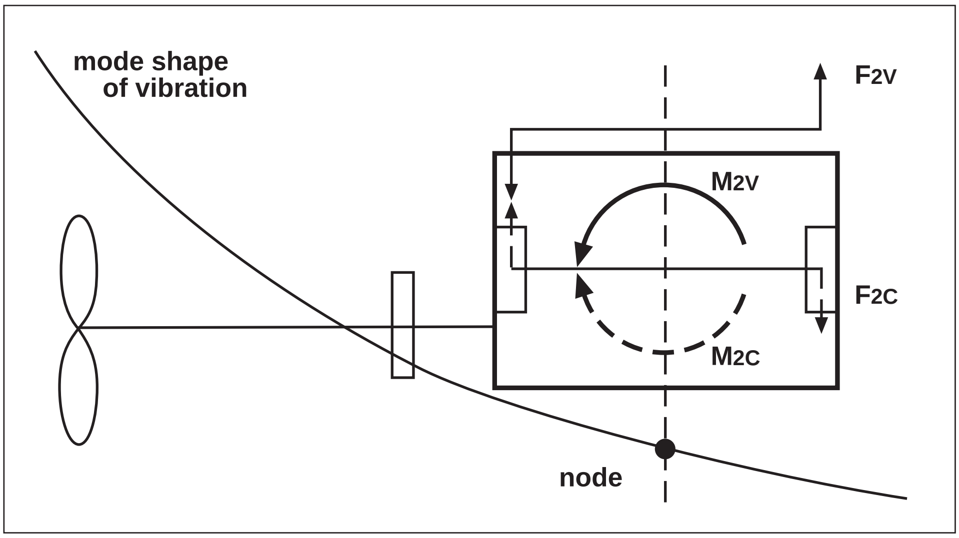


Fig. 39
For RTA48T, RTA58T and RTA68T engines, a new electric second-order balancer (ELBA) is available for fitting at the free end of the engine. It can be used in conjunction with an integral gear-driven Lanchester balancer at the driving end
[95#153]

Fig. 38
The fitting of Lanchester secondary balancers to both ends of an engine is suited to those cases where the engine is installed at a node of hull vibrations
[96#075]



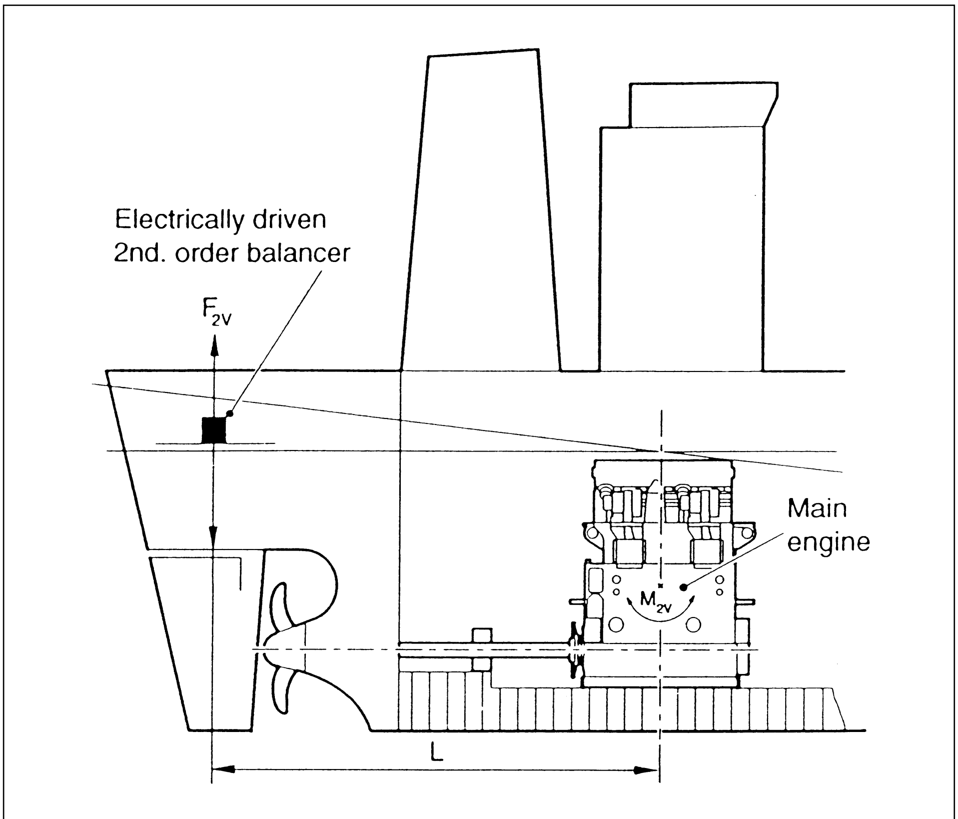



Fig. 40
The effect of an electrically-driven second-order balancer is enhanced by installing it as far aft as possible, usually on the steering gear flat
[96#078]

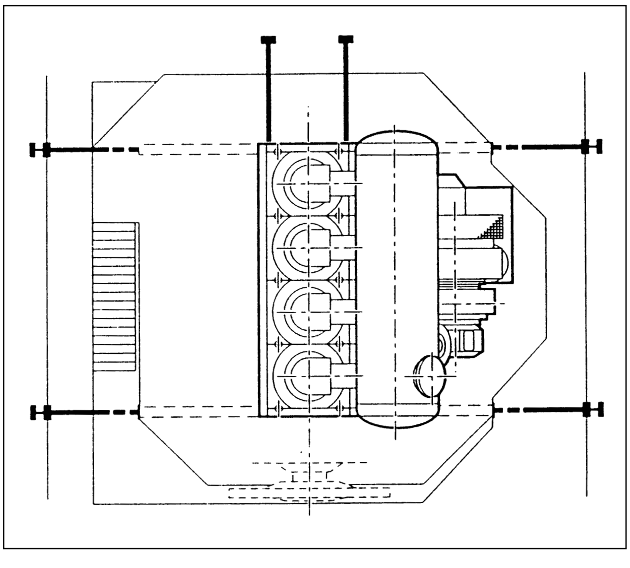
These countermeasures are explained below. Their advantage, however, is that compensation is provided directly at the source of excitation. Alternatively, compensation of the free moment $M_{2V}$ can be generated by an electrically-driven second-order balancer mounted in the stern of the ship (Fig. 40).

## Lateral forces and moments

Lateral forces act on each engine crosshead and crankshaft main bearing in counter phase. They result from the gas forces and mass inertia forces at each cylinder, and vary with the rotation angle of the crankshaft.

Depending on the number of cylinders and the harmonic order, these lateral force can generate engine vibration in either H- or X-form modes. The fundamental harmonic order (engine cylinder number), for example, gives H-mode vibration, or lateral rocking of the engine. Although this lateral vibration is not detrimental to the engine itself, it may lead to damage in attached parts such as the turbocharger supports, and may cause local vibration in the engine room and double-bottom structures. Appropriate countermeasures must then be taken.

Fig. 41
General arrangement of lateral and longitudinal stays (plan view)
[96#082]



The usual remedy for lateral vibration amplitudes is to fit side stays at the engine top. These are normally required for engines with five and eight cylinders while provision for fitting them ought to be made with six-cylinder engines in case they are shown to be necessary during the sea trials (Fig. 41).

Two types of lateral stays are standard as Sulzer engine equipment:

- Friction stays have to be fitted at one side of the engine. Their efficiency is very good if they are mounted at a very stiff locations in the hull structure and the tightening force is adjusted following the Wärtsilä NSD recommendations given in the standard engine documentation.
- Hydraulic stays cater well for the slow changes during cargo operations and ship deflections at sea. Their efficiency is very good if they are mounted following the recommendations.

Another method for reducing the lateral vibration amplitude is to fit a compensator at the top of the engine (usually for elimination of H-mode of vibration of five-cylinder engines) or two compensators at the top of eight-cylinder engines to eliminate the X-mode of vibration. Wärtsilä NSD has successful experience in using this technology to limit H- and X-mode of vibration.

## Second-order balancer

Second-order balancers are designed for all five and six cylinder RTA-8T engines. No modifications had to be introduced for the Version B.

At the aft end, two simple balance weights are incorporated in the camshaft gear train and they run in plain bearings which are bolted on the gear column.

At the free end of the engine, either a balancer mechanically driven from the camshaft, or an electrically-driven balancer (ELBA) as first introduced for the RTA-8T engines, can be fitted if required. The electrically-driven balancer was designed with the aim of being more flexible, being mechanical independent from the engine and reducing the balancer cost by more than 50 per cent.

Fig. 42
Electrical balancer ELBA on the 4RTA58T research engine
[7795-3080]

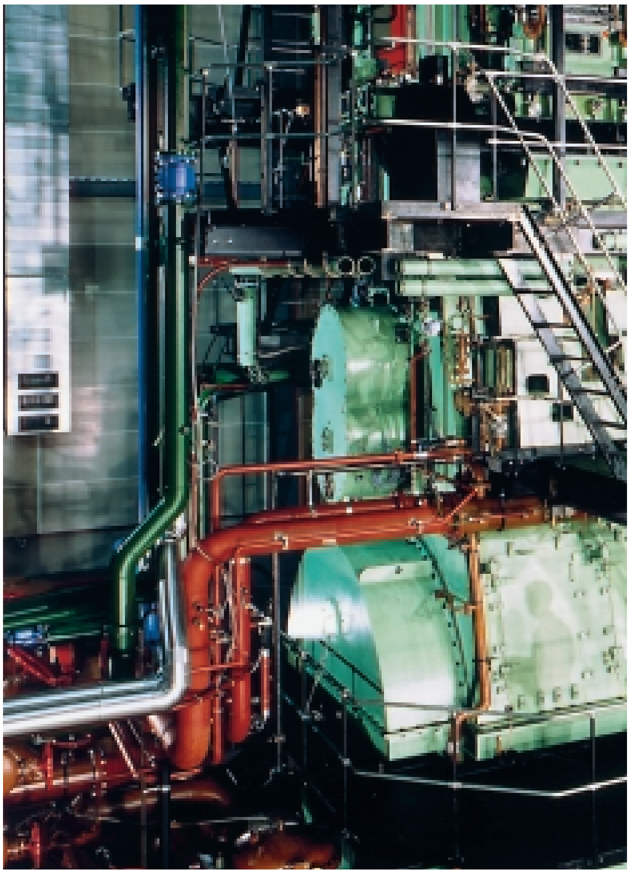
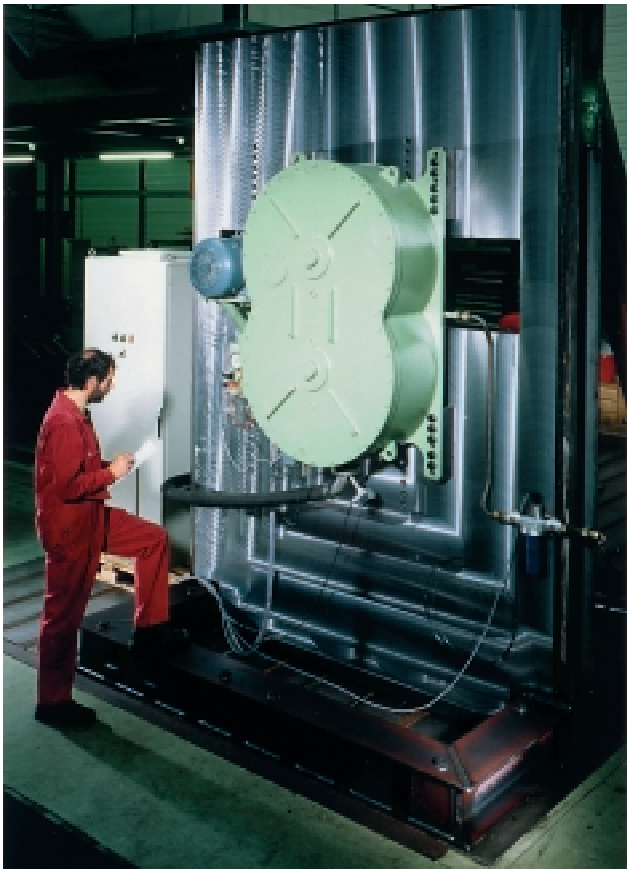


Fig. 43
Electrical balancer ELBA on the test rig
[7797-3041-2]



The first ELBA balancer was mounted on the 4RTA58T research engine in the Diesel Technology Center in Winterthur and was tested together with the engine (Fig. 42). Additionally, an endurance test of approximately 1000 hours has been carried out with the ELBA balancer on a test rig (Fig. 43). The first ELBA units have been ordered at H. Cegielski and Shanghai Shipyard, where the first engines equipped with ELBA balancers have now successfully passed their sea trials and are in service.

## Assistance to licensees and shipyards

Wärtsilä NSD Switzerland offers assistance regarding vibration calculations and measurements to all Sulzer licensees. In many cases, licensees take advantage of our offer to calculate vibrations jointly together with Wärtsilä NSD Switzerland.

# Engine management systems

**Key points**
- EMS (Engine Management Systems) is the intelligent way to save money through intelligent engine management. The EMS concept comprises different modules to meet the individual requirements of our customers in a modular way (Fig. 44). With two product families we secure a clear and integrated approach for associated products to suit the needs of operators (users) and shipyards (engines).
- All-electrical interface defined by the DENIS specification for all control and monitoring functions.
- Separate engine fitness systems modules in the MAPEX family for specialised functions such as:
  – Piston-running reliability supervision;
  – Vibration reliability supervision;
  – Management support for spare parts ordering, stock control and maintenance work.

## DENIS family for engine associated remote control and optimizing functions

The trend in modern shipbuilding has been towards automatic control from the bridge as standard. With the pressure for reduced manning, the automation tasks have been extended. Several arrangements of remote control systems are encountered; from the approach with a conventional engine control room close to the engine, up to very modern systems without a control room but with all control effected from the bridge.

To meet these different requirements, Wärtsilä NSD Switzerland has worked out a new concept with the following objectives:
- Clear definition of the signal interface between the engine and its remote control;
- Engine control reduced to local control and with the interface close to the engine;
- Interface to the remote control system (RCS) to be purely electrical.

The result of our efforts is the DENIS family (DENIS = **D**iesel **E**ngine co**N**trol and opt**I**mising **S**pecification) that was started in 1989 and introduced to the market in 1991. In the meantime, DENIS specifications have been made available for all modern types of Sulzer diesel engines. The RTA-8T engines use DENIS-6.

In fact, the change to a clearly defined, all-electrical interface is an important step towards an ‘intelligent engine’, as it is a basis for the integration of diverse control systems and automation levels into a unified ship management system.

The adoption of an all-electrical interface involved a change of philosophy. Instead of Wärtsilä NSD, as engine designer and builder, also providing the engine control system, it was decided to concentrate on the ‘engine’ and to co-operate with specialised partners who would provide the electronic systems. Nevertheless, Wärtsilä NSD is still active in those areas of control and monitoring which involve specific knowledge concerning diesel engines, as covered below.

## MAPEX family of user-associated engine fitness systems

MAPEX (**M**onitoring and m**A**intenance **P**erformance **E**nhancement with e**X**pert knowledge) is a growing family of products developed by Wärtsilä NSD to provide shipowners and operators with the tools which they need to improve the cost and operating efficiency of their engines through better management and planning. Because better efficiency means lower costs, MAPEX products translate into direct savings for the shipowner.

MAPEX products complement and expand upon the functions of standard remote control systems. They include features for monitoring, alarm and trend analysis, as well as management support for ordering spare parts, stock control and for the planning of maintenance. The specialised knowledge of engineers of Wärtsilä NSD is built into each member of the MAPEX family. This knowledge can now be put to work by the shipowner, virtually placing a trained Wärtsilä NSD engineer on board each ship full time.

## Benefits for the user

The MAPEX philosophy encompasses the following principles:
- Improved engine availability and performance through reduced down time;
- Monitoring of critical engine data – and intelligent analysis of that data;
- Advanced planning of maintenance work and management support for spare parts and for maintenance;

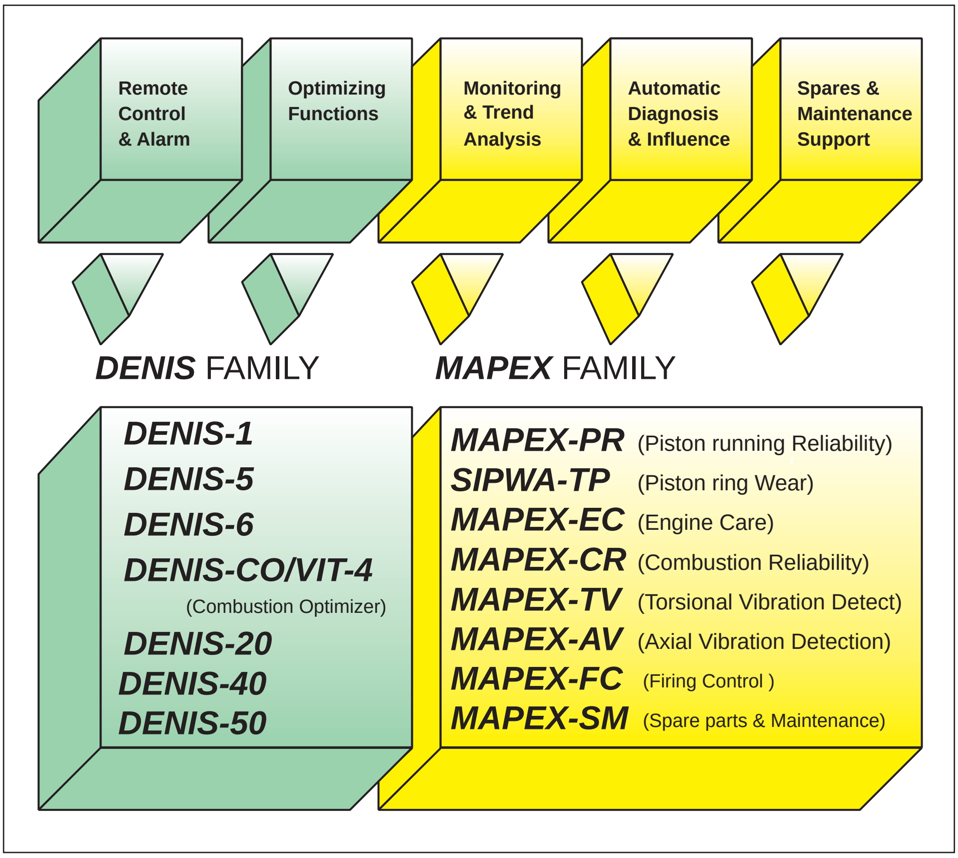



Fig. 44
Engine management systems for Sulzer low-speed engines are based on a modular concept with the DENIS interface specification for individual engine types and, in the MAPEX engine fitness family, a suite of engine performance enhancers
[97#243]

- Access on board ship to the knowledge of experts;
- Full support of data storage and transmission by floppy diskette and by satellite communications;
- Saving money through reduced costs and better efficiency.

The members of the MAPEX family are:
**MAPEX-PR** (Piston running Reliability)
- Attractive, unique tool to maintain the availability of the engine by monitoring the piston-running health.
- Should the piston-running condition suffer, alarms are activated to restore the availability of the engine in time.

**SIPWA-TP** (Wear Trend Processing)
- Well-known, widely used system to follow up the trend of piston ring wear.
- Using SIPWA-TP guarantees extended piston overhaul intervals.
- The treatment of the bunkered fuel oil is monitored.
- Cylinder lubricating oil savings are frequently reported from using this device.

**MAPEX-TV** (Torsional Vibration detector)
- Protects elastic couplings, engine components and gears, if they tend to suffer from engine misfiring.
- Alarm relay outputs with adjustable delay times are available to inform and to improve the situation in time.

**MAPEX-AV** (Axial Vibration detector)
- Monitors the function of axial vibration dampers and signals alarms if axial vibration levels exceed accepted values.
- Is designed for the acceptance tests of classification societies.

**MAPEX-SM** (Spare part and Maintenance management)
- Offers a broad range of services in the area of spares and maintenance management to cover users' needs.
- Maintains your fleet more efficiently at lower cost.

**MAPEX-CR** (Combustion Reliability)
- Monitors permanently the combustion process in each cylinder.
- Signals alarms if combustion quality is not within recommended limits.

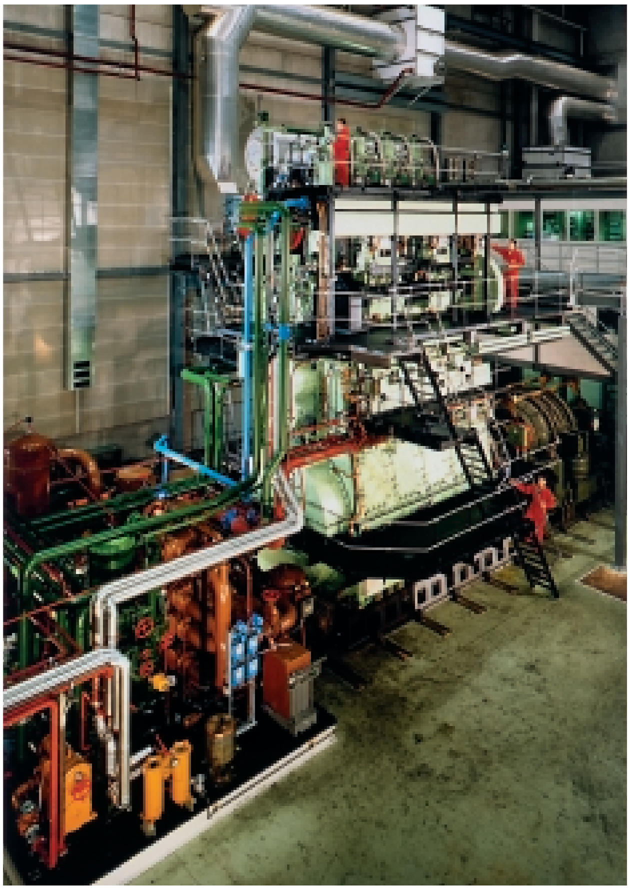
# Test results and service experience

**Key points**
With 20 RTA-8T engines in service at the end of December 1997, their service experience is proving to be very satisfactory, with:
- Low wear rates of cylinder liners, piston rings and grooves;
- Moderately clean pistons, piston rings, liners and ports in very good condition with only normal carbon deposits;
- All other components in very good condition.

The RTA58T research engine was started in the Diesel Technology Center in Winterthur at the end of September 1995, as the prototype of the RTA-8T engines (Fig. 45). By the end of 1997, the engine had accumulated about 1190 running hours of which some 300 hours have been run on heavy fuel oil. It has been operated at mean effective pressures up to 19.5 bar and mean piston speeds up to 8.6 m/s.

The engine showed very good behaviour during the running-in period, and the test programme proceeded through the planned schedule. After thermodynamic optimisation of the engine, the relevant performance characteristics were recorded.

Fig. 45
The 4RTA58T prototype engine in the Diesel Technology Center, Winterthur, where it has been tested since September 1995
[7795-3077-1]



The first fuel injection nozzles that were tested gave quite a good compromise on fuel consumption, $NO_X$ emissions and combustion chamber temperatures. However, the engine clearly showed that there was ample potential for further improvements. The specific fuel consumption figures of 173 g/kWh (127 g/bhph) for the RTA58T, with the usual three per cent tolerance, quoted in 1995, could be readily bettered. A reduction of 3 g/kWh was therefore announced shortly afterwards for the official values of production engines for all RTA-8T types.

Temperature measurements were also made on all combustion chamber components. As mentioned above, the achieved mean temperatures as well as the 'hot' spots on the cylinder cover, exhaust valve and piston are very moderate on the long-stroke RTA-8T engines. Consequently, no damage by hot corrosion is expected.

Some tests with different cylinder liner cooling arrangements were carried out to verify the optimum cylinder liner temperatures. The optimum solution found was then applied to the production engines.

A vast number of stress measurements were made on most of the engine components (Figs. 46 and 47). The results have confirmed the predictions made during the design stage. Only two components had to be modified to further improve the safety factors after their stress levels were found to be slightly higher than expected.

## RTA-8T production engines

The first 6RTA48T started its test-bed trial on 4 July 1996 at Diesel United Ltd in Aioi, Japan (Fig. 48). The engine reached full load after just eight hours of running in without any serious problems. In September 1996, the first seven-cylinder unit passed its shop test. They were followed in April 1997 by the first 7RTA58T. Key dates for the first few RTA-8T engines are given in table 6.

**Table 6: Key dates for the first few RTA-8T engines**

| Type | Cylinders | Ship | Engine Builder | Shop Trial | Sea Trial | Maiden Voyage | Inspection at hours: No. 1 | No. 2 |
|---|---|---|---|---|---|---|---|---|
| RTA48T | 6 | *Star Sea Bird* | DU | Oct 96 | Mar 97 | Mar 97 | 2478 | – |
| | 7 | *Win Trader* | DU | Sep 96 | Mar 97 | Mar 97 | 2078 | 5135 |
| | 7 | *Global Harmony* | DU | Oct 96 | Apr 97 | Apr 97 | 3048 | – |
| RTA58T | 7 | *Stena Commodore* | DU | Apr 97 | Sep 97 | Oct 97 | 1050 | – |
| | 6 | *CCNI Portrerillos* | HCP | Jul 97 | Dec 97 | Dec 97 | 1262 | – |
| | 6 | *CCNI Chagres* | HCP | Oct 97 | Oct 97 | Feb 98 | – | – |
| | 5 | *Song Shan Hai* | Shanghai | Aug 97 | Jan 98 | Jan 98 | – | – |

Diesel United, together with our team from Winterthur, tested the first RTA48T engine within a very short period. This was possible because the prototype 4RTA58T had started operation in Winterthur in September 1995 so that a great deal of the optimisation and measuring work had already been carried out. Four different rating points were consequently tested on the 6RTA48T. After fine tuning of the engine, a full range of thermodynamic, temperature and stress measurements were also carried out (Figs. 49, 50 and 51). The shop trial could be finished according to the planned schedule and the results achieved fully confirm the design objectives.

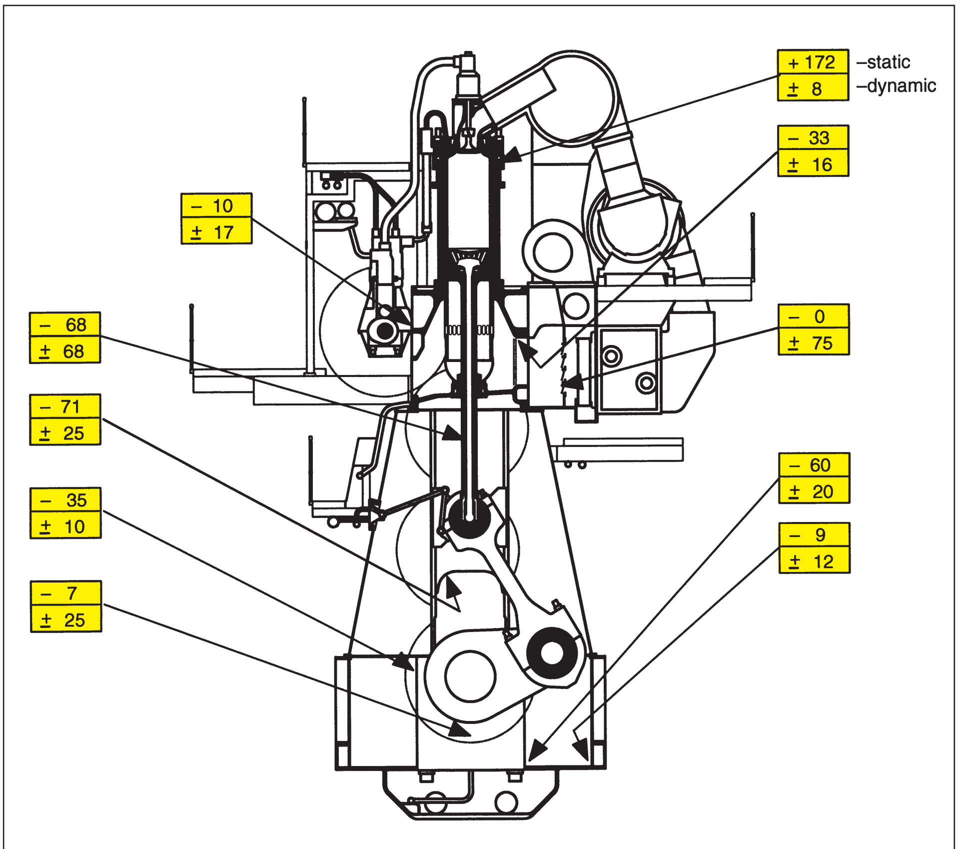



Fig. 46
Static and dynamic stresses (upper and lower figures respectively) measured on the principal components of the 4RTA58T research engine
[97#239]

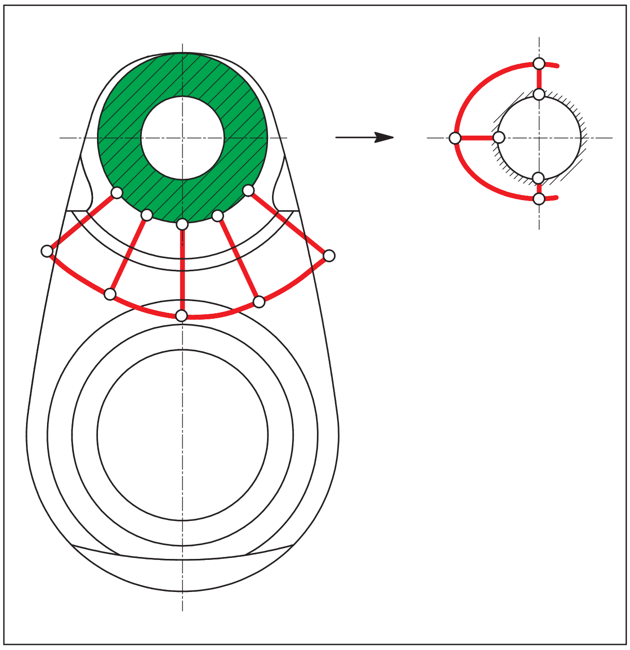


Fig. 47
Stresses measured in the crankshaft fillet area of the 4RTA58T research engine
[97#238]

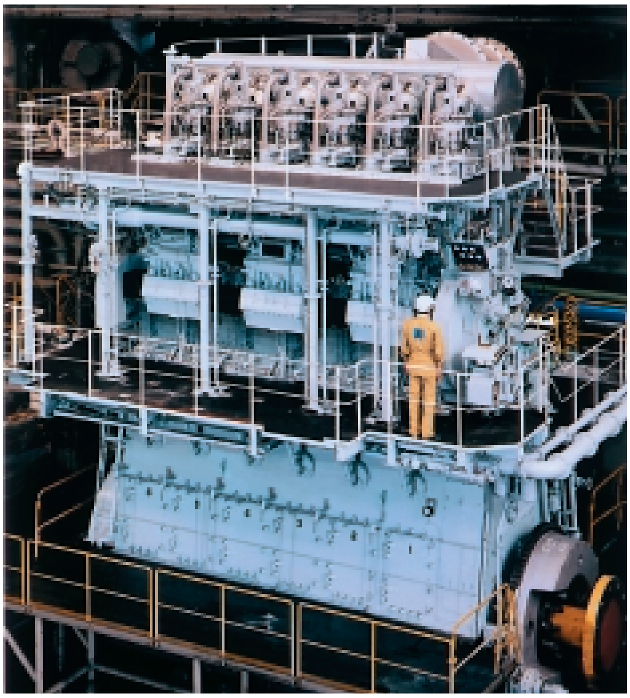
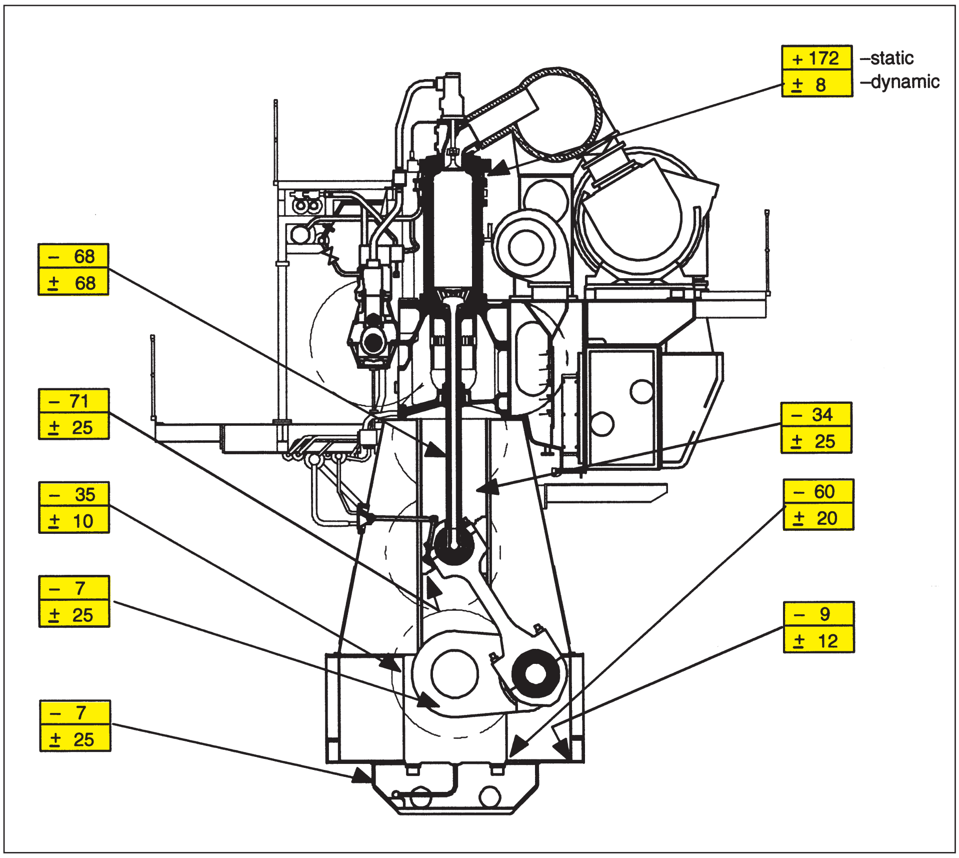


Fig. 48
The first RTA48T engine on the test bed of Diesel United Ltd in Aioi, Japan
[7796-3024]

Fig. 49
Static and dynamic stresses (upper and lower figures respectively) measured on the principal components of the first 6RTA48T engine
[98#111]



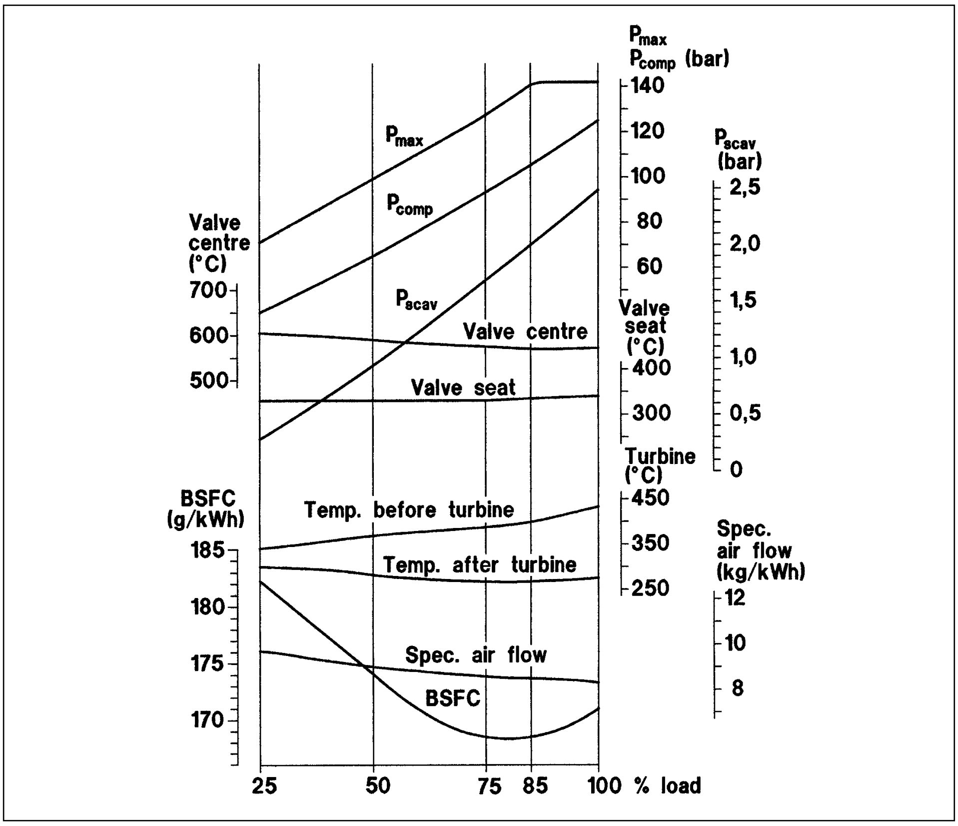



Fig. 50
Performance characteristics from the 4RTA58T prototype at 2000 kw/cylinder at 103 rev/min according to a propeller characteristic
[97#123]

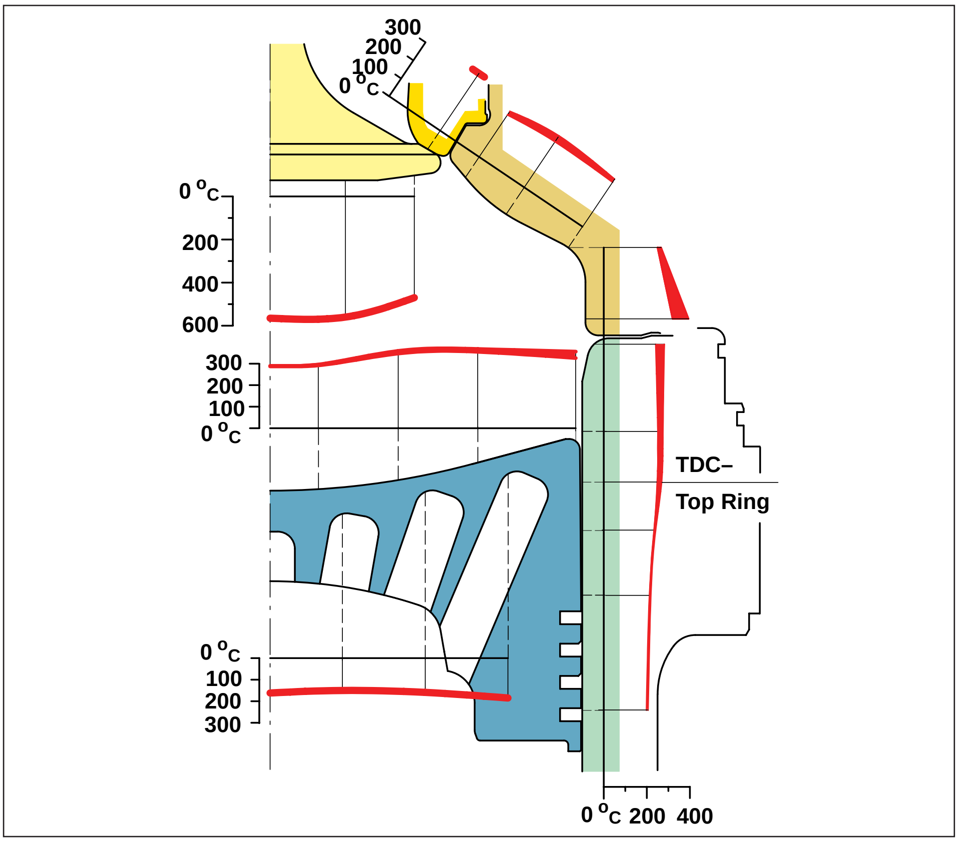



Fig. 51
Surface temperatures measured on the combustion chamber components of the RTA58T at the full-load R1 rating
[97#241]

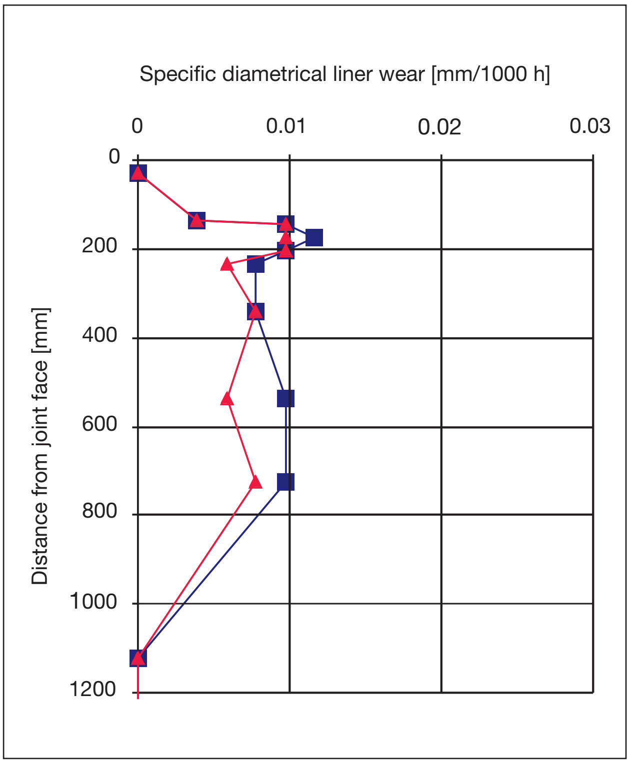
This first two RTA48T engines entered service at the beginning of 1997, both for Japanese owners. The six-cylinder engine is installed in the 42 000 tdw bulk carrier *Star Sea Bird* from I.H.I. (Fig. 52), while the seven-cylinder unit is in the Panamax bulk carrier *Win Trader*, built by Sumitomo H.I. (Fig. 53). The first RTA58T followed soon after. It is the seven-cylinder engine in the 106 000 tdw Aframax tanker *Stena*

Fig. 52
The 42 000 tdw bulk carrier *Star Sea Bird* powered by a 6RTA48T main engine [7797-3014-1]

Fig. 53
The Panamax bulk carrier *Win Trader* powered by a 7RTA48T main engine [7797-3015-2]


Fig. 54
The 106 000 tdw Aframax tanker *Stena Commodore* is equipped with a 7RTA58T main engine
[7798-3009]

Fig. 55
Liner wear rates, mm/1000 hours, measured in the 7RTA48T engine of the *Win Trader* after 5135 running hours
[98#117]




*Commodore* from NKK Corporation, which passed its sea trial at the end of September 1997 and was delivered on 28 October (Fig. 54).

As a standard procedure, service engineers from Wärtsilä NSD Switzerland accompanied the first three months of operation of each engine type. The first three RTA48T and the first two RTA58T have been closely followed from Winterthur, as well as from the respective licensees (and will continue to be followed). Inspections have been made at intervals of about 2000 running hours. All engines are giving highly satisfactory results.

From an inspection of the first 7RTA48T on the *Win Trader* after 5135 running hours, the maximum diametrical liner wear was found to be 0.012 mm/1000 hours (Fig. 55). This extremely satisfactory result is credited to the liner wall temperatures being optimised by the application of insulation tubes and mid-stroke insulation so that corrosive attack could be kept to a minimum.

During the inspections, the following components were inspected:

- Cylinder cover;
- Fuel valves and nozzles;
- Fuel pump block, stagnation pressure control valve (SPCV), spill valve;
- VIT function and parameters;
- High-pressure fuel pipes and couplings;
- Exhaust valves, exhaust valve seats and bush covers;
- Exhaust valve timing

- Liner (including wear measurements);
- Piston;
- Piston rings, grooves (including wear measurements);
- Piston undersides;
- Piston rod glands;
- Scavenge air receiver;
- Water separator;
- Condensate water drain;
- Crankcase check for white metal;
- Main bearings Nos. 5 and 6;
- Crosshead guide shoes;
- Camshaft and rollers;
- Gear wheels;
- Piston rods;
- Crosshead guides;
- Main bearing thermocouple cable supports;
- Lubricating and fuel oil samples.

The following information was reported by the inspection team that visited the *Win Trader* at 5135 running hours:

- The 7RTA48T engine is operated with an average load of about 80 per cent, while fuel with a sulphur content of 2.3 per cent and a viscosity of 380 cSt has been used. The actual lubricating oil consumption is set to 1.58 g/kWh, but it will be further reduced in the near future. The 7RTA48T of the *Global Harmony*, the 6RTA48T of the *Star Sea Bird* and 7RTA58T of the *Stena Commodore* are operated in a similar way.
- The cylinder liner No. 4 was inspected visually and Technovit replicas were taken. The machining marks are still visible, which is an indication of low liner wear. No lacquer formation and no carbon deposits could be detected neither on the liner surface nor in the lubrication grooves.
- The measured diametrical cylinder liner wear was found to be 0.012 mm/1000 hours. At the first inspection after 2078 running hours, the diametrical liner wear was measured to be 0.025 mm/1000 hours, confirming experience that liner wear during the running-in process is slightly higher than during normal operation.
- The piston crown shows normal carbon deposits (Fig. 56). Some little calcium deposits were found on both sides, in the direction of the fuel spray pattern.
- The water separator was dismantled and checked. From both sides, the inlet and outlet, the water separator was found to be very clean, without any signs of oil deposits or oil back flow.
- The piston rings of all seven cylinders were found to be in very good condition (Fig. 57). No sticking or broken rings could be detected. Moreover, all piston ring landings are free of carbon deposits, the ring packages are very clean. The piston crowns show only light carbon deposits down to the top ring. The piston skirts did not show any running marks. Some carbon deposits were found in the piston undersides, but the amount is absolutely at an acceptable level.

Fig. 56
Piston crown from the 7RTA48T of the *Win Trader* after 5135 hours' running
[98#122]

- The exhaust valve seat and the seat bush cover were inspected visually, showing absolutely no damage. On the inner diameter of the valve seat bushing, some little loss of material was visible but it was within an acceptable range for the time of operation. There was no loss of material from the exhaust valve disk.
- One fuel nozzle was removed, pressure tested and dismantled for inspection of the needle. The opening pressure was found to be 370 bar, which corresponds to the original level. On the needle, no cavitation could be detected.
- All piston ring packages where visually checked through the scavenging air inlet ports. With the ex-

Fig. 57
View through the scavenge ports of the piston rings in cylinder No. 2 of the 7RTA48T of the *Win Trader* after 5135 running hours
[98#114-2]

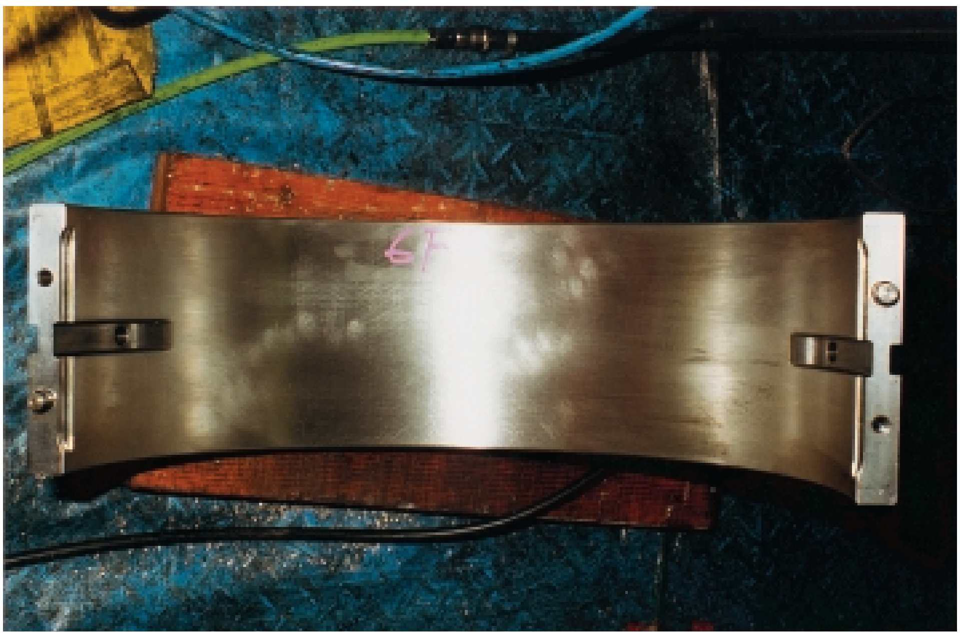



Fig. 58
Main bearing No. 6 shell of the 6RTA48T of the *Star Sea Bird* after 2478 running hours
[98#119]

ception of cylinder No. 2, no carbon deposits could be detected between the rings. On piston No. 2, the gap between top and second rings was slightly covered with carbon deposits, but within an acceptable amount.
- The cams, rollers and the gear wheel train were found to be in satisfactory condition with evenly distributed contact patterns.
- The high pressure fuel pipes and couplings were checked and were found to be fully tight.
- The regulating linkage of the VIT (variable injection timing) system was working properly without any malfunction.
- On all main bearings, the side surfaces were checked by steel wire for broken out white metal. All units were found to be undamaged.


Fig. 59
Piston ring package from the 6RTA48T of *Star Sea Bird* after 2478 running hours
[98#115]

Fig. 60
Piston crown from the 6RTA48T of *Star Sea Bird* after 2478 running hours
[98#113]

Additionally, the engine of *Star Sea Bird* was inspected after 2478 running hours and that of *Stena Commodore* after 1050 running hours. Basically the same information was collected as from the engine of the *Win Trader*. On *Star Sea Bird*, two pistons were pulled and one main bearing was inspected. The main bearing shell was found to be in satisfactory condition, with no loss of white metal (Fig. 58). The piston crown showed a normal level of carbon deposits, with no sticking or broken rings (Fig. 59). Only small calcium deposits were found on the piston surface (Fig. 60). Also the cylinder cover had a normal grade of carbon deposition (Fig. 61), with some small accumulation in the exhaust gas duct.

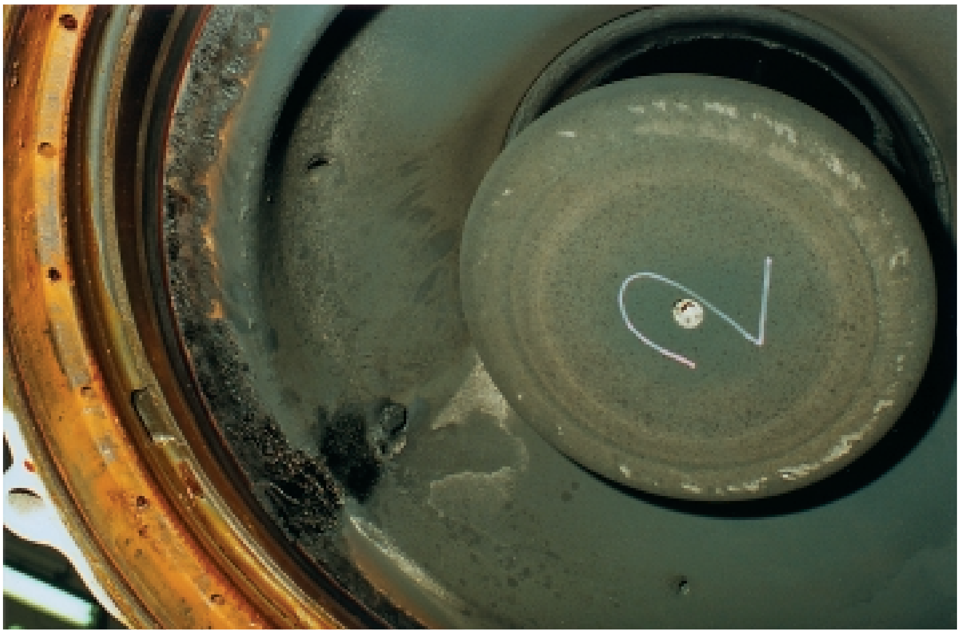


Fig. 61
Exhaust valve from the 6RTA48T engine of the *Star Sea Bird* after 2478 running hours
[98#112]

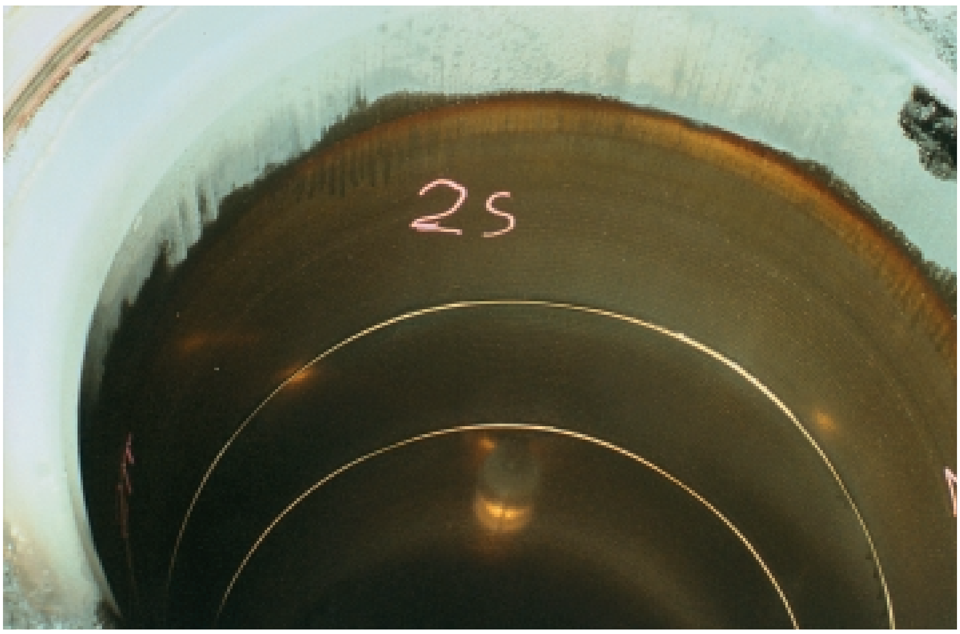



Fig. 62
Cylinder liner from the 6RTA48T of *Star Sea Bird* after 2478 running hours
[98#116]

Also on this engine, the liners still showed the machining marks, proving the excellent piston-running behaviour (Fig. 62). The piston ring wear was measured to be 0.14 mm/1000 hours (Fig. 55). The camshaft drive gear train was in a similar satisfactory condition (Fig. 63) to that in the *Win Trader*, with no damage at all.

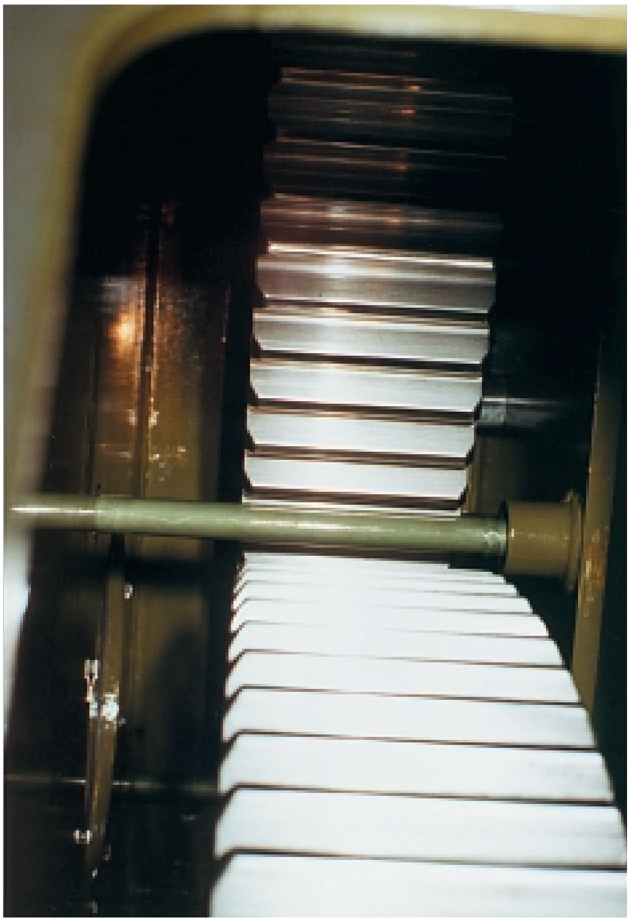


Fig. 63
Gear wheels of the 6RTA48T of the *Star Sea Bird* after 2478 running hours
[98#118]

# Conclusion

The Sulzer RTA-8T engines have been developed as tailor-made ‘workhorses’ for tankers and bulk carriers. The engine bores have been chosen to fulfil the required ship power with mainly six- and seven-cylinder engines. With stroke/bore ratios of 4.17 for the RTA48T and RTA58T and 4.0 for the RTA68T, they perfectly match the low propeller speeds required by these ship types.

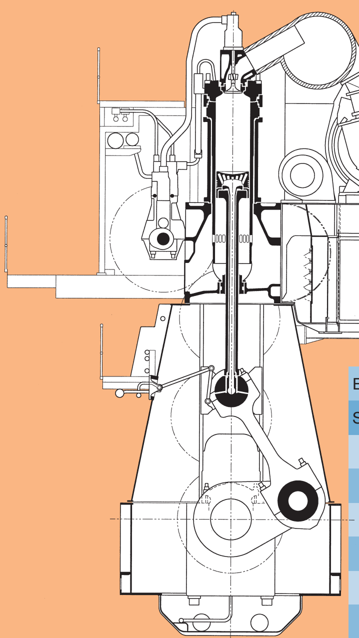
To secure the reliability of the new RTA-8T engines, the design is based upon the well-proven RTA-2 and RTA-U engines. New design features have been calculated by finite-element methods. Test results clearly confirm that the RTA-8T engines are achieving the levels of temperatures, stresses, strains and other parameters which will ensure that the new engines will reach the defined goals of reliability and times between overhauls (TBO).

Special efforts were also made in the engine design to offer cost and manufacturing-friendly engines to shipyards and engine builders. Furthermore, design simplifications were made to offer shipyards shorter times and lower costs for the installation of the engines on board ship.

The first service experience shows that the engines run to our full satisfaction, with diametrical liner wear well under 0.03 mm/1000 hours.

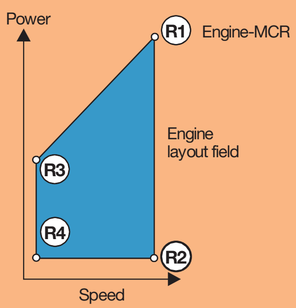
# References


1. J. Jenzer, ‘Some vibration aspects of modern ship installations’, Wärtsilä NSD Switzerland Ltd, July 1996.



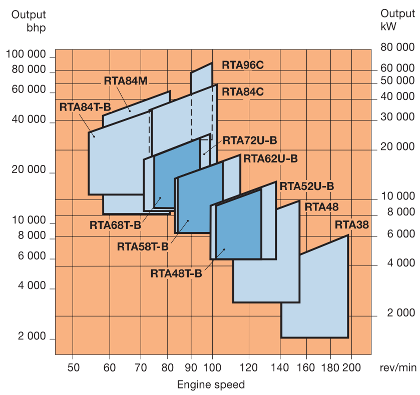
# Two-Stroke Marine Diesel Engines Main Data

| | | RTA68T-B | | | | RTA58T-B | | | | RTA48T-B | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| Bore x Stroke mm | | 680 x 2720 | | | | 580 x 2416 | | | | 480 x 2000 | | | |
| Speed rev/min | | 94 | 94 | 75 | 75 | 105 | 105 | 84 | 84 | 127 | 127 | 102 | 102 |
| Cyl. | Power | R1 | R2 | R3 | R4 | R1 | R2 | R3 | R4 | R1 | R2 | R3 | R4 |
| 5 | kW | 14 700 | 8 825 | 11 750 | 8 825 | 10 625 | 6 375 | 8 500 | 6 375 | 7 275 | 4 375 | 5 825 | 4 375 |
| | bhp | 20 000 | 12 000 | 16 000 | 12 000 | 14 450 | 8 675 | 11 550 | 8 675 | 9 900 | 5 950 | 7 925 | 5 950 |
| 6 | kW | 17 640 | 10 590 | 14 100 | 10 590 | 12 750 | 7 650 | 10 200 | 7 650 | 8 730 | 5 250 | 6 990 | 5 250 |
| | bhp | 24 000 | 14 400 | 19 200 | 14 400 | 17 340 | 10 410 | 13 860 | 10 410 | 11 880 | 7 140 | 9 510 | 7 140 |
| 7 | kW | 20 580 | 12 355 | 16 450 | 12 355 | 14 875 | 8 925 | 11 900 | 8 925 | 10 185 | 6 125 | 8 155 | 6 125 |
| | bhp | 28 000 | 16 800 | 22 400 | 16 800 | 20 230 | 12 145 | 16 170 | 12 145 | 13 860 | 8 330 | 11 095 | 8 330 |
| 8 | kW | 23 520 | 14 120 | 18 800 | 14 120 | 17 000 | 10 200 | 13 600 | 10 200 | 11 640 | 7 000 | 9 320 | 7 000 |
| | bhp | 32 000 | 19 200 | 25 600 | 19 200 | 23 120 | 13 880 | 18 480 | 13 880 | 15 840 | 9 520 | 12 680 | 9 520 |
| | | | | | | | | | | | | | |
| BSFC | | | | | | | | | | | | | |
| 85% | g/kWh | 166 | 160 | 167 | 162 | 166 | 160 | 167 | 162 | 167 | 161 | 168 | 163 |
| Load | g/bhph | 122 | 118 | 123 | 119 | 122 | 118 | 123 | 119 | 123 | 119 | 124 | 120 |
| 100% | g/kWh | 169 | 159 | 169 | 162 | 170 | 160 | 170 | 163 | 171 | 161 | 171 | 164 |
| Load | g/bhph | 124 | 117 | 124 | 119 | 125 | 118 | 125 | 120 | 126 | 119 | 126 | 121 |
| BMEP, bar | | 19.0 | 11.4 | 19.0 | 14.3 | 19.0 | 11.4 | 19.0 | 14.3 | 19.0 | 11.4 | 19.0 | 14.2 |

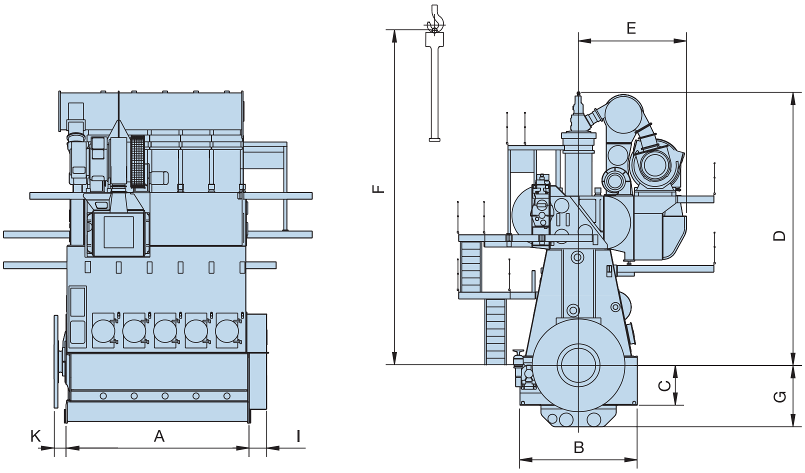



Definitions to all Sulzer diesel engines:

- R1, R2, R3, R4 = power/speed ratings at the four corners of the RTA engine layout field (see diagram).
- R1 = engine Maximum Continuous Rating (MCR).
- Contract-MCR (CMCR) = selected rating point for particular installation. Any CMCR point can be selected within the RTA layout field.
- BSFC = brake specific fuel consumption. All figures are quoted for fuel of net calorific value 42.7 MJ/kg (10 200 kcal/kg) and ISO standard reference conditions (ISO 3046-1), without engine-driven pumps and with the following allowances either:
  - Standard engine +3% on BSFC
  - or to comply with IMO emissions limit +5% on BSFC
- The values of power in kilowatts and fuel consumption in g/kWh are the official figures and discrepancies occur between these and the corresponding bhp values owing to the rounding of numbers.
- ISO standard reference conditions
  - Total barometric pressure 1.0 bar
  - Suction air temperature 25 °C
  - Charge air cooling-water temperature 25 °C
  - Relative humidity 60%




**Power and speed ranges of Sulzer RTA-series engines**




SULZER RTA-T

# Dimensions and Masses

(Millimetres and tonnes)

## RTA68T-B

| Cyl. | A | B | C | D | E*** | F* | G | I | K | Mass, t |
|---|---|---|---|---|---|---|---|---|---|---|
| 5 | 6960 | 4300 | 1520 | 10250 | 4340 | 12000 | 2340 | 610 | 480 | 412 |
| 6 | 8140 | 4300 | 1520 | 10250 | 4340 | 12000 | 2340 | 610 | 480 | 472 |
| 7 | 9320 | 4300 | 1520 | 10250 | 4340 | 12000 | 2340 | 610 | 480 | 533 |
| 8 | 10500 | 4300 | 1520 | 10250 | 4340 | 12000 | 2340 | 610 | 480 | 593 |

## RTA58T-B

| Cyl. | A | B | C | D | E** | F* | G | I | K | Mass, t |
|---|---|---|---|---|---|---|---|---|---|---|
| 5 | 5940 | 3820 | 1300 | 8810 | 3512 | 10880 | 2000 | 655 | 390 | 281 |
| 6 | 6946 | 3820 | 1300 | 8810 | 3512 | 10880 | 2000 | 655 | 390 | 322 |
| 7 | 7952 | 3820 | 1300 | 8810 | 3512 | 10880 | 2000 | 655 | 390 | 362 |
| 8 | 8958 | 3820 | 1300 | 8810 | 3512 | 10880 | 2000 | 655 | 390 | 407 |

## RTA48T-B

| A | B | C | D | E** | F* | G | I | K | Mass, t |
|---|---|---|---|---|---|---|---|---|---|
| 4929 | 3170 | 1085 | 7334 | 3292 | 9030 | 1700 | 573 | 371 | 171 |
| 5763 | 3170 | 1085 | 7334 | 3327 | 9030 | 1700 | 573 | 371 | 196 |
| 6597 | 3170 | 1085 | 7334 | 3327 | 9030 | 1700 | 573 | 371 | 221 |
| 7431 | 3170 | 1085 | 7334 | 3327 | 9030 | 1700 | 573 | 371 | 247 |

Definitions to dimensions and masses:
* Standard piston dismantling height, can be reduced with tilted piston withdrawal.
** Valid for R1-rated engines.
*** Valid for R1-rated engines with turbocharger VTR714.
- All dimensions in millimetres, not binding
- t = net engine mass, metric tonnes, without oil/water, not binding

Modifications may be introduced without notice.

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