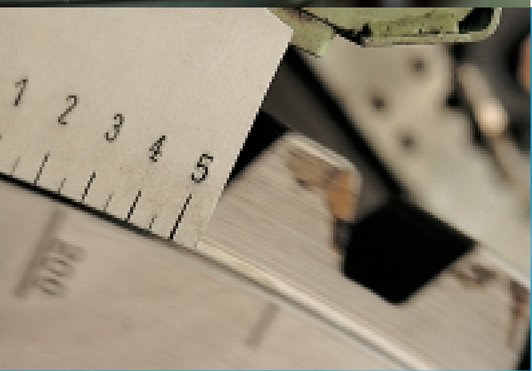
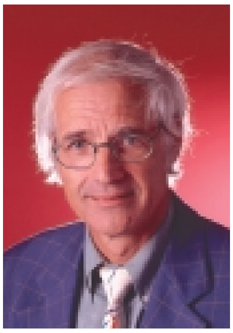
SULZER RTA-C

Technology Review



WÄRTSILÄ NSD
CORPORATION

Sulzer RTA84C and RTA96C engines:
The reliable driving forces for large, fast containerships



by Rudolf Demmerle
Product Manager, RTA-C Engines
Wärtsilä NSD Switzerland Ltd, Winterthur

October 1997

Contents **Page**

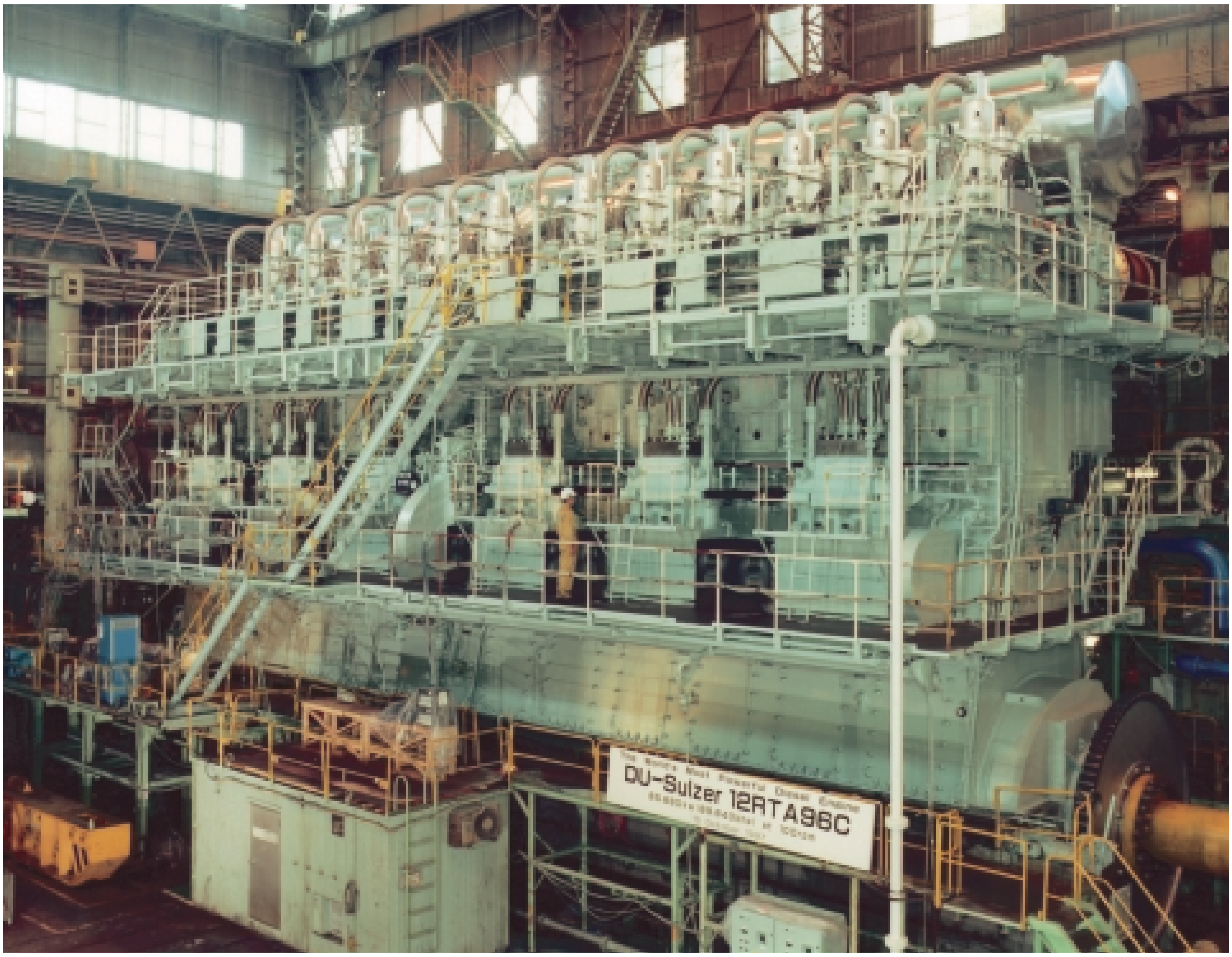


Fig. 1
The first 12-cylinder Sulzer RTA96C engine on test at Diesel United Ltd, Aioi, Japan
[7797-3038]

Summary

Containerships continue to be a most interesting sector of the newbuilding market. Not only is it still a strong market in terms of overall volume but it also puts continual demands on engine development. Larger, faster containerships require high-output propulsion engines which must operate most reliably with the least time out of service. Sulzer diesel engines have proved to be particularly successful in this market sector. To the RTA84C type which was introduced in 1988, and upgraded in 1993, has been added the RTA96C to extend the power range up to 89 640 bhp (65 880 kW) for the new generation of large ‘Post-Panamax’ containerships that load 6000 TEU or more.

The Sulzer RTA84C is the leader in this market sector, with 155 engines delivered or on order. Orders have already been received for eight RTA96C engines, including ten-, 11- and 12-cylinder models, and the first engine completed its shop trials in May 1997. A major landmark was the starting of the first 12-cylinder RTA96C in September 1997 as the world’s most powerful diesel engine.

This paper presents the development of both RTA-C engine types, together with accounts of the service experience of the RTA84C and test results from the first RTA96C engine, with 11 cylinders started in March 1997.

Key Points
Summaries of key points are given in boxes throughout the paper.

Introduction

The Sulzer RTA-C two-stroke diesel engines originated in September 1988 with the introduction of the RTA84C as a prime mover for the coming generation of large and fast containerships. It offered greater power outputs than the RTA84 which had already proved to be very popular for the propulsion of large containerships. The RTA84C, in turn, was readily accepted in the market and has become by far the leading design for this application.

Its reliability was acknowledged very quickly by the containership operators and led to a very good reputation, further applications and repeat orders. It thus became the market leader for this application segment.

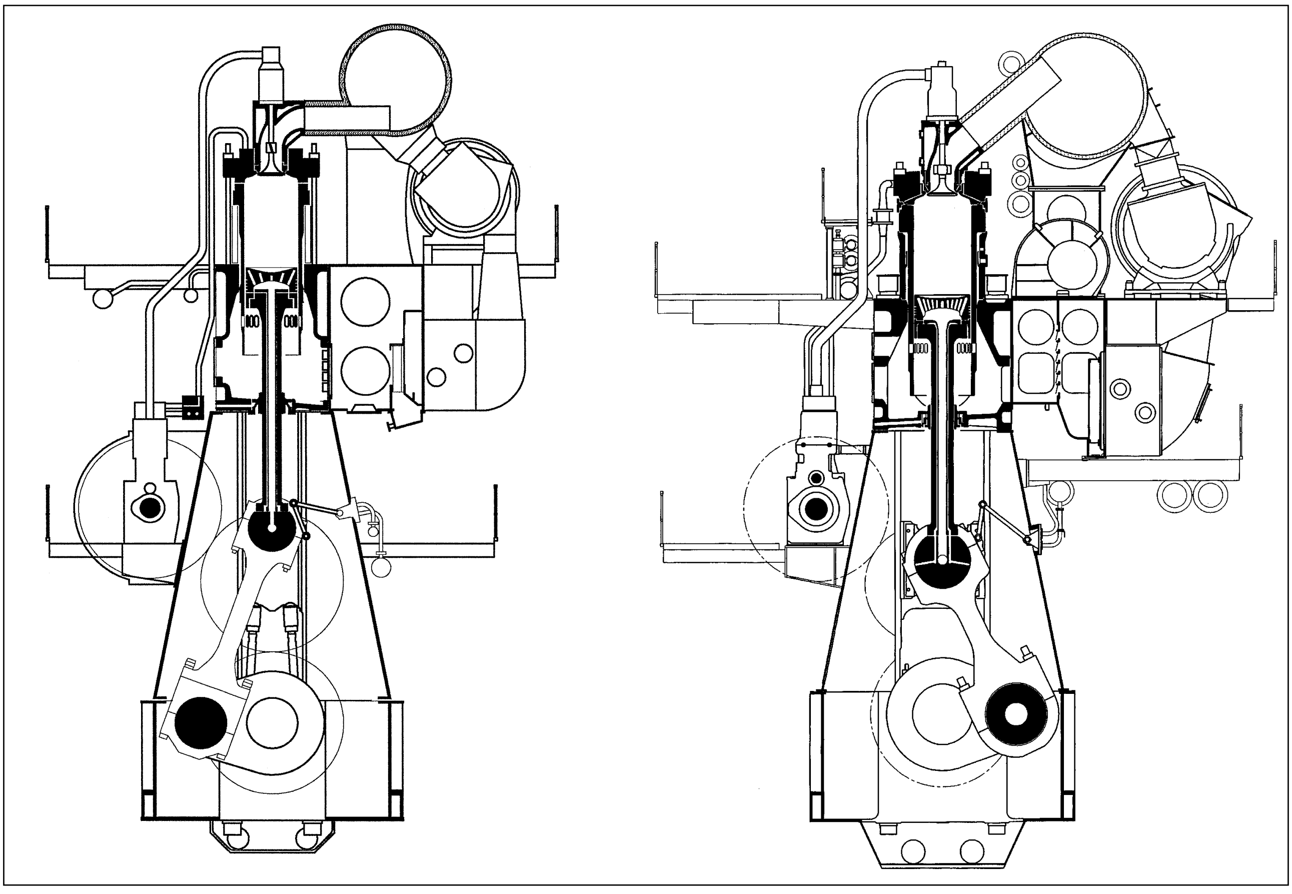
In 1992–1993, it was realised that there was a growing demand for even higher output engines in the containership market. As a consequence, the RTA84C engine was upgraded with a combination of design improvements to increase the proven reliability, as well as to provide a moderate six per cent increase in power (Figs. 2 and 3).

This upgraded RTA84C soon also attracted much interest from containership operators and built upon the earlier orders for the RTA84C. Now, some 155 RTA84C marine engines with a combined output of more than 7.86 million bhp (5.78 million kW) are in service or are on order for large containerships. These engines have been ordered by more than 20 different shipowners in the East Asia, Europe and the USA, and are being built by eight enginebuilders. It still remains the most popular prime mover in its power range.

The trend to ever larger containerships continued. Consequently, the power need in this market segment soared upwards thereby creating demand for a quantum jump in the engine bore size. This was studied by Wärtsilä NSD and it led to the launching of the Sulzer RTA96C containership engine in December 1994.

The new large-bore two-stroke engine extends the power spectrum of the RTA series up to almost 90 000 bhp (66.2 MW) in the 12-cylinder model at

Fig. 2
Cross sections of the upgraded RTA84C (left) and the new RTA96C (right) to the same scale
[7793-3034, 7797-3026]



100 rev/min (Figs. 2 and 3). Its design is based fully on the RTA84C to take advantage of the wealth of experience in theoretical design, test-bed research and operating service from the RTA84C and other previous RTA engines. The first RTA96C, an 11-cylinder engine, successfully completed its shop trials in May 1997 at the Aioi works of Diesel United. When this paper was being written, the first 12-cylinder RTA96C engines were being erected in the Aioi works for testing to begin in September 1997.

Together, the RTA84C and RTA96C two-stroke engines provide a comprehensive engine programme for all sizes of large containerships, from around 2500–3000 to 8500 TEU capacity.

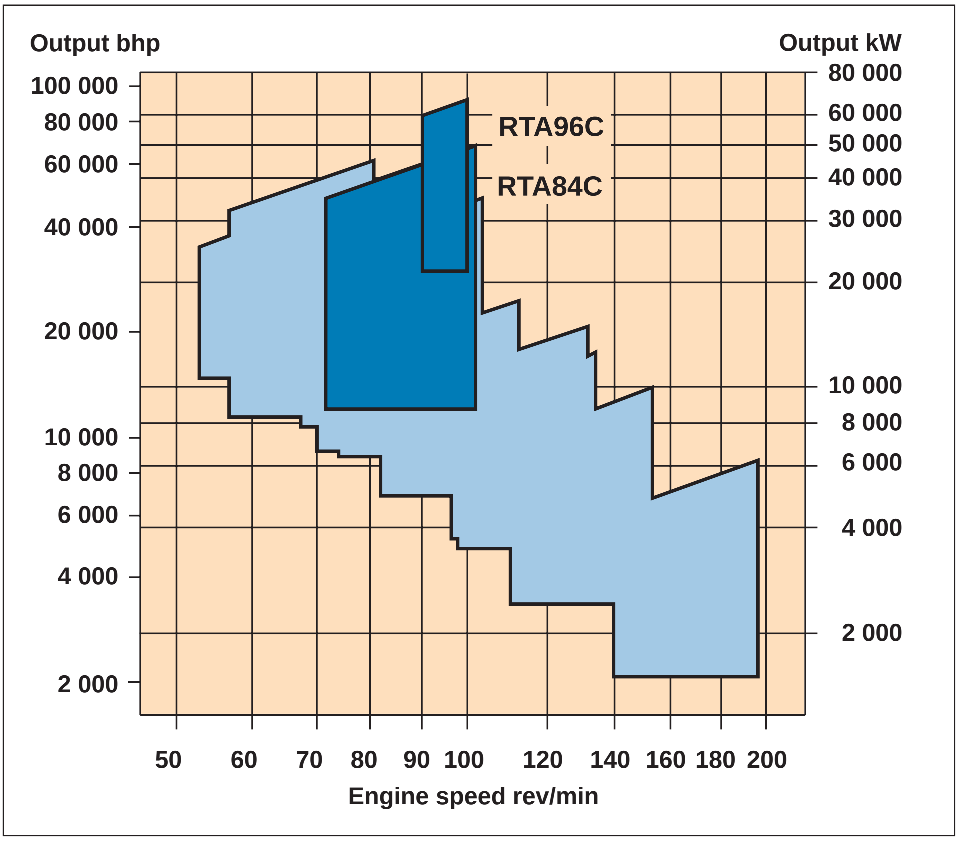


Fig. 3
Power and speed ranges of Sulzer RTA-C marine diesel engines superimposed on the overall range of the RTA series. The upper limit has been raised to 89 640 bhp (65 880 kW) in the 12-cylinder RTA96C
[97#210]

Market factors in containership propulsion

When the Post-Panamax ships with capacities of around 4000 TEU were created in the late 1980s, even larger container ships were not believed to be necessary. However, the past increases in the sizes of containerships, by either building new, larger vessels or adding extra midbody sections to 'jumboise' existing ships, had demonstrated the potential for 'economies of scale' to reduce the costs of container transportation. Therefore, owners began to investigate even bigger ships, with capacities of up to 6000 TEU or more, sailing at service speeds of 24 or even 25 knots. Now, ships of 5500-6600 TEU capacity are under construction or have already come into service while a number of shipbuilders have 8000–8500 TEU ships on the drawing board.

Key Points

The development of containerships has led to successive generations of larger, faster ships:

- Ship sizes increased by major capacity steps: 4000, 5000, 5500, 6000 and more TEU, with studies now for 8000-8500 TEU.
- Post-Panamax breadth became standard for ships above 4000 TEU.
- Service speeds settled at 24 or even 25 knots.
- The available maximum engine powers have thus risen:
  1981: 48 360 bhp (35 520 kW)
  1984: 54 000 bhp (39 720 kW)
  1988: 62 400 bhp (45 840 kW)
  1993: 66 120 bhp (48 600 kW)
  1994: 89 640 bhp (65 880 kW)
- The key requirements from the shipowners remain:
  – Reliable prime movers so that sailing schedules are maintained,
  – Long times between overhauls for minimum off-hire time.

For container capacities greater than 4000 TEU, there were seen to be distinct advantages in exceeding the limitations in dimensions imposed by the Panama Canal. The so-called 'Post-Panamax' hull dimensions, with a beam greater than Panamax breadth (usually of 32.2 m), allow a more efficient hull form in terms of the power required to propel a ship with a given container capacity at a given speed. The 'Post-Panamax' concept was pioneered by APL's 4340 TEU C-10 class (delivered in 1988 and powered by single Sulzer 12RTA84 engines each of 57 000 bhp, 41 920 kW) with their beam of 39.4 m. The concept has now become the norm for containerships of around 4500 TEU and larger.

Studies were also initiated at various institutions to design larger containerships, not only longer but also with greater beam with capacities for 6000 TEU or even more, possibly for 8000 TEU. The results were clearly positive, even when taking into account the logistic changes needed on shore at the 'main haul' ports served by the larger container liners, such as new, longer-reach cranes, the greater areas to store and move the enormous number of containers arriving and leaving within short times, and the increased capacities of block trains. There will also be a 'knock-on' effect with increases in the size of feeder containerships.

Such Post-Panamax ships of 6000 TEU capacity have overall lengths of some 300 m, breadths of 39–42 m or more, scantling draughts of 13.5–14 m and design draughts of 12.0–12.5 m. They run at service speeds of 24 to 25 knots, or faster.

Taking such dimensions and speeds into consideration, estimates for the propulsion power jumped to levels above 70 000 bhp, and settled at around 85 000 bhp (Fig. 4). This also takes into account a generous sea margin so that the ship will be able, at any time and under any circumstances, to fulfil the planned service schedule. Owners also wished to keep to a single-screw, single-engine configuration.

This was the background to the announcement in December 1994 of the new Sulzer RTA96C containership engine with its considerably greater bore to offer the higher propulsion powers needed in this market segment.

A key market factor in containership propulsion, however, is reliability. As might be expected, this is clearly the number one priority in designing a new large-bore containership engine. It is a priority driven by the ship operators' needs:

- The value of such a ship together with its cargo is exceptionally high, therefore all precautions have to be taken so that there is always sufficient pro-

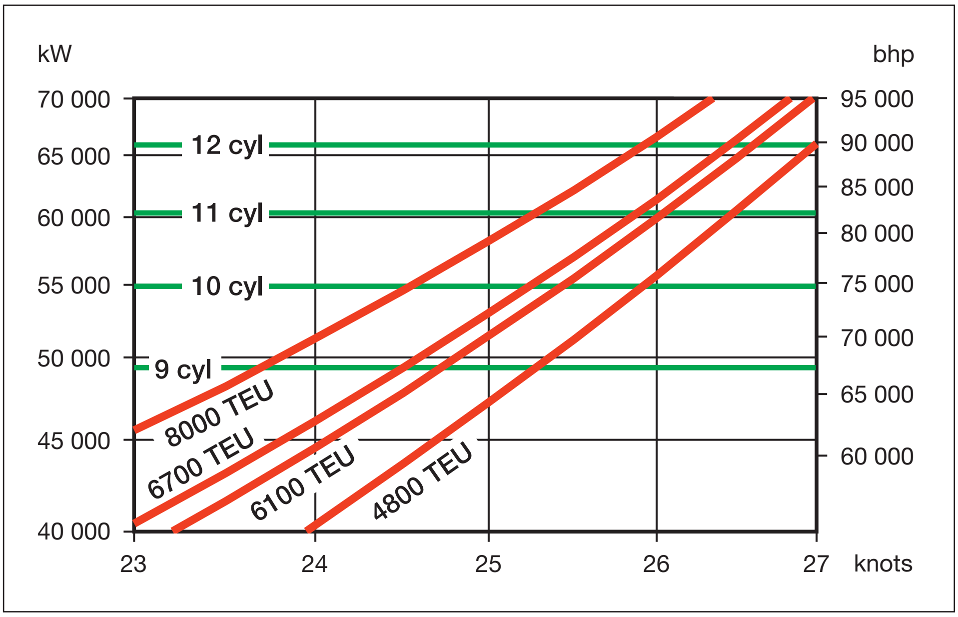


Fig. 4
Engine power requirement (1000 bhp) versus ship speed (knots) calculated for four typical containerships of 4800, 6100, 6700 and 8000 TEU. Sea margin of 20 percent without shaft generator. The power outputs of individual RTA96C models are marked
[97#224]

pulsion power readily available to keep the ship safe.

- The risk of missing the schedule with a ship carrying a huge quantity of high-value cargo is an economic threat of the first order. Such ships normally operate in a tightly timetabled, high-frequency schedule so that a delay with one ship might disrupt the whole service. Therefore, the propulsion power must always to be available to keep the ship on schedule, with ample power margin in hand to be able to catch up time, if necessary.
- Such valuable ships need to be in service for as long as possible without stopping, before they go to a pre-scheduled overhaul. The interval between ship overhauls should be taken as the basis for timing engine overhaul intervals. Therefore, the engine needs to run with clear and safe overhaul times for the main components (times between overhauls, TBO) to allow for the planning of maintenance work.

Market success of the RTA-C engines

The completion of the most powerful Sulzer diesel engine ever, a 12-cylinder RTA96C of 89 640 bhp (65 880 kW) is a major landmark in the history of diesel engineering. It follows, however, from the outstanding market success of its older but smaller brother, the RTA84C (Table I). The success is more than a matter of simple numbers but to the tremendous diversity of owners, ship designs, shipping routes and shipbuilders involved around the world.

The first RTA84C engine went into service in July 1990. It is a nine-cylinder unit in the *Katsuragi* of NYK Line (Figs. 5 and 6). Initially the nine-cylinder model was the most commonly ordered of the RTA84C type. These include the ten 4229 TEU R-class ships of Evergreen, nine 4000 TEU ships of Sea-Land Services, seven 4038 TEU ships of P&O Nedlloyd, seven 3800 TEU ships of COSCO, and six NYK ships of 3000–3800 TEU.

Table I: Numbers of RTA-C marine engines in service and ordered at end July 1997

Type	Cylinders	In service	Ordered
RTA84C	12	21	44
	10	14	18
	9	39	43
	8	14	21
	7	16	21
	6	6	8
	Total	110	155
RTA96C	12	–	4
	11	–	2
	10	–	2
	Total	–	8

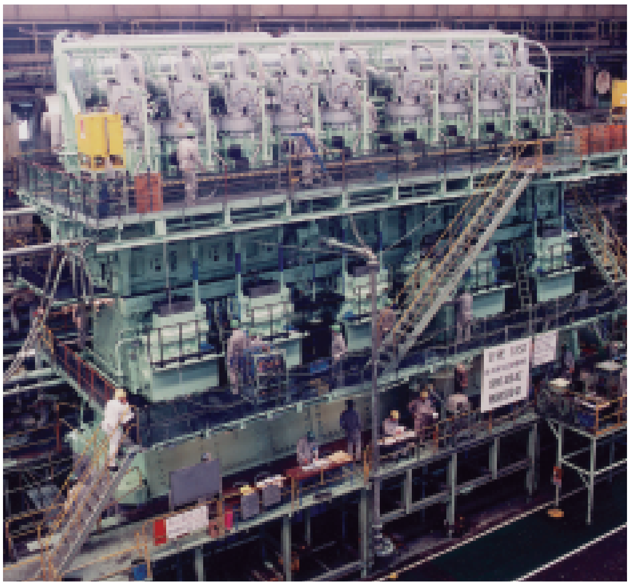


Fig. 5
Sulzer 9RTA84C engine on test
[7793-3030-9]

Fig. 6
Katsuragi powered by a 9RTA84C engine
[7790-3027-2]

The trend to higher unit outputs is demonstrated by the increasing number of 12-cylinder RTA84C engines ordered (Fig. 7). They are in ships with capacities in the range of 4112-5365 TEU (Fig. 8). The first 12RTA84C engine to enter service is in the 4112 TEU 'Post-Panamax' containership *Nedlloyd Hongkong* of P&O Nedlloyd delivered in February 1994, to which a sistership *Nedlloyd Honshu* was added in the following year. Most notably, Evergreen will have 23 ships powered by 12-cylinder RTA84C engines when their current newbuilding programme is completed in 1999. These include 13 U-class vessels of the 5365 TEU capacity and ten of the faster, 25-knot D-class Panamax ships of 4211 TEU. The first U-class vessel is the *Ever Ultra* which entered service in June 1996 (Fig. 9), while the first in the D class, *Ever Dainty*, was handed over in July 1997.

Fig. 7
Sulzer 12RTA84C engine on test
[7796-3014]

Fig. 8
NYK Procyon powered by a 12RTA84C engine
[7796-3027]

The RTA96C has already made a good start in the market with orders for eight engines together with a number of options. The most notable are the 12-cylinder engines at the full nominal maximum continuous rating of 89 640 bhp (65 880 kW) at 100 rev/min for four 6674 TEU containerships building for P&O Nedlloyd at Ishikawajima Harima Heavy Industries Co Ltd.

However, use will be made in smaller ships of the very high cylinder output of the RTA96C to install engines with fewer cylinders than is possible with available competing engines. In this respect, two 5300 TEU containerships contracted in Korea by Hanjin Shipping will each be powered by a ten-cylinder engine of 74 700 bhp (54 900 kW) output at 100 rev/min.

Fig. 9
The 5364 TEU Post-Panamax containership *Ever Ultra* powered by a 12RTA84C engine. Berthed ahead of her in Kaohsiung is the 2728 TEU *Ever Grace* of 1984 which has a Sulzer 6RLB90 main engine
[7797-3027]

Engine parameters

The RTA84C engine has been on the market in its current form since 1993. Its basic parameters (Table II) were developed from those of the RTA84 originally introduced in December 1981. When the RTA84C was upgraded in 1993, its output was increased but without exceeding basic parameters already employed in other existing RTA engines to safeguard the best basis for reliability and durability in the containership application (Fig. 10).

Since 1993, the RTA84C has had a maximum continuous rating (MCR) of 5510 bhp/cylinder (4050 kW) at 102 rev/min, corresponding to a brake mean effective pressure (BMEP) of 17.91 bar at 8.16 m/s mean piston speed. The result was six per cent more power with four per cent higher BMEP and two per cent higher mean piston speed. Thus, in models with four to 12 cylinders, the RTA84C today offers MCR power output up to 66 120 bhp (48 600 kW).

The basic engine parameters of the new Sulzer RTA96C engine were selected by carefully analysing the power requirements of the anticipated Post-Panamax containerships. The desired power output of some 90 000 bhp from no more than 12 cylinders led to the choice of a cylinder bore of 960 mm.

The RTA96C piston stroke is a little longer than in the RTA84C (2500 instead of 2400 mm) to enable the combustion chamber to have better proportions. By adopting a longer stroke, the depth of the combustion chamber can be proportionally increased to give more room for obtaining the best combustion and fuel injection parameters, and to obtain better control of temperatures in the combustion chamber components. These all have an influence on engine reliability and times between overhauls. Furthermore, with the slightly longer stroke, the design of the crankshaft is simplified because the shrunk-in main journals do not cut the journal fillets at the inner sides of the crank webs.

The selected BMEP of 18.2 bar is now proven in today's technology for two-stroke engines as a basis for very good reliability. It is already employed in the Sulzer RTA-U engines which have been in successful operation for more than two years. Accordingly, the maximum cylinder pressure was set to a level of 142 bar which is also backed by the relevant service experience of engines in operation.

Naturally, the optimum propeller speed was also the subject of detailed studies in view of the high power that was to be concentrated on a single propeller. It was found that the most favourable speed lies around 100 rev/min and therefore the stroke of the new engine was selected to be 2500 mm to make best use of the mean piston speed of 8.3 m/sec. Mean piston speeds of more than 8.0 m/s are now usual with satisfactory piston-running behaviour. The CMCR speed range of the RTA96C was set at 90–100 rev/min to give sufficient flexibility for matching the individual propeller and ship characteristics.

Fig. 10
Key parameters of the RTA84C and RTA96C shown in the context of the evolution of parameters of the Sulzer RTA engine series over the past 15 years. The advanced parameters of the 'Technology Demonstrator' (4RTX54) are also indicated [97#212]

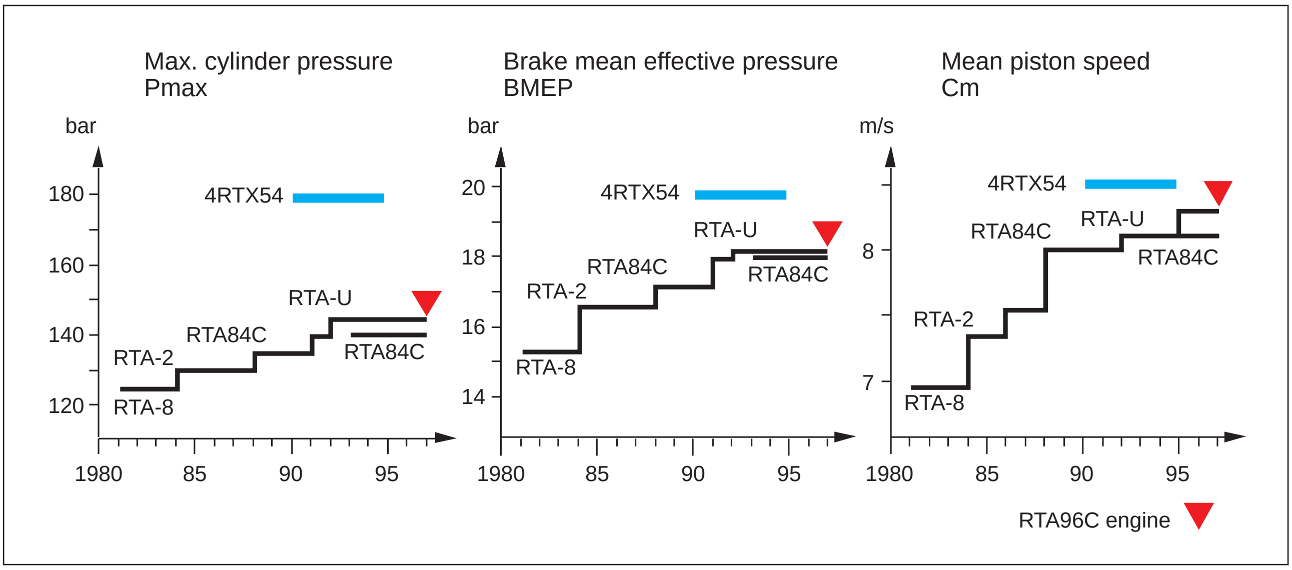


Table II: Main parameters for RTA-series containership engines

Engine type		RTA84	RTA84	RTA84C	RTA84C	RTA96C
Year introduced		1981	1988	1988	1993	1994
Cylinder bore	mm	840	840	840	840	960
Piston stroke	mm	2400	2400	2400	2400	2500
Stroke/bore ratio	–	2.86	2.86	2.86	2.86	2.6
Power/cylinder, MCR	bhp	4030	4760	5200	5510	7470
	kW	2960	3500	3820	4050	5490
Speed	rev/min	87	95	100	102	100
Mean piston speed	m/sec	6.96	7.6	8.0	8.16	8.33
Brake mean effective pressure	bar	15.35	16.6	17.2	17.91	18.2
Max. cylinder pressure, Pmax	bar	125	130	135	140	142
Brake specific fuel consumption,						
at full load, MCR:	g/bhph	127	126	126	126	126
	g/kWh	173	171	171	171	171

Development goals

The development goals for a large marine engine are many, often interact and in some cases even conflict with each other. The RTA84C and RTA96C are no exceptions. As the RTA84C might now be considered to be an established design, we shall here concentrate on the development goals for the new RTA96C. Apart from the goals coming from the market and application requirements, the following overall goals were defined for the RTA96C:

- To be as similar as possible to the RTA84C but incorporating the latest developments. The overall target was agreed at an early stage that the RTA96C should be at least as attractive as, or more than, its older, more experienced brother, the RTA84C.
- 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;
- Engine tuning to be oriented towards reliability;
- Lowest possible wear rates for cylinder liners and piston rings;
- Reasonably low specific cylinder lubricating oil consumption for good overall costs;
- Low fuel oil consumption;
- To comply with the IMO exhaust emissions regulations expected 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 TBN 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 proposed IMO regulations for the control of NO_X emissions that are expected to be introduced for new ships 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, there may need to be a wider tolerance in fuel consumption for engines that comply with the IMO regulations.

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 RTA96C, this fresh knowledge comes from:
  - The RTX54 'Technology Demonstrator' which was running at high parameter levels for more than 2500 hours in the early 1990s;
  - Combustion tests were simulated on an RTA84C engine on the test bed with a cylinder modified to give a combustion chamber with the same geometric proportions and concentration of fuel sprays as the RTA96C;
  - The latest results gathered from the 4RTA58T engine running since October 1995 in the Diesel Technology Center in Winterthur;
  - The first 6RTA48T, 7RTA48T and 7RTA58T engines built by Diesel United Ltd in Japan, running since the beginning of July 1996 for the first RTA48T, and April 1997 for the RTA58T.

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

Design features of RTA-C engines

The Sulzer RTA84C and thus also the RTA96C follow the well-proven design concept of all RTA-series two-stroke engines (Fig. 2). They are long-stroke uniflow-scavenged crosshead-type engines with a diaphragm and piston-rod gland separating the crankcase from the under-piston space. The single, central exhaust valve discharges into a manifold leading to one or more exhaust gas turbochargers operating on the constant-pressure system maintaining as much kinetic energy as possible. They deliver scavenge air down through coolers to the air manifold that runs the length of the engine, and then to the under-piston spaces and the air ports in the lower part of the liners. Fuel is injected to the cylinder through three fuel valves supplied by fuel injection pumps on the mid-height camshaft that is driven by gears from the aft end of the crankshaft. The fuel pumps and exhaust valve actuating pumps are arranged in blocks along the camshaft, with each block housing the units for one pair of cylinders.

It is imperative when designing engines with large bores such as the RTA-C types to identify and to pay particular attention to the major design priority areas. These are highlighted below.

Engine structure

In engines for containerships, the rigidity of the engine structure takes on a particular importance. They often have high cylinder numbers with nine to 12 cylinders in line to satisfy the high power requirement. Yet these long engine structures still need to have a completely vibration-free behaviour while safely carrying the usual gas forces (firing loads) and all

Key Points

The new RTA96C engine is as similar as possible to the thoroughly well-established RTA84C type. 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:
  - RTA84C: Hydraulic jack bolts
  - RTA96C: Elastic holding down bolts.

Running gear:
- Semi-built crankshaft;
- Main bearing lower halves: white-metal shells;
- Main bearing caps with cast-in white-metal layer;
- 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.

Fuel injection equipment:
- Three fuel-injection valves per cylinder;
- 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;
- Cladding of exhaust valve faces only applied in first RTA96C engines for experience and evaluation;
- Cladding of piston crowns still being evaluated in long-term tests but not anticipated for the RTA96C with its low surface temperatures.

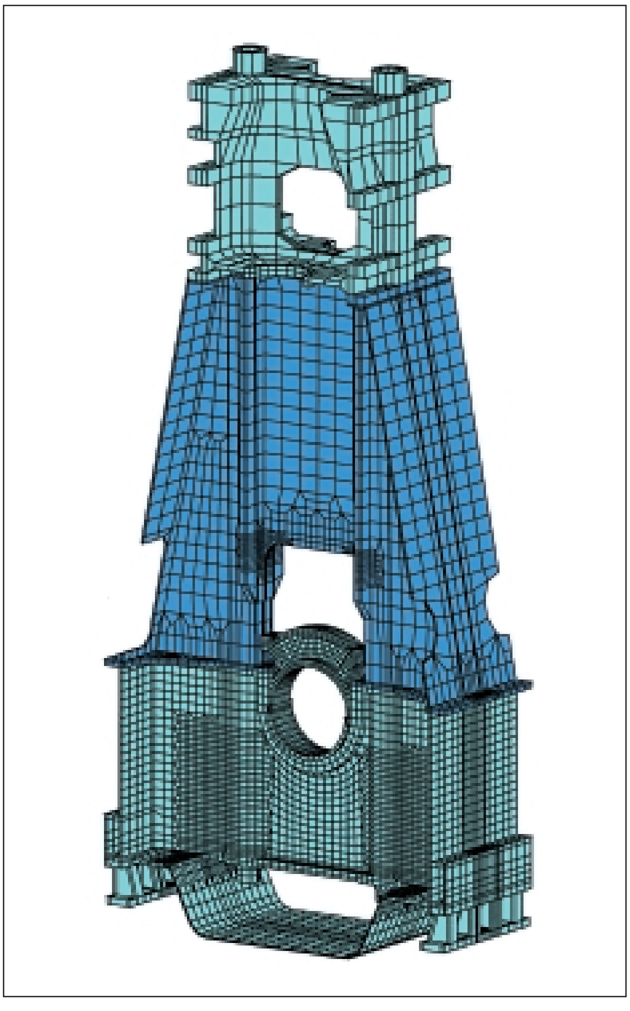
Piston-running behaviour:

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.

the internal forces and bending moments without any local over-stressing. Thus the structure must be carefully designed to give the necessary overall stiffness. At the same time, the basic structure of the engine is also expected to give a long life free of any failure.

The RTA-C engine structures are based on the proven and sturdy designs of former RTA engines. They comprise three main elements (Fig. 11): the welded gondola-type bedplate, the welded box-type columns and the cast-iron cylinder blocks, all bolted together to give a rigid structure. Assembly was made easier for the RTA96C by making all bolts between the columns, and between the columns and bedplate accessible from the outside.

Fig. 11
Finite-element model of the RTA96C engine structure for computer analysis: the ‘gondola’-type bedplate, welded box-type columns and individual cast-iron cylinder blocks [95#291]



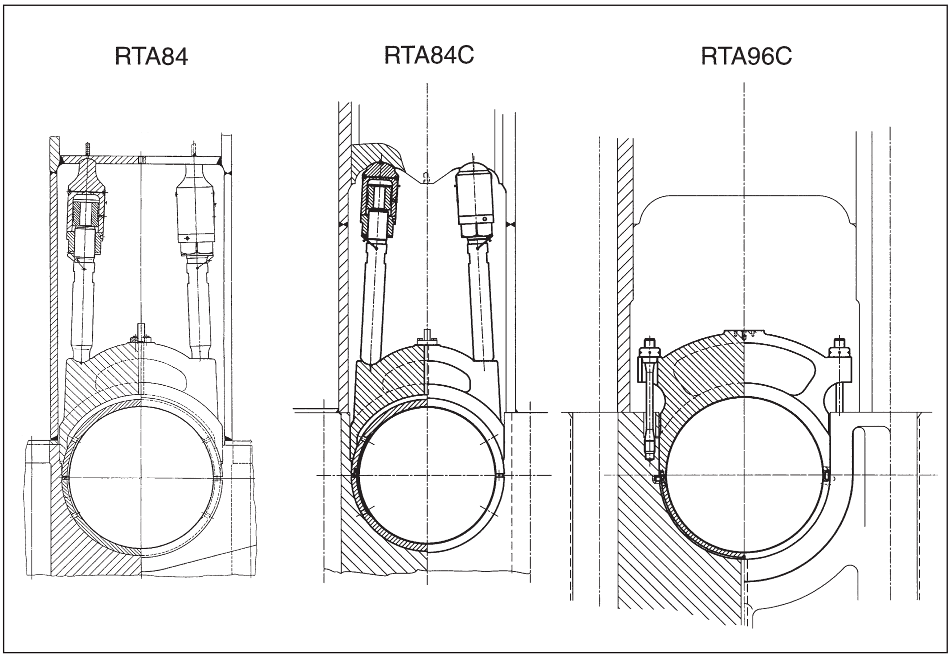
The principle of the design of welded structures for Sulzer RTA engines is to combine the highest rigidity with a low weight. Hence, stiff box-type elements are used for the column walls rather than open structures with many stiffening ribs. The advantage of using two thinner transverse plates, instead of a single thick one, is not only to have a stiffer structure but also to have the advantage of thin and easy-to-weld seams for a high quality standard instead of having the drawback of thick seams that are difficult to weld through to the root.

In the case of the new RTA96C, the opportunity has been taken to introduce simplifications in the whole engine structure assembly. For example, all bolts for connecting bedplate and cylinder block are fitted from the outside. However, careful attention was given in the RTA96C to past experience to eliminate drawbacks of former designs. Stress calculation techniques nowadays are powerful tools for investigating large structures under complicated assumptions, as well as for optimisation of small design details.

To take just one aspect as an example, the column aperture has been the subject of step-by-step improvements (Fig. 12). The aperture of the RTA84C was reviewed to reach a higher safety margin in the upgraded version of 1993. Here a more forgiving design was chosen for the transition between the girder of the crosshead guide and the transverse beam that supports the hydraulic jack bolts of the main bearing caps to allow for a wider scatter in manufacturing quality. This design solution had already been developed for the RTA84T type and the RTA-U series.

The next step is, in the RTA96C, the replacement of the hydraulic jack bolts by simpler elastic holding-down studs. They directly connect the bearing cap to the main bearing girder. This closing of the flow of forces within the column wall results in a very much simplified structure with reduced stress levels. This can improve reliability because, in that area, it often used to be difficult to achieve the required weld quality. However, great attention was given to the calculation of stresses and deformation in the main bearings with their new holding down arrangements. It is very important that the bearing deformation stays within certain limits under tie-rod pretension and engine full-load condition.

Fig. 12
Details of the column construction in three forms: the original RTA84C design (left), the upgraded design (centre) with the improved welding configuration for lower dynamic stresses, and with a further simplified structure in the RTA96C design with elastic holding-down studs instead of hydraulic jack bolts
[97#225]



Improved design for welding

In the RTA96C bedplate, the two transverse walls that connect the bearing girder to the longitudinal walls are now angled to widen the space between them and to improve access when welding the root layer of the welding seams (Fig. 13). In addition, this new arrangement of the transverse walls provides better load transfer and a more even distribution of stresses (Fig. 14).

To facilitate the welding process and to demonstrate the potential improvements, Wärtsilä NSD

Fig. 13
Cross girder of the RTA96C with angled transverse walls
[97#226]

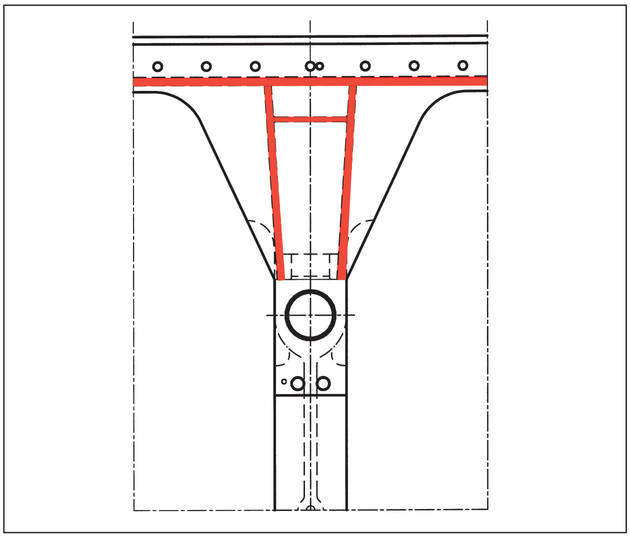


Fig. 14
Finite-element analysis of the cross girder of the RTA96C under full dynamic loading
[7795-3075-2]

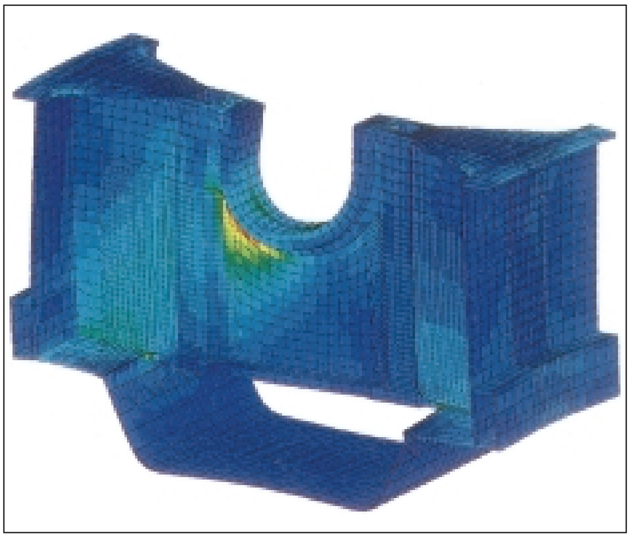
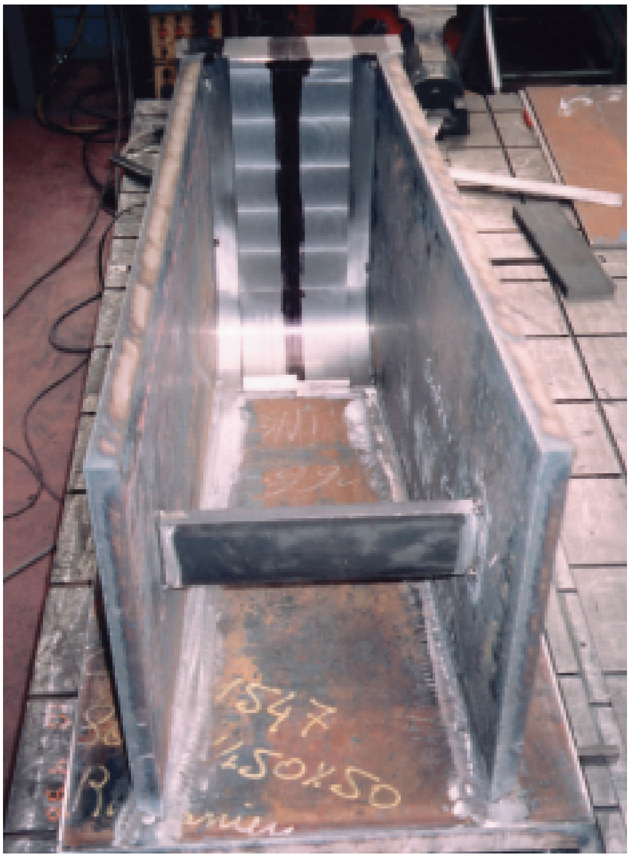


Fig. 15
Full-scale model of the angled transverse walls during welding
[7797-3028]

Fig. 16
Full-scale model of the angled transverse walls after welding
[7797-3029]



carried out welding trials for the transverse girder (Fig. 15). A full-size transverse girder was built in our welding test shop to assess the accessibility of welds which is the most decisive influence on the quality of the welds and thereby also their fatigue resistance (Fig. 16).

The welds were cut out of the test piece after being welded in realistic access conditions, and carefully inspected. The quality achieved was fully within expectations and specifications. It is therefore possible for Wärtsilä NSD to specify the welding procedures for the RTA96C precisely according to the experience made in this trial.

Running gear

Another area of great importance is the running gear, including the crankshaft (Fig. 17), connecting rods, crossheads, pistons, bearings, etc. These components must all function absolutely without trouble throughout the engine's running life.

The heart of the running gear is the crankshaft (Fig. 18). It collects all the power from the individual cylinders. It thus needs to be designed with utmost care. Both RTA-C engines have a semi-built crankshaft comprising combined crank pin/web elements forged from a solid ingot and the journal pins then shrunk into the crank webs.

When the engine BMEP and maximum pressure in the RTA84C were increased in 1993, all the geometrical dimensions could be kept the same as before with only the shrinkfit oversize needing to be adjusted to accommodate the increased torque. It is notable that, during the increases in output achieved by the RTA84C, the specific bearing pressures have been maintained at practically the same values.

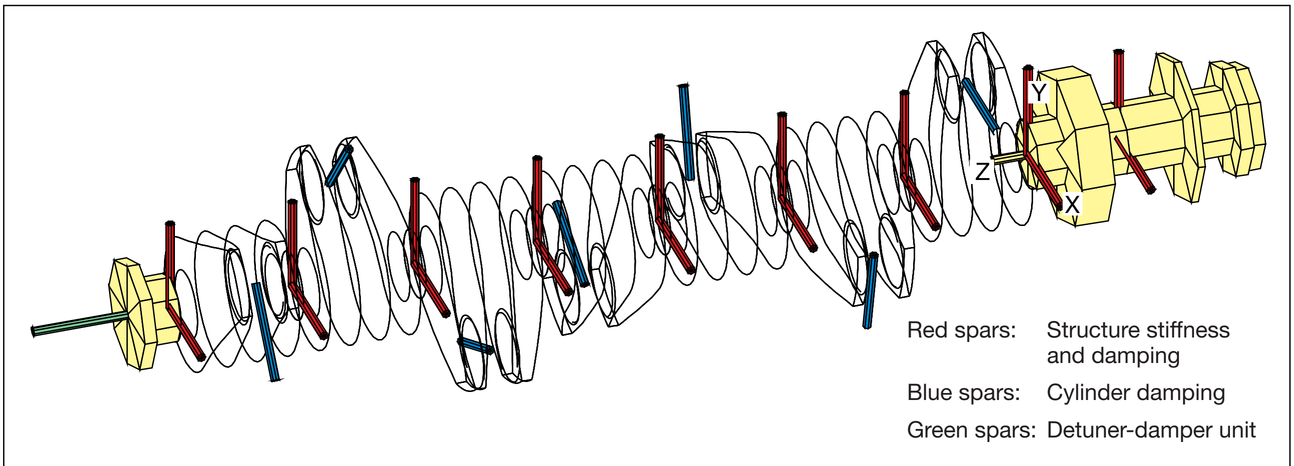
By choosing a stroke of 2500 mm for the RTA96C, it was possible to avoid the intersection of the shrunk-in main journal with the journal fillet in the crank web in spite of the appropriately larger journal diameter (Fig. 19). This allows the use of traditional design techniques for dimensioning the shrink fit (Fig. 20).

For the lower halves of the main bearings, white-metal shells are used in both RTA-C engines as in other RTA engines, whereas the main bearing cap has a cast-in white-metal layer. This solution is regarded as safe and tolerant of particles.

The RTA96C bearings are designed, despite the relatively short cylinder distance, with low 'relevant

Fig. 17
The two-part crankshaft being installed in the bed-plate of the first 11RTA96C [7797-3030]

Fig. 18 (below)
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 [97#227]



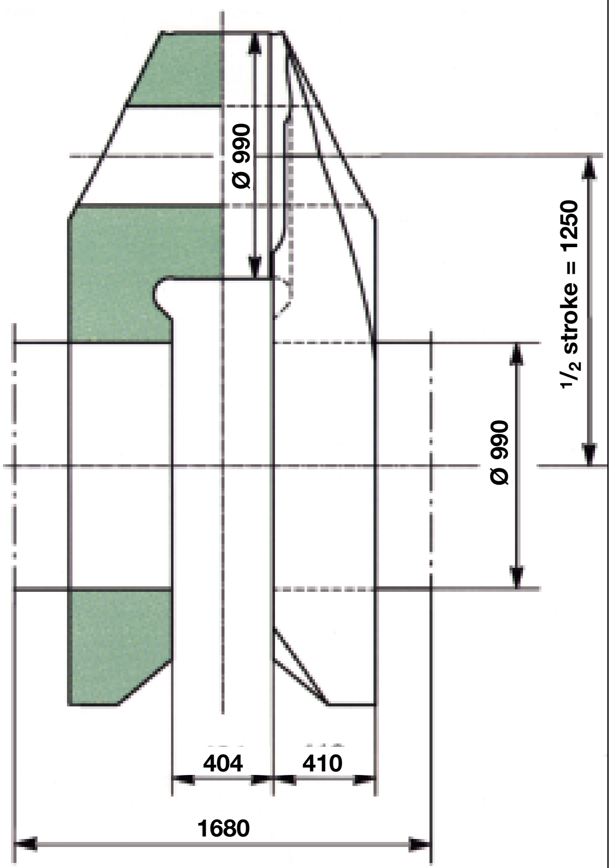


Fig. 19
Crank throw of the RTA96C. Dimensions in millimetres
[95#228]

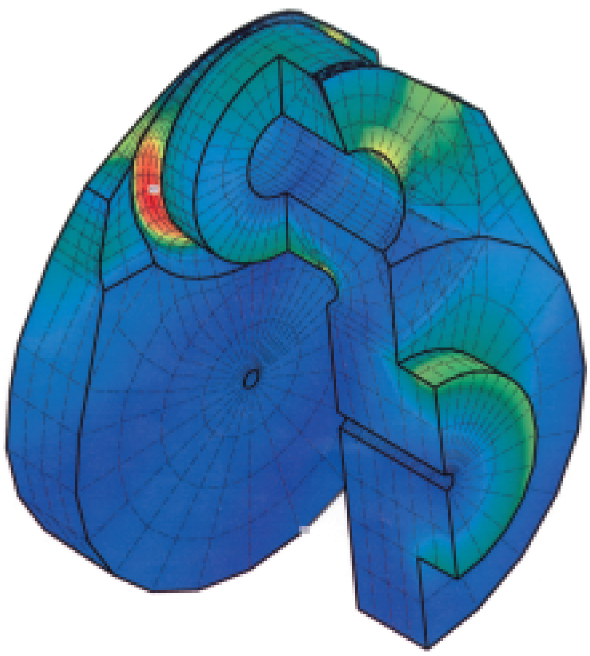


Fig. 20
Finite-element analysis of the crank throw of the RTA96C under full dynamic loading
[7796-3031]

Fig. 21
Relevant bearing loads: Main bearing. WM = white metal
[97#229]

type	RTA84M	RTA84C	RTA84T	RTA84C	RTA96C
year	1985	1988	1991	1993	1995
bhp/cyl	4700	5200	5280	5510	7470
P_{max}bar	130	135	140	140	142

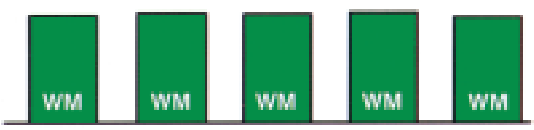
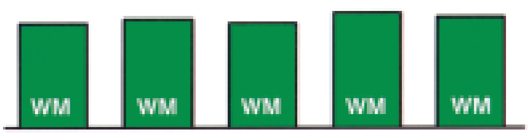


Fig. 22
Relevant bearing loads: Bottom-end bearing. WM = white metal
[97#230]

type	RTA84M	RTA84C	RTA84T	RTA84C	RTA96C
year	1985	1988	1991	1993	1995
bhp/cyl	4700	5200	5280	5510	7470
P_{max}bar	130	135	140	140	142



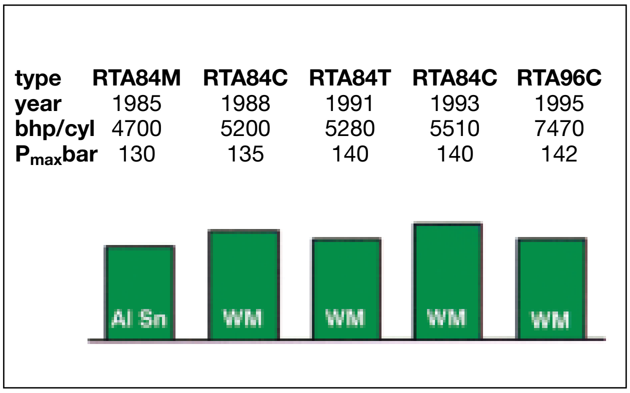
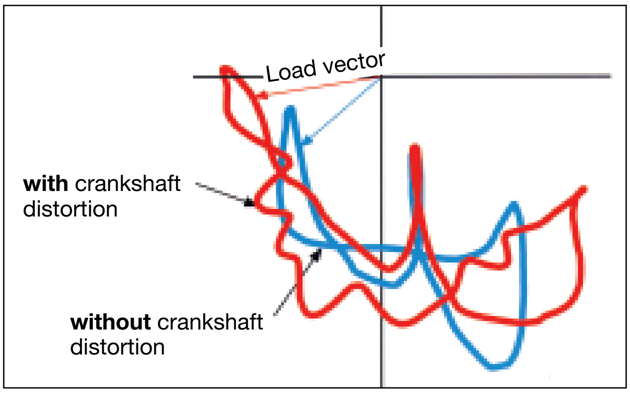


Fig. 23
Relevant bearing loads: Crosshead bearing
WM = white metal with galvanic overlay
AlSn = aluminium-tin alloy with galvanic overlay
[97#231]

Fig. 24
Load diagrams for the main bearing with and without crankshaft distortion taken into consideration during the calculations
[97#232]



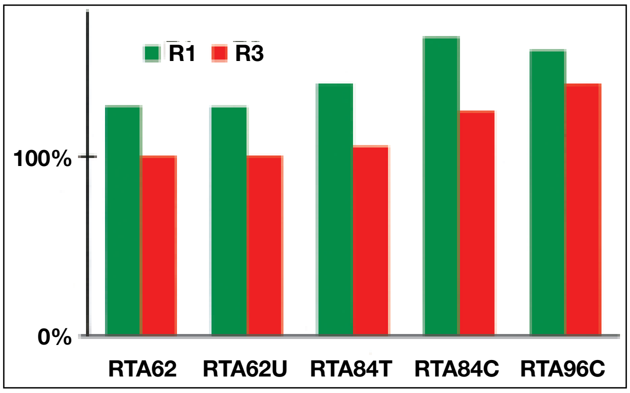
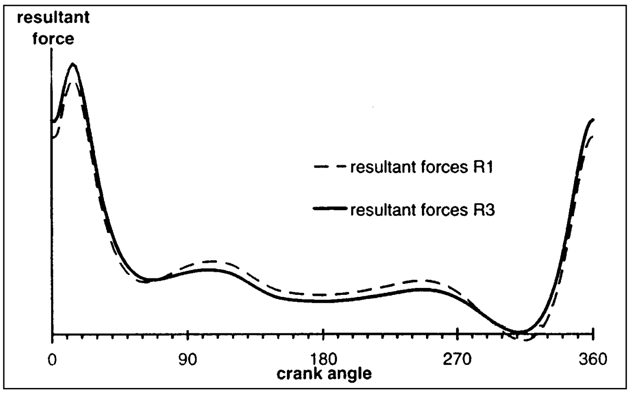


Fig. 25
Minimum oil film thickness for crosshead bearings
[97#233]

Fig. 26
Load in the RTA96C crosshead bearing during the engine cycle
[97#234]



bearing loads' as in the RTA84C to assure very good bearing behaviour (Figs. 21, 22 and 23). The so-called 'relevant bearing load' is a combination of the most important bearing behaviour factors, namely the minimum oil film thickness, the pressure gradient in the oil film, the maximum local pressure in the oil film and the specific pressure of the bearing which, in previous comparisons, used to be the only factor considered. To be able to compare the 'relevant loading' of the new bearing, this figure was also calculated for other RTA engines to assure safe transformation of this figure into real service behaviour of the bearings. The 100 per cent figure is always related to the engine found to be the highest loaded in the bearing considered and, at the same time, showing excellent bearing performance in actual operation.

Today's finite-element calculation techniques give very accurate calculated results for main bearing loadings. They use a full model together with the corresponding sub-structure and enable the boundary condition to be taken into consideration by incorporating the crankshaft stiffness into the calculation of the main bearing loadings. Depending on the crank angle between two neighbouring cylinders and mainly on the firing order sequence, the load vector can vary considerably. The influence of crankshaft stiffness can be seen in figure 24.

The RTA-C crossheads are designed according to the design principle used for other RTA engines and also feature a full-width 'bathtub' lower half bearing. The crosshead bearings of both RTA84C and RTA96C types have thin-walled shells of white metal for high load-bearing capacity.

Sulzer engines retain the use of a separate higher-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 principal design criteria is that the hydrostatic force created by oil pressure will lift the crosshead pin off the shell during each revolution to leave enough oil film thickness under the gas load (Figs. 25 and 26). This is achieved in the RTA96C even with its greater gas load. The better lift is assured because the RTA96C runs at 100 rev/min at the R1 rating and has a narrow speed-derating field with 90 rev/min at the R3 rating point, combined with the fact that the connecting rod is relatively shortened compared with the RTA84C.

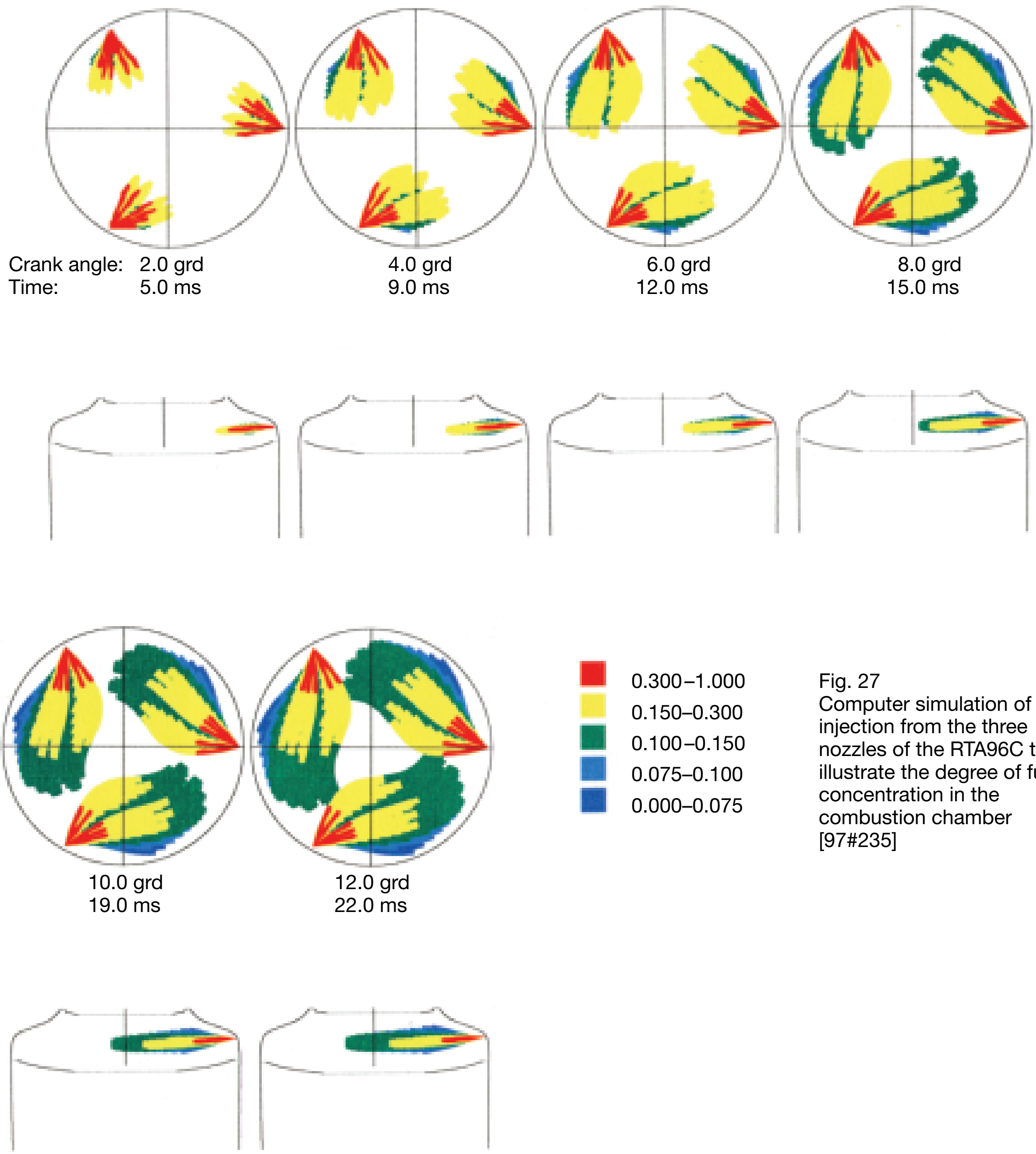


Fig. 27
Computer simulation of fuel injection from the three nozzles of the RTA96C to illustrate the degree of fuel concentration in the combustion chamber [97#235]

Camshaft, its drive and fuel injection equipment

The design of the RTA84C camshaft with its gear drive and fuel pumps is unchanged since the first engines, and the same concept is followed for the larger-bore engine. There was an improvement, however, to three fuel injection valves in each cylinder in the upgraded RTA84C which has been continued in the RTA96C (Figs. 27 and 28). In line with their adaptation throughout the RTA series in the mid 1990s, two-piece fuel nozzles were also introduced into the upgraded RTA84C instead of the previous shrink-fit nozzle, as it has been recognised that the life of the nozzle body is considerably longer than that of the nozzle tip. Similar two-piece uncooled nozzles are employed in the RTA96C. In both cases, the nozzle tips are of Stellite 6 material and have given excellent reliability results throughout the RTA series.

Part-load fuel consumption is minimised in the RTA-C engines by the use of variable injection timing (VIT) which has been employed for many years in Sulzer two-stroke engines. 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 mechanical VIT system.

Combustion chamber components

When designing the RTA96C, with its very large bore, the combustion chamber was recognised from the outset as the most important design area because of its major influence on the reliability of the engine. The principal reason is the high power concentration (in other words, the amount of fuel injected) in its cylinders. The RTA96C combustion chamber components, however, are firmly based on the well-proven designs in the RTA84C and other RTA engines while incorporating all the latest knowledge and experience (Fig. 29).

Although the cylinder cover was substantially unchanged in the upgrading of the RTA84C, the material specification was changed to enlarge the

Fig. 28
Analysis of fuel distribution and injection trajectories in the RTA96C cylinder. Colours indicate concentration with blue/green for the stoichiometric mixture. No combustion calculated
[97#236]

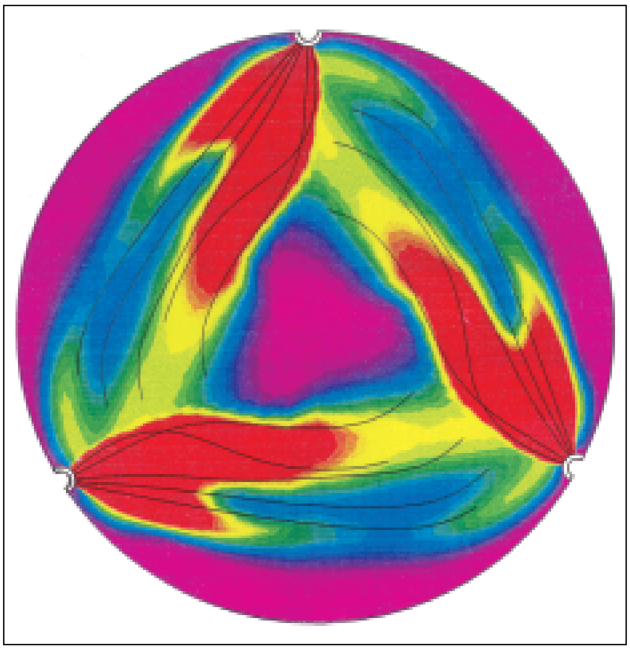
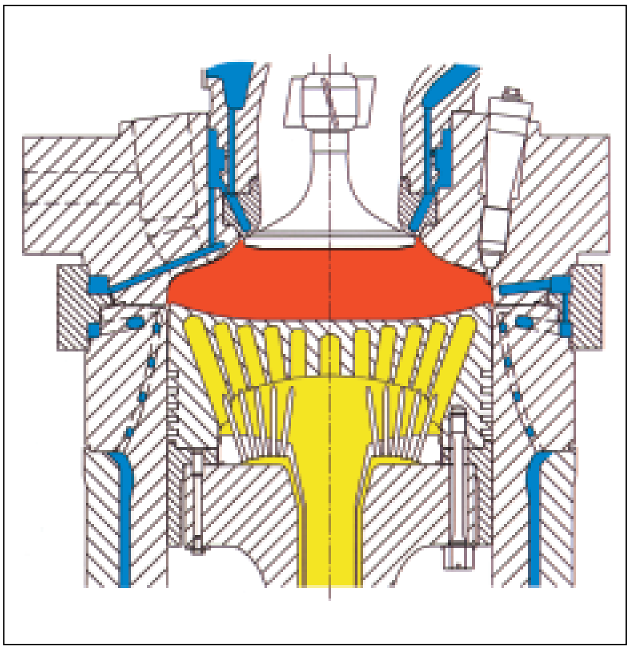


Fig. 29
The combustion chamber of the RTA96C is fully bore cooled
[97#237]



safety margin against low-cycle fatigue, and consequently also against corrosion fatigue, so that it is well above the factor that would be needed to cope with the increased power. The same material is used for the RTA96C cover.

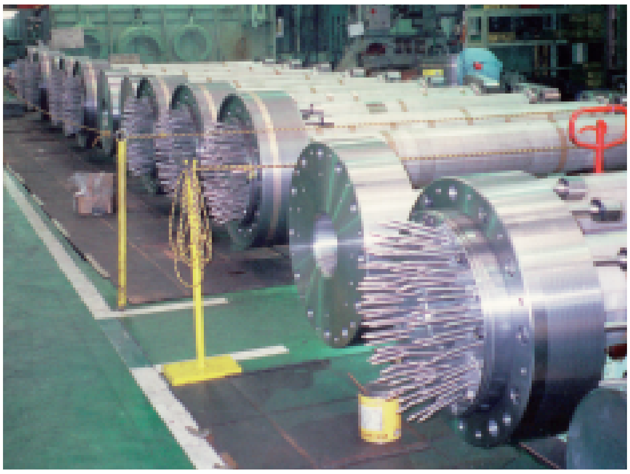
The RTX54 'Technology Demonstrator', with its ability to run at elevated parameter levels, added considerably to the knowledge of combustion chamber design. For example, the change from two to three fuel injection nozzles when the RTA84C was upgraded in 1993 arose from the positive effects of such a change that were observed in the RTX54 tests. It was seen both to improve fuel consumption and to give a more even temperature distribution around the combustion chamber components. This result was confirmed in the RTA84T and the RTA-U engines which, despite their increased BMEP of 18 bar and 18.2 bar respectively, also showed constant or even reduced component temperatures with the three-nozzle configuration.

The use of bore-cooling technology provides an escape from the rule that larger components (with reference to bore), when subjected to thermal loading, will also have higher thermal strains. Even in the RTA96C, the thermal strains in the cylinder cover, cylinder liner and piston crown could thus be kept fully within the values in previous generations of RTA engines. This also applies to the mechanical stresses in these components.

The comparatively short stroke/bore ratio of the RTA96C of about 2.60 compared to 2.86 of the RTA84C gives the new engine a rather shallow combustion chamber. Ways and means were thus sought to overcome the problem of injecting much more fuel

Fig. 30
Bore cooled pistons for the RTA96C
[7797-3032]

Fig. 31
RTA96C piston rods with cooling nozzles
[7797-3033]

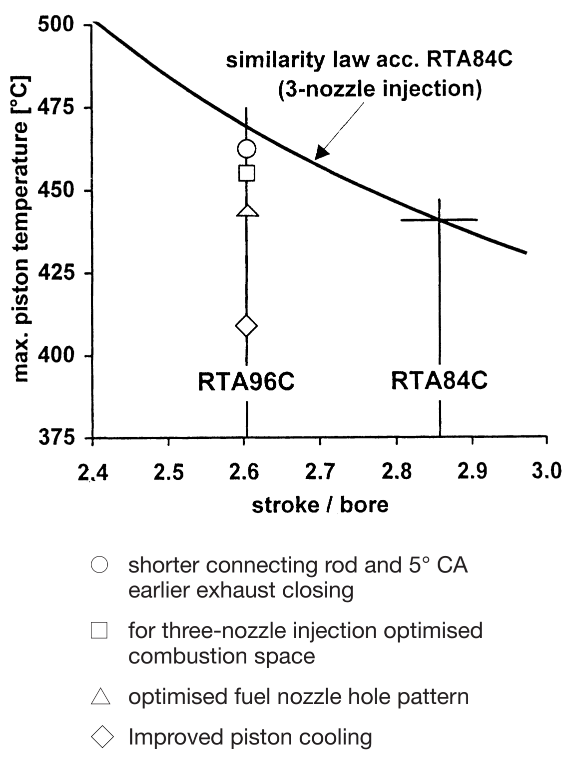


into a combustion chamber of shallower depth without any danger of the fuel flames impinging on component surfaces. Calculations based on previous measurements on engines with various stroke/bore ratios showed that, for example, if the RTA96C had been designed according to the law of similarity its piston surface temperature might be about 30 °C higher than in the RTA84C today, which might be above the limits for trouble-free operation (Fig. 32).

By variation of the effective compression ratio through earlier closing of the exhaust valve, however, the piston position could be lowered. The gain in depth, together with the use of three fuel injection valves and accordingly optimised shape of the piston crown, gave more freedom to adjust the fuel injection. Finally, improved piston cooling was also introduced, resulting in a further reduction of the piston crown temperatures. The temperature values reached in the RTA96C by these measures are actually lower than the temperatures in today's RTA84C, and therefore are fully expected to assure just as good, or better, performance for the RTA96C pistons in service.

An important contribution to achieving low surface temperatures in the piston crown was the improved cooling effect obtained through a better understanding of the combined jet-shaker cooling process that is employed in Sulzer RTA pistons (Figs. 30, 31 and 32). Cooling oil is directed as jets in to the cooling bores in the underside of the piston crown then it lingers in the cooling space of the crown before draining out. At certain stages in the engine cycle, cooling is predominantly by the jets acting in the bores whereas, at other stages, the shaker effect of oil in the cooling space and bores in the crown is more effective (Fig. 33). Understanding these cooling processes through theoretical investigations and experiments on a test rig has done much to enable the piston crown to be designed for the low temperatures necessary to avoid burning of the surface.

Fig. 32
Calculations show how the maximum piston crown temperature could be reduced cumulatively by various measures in the RTA96C from what it would be according to the law of similarity from the RTA84C with three-nozzle injection
[97#238]



Piston at TDC: Shaker cooling

Piston at BDC: Jet-cooling

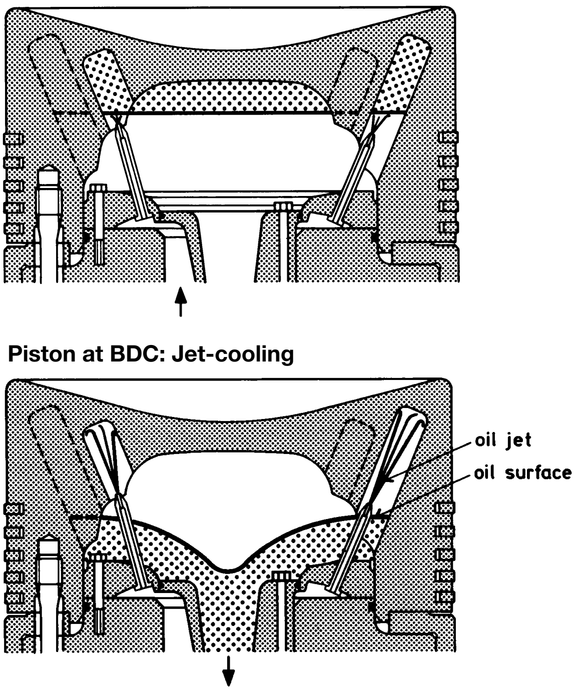


Fig. 33
Operating principles of jet-shaker cooling (drawing based on RTA-U engine)
[0789-3039-1/-2]

These low temperatures were achieved by optimising the oil cooling spray nozzles in terms of oil quantity and jet speed and direction, as well as by optimising the drainage arrangements for the cooling oil. The parameters have been adjusted to achieve the highest heat transfer possible by turbulent flow in an adequately shaped cooling bore geometry to fulfil both thermal and mechanical demands (Fig. 34).

Prototype testing for thermal loads

To obtain early real test results about the thermal load of a relatively shallow combustion chamber with a high power concentration, such as exists in the RTA96C, an RTA84C engine running at the Aioi works of Diesel United Ltd was modified to simulate the geometric proportions of the combustion chamber and the concentration of fuel sprays in the RTA96C (Fig. 35). The thermal load envisaged for the RTA96C could be derived from the temperature readings on the test engine components.

By finite-element calculation techniques, this thermal load distribution was imposed on the envisaged RTA96C combustion chamber. In the case of the

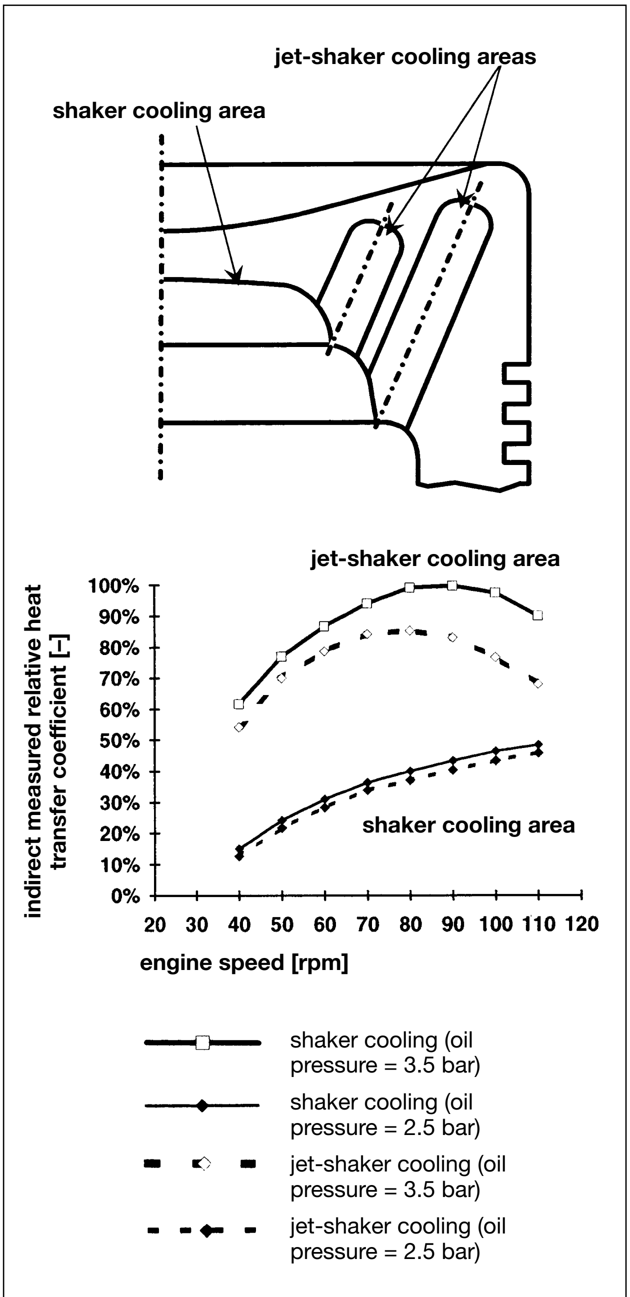


Fig. 34
Relative heat transfer coefficient measured from an oil-cooled piston crown based on an RTA-U engine at different oil pressures and in the shaker cooling area and the jet-shaker cooling area
[97#113-1/-2]

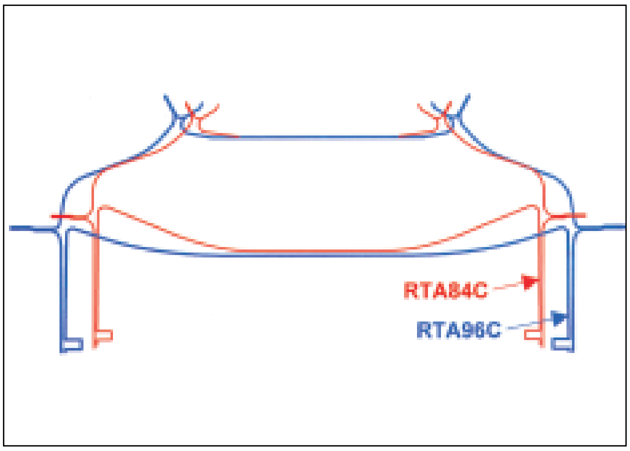


Fig. 35
Comparison of the shapes of the combustion chambers of the RTA84C and RTA96C engines to the same scale
[97#239]

piston design, it could be proved that, after slightly adapting the shape, number and depth of the cooling bores, as well as the intensity of the jet cooling, the temperatures on the combustion chamber surface of the piston crown were more evenly distributed and even slightly lower in value than in the RTA84C.

Component cladding

Even before the above adjustment of component temperatures in the RTA96C, an extensive research programme was started to investigate surface coatings for the protection of thermally-exposed components against hot corrosive or erosive attack within the long service intervals expected.

Cylinder covers of RTA-C engines are protected, as standard, against corrosion/erosion in the vicinity of the fuel injection valves by a welded cladding of corrosion-resistant material downstream of the injector tips.

In the case of the exhaust valve of the first RTA96C engines, it was further decided to protect its surface by a cladding to create more freedom in choosing fuel injection hole patterns to protect the surface of the piston crown. Protection of the exhaust valve surface is also more logical, because the thermal strains between the base material and cladding can be minimised by using nickel-based alloys as cladding material so that both the base and cladding materials have about the same coefficient of thermal expansion.

After evaluating the test measurements from the first 11- and 12-cylinder RTA96C engines, however, it will be discussed whether to omit the valve cladding as standard because the measured temperatures are very moderate, below 600 °C which will give a very satisfactory long life.

An investigation was also initiated to determine the best possible cladding material for the piston crown, by tests on pistons of the 8RTA84C engine of

Key Points

To summarise the design features for the RTA96C combustion chamber, four ways were simultaneously followed to improve the engine thermal load, based on experience from former RTA engine types:

- Adapted combustion chamber shape and further improved design of cooling spray nozzles and cooling bore geometry for piston crown, cylinder cover and cylinder liner.
- Carefully selected number of fuel injectors whose spray geometry is optimised as much for reliability as for low fuel consumption to give low surface temperatures.
- Engine tuning, in terms of turbocharger matching, scavenging port arrangement, valve timing and compression ratio, was similarly oriented towards reliability. It combined a low piston position with high air excess given by high scavenging pressure and early valve closing.
- Short connecting rod length with its corresponding effect on the piston movement.

the containership *Nedlloyd Asia*. The result after nearly 5000 running hours showed the benefit of the different claddings compared with the base material. It can be stated that the rate of material loss can be reduced to at least half or to even one third that of 'standard' material. The cladding therefore allows longer times between overhauls.

Plating of the piston crown was also considered for the RTA96C. It has to be stated that, by applying a cladding of 3 mm thickness onto the piston crown, the thermal strains during loading or unloading of the engine (low-cycle fatigue) are increased by about 30 per cent. Therefore, any possible cladding would need to be as thin as possible. This also means that the corrosion and erosion resistance has to be optimum to extend the life of the crown. Until now, there has been some unsatisfactory experience with plated piston crowns; for example, on an 8RTA84C engine after 17 700 running hours with the pistons in a high position to suit the engine rating point (Fig. 36).

Fig. 36
Piston crown with test cladding from an 8RTA84C showing cracks in the cladding after 17 700 hours' operation
[97#240]

Therefore, new tests were started, and revealed the potential of nickel-chromium alloys which reached 7000 running hours with only minor attack; for example, on the 8RTA84C of the *Nedlloyd America* (Fig. 37). It has to be said that these tests need to accumulate as many running hours as possible before any judgement can be made on the corrosion and erosion resistance of the cladding, as well as on its behaviour in low-cycle fatigue.

These cladding measure are of a provisional character and it is yet to be decided whether they will eventually be partially or fully introduced for future RTA-C engines. Today, piston crown cladding is unnecessary as the temperatures are sufficiently low in service to avoid material loss. This policy is based on the fact that the welding process used in cladding is itself very sensitive and, if the plating is not carried out with utmost care, the cladding material tends to crack under low-cycle fatigue thereby causing a worse situation than when material simply burns off.

Piston-running behaviour

Piston running is still one of the most important issues for the success of an engine in service. Wärtsilä NSD has been working hard on these difficult problems in recent years. A considerable quantity of evidence has been gathered, especially on large containerships equipped with RTA84C type engines. The available evidence demonstrates that most of the problems of the past are solved; this is a fact to which many owners agree.

Both the RTA84C and RTA96C incorporate all the well-proven features for good piston ring and cylinder liner behaviour used in all RTA-series engines built today (Fig. 38), namely:

- 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 (Fig. 39).

In addition, the RTA-C engines have extra features for good piston-running behaviour, namely:

- Top piston rings are plasma coated. It gives added safety for the running-in of new liners, as well as assured low wear rates for the piston ring and liner during subsequent operation of the engine.
- Three fuel injectors per cylinder, as noted above under 'Combustion chamber components'.

Reliability in two-stroke marine diesel engines is usually seen as synonymous with the times between overhauls (TBO) of major components of the engine. In the case of the RTA96C, the TBO was set to be at least two years, in other words more than 15 000

Fig. 37
Piston crown with a later test cladding from an 8RTA84C showing only minor attack after 6981 hours' operation
[97#241]

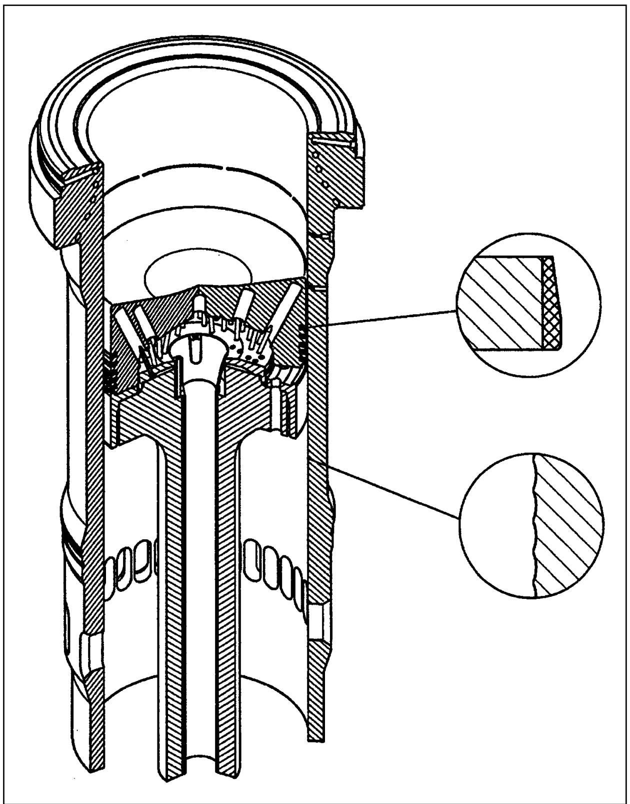


Fig. 38
Cylinder liner with multi-level lubrication, also showing the profile of the plasma-coated top piston ring (upper inset) and the surface profile of the liner
[93#143]

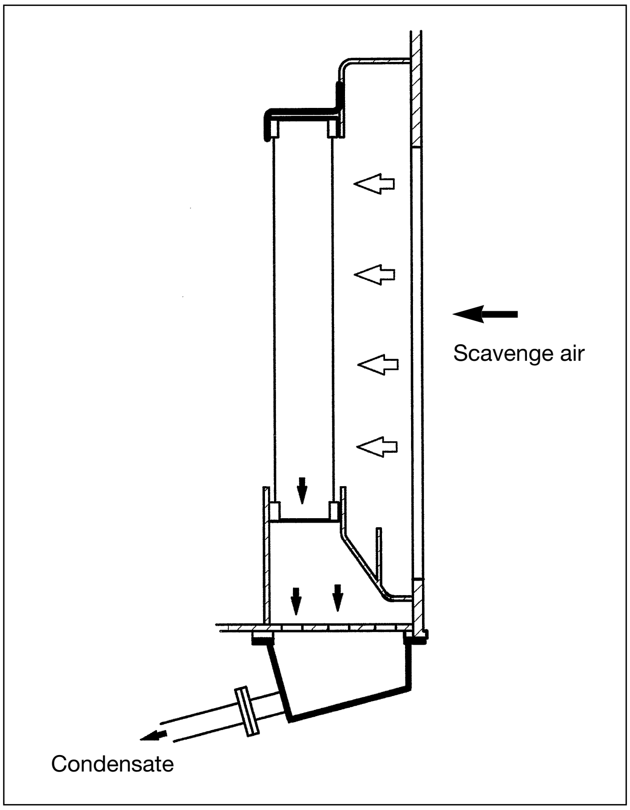


Fig. 39
Detail of the revised design of condensed water separator located after the scavenge air cooler, showing the higher position of the separator and the improved drain for condensate
[97#078]

hours' running as is already being surpassed by RTA84C engines. This TBO is equivalent to more than two years between the changing of piston rings (thus between the pulling of pistons), and therefore an easy and welcome matching of maintenance work for the engine and ship. The technology applied to reach this goal is already proven from the previous generations of RTA engines, finely tuned over the years and with the benefit of lessons learned from service experience.

Many parameters affect the piston-running behaviour, with some being controlled by the engine designer, some by the engine builder and subcontractors, and others by the operator. Trouble-free operation over a long service period can only be assured through close co-operation of all these partners. If any single link of the chain fails, the common success is in jeopardy.

The cylinder liner of the RTA84C has given excellent service results over the years with respect to tribology and strength, and a similar design is used in the RTA96C. In both cases, it remains crucial to use a cast iron with a high degree of ductility to cope with the thermal stress and at the same time to have the necessary properties to withstand tribological wear and tear in service. The liner is thus preferably of die-cast iron, featuring the necessary amount of wear-resistant hard-phase particles on the running surface and a smoothly machined and fully honed surface for quick and trouble-free running in.

Around the combustion chamber, all the main components are bore cooled to give low thermal strains and the smallest, circumferentially-symmetric deformations for good sealing between piston rings and liner.

In the liners, particular attention is given to the geometry of the cooling bores which are adapted to give the appropriate running-surface temperature distribution that is absolutely essential for corrosion-free running of the liner and piston rings. Additionally, the circumferential distribution of liner temperature benefits from using three fuel injection valves, with less variation and absence of areas with temperature peaks.

Standard stock liners are used for each RTA-C type. Adaptation of the liner temperature in the upper part of the piston stroke to the respective CMCR rating is made by inserting insulating tubes of appropriate length into the appropriate position in the liner's cooling bores, as in previous RTA engines.

The Sulzer multi-level accumulator system for cylinder lubrication has proved particularly beneficial to good piston-running behaviour. The cylinder oil is fed to the liner surface at two levels of quills. This gives an ideal distribution of the precious cylinder lubricating oil and is therefore considered to be efficient and economical. The lubricating oil pumps are driven by a frequency-controlled electric motor and the oil is distributed to the quill accumulators by oil distributors. With this solution, much less piping is required. The cylinder oil feed rate is controlled according to the engine load. Adjustments depending on the engine condition and for running-in can be easily made by using software in the engine control system.

Both RTA-C engines have five piston rings. The top piston ring is plasma coated to give the lowest wear rate to reach the goal of two-years' TBO with sufficient margin. The top piston ring also has an stronger base material. Gas-tight top rings are still being evaluated and tested.

Great care is dedicated to the removal of humidity from the scavenge air. Highly-efficient water separators are arranged after the scavenge air coolers with sufficiently large drains to assure that no water enters the cylinders with the danger of destroying the lubricating oil film on the cylinder liners.

Wärtsilä NSD has learned hard lessons from numerous test vessels running in service as part of the company's field testing and research programme. This important feedback is continuously transformed into an increasing amount of knowledge so that times between overhauls of up to two years are realistically expected for the RTA96C engine. The success of many RTA84C engines that are now running very well with low wear figures gives confidence that the RTA96C engine will reach similar TBOs.

Improved piston rod gland

Problems around the piston rod gland, such as excessive system oil losses, its contamination or heavy piston rod wear, are inconvenient if they occur. In most cases, such troubles start when the engine is apparently operating normally but with disturbed piston-running behaviour. The two phenomena are strongly interdependent.

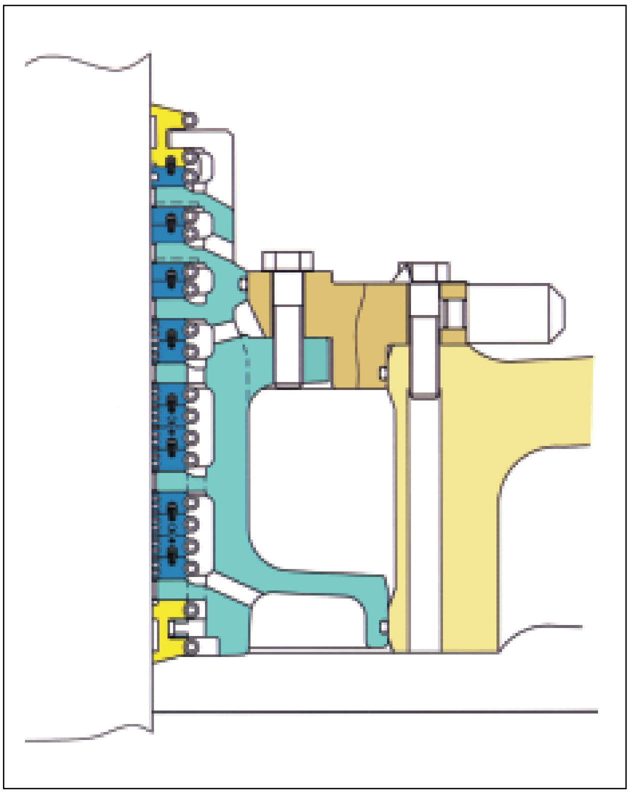
In an effort to solve the problems in this area, a new design of gland has been introduced (Fig. 40). The first examples have already been running trials for more than 10 000 hours in service. The main targets for the new gland design are:

- Minimal system oil consumption;
- Low leakage from the gland's neutral space; predominantly system oil and thus recyclable;
- No system oil contamination by particles and insolubles from the piston underside space;
- Almost non-existent increase of base number of system oil;
- Two-way dismounting, upwards and downwards.

The design modifications incorporated in the new piston rod gland comprise:

- Additional gas-tight top scraper package with a larger drain area;
- Stronger springs;
- Modified position for the neutral space;

Fig. 40
New piston rod gland
[97#242]



- Modified channel drains;
- Ring materials are bronze, with Teflon as the second standard;
- Hardened piston rods must be employed for the best results.

The above piston rod gland will be made available for all new engines as the new standard design. Retrofit solutions will also be made available for RTA84C engines already in service. These will use as many parts as possible from the glands already in the engines.

Maintenance aspects

Key Points

Maintenance work in the RTA-C engines is reduced and facilitated by several general principles:

- Long times between overhauls;
- High reliability, allowing dependable planning of work;
- New, smaller and lighter hydraulic tools on the RTA96C;
- Unified level of hydraulic pressure of 1000 bar on the RTA96C;
- Only eight holding-down studs on the cylinder cover;
- Access to the crankcase continues to be possible from both sides of the engine;
- The new piston rod gland box design allows a simpler dismantling procedure, allowing withdrawal either upwards or downwards.

Fig. 41
The guiding tool is bolted to the crosshead guides to serve as guide rails which ensure that the top end of the connecting rod and its bolts clear the crosshead pin as the connecting rod is lowered for inspection of the pin and its bearing, or during removal of the connecting rod [97#146]

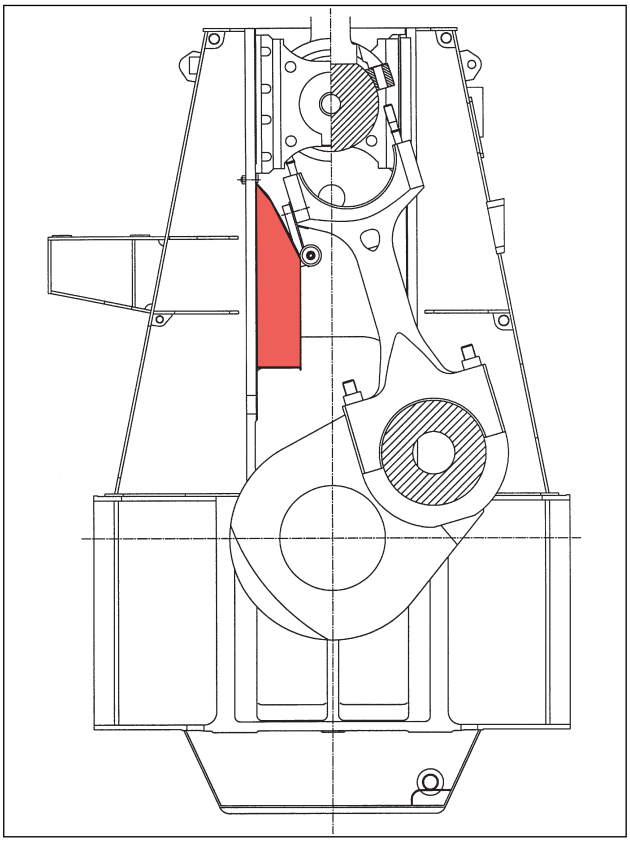
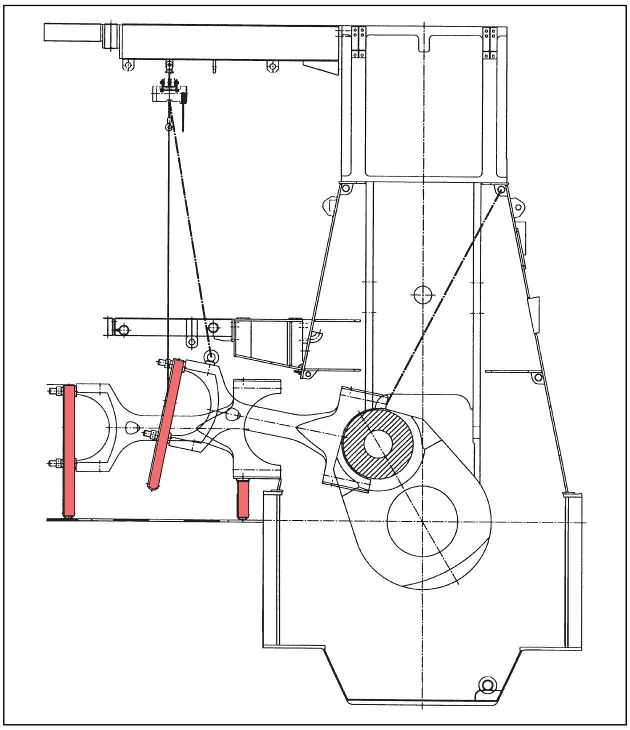


Fig. 42
If the connecting rod of the RTA96C engine is to be removed from the engine crankcase, the task is facilitated by trolley frames that can be fixed to each end of the rod. The connecting rod is lowered using the guiding tool in figure 41 until, at a certain crank angle, the connecting rod can be tilted into a completely horizontal position and finally be handled by the engine room crane. The wheeled trolley frames guide the rod as it is extracted from the crankcase and enable the rod to be wheeled along the floor plates as required
[97#147]



It is important for maintenance that the times between overhauls (TBO) are as long as practicable. In the case of marine diesel engines, at the present state-of-the-art they should operate trouble-free with only one intermediate inspection between classification surveys. This implies TBO of more than two years' operation, or around 15 000 hours' running, and gives ship operators more freedom to arrange the maintenance work to suit ships' sailing schedules. It has a considerable influence on the design of items such as the exhaust valves, cylinder liners, piston rings, bearings and piston rod glands.

The technical staffs of shipowners can make important contributions to design details that affect engine operation and maintenance. Where possible, their views are taken into consideration before engine designs are released for manufacture. One example of such collaboration is the arrangements for inspection of crosshead bearings and the removal of connecting rods on the new RTA96C engine. Special care was needed because of the heavy weights involved. Discussion between Wärtsilä NSD designers and the customer's technical staff resulted in a simple guide tool that is bolted to the crosshead guides to ensure that the top end of the connecting rod and its bolts clear the crosshead pin as the connecting rod is lowered using the turning gear (Fig. 41). Patents for this guide tool are pending.

Should a connecting rod need to be withdrawn completely from the crankcase then the task will be easier with the aid of trolley frames that can be fixed to each end of the rod (Fig. 42). The wheeled trolley frames guide the rod as it is extracted from the crankcase and enable the rod to be wheeled along the floor plates as required.

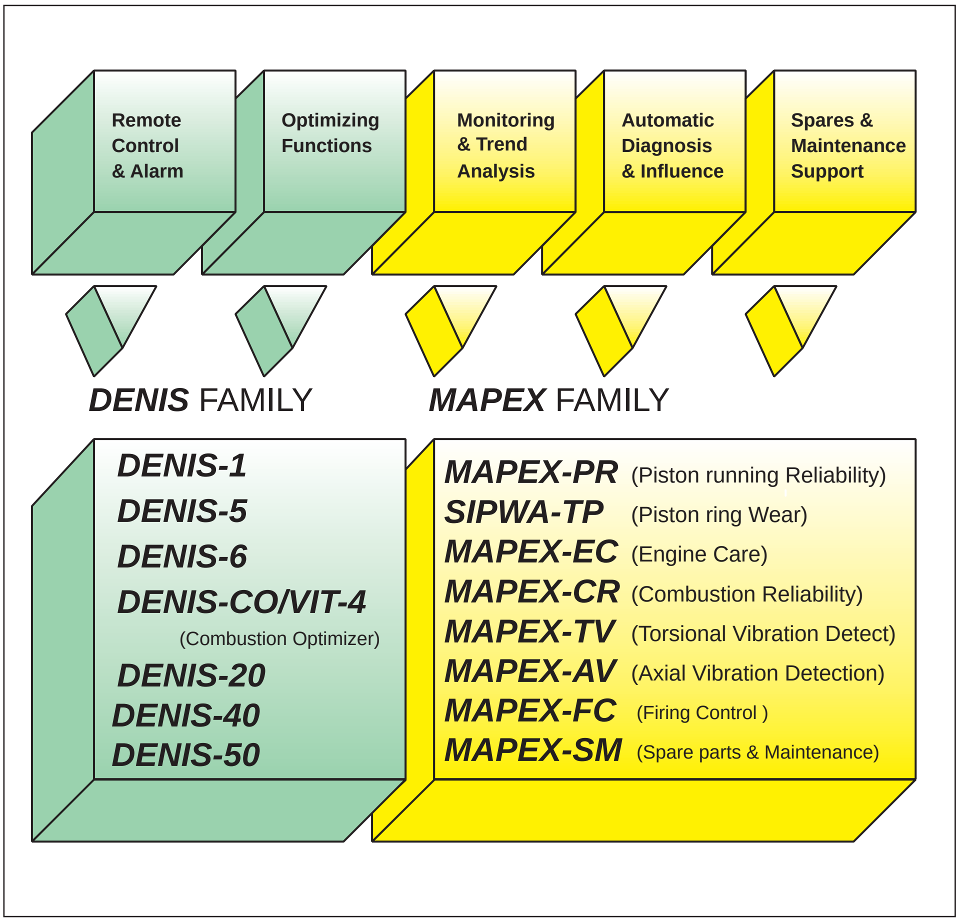
Engine management systems

Key Points
The engine management systems of RTA-C engines have the following key features:
- All-electrical interface defined by the DENIS specification for all control and monitoring functions;
- Separate engine fitness systems in the MAPEX family for specialised functions, including:
  - Piston-running parameters;
  - Cylinder liner wear;
  - Torsional vibration;
  - Axial vibration;
  - Management support for spare parts ordering, stock control, and maintenance work.

The engine management systems for Sulzer two-stroke engines comprise different modules to meet the individual requirements of each shipboard installation (Fig. 43). At the heart of these is an all-electrical interface for all control and monitoring functions that caters for the various arrangements of remote control systems encountered in today's ships. This interface provides the basis for remote control of the engine from the ship's automation system. The interface is described in the respective DENIS for each engine type (DENIS = **D**iesel **E**ngine co**N**trol and opt**I**mizing **S**pecification). For example, the RTA84C engines use DENIS-1 and the RTA96C uses DENIS-6.

The DENIS concept was introduced in 1991 and has the following objectives:
- Clear definition of the signal interface between engine and its remote control;
- Engine control reduced to local control and interface close to the engine;
- Interface to the remote control system to be purely electrical.

Fig. 43
Engine management systems for Sulzer two-stroke engines are based on a modular concept with an DENIS interface specification for individual engine types and, in the MAPEX family, a suite of engine performance enhancers
[97#243]



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, the company concentrates on the ‘engine’ and co-operates with specialist suppliers who provide the electronic remote control and monitoring systems. Nevertheless, Wärtsilä NSD is still active in those areas of control and monitoring which involve specific knowledge concerning diesel engines. These are grouped within the MAPEX (**M**onitoring and m**A**intenance **P**erformance **E**nhancement with e**X**pert knowledge) family of products developed by Wärtsilä NSD to provide shipowners and operators with the tools needed 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 shipowners.

MAPEX products complement and expand upon the functions of standard remote control systems. They include monitors dedicated to piston-running parameters, cylinder liner wear, torsional vibration, axial vibration and the combustion process, which include both alarm and trend analysis functions. MAPEX-SM offers management support for spare parts ordering, stock control, and maintenance work.

Overall, the MAPEX philosophy encompasses the following principles:

- Improved engine availability and performance through reduced downtime;
- Monitoring of critical engine data and intelligent analysis of that data;
- Advanced planning of maintenance work and management support for spare parts;
- Access on board ship to the knowledge of experts;
- Full support of data storage and transmission possible by either floppy diskette or satellite communications;
- Saving money through reduced costs and better efficiency.

Service experience of RTA84C engines

The first RTA84C engine was built by Diesel United Ltd in Aioi, Japan, and entered service in July 1990 in the containership *Katsuragi.* Since then, most of the engines ordered have entered service and give highly satisfactory results in demanding applications on regular liner trades between the East Asia and Europe as well as across the Pacific. Figure 44 shows the cylinder liner wear data available to Wärtsilä NSD for the RTA84C engines at the original rating. They are an excellent record with diametrical wear levelling out at less than 0.1 mm/1000 hours.

Also the performance of the top piston ring 'a' in service is very good. The average wear rates known to Wärtsilä NSD for the top ring are around 0.2 mm/1000 hours after running-in (Fig. 45). Figures 46 and 47 show the typical condition of pistons in the original RTA84C engines. The example shown is of a piston from the 8RTA84C engine of the containership *Nedlloyd Asia.*

Fig. 44
Summary of cylinder liner wear rates available in Wärtsilä NSD from RTA84C engines in the original version
[97#244]

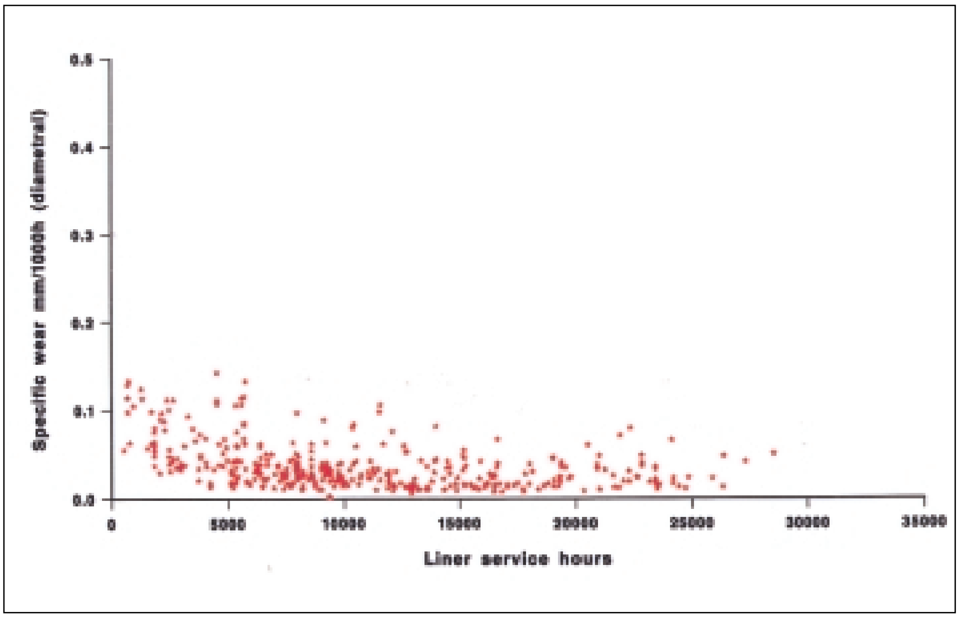
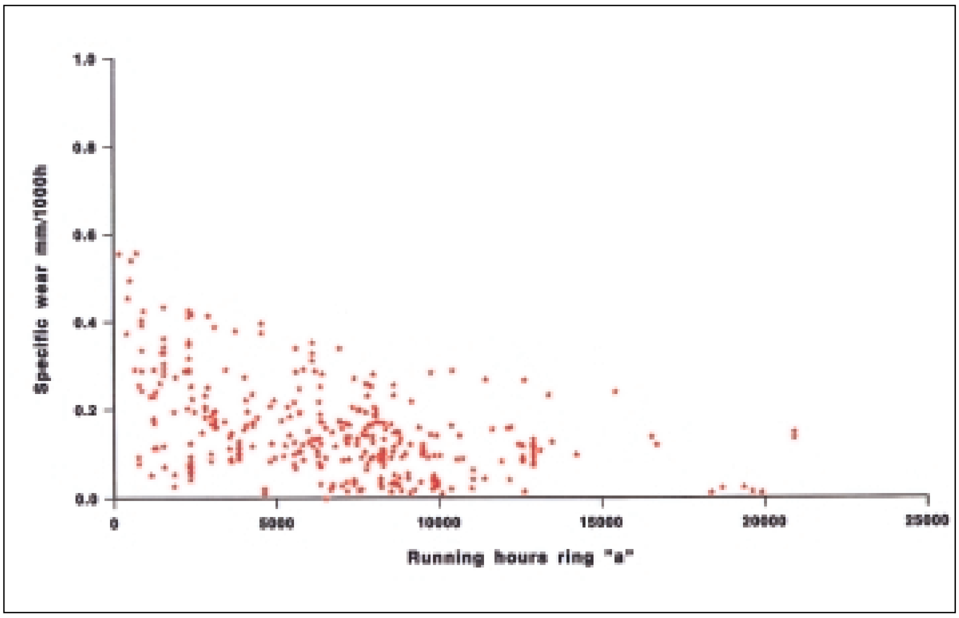


Fig. 45
Summary of wear rates of top piston ring 'a' available in Wärtsilä NSD from RTA84C engines in the original version
[97#245]



Service experience with upgraded RTA84C engines

The first upgraded RTA84C engine, built by Diesel United Ltd, Japan, entered service in June 1995 in the containership *Sea Land Champion.* Since then, 51 upgraded engines have already entered service.

The first engines to enter service have until now accumulated up to some 15 000 running hours. According to data available to Wärtsilä NSD, the average diametrical wear rates of the cylinder liners are generally good and are levelling out at values around 0.05 mm/1000 hours and below after running-in (Fig. 48).

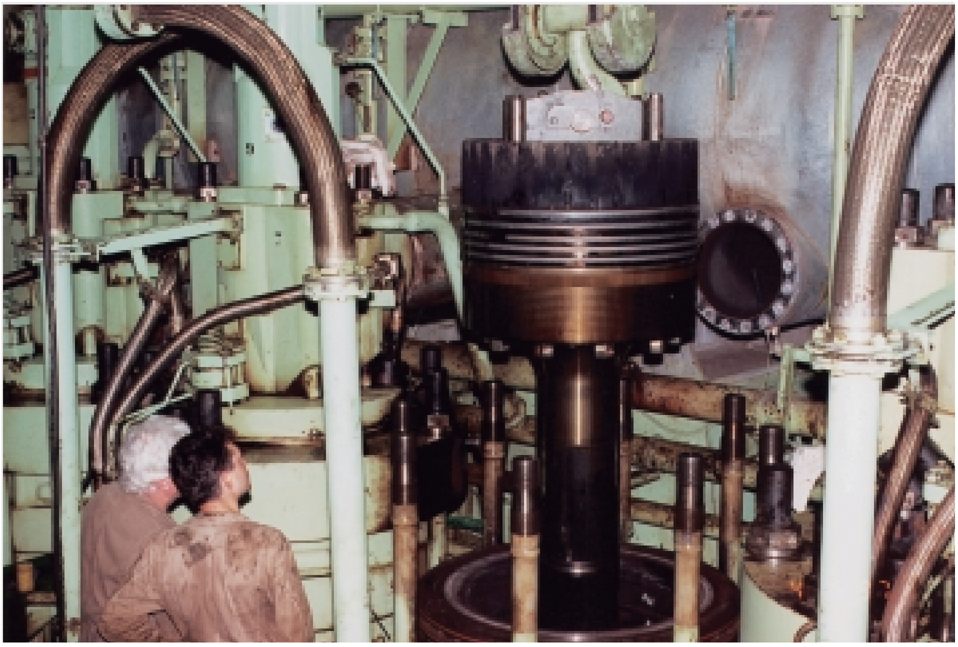


Fig. 46
Piston withdrawal at 38 667 total running hours, and 16 118 running hours since the previous inspection shows the typical condition of pistons in RTA84C engines at the original rating. The example is from the 8RTA84C engine of the containership *Nedlloyd Asia*
[7797-3034]

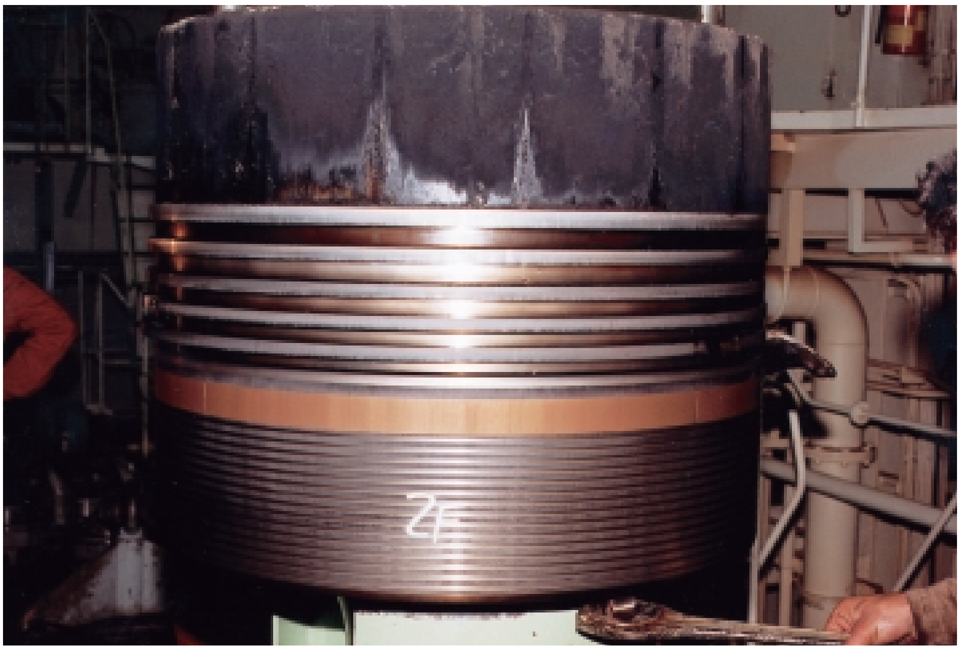


Fig. 47
Close-up view of the same piston shown in figure 46. The radial wear rate of the top ring is 0.08 mm/1000 hours at 16 118 running hours. The lower rings have wear rates of 0.02–0.04 mm/1000 hours. The diametrical wear rate of the cylinder liner is 0.026 mm/1000 hours.
[7797-3035]

It has to be mentioned that some liner scuffing has occurred on a limited number of liners which suffered wear rates that were far away from acceptable, even after the initial running-in time. These cases have been analysed and a combination of several factors were found to be the cause, such as damaged inner surface of cylinder liners by too rough machining and also insufficient protection against water carry-over with the scavenge air into the cylinder. Accordingly, counter-measures to eliminate such cases have been introduced, namely smooth machining of the cylinder liner followed by honing have been declared

Fig. 48
Summary of cylinder liner wear rates available in Wärtsilä NSD from RTA84C engines in the upgraded version
[97#246]

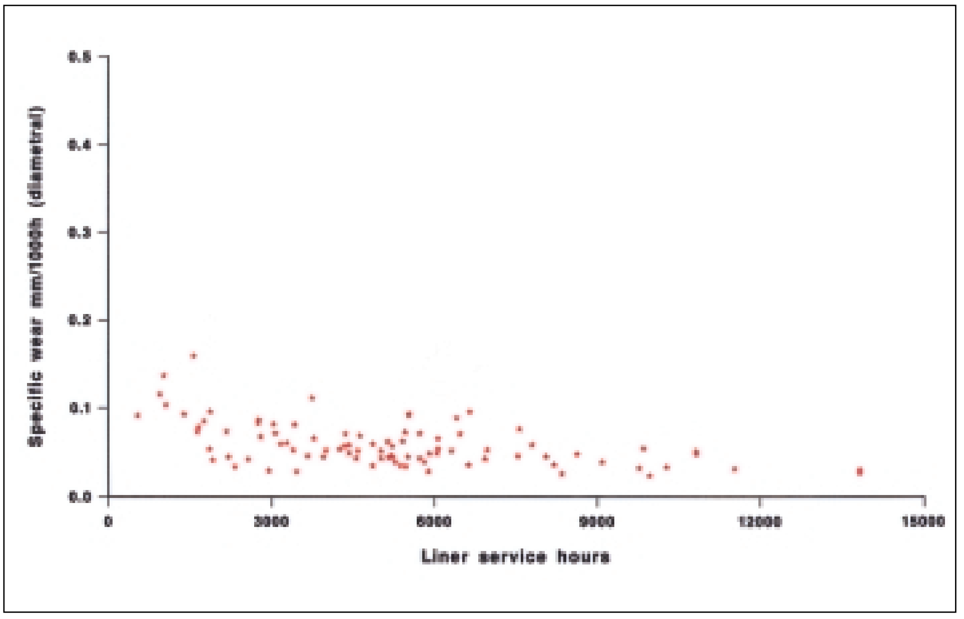
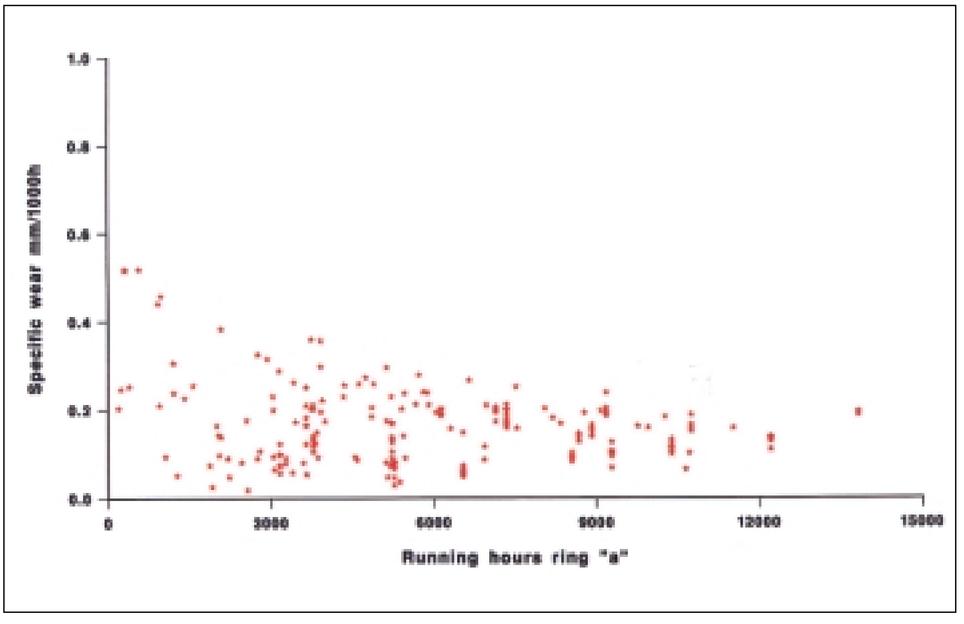


Fig. 49
Summary of wear rates of top piston ring ‘a’ available in Wärtsilä NSD from RTA84C engines in the upgraded version
[97#247]



standard and the water drain from the scavenge air receiver has been rectified. Very good results after modification prove the value of the measures taken.

The piston ring wear data are very satisfactory, mainly owing to the absence of corrosive attack (achieved by sufficiently high liner wall temperature) and also owing to the introduction of the plasma-coated top piston ring ‘a’. Specific wear rates for the top piston ring ‘a’ are below 0.2 mm/1000 hours (Fig. 49).

It has to be mentioned that, on some cylinders, the piston rings below the top one have worn with slight-

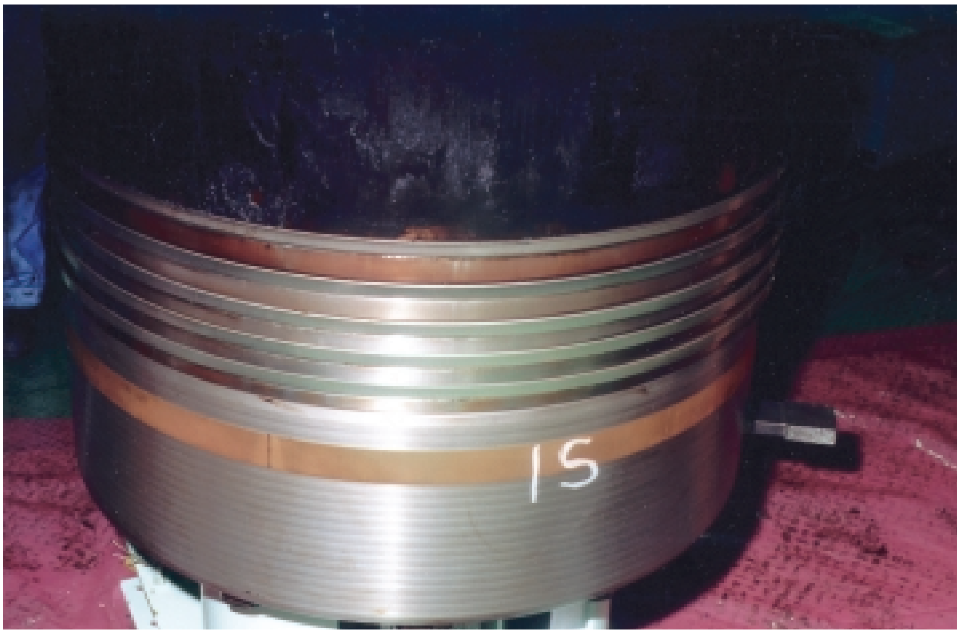


Fig. 50
Piston from the 12RTA84C engine of the *Ever Ultra* at 5019 hours’ operation. Top ring wear is 0.07 mm/1000 hours
[97#248]

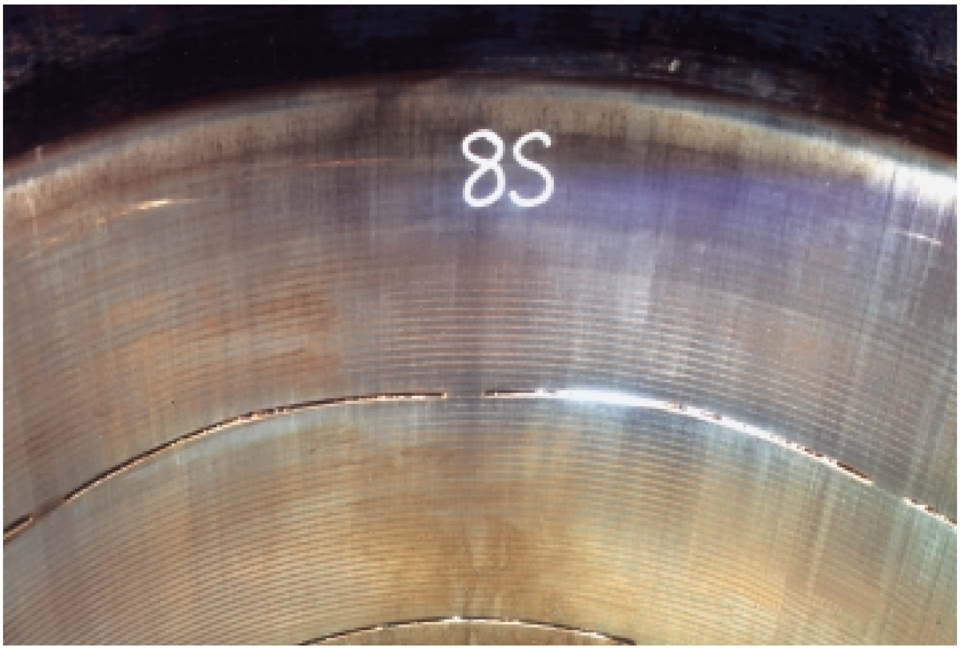


Fig. 51
Cylinder liner from the 12RTA84C engine of the *Ever Ultra* at 5019 hours’ operation. The maximum diametrical liner ring wear is 0.04 mm/1000 hours
[97#249]

ly higher wear rates than the plasma-coated top ring. The accuracy and the statistical relevance of these measurements are however today still questionable and need to be clarified.

On some engines, pistons equipped with all five rings, pre-profiled and plasma-coated, are running for test purposes with remarkable stability also in cases similar to the above-mentioned ones.

As a conclusion, it can be stated that the service experience with the upgraded RTA84C engines is in general excellent. The increased power level has been well absorbed by the upgrading measures which have proven to be effective in every respect.

Fig. 52
Underside of a piston crown from the 12RTA84C engine of the *Ever Ultra* at 5019 hours' operation
[7797-3036]

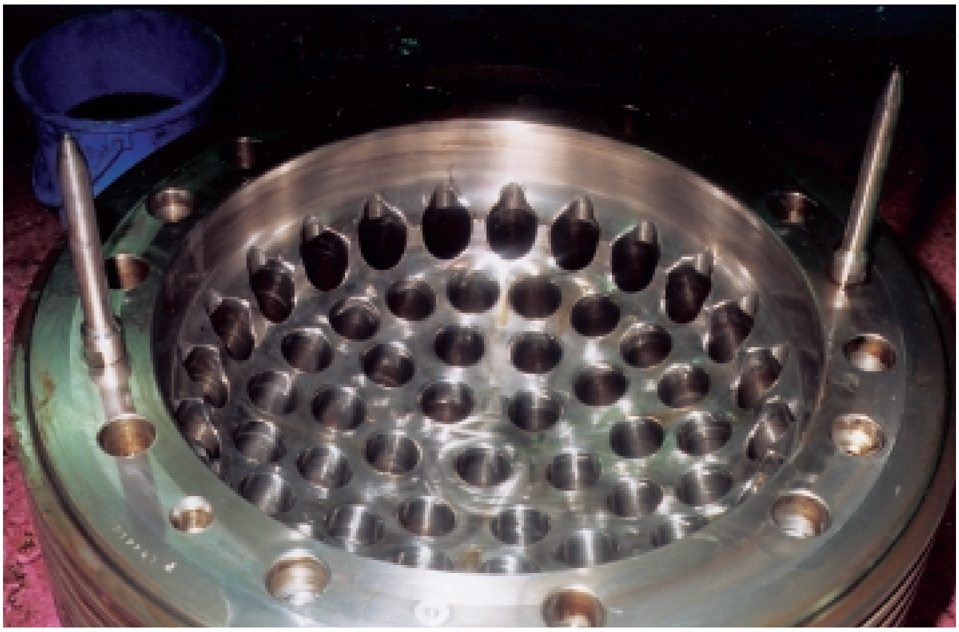
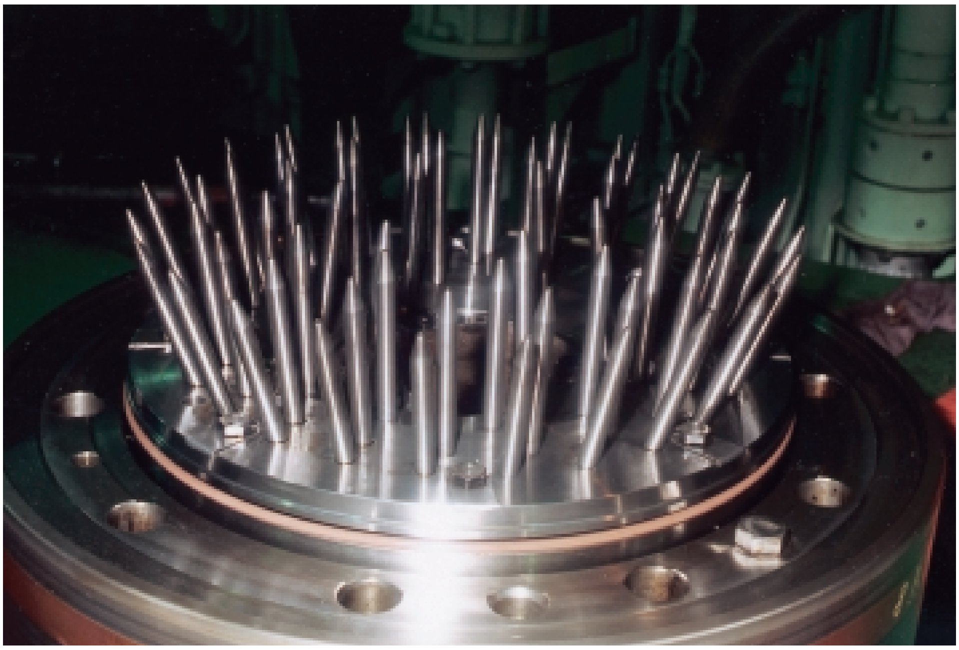


Fig. 53
Piston cooling nozzles from the 12RTA84C engine of the *Ever Ultra* at 5019 hours' operation
[7797-3037]



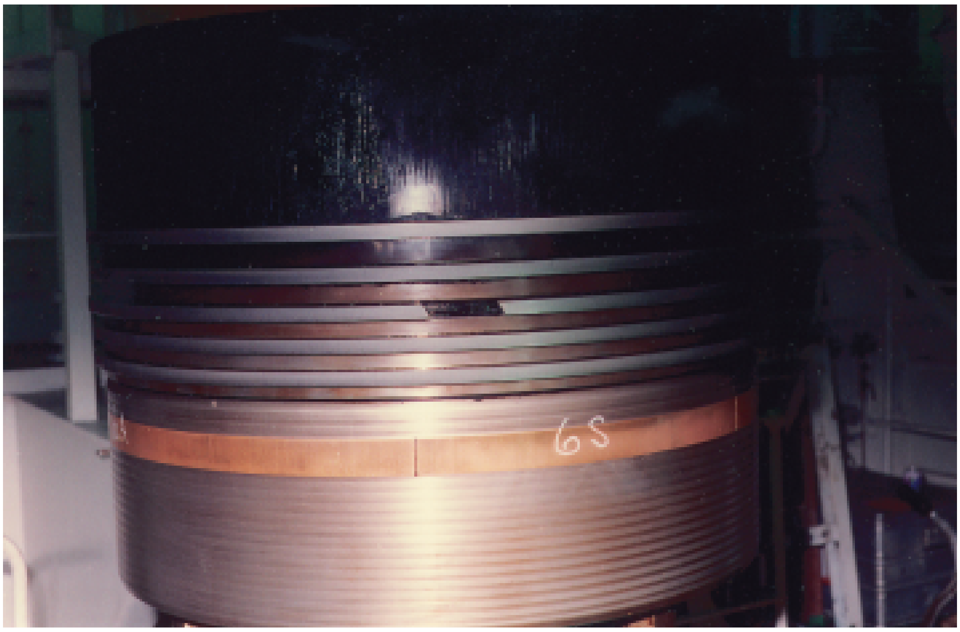


Fig. 54
Piston from the 10RTA84C engine of the *NOL Tourmaline* at 9998 hours' operation. Top ring wear is 0.17 mm/1000 hours [97#250]

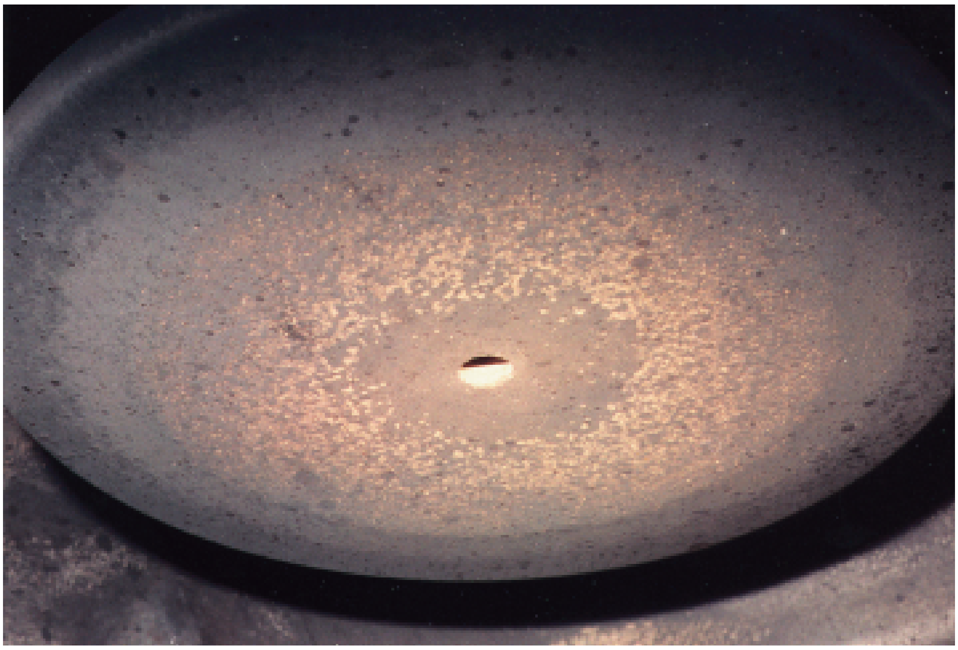


Fig. 55
Underside of an exhaust valve from the 10RTA84C engine of the *NOL Tourmaline* at 9998 hours' operation [97#251]

Test results from the first RTA96C

The first Sulzer RTA96C is a 11-cylinder engine which successfully completed its official shop test on 28 May 1997 at the Aioi works of Diesel United Ltd in Japan. At that time, it was the world's most powerful diesel engine ever built, with a nominal MCR output of 82 170 bhp (60 390 kW) at 100 rev/min.

The engine was built for NYK Line and will drive a 5750 TEU containership it has an installed contracted MCR of:

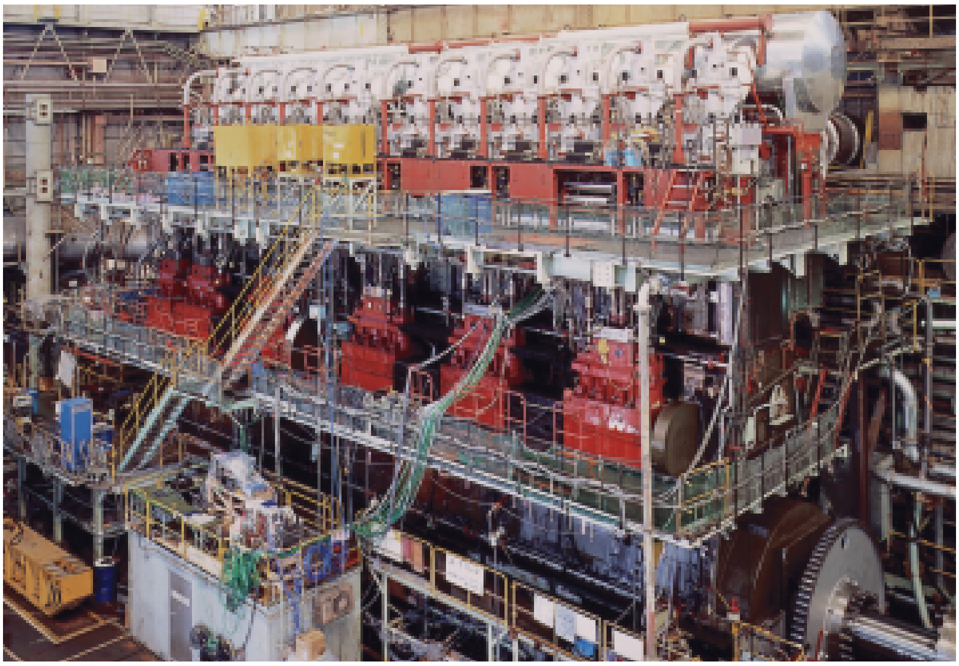
• CMCR	72 470 bhp (53 300 kW)	88.25 % nominal R1
• Speed	94 rev/min	94 % nominal R1
• BMEP	17.1 bar	94 % nominal R1

The test programme

To obtain comprehensive feedback from the tests, numerous temperature and strain gauges were applied during erection. Some 600 strain gauges for stress evaluation and 330 temperature measuring points were installed on structural, rotating, reciprocating and hot parts. Many of these measurements, such as valve lift, injection pressure and injection needle lift, as well as high and low pressure gas measurements, are state-of-the-art routine and do not need to be especially mentioned.

During the tests, the engine was run at the following rating points:

- Contracted MCR of 72 470 bhp (53 300 kW) at 94 rev/min, corresponding to 17.1 bar BMEP;



The first Sulzer RTA96C, an 11-cylinder engine, on test at Diesel United [7797-3013-1]

Key Points

- First RTA96C was an 11-cylinder engine, tests completed May 1997;
- Tested at three rating points: R3 for full bmep, CMCR, and at full speed with R3 power;
- Engine ran throughout the tests (196 hours' running) without any troubles:
- All performance characteristics have sound values;
- Very flat BSFC curve with good part-load fuel consumption;
- Moderate fuel injection pressures;
- Even, low surface temperatures on piston crown, 375 ±20 °C;
- Cylinder cover temperatures mostly around 315 °C to a maximum of 350 °C near joint with liner;
- Cylinder liner temperature 285 °C maximum, and 240 °C at TDC position of top piston ring;
- Exhaust valve surface temperature below 585 °C in centre and 540 °C at outer circumference;
- Exhaust valve seat temperature 315 °C;
- Stresses, strains and vibrations are all well within expected limits.

• The R3 rating point which corresponds to 73 920 bhp (54 340 kW) at 90 rev/min and 18.2 bar BMEP;
• An Rx rating point with the same power output as R3, but with the full speed of 100 rev/min, which corresponds to 16.4 bar BMEP.

The selection of the test ratings was based on the given CMCR and extended to other significant points and their corresponding propeller characteristics at which the engine could be run with the same turbocharger specification. The engine was not run at the nominal maximum continuous rating R1 owing to the complication of adapting all four turbochargers to that rating point.

The three test points, CMCR-R3-Rx, reflect on one hand full BMEP (18.2 bar) and, on the other, full engine speed. Therefore, the temperature and stress measurements cover most of the engine thermal and mechanical load range to be expected in service and are considered to be quite decisive for reliability. For example, overload at R3 showed very flat temperature and stress gradients well within safe expectations.

The following activities were executed during the test programme:
• Static stress measurements;
• Start of engine, running-in;
• CMCR: turbocharger matching;
• CMCR: engine tuning;
• CMCR: fuel injection variation, with temperature optimisation of combustion chamber;
• CMCR: performance data measurement;
• Heat balance;
• Exhaust emissions measurements;
• Vibration measurement;
• Dynamic stress measurement;
• R3: engine tuning;
• R3: fuel injection variation, with temperature optimisation of combustion chamber;
• R3: performance data measurement;
• Rx rating: fuel injection adaption;
• Rx: performance data measurement;
• Determination of final specifications;
• Parts inspection;
• Confirmation test;
• Official shop test.

The test programme was finished within 196 hours of running. The engine behaved very well indeed and went through these tests without any troubles (Figs. 56 and 57).

A careful running-in procedure was executed, taking just over 20 hours from the engine being started until it reached full load, thereby omitting any tribological problems with liners or piston rings.

Thermodynamics and comments on engine performance

The engine performance curves show very sound values that lie well within Wärtsilä NSD expectations or even positively supersede them (Fig. 58). Hence, based on these data, the long-term engine behaviour in service can be expected to be at least equal, or even better than the Sulzer RTA84C.

The very flat characteristics of temperatures before and after the turbine over the engine load range is remarkable. It is a consequence of the reliability-

Fig. 56
Piston from the first 11RTA96C after 196 hours' running on the test bed [97#252]

oriented engine tuning that was selected and of the characteristics of the four ABB VTR714D-32 type turbochargers used, together with the low flow resistance along the scavenge air and exhaust gas passages.

The level of temperatures before and after the turbine is somewhat lower in relation to previous engine types such as RTA84C. This is credited to the engine tuning and turbocharger match chosen with reliability in mind. The exhaust gas temperature level, however, is still significantly higher than those of competitors' engines.

The BSFC curve is very flat, with the good part-load consumption characteristic being remarkable. The part-load consumption benefits from the special, reliability-oriented engine tuning with a somewhat elevated ratio of scavenging pressure to mean effective pressure and of the corresponding valve timing. In addition, there was introduced a slightly reduced geometrical compression ratio, which was selected to gain more distance between piston crown and injection sprays in favour of low piston surface temperatures at full-load.

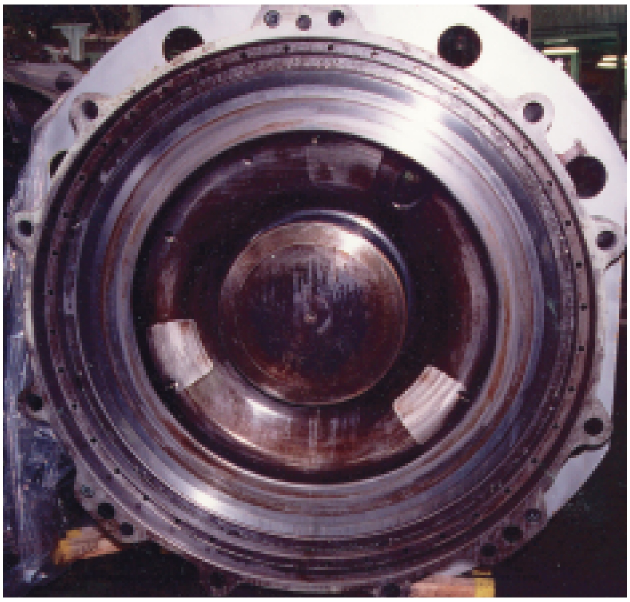
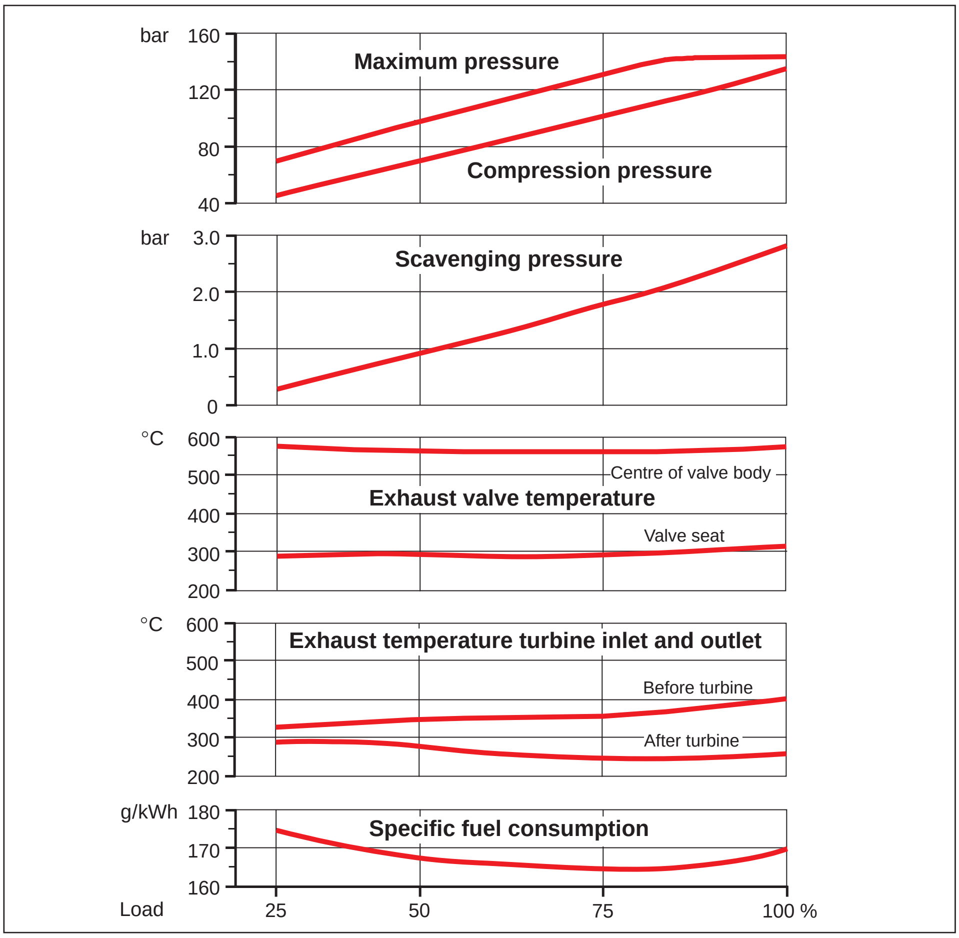


Fig. 57 (above)
Cylinder cover from the first 11RTA96C after 196 hours' running on the test bed. The three clad areas can be seen adjacent to the fuel injection valves
[97#253]

Fig. 58 (right)
Performance characteristics of the first 11RTA96C engine at the R3 rating, 73 920 bhp (54 340 kW), 18.2 bar BMEP, 90 rev/min, measured on the test bed at Diesel United Ltd in Aioi
[97#254]



Injection and combustion

Three fuel injection valves are fitted in each cylinder, helping to achieve a good mixture formation over the whole power range down to the lowest loads owing to the smaller nozzle hole diameters which would not be possible with just two injectors. In addition, the expected avoidance of hot spots on the piston surface, cylinder cover and liner owing to a more even distribution of fuel in combustion chamber was fully confirmed by measurements.

The fuel injection pressure is kept moderate; even at high engine speed there is ample margin for heavy-fuel operation (Fig. 59). An injection duration of about 18.5 degrees crank angle (CA) for R1 rating is considered to be appropriate for this type of engines. This judgement is based on the vast knowledge gained in tests with systematic variation of injection parameters on the RTX54 research engine.

The needle lift indicates that the injector needles close properly without after-injection and thereby avoid unnecessary fuel consumption, smoke and emissions (Fig. 59).

Temperature measurements

As combustion chamber surface temperatures, among other parameters, depend strongly upon the fuel nozzle specification, Wärtsilä NSD puts great emphasis in building up the skill to predict such dependent relationships. Computational fluid dynamic calculations have been carried out and more simple-to-use three-dimensional modelling of the fuel spray propagation prediction have been developed. Both are based on experience gained from numerous test engines and help for a better understanding of the complicated interactions during mixture formation.

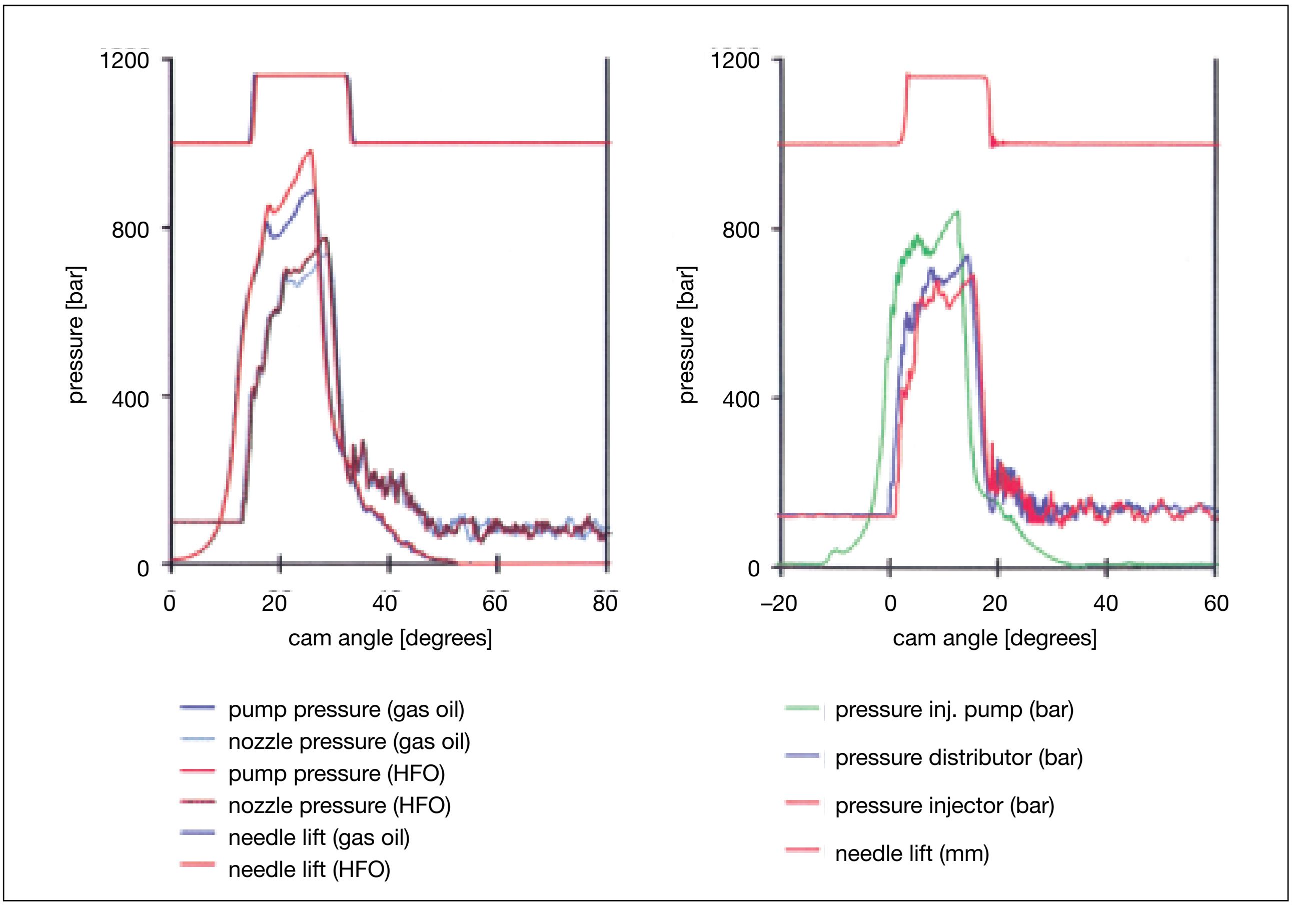


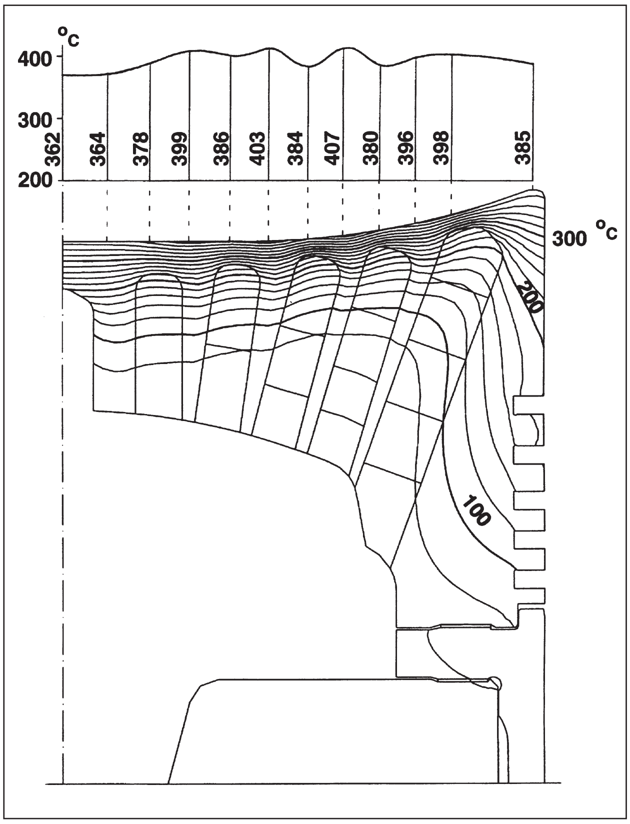
Fig. 59
Fuel injection pressure and needle lift diagrams for the RTA96C. Left gives a comparison of pressures calculated for both 380 cSt heavy fuel and gas oil. Right are the measured values from the 11RTA96C running at its CMCR of 72 470 bhp (53 300 kW) at 94 rev/min
[97#255]

The moderate temperature levels achieved in the combustion chamber components is a combined effect of optimising many parameters. It is based on techniques, such as injection modelling and with other simulation programs, and on an expert knowledge of gaseous and fluid heat transfer mechanisms, which also allow adequate computation of stress and strain under different thermal, static and dynamic loads.

Piston

Special care was given to keep the temperatures of the piston crown as low as possible. The temperatures measured fully confirm the assumptions for the piston design. As expected, surface temperatures of 375±20 °C are reported with a complete absence of hot spots (Figs. 60, 61 and 62). The temperatures measured in the inner cooling bore surfaces remain below 205 °C and thus well below any danger of oil coke formation. A consistently high heat transfer rate from piston crown to cooling oil can be expected over a long period of time. Owing to the improved jet cooling principle and thus constant piston crown temperature, long times between overhauls can be reasonably expected.

Fig. 60
Calculated temperature distribution in the piston of an RTA96C engine at MCR output assuming the same heat input as an RTA84C engine with three fuel injection valves
[95#256]



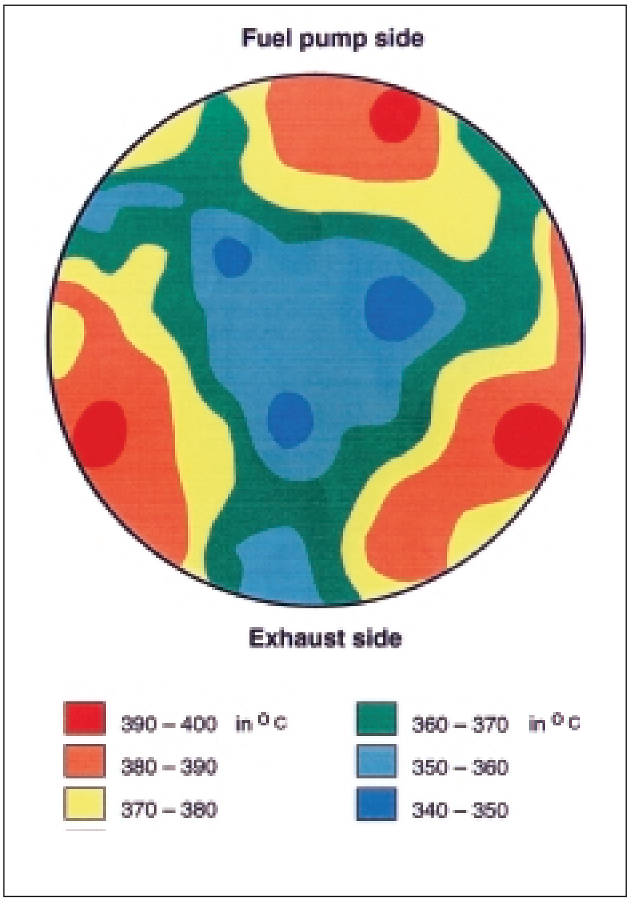


Fig. 61
Piston surface temperatures measured on the 11RTA96C at its CMCR of 72 470 bhp (53 300 kW) at 94 rev/min. The even distribution caused by the three-nozzle arrangement is noticeable
[97#257]

Cylinder cover

The cylinder cover temperatures range from around 315 °C for most of the surface to a slightly increased level of 350 °C towards the joint with the liner (Fig. 62). Downstream of the nozzles, the cover is partially protection clad.

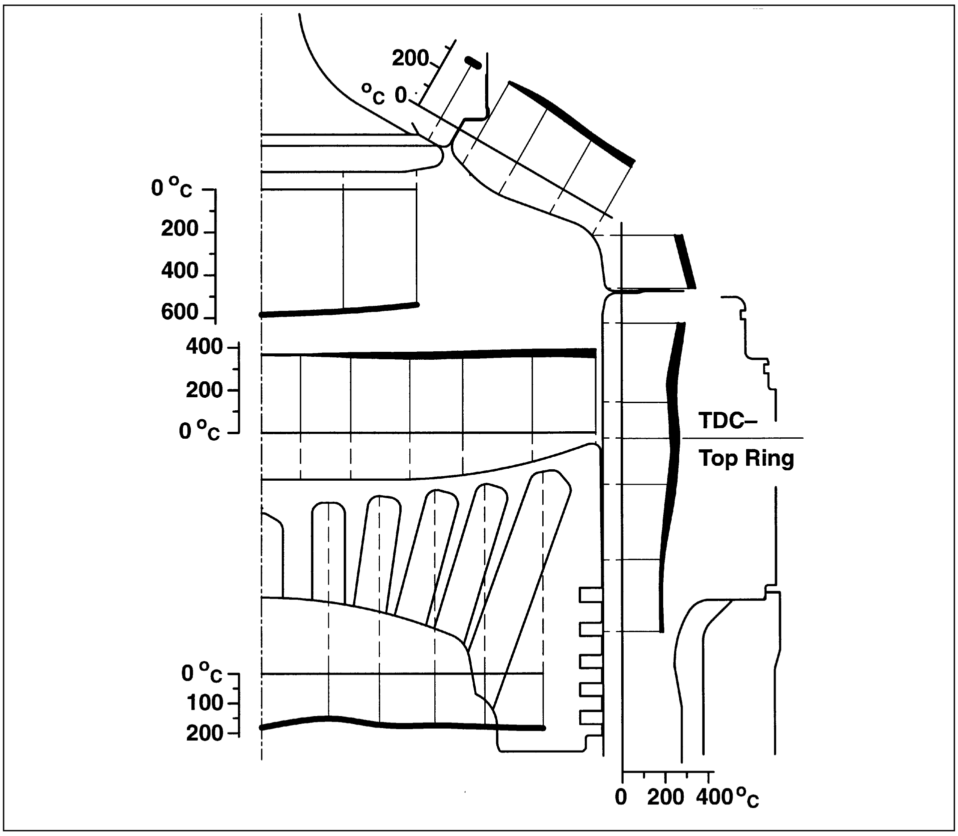


Fig. 62
Surface temperatures measured on the combustion chamber components of the 11RTA96C at the R3 rating, 73 920 bhp (54 340 kW), 18.2 bar BMEP, 90 rev/min
[97#258]

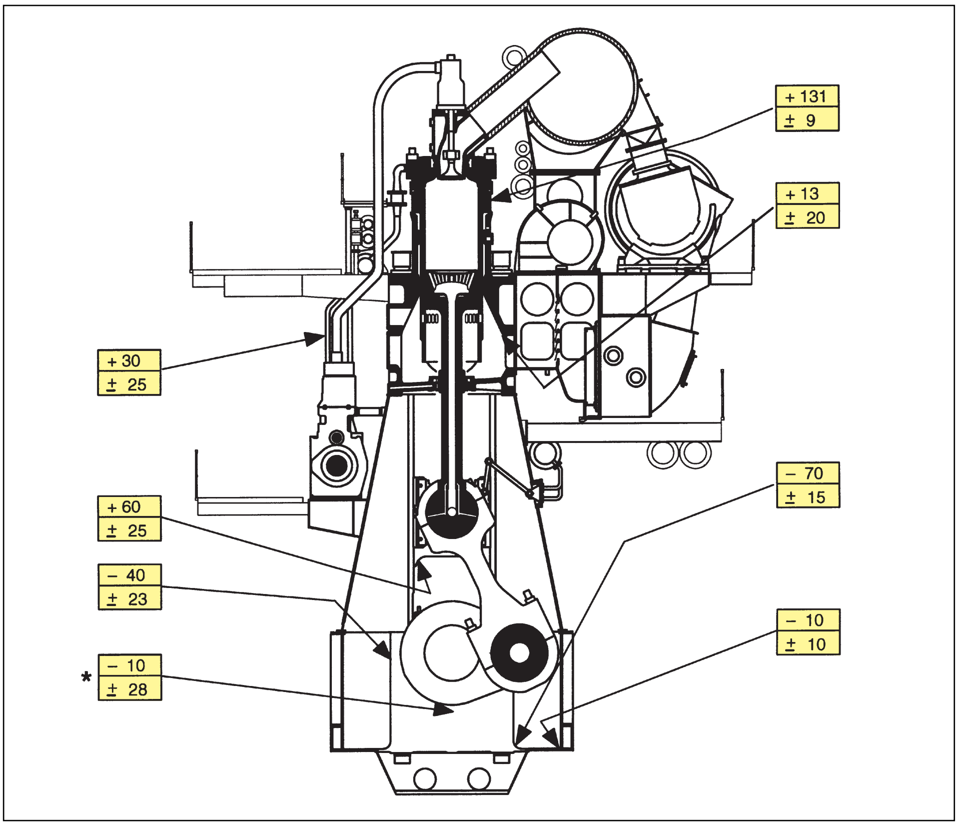


Fig. 63
Measured static and dynamic stresses in N/mm^2 on the RTA96C. The set marked (*) are calculated values
[97#259]

Cylinder liner

The absolute maximum liner surface temperature is measured close to the joint with the cover and is reported to be at a level of only 285 °C with a circumferential variation of ±20 °C (Fig. 62). This value and the corresponding combined static stress and thermal strain are well below the low-cycle fatigue limit of the material specified.

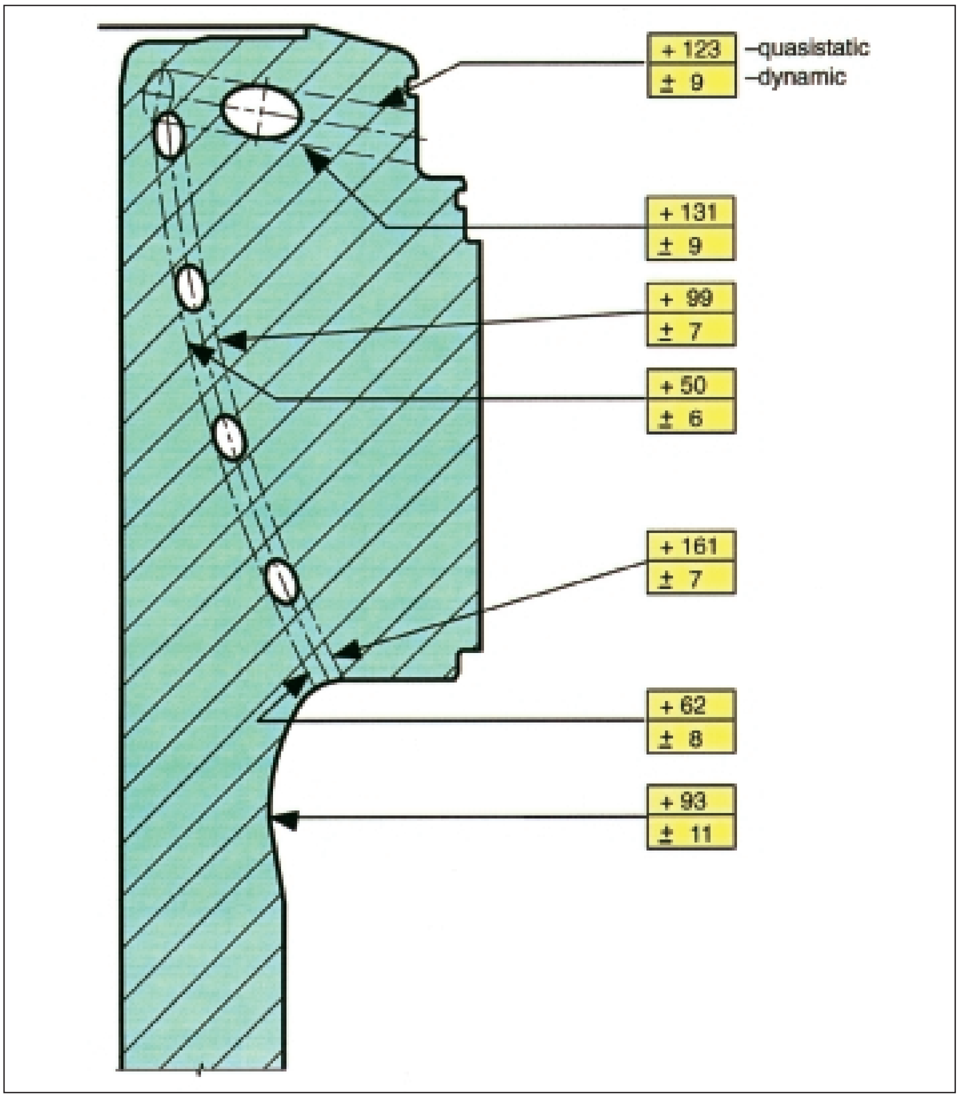
The surface temperature on the liner running surface is less than, or equal to 240 °C for the position where the top ring is at top dead centre (TDC). Along the stroke, the temperature is kept sufficiently high above the dew point that any corrosive attack can be avoided by applying insulating tubes of CMCR-dependent lengths into the liner cooling bores. The thermal prerequisites for successful piston-running are fulfilled.

Exhaust valve

The valve surface temperature is below 585 °C in the centre, decreasing to 540 °C at the outer circumference (Fig. 62). The valve seat temperature in the cover is as low as 315 °C. This temperature level allowed us to adopt, as the standard solution, an exhaust valve without cladding on the face, and still reach times between overhauls of far beyond two years.

Experience shows that engines with valve temperatures clearly above 600 °C measured on the test bed tend later in service to suffer from a certain amount of material loss, shortening their TBO. With valves, however, having a disc surface temperatures below 600 °C, a service lifetime of 50 000 hours and above was experienced in the past.

Fig. 64
Measured quasi-static and dynamic stresses in N/mm2 in the cylinder liner of the 11RTA96C at the R3 rating, 73 920 bhp (54 340 kW), 18.2 bar BMEP, 90 rev/min [97#260]



Engine dynamic measurements

The engine vibrations both in transverse and longitudinal direction were measured (Figs. 65 and 66). The values are well within expected limits and by far meet the VDI Guide Lines 2063. It was demonstrated that the engine structure is sturdy and, even with a stiff testbed foundation, shows good results, which in reality would lead to rather smaller displacements and lower local velocities on board ship owing to the damping effects that are prevalent.

The engine torsional vibration was there calculated for and measured at the testbed arrangement of the Diesel United's Aioi works. Owing to the limitations of the turbocharger specification used, the upper speed range above 94 rev/min had to follow a different propeller characteristic to measure mainly vibration up to 100 rev/min which is mainly dependent upon speed.

The recorded torsional synthesis shows two natural frequencies at a critical speed of 72.5 and 96.7 rev/min for the eighth and the sixth order

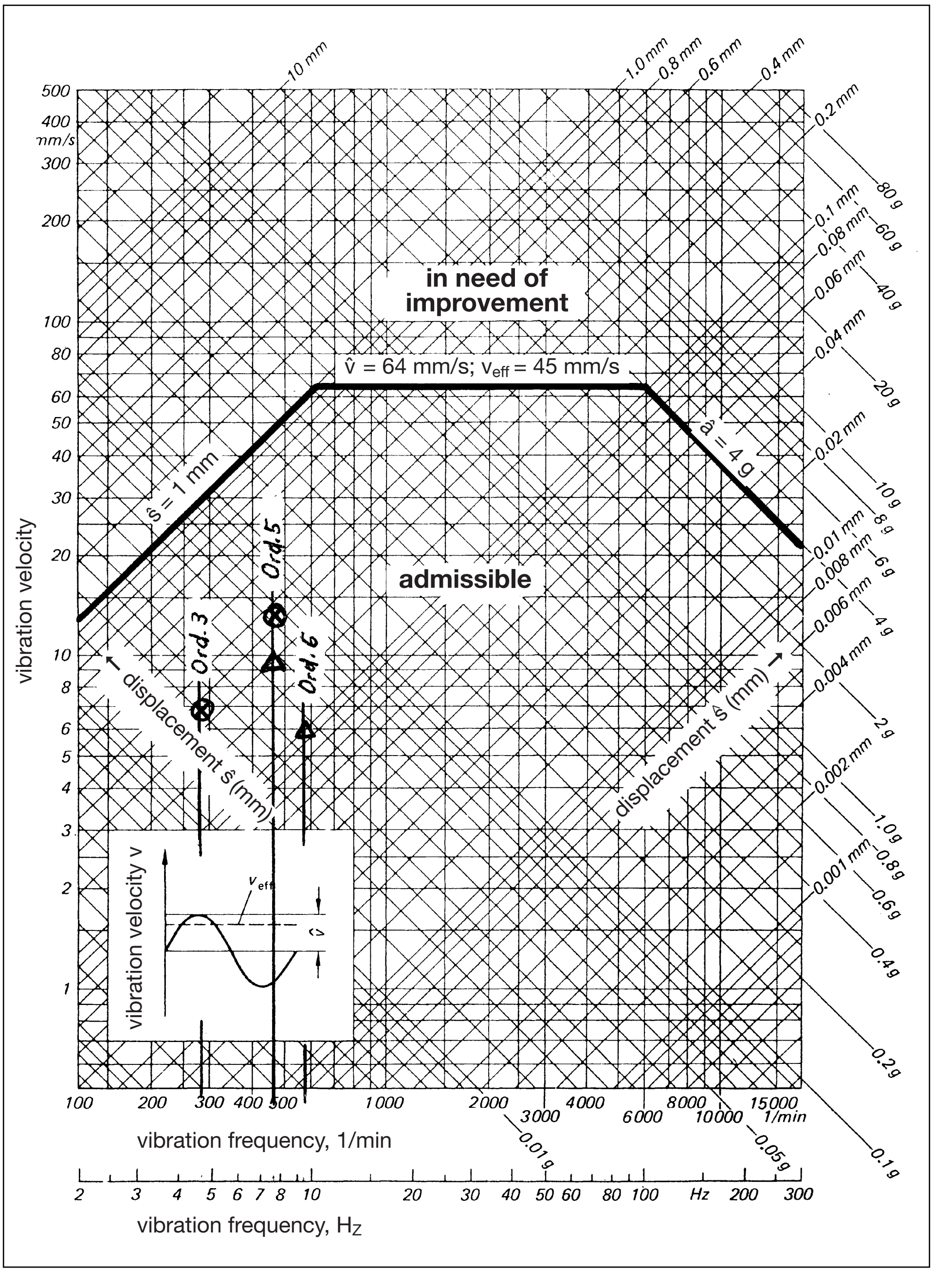


Fig. 65
Measured engine vibrations of the 11RTA96C engine on the test bed are well within the admissible values in VDI Guide Lines 2063
[97#261]

⊗ Lateral direction
△ Longitudinal direction

respectively which corresponded well with the calculated frequency amplitudes of the sixth and eighth orders. The respective stress amplitudes are well within IACS rules for all running conditions of this installation.

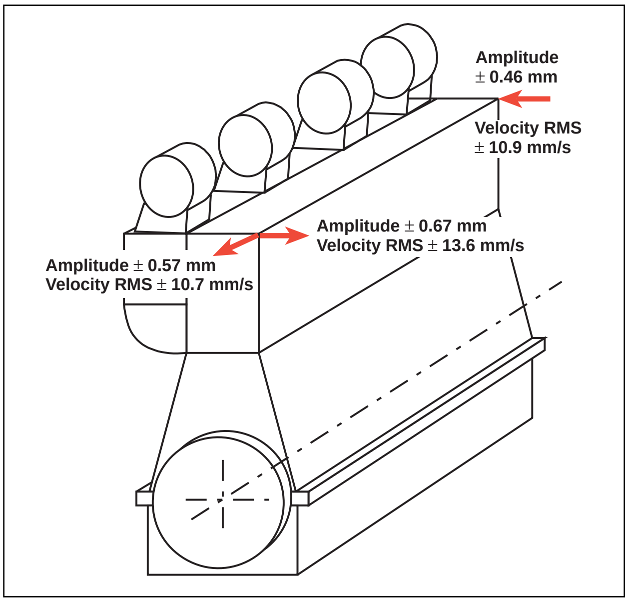


Fig. 66
Measured engine vibrations (longitudinal and lateral velocities and amplitudes) at cylinder top level of the 11RTA96C running on the test bed at its CMCR rating of 72 470 bhp (53 300 kW) at 94 rev/min
[97#180]

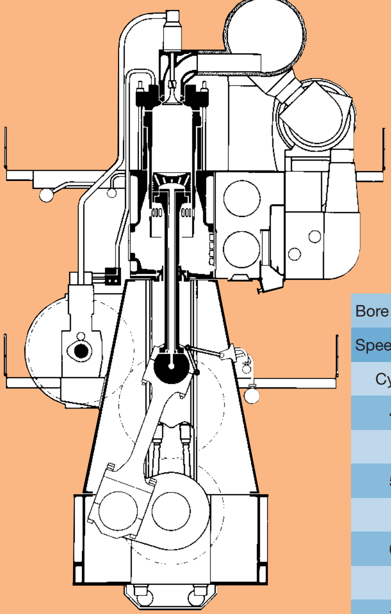
Conclusion

The Sulzer RTA-C two-stroke engines are tailored specifically to the needs of large containerships on line-haul services. Reliability is the key priority, together with times between overhauls of more than two years' operation. Such performance is now routinely achieved by RTA84C engines. With a design as similar as possible to the RTA84C, the new RTA96C has the definite objective of reaching the same, or even better levels of performance in service. Test results from the first RTA96C engines clearly confirm that the new design is 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 TBO. Wärtsilä NSD is dedicated to doing the right things to assure the full success of these prime movers.

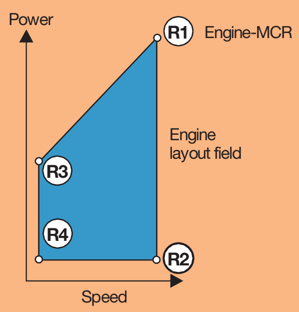
Bibliography

1. K. Aeberli, N. Mikulicic and E. Schaad, 'The development of a reliable and efficient two stroke engine range', CIMAC, Tianjin, China (PRC), 1989.
2. K. Aeberli: 'Results of the application of "high technology research methods" and their transformation into the design of simple and reliable two stroke engines', CIMAC, Florence, 1991.
3. K. Aeberli and N. Mikulicic: 'Reliability and economy of today's two-stroke diesel engine: Technology, solutions applied and results of the latest development of the Sulzer RTA engine range', CIMAC 1993.
4. K. Aeberli and A. Bitterli: 'The reliable and economic two-stroke diesel engine: Basic platform for evolution towards the intelligent engine', CIMAC, Interlaken, 1995.
5. S. Fankhauser: 'Advances in engine management systems keeping pace with change', CIMAC, Interlaken, 1995.
6. K. Aeberli: 'The Sulzer Containership Engines: The RTA84C and the RTA96C', Licensees' Meeting, October 1996.
7. K. Aeberli and L. Schnellmann: 'The Sulzer RTA-T engine series: Simple designs for simple operation and maintenance', 'The Motor Ship' conference, London, March 1997.

Two-Stroke Marine Diesel Engines Main Data



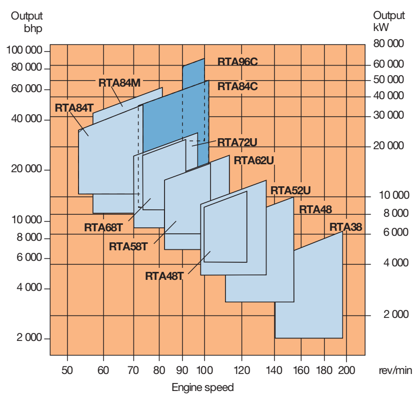
		RTA96C				RTA84C			
Bore x Stroke mm		960 x 2500				840 x 2400			
Speed rev/min		100	100	90	90	102	102	73	73
Cyl.	Power	R1	R2	R3	R4	R1	R2	R3	R4
4	kW					16 200	8 920	11 600	8 920
	bhp					22 040	12 120	15 760	12 120
5	kW					20 250	11 150	14 500	11 150
	bhp					27 550	15 150	19 700	15 150
6	kW	32 940	23 040	29 640	23 040	24 300	13 380	17 400	13 380
	bhp	44 820	31 320	40 320	31 320	33 060	18 180	23 640	18 180
7	kW	38 430	26 880	34 580	26 880	28 350	15 610	20 300	15 610
	bhp	52 290	36 540	47 040	36 540	38 570	21 210	27 580	21 210
8	kW	43 920	30 720	39 520	30 720	32 400	17 840	23 200	17 840
	bhp	59 760	41 760	53 760	41 760	44 080	24 240	31 520	24 240
9	kW	49 410	34 560	44 460	34 560	36 450	20 070	26 100	20 070
	bhp	67 230	46 980	60 480	46 980	49 590	27 270	35 460	27 270
10	kW	54 900	38 400	49 400	38 400	40 500	22 300	29 000	22 300
	bhp	74 700	52 200	67 200	52 200	55 100	30 300	39 400	30 300
11	kW	60 390	42 240	54 340	42 240	44 550	24 530	31 900	24 530
	bhp	82 170	57 420	73 920	57 420	60 610	33 330	43 340	33 330
12	kW	65 880	46 080	59 280	46 080	48 600	26 760	34 800	26 760
	bhp	89 640	62 640	80 640	62 640	66 120	36 360	47 280	36 360
BSFC									
85%	g/kWh	166	161	166	161	168	160	167	161
Load	g/bhph	122	118	122	118	124	118	123	118
100%	g/kWh	171	163	171	164	171	160	170	163
Load	g/bhph	126	120	126	121	126	118	125	120
BMEP, bar		18.2	12.7	18.2	14.1	17.9	9.9	17.9	13.8



Definitions to all Sulzer diesel engines:

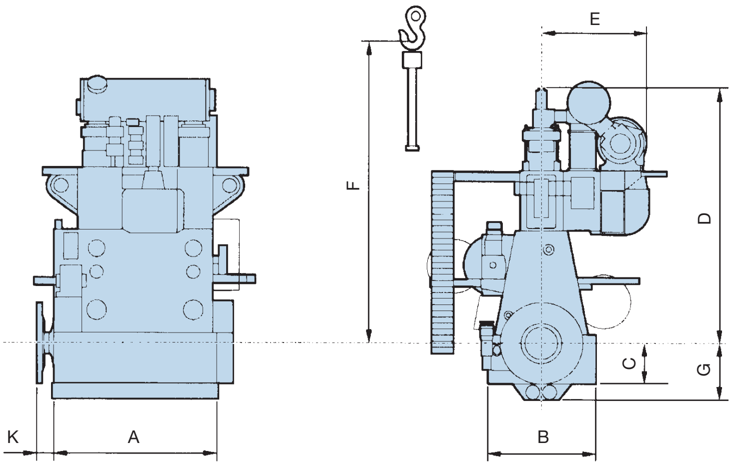
- R1, R2, R3, R4 = power/speed ratings at the four corners of the RTA engine layout field (see diagram).
- R1 = engine Maximum Continuous Rating (MCR).
- Contract-MCR (CMCR) = selected rating point for particular installation. Any CMCR point can be selected within the RTA layout field.
- BSFC = brake 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), with +3% allowance and without engine-driven pumps.
- 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-C**



Dimensions and Masses

(Millimetres and tonnes)

	RTA96C									RTA84C								
Cyl.	A	B	C	D	E**	F	G	K	Mass	A	B	C	D	E**	F*	G	K	Mass
4										7880	4320	1600	11315	4660	13130	2205	920	630
5										9480	4320	1600	11315	4191	13130	2205	920	740
6	11564	4480	1800	10925	5030	12880	2594	676	1150	11080	4320	1600	11315	4660	13130	2205	920	850
7	13244	4480	1800	10925	5030	12880	2594	676	1280	12680	4320	1600	11315	4660	13130	2205	920	960
8	15834	4480	1800	10925	5030	12880	2594	676	1460	15280	4320	1600	11315	4660	13130	2205	920	1110
9	17514	4480	1800	10925	5030	12880	2594	676	1600	16880	4320	1600	11315	4191	13130	2205	920	1230
10	19194	4480	1800	10925	5030	12880	2594	676	1740	18480	4320	1600	11315	4191	13130	2205	920	1350
11	20874	4480	1800	10925	5030	12880	2594	676	1890	20080	4320	1600	11315	4191	13130	2205	920	1460
12	22554	4480	1800	10925	5030	12880	2594	676	2030	21680	4320	1600	11315	4660	13130	2205	920	1570

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

Wärtsilä NSD Switzerland Ltd
PO Box 414
CH-8401 Winterthur
Switzerland

Telephone: +41-52-262 49 22
Fax: +41-52-212 49 17

WÄRTSILÄ NSD
CORPORATION

Wärtsilä NSD Switzerland Ltd
PO Box 414
CH-8401 Winterthur
Switzerland

Telephone: +41-52-262 49 22
Fax: +41-52-212 49 17

WÄRTSILÄ NSD
CORPORATION

23.86.07.40-Printed in Switzerland