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Sequential turbocharging of the MTU 1163 engine

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SYNOPSIS

The 1163 engine series is the largest in MTU's range, having a swept volume of 11.63 litres/cylinder producing a maximum of 370 kW/cylinder at 1300 rev./min. It is a 4-stroke engine available in 12, 16 and 20 cylinder versions in 60° vee-form configuration. The latest design incorporates a two-stage 'sequential' turbocharging system with multiple turbocharger groups mounted on the engine which are switched in and out of operation depending on engine power and speed demand. This new development resulted in a power increase of over 40% due to increasing the brake mean effective pressure from approx. 21 bar to 30 bar. In addition, the engine operating range was greatly expanded with advantages in performance and fuel consumption, particularly in the part-load condition. Such engines are almost exclusively used for the propulsion of fast vessels where power/weight and power/bulk ratios are of prime importance. The power/weight ratio has been increased by approx. 18% whereas the engine overall dimensions have actually been decreased due to the compactness of the new turbocharging system. This paper describes the design and development of the 1163 series sequential turbocharging system and its impact on the design of marine propulsion systems.

BACKGROUND

The 1163-02 engine series was originally conceived as the long-stroke version of the 956 series in order to compete in the generator set market with a speed of 1200 rev./min for power generation at 60 Hz. However, towards the end of the 1970s, it was decided to develop this engine and introduce an uprated version (1163-03) which could be used for the propulsion of fast vessels such as large patrol boats, corvettes and frigates. The power increase itself was not a major problem. The difficulty lay in additionally providing a compact unit with good torque characteristics and favourable fuel consumption. The following conditions were determined for the development activities.

1. Simple design using proven components and as far as possible retaining the basic engine design.
2. Turbochargers and intercoolers to be compactly mounted on engine.
3. Good starting and idling characteristics with low emission values.
4. Wide performance characteristics, i.e. high output torque even at low engine speed.
5. Rapid load acceptance.
6. Good consumption figures, in particular in the part-load range.
7. No increase in the maximum firing pressure (approx. 150 bar) and therefore no increase in the associated mechanical stresses.

Since the engine swept volume and piston speed were to remain unchanged, the increased power could only be achieved by increasing the brake mean effective pressure (b.m.e.p.), which required a new design of turbocharging system.

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COMPRESSION RATIO

It can be seen from Fig. 1, that the compression ratio must be reduced if the b.m.e.p. is to be increased without increasing firing pressure. In the early stages of the design, it was necessary to reduce the compression ratio to 8.5. Later, improvements in the fuel injection system, which included modifying the air inlet canals to create a swirl effect, permitted a shorter injection time (30° crank angle instead of 50°; see Fig. 2) and a reduction in the charge air pressure. This meant that the compression ratio could be raised to 9.7 without increasing the firing pressure. The higher compression ratio has of course had a beneficial effect on the low-power running and starting characteristics of the engine. Nevertheless, the measures originally introduced to overcome the problems associated with low compression ratio have been retained in the current design. These measures (cylinder cut-out and cylinder charge air transfer) are described in detail in refs. 1 and 2.

TWO-STAGE TURBOCHARGING

In order to increase the mean effective pressure to 30 bar, calculations had shown that a charge air pressure of around 5 bar

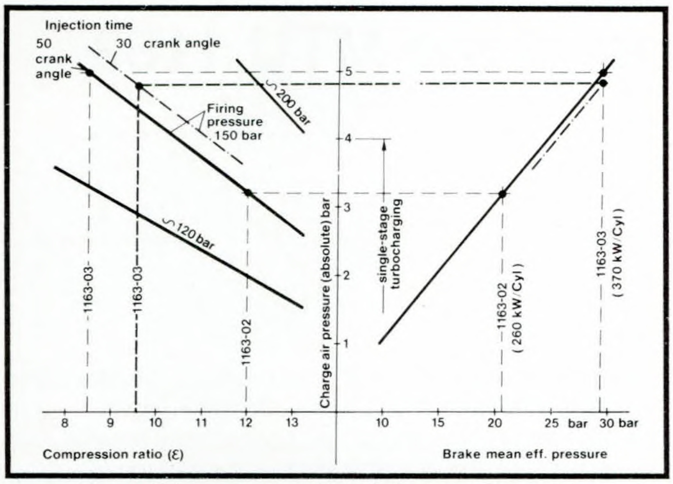

Fig. 1. Inter-relationship of design parameters (showing influence of injection time)

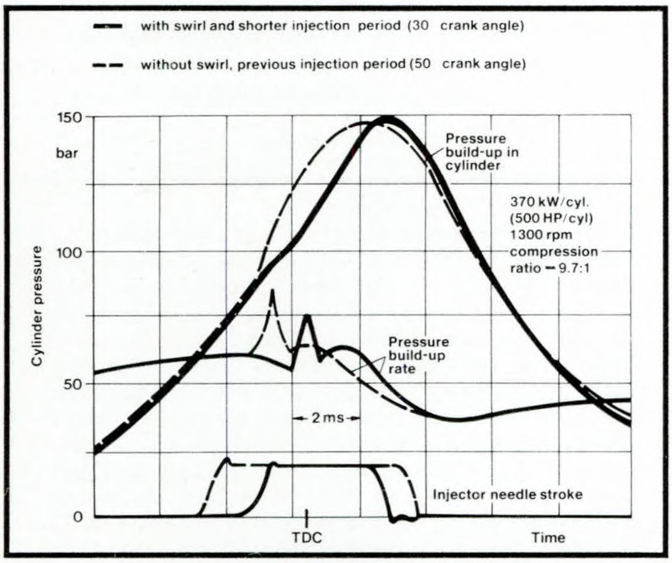

Fig. 2. Improved combustion process

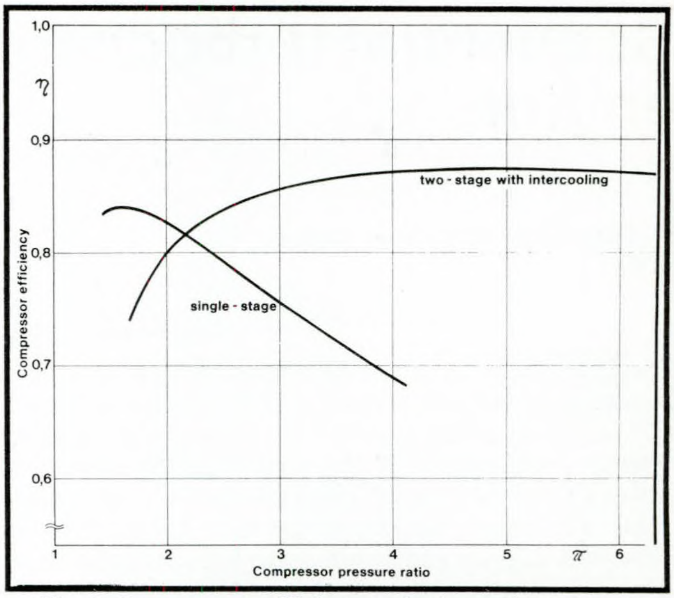

Fig. 3. Compressor efficiency, single- and two-stage supercharging

		1163-02	1163-03
P_{eff}	[kW/Zyl.] (mHP/Cyl.)	260 (354)	370 (503)
BMEP	[bar] (psi)	20.6 (299)	29.4 (426)
n	[rpm]	1300	1300
ε	[-]	12	9.7
λ	[-]	1.73	1.9
t [°C]	① Valve seat, cylinder head	260	245
	② Valve seat, valve	575	550
	③ Valve stem	260	250
	④ Cylinder head land	290	280

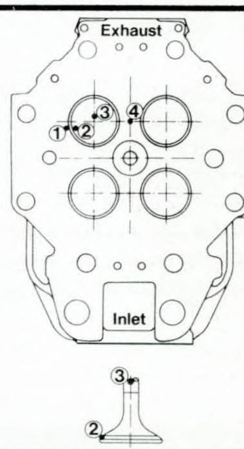

Fig. 4. Component temperature comparison using the same cylinder head

would be required. This exceeds the practical limit of single-stage turbocharging which lies in the region of 4 bar (Fig. 3), so it was necessary to design a two-stage system with intercooling and aftercooling. (Note: all pressures are absolute values.) Apart from providing the necessary charge air pressure, two-stage turbocharging enabled the air/fuel ratio (λ) to be increased. This assisted in the reduction of cylinder-head temperatures as shown in Fig. 4 (see also ref. 2).

SEQUENTIAL TURBOCHARGING

A wide performance characteristic is essential for a marine propulsion engine, particularly to satisfy the arduous operating profiles of naval craft. In addition to providing adequate ship speed and acceleration in various sea states, the propulsion engines should also be capable of supplying high torque at low speed for loitering duty or sonar towing. A conventional turbocharging system has only one optimum design point, namely at the engine's maximum continuous rating. At part-load, the turbine nozzle area is too large for the reduced exhaust gas flow and temperature and the compressor operating point moves closer to the surge line. The turbochargers therefore run inefficiently since the available energy in the exhaust gases cannot be effectively utilized, consequently, engine torque is limited. The problem is accentuated the more the specific power of engines is increased.

One solution is to mechanically vary the geometry of the turbine nozzles and compressor diffuser to compensate for the variations in exhaust gas and air flow. This is limited in it's effectiveness especially in the low-load region because the form of the turbine wheel and impeller cannot be similarly changed. After exhaustive studies of various turbocharging possibilities and consideration of the ways other manufacturers tackled this problem, the conclusion was reached that a most efficient solution would be to use a number of small turbochargers which can be 'sequentially' switched in and out of operation depending on engine speed and power demand. The system is in effect analogous to varying the geometry of the turbine nozzles, but without negatively affecting the efficiency, and is the only way in which the turbine geometry can also be matched to the available exhaust flow. The sequential turbocharging principle was first introduced on the single-stage turbocharged '538' engine series (Fig. 5). The concept of sequential turbocharging necessitated changing the basic turbocharging principle from a

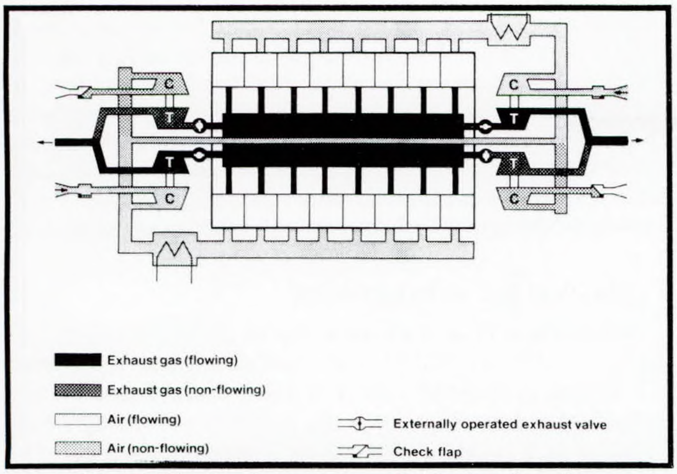

Fig. 5. Single-stage sequential turbocharging

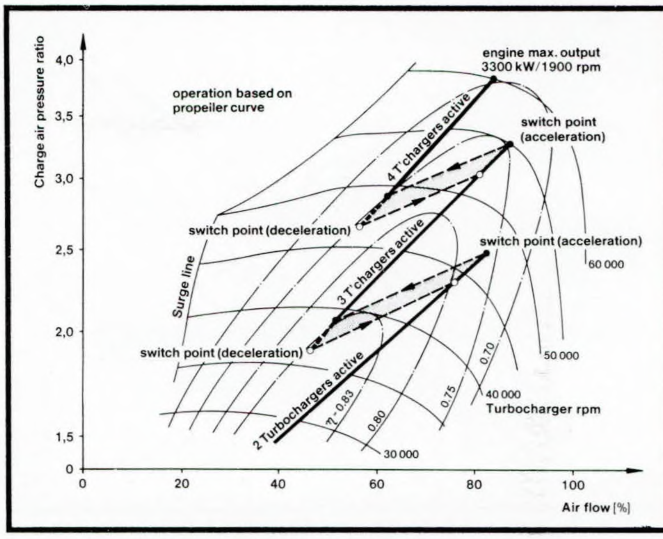

Fig. 6. Sequential turbocharging compressor characteristic with engine operating lines 16 V 538 TB 93/4 x ZR 170

'pulse' system to a 'constant pressure' system. Normally, MTU diesel engines with conventional charging systems work on the 'pulse' principle, whereby the exhaust outlets from particular cylinders are separately connected by short pipework to the turbochargers. The arrangement is such that the peak pressure pulses from the cylinders act in unison with the result that, especially in the part-load range, the maximum available exhaust energy can be utilized to drive the turbines. In this way, the engine performance (fuel consumption as well as torque) is optimized, but this becomes more difficult to achieve with increasing b.m.e.p. The 'constant pressure' principle, whereby the exhaust gases from all cylinders discharge into a common manifold before entering the turbochargers, can be used to give better efficiency at maximum power. However, with a conventional turbocharger arrangement, the efficiency in the part-load range is much worse than can be achieved with a 'pulse' system. The sequential turbocharging system effectively overcomes this disadvantage since the number of turbochargers in operation is always matched to the available exhaust gas flow, and therefore it always operates in a range of optimum efficiency, even at very low engine powers.

The turbochargers are switched in and out of operation by flap valves installed upstream of the turbine and compressor. The exhaust flap is pneumatically opened against spring pressure slightly before the air flap opens, which permits the turbine to accelerate rapidly under no-load. The charge air pressure between the non-return air flap and the compressor decreases, causing the flap to open. To make certain the air flap opens, and therefore ensure that turbine overspeeding cannot occur, a mechanical linkage is arranged between the actuating cylinder and the flap. The air flow provides resistance and retards turbine acceleration. This arrangement also ensures that the switching-in procedure is completed without loss of charge air pressure, and that compressor surging does not take place.

The turbocharger switching-in point for increasing engine speed is higher than the switching-out point for decreasing engine speed (Fig. 6). This hysteresis prevents continuous switching in/out of turbochargers at 'borderline' engine speeds. The exhaust flap valves are loose-fitting to allow a small amount of gas flow in the closed position. This keeps the turbochargers idling at operating temperature so that they are ready for immediate operation and not subjected to thermal shock.

Having proved the sequential turbocharging principle on the 538 engine series, it was a logical step to incorporate this in the 1163-03 design. The operating principle is identical, the only difference being that, in this case, groups of two-stage turbo-

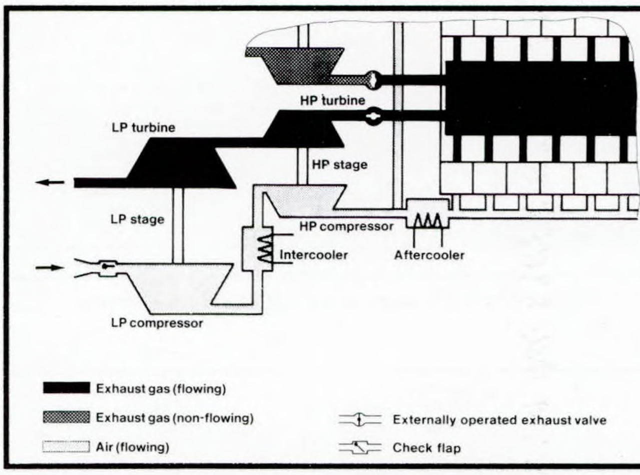

Fig. 7. Two-stage sequential turbocharging

chargers (h.p. and l.p.) are switched in and out of operation (Fig. 7). Figs. 8 and 9 show the performance curves and technical data of the 20 V 1163 TB 92 engine compared with the 20 V 1163 TB 93 engine with sequential turbocharging. The expanded operating range and power increase of the '03' version are clearly illustrated.

Turbocharger design

The former 1163-02 engine series incorporated MTU type 'AGL' turbochargers with axial-flow turbines and radial-flow impellers. Normally, the overall efficiency of several small turbochargers would be less than one or two large turbochargers. However, this disadvantage is compensated by intercooling between the h.p. and l.p. stages. The'ZR'-type turbochargers, originally designed for the smaller engine series, 396 (Fig. 10) and 538, incorporate radial-flow turbines as well as radial-flow impellers. For the 1163-03 engine series, the existing model ZR 170 (impeller wheel diameter) was suitable for the high-pressure stage, and a larger model, ZR 210 had to be designed for the low-pressure stage. The latter rotates in the opposite direction to the former to enable the turbochargers to be arranged in compact modular groups. Since the overall pressure ratio is divided into two stages, with an intercooler between, the air

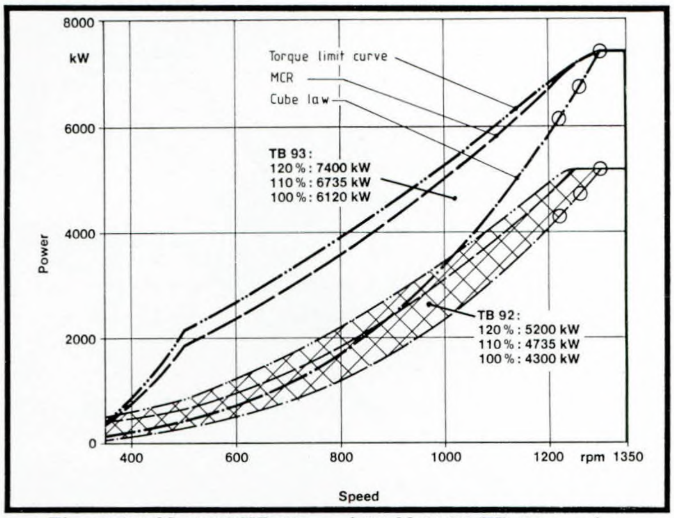

Fig. 8. 20 V 1163 TB 92 and 20 V 1163 TB 93 engine output graph

Sequential Turbocharging and Charge Transfer System								
	without				with			
No. of cyl.	BMEP bar	Output kW	Weight kg	Weight/ Power kg/kW	BMEP bar	Output kW	Weight kg	Weight/ Power kg/kW
	1163-02	Comp. Ratio 12.0:1			1163-03	Comp. Ratio 9.7 : 1		
12	20.6	3120	11400	3.7	29.4	4440	14200	3.2
16	20.6	4160	14350	3.5	29.4	5920	17400	2.9
20	20.6	5200	17050	3.3	29.4	7400	20900	2.8

* without Charge Transfer System

Fig. 9. 1163-03 compared with 1163-02 series

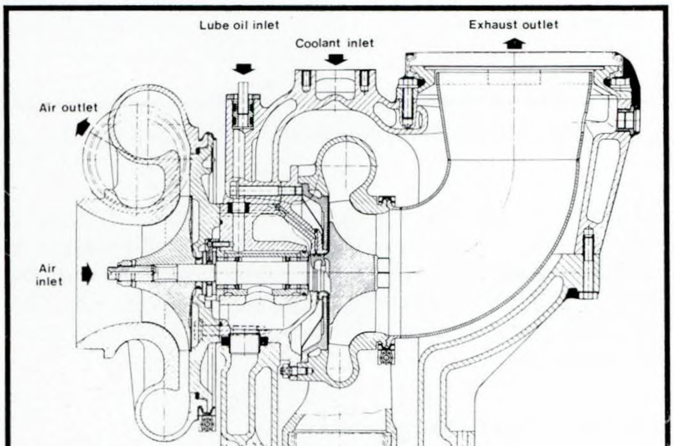

Fig. 10. Exhaust gas turbocharger ZR 170

temperature after compression is relatively low. The impeller wheels and diffusers of both h.p. and l.p. turbochargers could therefore be made from lightweight aluminium alloy. Due to the low moment of inertia of the impellers and turbines, the turbochargers can quickly accelerate and decelerate, thus improving the engine's load-change characteristics.

With conventional systems the turbocharger must be designed, not only for maximum efficiency but also to ensure that the compressor operates sufficiently far away from the surge line at low air flow rates. With the sequential turbocharging system, this design compromise is not so critical, since no individual turbocharger is required to cover the whole engine operating range. More emphasis can therefore be given to designing for maximum efficiency. Due to the lower pressure ratios, the diffusers could be designed without vanes, which, together with the swept-back impeller vanes, results in a wider operating field.

Turbocharger arrangement

For compactness, a modular layout of turbochargers was developed, which could be adapted to all engines in the series (12, 16 and 20 cylinder versions). Each module contains two turbocharger pairs (h.p./l.p.), the turbines themselves being enclosed in a common water-cooled housing (Fig. 11). The construction is neatly integrated with the engine and allows optimum flow of gases and charge air. Apart from the compact arrangement, the installation of the turbines in a common water-cooled housing has the advantage that the connections to the exhaust manifold do not need to be gas-tight, and therefore thermal expansion problems are eliminated. Also, the water-jacket minimizes heat radiation to the surroundings and reduces the surface temperature to meet the requirements of the Classification Societies.

The exhaust manifold housing in the engine vee is also water-cooled, and is fabricated from cast steel sections which can be built up to suit the number of cylinders. The heat-resistant steel exhaust piping inside the housing is installed in short sections which are not gas tight and serve only to provide an efficient flow path to the turbochargers. An air gap between the piping and housing reduces heat dissipation from the exhaust gas. The high-pressure intercoolers (one per cylinder bank) incorporate a preheating section integrated in the engine coolant system. This is automatically actuated under low-load and low-temperature conditions to raise the charge air temperature and provide optimum combustion conditions.

Fig. 12 shows a cross-section of the 03 version compared with the 02 version. The compactness of the turbocharger modules has even led to a reduction in height of the exhaust outlets. All components are contained within the original envelope of the 02 engine, with no off-engine mounted equipment. Fig. 13 shows the modular arrangement of turbochargers on the 12, 16 and 20 cylinder engines, which have three, four and five turbocharger pairs (h.p./l.p.) respectively. Investigations have confirmed that a 24 cylinder engine is technically feasible by using six pairs of turbochargers. This engine could be produced if the market demands.

Turbocharging control and monitoring

To ensure safe and correct operation of the turbochargers the following parameters are monitored.

1. Rotational speed of each h.p. turbine
2. Engine speed
3. Fuel rack position
4. Charge air pressure
5. Position of air intake flaps ('open' or 'closed')

The engine speed and fuel rack position are determined by an electronically controlled hydraulic governor. The electronic control system is programmed to limit the fuel rack position depending on which turbocharger group is in operation (sensed by limit switches on the air intake flaps and the speed of the h.p. turbine).

The various parameters are processed in an electronic control unit (ECS) mounted in the engine room control panel, which

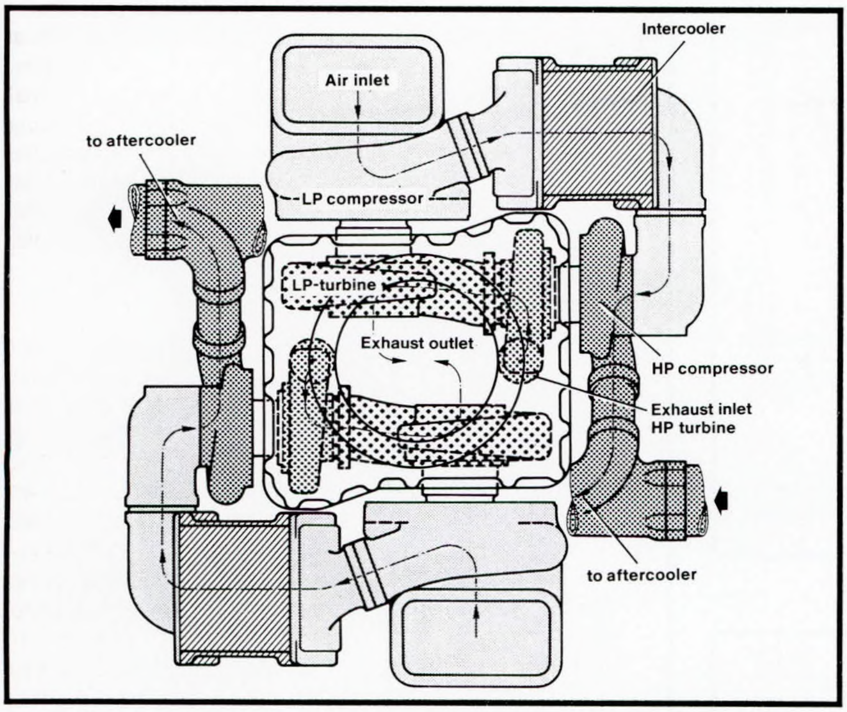

Fig. 11. Two-stage turbocharger arrangement, 1163-03 series

depending on customer's requirements. As an example, the turbocharger control of a 20 cylinder engine intended for application in a four-engine CODAD (Combined Diesel And Diesel) installation will be described. The resulting performance curves are shown in Figs. 15 and 16.

The first charger group to function when the engine is started, is the so-called 'base' group. This is identical to other groups, except that the exhaust and air flaps are locked in the 'open' position. The base group is therefore permanently in operation. The second group to function is activated depending on engine speed only, switching in at approx. 450 rev./min and out at approx. 430 rev./min. When accelerating, the third charger group can only be activated when both the required engine speed (700 rev./min) and charge air pressure (1.2 bar) are available. The reason for this is to ensure optimum acceleration and specific fuel consumption when operating off the cube-law curve near the maximum continuous rating (MCR) line. When decelerating, only one parameter (either an engine speed of 670 rev./min or a charge air pressure of 0.65 bar) must be fulfilled before the third charger group can be switched out of operation. When operating on the propeller curve, the charge air pressure is the over-riding parameter, and when operating near the MCR, it is the engine speed. The fourth and fifth turbocharger groups are activated only depending on charge air pressure, switching in at 2.4 and 3.1 bar, and switching out at 1.7 and 2.25 bar respectively.

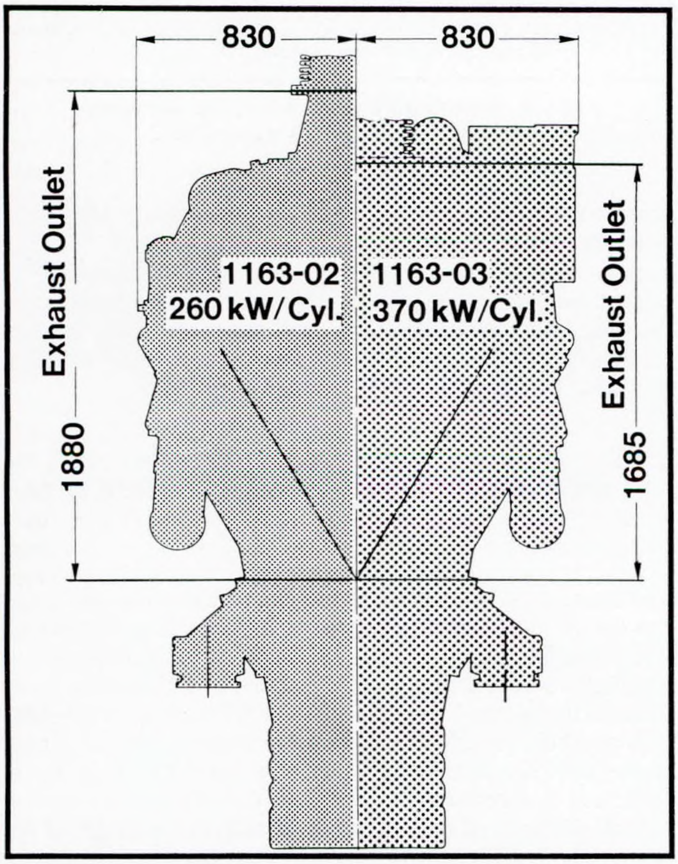

Fig. 12. Comparison of engine sizes 1163-02/1163-03

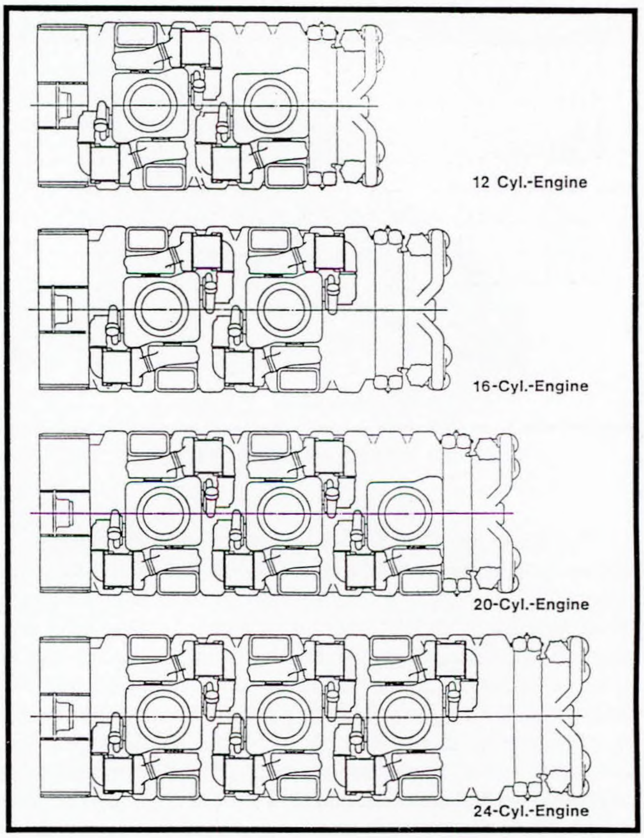

Fig. 13. Modular arrangement of turbochargers, 1163-03 series

controls the solenoid-operated pneumatic valves on the engine to open and close the exhaust and air flaps. Fig. 14 is a schematic of the ECS components and inter-related parameters that are described in detail in ref. 3. The engine performance can be precisely matched to suit the engine application, and optimized for low power running, high power running or fuel consumption,

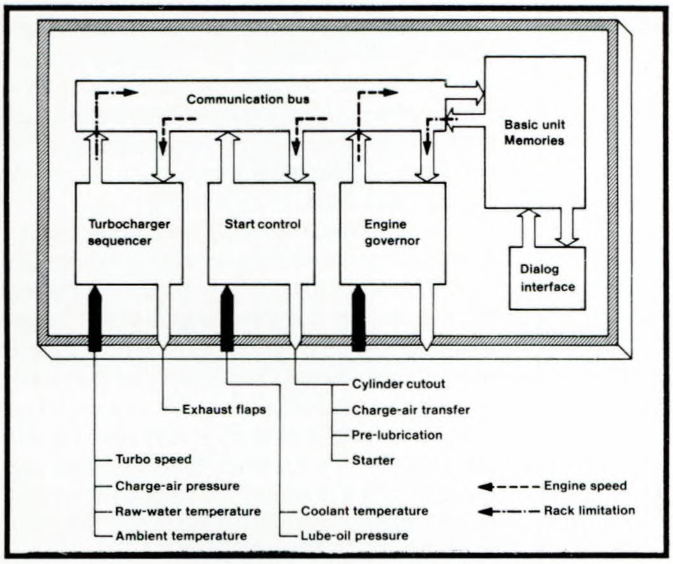

Fig. 14. ECS for series 1163-03

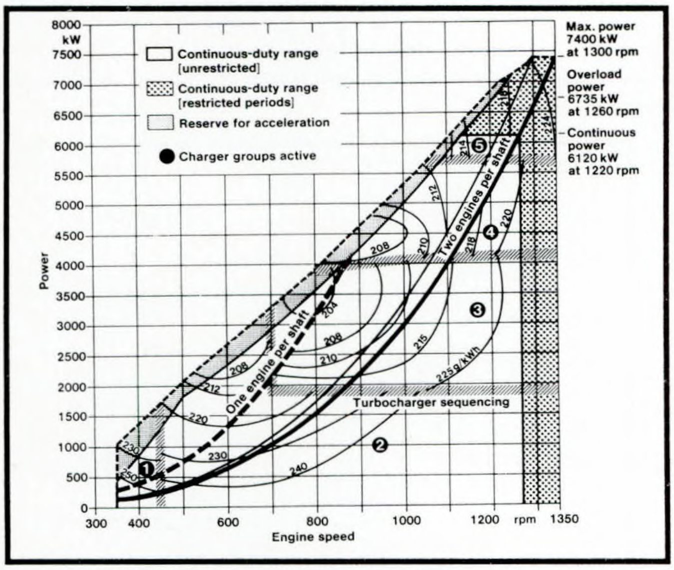

Fig. 15. 20 V 1163 TB 93 performance map

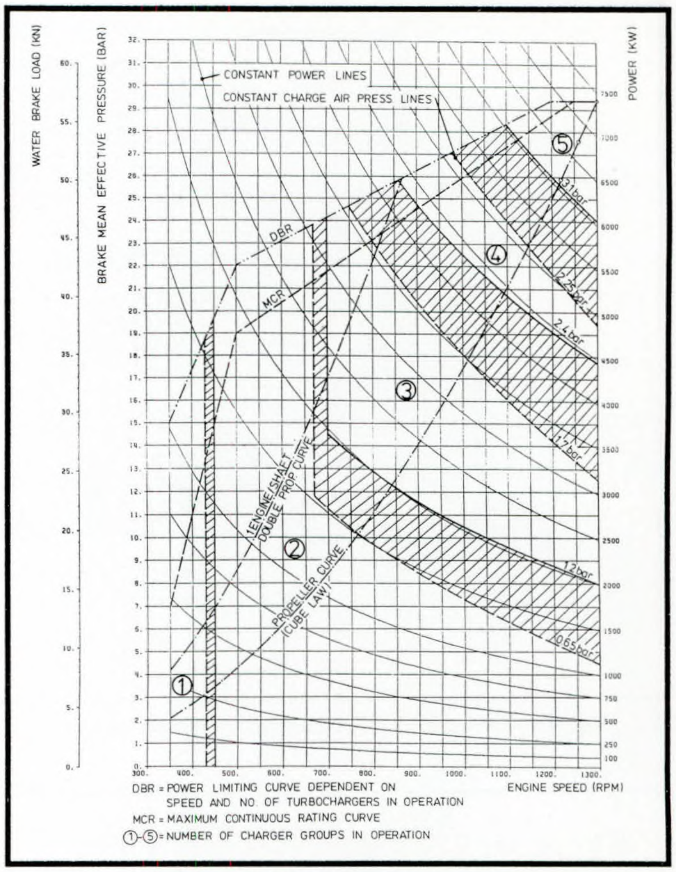

Fig. 16. Typical 20 V 1163 TB 93 performance curves for CODAD operation

The speed gradient and switching of turbochargers is continuously monitored to ensure overspeeding of the turbines does not occur. If the speed tolerance band is exceeded, the fuel rack travel is reduced and an alarm is initiated. In designing the sequential turbocharging system, the highest priority has been given to safety, reliability and redundancy aspects. All sensors monitoring the major operating parameters are duplicated, and the components themselves are of simple, well-proven design. The multi-turbocharger arrangement affords better redundancy compared with a conventionally turbocharged engine. Should failure of a charger unit occur, this can be cut out and the sequencing system rearranged so that the engine is still able to develop approximately 80% power. The changeover procedure can be effected by on-board staff within a short time.

Should failure of the 24 V D.C. power supply for the electronic system occur, the engines can still run since the solenoid valves controlling the air supply to the flap actuating mechanism are open in the de-energized position. The only effect is that the second charger group (normally activated depending on engine speed only) is continuously in operation with the base group, and the third group is activated only dependent on charge air pressure. This leads to a slight increase in fuel consumption in the low-load region, but has no influence on starting behaviour or the maximum power available. A back-up system to take into account failure of the compressed air supply is not considered to be essential, but can be offered if specified by the customer. In this case, alternative flap actuating cylinders are fitted that are capable of operation by using charge air pressure as the working medium.

IMPACT ON PROPULSION SYSTEM DESIGN

The uprated 1163-03 engine series has opened up new possibilities for the ship designer, due to the increase in power/weight and power/bulk ratios and the improved torque characteristics. The higher power density is a benefit not only for fast naval craft, but also for yachts, hydrofoils and SWATH (small waterplane area twin hull) vessels. The latter is illustrated in Fig. 17, which shows that the space around the 20 V 1163 TB 93 engine is sufficient to undertake top overhauls of the engine *in situ*. The outside diameter of only 3.8 m in relation to the length of the floating body does not increase the form resistance. The clear width of 1.8 m of the struts enables removal of the engine for main overhaul. Fig. 18 shows the propulsion system of an existing fast patrol boat, using four 16 cylinder MTU 956 engines, developing a total power of 13,200 kW. Using the 1163-03 series, the same boat could now be powered by three 12 cylinder engines with consequent saving in weight and space. The 1163-03 engine is ideally suited for the propulsion of medium-sized frigates.

For higher power requirements, a combined system using

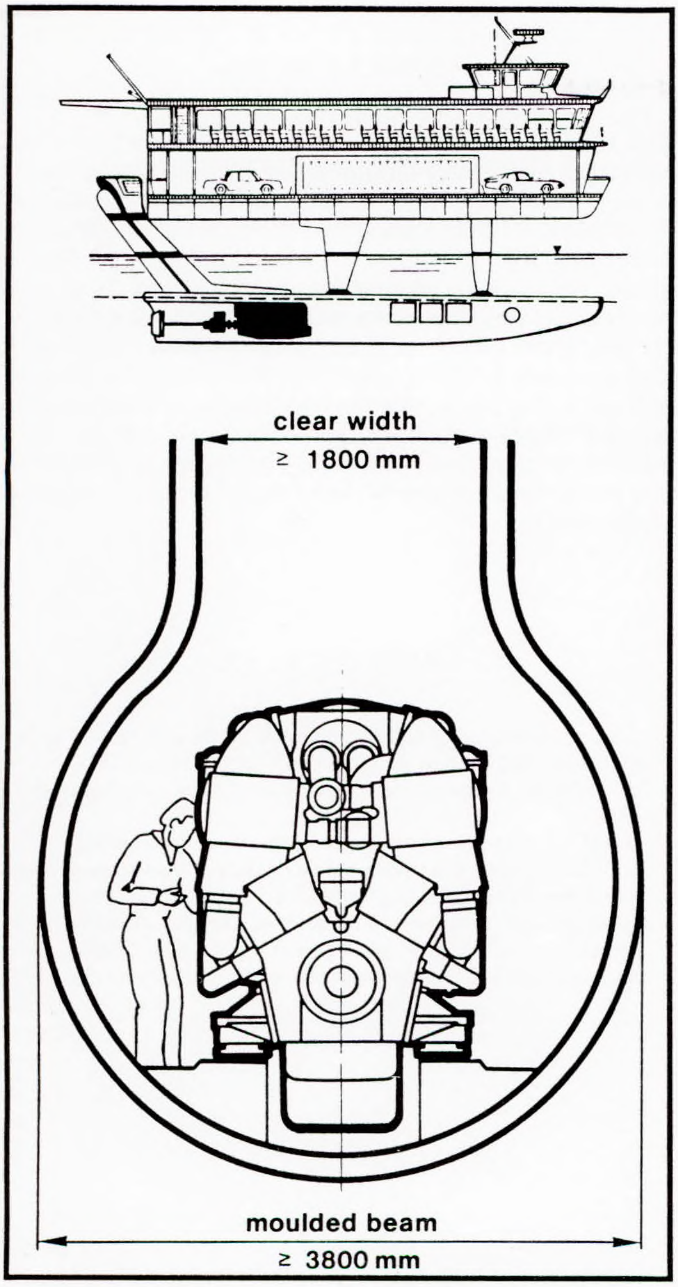

Fig. 17. Cross-section through SWATH strut with 20 V 1163 TB 93 (7400 kW/10000 h.p.)

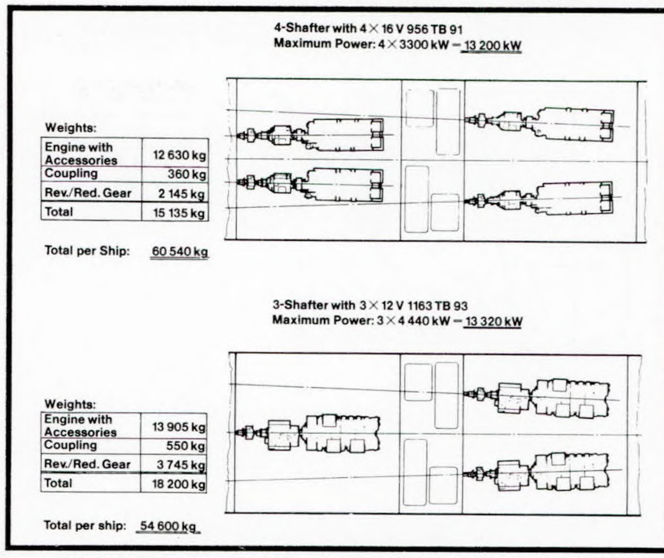

Fig. 18. Propulsion plants for 400 t fast patrol boat

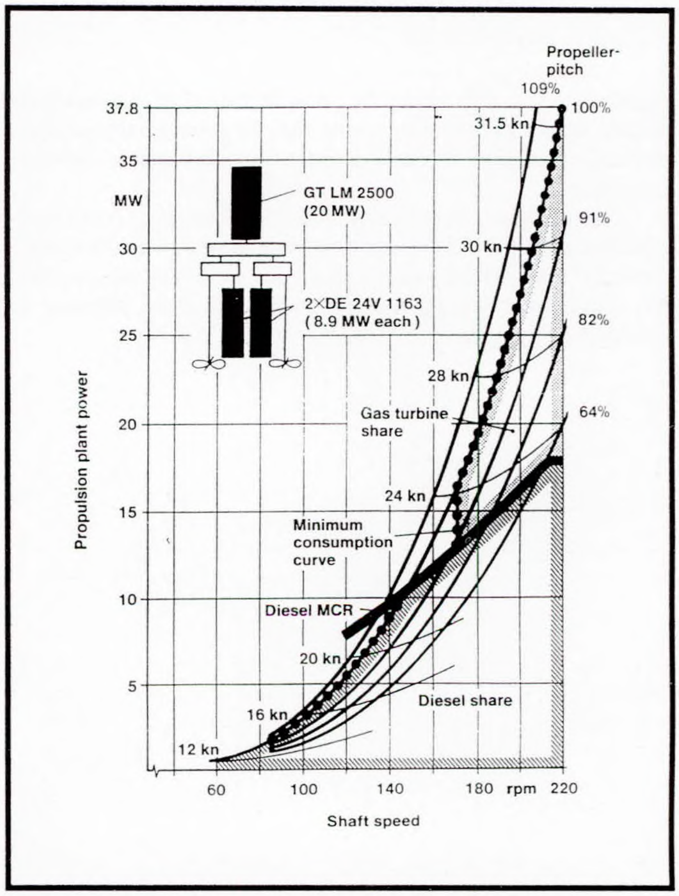

Fig. 19. CODAG propulsion plant with single-speed gearbox for 4000 t frigate

diesels and gas turbines is often installed, the diesels for low-speed and cruising duty because they have a lower fuel consumption than gas turbines operating at part load. The arrangement is normally CODOG (Combined Diesel Or Gas) with the full power of approx. 40,000 kW being delivered by two gas turbines. Since the operating profile of such a frigate generally results in very little running on gas turbines, a CODOG is not the most cost-effective installation. With the 1163-03 series, a very simple CODAG arrangement is now feasible, using two diesels, only one gas turbine and a single-stage gearbox. Formerly, CODAG systems were unpopular, mainly due to the fact that a two-stage gearbox was necessary because of the limited torque availability of the diesels. Fig. 19 shows a theoretical installation using two 24 V 1163 TB 93 diesels and one LM 2500 gas turbine to give a total power of 37,800 kW. The performance curve shows that a 4000 ton frigate could be driven at 23 knots in the diesel mode with slight reduction of propeller pitch, 27 knots in the gas turbine mode, and over 31 knots in the combined mode. With modern electronic technology, the control and monitoring of such a system presents no more difficulty than a CODOG system.

Another example which demonstrates the advantages of the improved torque characteristic of the 1163-03 series, is shown in Fig. 20. The resistance curve of a hydrofoil has a very pronounced 'hump' during the transition phase before the vessel becomes completely foil-borne. In some cases, the torque

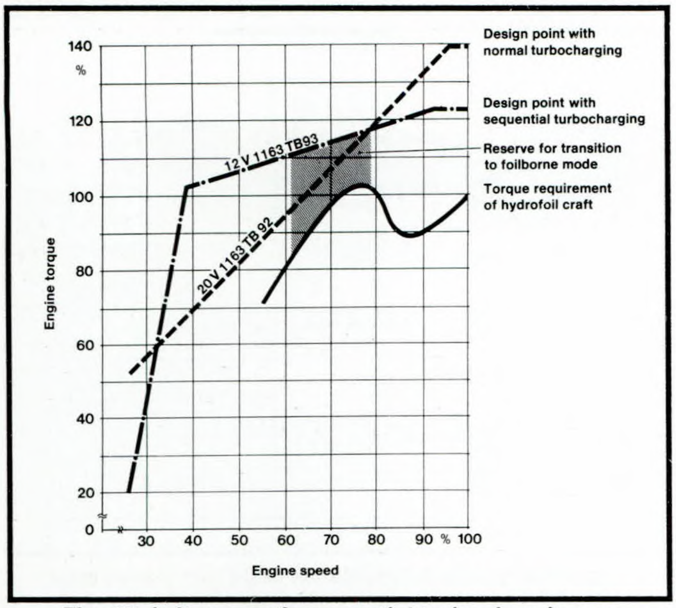

Fig. 20. Influence of sequential turbocharging on hydrofoil propulsion

requirement at this point is even higher than at maximum speed, which previously meant that relatively large engines had to be installed, the maximum power of which could never be utilized.

The example shows that a hydrofoil previously powered by two 20 cylinder 1163-02 engines can now be powered by two 12 cylinder 1163-03 engines. Apart from the initial cost savings, the consequent weight and space saving could be used for additional payload.

CONCLUSIONS

The introduction of the sequential turbocharging principle represents a major step forward in compact diesel engine technology. The development of higher-powered engines that can be installed in even less space is a challenge for engine designers – even more so when this is to be achieved without degrading the torque characteristics, the fuel consumption or the time between overhauls. In-field experience with many engines testifies to the success of this system, so much so that nowadays all the latest engine models in MTU's range (with the exception of the small 099 and 183 series) incorporate sequential turbocharging using the 'ZR' series of turbochargers. Combined with the most modern microprocessor technology, the operating characteristics of these engines can be 'tailor made' to suit the intended application.

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Discussion

S. G. CHRISTENSEN, C.Eng., F.I.Mar.E., B.Sc.: My contribution to this very interesting paper is more a series of questions made in an endeavour to learn more about this little publicized engine of high output, perhaps more importantly, where this high power output has been squeezed into what must be cylinders of relatively small linear dimensions.

Many of us formulate an idea of an engine by knowing its bore, stroke, and rev./min; the figures for the bore and stroke are missing both in the synopsis and in the paper. A figure for piston displacement given in some cubic measurement without a bore/ stroke ratio is not enough. From the data given we start by thinking the cylinder must over-square, the rest is conjecture.

The note on the reduction of the compression ratio is interesting, it was noticed in a later part of the paper that the fuel injection period had been reduced from 50 degrees to 30 degrees. Fig. 2 shows the injection period finishing earlier for the 30 degree period than for the 50 degree period. Can the author give us some idea of the combined effect of the change of compression ratio, the effect of earlier injection completion on the expansion ratio, and the value of its ultimate effect on the specific fuel consumption?

It is thought to be an excellent idea to keep the turbocharger rotor spinning when it is in a non-operative condition and the engine is operating with a low-power output.

It has been noticed on some large medium-speed engines kept in a state of immediate readiness for emergency duty that delay in rotor bearing lubrication on start-up led to early rotor bearing problems. The bearings were of the sleeve-type. This led to the necessity of supplying lubricant continuously to the rotor bearings when the engine was shut down; a secondary problem then arose with the rotor bearing sealing devices due to lubricant leakage into the other parts of the turbocharger. These problems were finally overcome by replacing the sleeve bearings with ball and roller bearings.

The paper states rotors are kept spinning to prevent thermal shock when the turbocharger is brought into use. Were there any other reasons for keeping the rotor spinning, such as problems similar to those just mentioned?

It is noticed in Fig. 10 showing the ZR 10 turbocharger that sleeve- or journal-type bearings are used. Is there any reason for the use of this type of bearing in preference to anti-friction or ball-and-roller bearings? Ball and roller bearings may be operated at a much higher temperature and therefore would appear to be safer. Cost and ease of availability is an obvious factor in the choice of sleeve bearings. Were there any other reasons?

Referring again to Fig. 2, detailing the improved combustion process, although the paper is on sequential turbocharging, the improvement in the combustion process raises many points of interest. It appears to be almost a paradox that one can increase the rate of injection by approximately 70% without increasing the maximum combustion pressure, even though some compensation is obtained from retarding the start of injection. Examination of the injector needle lift plotted on the time base shows the lift period has been increased from a little more than 1 ms to something slightly in excess of 1.5 ms. This obviously has some bearing on the improved combustion. When the slope of the plot of the injector needle lift is examined, it appears that the acceleration of the fuel pump ram has been considerably reduced, this then allows the supposition that something akin to a form of pilot injection has been obtained, thus improving the overall combustion. When problems arising out of the relationship of hertzian pressure and fuel-pump cam life are known, one may ask, why was the fuel-pump cam not designed in this manner from the start? Does cam geometry have any significance on hertzian stress in high-speed engines where the fuel injection pump parts appear to be much smaller than in low-speed and medium-speed engines? Can Mr. Herring tell us if the fuel-pump cam was in fact altered, and if any other changes were made in the injection system, such as an alteration in nozzle hole diameter, streamlining of the injector nozzle holes by chamferring or radiusing at the sac end of the hole to improve the coefficient of discharge, and/or alterations in piping elasticity and the like?

Another interesting point on this diagram is the negative slope shown on the rate of cylinder pressure rise for the 30 degree injection period. This occurs just before ignition, but at the point in the piston stroke where it is still rising 'slowly'. It is not know if this curve is based on a mathematical model for finding the rate of heat release or whether it has been obtained from the output of a piezoelectric transducer fed into a computer and then plotted after the cylinder pressure value has been divided by time. It might be expected to have some flattening-off effect of the rate of cylinder pressure rise due to the heat requirements for fuel particle surface vaporization. Perhaps Mr. Herring can tell us more about these points.

Turbocharging and the scantlings of crankshafts have an indirect relationship due to the increase in the mean effective pressure. The increase here is from approximately 21 bar to 30 bar. Maintaining the maximum combustion pressure at its original value will hold the range of alternating stress at about the same value as before. Obviously there are other unknown factors involved in this, but was it necessary to upgrade the crankshafts in any way by changes in dimensions, by material changes allowing higher working stresses, or by changes in surface hardening techniques, so giving a higher fatigue resistance?

It is noticed in the multi-engined geared propulsion drivers that controllable-pitch propellers are used in the machinery installations shown. With such arrangements, any engine may be operated under its designed conditions of power output and engine speed when operating alone. Has anybody considered using solid or fixed-pitch propellers with multi-engined geared propulsion installations and MTU 1163 engines? With such propellers the cylinder pressures will usually extend beyond the crankshaft design limits when operating on only one or less than the total number of installed engines unless the engine power is considerably reduced. Will the engine surveillance system and control equipment take care of such an operation and prevent the crankshaft from being overloaded?

On a final note, Mr. P. Herring and his company must be thanked for this interesting paper with the compliment that only such an interesting paper will elicit so many questions.

H. NIVEN, B.Sc.Eng., C.Eng., F.I.Mech.E. (Ricardo Consulting Engineers plc): I note that on the 03 version of the engine a heating section is incorporated into the h.p. intercooler, which is activated when the boost pressure drops below 1.6 bar. I presume that this is to help maintain regular and stable combustion at low load, and to control the rate of pressure rise within the cylinder.

Do you know what rate of pressure use (bar/crank degree) you achieve at light load to avoid detonation damage to cylinder components? We like to see a rate of pressure rise value of not greater than 10 bar/crank degree. We have measured values in excess of 30 bar/crank degree under part load, part speed conditions with very low air manifold temperatures, and such high values inevitably lead to detonation damage to the cylinder components.

Is the injection timing fixed, or do you have a light-load retard helix on the fuel pump plunger?

Lastly, could you tell us what the valve overlap is please?

J. M. THOMPSON (YARD Ltd.): The basic concept of two-stage turbocharging to obtain the highest power from a minimum size and weight of engine is not new, but the number of designs which have been developed to the point where they can be offered for general sale are very few. The author has described the only system of which I am aware that successfully combines high power with a good operating envelope, low specific fuel consumption throughout the operating range, and straightforward installation. The long list of sales described during the presentation is indicative of the good reception this somewhat specialized engine has obtained.

The only reservations I have are on the reliability and durability of the engine, particularly if full use is made of its capabilities. As others may share my concern I would be interested to have the author's comments on the following points.

(a) From Fig. 15, there is clearly a fuel consumption advantage in operating at or close to the left-hand boundary of the engine envelope, and indeed the author recommends this procedure in his description of a CODAG plant. As this is normally an area where high mechanical and thermal loads are encountered:
   (i) what parameter(s) set the limit of operation shown;
   (ii) what is the effect on bearings, exhaust valves, etc. of operating in this way compared with operation on the cube law?

(b) The 'base' turbocharger group is always in use, whereas the last group comes into use only at very high power and the remaining groups have intervening amounts of operation. Do MTU recommend periodically resequencing the order of operation to average out usage or are different turbocharger maintenance intervals recommended according to the preset sequence and operating profile?

J. TURVEY [Vosper Thornycroft (U.K.) Ltd.]: In the presentation the author mentioned white smoke, would he please elaborate on and quantify the smoke situation, both white and black?

M. J. NEEVES (Rolls-Royce plc): I would like to congratulate the author on a most interesting paper. It is fascinating to see the areas of similarity in the way that high-performance diesels and marine gas turbines are developing. The maintenance philosophy is similar with modular exchange in gas turbines corresponding to top overhauls in diesels and with removal for major overhaul recommended for both types of machine. Modern electronics technology, as the author points out, makes possible much more sophisticated controls and monitoring, whichever propulsion system is chosen. There are still significant areas of difference however. The gas turbine technologists are trying to match diesel fuel consumption characteristics with advanced cycle machines and the diesel designers, as described here by the author, are trying to approach the favourable torque characteristic of gas turbines.

It is a fact that there are, currently, still differences that make CODOG and CODAG systems attractive, and the author describes an interesting CODAG concept in his paper. I agree with him that CODAG makes economic sense when high-powered diesel cruise engines are installed - it seems wasteful not to use all the available power for the ship's full-speed condition. However, I believe he dismisses the possibility of two-speed gearboxes too lightly. In his example, a two-speed gear would permit the ship to do something like 25 kt on diesels before needing to start the gas turbine, and I do not think the system controls would be any more complex than those required to handle his propeller pitch scheduling. Admittedly an additional primary gear reduction would be needed, but only a small number of components similar to those used in the CODOG system would be required, as shown in the example in the diagram which shows one shaft set of machinery. I would be interested in the author' comments.

Author's reply

In answer to **S. G. Christensen**: I would like to thank Mr. Christensen for his keen interest in this paper, which is reflected in his in-depth questions.

The 1163 series has a bore/stroke ratio of 230/280 mm compared with 230/230 for its twin, the 956 series.

The problem of lubricating oil supply to the turbocharger bearings of quick-starting emergency generating sets is appreciated. For such an application the 956/1163-03 engines are continuously lubricated during the shutdown period, but a pressure control valve limits oil flow to the rotor bearings. When the engine is running, rotor sealing of all turbochargers (active and inactive) is supplemented by pressurizing the sealings with charge air.

The reasons for keeping the inactive rotors spinning are mainly to avoid thermal shock, and to permit rapid acceleration when switched into operation.

During the engine design phase, MTU chose 'floating bush'-type bearings since at the time, ball bearings could not satisfy the technical requirements. The main advantages were:
- good vibration and shock damping characteristics due to the double oil film;
- long life;
- high reliability;
- fine oil filtration unnecessary (30 mm adequate);
- cost and availability.

Most axial turbochargers utilize ball bearings but in this case, the heat source (turbine rotor blades) is more remote from the bearings. With radial turbochargers, the closer proximity of the heat source can present a problem for ball-bearings.

Mr. Christensen has read a great deal into Fig. 2, which is really only intended to illustrate the principle effects. The improvement in the combustion process is largely due to the delayed opening of the injector (the nearer TDC the better). The needle does in fact close earlier but this does not greatly influence the end of the combustion phase so the expansion ratio is hardly affected. The delayed injector opening enabled the compression ratio to be increased, and the more efficient combustion (also influenced by air intake swirl) resulted in a slight improvement in specific fuel consumption.

The Figure unfortunately gives the false impression that the rate of injection has been increased. The slope of the lines indicating injector needle opening should be almost identical and no form of pilot injection is intended. The maximum combustion pressure has not been increased due to the retarded start of injection, shorter injection time and improved combustion process. During the initial stages of the development, the existing fuel pumps, injectors, cams, etc. from the former '02' engine version were used. The required increase in output was reached, but the engine could be improved, so new equipment was designed. The injection pump bore was increased from 22 mm to 24 mm and total stroke (cam lift) increased from 15 mm to 24 mm. The nozzle and needle design were also modified and max. injection pressure increased from 1200 to 1450 bar. The cam and roller geometry is independent of engine speed, and is determined principally by the maximum injection pressure. The elasticity of the thick-walled piping is negligible.

If the cylinder compression (without combustion) were illustrated, the curve would be symmetrical, peaking at TDC or at approx. 100 bar. The associated pressure build-up rate would be similar to a sine wave intersecting the x-axis at TDC. The negative slope shown in the curve for the 30° injection period is therefore evident due to the retarded start of the combustion phase. The illustrated curve is derived from data obtained using a piezoelectric transducer and no appreciable flattening-off effect has been observed due to fuel particle surface vaporization.

No modification to the crankshaft was necessary since the design process already takes into account torsional vibration stresses, which, particularly in the case of one cylinder misfiring, are far higher than the increased stress due to the uprated output.

Controllable pitch propellers allow flexibility of operation, fast manoeuvring and are especially suitable for the varied operating profiles of modern warships. Fixed-pitch propellers do not offer such flexibility and demand either in reversing gearboxes or reversing engines for astern operation. When, in a multi-engined installation, one engine/shaft operation is called for, the engines must produce a higher torque for a given speed. Either for economical reasons (optimum SFC and propeller efficiency) or to minimize underwater noise, the CP installation will use full pitch. In this respect, the problems of one engine/shaft running are the same for CP or fixed-pitch propeller installations. For best performance, both demand a diesel engine which can produce high torque at low speed. Control and monitoring of engine speed and power is necessary to prevent overloading. Modern electronic control systems enable ship acceleration and deceleration to be programmed (for CPP and FPP systems) to ensure optimum operation under all conditions. The MCR and DBR (torque limit) curves shown in Figs. 8, 15 and 16 are also monitored to safeguard the engine.

In answer to **H. Niven**: Heating of the charge air using engine cooling water at boost pressures below approx. 1.6 bar is necessary to ensure stable combustion and hence minimize white smoke emission at low load. It also reduces the rate of pressure rise within the cylinder.

A rate of pressure rise of 10 bar/crank degree seems somewhat conservative. MTU has measured values of up to 15 bar/crank degree without observing any detrimental effects.

A degree of light-load retard helix is provided on the fuel pump plunger. This is not so pronounced since the same plunger is used for the twin 956 engine series (which, being used mainly for constant speed, genset application, actually requires no retard helix), hence the design is necessarily a compromise.

The valve overlap is 80°.

In answer to **J. M. Thompson**: Mr. Thompson is correct in pointing out that operation to the extreme left of the propeller curve results in higher mechanical and thermal loading. For this reason, MTU has reduced the torque-limit line for steady-state operation at lower engine speeds in favour of the reliability, in spite of the fact that higher torque easily can be achieved. The guaranteed TBO for a particular engine is dependent on the expected operating profile and is based principally on the life of the con-rod bearings. In Fig. 15, one engine/shaft operation on the MCR curve will reduce the bearing life by 50% compared with operation at the same power on the cube law. However, the resulting bearing life is still longer than the guaranteed engine TBO. In service, extended operation on one engine/shaft is usually carried out at 80% MCR, i.e. in Fig. 16, approximately 3600 kW/730 rev. per min, in which case the bearing life is more than doubled (15,000 hours). Apart from the effect on bearings, operation on the MCR curve also results is approx. 50°C higher exhaust gas temperatures compared with those at the same power on the cube law.

The question regarding resequencing the charger groups to even-out the running hours of the respective chargers is often posed. While this is indeed feasible, it is unnecessary since the turbocharger components and bearings are so designed that virtually no wear takes place. The thermal and mechanical loading is such that the stresses lie well below the LCF (low-cycle fatigue) limit of the material. Pre-lubrication prevents wear of the turbocharger bearings during the engine starting sequence.

In answer to **J. Turvey**: When the engine is operated at very low load, white smoke tends to be produced due to incomplete combustion. This occurs because of poor atomization of the fuel combined with a low compression temperature.

If the 20 V 1163 TB 93 would be allowed to idle (with clutch disengaged) with all 20 cylinders firing, the exhaust would have a hydrocarbon concentration of approx. 1000 p.p.m. With clutch engaged operating at 350 rev./min on the cube law, the hydrocarbon concentration is approx. 650 p.p.m.

In order to reduce the white smoke emission to below the visibility limit of approx. 500 p.p.m., the engine is operated in the idling condition with one cylinder bank cut out. The non-firing cylinders act as a compressor pumping charge air into the firing cylinders thereby increasing the firing pressure and temperature (charge air transfer system).

In addition, the combustion conditions are further improved by raising the temperature of the charge air in the heating section of the h.p. intercooler. By these means, the hydrocarbon concentration is reduced below the visibility limit.

Formerly, the cylinder bank cut-out system was only available for idling when the clutch was disengaged. This meant that for engine operation in the lower part-load range, the hydrocarbon concentration could exceed the visibility limit in the low-speed range of 350–550 rev./min.

The present engines are now equipped with a system allowing extended periods of operation under low-load conditions, incorporating a flap valve which shuts off the charge air supply to the non-firing cylinder bank and at the same time, allows air to be drawn from the engine room. The firing cylinders therefore receive more charge air and the compression temperature is increased.

Black smoke is produced when imperfect combustion is caused by a deficiency of air, i.e. the air/fuel ratio or distribution is no longer optimized. This is normally a transient condition which occurs when the engine is required to accelerate quickly.

To minimize the emission of black smoke, the control system for the 1163-03 engines includes an electronically controlled fuel rack limiting device which is activated depending on charge air pressure. The maximum engine acceleration rate is only marginally limited as the injected fuel is metered according to the available transient combustion air flow. Only the fuel surplus is cut off which would otherwise result in black smoke.

At low power the Bosch no. is approx. 1.0 and at MCR approx. 0.4.

In answer to **J. Neeves**: I am pleased that a representative of the gas turbine industry also supports the CODAG concept. The benefits compared with a CODOG system are so attractive it is strange that the system is not more widespread. I agree that a CODAG system with a two-speed gear involves no greater complexity than a CODOG system, however, if the operating requirements can be satisfied with a single-stage gear, this seems to me to be preferable. The changeover from diesel drive to CODAG and vice versa can then be accomplished without declutching the diesels.

Fig. 19 shows an example of one shipset of machinery incorporating one gas turbine and two diesels, driving two shafts. For higher power (as required for warships up to say 6000 t) a CODAG arrangement could be envisaged using two separate shaftsets, each incorporating one gas turbine and one diesel.

In conclusion, I would like to express my appreciation for the interest shown in this paper.