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ABSTRACT

IMPROVED PERFORMANCE OF COMPACT DIESEL ENGINES AND ITS IMPLICATION REGARDING NAVAL SHIPS

The large number of compact diesel engines so far delivered, in itself proves that these are most suitable for the propulsion of small warships. Over the years, they have always had economic advantages compared to the gas turbine. Recently, this advantage has become even more pronounced because of the fuel situation. Apart from specific fuel consumption, MTU has materialized further developments which will make compact diesel engines even more attractive. New turbocharging techniques result in still smaller engines for a given power, thereby enabling the machinery space to be further reduced and, in addition, the power range of compact diesel engines is extended. A corvette or small frigate can now utilise a CODAD installation with 4x 7400 kW diesels instead of a CODOG or CODAG arrangement, with the consequent advantages of using one type of prime-mover. By adopting a system of sequential turbocharging, principal disadvantages of very high M.E.P.'s are counter-balanced. Fuel consumption, especially under part-load, is improved so that fuel economy at cruising speed is even better, and endurance can be increased. Until now, the torque characteristic of a diesel engine was inferior to that of a gas turbine. This is no longer true for sequentially turbocharged engines, in which a higher intake air pressure is achieved so that more fuel can be burnt and thereby high M.E.P.'s attained. This system also helps to solve problems associated with the burning of low grade fuels.

IMPROVED PERFORMANCE OF COMPACT DIESEL ENGINES AND ITS IMPLICATION REGARDING NAVAL SHIPS

by Dipl.-Ing. Christian Günther

1 INTRODUCTION

It is a fact that diesel engines are the predominant propulsion units for the majority of ship applications. Out of an extremely extensive engine range, this paper is concerned only with compact diesels, that is, those with a rated speed of more than 1,000 rpm and with a power in excess of 400 kW. Due to the inherent high power/weight ratio, these engines are widely used for the propulsion of naval craft.

The history of the fast patrol boat is very closely linked to the development of high-speed diesel engines. However, such diesels have also demonstrated advantages in powering larger vessels such as corvettes and frigates. They are particularly used in combined propulsion plants to provide economic cruising speed.

2 TRENDS IN DIESEL ENGINE DEVELOPMENT

Depending on the different diesel engine types and applications, there are differing priorities regarding future development.

2.1 Increased Output

Higher engine output has always been one of the major objectives of compact diesel manufacturers. This situation arises due to two main reasons:

- Increased power to weight and power to bulk ratios in order to install more power in a given space or to reduce the dimensions of the machinery room.
- Reduced costs per horsepower to be more commercially attractive.

The latter is as important as the former because it is the only counteraction against the high cost of labour.

If the total swept volume of a diesel engine is kept constant, output can be increased by:

- Increasing rotational speed
- Increasing bmep

It is not rotational speed but piston speed which is a limiting factor for engine design. As stroke to bore ratio is more or less fixed within a certain range, rotational speed can be increased without affecting piston speed only by using smaller cylinder dimensions. Since, for operational reasons, the number of cylinders is somewhat restricted to a maximum of say, 24 for a V-type engine, it is evident that increased rotational speed will not be the most influential means of achieving higher power.

The other alternative, namely increased bmep, is therefore the area where all engine manufacturers concentrate their research and development work. In Fig. 1, two methods of increasing mean effective pressure are shown. With a given compression ratio, this can only be achieved by increasing the maximum cylinder pressure. If we want to keep the maximum pressure at the same, acceptable level, an increase of compression volume, which means a reduced compression ratio, is the only method.

Fig. 2 shows the correlation between mean effective pressure, charge air pressure and compression ratio. The figures relate in particular to the 1163 engine but the general principle applies to all engines.

Increasing mean effective pressure from 21 to 29 bar would lead to a maximum ignition pressure of approximately 200 bar with the existing compression ratio of 12 unchanged. Such a high pressure cannot be tolerated in existing engines and even in future designs this pressure will be limited since then the increased cylinder wall thickness might give rise to unacceptable thermal stresses.

For this reason, the maximum pressure must be kept more or less constant at a value of approx. 150 bar (which is valid for all existing engines). To achieve an mep of 29 bar therefore, whilst maintaining 150 bar max. pressure, the example shows that the compression ratio must be reduced to 8.5. The penalty for reducing the compression ratio however, is that the engine is normally

difficult to start under adverse conditions and idling tends to be erratic due to incomplete combustion. During recent years, engine builders throughout the world have evolved different methods to overcome these starting and idling problems; however, the systems used so far have been rather complex. At MTU we aimed to develop a simple solution using well-known and proven components.

2.1.1 Cylinder Cut-out System

Since 1975, MTU engines have been equipped with a cylinder cut-out system, Fig. 3. This was first introduced for rail traction engines to solve the problem of smoke emission while idling in railway stations. The fuel pumps of one bank are kept at zero-delivery position by means of a lube oil operated piston so that the remaining cylinders are supplied with approximately double the amount of fuel according to the command of the governor. The combustion process in this case is much better than with all cylinders firing. By this method, the idling problems are solved.

The cylinder cut-out principle is very simple for all engines with one block injection pump per cylinder bank or with individual injection units but it can also be applied to engines with a single block-type injection pump (Fig. 4). In this case the fuel rack is sleeved so that relative movement between the two parts is possible and half of the plungers can be kept in zero-delivery position. In this case the firing cylinders must not necessarily all be in one bank; on the contrary, it is possible to select the appropriate cylinders according to best firing order. Because of the greater angular distance of firing between two consecutive cylinders, the mechanical load on the crankshaft is higher than if all cylinders were firing. However, as power output is minimal while the cylinder cut-out is in use, this does not cause any harm bearing in mind that this operating mode has already been considered in the selection of the flexible coupling. Since the cylinder cut-out function is used only while idling, it is convenient to take the initiating control signal from the gearbox (neutral position). In other arrangements where there is no neutral, for instance diesel electric plant, a fuel-rack position signal can be taken.

If an engine is equipped with a cylinder cut-out device, the non-firing cylinders can then be used as an additional starting aid.

2.1.2 Charge Transfer System

With the charge transfer system shown in Fig. 5, a non-firing cylinder is used to increase the pressure of its opposite number in the other bank. By means of an externally controlled valve, compressed air is transferred via a small pipe to the corresponding cylinder. Thus the pressure in the 'power' cylinder is increased to enable proper starting. In Fig. 6, the effect of charge transfer can be seen. Of course, this system is only applicable within a certain range of 'V' angles.

The necessary valve is based upon the standard compressed air starting valve and has already proven its durability in a 500 hour-test which corresponds to an actual engine service time of more than 10,000 hours.

An extension of the charge transfer system has been investigated on research engines in which 2 or 3 cylinders in series compress the air fed to a firing cylinder. This improves the starting capability at low ambient temperatures or reduces the necessity of pre-heating an engine. The final compression temperatures reached with the two different methods of charge transfer are shown in Fig. 7.

The two systems just described (cylinder bank cut-out and charge transfer) permit starting and idling of a highly supercharged diesel engine with a low compression ratio without any additional energy from an external source.

2.1.3 Sequential Turbocharging

Inherent in highly supercharged engines, especially with single-stage turbocharging, is the comparatively narrow range of operation which can be allowed due to turbocharger matching. In order to extend the operating range of the engine, that is, increase the torque availability at low speed, MTU has introduced a new method of turbocharging, called 'sequential' turbocharging. As can be seen from Fig. 8, flap-valves are located at the turbine and compressor inlets for switching turbochargers into and out of operation. In the high power range, all turbochargers are active.

The switching off of a turbocharger is initiated by the actuation of the exhaust flap-valve positioned before the turbine inlet.

When the turbine is no longer subjected to exhaust gas pressure, the turbocharger speed drops rapidly, and the air flow is discontinued. The charge air from those turbochargers still subjected to exhaust gas pressure attempts to flow through the compressor of the inactive turbocharger in the reverse direction. This is prevented by the non-return flap located at compressor inlet. The loose fitting flap-valve on the turbine inlet allows a small amount of exhaust gas to flow to the turbine in order to keep the rotor free-wheeling and at operating temperature.

The exhaust gas supply to those turbochargers still in operation is increased when one turbocharger is switched off, thus providing the desired improvement in the charge air supply at constant loads and engine speed. When the engine speed is raised, the absorption capacity of the turbines must be increased. To achieve this, the exhaust flap-valve of an inactive turbocharger is reopened. The number of turbochargers in service is dependent on engine speed alone as long as charge air pressure has reached a certain value.

As can be seen in Fig. 9, sequential turbocharging means that the chargers themselves operate in their best efficiency range during the whole field of operation. This in turn leads to reduced fuel consumption, especially under part load conditions.

Sequential turbocharging may be used equally well for both single and two-stage turbocharging systems (Fig. 10). In the case of two-stage turbocharging, flaps are needed only at the inlet of the high pressure turbine and at the inlet of the low pressure compressor.

Until now, most large diesel engines have been equipped with turbochargers having an axial turbine wheel and a radial compressor

wheel. With sequential turbocharging, smaller units are used, having both radial turbines and radial compressors. These smaller chargers have so far only been used together with smaller engines, for instance with MTU engine series 331/396. With the introduction of sequential turbocharging, all MTU engines throughout the whole production range will incorporate turbochargers of an identical design (Fig. 11). One notable advantage is that both wheels, being integrally cast pieces, are less susceptible to blade vibrations and therefore permit higher circumferential speeds.

In addition to easier handling during maintenance, these small chargers can be compactly installed on the engine and permit finer staging and rapid acceleration when being switched into operation. The turbochargers are integrated into the intake air and exhaust sides of the engine's charging system. Instead of the pulse system which up to now was always employed on MTU engines, the charging system has been simplified by adopting a constant pressure system. This has the advantage of low fuel consumption at its design point but it is able to cover only a small range with acceptable efficiency. With sequential turbocharging, however, this range is widened by the different number of turbochargers in service thus enabling the constant pressure system to be used. The exhaust system itself is also much simpler compared to that of a pulse charging system or to any other because there is only one common duct for all cylinders.

All components necessary for charge transfer and sequential turbocharging are of simple and well-proven design and have been thoroughly tested.

2.1.4 Systems Arrangement

The turbocharger arrangement and the constant pressure exhaust manifold are very interesting from the design point of view.

The arrangement of the 5 turbochargers on the single-stage turbocharged 20 V 538 is shown in Fig. 12.

The two-stage turbocharging groups are attached to the 956/1163 engine in a modular arrangement (Fig. 13), identical components being used on 12, 16 and 20 cylinder engines. Each module comprises two pairs of high and low pressure turbines mounted in a water-cooled enclosure with a common exhaust outlet. By incorporating all hot parts inside the insulated module, the external temperature of the whole system is kept low.

The very simple design of the constant pressure exhaust manifold is shown in the sectional drawing (Fig. 14). It is manufactured from heat-resistant sheet steel sections each covering 4 cylinders, which are interconnected by sliding sleeves. The space between the exhaust manifold and the water-cooled walls of the housing is filled with insulating material which reduces heat dissipation, thereby retaining the energy of the exhaust gas. The identical exhaust elbows prevent gas flow disturbance between opposite cylinders. The connecting elements, designed as plug-in pipes, ensure that the gas carrying components are free

to expand individually. These connections need not be absolutely gas tight since the surrounding water jacket also serves as a gas seal. Expensive bellows joints are therefore unnecessary. Fig. 14 shows in addition the design and the location of the intake and exhaust flaps of the sequential turbochargers as well as the components for charge air transfer. Note the small diameter of the interconnecting pipe relative to the cylinder diameter.

This new development results in a considerable increase in power (Fig. 15). Instead of 260 kW per cylinder, the 1163 engine is able to deliver 370 kW per cylinder, an increase of more than 40 %. The 20 V 1163 TB 93 for fast patrol boats as 1 DS version, is able to deliver 7,400 kW (10,000 HP) and has a low weight/power ratio of 2.6 kg/kW (1.9 kg/HP). This represents a big step forward in diesel engine technology. It is important to note that this power increase is obtained without an increase in dimensions.

In fact, the overall height of the 956/1163 with two-stage turbocharging is less than that of the single-stage charged engine. It should be noted that all turbochargers, both low and high pressure, are integrated within the overall engine dimensions, and there are no 'off-engine' mounted components. As shown in Fig. 16, the exhaust outlet is approx. 200 mm lower which is obviously favourable for all installations.

As explained previously, mechanical stresses are kept to a constant level by using a low compression ratio. Thermal loads normally increase with higher engine output. However, in the case of the 1163 with two-stage turbocharging, it was possible to increase the air fuel ratio which results in a notable decrease in temperature despite the higher heat flow (Fig. 17).

2.2 Extension of the Operating Range

As mentioned before, the narrow range of operation which is a consequence of high pressure turbocharging, can be extended by using the sequential turbocharging system.

The increased range of operation is illustrated in Fig. 18 which shows the performance of a 16 V 538 TB 93 research engine fitted with a single-stage sequential turbocharging system.

Because of the ability to regulate the number of turbochargers in service, mep can be kept at a very high level throughout the whole range. For each number of turbochargers in operation, there is an optimum fuel consumption area.

2.3 Fuel Consumption

Fuel consumption has always been important because of its effect on ship endurance. This was in many cases the decisive factor in choosing a diesel engine instead of a gas turbine. In recent times, however, the fuel price situation has meant that ways to improve the specific fuel consumption have had an even higher priority in diesel engine research and development.

Fig. 19 shows the improvement in overall fuel consumption of a sequentially turbocharged engine compared to a conventional engine. The reduced fuel consumption at maximum power is the result of using a constant pressure turbocharging system. Such a system

can only be used favourably in combination with sequential turbocharging, which ameliorated the otherwise poor part-load consumption. The improvement in part-load operation is due to the fact that the operating turbochargers are always running in their range of optimum efficiency. This comparison is valid for single-stage turbocharged engines with similar compression ratios. With two-stage turbocharged engines, the sequential turbocharging system tends to counter-balance the lower efficiency which is due to the reduced compression ratio.

In Fig. 20, the figures for engine power and specific fuel consumption for the 20 V 1163 TB 93 are shown. The fuel consumption is comparable to published data for other two-stage turbocharged engines with low compression ratio. Further development work is presently being carried out with the aim of reducing fuel consumption and the results so far are very promising.

The greater flexibility afforded by sequential turbocharging allows the engine designer to optimise the engine according to specific customer requirements and therefore have the best fuel economy in the operating range where the engine is most frequently used.

2.4 Operation with Fuels of Various Grades

Until now, declining fuel quality does not seem to have been such an acute problem with naval applications as for instance in the merchant marine field. Nevertheless, the diesel engine of the future will have to cope with lower fuel specifications. Certainly the Cetane Number will decrease and the sulphur content will probably increase. Since MTU's new method of turbocharging provides more excess air for the combustion process, the combustion of poorer fuels is possible. The engine output, of course, will then be dependent on exhaust temperature which, being a function of the fuel quality used, will result in most cases in a reduction of engine power.

2.5 Ease of Maintenance and Time-Between-Overhauls

The time needed for maintenance is an important factor, since labour costs are becoming increasingly expensive. Also we must accept that the quality of labour varies greatly throughout the world. In designing a modern diesel engine, both of these facts must be recognised. It therefore behoves the designer to use his skill to find the best compromise between solutions which, at first sight, seem incompatible. A scheduled maintenance system, tailormade to meet the requirements of the special application, usually affords the best economic solution. With such a system, an overhaul might be expected after 6,000 hours for a fast patrol boat engine, or alternatively 24,000 hours for a work-boat engine.

In addition to scheduled maintenance, regular checking of lube oil condition assists in determining the optimum time for oil changes and has also proved effective in preventing engine malfunctions. Close cooperation between engine manufacturer and the end-user greatly assists in keeping availability to a maximum and through-life costs to a minimum.

2.6 System Integration

Engine installation can be greatly simplified if major subassemblies and associated circuits are integrated with the engine itself. This design principle, which has long been a feature of MTU engines, is very important for future development. The lube oil heat exchanger (engine mounted), is totally integrated in the engine cooling water circuit, as is the charge air intercooler on internally cooled engine versions of the engine family 331/396. The installer therefore has very few lube oil and water connections to make. The installation of only one water circuit, that is, engine water also used for charge air cooling, allows the simplest cooling system possible.

2.7 Noise and Shock Requirements

In the naval field, the requirements for low noise levels, particularly regarding structure-borne noise, are very stringent. Since there is little chance of reducing noise in the engine design itself, some form of noise attenuation is necessary. The compact diesel engine, by virtue of its low weight and bulk, is most suitable for such measures. A highly sophisticated solution to this problem was developed for the German Frigate F 122. In this case, the diesel engine is double-elastically mounted on a base frame in which a number of concrete blocks are fitted, thereby acting as an intermediate mass. In order to avoid discrete frequencies being transmitted from the engine foundation into the elastic mounting system, resonators have been fitted. These are specific spring/mass systems attached directly to the engine.

In addition to the precautions against structure-borne noise, air-borne noise is reduced by encapsulating the whole engine in a noise-proof enclosure. This enclosure is equipped with a seawater cooled air conditioning system to keep inside temperatures low in order to allow certain maintenance work within the module. The results achieved by this noise control system fully met the requirements of the German Navy and represent considerable progress in this field. Noise attenuation can be even further improved by using a special composition of concrete within the base frame.

Besides noise attenuation, high shock resistance is often a prerequisite for diesel engines in naval applications. Shock capabilities are normally proven by calculation or by actual shock testing. Recently, the 6 V 396 engine was approved by the US-Navy according to MIL 901 standard after successful completion of the respective tests within US-Navy facilities.

2.8 Other Trends

Apart from other avenues of diesel engine development, low load operation is very important for many applications. MTU's progress in this direction during recent years, for instance, has been demonstrated by a test run for the Federal German and the Dutch Navy in which a 20 V 956 engine ran continuously for 72 hours on no load with one cylinder bank cut-out.

It has been proven that in engines with cylinder cut-out, both the firing and non-firing cylinders wear less and have less carbon residuals than those of engines without cylinder cut-out,

assuming otherwise identical operation. To date, MTU has delivered more than 300 engines with this equipment.

The low lube-oil consumption, which is a prerequisite for favourable low-load operation, has been achieved through the intensive development of pistons, piston rings and cylinder liners.

One major advantage of compact diesel engines is the capability of rapid acceleration. With the introduction of sequential turbocharging, this advantage is maintained, despite the switching procedure, because of the low inertia of the turbocharger rotating parts. Fig. 21 shows the acceleration of a 16 V 538 TB 93 engine and in particular, the very short time required to speed up the turbocharger after switching in.

3 IMPLICATION OF IMPROVED PERFORMANCE DIESEL ENGINES FOR THE PROPULSION OF NAVAL SHIPS

3.1 Range of Operation

Fig. 22 shows the comparison between a normally turbocharged 16 V 538 TB 92 engine and a sequentially turbocharged TB 93 (single-stage engine). Apart from the increase of engine output from 3,000 to 3,300 kW due to better turbocharger effectiveness, the most important improvement is the broader range of operation, especially under part load conditions.

For better comparison, Fig. 23 shows the MCR curves of both engines in percentage terms. As can be seen, the TB 92 engine allows only a small margin between the actual power requirement of the fast patrol boat and the power the engine is able to deliver continuously. For instance at 1,000 rpm, the power demand is 23 % whilst the engine is capable of delivering 30 %. The new version of the engine with sequential turbocharging is able to deliver 40 % of its nominal power at the same speed. This means a considerable power reserve either to overcome increased displacement or higher resistance due to hull fouling. Besides the increased continuous power, the output limited by the governor is also much higher (Fig. 24). This allows a greater margin for acceleration or for short-time operation of a 4-shaft vessel with 1, 2 or 3 shafts. For 1-shaft operation this means that now approximately 60 % of engine power is available instead of around only 40 % with the former TB 92 version.

Increased engine torque at low engine speed is not only advantageous for Fast Patrol Boat application. Until now, for all fast vessels such as Surface-Effect-Ships or hydrofoils with a transition phase, the so called 'hump condition' was decisive in selecting engine output unless additional devices like torque converters were used. Now, with a broader engine torque range it is possible to equip such vessels with engines of lower maximum output as shown in Fig. 25. For bigger vessels with 4 diesel engines driving 2 propellers, two engines can be taken out of service under part load conditions without overloading the remaining engines. As Fig. 26 shows, for a power requirement according to the cube law curve, the vessel can be operated up to 64 % of max. engine speed which is approximately the same percentage of ship speed, whereas the previous engine designs

allowed only 39 %. This means 50 % instead of 12 % output per diesel engine. With MTU's latest engine development, a vessel of this kind can now be equipped with fixed pitch propellers, or alternatively, controllable pitch propellers can be used with their design pitch giving best efficiency.

In addition to fuel saving due to better efficiency of the propeller, the engine operates in a better sfc-range as shown in Fig. 27.

This broadening of the power range means that the diesel engine is now, from a practical point of view, just as flexible to operate as a gas turbine.

3.2 Output per Engine

The higher output per engine without an increase in dimensions, results in a more favourable weight to power ratio. This enables more propulsive power to be installed within a given machinery room or alternatively a smaller engine room (or more additional space), for the same engine power.

Fig. 28 compares two propulsion plants in a 400 t fast patrol boat. In one case, a 4-shafter is equipped with 4x 16 V 956 TB 91 engines with a maximum total power of 13,200 kW (as installed power in the German FPB S-143 craft). The other shows the same boat with only 3 shafts, powered by 3x 12 V 1163 TB 93 engines, developing a total maximum power of 13,320 kW. Besides the small space advantage, there is a weight saving of approx. 6 tons per ship, assuming that shaft and propeller weights remain the same.

If there is no requirement for separate engine rooms, two 20 V 1163 TB 93 engines with a total maximum power of 14,800 kW would be even more favourable. Of course, we accept that this solution is only viable without restrictions in propeller diameter, that is, for deep water navies only.

For corvette and frigate propulsion, the 7,400 kW diesel engine offers new prospects. Whereas until now these vessels were mainly equipped with combined propulsion plant, the 7,400 kW compact diesel permits an all-diesel installation (Fig. 29).

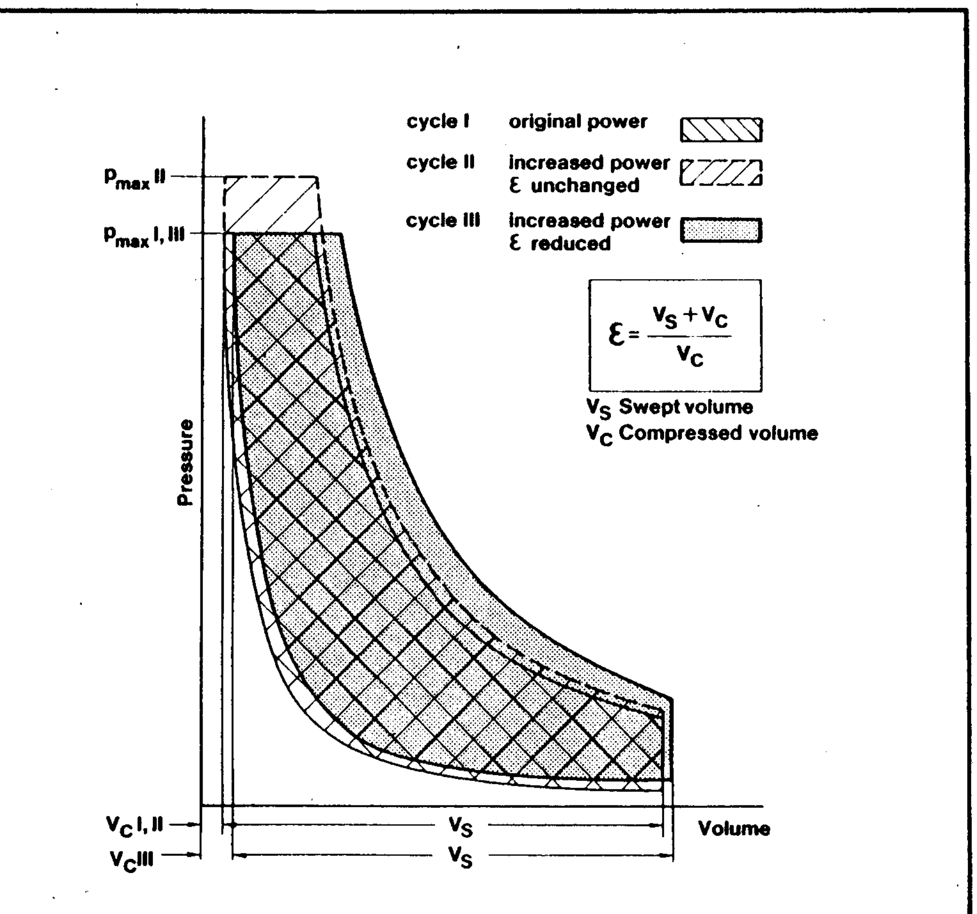
For the end-user, there are many logistic advantages in powering the ship with identical propulsion engines. For instance spare holdings, maintenance planning and staff training are simplified. In this case, the engines are operated on the highest rating (1 DS), as used for fast patrol craft propulsion. This is allowed due to the fact that this high power output is needed for a relatively short time only. Also, the engines can be alternately selected for the cruise mode so that, although the time between overhaul for each 1 DS rated engine is 6,000 hours, this time will not be accumulated on any one engine until a lapse of over say 10,000 hours of ship operation. This overhaul period is normally sufficient for most warships.

An additional advantage is that the ship's endurance is extended, especially at higher ship speeds because of the better part load SFC of the diesel compared to the gas turbine. Where stringent noise attenuation is required, the engines can be double-elastically mounted and enclosed by a sound-proof capsule.

All the advantages just described for propulsion engines are equally valid for on-board power generation as well. Fig. 30 shows the comparison of three generator modules in the 2,000 kW output range. The most interesting feature is the fact that even though there is a slight weight advantage in favour of the gas turbine module, this is more than compensated by the saving in fuel consumption of the diesel-powered genset.

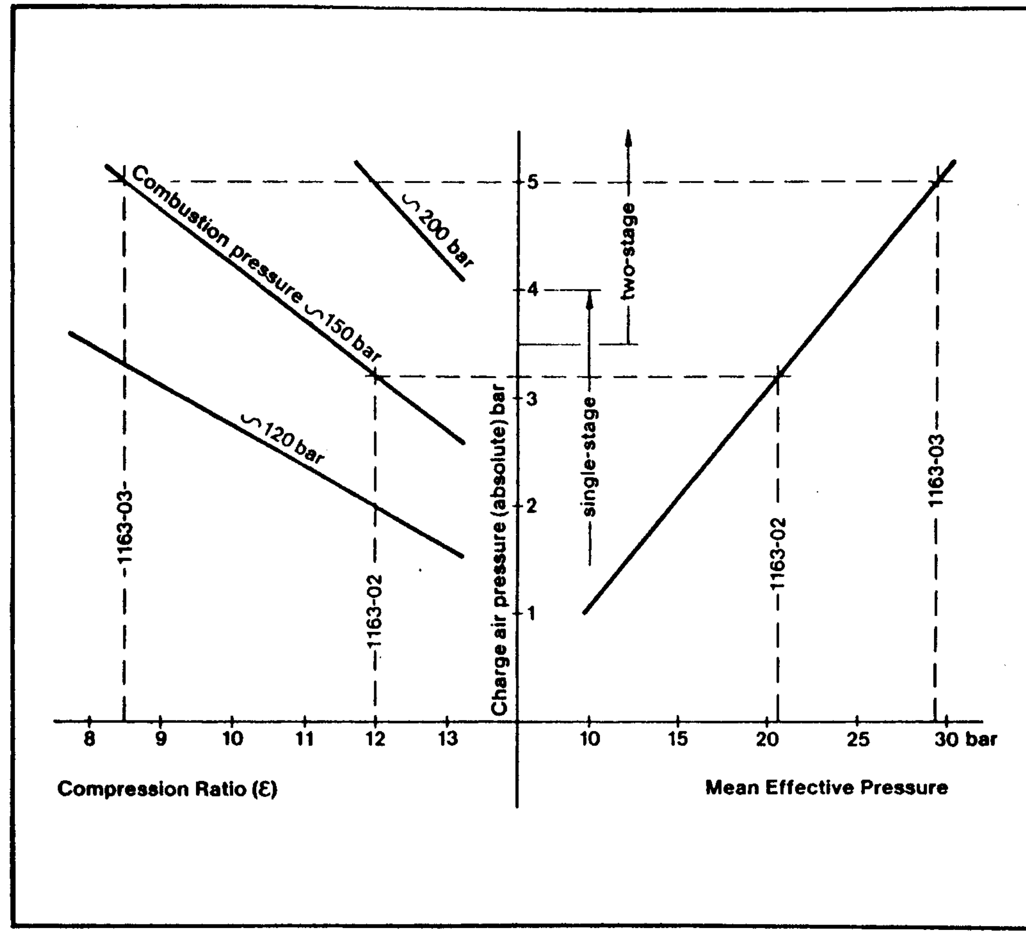
4 CONCLUSIONS

MTU is convinced that these developments have widened the potential for compact diesel engine propulsion by overcoming some of the restrictions which until now have limited their use. New propulsion arrangements are now feasible which are simpler and more cost effective for shipbuilders and navies alike.



cycle I original power
cycle II increased power ε unchanged
cycle III increased power ε reduced
$\varepsilon = \frac{V_S + V_C}{V_C}$
V_S Swept volume
V_C Compressed volume
p_{max} II
p_{max} I, III
Pressure
Volume
V_C I, II
V_C III
V_S
V_S

FIG. 1



Combustion pressure ∽150 bar
∽200 bar
∽120 bar
Charge air pressure (absolute) bar
single-stage
two-stage
1163-03
1163-02
Compression Ratio (ε)
Mean Effective Pressure
8
9
10
11
12
13
1
2
3
4
5
10
15
20
25
30 bar

FIG. 2

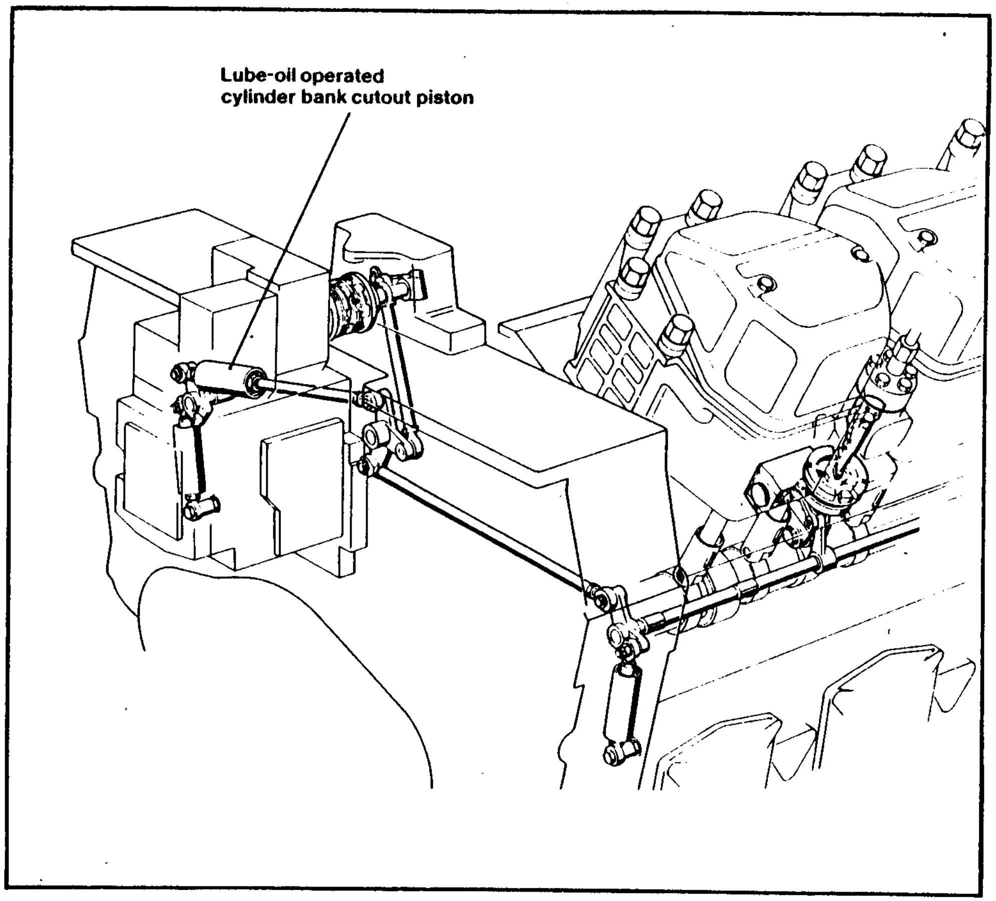

Lube-oil operated
cylinder bank cutout piston

FIG. 3

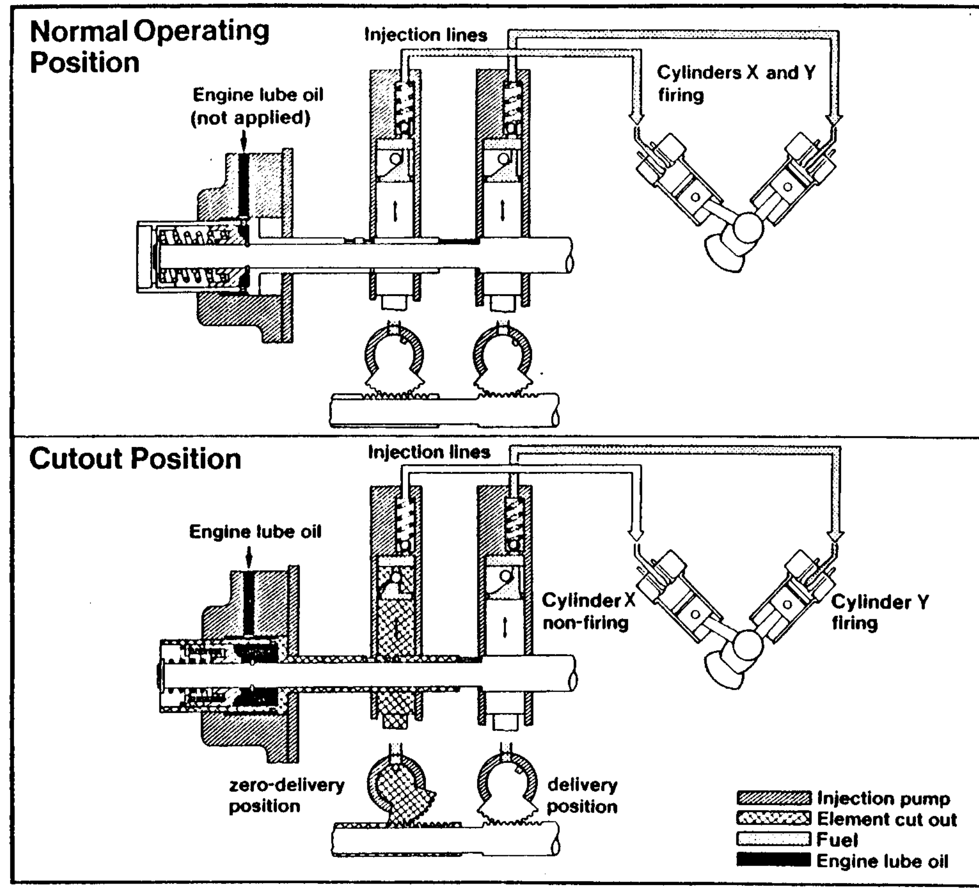

Normal Operating Position
Injection lines
Engine lube oil (not applied)
Cylinders X and Y firing
Cutout Position
Injection lines
Engine lube oil
Cylinder X non-firing
Cylinder Y firing
zero-delivery position
delivery position
Injection pump
Element cut out
Fuel
Engine lube oil

FIG. 4

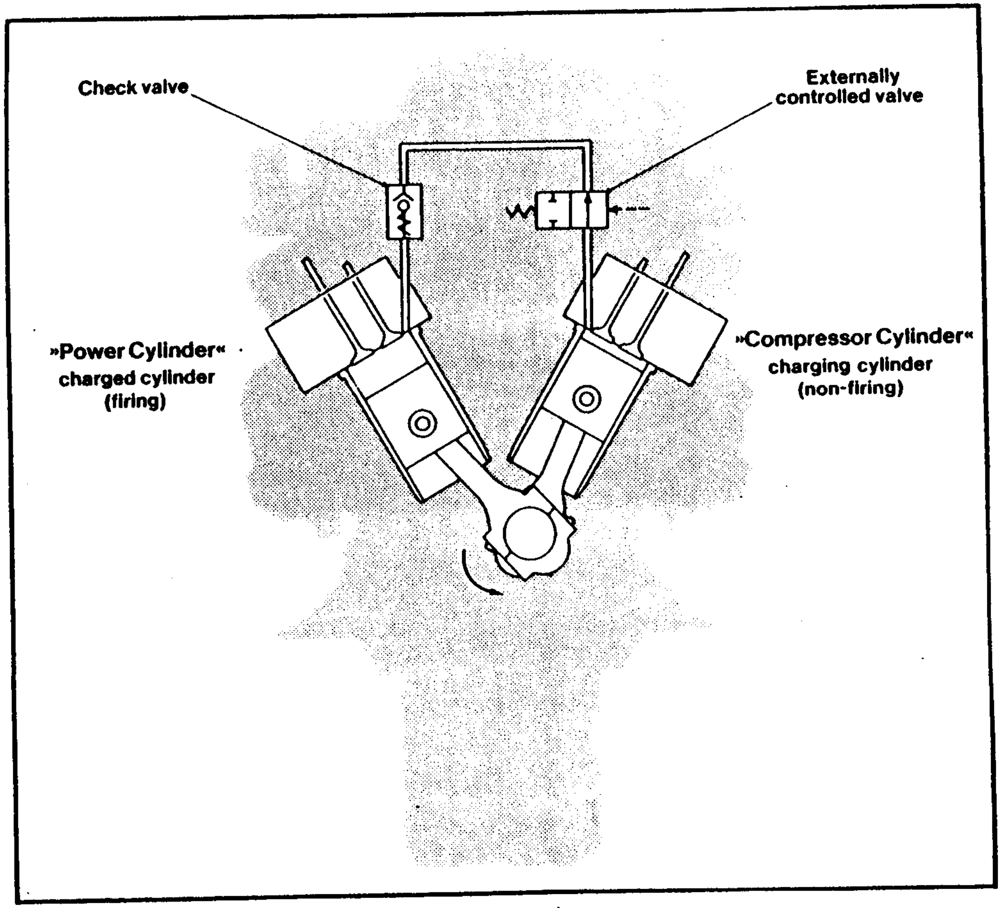

Check valve
Externally controlled valve
»Power Cylinder« charged cylinder (firing)
»Compressor Cylinder« charging cylinder (non-firing)

FIG. 5

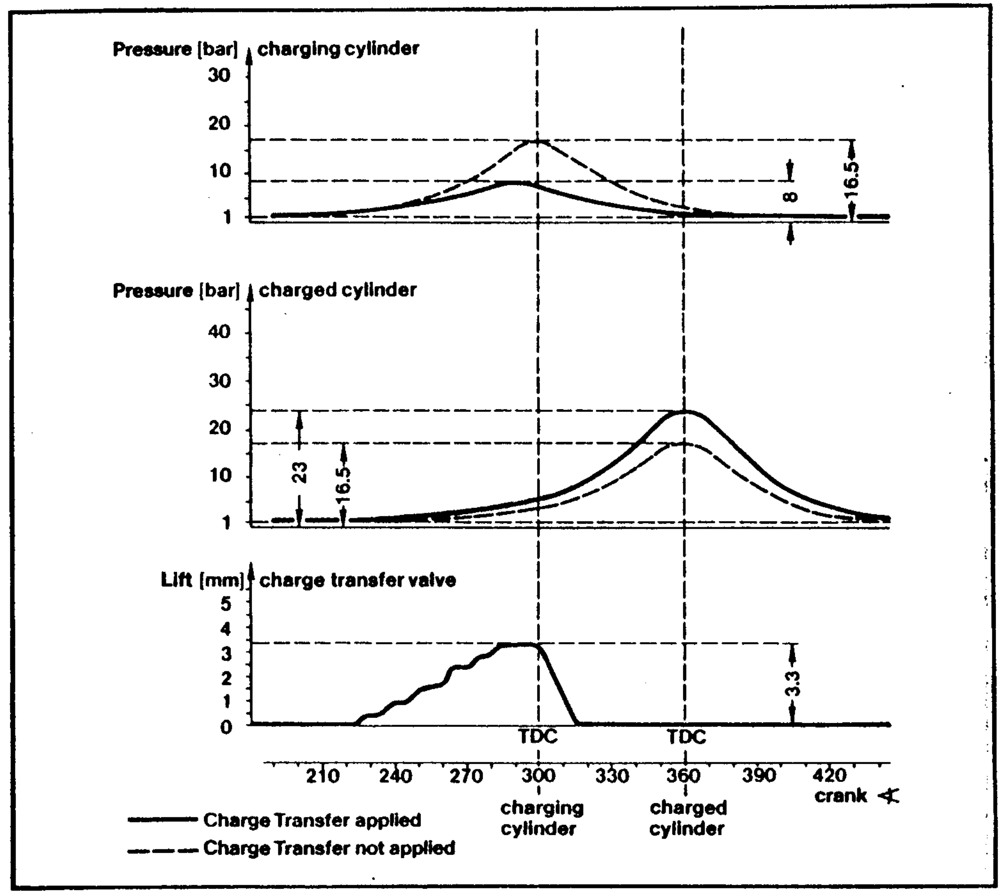

Pressure [bar] charging cylinder
30
20
10
1
8
16.5
Pressure [bar] charged cylinder
40
30
20
10
1
23
16.5
Lift [mm] charge transfer valve
5
4
3
2
1
0
3.3
TDC
TDC
210
240
270
300
330
360
390
420
crank ∢
charging cylinder
charged cylinder
Charge Transfer applied
Charge Transfer not applied

FIG. 6

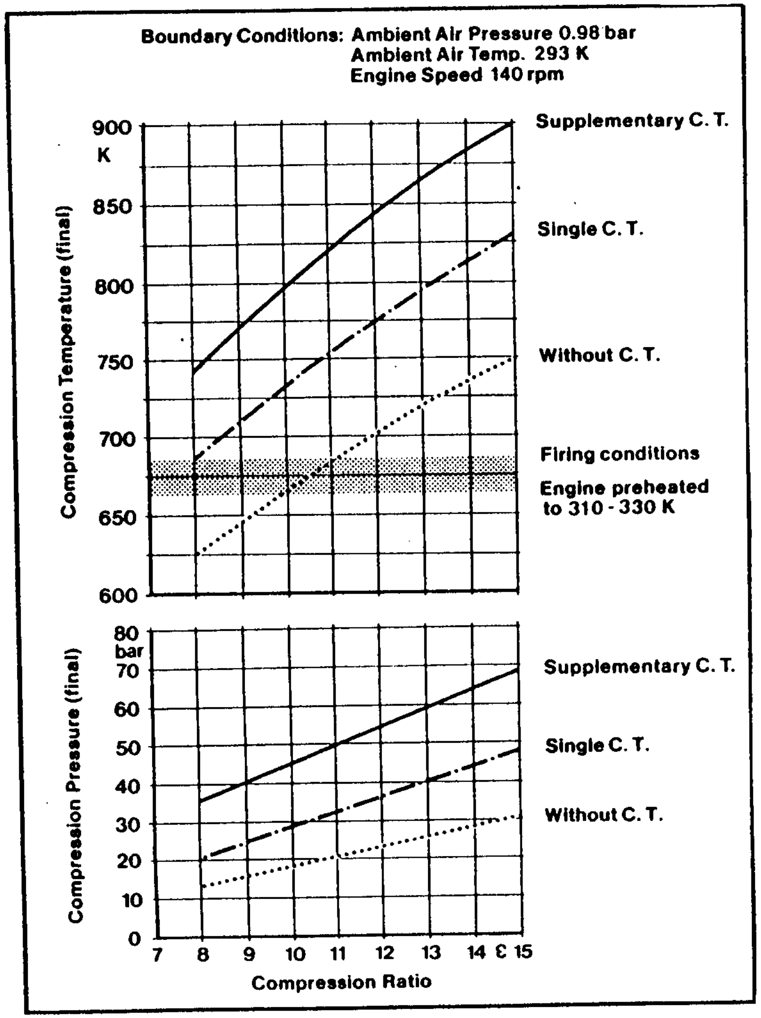

Boundary Conditions: Ambient Air Pressure 0.98 bar
Ambient Air Temp. 293 K
Engine Speed 140 rpm
Compression Temperature (final)
900 K
850
800
750
700
650
600
Supplementary C. T.
Single C. T.
Without C. T.
Firing conditions
Engine preheated to 310 - 330 K
Compression Pressure (final)
80 bar
70
60
50
40
30
20
10
0
7
8
9
10
11
12
13
14 ε 15
Compression Ratio

FIG. 7

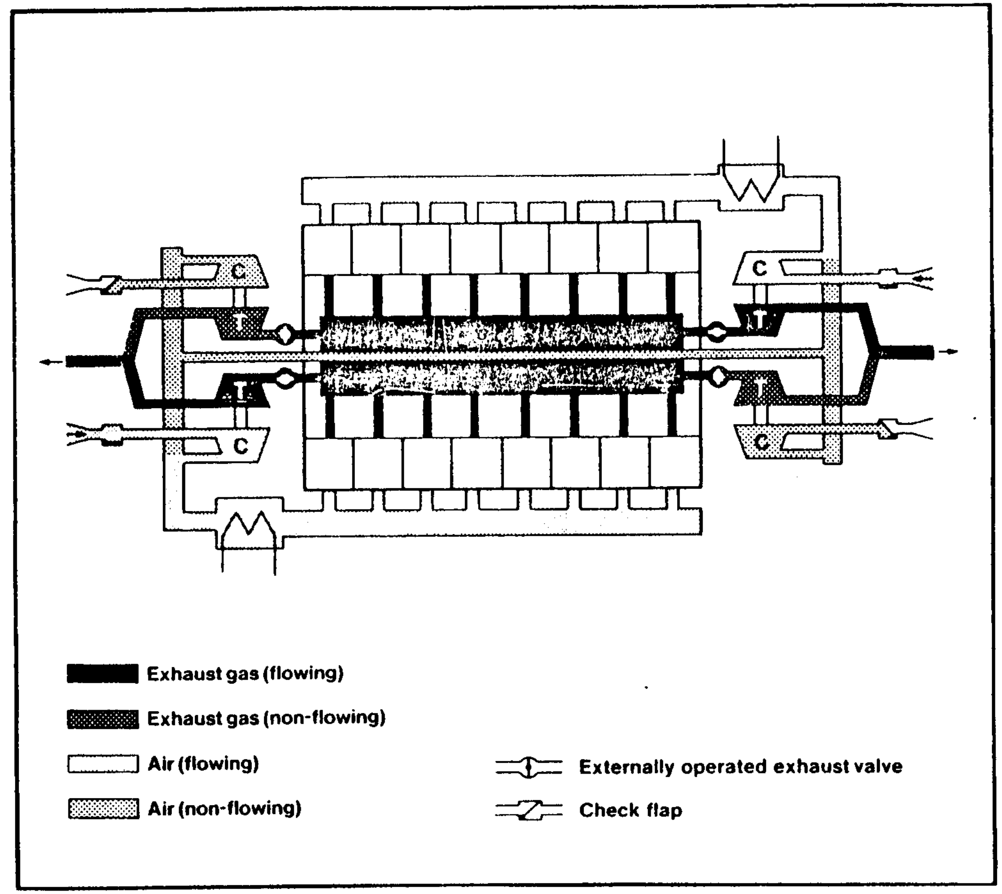

C
T
Exhaust gas (flowing)
Exhaust gas (non-flowing)
Air (flowing)
Air (non-flowing)
Externally operated exhaust valve
Check flap

FIG. 8

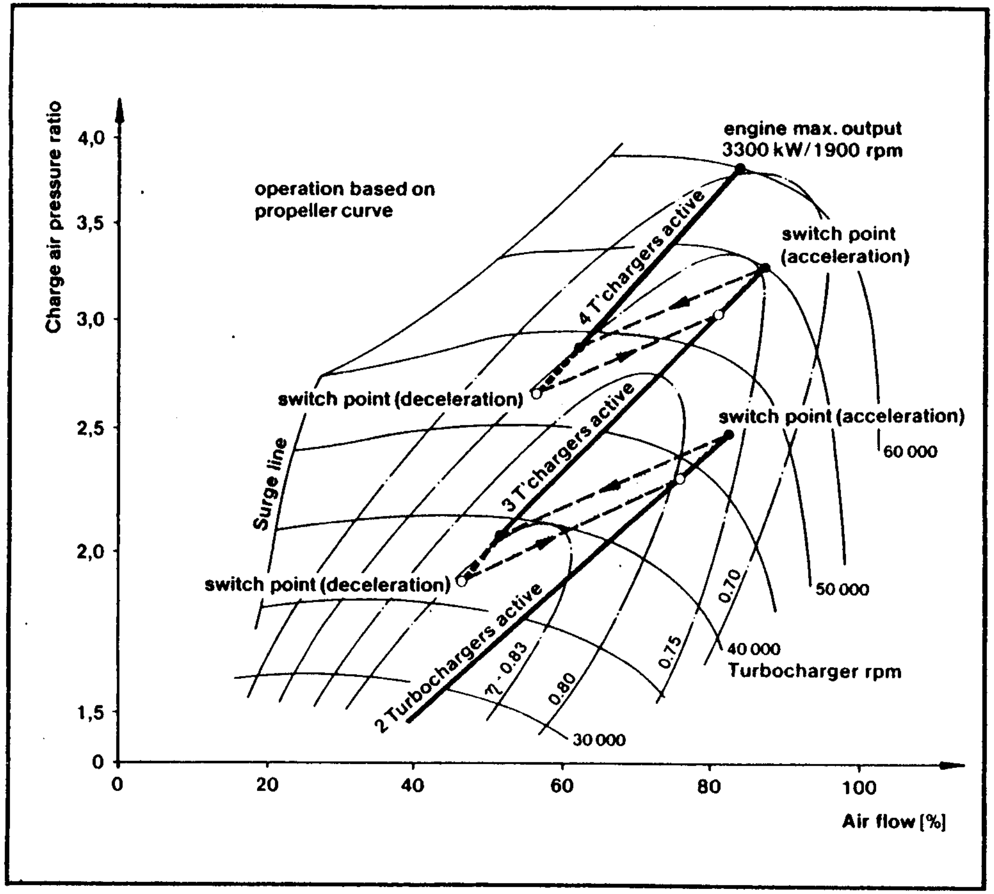

Charge air pressure ratio
4,0
3,5
3,0
2,5
2,0
1,5
0
0
20
40
60
80
100
Air flow [%]
operation based on
propeller curve
engine max. output
3300 kW/1900 rpm
switch point
(acceleration)
switch point (acceleration)
switch point (deceleration)
switch point (deceleration)
Surge line
4 T'chargers active
3 T'chargers active
2 Turbochargers active
η · 0.83
0.80
0.75
0.70
30 000
40 000
50 000
60 000
Turbocharger rpm

FIG. 9

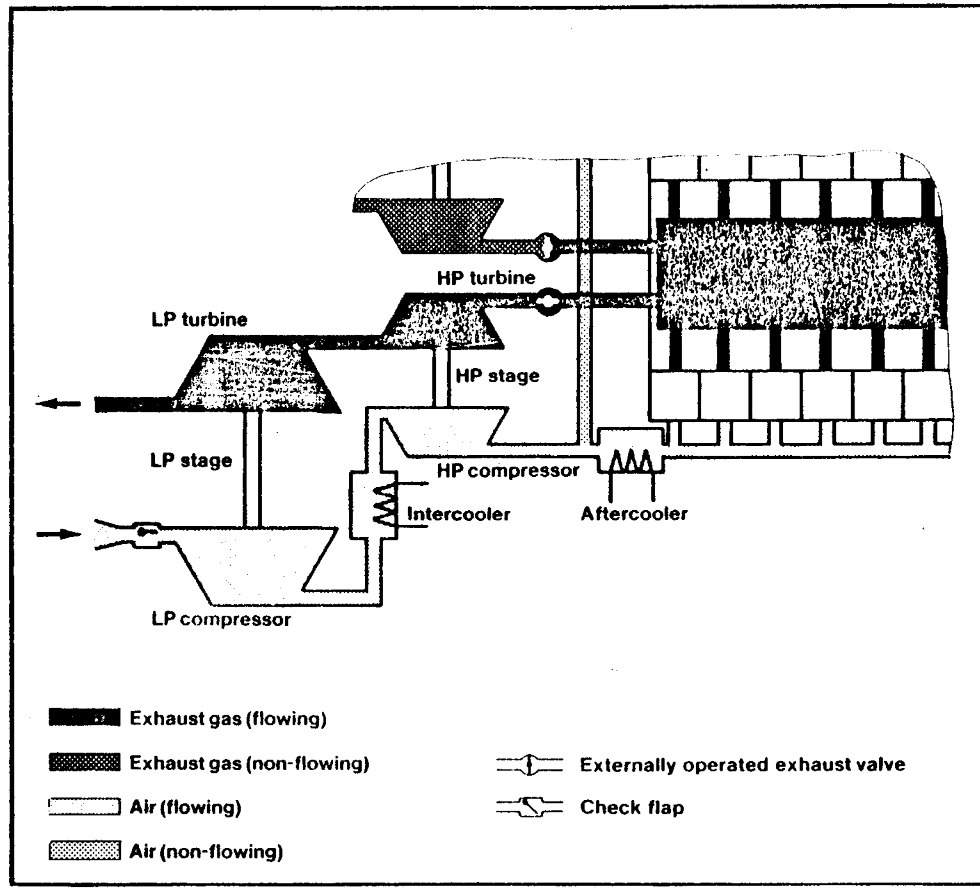

HP turbine
LP turbine
HP stage
LP stage
HP compressor
Intercooler
Aftercooler
LP compressor
Exhaust gas (flowing)
Exhaust gas (non-flowing)
Air (flowing)
Air (non-flowing)
Externally operated exhaust valve
Check flap

FIG. 10

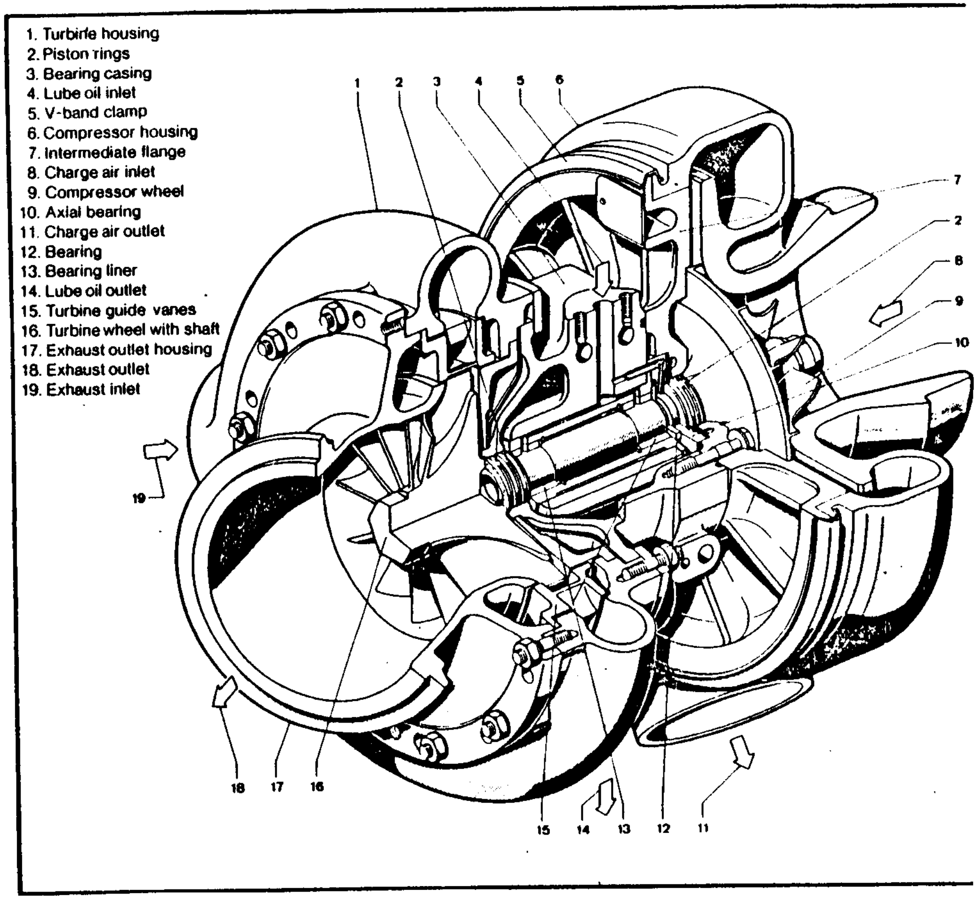

1. Turbine housing
2. Piston rings
3. Bearing casing
4. Lube oil inlet
5. V-band clamp
6. Compressor housing
7. Intermediate flange
8. Charge air inlet
9. Compressor wheel
10. Axial bearing
11. Charge air outlet
12. Bearing
13. Bearing liner
14. Lube oil outlet
15. Turbine guide vanes
16. Turbine wheel with shaft
17. Exhaust outlet housing
18. Exhaust outlet
19. Exhaust inlet

FIG. 11

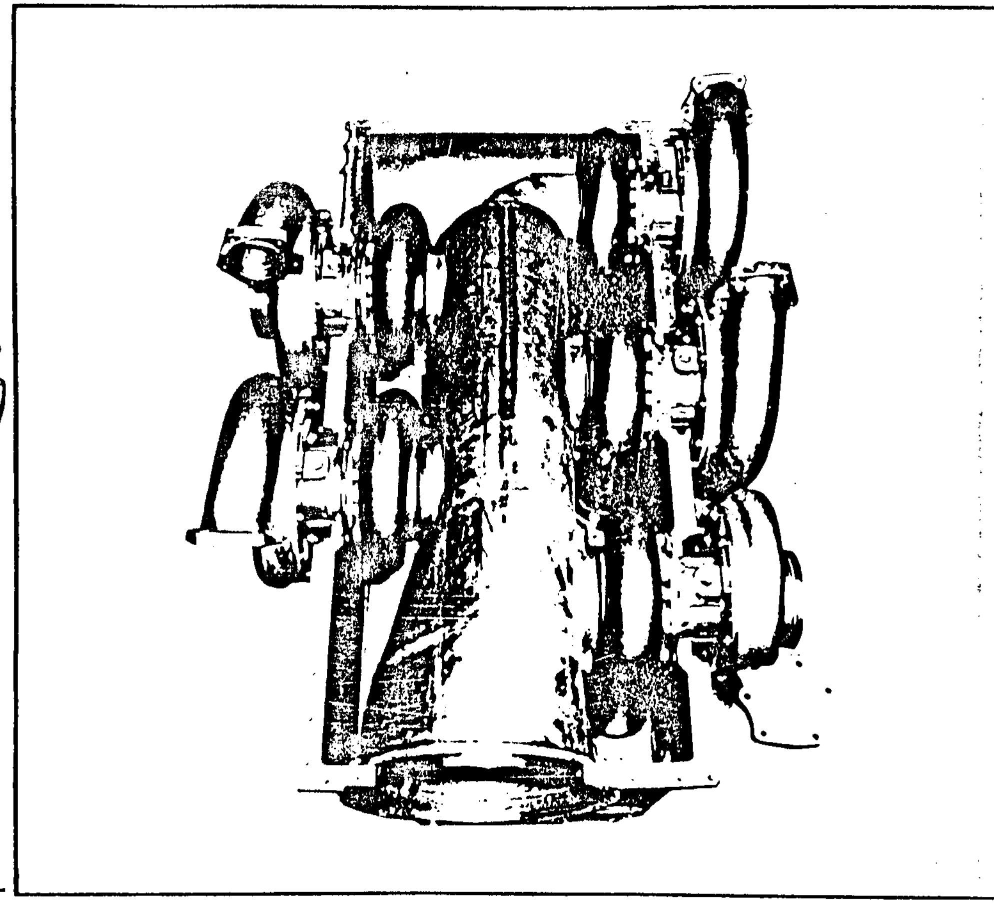


FIG. 12

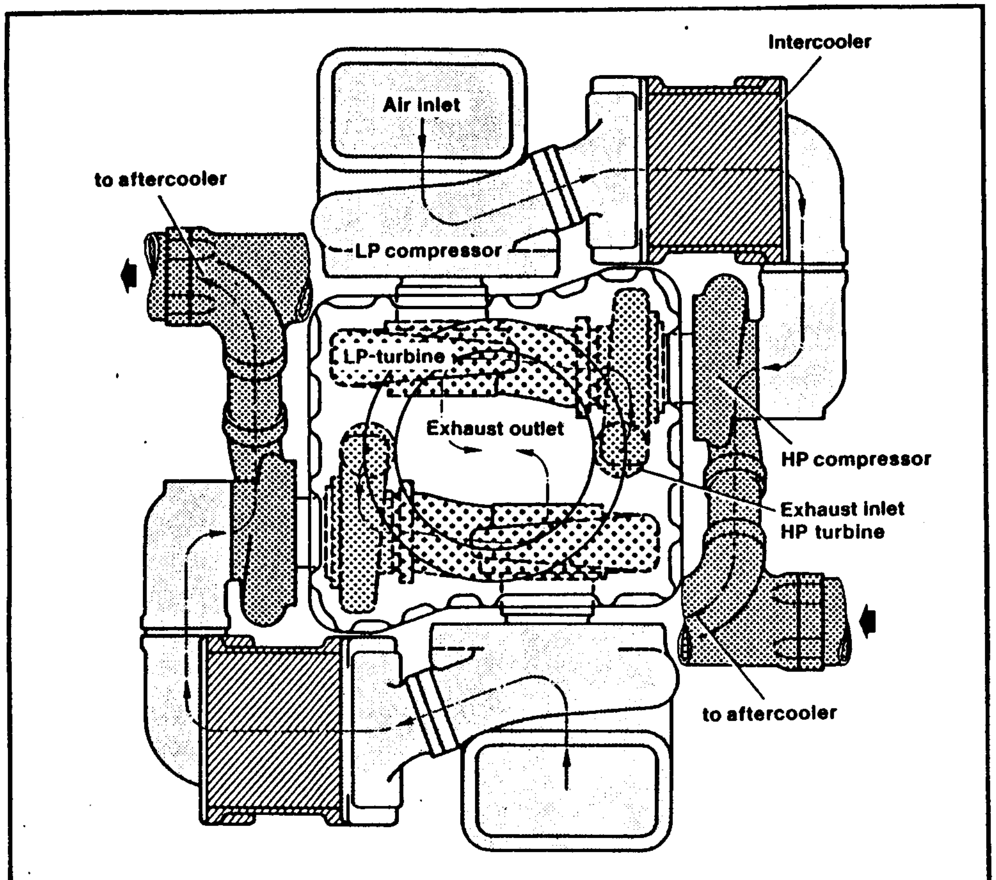

Intercooler
Air inlet
to aftercooler
LP compressor
LP-turbine
Exhaust outlet
HP compressor
Exhaust inlet
HP turbine
to aftercooler

FIG. 13

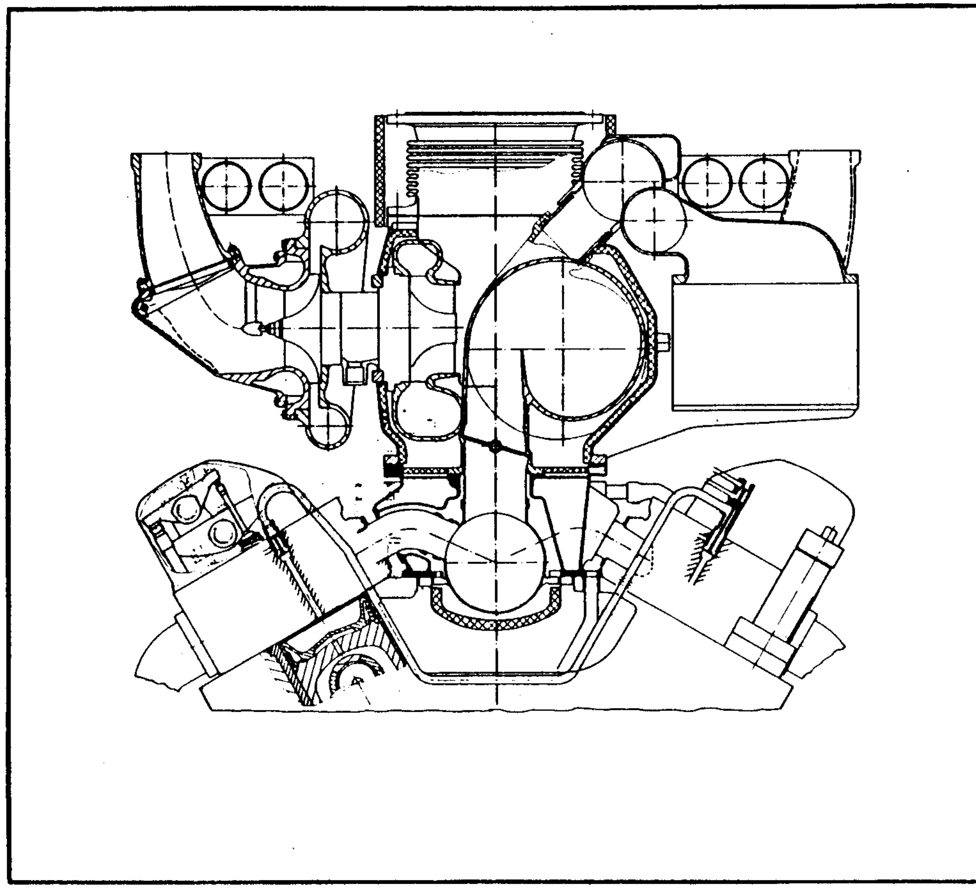


FIG. 14

	1163-02			1163-03		
	Power kW HP	Weight kg	Spec. Weight kg/kW kg/HP	Power kW HP	Weight kg	Spec. Weight kg/kW kg/HP
12 V	3120 4240	11 350	3.6 2.7	4440 6000	13 000	2.9 2.2
16 V	4160 5660	14 280	3.4 2.5	5920 8000	15 800	2.7 2.0
20 V	5200 7080	16 960	3.3 2.4	7400 10 000	19 400	2.6 1.9

FIG. 15

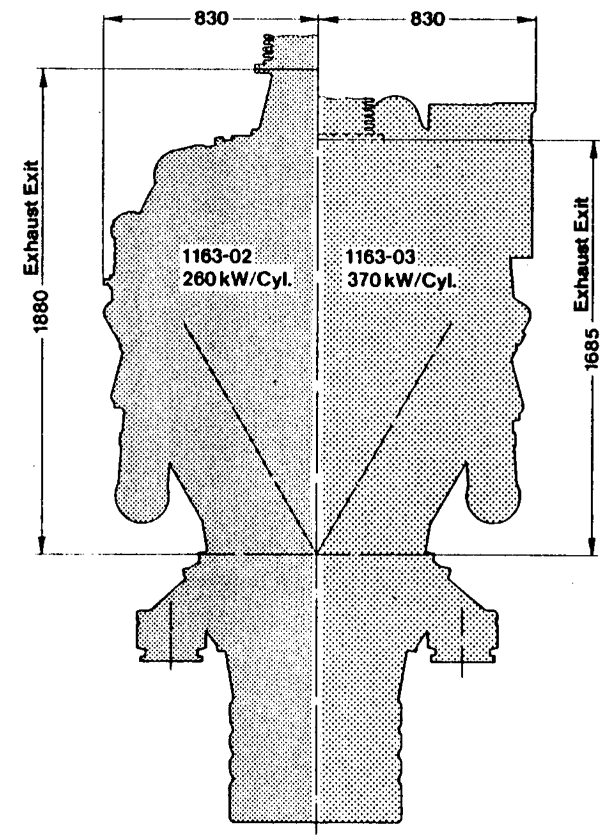


FIG. 16

		956-02	1163-03
P	[kW/cyl.]	245	370
bmep	[bar]	20.5	29.2
n	[min^{-1}]	1500	1300
ε	[-]	12	8.8
λ	[-]	1.72	2.03
t	[°C]	305	210
		280	215
		245	180
		155	160

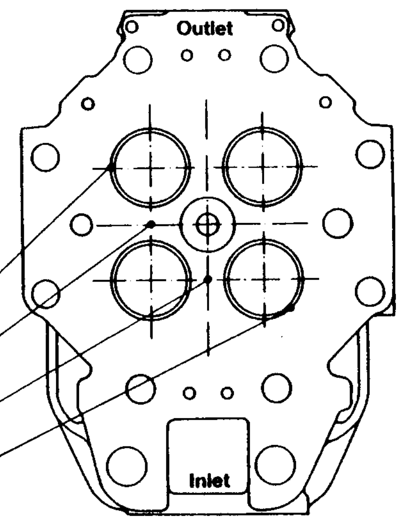


FIG. 17

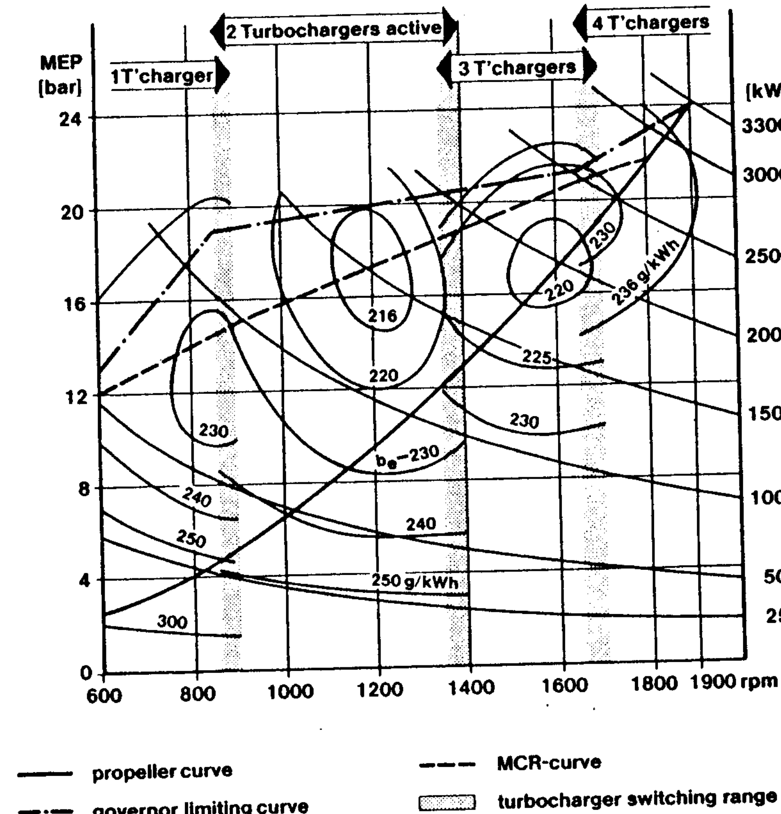


FIG. 18

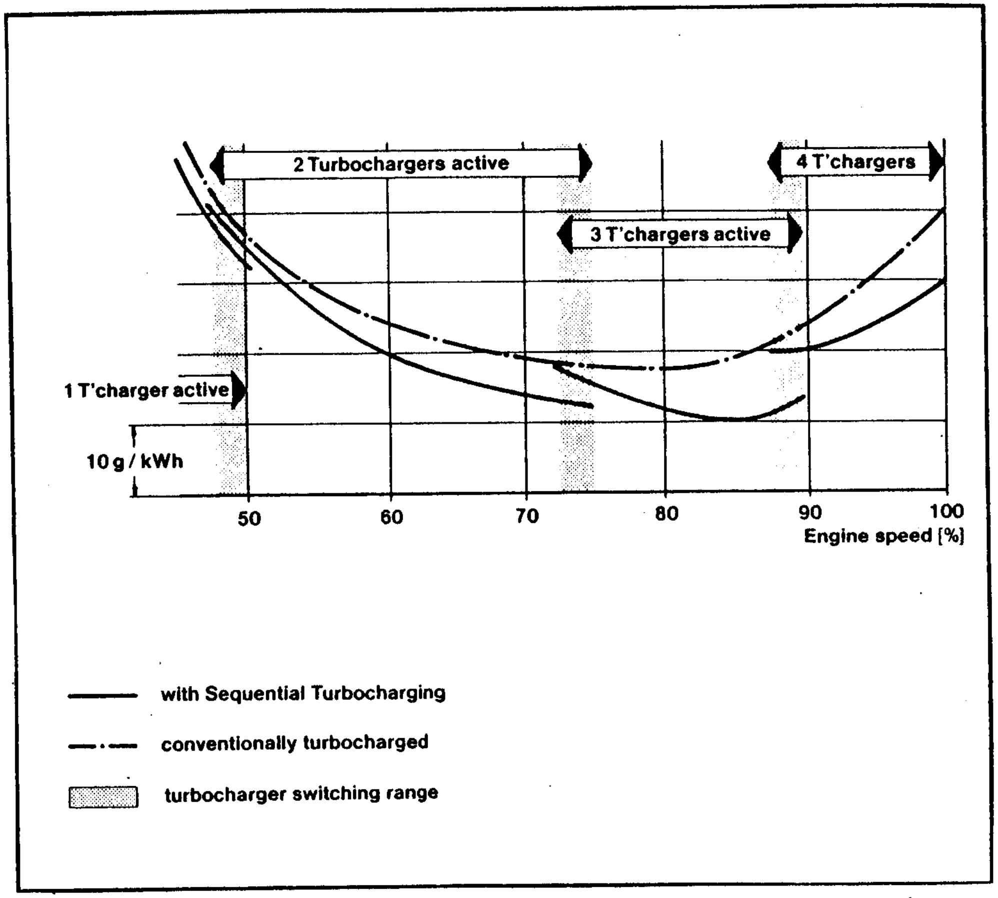

2 Turbochargers active
4 T'chargers
3 T'chargers active
1 T'charger active
10 g / kWh
50
60
70
80
90
100
Engine speed [%]
with Sequential Turbocharging
conventionally turbocharged
turbocharger switching range

FIG. 19

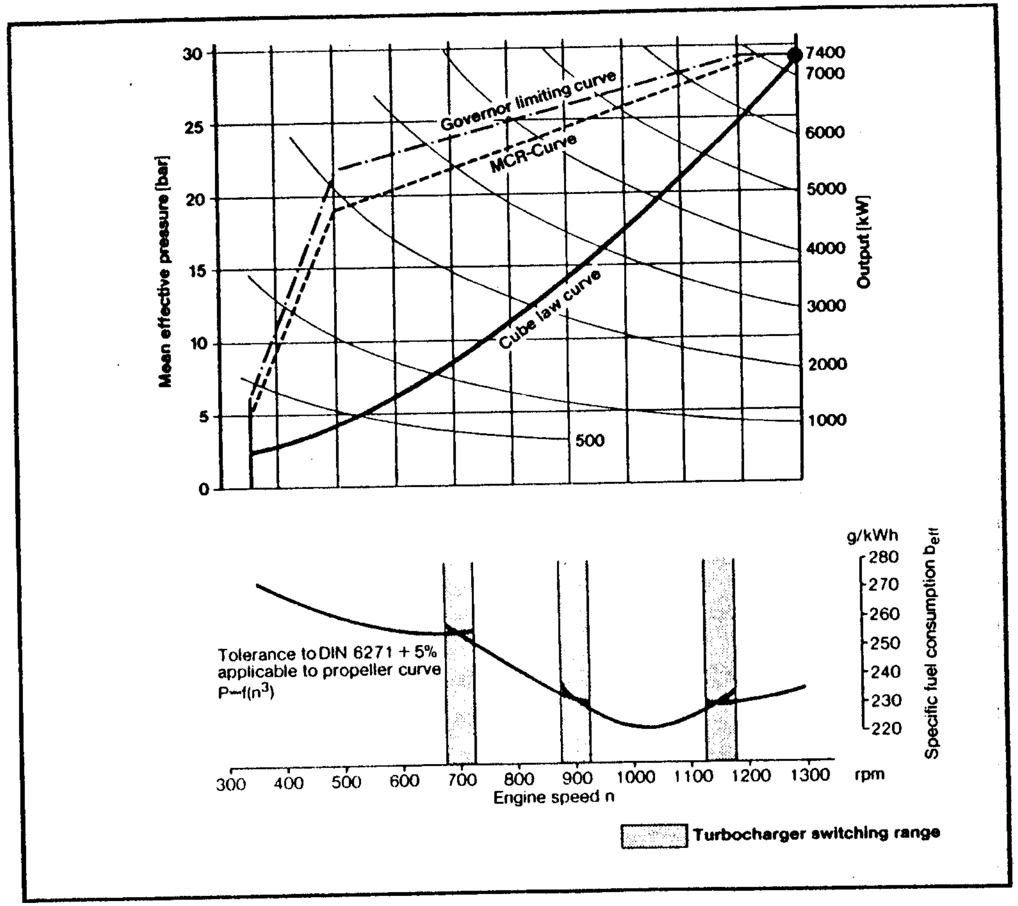

Mean effective pressure [bar]
Output [kW]
Governor limiting curve
MCR-Curve
Cube law curve
500
7400
7000
6000
5000
4000
3000
2000
1000
g/kWh
Specific fuel consumption b_{eff}
Tolerance to DIN 6271 + 5%
applicable to propeller curve
P~f(n³)
300 400 500 600 700 800 900 1000 1100 1200 1300 rpm
Engine speed n
Turbocharger switching range

FIG. 20

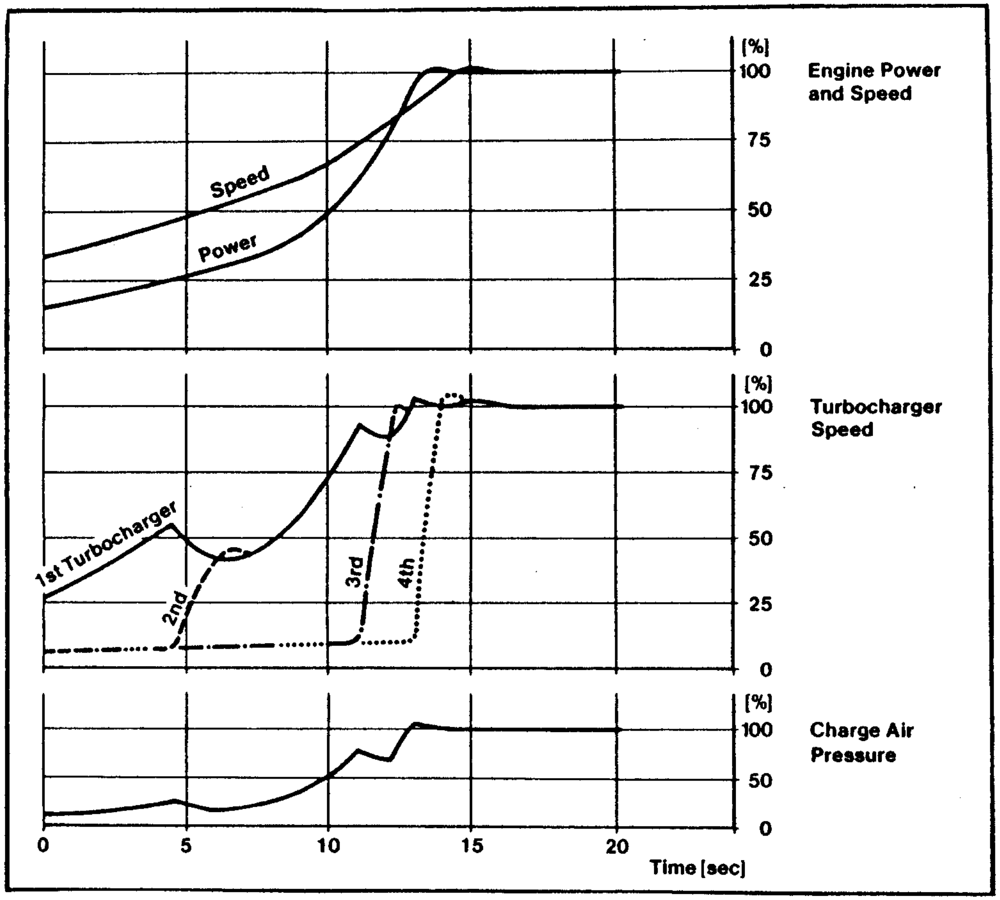

[%]
100
75
50
25
0
Engine Power and Speed
Speed
Power
Turbocharger Speed
1st Turbocharger
2nd
3rd
4th
Charge Air Pressure
0
5
10
15
20
Time [sec]

FIG. 21

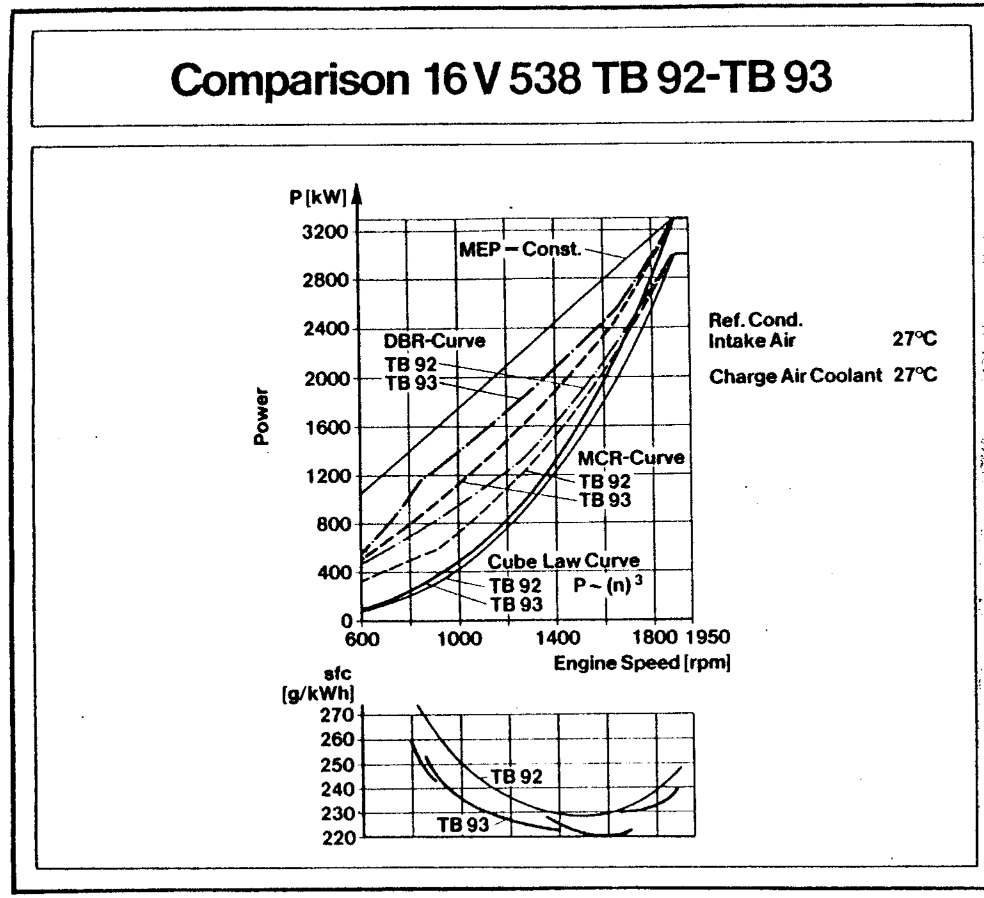

Comparison 16 V 538 TB 92-TB 93
P [kW]
Power
3200
2800
2400
2000
1600
1200
800
400
0
MEP – Const.
DBR-Curve
TB 92
TB 93
MCR-Curve
TB 92
TB 93
Cube Law Curve
TB 92
TB 93
P ~ (n)³
600
1000
1400
1800 1950
Engine Speed [rpm]
Ref. Cond.
Intake Air 27°C
Charge Air Coolant 27°C
sfc [g/kWh]
270
260
250
240
230
220
TB 92
TB 93

FIG. 22

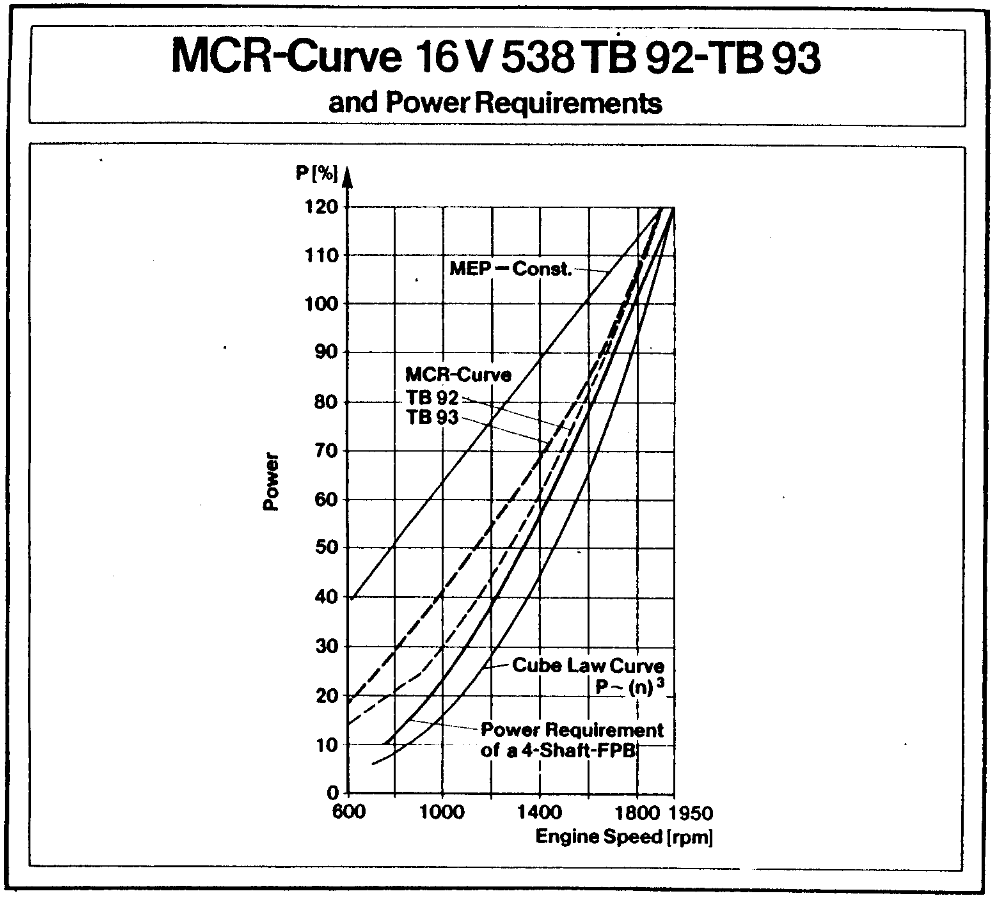

MCR-Curve 16 V 538 TB 92-TB 93
and Power Requirements
P [%]
Power
MEP – Const.
MCR-Curve
TB 92
TB 93
Cube Law Curve
P ~ (n)³
Power Requirement
of a 4-Shaft-FPB
Engine Speed [rpm]
120
110
100
90
80
70
60
50
40
30
20
10
0
600
1000
1400
1800
1950

FIG. 23

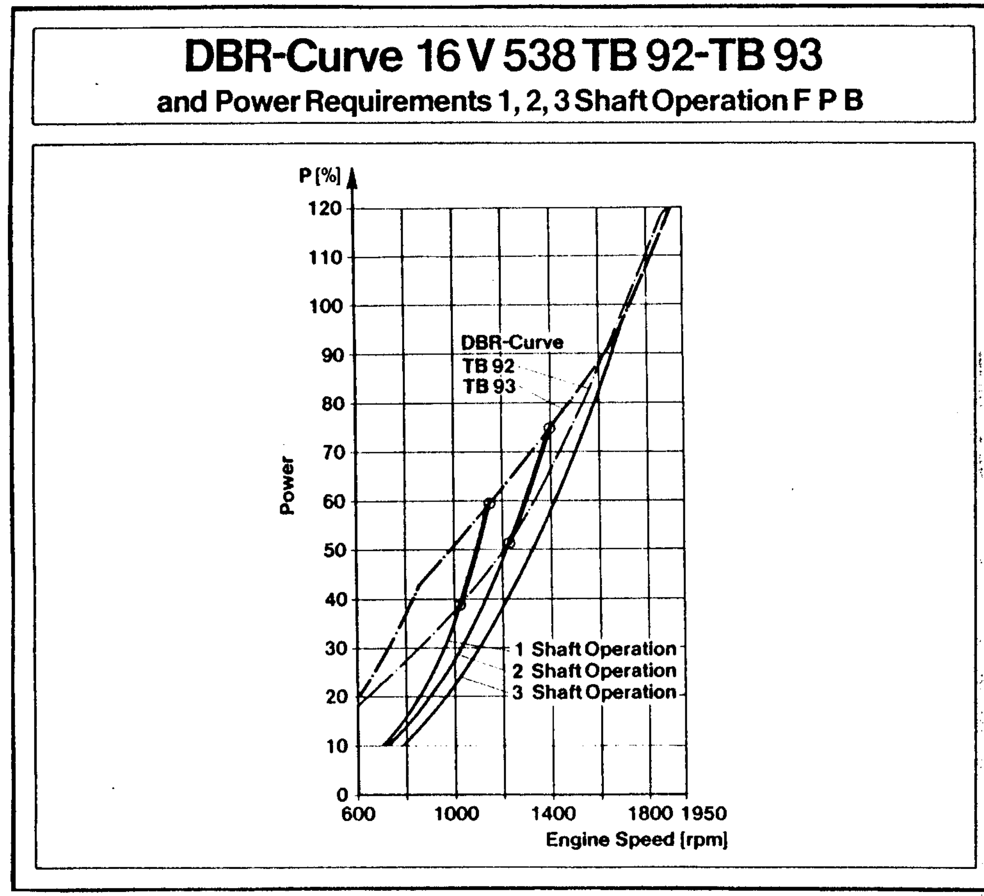

DBR-Curve 16 V 538 TB 92-TB 93
and Power Requirements 1, 2, 3 Shaft Operation F P B
P [%]
Power
DBR-Curve
TB 92
TB 93
1 Shaft Operation
2 Shaft Operation
3 Shaft Operation
Engine Speed [rpm]
120
110
100
90
80
70
60
50
40
30
20
10
0
600
1000
1400
1800
1950

FIG. 24

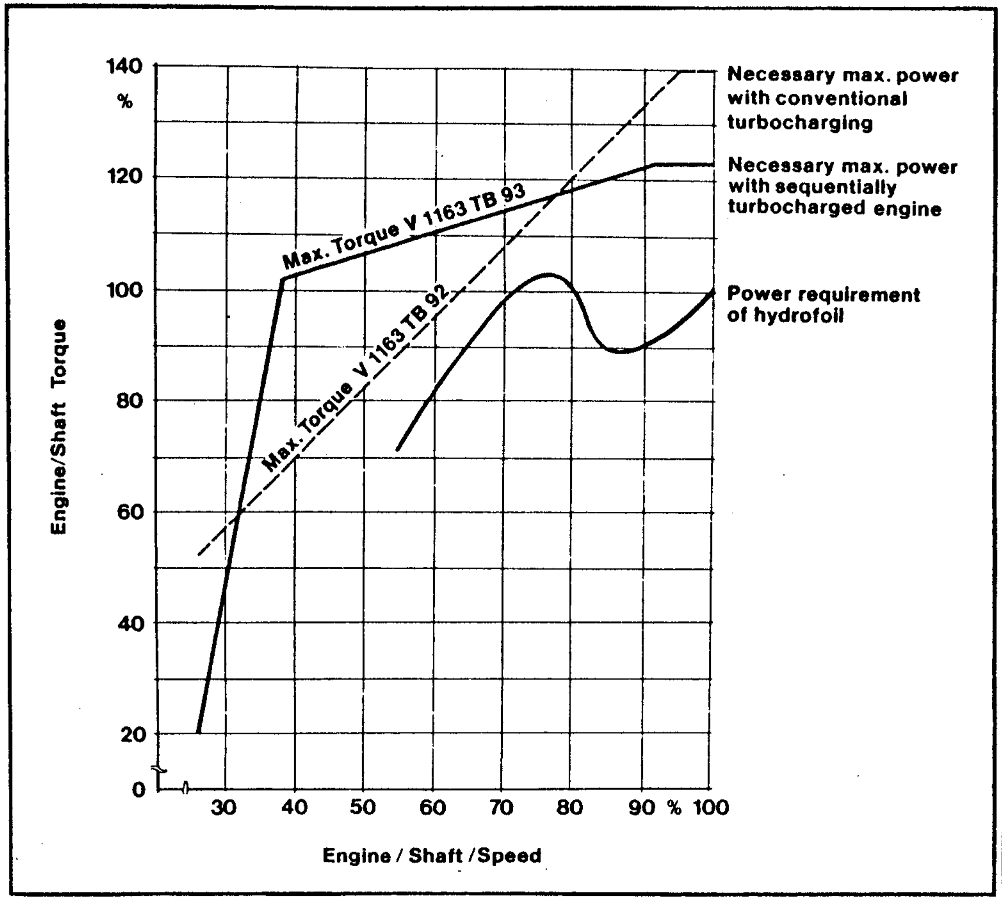

Engine/Shaft Torque
140
%
120
100
80
60
40
20
0
30 40 50 60 70 80 90 % 100
Engine / Shaft /Speed
Max. Torque V 1163 TB 93
Max. Torque V 1163 TB 92
Necessary max. power with conventional turbocharging
Necessary max. power with sequentially turbocharged engine
Power requirement of hydrofoil

FIG. 25

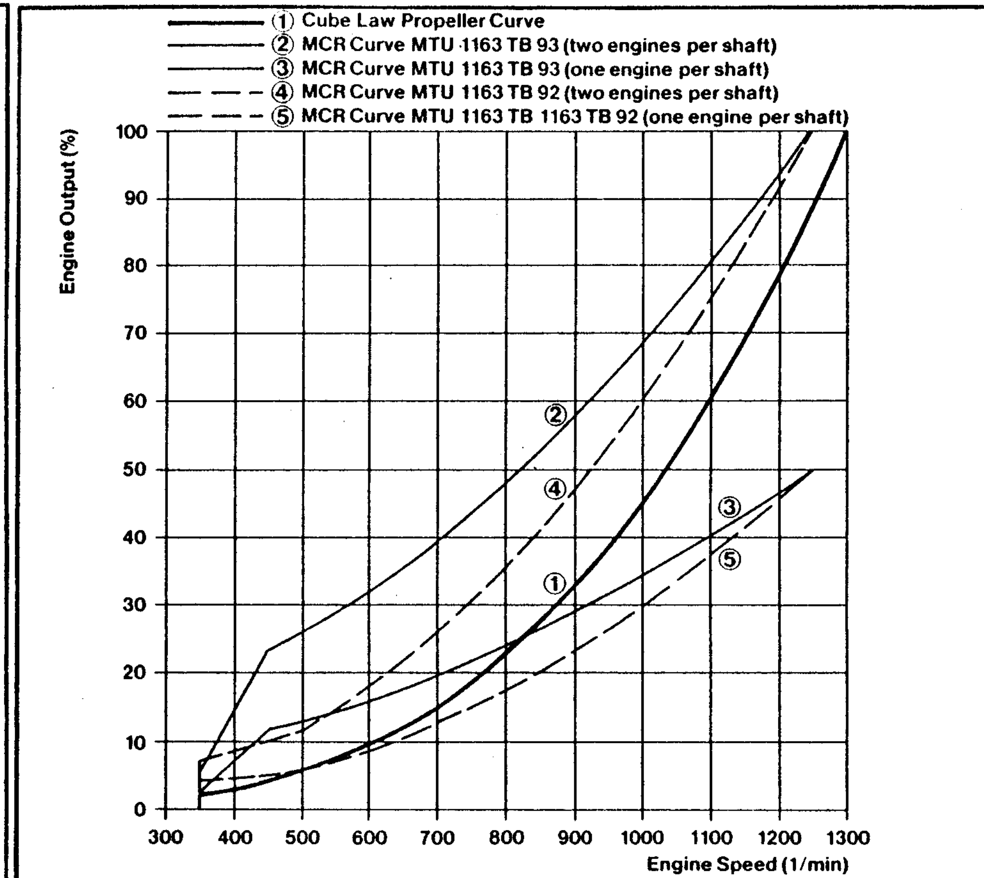

① Cube Law Propeller Curve
② MCR Curve MTU 1163 TB 93 (two engines per shaft)
③ MCR Curve MTU 1163 TB 93 (one engine per shaft)
④ MCR Curve MTU 1163 TB 92 (two engines per shaft)
⑤ MCR Curve MTU 1163 TB 1163 TB 92 (one engine per shaft)
Engine Output (%)
100
90
80
70
60
50
40
30
20
10
0
300 400 500 600 700 800 900 1000 1100 1200 1300
Engine Speed (1/min)
①
②
③
④
⑤

FIG. 26

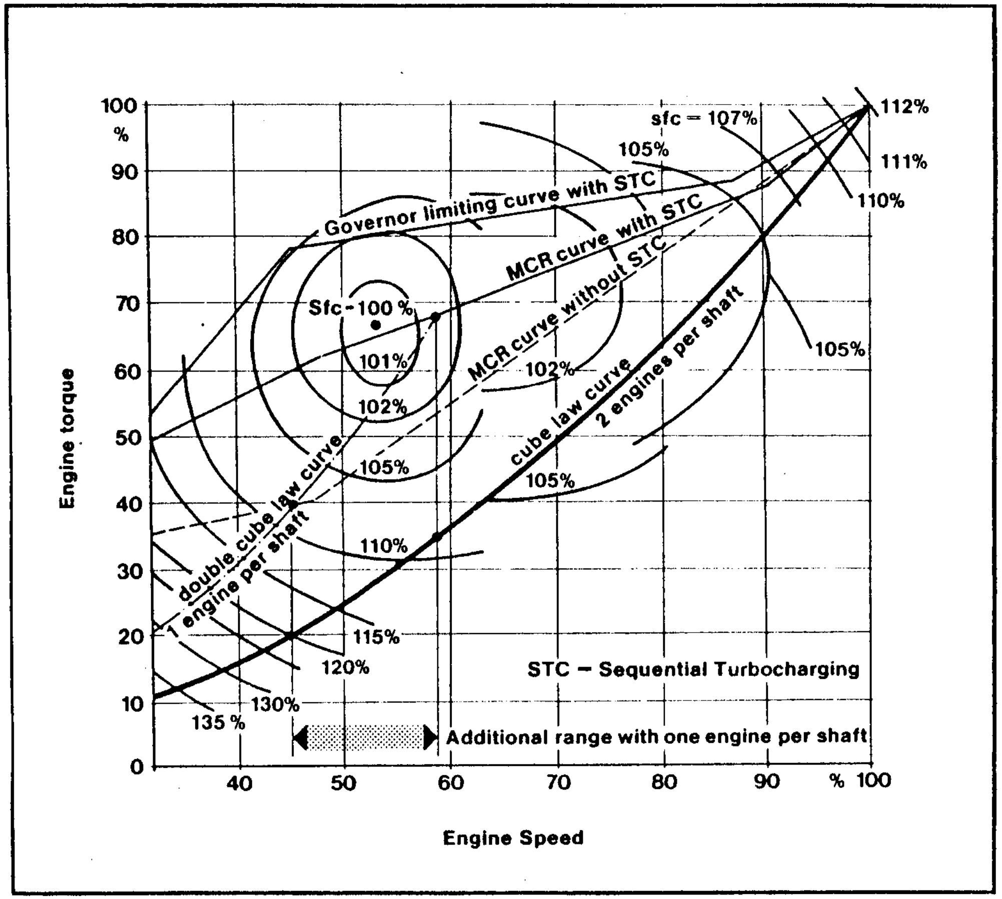
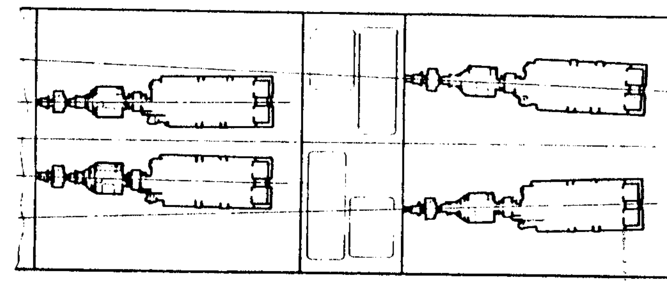


FIG. 27

4-Shafter with 4 × 16 V 956 TB 91
Maximum Power: 4 × 3300 kW = 13 200 kW

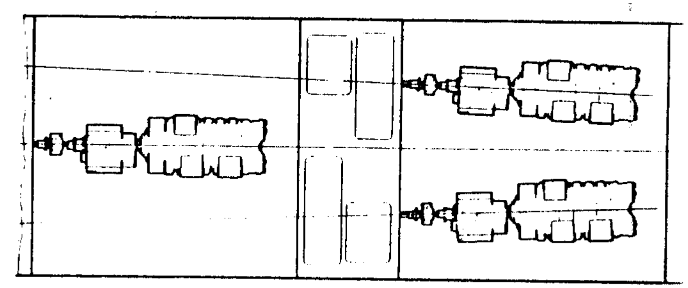


Weights:

Engine with Accessories	12 630 kg
Coupling	360 kg
Rev./Red. Gear	2 145 kg
Total	15 135 kg

Total per Ship: 60 540 kg

3-Shafter with 3 × 12 V 1163 TB 93
Maximum Power: 3 × 4 440 kW – 13 320 kW



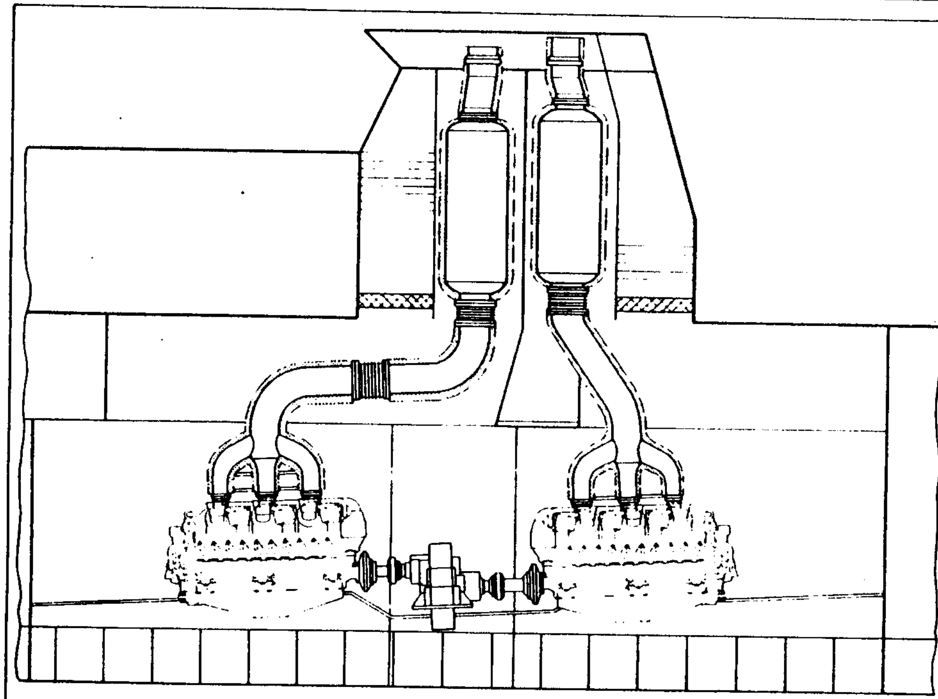
Weights:

Engine with Accessories	13 905 kg
Coupling	550 kg
Rev./Red. Gear	3 745 kg
Total	18 200 kg

Total per ship: 54 600 kg

FIG. 28

CODAD Propulsion Concept for Corvettes



4×20V 1163 TB 93, 10 000 HP (7400 kW) each

FIG. 29

Engine Type		MTU 12V 1163 TB52	ALLISON 501	TYPICAL 900 RPM DIESEL GENSET
Nominal Electric Output	kW	2200	2500	2480
Module Dimensions [1)] L x W x H	m	7.06 x 2.00 x 3.34	8.50 x 2.44 x 3.05	10.00 x 2.30 x 3.40
Bulk Volume [1)]	m^3	47.16	63.26	78.20
	%	100	134	166
Weight [1)]	t	31.0	28.6	74.6
	%	100	92.2	240
Specific Fuel Consumption at 80 % load	g/kWh	221	412	230
	%	100	186	104

1) estimated, including base frame, acoustic enclosure and double/or single elastic mounting

FIG. 30