Existing and Future Demands on the Turbocharging of Modern Large Two-stroke Diesel Engines

Klaus Heim [1]
Manager, R&D, Performance and Testing

Wärtsilä Switzerland Limited [2]
Zürcherstrasse 12, 8401 Winterthur, Switzerland

Summary

Turbocharger development remains one of the key issues for further performance development of large two-stroke diesel engines which power the majority of the world's shipping. This paper reviews the state of current technology in terms of turbochargers and their interaction with these large engines. Pressure ratio and turbocharger efficiency are the key parameters for further development of large two-stroke engine performance. The paper also looks forward to future requirements of turbocharging systems and comments on concepts such as variable turbine geometry, two-stage turbocharging, internal exhaust gas recirculation and turbochargers with auxiliary drive that have been proposed or are being developed for future application.

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[1] klaus.heim@wartsila.com
2 www.wartsila.com

1 Introduction

Large two-stroke diesel engines are predominantly employed for ship propulsion, with only a few being used for electricity generation in land power plants. In fact, they are now the dominant prime mover for the world's deep-sea shipping. Large four-stroke engines are applied in shipping generally only to smaller vessels and to those vessels with special requirements in power concentration or space restrictions such as cruise ships, ferries and Ro-Ro vessels. Steam turbines remain only in the niche of LNG carriers while gas turbines have had only a marginal position in merchant ship propulsion over the past half century.

Turbocharging diesel engines first came on to the market in the 1930s with four-stroke engines but it took until the mid 1950s before turbocharged two-stroke engines were in series production [1]. The first Sulzer turbocharged two-stroke engine in service was a Sulzer 6TA48 engine in 1946 in the works power house in Winterthur, while the first Sulzer turbocharged two-stroke marine engines were a 7RSAD76 and a 6SAD72 which both entered service in December 1956 [2].

That 20-year gap arose through the fundamental differences between four- and two-stroke engines. In the four-stroke cycle, the gas exchange process works by filling and emptying, with little valve overlap. Thus early turbochargers had adequate efficiency to deliver the necessary quantities of charge air required by the then four-stroke engines.

In contrast, the two-stroke cycle is a gas flow process. The working processes of intake, compression, combustion, expansion and scavenging must all be completed in one engine revolution. Instead of clearing the combustion gases by piston displacement as in the four-stroke cycle, in the two-stroke cycle they must be blown through by the scavenging air during the overlap between scavenge and exhaust openings. There needs to be a clear pressure drop between the scavenge air and exhaust manifolds to obtain the necessary flow through the cylinder for good scavenging. As a result, two-stroke engines call for greater efficiency from the turbocharger than four-stroke engines to deliver the required larger air quantity for scavenging while having comparatively less exhaust energy to drive the turbine.

Thus, it took some years of development before turbochargers were available with sufficient efficiency for large two-stroke engines, and for the engine builders to develop the techniques for matching engine and turbocharger. This co-operation between engine builders and turbocharger manufacturers is a crucial aspect of modern diesel engine development.

2 Turbocharging systems on two-stroke diesel engines

Over the past decades, there has been both a considerable advance in engine performance and also extensive rationalisation in engine types. Thus it is helpful, before progressing on to particular aspects of the turbocharging of two-stroke engines and its future development, to review the current state of the art, together with comparisons with the turbocharging of four-stroke engines.

Today for ship propulsion, we have two main engine types with the typical ranges of parameters as given in table I, which is based on the data for engines in the Wärtsilä marine programme. The examples of large two-stroke engines are taken from the Sulzer RTA series.

Table I: Comparison of engine parameters

Engine type	Sulzer low-speed	Wärtsilä medium-speed
Characteristics	two-stroke crosshead	four-stroke trunk-piston
Propeller drive	Direct	Gear or electric
Power/engine, kW	7275–80,080	720–23,280
Cylinders	5–14 L	4–9 L, 12–18 V
Speed, rpm	61–137	327–1500
Bore, mm	480–960	200–640
S/B ratio	2.6–4.2	1.2–1.4
BMEP, bar	12.5–19.5	21.2–26.1
Pmax, bar	155	200
Pscav, bar	3.5–3.9	4.5
Weight, tonnes	171–2300	7.2–432

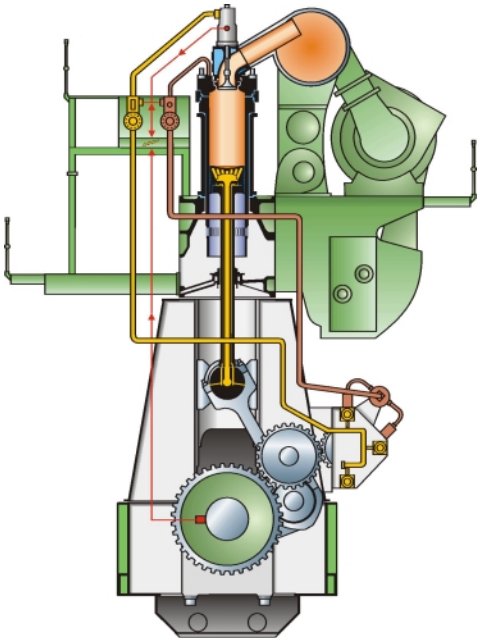

Fig. 1 Cross section of a modern two-stroke diesel engine

In terms of turbocharging and scavenging, the large two-stroke engines have reached a fairly standardised general arrangement (Figs. 1 and 2). The engines are now all uniflow scavenged with air entering the cylinder through ports around the full circumference of the liner at the bottom of the piston stroke, and exhaust from a single poppet-type valve in the centre of the cylinder cover. Sulzer two-stroke engines have been uniflow scavenged since 1982.

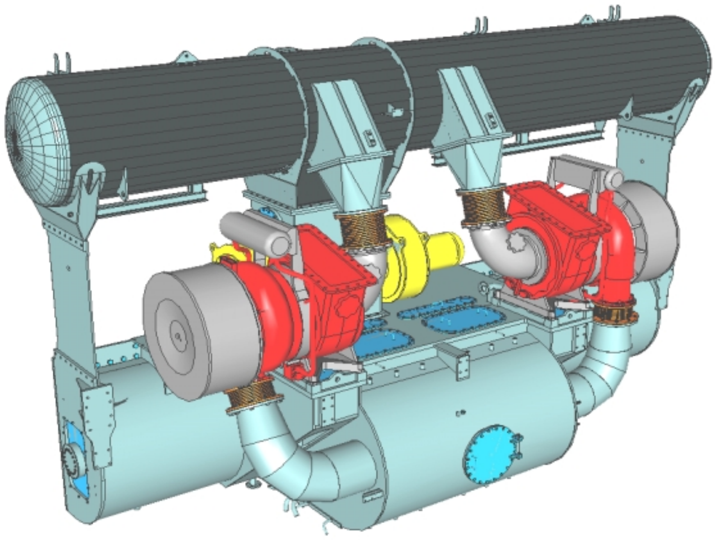

Fig. 2 The turbocharging system of a modern two-stroke diesel engine, with the exhaust manifold at the top and the scavenge air manifold at the bottom. The two turbochargers deliver air into the single air cooler and water separator unit

The turbochargers operate on a constant pressure system, with scavenge air pressures up to 3.9 bar. There are one to four turbochargers, all located high on the side of the engine, outboard and beneath the exhaust manifold. This allows an efficient arrangement with the scavenge air cooler and its associated water separator unit located immediately below the turbocharger and adjacent to the scavenge air space on the piston underside.

However, in some special cases with smaller engines having a single turbocharger, the turbocharger and air cooler can be arranged at the end of the engine.

Electrically-driven auxiliary blowers are provided to supplement the scavenge air delivery when engines are operating below 30 per cent load. This is necessary because, at such low engine loads and speeds, the turbochargers cannot deliver the necessary air for the gas flow process of the two-stroke cycle.

This paper only refers to marine engines directly driving fixed-pitch propellers. Thus the engines follow a propeller characteristic (power versus shaft speed) fixed by the ship/propeller relationship. Full load of the engine is the contracted maximum continuous rated (CMCR) power at the CMCR speed.

A comparison of two- and four-stroke turbocharging systems can be seen in table II. The differences might appear to be relatively small but note the higher specific air flow required for two-stroke engines and the consequently higher minimum efficiency.

Table II: Comparison of turbocharging systems

Engine type	Two-stroke	Four-stroke
Supercharging	constant pressure	Single-pipe exhaust (SPEX)
		or pulse
Number of turbochargers	1–4	1–2
Pressure ratio	up to 4.0	up to 4.5
Minimum TC efficiency, %	68	66
Spec. air flow, kg/kWh	~8	~7
TC mounting on engine	side or end	end
Weight of a turbocharger, tonnes	up to 14.5	up to 4

The most incredible difference is the large sizes of turbochargers used with large two-stroke engines to handle the large air mass flows involved. As an example, a Sulzer 12RTA96C engine of 68,640 kW MCR output uses three large turbochargers, such as the TPL85-B type from ABB Turbo Systems Ltd which weighs 10.5 tonnes each. Three such turbochargers together have an internal shaft power of 24,675 kW, or more than one-third of the engine shaft power. They deliver a total mass flow of 132 m^3/sec at full power.

The largest turbochargers currently available, however, are the TPL91-B type from ABB Turbo Systems with its maximum air flow of 55.7 m^3/sec, a shaft power of 10,450 kW and weight of 14.5 tonnes (Figs. 3 and 4).

Fig. 3 An ABB TPL91-B turbocharger

The turbochargers employed for large two-stroke engines from all today's manufacturers are broadly similar. They are single-rotor machines with a single radial compressor driven by a single-stage axial turbine. For today's pressure ratios up to 4.0, the compressors are of aluminium. The turbine blades are without a damping wire. The exhaust gas outlet casing is rectangular for high efficiency.

Fig. 4 Rotor from an ABB TPL91-B turbocharger

Cooling of the partition wall (back wall of the compressor wheel) is now ready to go into serial production. A small amount of compressed air is used to stop exhaust gas entering the bearing casing.

The turbocharger bearings are supplied with lubricating oil from the engine system but each turbocharger has its own integrated emergency lubricating oil tank in case there is a failure of the lubricating oil supply (the pumps are normally electrically driven).

3 Major requirements of turbochargers and their influence on engine performance

A proper turbocharger layout has several influences on engine performance. It gives:

- Optimised fuel consumption owing to a high mass purity as a result of a highly efficient scavenging process
- Low NO_X emissions through optimised low-NO_X tuning based on a well defined scavenge air pressure
- Low smoke emissions at part load by means of a high air excess rate
- Low exhaust valve temperatures at all loads for long component life.

The main criteria for specifying turbochargers for future engine developments are the turbocharger overall efficiency and the pressure ratio. Figure 5 shows the trend of engine brake mean effective pressure (BMEP) with respect to these two parameters.

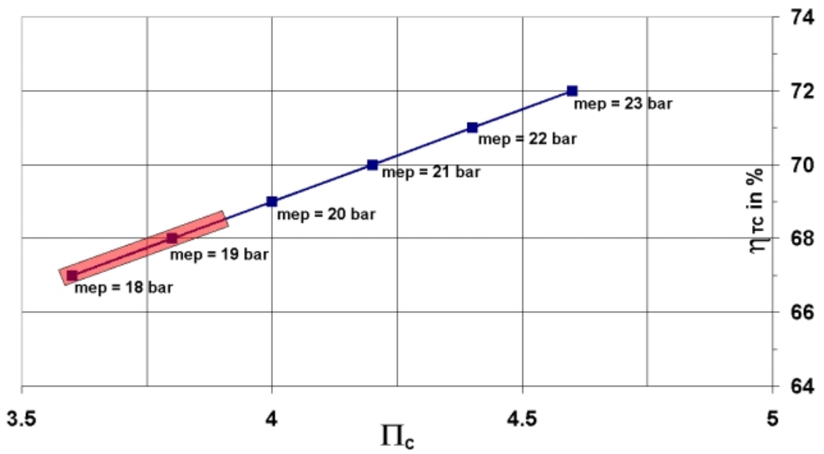

Fig. 5 Trend of engine BMEP with respect to turbocharger overall efficiency and pressure ratio

Today's engines operate at BMEP up to 19.5 bar, with turbocharger efficiencies of just over 68 per cent and pressure ratios up to 3.95. If the engine BMEP would be increased to 21 bar then the turbocharger efficiency would need to be raised to 70 per cent and pressure ratio to about 4.2.

The reason for this relationship between turbocharger performance and engine BMEP is because a higher BMEP requires a higher pressure ratio to keep the air excess ratio at the same level. At the same time turbocharger efficiency must also be increased to keep the air mass flow and cylinder mass purity at desired values.

In the case of four-stroke engines, turbocharger speed is a significant criterion because of the need to obtain higher pressure ratios. Turbocharger speed is not a major limiting factor for two-stroke engines.

3.1 Low-port concept

One of the most interesting developments of recent years in matching engines and turbochargers is the 'low-port' concept in 1997. This took advantage of the availability of new turbochargers of higher efficiency, about 68 per cent compared with the previous 65 per cent. Because the required scavenge air mass flow could be delivered using less exhaust gas energy, the height of the scavenge air ports could be reduced (Fig. 6) and the opening of the exhaust valve delayed, thereby lengthening the effective working expansion stroke and thus reducing specific fuel consumption (see table III). An important consideration was that this benefit could be obtained without raising key component temperatures. At the same time, the exhaust temperature was not reduced too low, so keeping up the potential for waste heat recovery from an economiser for heating services.

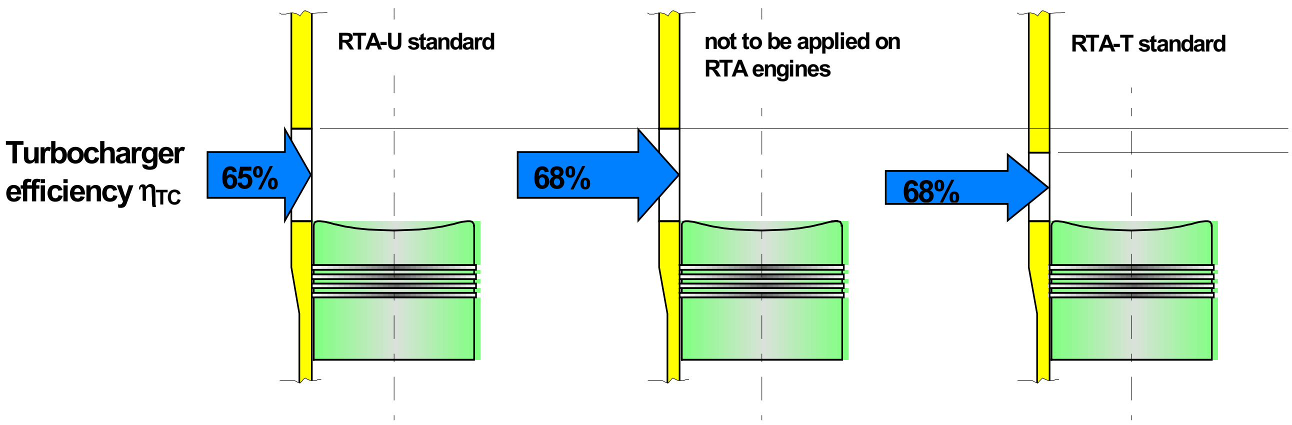

Fig. 6 Schematic of the port arrangement for the previous conventional standard, the same concept using higher efficiency turbochargers, and the low-port concept with the higher efficiency turbochargers

Table III: Key parameters for the 'low-port' concept

Turbochargers	Basis	High-efficiency	High-efficiency
TC overall efficiency, %	65	68	68
Ports	Basis	Basis	Low ports
Exhaust valve opens, °CA ATDC	119	119	122
Scavenge port opens, °CA ATDC	142	142	145
Exhaust temperature, °C	275	250	270
BSFC, g/kWh	173	172	170
Valve temperature difference, °C	Basis	– 6	±0
Exhaust gas flow, kg/kWh	8.2	9.0	8.3

3.2 Turbocharger matching and testing

Suitable turbocharger layouts are achieved through a well-established procedure. During the engine design process, an appropriate turbocharger specification is laid down based on the calculation of certain parameters – scavenge air pressure and the full-load and part-load turbocharger efficiencies (Fig. 7). The desired safety margin against surging, usually 15 per cent, is also specified.

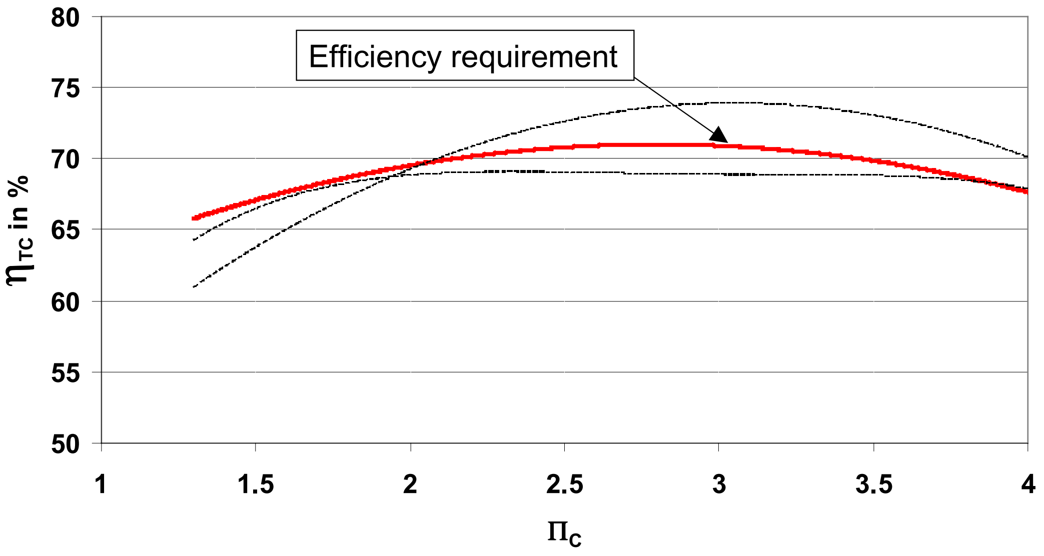

Fig. 7 Efficiency requirement of turbochargers for large two-stroke engines with respect to engine load as expressed as compressor pressure ratio

Then in the engine shop tests, the scavenge air pressure is adjusted based on checking the compliance with defined requirements, such as the level of turbocharger efficiency, shape of the efficiency curve and related tolerances. If necessary the turbine nozzle ring is changed.

Tests are also made to check the engine's stability against turbocharger surging. This is achieved in one of three ways. The exhaust back pressure can be increased at full engine power. Another method is to cut off fuel to one engine cylinder, which is close to the turbocharger, while the engine is running at part load. The third way is to suddenly reduce the dynamometer load by about 25 per cent from full load.

3.3 Surging

Turbocharging surging is a common hazard as it can be brought on by many events which result in a shifting of the turbocharger operating point towards the surge line (Fig. 8). The possible reasons for turbocharger surging include:

- Insufficient engine room ventilation
- Clogging of turbocharger intake silencer, compressor, scavenge air cooler, turbine gas inlet protection grid, turbine nozzle ring, and exhaust gas economiser and silencer, and deposits on turbocharger turbine blades
- Damage of scavenge air flaps
- Damage or wear of fuel injection valves
- Excessive lubricating oil feed rate
- Wear of turbocharger components such as nozzle ring, turbine blades, cover ring or shroud ring.

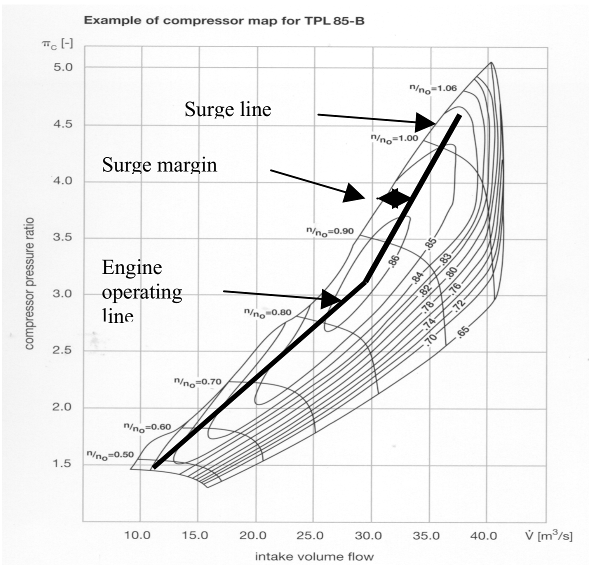

Fig. 8 Example of a compressor map showing the engine operating line and the surge margin with respect to the surge line

Thus it can be seen that the best prevention against surging is careful attention to engine maintenance with an important aspect being the cleaning of the turbocharger.

Fouling on the compressor wheel and diffuser as well as on the turbine nozzle ring and turbine blades degrades the turbocharger performance. The result can be increased exhaust temperatures, increased engine component temperatures and increased fuel consumption, or in extreme cases an increased danger of surging. The dirt deposited on the compressor side is the usual dust from the environment plus oil mist from the engine room. On the turbine side, the deposits are usually non-combustible components of the heavy fuel oil and surplus cylinder lubricating oil.

The solution is regular cleaning of turbocharger components at regular intervals. This is usually done while the engine is running, but at reduced speed. Dry cleaning of the turbine needs to be done every 24–48 hours using a granular medium. Wet cleaning of the turbine is then done every 48–500 hours using water alone. Wet cleaning of the compressor is done every 25–100 hours.

4 Future trends

The future trend for turbocharging large two-stroke reflects the symbiotic nature of the relationship between the engine and its turbocharger. Certain trends in engine parameters can be identified but these will need matching developments in turbocharger technology by the specialist manufacturers.

The following trends can be clearly identified:

- Higher specific power output to achieve less weight per kW
- Higher engine efficiency
- Lower exhaust emissions
- Improved engine reliability, and longer times between overhauls
- Lower manufacturing costs.

To achieve objectives in such trends, turbochargers will need further development in terms of:

- Higher scavenge air pressure
- Higher turbocharger efficiency
- Compact dimensions and competitive price.

The general trend in turbocharger parameters can be seen in figure 9. In 1990, engine BMEP was in the region of 17 bar in new engine types first delivered in that year. In 2000, it had moved up 19 bar. This year, Wärtsilä shall deliver the first engine running at 19.5 bar BMEP (the Sulzer RT-flex60C). Thus it might not be unreasonable to consider the possibility of engines running at 22 bar BMEP by 2010.

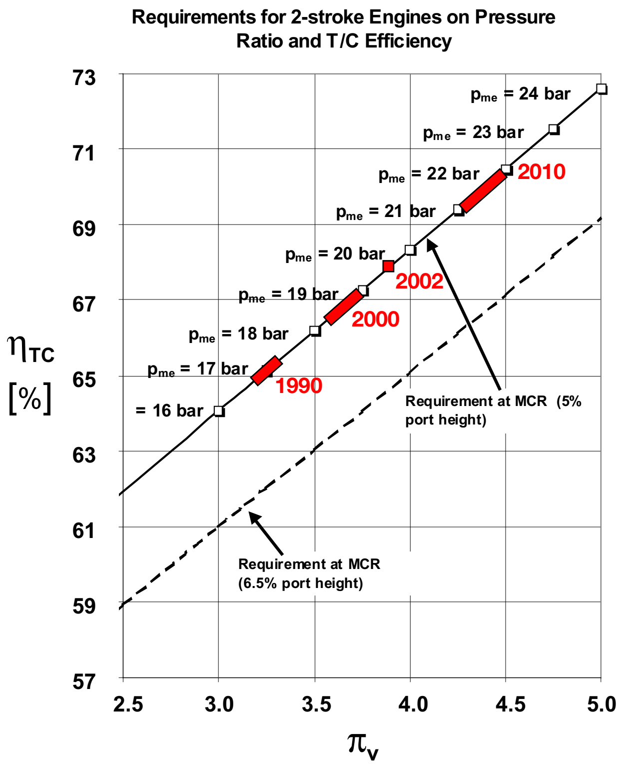

Fig. 9 Future development of large two-stroke engines will raise the requirements for turbocharger overall efficiency and pressure ratio

That level of BMEP makes a strong demand for increased pressure ratio and efficiency from the turbocharger because it is important that certain boundary conditions are maintained. The temperatures of engine components around the combustion chamber must remain at today's level. This is a crucial issue for engine reliability.

For the environment, the smoke level during engine part-load running must not increase.

Finally, the engine fuel consumption must remain the same. This has an effect on the environment but is an important economic matter for the engine user, the shipowner. Thus there is always pressure to reduce fuel consumption [3].

Various concepts are often mentioned when considering today's and future turbocharging developments. Some, such as waste gates, variable turbine geometry (VTG) and two-stage turbocharging, are envisaged for overcoming shortcomings of turbochargers at higher pressure ratios. Others such as exhaust gas recirculation are purely for reducing exhaust emissions. Perhaps a new concept such as turbochargers with auxiliary drives might be more fruitful.

4.1 Waste gate or air bypass

Exhaust waste gates or air bypasses are usually envisaged for improving the part-load performance of turbocharging systems.

However, such devices are not needed with large two-stroke engines owing to their general operating regime. In ship propulsion these engines operate mainly at around 85 per cent CMCR (contracted maximum continuous rating) power. In addition they have few transients during ocean passages, with long periods spent running at steady loads. In any case, auxiliary blowers are needed for operation at loads below about 30 per cent.

In fact, exhaust waste gates can be a definite disadvantage for large two-stroke engines because the loss of exhaust gas through the waste gate reduces the turbocharging efficiency which may lead to higher thermal loads in combustion chamber components.

4.2 Variable turbine geometry (VTG)

Variable turbine geometry has potential for improving part-load performance of the turbocharging system. It involves mechanical linkage to vary the angle of incidence of the turbine inlet guide vanes. However, at present, it is not considered to be a practical or economic proposition for engines burning heavy fuel oil.

The main problem is the fouling of the adjustable guide vanes by unburned fuel components and cylinder lubricating oil. Such fouling can readily accumulate during the long periods in which marine engines run at steady loads on long ocean passages.

As the great majority of ships run on heavy fuel oil, VTG is not practical with large marine two-stroke engines. However, fouling might be reduced if advanced cylinder lubrication systems are developed which would allow greatly reduced cylinder oil feed rates.

Another drawback of VTG is that the extra mechanism adds to the cost of the turbochargers. On a two-stroke engine, the system would be further complicated by a need to provide means of adjusting the compressor, perhaps by a variable orifice (VCG, variable compressor geometry), to avoid surging of the compressor while changing the angle of the VTG guide vanes.

4.3 Two-stage turbocharging

Two-stage turbocharging is another concept which has often been mooted in the past when available turbochargers appear to be reaching their limits in efficiency and pressure ratio.

Higher overall turbocharger efficiencies can be reached with two stages because it is possible to have intercooling between the two stages thereby reducing the compression work needed in second turbocharger stage.

It has the clear benefits of keeping within the strength limits of known materials, and allowing the operating fields of compressors and turbines to be optimised for a wide load range. However, the low-load efficiency of two stages of today's turbochargers is worse than is obtained with single-stage operation.

The major drawback of two-stage turbocharging is the complex arrangement of air and exhaust ducts (six ducts per turbocharging unit). It requires much more space which is, in any case, restricted in ships' engine rooms. Even the space required by today's single-stage systems is becoming critical.

It must also not be forgotten that two-stage turbocharging can result is less heat, or poorer quality heat owing to lower temperatures, being available from the scavenge air coolers for hot water for onboard use.

There is thus always the preference to remain with the simplest arrangement – single-stage turbocharging. Yet to keep up with the requirements of future engine developments, it requires higher technology turbochargers operating at the known limits of material strength and vibrations. It is currently possible to run at compressor pressure ratios of about 4.5, as determined by the compressor material, compared with the ratio of 4.0 currently used on two-stroke engines.

4.4 Water-cooled residual gas (WaCoReG)

Another future trend affecting turbocharging is exhaust gas recirculation (EGR). This is popularly known for small four-stroke engines as an important way to obtain further reductions in NO_X emissions. EGR reduces NO_X formation at source by reducing the oxygen available in the engine cylinder and increasing the specific heat of the gas in the cylinder. It also offers interesting benefits in large two-stroke engines but with the added benefit that the two-stroke cycle allows EGR to be applied in a simple, elegant way [3].

Whereas four-stroke engines need actual recirculation of exhaust gases through external manifolds and coolers, exactly the same effect can be obtained in two-stroke engines by simply adapting the engine scavenging process to decrease the purity of gas in the cylinder at the start of compression, as long as cylinder component temperatures are kept down by suitable means. Such 'internal EGR' is achieved by reducing the height of scavenge ports to reduce the scavenge air mass flowing through the cylinder. The lower scavenge ports also have the benefit of allowing greater expansion in the cylinder (see above under '3.1 Low-port concept') and thus improving fuel consumption.

Although no external hardware is required for exhaust gas recirculation on a two-stroke engine, internal EGR requires a direct water injection (DWI) system to reduce the cylinder gas temperatures which rise with the reduction of scavenge air mass flow. DWI thus provides the means of controlling component temperatures.

The combination of internal EGR and DWI is being developed by Wärtsilä as the water-cooled residual gas, or WaCoReG system (Fig. 10). DWI is also being developed as another technology for reducing NO_X emissions. However, the water is injected earlier in the compression stroke than when DWI is applied alone, so that it reduces temperature levels thereby keeping thermal loads much the same as when running without internal EGR.

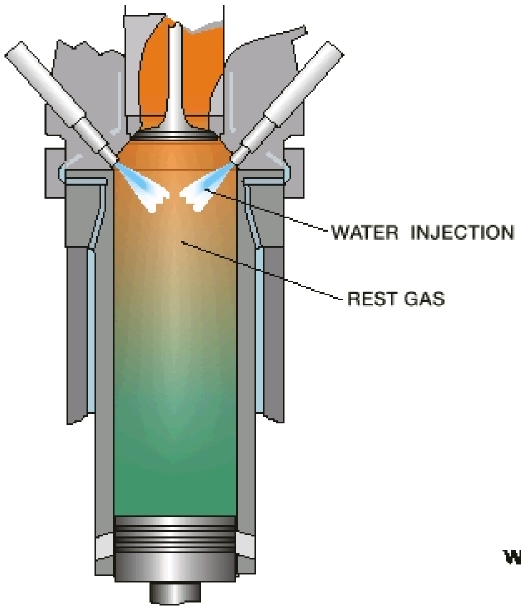

Fig. 10 Schematic of the WaCoReG system with direct water injection and the retention of residual gas in the cylinder

WaCoReG is expected to reduce NO_X emissions by up to 70 per cent from today's levels in engines which comply with the regulations in the MARPOL 73/78 convention, Annex VI.

There would be an added benefit if WaCoReG is applied to a Sulzer RT-flex engine as the electronically-controlled exhaust valve timing would allow easier adjustment of the EGR level.

The Sulzer RT-flex system is an electronically-controlled common-rail system for fuel injection and exhaust valve actuation [4]. It has been applied on a research engine since June 1998 and the first Sulzer RT-flex engine entered service in September 2001 [5].

4.5 Turbochargers with auxiliary drive

One idea which merits further investigation is to employ turbochargers with auxiliary drive. Each turbocharger would incorporate a motor/generator coupled to its rotor (Fig. 11). At times when the engine is running at a load for which the turbochargers cannot deliver sufficient scavenge air, then additional energy can be fed in by the motor drive to boost the scavenge air delivery (Fig. 12).

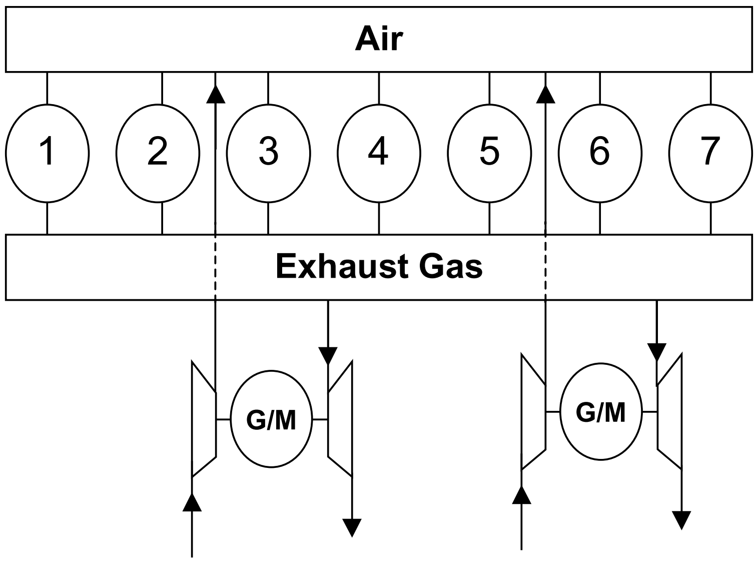

Fig. 11 Schematic for an arrangement with generator/motor units incorporated in each turbocharger

At higher engine loads when the available exhaust energy is more than adequate to deliver the necessary scavenge air flow, then the surplus energy could be abstracted by the integral auxiliary drive running as a generator.

Turbochargers with auxiliary drive as explained above offer some enticing benefits but at the cost of more expensive turbochargers and the added power management system for the integral motor/generator units.

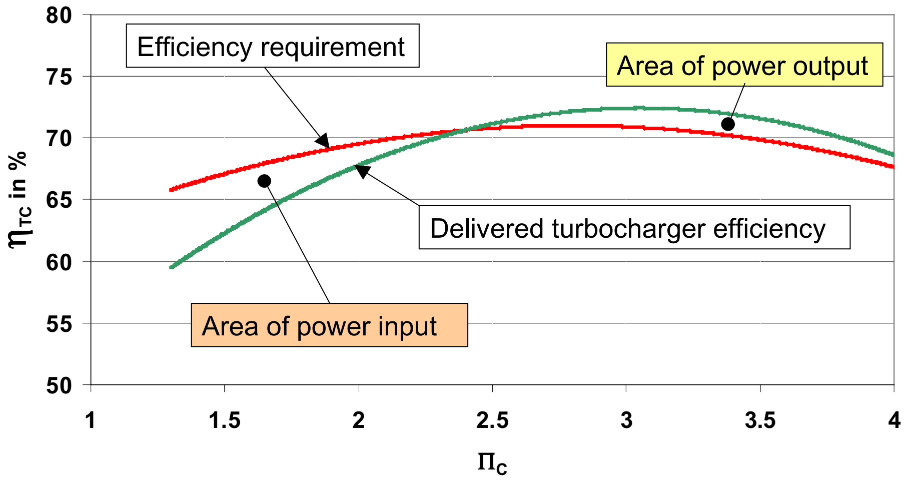

Fig. 12 Turbochargers with auxiliary drive would be able both to meet a shortfall the turbocharger efficiency requirement in large two-stroke engines at low loads while utilising the surplus energy for electricity generation at high loads. The curves are of for turbocharger efficiency with respect to engine load as expressed as compressor pressure ratio

This system would simplify the engine design. Auxiliary blowers and the scavenge air flaps could be omitted. The thermal load on the exhaust valves could be further reduced for certain engines during low-load operation. There would also be improved acceleration of the turbochargers and hence better combustion in the engine during transient part-load conditions. Something of a bonus would be given in the surplus electrical power which would become available in the full load region.

5 Conclusions

Turbocharging of large two-stroke engines is at an interesting stage of development. A further increase in pressure ratio and turbocharger efficiency at both full and part load is required to keep up with the pace of continuous engine performance development.

It is envisaged that single-stage turbocharging will remain the common supercharging principle for large two-stroke engines during the next eight to ten years.

New concepts such as internal exhaust gas recirculation (through WaCoReG) and turbochargers with auxiliary drive could help to meet future requirements in terms of reduced exhaust emissions, improved engine efficiency and a simple and reliable engine design.

Turbocharger development remains one of the key issues for further performance development of large two-stroke diesel engines. And we have every confidence that the turbocharger manufacturers will rise to this challenge.

6 Acknowledgements

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