Thermo Efficiency System (TES)
for Reduction of Fuel Consumption and CO_2 Emission

Contents:

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Introduction

Following the trend of required higher overall ship efficiency since the first oil crisis in 1973, the efficiency of the main engines has increased and, today, the fuel energy efficiency is about 50%. This high efficiency has, among others, led to a correspondingly lower exhaust gas temperature after the turbochargers.

Even though a main engine fuel energy efficiency of 50% is relatively high, the primary objective for the shipowner is still to lower the total fuel consumption of the ship and, thereby, to reduce the CO_2 emission of his ship.

Today an even lower CO_2 emission can be achieved by installing a Thermo Efficiency System. However, the main demand for installation of a Thermo Efficiency System is that the reliability and safety of the main engine/ship operation must not be jeopardised.

As an example, the heat balance diagram for the nominally rated 12K98ME/MC engine (18.2 bar) of the standard high-efficiency version is shown in Fig. 1a. Fig. 1b shows an example based on the Thermo Efficiency System, valid for a single-pressure steam system, together with the corresponding figures for a dual-pressure steam system shown in parenthesis.

The primary source of waste heat of a main engine is the exhaust gas heat dissipation, which accounts for about half of the total waste heat, i.e. about 25% of the total fuel energy. In the standard high-efficiency engine version, the exhaust gas temperature after the turbocharger is relatively low, and just high enough for production of the necessary steam for a ship's heating purposes by means of the exhaust gas fired boiler.

However, a main engine with changed timing and exhaust gas bypass – which redistributes the exhaust gas heat from high amount/low temperature to low amount/high temperature – increases the effect of utilising the exhaust gas heat, but at the same time may slightly reduce the efficiency of the main engine itself. Such a system is called a Thermo Efficiency System (TES).

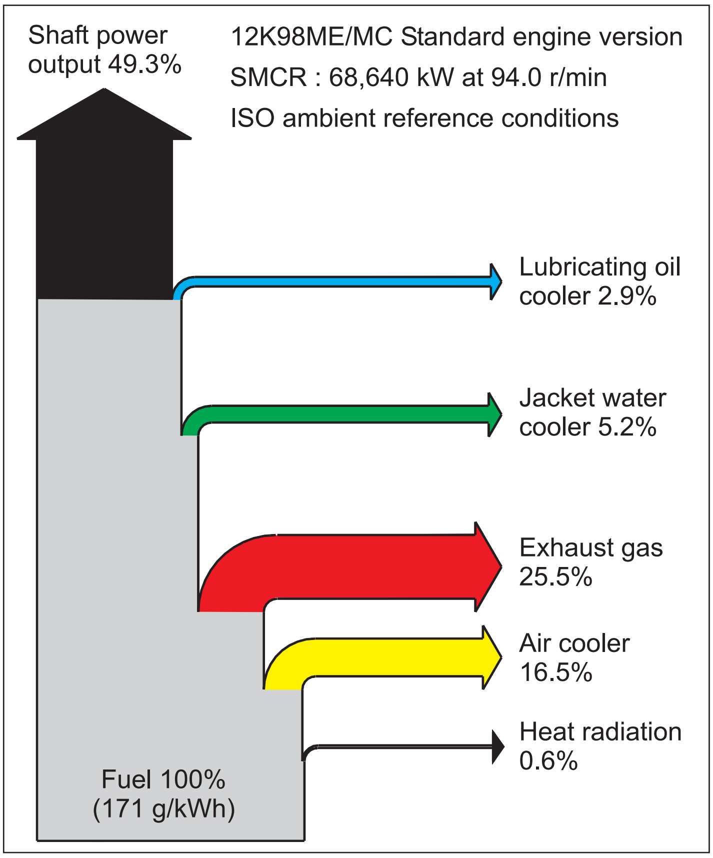

Fig. 1a: Heat balance diagram of the nominally rated 12K98ME/MC engine of the standard engine version operating at ISO ambient reference conditions and at 100% SMCR

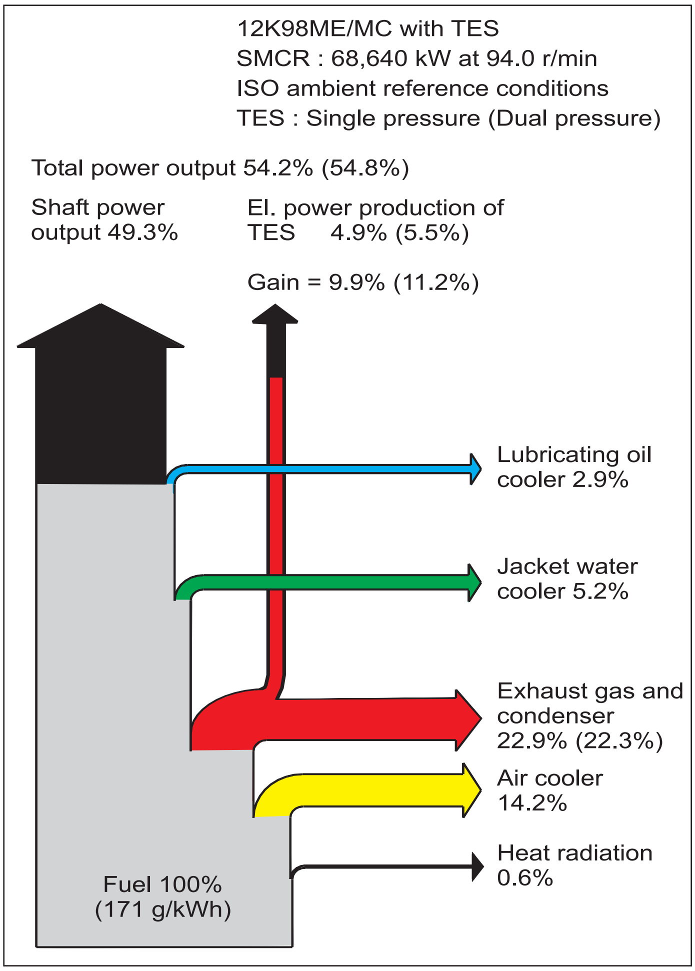

Fig. 1b: Heat balance diagram of the nominally rated 12K98ME/MC engine in Thermo Efficiency System (TES) version operating at ISO ambient reference conditions and at 100% SMCR

Description of the Thermo Efficiency System

Power concept and arrangement

The Thermo Efficiency System (TES) consists of an exhaust gas fired boiler system, a steam turbine (often called a turbo generator), an exhaust gas turbine (often called a power turbine) and a common generator for electric power production. The turbines and the generator are placed on a common bedplate. The system is shown schematically in Fig. 2a, and the arrangement of the complete turbine generating set as proposed by Peter Brotherhood is shown in Fig. 2b.

The exhaust gas turbine is driven by a part of the exhaust gas flow, which bypasses the turbochargers. The exhaust gas turbine produces extra output power for electric power production, which depends on the bypassed exhaust gas flow amount.

When a part of the exhaust gas flow is bypassed the turbocharger, the total amount of air and gas will be reduced, and the exhaust gas temperature after the turbocharger and bypass will increase. This will increase the obtainable steam production from the exhaust gas fired boiler.

The exhaust gas bypass valve will be closed for engine loads lower than about 50% SMCR, which means that the exhaust gas temperature will be reduced when operating below 50% SMCR.

The power output from the exhaust gas turbine is transmitted to the steam turbine via a reduction gear (see Figs. 2a and 2b) with an overspeed clutch, which is needed in order to protect the exhaust gas turbine from overspeeding in case the generator drops out.

The total electric power output of the TES – which reduces the ship's fuel costs – is only a gain provided that it can replace the power output of other electric power producers on board the ship. Otherwise, a shaft power motor connected to the main engine shaft could be an option, as also shown in Fig. 2a, but this extra system is rather expensive.

In general (without a shaft power motor installed), when producing too much electric power, the (high pressure) superheated steam to the steam turbine is controlled by a speed control governor through a single throttle valve, which means that the surplus steam is dumped via a dumping condenser. When the generator is operating in parallel with the auxiliary diesel generators, the gov-

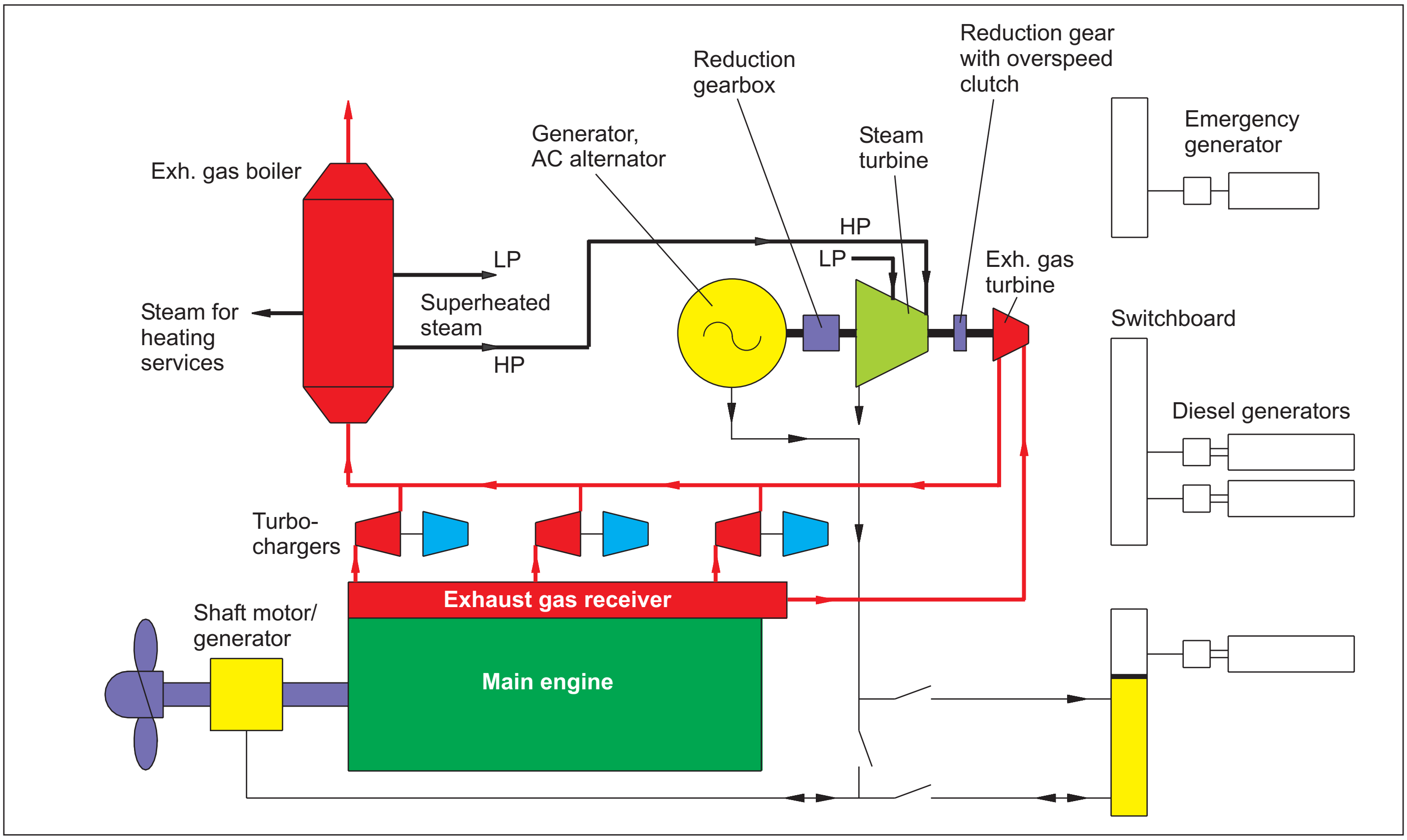

Fig. 2a: Power concept for the Thermo Efficiency System

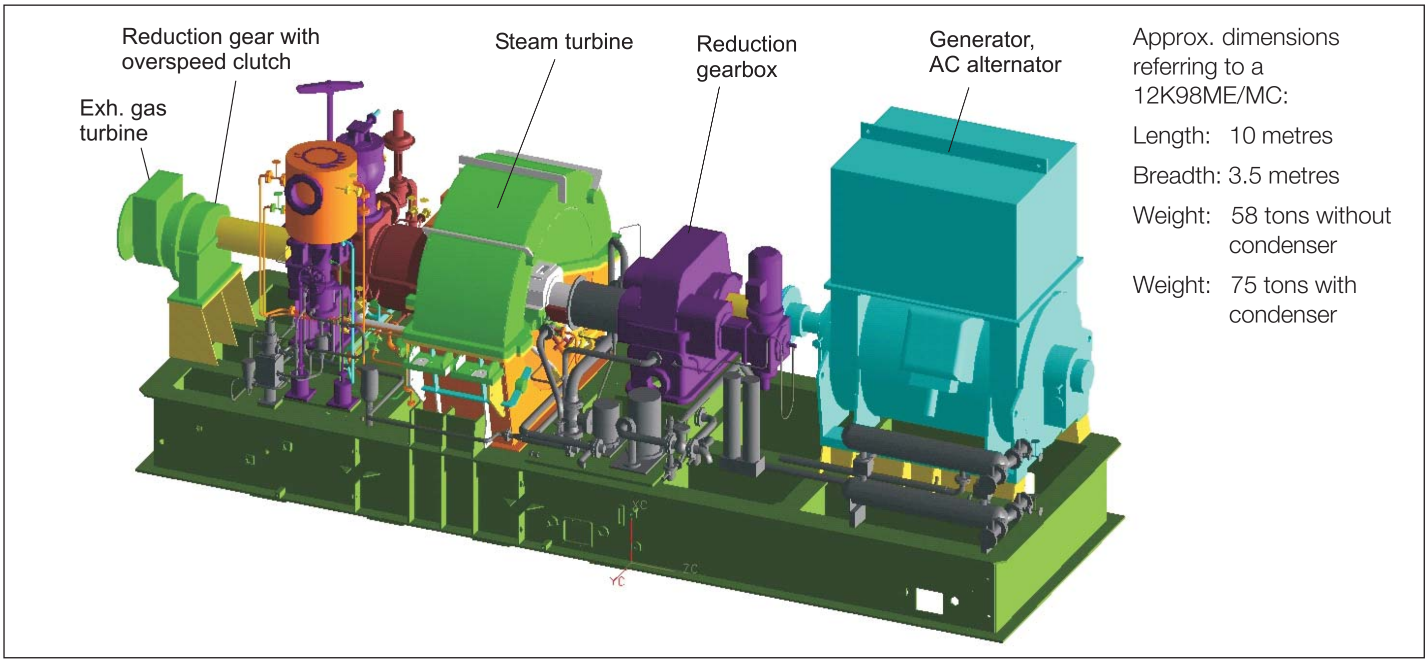

Fig. 2b: Arrangement of the complete turbine generating set as proposed by Peter Brotherhood Ltd

ernor operates in the normal way to give correct load sharing.

Main engine performance data

The exhaust gas bypass and turbine are available with the following approx. effects, compared with a standard high-efficiency main engine version without an exhaust gas bypass:

Parameters	Open exhaust gas bypass for exhaust gas turbine
Power output of exhaust gas turbine at 100% SMCR, up to	+4.6% SMCR power
Reduction of total exhaust gas amount, approx.	-13%
Total increase of mixed exhaust gas temperature after bypass, up to	+50°C
Increased fuel consumption	from 0.0% to +1.8%

The mixed exhaust gas temperature before the exhaust gas boiler, and valid for the TES and based on ISO ambient reference conditions, is shown as a function of the engine load in Fig. 3. When operating under higher ambient air temperatures, the exhaust gas temperature will be higher (about +1.6°C per +1°C air), and vice versa for lower ambient air temperatures.

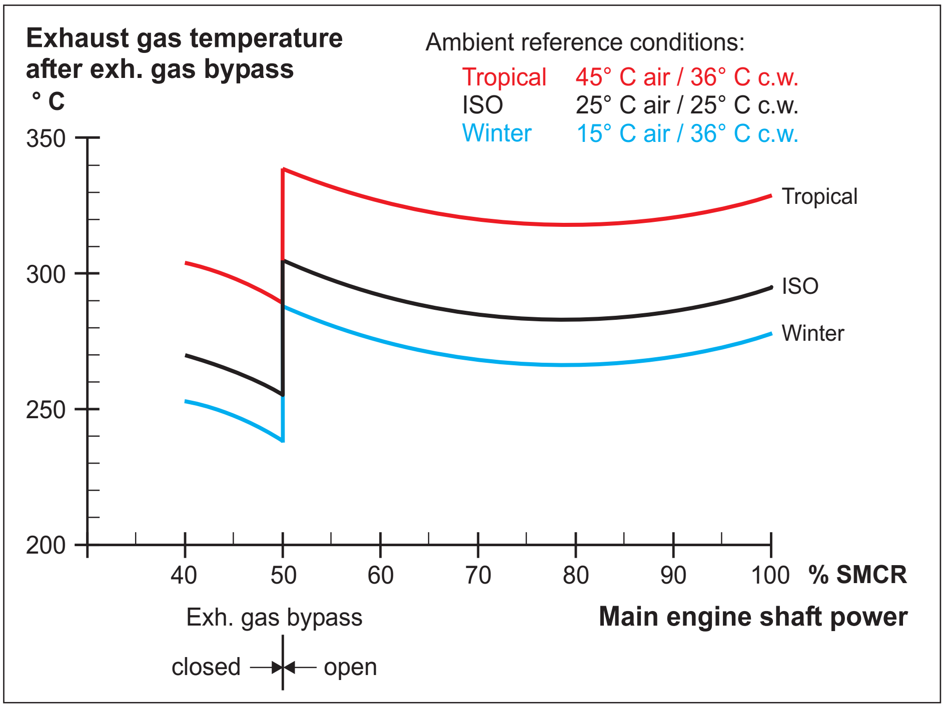

Fig. 3: Exhaust gas temperature after exhaust gas bypass for a main engine with TES

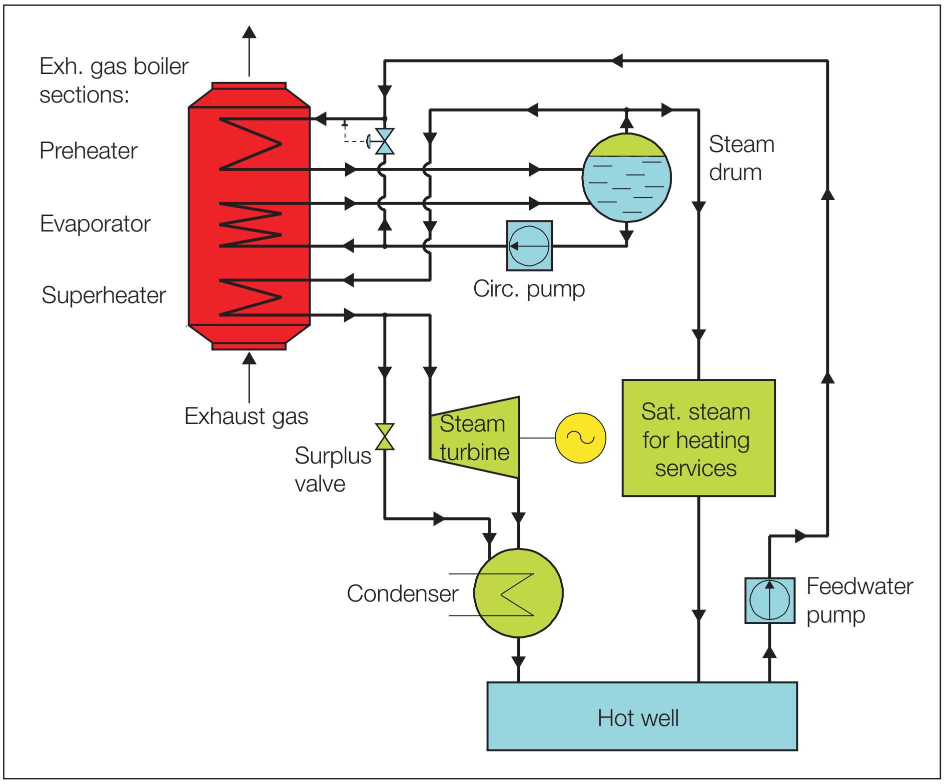

Fig. 4: Process diagram for the Thermo Efficiency System – single-pressure exhaust gas boiler system with a single-pressure steam turbine

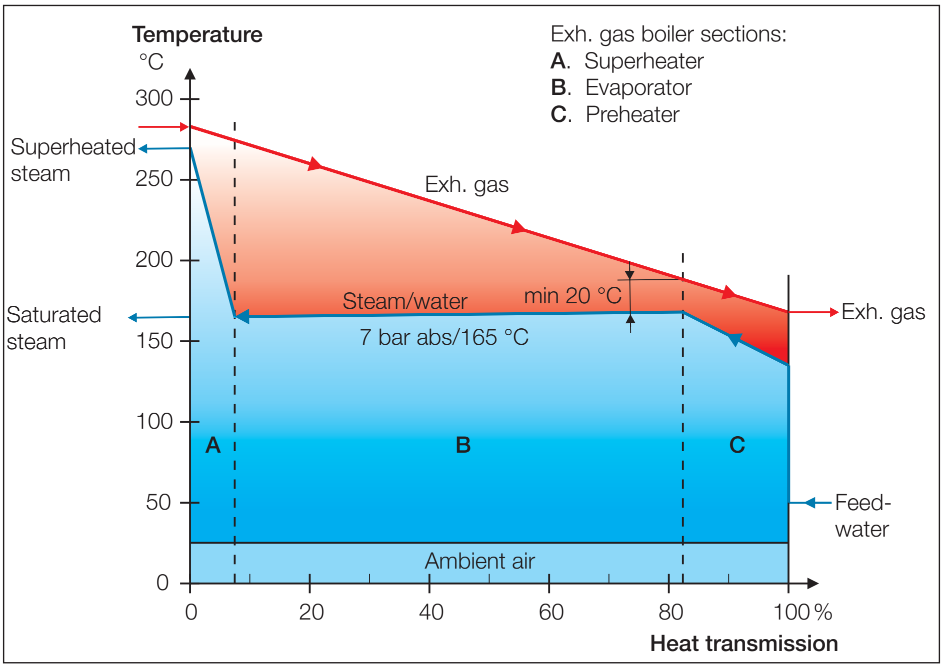

Fig. 5: Temperature/heat transmission diagram of an exhaust gas boiler with single-pressure steam system valid for a main engine with TES and operating at 85% SMCR/ISO

The increased fuel consumption of the main engine depends on the actual maximum firing pressure (P_{max}) used. The P_{max} used for TES will normally be increased – compared with a standard engine – and thereby an increase of the specific fuel oil consumption can be avoided when using TES.

Exhaust gas boiler and steam systems

The exhaust gas boiler and steam turbine systems analysed in this paper are based on the two types below:

1. Single-pressure steam system
   The simple single-pressure steam system is only utilising the exhaust gas heat. See the process diagram in Fig. 4 and the corresponding temperature/heat transmission diagram in Fig. 5. The steam drum from the oil fired boiler can also be used instead of a separate steam drum.

2. Dual-pressure steam system

When using the dual-pressure steam system, it is not possible to install an exhaust gas low-pressure preheater section in the exhaust gas boiler, because the exhaust gas boiler outlet temperature otherwise would be too low and increase the risk of wet (oily) soot deposits on the boiler tubes.

The more complex dual-pressure steam system, therefore, needs supplementary waste heat recovery (WHR) sources (jacket water and scavenge air heat) for preheating of the feedwater which, of course, will increase the obtainable steam and electric power production of TES. See the process diagram in Fig. 6 and the corresponding temperature/heat transmission diagram in Fig. 7.

If no alternative waste heat recovery sources are used to preheat the feedwater, the low pressure (LP) steam may be used to preheat the feedwater, involving an about 16% reduction of the total steam production.

The available superheated steam used for the steam turbine is equal to the surplus steam after deduction of the saturated steam needed for heating services.

The exhaust gas boiler has to be designed in such a way that the risk of soot deposits and fires is minimised, Ref. [1].

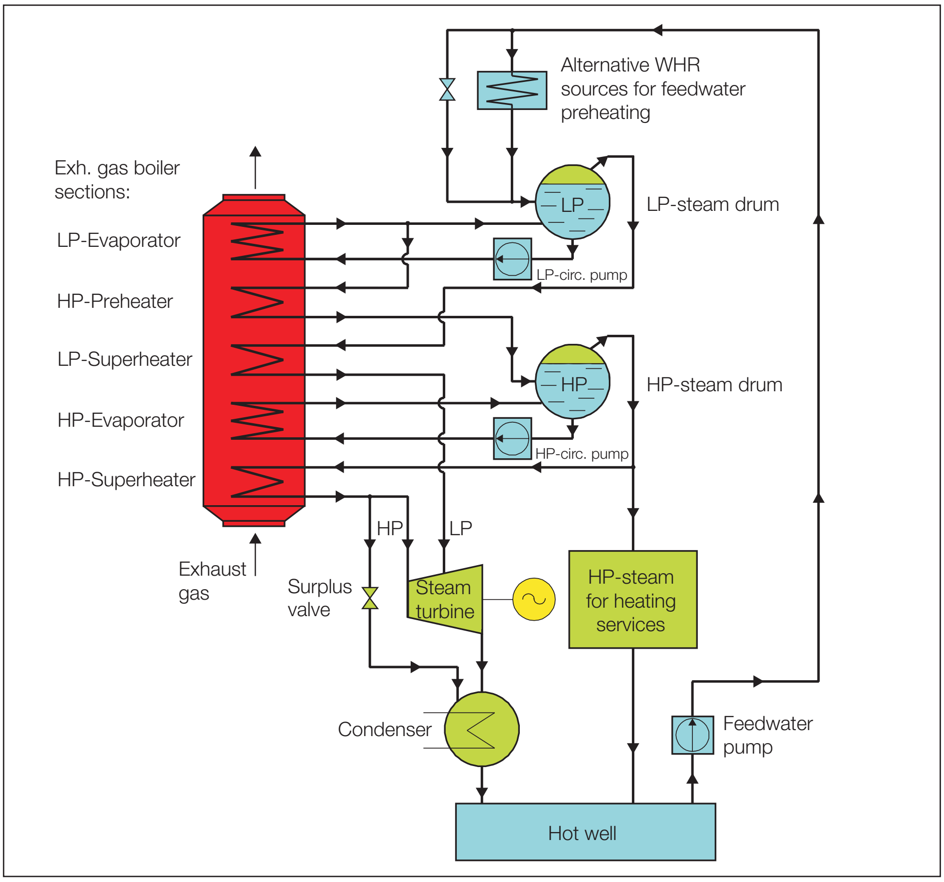

Fig. 6: Process diagram for the Thermo Efficiency System – dual-pressure exhaust gas boiler system with a dual-pressure steam turbine

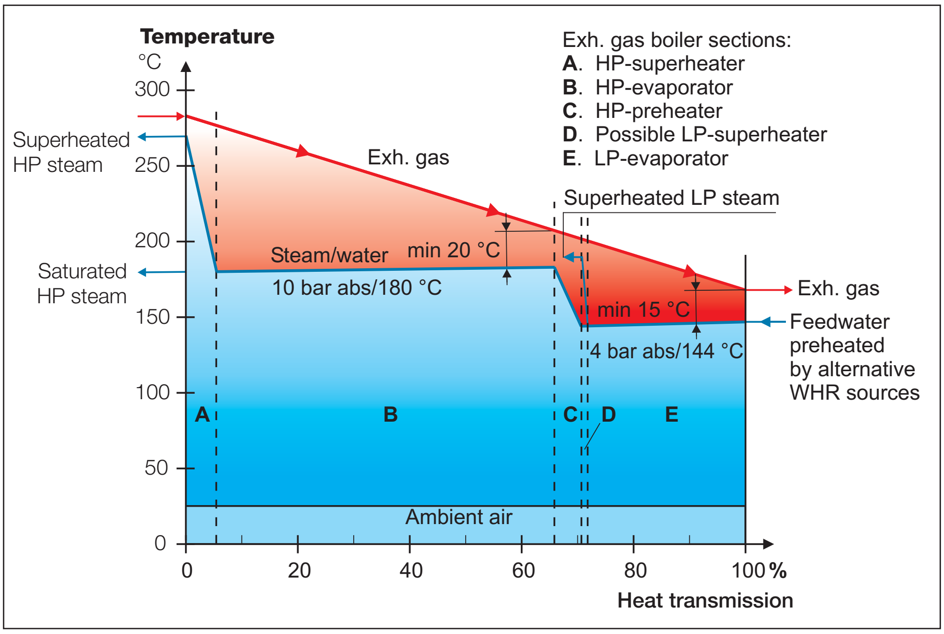

Fig. 7: Temperature/heat transmission diagram of an exhaust gas boiler with dual-pressure steam system valid for a main engine with TES and operating at 85% SMCR/ISO

Obtainable Electric Power Production of the Thermo Efficiency System

Exhaust gas turbine output

The exhaust gas bypass for the exhaust gas turbine has a bypass gas amount of approx. 12% of the total exhaust gas amount at 100% SMCR. This bypass-gas amount led through the exhaust gas turbine will typically produce an available power output of max. 4.6% of the SMCR power when running at 100% SMCR. The corresponding electric power output will be somewhat lower because of generator and gear losses.

At part load running of the main engine, the power output will be reduced by approximately the square root of the engine load.

As an example, the maximum available power output of the exhaust gas turbine valid for a nominally rated 12K98ME/MC (18.2 bar) engine is shown in Fig. 8, as a function of the engine load.

Exhaust gas and steam turbine generator output – single pressure

The single-pressure steam system is a system where all heat recovered comes from the exhaust gas heat only, which makes it relatively simple, see Figs. 4 and 5.

As low gas temperatures (risk of condensed sulphuric acid) and low gas velocities (risk of soot deposits) through the exhaust gas boiler may have a deteriorating effect on the boiler, Ref. [1], we have, in our studies, selected an exhaust gas boiler designed for a single-pressure steam system of minimum 7 bar abs (6 bar g) steam pressure (165°C), and minimum 20°C pinch point. The superheated steam temperature is about 270°C. The steam turbine is a multistage single-pressure condensing type. The alternator/generator is driven both by the steam turbine and the exhaust gas turbine.

As an example valid for the nominally rated 12K98ME/MC (18.2 bar) engine operating at ISO ambient reference conditions, we have calculated the steam production and the electric power production of TES, see Figs. 10 and 11.

The total electric power production in % of the main engine shaft power output is also shown as a function of the engine load, see Fig. 9.

The results for operation at 85% SMCR are shown in Fig. 14, together with the calculated ISO ambient temperature based results for three other main engine types. The corresponding results based on tropical ambient temperature conditions are shown in Fig. 15. However, it should be emphasised that it is probably more realistic to use the ISO ambient temperatures as the average ambient temperatures in worldwide operation. In Fig. 16, the ISO based total electric power production at 85% SMCR is also shown as a function of the main engine size measured in SMCR power.

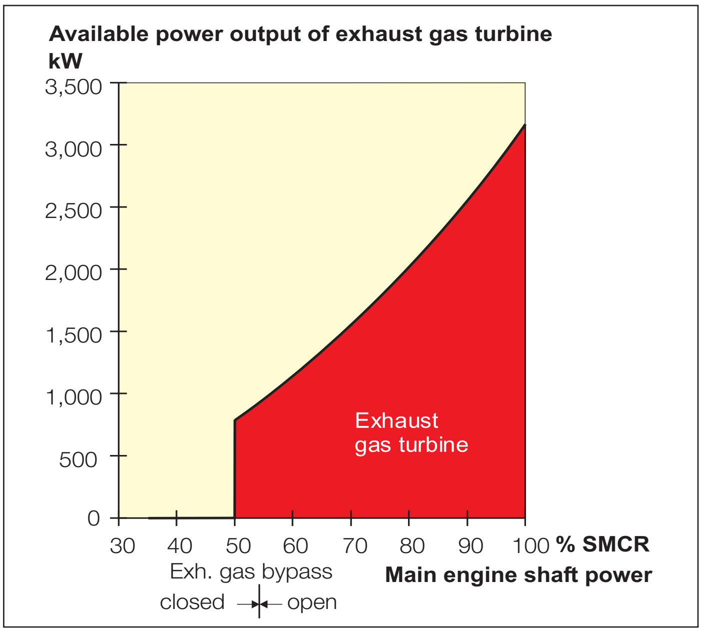

Fig. 8: Expected available power output of exhaust gas turbine for a 12K98ME/MC with SMCR = 68,640 kW x 94 r/min

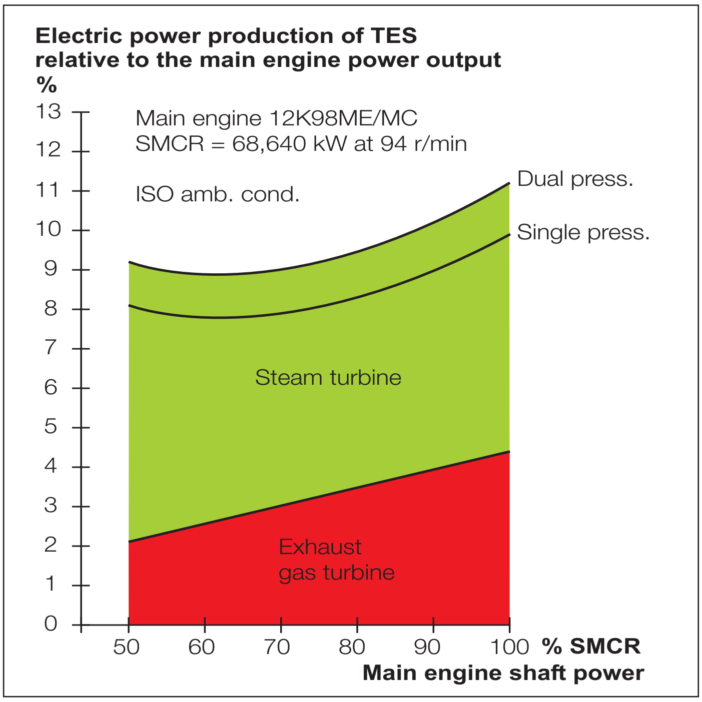

Fig. 9: Expected electric power production in % of the main engine shaft power output valid for a 12K98ME/MC with Thermo Efficiency System (TES) and based on ISO ambient reference conditions

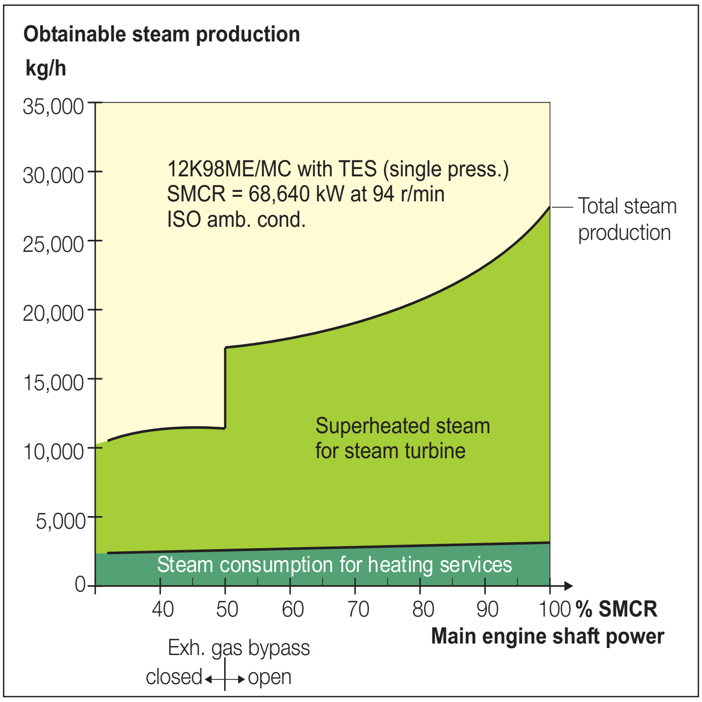

Fig. 10: Expected steam production of an exhaust gas boiler with single-pressure steam system valid for main engine 12K98ME/MC with TES and based on ISO ambient reference conditions

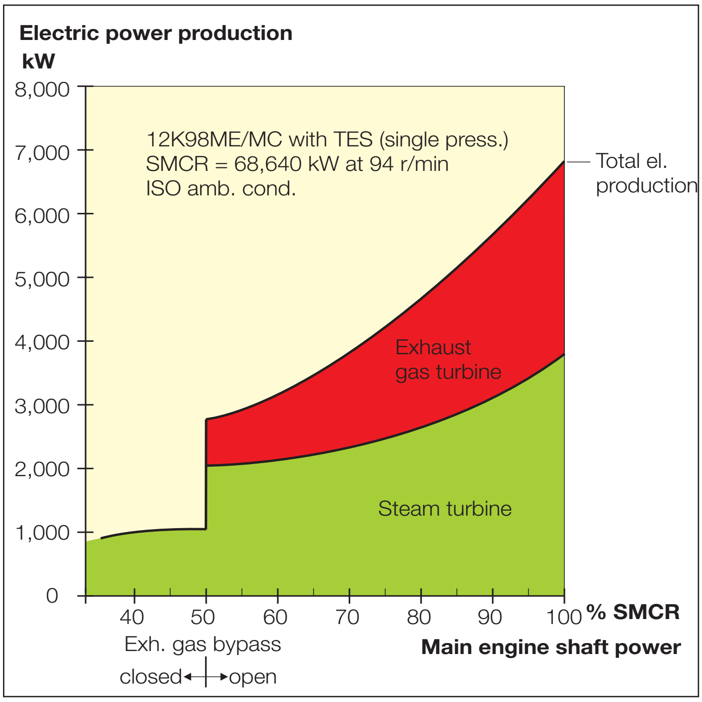

Fig. 11: Expected electric power production of the Thermo Efficiency System (TES) with a single-pressure steam system valid for main engine 12K98ME/MC and based on ISO ambient reference conditions

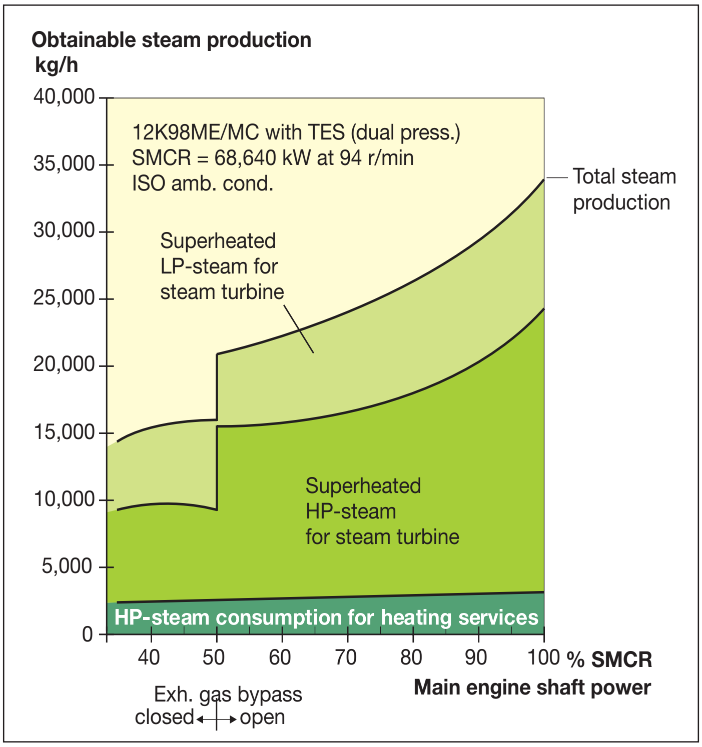

Fig. 12: Expected steam production of an exhaust gas boiler with a dual-pressure steam system valid for main engine 12K98ME/MC with TES and based on ISO ambient reference conditions

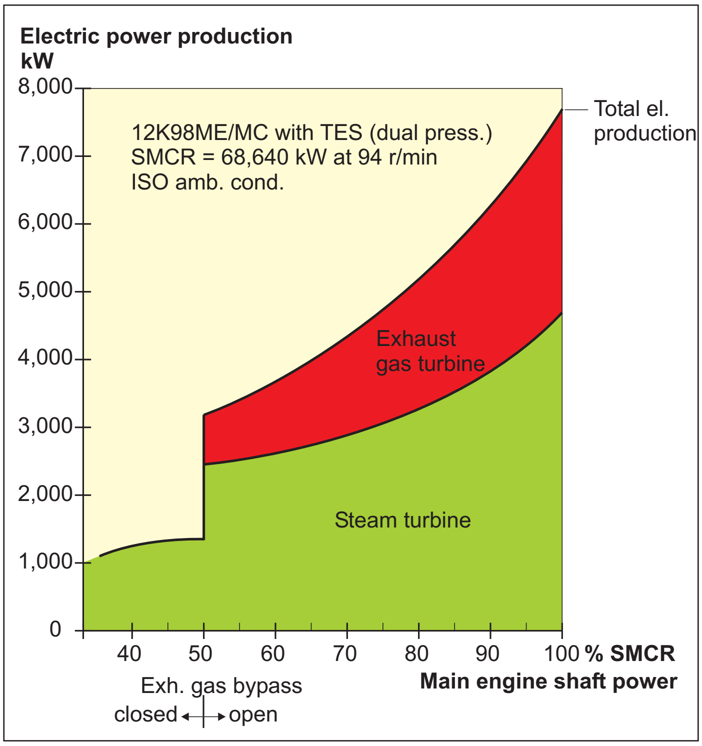

Fig. 13: Expected electric power production of the Thermo Efficiency System (TES) with a dual-pressure steam system valid for main engine 12K98ME/MC and based on ISO ambient reference conditions

Exhaust gas and steam turbine generator output – dual pressure

Besides the single-pressure steam system, a more complex and more expensive dual-pressure steam system is also available, see Figs. 6 and 7. The high and low steam pressures used are about 10-11 and 4-5 bar abs (9-10 and 3-4 bar g), respectively.

The steam turbine is a multi-stage dual-pressure condensing type. The alternator/generator is driven both by the steam turbine and the exhaust gas turbine.

Because of the low steam pressure and corresponding low saturated steam temperature (144°C/4.0 bar abs), there is no room for an LP-preheater section in the exhaust gas boiler for feedwater preheating, because the exhaust gas boiler outlet temperature has to be higher than about 160-165°C in order to avoid sulphuric acid corrosion of the boiler outlet.

The feedwater, therefore, has to be preheated by means of alternative heat sources such as the jacket water and scavenge air cooler heat.

Furthermore, the pinch point should not be too low (giving low gas velocities through the boiler) in order to protect the exhaust gas boiler against soot deposits and fires.

As an example valid for the nominally rated 12K98ME/MC (18.2 bar) engine operating at ISO ambient reference conditions, we have calculated the steam production and the electric power production of the TES, see Figs. 12 and 13.

The total electric power production in % of the main engine shaft power output is also shown as a function of the engine load, see Fig. 9.

The results for operation at 85% SMCR are shown in Fig. 14, together with the calculated ISO ambient temperature based results for three other main engine types. The corresponding results based on tropical ambient temperature conditions are shown in Fig. 15. However, it should be emphasised that it is probably more realistic to use the ISO ambient temperatures as the average ambient temperatures in worldwide operation. In Fig. 16, the ISO based total electric power production at 85% SMCR is also shown as a function of the main engine size measured in SMCR power.

Payback time of the Thermo Efficiency System

The payback time of TES depends very much on the size of the main engine and the trade pattern (main engine load and ambient temperatures) of the ship.

	Main engines operating at 85% SMCR and ISO ambient reference conditions					
ME = Main engine **EGT = Exh. gas turbine**	Ship type Main engine type Specified MCR (L1)	 kW	VLCC 6S90ME-C 29,340	4,500 teu 7K98ME-C 39,970	6,000 teu 12K90ME 54,840	8,000 teu 12K98ME 68,640
ST1 = Steam turbine **Single** steam pressure 7.0 bar abs	Main engine load Main engine power output	% SMCR kW	85 24,939	85 33,975	85 46,614	85 58,344
ST2 = Steam turbine **Dual** steam pressure 4.0 bar abs/10.0 bar abs Additional feed water preheating required	Steam consumption for heating services **EGT** electric power production, approx. in % of ME output	kg/h kW **%**	1,400 920 **3.7**	1,800 1,260 **3.7**	2,400 1,730 **3.7**	3,000 2,180 **3.7**
	ST1 electric power production in % of ME output	kW **%**	1,110 **4.5**	1,640 **4.8**	2,250 **4.8**	2,840 **4.9**
TES1 = EGT+ST1	**ST2** electric power production in % of ME output	kW **%**	1,360 **5.4**	2,020 **5.9**	2,800 **6.0**	3,520 **6.0**
TES2 = EGT+ST2 In normal service at 85% SMCR per year: 280 days	Total **TES1** electric power production in % of ME output Annual fuel savings payback time	kW **%** USD/year year	2,030 **8.2** 374,000 8.8	2,900 **8.5** 528,000 7.0	3,980 **8.5** 724,000 5.8	5,020 **8.6** 917,000 5.0
	Total **TES2** electric power production in % of ME output Annual fuel savings Payback time	kW **%** USD/year year	2,280 **9.1** 415,000 8.8	3,280 **9.6** 596,000 7.0	4,530 **9.7** 818,000 5.8	5,700 **9.8** 1,045,000 5.0

Fig. 14: Steam and electric power production and payback time of the Thermo Efficiency System (TES) when operating at 85% SMCR and ISO ambient reference conditions

	Main engines operating at 85% SMCR and tropical ambient conditions					
ME = Main engine **EGT = Exh. gas turbine**	Ship type Main engine type Specified MCR (L_1)	 kW	VLCC 6S90ME-C 29,340	4,500 teu 7K98ME-C 39,970	6,000 teu 12K90ME 54,840	8,000 teu 12K98ME 68,640
ST1 = Steam turbine **Single** steam pressure 7.0 bar abs	Main engine load Main engine power output	% SMCR kW	85 24,939	85 33,975	85 46,614	85 58,344
ST2 = Steam turbine **Dual** steam pressure 4.0 bar abs/10.0 bar abs Additional feed water preheating required	Steam consumption for heating services **EGT** electric power production, approx. in % of ME output	kg/h kW **%**	900 880 **3.5**	1,200 1,200 **3.5**	1,600 1,650 **3.5**	2,000 2,070 **3.5**
	ST1 electric power production in % of ME output	kW **%**	1,600 **6.4**	2,300 **6.8**	3,150 **6.8**	4,000 **6.9**
TES1 = EGT+ST1 **TES2 = EGT+ST2**	**ST2** electric power production in % of ME output	kW **%**	1,950 **7.8**	2,800 **8.2**	3,830 **8.2**	4,850 **8.3**
	Total **TES1** electric power production in % of ME output	kW **%**	2,480 **9.9**	3,500 **10.3**	4,800 **10.3**	6,070 **10.4**
	Total **TES2** electric power production in % of ME output	kW **%**	2,830 **11.3**	4,000 **11.8**	5,480 **11.8**	6,920 **11.9**

Fig. 15: Steam and electric power production of the Thermo Efficiency System (TES) when operating at 85% SMCR and tropical ambient conditions

When for example operating at tropical ambient conditions, the electric power output of the TES is higher than for ISO ambient conditions, which again has a higher TES output compared with winter ambient conditions.

Furthermore, the investment costs per installed kW_e output of a TES plant are relatively cheaper the bigger the plant is.

A simple estimation of the average saved fuel costs in service at ISO ambient temperature conditions for a 12K98ME/MC with TES (for single or dual pressure) compared to a standard 12K98ME/MC engine can be found by means of the already estimated relative TES1/TES2 gain of 8.6%/9.8% of the main engine output, see Fig. 14.

Based on the average operation in service at 85% SMCR = 58,344 kW in 280 days a year, SFOC = 0.00017 t/kWh and a fuel price of 160 USD/t the annual main engine fuel costs of the standard 12K98ME/MC engine are as follows:

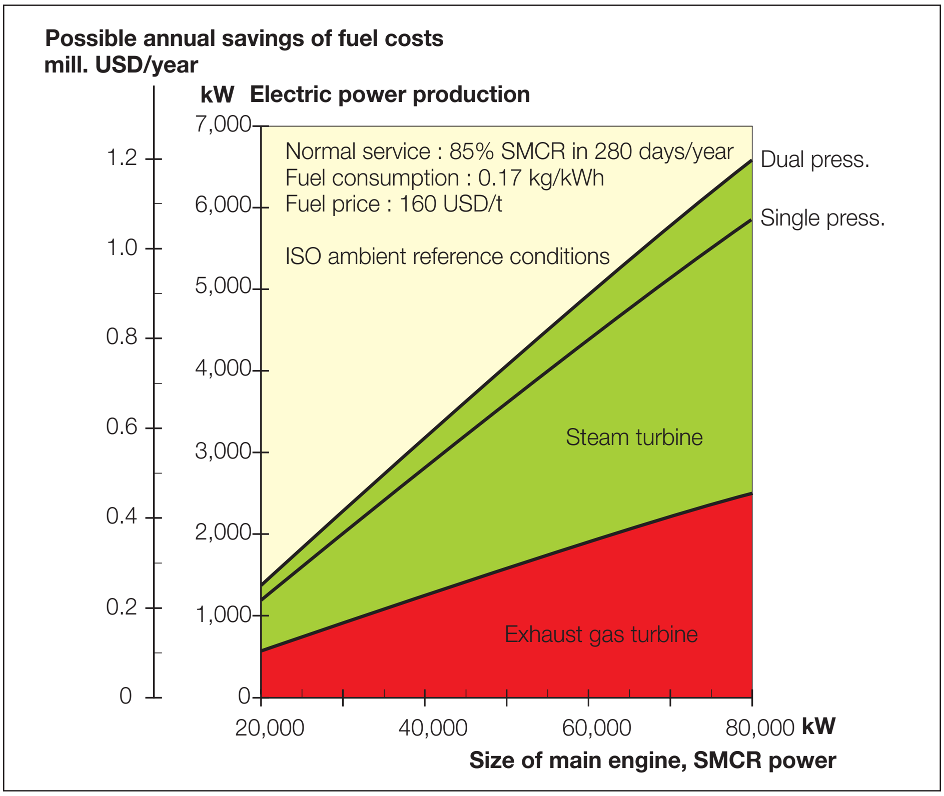

Fig. 16: Expected total electric power production and possible annual fuel cost savings of the Thermo Efficiency System (TES) based on ISO ambient reference conditions and 85% SMCR, shown as a function of the main engine size, SMCR power

Fuel costs = 280 days/y x 24 h/day
x 0.00017 t/kWh
x 58,344 kW
x 160 USD/t
= 10,664,000 USD/year

For the single and dual-pressure systems, respectively, the TES gain in saved fuel consumptions will then be as follows:

TES1 savings = 0.086 x 10,664,000
= 917,000 USD/year
TES2 savings = 0.098 x 10,664,000
= 1,045,000 USD/year

as shown in Fig. 14.

The similar fuel cost savings valid for the other three cases with smaller main engines are also stated in Fig. 14, and in curve form in Fig. 16 as a function of the main engine size, SMCR power.

Based on the extra investment costs of the TES plant (without installation of a shaft power motor on the main engine shaft and minus the investment cost of a normal exhaust gas boiler system) for the four main engine cases compared to a standard main engine installation, the estimated payback time found is as stated in Fig. 14, i.e. in principle the same for the single and dual-pressure systems. For the 12K98ME/MC engine, the calculated payback time is about five years.

In Fig. 17, the estimated payback time of the TES plant is shown as a function of the main engine size. The payback time is only valid provided all electric power production savings are used on board the ship.

It has been assumed that the increased specific fuel consumption of the TES main engine has been avoided by using an increased firing pressure, P_{max}.

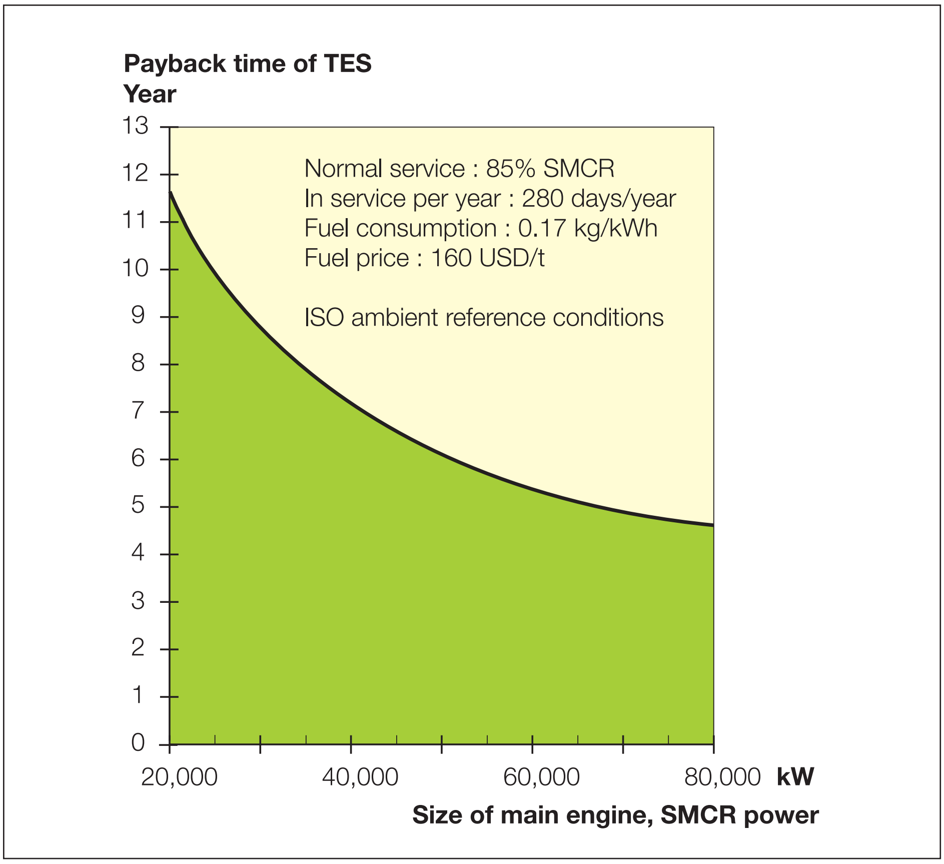

Fig. 17: Estimated payback time of the Thermo Efficiency System (TES) valid for both a single and a dual-pressure steam system

Summary

Our calculations indicate that for ISO ambient reference conditions, a reduction of the fuel consumption of 8-10% for a single-pressure system is possible in the normal service range of the main engine. The larger the engine load, the greater the possible reduction.

For the more complex dual-pressure system, the corresponding reduction is about 9-11%.

All calculations presume unchanged MTBOs (maintenance time between overhauls) compared to today's expectations.

However, if the ship sails frequently in cold weather conditions, the electric power production of the steam turbine will be reduced, and the above-stated reduction of fuel consumption and corresponding reduction of CO_2 emission might not be met.

The extra investment costs of the Thermo Efficiency System and the payback of the investment can be obtained by means of lower fuel/lube oil costs and, not to forget, the possibility of obtaining additional freight charters and higher freight rates, thanks to the 'green ship' image!

The payback time calculations based on a large container vessel with a 12K98ME/MC as main engine and as average operating at 85% SMCR and at ISO ambient reference conditions in normal service during 280 days/year indicate a payback time of about 5 years for the TES. The payback time is valid for both the single-pressure and the dual-pressure systems, because the increased electric power production gain of the dual-pressure system might correspond to about the same relative increase of the investment costs compared to the single-pressure system.

When selecting the type of Thermo Efficiency System, the correspondingly involved higher risks for soot deposits and fires of the exhaust gas boiler has to be considered, see Ref. [1].

Because of this, but also because of the more simple single-pressure steam boiler system, MAN B&W Diesel recommends using the single-pressure steam system, when installing TES.

Of course, the more complex and more expensive dual-pressure steam system, which gives a somewhat higher electric power output, may also be used in connection with MAN B&W Diesel's two-stroke main engine types.

The TES is rather expensive, and relatively more expensive the smaller the main engine and the TES are, giving a relatively higher payback time. Therefore, the installation of the TES is normally only relevant for the large merchant ships, such as the large container vessels.

The TES may be delivered as a package by Peter Brotherhood (turbines) in cooperation with Aalborg Industries (boiler) and Siemens (generator) or by other makers.

References

[1] Soot Deposits and Fires in Exhaust Gas Boilers, MAN B&W Diesel A/S, Copenhagen, Denmark, p. 280, March 2004.