EFFICIENCY INCREASE IN SHIP'S PRIMAL ENERGY SYSTEM USING A MULTISTAGE COMPRESSION WITH INTERCOOLING

by

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This paper focuses on an analysis of the potential increase of efficiency in ship's primal energy system using a turbocharger with multistage compression with intercooling, and diverting a greater flow of exhaust gases to power turbine of waste heat recovery system. Analysis of potential efficiency increase has been made for various stages of compression for a 100% main engine load, and an analysis of five stage compression with intercooling for a main engine load between 50% and 100%.

Key words: *waste heat recovery system, Diesel engine, turbocharger*

Introduction

Turbine technology has been approved and widely adopted on ships. Higher engine efficiency, lower NO_x and CO_2 emissions are among the key drivers of marine Diesel engine development. Higher engine efficiency is yielded by higher turbocharger efficiencies from two-stage turbocharging along with a more optimum division between compression and expansion strokes from the Miller cycle [1].

Saidur *et al.* [2], has identified that there are large potentials for energy savings through the use of waste heat recovery (WHR) technologies. The WHR entails capturing and reusing the waste heat from internal combustion engine and using it for heating or generating mechanical or electrical work. Aly [3] estimated that the heat recovered from Diesel engine exhaust gases can be used to obtain an additional 15-16% increase in the total power output. Shu *et al.* [4] performed a detailed analyse of WHR technologies on ships. These technologies include turbine, refrigeration, Rankine cycle, desalination, and combined cycle systems using more than two of these WHR technologies. Medica *et al.* [5] investigated turbocharger performance and efficiency reduction when the compressor's operating point approaches the surging limit due to heavy seas and in the event of contamination. More than 15% of fuel energy is lost as waste heat in scavenge air coolers. A typical turbocharger compresses air in one stage, after which the air is cooled to about 312 K before engine intake. In this case, excluding the main engine and a turbocharger, a WHR system which produces electricity, then about 10% of exhaust gases, which would usually go to the turbocharger, are diverted to the power turbine of the WHR system, decreasing the pressure of scavenge air [6]. The purpose of this study is to provide an analysis of potential efficiency increase in ship's energy system using a multistage compression with intercooling in a turbocharger, and diverting a greater flow of exhaust gases to the WHR system power turbine. This would leave us with the option of scavenge air pressure optimisation.

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Nowadays, engine manufacturers propose a WHR system that gives an efficiency increase of ship's primal energy system of about 5%. Main parts of WHR system are the power turbine, economiser, and the steam turbine. About 10% of exhaust gases flow is diverted to produce usable mechanical work in WHR system's power turbine which is connected via the gearbox to the steam turbine. Exhaust gases, after the turbocharger and power turbine, enter the economiser where they exchange heat for steam production. Steam flows to the steam turbine where it produces usable mechanical work and converts it to electricity in a generator, [7]. Figure 1 is remade schematic of WHR system used by MAN B&W.

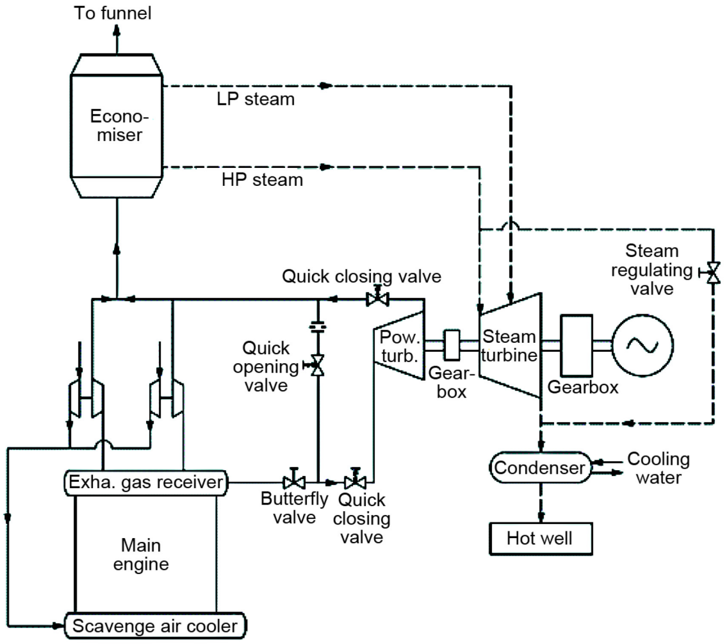

Figure 1. The MAN B&W WHR system

Theory and calculations

If we look at waste heat data, we can observe that air cooler heat amounts to about 15% of fuel energy (fig. 2, tab. 1). This is a result of high scavenge air pressure 3-4 bar at higher main engine loads [7-9].

Exhaust gases flow is divided between a turbocharger and a power turbine of the WHR system, meaning that the ship's overall efficiency is higher when greater flow of exhaust gases is diverted to the power turbine instead of the turbocharger, maintaining the same scavenge air pressure. This is accomplished using a multistage compression with intercooling in a turbocharger. If we take a look at the entire system, and not just its subsystems like a turbocharger, main engine, and a WHR system, then we can see that air is compressed in two stages with intercooling before engine ignition, first stage in a turbocharger compressor, and the second in compression stroke of the main engine. This paper examines a potential efficiency increase of ship's energy system taking into consideration the already produced engine so processes in the engine will not be considered. First stage of compression of the entire system compresses air to a pressure of 4.25 bar, and temperature of 485 K for 100% of the main engine load. Implementing multistage compression with intercooling reduces compression work

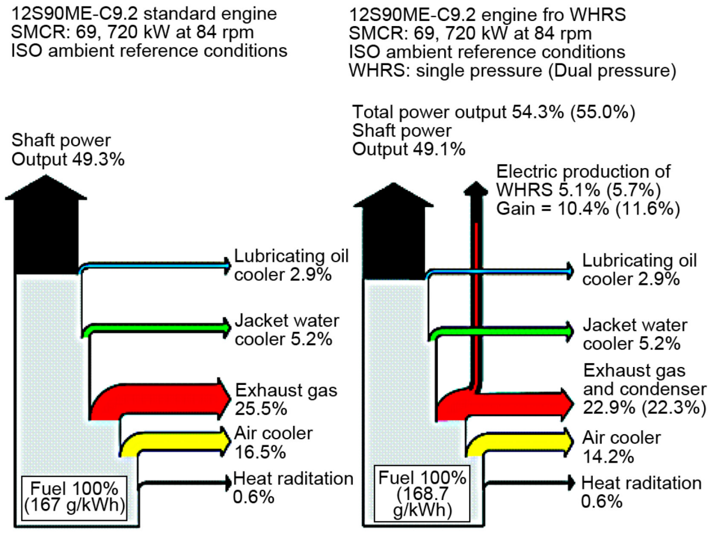

Figure 2. Sankey diagram for MAN B&W engine 12S90ME-C9.2 without and with WHR [7]

resulting in possible exhaust gases flow reduction to turbocharger and exhaust gases flow boost to the power turbine of the WHR system and efficiency increase of the WHR system.

Official data provided by engine manufacturer allows us to calculate the polytrophic compression coefficient. According to official data, electricity produced accounts for 80% and 100% main engine load [9]. This data can be transformed into an efficiency increase of the ship's primal energy system. As we have only those two sets of data, we need to make another assumption that the efficiency increase behaves linearly relative to main engine load. In order to make a model with existing data we use the following assumptions:

– polytrophic coefficient, and with it compressor small stage efficiency, is the same in the used as in the proposed compressor of turbocharger,
– all heat in the compression process is friction heat, there is no heat exchange (adiabatic process) with the surroundings,
– intake air has ISO standard properties,
– air is cooled to 312 K in the intercooler section, because of a sufficient temperature difference between compressed air and cooling water,
– air properties are constant relative to temperature,
– turbocharger turbine efficiency is equivalent to power turbine efficiency so all the work that is saved by multistage compression with intercooling in relation to single stage compression is transferred to an electric generator via power turbine, and
– proposed model works with the same air pressure and temperature at engine air intake similar to the main engine with a typical turbocharger.

Figure 3 represents MAN B&W's WHR system with our concept highlighted in red. Model calculations were made in MATLAB using standard equations from the field of thermodynamics. First model calculations were made for different stages of compression with intercooling for 100% main engine load and then model calculations for a selected number of stages of compression with intercooling variable to main engine load between 50% and 100%. The results are shown using diagrams. Equations used in the model are:

Table 1. The MAN B&W engine data for 14S90ME-C9.2-TII [8]

1	2	3	4	5	6	7	8	9	10	11
100	81,340	84.0	634,500	4.25	212	37	31,150	9,870	5,670	0.0
95	77,273	82.6	613,000	4.05	205	36	29,000	9,510	5,600	0.0
90	73,206	81.1	591,000	3.84	197	34	26,840	9,140	5,510	0.0
85	69,139	79.6	568,200	3.64	188	33	24,670	8,780	5,420	0.0
80	65,072	78.0	544,400	3.43	180	32	22,480	8,410	5,310	0.0
75	61,005	76.3	519,400	3.23	171	31	20,280	8,040	5,200	0.0
70	56,938	74.6	493,200	3.02	161	30	18,070	7,680	5,070	00
65	52,871	72.8	465,500	2.82	151	30	15,870	7,310	4,920	0.0
60	48,804	70.8	436,300	2.61	141	29	13,690	6,940	4,770	0.0
55	44,737	68.8	405,700	2.41	130	28	11,550	6,580	4,600	0.0
50	40,670	66.7	373,600	220	118	27	9,470	6,210	4,410	0.0
45	36,603	64.4	340,100	2.00	106	27	7,520	5,840	4,210	0.0
40	32,536	61.9	305,200	1.82	94	27	5,780	5,480	3,990	0.0
35	28,469	59.2	269,100	1.66	83	26	4,260	5,110	3,740	0.0
30	24,402	56.2	278,000	1.51	72	33	3,570	4,750	3,480	0.0
25	20,335	52.9	234,800	1.38	61	33	2,360	4,380	3,180	0.0
20	16,268	49.1	199,900	1.27	52	32	1,490	4,010	2,850	0.0
15	12,201	44.6	167,600	1.18	44	32	870	3,650	2,460	0.0

ISO condition; Ambient air, 25.0 °C; Scavenge air coolant, 25.0 °C
Loads below 35% are associated with larger tolerances
1 – Engine load [% of SMCR]; 2 – Engine power [kW]; 3 – Engine speed [rpm]; 4 – Scavenge air amount [kg/h] ± 5%; 5 – Scavenge air pressure [bar abs]; 6 – Scavenge air temperature before cooler [°C]; 7 – Scavenge air temperature after cooler [°C]; 8 – Scavenge air cooler heat [kW]; 9 – Jacket water cooler heat [kW]; 10 – Main lubrication oil heat [kW]; 11 – Condensed water [t/24 h]

$$n_{\text{air}} = \frac{1}{1 - \left(\dfrac{\ln T_0 - \ln T_1}{\ln p_0 - \ln p_1} \right)} \tag{1}$$

Equation (1) is derived from a standard relation of pressures, temperatures and a polytrophic coefficient:

$$w_{\text{compr\_r}} = c_p (T_1 - T_0) \tag{2}$$

Equation (2) presents compression work as a difference of enthalpy because there is no heat exchange in the process:

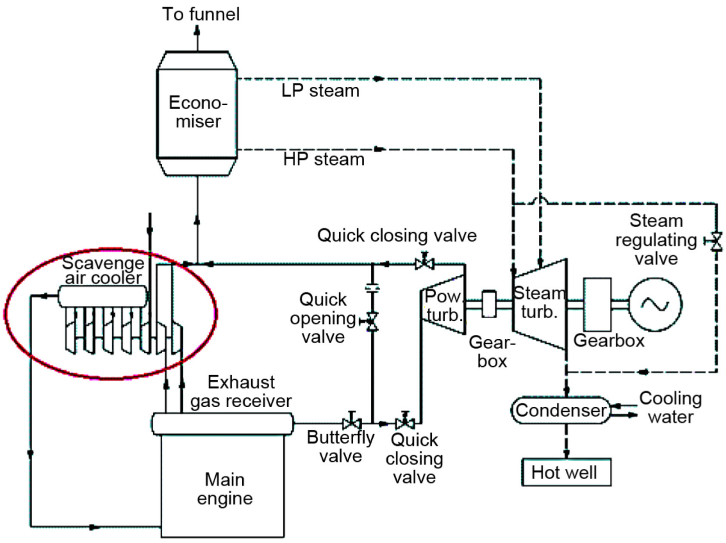

Figure 3. Concept of improved model of ship's primal energy system

$$\varepsilon = \sqrt[z]{\frac{p_1}{p_0}} \tag{3}$$

Equation (3) presents compression ratio of one stage in multistage compression as a z^{th} root (z is the number of stages) of total compression ratio:

$$T_{\text{st.1}} = T_0 \varepsilon^{\frac{n_{\text{air}}-1}{n_{\text{air}}}} \tag{4}$$

$$T_{\text{st.2}} = T_{\text{cool}} \varepsilon^{\frac{n_{\text{air}}-1}{n_{\text{air}}}} \tag{5}$$

Equations (4) and (5) are standard relations of temperatures, one stage compression ratio, and a polytrophic coefficient:

$$w_{\text{compr\_r\_multistage}} = c_p[(T_{\text{st.1}} - T_0) + (z-1)(T_{\text{st.2}} - T_{\text{cool}})] \tag{6}$$

Equation (6) presents compression work as an added difference of enthalpy of all stages of compression, because there is no heat exchange in the process. In the first stage of compression, air temperature is 298 K, but in all other compression stages air has a different temperature at the start of compression:

$$\Delta w_{\text{compr}} = w_{\text{compr\_r}} - w_{\text{compr\_r\_multistage}} \tag{7}$$

Equation (7) presents the assumption that all saved work in a turbocharger compression is transferred to the power turbine:

$$\eta_{\text{increase}} = \frac{\Delta w_{\text{compr}} \eta_{\text{gen}}}{E_{\text{f\&o}}} \tag{8}$$

$$\eta_{\text{overall}} = \eta_{\text{main\_engine}} + \eta_{\text{WHR}} + \eta_{\text{increase}} \tag{9}$$

Equations (8) and (9) present the efficiency increase of the proposed system.

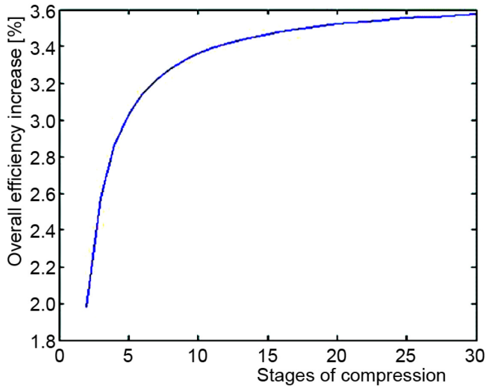

Figure 4. Potential overall efficiency increase in relation to the number of compression stages

Results

Figure 4 shows the ship's primal energy system efficiency increase of 2% for two stage compression, and efficiency increase approaches 3.6% as we increase the number of stages. This data is valid for a 100% main engine load.

It is necessary to determine an optimal number of compression stages, in which investment cost and specific fuel consumption reduction due to overall efficiency increase will be compared. Five stage compression with intercooling is selected for further consideration, for which we can assume to be optimal by the shape of the graph, before analysis. After calculations of WHR efficiency increase and proposed model efficiency increase we get the graph of all models.

Figure 5 shows greater potential overall efficiency increase of the proposed model at higher main engine loads, greatest being at 100% engine load. This is because of higher scavenging air pressure at higher main engine loads.

Figure 6 shows the potential of the proposed model with the remark that this efficiency data is valid if we compress the scavenge air at the same pressure as in a case without the proposed model. Given results are valid with previously specified assumptions and more detailed calculations, based on experimental data, should be made for more precise results.

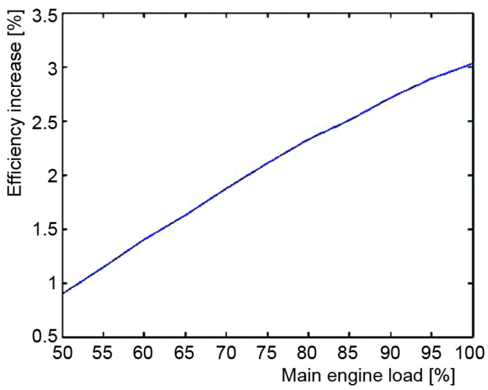

Figure 5. Potential overall efficiency increase relative to main engine load for five-stage compression

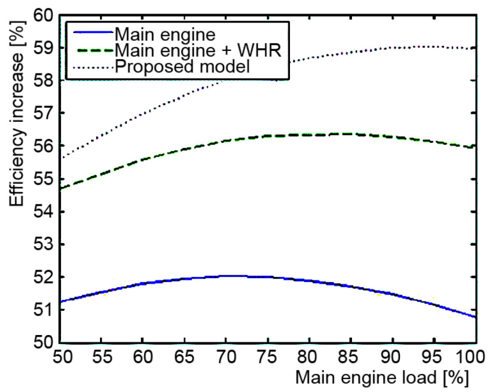

Figure 6. Efficiency of all three models

Discussion

Although present thermal efficiency is very high relative to the past, we see that there still is room for improvement. Efficiency of systems which include internal combustion engines has reached 55%. Most recent addition to this system, the WHR system which uses exhaust gas energy to produce electricity, has improved the efficiency by approximately 5%. The primal system has three subsystems: main engine, turbocharger, and WHR system. The principal point of this paper is that the system should be analysed as a whole, and its efficiency increase, not only of listed subsystems. If we monitor the air from intake through the fuel combustion to the release of exhaust gases, we can see three processes: compression, combustion, and expansion. Compression occurs in two sub-processes: compression in a turbocharger and compression in a compression stroke of the main engine, with intercooling between. Expansion also occurs in two sub-processes: expansion in an expansion stroke of the main engine and expansion in a turbocharger.

Two-stroke engines need pre-compressed air, so an ideal set of processes would look like this: isothermal compression, adiabatic compression, and combustion and adiabatic expansion. Isothermal compression of air is impossible to perform in reality so we need to be content with the second best, and that is multistage compression with intercooling. The number of stages should be determined on the basis of economic analysis. This paper has given a concept of improvement on that basis and elaborated an analysis of potential efficiency increase with previously mentioned assumptions.

The scavenge air pressure is calculated at exact values as available in official data. At main engine load higher than 80%, scavenge air pressure exceeds maximal values, so it must be regulated with a variable exhaust valve opening in main engine. This way work on compression is spent on higher pressures than necessary. This can be avoided by releasing a greater amount of exhaust gases to the WHR system power turbine, thus raising the overall efficiency. A possibility has presented itself with multistage compression and intercooling in a turbocharger, thus giving maximum compression pressure even on partial loads lower than 80%, where a standard turbocharger compresses air to lower pressures than its possible maximum. Additional tests should be conducted in a laboratory to determine what will be the rise of overall efficiency.

This paper has presented a concept for overall energy efficiency increase in a ships primal power system, proved by some standard thermodynamics equations and calculations, but further study is needed to explore the possibilities of this kind of improvement.

Conclusion

Results of this paper show a potential efficiency increase from 2.2% to 4% for 100% main engine load, relative to number of compression stages. It was also shown that efficiency increase rises with higher main engine load on a five stage compression example. At main engine load higher than 80% scavenge air pressure exceeds maximal values so it must be regulated with variable exhaust valve opening. But this way work on compression is unnecessarily spent on higher pressures. This can be avoided by releasing greater part of exhaust gases to the WHR system power turbine, raising the overall efficiency. A possibility should be explored, to compress air to its maximum value before a compression stroke intake on main engine loads where previous systems have not, to see if system efficiency will rise.

Nomenclature

c_p – heat capacity (at constant pressure), [kJkg^{-1}K^{-1}]
$E_{\text{f.\&o.}}$ – fuel and oil energy, [kJkg^{-1}]
n_{air} – polytrophic coefficient of air during compression
p_0 – ambient air pressure, [bar]
p_1 – air pressure before engine intake, [bar]
T_0 – ambient air temperature, [K]
T_1 – air temperature before aftercooler and engine intake, [K]
T_{cool} – air temperature after intercooler, [K]
$T_{\text{st.1}}$ – air temperature after first stage of multistage compression, [K]
$T_{\text{st.2}}$ – air temperature after other stages of multistage compression, [K]

$w_{\text{compr\_r multistage}}$ – real compression work per mass of proposed turbocharger with intercooling, [kJkg^{-1}]
$w_{\text{compr\_r}}$ – real compression work per mass of used turbocharger, [kJkg^{-1}]
Δw_{compr} – saved work in multistage compressor, [kJkg^{-1}]
WHR – waste heat recovery system
z – number of compression stages

Greek symbols

ε – compression ratio of one stage in multistage compression
η_{gen} – electric generator efficiency
η_{increase} – proposed model efficiency rise
$\eta_{\text{main\_engine}}$ – main engine efficiency
η_{overall} – overall system efficiency
η_{WHR} – waste heat recovery system efficiency rise

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