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C O N T E N T S

ENERGY EFFICIENCY AND ENERGY POWER LOSSES OF THE TURBO-GENERATOR STEAM TURBINE FROM LNG CARRIER PROPULSION SYSTEM

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Abstract: *Turbo-generator (TG) steam turbine energy efficiency and energy power losses in a wide range of turbine loads were presented in this analysis. For TG steam turbine was investigated influence of steam specific entropy increment from the real (polytropic) steam expansion on energy power losses and energy efficiency. TG turbine energy power losses, during the all observed loads, were in the range from 646.1 kW to 685.5 kW. The most influenced parameter which defines change in TG turbine energy power losses is steam mass flow change, while for small steam mass flow changes, influence of steam specific entropy increment on steam turbine energy power losses is the most influential. Steam specific entropy incremental change can be used to estimate the change of TG steam turbine energy efficiency. Increase in steam specific entropy increment resulted with a decrease in TG turbine energy efficiency and vice versa. Analyzed steam turbine energy efficiency ranges from 53.84 % to 60.12 %, what is an expected range for low power steam turbines.*

Keywords: *TURBO-GENERATOR, STEAM TURBINE, ENERGY EFFICIENCY, ENERGY POWER LOSSES*

1. Introduction

Marine steam turbine propulsion plants nowadays can be found in a number of LNG carriers [1]. Such steam propulsion plant consists of many constituent components [2] and one of them is turbo-generator (TG) which steam turbine is analyzed in this paper from the aspect of energy.

The analyzed LNG carrier has at disposal two identical turbo-generators which are designed to cover all ship requirements for electrical power. Each TG turbine has identical operating parameters (inlet and outlet temperatures, pressures and mass flows) and for the analysis is selected one of them. Steam turbine for each electric generator comprises of nine Rateau stages. Steam turbines with Rateau stages and their complete analysis can be found in [3]. Many details of the classic and special designs of marine steam turbines and their auxiliary systems are presented in [4] and [5].

The goal of the TG steam turbine analysis was to determine the specific entropy increment increase during steam expansion from the real exploitation for different steam turbine loads. Increase in steam specific entropy increment, usually indicate an increase in system energy power losses (in this analysis system is a TG steam turbine). It was examined the influence of steam specific entropy increment change on TG turbine energy power losses and energy efficiency change, at each observed operating point.

Main characteristics of the LNG carrier in which steam propulsion system is mounted analyzed turbo-generator are presented in Table 1.

Table 1. *Main specifications of the LNG carrier*

Dead weight tonnage	84,812 DWT
Overall length	288 m
Max breadth	44 m
Design draft	9.3 m
Propulsion turbine	Mitsubishi MS40-2 (max. power 29420 kW)
Turbo-generators	2 x Shinko RGA 92-2 (max. power 3850 kW each)

2. Equations for steam turbine energy analysis

2.1. General equations for the energy analysis

Energy analysis is defined by the first law of thermodynamics, which is related to the conservation of energy [6]. Mass and energy balance equations for a standard volume in steady state disregarding potential and kinetic energy can be expressed according to [7] and [8] as:

$$\sum \dot{m}_{\mathrm{IN}} = \sum \dot{m}_{\mathrm{OUT}} \tag{1}$$

$$\dot{Q} - P = \sum \dot{m}_{\mathrm{OUT}} \cdot h_{\mathrm{OUT}} - \sum \dot{m}_{\mathrm{IN}} \cdot h_{\mathrm{IN}} \tag{2}$$

Energy power of a flow for any fluid stream can be calculated according to the equation [9]:

$$\dot{E}_{\mathrm{en}} = \dot{m} \cdot h \tag{3}$$

Energy efficiency may take different forms depending on the type of the system. Usually, energy efficiency can be written as [10]:

$$\eta_{\mathrm{en}} = \frac{\text{Energy output}}{\text{Energy input}} \tag{4}$$

2.2. Turbo-generator turbine energy efficiency and energy power losses

Steam turbine for each turbo-generator drive is condensing type and consists of nine Rateau stages [11]. Schematic view of steam turbine directly connected to an electric generator (the whole set of steam turbine and electric generator is called turbo-generator) is presented in Fig. 1. In Fig. 1 is also presented steam mass flow along with specific enthalpy and specific entropy at the steam turbine inlet and outlet.

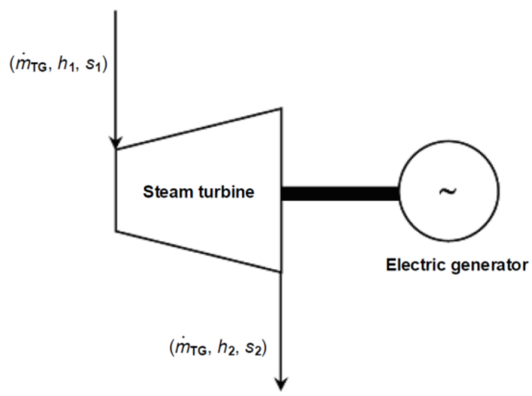

Fig. 1. *Specific enthalpy, specific entropy and steam mass flow through the turbo-generator turbine*

Steam mass flow in relation to the real developed turbine power for TG turbine, regarding the producer specifications, is presented in Fig. 2.

Real TG turbine power calculation at different loads was necessary for the TG turbine correct energy analysis. The turbine real developed power curve was approximated by the third degree polynomial using data from Fig. 2:

$$P_{\mathrm{TG,RE}} = -4.354 \cdot 10^{-10} \cdot \dot{m}_{\mathrm{TG}}^{3} + 6.7683 \cdot 10^{-6} \cdot \dot{m}_{\mathrm{TG}}^{2} + 0.251318 \cdot \dot{m}_{\mathrm{TG}} - 256.863 \tag{5}$$

where $P_{TG,RE}$ was obtained in (kW) when $\dot{m}_{TG}$ in (kg/h) was placed in the equation (5). Steam mass flow through the TG turbine ($\dot{m}_{TG}$) was measured component, while the developed real TG turbine power was calculated according to equation (5).

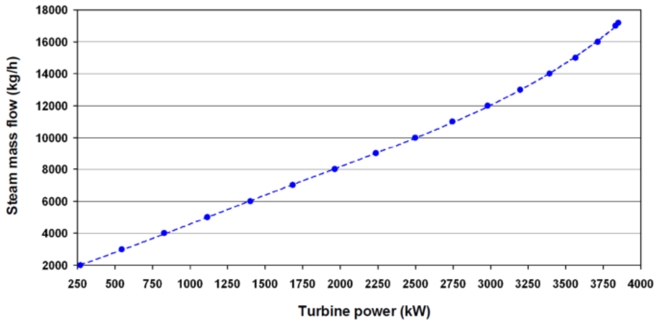

Fig. 2. *Real TG turbine power in relation to the steam mass flow [11]*

During measurements, no steam leakage on the analyzed TG turbine was observed, so the mass balance for the TG steam turbine inlet and outlet is valid as:

$$\dot{m}_{TG,1} = \dot{m}_{TG,2} = \dot{m}_{TG} \quad (6)$$

According to Fig. 1 and Fig. 3, h_1 is steam specific enthalpy at the turbine inlet, and h_2 is steam specific enthalpy at the turbine outlet after real (polytropic) expansion. Steam specific enthalpy at the turbine inlet was calculated from the measured pressure and temperature. Steam specific entropy at the turbine inlet s_1 was also calculated from measured steam pressure and temperature at the turbine inlet. Steam real specific enthalpy at the turbine outlet was calculated from the turbine power $P_{TG,RE}$ in (kW) and measured steam mass flow $\dot{m}_{TG}$ in (kg/s) according to [12] by using an equation:

$$h_2 = h_1 - \frac{P_{TG,RE}}{\dot{m}_{TG}} \quad (7)$$

The steam real specific entropy at the turbine outlet s_2 was calculated from steam real specific enthalpy at the turbine outlet h_2, calculated by using equation (7), and measured pressure at the turbine outlet.

Steam specific enthalpy after isentropic expansion h_{2S} was calculated from the measured steam pressure at the turbine outlet p_2 and from known specific entropy at the turbine inlet s_1. Ideal isentropic expansion assumes no change in steam specific entropy ($s_1 = s_{2S}$), Fig. 3.

Steam specific enthalpy at the turbine inlet, steam specific enthalpy at the end of isentropic expansion and both steam specific entropies (at the turbine inlet and outlet) were calculated by using NIST REFPROP 8.0 software [13].

To proper described TG turbine energy power losses, in any steam turbine operating range, it must be known the real turbine developed power and isentropic power, which can be developed in the ideal situation (when the change in steam specific entropy does not occur). Isentropic steam turbine power, according to Fig. 3, should be calculated as:

$$P_{TG,IS} = \dot{m}_{TG} \cdot (h_1 - h_{2S}) \quad (8)$$

Isentropic steam turbine power will always be higher than the real developed power, because of higher specific enthalpy difference (increment) during the isentropic expansion in comparison to the real polytropic expansion.

Steam turbine (TG turbine) energy power losses can be calculated as:

$$\dot{E}_{TG,en,PL} = P_{TG,IS} - P_{TG,RE} = \dot{m}_{TG} \cdot (h_2 - h_{2S}) \quad (9)$$

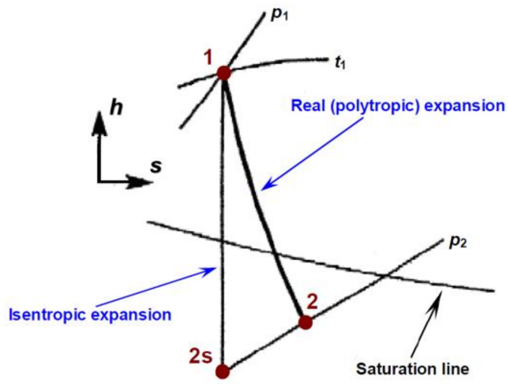

Fig. 3. *Turbo-generator steam turbine real (polytropic) and ideal (isentropic) expansion*

Energy efficiency of TG steam turbine can be calculated according to [14] and [15] by using the following equation:

$$\eta_{TG,en} = \frac{(h_1 - h_2)}{(h_1 - h_{2S})} = \frac{P_{TG,RE}}{P_{TG,IS}} \quad (10)$$

3. Measurement results and measuring equipment of the analyzed TG steam turbine

Measurement results of required operating parameters for TG turbine are presented in relation to the propulsion propeller speed, Table 2. Propulsion propeller speed is directly proportional to steam system load, higher propulsion propeller speeds denote higher steam system loads and vice versa.

Table 2. *Measurement results for TG turbine*

Propulsion propeller speed (rpm)	Steam pressure at the TG turbine inlet (MPa)	Steam temperature at the TG turbine inlet (°C)	Steam pressure at the TG turbine outlet (MPa)	Steam mass flow through TG turbine (kg/h)
25.00	6.21	491.0	0.00541	4648.83
41.78	6.22	491.0	0.00489	4556.16
56.65	5.97	490.5	0.00425	4000.58
65.10	6.07	491.0	0.00392	3838.78
70.37	6.07	502.5	0.00397	3778.91
76.56	6.01	504.5	0.00420	4070.84
80.44	5.89	501.5	0.00554	4689.03
83.00	5.90	493.5	0.00561	4487.93

All the measurement results were obtained from the existing measuring equipment mounted on the TG turbine inlet and outlet. List of all used measuring equipment was presented in the Table 3.

Table 3. *Used measuring equipment for the TG turbine analysis*

Steam temperature (TG inlet)	Greisinger GTF 601-Pt100, Immersion probe [16]
Steam pressure (TG inlet)	Yamatake JTG980A, Pressure Transmitter [17]
Steam pressure (TG outlet)	Yamatake JTD910A, Pressure Transmitter [18]
Steam mass flow (TG inlet)	Yamatake JTD960A, Pressure Transmitter [18]
Propulsion propeller speed	Kyma Shaft Power Meter, (KPM-PFS) [19]

4. TG steam turbine energy analysis results with the discussion

Steam specific entropy difference (increment) between the inlet and outlet of the TG steam turbine is presented in Fig. 4 for all the observed steam system loads. As the steam specific entropy at a TG turbine inlet is almost constant during the all propulsion propeller speeds, specific entropy increment is the most influenced by steam specific entropy at the TG turbine outlet.

From the lowest to the highest observed propulsion propeller speeds, TG turbine steam specific entropy increment (difference) firstly increases from 1.69 kJ/kg·K at 25.00 rpm up to 2.02 kJ/kg·K at propulsion propeller speed of 70.37 rpm, after which decreases to the lowest value of 1.69 kJ/kg·K at 83.00 rpm.

Increase in steam specific entropy increment, usually indicates an increase in system energy power losses, for a large number of different systems [14]. It will be interesting to analyze does the same conclusion is valid for the TG steam turbine.

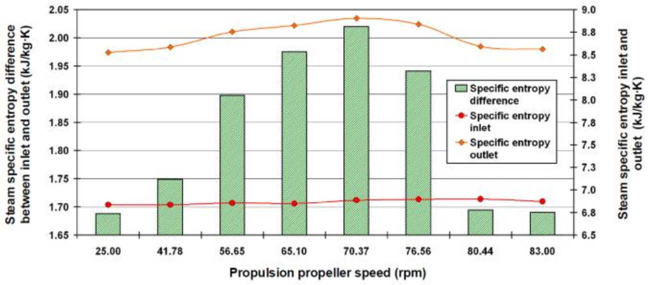

Fig. 4. Steam specific entropy change at the TG turbine inlet and outlet along with specific entropy difference (increment) between inlet and outlet

TG steam turbine isentropic and real power is presented in Fig. 5. The change in both TG turbine power for all observed propulsion propeller speeds must have the same trend. Isentropic TG turbine power can theoretically be developed by using the real operating parameters, but without any losses (without change in steam specific entropy). TG real power is the power developed according to real measured operating parameters in the LNG carrier propulsion system during navigation.

In the whole range of observed steam system loads, isentropic TG turbine power varies from 1423 kW up to 1711 kW, while in the same load range real TG turbine power varies from 766 kW up to 1025.5 kW. The real TG turbine power depends on the current need for electricity and it changes depending on the inclusion or exclusion of the individual electrical consumers.

The difference in isentropic and real TG turbine power represents energy power losses of the real TG steam turbine process in comparison with ideal one.

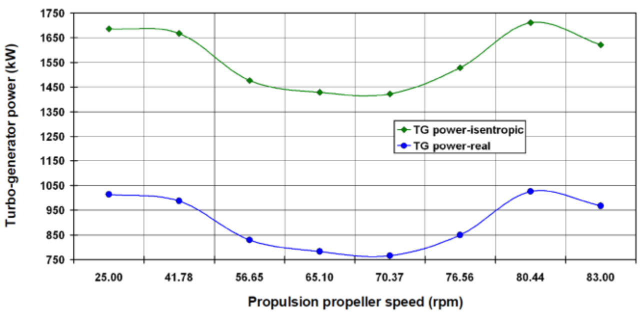

Fig. 5. Change in TG turbine power (real and isentropic) for all observed propulsion propeller speeds

Increase in steam specific entropy increment reduces available steam specific enthalpy difference which will be used in steam turbine. As a result, increase in steam specific entropy increment will cause a decrease in real developed steam turbine power.

From Fig. 4 and Fig. 6 can be seen that the change in TG turbine specific entropy increment does not have the most significant influence on TG steam turbine energy power losses. For example, between propulsion propeller speed of 41.78 rpm and 56.65 rpm, steam specific entropy increment increases, Fig. 4, while for the same propulsion propeller speeds TG turbine energy power loss decreases, Fig. 6. The same occurrence is visible between propulsion propeller speeds of 76.56 rpm and 80.44 rpm when steam specific entropy increment decreases while for the same propulsion propeller speeds TG turbine energy power loss increases.

The most significant influence on TG steam turbine energy power losses has steam mass flow through the turbine. In general, increase in steam mass flow will increase TG turbine energy power losses, while a decrease in steam mass flow will decrease TG turbine energy power losses. TG steam turbine energy power losses, between some observed operating points, during a small change in steam mass flow are also influenced by steam specific entropy increment.

Between propulsion propeller speed of 41.78 rpm and 56.65 rpm, TG turbine energy power losses decrease, Fig. 6, because of a noticeable decrease in steam mass flow from 4556.16 kg/h to 4000.58 kg/h, Table 2. Increase in steam specific entropy increment between these two propulsion propeller speeds does not have significant influence on energy power losses change, Fig. 4.

Also, between propulsion propeller speeds of 76.56 rpm and 80.44 rpm TG turbine energy power losses increases, Fig. 6, because of noticeable increase in steam mass flow from 4070.84 kg/h to 4689.03 kg/h, Table 2, regardless of steam specific entropy increment noticeable decrease, Fig. 4.

On the other side, for a small change in steam mass flow, steam specific entropy increment can have an important influence on TG turbine energy power losses change. When compared propulsion propeller speeds of 65.10 rpm and 70.37 rpm, steam mass flow decreases from 3838.78 kg/h to 3778.91 kg/h, Table 2, but this steam mass flow decrease does not cause a decrease in TG turbine energy power losses, Fig. 6. Between these two propulsion propeller speeds, increase in TG turbine energy power losses occurs because of a notable increase in steam specific entropy increment, Fig. 4.

Final conclusion which can be derived is that the most influenced parameter on TG turbine energy power losses is steam mass flow. For a small change in steam mass flow, steam specific entropy increment takes a leading role in affecting the change of TG turbine energy power losses.

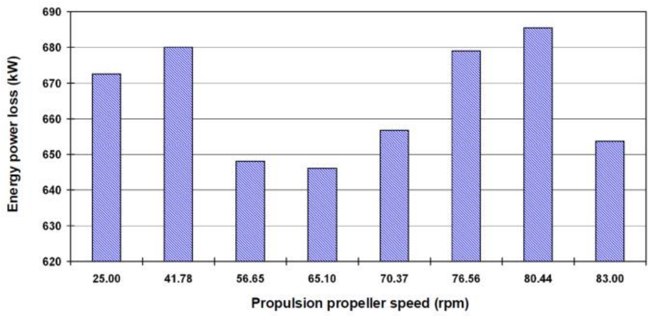

Fig. 6. TG turbine energy power loss change in all observed propulsion propeller speeds

Steam specific entropy increment can be used as an essential parameter for evaluation of TG steam turbine energy efficiency change. During the increase in steam specific entropy increment, TG turbine energy efficiency decreases and during the decrease in steam specific entropy increment, TG turbine energy efficiency increases. This conclusion is valid for every two observed propulsion propeller speeds, during the whole investigated TG turbine load range, Fig. 4 and Fig. 7.

The highest TG turbine energy efficiency of 60.12 % was obtained for the lowest steam specific entropy increment of 1.69

kJ/kg·K at the propulsion propeller speed of 25.00 rpm (the lowest observed TG turbine load), Fig. 7. The lowest TG turbine energy efficiency of 53.84 % was obtained for the highest steam specific entropy increment of 2.02 kJ/kg·K at the propulsion propeller speed of 70.37 rpm.

The analyzed TG steam turbine is a low power steam turbine. Its energy efficiency, for the observed loads, ranges from 53.84 % to 60.12 %, what is an expected range of energy efficiency for low power steam turbine in general [9].

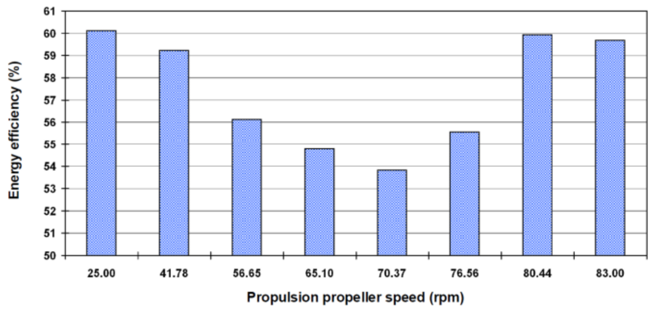

Fig. 7. *TG turbine energy efficiency change in all observed propulsion propeller speeds*

5. Conclusions

This paper presents an analysis of energy efficiency and energy power losses for low power steam turbine, in a wide range of turbine loads. For the analyzed TG steam turbine was investigated influence of steam specific entropy increment from the real (polytropic) expansion process on energy power losses and energy efficiency.

TG steam turbine energy power losses were calculated as a difference between steam turbine real developed power (polytropic steam expansion) and power which can be developed in an ideal situation without any specific entropy increment (isentropic steam expansion). It was found that TG turbine energy power losses, during the all observed loads, were in the range from 646.1 kW to 685.5 kW. Steam specific entropy increment does not have a major influence on TG turbine energy power losses change in general, but for small steam mass flow change, influence of steam specific entropy increment on steam turbine energy power losses is dominant. The most influenced parameter which defines change in TG turbine energy power losses is steam mass flow - increase in steam mass flow caused an increase in TG turbine energy power losses and vice versa.

Steam specific entropy increment change can be used to estimate the change of TG steam turbine energy efficiency. Increase in steam specific entropy increment resulted with a decrease in TG turbine energy efficiency and decrease in steam specific entropy increment resulted with an increase in TG turbine energy efficiency.

NOMENCLATURE

Abbreviations:		**Greek symbols:**	
LNG	Liquefied Natural Gas	η	efficiency, -
TG	Turbo generator		
Latin Symbols:		**Subscripts:**	
$\dot{E}$	stream flow power, kJ/s	en	energy
h	specific enthalpy, kJ/kg	IN	inlet
$\dot{m}$	mass flow, kg/s or kg/h	IS	isentropic (ideal)
p	pressure, MPa	OUT	outlet
P	power, kJ/s	PL	power loss
$\dot{Q}$	heat transfer, kJ/s	RE	real
s	specific entropy, kJ/kg·K		

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