NUMERICAL ANALYSIS OF TURBO-GENERATOR STEAM TURBINE ENERGY EFFICIENCY AND ENERGY POWER LOSSES CHANGE DURING THE VARIATION IN DEVELOPED POWER

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Abstract: *Developed power variation of turbo-generator (TG) steam turbine allows insight into the change of turbine energy efficiency and energy power losses. Measurements were performed in five different TG steam turbine operating points and analysis is presented in three randomly selected operating points. Turbine developed power was varied from 500 kW until the maximum power of 3850 kW in steps of 100 kW. Turbine energy efficiency increases from 500 kW to 2700 kW and maximum energy efficiency was obtained at 70.13 % of maximum turbine power (at 2700 kW) in each operating point. From 2700 kW until the maximum of 3850 kW, TG turbine energy efficiency decreases. Change in TG turbine energy efficiency is caused by an uneven intensity of increase in turbine power and steam mass flow. For all observed operating points, energy efficiency during turbine exploitation is approximately 10 % or more lower than the maximum obtained one. A continuous increase in turbine energy power losses during the developed turbine power increase are the most influenced by the continuous increase in steam mass flow through the turbine.*

KEYWORDS: STEAM TURBINE, ENERGY EFFICIENCY, ENERGY POWER LOSSES, POWER VARIATION

1. Introduction

Steam turbine propulsion plants are not a rarity for a number of LNG carriers [1]. Such steam propulsion plants have many essential components, not only for ship propulsion, but also for electricity and heat production. Each component from the steam propulsion plant can and should be investigated and optimized to achieve the optimal operating parameters. One of the constituent components of such marine steam propulsion plant is turbo-generator (TG) which steam turbine is analyzed in this paper from the energy aspect [2].

Measurements of required TG steam turbine operating parameters were performed on conventional LNG carrier. Every LNG carrier with steam propulsion system has at disposals at least two or more turbo-generator sets which are designed to cover all ship requirements for electrical power.

On the analyzed LNG carrier is mounted two identical TG operating sets. Each TG turbine has identical operating parameters (inlet and outlet temperatures, pressures and mass flows). For the analysis in this paper is selected one TG steam turbine. Steam turbine, which drives an electric generator on the analyzed LNG carrier comprises of nine Rateau stages. Steam turbines with Rateau stages, analysis of their operation and its characteristics is presented in [3]. Usual and specific designs of marine steam turbines along with their auxiliary systems are presented in [4].

The main goal of the TG steam turbine analysis in this paper was to present change in steam turbine energy efficiency and energy power losses during the change in turbine developed power. Measurements of necessary operating parameters were provided in five different turbine operating points, at five different loads. In each turbine operating point was varied turbine developed power from the lowest to the highest one. During the power variation was calculated turbine energy efficiency and energy power losses. The results of the analysis were presented for three selected turbine operating points, but presented conclusions are valid also for all the other operating points. Steam turbine developed power variation allows detecting optimal turbine loads with the highest energy efficiency, for each operating point. It was compared turbine energy efficiency and energy power losses from the real exploitation (measured operating parameters) with achieved optimal operating conditions when the turbine has the highest energy efficiency. TG steam turbine load depends on ship electrical consumers and their current needs for the electrical power. From the aspect of energy efficiency, for the analyzed TG steam turbine will be better to be more loaded to achieve maximal energy efficiency in each operating point.

Main characteristics and specifications of the LNG carrier in which steam propulsion system is mounted analyzed TG steam turbine are presented in Table 1.

Table 1. *Main characteristics of the analyzed 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 29.420 kW)
Turbo-generators	2 x Shinko RGA 92-2 (max. power 3.850 kW each)

2. Low power steam turbine energy analysis

2.1. Steam turbine energy analysis equations

The first law of thermodynamics defined energy analysis of any steam system component [5]. Mass and energy balance equations for a standard volume in steady state disregarding potential and kinetic energy can be expressed according to [6]:

$$\sum \dot{m}_{\mathrm{IN}} = \sum \dot{m}_{\mathrm{OUT}} \quad (1)$$

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

Energy power of a flow for any fluid stream can be calculated according to the [7] by using the equation:

$$\dot{E}_{\mathrm{en}} = \dot{m} \cdot h \quad (3)$$

Energy efficiency may take different forms and types. Usually, energy efficiency can be written as [8]:

$$\eta_{\mathrm{en}} = \frac{\text{Energy output}}{\text{Energy input}} \quad (4)$$

2.2. Energy efficiency and energy power losses for the TG steam turbine

Steam turbine for a turbo-generator drive is a condensing type [9]. Schematic view of steam turbine connected to an electric generator (the whole set of steam turbine and electric generator is usually called turbo-generator) is presented in Fig. 1. Superheated steam mass flow, specific steam enthalpies and specific steam entropies at the TG steam turbine can also be seen in Fig. 1. All variables important for TG turbine numerical analysis were marked with 1 for inlet variables and with 2 for outlet variables.

According to producer specifications [9], TG turbine power can be expressed with the following third degree polynomial:

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

where P_{TG} 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 TG turbine power was calculated according to equation (5).

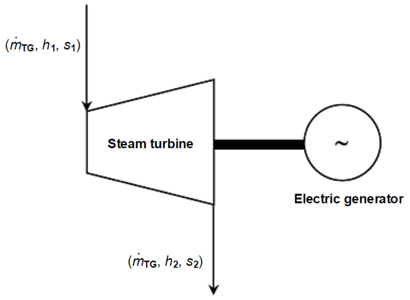

Fig. 1. *Inlet and outlet variables for the TG steam turbine*

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:

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

According to Fig. 1 and Fig. 2, 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 at each operating point. 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} in (kW) and measured steam mass flow $\dot{m}_{TG}$ in (kg/s) according to [10] by using an equation:

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

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

Steam specific enthalpy at the turbine inlet, steam specific enthalpy at the end of isentropic expansion and steam specific entropy at the turbine inlet were calculated by using NIST REFPROP 8.0 software [11].

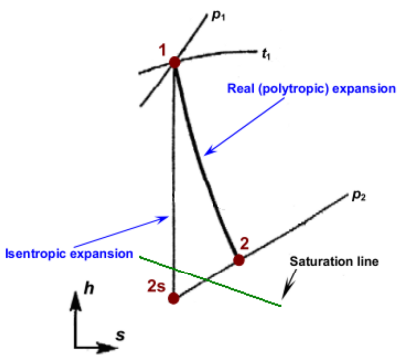

Fig. 2. *TG real (polytropic) and ideal (isentropic) expansion*

TG steam turbine energy power losses in each operating point can be calculated according to Fig. 2 as:

$$\dot{E}_{TG,en,PL} = \dot{m}_{TG} \cdot h_2 - \dot{m}_{TG} \cdot h_{2S} = \dot{m}_{TG} \cdot (h_2 - h_{2S}) \quad (8)$$

Energy efficiency of TG steam turbine can be calculated according to [12] by using the following equation:

$$\eta_{TG,en} = \frac{(h_1 - h_2)}{(h_1 - h_{2S})} \quad (9)$$

2.3. The principle of the TG developed power variation

TG steam turbine real developed power can be calculated according to Fig. 2 by an equation:

$$P_{TG} = \dot{m}_{TG} \cdot (h_1 - h_2) \quad (10)$$

Three different methods can be used for the power change of TG turbine (if it is assumed always the same inlet pressure and temperature and the same outlet pressure):

1) Change in steam mass flow through the TG steam turbine
2) Change in the value of steam specific enthalpy at the steam turbine outlet (h_2)
3) Combination of method 1 and 2

To present the change of TG steam turbine energy efficiency and energy power losses in this paper is selected combined method (method 3) for each operating point.

Turbine developed power was varied from 500 kW up to a maximum of 3850 kW in steps of 100 kW. Power change requires a change in steam mass flow through the turbine, so the adequate steam mass flow for any turbine power was calculated by using the reversed equation (5). In each operating point, steam pressure and temperature at the turbine inlet and steam pressure at the turbine outlet remain identical to the measured data. Steam enthalpy at the turbine outlet (h_2) was calculated for each turbine power and mass flow by using equation (7). Change in steam enthalpy at the turbine outlet (h_2) along with the change of steam mass flow causes the change of TG steam turbine energy efficiency and energy power losses, equations (8) and (9).

3. Measurement results of the analyzed TG

Measurement results for TG steam turbine at different loads are presented in Table 2. Measured operating parameters were: steam pressure at the TG turbine inlet and outlet, steam temperature at the TG turbine inlet and the steam mass flow through TG turbine.

Table 2. *Measurement results for TG steam turbine at several loads*

Operating point	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)
1	5.97	490.5	0.00425	4000.58
2	6.07	491.0	0.00392	3838.78
3	6.07	502.5	0.00397	3778.91
4	6.02	504.0	0.00412	3951.37
5	5.80	493.0	0.00557	4428.43

4. Used measuring equipment

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

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

Steam temperature (TG inlet)	Greisinger GTF 601-Pt100 - Immersion probe [13]
Steam pressure (TG inlet)	Yamatake JTG980A - Pressure Transmitter [14]
Steam pressure (TG outlet)	Yamatake JTD910A - Differential Pressure Transmitter [15]
Steam mass flow (TG inlet)	Yamatake JTD960A - Differential Pressure Transmitter [15]

5. Energy efficiency and energy power losses during TG turbine developed power variation

Change in TG steam turbine energy efficiency and energy power losses during the turbine developed power variation was presented in three operating points from Table 2 – operating points 1, 3 and 5. Obtained conclusions and trends are also valid for the other TG steam turbine operating points.

5.1. Developed power variation for operating point 1

Change in energy efficiency for TG turbine in operating point 1 (Table 2), during the developed power variation is shown in Fig. 3. Increase in turbine developed power causes an increase in energy efficiency until the maximum value, after which follows a decrease in turbine energy efficiency. Maximum turbine energy efficiency is obtained at power of 2700 kW (70.13 % of maximum turbine power) and amounts 67.82 %. At the highest turbine load of 3850 kW, energy efficiency amounts 65.72 % in this operating point.

Turbine energy efficiency in each operating point, as well as in operating point 1, is calculated by using equation (9). For each operating point, energy efficiency change is affected only with the change in steam specific enthalpy after real polytropic expansion (h_2) which is calculated according to equation (7). Steam mass flow through the TG turbine in equation (7) is calculated by using the reversed equation (5) where the turbine power is known, and steam mass flow is an unknown variable. Values of steam specific enthalpy after real polytropic expansion (h_2) decreases in the turbine power range from 500 kW until the 2700 kW, because the intensity of increase in turbine power is higher in comparison with an increase in steam mass flow through the turbine. In the turbine power range from 2700 kW until the highest turbine load of 3850 kW, steam specific enthalpy after real polytropic expansion (h_2) increases because the intensity of increase in turbine power is lower in comparison to an increase in steam mass flow through a turbine in that operating area.

TG steam turbine load depends on ship electrical consumers and their current needs for the electrical power. In operating point 1, TG steam turbine energy efficiency during LNG carrier exploitation amounts only 56.13 % what is much lower energy efficiency than possible maximum one for this operating point. To obtain better energy efficiencies of TG steam turbine in exploitation, it can be recommended that TG turbine should be more loaded, but not more than 2700 kW.

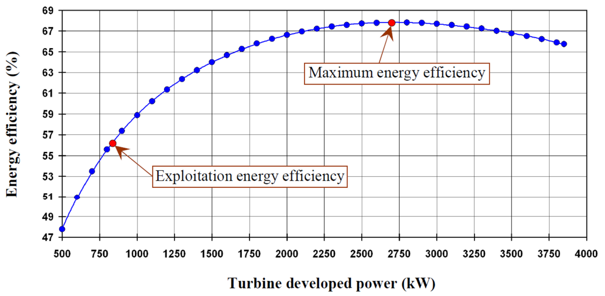

***Fig. 3.** Steam turbine energy efficiency change during the developed power variation for operating point 1*

TG steam turbine energy power loss is calculated by using equation (8) for each observed operating point. Turbine energy power loss is the most influenced by steam mass flow through the turbine. Continuous increase in steam mass flow during the TG turbine power increase from 500 kW to 3850 kW causes a continuous increase in turbine energy power loss, Fig. 4.

During LNG carrier exploitation in operating point 1, TG steam turbine energy power loss amounts 648 kW, while at TG turbine maximum energy efficiency in this operating point (at turbine developed power of 2700 kW) turbine energy power loss amounts 1281.16 kW. At maximum turbine power of 3850 kW, energy power loss is the highest and amounts 2008 kW.

For TG steam turbine is not valid a conclusion that the lowest energy power losses are obtained at the highest energy efficiency. TG steam turbine developed power variation showed that energy power losses are proportional to turbine load - higher load results with the higher energy power losses and vice versa. Energy power losses are not proportional to the energy efficiency of the TG steam turbine.

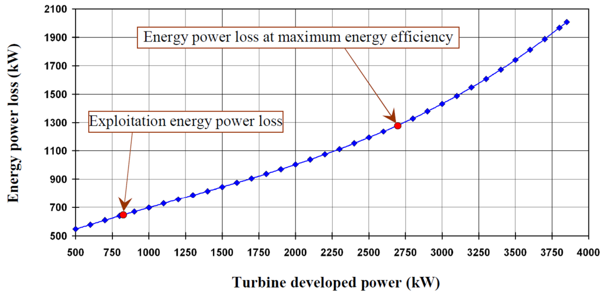

***Fig. 4.** Steam turbine energy power loss change during the developed power variation for operating point 1*

5.2. Developed power variation for operating point 3

Change in energy efficiency for TG turbine in operating point 3 (Table 2), during the developed power variation is shown in Fig. 5. As in observed operating point 1, an increase in turbine developed power causes an increase in energy efficiency until the maximum value, after which follows decrease in turbine energy efficiency.

In operating point 3, maximum energy efficiency is obtained also at turbine developed power of 2700 kW and amounts 66.50 %. For this operating point, at the highest turbine load of 3850 kW, energy efficiency amounts 64.44 %, while during LNG carrier exploitation turbine energy efficiency amounts only 53.84 %. The reasons for such TG turbine energy efficiency change are identical as in operating point 1 described earlier.

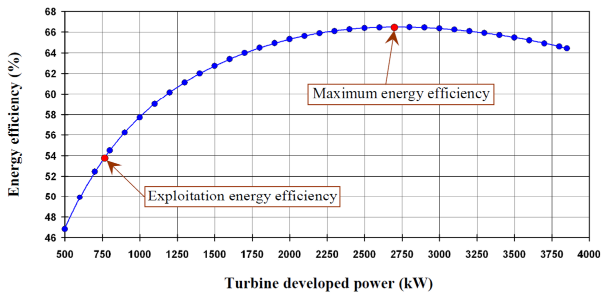

***Fig. 5.** Steam turbine energy efficiency change during the developed power variation for operating point 3*

Continuous increase in steam mass flow during the TG turbine power increase from 500 kW to 3850 kW causes a continuous increase in turbine energy power loss, as presented in Fig. 6, also in TG turbine operating point 3.

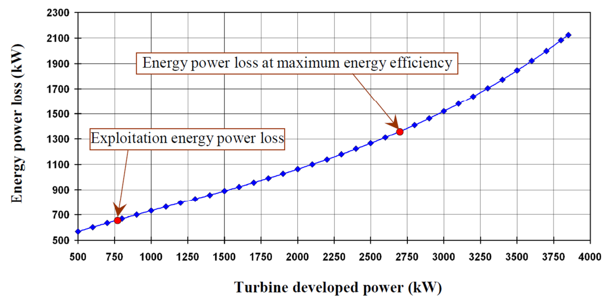

***Fig. 6.** Steam turbine energy power loss change during the developed power variation for operating point 3*

TG steam turbine energy power loss during LNG carrier exploitation in operating point 3 amounts 656.86 kW. At maximum energy efficiency (2700 kW) turbine energy power loss amounts 1360.24 kW, while at maximum turbine power of 3850 kW, energy power loss is the highest and amounts 2124.40 kW in this turbine operating point. As in TG turbine operating point 1, energy power losses are proportional to turbine load - higher load results with the higher energy power losses and vice versa.

5.3. Developed power variation for operating point 5

The same trends and conclusions obtained from TG steam turbine operating points 1 and 3 are also valid for operating point 5 (Table 2). In operating point 5 maximum turbine energy efficiency amounts 69.37 % and as before, is obtained at turbine developed power of 2700 kW. At the highest turbine load (3850 kW) in this operating point energy efficiency is 67.22 %, while during LNG carrier exploitation TG turbine energy efficiency is 59.50 %, Fig. 7. TG turbine operating point 5 also confirmed conclusion that energy power losses are proportional to turbine load - higher load results with the higher energy power losses and vice versa, Fig. 8.

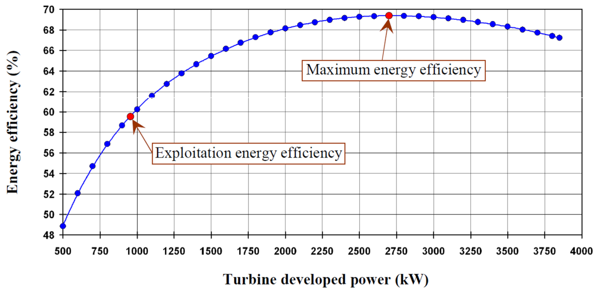

Fig. 7. *Steam turbine energy efficiency change during the developed power variation for operating point 5*

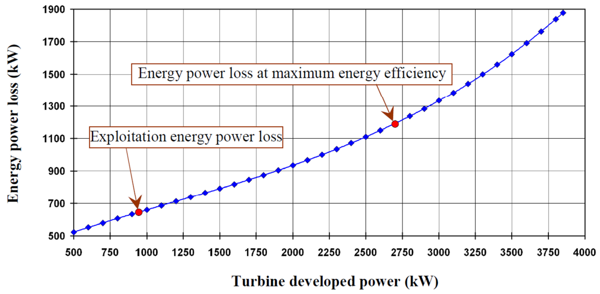

Fig. 8. *Steam turbine energy power loss change during the developed power variation for operating point 5*

6. Conclusions

The paper presents numerical analysis of TG steam turbine energy efficiency and energy power losses change during the variation in turbine developed power. Measurements were performed in five different TG steam turbine operating points and analysis is presented in three randomly selected operating points, but major conclusions are valid in each of them.

Analyzed TG steam turbine energy efficiency increases from 500 kW to 2700 kW of developed power and maximum energy efficiency was obtained at 70.13 % of maximum turbine power (at 2700 kW) in each operating point. From 2700 kW until the maximum of 3850 kW, TG turbine energy efficiency decreases. Increase and decrease in TG turbine energy efficiency is caused by an uneven intensity of increase in turbine power and steam mass flow. For all observed operating points, turbine energy efficiency in LNG carrier exploitation is approximately 10 % or more lower than the maximum obtained ones.

TG steam turbine energy power losses are proportional to turbine load - higher load results with the higher energy power losses and vice versa. The main reason for continuous increase in turbine energy power losses during the developed power increase are found in continuous increase in steam mass flow through the turbine.

7. Acknowledgment

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NOMENCLATURE

Abbreviations:

LNG	Liquefied Natural Gas
TG	Turbo-generator

Latin Symbols:

$\dot{E}$	stream flow power, kJ/s
h	specific enthalpy, kJ/kg
$\dot{m}$	mass flow rate, kg/s
p	pressure, MPa
P	power, kJ/s
$\dot{Q}$	heat transfer, kJ/s
s	specific entropy, kJ/kg·K

Greek symbols:

η	efficiency, -

Subscripts:

en	energy
IN	inlet
OUT	outlet
PL	power loss

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