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ASSESSMENT OF MARINE ENGINES TORQUE LOAD WITHOUT USING OF THE TORQUEMETER

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Abstract

One of the aspects of marine engine operation is defining its torque load. It is crucial for both the general estimation of the engine working parameters value reflecting the engine's operational condition and for comparing its current condition with the previous one (recorded during the last check of the engine performance) or its state during the trials at the engine test bed. In the paper there have been analyzed the well known and recommended by marine engine manufacturers methods of estimating marine engine torque load without any torquemeter used, based exclusively on measurable working parameters of marine engines. To most marine engines the above mentioned indirect methods apply since so far the systems of taking torque direct measure have been neither common nor standard engine room equipment. Methods based on parameters obtained during engine indicating (engine performance check), fuel rack setpoint or load indicator reading and turbocharger system working parameters have been presented. Example nomograms for engine torque load estimation for engines manufactured by famous makers have been shown. Some of the methods have been compared based on parameters obtained during engine performance check. Discussion on the advantages and disadvantages presented methods has been presented.

Keywords: *marine engine, torque load assessment, effective power, torquemeter*, indirect estimation

1. Introduction

Methods of evaluating engine effective power can be divided into direct (making use of brakes or simultaneous measure of torque and revolutionary speed) and indirect ones (approximate). Brakes are used only during the manufacturer's engine trials whereas the torque to be measured during the engine operation needs additional devices which so far have not been comprised by the standard engine room equipment. That is why engine room operators often can make use only of approximate methods.

Accessible literature presents various approximate methods of engine torque load, like in [12, 13, 14]. No synthetic analysis containing applied methods comparison has been found in the literature. There are numerous publications presenting the subject in a general way; e.g. [1, 2, 3, 5, 7, 8, 9, 10]. On the one hand there has not existed any review material summarizing the topic and on the other hand the usefulness of the engine torque load methods for the engine room operators is significant which has made the author of the paper present the most common approximate methods of estimating power.

Basic methods of engine torque load assessment without the use of torquemeters (devices using the phenomenon of intermediate shaft torsion in order to measure the engine torque) have been methods making use of energetic processes analysis results (engine indicating), reading fuel rack setpoint (load indicator) and using parameters measure of engine supercharging system. Possible calculation of effective power based on the measure of active generator power regarding marine generator sets here is not taken into account because the method is endangered to significant error connected with the lack of information concerning the alternator's efficiency (greatly affected by the loads). Furthermore, in the paper a short characteristic of each of the methods together with selected examples have been presented.

2. Load estimation based on engine indicating results

One of more frequently applied methods of defining engine effective power P_e without the use of torquemeter is evaluation based on the results of engine indicating (determined indicated power P_i) [3, 4, 7, 11]. There is a relation between effective and indicated power:

$$P_e = P_i - P_r \text{ [W]}, \tag{1}$$

where:

P_r – the stream of energy used to overcome friction resistance and for suspended mechanisms drive/propulsion [W].

Indicated power is the power output of the engine working spaces (of all particular cylinders) in certain established environment. For one cylinder it is:

$$P_{1i} = C_1 p_i n \text{ [W]}, \tag{2}$$

where:

C_1 – the cylinder constant taking into consideration the piston area, its stroke and number of ignitions assigned for one crank shaft revolution [m^3],
P_i – mean indicated pressure [Pa],
n – speed of the crank shaft [s^{-1}].

The indicated power of a k-cylinder engine is calculated as a sum of particular cylinders indicated power. Using the results based on indicating values of mean indicated pressures *pi* or particular cylinders indicating power it is possible to determine mean effective pressure p_e or effective power P_e according to the relationships:

$$p_e = \eta_m p_i \text{ [Pa]}, \tag{3}$$
$$P_e = \eta_m P_i \text{ [W]}, \tag{4}$$

where:

η_m – engine mechanical efficiency [-].

Having the results of indicating the engine and possessing the knowledge of engine mechanical efficiency it is possible to evaluate the effective power of the engine. It is so much the simpler that nowadays vessels are equipped with modern marine engine control indicating systems. The systems make use of computer programs supporting working out, archiving, visualization and comparing the results of indicating with the standard (reference) state .

They often make other functions accessible which with the knowledge of the engine mechanical efficiency run in the function of load indicator or the engine rotating speed allow to estimate the engine effective power. For example the electronic cylinder pressure measuring system *Premet* [15] allows estimating the load with the use of formerly introduced data concerning the engine mechanical efficiency plot in the function of the engine load indicator (Fig. 1).

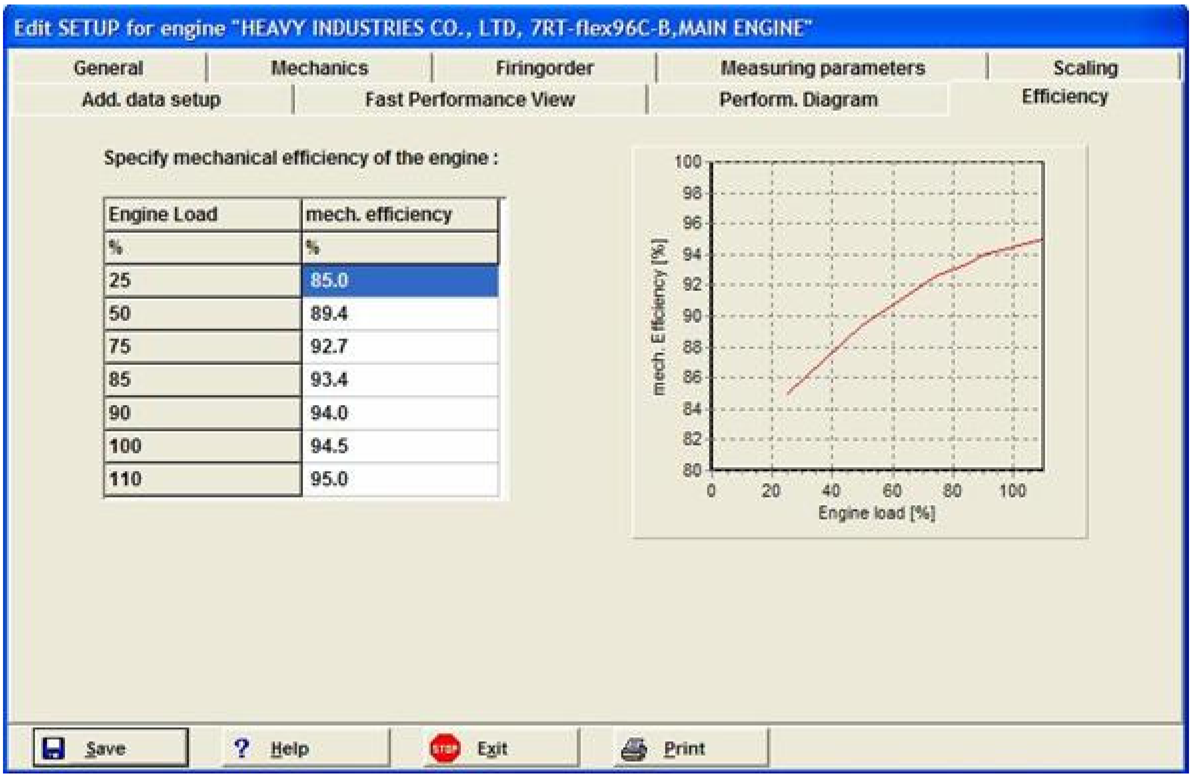

Fig. 1.Bookmark of defining the engine mechanical efficiency plot in the function of load indicator in the setup window of the computer program operating Premet system

Knowing the mean indicated pressure and the pressure of engine friction loss it is possible to assess the mean effective pressure. Assuming the mean pressure of engine friction loss C_2 as equal to 10^5 Pa (1 bar) [12, 13] in a low speed two stroke engine based on an operational experience of *MAN B&W* the mean effective pressure can be determined in relation to:

$$p_e = p_i - C_2 = p_i - 10^5 \text{ [Pa]}. \tag{5}$$

Due to the estimation approximate torque value T and effective power P_e can be calculated in relation to the following:

$$T = C_1 p_e \text{ [Nm]}, \tag{6}$$

$$P_e = C_1 p_e n \text{ [W]}. \tag{7}$$

In order to assess the possibilities of using indirect methods of defining the engine torque load it is necessary to compare the estimated values with the exact measures obtained by means of the torquemeter. This is possible only on ships with proper measuring devices. Table 1 presents the examples of power assessments with reference to (5) and (7) compared with the power estimated on the basis of the engine rotating speed and torque measured with a torquemeter. The comparison was carried out due to the results obtained from the 6 consecutive *MAN B&W 7K80 MC-C* engine performance checks. Particular data series have been arranged according to the growing value of the average (from all cylinders) mean indicating pressure.

Tab. 1. Comparison of MAN B&W 7K80 MC-C engine effective power estimations obtained by means of two selected methods

Average mean indicator pressure p_i [MPa]	Average mean effective pressure based on formula (5) p_e [MPa]	Engine rotating speed n [rpm]	Effective power determined due to the use of a torquemeter P_{e1} [kW]	Effective power determined on the basis of formulas (5) and (7) P_{e2} [kW]	$\left\lvert\frac{P_{e1}-P_{e2}}{P_{e2}}\right\rvert 100$ [%]
1,57	1,47	102,1	20842	20243	2,87
1,61	1,51	101,0	20800	20,569	1,11
1,62	1,52	101,8	20100	20,870	3,83
1,64	1,54	102,0	21416	21186	1,07
1,69	1,59	103,0	22264	22088	0,79
1,72	1,62	102,1	22313	22308	0,02

The above presented results of the comparison prove high exactness of the indirect method based on (5). The percentage of the results discrepancies in this case does not exceed 4%. To a great extend such result is influenced by the torque and rotating speed exact measure as well as up-to-date precise system for carrying out engine indicating. High coherence between assessment results and the results obtained by means of the torquemeter probably refers to a certain range of engine loads close to the rated load. The presented measures were achieved in the conditions in which engine performance should be checked, that is adequately high torque load (70-80%). Error increase of effective power estimation for working at partial load does not diminish the possibility of the method application since it is usually applied during regular operation (operational load), not during a ship's maneuvering.

3. Load assessment based on the fuel rack setpoint

One of the groups of marine engines features are speed characteristics at the same/constant fuel dose. For the engine work the most important characteristic is the rated power characteristic which is done for the constant fuel dose setting corresponding with the rated power and rotary speed. Engine effective efficiency in the tested range of rotary speeds is often assumed to be constant (so called theoretical characteristic). The example of such characteristics presenting power and torque in the function of rotary speed has been shown in Fig. 2.

Change of power for certain rotary speed of an engine is possible only due to a fuel dose change [7, 10].Engine effective power at its constant effective efficiency assumed can be presented as follows:

$$P = C_3 g_w n \,[\mathrm{W}], \tag{8}$$

where:

$C_3 = zW\eta_e$ – constant value [J/kg],
z – number of ignitions inside the cylinder per one crank shaft revolution [-],
W – lower heating value of the fuel [J/kg],
η_e – rated effective efficiency [-],
g_w – fuel dose per one working cycle [kg],
n – engine rotary speed [s^{-1}].

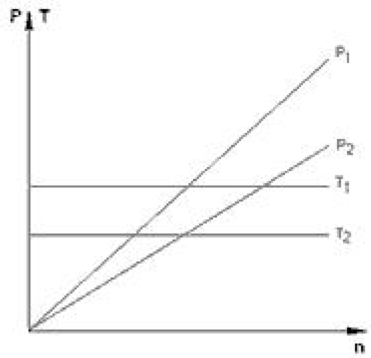

Fig. 2. An example of theoretical characteristics at the constant fuel dose in torque T and effective power P system in the function of rotary speed n for two various fuel doses $g_{w1}>g_{w2}$

Actually the engine effective efficiency is not constant so the power *P*, torque *T* and other working parameters of the engine depend not only upon the fuel dose but also upon the efficiency of energy conversion [5, 7, 10]. Effective efficiency is not constant and varies within the whole range of the engine operational rotary speeds. In Fig. 3 an example of actual characteristics of outside power and torque in the function of rotary speed have been presented.

Taking into account variable value of effective efficiency formula (8) has the following shape:

$$P = C_4 \eta_{e2} g_w n \,[\mathrm{W}], \tag{9}$$

where:

η_{e2} – effective efficiency (affected by rotary speed and the load) [-],

$C_4 = \dfrac{C_3}{\eta_e}$ – constant value[J/kg].

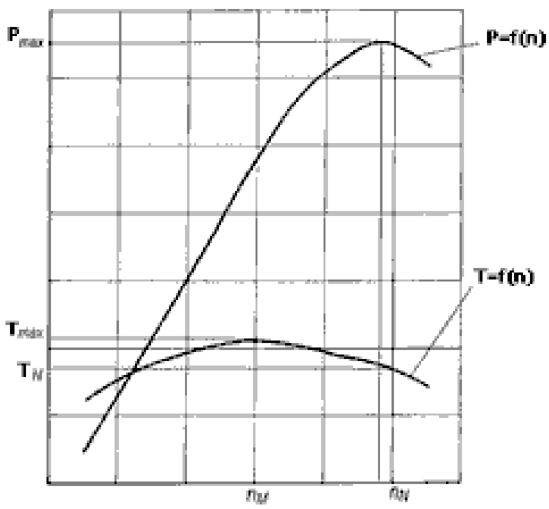

Fig . 3. Real characteristic of a constant fuel dose in power and torque system in the function of the engine rotary speed

In a considerable range of rotary speeds characteristics may be considered to be linear; thus the assumption of the engine constant efficiency value and the use of information concerning the fuel dose value is usually to be read in relative units as the fuel rack setpoints read from the load indicator or by means of each pump reading the positions of fuel racks in millimeters a defining the mean value. In case of the main propulsion engines the reading may cause problems in bad hydro-meteorological conditions.

The fuel pump capacity h is usually read from the indicator fixed at the terminal shaft of governor or fuel pumps actuator, load indicator on the governor or it may be a remotely reading system with the use of selsyns from the indicator located in the Engine Control Room. Indications of the load indicator are usually given in percentage of the maximum fuel dose value of the fuel settings (scale 0-100%) or relative units in respect to maximum fuel dose (scale 1-10). When comparing various states of engine loads it is recommended to always apply the same way of engine load evaluation.

The torque developed by the engine depends upon fuel dose (9). In order to relate current pump settings to the setting values obtained during the trials at the engine test bench, correction of the current load indicator value to the values obtained during the trials at the engine test bench h_{ham} is based on the formula [12]:

$$h_{ham} = \frac{h\,W_{ham}}{W}\,[\%], \tag{10}$$

where :

h – current load grade [%],
W – lower heating value of the currently used fuel [J/kg],
W_{ham} – lower heating value of the fuel used during trials at the engine test bench [J/kg].

Correction of the current load indicator value may also include current fuel density ρ [kg/m^3] and the density of the fuel used during trials at the engine test bench ρ_{ham} [kg/m^3] – then the shape of the relationship is the following:

$$h_{ham} = h \cdot \frac{\rho_{ham}\,W_{ham}}{\rho\,W}\,[\%]. \tag{11}$$

Marine engine manufacturers provide engine test bench trial reports as a part of documentation also containing fuel grades of the fuel used during delivery trials which makes corrections of current load indication to the environment during the trials possible. The obtained value of the corrected load indication allows for the assessment of the engine current mechanical load.. The well known marine engine manufacturers *Wartsila* and *MAN B&W* provide their customers with nomograms allowing for approximate assessment of the power developed by the engine. Fig 4 presents an example of such a nomograms provided by *MAN B&W* allowing for approximate assessment of mechanical load based on fuel pump setting [13].

For a corrected value of load indicator the point of the crossing defined by the load indication value and the engine test bench trials curve is read. Next we read the estimated value of the average mean effective pressure which is directly proportional to the engine torque. Then from the bottom nomogram it is read the value of the effective power specified by values of the determined average mean effective pressure and the line representing current rotary speed of the engine. On the nomograms power is expressed in brake horse power (BHP), engine speed in revolutions per minute (RPM) and mean effective pressure in bar since these are units commonly used in marine engines operating practice.

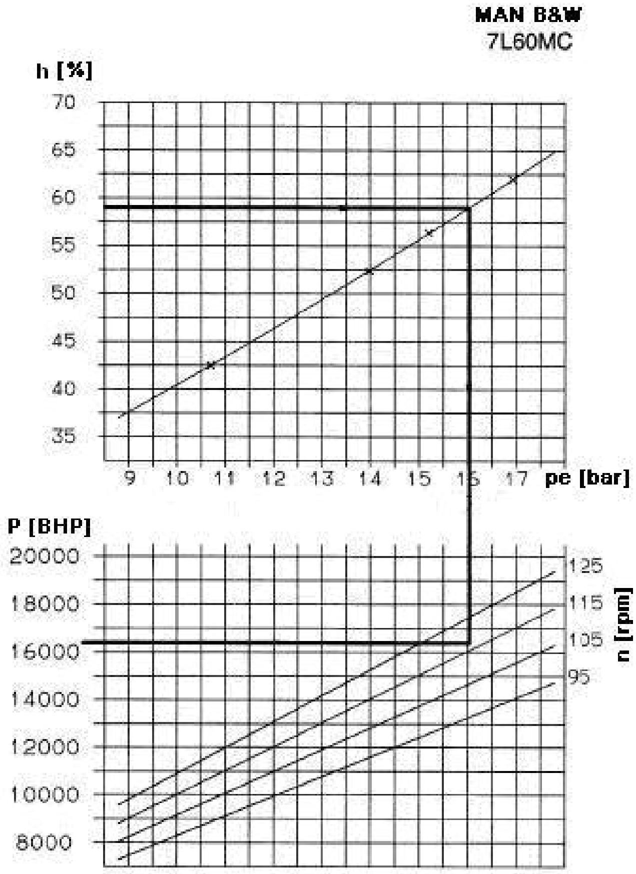

Fig. 4. The way of effective power Pe estimaton based on the load indicator value L and the rotary speed of n of MAN B&W 7L60MC engine [13]

Actually, the load indication value is affected not only by the fuel specific energy but also by other fuel properties. Lower viscosity grade shall lead to bigger leaks of fuel pumps and this shall call for increased fuel dose in order to inject the same volume of fuel. Apart from that the value of fuel dose setting shall be affected by all factors varying fuel consumption like: outside conditions, maximum combustion pressure [9,13, 14]. Thus, the varying conditions during the load assessment process in relation to the conditions at the engine test bench should be taken into account.

Wartsila company suggests a different way of assessing the engine effective power. The power value corresponds to the indication which is a product of load indicator *h* (directly proportional to the fuel dose g_w) and the rotary speed according to (9). Fig. 5 [14] shows an example of *Wartsila 7RT-flex 97C-B* engine diagram. Power here is also expressed in BHP and engine speed in RPM.

The comparison of approximate methods is relatively difficult due to the lack of particular parameter values for the same load states, especially when there is no exact information about actual torque load of the engine (specified by means of a torquemeter). However, the comparison of power estimations with the use of indirect methods for various fuel rack setpoints allows evaluating the tendency of changes in value differences obtained by means of various methods and the value of the differences. Table 2 presents the comparison of power estimations for which the relationship (4) and the method of nomogram presented in Fig. 5 have been used. The comparisons have been done for *Hyundai 7 RT-flex 96C-B* type of engine.

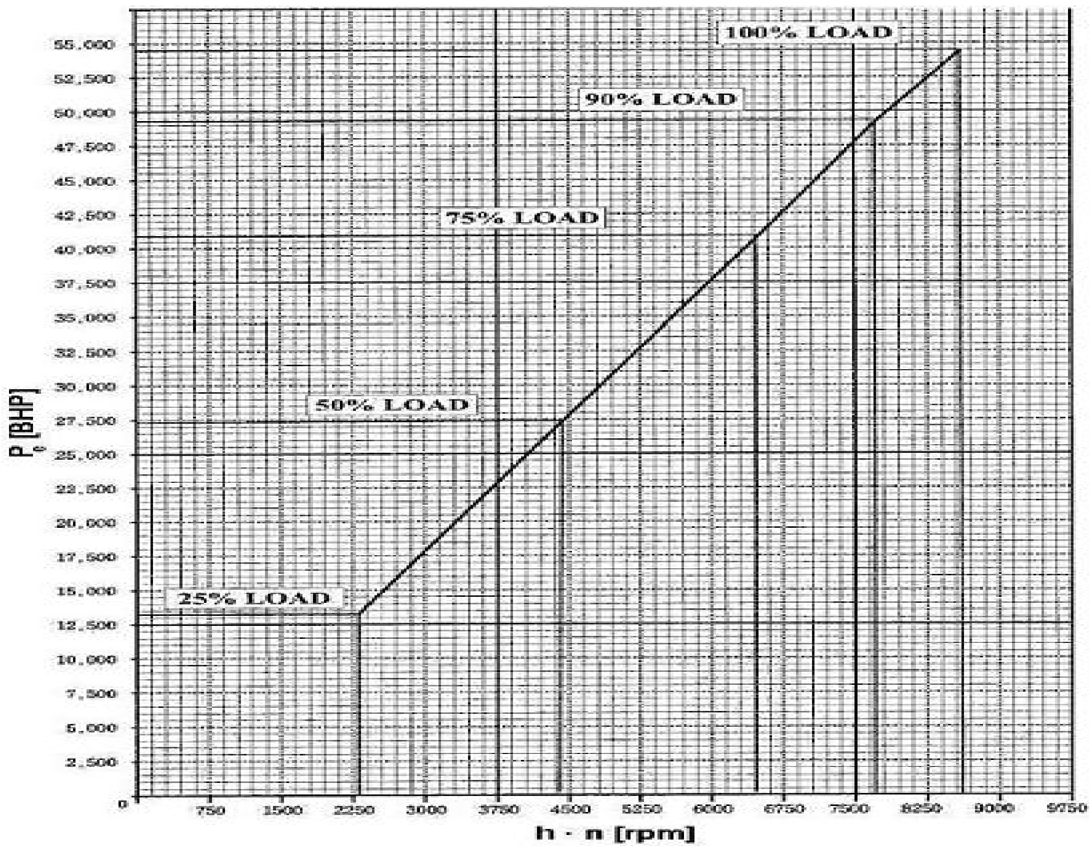

Fig. 5. The way of assessing effective power P_e based on load indication value h and rotary speed n for Wartsila 7RT-flex 97C-B engine [14]

The estimations have been carried out due to periodical reports of performance checks of a real working engine. The necessity for simultaneous register of numerous parameters, which is actually impossible because in real working conditions the parameters undergo continuous changes, makes the comparison of presented methods difficult. Another calculating error arises due to errors of measuring tools and possible reading errors concerning mainly some mean values (e.g. load indicator).

Tab. 2. The comparison of power estimations for Hyundai 7 RT-flex 96C-B engine carried out by means of two selected methods

Load indicator [%]	Rotary speed [rpm]	Power determined on the basis of indicating results from the relationship (3): P_{e1} [BHP]	Power determined on the basis of nomogram presented in Fig. 5: P_{e2} [BHP]	$\left\lvert\dfrac{P_{e1}-P_{e2}}{P_{e2}}\right\rvert 100\,[\%]$
42,8	86	24238	22000	9,23
47,4	90	25771	26000	0,89
57,9	97	31458	35000	11,26
69,3	100	40949	44500	8,67
83,5	101	45462	52500	15,48
91,4	108	49800	55000	10,44

The value of discrepancies in power estimations with the use of relationship (3) expressed in percents range within 0,89÷15,48. The differences are relatively small and allow for the conclusion that both methods show approximate exactness of the estimation. The presented results of the estimation need to be considered as initial and depend upon numerous factors. That is why in order to draw detailed conclusions regarding the type of estimation changes in the function of real load, detailed research concerning these aspects should be carried out

4. Engine load estimation based on work charging system parameters

MAN B&W company also recommend the way of effective power estimation based on engine charging system parameters. Fig.6 presents a nomogram used for specifying the effective power based on the turbocharger rotary speed and the charging air temperature after the cooler.

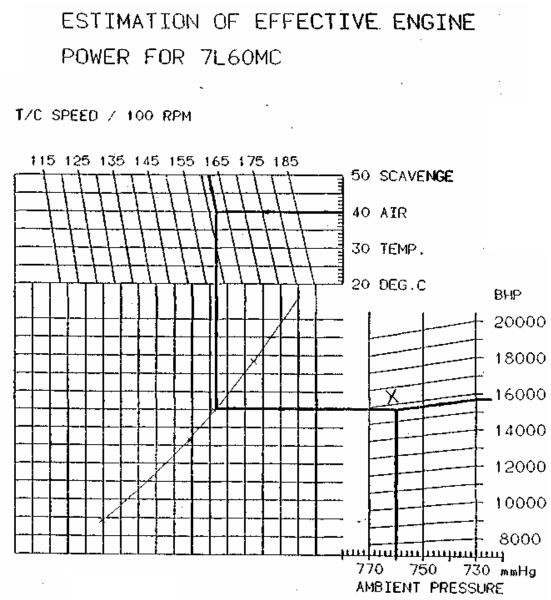

Fig. 6. The example of effective power estimation based on the turbocharger rotary speed and the charging air temperature and the atmospheric pressure of the MAN B&W 7L60MC engine [13]

The values determine the points which together with the engine trials curve the current value of the atmospheric pressure define the value of the engine effective power. The way turns out to be more precise than the formerly presented methods using the load indicator value. [13].

5. Final conclusion

In the paper the basic methods of torque engine load estimation have been presented. In fact for the main propulsion engines more and more often torquemeters have been installed. However, there have still existed many vessels without such equipment. In such cases the above presented methods are applied..

In order to verify the estimation values the methods can be connected, that is used together. Fig.7 presents the comparison of torque estimation run based on fuel rack setpoint and trial scale lines from the engine test bench and the line obtained due to the check indicating results [2]. Operational conditions undergo changes and they are often different from the conditions during the engine test bench [5, 6, 10].

Engine components including fuel pumps undergo wear causing changes of relationships between fuel rack position and the actual instantaneous fuel dose provided by the pump. That is why it is so meaningful to carry out periodical indication for the purpose of methods calibration.

Each of the presented methods has its advantages and disadvantages and a specified range of its application.. To the most universal methods belong the ones making use of indicators and readings of fuel rack setpoint values with the first ones which appear to be the most useful especially when the operator does not possess any engine test bench trial reports.

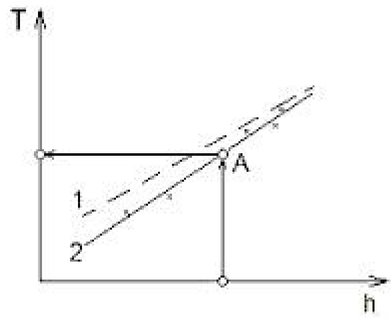

Fig. 7.General presentation of torque estimation based on of fuel rack setpoint .T –torque, h – fuel rack setpoint, 1– scaling line of engine test bench trials, 2 – scaling line based on indicator check results

Estimation based on effective power measure on the main switch board refers only to electric power generating engines. However, during the operation the method does not seem to be so significant because it requires the knowledge of generator efficiency and as a matter of fact the relative changes in engine torque load are reflected precisely enough in direct reading of the generator active power.

The method using the charging system parameters requires measure of atmospheric pressure. Its advantage is significantly higher precision of estimation when compared to other methods based on load indicator reading or fuel rack setpoint. [13].

References

1. Jędrzejowski, J., *Obliczanie tłokowego silnika spalinowego*. WNT, Warszawa 1984.
2. Kluj, S., *Diagnostyka urządzeń okrętowych*, Studium Doskonalenia Kadr WSM w Gdyni. Gdynia 2000.
3. *Komputerowe wspomaganie badań silników spalinowych*. Praca zbiorowa pod red. Z. Sroki. Oficyna Wydawnicza Politechniki Wrocławskiej, Wrocław 1996.
4. Nagawiecki, J., *Naprawy silników wysokoprężnych*. Wydawnictwo Morskie, Gdynia 1964.
5. Niewiarowski, K., *Tłokowe silniki spalinowe*. WKŁ, Warszawa 1983.
6. Niziński, S., *Elementy eksploatacji obiektów technicznych*. Wydawnictwo Uniwersytetu Warmińsko-Mazurskiego, Olsztyn 2000.
7. Piotrowski, I., Witkowski, K., *Okrętowe silniki spalinowe*. Trademar, Gdynia 1996.
8. Piaseczny, L., *Technologia naprawy okrętowych silników spalinowych*. Wydawnictwo Morskie, Gdańsk 1992.
9. Rychter, T., Teodorczyk, A., *Modelowanie matematyczne roboczego cyklu silnika spalinowego*. PWN, Warszawa 1990.
10. Wajand, J. A., Wajand, J. T., *Tłokowe silniki spalinowe średnio- i szybkoobrotowe*. WNT, Warszawa 1993.
11. Wimmer, A., Glaser, J., *Indykowanie silnika*. Instytut Zastosowań Techniki, AVL Warszawa 2004.
12. *50-108 ME Engines – Operation*. MAN B&W Instruction, Copenhagen 2005.
13. *7L60MC Engines – Operation*. MAN B&W Instruction, Copenhagen 2005
14. *Hyundai-Sulzer RT-flex96C-B Diesel Engine Operating Instruction*. Hyundai Heavy Industries. Ulsan 2006.
15. *Operator manual for the electronic cylinder pressure measuring system PREMET ONLINE SL Version 1.02.* LEHMANN & MICHELS GmbH, Hamburg 16.06.2004.