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Possibilities of reliable and safe main engine load evaluation on board ship

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Abstract

Proper load control of contemporary highly rated marine diesel engines is of paramount importance. The paper concentrates on the load diagrams of older and to days engines. Further means of controlling the engine power by a load control system are discussed. In the absence of a torquemeter on board the question is answered whether readings taken from a fuel pump rack or the engine load indicator are accurate enough to determine the power of the engine. Examples of discrepances between torquemeter and load indicator readings based on concrete examples leading to serious consequences are given.

1. Introduction

Contemporary marine diesel engines especially of the slow speed type since their introduction in 1983 whether it be a MAN–B&W or Wärtsila NDS engine have been continuously uprated throughout the past 15 years reaching a high specific output from a cylinder unit, characterised by the mean effective pressure reaching now a level up to 19 bars. This imposes in turn high thermal loads on the engine combustion space.

To prevent the engine from overloading in conditions such as heavy weather, fouled hull, shallow water, too heavy propeller layout or excessive shaft generator output the operator should keep his engine within the limits of the load diagram. With the development of the marine diesel engine the load diagram limits have also been changing.

Figures 1, 2 (1), (2) present the load diagrams of engines in the 1970's wheras Fig. 3, (3) 4, 5 (4) the load diagrams of engines in the 90's. It is worthwhile to notice that even the latest generation of MAN–B&W MC engines have modified their load diagrams. Diagram on Fig. 4 is valid for practically all MC engines installed in ships delivered up to and including 1991, whereas diagram on Fig. 5 is valid for subsequent installations.

2. Load Control

Both engine makers MAN–B&W and Sulzer have experienced cases where operation has occurred outside the limits of the load diagram. As a consequence of running the engine above the torque speed limit curve (4) see Fig. 4, 5 developed high thermal load has lead in some cases to cylinder liner cracks and burnt out piston crowns. To verify this a series of long term measurements were carried out in service on different ship types by MAN–B&W engine maker. A three month continuous measurement of engine load (a 6 S60MC engine) on one ship is illustrated on Fig. 6 (5). The measurements have documented that wind and wave action, together with hull fouling, shallow water and too heavy propeller layout or too large shaft propeller have an important influence on the daily loading of the engine. Up to 20 % higher load has been recorded due to influence of above mentioned factors. The recorded points on Fig. 6 show that on this ship the engine was continuously operating along limit 4 sometimes even crossing it over. The limit would obviously by exceeded if not for a load control system with a built – in limiter on the governer, whose function was to prevent overloading. The intention of MAN–B&W is to incorporate in future governors a limiter device as an integral part acting as a limiter. Based on carried out load measurements MAN–B&W has also changed their recent load diagram recommending a propeller layout with 2.5 – 5 % light running and very recently pushed the margin even to 3 – 7 % light running.

Fig. 7 (6) illustrates a load control system developed by MAN–B&W and tested on several ships, the interesting measurement results can be seen on Fig. 6 as well as on Fig. 8. From Fig. 7 it can be concluded that a load control system from which reliable measuring results are expected must contain a torsionmeter.

Unfortunately on the majority of ships in service as well as on new built a torquemeter placed on an intermediate propeller shaft is not a regular outfit of a ship propulsion plant. This is quite a difficult to understand attitude of the shipowners who order ships equipped with latest generation modern marine diesel engines but don't care so much about a more sophisticated monitoring equipment for the main propulsion unit.

What may discourage the shipowners from fitting torquemeters on board ships as a standard propulsion plant outfit is the problem of achieving perfect transmission of measured signals from rotating machine parts to the recording and data logging instruments as well as perfect calibration during a set-up of a torque meter. Signal transmission with a slip ring was often not a satisfactory solution and was as well failure prone in an environment exposed to water, oil and high temperatures on board ships. But in the meantime several contact less transmission system have been developed which offer accuracy of torque measurements between 1 – 2 % with a properly carried setting up. Let's hope that over the years more and more torquemeters will be installed on board ship becoming a conventional measuring instrument of a marine diesel engine power output. So in the absence of a torquemeter in the propulsion plant is the crew having some other reliable means to determine the operating point?

3. Engine load control system without a torquemeter

If a torquemeter is not available on board then there still exists some way to determine in a quite precise way the power of an engine. Namely the engine load indicator indications multiplied by the engine revolutions are an accurate way to calculate the engine power. This statement needs some further considerations substantiating the above made assumption. To get orientated what convergence do exists between the engine torque M_o (or mean effective pressure p_e) and the engine load indicator "L" in order to be able to ascertain eventual problems during transfering the parameter values expressed as a function of M_o into the engine load indicator indications some following considerations have to be done.

In accordance with the theoretical propeller curve the engine has to develop a power defined by the equation

$$N_e = \mathrm{C} \cdot n^3 \tag{1}$$

where: C – constant, n – engine revolutions

The needed (service) torque is

$$M_e = \frac{N_e}{n} = \mathrm{C}_1 \cdot n^2 \tag{2}$$

where: C_1 – constant

The needed nominal torque

$$M_{en} = \mathrm{C}_1 \cdot n_n^2 \tag{3}$$

where: n_n – nominal revolutions

hence

$$\frac{M_e}{M_{en}} = \left(\frac{n}{n_n}\right)^2 \tag{4}$$

The needed power developed by the engine can be expressed as

$$N_e = \mathrm{K} \cdot n \cdot p_e \tag{5}$$

where: K – engine constant, p_e – mean effective pressure

Assuming the engine load indicator

$$L \sim p_e \tag{6}$$

therefore:

$$N_e = \mathrm{K} \cdot n \cdot L \tag{7}$$

The needed (service) torque developed by the engine

$$M_e = \frac{N_e}{n} = \mathrm{K} \cdot L \tag{8}$$

The needed nominal torque developed by the engine

$$M_{en} = K \cdot L_n \tag{9}$$

therefore

$$\frac{M_e}{M_{en}} = \frac{L}{L_n} \tag{10}$$

If the assumption $L \sim p_e$ is valid then the following equation should be fulfilled

$$\left(\frac{n}{n_n}\right)^2 = \frac{L}{L_n} \tag{11}$$

The deviaton that $L \sim p_e$ is

$$\Delta = \frac{\frac{L}{L_n} - \left(\frac{n}{n_n}\right)^2}{-\left(\frac{n}{n_n}\right)^2} = \frac{\frac{L}{L_n}}{\left(\frac{n}{n_n}\right)^2} - 1 \tag{12}$$

Build on the above equation an analysis was performed to verify the assumption that $L \sim p_e$ based on obtained results from engine test bed. The results have been compiled in table 1 (7) in which, load indicator position and calculated value of Δ is given. During the study of this issue it was assumed that for $N_e = N_{en}$ at $n = n_n$ the position of the load indicator corresponds entirely with the mean effective pressure p_e $i \cdot e$. $L \sim p_e$. From this assumption it becomes obvious that for the nominal load $\Delta = 0$.

Analysing the results in table 1 it can be stated that between the four engines the deviation $\Delta = f(n)$ assumes positive and negative values within the limits + 8.4 % to − 6.8 %. The average for all for engines is about + 5 % this can be considered as a rather moderate deviation and measuring error, what in turns allows to consider the load indicators as a tool sufficiently determining the engine operating point in the load diagram. The quoted figures in table 1 are for ships and engines (Sulzer RND type) built in the 1970'S.

A quite interesting and striking results contains table 2 (8). The given in this table data stems from a recent (July 98) sea trial results of a new built ship in one of a well known shipyard. During the sea trial of this ship (a $45 \cdot 10^3$ DWT bulk carrier) for the measurement of the engine (a 6RTA 58T) torque three torquemeters were installed while the fourth engine torque value was calculated from the product load indicator x engine revolutions ($L \cdot rpm$).

The reason for such unusual measurement arrangement was a heated dispute between the shipyard and propeller maker who insisted that the calculations of engine power by using the formula $L \cdot rpm$ is an accurate method of power calculation and comparable with the results obtained from torquemeters readings. The dispute actually started when during sea trials of a previous ship of the same class an unaccepted difference between shipyard torquemeter readings and the $L \cdot rpm$ readings did occur see table 3 (9). As can be seen from table 3 the difference in power calculation by the torquemeter and the $L \cdot rpm$ formula

gave quite satisfactory results for the first ship. But as the second ship is concerned the difference during the first sea trial appeared to be quite large and the shipyard didn't accept the method of $L \cdot rpm$ as a credible way to calculate the engine power (on what the propeller maker insisted). So based on the torquemeter results it was concluded that the propeller is too heavy and a cutting of propeller blades was carried out. This ship went then for a second sea trial during which the discrepancy between the torquemeter and $L \cdot rpm$ formula became really horrendous. The propeller turned out to be now too light and it was difficult to achieve the contracted speed with initially set up in the contract engine revolutions.
From this rather unprecedented case it can be concluded that if the calibration of a torquemeter is not correctly done quite a significant error may occur.

4. Conclusions

Table 2 provides and undoubted proof how accurate is the estimation of engine power by using the $L \cdot rpm$ formula as well that there are differences between each of the installed torquemeters but the range of deviations lies within an accepted limits. It can be concluded from table 2 that the formula $L \cdot rpm$ has given in all power ranges higher values then torquemeters recordings but the average deviations were about + 2.5 % thus by getting a little higher values the operator of the engine calculating the power output by the $L \cdot rpm$ formula is on the safe side.
Finally it should be remembered that there are certain factors which have however, to be considered when using the formula $L \cdot rpm$ to obtain trustworthy results. These factors to be considered are as follows: – fuel calorific value of presently used fuel oil and density comparable with the one used on the shop trial. If not a correction of the calculated value has to be carried out.
Important parameters such as scavenging air pressure turbocharger speed and exhaust gas pressures have to be compared with the shop trial values. The state of the injection system, injection pumps, nozzles and injection timing must be as during shop trials. The reason for this is that for example a worn linkage would result in a different load indicator reading result. As long as above parameters of the measurement and the shop trial are comparable there is no reason to doubt the power calculation.

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(8) Sea trials results recorded by the author attending the trial
(9) Personal Correspondence with engine manufacturer H. Cegielski

Table 1. Analysis of assumption correctness that $L_{ind} \sim p_e$ for a theoretical propeller curve within a range $\frac{N_e}{N_{en}} = 0.5 \div 1.1$

N_e	n	$\left(\frac{n}{n_n}\right)^n$	Ship	L_{ind}	$\frac{L_{ind}}{L_{ind_{nom.}}}$	Δ
0.25	76.8	0.3960	Ship No. 1	3.0	0.4000	+0.010
0.50	96.8	0.6296		4.4	0.5867	- 0.068
0.75	110.8	0.8248		6.5	0.8667	+0.051
0.90	117.8	0.9323		7.0	0.9333	+0.001
1.0	n_n=122	1.0000		W_n=7.5	1.0000	0
1.1	125.9	1.0650		8.1	1.0800	+0.014
			Ship No. 2	-	-	-
				4.7	0.61840	-0.0178
				6.1	0.82026	-0.0269
				7.0	0.92100	-0.0121
				W_n=7.6	1	0
				8.2	1.07890	+0.0130
			Ship No. 3	-	-	-
				5.05	0.6824	+0.0839
				6.15	0.8311	+0.0076
				7.0	0.9459	+0.0146
				W_n=7.4	1	0
				8.0	1.0811	+0.0151
			Ship No. 4	-	-	-
				4.8	0.6233	-0.0100
				6.3	0.8182	0.0080
				7.1	0.9220	-0.0110
				W_n=7.7	1	0
				8.2	1.0650	0

Remark:
The analysis was carried out based on documents from the shop trials carried out at engine the maker test bed

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Table 2. Comparison of engine power measurement by the load indicator x RPM formula and three torquemeters

No.	Load Indicator Position	RPM	L_{ind} x revs KW	Shipyard Torquemeter		Classification Society Torquemeter		Research Institute Torquemeter	
				RPM	Power (KW)	RPM	Power (KW)	RPM	Power (KW)
1.	4.90	90.4	443.0 6300	90.4	6090	90,0	5850	90.43	6002
2.	5.10	90.4	461.0 6550	90.4	6361	90.0	6113	90.42	6267
3.	5.35	95.0	508 7300	95.0	7124	95.0	7001	95.05	7071
4.	5.45	95.0	548 7550	95.0	7361	95.0	7268	94.86	7325
5.	5.95	101.7	605 9000	101.7	8883	102.0	8927	101.74	8881
6.	9.95	101.7	605 9000	101.7	8901	102.0	8874	104.79	8881
7.	6.25	101.7	636 9550	101.7	9201	102.0	9143	101.71	9177
8.	6.15	101.7	625 9400	101.7	9217	102.0	9168	101.72	9186
9.	6.30	104.7	9800	104.7	9676	105.0	9671	104.7	9697
10.	6.45	104.7	10040	104.7	9999	105.0	9973	104.7	10010

Table 3. Comparison of M.E. Power Evaluation by load Indicator and Torque meter Indications

Ship	Load Ind.	Revs. revs/min	Power by LI x n (KW)	Power by Terque meter (KW)	Difference in Power Readings $\Delta P = L_{ind}$ − Torque meter (KW)
No.1	5.7	90.2	7400	7335	+ 65
	5.4	90.1	6900	6950	- 50
	6.0	96.1	8450	8485	+ 35
	6.2	96.0	8800	9012	- 212
	6.8	101.0	10300	10270	+ 30
	6.8	101.0	10300	10280	+ 20
No.2 First speed trial	5.9	90.5	7660	8119	- 459
	5.8	90.6	7500	8240	- 740
	6.3	95.5	8850	9631	- 781
	6.4	95.7	9040	9598	- 558
	7.1	101.6	11000	11330	- 330
	7.1	101.2	10930	11380	- 450
No.2 Second speed Trial	5.00 (5.75)	90.4	6300	7365	- 1065
	5.35 (6.2)	95.2	7300	8672	- 1372
	5.95 (6.7)	101.7	8900	10200	- 1300
	6.00 (6.7)	101.8	9020	10260	- 1240
	5.90 (6.4)	101.8	8870	9628	- 758
	5.80 (6.4)	101.9	8700	9632	- 932
	5.30 (5.7)	95.4	7210	7848	- 638
	4.80 (5.3)	90.2	5950	6732	- 782

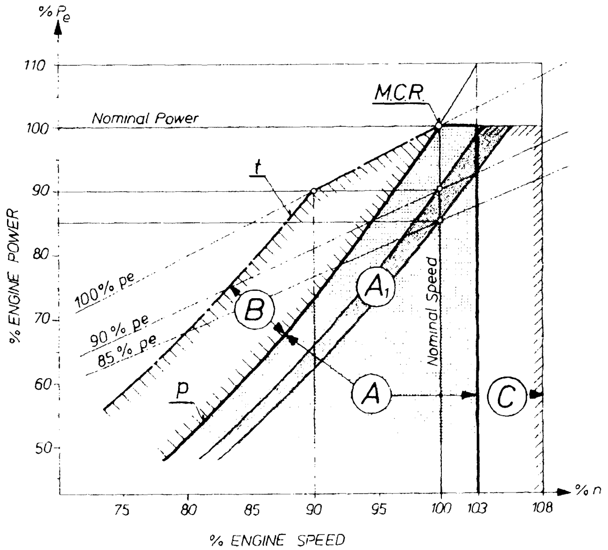

Fig. 1. Load Diagram of Sulzer RND engine
A : Optimum range for continuous operation, A_1 : Range for engine characteristic on sea trial with fair weather, ship fully laden and clean hull, B : Working range for restricted time only (max. 2000 hours), C : Upper speed range for sea trial only, p : Engine characteristic on shop trial, i.e. approximated propeller curve through the point of M.C.R.

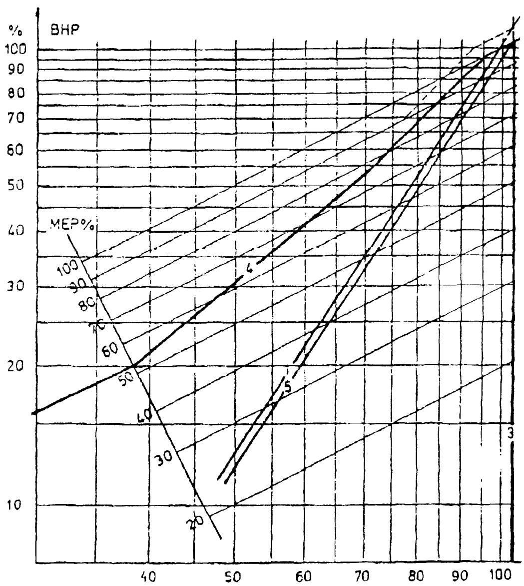

Fig. 2. Load Diagram of a B&W K/L – GFCA engine

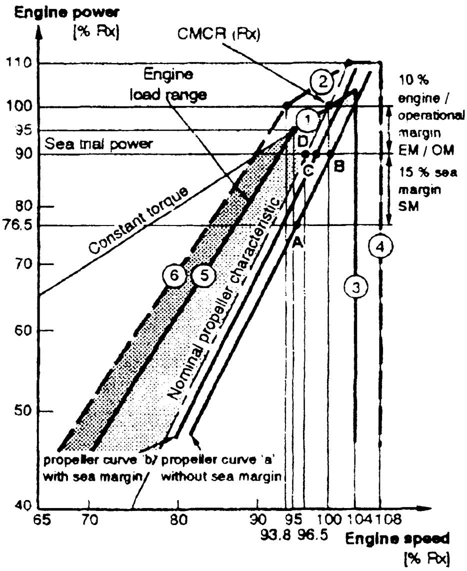

Fig. 3. Sulzer RTA 48T/RTA 58T Load diagram

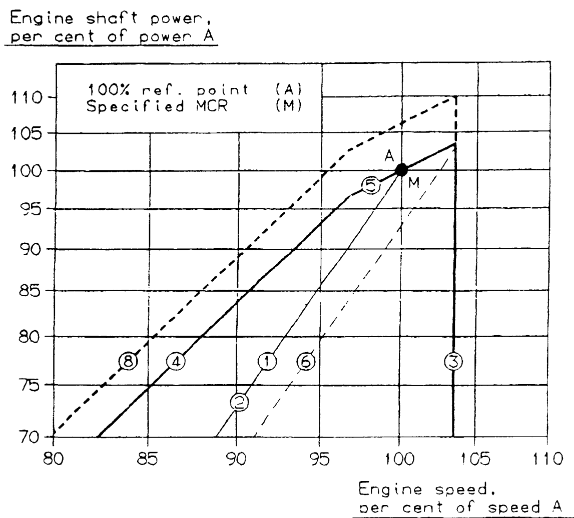

Fig. 4. Load Diagram for MC engines up to 1991Line
1 : Propeller curve through point A, Line 2 : Propeller curve, fouled hull – heavy running,
Line 3 : Speed limit, Line 4 : Torque/speed limit, Line 5 : Mean effective pressure limit,
Line 6 : Propeller curve, clean hull – light running (Range : 0 – 3 %),
Line 8 : Overload limit.

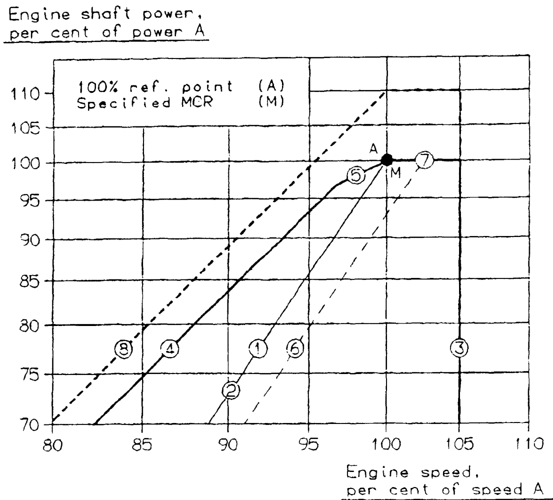

Fig. 5. Load Diagram for MC engines from 1992
Line 1 : Propeller curve through point A, Line 2 : Propeller curve, fouled hull – heavy running, Line 3 : Speed limit, Line 4 : Torque/speed limit, Line 5 : Mean effective pressure limit, Line 6 : Propeller curve, clean hull – light running (Range : 2.5 – 5 %), Line 7 : Power limit for continuous running, Line 8 : Overload limit.

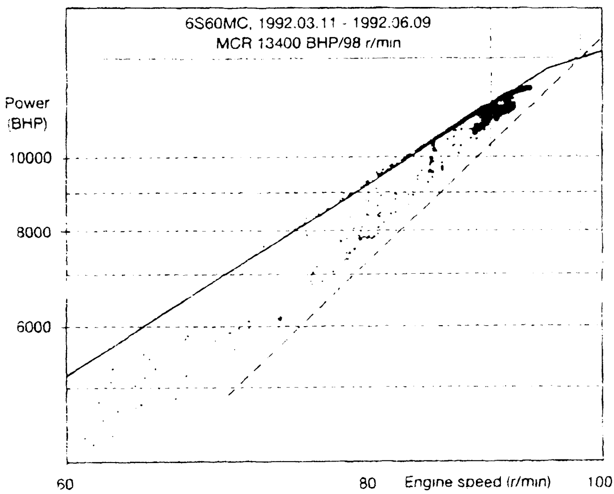

Fig. 6. Hourly recording of engine load over a three month period

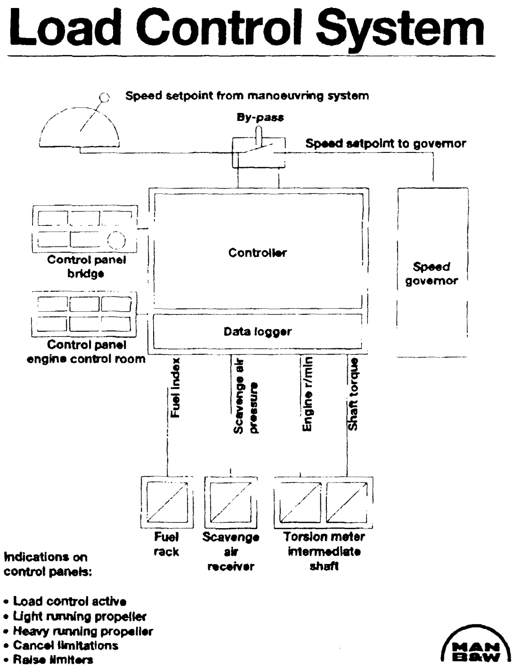

Fig. 7. Load Control System proposed by MAN – B&W

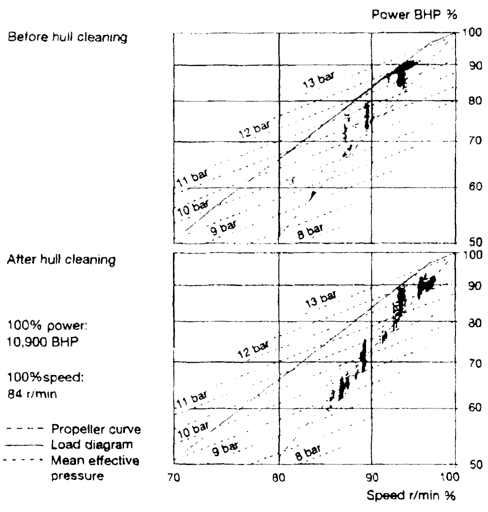

Fig. 8. Engine load conditions on 5L70MCE before and after hull cleaning February-May 1992