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ANALYSIS OF MAIN PROPULSION ENGINE SEATINGS IN SHIP POWER PLANTS

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Abstract

In the circumstances of information gap from the manufacturer of engines mounted in ship power plants about the tension of foundation bolts resulting from normal engine work in ship operation, the author resolved to check on ships constructed in Poland how the applied pre-tensions of foundation bolts relate to different parameters of engines. On the basis of research and control calculations of the seating of main propulsion engine series MAN B&W type L35MC on foundations it was shown that the number and tension of foundation bolts are directly proportional to the weight of seated engine.

Keywords: *marine engines, seating, foundation plates, pre-tension*

1. Introduction

Main propulsion diesel engines are seated on foundation plates in ship power plants by metal or polymer chocks [1]. According to regulations [2] two bolts should pass through each chock, with which the engine is mounted on the foundation plate. The bolts are pre-tightened, usually with a hydraulic jack, to a particular force value (usually 60-70% of ductility limit of the bolt material, frequently 42CrMo4), with controlled elongation, of usually elongated and reduced bolt shank. The larger the elongation when pre-tightening the bolts, the larger the security before loosening the bolt nuts during engine's work. This elongation is controlled during the engine's operation to make sure if there is no relaxation in the bolts' pre-tension.

The number of foundation bolts, their materials and dimensions and the pre-tension values are determined by the engine manufacturer. He also usually provides a hydraulic jack for pre-tightening of bolts with established piston surface in the cylinder and adjustable oil pressure. With proper selection of oil pressure the required tension of foundation bolts is obtained.

Regulations [1] require that the pre-tension of foundation bolts applied when seating the engine foundation in the ship power plant should be larger (without specifying how much larger) than the tension resulting from normal engine work at full load. What this last-mentioned is for a particular engine, is not given by the engine manufacturer and it has to be presumed that it is smaller than the pre-tightening of foundation bolts required by the manufacturer.

In literature [5] suggestions are made that when seating the engine on the foundation plate with metal chocks (usually made of steel) the tension sum of all foundation bolts should be ten times larger than the engine weight, due to the relatively small (0.1) coefficient of static friction of the metal chock against the foundation plate. When applying polymer chocks, due to the relatively large (0.7) coefficient of static friction of the polymer against the foundation plate it is suggested that for a certain position of the engine on the foundation it is sufficient for the tension sum of all foundation bolts to be five times larger than the engine weight.

Firm ITW Polymer Technologies [2] is of the opinion that for a certain seating of the engine on the foundation with polymer chocks in the ship power plant it is sufficient for the tension sum of all foundation bolts to be two and a half times larger than the engine weight.

2. Own research

The object of analysis is the seating in ship power plants of eight engines MAN B&W type L35MC differing as to the number of cylinders (5, 6, 7, 8, 9, 10, 11 or 12), power and weight. The engines were seated on foundation plates with cast polymer chocks. The distribution and dimensions of chocks are shown in Figs 1 and 2.

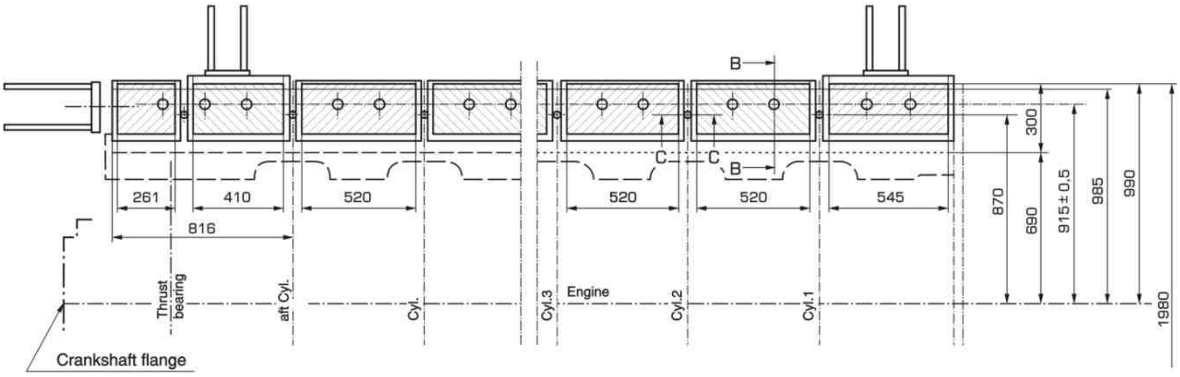

Fig.1. Scheme of seating engines MAN B&W type L35MC on foundation plates in ship power plants

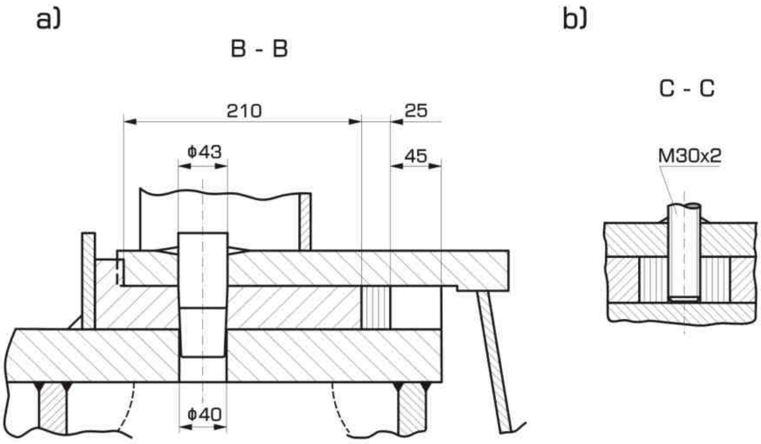

Fig. 2. Chock cross sections: a) section B-B showing position and dimensions of the chock hole for the foundation bolt, b) section C-C showing the position of bolt aligning between chocks

When aligning with the shaft line the engine is vertically removed from the foundation plate by means of 10-24 M30x2 aligning bolts (Fig. 2b).

Depending on the number of cylinders the engine is mounted on the foundation plate of the ship power plant by means of 26-54 M30x2 foundation bolts (Fig. 3) with reduced bolt shanks (∅ 26 mm), facilitating to obtain the required elongation of pre-tightened bolts. The obtaining of clear elongation is also enhanced by increasing the length of bolts combined with mounting a spacing sleeve (4).

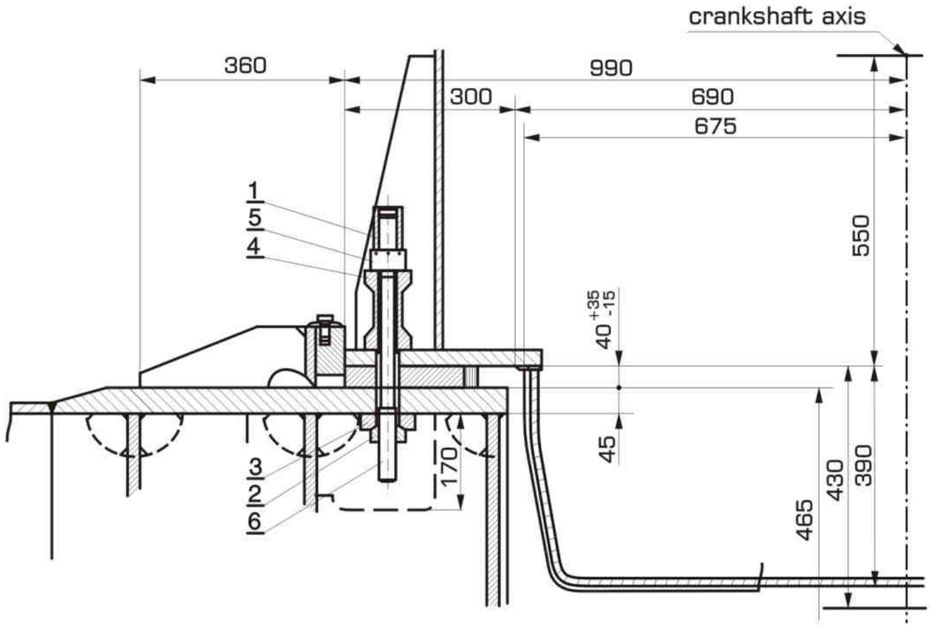

Fig.3. Foundation bolt and side stopper of engine MAN B&W type L35MC: 1 - protective shield, 2 - spherical nut, 3 - spherical chock, 4 - spacing sleeve, 5 – circular nut, 6 – foundation bolt

The certitude of the engine's position on the foundation plate is also ensured by four side stoppers and two head stoppers (Figs 1 and 2). This is important as there are forces trying to push off the engine from the foundation plate with the vessel's rolling and pitching.

For measuring seating parameters the weights of four engines type L35MC series were taken from MAN B&W products catalogue [3]. The number of foundation bolts for particular engines in the series were reckoned from Fig.1.

In control calculations of seatings of engine series MAN B&W type L35MC from dimensions given in Figs 1 and 2 there was calculated the effective surface of A_e polymer chocks for each engine. The engines were assumed to be seated on polymer chocks admitted by classification societies for load up to 3.5 N/mm^2 [1] (currently polymers are applied accepting load up to 5 N/mm^2).

Seating parameters of particular engines in the series were calculated from formulae given in regulations [1]. Changes of seating parameters with the change engine weights have been presented in Fig. 4.

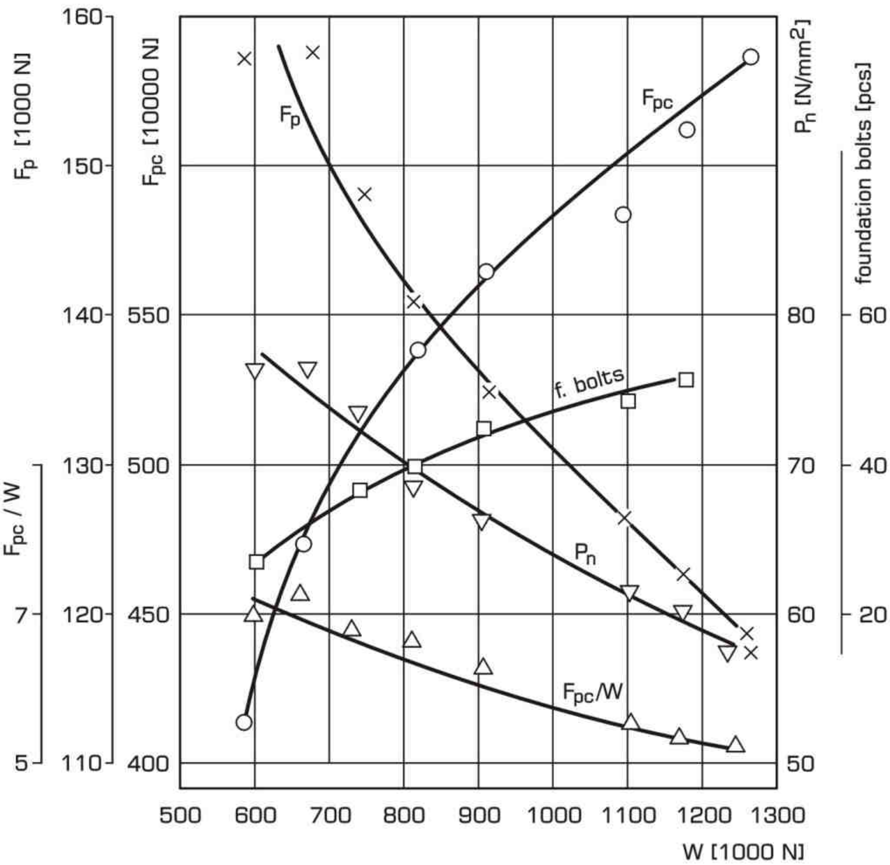

Fig. 4. Change of seating parameters in the weight function of engine MAN B&W type L35MC

Data system shown in Fig. 4 proves that all seating parameters of engine series MAN B&W type L35MC are directly or inversely proportional to the engine weight. The proportion of pre-tensions sum of foundation bolts F_{pc} to the engine weight decreases with the increase of engine weight from 7.11 to 5.08. This proportion can be considered as safety coefficient of a certain measure before tearing off the engine from the foundation plate, that is, the larger the engine weight the more certain its seating on the foundation in the ship power plant.

The dynamics of ship diesel engines [4] indicates that the forces tearing out the engine from the foundation plate can be the impacts of diesel gases on the head at ignition moment in the cylinder and vertical vibrations caused by unbalanced moments of inertial forces and centrifugal forces active on each crank. For engines of up to 300 rev/min, stiffly seated on foundations, vibrations in the frequency range 1-10 Hz with amplitude max 0.16 mm are considered admissible [3]. The operation of 6L35MC engine on a ship showed that engine vibrations were smaller than admissible ones, hence the impacts of gases against the head can be considered as the main force attempting to tear out the engine from the foundation plate.

It was established during research on a ship that with full load and 210 rev/min of 6L35MC engine the largest pressure of gases at ignition moment in the cylinder is equal to 14.5 N/mm^2. The wave of gases with this pressure strikes simultaneously against the piston crown, i.e. 96162.5 mm^2. The force of gases impact against the engine head is thus equal to F_g=1394356 N. In the 5L35MC engine currently analysed there are 26 foundation bolts, each of them being loaded with force F_{gj}=46479 N.

In the work of L35MC type engines, foundation bolts are also loaded with forces emerging from unbalanced external moments of first and second order (Fig. 5) and forces from transverse moment of type H bending the engine, arising from the pressure of the crosshead against the guide (Fig. 6a). The simultaneously arising moment of type X (Fig. 6b) does not result in loading of the foundation bolts.

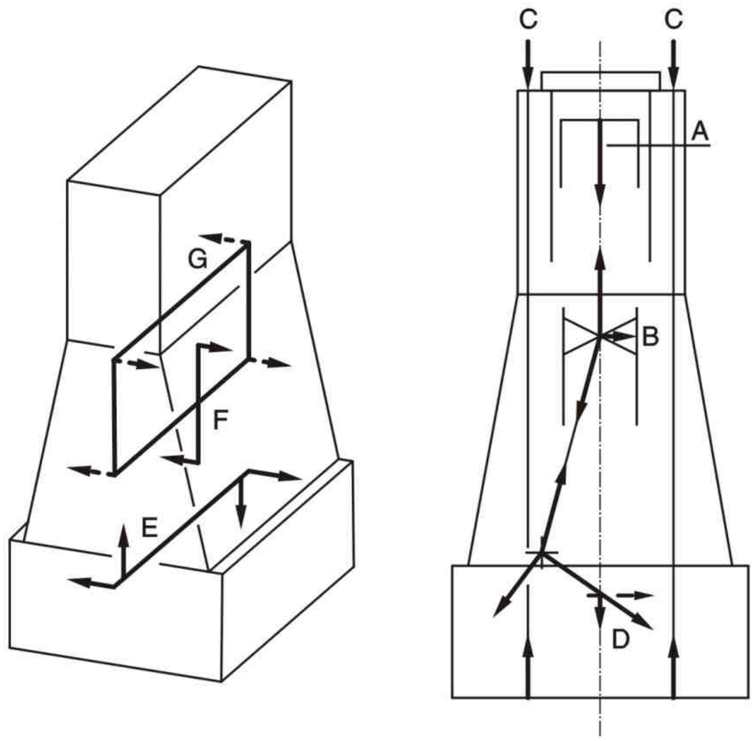

Fig. 5. Unbalanced external moments and force moments of the crosshead pressure on the guide: A – combustion pressure, B – crosshead pressure on the guide, C – bolt tension, D – forces on journal bearings, E – external vertical longitudinal moments of the first and second order, F – H type moment of pressure force of crosshead on the guide, G – X type moments of crosshead pressure force on the guide

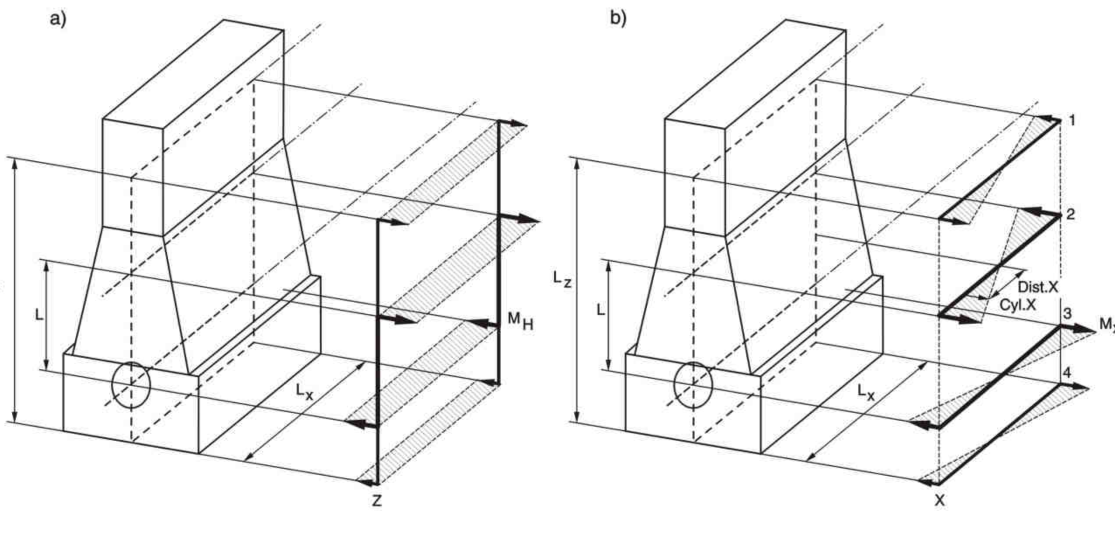

Fig. 6. H type (a) and X type (b) moments of crosshead pressure on the guide: 1 – engine top level, 2 – mean position of crosshead, 3 – level of crankshaft axis 4 – engine feet level

Unbalanced external moments arise from the engine's unbalanced elements masses performing rotary motion or reciprocating motion. External moments of the first order (one cycle per revolution) are considered first of all for engines with a small cylinder number (up to six cylinders). Inertial forces in engines with more than six cylinders get more or less self-neutralised.

The remedies (reducers of unbalanced external moments, Fig.7) are applied when the resonance of ship's hull vibrations occurs in the range of operational speed of the engine's revolutions and if the level of vibrations leads to accelerations and/or speeds higher than those established in international standards. The natural frequency of vessel's hull vibrations depends on the hull's stiffness and distribution of mass, whereas vibrations level at resonance depends mainly on the values of unbalanced external moments and the engine's position in relation to the vessel hull's vibration nodes.

In practice, the application of reducers of unbalanced external moments concerns engines with cylinder bores of 460 mm and more [5].

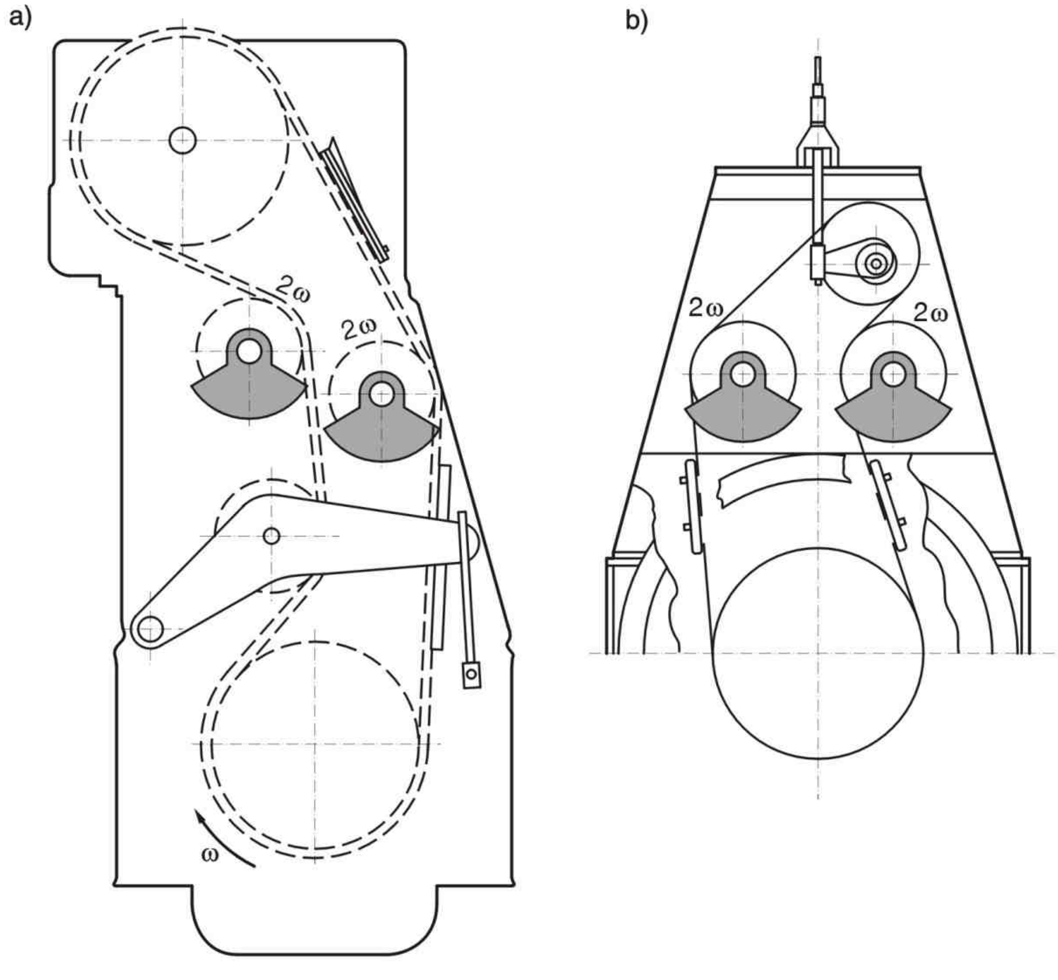

Fig. 7. Reducers of external moments of the first and second order mounted on MAN B&W engines: a) reduce mounted on the stern wall of the engine, b) reducer mounted on the bow wall of the engine

For the assessment of risk if external moments of the first and second order will disturb the vessel hull's vibrations, the PRU value (**P**ower **R**elated **U**nbalance) is applied as index, the value being the proportion of the external moment T to the engine's power P:

$$PRU = \frac{T}{P} \qquad \left[\frac{N \cdot m}{kW}\right] \tag{1}$$

according to manufacturer's data [3] the PRU values for the analysed engine series type L35MC have been given in Table 1. The Table also gives the values of the engine's unbalanced external moments reckoned at 650 kW/cylinder power obtained from those engines with 210 rev/min and full load.

Tab. 1. Values of PRU and unbalanced external moments of engines type L35MC with power 650 kW/cyl at 210 rev/min

Feature	Unit	5 cyl.	6 cyl.	7 cyl.	8 cyl.	9 cyl.	10 cyl.	11 cyl.	12 cyl.
PRU1	N·m/kW	9.1	0.0	3.9	5.7	10.3	2.3	1.4	0.6
PRU2	N·m/kW	88.0	51.0	12.6	0.0	11.2	0.5	1.8	0.3
T1	kN·m	29.575	0.0	17.745	29.640	60.255	14.950	10.01	4.680
T2	kN·m	286.00	198.90	57.33	0.0	65.52	3.25	12.87	2.34
PRU1 – first order moment index, PRU2 – second order moment index, T1 – first order moment, T2 – second order moment.									

In the analysed engine 5L35MC five unbalanced external moments of the first and second order are active, attempting to turn the engine on the foot edge of the engine's head wall on the stern side of the ship. The arms of these moments are distances of axes of particular cylinders from the foot edge of the engine's head wall. According to Fig.1, for the engine 5L35MC the moment arms are equal to: for cylinder 5 – 816 mm, for cylinder 4 – 1416 mm, for cylinder 3 – 2016 mm, for cylinder 2 – 2616 mm and for cylinder 1 3216 mm. This permits the writing down of the engine's external moments sum in the equations:

$$T1(T2) = F_b \cdot L = F_b(0.816 + 1.416 + 2.016 + 3.216) \qquad [N \cdot m] \tag{2}$$

After substituting in these equations the values M1 and M2 from Table 2 there are obtained the sums of forces acting on the engine's foundation bolts: F_{b1}=2934 N and F_{b2}=28373 N. As the 5L35MC engine has 26 foundation bolts, the forces loading a single bolt are equal to: F_{b1j}=112.8 N and F_{b2j}=10912.7 N.

In engine type L35MC the largest M_H moment appears when the crank R=525 mm sets perpendicularly to the cylinder axis, and crankshaft K=1260 mm is deflected from the cylinder axis by angle α emerged after the piston has performed half a stroke (Fig. 6). In this system, sinα=K/R=0.4167, and angle $\alpha \approx 24^o 40'$.

Engines of type L35MC, with piston travel s = 1050 mm have cylinder displacement volume V_s = 100970620 mm^3. With compression degree ε = 15, the volume of compression chamber is equal to $V_k = V_s/(\varepsilon - 1)$ = 7212187 mm^3. Maximum gas pressure in the compression chamber after fuel ignition with work at full load is equal to p_{ks} = 14.5 N/mm^2. It has been determined from Bernoulli equation that:

$$V_k \cdot p_{ks} = (V_k + 0{,}5V_s)p_{ps} \tag{3}$$

halfway of the piston stroke the gas pressure is equal to p_{ps} = 1.8125 N/mm^2. The gas force at this place is A_{ps} = 174294.5 N, hence B = A_{ps}·tg α = 80036 N. This force is active on arm L = K·cos α = 1145 mm and creates moment M_H = B·L = 91641220 N·mm. To find the force loading the row of the engine's foundation bolts, the distance of both bolt rows was accepted, i.e. 1830 mm, as the moment's arm (Fig. 1). Hence, the row of foundation bolts is loaded with force F = 50077.169 N. As the 5-cylinder engine produces five MH moments, the row of foundation bolts is loaded with force F_H = s·F = 250385.84 N. In a row of a 5-cylinder engine there are 13 foundation bolts, hence the load of a single bolt is equal to F_{Hj} = 19260 N.

Cumulatively, the operational load of a single bolt of engine 5L35MC is equal to:

$$F_a = F_g + F_{b1} + F_{b2} + F_H = 46479 + 2934 + 28373 + 2572 = 80358 \text{ N}.$$

According to regulations [1] the operational load of foundation bolts has to be smaller than their pre-tension. As the pre-tension of a single foundation bolt is equal to 1581 N (Table 1), this condition is fulfilled for engines MAN B&W 5L35MC.

3. Conclusions

The conducted research and control calculations of seatings on foundations in ship power plants of engines series MAN B&W type L35MC permit the statement that:

1. The sum of foundation bolts tension is directly proportional to the engine weight.
2. With the increase of engine weight there increases the certitude of engine seating on the foundation in the ship power plant measured by the relation F_{pc}/W.

References

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