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THE INFLUENCE OF VARIABLE COMPRESSION RATIO ON THE CONNECTING ROD’S STRAINS

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***Abstract**: We can see, at this moment, a worldwide trend concerning the fuel economy and limited pollutant emissions, especially when we are thinking to naval thermal engines. One of the various options in this direction is to alter the running cycles by correlating the compression ratio with engine`s load.*
This paper presents the results of a study carried for a four stroke naval engine, where were calculated the main parameters and their relation with the value of the compression ratio.
In the first step, there was realized a theoretical study, using a basic thermodynamic calculus and after correlating the results with the basic aim (the relation between performance and consumption) is presented a starting point for developing a concept of variable compression ratio naval engine.
***Keywords**: engine, compression, ratio, efficiency, naval*

1. INTRODUCTION

Engines are more sophisticated and efficient in 2010 than they were in the past but less than they will be in the future. They are equipped with many sensors that send information to the engine control unit (ECU) indicating the intake airflow, the oxygen content in the exhaust gases or even the engine speed. The ECU manages the operation of the engine via actuators that it controls electrically.

These actuators are, for instance, used to regulate the opening of the throttle valve, the amount of fuel injected, the valve timing or the ignition timing.

There is, however, an essential parameter over which the ECU has no control: the compression ratio. Yet, this parameter directly conditions engine efficiency, pollutant emissions and the engine’s torque and power performances. Compression ratio control has so far never been possible for lack of an appropriate technology. This shortcoming has led to increased fuel consumption, limited torque and power and less control over pollutant emissions.

2. MAIN CHARACTERISTICS FOR ALCO 12 R 251 FMA ENGINE

Bore: 228.6 mm
Stroke: 266.7 mm
Cylinders: 6L, 8L, 12V, 16V
Cylinder`s effective power: 147.75 kW
Effective pressure: 16.78 bar

This type of engine is used both in naval and onshore applications. For naval branch, the engine in both used as main and auxiliary engine. On the other hand, the engine is used in railways applications and as stationary generators.

3. THE VARIATION OF THE COMPRESSION RATIO WITH STROKE`S LENGTH VARIATION

First, we will consider a variation of $\Delta h_l = 0 \ldots 3\ [mm]$ for the connecting rod`s length, obtaining the following variation for the combustion chamber`s volume and for the value of the compression ratio.

$$V_c(h_l) = \frac{\pi \cdot D^2 \cdot h_{V_c}(h_l)}{4}, \quad (1)$$

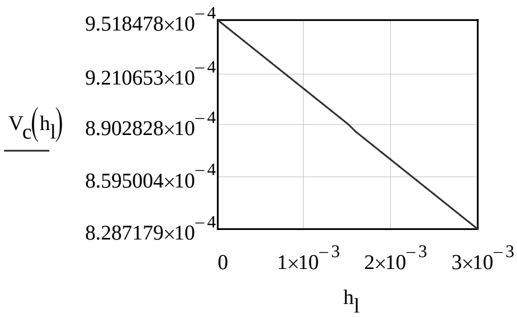


Figure 1 – The variation of the combustion chamber`s volume with the connecting rod`s length

Furthermore, there will be achieved a variation for the compression ratio, as follows:

$$\varepsilon(h_l) = 1 + \frac{V_s}{V_c(h_l)}, \quad (2)$$

$$\varepsilon(h_l) = 12{,}5 \ldots 14{,}2086$$

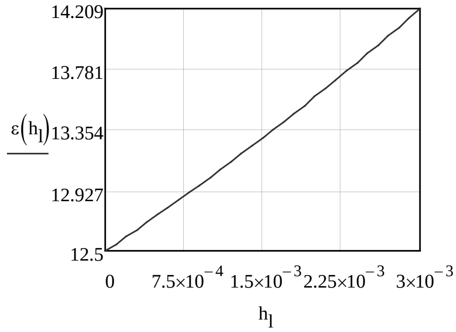


Figure 2 – The variation of the compression ratio value with the connecting rod`s length

4. THE VARIATION OF THE DIESEL CYCLE INDICATED PARAMETERS WITH THE COMPRESSION RATIO

$$P_i(\varepsilon) = \frac{\phi_T \cdot L_i'(\varepsilon)}{V_a(\varepsilon)-V_c(\varepsilon)}[Pa] \quad (3)$$

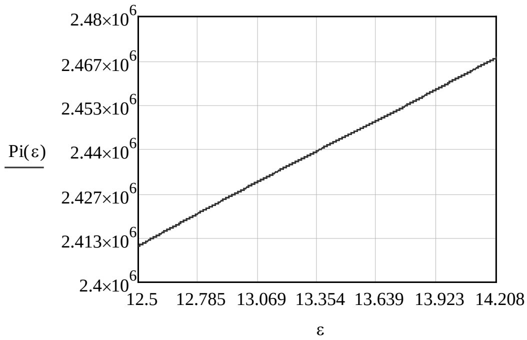


Figure 3 – The variation of the indicated pressure with the variation of the compression ratio

$$\eta_i(i) = \frac{8.31434 \cdot L \cdot T_S \cdot P_i(\varepsilon)}{P_s \cdot \lambda_v(\varepsilon) \cdot Q_i} \quad (4)$$

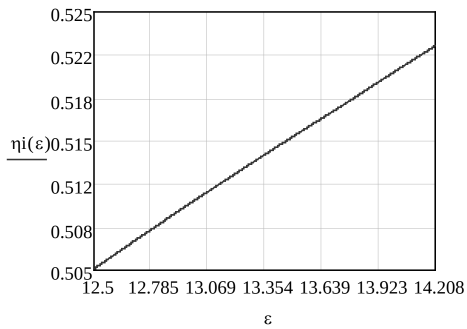


Figure 4 – The variation of the indicated efficiency with the variation of the compression ratio

$$c_i(\varepsilon) = \frac{3600}{\eta_i(\varepsilon) \cdot Q_i}\left[\frac{kg}{kWh}\right] \quad (5)$$

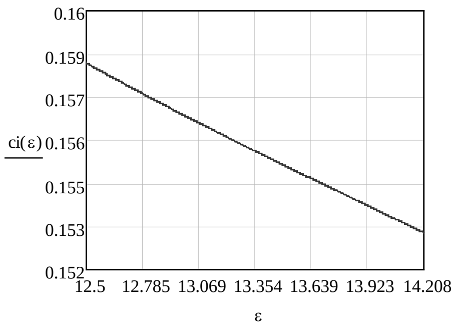


Figure 5 – The variation of the indicated fuel flow rate with the compression ratio variation

5. THE INITIAL CONDITIONSFOR GENERATING AND STUDYING THE MODELS AND THE CONNECTING ROD`S STRESSES IN CONSIDERED SITUATIONS

First, we will consider as an initial data set, the values obtained from the ALCO 12 R 251 FMA engine.

The main data for ALCO 12 R 251 FMA engine

Table 1

Bore	228,6	mm
Cranckshaft speed	1000	rpm
Athmospheric pressure	1,632	bar
The pressure at the end of the compression process	52,5519	bar
The maximum burning pressure	122,955	bar
The pressure at the end of the expansion process	5,7087	bar
The exhaust gas pressure	1,36	bar
The connecting rod`s weight	32,6	kg
The connecting rod`s weight correspondingly to the piston	9,78	kg
The connecting rod`s weight correspondingly to the crankshaft	22,82	kg

The material specifications

Table 2

The element	Alloy type	Yield strenght	Breaking strenght
The crankshaft	30MoC10	745	865
Connecting rod	40MoCN15	915	1020
Piston`s head	30MoVC12	905	990
Wrist pin	18MnCr13	840	925

The functional parameters used in the following analysis was obtained during some tests carried in the Naval Academy from Constanta. We will consider two situations.

In the first case, we will analise the situation of the real engine, when the compression ration value is 12,5 (figure 9), the real situation and in the second case we will consider the situation when the compression ratio value is 14,208.

At the end, we will compare the results obtained in both situations, considering the stresses resulted with specified variations.

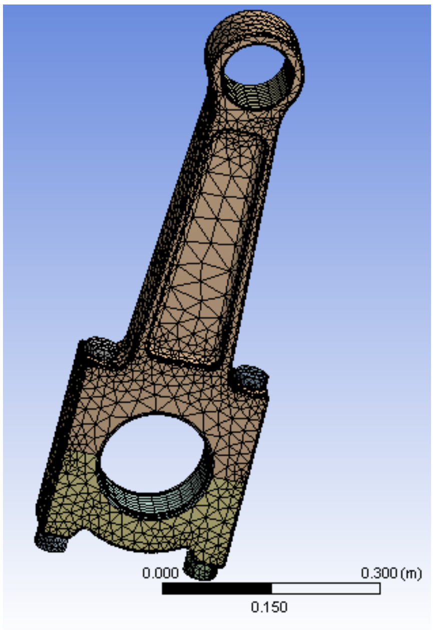


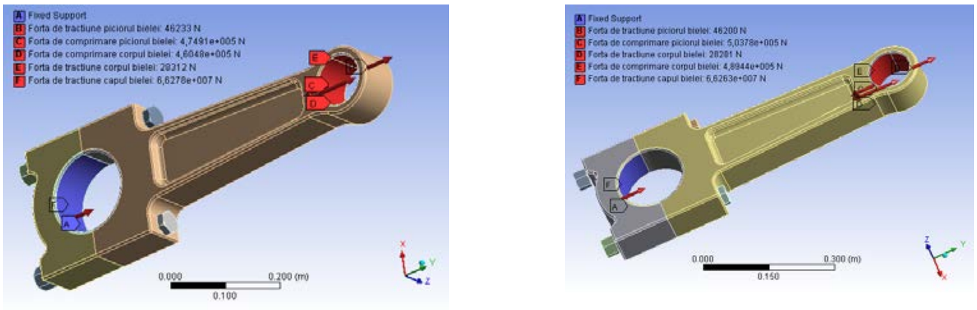
Fig. 6. The connecting rod for ALCO 12 R 251 FMA engine with ε=12,5

6. THE FORCES AND LOADS SYSTEM FOR THE STUDY MODEL

The required elements in order to generate the load system was obtained from the developement of cynematic, dynamic and resistance calculus, for each case (more precisely the case with a value of the compression ratio of 12,5 and respectively of 14,208). The results are presented in Table 3:

The load system for both considered cases *Table 3*

The stress	ε=12,5	ε=14,208
The traction stress on the connectig rod`s base [N]	46233	46200
The compression stress on the connecting rod`s base [N]	$4,7491*10^5$	$5,0378*10^5$
The compression stress on the connecting rod`s body [N]	$4,6048*10^5$	$4,8944*10^5$
The traction stress on the connecting rod`s body [N]	28312	28281
The traction stress on the connecting rod`s head [N]	$6,6278*10^7$	$6,6263*10^7$

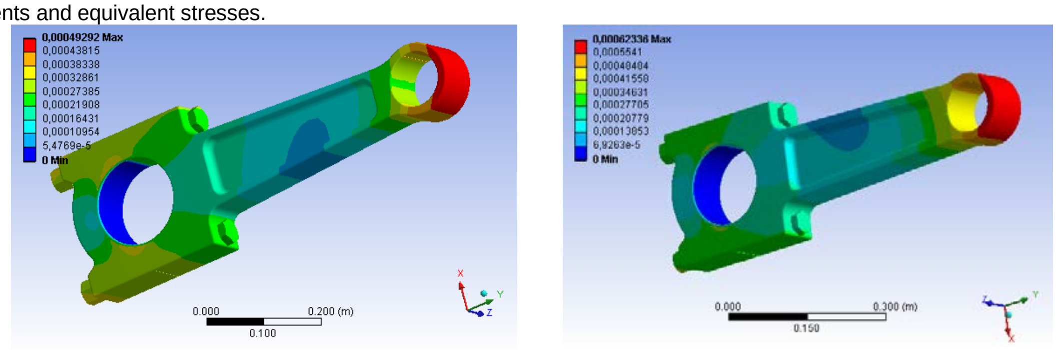


a) b)

Fig. 7 The load system
a)for the real engine; b)for the obtained model

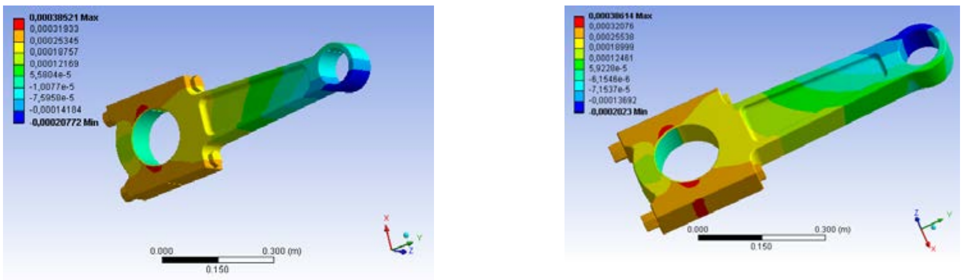
7. THE RESULTS OBTAINED DURING THE ANALISYS

After finalising the load system as in figures 8 and 9, the analisys was started, and the main results occured were total displacements and equivalent stresses.



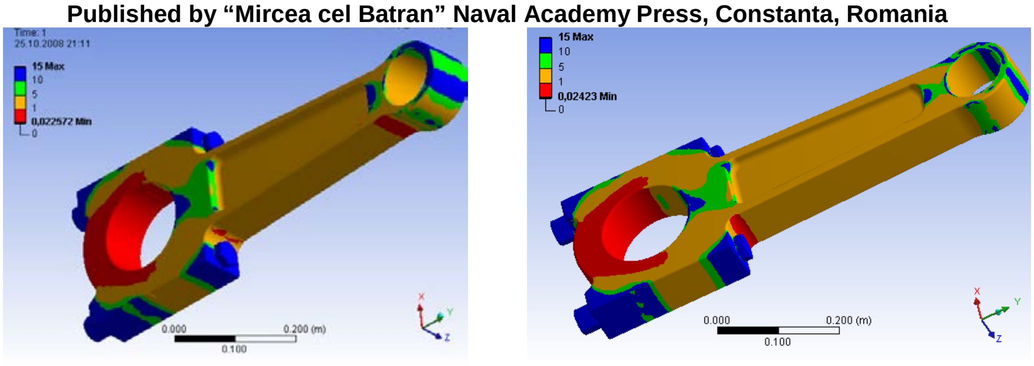
a) *b)*

Fig. 8. Total displacements
a) for the real engine; b)for the obtained model



a) *b)*

Fig. 9 The Oy axis displacements a) for the real engine; b)for the obtained model



a) *b)*

Fig.10. Safety coefficient
a)for the real engine; b)for the obtained model

8. CONCLUSIONS

First, according to the results achieved from the first set of the structural analisys, it was necessary to modify the connecting radius for the connecting rod, in the case of the increasing value of the compression ratio to 14,208, beacause of the stresses concentrators.

At the end, the results show a variation of the safety factor from a minimun of 0.0225, corresponding to a value of the compression ratio of 12,5 to a minimum of 0.0242 for a value of 14,208. Therefore, we can condiser that the connecting rod will undertake the stresses generated by the superior pressure value without major risks.

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