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Initial tightening of a cylinder head assembley on a marine engine block

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Abstract. When attaching the cylinder head to the engine block, to internal combustion engines, the force with which the mounting screws are tightened is of particular importance (in case of too tight a tightening the screws may break, and in the case of too small tightening the fluid may leave the container due to removal. the flanges one to the other).
In the case of slow, high-power engines, the engine block is individual, containing a single cylinder shirt. The intermediate solution of the engine block for a group of cylinders is also used.
The two-stroke engine chops are more strongly heat demanded, due to the doubling of the number of cycles in the unit of time, respectively by combustion processes in the unit of time.
Requesting the cylinder head bolts and other components of the assembly by force due to the pressure developed during the process of gas exchange and combustion in the engine cylinders is a demand with a pulsating character.

1. Introduction

As a result of the introduction of international environmental protection norms, in the last decades the manufacturers of internal combustion engines have been forced to produce engines whose pollutant and noise emissions decrease severely.

Internal combustion engines have made significant progress through:

- significant reduction of fuel consumption,
- increasing the durability of the components,
- reducing the specific consumption of material

These achievements have led to increasing demands on the components and subassemblies of current engines and the modification of traditional construction solutions.

The widespread use of aluminum alloys allowed the construction of lightweight constructions that increased the specific power of the engines. Due to the lower resistance compared to cast iron, in conjunction with the need to obtain higher sealing pressures due to the level of pressures and temperatures developed in the combustion chamber, it was necessary to recalculate the forces from the cylinder head bolts.

2. Operation of the prestressed threaded assembly

When attaching the cylinder head to the engine block, to internal combustion engines, the force with which the mounting screws are tightened is of particular importance (in case of too tight a tightening the screws may break, and in the case of too small tightening the fluid may leave the container due to removal flanges one to the other).

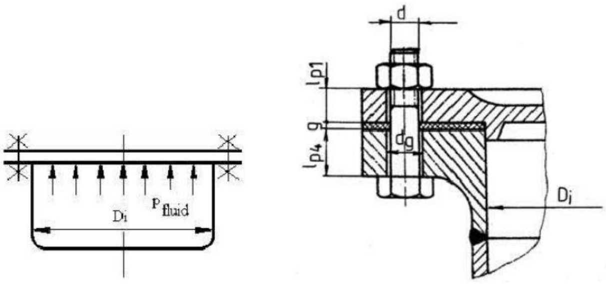

Figure 1 – Assembly with screws with initial tightening

The operation diagram of the assembly (Figure 2) Shows the dependence between the force F_0 and the two deformations for the screw and for the elements tightened at the operating point A.

The initial clamping force (clamping) with which each screw-nut assembly tightens the package of parts, consisting of flanges and gasket, simultaneously determines an elongation of the screw rod Δl_s and a compression of the package of parts Δl_p, considered to take place in the field Hooke's law.

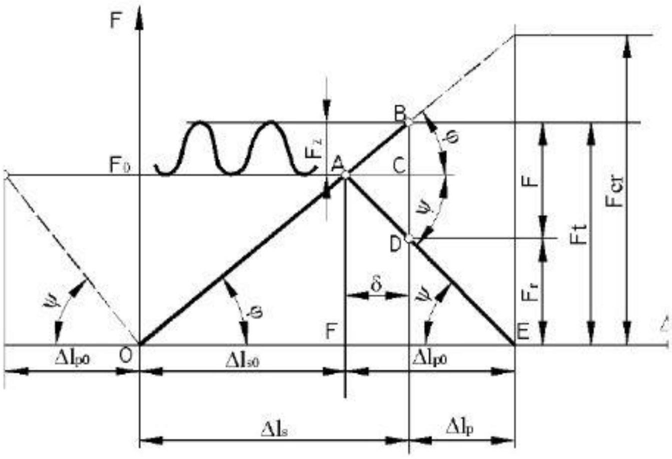

Figure 2 – Function diagram of the assembly with initial tightening

In the case of assembling the cylinder head gasket on the engine block, a number n_s of identical screws are used, during operation when a pressure of the fluid p_{fluid} is created inside the container, a force due to the circulation of the fluid tends to remove the lid.

The operating point on the diagram moves to B. This force will additionally require the traction screw and accordingly the parts package will relax with Δl due to the decrease of the clamping force to the value F_{rem} (remaining force). The remaining force is calculated based on the external force to ensure the tightness of the assembly.

Thus the screw is subjected to a total operating force that represents the sum of the two forces, the external force F respectively the remaining force F_{rem}or the sum between the initial clamping force F_0 and the additional force F_z.

If there is no remaining force in the package then the assembly loses its tightness and the entire external request is the screw:

$$F_{rem} = 0 \quad => \quad F_t = F$$

3. The internal combustion engine

The internal combustion engine is a thermal machine (it converts the energy produced by combustion of a fuel into mechanical work through a fluid, called a motor fluid) in which the products of combustion enter the composition of the engine fluid, and its evolution is accomplished by means of a piston, a whose alternative rectilinear motion within a cylinder is transformed into rotational motion by the crank mechanism.

Engine type MAN B&W 8S70MC-C8.2

The ship's propulsion is provided by a MAN type B&W 8S70MC-C8.2 engine, 2-stroke, slow and reversible, with a constant supercharging pressure that develops a rated power of 31,620 kW at a maximum speed of 72 rpm. The ship moves at a maximum speed of 15.70 Nd.

Figure 3 – Main type engine MAN B&W 8S70MC-C8.2

The S70MC-C8.2 engines are among the largest and most powerful engines built by MAN B&W in the two-stroke shipbuilding sector.

The characteristics of the main engine are as follows:

- Type: 8S70MC-C8.2
- MAC type engine, slow 2-stroke, with normal intake
- Number of cylinders: 8 in line
- Diameter of the cylinder: 900 mm
- Piston stroke: 3199 mm
- Piston speed: 7.77 m / s
- Injection pressure: 32 Mpa
- Maximum power: 31620 kW
- Maximum speed: 72 rpm
- Engine steering: electric motor

The 8S70MC-C8.2 engine is a slow, 2-stroke, normal-intake, 8-cylinder MAC (compression-ignition) engine with the cylinder head gasket fitted to each individual cylinder.

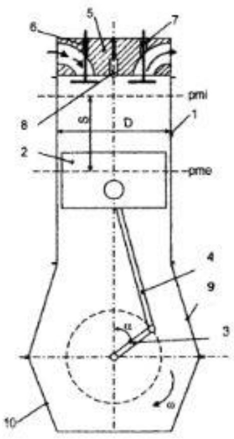

Figure 4 – Schematic diagram of a cylinder

The diagram of the engine in Figure 4 includes the engine mechanism and distribution system.

1 – cylinder	6 – intake valve
2 – piston	7 – exhaust valve
3 – crankshaft	8 – injector
4 – rod	9 – upper casing
5 – cylinder head	10 – lower casing

The piston that is connected to the crankshaft moves into the cylinder through the connecting rod. The cylinder is closed at the top with a cylinder in which there are 3 holes in which are arranged:

- ❖ intake valve
- ❖ exhaust valve
- ❖ injector

The lower part of the cylinder is fixed to the engine casing consisting of two parts:

- ✓ the lower casing from which the crankshaft bearings are suspended
- ✓ the upper casing that may contain the oil bath

During operation, the piston moves between two limit positions, called dead points (points where the speed of the moving parts is zero). The position of the piston that corresponds to the minimum volume occupied by the engine fluid is called the internal deadlock and the position of the piston that corresponds to the maximum volume occupied by the engine fluid is called the external deadlock.

The space traveled by the piston between the two dead points is called the stroke of the piston *S* and the diameter of the cylinder *D*.

The volume generated by the displacement of the piston in race *S* is called unit cylinders:

$$V_s = \frac{\pi \cdot D^2 \cdot S}{4}\ dm^3$$

The sum of the cylinders of a polycylindrical motor represents the total cylinder

$$V_t = i \cdot V_s\ dm^3$$

Where

i – number of cylinders

The ratio between the two volumes is called the compression ratio

$$\varepsilon = \frac{V_a}{V_c} = 1 + \frac{V_s}{V_c} = 1 + \frac{S}{h_{min}}$$

The accomplishment of a motor cycle supposes first of all to be introduced in the fresh fluid cylinder (air); the introduction of fresh fluid represents the intake process.

The release of the chemical energy of the fuel takes place in the combustion process. In order to increase the efficiency of this process (implicitly the efficiency of the operating cycle), a compression process of the motor fluid is interposed.

The useful mechanical work is interposed by the action of the flue gases on the piston, in the relaxation stroke. After the combustion process, the relaxation process takes place. To resume the engine cycle, the combustion gases are removed from the cylinder.

Removing the flue gas from the cylinder is the process of evacuation. Exhaust and intake processes together constitute the so-called gas exchange processes (distribution processes). All the processes that make up the motor cycle are called thermal processes.

In the case of slow, high-power engines, the engine block is individual, containing a single cylinder shirt. The intermediate solution of the engine block for a group of cylinders is also used.

In two-stroke engines, the sealing of the cooling spaces in the windows area is done with copper rings for gas, followed by one or two rubber rings.

The construction of the cylinder head is determined by the architecture of the combustion chamber and the inlet and outlet galleries, by the necessity of efficient cooling of the hottest areas, minimizing the thermal demands, by the conditions of resistance and rigidity, by technological considerations. In two-stroke engines, the cylinder head construction is simpler, as all or part of the distribution valves are missing.

The cylinder head can be made in a common body for all cylinders, for groups of cylinders or for a single (individual) cylinder.

The two-stroke engine chops are more strongly heat demanded, due to the doubling of the number of cycles in the unit of time, respectively by combustion processes in the unit of time.

The commonly used materials for making the cylinder head are cast iron and aluminum alloys. Cast iron chippers generally equip high-power engines with high mechanical and thermal demands. The cast iron has high mechanical properties (which are maintained at high temperatures) and ensures high rigidity of the cylinder head. Generally, gray cast iron or special cast iron alloyed with Cr, Ni, Mo, Cu is used.

4. Calculation of the cylinder head gasket assembly on the engine block

4.1. Input data

1. Characteristics of the screw

- ⇨ Screw material - OL60 laminated steel
- ⇨ Group of materials for screw - 5.6
- ⇨ Type thread - M16

2. Adopted bolt and nut elements

- ➢ Thread dimensions:
  - ✓ $p = 2\ mm$ - thread pitch
  - ✓ $d = 16\ mm$ - outer diameter (nominal diameter of threaded assembly)
  - ✓ $d_2 = 15{,}026\ mm$ - average diameter, respectively of the nut
  - ✓ $d_1 = 14{,}376\ mm$ - minimum inside diameter, respectively minimum outside diameter of the nut

- The dimensions of the fixing screw:
  - ✓ $k = 10\ mm$ - head height
  - ✓ $l = 60\ mm$ - total length
  - ✓ $b = 30\ mm$ - thread length
  - ✓ $e = 26{,}5\ mm$ - outer dimension of the head
  - ✓ $d_g = 18\ mm$ - diameter of the screw hole for the screw

- Tightening nut dimensions:
  - ✓ $m = 13\ mm$ - the height of the nut
  - ✓ $S = 24\ mm$ - the keyhole opening of the hexagon nut
  - ✓ $D_1 = 0{,}95 \cdot S = 0{,}95 \cdot 24 = 22{,}8\ mm$ - diameter of bearing nut surface

3. **Number of screws**

$$n_s = 15 \text{ bucăți}$$

4. **The inner diameter of the cylinder**

$$D_i = 900\ mm$$

5. **The fluid pressure in the container**

$$p_f = 6\ bari = 0{,}6\ MPa$$

6. **Gasket**

→ Gasket material: brass
→ The thickness of the gasket

$$g_g = 5\ mm$$

7. **The flow limit of the screw material**

$$\sigma_c = 300\ MPa$$

8. **Tensile strength of screw material**

$$\sigma_r = 500\ MPa$$

9. **The thickness of the flanges of the container**

$$l_{p1} = l_{p4} = 20\ mm$$

10. **The modulus of elasticity of the screw material**

$$E_s = 2{,}1 \cdot 10^5\ MPa$$

11. **The modulus of elasticity of the gasket material**

$$E_{p2} = E_{p4} = 0{,}9 \cdot 10^5\ MPa$$

4.2. Calculated element

1. **Force due to fluid pressure**

$$F_{pf} = p_f \cdot \frac{\pi \cdot D_i^2}{4} = 0{,}6 \cdot \frac{3{,}14 \cdot 900^2}{4} = 381{,}5 \cdot 10^3\ N$$

2. **External force on a screw**

$$F_s = \frac{F_{pf}}{n_s} = \frac{381{,}5 \cdot 10^3}{15} = 25{,}43 \cdot 10^3\ N$$

3. Coefficient χ for sealing

χ - is recommended depending on the type of gasket

Gasket type – brass

$$\chi = 3$$

4. Remaining force

$$F_{rem} = \chi \cdot F_s = 3 \cdot 25{,}43 \cdot 10^3 = 76{,}29 \cdot 10^3\ N$$

5. Total strength of the screw

$$F_t = F_0 + F_z = F_s + F_{rem} = 25{,}43 \cdot 10^3 + 76{,}29 \cdot 10^3 = 101{,}72 \cdot 10^3\ N$$

6. Calculation lengths for screw

→ the length of the non-threaded section

$$l_{tn} = \frac{1}{3} \cdot k + (l - b) = \frac{1}{3} \cdot 10 + (60 - 30) = 33{,}3\ mm$$

→ threaded section length

$$l_{tf} = \frac{1}{2} \cdot m + \left(l_{p1} + g_g + l_{p4}\right) - (l - b) = \frac{1}{2} \cdot 13 + (20 + 5 + 20) - (60 - 30) = 21{,}5\ mm$$

7. The cross-sectional area of the screw

→ area of non-threaded section

$$A_{tn} = \frac{\pi \cdot d^2}{4} = \frac{3{,}14 \cdot 16^2}{4} = 201{,}06\ mm^2$$

→ threaded section area

$$A_{tf} = \frac{\pi \cdot d_1^2}{4} = \frac{3{,}14 \cdot 14{,}376^2}{4} = 162{,}31\ mm^2$$

8. Screw stiffness

$$\frac{1}{c_s} = \frac{1}{E_s} \cdot \left(\frac{l_{tn}}{A_{tn}} + \frac{l_{tf}}{A_{tf}}\right) = \frac{1}{2{,}1 \cdot 10^5} \cdot \left(\frac{33{,}3}{201{,}06} + \frac{21{,}5}{162{,}31}\right) = \frac{1}{2{,}1 \cdot 10^5} \cdot (0{,}1656 + 0{,}1324) = \frac{0{,}2980}{2{,}1 \cdot 10^5}$$

$$\Rightarrow\ c_s = 7{,}04 \cdot 10^5$$

Where:

E_s - modulus of elasticity of the screw material

$l_{tn}; l_{tf}$ - the lengths of the distinct sections on the screw

$A_{tn}; A_{tf}$- the areas of the separate sections on the screw

9. Calculation lengths for the pack of tight parts

→ length calculation of the flange

$$l_{p1} = l_{p4} = 20\ mm$$

→ the calculation length of the gasket

$$l_{p2} = l_{p3} = \frac{1}{2} \cdot g_g = \frac{1}{2} \cdot 5 = 2{,}5\ mm$$

10. Flange stiffness

The calculation of the rigidity of each piece in the package is done by replacing the cone trunk with an equivalent cylinder (having the same height and volume) that passes through the middle of the cone trunk generator.

→ equivalent diameter

$$D_{ech1} = D_1 + l_{p1} \cdot tg\lambda = 22{,}8 + 20 \cdot tg17° = 22{,}8 + 20 \cdot 0{,}306 = 22{,}8 + 6{,}12 = 28{,}92\ mm$$

Where:

$\lambda = 17°$ – the semicircle of the cone

→ the equivalent area

$$A_{p1} = \frac{\pi}{4} \cdot \left(D_{ech1}^2 - d_g^2\right) = \frac{\pi}{4} \cdot (28{,}92^2 - 18^2) = \frac{3{,}14}{4} \cdot (836{,}37 - 324) = = \frac{3{,}14}{4} \cdot 512{,}37 = 402{,}21\ mm^2$$

→ rigidity

$$c_{p1} = \frac{E_{p1} \cdot A_{p1}}{l_{p1}} = \frac{2{,}1 \cdot 10^5 \cdot 402{,}21}{20} = 44{,}12 \cdot 10^5$$

11. Rigidity of the gasket

→ equivalent diameter

$$D_{ech2} = D_2 + l_{p2} \cdot tg\lambda = 34 + 2{,}5 \cdot tg17° = 22{,}8 + 20 \cdot 0{,}306 = 34 + 0{,}764 = 34{,}76\ mm$$

→ the equivalent area

$$A_{p2} = \frac{\pi}{4} \cdot \left(D_{ech2}^2 - d_g^2\right) = \frac{\pi}{4} \cdot (34{,}76^2 - 18^2) = \frac{3{,}14}{4} \cdot (1208{,}25 - 324) = = \frac{3{,}14}{4} \cdot 884{,}25 = 694{,}14\ mm^2$$

→ rigidity

$$c_{p2} = \frac{E_{p2} \cdot A_{p2}}{l_{p2}} = \frac{0{,}9 \cdot 10^5 \cdot 694{,}14}{2{,}5} = 249{,}89 \cdot 10^5$$

12. Rigidity of the package

$$\frac{1}{c_p} = \frac{2}{c_{p1}} + \frac{2}{c_{p2}} = \frac{2}{44{,}12 \cdot 10^5} + \frac{2}{249{,}89 \cdot 10^5} = \frac{0{,}045}{10^5} + \frac{0{,}008}{10^5} = \frac{0{,}053}{10^5}$$

$$=> \ c_p = 18{,}87 \cdot 10^5$$

13. Rigidity ratio

$$k_r = \frac{c_s}{c_s + c_p} = \frac{7{,}04 \cdot 10^5}{7{,}04 \cdot 10^5 + 18{,}87 \cdot 10^5} = \frac{7{,}04 \cdot 10^5}{25{,}91 \cdot 10^5} = 0{,}27$$

14. Extra force from the screw

$$F_z = k_r \cdot F_s = 0{,}27 \cdot 25{,}43 \cdot 10^3 = 6{,}87 \cdot 10^3$$

15. Initial tightening force of the screws

$$F_0 = F_t - F_z = 101{,}72 \cdot 10^3 - 6{,}87 \cdot 10^3 = 94{,}85 \cdot 10^3\ N$$

If we graphically represent in a coordinate system $((F_s, F_t)$ the diagram in figure 5 is obtained. The right AB corresponds to the relation

$$F_t = F_0 + \frac{c_s}{c_s + c_p} \cdot F_s$$

and the right BC is given by the relation

$$F_{rem} = 0 \quad => \quad F_t = F$$

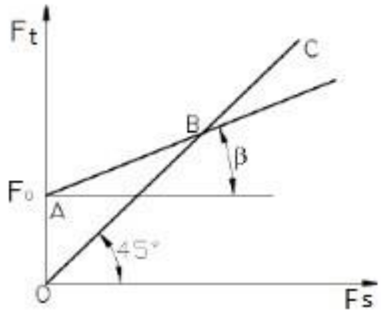

Figure 5 – Diagram of variation of forces

16. Tensile stress

$$\sigma_t = \frac{F_t}{A_{min}} = \frac{4 \cdot F_t}{\pi \cdot d_1^2} = \frac{4 \cdot 101{,}72 \cdot 10^3}{3{,}14 \cdot 14{,}376^2} = \frac{406{,}88 \cdot 10^3}{648{,}94} = 117{,}36\ MPa$$

17. The thread winding angle of the thread

$$\beta_m = arctg\frac{p}{\pi \cdot d_2} = arctg\frac{2}{3{,}14 \cdot 15{,}026} = arctg0{,}042 = 2{,}42°$$

18. The coefficient of friction between the turns

Recommended $\mu = 0{,}1 \ldots 0{,}15$

$$\mu = 0{,}12$$

19. The angle of friction in the thread

$$\varphi' = arctg\frac{\mu}{cos\frac{\beta}{2}} = arctg\frac{0{,}12}{cos\frac{15{,}11°}{2}} = arctg\frac{0{,}12}{0{,}991} = arctg0{,}121 = 6{,}90°$$

Where:

$$tg\beta = \frac{F_t - F_0}{F_s} = \frac{c_s}{c_s + c_p} = 0{,}27$$
$$=> \beta = arctg0{,}27 = 15{,}11°$$

To ensure self-braking the condition is as:

$$\beta_m \leq \varphi'$$
$$2{,}42° < 6{,}90°$$

⇨ The self-braking condition is fulfilled

20. The moment of friction in the thread

$$M_{t1} = F_0 \cdot \frac{d_2}{2} \cdot tg(\varphi' + \beta_m) = 94{,}85 \cdot 10^3 \cdot \frac{15{,}026}{2} \cdot tg(6{,}90° + 2{,}42°)$$
$$= 94{,}85 \cdot 10^3 \cdot 7{,}513 \cdot tg9{,}32° = 94{,}85 \cdot 10^3 \cdot 7{,}513 \cdot 0{,}164$$
$$= 116{,}87 \cdot 10^3\ N \cdot mm$$

21. Torsion tension

$$\tau_t = \frac{M_{t1}}{W_{p\,min}} = \frac{16 \cdot M_{t1}}{\pi \cdot d_1^3} = \frac{16 \cdot 116{,}87 \cdot 10^3}{3{,}14 \cdot 14{,}376^3} = \frac{1869{,}92 \cdot 10^3}{9329{,}19} = 38{,}24\ MPa$$

22. The equivalent stress

$$\sigma_{ech} = \sqrt{\sigma_t^2 + 3 \cdot \tau_t^2} = \sqrt{117{,}36^2 + 3 \cdot 38{,}24^2} = \sqrt{13773{,}37 + 3 \cdot 1462{,}29} =$$
$$= \sqrt{13773{,}37 + 4386{,}87} = \sqrt{18160{,}24} = 134{,}75\ MPa$$

23. The coefficient of safety

$$c = \frac{\sigma_c}{\sigma_{ech}} = \frac{300}{134{,}75} = 2{,}23$$

Permissible safety factor:

$$c_a = 1{,}5 \dots 2$$

The condition is $c \geq c_a$

$\mathbf{2{,}23 > 2}$ **=> the screw holds**

5. Conclusion

At these assemblies, during the assembly, an axial force F_0 called the prestressing force is induced, which puts the screw to the traction and compresses the assembled parts. During operation, an operating force appears which has the same direction as the prestressing force, overlapping it.

Functionally, this type of assembly must ensure tightness (in containers, pipes, cylinder heads, etc.) or prevent relative movement (at connecting rods and sliding bearings).

The present work offers elements for establishing the values of the prestressing forces characteristic of the screws of the heat engine cylinders, so that after this operation there is sufficient reserve for the surplus of demand that appears in operation. The precision regarding the intensity of these forces is of great importance, in order to avoid extremely hard accidents.

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