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OPTIMAL INJECTOR SPRING DESIGN FOR MAN B&W AG ENGINE

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Abstract **:** The injector is the machine part that performs the function of atomizing the fuel and introducing it into the combustion chamber. Mechanical injection is used in almost all types of diesel engines.

Springs are machine parts that provide an elastic connection between certain parts or subassemblies of a machine. When designing and calculating springs, the number of parameters that must be taken into account is greater than the number of relationships between them. Various methods are used to adopt some parameters and the rest are calculated.

Thus, the dimensioning of a spring can be carried out partly on the basis of such methods and partly by strength calculations, knowing the load that loads the spring, the type of stress, the shape of the spring.
This paper presents a method for the optimal design of the spring that is part of a MAN B&W AG engine injector. The theoretical part of the paper includes notions related to the injection system, types of injection pumps and their role, as well as types of injectors.

The case study of the paper includes the components of the MAN B&W AG engine injector, then starting from the initial input data, the helical cylindrical compression spring for the injector was designed and the necessary checks were made to determine whether the chosen spring meets the technical requirements for operating the injector.
Key words : Injector, diesel engine, compression spring, spring parameters.

1. INTRODUCTION

The diesel engine is an internal combustion engine in which the fuel is ignited by the high temperature created by the compression of the air necessary for combustion, and not by the use of an auxiliary device, such as the spark plug in the case of a spark ignition engine.

The engine works on the basis of the Diesel cycle.[1]

The engine was named after German engineer Rudolf Diesel at the suggestion of his wife, Martha Diesel, who in 1895 advised him: Nenn ihn doch einfach Dieselmotor! ("*Just call it the Diesel engine*!"), thus making it easier for Diesel to find a name for the engine, which he invented in 1892 and patented on February 23, 1893. Diesel intended his engine to run on a wide range of fuels, including coal dust. Diesel presented his invention running at the 1900 World's Fair on peanut oil.[2]

A vital component of diesel engines is the speed governor, mechanical or electronic, which regulates the engine speed by correctly dosing the injected diesel fuel. Unlike spark ignition (Otto) engines, the amount of air sucked in is not controlled, which leads to engine overspeed. Mechanical governors use different mechanisms depending on the load and speed. Modern, electronically controlled engine governors control fuel injection and limit engine speed via a central control unit that constantly receives signals from sensors, correctly dosing the amount of diesel fuel injected.[1]

Precise control of injection timing is the secret to reducing fuel consumption and pollutant emissions. Injection timing is measured in crankshaft rotation angles before top dead center. For example, if the central control unit initiates injection 10 degrees before top dead center, this is an injection timing of 10 degrees. The optimal injection timing is determined by the design, speed and load of the respective engine. [1]

2. HISTORICAL SHIELD OF THE DEVELOPMENT MECHANICALLY AND ELECTRONICALLY CONTROLLED INJECTION

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Older engines used a mechanical pump and a valve mechanism driven by the crankshaft, usually via a chain or timing belt. These engines used simple injectors, with a valve and spring, that opened/closed at a certain fuel pressure. The pump consisted of a cylinder that compressed the diesel fuel and a disc-shaped valve that rotated at half the speed of the crankshaft. The valve had a single opening on one side, for the pressurized fuel and another for each injector. As it rotated, the valve disc distributed a precise amount of high-pressure fuel to each injector. The injector valve was actuated by the pressure of the injected diesel fuel as long as the disc delivered fuel to the respective cylinder. The engine speed was controlled by a third disc that rotated only a few degrees and was operated by a lever. This disc controlled the opening through which the fuel passed, thus dosing the amount of diesel fuel injected.

Old diesel engines could be started in reverse, by mistake, although they operated inefficiently due to the misaligned ignition sequence. This was usually the result of starting the car in the wrong gear.

Modern engines have an injection pump that provides the necessary injection pressure. Each injector is electromagnetically actuated by a central control unit, which allows precise control of the injection depending on the speed and load, resulting in increased performance and low consumption. The simpler technical solution of the pump-injector assembly has led to the construction of more reliable and quieter engines.[1]

In the case of the indirect injection diesel engine, the diesel fuel is not injected directly into the combustion chamber, but into a pre-chamber where combustion is initiated and then extends into the main combustion chamber, driven by the turbulence created. The system allows for quiet operation, and, since combustion is favoured by turbulence, the injection pressure can be lower, so high rotation speeds are allowed (up to 4000 rpm), much more suitable for cars. The pre-chamber had the disadvantage of high heat losses, which had to be borne by the cooling system, and of low combustion efficiency, up to 5-10% lower than direct injection engines. Almost all engines had to have a cold start system, such as glow plugs. Indirect injection engines were widely used in the automotive and marine industries from the early 1950s until the 1980s, when direct injection progressed significantly.

Indirect injection engines are cheaper and easier to build for industries where emissions are not a priority. Even with new electronically controlled injection systems, indirect injection engines are slowly being replaced by direct injection engines, which are much more efficient.

3. INJENCTION PUMPS

3.1 Types. Role. Description

The injection pump is the organ that gives the fuel the high pressure it needs to enter the combustion chamber of the engine cylinder at high speed.

The technical conditions for injection pumps are the following:

→ The amount of fuel injected into each cylinder must be constant and the same for all cylinders, not being allowed to vary over time or from cylinder to cylinder, for the same speed and load

→ The injection advance angle must not vary over time, for the same speed, it must be adjustable, increasing with increasing speed

→ The injection pressure must be as constant as possible throughout the injection period

→ The flow rate must be easily adjustable, and, for the same adjustment position, the amount of fuel injected at each stroke of the pumping element must not vary with the pump speed

→ The power consumption for driving the injection pump must be as low as possible

Injection systems can be classified into two groups:

→ the pulse injection system, in which the pressure necessary to atomize the fuel and introduce it into the combustion chamber is produced only at the moment of injection and lasts as long as the injection lasts;

→ the pressure accumulation injection system, in which the injection pump maintains the pressure necessary for atomization, at a constant value, in an accumulator, and the injection of fuel into the combustion chamber is controlled by another organ (the injector, which, as will be shown, performs the function of a valve with controlled opening, the duration of the opening representing the duration of the injection).

Injection pumps that are part of injection equipment have a complex and varied role. First, to obtain optimal characteristics of the fuel jet injected into the engine cylinder, injection pumps must develop very high discharge (injection) pressures (300…1100 bar and sometimes higher – for example, in injector pumps). Second, injection pumps must allow dosing the amount of fuel per cycle in accordance with the operating mode, while ensuring uniformity of the fuel dose in all cylinders of multi-cylinder engines. Third, injection pumps must ensure optimal injection advance, limitation of injection duration and optimal injection characteristic.

Also, injection pumps must be reliable, durable, have simple construction, low cost, and as simple to operate as possible.

The essential problem of injection pumps is the achievement of high injection pressures, required by the need for fine atomization of the fuel. These pressures can only be provided by piston pumps. For these values of injection pressure (hundreds of bars), however, special

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problems arise in connection with the precision of execution of the piston and cylinder of the pump, as well as with the sealing of this pair of parts from the external environment. The only way to effectively seal is to reduce the clearance between the piston and the cylinder of the pump to values of 1.5…3 μm and to practice an execution with an increased piston length in relation to its diameter. This involves fine grinding operations, with extremely tight deviations in shape (from the quality of surface processing and from their mutual position), as well as lapping operations and "pairing" of the piston with the cylinder, which thus become non-interchangeable. [5]

3.2. Operating principle

To carry out the injection process, the pump piston performs a suction stroke and a discharge stroke. The piston is driven rigidly in the discharge stroke by a profiled cam. This has the main advantage of being able to choose the piston movement law so as to ensure the optimal injection characteristic and to satisfy the conditions for good fuel atomization. For this reason, the piston drive with a profiled cam is currently almost universally used. The piston can also be driven elastically in the discharge stroke by a spring. However, with elastic drive, the piston movement law cannot be controlled; instead, the injection process is removed from the influence of the speed, which also favours the uniformity of the injected fuel doses at low speeds. [4]

The schematic diagram of a piston injection pump is shown in Figure 1 below. The pumping element consists of the cylinder (1) and the piston (2). The fuel is sucked, through the suction valve (3), into the space (C), when the piston moves in the suction stroke under the action of the spring (4). The movement of the piston in the discharge stroke takes place under the action of the cam (5) and the tappet (6). The fuel in the space (C) is compressed and discharged at pressures of hundreds of bars to the space (R), through the discharge valve (7).

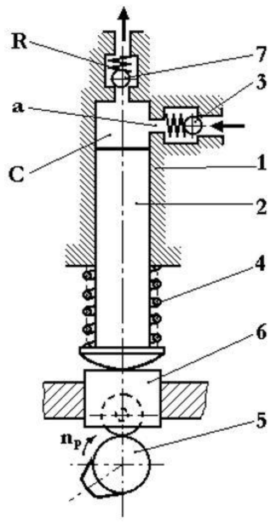

Figure 1 Schematic diagram of a piston injection pump [4]

The size of the discharged dose is established in accordance with the needs of the engine operating regime, by dividing the piston discharge stroke into a variable useful stroke s_u (Figure 2) and, most often, into two dead strokes s_m1 and s_m^2, one fixed and the other variable or both variable. The useful stroke s_u is usually placed in the maximum piston speed portion.

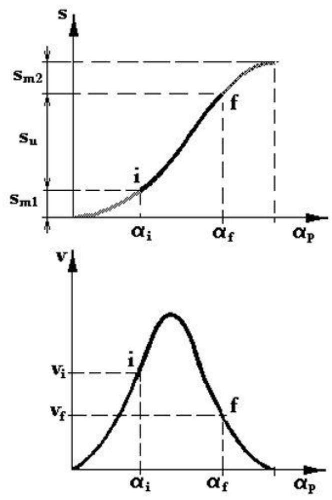

Figure 2 Engine operating mode [4]

In the form shown in figure 1., the piston injection pump discharges into space (R) the entire quantity of fuel sucked into space (C). To correlate the dose of fuel discharged with the engine operating mode, either the flow section (a) is externally controlled, or an axially movable variable profile cam is used.

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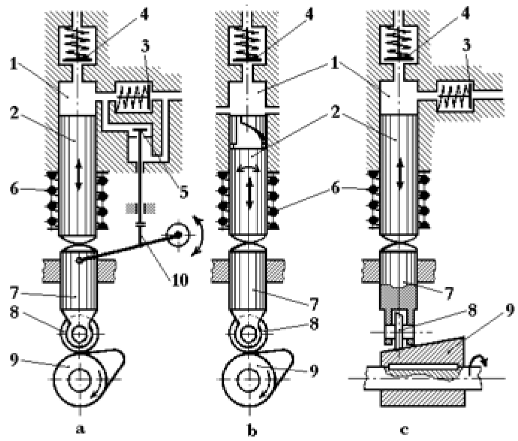

Figure 3 Types of valve pumps [4]
a-valve pump, with constant piston stroke
b-rotary plunger injection pump, with constant piston stroke
c-injection pump, with adjustable piston stroke

The composition and operation of the mentioned injection pumps (figure 3) is as follows: by rotating the cam (9) the roller (8) and the tappet (7) are activated, which determine the movement of the piston (2) and the pressing of the fuel in the cylinder (1). Through the discharge valve (4), the fuel pressed at the injection pressure level is sent to the injector. In the case of the injection pump in figure 3. a), the pressing is interrupted when the pressing valve (5) opens, operated by the regulating device (10).

Due to the potential energy of deformation of the spring (6), the piston (2) performs the filling stroke, the fuel enters the pump cylinder through the suction valve (3) in Figure 3. a) and c). In the piston-slide pump (figure 3. b), the suction valve (3) is missing, and the amount of fuel discharged is adjusted by rotating the piston (2). In the pump in Figure 3. c, the adjustment is made by axial movement of the cam (9).

In the case of the injection pumps in Figure 3. a) and b), only a fraction of the amount of fuel sucked into the cylinder (1) is discharged. These are pumps with invariable suction and partial discharge. Their main advantage is the discharge of fuel at high piston speeds. In this sense, the useful stroke s_u (figure 3. b) is placed between the dead strokes s_m1 and s_m2, where the speeds are reduced.

The pump in figure 3. c, is of the variable suction and total discharge type. The pump with a single discharge element and a rotary distributor, currently widespread in high-speed MACs used in road traction, is also built according to this principle of regulation. In these pumps, the fuel dose corresponding to each engine operating mode is sucked entirely into the suction space, after which it is completely discharged to the injector.

4. SPRINGS

4.1. Definition. Area of use

Springs are machine parts that create an elastic connection between certain parts or subassemblies of a machine. Due to their shape and the special mechanical characteristics of the materials they are made of, springs have the ability to deform under the action of an external force, taking over its mechanical work and storing it in the form of deformation energy. When the external load disappears, the stored energy is returned to the mechanical system of which the spring is a part.[6]

The fields of use of springs are varied, the most important being:

→ shock and vibration damping (in vehicle suspensions, elastic couplings, machinery foundations, etc.);

→ accumulation of energy that must be released gradually or in a short time (in spring clocks, internal combustion engine valve springs, etc.);

→ exerting a permanent, elastic force (in friction safety couplings, friction clutches, etc.);

→ regulating or limiting forces (in presses, control valves, etc.);

→ measuring forces and moments by using the dependence between load and deformation (dynamometric).

→ changing the natural frequencies of some machine parts.

4.2. Cylindrical compression coil springs

Figure 4 presents several solutions for helical compression springs, and Figure 5. presents the geometric elements of cylindrical helical compression springs with a round coil section.

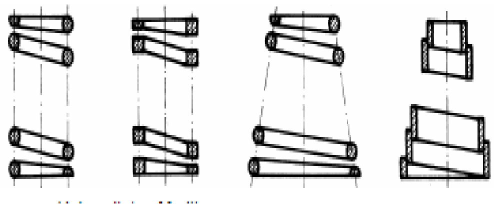

Figure 4 Types of compression coil springs [7]

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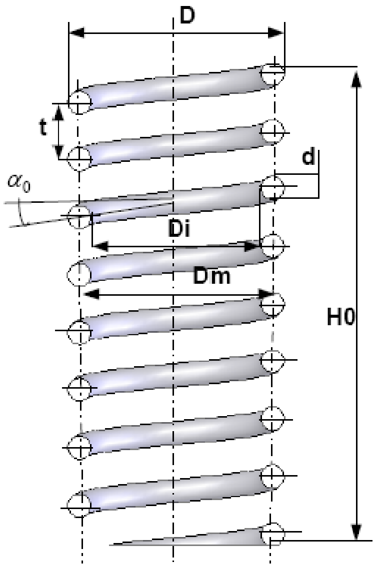

Figure 5 Geometric elements of cylindrical compression coil springs [7]

The terminology used for cylindrical compression helical springs with a round coil section (also applicable to a rectangular coil section) is:
d – diameter of the spring coil
D_i – inner diameter
D_m – average diameter
D – outer diameter
t – coil pitch
H_0 – length of the spring in the free state
α_0 – angle of inclination of the coil in the free state

5. OPTIMAL COMPRESSION SPRING DESIGN

5.1. Injector presentation

Compression springs are produced according to quality standards regulated by DIN 2095 in a wide variety of sizes.

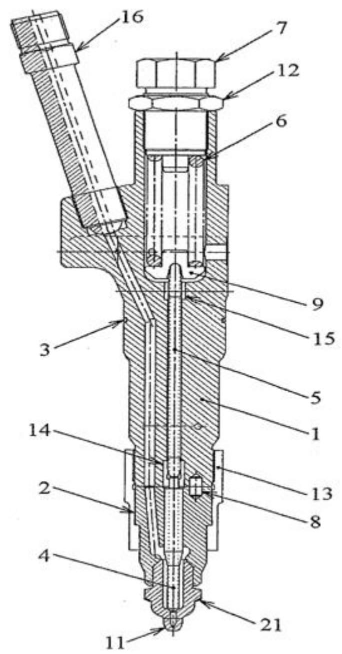

Figure 6 Injector section [8]
1-Injector body; 2-nozzle; 3- o-ring; 4-nozzle valve; 5- needle; 6-spring; 7-adjustment screw; 8-parallel pin; 9-spring lock component; 10-conical disk; 11-nozzle tip; 12-lock washer; 13-nut; 14, 15-spring guide, 16-duct; 17-bolt; 18-washer; 19-spring tube; 20-pine; 21- gasket

In this case, helical springs made of corrosion-protected stainless spring steel are used, cylindrical springs made of round wire with a constant diameter in a range of colours from dark grey to black.

The strength of the springs depends on several factors: period of use, elastic limit, load, etc.

During the operation of springs, residual stresses occur which place an additional load on the spring. Coil springs can lock if the elastic limit of the material is exceeded and plastic deformation can occur. Very long compression coil springs can buckle under higher loads.

Since it is impossible to take into account all real dynamic influences, the strength of the springs cannot be fully guaranteed.

To evaluate the properties of the spring, the "Spring characteristic" is taken into account, whose force F represents the dependence of the spring displacement. The ratio between the spring force and the spring stroke is called the spring constant. In cylindrical helical springs, a linear relationship is formed.

5.3. Initial calculation data

- Maximum load: $F_{max} = 1850\ N$
- Minimum load: $F_{min} = 860\ N$
- Working stroke: $s = 55\ mm$

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- Material density: $\rho = 7{,}87\ kg/mm^3$,
- Longitudinal modulus of elasticity: $E = 2{,}1 \cdot 10^5\ MPa$,
- Transverse modulus of elasticity: $G = 0{,}81 \cdot 10^5\ MPa$,
- Bearing coefficient: $\nu = 0{,}5$,
- Spring wire diameter: $d = 5\ mm$,
- Arc index: $i = 8$
- Coefficient of the distance between the turns: $k = 0{,}14$

Based on these data, two springs with the following characteristics are chosen from the standard:

Table 1. Characteristics of the spring

Symbol	Label	Unit of measurement	Spring 1	Spring 2
d	Spring wire diameter	mm	5	5
D_e	Outer diameter	mm	45	37
D_m	Average diameter	mm	40	32
D_i	Inner diameter	mm	35	27
L_0	Unstressed spring length	mm	95.5	110
n	Total number of turns		5.5	8.5
D_d	Guide mandrel diameter	mm	34.2	26.4
D_h	Guide bushing diameter	mm	46.3	38
R	Spring constant	N/mm	15.85	20.03
s	Bow stroke	mm	54.2	52.5
L_n	Permissible length related to force	mm	41.3	57.5
F_n	Spring force relative to length	N	859	1051.2
M	Spring weight	g	147.14	164.77

We calculate the weight of the two springs

$$M_1 = l_1 \cdot \frac{\pi \cdot d^2}{4} \cdot \rho = 147{,}14\ g \qquad (1)$$

$$M_2 = l_1 \cdot \frac{\pi \cdot d^2}{4} \cdot \rho = 164{,}77\ g \qquad (2)$$

One of the criteria for choosing the spring is: the lowest possible weight

Due to this criterion we will choose Spring 1 for the injector.

5.4. Parameters calculation

- Spring stiffness

$$c = \frac{F_{max} - F_{min}}{s} = \frac{1850 - 860}{55} = 18\ N/mm \qquad (3)$$

- Minimum arrow

$$f_{min} = \frac{F_{min}}{c} = \frac{860}{18} = 47{,}8\ mm \qquad (4)$$

- Maximum arrow

$$f_{max} = \frac{F_{max}}{c} = \frac{1850}{18} = 102{,}8\ mm \qquad (5)$$

- Minumum winding diameter

$$D_m = i \cdot d = 8 \cdot 5 = 40\ mm \qquad (6)$$

- Outer diameter

$$D_e = D_m + d = 40 + 5 = 45\ mm \qquad (7)$$

- Inner diameter

$$D_i = D_m - d = 40 - 5 = 35\ m \qquad (8)$$

- Number of active turns

$$n_c = \frac{G \cdot d}{8 \cdot c \cdot i^3} = \frac{0{,}81 \cdot 10^5 \cdot 5}{8 \cdot 18 \cdot 8^3} = \frac{81000 \cdot 5}{73728} = 5{,}5 \qquad (9)$$

- Number of support turns

$$n_r = 1{,}5 \text{ pentru } n_c < 7$$

$$n_r = 2{,}5 \text{ pentru } n_c > 7 \qquad (10)$$

- Total number of turns

$$n_t = n_c + n_r = 5{,}5 + 1{,}5 = 7 \qquad (11)$$

- Length of locked spring

$$H_b = n_t \cdot d = 7 \cdot 5 = 35\ mm \qquad (12)$$

- Minimum distance between active turns

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$$S_a = \left(0{,}0015 \cdot \frac{D_m^2}{d} + 0{,}1 \cdot d\right) \cdot n_s = \left(0{,}0015 \cdot \frac{40^2}{5} + 0{,}1 \cdot 5\right) \cdot 5{,}5 = 5{,}39\ mm \quad (13)$$

- Average distance between turns

$$m = \frac{L_0 - d}{n_s} = \frac{95{,}5 - 5}{5{,}5} = \frac{90{,}5}{5{,}5} = 16{,}45\ mm \quad (14)$$

- Unstressed spring pitch

$$t = d + \frac{f_{max}}{n_c} + d \cdot k = 5 + \frac{102{,}8}{5{,}5} + 5 \cdot 0{,}14 = 5 + 18{,}69 + 0{,}7 = 24{,}39\ \ mm \quad (15)$$

- Unstressed spring pitch

$$H_0 = H_b + n_c \cdot (t - d) = 35 + 5{,}5 \cdot (24{,}39 - 5) = 141{,}65\ mm \quad (16)$$

- Length of the assembled spring

$$H_{min} = H_0 - f_{min} = 141{,}65 - 47{,}8 = 93{,}85\ mm \quad (17)$$

- Spring length at maximum load

$$H_{max} = H_0 - f_{max} = 141{,}65 - 102{,}8 = 38{,}85\ mm \quad (18)$$

- The angle of inclination of the unstressed spring coil

$$\alpha_0 = \frac{180}{\pi} \cdot \alpha \cdot tan\left(\frac{t}{\pi \cdot D_m}\right) = 8{,}5\ grade \quad (19)$$

- Length of spring wire

$$l_s = \frac{\pi \cdot D_m \cdot n_t}{cos(\alpha_0)} = \frac{3{,}14 \cdot 40 \cdot 7}{\cos 8{,}5^\circ} = \frac{879{,}2}{0{,}989} = 888{,}98\ mm \quad (20)$$

- Arrow at spring lock

$$f_b = min(H_0 - H_b) = \min(141{,}65 - 35) = 106{,}65\ mm \quad (21)$$

- Arrow at spring lock

$$F_b = c \cdot f_b = 18 \cdot 106{,}65 = 1919{,}7\ N \quad (22)$$

- Slenderness coefficient

$$\lambda = \frac{H_0}{D_m} = \frac{141{,}65}{40} = 3{,}54 \quad (23)$$

- Theoretical buckling deflection coefficient

$$c_f = \frac{0{,}5}{1 - \frac{G}{E}} = \frac{0{,}5}{1 - \frac{0{,}81 \cdot 10^5}{2{,}1 \cdot 10^5}} = 8{,}13 \quad (24)$$

- Critical slenderness ratio

$$\lambda_c = \frac{\sqrt{c_f}}{v} = \frac{\sqrt{8{,}13}}{0{,}5} = \frac{2{,}85}{0{,}5} = 5{,}7 \quad (25)$$

- Natural operating frequency of the spring

$$f = \frac{3{,}5 \cdot 10^5 \cdot d}{n_c \cdot D_m^2} = \frac{3{,}5 \cdot 10^5 \cdot 5}{5{,}5 \cdot 40^2} = \frac{1750000}{8800} = 198{,}86\ Hz \quad (26)$$

6. CONCLUSIONS

In high-power marine engines, heavy fuel injectors are used that do not differ essentially from diesel injectors. Due to the higher thermal regime of the atomizer, heavy fuel injectors must be cooled. In this regard, an additional channel is made in both the injector body and the atomizer, in which oil or water circulates. Water (distilled or treated, to avoid corrosion and deposits) is preferred for safety reasons. The cooling circuit of the injectors must be independent of the engine cooling circuit.

The injector body is made of OLC of cementation or improvement quality, the semi-finished product being obtained by die forging. The seating surface is carbonitrided and hardened to avoid deformation and ensure proper sealing.

The injector spring is required to have precise characteristics, and stabilization treatments are necessary to ensure that its qualities are maintained over time.

Based on the initial design data, two springs were chosen.

The weight of the two springs was calculated. In the end, one of the main criteria for choosing the spring was taken into account: the lowest possible weight, so Spring 1 was chosen.

The calculation of the parameters was performed and then it was checked whether the spring was chosen correctly.

The condition that is imposed

$$\boldsymbol{\lambda < \lambda_c => 3{,}54 < 5{,}7}$$

=> ***The condition is being checked.***

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When overloaded, cases encountered in practice, the equivalent stresses do not exceed the breaking strength of the wire, so the spring resists and does not fail during operation.

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Methodology: T.M.;

Project administration: T.M.;

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Supervision: T.M.;

Validation: T.M.;

Visualization: T.M.;

Writing – original draft: T.M.;

Writing – review & editing: T.M..

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