NUMERICAL SIMULATION AND MODELING OF HYDRO-MECHANIC CONVERTERS

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Convertoarele mecanohidraulice sunt componente ale echipamentelor de comandă şi control din structura sistemelor de acţionare. În lucrare sunt prezentate rezultatele cercetărilor teoretice privind modelarea matematică a convertoarelor mecanohidraulice şi simulare numerică a curgerilor prin orificiile variabile de tip duză-paletă.. Modelul matematic prezentat în lucrare poate fi utilizat la sinteza convertoarelor în vederea optimizării funcţionării acestora. De asemenea modelele de simulare a curgerilor prin ansamblul duză-paletă prezentate în lucrare pun în evidentă distribuţia de presiuni şi spectrele de viteză pentru diferite configuraţii ale ansamblului şi pot fi utilizate pentru oprimizarea dimensionala a convertoarelor mecanohidraulice.

The hydro-mechanic converters are components of action and control equipments from the structure of the power systems. This paper presents the results of theoretical research regarding the mathematical simulation of hydro-mechanic converters and the numerical simulation of the flow through some variable nozzle-flapper orifice type. The mathematical model presented in this paper can be used in converter synthesis in order to optimize their operation. Also the simulation models of the flow through the nozzle-flapper assembly presented mark out the pressure and velocity distribution for different types of assembly configurations and can be used to dimension optimization for the hydro-mechanic converters.

Keyword: hydro-mechanic, converter, nozzle-flapper, simulation, modeling

1. Introduction

The hydro-mechanic converters are components of action and control from the structure of fluid power systems. These ones are converting a mechanical input, displacement or force, into a hydraulic one, flow or pressure. There are

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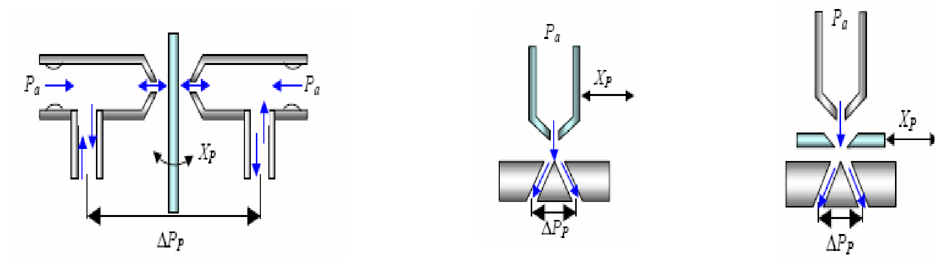
many solutions to realize this convertor, but the most used are the nozzle-flapper type, the convertors with oscillatory jet and the convertors with deflecting jet (fig.1). From technological point of view the most convenient are the hydro-mechanical nozzle-flapper type convertors used in a differential configuration. These are relative simple to realize, the only difficulty being the necessity of making identical pairs of nozzles from hydraulic point of view; $Q = f(p)$.



a b c

Fig.1. Hydro-mechanic converters.
a- nozzle-flapper convertors, b- convertors with oscillatory jet, c- convertors with deflecting jet

This difficulty is generally solved by manufacturing a set of nozzles, experimental determination of their hydraulic characteristics and then choosing a pair of nozzles that have relative the same characteristic. The major disadvantage of this type of convertors is the great sensibility for the oil contamination that can cause an "active deficiency". Total obstruction of an orifice causes an uncontrollable pressure difference and also bringing the execution element into an extreme position.

Manufacturing the convertors with oscillatory jet is relative difficult because the pipe that generates the jet (the injector) must make oscillatory movements around a fixed point. The advantage of this convertor type is conferred by its reduced sensibility at oil contamination and by the fact that generates a "passive deficiency". Injector obstruction causes the pressure difference to be canceled and bringing the execution element into a neutral position.

The convertors with deflecting jet are easier to manufacture than the convertors with oscillatory jet and have the same behavior when the injector is obstructioned.

2. Mathematical simulation of nozzle-blade convertor

The hydro-mechanical nozzle-flapper convertor, used in a differential configuration has very good results when functioning in the structure of the electro-hydraulic amplifier of "servovalve" type and is named in generally hydraulic pre-amplifier. The convertor is made out of two fix orifices (fix throttles), two nozzles and one common flapper that can move in front of the nozzles. The nozzles-flapper assembly creates a pairs of variable orifices (variable throttles) that work in opposition. To improve the hydro-mechanical convertor performances the pressure from its returning circuit has a value of near 35 bar. The convertor performances and attitude in static and dynamic regimes can be determined theoretical and experimental. Using the notation from figure 2 we can deduce the convertor mathematical model, writing the various relations that characterizes its functioning.

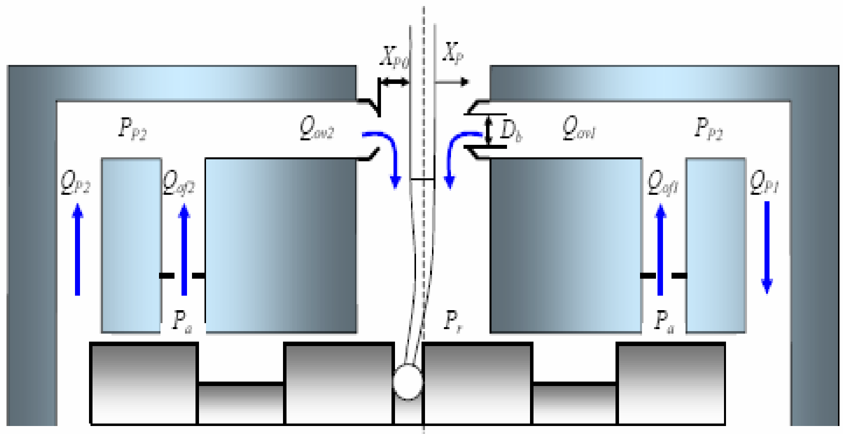


Fig. 2. Principial scheme of a hydraulic nozzel-flapper pre-amplifier

2.1. Flow characteristic of a fix orifices

The relations used to determine the oil flow that passes through the two fix throttles of the hydro-mechanical convertor are:

$$Q_{of1} = C_{qf} \cdot \frac{\pi \cdot D_{of}^2}{4} \cdot \sqrt{\frac{2}{\rho} \cdot (P_a - P_{P1})} \tag{1}$$

$$Q_{of2} = C_{qf} \cdot \frac{\pi \cdot D_{of}^2}{4} \cdot \sqrt{\frac{2}{\rho} \cdot (P_a - P_{P2})} \tag{2}$$

where:

Q_{of1}, Q_{of2} - the flow through the fix orifices 1 and 2 measured in (m^3/s);

C_{qf} - flow coefficient of fix orifices;
D_{of} - the inside diameter of the fix orifices measured in (m);
P_{P1}, P_{P2} - the pressure from the convertor operating chamber measured in (Pa);
P_a - inlet pressure measured in (Pa);
ρ - oil density measured in (kg/m^3).

2.2. *Flow characteristic of a variable throttle*

The relations used to determine the oil flow that passes through the two variables nozzle-blade throttles are:

$$Q_{ov1} = C_{qv} \cdot \pi \cdot D_b \cdot \left(X_{P0} - X_P\right) \cdot \sqrt{\frac{2}{\rho} \cdot \left(P_{P1} - P_r\right)} \tag{3}$$

$$Q_{ov2} = C_{qv} \cdot \pi \cdot D_b \cdot \left(X_{P0} + X_P\right) \cdot \sqrt{\frac{2}{\rho} \cdot \left(P_{P2} - P_r\right)} \tag{4}$$

where:
Q_{ov1}, Q_{ov2} - the flow through the variable orifices 1 and 2 measured in (m^3/s);
C_{qv} - flow coefficient of variable orifices;
D_b - the inside diameter of the nozzle measured in (m);
X_{P0} - the distance between the nozzle and the blade placed in the neutral position (medium position), measured in (m);
X_P - blade displacement relative to neutral position, measured in (m).

2.3. *Continuity equation*

The continuity equation corresponding to the two hydraulic circuits of the hydraulic convertor in stationary regime are:

$$Q_{P1} = Q_{of1} - Q_{ov1} = C_{qf} \cdot \frac{\pi \cdot D_{of}^2}{4} \sqrt{\frac{2}{\rho} \cdot \left(P_a - P_{P1}\right)} - \ldots$$
$$\ldots - C_{qv} \cdot \pi \cdot D_b \cdot \left(X_{P0} - X_P\right) \cdot \sqrt{\frac{2}{\rho} \cdot \left(P_{P1} - P_r\right)} \tag{5}$$

$$Q_{P2} = Q_{ov2} - Q_{of2} = C_{qv} \cdot \pi \cdot D_b \cdot \left(X_{P0} + X_P\right) \cdot \sqrt{\frac{2}{\rho} \cdot \left(P_{P2} - P_r\right)} - \ldots$$

$$\ldots. - C_{qf} \cdot \frac{\pi \cdot D_{of}^2}{4} \sqrt{\frac{2}{\rho} . \left(P_a - P_{P2}\right)} \tag{6}$$

If we accept the hypothesis that the command pressures P_{P1} and P_{P2} have a symmetrical evolution versus the pressure P_a and P_r semi sum and note with:

$$\Delta P_P = P_{P1} - P_{P2} \tag{7}$$

the differential command pressure of the pre-amplifier, then we can write

$$P_{P1} = \frac{P_a + P_r}{2} + \frac{\Delta P_P}{2} \tag{8}$$

$$P_{P2} = \frac{P_a + P_r}{2} - \frac{\Delta P_P}{2} \tag{9}$$

If we note with:

$$Q_P = \frac{Q_{P1} + Q_{P2}}{2} \tag{10}$$

the pre-amplifier command flow, from relations (5), (6), (7), (8) and (9) results:

$$Q_p = \sqrt{\left(P_a - P_r\right) - \Delta P_P} \cdot \left(\frac{k_1}{2} + \frac{k_2}{2} \cdot \left(X_{P0} + X_P\right)\right) - \ldots$$

$$\ldots - \sqrt{\left(P_a - P_r\right) + \Delta P_p} \cdot \left(\frac{k_1}{2} + \frac{k_2}{2}\left(X_{P0} - X_P\right)\right) \tag{11}$$

where:

$k_1 = C_{qf} \cdot \frac{\pi \cdot D_{of}^2}{4} \cdot \sqrt{\frac{1}{\rho}}$ - is a specific coefficient of the fix orifice $\left(m^2 / \sqrt{Kg / m^3}\right)$;

$k_2 = C_{qv} \cdot \pi \cdot D_b \cdot \sqrt{\dfrac{1}{\rho}}$ - is a specific coefficient of the variable orifice $\left(m^2 / \sqrt{Kg / m^3}\right)$;

Linearizing the equation (11), by developing in a Taylor series around the functioning point Q_{P0}, X_{P0}, ΔP_{P0} leads us to linear equation

$$\Delta Q_P = K_{QXp} \cdot X_P + K_{QPp} \cdot \Delta P_P \tag{13}$$

where:

$$K_{QXp} = \frac{\partial Q_P}{\partial X_P} = \frac{k_2}{2} \cdot \left(\sqrt{(P_a - P_r) - \Delta P_P} + \sqrt{(P_a - P_r) + \Delta P_P} \right) \tag{14}$$

$$K_{QPp} = \frac{\partial Q_P}{\partial \Delta P_P} = -\frac{\left(\dfrac{k_1}{2} + \dfrac{k_2}{2} \cdot (X_{P0} + X_P) \right)}{2 \cdot \sqrt{(P_a - P_r) - \Delta P_P}} - \frac{\left(\dfrac{k_1}{2} + \dfrac{k_2}{2} \cdot (X_{P0} - X_P) \right)}{2 \cdot \sqrt{(P_a - P_r) + \Delta P_p}} \tag{15}$$

For $X_P = 0$ and $\Delta P_P = 0$ the coefficient have the following values

$$K_{QXp} = \left. \frac{\partial Q_P}{\partial X_P} \right|_{\Delta Pp = 0} = k_2 \cdot \sqrt{P_a - P_r} \tag{16}$$

$$K_{QPp} = \left. \frac{\partial Q_P}{\partial \Delta P_P} \right|_{X_P = 0, \Delta P_P = 0} = -\frac{k_1 + k_2 \cdot X_{P0}}{2 \cdot \sqrt{P_a - P_r}} \tag{17}$$

When functioning in a dynamic regime the continuity equation corresponding to the pre-amplifier operating chamber has the following form:

$$\Delta Q_P = \frac{V_p}{2 \cdot \beta} \cdot \Delta \dot{P}_P + S_t \cdot \dot{X}_t \tag{18}$$

where:

V_p - operating chamber volume (m^3);

β - oil compressibility coefficient (Pa);

X_t - slide valve displacement (m);

S_t - slide valve area (m^2);

Also the displacement equation of the servo-valve slide has the form:

$$M_t \cdot \ddot{X}_t = F_{pt} - F_{hr} - F_{hvt} - F_{ra} \tag{19}$$

where

M_t -slide valve weight (kg);

$F_{pt} = \Delta P_P \cdot S_t$ - the command pressure force applied on the slide valve (N);

$F_{ht} = F(Q)$ - hydraulic force developed by the fluid jet applied by the pressure (N);

$F_{hvt} = f_{vt} \cdot \dot{X}_t$ - viscous friction force at the slide valve level (N);

$F_{ra} = K_{ra} \cdot X_t$ - reaction force made by the feedback bar over the slide valve (N).

The theoretic flow characteristic of hydro-mechanical convertor has a linearity area around the neutral position of the blade, a nonlinearity area and a saturation one for the extreme positions of the blade and it is influenced by the differential command pressure (figure 3).

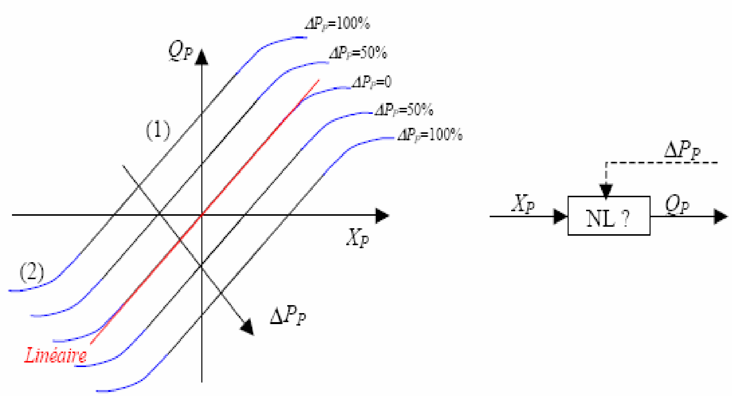


Fig. 3. Static characteristic of a hydraulic pre-amplifier.

3. Numerical simulations of the flow through the hydro-mechanical convertor.

The numerical simulation of the flow through the preamplifier was made in ANSYS CFX. The geometrical dimensions of nozzle-flipper assembly were considered in usual limits, which can be found in the specialty literature and in the technical card of servo valves. The fix orifice (hrottle) was considered to have a diameter of $\phi = 0{,}5\,mm$ and the variable orifice (throttle) was considered to have a diameter of $\phi = 1\,mm$. The simulations were made for three different values of the flipper position: 0.03 mm, 0.05 mm, 0.07 mm. There were considered two types of slots: a slot with a straight edge and one with an edge at 45˚.
With this simulation we wanted to mark out the pressure and velocity spectrums for different configurations of nozzle-flipper assembly. To reduce the calculus effort the simulations were made for a simple nozzle-flipper assembly (simple hydraulic potentiometer). Also the simulations were made considering a symmetrical fluid domain, being presented only half of the fluid domain.
The fluid domain discretization was made with ANSYS Workbench, the mesh being a tetrahedral one, creating an average number of cells of 550,000 for each case. In the symmetry plane a smoother discretization was made, the dimensions of the elements edge being in the field of 0.1…0.5 mm and in the slot area being in the field of 0.002…0.004 mm.
In the specialty literature the pressure and velocity variation through the nozzle-flapper assembly is presented in figure 4. The simulations results are presented in the figures 5…13.

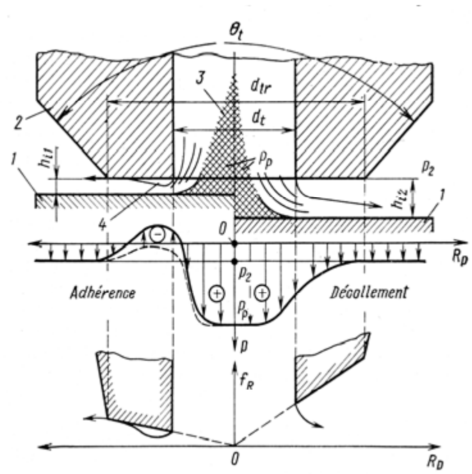


Fig.4. Pressure and velocity variation through the nozzle-flapper assembly

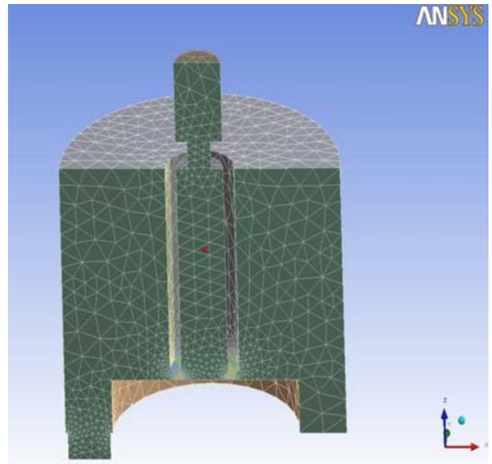


Fig.5 Discretization of the nozzle-flapper assembly

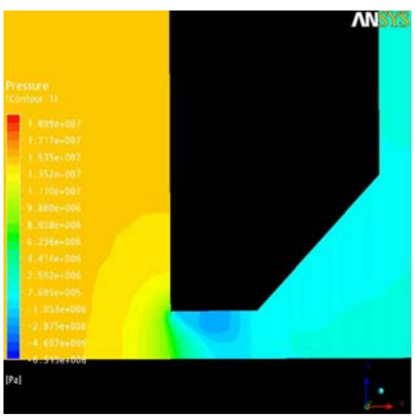


Fig.6. Pressure spectrum through the nozzle-flapper assembly. Detail. opening 0.05 mm

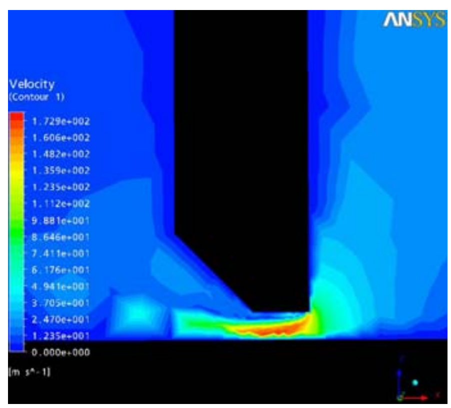


Fig.7. Velocity spectrum through the nozzle-flapper assembly. Slot with edge at 45˚. Detail. Opening 0.05 mm.

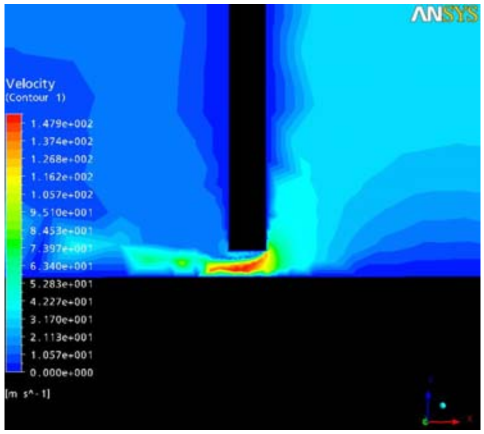


Fig.8 Velocity spectrum through the nozzle-flipper assembly. Slot with edge at 90˚. Detail. Opening 0.07 mm.

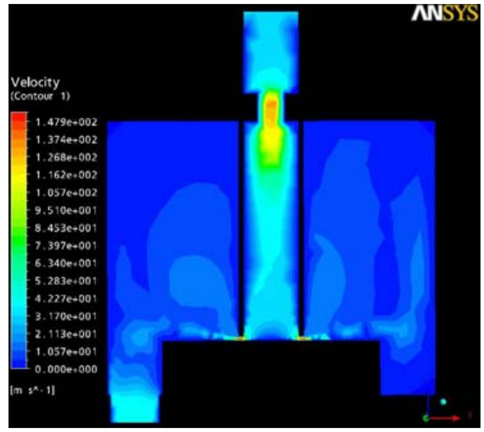


Fig.9. Velocity spectrum through the nozzle-flipper assembly. Slot with edge at 90˚. Detail. Opening 0.07 mm.

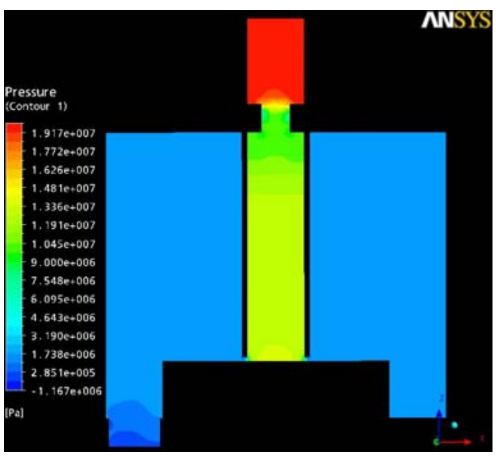


Fig.10. Pressure spectrum through the nozzle-flapper assembly. Slot with edge at 90˚. Opening 0.07 mm

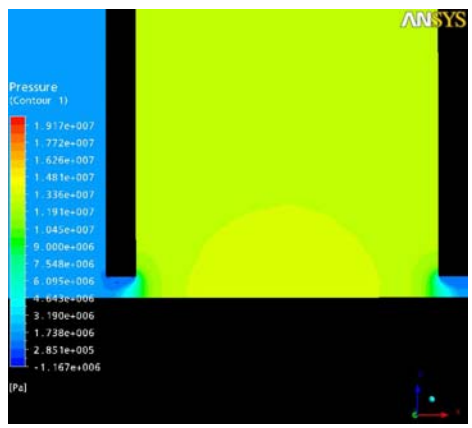


Fig.11. Pressure spectrum through the nozzle-flapper assembly. Slot with edge at 90˚.Detail. Opening 0.07 mm

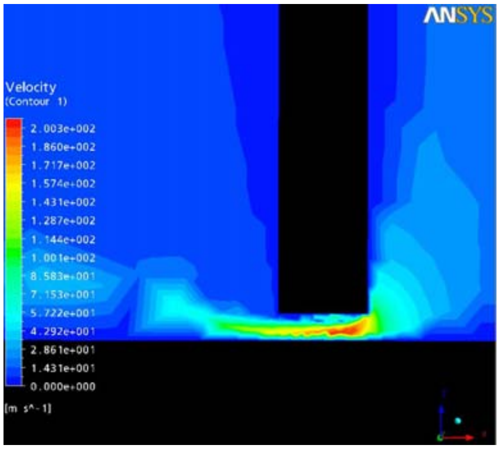


Fig.12.Velocity spectrum through the nozzle-flipper assembly. Slot with edge at 90˚. Detail. Opening 0.03 mm.

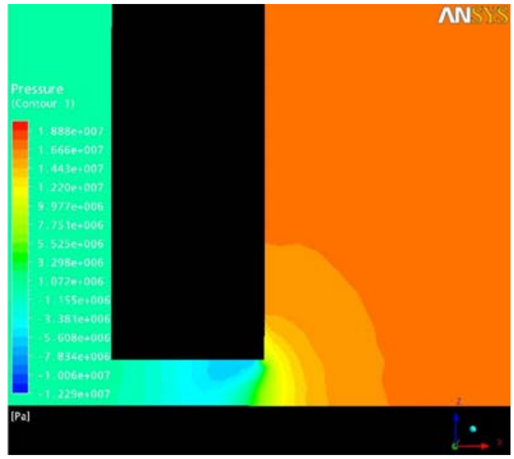


Fig.13. Pressure spectrum through the nozzle-flapper assembly. Slot with edge at 90˚.Detail. Opening 0.03 mm

3. Conclusions

- Mathematical modeling and numerical simulation of hydraulic equipments operation represents a base instrument for analyze and synthesis of the hydraulic action systems. The results obtained with the help of the numerical simulation and from experimental research can become data base that have direct implications over the costs and the duration to manufacture a product.
- The numerical simulations through the hydro-mechanical convertor of nozzle-flipper type were made with ANSYS CFX. The shape and the geometrical dimensions of the equipments were choose in such way, to respect as much as possible the real shape and dimensions of the convertors and also to allow a simple modelling without affecting the quantity and quality of the parameters flow through the nozzle.
-For flow domain discretization was used tetrahedral elements. Because in some area the pressure gradients have a great value it is necessary to use volume elements with a parallelepiped shape to improve the mesh quality.

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