Optimization of the intake system of Z-Engine

Technical report

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Content

1. Introduction

The target of the current study is an investigation of intake process of the z-engine to make it possible the improvement of engine volumetric efficiency at maximum RPM operating points by optimization of the non steady gas flow in the intake port of the engine having very much density of fresh charge and very short intake period.

At the intake valve opening the low-pressure wave starts to be moving from the valve gap up to the intake port to the receiver. If the receiver has sufficient volume than this wave reflects from opened port mouth as high-pressure wave and returns to the popped valve. If the valve is opened at the moment when the wave arrivals then an additional portion of the fresh charge enters into the cylinder. A rational combination of intake port length and diameter should be done to use this phenomenon. A previous research has shown: a problem of insufficient cylinder fresh charge is actual at maximum engine speed. So the current research was done for engine maximum power operating mode.

Table 1 - Engine specification

Cylinder bore	72 mm
Piston stroke	70 mm
RPM	3600
CR	15,5
Average intake receiver pressure	19,2 bar
Average intake receiver temperature	344,4 K
Intake maximum effective flow area (2 valves)	4,87 cm^2
Intake duration	19° CA

In the current research there was used author's program "OneDimFlow" developed in Bauman Moscow State Technical University. The program realizes

Godunov’s method for 1D flow of compressible gas with account the friction and heat exchange. To account the friction the approach of Darcy – Weisbach is used. The heat exchange with walls is calculated with Newton equation where heat transfer coefficient is calculated with phenomenological expression as function of Reynolds number.

To provide the engine parameters estimation the code includes modules:

- simulation of flow in the valve (prof. Yuri Grishin);
- simulation of flow in the junctions (prof. Yuri Grishin);
- 0D thermodynamic engine cylinder model.

The traditional system of equations being used in CFD for viscous, compressible gas is:

$$\frac{\partial r}{\partial t} + div(r \cdot \bar{u}) = 0$$

$$\frac{\partial ru}{\partial t} + div(ru \cdot \bar{u}) = -\frac{\partial p}{\partial x} + div(m \cdot grad(u)) + S_{Mx} \text{ a}$$

$$\frac{\partial rv}{\partial t} + div(rv \cdot \bar{u}) = -\frac{\partial p}{\partial y} + div(m \cdot grad(v)) + S_{My}$$

$$\frac{\partial rw}{\partial t} + div(rw \cdot \bar{u}) = -\frac{\partial p}{\partial z} + div(m \cdot grad(w)) + S_{Mz}$$

$$\frac{\partial re}{\partial t} + div(re \cdot \bar{u}) = -p \cdot div(\bar{u}) + div(k \cdot grad(T)) + \Phi + S_e$$

The method of decay of an arbitrary discontinuity uses same base equations of state, mass, pulse and energy conservation but in integral form:

$$\oint (r dx - rvdt) = 0\,; \qquad (1)$$

$$\oint \left[rvdx - \left(p + rv^2\right)dt\right] = -\iint l_T \frac{rv|v|}{2D} dxdt\,; \qquad (2)$$

$$\oint \left[redx - rv\left(e + \frac{p}{r}\right)dt \right] = -\iint 4a_w \frac{T - T_c}{D} dxdt\,. \qquad (3)$$

where: x, t are coordinate and time, p, ρ, T, v, R, c_p, c_v, $e = u+v^2/2$, λ_m, α_w, T_c, D are pressure, density, temperature, velocity of flow, gas constant, isobaric and isochoric heat capacity, specific energy (sum of specific internal energy $u = c_vT$, and kinetic energy $v^2/2$), friction coefficient, heat transfer coefficient between gas and port wall, port wall temperature and hydraulic diameter of port.

In this study there was used a model of thermodynamic ideal and caloric perfect gas ($p = r\ R\ T$, c_p = const, c_v = const, $R = c_p - c_v$).

A smooth distribution of parameters along the port length (which takes place in practice) is replaced by a discontinuous distribution (each border has arbitrary gap of parameters). To write the finite-difference relation for the integral conservation laws (1-3), the parameters at the boundaries between cells are approximated by linearized ratios of the decay of an arbitrary discontinuity. The parameters on the cell side *i+1/2* placed between cell number *i* and cell number *i+1* are calculated with expressions:

$$p_{i+1/2} = \frac{p_i^n + p_{i+1}^n}{2} + (ar)_c \frac{v_i^n - v_{i+1}^n}{2}; \tag{4}$$

$$v_{i+1/2} = \frac{v_i^n + v_{i+1}^n}{2} + \frac{p_i^n - p_{i+1}^n}{2(ar)_c}. \tag{5}$$

where: $$(ar)_c = \frac{a_{i+1}^n r_{i+1}^n + a_i^n r_i^n}{2}. \tag{6}$$

To define the flow density the formulas of linear approaches of adiabatic relations are used:

$$r_{i+1/2} = r_i^n \left[1 + \frac{1}{k}\left(\frac{p_{i+1/2}}{p_i^n} - 1\right)\right] \quad \text{if} \quad v_{i+1/2} \geq 0; \tag{7}$$

$$r_{i+1/2} = r_{i+1}^n \left[1 + \frac{1}{k}\left(\frac{p_{i+1/2}}{p_{i+1}^n} - 1\right)\right] \quad \text{if} \quad v_{i+1/2} < 0, \tag{8}$$

After definition of cell borders parameters the in-cell parameters on the next time step are defined from expressions:

$$M_i^{n+1} = M_i^n + M_{i-1/2} - M_{i+1/2}; \tag{9}$$

$$K_i^{n+1} = K_i^n + (K + I)_{i-1/2} - (K + I)_{i+1/2}; \tag{10}$$

$$E_i^{n+1} = E_i^n + (E + L)_{i-1/2} - (E + L)_{i+1/2}. \tag{11}$$

where: $M_{i-1/2} = (\rho v)_{i-1/2} \Delta t$; $K_{i-1/2} = (Mv)_{i-1/2}$; $E_{i-1/2} = (Me)_{i-1/2}$;

$I_{i-1/2} = p_{i-1/2} \Delta t$; $L_{i-1/2} = (pv)_{i-1/2} \Delta t$.

The parameters on the right side of cell *i* having index *i+1/2* are calculated in a similar. The described above method is easy, fast and stable at computing. It allows make computations at high Courant number (even up to 0.5 – 07).

2. Boundary and Initial Conditions

The engine scheme being used in this research is presented in fig. 1. The scheme includes the single cylinder, the intake popped valve and individual intake port having length *L* and diameter *d*. The reason why the single cylinder concept was used is the necessity to use large volume intake receiver to improve effect of reflection of pressure waves. On the other hand, the large receiver reduces the influence of the cylinders on each other.

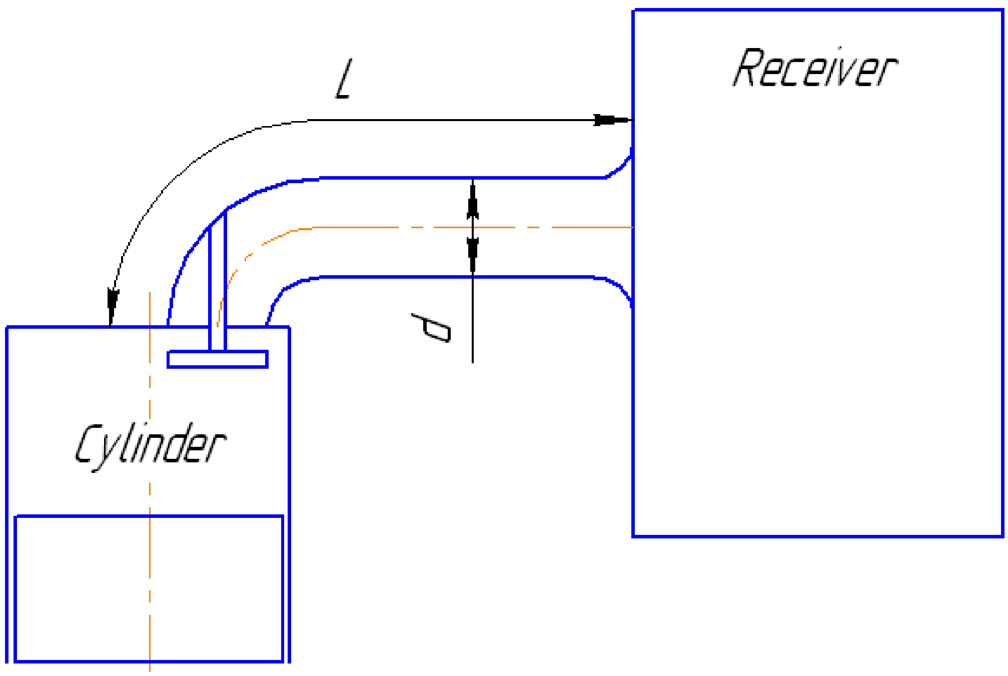

Fig.1 The engine scheme being used for analysis

The gas parameters in the cylinder at the opening of the intake valve are defined by thermodynamic engine simulation with program Diesel-RK [1]. At the beginning of intake (67^0 CA BTDC) the in-cylinder pressure and temperature are 6.05 bar and 929 K. (It is the full power operating mode @ 3600 RPM.)

The gas parameters in the inlet receiver were taken as constant and equal: 19.2 bar and 344.4 K.

The initial state of the gas in the port was taken as equal to the parameters in the receiver. 3 engine intake cycles was simulated at every point to provide a periodicity of processes in the intake system.

3. Effect of the port length on the volumetric efficiency

During the parametrical research of cylinder induction the port length *L* and diameter *d* were varied as it shown in the Table 2.

Table 2 – Variations of port parameters

Parameter	Symbol	Range of variation
Port length, mm	*L*	50 – 600
Port diameter, mm	*d*	30 – 80

Effect of port length on the volumetric efficiency of the engine is shown in figure 2.

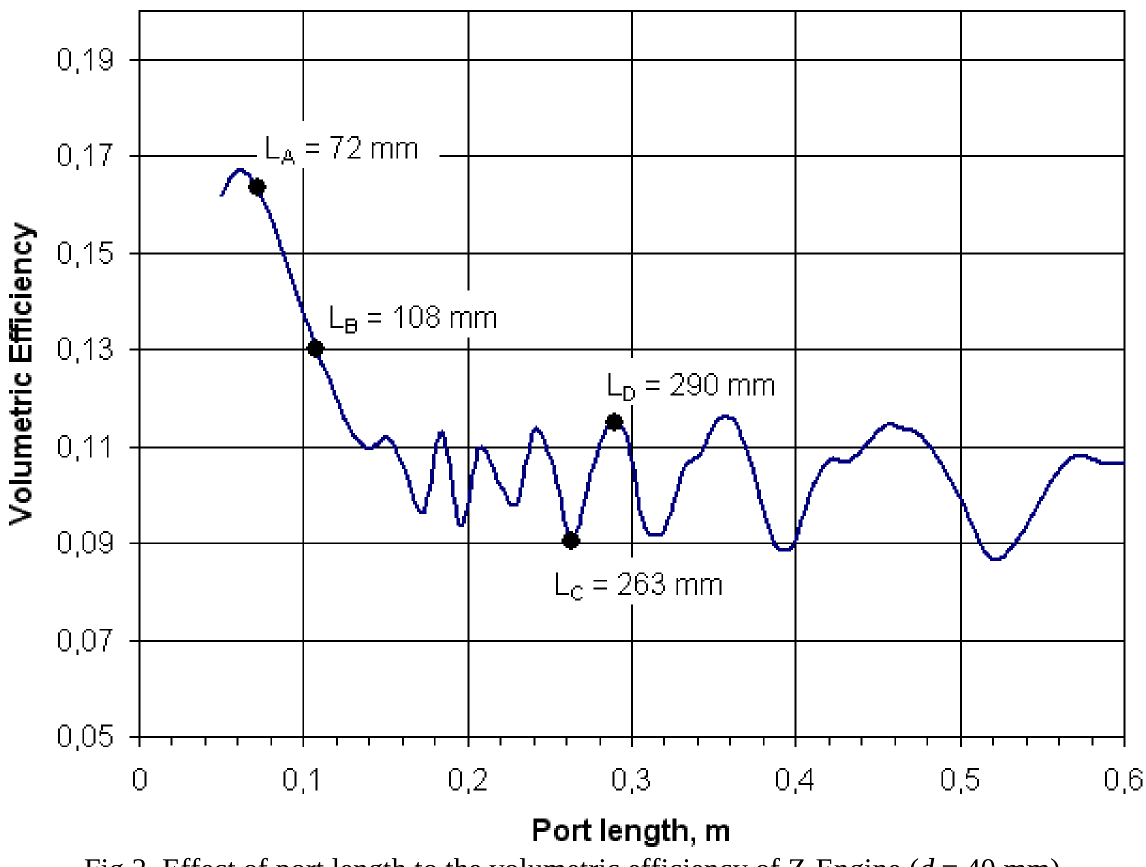

Fig.2. Effect of port length to the volumetric efficiency of Z-Engine (*d* = 40 mm)

Presence of volumetric efficiency oscillations on right side of the curve is typical when wave effects in the intake system are used for dynamic boosting. Similar phenomena are observed in the traditional four-stroke engines. As an example, Figure 3 shows a graph of volumetric efficiency for four-stroke single cylinder experimental diesel engine with a cylinder diameter D/S = 85 / 110 mm at 1500 RPM.

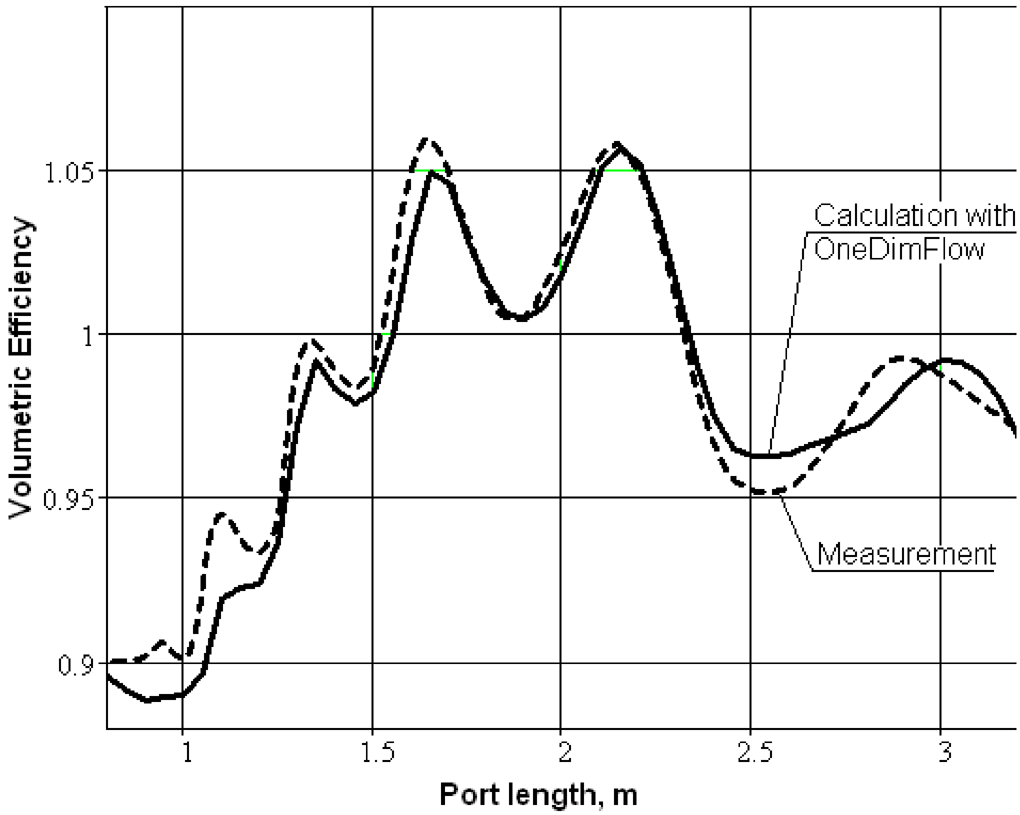

Fig.3. Intake manifold resonance charging for single-cylinder experimental diesel D/S=85/110 [2,3]

Figure 4 shows diagrams of cylinder pressure and intake port pressure (near the valve) as functions of CA calculated for port lengths L_A = 72 mm (dark blue line) and L_B = 108 mm (brown line).

As shorter the port than quicker the reflected high-pressure wave reaches to the valve. As a result the time-averaged pressure before valve is larger, the mass flow rate is larger and total volumetric efficiency h_v is higher.

At the larger port length (L_B =108 mm) high-pressure wave comes at the valve later (brown line), in comparison with a wave of shorter port (dark blue line). As a result the time-averaged pressure before the valve is less than the pressure corresponding with the shorter port L_A =72 mm. Therefore, the larger volumetric efficiency corresponds with smaller length L_A = 72 mm. Oscillatory process in a short port has enough time to be completely damped before the next opening of the valve due to higher frequency corresponding with small port length.

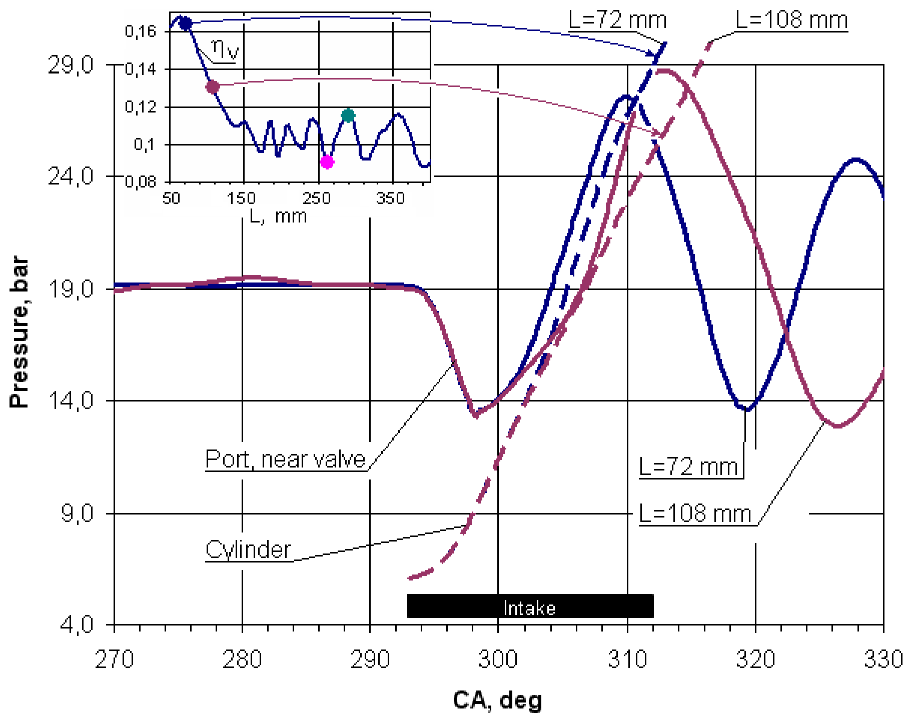

Fig.4. Pressure in cylinder and in intake port (near valve) on small port lengths (variants L_A=72 mm and L_B=108 mm, fig.1)

At long port condition (L > 150 mm) a reflected high-pressure wave is coming too late (valve is already closed), as a result, the volumetric efficiency decreases down to 0.09 – 0.12.

Figure 5 shows diagrams of the pressure for port lengths L_C = 263 mm and L_D = 290 mm. If the port length will be increased over 150 mm the mass of in-cylinder fresh charge is defined mainly by the free pressure oscillations in the port remaining since the previous cycle. Depending on the phase of the oscillation, local minima and maxima appear. (It depends on the time moment and pressure of the reflected wave coming to the valve.) If the average pressure before the valve increases (curves corresponding L_D) the volumetric efficiency increases too and vice versa (curves corresponding L_C).

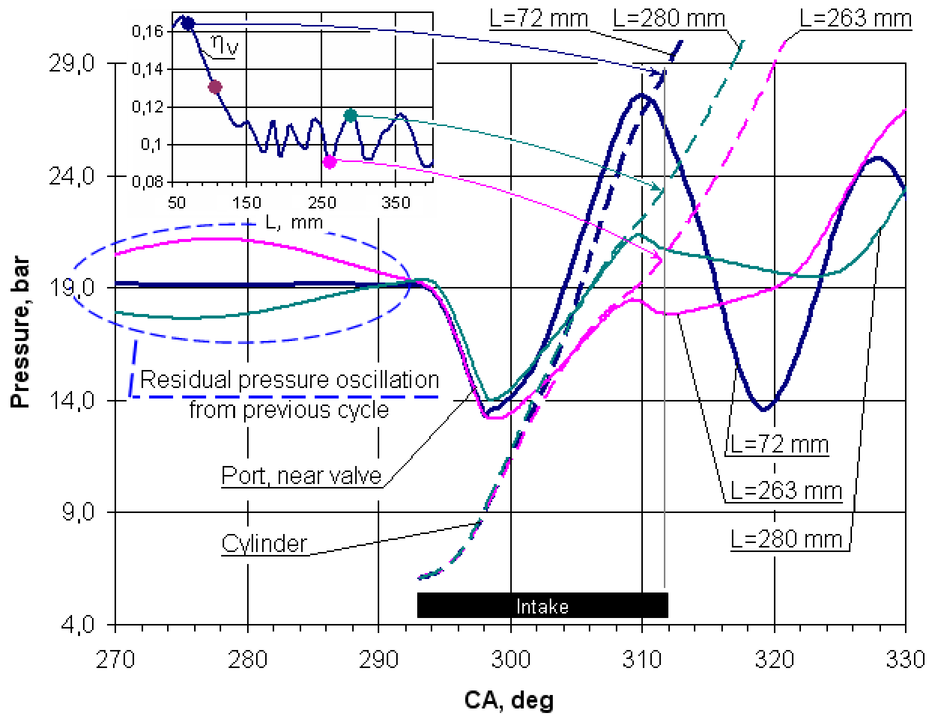

Fig.5. Pressure in cylinder and in intake port (near valve) on big port lengths (variants L_C=263 mm and L_D=290 mm, fig.1)

In the above numerical simulations the temperature of the port wall was assumed to be constant and equal to 347 K, and velocity coefficient, which takes into account losses in the junction of the port and the receiver, was assumed φ = 0.98 (that corresponds to a well shaped port mouth). These two factors affect on the induction process obviously. At the current stage of the research there was checked do they affect qualitatively on oscillatory processes in the port. Figure 6 shows graphs of volumetric efficiency vs port length at different wall temperature: 347 K and 447 K, respectively. It may be seen that raising the wall temperature reduces slightly the mass of fresh air in cylinder due to the small length of port and the short duration of intake process and do not change the qualitative behavior of volumetric efficiency curve.

Figure 7 shows graphs of the volumetric efficiency vs port length for different values of velocity coefficient at the entrance of port. It may be seen that this parameter affects on volumetric efficiency only if the port is short and does not affect if port is long.

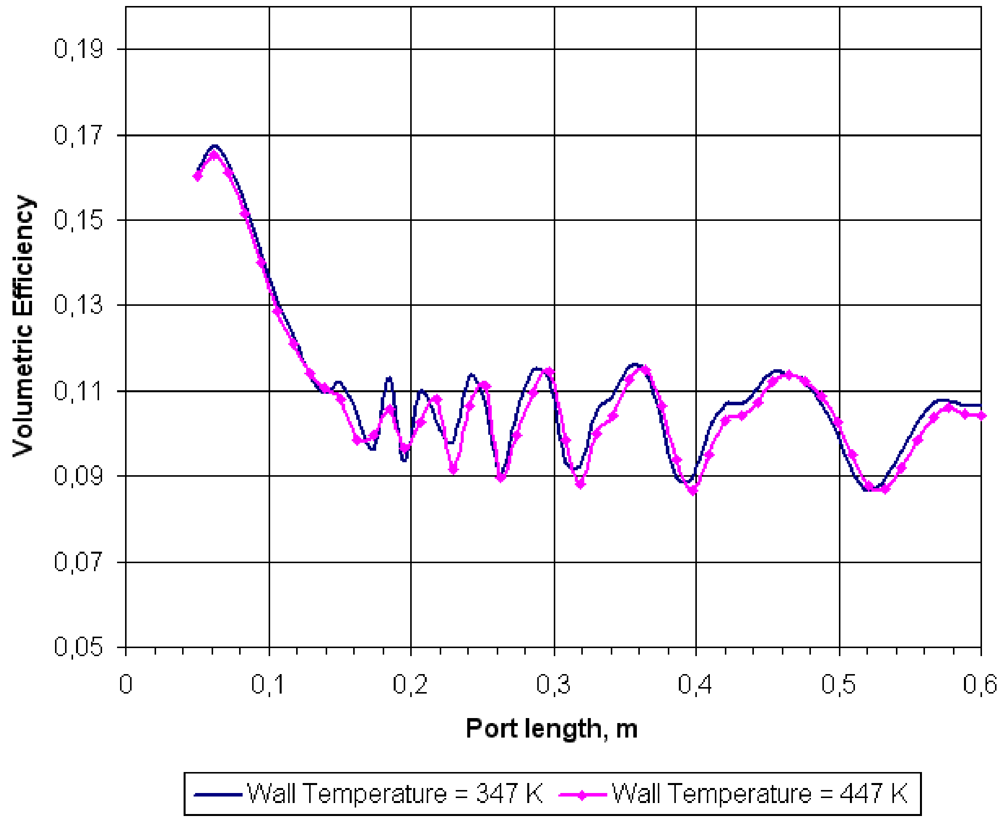

Fig.6. Effect of port length on volumetric efficiency at various temperatures of port wall

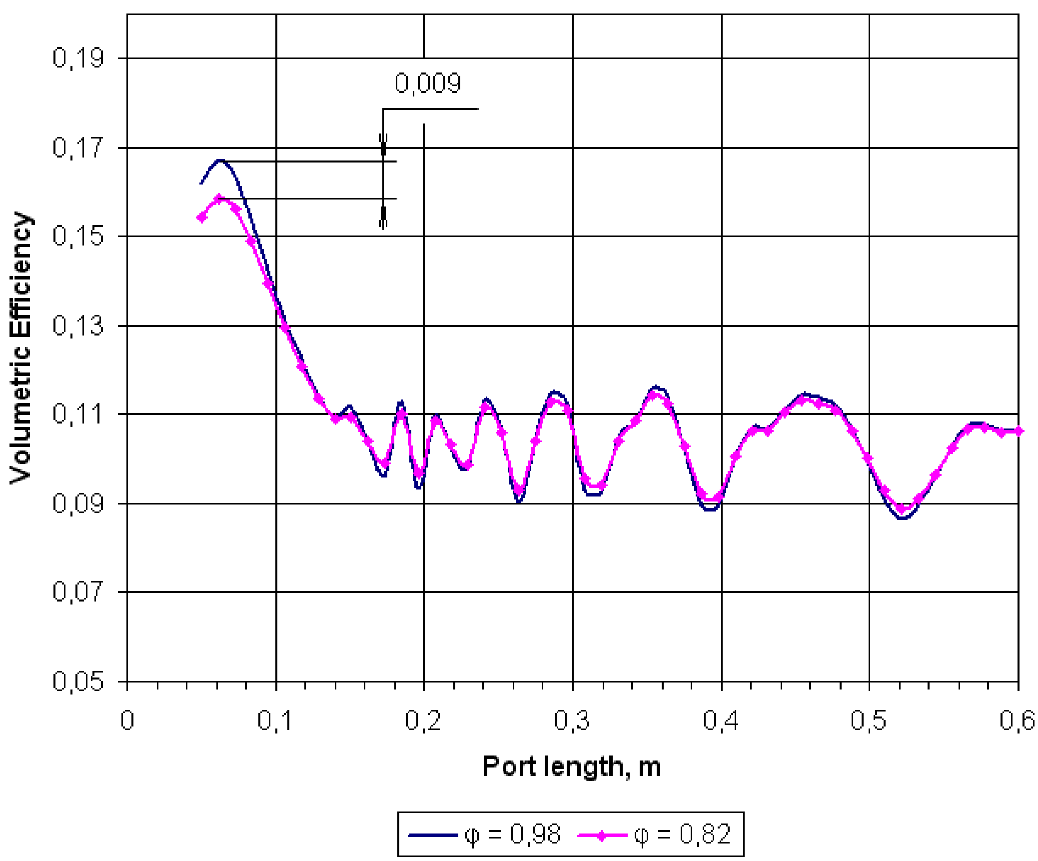

Fig.7. Effect of port length on volumetric efficiency at various velocity coefficient of the entrance of port

4. *Effect of Port Diameter*

A second parameter considerably affecting on the oscillation process is the port diameter ***d***. Figure 8 shows the volumetric efficiency vs port length at three different diameters. It may be seen that an optimal value of port diameter is ***d*** = 40 mm, but its value depends on the port length.

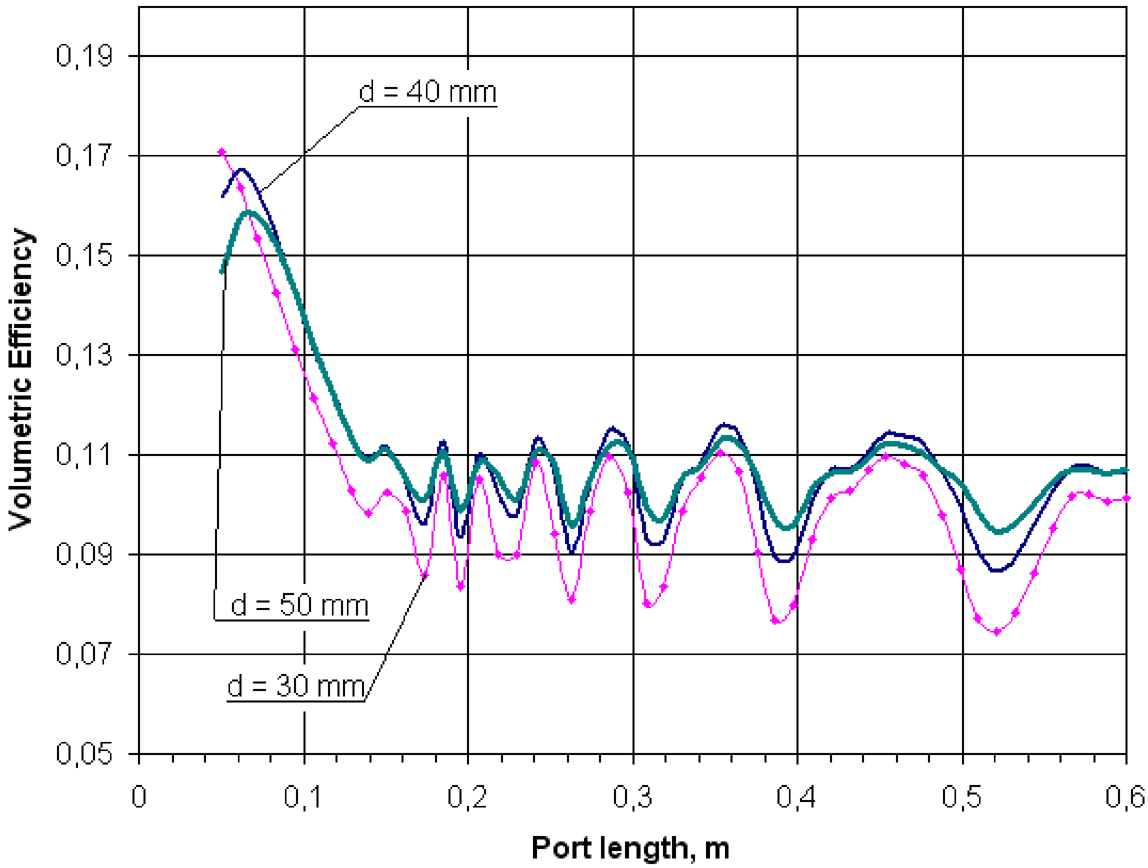

Fig.8. Effect of port length on volumetric efficiency at various port diameters

The smallest possible port length is bounded by engine intake system design. At increasing the port length its optimum diameter increases as well. (In terms of volumetric efficiency). The figure 9 shows plots of the volumetric efficiency vs the port diameter for port lengths: 72 mm, 108 mm and 144 mm. The best result may be achieved with *L* = 72 mm and ***d*** = 40 mm.

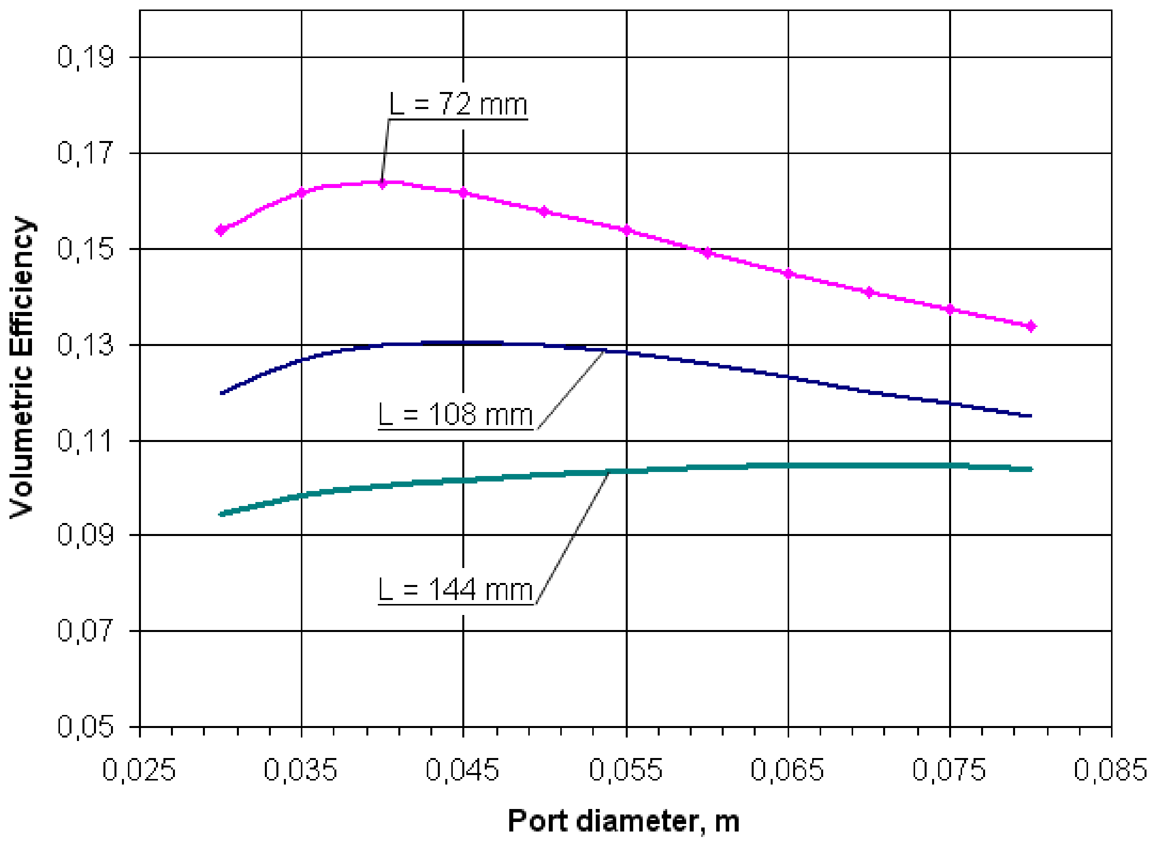

Fig.9. Effect of port diameters on volumetric efficiency at various port lengths

5. *Effect of Valve Lift Profile*

In the study presented above there was used assumption about conventional design of intake valve seat. So the effective valve lift profile has segments with a small effective flow area at the beginning and ending of intake. However in the z-engine, where the very high difference of pressure takes place at both the beginning and ending of intake, these segments may have essential negative effect. To estimate this effect there was investigated two alternative designs: base or normal design of valve & seat and valve in a recessed seat, fig. 10. Recessed seat allows exclude segments of injection profile with small effective flow area and use more effective valve capabilities.

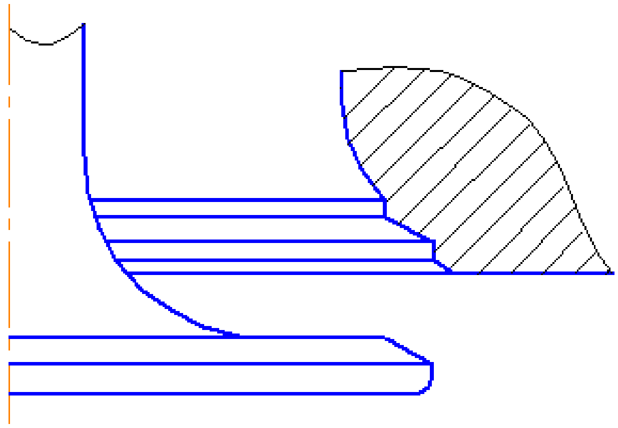

Fig. 10. The intake valve in recessed seat.

There were investigated two variants of recessed seat: with same valve lift as in normal design and extended profile where valve lift is increased on value of recess. All variants of effective flow area are presented in fig. 11.

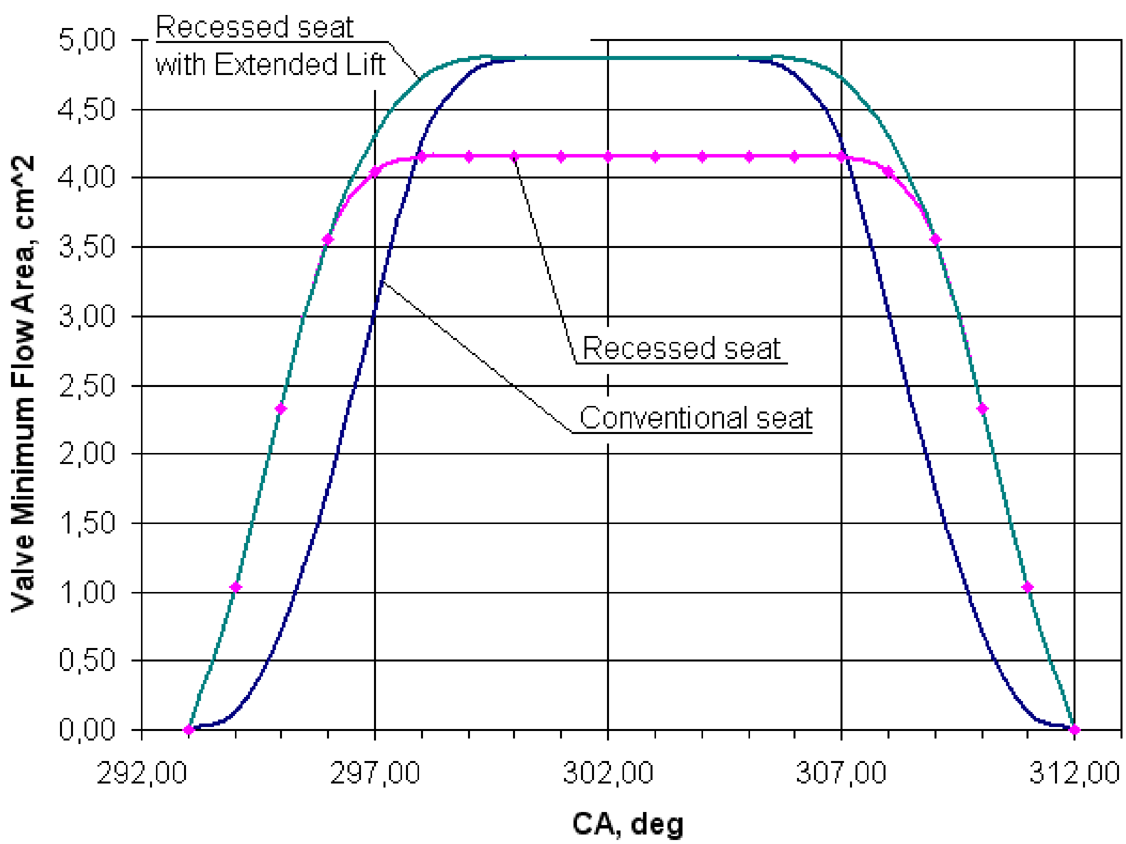

Fig.11. Valve lift profiles

In practice the extended duration of intake increases optimal port length, fig. 12 and increases volumetric efficiency. More over, the recessed variant may be more effective if so short port may not be realized in real engine design. Fig. 13 compares air flow diagrams obtained for three investigated variants of valve effective flow area. The selection of the end variant of valve timing and port & seat design should be done after 1D thermodynamic optimization of port timing at different operating modes and analysis of valve train mechanism.

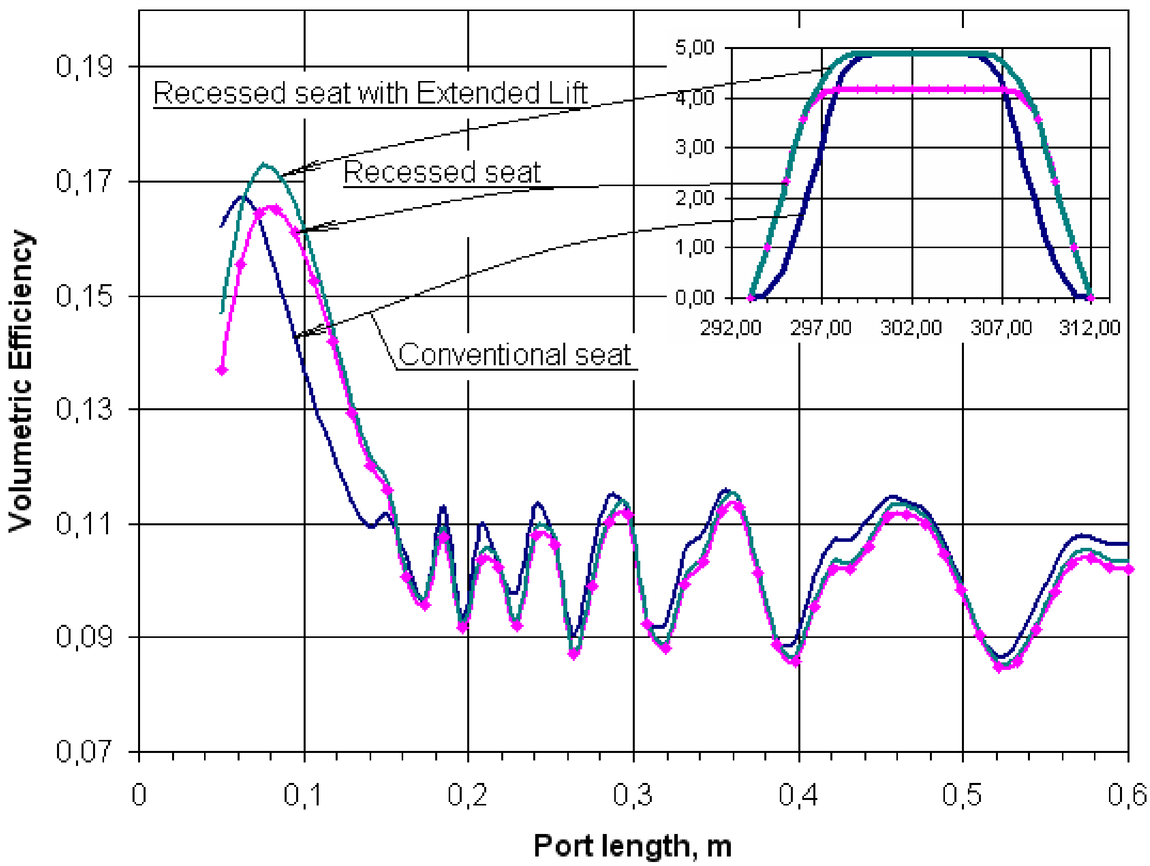

Fig.12. Effect of port length on volumetric efficiency at various valve lift profiles

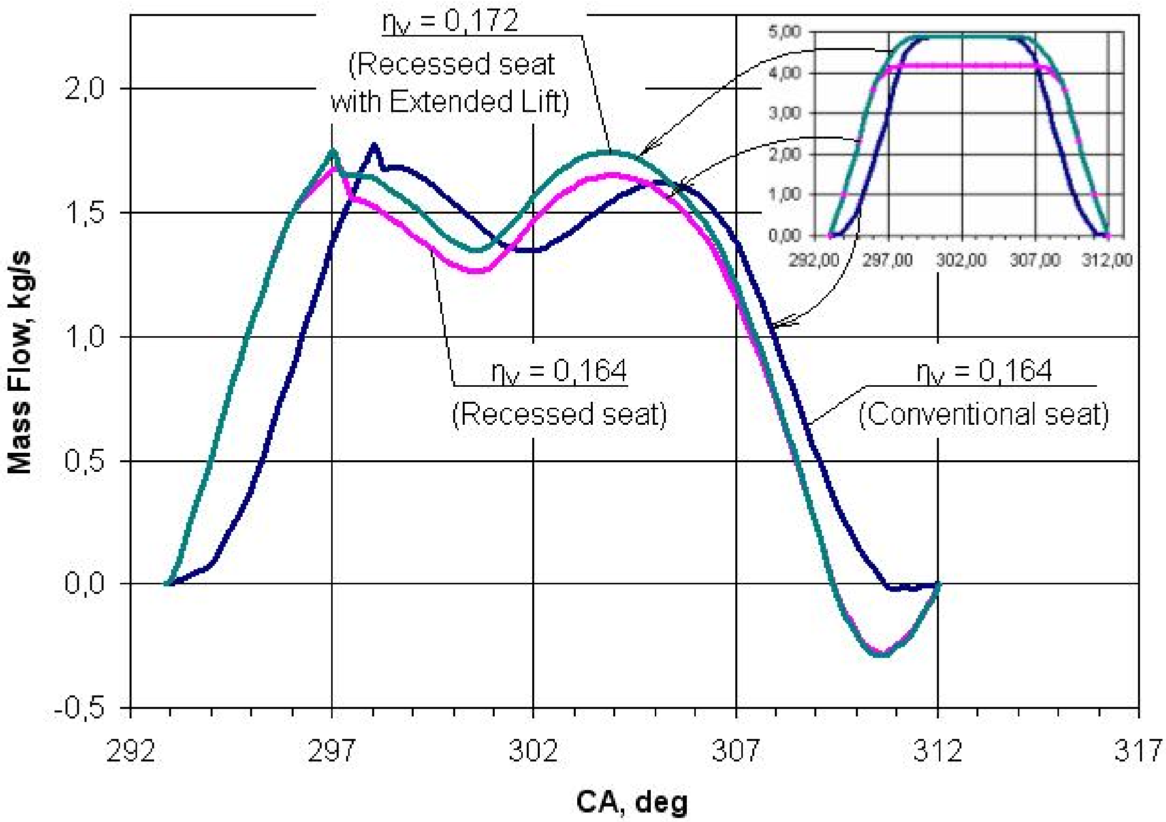

Fig.13. Effect of valve lift profile on intake mass flow (L=72 mm d=40mm)

6. Conclusions

The optimum intake port length of the z-engine is 60 – 70 mm. This value is obtained in study of intake process at full power operating mode @ 3600 RPM. This operating mode is most critical to required in-cylinder fresh charge.

The recessed valve seat design is effective to improve z-engine volumetric efficiency.

Temperature of intake port wall has not observable effect on volumetric efficiency.

The optimum value of average intake port diameter is 40 mm.

The detailed design of intake ports and their junction with a receiver should be investigated in 3D approach. However this work should be done after the general concept of the z-engine design is known. The draft cross section of the z-engine intake system developed on the base of presented research is shown in the fig. 14.

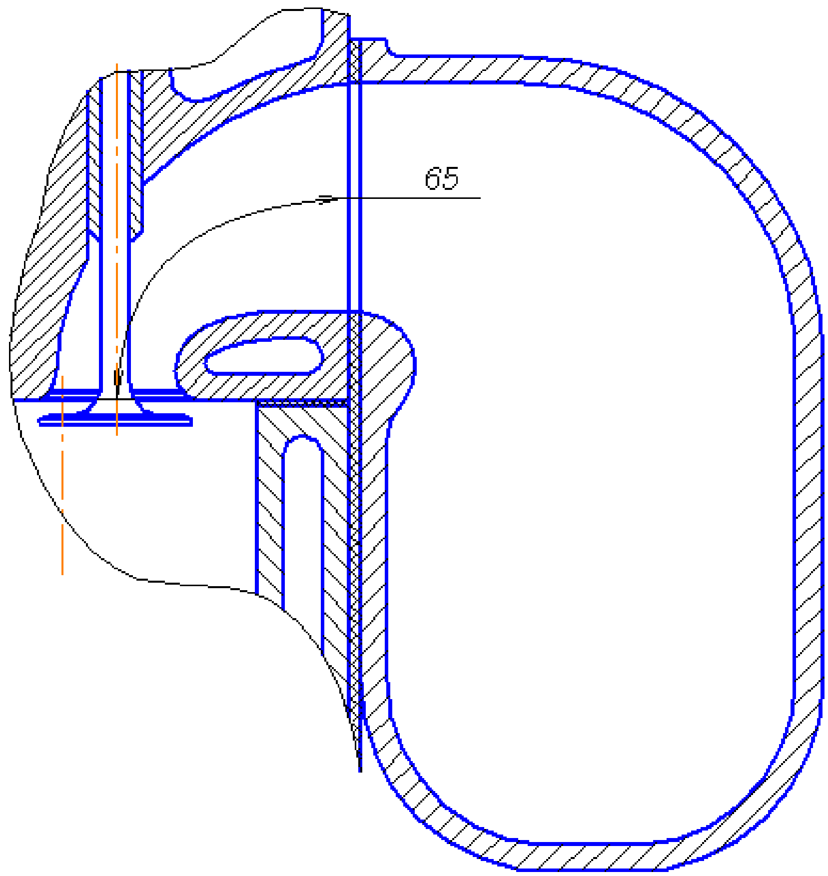

Fig. 14. Variant of cross section of the Z-engine intake system

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