Numerical simulation of non-stationary flows in water jet cutting nozzles

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The use of a high-speed water jet is one of the modern technologies for cutting and machining the material. High efficiency can be achieved by either by increasing the operating pressure or by using a non-stationary mode created by an ultrasonic generator. The use of ultra-high pressures in continuous mode induces extreme loading to high pressure parts of the cutting machine. The alternative approach using pulsating jets from an ultrasonic generator can increase the efficiency of the cutting without the need of such high water pressures. The impact of droplets of the pulsating jet generates much higher impact pressure than the stagnation pressure generated by the action of continuous jet. The efficiency of the cutting is further increased by cyclic loading of the target as well as by large shear stresses generated by the jet, see [1].

The efficiency of the pulsating jet cutting is highly influenced by the amplification of the pressure oscillations created by the ultrasonic generator in the properly shaped domain. The higher amplification is obtained in the case of resonance in the high-pressure system. The propagation and amplification of these pulsations were previously studied using analytical and numerical models for the case of single nozzle systems (see, e.g., [1] or [2]).

The current article deals with the numerical simulation of the pulsating flows in the high pressure part of a cutting system with five nozzles. The final aim of the work is the shape optimization of the system in order to achieve high amplification of the pressure waves and hence the high amplitudes of jet velocities. First of all two numerical models are cross compared for the case of reference geometry (see Fig. 1).

The first model assumes compressible liquid with the water density and sound speed given by

$$\rho = C - \frac{1}{k^2 \Delta p + kq}, \tag{1}$$

$$a = k\Delta p + q, \tag{2}$$

where $C = 1402.4\,\mathrm{kg/m^3}$, $k = 1.669 \times 10^{-6}\,\mathrm{m^2 s/kg}$, $q = 1481.98\,\mathrm{m/s}$ and $\Delta p = p - 101\,325\,\mathrm{Pa}$. The model is based on the solution of Reynolds-averaged Navier-Stokes equations with an additional two-equation RNG $k - \epsilon$ turbulence model with non-equilibrium wall functions. The ultrasonic generator is modeled using moving mesh strategy as an oscillating wall with the frequency $f = 20\,\mathrm{kHz}$ and amplitude $A = 6 \times 10^{-6}\,\mathrm{m}$. The spatial discretization is achieved with the finite volume approach with a second order interpolations and the temporal derivatives are discretized by a first or second order implicit scheme. The model is the ANSYS Fluent package.
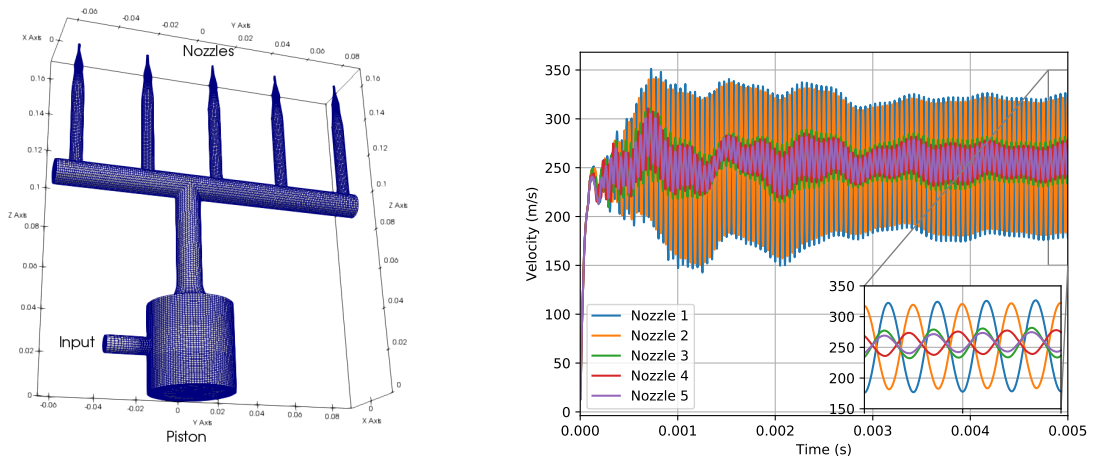
Fig. 1. Geometry of the cutting nozzle system and the maximal velocities in the nozzle throats

The second model uses simplified approach without turbulence model and modeling the ultrasonic generator with fixed grid prescribing the fluid velocity at the piston head as $v(t) = 2\pi f A \cos(2\pi f t)$. The density is given by linearized equation of state

$$\rho = \frac{p}{a^2} + \rho_0 \qquad (3)$$

with $\rho_0 = 997.95\,\text{kg/m3}$ and $a = 1484.8\,\text{m/s}$. The relative difference of the density (3) and (1) is less than 1×10^{-6} for pressures in the range $0\,\text{MPa}$ to $35\,\text{MPa}$.

Both models predicts similar amplitudes of the velocity in the nozzles, however the second one requires less computational time than the first one.

An optimization was carried out with the first model using the objective function $J = \sum_{i=1}^{5}(\max w_i - \min w_i)/5$ (here w_i is the z-component of the velocity in the nozzle, maximum and minimum is taken over last few pulses). The achieved amplitudes are in the order of $120\,\text{m/s}$, whis is roughly $50\,\%$ of the average speed in the nozzles. Similar results were obtained also with the second model using objective function based on the amplitudes of pulses in the mass flux rate.

Acknowledgement

References

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