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THEORETICAL BASICS AND EXPERIMENTAL INVESTIGATIONS OF STEAM – WATER JET PUMP

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SUMMARY

The aim of the paper is investigation possibilities of highly compressible steam in supersonic steam-water jet pump's mixing chamber, for simultaneous achieving high pressures and hot water at the jet pump's outlet.

The steam-water jet pump is complex flow device, since in it is happening two phase supersonic-subsonic flow and at the same time, very quick condensation and transformation of heat energy into kinetic and pressure energy.

The paper presents the main conclusions which were appeared during experimental investigation of steam-water jet pump which were carried out of steam-water jet pump in the oil refinery Novi Sad, during August and September 2003. On the basis of experimental investigation it is concluded next:

- *experimental investigation proved possibilities of achieving high pressures and temperature at the outlet of steam-water jet pump (results are in the frame of well-read data),*
- *according to given data, it was determinate efficiency rate of mixing chamber,*
- *high compressible of two-phase region was proved and it was emphasized Mach number as basic index of compressibility,*
- *hypothesis of changing the Mach number was given.*

KEY WORDS:
Jet pump, two-phase region, compressible fluid

1. Introduction

Steam - water jet pump is actually heat pump because it replaces a heat excanger and pump. It consists of:

- converging – diverging Laval's nozzle for steam expansion,
- supersonic steam – water mixing chamber and
- discharge diffuser.

Steam – water jet pump transforms steam heat into kinetic energy of liquid flow. Following assumptions are of the importance:

- processes in Laval's nozzle and outlet diffuser are isentropic.
- process in mixing chamber is unisentropic just for the part of losses concerning mixing of steam and water.
- all lows are valid (mass conservation equation, momentum equation, energy equation, equation of state, equation of phase transitions).

2. Experimental investigation

Experimental investigation have been conducted on steam-water jet pump prototype with exploitation scale (2 MW power). The outlook of jet pump is given in Fig.1. The efficiency rate, as well as other characteristics of supersonic steam-water jet pump justifies its application.

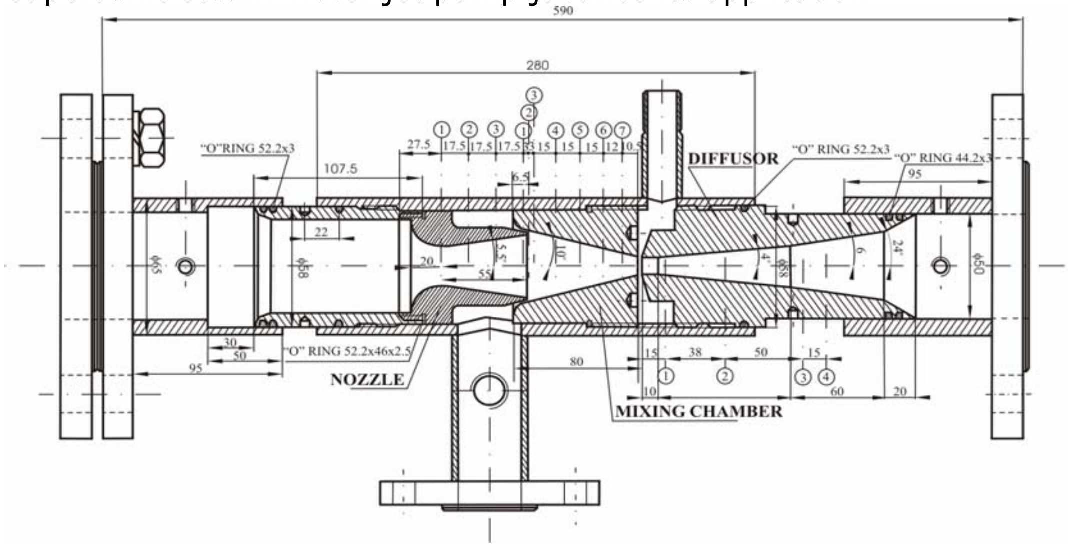

Figure 1. The outlook of steam-water jet pump

Conducted experiments have proved as necessary for correct direction of whole investigation work. Without experiments, a lot of dilemmas, which disturb diagnostics and theoretical bases of this problem, would remain. The experiments have confirmed two basic questions:

- possibility of gaining higher outlet pressures of hot water from steam-water jet pump, compared to pressures of steam and injected cold water.
- achieving subsonic-supersonic-supersonic-subsonic condensation regime through steam-water jet pump, what represents the justified started base for the explanation of the investigation theses.

With experiments could also be determined:

- approximate efficiency rate of jet pump, what enables to locate geometry and flow regime areas with considerable losses. In that way, the time for investigation of marginal influences is not lost,
- the borders of operating areas and the area of self regulation,

- manner of acceptable and efficient regulation for injector stable operation in supersonic regime,
- the adequate geometry and application of jet pump in different fields of human activities.

Table 1. Supersonic flow through steam-water jet pump's mixing chamber with and without damping, Date: 25.11.2003, Air temp. 14°C; Air pres. 101400 Pa; Air hum. 65%

Cold water						
	p_{Wm} [bar]	t_W [°C]	$\dot{m}_W$ [kg/s]	e_{Wkin} [J/kg]	e_{Wp} [kJ/kg]	q_W [kJ/kg]
1	1,1	12,5	1,08	0,15	0,21	52,375
2	1,1	12,5	1,08	0,15	0,21	52,375
3	1,1	12,5	1,03	0,14	0,21	52,375
4	0,95	12	1,08	0,15	0,195	50,28
5	0,9	12	1,07	0,15	0,19	50,28
6	0,9	12	1,08	0,15	0,19	50,28
7	0,9	12	1,08	0,15	0,19	50,28

With stable flow: maximal pressure at the exit of jet pump (with damping at the exit) is 8,5 bar, maximal pressure (without damping) is 1,7 bar, max. temp. of hot water is 82 °C.
At the start chosen flow rate of cold water 1,08 l/s (12 °C, p_{Wm}≈1 bar) was constant during each regime of investigation.

Steam								
	p_{Sm} [bar]	t_S [°C]	$\dot{m}_S$ [kg/s]	p_{Smg} [bar]	p_t/p_S [-]	e_{Skin} [kJ/kg]	e_S [kJ/kg]	$q_S + e_{Sp}$ [kJ/kg]
1	1,5	127,5	0,5	0,22	0,488	0,183	179,47	2717
2	1,5	127,5	0,24	0,19	0,476	0,088	179,47	2717
3	1,5	127,5	0,16	0,19	0,476	0,058	179,47	2717
4	1,5	127,5	0,22	0,20	0,48	0,080	179,47	2717
5	1,5	127,5	0,19	0,21	0,484	0,069	179,47	2717
6	1,5	127,5	0,19	0,21	0,484	0,069	179,47	2717
7	1,5	127,5	0,20	0,21	0,484	0,073	179,47	2717

Hot water - mixture									
	p_{Mm} [bar]	t_M [°C]	$\dot{m}_M$ [kg/s]	e_{Mkin} [J/kg]	e_{MS} [kJ/kg]	q_M [kJ/kg]	$\dot{m}_M / \dot{m}_S$ [-]	η [-]	flow regime
1	8,5	70	1,58	0,324	0,95	293,3	3,16	0,31	damping
2	1,8	80	1,32	0,226	0,28	335,2	5,5	0,59	
3	1,8	84	1,24	0,199	0,28	352,0	7,75	0,84	
4	1,7	81	1,30	0,219	0,27	339,4	5,91	0,64	
5	4,3	82	1,26	0,206	0,53	343,6	6,63	0,71	damping
6	5,2	82	1,27	0,209	0,62	343,6	6,68	0,74	damping
7	7,7	82	1,28	0,212	0,87	343,6	6,4	0,74	damping

The experiments showed that after achieving stabile operating regime, the flow process and phase transformation are conducted in optimal way, i.e. without vibrations and smooth. The vibrations which are the measure of un-stationary, can be observed by very sensitive touching the housing of jet pump. The stability of flow can be observed by watching outlet flow of hot-water. The stable flow proves that transitions from supersonic into sonic regime (and vice versa) is happening without shockwaves.

3. Efficienty rate of the jet pump

For determining total efficiency rate of a jet pump it is neccesary knowledge of efficiency rate individual parts of a jet pump. Based on experimental investigation number 7, Table1, total efficiency rate of the jet pump is: $\eta = \eta_{mcinl}\eta_{mc}\eta_d = 0,74$.

Lavals nozzle efficienty rate for h_L=250 kJ/kg η_L = 0,98

Ringlike nozzle cold water efficiency rate according to geometry of nozzle η_w = 0,90

To entrance mixing chamber efficiency rate $(\eta_L m_s+\eta_w m_w)/m_M$ η_{mcin} = 0,91

Mixing chamber efficiency rate $\eta_{mc}=\eta/(\eta_{mcin}\eta_d)$ η_{mc} = 0,90

Discharge diffuser efficiency rate η_d= 0,90

During calculating efficiency rate it didn't take into consideration losses due to exchange of heat with environmental (0,5 kJ/kg). Although separation of estimated efficiency rate of mixing chamber (0,9) on sources of losses in this moment isn't possible ,some remarks are exused.

Diffuser's function of mixing chamber, similarly like classical diffuser, has corresponding loss. The maximal magnitude of classical diffuser losses during streaming of homogeneous fluids is about 0,94. It can be acceptable as maximal magnitude of mixing chamber. Taking losses are more related to change of geometry then friction into consideration, in this case friction doesn't introduce loss, because of transformation into usefull heat.

The input loss in mixing chamber is demonstrated by uncoincidence direction of velocity with axis of jet pump. As this angle is smaller then 5^0 component of radial velocity which disturb flow and lower mixing chamber efficiency rate , can be neglected. For other sources of losses related to complex processes in mixing chamber, there are no available reliable facts.

4. Sound velocity in two phase region

There are two interphase processes which influence the sound velocity: energy change and momentum change, with possible violation of their equilibrium. Sound velocity in two phase medium depends on: state parameters of phases (pressure and temperature), physical characteristics of each phase, mass concentration of gas phase *x*, medium structure (shape and dimensions of water droplets, distances between droplets etc.), parameters of sound impulses (frequent-flow parameter ωt_r [-] where ω is angular velocity [s^{-1}], and t_r [s]- relaxation time of liquid phase flow, i.e. time to establish constant velocity of liquid phase, which represents other relaxation times).

It should be noted that boundary condition $\omega t_r \to \infty$ is fictitious. Upper limit of sound dispersion is not boundless. For example, in humid vapor at t = 100 ^oC and x = 0,1 with calculation error of 0,5 % upper limit of dispersion of parameter ωt_r is only 100.

In Fig. 2. is given change of sound velocity during isobaric transition from water into steam. In two phase region sound velocity is presented as a function of *x*, and in one phase region as a function of bulk temperature

T. Designations of velocity curves for different ωt_r in Fig. 2 are: 1 - $(\omega t_r)_{max}$ (upper limit of sound dispersion); 2 -ωt_r = 4.0; 3 - ωt_r = 1.0; 4 - ωt_r = 0 (lower limit of sound dispersion); 5 - sound velocity in water; 6 - sound velocity in overheated steam.

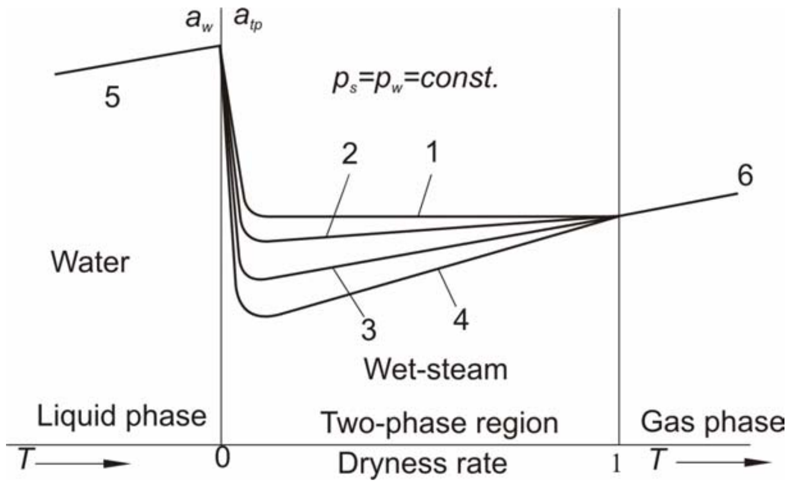

Figure 2. Change of sound velocity during isobaric transition from one phase to two phase region

If it is accepted that impulse transition is complete and if it is assumed that pressures in phases are the same, could be gotten following:

$$\left(\frac{a}{a_S}\right)^2_{ul} = \frac{x + (1-x)(\rho_S)^2}{x + (1-x)\left(\frac{\rho_S}{\rho_W}\right)^2\left(\frac{a_S}{a_W}\right)^2} \quad (1)$$

where: *a* -sound velocity in two phase region [m/s], and *ul* refers to upper limit of sound dispersion.

If it is considered that interphase transitions totally lack, the density of liquid phase is considerably higher than steam density and particles of liquid phase are spherical for dryness level of 0,1 < *x* < 1,0; when their movement is conducted under Stock's low, and hence the equation (4.1) is simplified.

5. Conditions for maintenance of maximal compressibility (minimal sound velocity)

Change of state of water-steam mixture in injector's mixing chamber consists of heating of cold water and condensation of steam. It can be considered that condensation of steam is conducted isothermal or isobaric, since the curves of isothermal and isobaric change are pretty the same in the whole two phase region. Heating of cold water is conducted by convective heat transition. It can be considered that compressibility of steam, during realistic velocities of water and steam, depends on sound velocity. Lower sound velocity correspondents to higher compressible effects and higher Mach numbers. Mixture of water and steam is more compressible then steam. For same flow velocities the Mach number of mixture is a lot larger then the Mach number of steam. For example, for the same velocity of mixture and steam *v* = 200 m/s:

$M_1 = \frac{200}{500} = 0,4$ - the Mach number of the steam;

$M = \frac{200}{50} = 4$ - the Mach number of the mixture;

Explaination of this appearance we can find in dependence of sound velocity and dryness rate on Fig.3. The sound velocity in the mixture (about 40 – 50 m/s) is smaller then the sound velocity in the steam (about 500 m/s). As the Mach number is defined as:

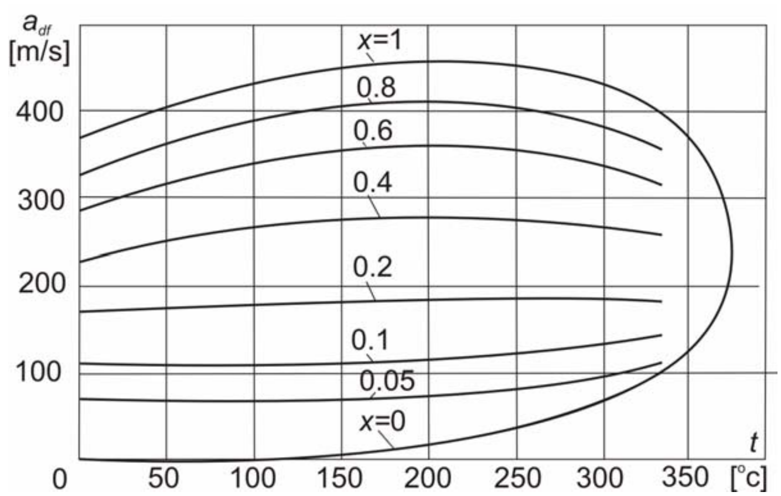

Figure 3. Dependence sound velocity on dryness rate ane temperature

$$M = \frac{v}{c} \quad (2)$$

it's possible achieving higher Mach numbers by decreasing sound velocity instead of increasing flow velocity (it's related with high losses).

According to laws of compressible flow higher Mach numbers should provide achievement of higher pressures (proportional to M^2) in downstream sections of mixing chamber.

As at the exit of mixing chamber is expected hot water (about 110 °C), the two phase region with $x \leq 0,15$ is near to boundary saturation line, what can be seen from *h-s* diagram.

It is assumed that sound dispersion covers region of natural conditions for forming dispersion of sound waves, which are in harmony with physical and thermodynamically laws. Also is assumed that the frequency of sound wave *n* [Hz] can be influenced artificially by mechanically made oscillations at certain segments of mixing chamber, what is going to be examined in the future investigations.

Variable ω, angular velocity of gas particles, during meeting with disturbance wave is:

$$\omega = 2\pi n = 2\pi/T \; [s^{-1}] \qquad (3)$$

where are:

n - wave frequency [Hz]

T - time of oscillation [s] (T=wave length/wave velocity=L/a)

The time of relaxation t_r, in most cases, represents time needed to establish stationary state of some occurrences or some segment of change. In this case time of relaxation denotes "inertia" of process for establishing state of constant phases, i.e. $\Delta v = v_S - v_W = const$.

Variable t_r represents also the time of relaxation of spherical water droplet in gas region.

6. Characteristic changes of *p, M* and *v* through investigated jet pump

Characteristic results of supersonic flow through steam-water jet pump are shown in Table 1. Diagrams of Mach number, pressure and mixture velocity in jet pump (on the basis of investigation number 7 in Table1.) are shown in Fig. 4.

Credible pressure lines are those through Laval's nozzle and outlet diffuser, with their borders at entrance and exit of mixing chamber. Pressure gradient is positive, with point of inflection in front of the throat of mixing chamber. Stable work with supersonic zone in mixing chamber is from 1.7 to 8.5 bar gauge pressure, with the same initial parameters of all investigations.

Appearance of pressure discontinuity due to appearance of shockwaves at the exit of Laval's nozzle as well as in mixing chamber isn't registered during proper work of the device. Stable work requires smooth pressure line throughout jet pump, without shockwaves in any section.

The parts of pressure curve in the throat of Laval's nozzle as well as in the throat of mixing chamber (or in its vicinity) have a high gradients. It is agreeable to transition through sound border according to some unknown laws, similar to isentropic flow shown in Fig. 5.

Changes of the Mach number through mixing chamber is more uncertain than the pressure lines. The parts of lines through the Laval's nozzle, throat of mixing chamber and outlet diffuser are known; while the shape of the Mach number curve is uncertain in the mixing chamber. Future experiments are not likely to give more information about location of maximal magnitude of the Mach number in the mixing chamber, as well as contour of the changes.

Authors consider that maximal Mach number isn't located at the exit section of Laval's nozzle but in some middle section of mixing chamber. Because of that pressure rise toward throat of mixing chamber requires decrease of Mach number.

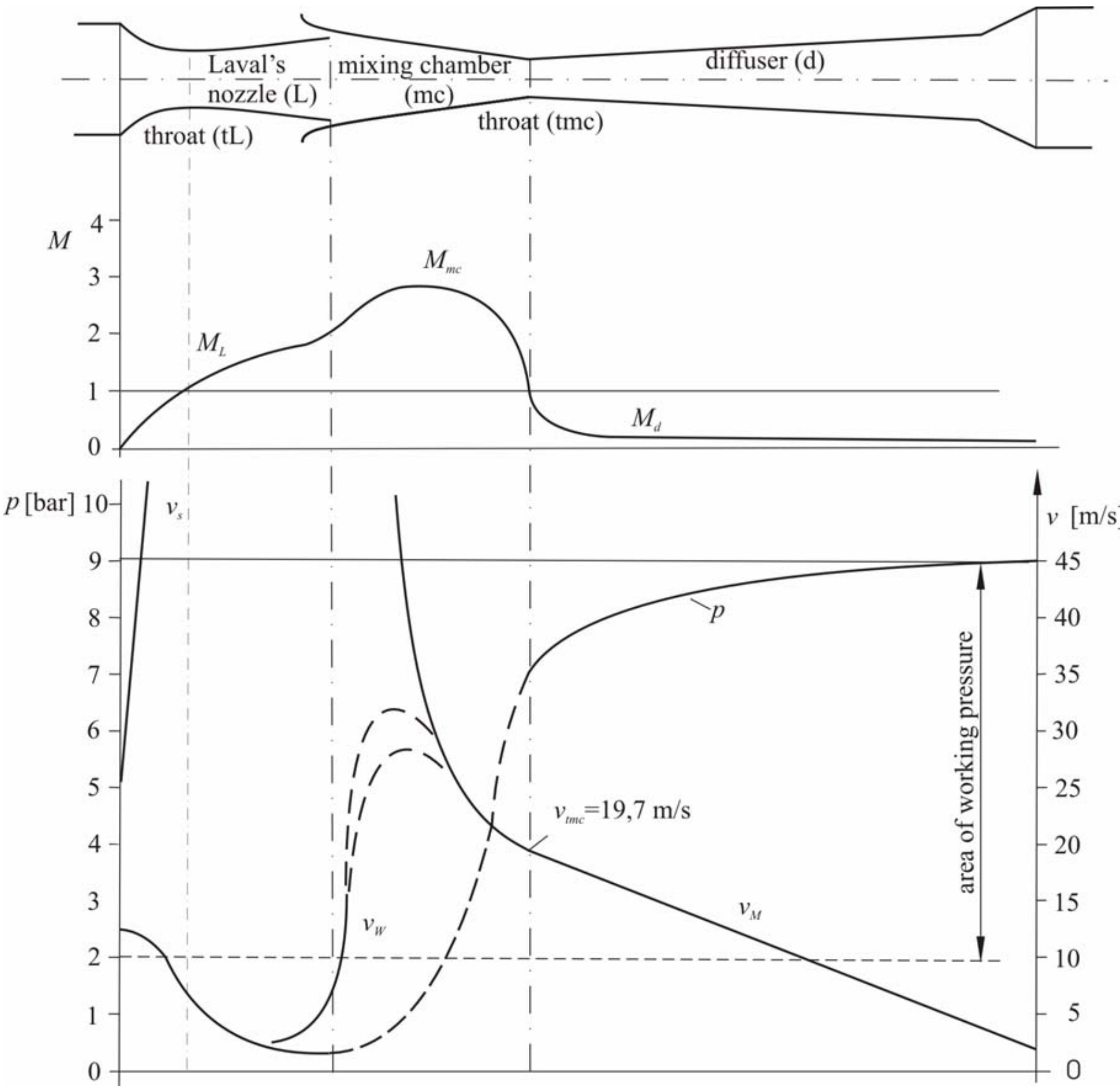

Figure 4. Diagrams of Mach number, pressure and mixture velocity in jet pump

The relation between pressure and Mach number could be recognized in known laws of compressible flow:

$$\frac{p_t}{p} = \left(1 + \frac{k-1}{2} M^2\right)^{\frac{k}{k-1}} \tag{4}$$

where coefficient (κ-1)/2 and exponent κ/(κ-1) should be exchanged with variable and, in some way, independent coefficients (κ-1)/2=*m* and κ/(κ-1)=*n.*

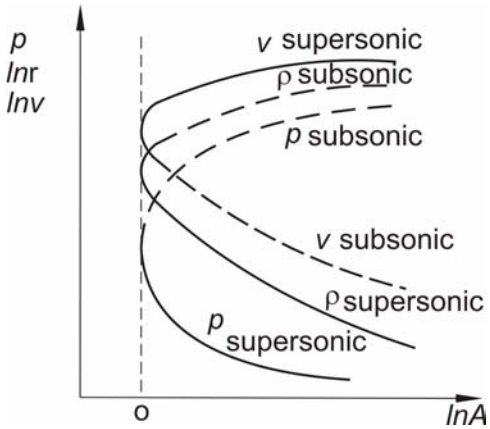

Figure 5. Changes of flow parameters during supersonic-subsonic transition depending on nozzle geometry change

If equation (4) is used for area in front of mixing chamber, the total pressure in the throat will be less for the value of dynamic pressure. It is known because velocity and density of water in the throat are known. Taking into consideration that upstream mixture velocities in mixing chamber are significantly larger than those in the throat of mixing chamber, kinetic energy in the throat can be neglected. Actually, the aim of jet pump isn't achieving high velocities in the throat of mixing chamber. Large difference between velocity of hot water in front of diffuser and behind it could bring higher losses through outlet diffuser.

Related to comment above and unfortified assumption that *n* is about 1, by multiplying with upstream pressure *p* is given:

$$\Delta p = p_t - p = mpM^2 \quad (5)$$

This relation can be for supersonic area of steam M>1. The future experimental results about disposition of pressure through mixing chamber will contribute to analysis of this relation. The starting point of mixture velocity distribution (Fig.4.) in the mixing chamber is indefinite, because it refers to section in which are completely mixed water drops with steam. In that section velocities of water drops (which are accelerated by steam) are considerable lower than steam velocities.

Condensation must be finished in the throat of mixing chamber. Opposite, compressible effects would be absent. Because of that, there is no doubt that in front of the throat of mixing chamber the Mach number suddenly decreases similar to Fig.5.

7. Suggestions for further activities

Predicted and finished program of experimental investigation was related to measure the input and output fluid flow parameters, with pressure control in the throat of Laval's nozzle. Derived results justify continuance of investigation in following directions:

- simultaneous developing of mathematical model (with experiments),

- investigation of usage of classical *h-s* diagram for stream processes with introduction of secondary fluids,
- characteristics of mixing chamber:
    - work synchronization of jet pump with subsonic and supersonic flow,
    - influence of pressure increase to change of compressible properties of wet-steam with a view to determine optimal limits of utilization,
    - shaping of mixing chamber according to desired input and output parameters; for definite purposes
    - determining of maximal sound mixture velocity in the throat of mixing chamber and its dependence on compressibility and other parameters
    - investigation of inertia of condensation process behind the throat of mixing chamber (without output diffuser).

8. Conclusion

Steam-water jet pump is actually heat pump because it replaces a heat exchanger and pump. Compared to standard equipment for this purpose, steam-water jet pump is less expensive, easier to maintain, without parts which requires usage of potentially hazardous liquids, produces less noise and saves energy.

Conducted experimental investigation as well as presented conclusions don't have only academic significance then open possibilities of new applications of steam-water jet pump. Applications of steam–water jet pumps is a matter of great importance both for region and for European Community generally. The paper is in this way completely harmonized with European efforts to save energy, which is simultaneous one of the biggest problems for region in which we live.

References

1. Астахов, И.В. (1990) Топливные системы и экономичность дизелей, Машиностроение,, Москва.
2. Bukurov M.(2004)Istraživanje svojstava nadzvučnih paro-vodenih injektora, doktorska teza, Ph.D. Theses, University of Novi Sad, Faculty of Technical Sciences
3. Дейч М.Е.,Филипов Г.А. (1968) Газодинамика двуфазных сред, Энергия, Москва
4. Beithou, N., Aybar, H.S. (2000) A Mathematical Model for Steam-Driven Jet Pump, Int. J. Mul. Flow 26, 1609-1619