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COMPUTATIONAL STUDY OF EJECTOR-PUMPS

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ABSTRACT

Computational results of 3D turbulent compressible gas flow in a single-nozzle ejector are compared with experimental data. Full Navier-Stokes equations and k-ε model of turbulence are used for mathematical model of gas flow. In computations the suction gas flow rate was determined and compared with experimental one. Two computational grids – coarse and fine are used to perform simulation. The fine grid is differ from coarse one by adaptation near the nozzle of active gas.. Comparison of results carried out on coarse and fine grids shows that the accuracy of coarse grid is enough to get reliable results. Difference of computed and experimental results is less then 4% for the flow rate of passive gas.

These results enable to make computational study of the multi-nozzle water-steam ejector. Condensation of steam is taken into account by introducing the equilibrium model of condensation. It is found that location of nozzles and its length are the important parameters of ejector influencing considerably its characteristics. The process of the condensation of water vapor significantly influences the work of ejector with an increase of the suction flow rate by a factor of 2.

NOMENCLATURE

M	Molecular weight of vapor	[kg kmol^{-1}]
R_0	Universal gas constant	[J kmol^{-1} K^{-1}]
T	Temperature	[K]
ρ	Gas-liquid mixture density	[kg mol^3]
h	Specific enthalpy of gas and liquid mixture	[J kg^{-1}]
$\rho^{(L)} = c\rho$	Density of the liquid phase of vapor	[kg m^{-3}]
$\rho^{(G)} = (1-c)\rho$	Density of gas phase of vapor	[kg m^{-3}]
h	Specific enthalpy of vapor	[J kg^{-1}]
$C_P^{(G)}.\ C_P^{(L)}$	Specific heat of the gas and liquid phase of vapor	[J kg^{-1} K^{-1}]
c	Mass fraction of liquid phase	

GOVERNING EQUATIONS

Gas is described like ideal gas with equation of state

$$P = \rho^{(G)} R_0 T / M$$

Full Navier-Stokes equations and k-e model of turbulence are used for mathematical model of gas flow.

According to Dalton's law the vapor pressure can be represented in the form

$$P = P^{(G)} + P^{(L)} \tag{1}$$

Where $P^{(G)}$ and $P^{(L)}$ - partial pressures of the gas and liquid phases of vapor respectively. Subsequently we will assume that

$$P^{(L)} << P^{(G)} \tag{2}$$

i.e., the partial pressure of liquid phase can be disregarded. Partial pressure $P^{(G)}$ of gas phase is equal to

$$P^{(G)} = \rho R_0 T (1-c) / M \tag{3}$$

Specific enthalpy of vapor can be represented in the form

$$h = (1-c) h^{(G)} + c h^{(L)}, \tag{4}$$

where $h^{(G)}$ and $h^{(L)}$ are specific enthalpy of gas and liquid phases respectively.

The described model is equilibrium vaporization model, so the following condition is always satisfied:

$$P \le P_H(T), \qquad (5)$$

where $P_H(T)$ is well known experimentally water vapor saturation pressure.

Fraction of liquid phase in the mixture is equal to 0 at $P < P_H(T)$. If $P = P_H(T)$ the concentration c and temperature T are found from the solution of system of equations (3, 4).

This model is implemented in commercial CFD code FlowVision [1,2]. Modeling gas injector is based on solution of full Navier-Stokes equations with k-epsilon turbulence model.

GAS EJECTOR MODELING

The general form of ejector is shown in Fig. 1. Ejector consists of a nozzle for motive air and a long mixing tube.

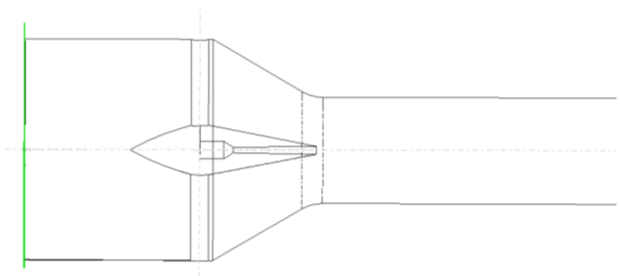
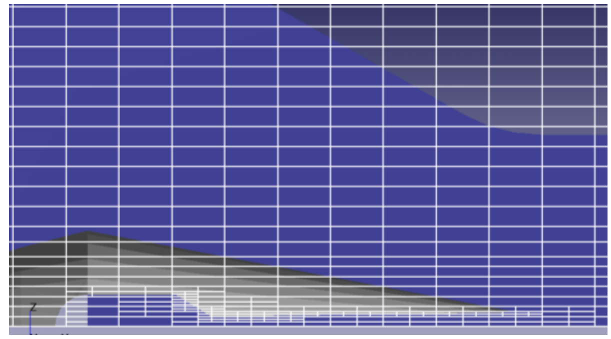
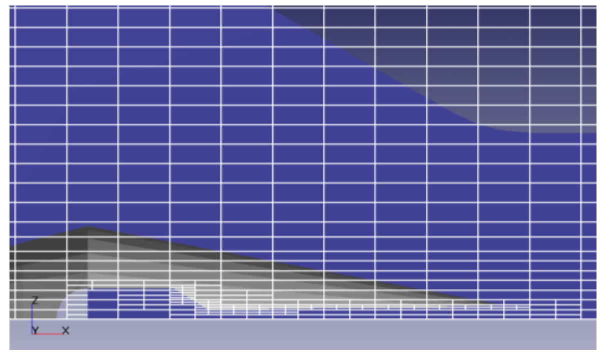


Fig. 1: Sketch of the air test ejector

Computations are completed on two grids - coarse and fine. These grids are shown in Figure 3. Fine grid differs from coarse one by adaptation near the nozzle of active air. Because of the geometry has two symmetry planes simulation is performed in quarter of ejector. Comparison of results carried out on coarse and fine grids shows that accuracy of coarse grid is enough to get reliable results.



a)



b)

Fig. 3: a) Coarse grid b) Fine grid

In Figure 4 and 5 the air flow near active air nozzle and entrance of mixing diffuser is shown. One can see that the air flow has no recirculation zones, that leads to high performance of the ejector. All pictures below are for simulation with an active air pressure 4.2 Bar.

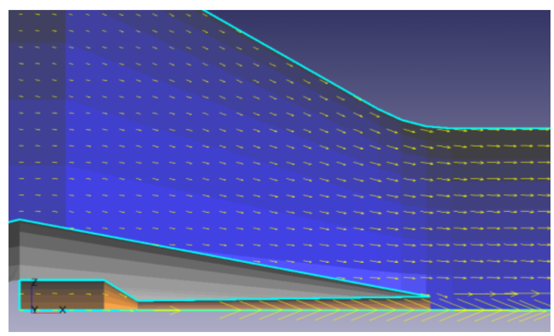


Fig. 4: Airflow near the active air nozzle and entrance of mixing diffuser

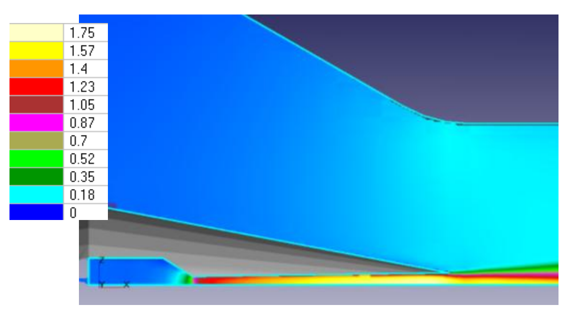


Fig. 5: Mach number distribution near the nozzle

Mach number distribution in mixing diffuser is shown in Fig. 6. One can see mixing of active air jet from nozzle with passive air.

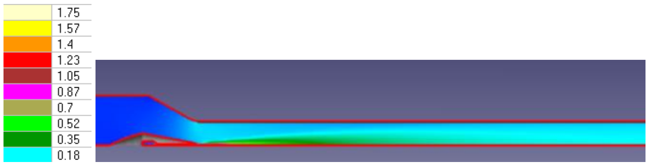


Fig. 6: Mach number distribution in mixing diffuser

Pressure of motive air was changed in experiment and in computation.

Experiment 1

	MOTIVE AIR	SUCTION
PRESSURE (relative values)	4.2Bar	0
TEMPERATURE ºC	18	18
FLOW RATE (simulated result, coarse grid) (simulated result, fine grid)	0.00154* kg/s (0.00136 kg/s) (0.00138 kg/s)	0.0288 kg/s (0.0276 kg/s) (0.0278 kg/s)

Experiment 2

	MOTIVE AIR	SUCTION
PRESSURE (relative values)	3.2Bar	0
TEMPERATURE ºC	18	18
FLOW RATE (computational result, coarse grid)	0.00124 kg/s (0.00112 kg/s))	0.0242 kg/s (0.0234 kg/s)

In computations the flow rate of motive air was calculated through formula for the Laval ideal nozzle and can differ from the measured one.

Comparison of experimental and computed flow rates is shown in Fig. 2. One can see good agreement of simulated results with experiment.

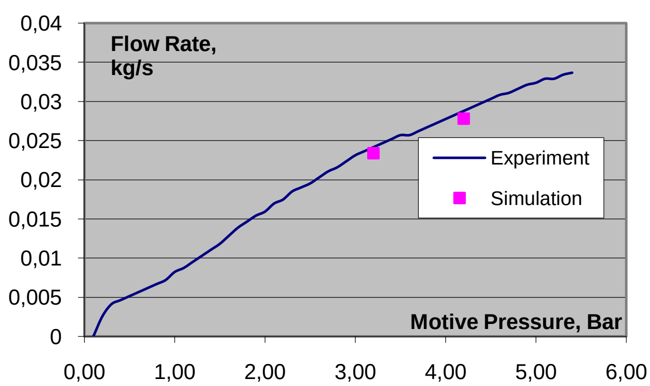


Fig. 2: Dependence of flow rate of passive air on pressure of active air

Difference of simulated and experimental results is less then 4% for the flow rate of passive gas.

STEAM EJECTOR MODELING

After successful modeling of the gas ejector, computations are aimed to study design alternatives for the actual steam multi-nozzle ejector.

Two series of computational tests are completed for two designs – ejector A and ejector B.

The general form of ejector is shown in Fig. 7.

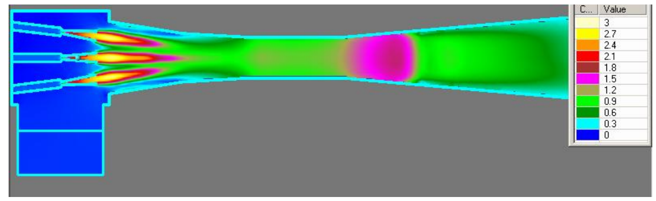


Fig. 7: Mach number distribution

Ejector A

For ejector A influence of two parameters is under the study:

d - the length of nozzle;

δ -the distance between nozzle exit section and input into the neck of ejector.

The purpose of calculations is determination of pressure in SUCTION.

Boundary conditions

Boundary conditions were preset as follows on the base of known parameters:

Designation of boundary condition	Type of the boundary condition		
	Flow rate	Temperature. ºC	Pressure
MOTIVE STEAM	5850 kg/hr	155	*Not defined*
SUCTION	3450 kg/hr	62	*Not defined*
DISCHARGE	*Not defined*	*Not defined*	256 mm Hg abs

Those parameters, which are noted in the table " not defined ", were determined in the code automatically. For example, on going on a stationary mode of operation pressure in THE MOTIVE STEAM differs from data of experiment not more than by 0.5%.

EJECTOR B

Boundary conditions

With the definition of boundary conditions were used experimental data, given below in the table.

Boundary conditions were preset as follows:

Designation of	Type of the boundary condition

boundary condition	Flow rate	Temperature. ºC	Pressure
MOTIVE STEAM	45500 kg/hr	147.2	*Not defined*
SUCTION	*Not defined*	47	80.0 mm Hg abs
DISCHARGE	*Not defined*	*Not defined*	*Not defined*

Those parameters, which are noted in the table " not defined ", were determined in the code automatically. For example, on going on a stationary mode of operation pressure in THE MOTIVE STEAM differed from data of experiment not more than by 0.5%.

The purpose of calculations was the determination of flow rate in SUCTION taking into account and without taking into account the process of the condensation of water vapor.

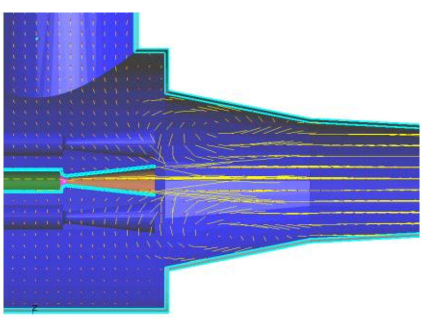
RESULTS OF CALCULATIONS

Ejector A

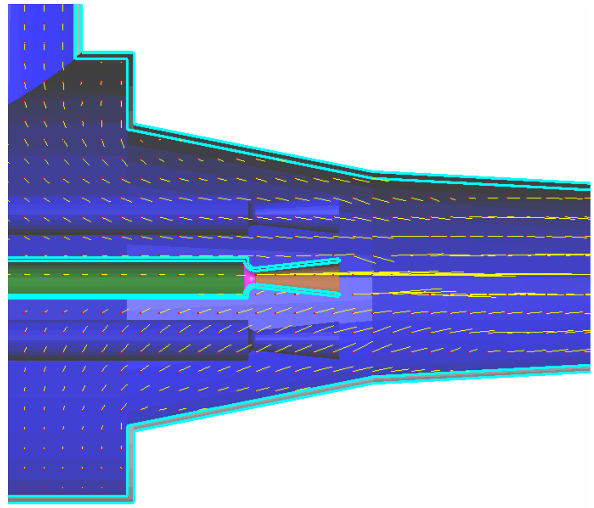
Below table gives the results of the calculations of six cases.

Case	Nozzle length d (m)	Distance δ (m)	Pressure in SUCTION (kPa)
0	0.25	0.4	26.5
1		0.2	21.1
2		0.1	21
3	0.15	0.2	18
4		0.1	16
5		0.05	15

Let us give the distribution of the velocities in the region of the input of jets into the neck of ejector for the case 0 and for the case 5. On the below figures SUCTION is located in the top.



Case. 0



Case. 5.

From the results obtained that recirculation zones substantially are reduced or eliminated completely in proportion to the decrease of distance. As it is known, the presence of recirculation zones leads to losses of pressure. The less these zones, losses of pressure less. And, as a result, in proportion to the decrease of recirculation zones pressure in SUCTION is reduced.

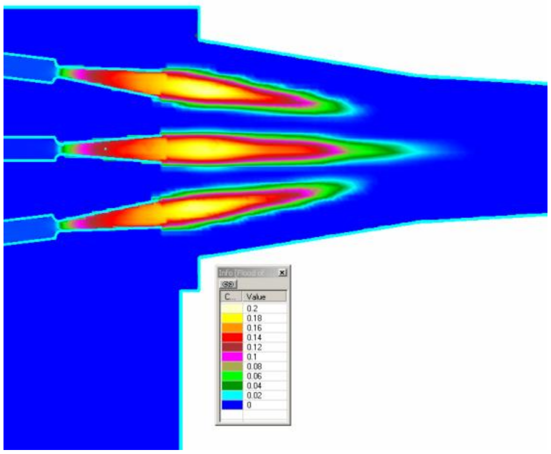
It is evident from the tabular results that the length of nozzle also substantially effects pressure in SUCTION.

Ejector B

Below table gives the results of the calculations of two versions taking into account and without taking into account the process of the condensation of water vapor.

	Flow rate in SUCTION (kg/h)
Without taking into account condensation	4680
Taking into account condensation	9936
Experiment	21261

As we see, the account of the process of condensation substantially influences the parameters of ejector. Lower figure shows the distribution of the condensate of water vapor in the ejector.



It is evident that the process of condensation occurs in the nozzle and on leaving from it. As it is known, the presence of condensate leads to lowering of pressure by this an improvement in the parameters of ejector upon consideration of the process of condensation is explained.

It is necessary to note that with this geometry of ejector recirculation zones appear in the region of the input of jets into the neck of the passive part of the ejector. Below there is presented distribution of longitudinal X- component of the velocity vector of jets in different cuts 1÷4 (see Fig. 3) for the case, which considers the condensation of vapor. Negative value the X- component of velocity vector means that the flow occurs towards inducing jets.

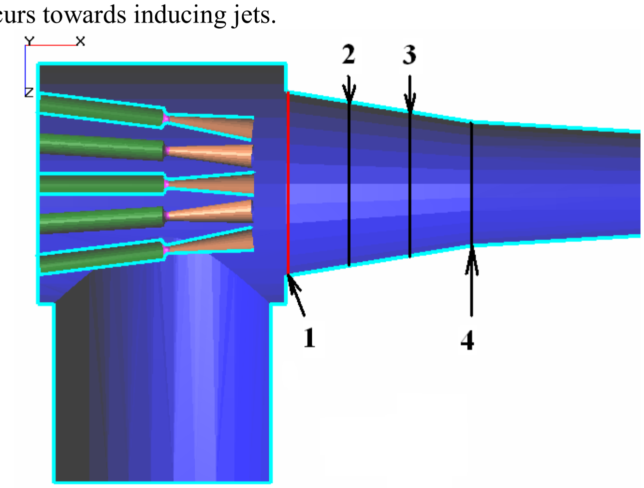
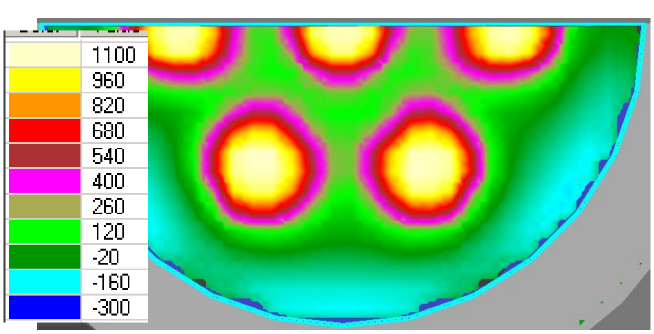
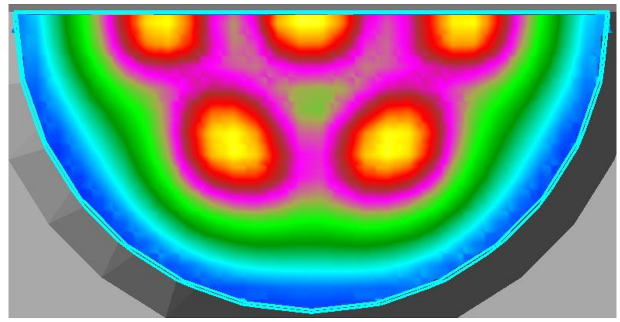


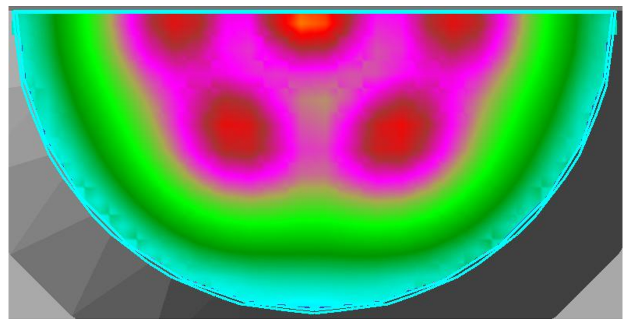
Fig. 3



Cut 1



Cut 2



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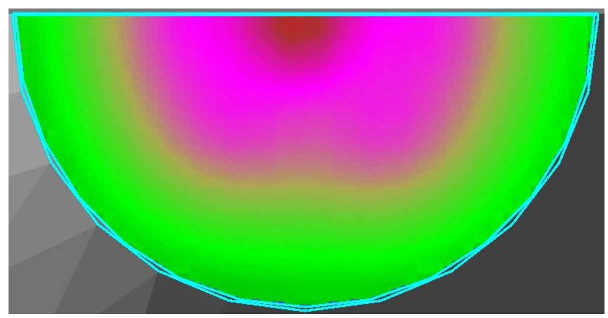

Cut 4

In case without condensation the structure of flow is analogous the one given above with the only difference that the maximum and minimum values of the axial- component of velocity vector are by 20% less.

It is evident from the given figures that the jet of vapor from the central nozzle is isolated from the outer flow by jets from surrounding nozzles. Since the effect of ejection is caused by the suction of the gas through the surface of jet, one should expect that the central jet practically does not provide the effect of ejection.

CONCLUSION

On the basis of carried out calculations for ejectors A and B it is possible to make the following conclusions:

Location of nozzle and its length is the important parameter of ejector. A change of these parameters in the ejector A in the limits of 100-300 mm leads to the essential – by a factor of 2-3 change in such parameters of ejector as the flow rate of the suction vapor. It is necessary to study also such parameters as the distance between the nozzle tubes, their diameter and angle of inclination from the axial direction.

The process of the condensation of water vapor significantly influences the work of ejector. Its account in the ejector B leads to an increase of the flow rate of suction by a factor of 2.

LITURATURE

[1] FlowVision User Manual, Capvidia, 2005, 310 p

[2] Aksenov A, Dyadkin A, Pokhilko V. Overcoming of Barrier between CAD and CFD by Modified Finite Volume Method, Proc. 1998 ASME Pressure Vessels and Piping Division Conference, San Diego, ASME PVP-Vol. 377-1., 1998