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EXPERIMENTAL INVESTIGATION OF AN INJECTOR DEVICE*

by

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The present work has been conducted within the framework of the program of studying injector accelerating devices with the use of steam-water models. The basic purpose of the work is to study the features of the characteristics [i.e. performance] of an injector using wet steam [vapor]. These investigations are of interest in connection with the fact that thermodynamic analysis shows that the optimum efficiency cycles of liquid-metal MHD plants [installations] must be accomplished on fairly wet steam. The operation of a wet steam injector along with the mixing chamber and diffuser [exit cone] includes the necessity of studying the operation of the steam [vapor] nozzle. This study is suitable both for conducting the study on a nozzle alone and on a nozzle coupled with the mixing chamber of the injector.

The results of the appropriate investigations are presented below.

1. *Operation of a wet steam nozzle*

The investigation of the most important characteristics of the flow of a steam-water mix in an axially symmetric supersonic nozzle was conducted in an especially developed plant. Wet water vapor served as the working medium; its initial degree of dryness was within the limits of x_0 = 1.0-0.25 at an initial pressure of p_0 = 1.2 atm abs.

A supersonic axially symmetric nozzle geometrically similar to the acceleration nozzle of the injector was the subject of the investigation. The calculated Mach number is equal to 1.8 at K = 1.3 for the nozzle. In order to investigate the distribution of the static pressure along the length of the nozzle, its wall was drained at 22 points, the diameter of the drain aperture 0.8 mm. The basic dimensions of the nozzle are clear from figure 2. The system of preparing the working mix made it possible to have practically a stable two-phase flow obtained by mixing saturated steam and saturated water (at pressure p_0) injected into the steam flow via centrifugal jets at the input into the nozzle. In order to determine the energy characteristics [performance curve] of the nozzle, a weight unit [literally: weight block] was developed that makes it possible to measure the reactive force and is a single-component tensometric scale using a non-balanced four-arm bridge with silicon conductor pickups. In accordance with estimates, the error in determining the coefficient of velocity is 3.5-4 per cent.

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In order to interpret the experimental results, it is necessary to select one or another theoretical model having a two-phase medium flow in the nozzle. The selection of this model is indispensible, for the strict theoretical calculation of the two-phase flows in the nozzle is not possible. Two possible models of two-phase flow are examined below. One of them presenting the most interest for evaluation of the ideality of the process is extremely stable. The other corresponds to a definite non-stability and at the same time is closer to the results of the experiments.

A. Extremely stable model

Assumptions concerning the isentropic character of the flow of the mix under a condition which both phases at any moment are located in equilibrium and have equal velocities are presented on the basis of thermodynamically stable model of the two-phase flow.

In calculating the isentropic expansion of the two-phase mix (particularly the wet water vapor), the analytic description of the specific flow rate $\bar{q}$ as a function of the ratio of pressures is impossible without including empirical relationships yielding a large error. Therefore, the numerical determination of the function $\bar{q}$ (ε_a) with the following graphic location of the critical values q and ε_a is uniquely possible by means of calculation. The accuracy of the calculation here depends on the accuracy of the initial data on the thermodynamic properties and the step in magnitude of ε_a. Such calculations were conducted for water vapor at p_0 = 1.2 atm. abs. and $0 < x_0 < 1$ on the "Minsk-22" computer, while the step in magnitude of the finite pressure p_0 was 0.01 atm abs, and magnitude of x_0 0.05. As a result of these calculations, generalizing dependences are obtained for critical values of $\bar{q}_{cr}(x_0)$ and $\varepsilon_{cr}(x_0)$ which are used below to compare experimental data with the theoretical as to flow rate of the impulse and the distribution of pressures.

B. Model of a flow of fixed composition

This model corresponds to the following assumptions [ref. #1]:

a) The velocities of the phases are equal;

b) Thermal and mass exchange between the phases are absent;

c) The expansion of the vapor occurs according to the adiabatic law at constant K (it was understood $K = 1.3$ and $K - 1.135$);

d) All kinetic energy of the flow is obtained as a result of the expansion of the vapor [steam];

e) The flow rate and the distribution of pressures is determined by the geometry of the nozzle and the properties of the vapor phase.

It can be shown that under these assumptions the basic calculation formulas have the form:

critical velocity of the mix:

$$C_{cr\ mix} = \sqrt{x_0 2 p_0 V_{v vap} \frac{K}{K-1}\left[1 - \varepsilon_{cr}^{\frac{K-1}{K}}\right]} = C_{cr\ vap}\sqrt{x_0} \quad (1)$$

where

$$\varepsilon_{cr} = \left(\frac{2}{K+1}\right)^{\frac{K}{K-1}}$$

$C_{cr\ vap}$ = the critical velocity in the flowing of the vapor

V_{vvap} = the specific volume of the vapor phase at p_0.

critical volume of the mix:

$$V_{cr \cdot mix} = x_0 [(\frac{1}{\varepsilon_{cr}})^{\frac{1}{K}} \cdot V_{vvap} \cdot V_{vliq}] + V_{vliq} = x_0 \cdot V + V_{vliq} \quad (2)$$

where

V_{vliq} = specific volume of the liquid phase,

critical flow rate of the mix through the nozzle

$$G_{cr\ mix} = \frac{C_{cr\ vap} \sqrt{x_0}}{x_0 (\Delta V + V_{liqn})} \cdot F_{cr} \quad (3)$$

where

F_{cr} = the area of the critical cross section of the nozzle.

The condition that $G_{cr} = 0$ when $x_0 = 0$ and the function $G_{crmix}(x_0)$ has a maximum when $x_0^m = V_m/\Delta V$ does not correspond to the actual pattern of the process, but is connected with the assumption that the kinetic energy of the mixture flow is obtained only due to the expansion of the vapor phase. It is natural that this assumption, and that means also formulas (1)-(3) are more justified, the greater x_0 is. Note that at low pressures of the water vapor the value $x_0{}^m$ is very small (when $p_0 = 1.2$ atm abs, $x_0{}^m \approx 0.045$ per cent). It is easy to show that under the assumptions made, the regime corresponding to the calculated value ε_a at an understood [assumed] K_0 will be the "calculated" regime of operation of the nozzle. The results of the calculations in the model of fixed composition are also used below for a comparison with the experiment.

Figure 1 shows the experimental data on the flow rate of the vapor through the nozzle. From the figure it is evident that both models of the flow yield smaller values of the flow rate so that the coefficients of the flow rate $\mu = \frac{G_{exper}}{G_{theor}}$ are substantially larger than unity. In comparison with the stable process, μ varies from 1.19 when $y_0 = 1.13$ ($y = 1 - a$) up to 1.32 when $y_0 = 0.73$, and in the model of fixed composition from 1.1 to 1.25 respectively.

These results differ from the data of [ref. #1] obtained at high p_0 (up to 70 atm abs) which are located between the indicated theoretical curves. As a result of this, one should come to the problem of whether the given theoretical models are extreme cases. As far as the stable model of the two-phase flow is concerned, the fact of exceeding the real flow rate above this theoretical value is quite widely known, however, it seems that under certain conditions such factors as structure of the flux and its inhomogeneity, the presence of low velocity film on the walls of the nozzle, initial slippage, absolute dimensions of the duct, and the regime of operation of the nozzle can lead to a decrease in the flow rate of the mixture in comparison to the stable one. Similar reasons should also relate to the model of fixed composition which, as is clear from the assumptions made in it, does not include in itself all possible conditions of accomplishing the maximum flow rate.

Figure 2 shows the experimental distribution curves of the static pressure along the length of the nozzle when p_0 = 1.2 atm abs, $\&_a$ = 0.15 when the degrees of wetness are different. For comparison, calculated cuves are drawn in graphs for models of fixed composition when K = 1.3 and K = 1.135 along with curves of the stable expansion process.

At small values of y_0 in the expanding part of the nozzle there is a jump in condensation transferring the flow from the recooled state to the stable. Before the condensation jump, the experimental curve is quite close to the calculated one at K = 1.3, which corresponds to the earlier obtained data of [ref. #2]. It is known that the place and intensity of the condensation jump is a function of the geometry of the nozzle, the initial parameters, the dispersity of wetness, and the like. In the present work this effect was not the subject of investigation.

At an initial wetness of $y_0 > 0.5$, apparently condensation jump processes do not take place since strong supercooling of the vapor flow is made difficult by the presence of a large quantity of the liquid phase. This result agrees with the data of the paper [ref. #1] where, under similar conditions, jump phenomena were not detected. Note that the comparison of the calculated values with respect to the drop in pressures at the nozzle at large values of y_0 with the data of [ref. #1] also agree very well.

From the presented graph, a significant growth of the relative pressure in the converging cross sections of the nozzle is clearly evident with an increase of the initial wetness and, consequently, a change in the calculated pressure drop. Thus, when y_0 = 0.22, $\&_p \approx 0.225$, and when $y_0 \approx 0.7$, $\&_p$ increases to 0.3. This condition is especially important for the correct selection of the degree of expansion of the acceleration nozzle and determining the conditions in the input cross section of the injector mixing chamber.

As has already been noted, a reactive force (thrust) during operation of the nozzle was measured in the described installation [plant]. This value can be represented by the equation

$$R = \frac{G_0}{d} \overline{C}_{out} + F_{out}(p_{mean} - p_a) \tag{4}$$

where

G_0 = is the experimental flow rate

$\overline{C}_{out(?)}$ = is the mean velocity of the flow in a section of the nozzle.

Furthermore, we suppose that the profile velocity in a section of the nozzle is so homogeneous that the mean velocities in the flow rate, impulse, and energy can be assumed to be identical;

F_{out} = the area of the output cross section,

$(p_{mean} - p_0)$ = the difference of pressures in the section and the surrounding environment due to the incalculability of the outflow regime which is also determined experimentally.

Let us introduce the coefficient of velocity of the nozzle y_n, defined as $y_n = \frac{\overline{C}out}{C_t}$, where C_t is a theoretic velocity calculated according to the stable temperature drop suggested from p_0 to p_a. Thus, the coefficient y_n characterizes not only the losses inthe duct of the nozzle, but also the incalculability of the nozzle of a given geometry in the regime in question, although the static term of equation (4) must also be included in the magnitude [value] of the impulse introduced by the nozzle into the mixing chamber of the injector. The experimental data are presented in figures 3a and 3b.

We shall call this, as is to be assumed, the calculated regime, when the pressure on the section of the nozzle coincides with the counterpressure ($p_c = p_a$). The experiment shows that in the calculated regime y_n is maximum, since losses are defined only by losses specifically in the nozzle duct. In the regimes of non-expansion ($p_c > p_a$), the losses in the nozzle are almost constant (the absolute value of the impules is constant), but y_n falls due to the growth of C_t. In superexpansion, it is seen from figure 3 that the coefficient of the velocity begins to fall. This phenomenon is observed also for nozzles operating on a single-phase gas medium and is associated with the change of the wave or eddy losses in the nozzle [ref. #3].

The experimental data presented on the flow rate and energy characteristics of the nozzle make it possible to estimate the characteristics and parameters of the flow at the input into the mixing chamber, parameters that are necessary for the further calculation of processes of the injector.

2. *Operation of a wet steam [vapor] injector*

In this paper the steam-water injector having a vapor [steam] nozzle, located along an axis, having conical mixing chambers of various length (from 100 to 600 mm) and a conical diffuser [exit cone] with cylindrical throat is investigated [The terms "steam" and "vapor" are used interchangeably in this paper, the Russian word for steam is simply [water] vapor.] The flow pattern of the injector is shown in figure 4.

The two-phase flow at the input into the vapor nozzle was created in the humidifying circuit by means of either eddy jets 1 (total number 26), or directly before the input into the vapor nozzle. By means of steam-blown jets II, the experiment showed no advantages of any of the examined types of humidifying devices. In order to investigate the efficiency of the operation of the injector, pressures of retardation of vapor p_{vvap} and retardation of water p_{vliq} at the input into the injector, the retardation pressure at the output of the diffuser p_{dif}, static pressure in the mixing chamber p , flow rates of the ejcted water G_w, superheated steam G_s, water at jets G_w, are measured. The initial degree of dryness at the input into the vapor nozzle x_0 varied within the limits of $x_0 = 1—0.01$ and was defined from the equations of heat and mass equilibrium of the humidifying devices. The efficiency of the injector was estimated by the internal relative efficiency calculated following the formula

[s = vap]

$$\eta_{0i} = \frac{G_{vap}(p_{dif} - p_{vvap}) + G_w(p_{dif} - p_{vliq})}{427\Delta H_0 \ \gamma_w} \quad (5)$$

where γ_w is the specific weight of the water, and

$\Delta H_0 = i_{vvap} - i_{liq(?)}$ is the assumed heat drop of the steam.

The working pressure of the steam was 3 kg/cm^2. The velocity of the liquid is w_{liq} = 6-8 m/sec, therefore, in the formula for η_{0i}, the value of the kinetic energy of the injected liquid is insignificantly small in comparison with the kinetic energy of the steam. The procedure [methodology] of the experiments assumed and the measuring instruments make it possible to determine this with an error that does not exceed 5%.

Operation of the mixing chamber

As is seen from figure 5a in which the distribution of static pressure is presented along the mixing chamber for dry saturated and wet (x_0 = 0.2) steam, the operation of the mixing chamber is not a function of the initial wetness of the steam in practice. The pressure on the nozzle section p_n for wet steam is greater than for dry saturated (or superheated) steam when the pressure in the mixing chamber p_K turns out to be lower for wet steam. This result is reflected in figure 5b where the dependences of p_n and p_K as a function of the initial degree of dryness of the steam are given at the input into the nozzle x_0.

The further path of the curve of static pressure along the mixing chamber and diffuser when x = 0.2 turns out to be the same as in the case of superheated steam; in front of the throat of the diffuser the static pressure rises smoothly, and at the throat there is assumed a condensation jump at which practically all the steam phase disappears. It is important to note that the condensation jump in the diffuser is preserved at all initial degrees of dryness from x_0 = 1 to x_0 = 0.05 and at different investigating lengths of the mixing chamber from 200 mm to 600, the intentensity of the jump (degree of increase of the pressure in the jump) in practice not being a function of x_0. There are reasons to assume that it is impossible to exclude this jump by any effect of regime parameters. It is not excluded that the indicated jump is associated with supersonic character of the flow of very wet or strongly moistened [wetted] steam (in such a flow the velocity of sound is 20 to 40 m/sec) and is a direct jump in condensation that carries the supersonic flow into the subsonic. At the present time the jump in the injector is being studied by us in detail. The results of this research will be published later.

An analysis of the distribution of static pressures along the length of the mixing chamber in different operating modes makes it possible to establish that for each mode the pressure in the chamber p_K is practically constant, however, in the transition from one regime [mode] to another x_0 changes by rendering sufficiently complicated the function of the coefficient of injection, temperature of the injected medium, initial pressure of dryness x_0 and geometric characteristics of the mixing chamber. In other words, the magnitude of the pressure p_K for each regime cannot be found a priori, but must be determined experimentally. However, in the first approximation it is shown that the mean magnitude of pressure p_K can be found as the saturation pressure defined by the temperature of the liquid at the output of the apparatus t_{ap}. This temperature t_{ap} is the basis of the thermal balance:

$$(1 + u)t_{mix} \approx i_{vvap} + u_{iliq}$$

Then the pressure in the chamber is equal to $\bar{p}_K = p_s(t_{mix})$. A comparison of the calculated magnitudes of $\bar{p}_K$ with the multiterm experimental curves of distribution of static pressure along the length of the mixing chambers showed that $\bar{p}_K$ characterizes the mean level of pressure in the chamber (see for example figure 5a). This condition makes it possible to conduct all programming of experiments uniquely according to the pressure $\bar{p}_K$, as well as using the results of these tests in designing the corresponding equipment [appropriate equipment].

Tests with the injector show that the steam nozzles [vapor nozzles] in all regimes except rupture (vaporization) operate with non-expansion, the pressure at section p_n growing with the rise in wetness when the pressure p_K decreases. Consequently, in the transition from the superheated steam to the wet, a redistribution of heat drops occurs [that are] developed at the nozzle H_n(figure 5c) and in the chamber beyond the section; the effectively used heat drop H_n decreases, and the losses associated with the non-calculated expansion of the jet beyond the section of the nozzle, equal to $(H_0 - H_n)$, increases. Obviously here the efficiency of the apparatus must decrease as a whole.

External characteristics of the injector in comparison with the theory

Figure 6a shows graphs of the pressure at the output of the apparatus p_d as a function of the coefficient of injection u for different initial degrees of dryness of x_0 (experimental curves are represented by solid lines). From the graph it is evident that the maximum value of p_d, corresponding to the optimum coefficient of injection is in practice not a function of the initial wetness x_0, i.e., for a given chamber geometry p_d remains practically constant.

The length of the mixing chamber and the length of the throat substantially affect the value of p_d. As is seen from figure 6c, the optimum length of the chamber L is 200 mm with a short throat (approximately 15 mm). For shorter chambers and longer p_d it turns out to be lower than for L = 200 mm. A chamber with a length of 600 mm with a throat diameter of 12 mm could not be started (inserted into a regime at which the pressure p_d is greater than the pressure p_{vvap}). The optimum chamber from the point of view of its external characteristics turned out to be a chamber with a length $L \approx$ 100 mm, with a throat length of 115 mm. The experimental points agree well with the calculations for this chamber.

This calculation was made following the procedure according to which the calculated magnitude [value] of the pressure beyond the diffuser p_d is determined by the formula:

$$p_d = p_n \quad \varphi_n^2 \frac{C_t^2 G_{vap}^2 \gamma_w}{d(C_{vap} + G_w)^2} \cdot \frac{2+\eta}{8} \qquad (6)$$

where

η = efficiency of the diffuser, and

γ = specific weight of the liquid

A comparison of the theory with the experiment showed that for a chamber of L = 215 mm, the agreement of the theory and the test is quite good, while the experimental data for the other mixing chambers lie below the theoretical. This is obviously explained by the fact that with the derivation of formula (6) friction is disregarded in the mixing chamber. In addition, formula (6) assumes that the condensation jump [jump in density] in the diffuser has a zero length while in actuality its length is quite great.

Figure 7 shows curves of the theoretical and experimental coefficients of efficiency. From the graphs it is evident that with the growth of wetness, the efficiency of the injector increases very slightly. The experimental and theoretical values of η_{0i} are 2-3 per cent. Such a low value of the efficiency is explained by the following causes:

1) Losses of energy in the nozzle increase sharply with the growth of wetness due to the fact that the efficiency of the injector becomes small.

2) As analysis shows, in order to obtain an optimum regime of operation of the injector simultaneously with the growth of the initial wetness at the input into the nozzle, the passage cross section of the throat of the diffuser must be reduced. Since the present experiments are being conducted at a constant diameter of the diffuser throat, the injector deviates from the calculated regime all the more with the growth of y_0.

3) In spite of the great moistures [wetnesses] in the vapor nozzle, the ratio of velocities of vapor and liquid in the mixing chamber is 70-20. The shock losses under such ratios are very great. Therefore, further development of operations must be conducted in the direction of increasing the velocities of the liquid at the input into the mixing chamber.

References

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Figures

1. Critical passage of wet steam through a nozzle (p_0=1.2 atm)

2. Distribution of static pressure along the length of a steam nozzle

3. (a) Dependence of the coefficient of nozzle velocity y_n on the operating conditions (y_o(p_0=1.2 atm)

   (b) The value y_n^{max} depends on the initial stages of moisture

4. Schematic of the injector plant

5. (a) Distribution of static pressure along the length of the circulating parts

   (b) The influence of the initial moisture on the static value

   (c) The influence of initial moisture in the process of vapor expansion in the injector

6. Influence of the initial degree of dryness on the output pressure

7. Influence of the initial degree of dryness on the efficiency of the injector.