OPTIMIZATION OF GAS-JET EJECTORS WITH CONVERGING CHAMBER

A.V. Sobolev

Khristianovich Institute of Theoretical and Applied Mechanics, SD RAS, Novosibirsk 630090, Russia

In the development of gas-jet ejectors, the choice problem has to be solved for quantities defining the trade-off between the possibility to start up the ejector and the desire to achieve maximum possible ejector discharge characteristics.

The properties and stagnation parameters of the primary gas and those of the secondary gas are normally assumed known. For a unique solution to be separated out, one has to additionally set the normalized velocity of the secondary flow in the aerodynamic throat $\lambda_s$ and the normalized velocity of the mixed flow at the mixing-chamber outlet $\lambda_3$. For these quantities to be determined accurate to (5 - 10)$\%$, an account of losses is required. With the optimal values of normalized velocities $\lambda_s$ and $\lambda_3$ set together with the losses, it becomes possible to uniquely determine the maximum mass flow rate ratio that can be achieved at a given compression ratio $\varepsilon$, and also geometric and gas-dynamic characteristics of the ejector.

Initially, properties of gas ejectors were examined experimentally. Several air-ejector designs with central primary nozzle were put to tests. The varied parameters in these tests were the mixing intensity, the length and contour of the mixing chamber, and the throat cross-sectional area. As mixing intensifiers, tabs were used [1]. Ejector designs providing for best discharge characteristics were chosen. For these designs, calculations were carried out, and the obtained data were then generalized with empirical dependences. The normalized velocity in the aerodynamic throat was shown to depend on the mass flow rate ratio $k$ linearly, $\lambda_s=0.35+0.9k$, and the loss factor, to be defined predominantly by the wall friction, $\zeta=0.02(l_{23}/d_2+1)$, where $l_{23}$ is the supersonic flow length between the complete-mixing section and the throat inlet.

The calculations were performed for the ejector design shown in Fig. 1. Over the section between inlet section 1 and the aerodynamic throat, the primary and secondary gas flows undergo acceleration. The static pressures of the two flows are assumed to equalize in the aerodynamic throat. Over the section between the aerodynamic throat and complete-mixing section 2 the gas flows undergo mixing at constant pressure. The distribution of flow quantities in section 2 is one-dimensional, and the flow velocity is supersonic. The supersonic flow is decelerated prior to entering throat 3. In throat 3-4 the supersonic flow transforms in a subsonic flow as it traverses a pseudo-shock. Next, in the subsonic diffuser the flow gets decelerated to small subsonic velocities, and the static pressure increases to finally reach the total-pressure value.

First, the flow quantities in the aerodynamic

\[ \lambda_p; p_p = p_s \]

\[ \lambda_s; p_s \]

Fig. 1. The ejector contour.
throat are to be calculated. The pressure in the aerodynamic throat is \( p = p_0 \pi(\lambda_s) \). The characteristics of the primary gas flow are to be calculated using the equilibrium-condensation relations. The amount of the heat released due to condensation without allowance for losses is given by the formula \( \mu r = (S_g - S_0) T_c \). Here, \( \mu \) is the mass fraction due to precipitated liquid, \( r \) is the heat of evaporation, \( S_g \) is the entropy of the gas phase of the primary substance in the aerodynamic throat, \( S_0 \) is the entropy of the primary gas in the volume upstream of the nozzle, and \( T_c \) is the dew-point temperature at aerodynamic throat pressure \( p \). The velocity of the primary substance in the aerodynamic throat is

\[
\quad u_p = \varphi \sqrt{2(1 - h_0 - h_g + \mu r)}.
\]

Here, \( h_0 \) is the primary-gas enthalpy at the nozzle inlet, \( h_g \) is the temperature of the gas phase of the primary substance in the ejector aerodynamic throat, and \( \varphi \) is the nozzle velocity ratio.

The flow velocity in complete-mixing section 2 is

\[
\quad u_2 = \frac{u_p + k a_s \lambda_s}{(1 + k)(1 + \zeta / 2)}.
\]

Here, \( k \) is the mass flow rate ratio, \( a_s \) is the critical sound velocity in the secondary gas flow, and \( \zeta \) is the loss factor. The total pressure in the gas mixture is

\[
\quad p_{02} = p / \pi(\lambda_2).
\]

With precipitated liquid present at section 2 the calculation procedure for the total pressure becomes more complicated; in the latter case the total pressure is to be determined as the pressure of an isentropically decelerated liquid-gas mixture. The total pressure in section 3, at the throat inlet, remains unchanged, \( p_{03} = p_{02} \), and the throat pressure falls in value like that across a normal shock, \( p_{04} = \sigma(M) p_{03} \). At design flow regimes the losses in the subsonic diffuser were taken into account via the relation \( p_{04} = 1.08p_h \).

The existence of an optimal value of \( \lambda_s \) is defined by properties of the isobaric mixing process. With increasing \( \lambda_s \), the normalized velocity \( \lambda_2 \) also increases, thus leading to increased pressure \( p_{02} \). Yet, in the latter situation the static pressure in the chamber decreases, causing a decrease of \( p_{02} \). The joint action of the two factors results in that the discharge characteristics reach a maximum at some optimal value of \( \lambda_s \). Figure 2 shows the total pressures \( p_{02} \) calculated by formulas (2) and (3) and normalized to the maximum pressure values. For the velocity of the primary gas flow, a value \( u = 2.1a_s \) was adopted. With increasing the mass flow rate ratio, the total-pressure maximum is displaced towards high values of \( \lambda_s \).

The optimal values of \( \lambda_s \) in air ejectors with regard for all factors were revealed in calculations performed at given total pressures of the two gas flows. The loss factor was calculated by the above formula, with the length \( l_{23} \) found for the cone generatrix sloping angle equal to \( 3^\circ \). In all subsequent calculations the normalized flow velocity at the throat inlet was fixed, equal to \( \lambda_3 = 1.4 \). The calculated optimal values of \( \lambda_s \) in ejectors with identical total temperatures of

![Fig. 2.](image-url)
the primary and secondary gas flows at mass flow rate ratios \( k > 0.2 \) turned out to be over 0.8, see Fig. 3. These values somewhat exceed experimentally obtained values. Possible cause for this is incomplete mixing of the two flows in the ejector. With incomplete mixing assumed, the mixing-chamber pressure increases, and the normalized velocity \( \lambda_s \), decreases.

In the development of gas ejectors, the following approximate law of similarity is often used:
\[
k = \frac{k_0}{\sqrt{\theta}} .
\]
Here, \( k_0 \) is the mass flow rate ratio at \( \theta = 1 \). At temperature ratios between the secondary and primary gas flows \( \theta = T_{0s}/T_{0p} \) strongly differing from unity the approximate law of similarity is violated. The calculated mass flow rate ratios are notably lower than the approximate values, which at \( \theta = 0.25 \) and 4 are multiples, with the factor two, to the values at \( \theta = 1 \), see Fig. 4, a. The approximate values results with the critical sound velocity in the complete-mixing section \( a_{s2}/a_{sp} = \sqrt{(k\theta + 1)/(k + 1)} \) (see formula (2) with \( u_2 = \lambda_2 a_{s2} \) substituted into the left-hand side) replaced with the approximate value of this velocity
\[
a_{s2}/a_{sp} = (k\sqrt{\theta} + 1)/(k + 1). \]
The exact values of the critical sound velocity are up to 5% greater than the approximate values. At fixed mass rates of the two flows, the exact value of \( \lambda_2 \) in the complete-mixing section turns out to be lower by the same value. Small variations of the normalized velocities at high compression ratios dramatically affect the total pressure, this strong influence providing an explanation why the exact mass flow rate ratios differ so much from their approximate values. The violation of the approximate relations makes the choice of ejector parameters difficult a problem. For the parameter values to be properly chosen, calculations with varied \( \theta \) are required.

The calculated relative throat cross-sectional areas for ejectors intended for use at \( \varepsilon = 10 \) and \( \theta = 1 \) are small, \( A_3 < 0.37 \), see Fig. 4, b. The possibility of ejector starting being defined by the rela-

![Fig. 3. Variation in mass flow ratio with normalized velocity in the aerodynamic throat.](image)

![Fig. 4. Characteristics of air ejectors operated at \( p_{0p}=35 \) bar and \( p_{0s}=0.1 \) bar. Quantities plotted along the vertical axes are the mass flow rate ratio (a) and the relative throat cross-sectional area (b).](image)
tive throat cross-sectional area, the startup of such ejectors seems to be problematic. The starting characteristics can be improved by increasing the normalized velocity $\lambda_3$ and the throat cross-sectional area; yet, as a result of these measures, the discharge characteristics become deteriorated. Ejectors with one of the two gas flows being a hot flow are believed to allow starting and have good discharge characteristics at $\lambda_s=0.5-0.6$.

To elucidate the effect due to $\theta$ in air ejectors, calculations of two types were performed. In the calculations of the first type the primary-flow temperature was fixed, equal to $T_{0p}=290\ K$, whereas the secondary flow temperature was a variable quantity (open circles in Fig. 5). In the calculations of the second type the secondary-flow temperature was fixed, equal to $T_{0s}=290\ K$ (squares in Fig. 5). It should be noted here that the value $k=0.235$ at $T_{0p}=T_{0s}$ is an experimentally validated value. With increasing the temperature ratio $T_{0p}/T_{0s}$, the mass flow rate ratio grows in value to asymptotically approach the value $k=0.34$ at $T_{0p}/T_{0s}=4$. Apparently, the use of primary gas flows with a temperature more than twice greater than the secondary-flow temperature makes the device performance deteriorated because of increased energy consumption. At temperatures of the primary air flow 200 - 240 K the reduction displayed by $k$ gets slowed down at the level of $k=0.225$ due to oxygen and nitrogen precipitation in the primary nozzle. In Fig. 5, the straight line passing through the origin shows the approximate relation $k=k_0/\sqrt[\theta]{(T_{0p}/T_{0s})}$. The exact mass flow rate ratios are lower than the approximate values by 8-8.5% at a twofold difference between the temperatures of the two flows.

Quite a different behavior is exhibited by the mass flow rate ratio versus temperature in water-vapor-driven ejectors. Figure 6 shows the calculation data for ejectors with $\varepsilon=10$. Here, the primary gas is water vapor, and the secondary gas is air with temperature $T_{0s}=290\ K$. At the rightmost point, $T_{0p}=1073\ K$, the water vapor outflows from the nozzle experiencing no condensation. The mass flow rate ratio in the case of suction performed with a 15-bar water-vapor flow is the same magnitude as in the case of suction with 35-bar airflow, $k=0.34$. On decreasing the total water-vapor temperature, the outflow from the nozzle becomes accompanied with condensation, and at temperatures below 950 K the mass flow rate ratio starts increasing. The heat release due to condensation acts to increase the normalized velocity of the wet steam flow. The heat evolving during the condensation and the normalized velocity of the wet stream flow are shown in the upper graph of Fig. 6 respectively with triangles and rhombuses. The absolute velocity of the primary jet flow decreases with decreasing the temperature $T_{0p}$; yet, a quantity of even greater importance here is the normalized primary-flow velocity. For this to be figured out, consider formula (2) that shows that the normalized velocity of the mixed flow in the complete-mixing section depends linearly on the normalized velocity of the primary water-vapor flow, $\lambda_2=a\lambda_p+b$. Despite the fact that, at a fixed mass rate flow ratio, the coefficients $a$ and $b$ both depend on gas parameters, a qualitative behavior displayed by the above dependence during variation of the primary-flow stagnation temperature is retained here. According to formula (3), the total pressure in the mixed flow $p_{02}$ also becomes a function of $\lambda_p$. As a result, the increased total pressure $p_{02}$ resulting from increased value of $\lambda_p$ improves the ejector discharge characteristics.
At temperatures $T_0 < 670$ K the condensed phase first appears at complete-mixing section 2 (open circles in Fig. 6). As previously, here the mass fraction of precipitated liquid $\mu_2$ is expressed as the mass fraction of the primary water-vapor flow. The precipitation in the mixing chamber proceeds with increased temperature, decreased Mach number, and decreased total pressure of the mixed flow in comparison with the values the same quantities have in supercooled mixture. With increased mass fraction of the water precipitated in the chamber the growth displayed by the mass flow rate ratio becomes decelerated. At water-vapor temperatures below 495 K a small amount of water appears at the throat inlet. The influence the precipitated water has on the recovery of total pressure in the throat was ignored in the present calculations. The mass flow rate ratio attains its highest, $k = 0.465$, at the nozzle-inlet water-vapor temperature 473 K. At this temperature, the water vapor at the inlet to the nozzle becomes saturated.

The mass fraction of the water precipitated at the complete-mixing section can be determined by the method of successive approximations. At a chosen value of $\mu_2$ we calculate the mass fractions of the gaseous water vapor and air in the mixture and, subsequently, the molar fractions of these mixture components. Then, the partial pressures due to the components are to be calculated. From the water-vapor partial pressure we determine the dew-point temperature. The equilibrium temperature of the mixture is equal to the dew-point temperature. Now, using the calculated temperature and pressure values, we calculate the enthalpies of all mixture components and the total enthalpy of the mixture. The refined value of $\mu_2$ can be determined from the enthalpy balance between the ejector inlet and the complete-mixing section.

The total pressure in the gas-liquid mixture was calculated as follows. First, we calculate the entropy $S_2$ of the gas-liquid mixture at the complete-mixing section. Then, we calculate the molar fractions of stagnant components under no-condensation conditions. From the energy equation, we then determine the stagnation temperature of the mixture $T_{02}$. Then, the tabulated stagnation-pressure value for the gas-liquid mixture and the tabulated stagnation-temperature value are to be set as $p_t \approx 1.2 p_h$ and $T_t \approx T_{02}$. Then, the entropies $S_p$ and $S_s$ of the mixture components are to be determined for the stagnation temperature and pressure values. Then, we calculate the partial pressures $p_p$ and $p_s$ of the components at the tabulated value of $p_t$, and also the entropy values at known partial pressures of the components and at tabulated stagnation temperature:

$$S_p = S_{p0} + R_p \ln(p_t / p_h); \quad S_s = S_{s0} + R_s \ln(p_t / p_h).$$

From the component entropies, we calculate the entropy of the mixture at tabulated stagnation-pressure and stagnation-temperature values:

$$S_t = S_{pg} + S_{gs}.$$  

Here, $g_p$ and $g_s$ are the mass concentrations of the components. With the obtained values, the perfect-gas total pressure $p_{02}(T_{02}, S_2)$ can be calculated as

$$p_{02} / p_t = (T_{02} / T_t)^{\gamma / (\gamma - 1)} e^{S_t - S_2} / R.$$  

Here, $R$ is the gas constant, and $\gamma$ is the ratio of specific heats of stagnant mixture.
The mean specific heat at constant pressure of the wet steam expanding in the nozzle (see formula (1)) is
\[ c_p = \frac{h_0 - (h_g - \mu r)}{(T_0 - T_g)}, \]
and the ratio of specific heats is \( \gamma = \frac{c_p}{(c_p - R)} \). With the obtained value of \( \gamma \), we can calculate, in a traditional manner, using gas-dynamic functions, the parameters of the wet steam flow. The values of \( \gamma \), are shown in Fig. 6 as overturned triangles. The ratio of specific heats increases with increasing water-vapor temperature from \( \gamma = 1.10 \) to \( \gamma = 1.28 \). It is in this way, using the value of \( \gamma \) obtained on the basis of water-vapor enthalpy-entropy diagram, the nozzle discharge velocity in [2] was calculated.

The relative throat cross-sectional area at the maximum mass flow rate ratio is small, \( A_3/A_1 = 0.25 \) (squares in Fig. 6). The startup of ejectors with such narrow throat may appear difficult; in this connection, here we present calculation data for ejectors with lower pressure ratio, \( \varepsilon = 5 \), see Fig. 7. The calculations were performed for three temperatures of the secondary air flow. With increasing the latter temperature, the condensate mass fraction \( \mu_2 \) at the complete-mixing section decreases. As a result, the maximum mass flow rate ratios achievable with saturated water vapor decrease in value much more slowly than \( T_0 s^{-1/2} \). The relative throat cross-sectional area in these ejectors is over 0.34.

The detrimental effect owing to the water precipitated in the mixing chamber becomes evident if we compare the properties of water-vapor-driven ejectors with air ejectors. The effect owing to the water precipitated in the ejector chamber is insignificant at the secondary-flow temperature 700 K. Here, the mass flow rate ratio in the water-vapor-driven ejector is 0.66. The mass flow rate ratio in the air ejector with \( \varepsilon = 5 \) operated at the temperature ratio \( \theta = 700/290 = 2.41 \), with other parameter values as indicated in Fig. 5, is 0.24. The increase of the mass flow rate ratio in the water-vapor-driven ejector amounts to 175%. The amount of the water precipitated in the chamber becomes considerable at a secondary-flow temperature of 290 K. The mass flow rate ratio in the water-vapor-driven ejector operated at \( \theta = 290/473 = 0.725 \). The mass flow rate ratio in the air ejector operated at \( \theta = 290/290 = 0.42 \). The increase in the mass flow rate ratio for the water-vapor ejector with the condensate present in the chamber is much lower, equal to 71%. These examples show that the use of water-vapor ejectors is more advantageous at increased secondary flow temperatures.

It should be noted finally that the present calculations were performed for the nozzle velocity ratio value \( \phi = 0.97 \). In the case of a two-phase mixture the losses in the nozzle increase and, as a result, the velocity ratio may assume lower values [3]. By way of example, ejectors with water-vapor temperature 473 K and nozzle velocity ratio \( \phi = 0.95 \) were calculated. For an ejector with
ɛ = 10, a mass flow rate ratio \( k = 0.42 \) instead of \( k = 0.465 \) (see Fig. 6), i.e. a 9.7-% lower value, was obtained, whereas for an ejector with \( ɛ = 5 \) the calculations yielded a value \( k = 0.67 \) instead of \( k = 0.725 \) (see Fig. 7), i.e., a 7.6-% lower value.

**Conclusions**

Critical regimes of air- and water-vapor-driven ejectors were calculated in a broad range of temperatures of the primary and secondary gas flows with regard for equilibrium condensation processes and with allowance for losses in the primary nozzle and in the ejector channel.

It is shown that in air-jet ejectors operated with different temperatures of the primary and secondary flows the mass flow rate ratios are lower than the rate ratios predicted by the approximate-similarity theory. The growth the mass flow rate ratio exhibits with increasing the temperature of the primary air flow is limited, displaying saturation at a constant level.

The calculated ejector discharge characteristics can be improved by increasing the normalized velocity of the secondary gas flow provided that the mass flow rate ratio is greater than 0.2. As an optimum values of the latter velocity, based on experimental data, the value \( \lambda_s = 0.6 \) was chosen.

In the case of water-vapor-driven ejectors, an inverse dependence on temperature was obtained, with highest mass flow rate ratios obtained at the temperature of dry saturated water vapor. The absolute and normalized velocities of the water-vapor jet flow are notably higher than the velocities of the air jet flow; as a result, better discharge characteristics can be achieved with water-vapor-driven ejectors. The advantage the water-vapor-driven ejectors have over air ejectors becomes most distinctly pronounced when the ejectors are used to suck hot secondary gases. This advantage, however, vanishes when the ejectors are used to suck cold gases because of the condensed phase precipitating in the mixing chamber and deteriorating the pressure recovery.

This work was supported by the Russian Foundation for Basic Research (Grant No. 05-08-01215).

**REFERENCES**