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# Experimental investigations of the effect of nozzle geometry on performance of water-air ejectors

The paper deals with experimental investigations of a two-phase water-air ejector. The aim of research is to find out the influence of the ejector geometry on its performance. Particular attention is paid to the influence of the nozzle geometry on the gas suction rate and the ejector efficiency.

## Nomenclature

$A$ – cross-sectional area, m$^2$,
$b$ – nozzle/chamber area ratio, $b = A_n/A_{th}$
$D$ – diameter, m
$K$ – constant, see Eq. (5),
$K_{th}$– friction loss coefficient for mixing chamber, –
$L$ – length, m
$\dot{m}$ – mass flow rate, kg/s
$p$ – pressure, Pa
$T$ – temperature, °C,
$\dot{V}$ – volumetric flow, $m^3/s$

$w$ – velocity, m/s
$\gamma_{aw}$– density ratio, $\gamma = \rho_{ai}/\rho_w$
$\gamma_{12}$ – density ratio, $\gamma = \rho_{2i}/\rho_1$
$\chi_i$ – volumetric flow ratio, $\chi = \dot{V}_a/\dot{V}_w$
$\eta_T$ – isothermal efficiency, see Eq. (11),
$\rho$ – density, kg/m$^3$
$\Pi$ – dimensionless compression, $\Pi = (p_d - p_{ai})/(p_{wi} - p_{ai})$
$\phi$ – velocity coefficient, dimensionless

### Subscripts

$a$ – air,
$ai$ – air at suction chamber,
$b$ – barometric,
$d$ – diffuser,
$n$ – nozzle,
$o$ – mixing chamber entrance,
$s$ – saturation,
$t$ – mixing chamber exit,

$th$ – mixing chamber,
$w$ – water,
$wi$ – water at the nozzle entrance,
$1$ – motive fluid,
$2$ – secondary fluid,
$1i$ – motive fluid at the nozzle entrance,
$2i$ – secondary fluid at suction chamber.

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## 1. Introduction

During recent years one can observe widespread use of two-phase ejectors. This situation takes place in various branches of practical applications such as: conventional power, refrigeration, chemical engineering, sanitary technique, food processing industry, vacuum technique and others. Gas-liquid two-phase ejectors enjoy a particular favour, where a liquid phase is a motive medium and gas or vapour is sucked [1].

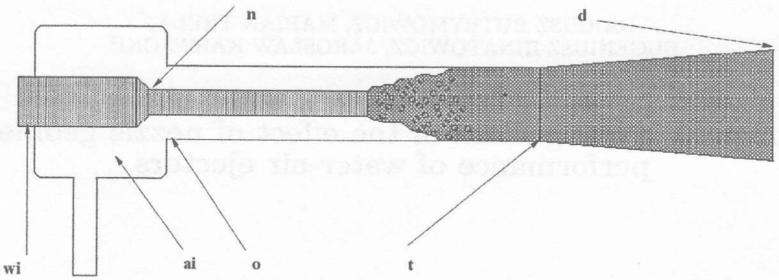



Fig. 1. Pictorial diagram of liquid-jet ejector.

The merit of the present paper is to investigate a two-phase liquid-gas ejector, schematically presented in Fig. 1. The ejector is fed by the stream of the motive liquid. The liquid inlet cross-section to the ejector is denoted by the symbol $wi$. Motive liquid expands in the nozzle. The outlet cross-section is denoted by $n$. The stream of gas is supplied to the suction chamber of the ejector. The inlet cross-section is denoted by the symbol $ai$. The entrance cross-section of the mixing chamber is denoted by the symbol $o$. At the entrance of the mixing chamber the annular flow of liquid and gas occurs, where two phases flow separately. At some location in the mixing chamber the liquid stream looses its stability and the split of the stream takes place. The mixing section is filled out with a water-air froth. This is the region of formation of the shock wave, which causes mixing, beyond which we are dealing with a homogeneous two-phase flow (wispy flow) of liquid and gas. At this location there is an increase of the static pressure of the two-phase medium. It ought to be stressed that in the entry part of the mixing chamber the reversing currents of the froth occur, which reverse against the main stream and then are captured by the main stream and conveyed to the final part of the mixing section and further to the diffuser. The end of the mixing section is denoted in Fig. 1 by the symbol $t$. Due to the increase of the cross-section in the diffuser there is observed an increase of the pressure of the two-phase medium. The outlet from the diffuser is denoted by the symbol $d$.

It should be stressed that processes taking place in the ejector are very complex [2,3], primarily due to the presence of a two-phase shock wave, as well as the change of the flow structure in the mixing chamber. If the two-phase flow formed in the region of the shock wave will be regarded as homogeneous, then we can prove that the flow upstream the shock wave is subsonic. Witte [4] calls that phenomenon as mixing shock. Mixing shock shows some similarities and differences, when compared against a shock wave of a compressible fluid. The similarities are as follows:

- In the region of shock mixing there is the pressure increase and the decrease of velocity.
- Phenomena of the pressure increase and the velocity decrease are accompanied by the kinetic energy dissipation.

On the other hand the differences are as follows:

- Processes taking place in the mixing wave can be regarded as isothermal.
- Upstream to the mixing wave the interface slip usually occurs.
- The length of the mixing wave is much more greater than the thickness of a gasdynamic shock wave.

It must be stressed that the present state of knowledge in the area of physical phenomena taking place in the liquid-gas ejectors is highly unsatisfactory as far as prescription of accurate performance characteristics is concerned. Particularly important, from the practical point of view, is the knowledge of the influence of principal geometrical parameters of the ejector on its performance and available efficiency.

In the up to date publications the diameter ratio $b$ of the motive nozzle to the mixing chamber is regarded as the most important geometrical parameter [2÷16].

Systematic investigations of the influence of the ratio $b$ as well as the motive nozzle diameter profile on the efficiency was conducted by Cunningham [11]. A conclusion may be drawn from these investigations, that from the point of view of the efficiency the most appropriate is to use a sharp-edged nozzle rather than convergent nozzle, for example of Bendemann type. This result was explained by Cunningham in such a way, that in the case of convergent nozzles he obtained lower values of the velocity coefficient. Additionally, he suggested that in the case of convergent-divergent nozzles there is a sudden flow distribution in the suction chamber, which results in impairment of gaseous phase supply to the mixing chamber. In the case of a nozzle with a conical profile one can expect existence of significant friction losses close to the nozzle outlet, primarily due to the boundary layer separation. Cunningham [11] pointed out attention to the fact, that in the case of the nozzles with different geometries, an optimal length of the mixing chamber varies, which seems to be consistent with the results of experimental investigations presented by the authors in the paper [1].

Explanation given by Cunningham [11] cannot be regarded as fully reliable in the case of application of a sharp-edged nozzle from the point of view of physical phenomena, which take place in the ejector, because a key problem in that case is the influence of a nozzle geometry on the flow split in the mixing chamber, which was one of the incentives to conduct present investigations. It can be expected, that even in the case of lower value of the velocity coefficient of the motive nozzle, the higher value of the ejector efficiency could be achieved.

It ought to be noted, that some researchers [13] suggest, that the effect of the diameter of the motive nozzle is independent on mixing section diameter, i.e. for the same value of the ratio $b$ obtained can be different values of the compression ratio and the ejection coefficient at similar parameters of liquid.

It should also be added, that one of the methods of improvement of the ejector efficiency is application of a multi-nozzle system. Such solution has been applied for the first time in single phase liquid ejectors, however experimental investigations proved a high efficiency of such solution in the case of two-phase ejectors. Witte [4] obtained the highest efficiency of the ejector using liquid feeding through 19 nozzles.

In practice swirling tapes prior or beyond the nozzle [17] are used apart from the solutions described above, which are aimed at intensification of momentum transfer between phases and in effect improvement of the mixing process and the efficiency of the ejector operation.

It should be noted, that in principal, remaining geometrical parameters received considerably less attention, and in that case a very little attention has been devoted to the length of the mixing chamber, which has been pointed out by the present authors in the paper [1].

It must be however stressed that such state results from the widely assumed point of view that two-phase liquid-gas ejectors can be modelled using the theory developed for single phase liquid ejectors [2, 3, 5, 12], where the diameter ratio $b$ is a principal parameter. Derived also have been equations for liquid-gas ejectors after some relevant changes to single phase flow descriptions and in some cases satisfactory results have been obtained [5]. That issue has been critically discussed in the following part of the present paper.

In the light of own experimental investigations presented in the paper [1] we can conclude that there exist important deficiencies of above mentioned approach. It is also important, that the amount of available in the literature results of systematic investigations in the area of the influence of the motive nozzle geometry on the ejector performance has to be regarded as highly unsatisfactory. Taking the above into account, the issue of the influence of fundamental geometrical parameters and the motive nozzle profile remains still open and requires systematic experimental investigations. A principal aim of investigations presented in the present work was to determine the influence of the motive nozzle profile and the value of the module $b$ on the efficiency of the two-phase ejector.

## 2. Research stand and measurement procedure

The testing rig as well as the measurements procedure are described in detail in the accompanying paper [1].

In presented investigations utilised have been two types of motive nozzles: a sharp-edged one and a Bendemann one. The shapes of these nozzles are showed in

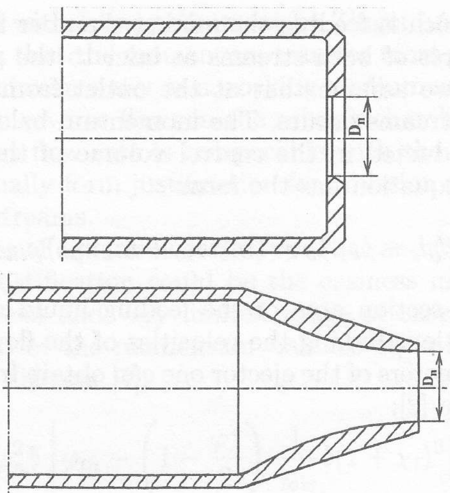



Fig. 2. The geometry of investigated nozzles.

Fig. 2. Measurements have been conducted for three different outlet diameters of the nozzles: $D_n = 3.45$ mm; 6.00 mm; 9.50 mm. In all measurements the distance of the nozzle outlet cross-section from the entry to the mixing chamber was the same and was equal to 42 mm. The results of experimental investigations available in the literature precisely indicate [2, 3] that the distance between the nozzle location with respect to the entrance of the mixing chamber has not influence on operation of a two-phase ejector.

Similarly as in [1] experimental investigations conducted have been in a two-fold manner: investigations of performance characteristics of the ejector and investigations of the ejector performance in limiting cases. These cases are defined in the accompanying paper [1].

For a longer mixing chamber ($L_{th} = 840$ mm) performed have also been investigations of performance characteristics obtained by the change of back-pressure. The change of discharge pressure was rendered by flow throttling in a valve located on the outlet pipe (see Fig. 3 in [1]).

## 3. Characteristics of ejector operation

As it has been indicated in the Introduction there exists a widespread opinion [2-4], that in the calculation of the ejector performance characteristics there can be used an equation of momentum conservation in a similar form as in the case of single phase ejectors.

In order to develop of momentum equation for liquid ejectors we assume that the liquid streams at inlet to the mixing chamber have yet to decrease, and hence we are dealing with an axisymmetrical flow of motive liquid stream $\dot{m}_1$ and feeding liquid stream $\dot{m}_2$, which is feeding the mixing chamber in the annular way. It is assumed that pressures of both streams at inlet to the mixing chamber are the same. Additionally, we assume that at the outlet from the mixing chamber a complete mixing of streams occurs. The momentum balance of both motive and feeding media is conducted in the control volume of the mixing chamber. The momentum balance equation has the form:

$$(p_t - p_o)A_n + (p_t - p_o)A_o = (\dot{m}_1 w_{1o} + \dot{m}_2 w_{2o})\varphi_{th} - (\dot{m}_1 + \dot{m}_2)w_t \ , \tag{1}$$

where: $A_o$ is a cross-section area of the feeding liquid stream: $A_o = A_t - A_n$. Having used the relations linking the velocities of the flow and the pressure with the geometrical parameters of the ejector one can obtain from Eq. (1) the following dimensionless formula [2]:

$$\frac{p_d - p_{2i}}{p_{1i} - p_{2i}} = \varphi_n^2 \frac{A_n}{A_{th}} \left[ 2\varphi_{th} + \left( 2\varphi_{th} - \frac{1}{\varphi_i^2} \right) \frac{A_n}{A_{2i}} \gamma_{12} \chi_i^2 - (2 - \varphi_d^2) \frac{1}{\gamma_{12}} \frac{A_n}{A_{th}} (1 + \gamma_{12}\chi_i)^2 \right] . \tag{2}$$

In the above relation $\varphi_n$ is a nozzle coefficient of velocity, and respectively $\varphi_{th}$ – mixing chamber one, $\varphi_d$ – diffuser one, and $\varphi_i$ – suction chamber with the intake nozzle. Under conditions of a hypothetical two-phase ejector with the same densities of media (i.e. for $\gamma_{12} = 1$) [2,12] we obtain a simplified version of Eq.(2):

$$\frac{p_d - p_{2i}}{p_{1i} - p_{2i}} = \varphi_n^2 \frac{A_n}{A_{th}} \left[ 2\varphi_{th} + \left( 2\varphi_{th} - \frac{1}{\varphi_i^2} \right) \frac{A_n}{A_{2i}} \chi_i^2 - (2 - \varphi_d^2) \frac{A_n}{A_{th}} (1 + \chi_i)^2 \right] . \tag{2a}$$

An important fact, from the point of view of conversion of Eq.(2) to the case of liquid-gas ejectors, is that on the flow path of gaseous phase from the suction chamber to the mixing chamber there is no pressure drop. This fact can be interpreted as [2] that $A_{2i} \to \infty$, hence $A_n/A_{2i} \approx 0$ and the term describing the pressure drop at inlet to the mixing chamber can be omitted. In effect we assume the following equation of momentum conservation for the case of liquid-gas ejectors:

$$\Pi = 2\varphi_n^2 b \left[ \varphi_{th} - \left( 1 - \frac{\varphi_d^2}{2} \right) \frac{1}{\gamma_{aw}} b (1 + \gamma_{aw}\chi_i)^2 \right] . \tag{3}$$

It should be noted that Zinger and Sokolov [2, 12] introduced the momentum balance for the case of the two-phase ejector based on relation (2a) for a hypothetical ejector, and hence it does not include the difference of densities between liquid stream and feeding gas. Then:

$$\Pi = 2\varphi_n^2 b \left[ \varphi_{th} - \left( 1 - \frac{\varphi_d^2}{2} \right) b (1 + \chi_i)^2 \right] . \tag{3a}$$

Equation (3a), suggested without any comment by Zinger and Sokolov [2], can be explained by a suspicion, that the momentum transfer most probably takes place in the mixing chamber between water as a working medium and foam surrounding that stream coming from reverse flows in the mixing chamber. Velocity of foam is small, however its mass flow rate is comparable with the stream of a working medium. This can eventually form justification for omission of a ratio of densities of feeding and working streams.

An issue of correctness of equations of the type (2a) and (3a) is, in our opinion, doubtful and the only justification could be the easiness in application of such equations in some practical and very limited range of cases. A similar arbitral approach in construction of the momentum balance equations for a two-phase ejector is presented by Filipczak [4]:

$$\Pi = 2\varphi_n^2 b\left[\varphi_{th} - \left(1 - \frac{\varphi_d^2}{2}\right)\frac{1}{\gamma_{aw}} b(1+\chi_i)^2\right], \tag{3b}$$

hence he arbitrarily assumed a volumetric ejection coefficient instead of a mass one. Berman and Efimotschkin [16] whilst considering the tubular ejectors, i.e. without a diffuser ($\varphi_d = 0$), also omit the ratio of densities of media and suggest a following form of momentum conservation equation for the case of a mixing chamber:

$$\Pi = 2\varphi_n^2 b\left[\varphi_{th} - b\left(1 + \chi_i \frac{p_{ai} - p_s(T_{wi})}{p_t - p_s(T_{wi})}\right)^2\right], \tag{3c}$$

where: $p_s(T_{wi})$ is a saturation pressure wich depends on water temperature $T_{wi}$.

The momentum equation for a two-phase ejector has been most rationally and independently from the above mentioned ones presented by Cunningham [10]. He assumed that up to the mixing zone (shock wave) both phases flow separately and after transition through the mixing wave the two-phase flow is homogeneous and the interface slip is equal zero. Additionally, the pressures at the entrance of the mixing chamber and in the suction chamber are equal and hence we can assume $\chi_i = \chi_o$. Bearing in mind the assumptions of Cunningham [10] it can be shown that the operational characteristic of a turbular ejector takes the following form:

$$\frac{\Pi}{\varphi_n^2} = 2b - (2 + K_{th})b^2\left(1 + \frac{p_{ai}}{p_t}\chi_i\right)(1 + \gamma_{aw}\chi_i) + 2\gamma_{aw}\chi_i^2\frac{b^2}{1-b}\,. \tag{3d}$$

There exists a density ratio $\gamma_{aw}$ of both phases in the ejector.

As can be seen from the presented literature survey the abundant number of available forms of the momentum balance equation for the mixing chamber can be a reason for significant discrepancies in obtained results in the range of the influence of the ejector geometrical parameters and the ratio $b$ in particular, on its performance.

Apart from the influence of the form of equation describing the ejector performance characteristics an important impact have also the velocity coefficients of the nozzle, mixing chamber and diffuser. Zinger and Sokolov [2] suggest to use in calculations following values of the velocity coefficients: $\varphi_n = 0.950$; $\varphi_{th} = 0.975$; $\varphi_d = 0.900$. Available in the literature [16] data on the velocity coefficient for sharp-edged nozzles fall in the limits from $\varphi_n = 0.93 \div 0.98$, and values of the coefficient $\varphi_{th}$ are usually close to 0.90.

As a comment, it should be noted that the relations in the form of Eqs. (3) are practical to implement when there is a need to calculate the pressure after the diffuser at given pressures of both media at inlet, the ejection coefficient and ejector geometry. In the case of designing an ejector destined for gas suction (for example of air from the vacuum system of the condenser) we have to deal with another problem, i.e. we know the air mass flow rate, water and air pressures at inlet and the water-air mixture at outlet, and we are searching for the ejection coefficient, corresponding water flow rate and principal geometrical parameters of the ejector. It turns out then [16], that using relations (3) we obtain sometimes significant discrepancies between calculated results and measured values. It ought to be noticed also, that the accuracy of prediction of the velocity coefficients in the presented case is of great importance. Mentioned discrepancies between theoretical and experimental values can be explained by the fact that in the case of large values of the ejection coefficient $\chi_i$ the corresponding values of compression ratio $\Pi$ change insignificantly and hence small range of compression ratio variation is accompanied by a significant range of variation of the ejection coefficient. These circumstances force some researchers to investigate empirical relations between the compression ratio and the ejection coefficient, which often are not based on the momentum conservation equation.

Taking into account the velocity coefficients assumed by Zinger [12], based on equation (3a) we obtain the following equation of performance characteristic of a two-phase ejector [2, 3, 11]:

$$\Pi = 1.77b - 1.12b^2(1 + \chi_i)^2 \, . \tag{4}$$

Sokolov and Zinger [2] suggest also an empirical relation for volumetric ejection ratio as a function of the compression ratio in the form:

$$\chi_i = K\frac{1}{\sqrt{\Pi}} - 1 \, , \tag{5}$$

where in the case of two-phase ejectors they assumed: $K = 0.85$. Relation (5) is, in some approximation, an envelope of characteristics obtained for particular values of geometrical parameter $b$ of ejectors. Points of tangency of a curve given by (5) with performance characteristics described by (4) correspond roughly to a value of the ratio $b$ of cross-section areas of the nozzle and mixing chamber often regarded in the literature [2,3] as optimal. Its value can be determined from the condition:

$$\frac{\partial \Pi}{\partial b} = 0 \, . \tag{6}$$

From Zinger and Sokolov equation (3a) and using the empirical relation (5) we obtain:

$$b = \frac{\varphi_{th}}{(2-\varphi_d^2)K^2}\Pi\,. \tag{7}$$

Assuming the velocity coefficients suggested by Sokolov and Zinger [2] $\varphi_n = 0.90$ and $\varphi_d = 0.90$, $K = 0.85$, we obtain from (7):

$$b = 1.05\Pi\,. \tag{8}$$

Performance characteristics of the two-phase ejector obtained in own investigations for different values of the module $b$ and two different nozzle profiles are presented in Fig. 4. Detailed information on the parameters of working and feeding media used in investigations are contained in reports [18, 19]. In the graphs presented are also curves determined based on experimental relations due to Zinger and Sokolov (4) and (5). It ought to be stressed that the measurements for a given nozzle in the ejector have been performed at two different volumetric flow rates.

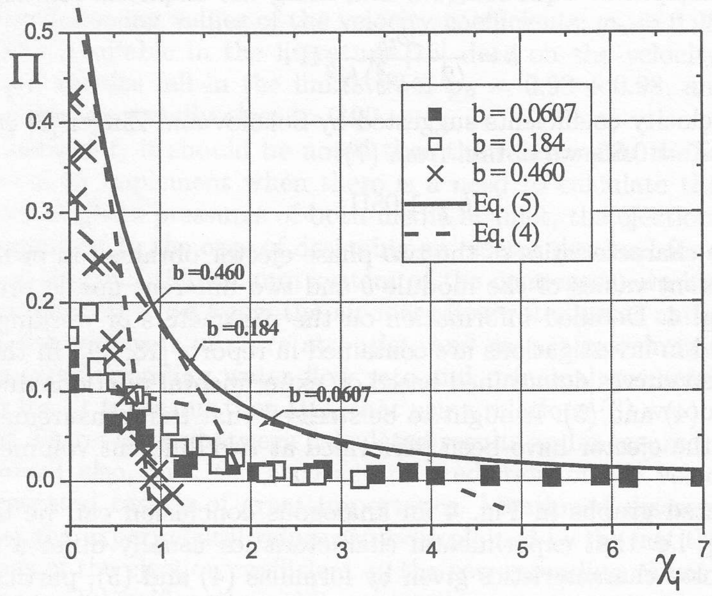
From presented graphs in Fig. 4 an analogous conclusion can be drawn as in the paper [1], i.e. that experimental characteristics usually differ a lot from Zinger and Sokolov characteristics given by formulas (4) and (5), particularly in the case of lower values of the module $b$. This means that the relations (4) and (5) provided by Zinger and Sokolov can only serve for preliminary calculations. Additionally, a characteristic feature of the distributions of characteristics consisting of experimental points is that for a given module $b$ the points are aligning along some curves, which are practically independent from the water flow rate. Due to that it can be concluded that characteristics in the form $\Pi = f(\chi_i)$ are a good description of the performance of the two-phase ejector and the additional necessary parameters for their prescription are: module $b$ and, in line with the results obtained in [1], a non- dimensional length of the ejector mixing chamber $L_{th}/D_n$. Another conclusion based on presented experimental results can be drawn is that for a given value of the ejection coefficient $\chi_i$ in the case of a sharp-edged nozzle obtained have been lower values of the compression ratio $\Pi$, compared with a Bendemann profile nozzle.

## 4. Influence of nozzle geometry on ejector efficiency

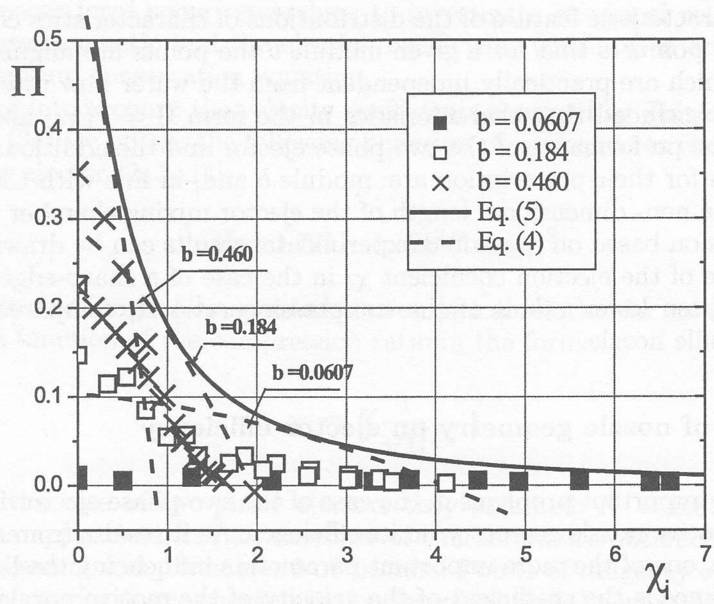
One of more important problems in the case of the two-phase ejector is the influence of the motive nozzle geometry on its efficiency. As it results from relations presented above, one of the more important parameters influencing the liquid-gas ejector performance is the coefficient of the velocity of the motive nozzle.

In Table 1 presented are values of the velocity coefficients for investigated nozzles. These coefficients have been calculated from the following relation:

$$\varphi_n = \frac{4}{\pi}\frac{\dot V_w}{D_n^2}\sqrt{2\frac{p_{wi}-p_{ai}}{\rho_w}}\,. \tag{9}$$




a)




b)

Fig. 3. Ejector performance curves $\Pi = f(\chi_i)$ for length of the mixing chamber $L_{th} = 530$ mm, the nozzle diameter is a parameter: a) the Bendemann nozzle, b) sharp edged nozzle.

Table 1. Velocity coefficient $\varphi_n$ of investigated nozzles

| Nozzle profile | $D_n = 345$ mm | $D_n = 6.00$ m | $D_n = 9.50$ mm |
|---|---|---|---|
| Bendemann | 0.97 | 0.95 | 0.94 |
| Sharp-edged | 0.62 | 0.70 | 0.64 |

At this point we face a problem of prediction of the optimal value of the module $b$. We may bear in mind for ths purpose a value of maximum compression ratio or the ejector efficiency. In Section 3 presented has been a relation (5) describing the optimal value of the module $b$ with respect to the compression ratio proposed by Zinger and Sokolov [2, 3]. In the opinion of present authors a more rational definition of the optimal value of the module $b$ is such a value at which a maximum efficiency is obtained in the ejector, i.e.

$$\left(\frac{\partial \eta_T}{\partial b}\right)_{\Pi=idem} = 0\,. \tag{10}$$

The most rational efficiency of the ejector is defined by the following equation:

$$\eta_T = \chi_i \frac{p_{ai} \ln \frac{p_d}{p_{ai}}}{p_{wi} - p_d}\,. \tag{11}$$

Bearing in mind, that $\gamma_{aw}$ is at most of the order of $10^{-3}$, which enables to make an approximation: $1 + \gamma\chi_i \approx 1$, from (3d) we obtain:

$$\chi_i = \frac{p_t}{p_{ai}} \left[\frac{2b - \Pi/\varphi_n^2}{b^2(2 + K_{th})} - 1\right]\,. \tag{12}$$

Then, from equations (10), (11) and (12) we get a following value of the optimal ratio $b$:

$$b_{opt} = \Pi/\varphi_n^2\,. \tag{13}$$

This result is comparable to the one obtained by Zinger and Sokolov [2, 3] – see Eq. (8). Obtained result means that in order to achieve high value of the compression ratio ($\Pi \to 1$) we must use nozzle relatively higher diameter $D_n$, and in the case of searching for high value of the ejection coefficient $\chi_i$ (i.e. $\Pi \to 0$) it is desired to use nozzle of lower diameter with respect to the diameter of the mixing chamber.

Results of own experimental investigations of the influence of the motive nozzle diameter on available ejector efficiency have been presented in Fig. 4 for the ejector equipped with a mixing chamber of the length $L_{th} = 530$ mm, whereas in Fig. 5 for a mixing chamber of the length $L_{th} = 840$ mm. Curves presented in these figures are an approximation of the points obtained in experiments. From presented graphs one can conclude that in principal the highest efficiencies have been obtained for the nozzle of the largest diameter, i.e. for the highest value of the module $b$, independently of the nozzle profile. Only in the case of a longer investigated mixing chamber, in the case of a Bendemann nozzle, obtained have been the ejector efficiencies close to the maximum value for nozzles of the diameters $D_n = 6.00$ and $D_n = 9.50$ mm.

It is an important issue, that despite significantly lower values of the coefficient of velocity in the case of a sharp-edged nozzle, higher values of the efficiency have been obtained. The similar conclusion has also been reached by other researchers, for example [11]. Another important conclusion resulting from Fig. 4 and 5 is that there is a value of the ejection coefficient, where a maximum value of the efficiency is reached, which when exceeded, in the case of nozzles characterised by a higher module $b$, an abrupt decrease is observed. This is a very important conclusion from the practical point of view.

In Fig. 6 presented are values of the ejection coefficient, which have been obtained during investigations, as a function of the water flow rate for different nozzle geometries and two different lengths of the mixing chamber. As results from enclosed graphs, at identical water flow rate obtained have been, in the case of a sharp-edged nozzle, significantly higher values of ejection coefficient $\chi_i$ independently from the length of the mixing chamber. This effect is responsible for reaching higher efficiencies in the case of sharp-edged nozzles, as values of the compression ratio differ only a little between both investigated profiles, because presented results pertain to the first limiting case of the ejector performance.

It ought to be stressed that a primary objective of present investigations was an empirical determination of the influence of the nozzle profile and value of the module $b$ on the efficiency and not reaching of a maximum value of the ejector efficiency. Presented above results of investigations obtained have been under pressure conditions at outlet from the ejector close to atmospheric pressure and the volumetric stream of intaken air and pressure was controlled by a throttling valve. In this way obtained have been the conditions of operation close to those, which exist under operating conditions during suction of air from vacuum systems of power plant condensers.

Based on own and available from literature experimental investigations [11, 13] we may suspect that the availability of high values of the efficiency is most often a pre-requisite of obtaining an increase of back-pressure after the ejector. In Fig. 7 presented is a comparison of the ejector efficiency obtained during investigations under conditions of control of the amount of intaken air by means of throttling at the intake (continuous curves are an approximation of experimental points) and by regulation of back-pressure by throttling at outlet (a set of filled in points). It can be concluded from presented graphs that the higher values of the efficiency have been obtained under conditions of the back-pressure increase, where a different character of the relation $\eta_T = f(\chi_i)$ can be observed in comparison against data obtained by throttling of intaken air. This is a very important conclusion, which casts a new light on the issue of two-phase ejectors operating under different conditions. This issue has yet to be considered from that point of view in the literature of the subject.

In Fig. 8 presented are very interesting photographs of the structures of

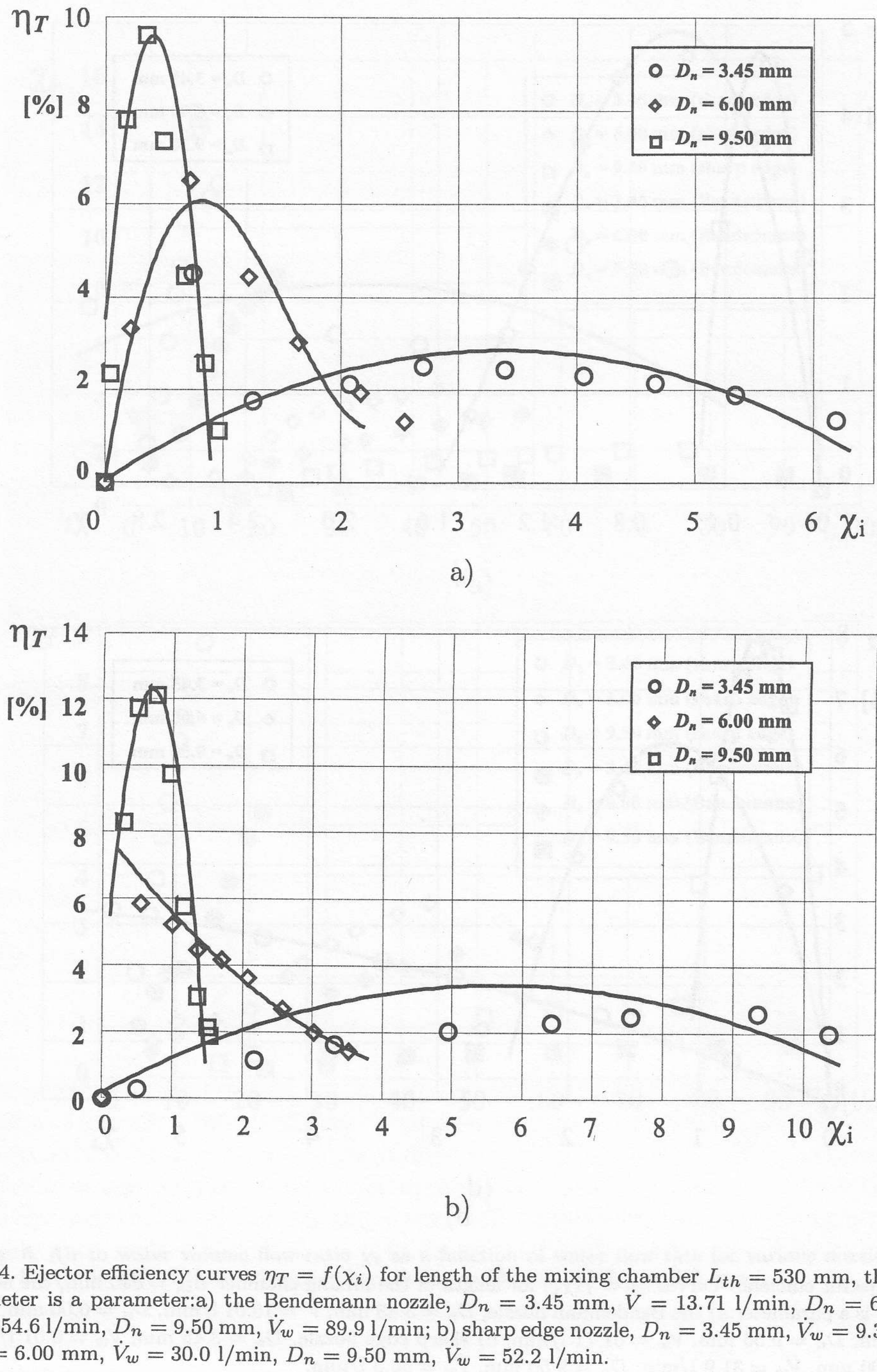



Fig. 4. Ejector efficiency curves $\eta_T = f(\chi_i)$ for length of the mixing chamber $L_{th} = 530$ mm, the nozzle diameter is a parameter:a) the Bendemann nozzle, $D_n = 3.45$ mm, $\dot{V} = 13.71$ l/min, $D_n = 6.00$ mm, $\dot{V} = 54.6$ l/min, $D_n = 9.50$ mm, $\dot{V}_w = 89.9$ l/min; b) sharp edge nozzle, $D_n = 3.45$ mm, $\dot{V}_w = 9.31$ l/min, $D_n = 6.00$ mm, $\dot{V}_w = 30.0$ l/min, $D_n = 9.50$ mm, $\dot{V}_w = 52.2$ l/min.

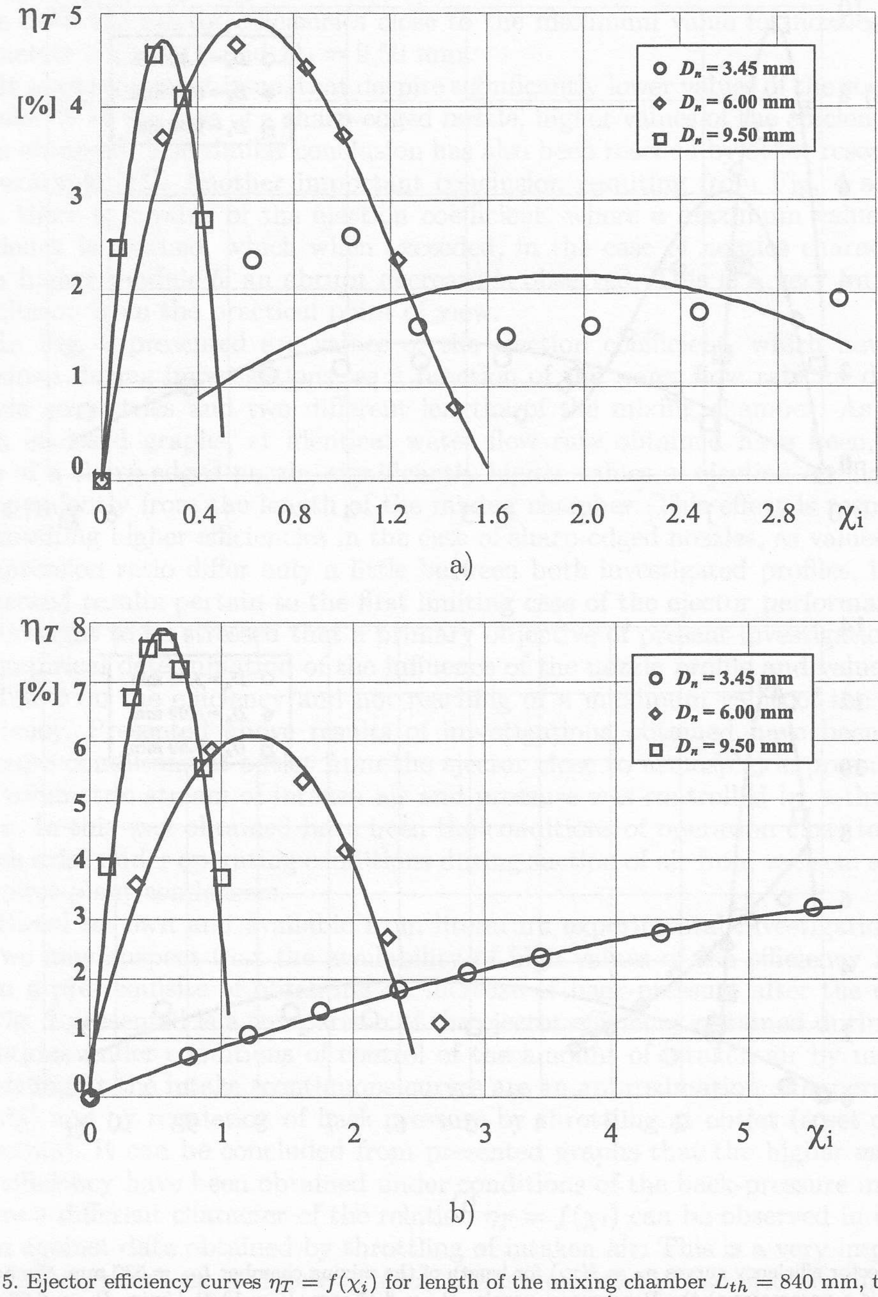
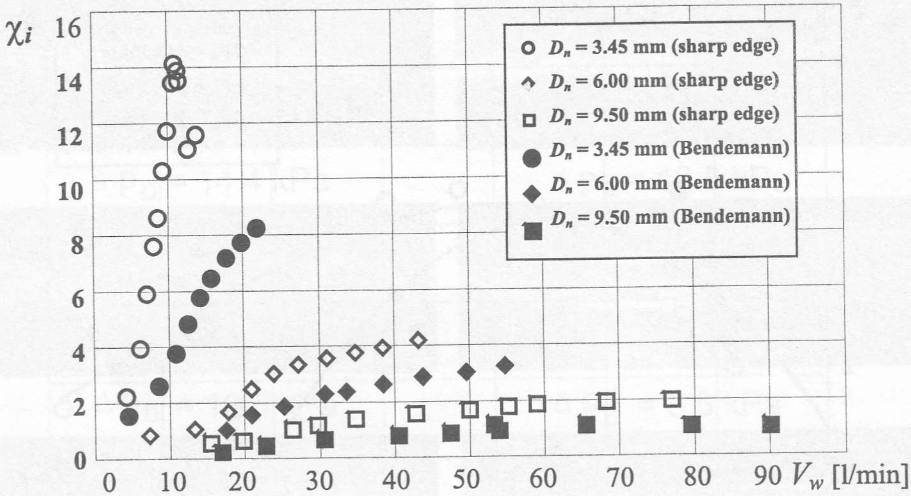
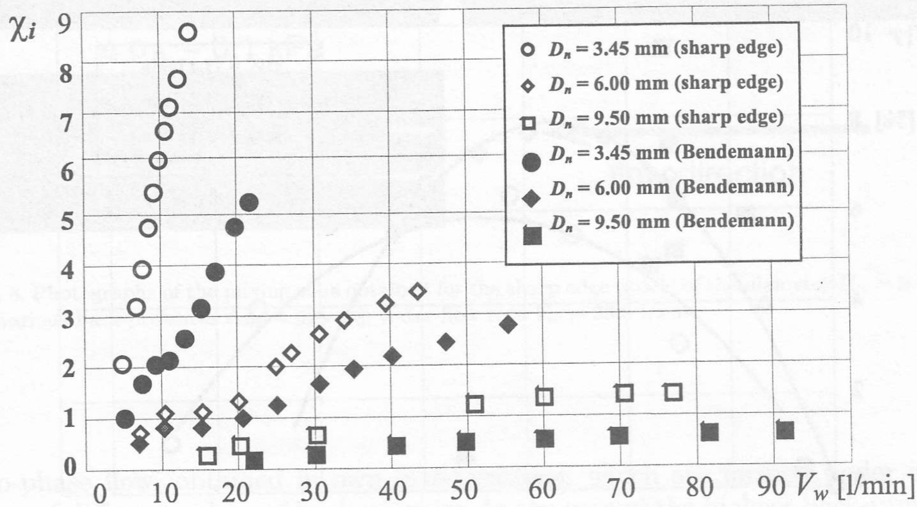



Fig. 5. Ejector efficiency curves $\eta_T = f(\chi_i)$ for length of the mixing chamber $L_{th} = 840$ mm, the nozzle diameter is a parameter:a) the Bendemann nozzle, $D_n = 3.45$ mm, $\dot{V} = 15.01$ l/min, $D_n = 6.00$ mm, $\dot{V} = 33.9$ l/min, $D_n = 9.50$ mm, $\dot{V}_w = 61.71$ l/min; b) sharp edge nozzle, $D_n = 3.45$ mm, $\dot{V}_w = 9.01$ l/min, $D_n = 6.00$ mm, $\dot{V}_w = 31.9$ l/min, $D_n = 9.50$ mm, $\dot{V}_w = 50.5$ l/min.




a)




b)

Fig. 6. Air to water volume flow ratio $\chi_i$ as a function of water flow rate for various nozzle gemetries and for the length of the mixing chamber: a) $L_{th} = 530$ mm, b) $L_{th} = 840$ mm.

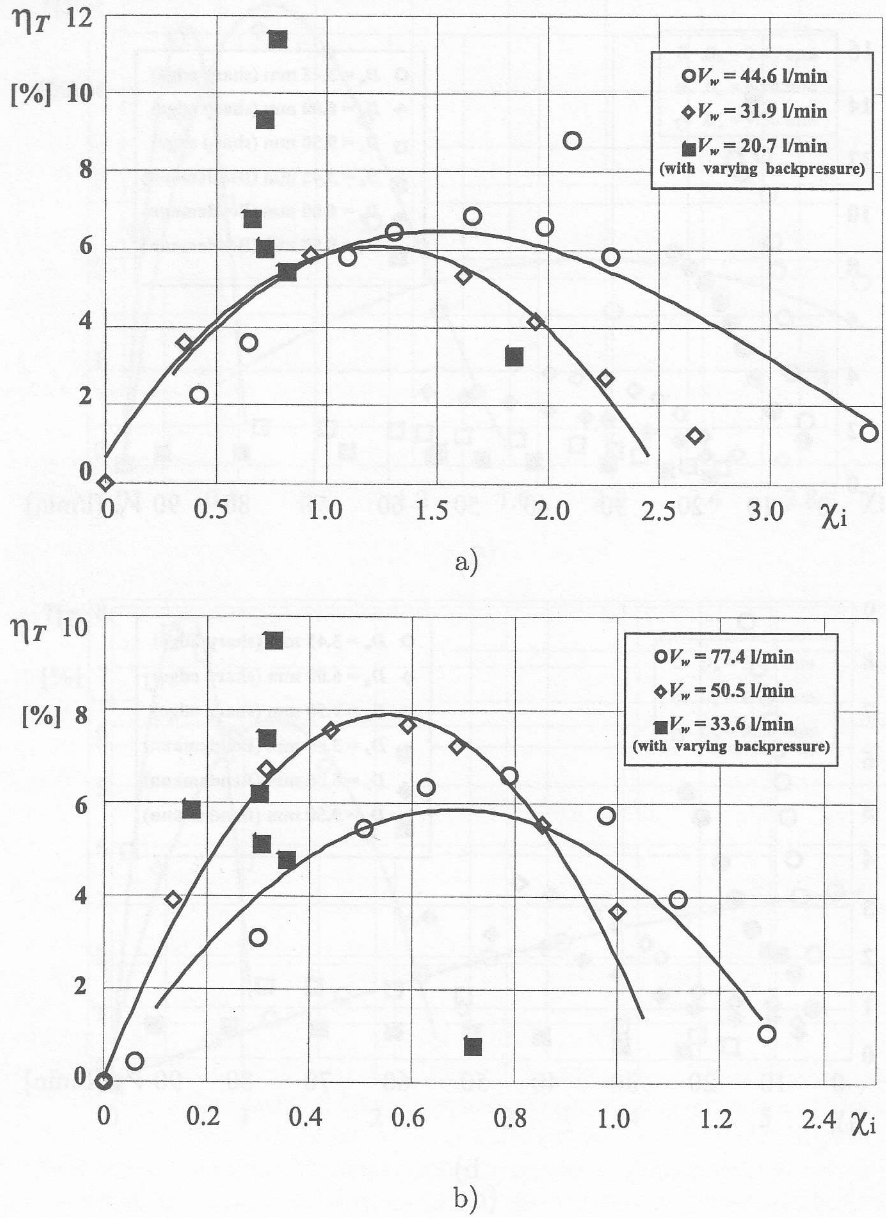



Fig. 7. Ejector performance curves $\eta_T = f(\chi_i)$ for length of the mixing chamber $L_{th}$ = 840 mm, the nozzle diameter is a parameter: a) the Bendemann nozzle of the diameter $D_n$ = 6.00 mm, b) the sharp edged nozzle of the diameter $D_n$ = 9.50 mm.

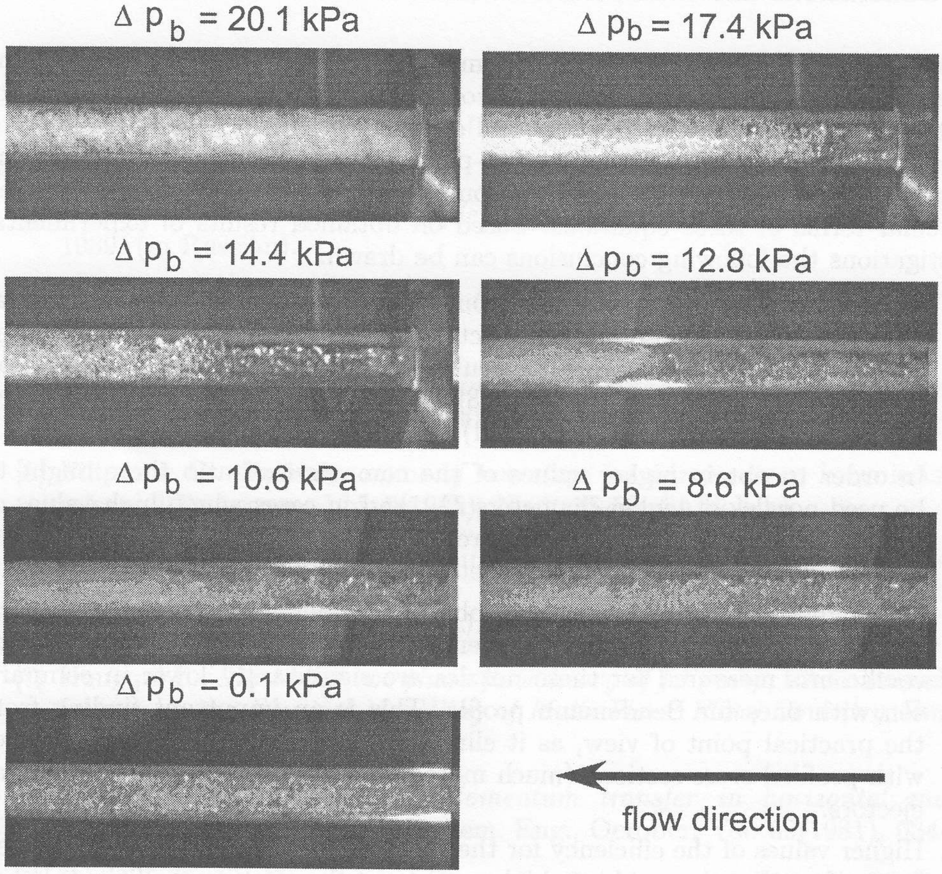



Fig. 8. Photographs of the mixing zone obtained for the sharp edge nozzle of the diameter $D_n = 9.50$ mm for various back-pressures $\Delta p_b = p_d - p_b$; water flow rate $\dot{V}_w = 33.6$ l/min.

two-phase flow, obtained in own investigations, which are formed under conditions of different values of back-pressure. In the case of the highest back-pressures there can be observed a a beginning of a mixing wave, which is located at the entrance to the mixing chamber. Reduction of the back-pressure renders movement of the beginning of the mixing wave in the direction of the diffuser and at the same time increase in the length of the mixing wave is observed. Under conditions of small back-pressures or its complete lack in the mixing chamber, there cannot be discerned an unanimous location of the mixing wave and also observed is a formation of a homogeneous flow with bubbly structure with visible reverse flows taking place in the vicinity of the chamber wall.

## 5. Conclusions and final remarks

In the work presented are results of own extensive experimental investigations of the influence of the two-phase flow ejector geometry on its performance characteristics and the efficiency. Presented and critically analysed have been different forms of performance equations of the two-phase ejector which can be found in the literature. Presented have been assumptions, based on which obtained have been particular forms of these equations. Based on obtained results of experimental investigations the following conclusions can be drawn:

1. There is an optimal value of the geometrical module $b$ with respect to available compression ratio and the ejector efficiency. It has been showed, that values of the optimal module $b$ calculated from the characteristic equation proposed by Zinger and Sokolov [2, 3] are similar to values calculated based on Cunningham model [10], Eq. (3d).
2. In order to obtain higher values of the compression ratio there ought to be used nozzles of higher diameters $D_n$, and in cases where high values of the ejection coefficient $\chi_i$ are required it is desired to use nozzles of lower diameters with respect to the diameter of the mixing chamber.
3. The highest ejector efficiencies are obtained under conditions of using nozzles with sharp-edged profile, independently from the fact that the velocity coefficients measured for these nozzles are significantly lower in comparison with ones for Bendemann profile. This is an important finding from the practical point of view, as it eliminates the necessity of using nozzles with profiled cross-sections (much more expensive) in two-phase liquid-gas ejectors.
4. Higher values of the efficiency for the sharp-edged nozzles can be explained by the fact that these achieve higher values of the ejection coefficient. It has been confirmed by the results of investigations obtained under conditions of the first limiting case of the ejector operation.
5. Increase of the back-pressure at the outlet of the ejector diffuser renders major changes in the picture of two-phase flow structures in the mixing chamber (location of the mixing wave), where at the same time there is a general increase of the efficiency of ejector operation.
6. The mixing wave can be influenced by different two-phase flow water-air structures. This is not only a homogeneous flow as it is usually assumed in modelling. Sometimes there appeared a separated flow. It can be expected that such aspect should also be considered in future investigations, as it influences the ejector operation.

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