INSTITUTE OF FLUID-FLOW MACHINERY
POLISH ACADEMY OF SCIENCES

TRANSACTIONS OF THE INSTITUTE OF FLUID-FLOW MACHINERY

108

GDAŃSK 2001

TRANSACTIONS OF THE INSTITUTE OF FLUID-FLOW MACHINERY Appears since 1960

Aims and Scope

Transactions of the Institute of Fluid-Flow Machinery have primarily been established to publish papers from four disciplines represented at the Institute of Fluid-Flow Machinery of Polish Academy of Sciences, such as:

- Liquid flows in hydraulic machinery including exploitation problems,
- Gas and liquid flows with heat transport, particularly two-phase flows,
- Various aspects of development of plasma and laser engineering,
- Solid mechanics, machine mechanics including exploitation problems.

The periodical, where originally were published papers describing the research conducted at the Institute, has now appeared to be the place for publication of works by authors both from Poland and abroad. A traditional scope of topics has been preserved.

Only original and written in English works are published, which represent both theoretical and applied sciences. All papers are reviewed by two independent referees.

EDITORIAL AND PUBLISHING OFFICE

IFFM Publishers (Wydawnictwo IMP), Institute of Fluid Flow Machinery, Fiszera 14, 80-952 Gdańsk, Poland, Tel.: +48(58)3411271 ext. 141, Fax: +48(58)3416144, E-mail: esli@imp.gda.pl

Financial support of publication of this journal is provided by the State Committee for Scientific Research, Warsaw, Poland

Terms of subscription

Subscription order and payment should be directly sent to the Publishing Office

Warunki prenumeraty w Polsce

Wydawnictwo ukazuje się przeciętnie dwa lub trzy razy w roku. Cena numeru wynosi 20,- zł + 5,- zł koszty wysyłki. Zamówienia z określeniem okresu prenumeraty, nazwiskiem i adresem odbiorcy należy kierować bezpośrednio do Wydawcy (Wydawnictwo IMP, Instytut Maszyn Przepływowych PAN, ul. Gen. Fiszera 14, 80-952 Gdańsk). Osiągalne są również wydania poprzednie. Prenumerata jest również realizowana przez jednostki kolportażowe RUCH S.A. właściwe dla miejsca zamieszkania lub siedziby prenumeratora. W takim przypadku dostawa następuje w uzgodniony sposób.

ISSN 0079-3205

TRANSACTIONS OF THE INSTITUTE OF FLUID-FLOW MACHINERY

No. 108, 2001, 113-136

D. BUTRYMOWICZ, M. TRELA, E. IHNATOWICZ, J. KARWACKI[1]

Experimental investigations of the influence of mixing chamber geometry on performance of water-air ejector

The paper deals with the experimental investigations of a two-phase water-air ejector. The experimental apparatus and procedure are described. The aim of the research was to investigate the influence of the ejector geometry on its performance. A special attention was paid to the influence of the mixing chamber length on the gas suction rate, ejector efficiency as well as on the friction loss coefficient of the mixing chamber.

Nomenclature

A – cross sectional area, m^2

b – nozzle/mixing chamber area ratio, $b = A_n/A_{th}$,

c – gas annulus/nozzle area ratio, $c = (1-b)/b$,

D – diameter, m

k – coefficient, see Eq. (2), Pa·s/kg

K_{th} – friction loss coefficient for mixing section dimensionless,

L – length, m

χ – volumetric flow ratio, $\chi = \cdot V_a/\cdot V_w$

η_T – isothermal efficiency, see Eq. (14),

τ – friction shear stress, Pa

L_m – distance between the origin of the mixing zone (or condensation zone) from the origin of the mixing chamber, cm

$\dot{m}$ – mass flow rate, kg/s

p – pressure, Pa

R_a – individual gas constant, J/(kg·K)

T – temperature K,

$\dot{V}$ – volumetric flow, m^3/s

w – velocity, m/s

γ – density ratio, $\gamma = \rho_{ai}/\rho_w$

ρ – density, kg/m^3

Π – dimensionless compression coefficient, $\Pi = (p_d - p_{ai})/(p_{wi} - p_d)$

Subscripts

a – air,

ai – air at suction chamber,

b – barometric,

d – diffuser,

n – nozzle,

o – mixing section entrance,

s – saturation,

t – mixing chamber exit,

th – mixing chamber,

w – water; wall of the mixing chamber,

wi – water at the nozzle entrance,

wo – water at the mixing chamber

[1]Institute of Fluid-Flow Machinery, Fiszera 14, 80-952 Gdańsk, Poland

1. Introduction

Recently one can observe the growing interest in using ejectors in many technical cases such as: conventional and nuclear power, refrigeration, chemical engineering, sanitary technique, food processing industry, vacuum technique and others. Practically in all cases a decisive factor about implementation of the ejector is not its efficiency, but technical and operational features, particularly the operational reliability. Such situation determined in the past a little interest in research directed towards to the increase of the efficiency of ejectors [1-3]. This situation has been changed during last years with respect to two-phase ejectors with regard to their potential application in the cooling systems of nuclear reactors, particularly of BWR type [26].

In the present paper a two-phase water-air ejector is considered (Fig. 1). The

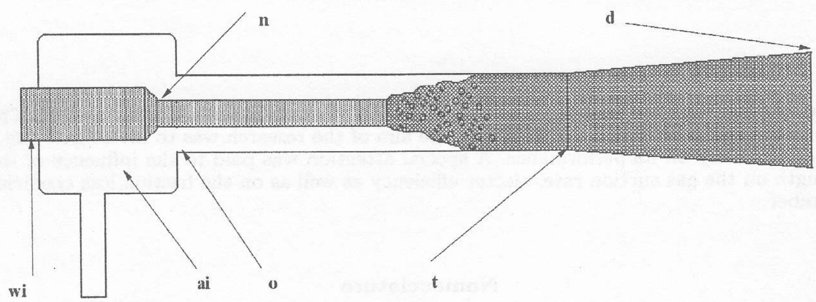


Fig. 1. Pictorial diagram of a liquid-gas ejector.

ejector is fed by a stream of the working liquid. The liquid inlet cross-section to the ejector is denoted by the symbol wi. The working liquid expands in the nozzle. The outlet cross-section of the nozzle is denoted by n. The stream of gas is supplied to the suction chamber of the ejector. The inlet cross-section of this phase is denoted by the symbol ai. The entrance of the mixing chamber is denoted by the symbol o. At the entrance of the mixing chamber we are mostly dealing with annular flow of liquid and gas, 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 chamber is filled out with white water-air froth. This is the area 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 flow. It ought to be stressed that in the entry part of the mixing chamber there are observed reversing currents of the froth, which reverse against the main stream and then are captured by the main stream and conveyed to the final part of the mixing chamber and further to the diffuser. The end of the mixing chamber 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 flow. The outlet from the diffuser to the outlet pipeline is denoted by the symbol d.

At present there is relatively a small number of papers referring to modelling of the two- phase water-air ejector [4-15]. In principal, the models are based on the lumped parameters with many coefficients describing processes taking place in the nozzle, the mixing chamber and the diffuser. Values of these coefficients are adjusted to experimental data. In the majority of cited above papers description of operation of the two-phase ejector is based on equations derived for the case of liquid ejectors, which lead to not very good results. Regretfully, the number of experimental works regarding the two-phase water-air ejectors is relatively small, which results in the inability to produce more accurate performance characteristics of the ejector.

From the available experimental data it may be concluded that the principal ejector geometry parameters influencing the operation are: the length of the mixing chamber and the shape and dimensions of the nozzle and diffuser. Other geometrical parameters such as the distance between the nozzle and the entry to the mixng chamber do not play significant role [1, 17] if only the ratio L_n/D_n falls in the range $0.0 \div 7.3$.

It ought to be stressed that relatively most of attention has been paid to the problem of the influence of the ratio of the feeding nozzle to the mixing chamber diameter on the performance of the two-phase ejector [22], neglecting the effect of the length of the mixing chamber. Based on the available reports we can probably expect, that the influence of the latter parameter can prove to be very significant [9, 16, 21].

Scarce reports on that topic enable to conclude that the most optimal is such a length of the mixing chamber, where the shock wave is appearing in the final part of the mixing chamber just prior to the entrance to the diffuser (Fig. 2a) [9, 16]. If the length of the mixing chamber will be too short, then, as shown in Fig. 2b, the shock wave will be formed inside the diffuser and even partially in the outlet pipe beyond the diffuser. We must expect that the available ejector performance will be lower in this case. According to [21], obtaining of higher efficiencies for longer mixing chambers must be explained by the fact that the increase of the pressure in the mixing chamber takes place over a significantly longer distance and all losses concerned with the presence of the shock wave will then be relatively lower.

In the literature there is a lack of relations enabling theoretical determination of the optimal length of the mixing chamber which guarantees a maximum performance of the ejector under given conditions. There has also to be noted that experimental data for the discussed case are scarce and incomplete. Additionally, in the reports devoted to the problems of calculation of two-phase ejectors, the issue of the length of the mixing chamber is omitted [6], [10-13], [15], or is modelled by generalisation of experimental data [14], or is accounted for by introduction of a relevant coefficient obtained during experiments [5, 7, 9].

Unsatisfactory state of knowledge on the influence of geometrical parameters of two- phase liquid-gas ejector and the length of the mixing chamber in particular,

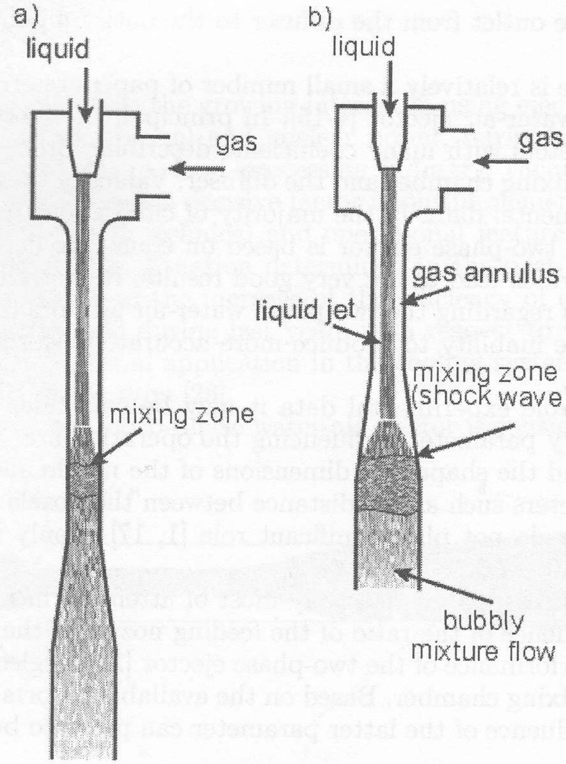


Fig. 2. Schematic diagram of liquid-gas ejector: a) with elongated mixing chamber, b) with short mixing chamber.

on the ejector efficiency served as a basis for carrying out systematic experimental investigations in this area. The results of these investigations, comparisons with other authors data and conclusions are presented in this work.

2. Research stand and measurement procedure

Research stand, where investigations have been performed, has schematically been presented in Fig. 3. The rig conforms to ISO 9001 quality standards implemented at the Institute of Fluid-Flow Machinery PASci. The main components of the rig are: a horizontal liquid-gas ejector, water tank and pumps. Investigated ejector consists of the suction chamber (1) with attached feeding nozzle (9), mixing chamber (2) and a diffuser (3). A design of the suction chamber enables change of the feeding nozzle location with respect to the entry to the mixing chamber.

Water is pumped from the main tank (5) to the nozzle feeding the ejector by means of, coupled in series, two rotating pumps. At the same time the ejector sucks the atmospheric air into the suction chamber (1). There is a possibility of throttling of sucked air using the control valve. A gas-liquid mixture flows from the ejector to the outlet pipe (4) and further to the main tank (5). Beyond the outlet pipe there is also a control valve mounted to vary the back-pressure at the outlet from the ejector.

The design of the ejector enables the change of the feeding nozzle, change of

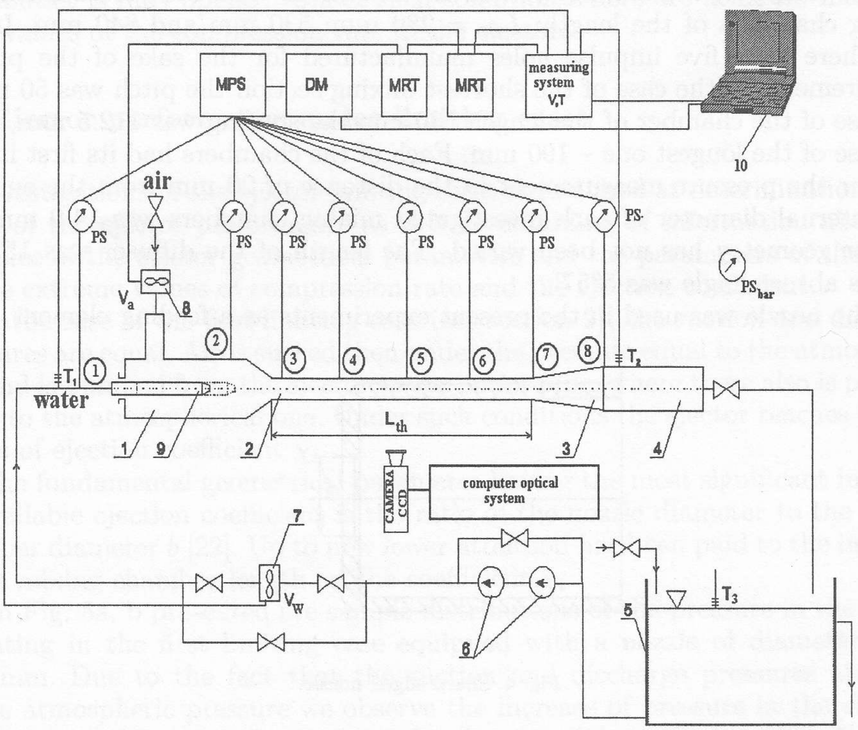


Fig. 3. Schematic diagram of the testing rig: 1 – suction chamber, 2 – mixing chamber, 3 – diffuser, 4 – outlet pipe, 5 – water vessel, 6 – water pump, 7 – water turbine flow meter with sensor, 8 – air turbine flow meter with sensor, 9 – nozzle, 10 – computer, PS – pressure transducers, MRT – programmable indicator/counter of water and air flow rates, DM – multi-manometer, MPS – multiplier, T – thermocouples (the numbers of pressure measuring points are circled).

its location with respect to the mixing chamber and the exchange of the mixing chamber. Investigated mixing chambers have been made of perspex, which enabled visualisation of two-phase flow during investigations.

The following quantities have been measured during measurements:

- absolute pressure of fluids in the ejectors (Fig. 3) and the atmospheric pressure by means of electronic pressure sensors PS-2 with the measuring accuracy class of the entire measurement line of 0.1,
- water temperature in points denoted in Fig. 3 by the thermocouples with the measuring error of the entire measurement line of 0.2 K,

- volumetric water flow rate by means of the turbine flow-meter with the magnetic sensor of accuracy class of 0.1,
- volumetric air flow rate by means of turbine flow-meter with a modulation sensor of accuracy class of 0.1.

A fundamental geometrical parameter, which was varied during measurements was the length of the mixing section. Investigations were conducted using three mixing chambers of the length: L_{th} = 280 mm; 530 mm and 840 mm. In each case there were five impulse holes manufactured for the sake of the pressure measurement. In the case of the shortest mixing section the pitch was 50 mm, in the case of the chamber of the length 530 mm the spacing was 112.5 mm, and in the case of the longest one – 190 mm. Each of the chambers had its first impulse hole for the pressure measurement in the distance of 20 mm from the entrance. The internal diameter of each investigated mixing chambers was 14.0 mm. The diffuser geometry has not been varied. The length of the diffuser was 152 mm, and its abtuse angle was $5^o57'$.

The nozzle was used in the present experiments as a feeding element, which

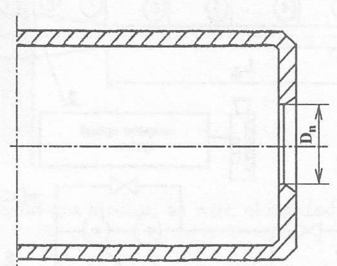


Fig. 4. Sharp edged nozzle.

shape is shown in Fig. 4. Measurements have been conducted for three different outlet diameters: D_n = 3.45 mm; 6.00 mm; 9.50 mm. In all measurements the distance of the nozzle outlet cross-section from the inlet mixing chamber was the same and was equal to 42 mm.

Conducted have been two kinds of experimental investigations: investigations of performance characteristics of the ejector and investigations of the ejector performance in limiting cases.

Investigations of performance of the ejector equipped with the mixing chamber and a nozzle with adjusted geometrical features have been conducted in such a way that in the case of a steady water flow rate the compression rate was controlled by throttling of intaken air by means of the control valve located prior to the suction chamber (see Fig. 3). Further investigations have been repeated for the second value of the water flow rate.

The second part of investigations encompassed investigations in two limiting cases of the ejector operation:

- The first limiting case occurs, when the suction pressure and the outlet pressure are equal to the atmospheric pressure. In such case the control valves on the suction side of air and beyond the balancing pipe are fully opened. The ejector sucks under such conditions the maximum amount of air.
- The second limiting case is when the ejector does not intakes air. Then the suction control valve is fully closed and the control valve at the ejector outlet is fully opened. Under such conditions there are achieved maximum values of the compression rate in the ejector.

3. Ejector performance under limiting cases

Investigations of the ejector limiting cases were aimed at determination of the range of the ejector performance as well as obtaining of information about the influence of the ejector geometrical parameters and the parameters of the media on the extreme values of compression rate and the ejection coefficient.

In the case of the first limiting case (see section 2), the suction and discharge pressures are equal. Air is sucked then under the pressure equal to the atmospheric one and is pumped from the ejector to the outlet pipe, where there also is pressure equal to the atmospherical one. Under such conditions the ejector reaches highest values of ejection coefficient χ_i.

The fundamental geometrical parameter having the most significant influence on available ejection coefficient is the ratio of the nozzle diameter to the mixing chamber diameter b [22]. Up to now lower attention has been paid to the influence of the mixing chamber length on the coefficient χ_i.

In Fig. 5a, b presented are sample distributions of the pressure in the ejector operating in the first limiting case equipped with a nozzle of diameter $D_n =$ 9.50 mm. Due to the fact that the suction and discharge pressures are equal to the atmospheric pressure we observe the increase of pressure in the chamber followed by its decrease. In the case of a shorter mixing chamber (Fig. 5b) there has been observed a higher pressure increase in its middle part. In the case of the longer mixing chamber (Fig. 5b) there has been achieved a significantly higher pressure increase, and its maximum falls in its final part.

The relations of the maximum ejection coefficient as a function of water flow rate are presented in Fig. 6. As indicated in [22], the ejection coefficient significantly depends on the ratio of the feeding nozzle diameter and a diameter of the mixing chamber, and the highest values can be obtained for the nozzles with the lowest diameters. Based on obtained results we can conclude that the influence of the mixing chamber length L_{th} on values of the ejection coefficient χ_i can be regarded as significantly lower, particularly for lower values of the diameter ratios b. It can, however, be concluded from presented diagrams that highest values of ejection coefficient have been obtained for the mixing chamber of the length 530 mm. It is highly probable that there exists an optimal length of the mixing chamber, where the obtained value of χ_i will be highest. This issue,

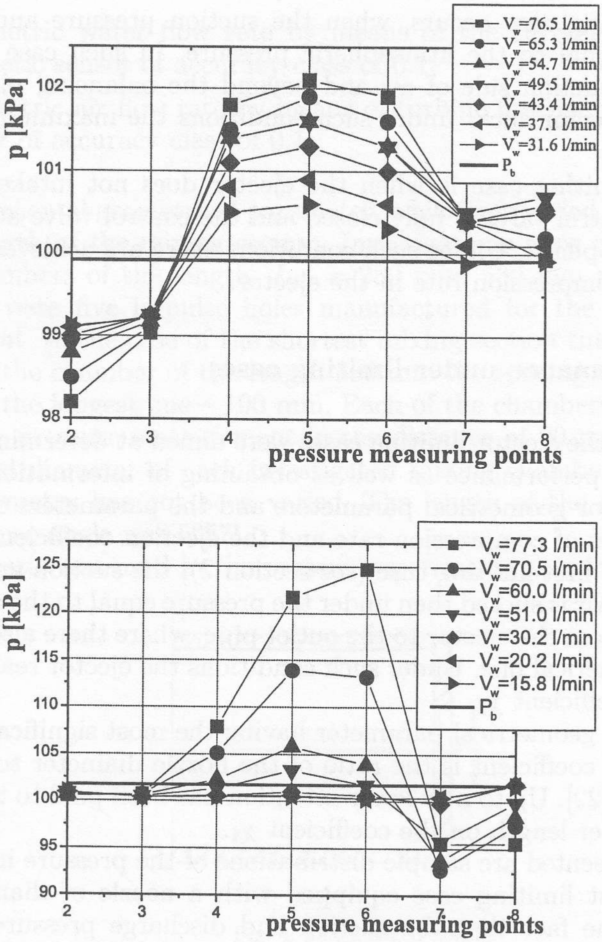


Fig. 5. Pressure distribution within the ejector for the case of the maximum air flow rate (the first limiting case) for various water flow rates: a) $D_n = 9.50$ mm, $L_{th} = 280$ mm; b) $D_n = 9.50$ mm, $L_{th} = 840$ mm.

however, requires further more systematic investigations.

The second limiting case is the operation of the ejector under conditions of the ejection coefficient χ_i equal zero, i.e. without suction of air. In this case, the highest compression ratio is obtained, as in the mixing chamber there exists lowest possible pressure. In the case of investigated liquid-gas ejector the lowest pressure, which can arise on the suction side of the ejector, is equal to the saturation pressure of water at given temperature. In Fig. 7a, b presented are representative data of the pressure distributions in the ejector equipped with a nozzle of diameter $D_n = 9.50$ mm. For the shortest mixing chamber in the case of majority of measurements we can observe the beginning of the pressure increase zone not earlier than in the diffuser – see Fig. 7a. In the case of the longest mixing section, the beginning of a sudden increase lied in the middle or final part and was a

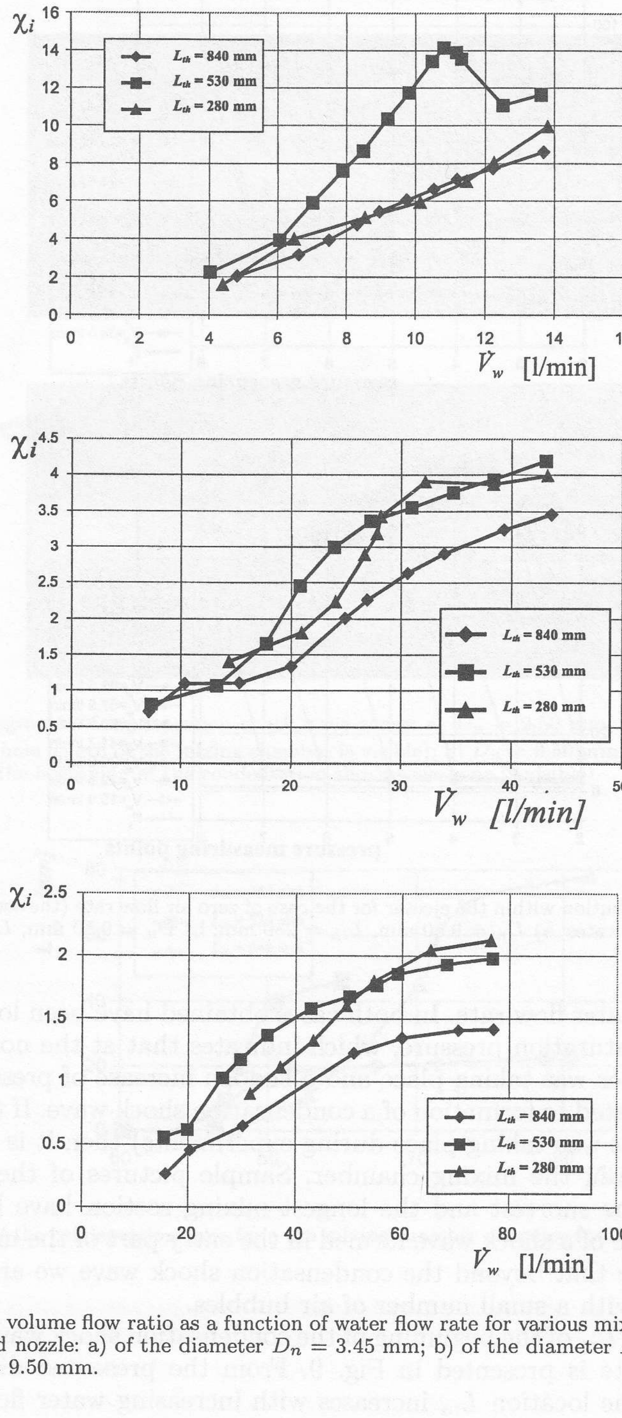


Fig. 6. Air to water volume flow ratio as a function of water flow rate for various mixing chamber lengths and for sharp edged nozzle: a) of the diameter $D_n = 3.45$ mm; b) of the diameter $D_n = 6.00$ mm; c) of the diameter $D_n = 9.50$ mm.

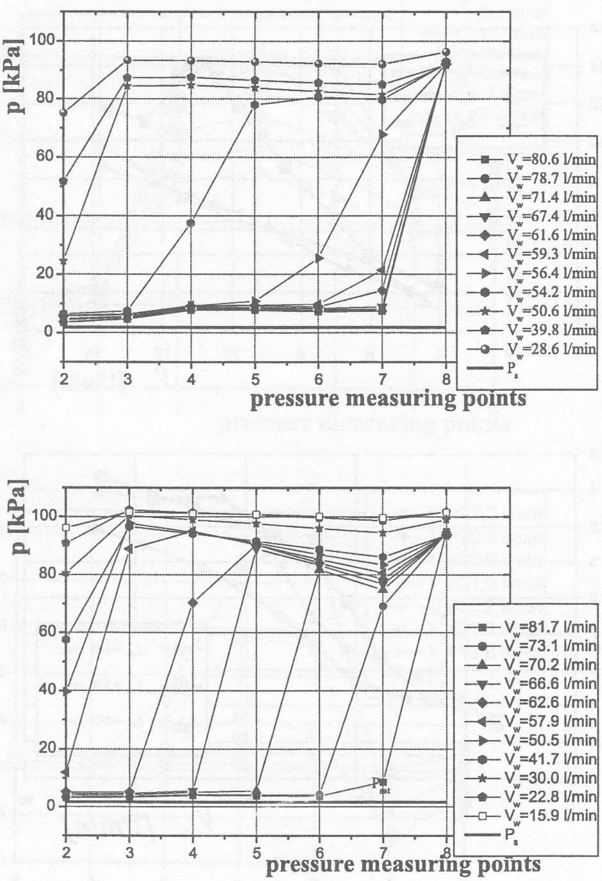
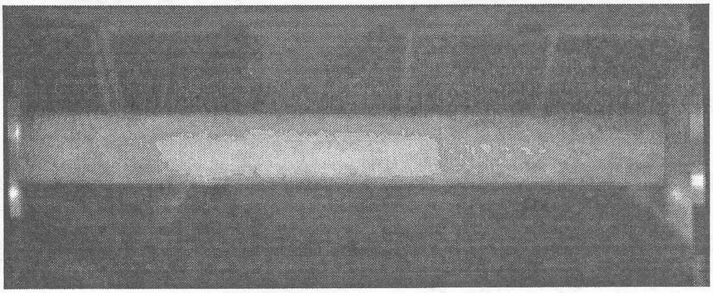


Fig. 7. Pressure distribution within the ejector for the case of zero air flow rate (the second limiting case) for various water flow rates: a) $D_n = 9.50$ mm, $L_{th} = 280$ mm; b) $D_n = 9.50$ mm, $L_{th} = 840$ mm.

function of the water flow rate. In both cases obtained have been lowest pressures equal to water saturation pressure, which indicates that at the nozzle outlet the expansion of water was taking place and a sudden increase of pressure should be considered as related to formation of a condensation shock wave. If there is diluted air in water (as it was taking place during experiments) then it is being released from the stream in the mixing chamber. Sample pictures of the condensation shock wave in the shortest and the longest mixing section have been shown in Fig. 8. In the case of a shock wave formed in the entry part of the mixing chamber we could observe that beyond the condensation shock wave we are dealing with a flow of water with a small number of air bubbles.

The location L_m of the beginning of the condensation shock wave as a function of water flow rate is presented in Fig. 9. From the presented diagram we can conclude, that the location L_m increases with increasing water flow rate. In the



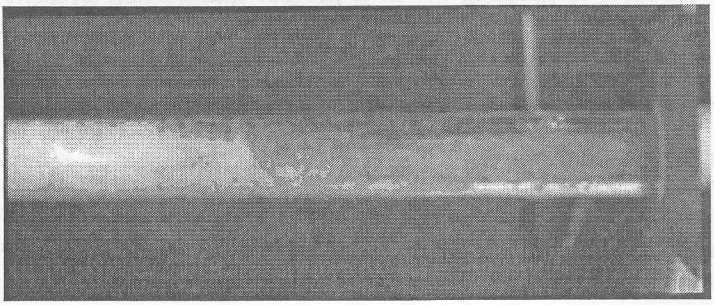


Fig. 8. The photographs of condensation shock wave zones: a) $D_n = 9.50$ mm, $L_{th} = 280$ mm, $\dot{V}_w = 56.4$ l/min (the whole length of the mixing chamber is visible); b) $D_n = 9.50$ mm, $L_{th} = 840$ mm, $\dot{V}_w = 62.7$ l/min (only the beginning of the condensation shock wave zone is visible).

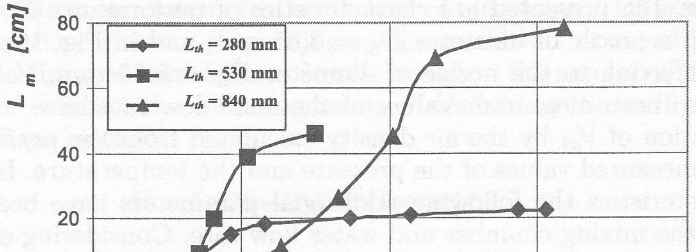


Fig. 9. Distance of the condensation zone from the mixing chamber entrance for various mixing chamber lengths.

case of highest rates of water the condensation shock wave is located in the final part of the mixing chamber and the location of its beginning is almost independent of water flow rate.

4. Characteristics of the ejector performance

The performance characteristic of a liquid-gas ejector in the mode of vacuum pump is presented usually in the form of the dependence of the pressure in the suction chamber as a function of the mass flow rate of intaken gas. Taking into account the geometry of the ejector, the pressure in the suction chamber is regarded as constant and equal to the pressure at the outlet from the nozzle of the working medium. Regarding air as a perfect gas we can present this relation in the form

$$p_{ai} = p_s(T_{ai}) + k(T_{ai})\dot{m}_{ai}. \tag{1}$$

A value of the coefficient k can be calculated from the formula

$$k = \frac{R_a}{\dot{V}_{ai}} T_{ai}. \tag{2}$$

According to [17], for given geometrical parameters of the ejector, constant temperatures of water and air as well as water pressure at inlet results in a value of coefficient k being constant. Only in the range of very small water flow rates we can expect some changes of this coefficient. It results from there, that if mentioned above quantities are constant, then the volumetric flow rate remain also constant. The above conclusion means also that the characteristic $p_{ai} = f(\dot{m}_{ai})$ should be linear in almost entire range of the air flow rate.

In Fig. 10a presented are characteristics of performance of the ejector equipped with a nozzle of diameter $D_n = 3.45$ mm, and in Fig. 10b similar characteristics referring to the nozzle of diameter $D_n = 9.50$ mm. Volumetric rate of air has also been measured. Values of the mass flow rate have been obtained by multiplication of $\dot{V}_{ai}$ by the air density calculated from the perfect gas equation based on measured values of the pressure and the temperature. In preparation of the characteristics the following additional parameters have been included: the length of the mixing chamber and water flow rate. Considering obtained characteristics we can conclude that most of them does not have a linear character. This means that the coefficient k cannot be regarded constant. In presented investigations it varied, for different rates of intaken air, even more than one order of magnitude. Such large changes cannot be explained by unavoidable temperature fluctuations of intaken air or water in the range of 1-3 K. Additional conclusion, which can be drawn from obtained results is that for the equal water flow rate and different lengths of the mixing sections the characteristics in almost entire range of air mass flow rate are the same. The only exception is shown in Fig. 10a regarding a nozzle with the smallest diameter and small water flow rates. In this case, for small values of $\dot{m}_{ai}$ obtained characteristics differ significantly between

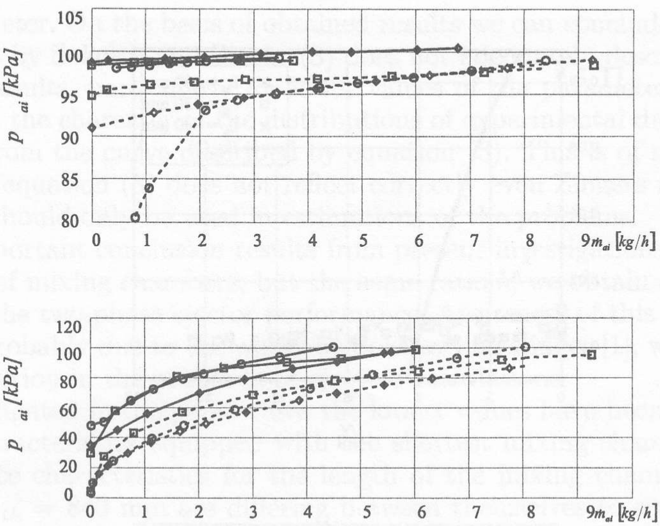


Fig. 10. Ejector performance: a) nozzle diameter $D_n = 3.45$ mm, $\bar{\dot{V}}_w = 13.7$ l/min (dashed line), $\bar{\dot{V}}_w = 9.3$ l/min (solid line), $L_{th} = 280$ mm, $L_{th} = 530$ mm, $L_{th} = 840$ mm; b) nozzle diameter $D_n = 9.50$ mm, $\bar{\dot{V}}_w = 76.5$ l/min (dashed line), $\bar{\dot{V}}_w = 51.5$ l/min (solid line), $L_{th} = 280$ mm, $L_{th} = 530$ mm, $L_{th} = 840$ mm.

themselves. It must be stressed, however, that obtained under-pressures are relatively small in this case and the divergence in characteristics does not play an important role in practical applications.

Another type of the performance characteristics of the two-phase ejector is suggested by Sokolov and Zinger [1] in the form of dependence of non-dimensional compression rate Π on the volumetric coefficient of ejection χ_i. Based on a theory of the single phase ejectors they adjusted the coefficient of velocity and the coefficient of the mixing section friction losses, based on the results of their experiments, and put forward the following relation describing the performance of two-phase liquid-gas ejectors

$$\Pi = 1.77b - 1.12b^2(1+\chi_i)^2. \tag{3}$$

Sokolov and Zinger postulate [1], that for a given geometrical parameter b, the performance characteristics of the ejector calculated according to (3) should be the same independently of the changes of water and air parameters and geometrical parameters, including the length of the mixing chamber. It must be stressed that the relation (3) was constructed based on experimental data, where only the influence of the diameter ratio b was considered assuming other geometrical parameters of the ejector constant. More detailed considerations regarding the characteristic $\Pi = f(\chi_i)$ have been presented by the present authors in [22].

In Fig. 11a-11c presented are experimental characteristics $\Pi = f(\chi_i)$ prepared for a given diameter ratio b and assuming the length of the mixing chamber

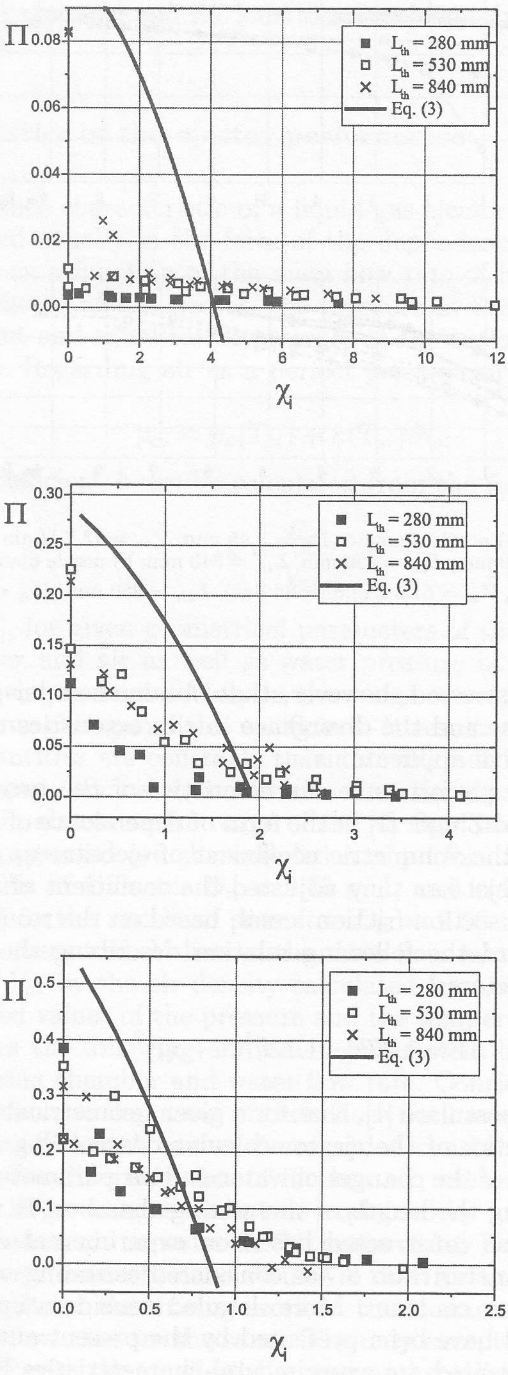


Fig. 11. Ejector performance curves $\Pi = f(\chi_i)$ with mixing chamber length as a parameter: a) $D_n = 3.45$ mm, b) $D_n = 6.00$ mm, c) $D_n = 9.50$ mm.

L_{th} as a parameter. On the basis of obtained results we can conclude that the relation provided by Sokolov and Zinger (3) does not adequately describe obtained experimental results, particularly for lower values of the parameter b. It is also important that the character of the distributions of experimental data is completely different from the curve described by equation (3). This is of no surprise to the authors as equation (3) does not reflect correctly even Zingers results [1, 10] and therefore should only be used in estimations of the problems.

A more important conclusion results from present investigations that for different lengths of mixing chambers, but the same ratio b, we obtain different characteristics of the two-phase ejector performance. Awareness of this fact is rather little known, probably due to the works by Sokolov and Zinger [1], which have an established position in the problems of ejectors calculations.

In all investigated in the paper cases the lowest values have been obtained for the ejector characteristics equipped with the shortest mixing chamber, whereas the performance characteristics for the length of the mixing chamber of L_{th} = 530 mm and L_{th} = 840 mm are differing between themselves to a much smaller extent, particularly in the region of large values of the ejection coefficient χ_i.

5. Friction losses in the mixing section

Bearing in mind considerations presented in the former chapter there arises now an issue of including a parameter, which would describe the influence of the length of the mixing chamber on the ejectors performance. A small number of investigators [5, 7, 9] consider the length of the mixing section by introduction of a relevant coefficient obtained experimentally. In the works [5, 7] it is assumed that proposed there coefficients of friction losses in the chamber are constant for a given ejector geometry, i.e. they are independent on the ejection coefficient χ_i.

The influence of the length of the mixing chamber on the coefficient of friction losses K_{th} will be analysed here using some elements of Cunninghams model [9]. In this model it is assumed that air compression in the ejector is isothermal. Additionally it is assumed [9], that up to the mixing zone (shock wave) both phases flow separately and beyond the transition through a shock wave the two-phase flow is homogeneous and interfacial slip is equal zero. It is considered also that during the flow of both phases through the ejector the solubility of the gaseous phase in the liquid one does not change. Additionally, at the beginning of the mixing chamber there is the pressure equal to the pressure in the suction chamber and hence it can be assumed that $\chi_i = \chi_0$. The momentum equation for the mixing chamber can be written as follows

$$(p_0 - p_t)A_t - \tau A_w = (\dot{m}_w + \dot{m}_a)w_t - \dot{m}_w w_{w0} - \dot{m}_a w_{ai}. \tag{4}$$

Bearing in mind the above assumptions (4) will be transformed to [9]

$$(p_0 - p_t) - \tau\frac{A_w}{A_t} = \Big[(1+\gamma\chi_0)(1+\chi_t)b^2 - \Big(1+\gamma\frac{\chi_0^2}{c}\Big)b\Big]\rho_w w_{w0}^2. \tag{5}$$

Friction losses appearing on the left hand side of equation (5) can be calculated using the relation [9]

$$\frac{A_w}{A_{th}}\tau = 4\frac{L_{th}}{D_{th}}\tau = \frac{1}{2}K_{th}\rho_t w_t^2, \tag{6}$$

which is a definition of the friction loss coefficient K_{th} in the mixing chamber. Including the relations linking the air parameters of state during its isothermal compression, from relation (6) we can obtain [9, 23]

$$\frac{A_w}{A_{th}}\tau = \frac{1}{2}K_{th}(1+\gamma\chi_0)(1+\chi_t)b^2\rho_w w_{w0}^2. \tag{7}$$

From (5) and (7) we get a sought formula describing the friction loss coefficient in the mixing chamber

$$K_{th} = \frac{2b + 2\gamma\chi_0^2\frac{b}{c} - \frac{p_t - p_0}{\rho_w w_{w0}^2}}{b^2(1+\gamma\chi_0)(1+\chi_t)} - 2. \tag{8}$$

As can be seen, the coefficient K_{th} is a function of geometrical and flow parameters of the ejector. A value of the ratio of gas and water volumetric flow rates can be calculated based on the assumption of isothermal air compression

$$\chi_t = \chi_0\frac{p_0}{p_t}. \tag{9}$$

Cunningham [16] obtained in his experimental investigations the values of friction loss in the range from $K_{th} = 0.15$ at $L_{th}/D_{th} = 16$ to $K_{th} = 0.43$ at $L_{th}/D_{th} = 28$, and the measurements were conducted at the diameter ratio $b = 0.30$. A characteristic feature of results obtained by Cunningham [16] was that K_{th} remained constant in the range of the ejection coefficient χ_i from zero up to some threshold value obtained at the level of about 1.5 to 1.75 (greater values for longer mixing chambers). Above this value of χ_i the coefficient of frictional losses first slightly rises and than decreases. It must be stressed, however, that Cunningham [16] conducted his experiments in a narrow range of the ejection coefficient, i.e. up to $\chi_i = 2.00$ and only for one diameter ratio b. Therefore, his results cannot provide a sufficient database for the assessment of the influence of the length of the mixing chamber and the ejection coefficient on a value of the friction loss coefficient in the mixing chamber. It ought to be noticed also, that Cunningham [16] suggests, that in calculations of the two-phase ejectors a constant value of the friction loss coefficient should be taken, however, without its specification.

Based on relation (8) we could attempt to assess the extreme values, which could be taken by the friction loss coefficient. Bearing in mind that the density ratio γ is of the order of no more than 10^{-3}, the following simplification is justified:

$1+\gamma\chi_o \approx 1$. If we were to consider the first limiting case, i.e. assume $p_t - p_0 = 0$, than based on (8) and (9) we obtain

$$K_{th1} = \frac{2\dfrac{b}{\chi_0}}{b^2\left(\dfrac{1}{\chi_0}+1\right)} - 2. \tag{10}$$

If we respect a fact that in the considered limiting case the ejection coefficient reaches values tending to infinity then we get

$$K_{th1} = -2. \tag{11}$$

Such value of the friction loss coefficient is concerned with a limiting case, where the amount of air supplied to the mixing section is so large that it is not possible to pump it through the mixing chamber and the entire flow goes in the direction of the suction chamber. Obviously, such case is practically unattainable, as the amount of intaken air is linked with achieving of the speed of sound in a two-phase flow in the mixing chamber [9, 23].

In the other limiting case we have a value of the ejection coefficient equal zero. Than on the basis of (8) and (9) we get

$$K_{th2} = 2c + \frac{p_t - p_0}{b^2 \rho_w w_{w0}^2}. \tag{12}$$

The values of the friction loss coefficient must be contained between discussed above limiting cases and hence

$$K_{th1} \leq K_{th} \leq K_{th2}. \tag{13}$$

It must be borne in mind that outlined above relations are valid only when at the outlet from the mixing chamber there is a homogeneous two-phase flow. Observations of the structure of two-phase flows existing inside the mixing chamber prove that such situation is not taking place in all considered cases, as sometimes we have to deal with at least a partially separated flow. Such situation inclines to deduce, that we can expect significant discrepancies in analysed coefficient K_{th} in cases of other geometries and operational conditions.

In Fig. 12 presented are experimental values of K_{th}, calculated using (8), for two different diameters of feeding nozzle ($b = 0.0607$ and $b = 0.430$) and different lengths of the mixing chambers ($L_{th}/D_{th} = 20.00$; 37.86; 60.00). Therefore, presented results encompass a much wider range of parameters in comparison with described above Cunninghams investigations [16].

It is very interesting that in the results presented in Fig. 12b, there appears an influence of water mass flow rate at the ejection coefficient $\chi_i < 0.8$. Usually the influence of water flow rate on the friction losses is neglected by many investigators. As can be seen, such influence exists and under some conditions is quite

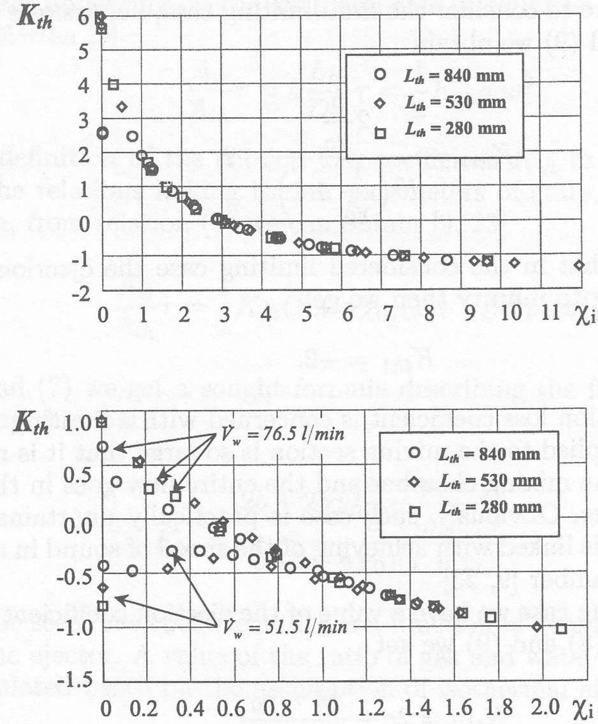


Fig. 12. Friction loss coefficient of the mixing chamber for nozzle diameters: a) D_n = 3.45 mm, b) D_n = 9.50 mm.

perceptible.

The following conclusions result from presented distributions of K_{th}. Firstly, the coefficient K_{th} varies significantly with the change of a volumetric ejection coefficient χ_i, which means, that its constant value cannot be assumed, as it has been done in [5, 7, 9].

The second very important conclusion is a fact, that a negative value of this coefficient has been obtained in the majority of experimental points. This can be explained in a twofold manner. The first is that in a final part of the mixing chamber there exists a non- homogeneous structure of the two-phase flow, which causes that the necessary conditions in the relation (8) are not fulfilled. A second circumstance is that by means of observations of the two-phase flow there have been found the secondary flows in the vicinity of the mixing section walls. Existence of such flows is a well known phenomenon [1]. In such situation it can be expected that the direction of shear stresses on the wall will be negative to the direction present when the flow in the entire volume of the mixing chamber would flow toward the diffuser.

A next conclusion, which can be drawn on the basis of graphs presented in Fig. 12 is that in the range of larger values of the ejection coefficient there is a surprisingly good consistency between the curves obtained for different lengths of mixing chambers, particularly in the light of discussion presented above about the two-phase flow structure. It can therefore be said, that the suggested relation describing the friction loss coefficient K_{th} has been adequately specified and is suitable for calculations of friction losses in the mixing chamber of the two-phase flow ejectors. It can therefore be presumed, that in this case lack of influence of the length of the mixing chamber results from the extent of friction losses. These losses, in the two-phase flow ejector, are only decisive in the area of the shock wave, and prior to its occurrence they can be neglected [24].

6. Efficiency of ejector performance

From the point of view of the influence of the length of the mixing chamber on the ejector performance it is of importance to investigate the level of available efficiency with respect to the length of the ejector. It must be stressed, that the issue of definition of the efficiency of the ejector may still be a matter of discussions [25]. In the present work, as the most appropriate definition we have assumed a ratio of isothermal power of gas compression and the power of isochoric pumping of motive liquid [23]

$$\eta_T = \chi_i \frac{p_{ai} \ln \dfrac{p_d}{p_{ai}}}{p_{wi} - p_d}. \tag{14}$$

In Fig. 13 presented are graphs of the dependence of efficiency on the ejection coefficient for two values of nozzle diameter ($D_n = 3.45$ mm and $D_n = 9.50$ mm) and three different lengths of the mixing chamber under two different values of water flow rate. It results from the graphs that there is a significant influence of the mixing chamber length on the isothermal efficiency. The lowest values of the efficiency have been obtained in the case of the shortest investigated mixing chamber. In the case of the nozzle of diameter $D_n = 3.45$ mm the most efficient was the longest mixing chamber, and in the case of a nozzle of diameter $D_n =$ 9.50 mm the highest efficiencies have been obtained for the length of the mixing chamber $L_{th} = 530$ mm. It should be recalled (see Fig. 6), that in the case of this chamber there were obtained the highest values of the ejection coefficient in the majority of runs. A second conclusion which can be drawn, based on presented results, is that the influence of the mixing chamber length on the efficiency is similar for all cases, independently of water flow rate, even though the efficiency itself depends on such flow rate. A highest efficiency of the order of about 12% has been obtained in the case of the ejector equipped with a mixing chamber of the length $L_{th} = 530$ mm and a nozzle of diameter $D_n = 9.50$ mm and a volumetric flow rate of $\dot{V}_w = 51.5$ l/min (see Fig. 13d).

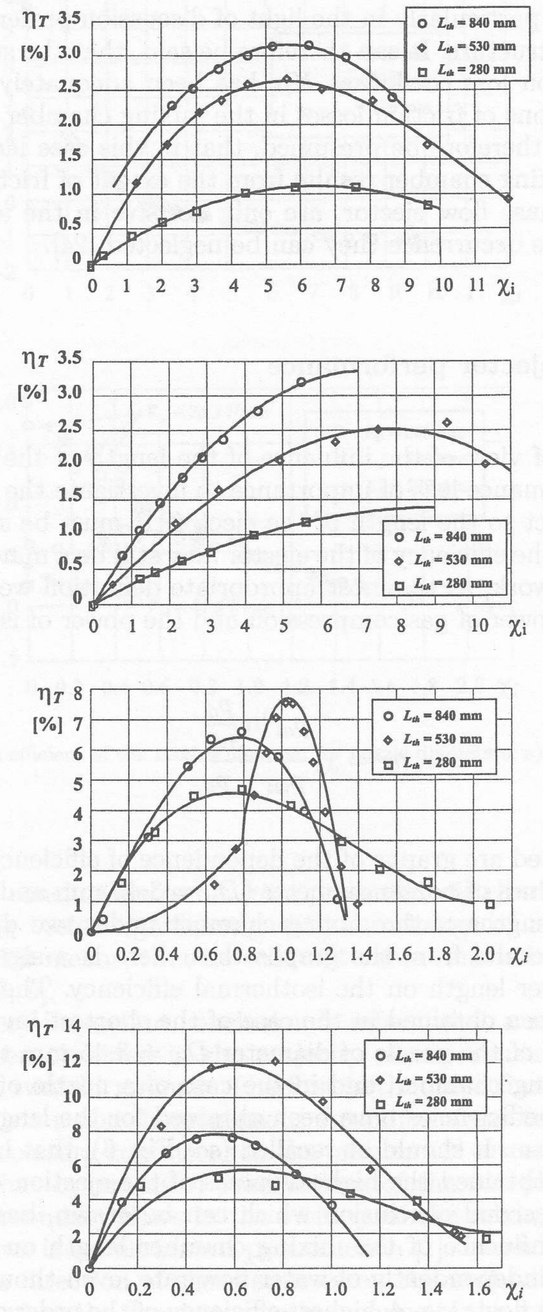


Fig. 13. Ejector efficiency: a) $D_n = 3.45$ mm, $\bar{\dot{V}}_w = 13.7$ l/min; b) $D_n = 3.~45$ mm, $\bar{\dot{V}}_w = 9.3$ l/min, c) $D_n = 9.50$ mm, $\bar{\dot{V}}_w = 76.5$ l/min; d) $D_n = 9.~50$ mm, $\bar{\dot{V}}_w = 51.5$ l/min.

7. Conclusions and final remarks

There are presented the results of extensive experimental investigations of the influence of the two-phase flow ejector geometry on its operational characteristics and the efficiency. The analysis of the distribution of the coefficient of friction losses K_{th} in the mixing chamber for different cases of the ejector performance has been investigated. Based on obtained results of experimental investigations we can draw following conclusions:

- There exist two limiting cases of operation of a two-phase flow ejector. In the first limiting case obtained are the highest values of the ejection coefficient and in the second limiting case obtained are extreme values of compression ratio.
- There is an influence of the length of the mixing chamber on maximum values of the ejection coefficient, and in the majority of measurements the highest values of this coefficient have been obtained in the case of a chamber with intermediate length of all investigated chambers.
- A condensation shock wave, which is formed in the second limiting case, changes its location with respect to water flow rate. In the case of the highest values of volumetric water flow rate, the beginning of the shock wave is located at a final part of the mixing chamber, just prior to the entrance to the diffuser.
- The influence of the length of the mixing chamber on operational characteristics of ejector in the form $p_{ai} = f(\dot{m}_{ai})$ and $\Pi = f(\chi_i)$ is not very significant, if the shock wave occurs in the mixing chamber.
- Characteristics $p_{ai} = f(\dot{m}_{ai})$ are not linear, which means that the proportionality coefficient in relation (1) is not constant, contrary to the contemporary belief.
- Characteristics of the ejector operation prepared on the basis of widely accepted relations due to Sokolov and Zinger (3) do not satisfactorily describe experimental data, mainly due to the fact that they do not account for the influence of the length of the mixing chamber.
- Based on conducted analysis of the coefficient of losses K_{th} in the mixing chamber it has been shown that it is variable (and not constant as usually assumed) and a range of its variations has been determined.
- In the case of higher values of the ejection coefficient obtained are negative values of the coefficient of frictional losses K_{th}, which can be explained by the existence of secondary flows in the mixing chamber and a non-homogeneity of the two-phase flow structure.
- There has not been observed influence of the length of the mixing chamber on a value of the friction loss coefficient, if the ejection coefficient is beyond some limiting value. In the case of conditions determined in Fig, 12a the limiting value χ_i is close to zero, whereas in the case of conditions presented in Fig. 12b it is equal to about $\chi_i = 0.8$.

- There is a strong dependency between the length of the mixing chamber and the ejector efficiency. The lowest values of the efficiency have been obtained in the case of the ejector fitted with the shortest mixing chamber.

Above remarks and conclusions refer to a two-phase water-air ejector, with a sharp-edged nozzle. It should be expected that, due to the influence of the shape of the nozzle on the flow stability, there will also be the influence of the shape of a nozzle on performance of the ejector.

Akcnowledgement The paper was supported by the Polish Committee for Scientific Research, Grant No 8T 10B 02413.

Received 15 March 2001

References

[1] Sokolov E. J., Zinger N. M.: *Jet apparatuses*, Energoatomizdat, Moskva, 1989, (in Russian).

[2] Goliski J. A., Troskolaski A. T.: *Ejectors*, WNT, Warszawa, 1979, (in Polish).

[3] Filipczak G.: *Jet devices – review of design and application*, Inżynieria i Aparatura Chemiczna, No 4, 1985, 3-7, (in Polish),.

[4] Cramers P. H. M. R., Smit L., Leuteritz G. M., van Dierendonck L. L., Beenackers A. A. C. M.: *Hydrodynamics and local mass transfer characteristics of gas-liquid ejectors*, The Chem. Eng. J., Vol. 53, 1993, 67-73.

[5] Hongqui L.: *Liquid gas two phase flow theory of jet pump*, Inter. Conf. on the *Physical Modelling of Multi-Phase Flow*, Coventry, England, 1983, 439-451.

[6] Witte J.H.: *Efficiency and design of liquid-gas ejectors*, British Chemical Engineering, No 9, Vol. 10, 1965, 602-607.

[7] Biswas M. N., Mitra A. K.: *Momentum transfer in horizontal multi-jet liquid-gas ejector*, The Can. J. of Chem. Eng., October, Vol. 59, 1981, 634-637.

[8] Henzler H.-J.: *Das Sogverhalten von Strahlsaugern für das Stoffsystem: flüssig – gasförmig*, Verafahrenstechnik, Vol. 15, No 10, 1981, 738-749.

[9] Cunningham R. G.: *Gas compression with the liquid jet pump*, Transactions of the ASME, Journal of Fluids Engineering, No 3, Vol. 94, 1974, 203-215.

[10] Zinger K. M.: *Investigation of water-air ejector*, Teploenergetica, No. 8, 1958, 26-31, (in Russian).

[11] Berman L. D., Efimotschkin G. I.: *Relationships for calculation of water-jet ejectors*, Teploenergetica, No. 7, 1964, 44-48, (in Russian).

[12] Berman L. D., Efimotschkin G. I.: *Method of calculation of water-jet ejector*, Teploenergetica, No. 8, 1964, 92-94, (in Russian).

[13] Berman L. D., Efimotschkin G. I.: *On calculation of water-jet ejectors for condensation plants*, Teploenergetica, No. 7, 1983, 57-58, (in Russian).

[14] Efimotschkin G. I., Korhennov B. E.: *Calculation method of water-air ejector with elongated mixing chamber*, Teploenergetica, No. 1, 1976, 84-86, (in Russian).

[15] Efimotschkin G. I.: *Design and calculation of water-jet ejectors with elongated mixing chamber*, Teploenergetica, No. 12, 1989, 48-51, (in Russian).

[16] Cunningham R. G.: *Jet breakup and mixing chamber lengths for the liquid jet gas pump*, Trans. of the ASME, J. of Fluids Eng., No. 3, Vol. 94, 1974, 216-226.

[17] Berman L. D., Efimotschkin G. I.: *Experimental investigation of water-jet ejector*, Teploenergetica, No. 9, 1963, 9-15,(in Russian).

[18] Berman L. D., Efimotschkin G. I.: *Peculiarities of working process and work parameters of water-jet ejector*, Teploenergetica, No. 2, 1964, 31-35, (in Russian).

[19] Bazhenov M. I.: *Experimental investigation of transparent model of water-air jet device*, Izvestha Vissikh Uchebnikh Zaveedeni, No 3, 1966, 82-86.

[20] Korhennov B. E.: *Working process in gas-liquid ejector*, Teploenergetica, No. 1, 1977, 59-65, (in Russian).

[21] Berman L. D., Efimotschkin G. I.: *The effect of mixing chamber on work parameters and economic efficiency of water-jet air ejector*, Teploenergetica, No. 12, 1978, 66-71, (in Russian).

[22] Butrymowicz D., Trela M., Ihnatowicz E., Karwacki J.: *Experimental investigations of the effect of the nozzle geometry on work of water-air ejector*, Transactions of IFFM, No. 107 (2000), 1-22.

[23] Trela M., Butrymowicz D., Rybnik R.: *Modelling of liquid-gas ejector*, Part 1, Report of IF-FM, No. 125/98, Gdańsk, 1998, (in Polish).

[24] Kwidziński R.: *Stationary shock wave in the flow of a water-vapour mixture*, Scientific Notes of IFFM, No. 498/1458/99, Gdańsk, 1999.

[25] Paliwoda A.: *Ejector refrigeration devices*, WNT, Warszawa, 1971, (in Polish).

[26] Cattadori G., Galibati L., Mazzochi L., Vanini P.: *A single-stage high pressure steam injector for next generation reactors: test results and analysis*, Inter. J. of Multiphase Flow, Vol. 21, No. 4, 1995, 591-606.