Performance of Low-Cost Ejectors

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Abstract: Despite its widespread use in agricultural systems, the Venturi-type ejector presents lower efficiency than that obtained by conventional jet pumps. In this study, low-cost ejectors built from PVC T-junctions, with geometry similar to that of the conventional jet pumps, were developed and evaluated experimentally. The tests were carried out on ejectors with nominal diameters of 25 and 32 mm with area ratios of 0.25, 0.35, and 0.53. For each diameter, a Venturi-type ejector with an area ratio of 0.35 was also employed. The ejectors developed in this research presented efficiency between double and triple of that of the Venturi type, with a maximum of 30.5% being reached with the ejector of 25 mm and area ratio of 0.35. The head-loss coefficients for each component of the ejectors were evaluated by fitting a one-dimensional model. The results showed that the low-cost ejectors developed here presented an operation comparable to that of conventional jet pumps and had a better benefit/cost ratio than the Venturi-type ejectors.

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Introduction

Ejectors, also called jet pumps, are devices usually manufactured with foils and metallic tubes modeled in a power lathe. Composed basically of five parts, namely, driving and suction nozzle, suction chamber, mixing chamber or throat, and diffuser, as schematically shown in Fig. 1, these accessories are used in the fields of civil, mechanical, chemical, and industrial engineering for the suction and the elevation of liquids, gases, or even granular solids. Their most frequent applications are well-pumping, hydraulic dredging, priming of pumps or siphons, drainage of ditches, mixing or dilution of fluids, and aeration of tanks. The commercially available jet pumps used in these cases, similar to the ones tested by Winoto et al. (2000), present maximum efficiency of the order of 30% and their cost, depending on the manufacture complexity and the type of materials employed, can vary from $200 to $500 (U.S. dollars throughout), in local market, for a 32 mm nominal diameter device.

With the appearance of the Venturi-type ejectors manufactured in PVC or polyethylene, which are more compact and cheaper than the conventional jet pumps, it is found that these accessories are being increasingly employed in agricultural systems for the injection of chemical products in irrigation pipelines. Although the cost of these devices is around $60 for nominal diameters of 32 mm, the maximum efficiency is usually less than 15%, according to Ferreira (1994), who tested Venturi-type ejectors available in the market.

In an effort to reach larger efficiency for chemical product applications in agricultural systems, ejectors of simpler construction and reduced cost were evaluated as well as ejectors of the Venturi type for comparison purposes. The research experimentally evaluated each ejector under various pressure and flow conditions and investigated the influence of the driving nozzle/throat area ratio on the functioning of the ejectors. A theoretical formulation was developed to determine the efficiency and head-loss coefficients for each component of the ejectors, from which theoretical efficiency curves for several area ratios on the basis of these coefficients were developed.

All analyses were made for primary and suction fluids of the same density. For fluids of different densities, high viscosity or with sediments in suspension, the theory should be modified. Researchers such as Hatziavramidis (1991) and Cunningham (1995) analyzed the performance of the ejectors operating with different fluids.

Material

The ejectors developed in this work were composed of two pieces shaped from PVC bar, which were inserted in a tee junction of the same material. In Figs. 2 and 3, where various geometrical dimensions are shown, the hatched parts correspond to the shaped pieces and the dashed lines represent the tee. Fig. 2 shows the

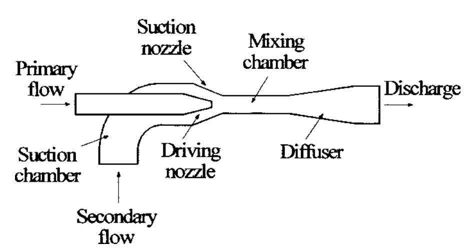

Fig. 1. Components of conventional jet pump

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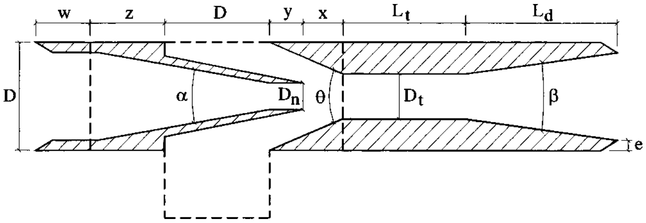

Fig. 2. Type A ejector

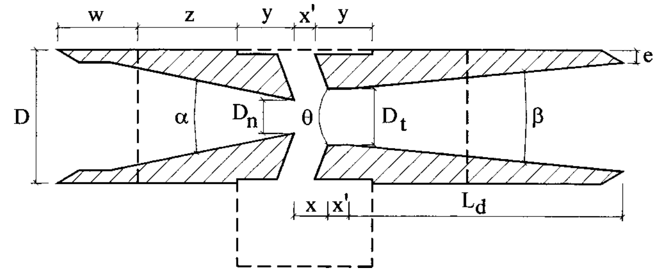

Fig. 3. Type B ejector (Venturi)

Type A ejector, with geometry similar to that of the conventional jet pumps, while Fig. 3 shows the Type B ejector, a more compact ejector without a mixing chamber and similar to the Venturi-type ejectors developed by Rojas (1995).

For each nominal diameter D of the tees, of 25 and 32 mm, four ejectors were built, three Type A and one Type B. Representing the nominal diameters by numbers in the parentheses, the tees are denominated as having the dimensions $z(25)=18.0$ mm and $z(32)=22.5$ mm. In order to provide for sufficient overlap between the pipeline and the ejector entrance, the overlap lengths were taken as $w(25)=23.4$ mm and $w(32)=30.0$ mm. It was also decided that the thickness e in any part of the pieces, except in the chamfered areas, should be larger than 2 mm in order to avoid cracks in the material during the machining process. Therefore, $e(25)=2.34$ mm and $e(32)=3.00$ mm were adopted, with the other dimensions being similar to the limits recommended by researchers, like Mueller (1964) and Sanger (1970), who evaluated the efficiency of the ejectors in terms of their geometry. The parameters adopted for the various ejectors are presented in Table 1, R being the ratio between the areas of the driving nozzle (A_n) and the throat (A_t).

The shaped pieces and a short tube of PVC were glued inside the tees. Three splices were also glued to the ends of the ejectors so that these could be connected to the main and the suction lines. The total cost of construction of these ejectors, including labor, was around $30.

The items necessary for carrying out the tests were a centrifugal pump with a capacity of 8 m^3/h and a manometric head of 60 m H_2O; two digital discharge meters with transducers of inductive magnetic type, with diameters of 25 and 32 mm and capacities respectively of 8 and 7 m^3/h, both with an accuracy of $\pm 1\%$; two digital manometers, with a precision of two decimal points and a bottom scale of 60 m H_2O; a mercury column vacuum meter, with millimeter scale and bottom scale of 750 mm Hg; two 100 L water tanks in addition to gate, globe, and needle valves.

The experimental setup is shown in Fig. 4. The pump (1) delivered the water from the feed tank to the ejector (9). The secondary fluid was lifted from the water tank (11) with the delivery tank being supplied by the resulting flow. The water tank (11) was fed by the auxiliary tank (13) to avoid turbulence problems in the suction. The auxiliary pump supplied this tank (13) as well as the feeding tank. The delivery water tank returned to the reservoir of the auxiliary pump through an overflow pipe. The exit pipeline from the pump (1) was of 50 mm diameter, which was also the size of the feeding, suction, and discharge pipes, as well as of the needle and globe valves and the tees used for the construction of the ejectors. The total length of the pipelines varied in accordance with the length L_e of each ejector.

Experimental Study

Two types of tests were conducted for each kind of ejector in order to analyze the ranges of maximum efficiency and the head-loss coefficients in each component of these accessories.

Test for Maximum Efficiency

The pressure head in the first manometer (Point A) varied from 10 to 50 m H_2O in increments of 5 m H_2O. This was obtained through adjustments in the needle valve upstream of the ejectors keeping the other valves open. The efficiency of the ejectors was analyzed as a function of the Reynolds number (R) in the feed pipes.

Test for Efficiency Curves

The pressure heads in the first manometer (Point A) and in the tee (Point B) were fixed in respectively 50 and -1.60 m H_2O. The suction and discharge valves were manipulated to obtain various discharges in the secondary flows. Further, the primary flow was also fixed for each ejector in view of the fact that at driving pressure heads higher than 40 m H_2O, the pressures as well as the driving flows did not change when the valves were adjusted. With a fixed number of 12 experimental data points, efficiency (η) versus flow ratio (M) curves were obtained and the head-loss coefficients in each component of the ejectors were determined by fitting a one-dimensional model.

Table 1. Ejector Dimensions

Ejector	$R=A_n/A_t$	α	β	θ (deg)	D_n/D	x/D	x'/D	y/D	L_t/D	L_d/D
B(25)	0.35			140.0			0.156	0.422		2.240
A1(25)	0.25			38.3				0.470	2.500	1.780
A2(25)	0.35			43.9				0.470	2.100	2.240
A3(25)	0.53	20°	10°	49.0	0.250	0.250		0.470	1.720	2.672
B(32)	0.35			140.0			0.156	0.422		2.234
A1(32)	0.25			39.2				0.453	2.500	1.781
A2(32)	0.35			44.7				0.453	2.109	2.234
A3(32)	0.53			50.0				0.453	1.719	2.672

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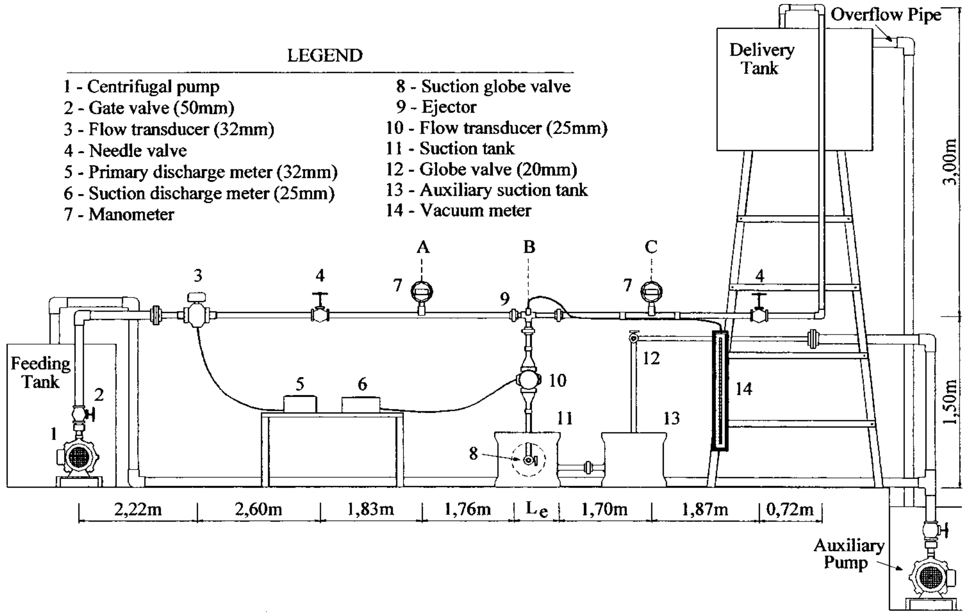

Fig. 4. Schematic of experimental setup

Theoretical Analysis

Efficiency Equation

In Fig. 5, Sections 1, 2, and 3 respectively represent the ejector's entrance, suction, and discharge locations. The driving and suction flows and the pressures at Points A and C (Fig. 4) were measured and the heads at ejector Sections 1 and 3 (Fig. 5) were calculated by the energy equation. The friction head losses involved in energy balance were calculated by the Darcy-Weisbach equation.

$$h=f\frac{L}{D}\frac{V^2}{2g} \tag{1}$$

The friction factor f was calculated by the formula of Swamee and Jain (1976) considering the kinematic viscosity $\nu = 10^{-6}\,\mathrm{m^2/s}$ (water at 20°C) and the absolute roughness $\varepsilon = 0.0015$ mm (tube of PVC).

$$f=\frac{0{,}25}{\left\{\log\left[\frac{\varepsilon}{3{,}7D}+\frac{5{,}74}{(VD/\nu)^{0{,}9}}\right]\right\}^2} \tag{2}$$

In addition, it was assumed (Lima Neto 2001) that the suction pressure in the PVC ejector (Section 2 in Fig. 5) was the same as that at Point B (Fig. 4).

The efficiency equation used in this work was the same proposed by Silvester (1961) for a centrifugal pump-ejector association expressed in terms of the flow ratio M and head ratio $N = (H_3-H_2)/(H_1-H_3)$.

$$\eta = MN = \frac{Q_2}{Q_1}\cdot\frac{H_3-H_2}{H_1-H_3} \tag{3}$$

This equation was derived through a power budget among Sections 1, 2, and 3 (Mueller 1964).

Similar to models developed by Reddy and Kar (1968), Sanger (1970), Hatziavramidis (1991), Cunningham (1995), and Winoto et al. (2000), the one-dimensional model proposed in this study was based on the conservation equations for energy, momentum, and mass. Therefore, these equations were applied between sections 1, 2, 3, 4, and 5, then the theoretical head ratio N' expressed in terms of M, R and the head-loss coefficients in each component of the ejectors was obtained.

$$N'=\frac{2R+\frac{2R^2M^2}{1-R}-(1+K_t+K_d)R^2(1+M)^2-(1+K_s)\frac{R^2M^2}{(1-R)^2}}{1+K_n-2R-\frac{2R^2M^2}{1-R}+R^2(1+M)^2(1+K_t+K_d)} \tag{4}$$

From Eq. (3), the theoretical efficiency is given by

$$\eta' = MN' \tag{5}$$

It was assumed that the jet influences the suction flow only at the throat entrance, that the mixing flow was one-dimensional, the

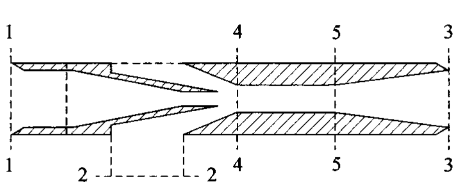

Fig. 5. Ejector sections

Table 2. Coefficient Bounds on Each Ejector Type

Type A	Type B
$0.041<K_n<0.181$	$0.156<K_n<0.262$
$K_s>0.90$	$K_s>0.90$
$0.060<K_t<0.075$	$K_t=0$
$0.10<K_d<0.30$	$0.20<K_d<0.40$

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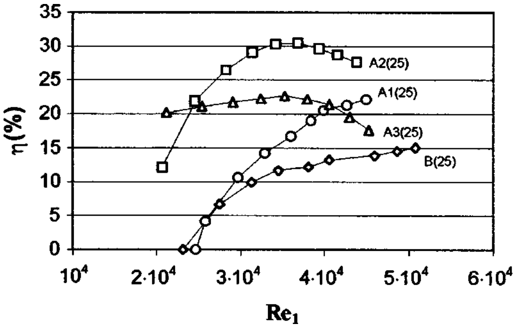

Fig. 6. Efficiency of ejectors with 25 mm diameter

pressure was constant at the throat entrance and exit, that the average throat velocity was constant and equal to the velocity at the diffuser entrance, and that the primary and secondary fluids were of the same density, incompressible, and were at the same temperature.

The terms K_n, K_s, K_t, and K_d respectively correspond to the head-loss coefficients in the driving nozzle entrance, suction chamber, throat, and diffuser. Whereas the coefficient K_s depends on the geometry of the suction chamber, K_n, K_t, and K_d can be expressed in the following forms:

$$K_n=\left(\frac{1}{C_d^2}-1\right) \tag{6}$$

$$K_t=f\frac{L}{D} \tag{7}$$

$$K_d=(1-\eta_d) \tag{8}$$

Eqs. (6) and (7) were suggested by Mueller (1964), where C_d is the discharge coefficient of the driving nozzle and f, L, and D are respectively the friction factor, length, and diameter of the throat. Eq. (8) was suggested by Citrini (1956), where η_d is the efficiency of the diffuser.

Curve Fitting

The head-loss coefficients K_n, K_s, K_t, and K_d were obtained by fitting Eq. (5) to the experimental data. For each of the 12 measured M values, corresponding N and N' values were obtained. Thus, $\eta=MN$ and $\eta'=MN'$, the objective function was expressed in the following form and subject to the bounds shown in Table 2:

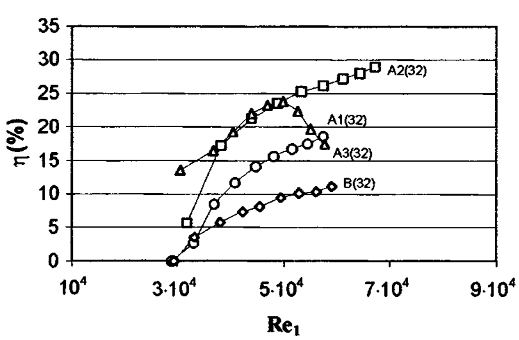

Fig. 7. Efficiency of ejectors with 32 mm diameter

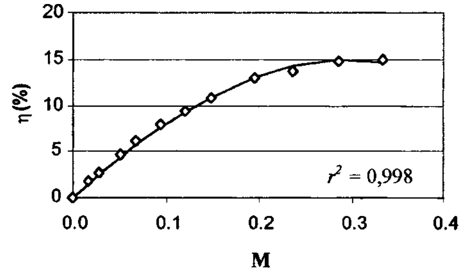

Fig. 8. Observed and fitted efficiency—Ejector B(25)

$$\underset{K_n,K_s,K_t,K_d}{\text{Minimize}}:\sum_{i=1}^{12-c}(\eta_i-\eta_i')^2 \tag{9}$$

These limits were applied on the basis of the values of C_d, f, and η_d obtained from Engel (1963), Mueller (1964), and Silvester and Mueller (1968). As the type B ejectors were without throat, $K_t=0$ and their efficiency was lower. Hence, the limits of K_n and K_d considered for these ejectors corresponded to smaller values of C_d and η_d. In both cases, a lower bound for K_s of 0.90 was the loss in the suction chamber, whereas it was larger than the loss in a 90° PVC elbow.

The variable c in the sum of squared deviations in Eq. (9) corresponded to the number of experimental points in which the flow entered the cavitation regime. These points were considered to be in the region in which efficiency suffers a sudden decline and which is characterized by typical cavitation phenomenon noises.

To evaluate the level of adjustment of the theoretical curves, the coefficient of determination r^2 was calculated by the following equation:

$$r^2=\frac{[(12-c)(\Sigma_{i=1}^{12-c}\eta_i\eta_i')-(\Sigma_{i=1}^{12-c}\eta_i)(\Sigma_{i=1}^{12-c}\eta_i')]^2}{[(12-c)\Sigma_{i=1}^{12-c}\eta_i^2-(\Sigma_{i=1}^{12-c}\eta_i)^2][(12-c)\Sigma_{i=1}^{12-c}\eta_i'^2-(\Sigma_{i=1}^{12-c}\eta_i')^2]} \tag{10}$$

Results and Discussion

Experimental results presented in Figs. 6 and 7 show the efficiency of the ejectors as a function of the Reynolds number in the entrance tube. It was verified that Type A ejectors with the area ratio $R=0.35$ were more efficient than the others, with a maximum efficiency of 30.5% for the A2(25) ejector. However, it can be seen through the inclination of the curve presented in Fig. 7 for the A2(32) ejector that the efficiency did not reach its maximum point because of lower pump capacity. Therefore, it could be in-

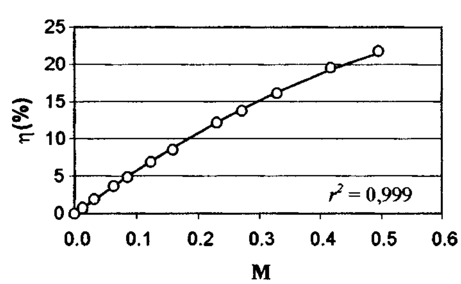

Fig. 9. Observed and fitted efficiency—Ejector A1(25)

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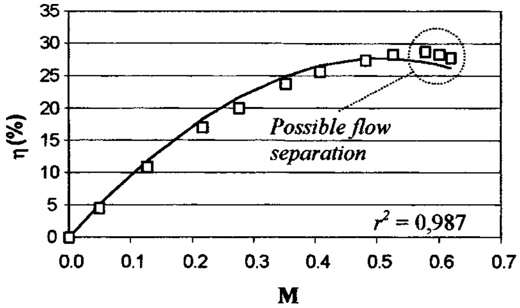

Fig. 10. Observed and fitted efficiency—Ejector A2(25)

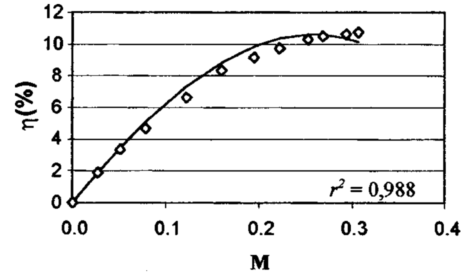

Fig. 12. Observed and fitted efficiency—Ejector B(32)

ferred that the maximum efficiency of the A2(32) ejector is even larger than that observed in its smaller equivalent, namely, the A2(25) ejector.

While analyzing the ejectors of area ratio $R=0.35$, it was verified that the efficiency of the Type A equipment was about two to three times the efficiency of the Type B, which reached a maximum efficiency of about 15% with the B(25) ejector.

Figs. 8–15 present the experimental data together with the fitted theoretical curves. In Figs. 10, 11, and 15, regions of possible flow separation and cavitation phenomenon are also indicated. According to Sanger (1970), the continuation of mixing into the diffuser results in larger head losses and, perhaps, in flow separation. Further, the A2(25) and A3(25) ejectors presented regions of gradual efficiency decline caused by a considerable increase of the head losses due to turbulence and it was probable that, in these regions, flow-separation phenomenon took place. In the Type A3 ejectors, it was observed that a characteristic cavitation noise coincided with the points of abrupt efficiency decline. According to Cunningham et al. (1970), the cavitation is caused by at least one of the following three factors: high jet velocity, low suction pressure, or low discharge pressure. As in the tests, the pressure heads in the first manometer and in the ejector suction were fixed, the cavitation was caused by high flow ratios and, consequently, by low delivery pressures.

The theoretical efficiency in Eq. (5) presented a good fit for all of the ejectors since the coefficient of determination r^2 varied from 0.934 to 0.999. It was observed that a much better agreement between the experimental and theoretical curves was obtained for smaller flow ratios. This may be due to the fact that the higher the flow ratio, the less uniform the total flow, the more variable the pressure at throat entrance and exit, and the less accurate the model.

In the A1, A2, and A3 ejectors with 25 mm diameter, the adjusted discharge coefficients C_d were respectively 0.940, 0.961, and 0.942. The coefficients K_s, f, and η_d were equal to 0.90, 0.012, and 0.90 for the three ejectors. In the A1, A2, and A3 ejectors with 32 mm diameter, the adjusted discharge coefficients were respectively 0.936, 0.980, and 0.927 and the other coefficients being equal to the ones obtained with the 25 mm ejectors.

Similarly, for the Type B ejector with 25 mm diameter, curve fitting produced the coefficients C_d, K_s, and η_d, respectively, of 0.930, 3.96, and 0.73, and for the ejector with 32 mm diameter, the values of 0.890, 5.25, and 0.60.

Figs. 16(a and b) and 17(a and b) present the theoretical curves N' versus M and η' versus M generated by using the average adjusted head-loss coefficients for the Types A and B ejectors for various area ratios R. Thus the following sets of coefficients were used for the Type A ejectors: $K_n=0.11$, $K_s=0.90$, $K_t=0.06$, and $K_d=0.10$; and $K_n=0.21$, $K_s=4.61$, and $K_d=0.33$ were used for the Type B ejectors.

The curves N' versus M generated for the Type A ejectors [Fig. 16(a)] were practically linear, while the ones generated for the Type B ejectors [Fig. 16(b)] were slightly parabolic. It was verified that the head ratio N' (and consequently the discharge head H_3 since H_1 and H_2 were fixed) was inversely dependent on the flow ratio M (and consequently on the suction flow Q_2 since Q_1 was fixed). Hence, for the same flow ratios, the Type A ejectors caused smaller head losses than the Type B ejectors. It could be seen for both types of ejectors that the lower the area ratio R, the lower the head ratio N', but the higher the range of flow ratio M.

The curves η' versus M [Figs. 17(a and b)] presented a parabolic form with small asymmetry. It was verified that as the area ratio R decreased, maximum efficiency η' increased and then decreased for both types of ejectors. This was due to the fact that the lower the area ratio R (higher throat diameters D_t since D_n was fixed), the lower the friction head loss was at the annular area of the secondary inlet until the point that the efficiency began to decline because the driving jet could not exchange its momentum very effectively with the secondary stream near the throat wall. Therefore, the maximum theoretical efficiency for the Type A ejectors, around 26%, was reached for area ratios R between 0.30

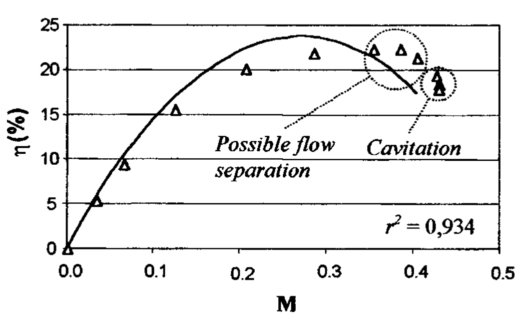

Fig. 11. Observed and fitted efficiency—Ejector A3(25)

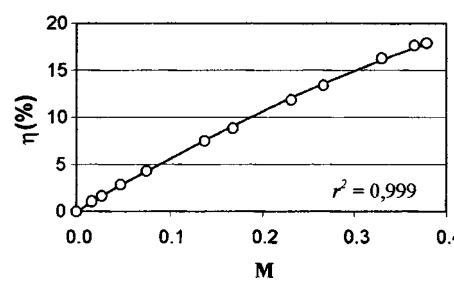

Fig. 13. Observed and fitted efficiency—Ejector A1(32)

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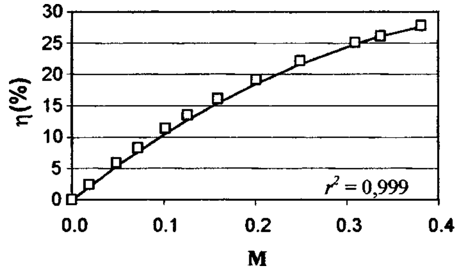

Fig. 14. Observed and fitted efficiency—Ejector A2(32)

and 0.40 and flow ratios M between 0.50 and 0.70. The maximum theoretical efficiency for the Type B ejectors, around 13%, was reached with $R=0.30$ and $M=0.35$. The value of $R=0.20$ allowed the equipment to operate at higher range of flow ratio M, the best operation point for the Type A ejectors being around $M=1.00$, and for the Type B ejectors, around $M=0.60$.

Since the average adjusted head-loss coefficient in the throat was the lowest one ($K_t=0.06$), its effect on the efficiency of the Type A ejectors was lower than 3% [Eq. (4)]. Thus the efficiencies of the Type A ejectors were higher than those of the Type B ejectors because the other average adjusted head-loss coefficients on the former case ($K_n=0.11$, $K_s=0.90$, and $K_d=0.10$) were lower than those on the latter case ($K_n=0.21$, $K_s=4.61$, and $K_d=0.33$). It could be inferred that the adjusted head-loss coefficients, K_s and K_d, were higher because on the Type B ejectors the suction inlet line joined the driving jet at a greater angle θ and there was a continuation of turbulent mixing into the diffuser (Figs. 2 and 3). As a consequence, the flow ratios M on the Type B ejectors were lower and the driving flows as well as the adjusted head-loss coefficients K_n were higher.

Summary and Conclusions

A simple methodology of design and construction of low-cost ejectors made from PVC was presented. Ejectors of nominal diameters of 25 mm and of 32 mm, with geometry similar to that of the conventional jet pumps (Type A) and with different area ratios, were built and evaluated experimentally under various pressure and flow conditions. A more compact ejector without a mixing chamber, similar to the Venturi-type ejector (Type B), was also built and evaluated.

It was verified that the ejectors developed in this work present similar operation to that of the conventional jet pumps. All of the ejectors tested were able to operate with applied heads of more than 15 m H_2O and with Reynolds numbers larger than $3\cdot10^4$. The Type A ejectors reached efficiency above 30% and the Type

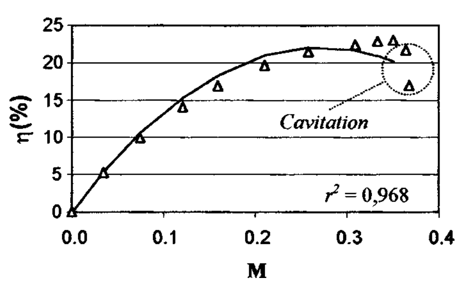

Fig. 15. Observed and fitted efficiency—Ejector A3(32)

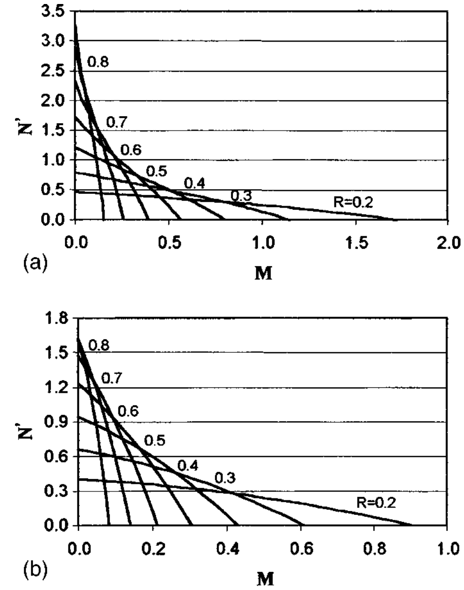

Fig. 16. Theoretical curves N' versus M—(a) Type A ejector and (b) Type B ejector

B between one-third and half of that value. Further, the cost of these accessories was around half of that of the Venturi-type ejectors used to apply chemicals into irrigation pipelines, the least expensive now available in the local market.

A one-dimensional model was proposed to evaluate the performance of the ejectors. The head-loss coefficients in each component of these accessories were obtained by fitting theoretical efficiency curves to the experimental data. The calibration achieved coefficients of determination r^2 varying between 0.934 and 0.999, which gave credence to considerations employed in the theoretical formulations. The theoretical performance of ejectors, for a wide range of area ratios in the form of efficiency curves, was generated on the basis of average coefficients of head loss determined through calibration. It was found that the lower the area

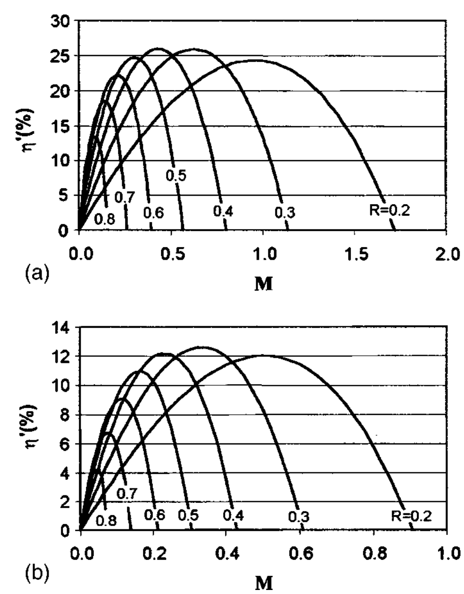

Fig. 17. Theoretical curves η' versus M—(a) Type A ejector and (b) Type B ejector

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ratio value, the higher the range of flow ratio reached with the ejectors. Further, the maximum theoretical efficiencies of 26 and 13% were obtained with area ratios around 0.30 for the Type A and the Type B ejectors, respectively, but the latter attained its maximum efficiency with smaller flow ratio than the former.

The experimental and theoretical studies showed that the Type A ejectors with area ratio of 0.35 are the most efficient design. However, it is advisable that the flow ratio on these accessories does not exceed the value of 0.50 in order to avoid efficiency declines caused by flow separation and cavitation. On the other hand, when higher flow ratios are required, the use of the Type A ejectors with area ratio of 0.20 is recommended because these accessories can operate with flow ratios of up to 1.00 without causing efficiency declines. Therefore, it is believed that the ejectors developed in this research could replace the Venturi-type ejectors in the drip-irrigation industry in order to reduce the costs and improve the performance of chemical application systems.

Notation

The following symbols are used in this paper:

A = area (mm^2);
A1(25) = Type A ejector of 25 mm ($R=0.25$);
A2(25) = Type A ejector of 25 mm ($R=0.35$);
A3(25) = Type A ejector of 25 mm ($R=0.53$);
A1(32) = Type A ejector of 32 mm ($R=0.25$);
A2(32) = Type A ejector of 32 mm ($R=0.35$);
A3(32) = Type A ejector of 32 mm ($R=0.53$);
B(25) = Type B ejector of 25 mm ($R=0.35$);
B(32) = Type B ejector of 32 mm ($R=0.35$);
C_d = discharge coefficient;
D = diameter (mm);
f = friction factor;
H = total head (m H_2O);
K = fitting loss coefficient;
L = length (mm);
M = flow ratio$=Q_2/Q_1$;
N = head ratio$=(H_3-H_2)/(H_1-H_3)$;
N' = theoretical head ratio;
P/γ = pressure head (m H_2O);
Q = flow (L/s);
R = Reynolds number;
R = driving nozzle-throat area ratio;
r^2 = coefficient of determination;
V = velocity (m/s);
x = distance of driving nozzle exit to throat entrance (mm);
x,y,z,w,e,x' = ejectors dimensions shown in Figs. 2 and 3 (mm);
α = driving nozzle angle (deg);
β = diffuser angle (deg);
ε = absolute roughness of tube (mm);
η = experimental ejector efficiency (%);
η' = theoretical ejector efficiency (%);
η_d = diffuser efficiency;
ν = kinematic viscosity (m^2/s); and
θ = suction nozzle angle (deg).

Subscripts

1,2,3,4,5 = ejector sections shown in Fig. 5;
d = diffuser;
n = driving nozzle;
s = suction chamber; and
t = throat or mixing chamber.

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