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Estimation and Modeling of Multi-Stage Ejectors in the Micro Scale

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Abstract: *Ejectors are used for the suction of secondary fluids through momentum transfer and the energy from high-speed primary jets. Ejectors operating with incompressible fluids (liquids) are known as jet pumps or eductors. In other words, the term "ejector" is used when the equipment operates with compressible fluids (gasses and vapors). These phenomena are associated with considerable thermodynamic complexities in the mixer and diffuser. Microscale multistage ejectors are able to reduce losses and increase the efficiency of those MEMs electronic devices requiring liquid pumping. In this thesis, various types of multistage ejectors were studied and then the microscale multistage ejectors were mathematically modeled and simulated with the help of computational fluid dynamics (CFD) techniques. The secondary objective of the present study was to investigate the efficiency of microscale multistage ejectors. Microscale modeling of this type of ejectors is in fact an important aspect of this thesis. The fundamental equations of the model were solved using the commercial FLUENT software package. A reasonably good agreement was observed between the numerical, experimental and analytical results.*

Keywords: *Modeling, Multi Stage ejectors, Micro*

INTRODUCTION

Microelectromechanical systems (MEMS) enable the downsizing of many of the macro-sized devices, including power generation systems, electromotive force systems and current measurement, analysis and processing systems.

These devices include the required fluid flow providing for the downsizing of the components constituting the fluid and they are very similar to macro-analog models. One important use of such a technique is in the MEMS-based Rankine cycle engines' ejector pump which is very similar to locomotive engine with its only motive segment being the outward force transfer axle.

Steam-based jet ejectors with their characteristic low traction were first used during 19th century for creating air stream inside the coal-burning steam-based locomotive engines' combustion chamber for exhausting the gases resulting from combustion through diffuser pipes (Semmens Goldfinch, 2004).

The absence of motive segments in ejectors has simplified the designers' task for designing a system capable of pumping liquids containing particles. In contrast to vacuum pumps, ejectors impose lower initial costs and are simpler to maintain. Since ejectors have no movable part, they need no repair in case that they are not subjected to corrosion. It is easy to install the ejectors and the operation control is simple, as well. One attribute of these ejectors is the mixing of the motive fluid with the processual fluid, which is a very important consideration in designing the process. During the recent years, the use of MEMS instruments has drawn a lot of attentions. Since vacuum is widely applied in micro-dimensions, the importance and necessity of designing multistage ejectors for generating the required high-output vacuum is amongst the research works needed in the current technologies. In micro dimensions, the multistage ejectors can reduce the wastage and increase the output in some electrical MEMS-based devices.

Ejector's accidental emergence in steam locomotives dates back to 1820, when the designers used to inject the vapor exhausted from the cylinder into the chimney (Semmens Goldfinch, 2004). They had noted that steam enters the vapor chamber in high pressure, undergoes expansion upon passing through the vapor nozzle, and leaves it in a very high speed. The designers enlarged the space in front of the boiler so as to embed the ejector to help the existence of the ash that might otherwise be transferred through the fire pipes into the boiler; this process was named smoke box (Semmens Goldfinch, 2004).

The operators found out that the increase in the vapor discharge velocity caused an increase in the current inn of the combustion chamber. The higher the temperature of the combustion chamber, then the more vapor was created and this process continued automatically. In steam ejectors, the driving fluid is water vapor and they essentially work similar to the other ejectors. Steam ejectors easily dislocate the currents with low pressure and high volumetric mass. The performance of these ejectors depends on the area of the driver jet as well as the speed of the jet that is a function of the supplied pressure. High pressure causes an increase in the system's performance but the system's output is reduced. Hence, due to the reduction in potential energy of the pressure in respect to the useful work by piston, there is a need for balance.

Since locomotives are operated in more than one velocity domain, the various kinds of air stream control systems were put into work and the designs took the form of multiple nozzle and multistage systems. A successful version was the double exhaust Kylchap blast-pipe that was designed by a Finnish engineer, Kylala, and a French engineer, Chapelon (Semmens Goldfinch, 2004).

The system was successfully operated in England on A4 Pacific in 1936 and it was subsequently verified in 1950 (Semmens Goldfinch, 2004).

In (Semmens Goldfinch, 2004), a succinct investigation has been presented for the historical development of the early ejector systems. The book by Professor Gus expresses the empirical works carried out on many of the locomotive components and systems (William, 1907).

It can be observed in an investigation of the articles published during the recent years that the computational fluid dynamics (CFD) has been employed as a useful tool by the researchers. The biggest advantage of numerical investigation in contrast to the laboratory research is the low cost and feasibility of extensive examinations within a short period of time. To better understand the flow inside the ejector and precisely predict the ejector's performance, many researchers have dealt with the simulation of the flow inside ejector by the assistance of CFD. Riffat et al. used this method to examine the initial position of nozzle and obtain the optimal position for it (Riffat and Omer, 2001).

Russley and others modeled several ejectors so as to investigate the flow behavior resulting from the geometrical changes of the ejector. They concluded that the maximum mass ratio could be obtained a short time before shock occurrence. They also came to the conclusion that the nozzle position was one of the important parameters in designing ejectors. Amongst the other issues that attracted the researcher's attention was the selection of the turbulence model in the numerical simulation of the flow inside the ejector (Sriveerakul, Aphornratana and Chunnanond, 2007).

Srivirakole and others investigated the process of flow mixing in a steam ejector during the refrigeration cycle in laboratory and compared the results with those obtained in the simulation of the flow based on CFD. The simulation results were consistent with the laboratory ones (Pianthong et al., 2007).

Similar results were obtained by Piantong and others, who showed that there was a difference by about 5% between the numerical results and the laboratory results of the mass ratio. They concluded that CFD could precisely predict the ejector's performance and reveal the effect of functioning conditions on the effective area that was directly associated with its performance (Pianthong et al., 2007).

Bala Morgan performed a series of experiments and numerical simulations to understand the hydrodynamic specifications of the ejector's geometry. He showed that there was an optimum ratio for the nozzle area to nozzle throat area and that the suction was in its maximum rate therein. In diverse geometries and different functioning conditions, the ratio of the sucked liquid rate was associated with the pressure difference between the water surface in the suction chamber and the throat outlet (Balamurugan, Gaikar and Patwardhan, 2008).

Srivirakole and others used CFD method to numerically investigate the flow inside the ejector and examined the effect of functional conditions and ejector's geometry based on the flow phenomena (Sriveerakul, Aphornratana and Chunnanond, 2007).

Lee et al., used computer-based simulation to investigate the reason for the low performance of a thermos-compressor and the way it could be optimized. In their study, the effects of such parameters as the position of the outflowing jet, diffuser's shape, suction mouth size and the downstream resistance were investigated on the ejector's suction current rate (Li, Wang and Day, 2010).

Analytical Design

The analytical solutions for ejectors are important in regard of achieving a preliminary design (not necessarily exact and optimum). At present, there are two methods for designing an ejector. One method is using empirical data offered by Power (1994) and the other is the application of analytical method. Kinen et al., (1950) carried out one of the first researches on ejectors. In their work, a theoretical analysis of a pneumatic ejector with a fixed mixing zone and without diffuser was conducted. Firstly, they developed a 1D flow theory based on the dynamics of the ideal gases' current using the principles of mass conservation, impulsion and energy. In this theory, the primary and secondary currents of perfect gas were taken into account and an isentropic ejector process was also considered. The outputs of isentropic process in the preliminary nozzle, mixture region and diffusion process had to be in such a way that the analytical results provided the best match with the laboratory results. The continuation of research on ejectors enabled the extensive use of fixed pressure mixing chamber and diffuser. Another analytical work on ejectors was the one by El Desuki et al., (2002) presenting a new analytical method therein.

Fluid Equations Governing the Ejectors

The numerical model of the simulated ejector was constructed using Gambit 2.4.6 software for generating the geometry and network. Moreover, Fluent 17.2 software was applied for numerical solving of the equations based on the final element method. The produced geometry of the multistage ejector has been illustrated in figure 1 along with its dimensions. All of the dimensions were in micrometers (Omidvar, Ghazikhani and Razavi, 2014).

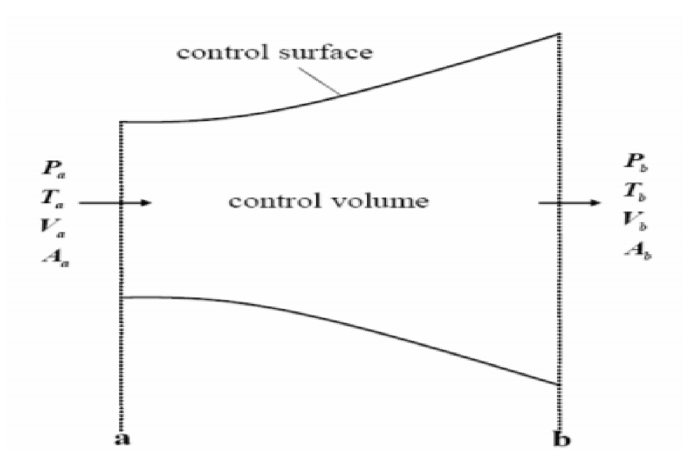

Figure 1: The geometry of the multistage ejector (William, 1907)

Figure 2 displayed a view of an ejector. In this figure, the high-pressure primary fluid (P) was accelerated in the initial nozzle and its velocity reached sonic value. In this position, gas becomes expanded and when exiting (position1), its pressure is reduced.

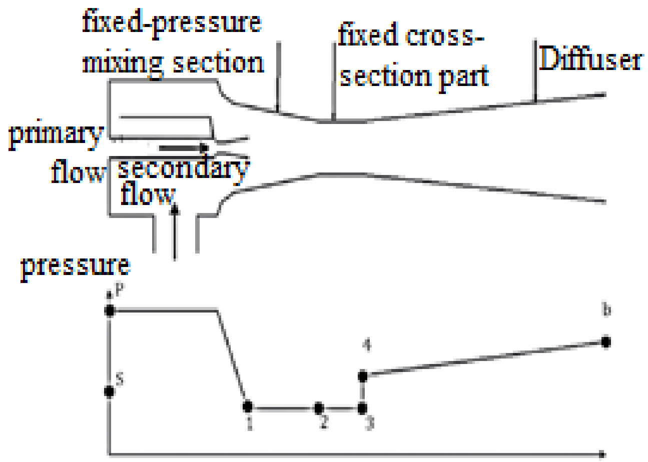

Figure 2: Schematic view of the ejector and pressure distribution therein (William, 1907)

This low pressure region sucks the secondary flow (S) into the mixing chamber. The primary flow exiting the nozzle and the sucked secondary flow are mixed with one another and the flow velocity reaches ultrasonic values in the ending part of the mixing section (position 3). In this state, a vertical shock causes the creation of condensing effects and the flow velocity is reduced to below the sonic values. This pressure increase is strengthened with the current passed through the diffuser.
Assume a control volume with a variable cross-section as demonstrated in figure (1). A given fluid with pressure Pa enters cross-section Aa and exits the cross-section Ab with the pressure Pb.
Mass conservation equation for this control volume with a variable cross-section, shown in figure (1), takes the following form:

$$m = \rho_{\mathrm{a}} V_{\mathrm{a}} A_{\mathrm{a}} = \rho_{\mathrm{b}} V_{\mathrm{b}} A_{\mathrm{b}}$$

Where, ρ is the density and V is the fluid's velocity. It is worth mentioning that the relation is expressive of the idea that the mass discharge rate should be equal in the control volume's inlet and outlet.
Since the fluid has been presumed in the present study to be non-condensable, the fluid density is fixed. Hence, the mass conservation equation takes the following form:

$$m = V_{\mathrm{a}} A_{\mathrm{a}} = V_{\mathrm{b}} A_{\mathrm{b}}$$

Regarding this control volume, the momentum conservation equation takes the following form:

$$P_{\mathrm{a}} A_{\mathrm{a}} + m_{\mathrm{a}} V_{\mathrm{a}} + \int_{A_{\mathrm{a}}}^{A_{\mathrm{b}}} P dA = P_b A_b + m_b V_b$$

Moreover, in cases that there are cross-section variations, there is a need for defining a Mach number. This number is the ratio of the fluid velocity to the local sound velocity.

$$M = \frac{V}{C}$$

Where, V is the fluid velocity and c denotes the local sound velocity that is obtained from the following relation:

$$c = \sqrt{\gamma RT}$$

In the above relation, γ is the Cp:Cv ratio, R is the global gas ratio and T is the absolute gas temperature.
The following relation also holds for an ideal gas during an isentropic process:

$$\frac{P}{\rho^{\gamma}} = constant$$

Using the above equations and considering the static pressure conditions, the following equation can be obtained for the fluid's local pressure:

$$\frac{P_0}{P} = \left(1 + \frac{\gamma - 1}{2} M^2\right)$$

The subscript 0 pertains to the properties in static state. Using this relation and considering the input pressure, the pressure can be obtained in any point of this control volume.
In the present study, the fluid was air, so γ was equal to 1.4 and the static state pressure was 1.12 bars. Mach number was also directly correlated with fluid velocity, since the velocity was changed with the change in the cross-section based on mass conservation equation; the pressure was also changed with the change in cross-section. The following diagram shows the pressure variations in respect to Mach number.

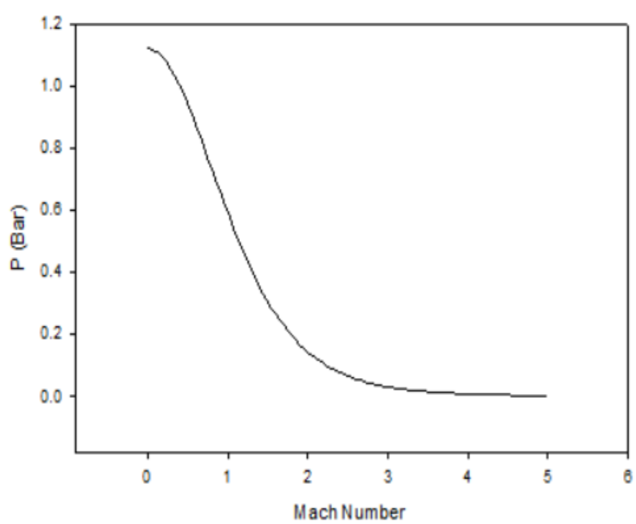

Figure 3: The diagram of pressure variations in respect to Mach number (Hanafi et al., 2015)

It is evident that the increase in the fluid velocity reduced the pressure and, since the velocity increase occurred in the convergent parts of the ejector, pressure reductions happened regions as a result of which a relative vacuum was created. This vacuum caused suction that pulled in the fluid from the inlets.

The above equations expressed the quality of the process inside an ejector. The duty of an ejector was sucking in the fluid from the suction inlets. The process occurred with the creation of a relative vacuum in the throats as a result of the velocity increase followed by the pressure decrease. The higher the rates of the velocity increase or pressure reduction, the higher the amount of suction was.

Numerical Simulation

To perform numerical simulation, Gambit 2.4.6 software set was used for producing the geometry. Moreover, commercial Fluent 17.2 software pack was utilized for numerically solving the equations based on finite element method. The generated geometry of the multistage ejector along with its dimensions have been depicted in figures 4 and 5 (all of the dimensions were in micrometers).

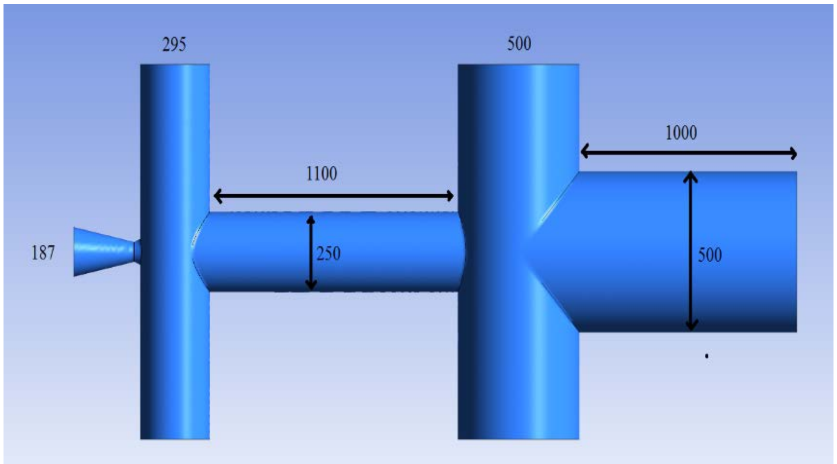

Figure 4: Simulated multistage ejector's model

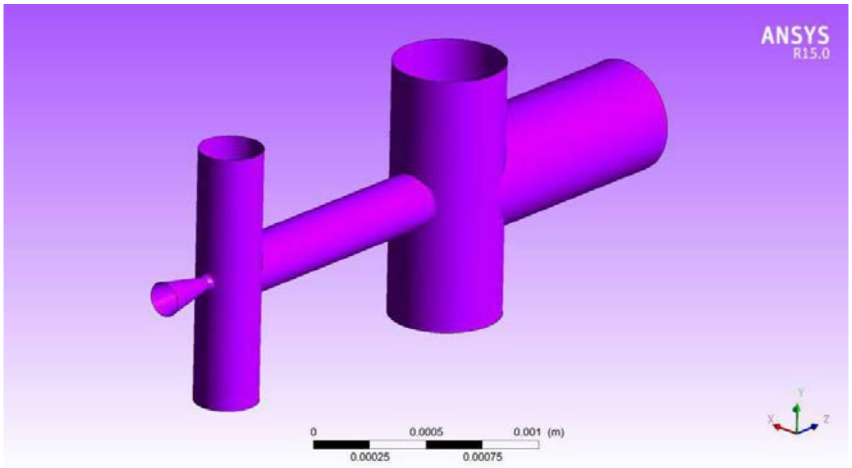

Figure 5: 3D model of the simulated multistage ejector

Simulation was done in 3D and a total of 188654 cells were used for performing the numerical calculations. In this simulation, K-ε of RNG type was employed for numerical description of the fluid's turbulent behavior inside the ejector and mass flow inlet boundary conditions and other pressure-inlet boundary conditions were taken into consideration. In addition, the air was the fluid of choice. Simulations were undertaken in steady-state and 3×10^{-5} was the convergence condition set for the continuity equation and, as it is known, it was a high standard hence the simulation results could be considered completely credible.

Table 1: Designing the geometry of ejector based on reference (Gardner, 2011)

Expansion ratio ε_m	Expansion angle α_m	Nozzle L/D	Ejector area ratio α	Mixer L/D
2.5	15°	4.3	9	4

Table 2 gives the designing of motive nozzle based on the geometry existent in (Gardner, 2011).

Table 2: Motive ejector's design

Motive Nozzle	Design throat width D_m^*	Design expansion ratio ε_m	Actual throat width D_m^*	Actual expansion ratio ε_m
Small	50µm	2.5	64µm	4.2
Medium	168µm	2.5	187µm	2.7
Large	755µm	2.5	733µm	2.2

Table 3: Comparison of the geometry designs of motive nozzles

Motive nozzle exit diameter	Design ejector area ratio	Actual ejector area ratio
129µm	9	7.6
309µm	9	9
1097µm	9	9

Using the symbols in figure 4, the total additive bubble ratio to the total suction current and the motive mass current could be obtained as demonstrated below:

$$\alpha = \frac{1}{\dot{m}_1} \sum \dot{m}_{s,n}$$

The final ratio of ejector could be defined for every stage based on $A_d = A_m + A_s$. The regional ratio of a reasonable multistage ejector could also be produced and expressed as:

$$\alpha = \frac{A_{d,2}}{A_{m,1}} \left(1 = \frac{A_{S,1}}{A_{m,1}}\right) \left(1 + \frac{A_{s,2}}{A_{m,2}}\right)$$

Validation based on Laboratory Results and Discussions:

The experiment has been simulated based on a similar test (Gardner et al., 2010; Gardner et al., 2010). When nitrogen entered in a fixed pressure into the motive nozzle, the additive bubble ratio was changed using a liquid sucker needle valve. Thus, the measurements using MKS Alta-180 showed a mass flow range between 0sccm and 20000sccm. The rate of the motive liquid's mass flow was calculated based on isentropic flow in a motive nozzle. There was observed a little difference between the prior experiment's result and the measured flow rate (Gardner et al., 2010; Bayt, 1998).

In the previous study, a single-stage ejector with a small 10µm throat diameter had been used and the effects of scale were predicted for throat and mixer's Reynolds numbers near 1300 and 400, respectively (Mehra, 1997). The other studies on 2D micro-nozzles reported the viscosity loss for Reynold numbers near 2000 (Gardner et al., 2010). Using Reynolds numbers, the throat of the motive nozzle could be calculated. Moreover, the secondary Reynolds number of the mixer could be obtained based on geometry and additive bubble ratio.

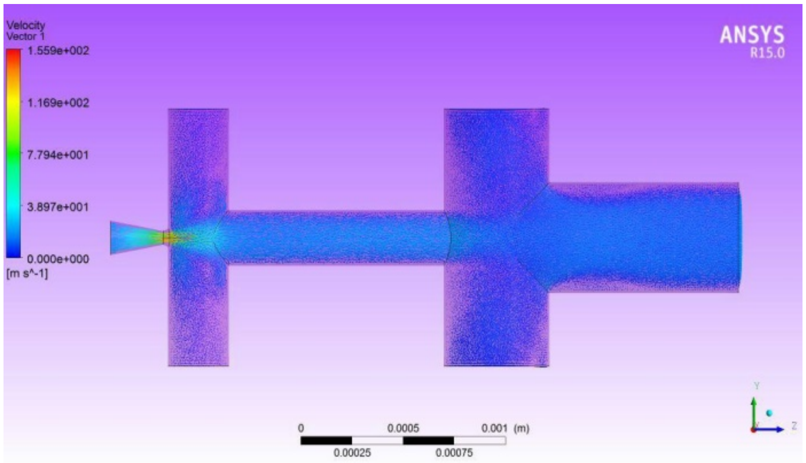

Figure 6: Velocity vector

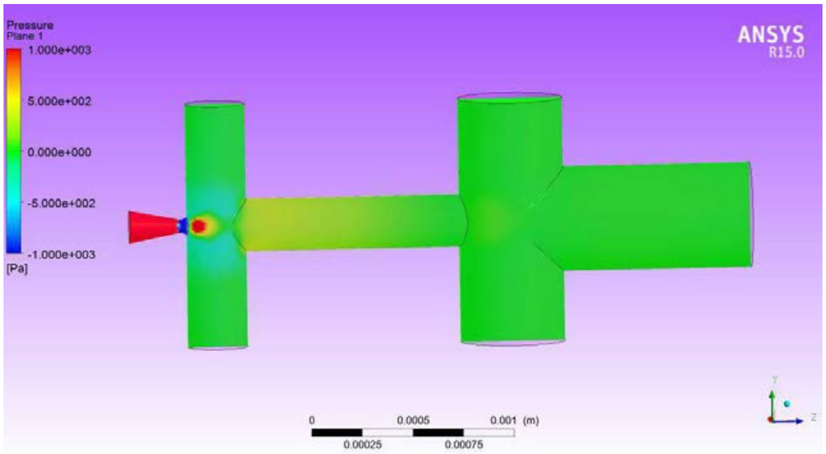

Figure 7: Pressure vector

Figure 8 presented the Cp ratio of the additive bubble in three various ejectors. The ejector with a 63μm throat did not have a suitable performance due to the nozzle's overwork. As it is observed in figure 8, the Cp ratio of the additive bubble for the ejector with 63μm throat was indicative of its failure in contrast to the ejectors with 187μm and 733μm throats. Pressure correction has been calculated using the nozzle's static output pressure. When the nozzle's static pressure was decreased and the flow turned into droplets in the vicinity thereof, the output flow ratio was decreased.

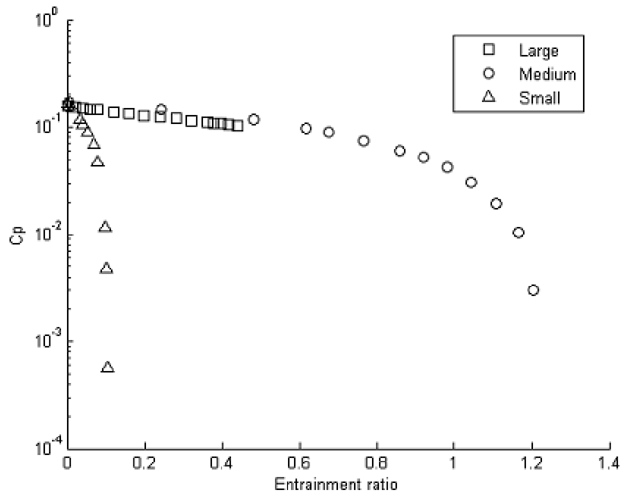

Figure 8: Cp variation rate in respect to entrainment ratio

Figure 9 portrayed the calculation of the thrust coefficients based on the additive bubble ratio in proportion to the suction pressure measurement. As it can be observed, the results of the expansion effects of the nozzle having 63µm throat have been compared in various behaviors with the nozzles having 187µm and 733µm. The design comparisons for both expansion ratios have been shown in table 2.

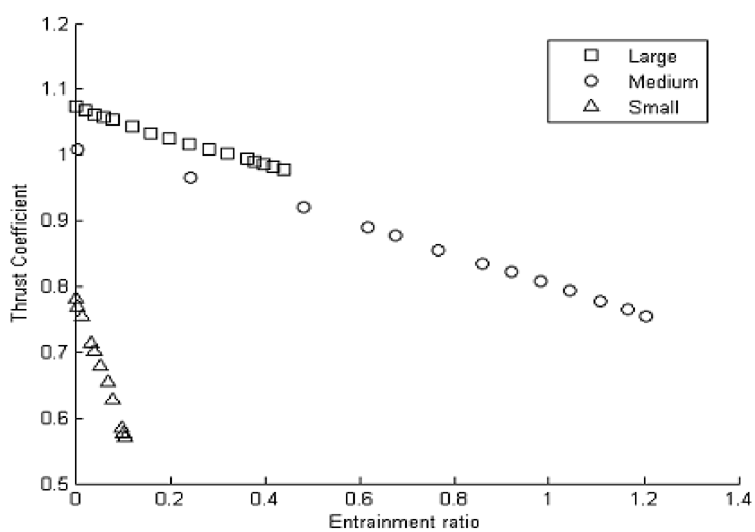

Figure 9: calculation of the isentropic thrust coefficients based on the additive bubble ratio (Mehra, 1997)

To test the ejector in this stage, $A_{s,1}/A_{m,1} = 3.5$, $A_{s,2}/A_{m,2} = 1$ have been utilized.

Comparison of the Primary Mixer with the Secondary Mixer:

As it can be understood from the name of this equipment, in two-stage ejector, suction process occurred through two paths. The effectiveness of the first and second mixers can be compared through comparing the pressure drop in the first and second throats as well as the mass discharge rates of the fluid sucked from the first and second suction inlets. Table 4 provided the values of the various operational parameters in the first and the second mixers.

Table 4: Various operational parameters of the first and second mixers

Parameter	Primary mixer	Secondary mixer
Fluid velocity in throat (m/s)	143	31.2
Fluid pressure in throat (Pa)	-845	-4.16
Velocity of the sucked fluid	4.157	1.775
Pressure of the sucked fluid (Pa)	-11.59	-2.1
Mass discharge rate of the sucked fluid (mg/s)	0.7	0.9

Figure 10 displayed the pressure variations in ejector. The velocity increase in the throats followed by pressure decrease in them caused considerable suction of fluid from outside to the inside of the ejector.

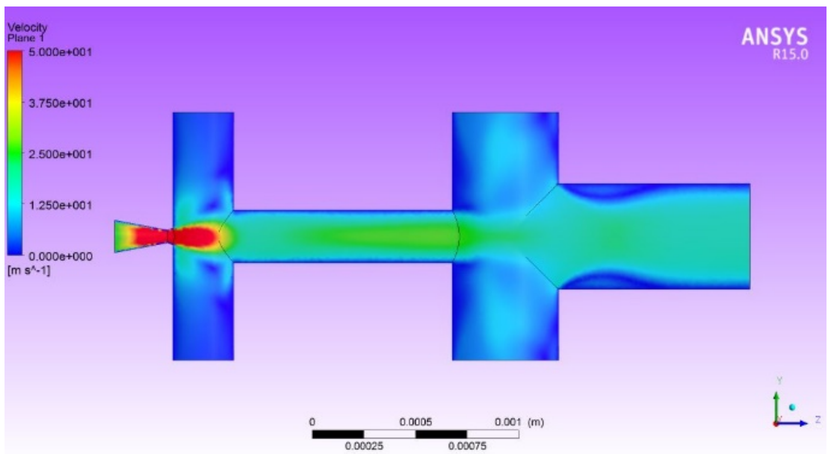

Figure 10: Pressure changes in the ejector

Moreover, figure 11 exhibits the velocity vectors indicating the fluid inflow from both suction paths.

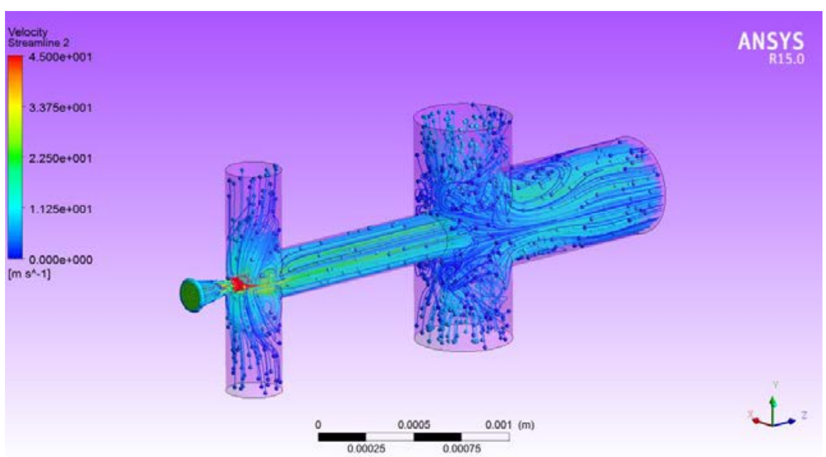

Figure 11: Velocity vectors indicating the fluid inflow from both suction paths

The prior studies on multistage ejectors (Presz and Werle, 2002) showed that the output was increased in them. Furthermore, these ejectors were provided by higher suction in contrast to single-stage ejectors in a final motive pressure and identical peripheral ratio. The same corrections have been made for both of the stages of a two-stage ejector for an identical peripheral ratio, which were usable in short mixers for a length to diameter ratio of L/D≈ 0.5 − 1. So, the conventional ejectors featured this same function in the mixer half due to 3D pressure (Waitz et al., 1997; Presz, Morin and Gousy, 1988).

The results of the two-stage ejector presented herein were not better than those of the single-stage ejector for an identical peripheral ratio and similar Reynolds number (Gardner et al., 2010), which was due to the inappropriate design for similar peripheral proportions. Different primary and secondary peripheral ratios were taken into account for the various parts of the test and this might cause an improvement in the performance as compared to single-stage ejectors.

Conclusions

According to the table of comparison between the primary and secondary mixers, it is clear that the suction rate was a lot higher in the primary mixer than the secondary mixer. So, it seems in the first glance that the amount of fluid sucked into the primary mixer was higher. However, it is worth mentioning that the amount of the fluid sucked into the secondary mixer was significant, since the level of the suction was high and even more than the primary mixture. So, the efficiency of the mixer can be increased by increasing the suction surface. According to the abovementioned materials, it can be concluded that operationalizing the ejector in two stages causes the total amount of the sucked fluid to be increased by over two times. This is reflective of the efficiency of operationalizing the ejectors in several stages. In other words, after passing through the first throat, the fluid still had the ability to create vacuum in the other throats, as well. However, if the second and/or the third throats were not added to it, this energy would be wasted with the fluid's exit of the outlet. So, the use of more stages could cause an increase in the ejector's output as long as the motive fluid did not enter the suction chamber. It can be stated that there was an optimum number of the stages for each ejector according to its geometry, velocity and pressure of the inflowing current, which had to be accurately specified so that the highest output could be attained.

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