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Experimental study of the effect of primary nozzle position on the performance of jet ejector

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ABSTRACT

In this work the effect of primary nozzle positions relative to the mixing section in an ejector on the performance of the air ejector are investigated. Air jet ejector and two size of circular nozzles are designed and constructed.. An apparatus is designed and constructed to achieve the ejector performance (enhancing the entrainment ratio). A data acquisition system is used to record different measurements such as pressure, temperature, and velocity at different locations on the test apparatus. The convergent nozzle can be moved to study the effect of nozzle positions on the ejector performance. Tests are carried out at three different positions for each nozzle. At the 1st position, the exit section of the primary nozzle is located just at the inlet plane of the mixing section of the ejector. At the 2nd position, the exit section of the primary nozzle is shifted 8.5 mm before the inlet plane of the mixing section. For the 3rd position, the exit section of the primary nozzle is shifted 17 mm before the inlet plane of the mixing section. Results show that the performance of the ejector is affected by the nozzle position, and the optimum position which achieves a maximum entrainment ratio for all nozzles is at position (1), (nearest position to mixing section of the ejector. But the entrainment ratio increases when the nozzle is placed at position (2) in case of the small nozzle. Position (3) is the worst position for all nozzle and achieves minimum entrainment ratio. Regardless the position of the primary nozzle, there is a strong relation between the nozzle exit area and the performance of the ejector, i.e., decreasing of primary nozzle exit area increases ejector entrainment ratio.

Keywords: *Ejector; Air Working Fluid; Nozzle Positions; Circular Nozzle; Entrainment Ratio.*

1. INTRODUCTION

An ejector is used in upstream processing to compress or boost the pressure of an entrained fluid. It is an alternative to a vapor recovery unit (small compressor recovering gas) in some applications. It can be less capital cost, lower operational costs, and less maintenance intensive. Air ejectors are, practically, applicable devices used to remove air following a membrane integrity test (MIT) A lot of studies on the performance of the ejector were reported

Flugel [1] produced the first recognized ejector design method. The use of ejector was wide spread, but their design was based largely on experience. **Keenan and Neumann** [2] took a step forward with their analysis of the simplest form of ejector. They calculated the performance of ejector using the one – dimensional continuity, momentum and energy equations for a constant area ejector without diffuser. Although the analysis was simplified, the results were consistent and compared well with experimental results.

Elord [3] developed the theory of mixing under constant pressure, in which the factor of entropy was further considered. He attempted to design an ejector by applying theoretical analysis. However; he provided no experimental data to corroborate the correctness of his analysis. He also defined ejector efficiency as the actual flow ratio divided by limitation flow ratio.

Kroll [4] summarized several important features of ejector design and some experiments. He made the following suggestions concerning ejector design:

1- Contracting angle of mixing section should be 20 degrees to avoid harmful shock wave.
2- Ejector must have constant area section and its ideal length is 7-9 times of its diameter.
3- Ideal diffuser angle is 4-10 degree and, ideal length of diffuser is 4-8 times the diameter of the constant-area section.

Keenan et al [5] developed one dimensional ejector theory based on the gas dynamics of an ideal gas in conjunction with the principles of mass, momentum, and energy conservation and ignored heat and friction loss. Their theory assumed a constant pressure mixing process in the mixing section. They also assumed that the two streams had the same molecular weight and specific heat ratio.

Hoggarth [6] reported that varying the position of the driving nozzle gives marked improvement in performance at off design conditions. However he didn't come with a recommendation for the ideal position of the primary nozzle.

Watanabe [7] performed experiments to determine the effect of the nozzle position and the length of the diffuser. He found that the nozzle has an –optimum position within the mixing section in ejector, and this maximizes the efficiency of an ejector. An increase in the diffuser length was shown to improve diffuser efficiency but made little improvement in the effect of the ejector efficiency. He stated that the effect of the nozzle position on ejector performance was beyond the scope of any theory at that time.

Vyas and Kar [8] also made observations regarding the effect of nozzle position on ejector performance . They reported that as the nozzle is moved further away from the entrance of the constant - area mixing section, entrainment ratio falls off. Although this is a very limited investigation, they were able to conclude that the nature of decay of centerline velocity was the same for all nozzle positions.

Henzler [9] showed that ,the losses in the nozzle increase if the distance between the jet nozzle and the mixing tube is excessive. In this case, efficiency of the diffuser can also be reduced if this distance is too small because low distance leads to the momentum transfer may possibly fail to reach completion within the mixing tube. The optimum distance depends on the contours of the entrance to the ejector.

Charles [10] ,suggest that, the more a flow is double-choked and comprised of a longer compound supersonic section, the more the NXP is deemed optimal. Moreover, the compound supersonic section length is found to vary significantly with the change in NXP.

Arun[11], presented that the variation of entrainment ratio with change in primary stream pressure. Moreover, the shock wave patterns in constant area mixing chamber and diffuser for set of inlet and outlet conditions have been visualized and presented**.**

V. Kumar[12] It is observed that the global performance parameter entrainment ratio (ω) is very sensitive to the NXP for a fixed geometry and operating condition. Results indicate that the numerical and experimental maximum entrainment ratios (i.e., 0.591 and 0.512) occur at NXP 0 (on-design) and 1 cm, respectively. The overall best performance can be achieved at on-design working conditions (i.e., NXP = 0) for variable area mixing ejector.

N**. Ruangtrakoon**, [13]. Some results show that the nozzle with the largest throat diameter obviously supplies more fluid mass flow rate, resulting in less secondary fluid being moved because of less flow area being available for the 19 secondary flow in the mixing chamber. Larger nozzle throat diameter can also lead to a higher exit pressure, so the critical condenser pressure increases.

1.1 Aim of this work

The aim of this work, study the effect of some design parameters such as nozzle positions relative to the mixing section in an ejector, and size of this primary nozzle on the performance of the air ejector. Air jet ejector and two sizes of nozzles are designed and constructed to achieve this goals.

2. EJECTOR DESIGN

The design of ejector can be developed using one –dimensional compressible flow theory. The first models were presented by **Keenan et al [5]** to analyses air ejectors. Their first model was a one-dimensional model on ideal gas dynamics in conjunction with the principles of conservation of mass, momentum and energy and friction losses were ignored. In this current study, Keenan's model is modified to include irreversibilities associated with the primary nozzle, mixing section and diffuser.
The analysis carried out in this work is based on the following assumption:

- Primary and secondary fluids have the same molecular weight and specific heats ratio.
- Secondary streams are supplied at zero velocities.
- Mixing occurs at constant pressure in the mixing section.

Figure (1) shows pressure distribution along axis of the air ejector used.
For the constant area section, assume pressure at inlet P_3 equals pressure at exit P_4, also temperature at inlet T_3 equals temperature at exit T4.

3. EJECTOR OPTIMIZATION

Ejector design based on one dimensional theory does not yield the optimum geometry in terms of diameter ratios, length of different sections or angles. The optimum design has been obtained from experimental measurements in previous work, [4].

It has been found that, the best entrance of the ejector is elliptical or conical shape, so the nozzle, mixing section and diffuser section of the ejector are chosen to be conical shape, **Henzler [9].**

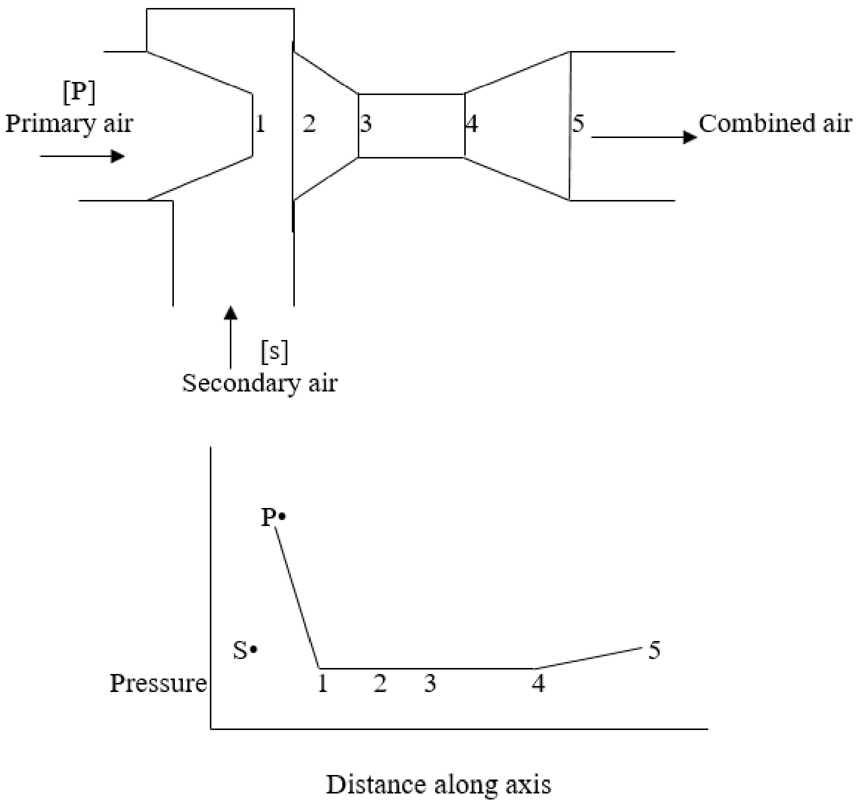


Figure (1) Schematic representation of Ejector

Nozzle cone angle for steam jet refrigeration work lies between 10 deg.and 12 deg., **ASHRAE [14].** In the present work, for air as motive fluid, the nozzle cone angle is taken 15 deg. for large circular nozzle and 18.75 deg. for small circular nozzle.

The length of mixing section of ejector must be sufficient to transfer the momentum completely at the time the air reaches the diffuser throat. This length is proportional to the primary pressure, **ASHRAE [14].** Because the primary pressure is small (1.2 to 1.8 bar gauge) in the present work, the length of mixing section is taken 1.5 times the throat of the constant area section (D). Also the optimum mixing section cone angle decreases with decreasing ejector design pressure ratio, larger angle will result in a decrease of ejector efficiency and vice versa, **ASHRAE[14],** so it is taken in present work equal to 6 deg.

The length of the constant area section, from previous works, is ranged between 3 to 5 (D), **ASHRAE [14],** in the present work it is taken 3(D).

The diffuser length has a large effect on diffuser efficiency. The optimum length of diffuser lies between 3 to 8 (D), **ASHRAE [14].** In the present work the length of the diffuser is taken 3(D). Also, it has been found that the diffuser angle lies between 5 and 12 degrees, **Henzler [9],** in present work it is taken 5 degrees. Figure (2) shows the relative dimensions of ejector.

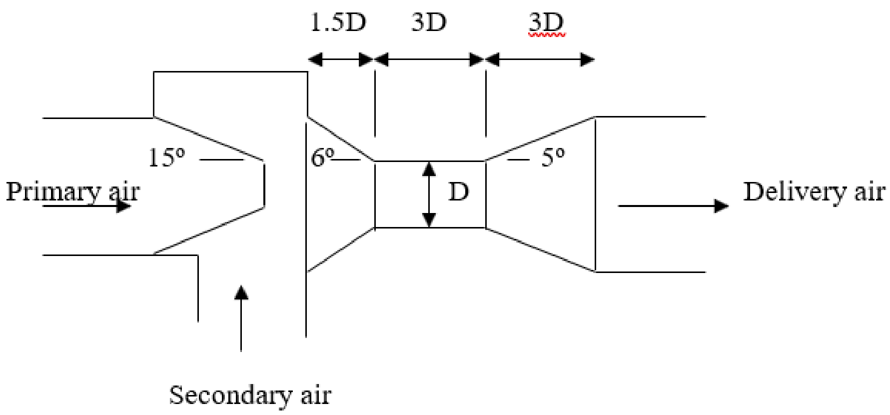


Figure (2) Ejector Optimization

The proposed air ejector, made of brass, consists of five parts; the convergent nozzle, the inlet section, the mixing section, the constant area section and the diffuser section.

Figure (3) shows the construction drawing of the ejector including all dimensions.

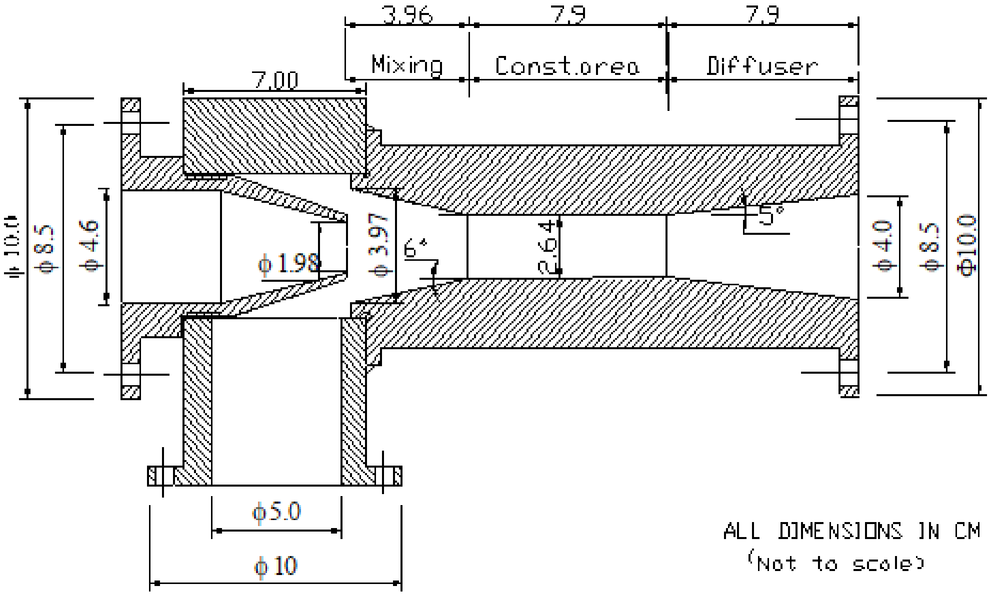


Figure (3) Ejector Dimensions

The convergent nozzle is conical in shape and 9 cm in length. The inlet diameter of the nozzle is fixed and equal 46 mm. Two circular primary nozzles were used, the first has an exit diameter of 1.98 cm and the second has an exit diameter of 1.25 cm. The convergent angle of the first nozzle is 15 deg. and that of the second nozzle is 18.75 deg. Details of the two nozzles are shown in figure (4). The inlet section of the ejector consists of three parts: one part for the converging nozzle, the second part for the suction pipe and the third part for the mixing section. The convergent nozzle is screwed into the inlet section. It is possible to change the location of the convergent nozzle inside the inlet section by unscrewing the nozzle. The length of this screwed part is 20 mm.

The suction connection and mixing section are also connected to the inlet section by welding. The mixing section, constant area section and diffuser section are machined from one piece. The mixing section is conical in shape with a converging angle of 6° and a length of 3.96 cm. The exit diameter of this section is 2.64 cm. This is followed by a constant area section, 2.64 cm in diameter also, and its length 7.9 cm.

The diffuser is also conical in shape with a divergence angle of 5° and length of it is 7.9 cm, the diffuser exit is connected to 2″ steel pipe through a flanged connection. The exit pipe diameter is further increased to 3” by using a symmetric expander 2″ /3″ fitting.

Design program is used to obtain the properties of air at different sections of nozzle and all dimensions of ejector parts. As mentioned above. The basic equations of ejector design in this work were summarized in this program, there are many assumptions and constant were used. This program run under Basic.

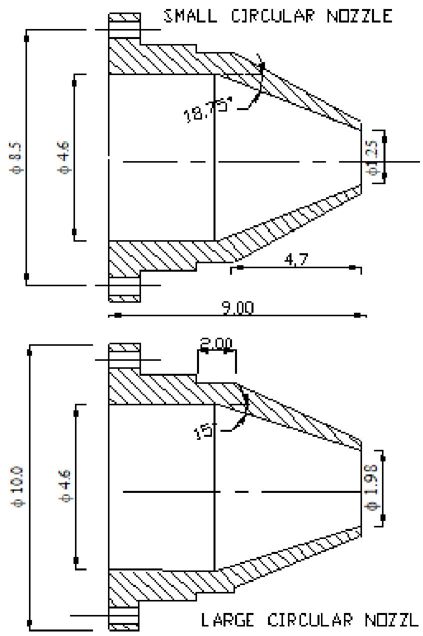


Figure (4): primary Nozzles shapes

Figure (4-a) large circular nozzle photo

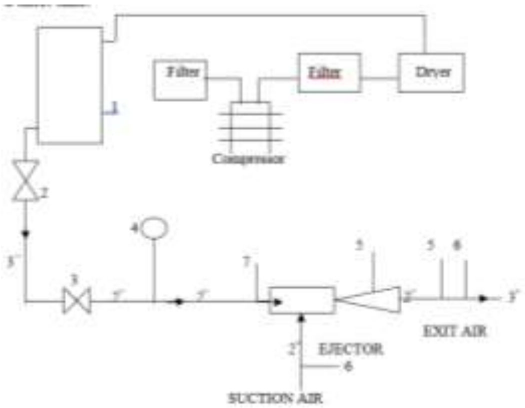
4. TEST APPARATUS

Figure (5) shows a schematic drawing of the used test apparatus which established at fluid labratory, Ain Shams Engineering Faculty, it consists of five parts: air compressor with a receiver, piping system and valves, air ejector, measuring instruments and data acquisition system. A 5 hp compressor is used for supply air to the test apparatus.

The stored air is transferred to the test apparatus by 3″/2″ galvanized steel pipes. Ball valves are used to isolate the test apparatus if necessary and to adjust the upstream pressure.

The ejector used, shown in figure (4), consists of inlet nozzle, mixing section, suction pipe, diffuser and exit pipe. The size of the inlet and exit pipes are 2″and 3″, respectively. Both pipes are made from galvanized steel. The instruments shown in figure (5) include: pressure gauges and pressure transducers, thermocouples and velocity transducers.

The power sources for the transducers were arranged by using AC/DC converters as well as AC transformers. Due to short duration in any one run of the experiment, it is not possible to read the output from measuring transducers using digital or analogue readout. Therefore, all analogue signals coming from transducers are converted to digital signals through an A/D converter which is built in a data acquisition card. The converted signals are then recorded by a personal computer. This arrangement allows the collection and recording of instruments signals in a short time.



1-AIR COMPRESSOR RESERVOIR	2- BALL VALVE AT RESERVOIR OUTLET
3-BALL VALVE AT INLET OF THE EJECTOR	4-PRESSURE GAUGE
5-THERMOCOUPLES	6-AIR VELOCITY TRANSDUCERS
7-PRESSURE TRANSDUCER	

Figure (5): Layout of the Test Apparatus

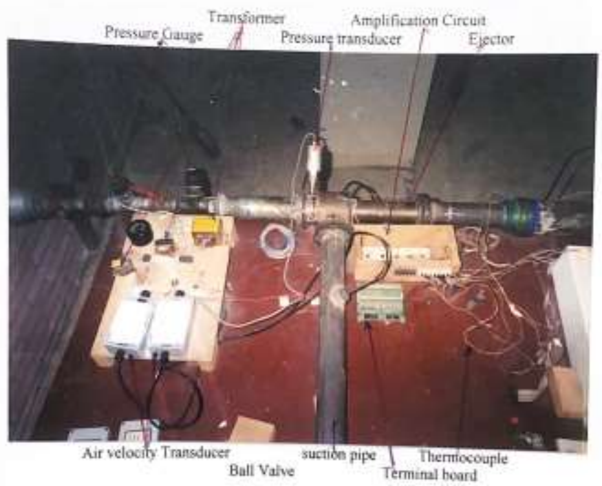


Figure (5-a) Test apparatus

Measuring Instruments

The average air velocity inside the suction and discharge pipes of the ejector were measured using calibrated two hot-wire velocity transducers have a maximum range of 50 m/s and an accuracy of 2% of reading. Thermocouples type J (Copper-Constantan) are used with a wire diameter of 0.5 mm. There are two thermocouples measuring the temperature at two points on the test rig. The position of these thermocouples are as follows; the 1st at middle point of constant area section of ejector (called Tmix), the 2nd at the delivery tube, about 30 cm from ejector exit (called Tdel.). All thermocouples have been calibrated and connected to an amplification circuit which is shown in figure (4-6) in order to become suitable to data acquisition card in which the input Volt is ranged from 0-10 Volt. The gain of amplification circuit is 1:1000 to amplify the signal of thermocouples.

5. RESULTS AND DISCUSSION

In this section, the experimental measurements obtained in this study are presented. For each test case, the measuring of static pressure, temperature and velocity at different locations are plotted versus time. The locations of measuring instruments are shown in figure (6), and the positions of nozzle are shown in figure (7).

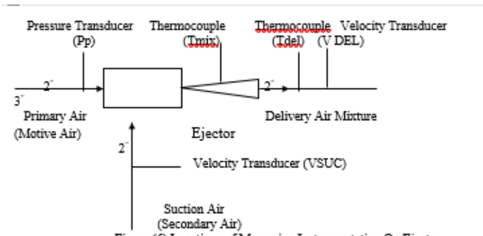


Figure (6) Locations of Measuring Instrumentation On Ejector

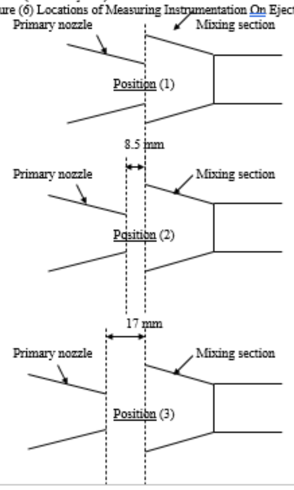


Figure (7): Positions of Nozzle

The measurements were used to calculate the entrainment ratio of the ejector for different cases. This parameter is plotted against the ejector pressure ratio. Comparison between the measured entrainment ratios for the different cases are made to show the effect of nozzle area ratio, nozzle position and nozzle shape on the ejector performance.

5.1 Temperature Measurement

Figure (8) shows the variation of the measured temperatures at two locations versus time for large circular nozzle. The first location is at middle point of the constant area section in ejector (Tmix) and the other is at delivery pipe (Tdel.). The figure shows that Tmix is lower than Tdel because the pressure of mixture air is lower than delivery air. The measured temperature at both locations changes very slowly with time. This indicates that the thermal conditions inside the ejector are nearly steady.

The behavior of the different nozzles at all positions is similar to that for large circular nozzle.

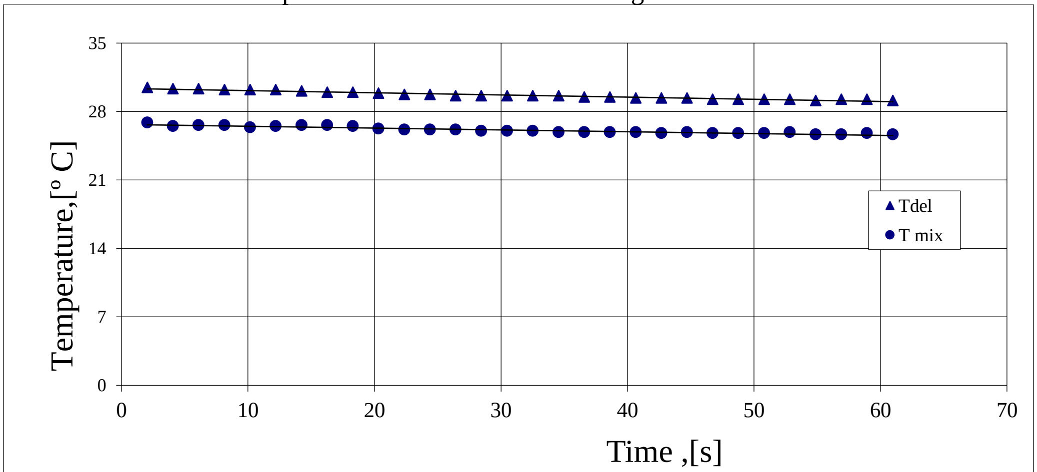


Figure (8): Variation of temperatures versus time for large…

5.2 Pressure Measurements

Figures (9) to (11) show the variation of the measured primary pressure versus time for two test cases. For all test cases, the primary pressure decreases continuously with time due to emptying of the air storage tank. The duration of the experiment was nearly 60 seconds for most cases. The initial value for any case varied from each other to ensure that the exit velocity of ejector does not exceed 50m/s which is the maximum value that can be measured by the air velocity transducer used.

Figures (9) and (11) show the variation of the measured primary pressure versus time for circular nozzle .The figures show that initial value of the primary pressure during the experiment varied between 1.65 bar absolute to 2.8 bar absolute. The behavior of the small circular nozzle at all positions is similar to that for large circular nozzle.

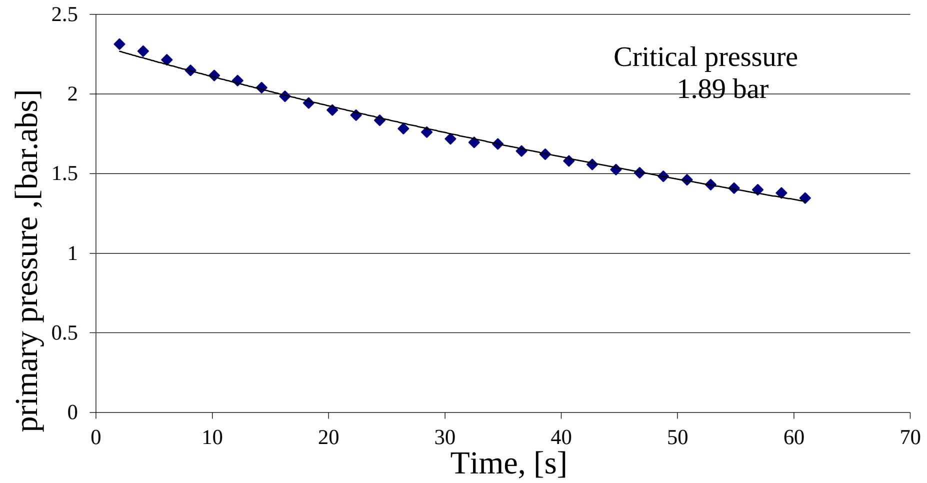
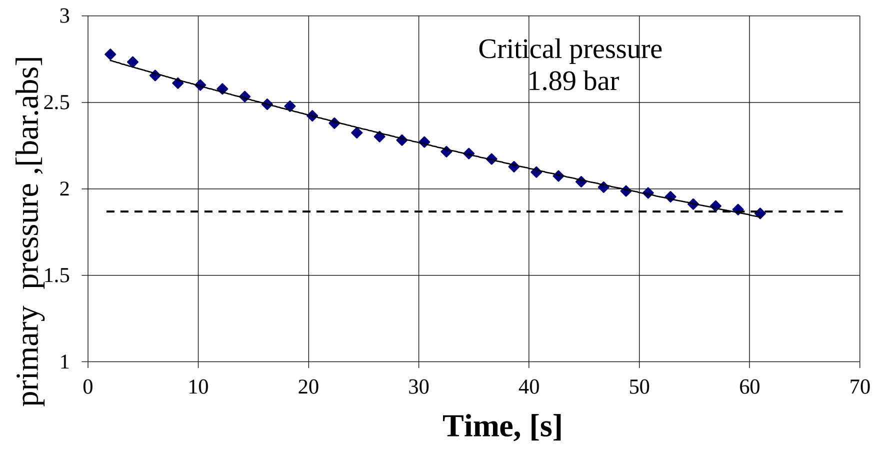
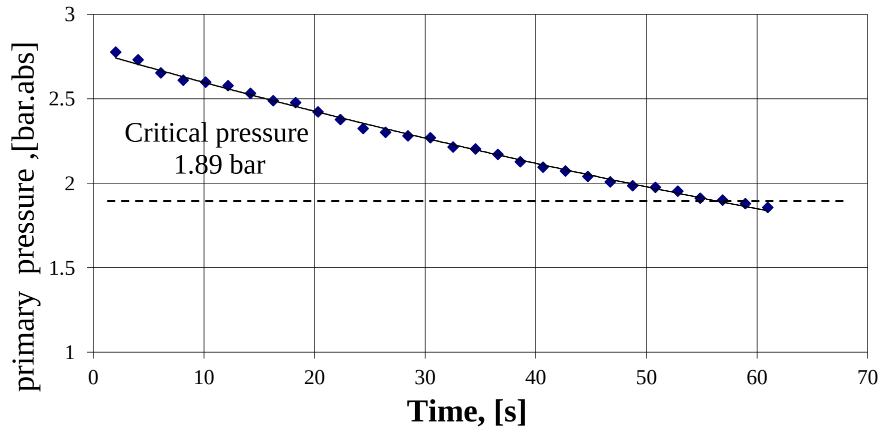


Figure (9): Variation of primary pressure versus time…



Figure(11): Variation of primary pressure versus…

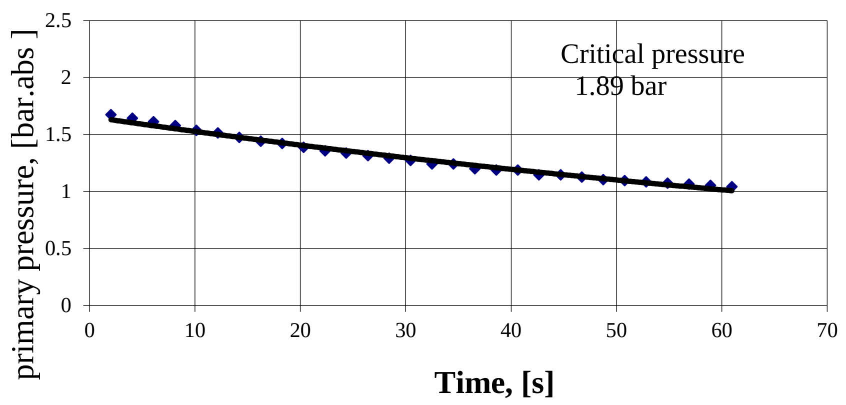


Figure(11): Variation of primary pressure versus…

5.3 Velocities Measurements

Figures (12) to (14) show the measured average air velocity variation with time inside the suction pipe (VSUC) and delivery pipe (VDEL) of the ejector for the circular ejector nozzle at different nozzle positions.

In the three cases, the velocity in the delivery pipe (VDEL) decreases with time. This is due to the fact that the primary air pressure also decreases with time during the experiments as discussed in the previous section. This leads to a continuous decrease in the primary mass flow rate and hence total mixture mass flow rate.



Figure(10): Variation of primray pressure versus time…

The suction velocity (VSUC) slightly increases with time, although it's value much lower than the delivery air velocity. This gradual increase in suction velocity indicates a slow decrease in pressure in the mixing section of the ejector below the atmospheric pressure which drives the suction air into the ejector. This pressure decrease is associated with more efficient entrainment of secondary air into the primary jet as the velocity of this jet decrease with time.

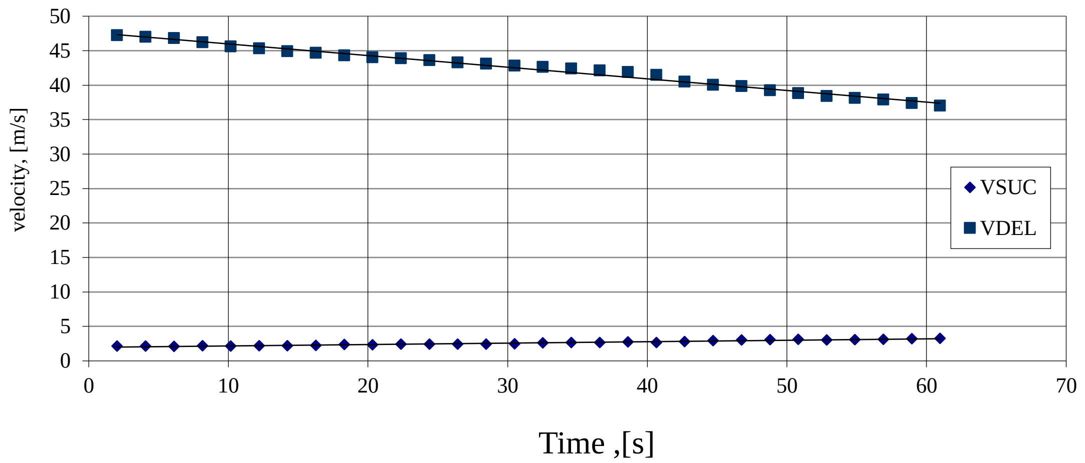


Figure (12) :Variation of suction air velocity (VSUC) and delivery…

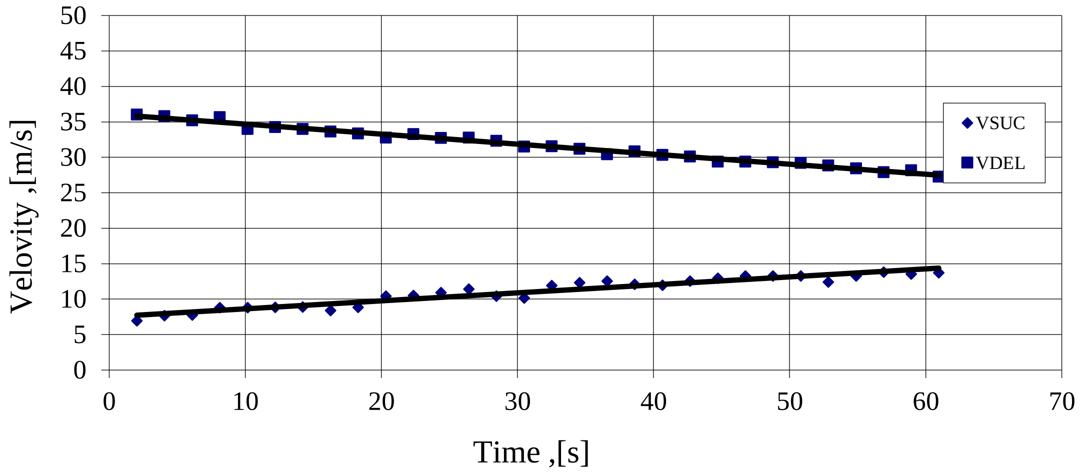


Figure (13) :Variation of suction air velocityr (VSUC) and…

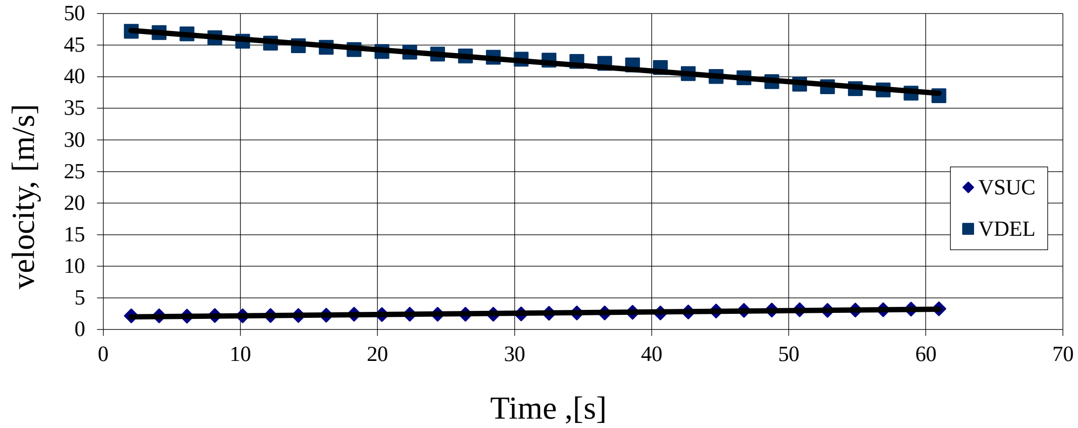


Figure (14) :Variation of suction air velocity (VSUC) and…

Figures (15) to (17) show the measured variation of suction and delivery velocities with time for ejector with small circular nozzle, again at different positions. In this case, the velocity in both suction and delivery pipes are of the same order of magnitude. For position (1), the suction pipe velocity is actually higher than the delivery pipe velocity. This clearly shows that the small nozzle leads to higher entrainment rates, compared to large nozzles.

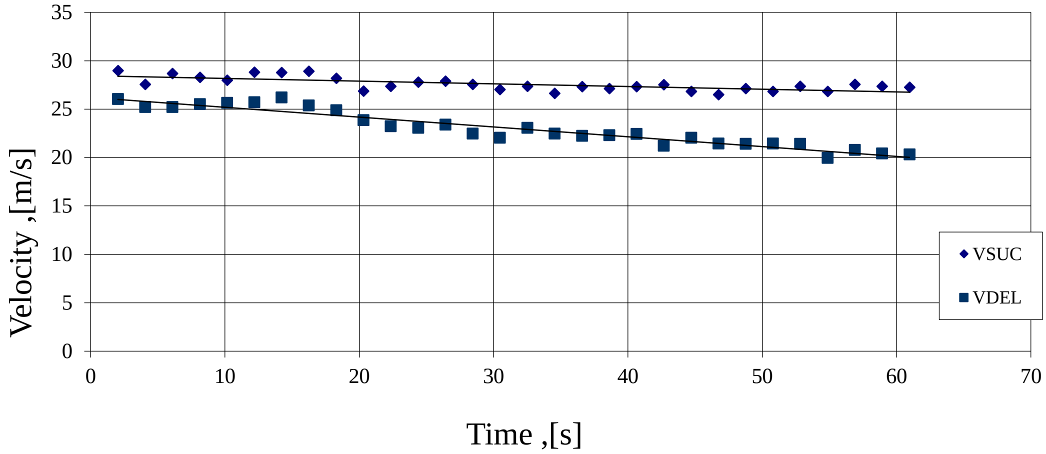
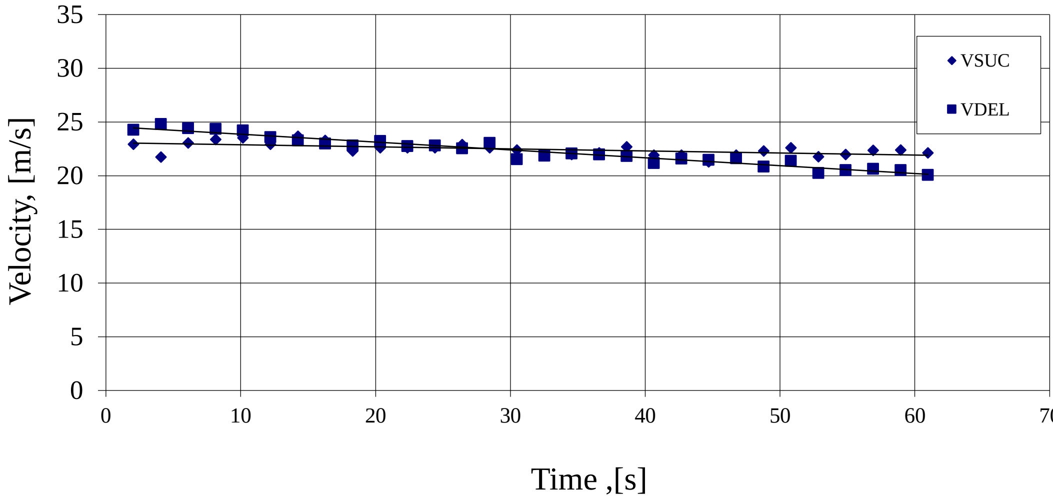
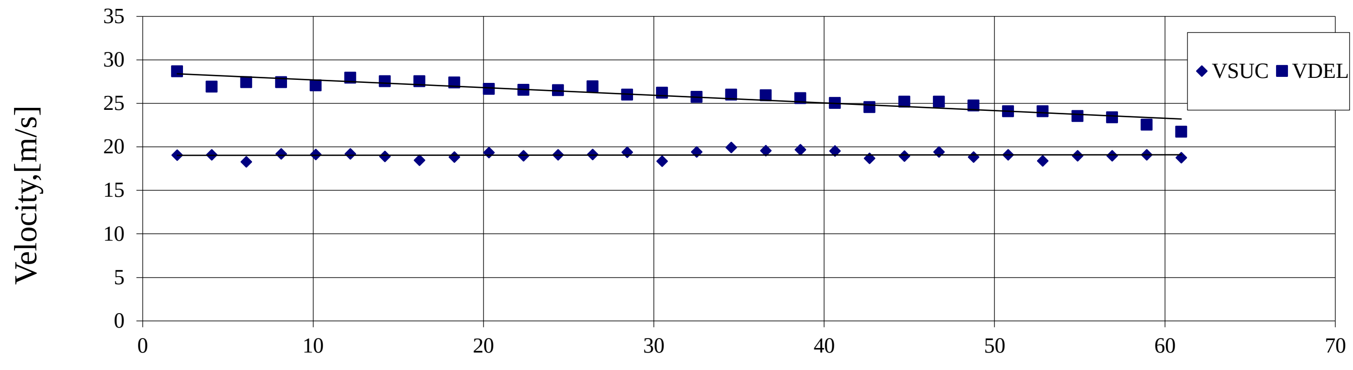


Figure (15):Variation of suction air velocity (VSUC) and delivery mixture air velocity (VDEL) versus time for small circular…



Figure(16): Variation of suction air velocity (VSUC) and delivery mixture air velocity (VDEL) versus time for small circular…



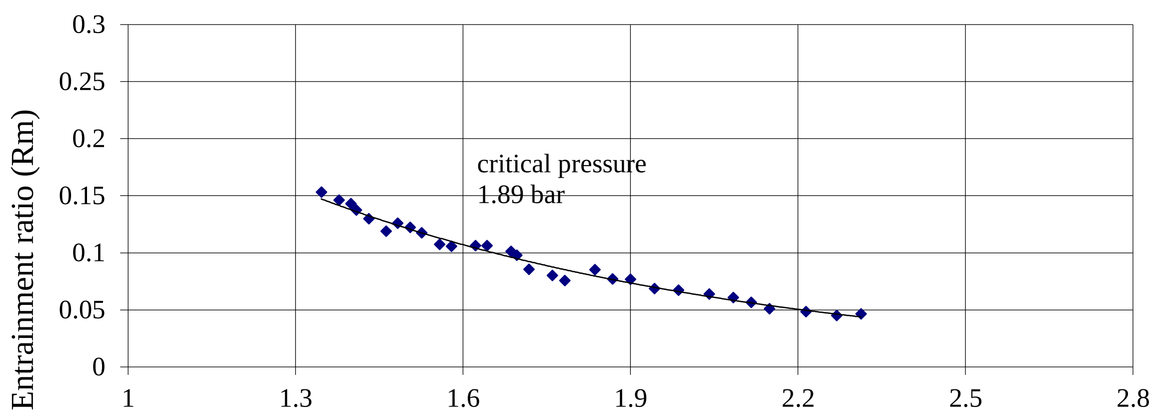
Time, [s]

Figure (17): Variation of suction air velocity (VSUC) and delivery mixture air…

5.4 Entrainment Ratio (Rm)

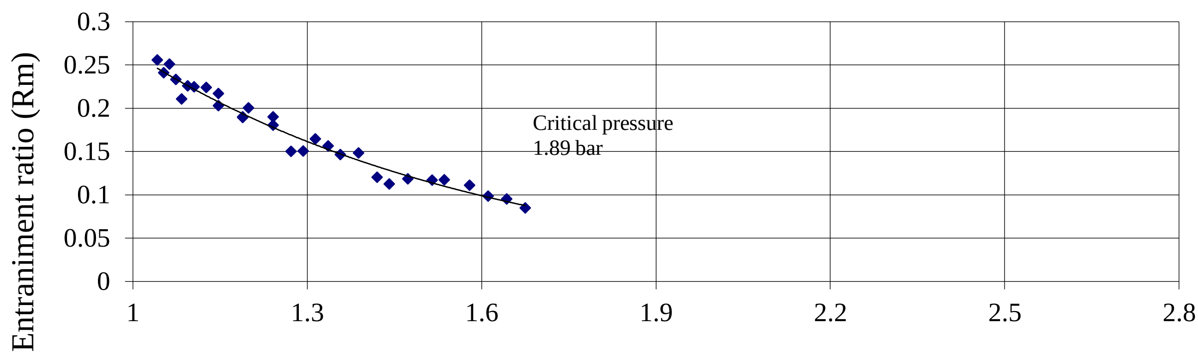
The measurements were used to calculate the entrainment ratio for all test cases. This ratio is defined as Rm=suction mass flow rate (m_s) /primary mass flow rate (m_p).

Figures (18) to (20) show the variation of the entrainment ratio (Rm) versus pressure ratio (Pr) (Pr = Primary pressure /ambient pressure) for circular nozzles at different positions. As can be seen, large nozzles produce low entrainment ratios. The entrainment ratio increases for low pressure ratios. This is because when pressure ratio decreases, the suction velocity (VSUC) increases hence (m_s) increases and delivery mixture air velocity (VDEL) decreases hence (m_p) decreases. Generally, Rm increases at low pressure ratio.



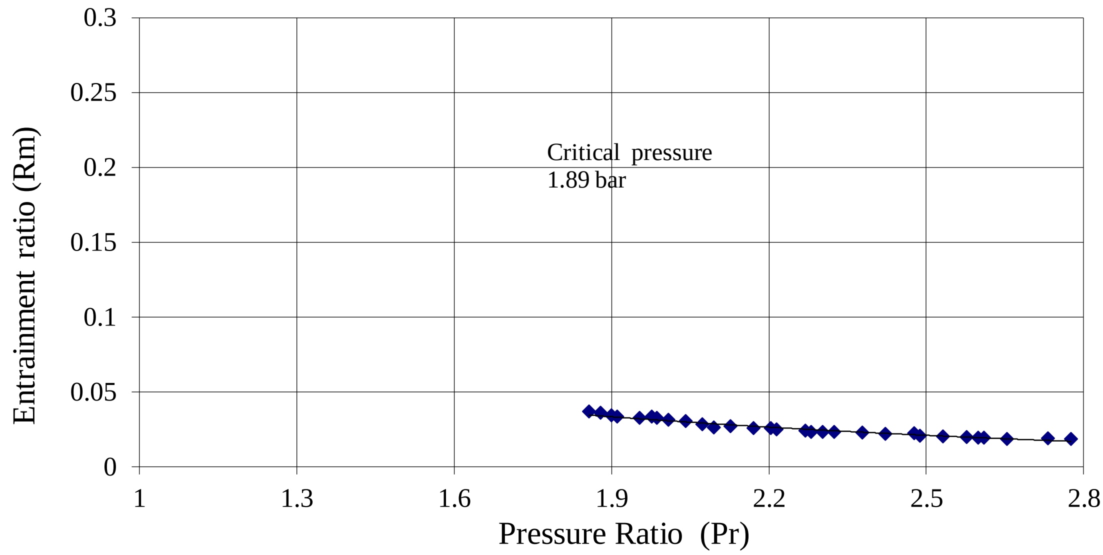
Pressure Ratio (Pr)

Figure(18) : Variation of entrainment ratio (Rm) versus Pressure Ratio (Pr)…



Pressure Ratio (Pr)

Figure(19): Variation of entrainment ratio (Rm) versus pressure ratio (Pr) for…



Figure(20): Variation of entrainment ratio (Rm) versus pressure ratio (Pr) for large circular nozzle at position (3)

The same trend is observed for small circular nozzle although much higher values of Rm are observed for small nozzles, figures (21) to (23). Constant values of Rm were observed for pressure ratios above the critical pressure ratio for small nozzles only. That is due to (VSUC) and (VDEL) were held constant at different positions for small nozzles.

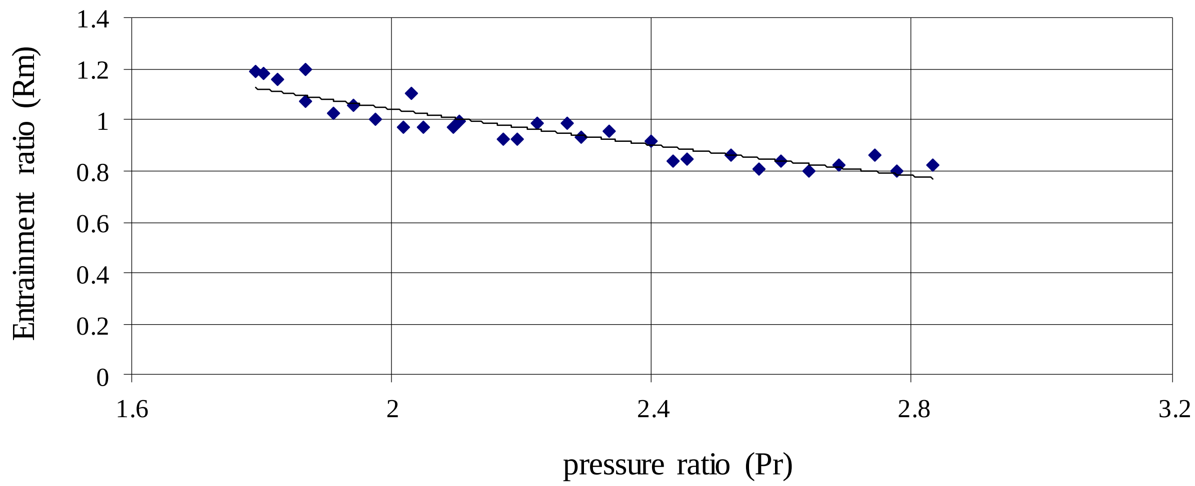


Figure (21): Variation of entrainment ratio (Rm) versus Pressure ratio (Pr) for small circular nozzle at position (1)

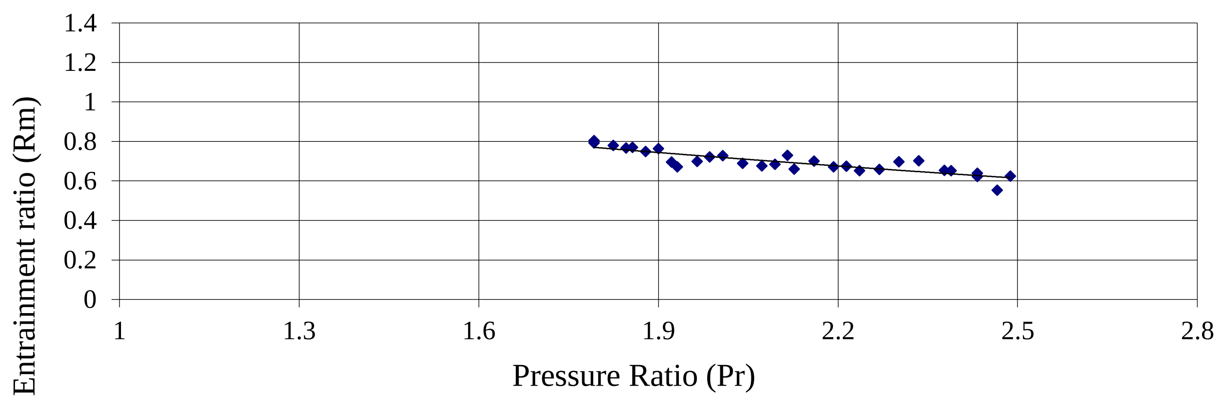


Figure (22) : Variation of entrainment ratio (Rm) versus pressure ratio for…

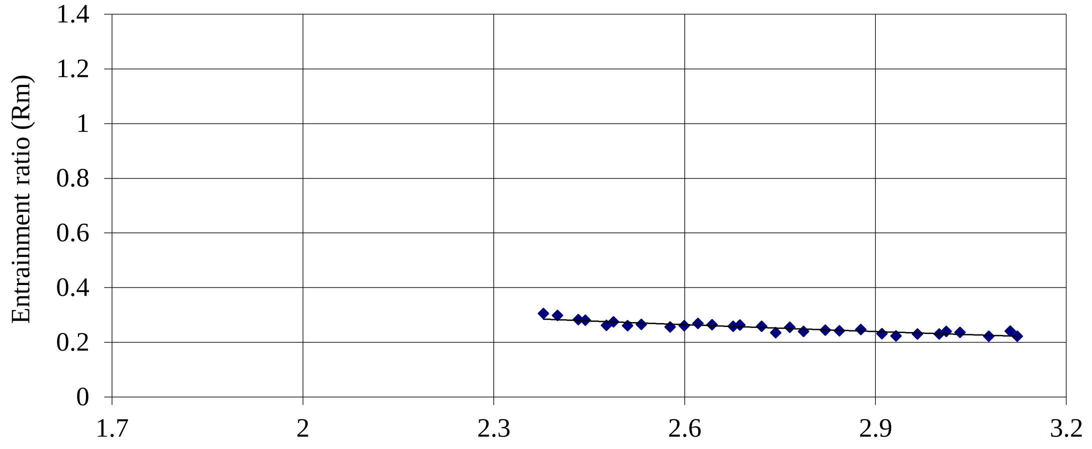


Figure (23): Variation of entrainment ratio (Rm) versus pressure ratio (Pr) for small circular nozzle at position (3)

6. DISCUSSION OF RESULTS

6.1 Nozzle position

There are several work has been carried out to state the optimum positions of nozzle such as, Watanabe [7] , Vyas and Kar [8] , Henzler [9] , Charles[10] and V. Kumar[12 . They state that the best performance of ejector when position of nozzle is at entrance of mixing section, (at certain operating conditions and ejector dimensions). Hoggarth [6] ,Watanabe [7] and V. Kumar[12] reported that, varying the position of the driving nozzle could give marked improvement in performance of ejector but it is inapplicable to generalize an optimum nozzle position for all ejectors. For the different cases that has been investigated in this work, the effect of nozzle positions on ejector performance are illustrated below.

Figures (24)) show the measured entrainment ratio (Rm) versus pressure ratio at all positions for large circular nozzle). Position (1) corresponds to a nozzle tip positioned exactly at the entrance plane of the mixing section, see figure (5.1.b). Positions (2) and (3) correspond to nozzle tip displaced 8.5 mm and 17 mm backward from the entrance plane of the mixing section. As can be seen from figure (24) the entrainment ratio measured for position (1) is the highest among the three positions. This is followed by position (2) and then position (3) which gives the lowest entrainment ratio.

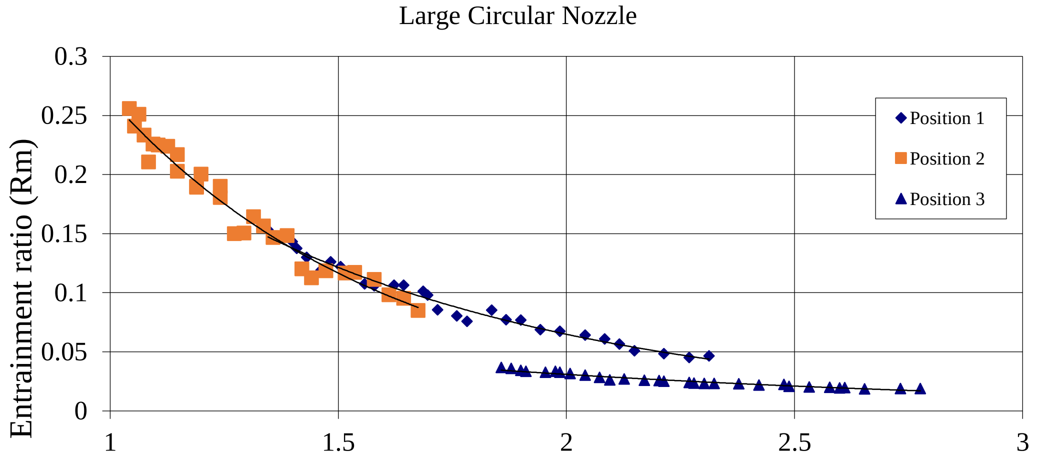


Figure (24): Variation of entrainment ratio (Rm) versus pressure ratio (Pr) for all positions of large circular nozzle (area ratio = 0.57)

For the small circular nozzle (Ar=0.22), figure (25) shows the entrainment ratio (Rm) versus pressure ratio at all positions. It is clear that position (1) is the best position. From this figure, one can note that the rate of change of entrainment ratio with pressure ratio depends strongly on the nozzle position.

6.2 Nozzle Size

To study the effect of nozzle size for different nozzles on entrainment ratio, the exit diameter of the large circular nozzle is 1.98 centimeters, and the exit diameter of the small circular nozzle is 1.25 centimeters.

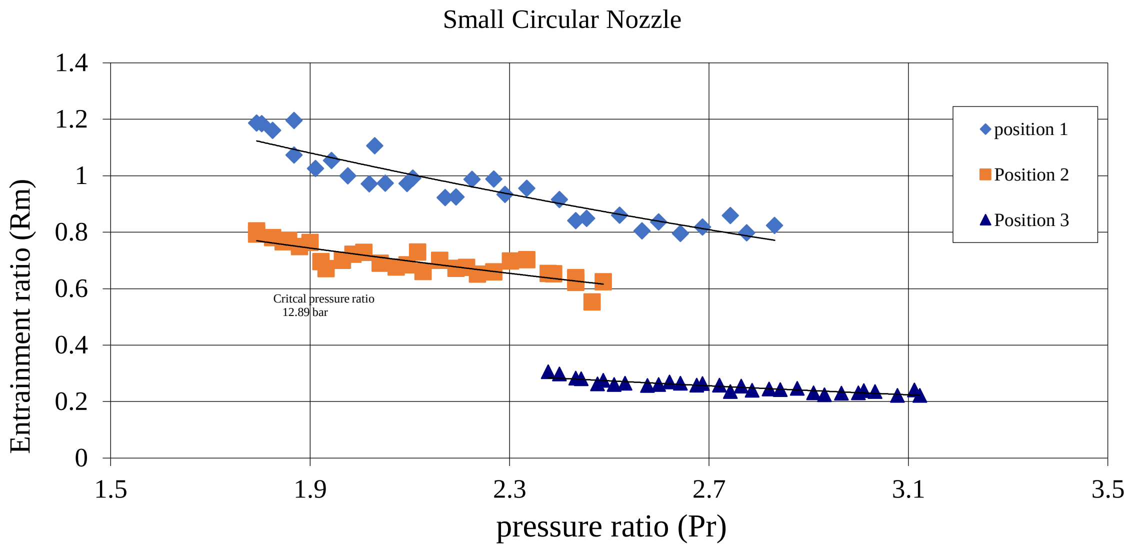


Figure (25): Variation of entrainment ratio (Rm) versus pressure ratio (Pr) for all positions of small circular nozzle (area ratio=0.22)

Figure (26) shows the effect of nozzle sizes on the entrainment ratio at all positions for large circular nozzle and small circular nozzle respectively, the area ratio (A_r) for large circular nozzle =0.57and (A_r) for small circular nozzle=0.22 There are several work has been carried out to state the primary nozzle size such a N**. Ruangtrakoon** s .[13] show that the nozzle with the largest throat diameter obviously supplies more fluid mass flow rate, resulting in less secondary fluid being moved

It is clear that the change in area ratio have a great effect on the entrainment ratio of the ejector for all positions at same operating conditions, i.e the small circular nozzle have a much better performance than large circular nozzle. This much increasing of entrainment ratio for small nozzle due to the fast decreasing of the suction pressure into mixing section of ejector which leads to higher suction mass flow rate and hence increase of entrainment ratio for small nozzle . The fast decreasing of suction pressure in case of Ar=0.22 (small size) due to the nozzle exit does not block the entrainment area (entrainment area is the area difference between inlet mixing section and exit nozzle area). This provides less resistance to entrained flow.

7. CONCLUSIONS

In this work the effect of some parameters such as nozzle positions relative to the mixing section in an ejector on the performance of the air ejector are investigated. Air jet ejector and two size of circular nozzles (large and small) are designed and constructed. The results presented above lead to the following conclusions:

1. Entrainment ratio increases by reducing the pressure ratio. Although there is no optimum pressure ratio, the highest entrainment ratio appears to occur near the critical pressure ratio in case of small circular nozzle.
2. The general optimum position for ejector convergent nozzles, is at the entrance plane of the mixing section.
3. The study showed that area ratio (Ar) (area of nozzle exit / area of constant area section) seriously effects the ejector performance. However, no optimum value can be recommended at this stage.

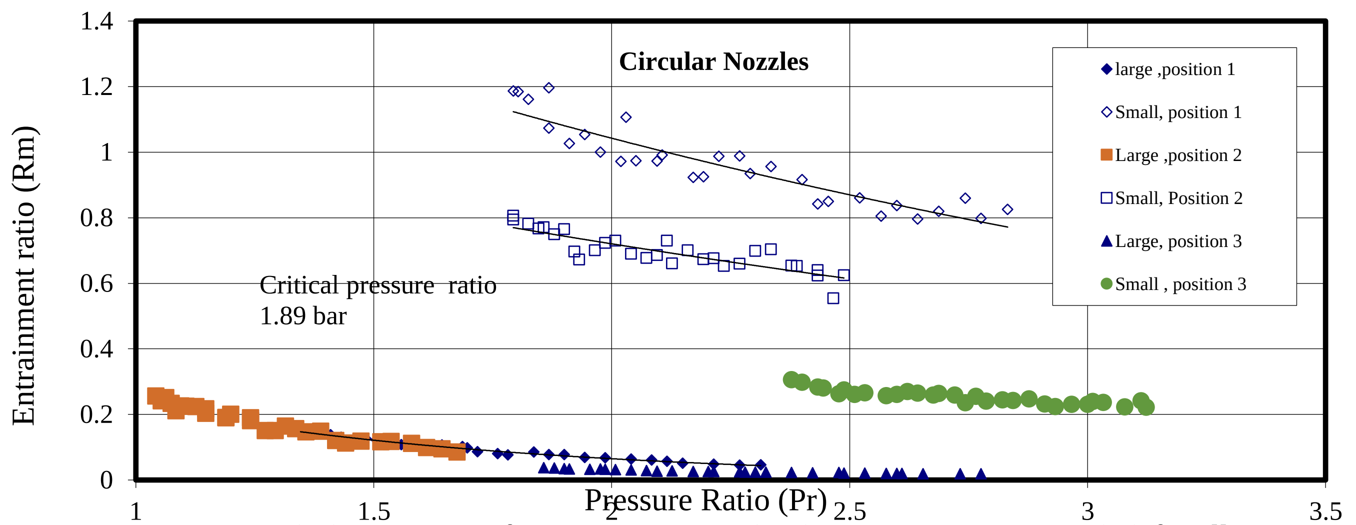


Figure (26) :Variation of entrainment ratio (Rm) versus pressure ratio (Pr) for all positions of large circular nozzle (area ratio = 0.57) and small circular nozzle…

8. LIST OF ABBREVIATIONS

A Area, m^2
A_C Area of the constant area section, m^2
A_r Area ratio (area of primary nozzle exit /A_C)
D Constant area section diameter, m
h Specific enthalpy, kJ/kg
K Specific heat ratio
L Length, m
M Mach number
m Mass flow rate, kg/s
NXP Nozzle exit position
P Pressure, bar
Pr Pressure ratio (primary pressure/ambient pressure)
Rm Entrainment ratio
t Time, s
T Temperature, K
V Velocity, m/s
V_{DEL} Delivery velocity of air mixture, m/s
V_{SUC} Suction velocity of air, m/s

Greek letters

θ Angle, degree
η_d Diffuser efficiency
ρ Density, kg/m^3
η_m Mixing section efficiency
η_n Nozzle efficiency

Subscripts

Primary nozzle exit plane
Upstream of the mixing section
Upstream of constant area section
Upstream of diffuser section
Downstream of subsonic diffuser
p Primary fluid
s Secondary fluid
i Nozzle inlet
e Nozzle exit

Superscripts

/ Primary fluid at nozzle exit
// Secondary fluid at nozzle exit

9. DECLARATION

The work included in this thesis is carried out by the author, at laboratories of Mechanical Power Engineering Department, Ain Shams University.
No part of this work has been submitted for degree or qualification at any other university.

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