AIR EJECTOR WITH A DIFFUSER THAT INCLUDES BOUNDARY LAYER SUCTION

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The article deals with axial-symmetric subsonic air-to-air ejector with a diffuser adapted for boundary layer suction. The diffuser, which is placed after the mixing chamber of the ejector, has a high divergence angle and is therefore inefficient. To increase efficiency, the diffuser is equipped with a slot enabling boundary layer suction. The effect of boundary layer suction on the airflow in the ejector was measured, as were the static pressure distribution on the mixing chamber wall and ejector characteristics. Both diffuser and ejector efficiency were calculated. Although the efficiency of the diffuser was increased, the efficiency of the ejector itself remained low.

Keywords : *air ejector, diffuser, boundary layer suction*

1. Introduction

Diffusers often play an essential role in many applications, and so a lot of research has been carried out into diffuser design, as summarized by Japikse and Baines in their work [1]. The efficiency of diffusers with high enlargement can be improved by boundary layer suction. For example, Furuya et al. published a detailed quantitative investigation [2] of a simple conical diffuser with an inlet suction similar to the one shown in Fig. 1. They found that the efficiency of the diffuser could be improved substantially, especially at large divergence angles using fairly modest suction levels of 2–5 %. By experimentation and detailed measurement, it was found that the optimum rate of suction corresponded roughly to the condition where the initial boundary layer thickness was decreased to zero by suction through a single slit. The results for a diffuser with a divergence angle of 40°, inlet diameter of 80 mm and enlargement ratio of 3.52 are shown in Fig. 10.

Boundary layer suction is also applied when using a Griffith diffuser, where the suction causes a sudden deceleration in the fluid near the wall to a low velocity which is maintained constant through the diverging section. Authors Yang and Nelson [3] measured the diffuser efficiency, after correcting for suction flow, to be in the range of 90–95 % for conical and annular diffusers.

Another approach was used by Rockwell [4], who used perforated walls for boundary layer suction, but the results were less impressive. By contrast to the techniques described above, the suction rates were quite high and the flow stability was limited.

If the diffuser is a part of an ejector, then boundary layer suction can be achieved by the ejector itself. For example, Anderson, [5], used this arrangement while investigating a supersonic ejector, but boundary layer suction did not bring any improvement. At first,

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a diffuser with a low divergence angle of 6° was used. Secondly, the sucked fluid was returned into the suction chamber in front of the ejector. Thus, the energy obtained by the fluid in the mixing chamber before suction, was dissipated.

2. Methods

Based on the knowledge obtained in [2] by Furuya et al and [5] by Anderson, a diffuser was designed with enlargement angle of 40° fitted with an adjustable slot for boundary layer suction, as shown in Fig. 1. The diffuser was manufactured by spinning from nylon. As was proved by Anderson [5], the sucked fluid must be brought back into the mixing chamber and accelerated in the direction of the main flow. Firstly, to use the energy of the sucked gas, and secondly, to enhance the mixing process when it starts. The problem lies in a thorough design of such system, because there are several unknown construction parameters. In the case studied here, a system using 4 nozzles with diameter of 5 mm inclined at an angle of 15° to the ejector axis was chosen. This inlet part of the mixing chamber including the recovery nozzles were manufactured by rapid prototyping and its dimensions can be seen in Fig. 1.

The experimental arrangement is shown and described in Fig. 2. We used a primary nozzle with diameter of 19.2 mm and a mixing chamber of diameter 40 mm, i.e. the inlet area ratio of nozzles is $\mu = A_1/A_2 = 0.3$. The length of the mixing chamber was 9D = 360 mm, the diffuser had a divergence angle of 40° and an enlargement ratio given as $\mu_D = A_4/A_3 = 3.15$. Three mass flow rates were measured: The primary mass flow rate m1 was measured with a nozzle, the mass flow rate after the ejector m_4 was measured by an orifice and finally the suction mass flow rate $m_3 - m_4$ was measured by a velocity probe situated in the suction tube and calibrated by a rotameter, see Fig. 2. The primary airflow was supplied by a compressor at pressure of $p_{01} - p_{02} = 1$ kPa, while the secondary airflow was sucked straight from the laboratory, hence the secondary stagnation pressure was equal to atmospheric pressure. A Druck LP 1000 pressure sensor with range 100, 500, 1000 and 2000 Pa was used to measure the pressure. These low pressure sensors with high accuracy 0.25 % are slow, so only the mean pressures were measured.

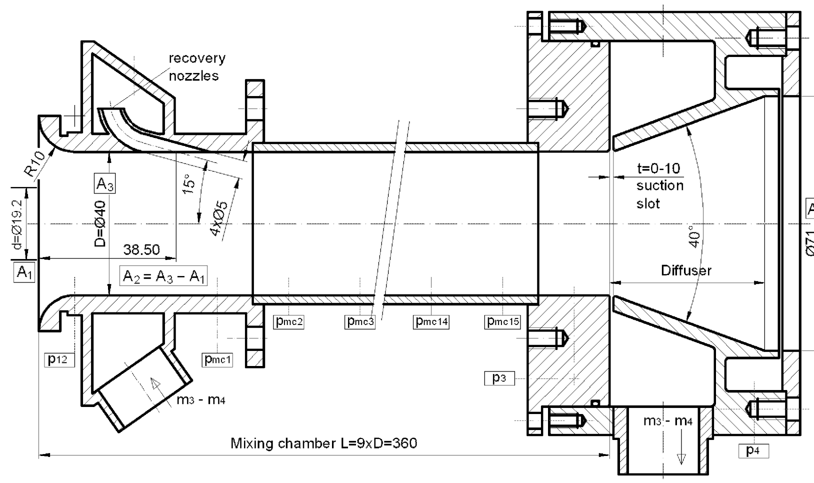

Fig. 1: Dimensions of ejector parts and positions of static pressure taps of the experimental air ejector with diffuser with an adjustable suction slot for boundary layer suction in the inlet of the diffuser

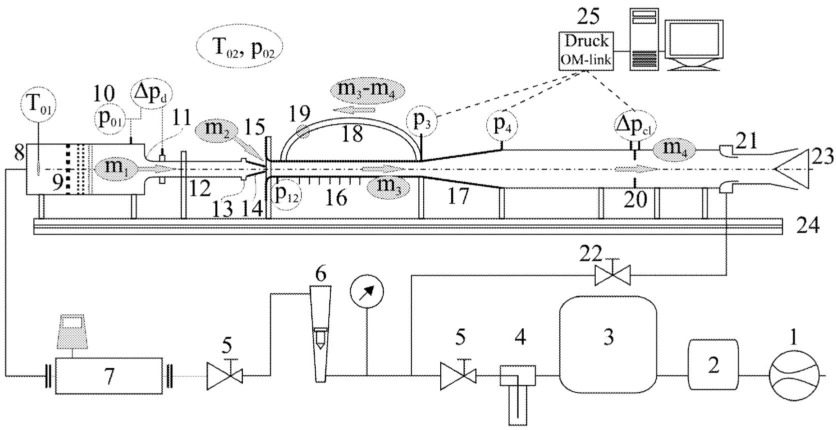

Fig.2: Experimental arrangement: 1 – compressor, 2 – air dryer, 3 – tank, 4 – filter, 5 – pressure regulator, 6 – rotameter, 7 – Coriolis mass flow meter, 8 – stilling chamber, 9 – stilling riddles, 10 – measuring of primary stagnation pressure p_{01}, 11 – measuring of primary mass flow rate, 12 – primary flow supply tube, 13 – holder of primary nozzle, 14 – primary nozzle, 15 – secondary nozzle, 16 – mixing chamber with static pressure taps, 17 – diffuser with suction slot, 18 – suction tube, 19 – velocity probe, 20 – measuring of total mass flow rate, 21 – suction ejector, 22 – control valve, 23 – chocking, 24 – base, 25 – measuring instrument

To calculate ejector efficiency, the following equation was used

$$\eta = \frac{m_2}{m_1} \, \frac{\left(\dfrac{p_4}{p_{02}}\right)^{\frac{\kappa-1}{\kappa}} - 1}{1 - \left(\dfrac{p_4}{p_{01}}\right)^{\frac{\kappa-1}{\kappa}}} \, \frac{T_{02}}{T_{01}} \approx \frac{m_2}{m_1} \, \frac{p_4 - p_{02}}{p_{01} - p_4} \;, \tag{1}$$

where T_0 is stagnation temperature, p_0 stagnation pressure, p static pressure, m mass flow rate, κ ratio of specific heats, 1 denotes conditions of primary air, 2 of the secondary air and 4 behind the diffuser, while p_4 is the back pressure. The first term in equation (1) is for compressible fluid and the second one for incompressible. Both terms can be used, because the Mach number is lower than 0.15 and the stagnation temperatures are equal: $T_{01} = T_{02}$. As can be seen from equation (1), the kinetic energy after the ejector is considered as dissipated. Similarly, these two equations were used to derive the static pressure recovery coefficient of the diffuser

$$C_\mathrm{p} = \frac{m_4}{m_3} \, \frac{1 - \left(\dfrac{p_3}{p_4}\right)^{\frac{\kappa-1}{\kappa}}}{1 - \left(\dfrac{p_3}{p_{03}}\right)^{\frac{\kappa-1}{\kappa}}} \, \frac{T_4}{T_{03}} \approx \frac{m_4}{m_3} \, \frac{p_4 - p_3}{p_{03} - p_3} \;, \tag{2}$$

where 3 denotes the conditions after the mixing chamber and 4 after the diffuser. The difference between m_3 and m_4 is the mass flow rate through the suction slot. The theoretical maximum static pressure recovery coefficient without suction is derived from the enlargement of the diffuser as

$$C_{\mathrm{p\,ideal}} = 1 - \frac{1}{\mu_\mathrm{D}^2} = 1 - \left(\frac{A_3}{A_4}\right)^2 \;. \tag{3}$$

In this case it is 0.899. The diffuser efficiency is simply the ratio between the actual recovery and the ideal recovery

$$\eta_{\mathrm{D}} = \frac{C_{\mathrm{p}}}{C_{\mathrm{p\,ideal}}} \; . \tag{4}$$

The problem in computing the actual pressure recovery is to evaluate properly the dynamic pressure in the diffuser inlet p_{d3}. We calculated it simply from the mass flow rate by using this equation

$$p_{\mathrm{d3}} = \frac{\varrho_3}{2} \left(\frac{4\, m_3}{\pi\, D^2\, \varrho_3} \right)^2 \; , \tag{5}$$

where D is the diameter of the mixing chamber.

3. Results and discussion

The results of this experimental investigation of the ejector performance are given in Fig. 3. It shows the ejector efficiency for various settings of the suction slot and a comparison with the results obtained on the same ejector with a 6° diffuser [6]. Ejector efficiency is plotted as a function of the ejection ratio, i.e. a ratio of mass flow rates m_2/m_1. The ejector with a 40° diffuser without suction through a slot of width 0 mm had the lowest efficiency. Ejector efficiency was increased by applying boundary layer suction through the slot. Efficiencies did not vary significantly for various settings of the suction slot. The highest efficiency was found using the narrowest slot of width 0.5 mm and the efficiency decreased as the opening of the suction slot was widened. The results of pneumatic measurements of the expansion pressure p_{12} are in Fig. 4. It can be seen that the slot opening and design of the diffusers have no influence on pressure p_{12}. The expansion pressure p_{12} is measured in the beginning of the mixing chamber and is fully determined by the ejection ratio. Neither boundary layer suction nor the return of the fluid into the mixing chamber after the point where the expansion pressure is measured influences it.

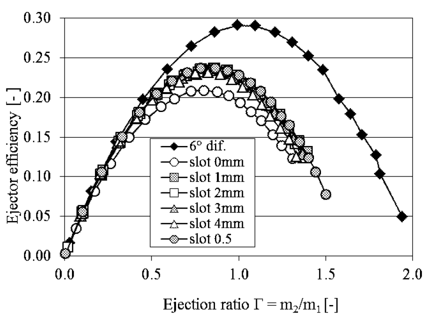

Fig.3: Ejector efficiency for different settings of the suction slot and a comparison with ejector with 6° diffuser [6]

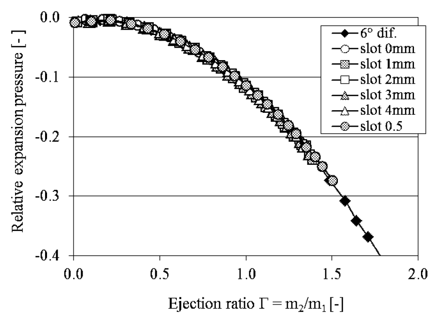

Fig.4: Relative expansion pressure $(p_{12} - p_{02})/$ $/(p_{01} - p_{02})$ in the beginning of the mixing chamber for various settings of suction

The mixing pressure p_3 measured at the end of the mixing chamber, i.e. in the diffuser inlet upstream from the suction slot, is shown in Fig. 5. It shows that the mixing pressure was only slightly affected by boundary layer suction, no bigger differences are obvious from the graph. When suction was applied, the pressure p_3 in the mixing chamber decreased and it was lower for the same ejection ratio. It is most likely caused by the additional fluid which

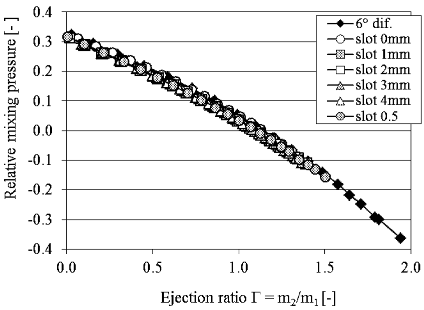

Fig.5: Relative mixing pressure $(p_3-p_{02})/(p_{01}-p_{02})$ at the end of the mixing chamber for various settings of the suction slot

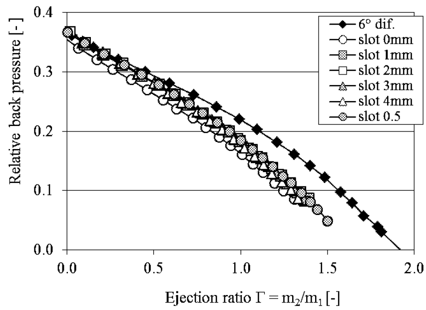

Fig.6: Relative back pressure $(p_4-p_{02})/(p_{01}-p_{02})$ after the diffuser for various settings of the suction slot

flowed through the mixing chamber and thus the dynamic pressure p_{d3} was increased and the static pressure p_3 decreased.

The measurements of the back pressure in Fig. 6 show that the main differences in flow are in the diffusers. The highest back pressures were measured when the diffuser was used with a divergence angle of 6° and the lowest with a 40° diffuser without suction. According to the relation (1), the resultant back pressure is crucial for ejector efficiency for a given ejection ratio. For an ejection ratio lower than 0.4, the diffuser combined with suction had a comparable performance to a 6° diffuser.

The efficiency of the diffuser itself is plotted in Fig. 7 and 8. The diffuser efficiency is derived from the back pressure measured in the exit section of the diffuser as shown in Fig. 7. It follows from the results that the efficiency of the diffuser with a divergence angle of 40° remains low compared to a 6° diffuser even when boundary layer suction is applied. The problem lies in the correct evaluation of the back pressure p_4. It was found that the process of flow deceleration in the diffuser was not finished before the diffuser exit, but continued further. To calculate the backpressure correctly, it was measured 300 mm (4.2 tube diameters) past the diffuser exit. This distance was sufficient to ensure complete deceleration of the airflow due to cross section enlargement in the diffuser, while the pressure loss due to friction was negligible. The resulting diffuser efficiency is shown in Fig. 8. The average efficiency of a 6° diffuser increased from 0.87 to 0.92 and the efficiency of a 40° diffuser without suction increased from 0.29 to 0.61. Also the efficiency of a diffuser with boundary layer suction increased significantly.

The suction ratio, which is defined as a ratio of mass flow rates $(m_3-m_4)/m_3$, is plotted in Fig. 9 as a function of ejection ratio. Suction mass flow rates are mostly determined by the pressure difference p_3-p_{12}, i.e. the pressure difference between the suction slot and the recovery nozzles, while the width of the slot opening is not so important. This pressure difference decreases with higher ejection ratios, but this is not the only influencing factor. It can be seen that diffuser efficiency is high for high pressure differences, high suction ratios, low ejection ratios and also fast mixing.

The resulting diffuser efficiency as a function of suction ratio is plotted in Fig. 10. There is also a comparison with the results published by Furuya et al. [2] for 40° diffuser. It follows from the results in Fig. 10 that: the influence of boundary layer suction on diffuser efficiency is significant. The influence of the opening of the suction slot on the suction mass flow rate is insignificant. It seems that the optimum slot width was 0.5 mm. When the slot opening

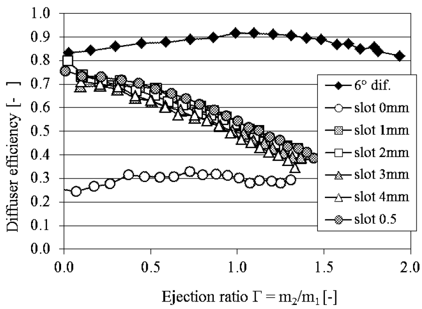

Fig.7: Diffuser efficiency for backpressure measured at the exit of the diffuser

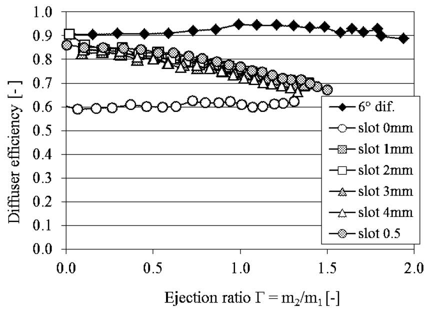

Fig.8: Diffuser efficiency for backpressure measured 300 mm past the diffuser

was wider, the suction mass flow rate was increased only slightly, but the efficiency of the diffuser was decreased. Evidently the suction slot affects the flow in the diffuser adversely and so the narrower the slot, the more advantageous. Only a small slot opening is sufficient, because of the recovery nozzles. If the suction slot is wider than 0.6 mm, its cross section is larger than the cross section of the recovery nozzles through which the fluid sucked through the slot returns to the mixing chamber. To decrease the suction ratio more significantly, the slot opening should be smaller than 0.5 mm.

Fig. 10 shows that diffuser efficiency increases until the suction ratio is 0.06 and then remains constant or increases only negligibly. Unfortunately these suction ratios are obtained only for ejector regimes with low ejection ratios, as is obvious from Fig. 9. For higher ejection ratios, the backpressure is lower and also the pressure difference $p_3 - p_{12}$ is lower and the suction ratio decreases rapidly below sufficient values.

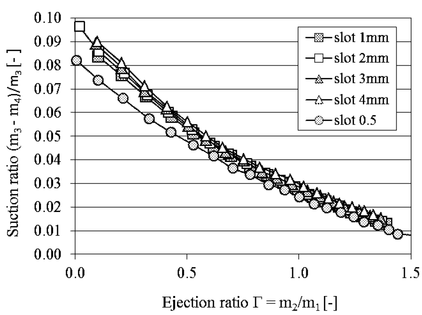

Fig.9: Suction ratio for various settings of the suction slot and ejection ratio

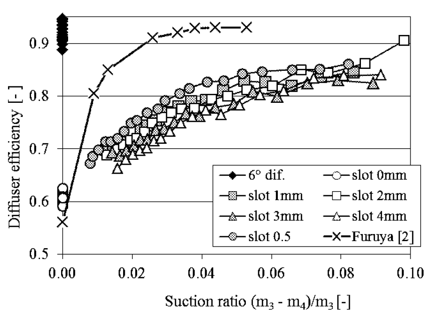

Fig.10: Diffuser efficiency as a function of suction ratio and slot width

The static pressure distribution on the mixing chamber wall was measured to investigate the effect of boundary layer suction and the returning of sucked fluid back into the mixing chamber. The results of measuring the static pressure in the mixing chamber, diffuser and the tube after the diffuser are in Fig. 11. Static pressure was measured in two positions after the diffuser, 30 and 300 mm. Regimes with similar expansion and mixing pressure, p_{12} and p_3 respectively, were chosen to illustrate the influence of various settings of the suction slot on flow in the diffusers. Fig. 11a shows flow in the ejector for low backpressure and the ejection ratio approximately equal to 1.3. The static pressure rise is low in the mixing chamber for this regime, while the mass flow rate as well as the dynamic pressure p_{d3} are high, and therefore the rise in pressure in the diffusers is correspondingly high as well. Fig. 11a shows that only 47 % of the static pressure rise is realized in the 40° diffuser without suction itself, while it is 56 % with suction and 96 % for a 6° diffuser. Fig. 11b presents the results for a regime with an ejection ratio of 0.9. Here, the static pressure rise was bigger in the mixing chamber and the pressure rise in the diffusers was decreased. Finally, Fig. 11c shows static pressure distributions for a regime with high backpressure and an ejection ratio of 0.2. The static pressure rise in the mixing chamber and also the pressure difference $p_3 - p_{12}$ were high for this regime and a diffuser with suction is highly efficient. Here, 85 % of the static pressure rise was realized in a 40° diffuser with suction while it is only 45 % without suction. The static pressure distributions presented in Fig. 11 again confirm that the main and dominant differences in flow throughout the whole ejector occur in the diffuser, which is the crucial part of the ejector.

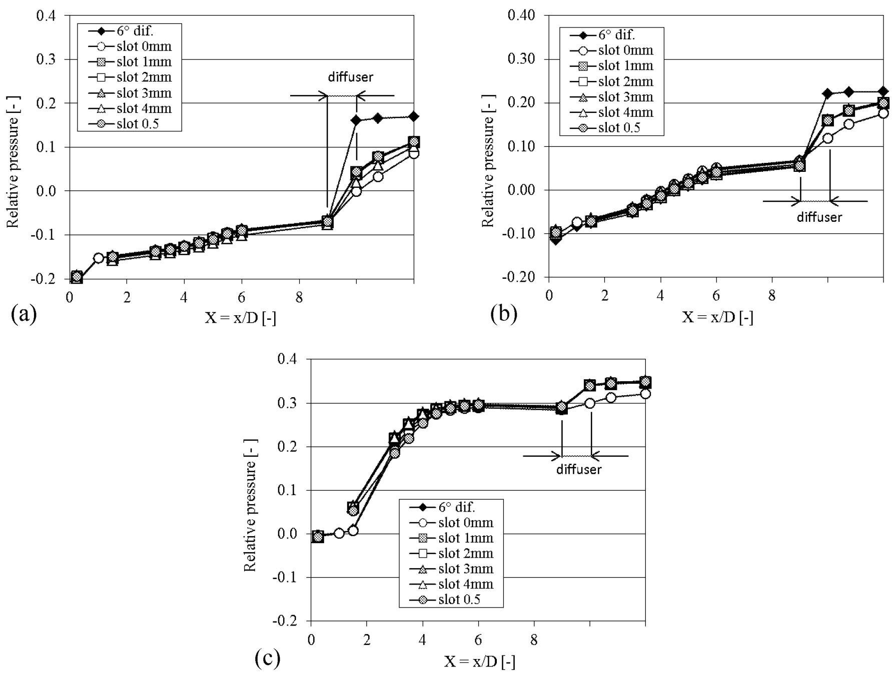

Fig.11: Static pressure distribution on the mixing chamber wall, diffuser and tube after the diffuser for three values of mixing pressure; relative pressure is defined as $(p - p_{02})/(p_{01} - p_{02})$; $X = x/D$ is a dimensionless axial coordinate

4. Conclusions

The effect of boundary layer suction in the inlet of a diffuser with a divergence angle of 40° on flow in the ejector was investigated. Both diffuser and ejector efficiency were calculated. It was found that boundary layer suction can improve the efficiency of the diffuser and thereby of the whole ejector significantly. The suction ratio is dependent on the regime of the ejector, i.e. on the ejection ratio. The diffuser efficiency increases for higher suction ratios and remains almost constant if it is greater than 0.06. Therefore, the suction is inefficient for low backpressures and high ejection ratios of the ejector. The effect of the size of the suction slot opening was investigated too. It was found that a narrower slot is preferable to a wider slot even though the suction ratio is decreased for a narrower slot. A higher suction ratio and more efficient suction cannot be obtained for this configuration of the ejector, because the recovery nozzles are too small and the suction flow rate is consequently limited.

When a diffuser with a divergence angle of 40° was used, the process of flow deceleration was not finished in the diffuser outlet. Generally, it was significant for cases with inefficient diffusers, probably because of flow separation. Therefore, the pressure recovery of the diffuser was calculated after the diffuser exit. Static pressure distributions on the mixing chamber wall were measured and it seems that the sucked fluid which is returned to the mixing chamber does not enhance mixing.

The next academic work focuses on mathematical modelling to obtain a more detailed view of the problem. The optimization of the ejector configuration can yield higher ejector efficiency. It seems that bigger recovery nozzles would be beneficial. Also repositioning the suction slot further downstream in the diffuser can solve the problem with low pressure difference of high ejection ratio regimes.

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