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The Experimental Study on the Pneumatic Ejector, with Special Reference to the Effect of the Lengths of the Diffuser and the Parallel Part of the Mixing Tube on Performance Characteristics

(3 rd and 4 th Report)

(Received September 15, 1956)

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Abstract

Experiments on the effects of the lengths of the diffuser and the parallel part of the mixing tube upon the performance characteristics of a pneumatic ejector, were conducted. The area ratio of the ejector was 2.923, while the distance a' from the nozzle exit section to the inlet of the parallel part was $a'=8.5$.

The lengths of the diffuser were varied. viz. 60.2mm, 160.2mm and 260.2mm, maintaining the divergence angle constant (8 degrees). The vacuum obtained and the ejector efficiency were found to remain unaffected by the length of the diffuser. Then, the lengths l of the parallel part of the mixing tube were varied. viz. $l/e=0.9$, 3.4, 5.9, 8.4,10.9 and 13.4, where e means the inner diameter of the parallel part ($e=10$mm). The results were that the optimum value of the length l tended to increase as the velocities of the driving jet as well as the weight flow ratio G_2/G_1 became larger, where G_1 and G_2 denotes respectively the weight flow of the driving air and the secondary air. Further, it was found that the optimum values of l/e lie in the range $l/e=8.5\sim5$, coinciding with the experimental results by L. J. Kastner and J. R. Spooner.

I. Introduction

The experiments were performed to examine the effects of the lengths of the diffuser and the parallel part of the mixing tube upon the performance characteristics of a pneumatic ejector. The set and the method of experiments were similar to those described in the second report. An ejector, with the area ratio

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2.923 and the distance from the nozzle exit section to the entrance of the parallel part of the mixing tube $a'=8.5$mm, was employed.

The divergence angle, 8 degrees, of the diffuser was put to use, and the length of the diffuser was varied viz. $L_d=60.2$mm, 160.2mm and 260.2mm. It was found by these experiments that, although increments of the diffuser efficiency were made clear when the diffuser length was increased, little improvement was found out in the vacuum obtained and the ejector efficiencies.

In order to find out the effect of the length l of the parallel part of the mixing tube, we used various lengths l viz. 9mm, 34mm, 59mm, 84mm, 109mm and 134mm. As the inner diameter of the parallel part $e=10$mm, the above alterations of lengths are expressed in non-dimensional form as follows; $l/e=0.9$, 3.4, 5.9, 8.4, 10.9 and 13.4. It was found by the experiments, that it was necessary to increase the length l as the driving pressure p_1, in other words, the velocity of the driving fluid as well as the weight flow ratio G_2/G_1 became larger. The optimum values l/e for the pressure ratio (vacuum) and for the ejector efficiency coincide with each other, lying in the ranges $l/e=8.5\sim5$. These figures coincide with those obtained by L. J. Kastner and J. R. Spooner[1] on the pneumatic ejector.

II. The Experiments on the Effect of the Diffuser Length of the Mixing Tube

2・1 The Set and the Method of the Experiment.

The set put to use was the same shown in the second report. The ejector nozzle was of a convergent type with the exit diameter 5.85mm. The mixing tube used was provided with a nozzle-shaped entrance with the inner diameter of the parallel part 10mm, as shown in fig. 12 in the second report. Thus the area ratio yields to: $m=(10)^2/(5.85)^2=2.923$. In this case, the length of the diffuser was varied, maintaining the distance a' from the nozzle exit section to the inlet of the parallel part of the mixing tube constant ($a'=8.5$mm). In order to change the total length of the diffuser, the original diffuser with the divergence angle 8 degrees and the length 60.2mm, shown in fig. 12 in the second report was fitted with the succeeding diffusers having the same divergence angle (8 degrees), and thus the length L_d was altered to 160.2mm and 260.2mm respectively as shown in fig. 5. Namely, the length of the diffuser was varied as follows; $L_d=60.2$mm, 160.2mm and 260.2 mm. Further, the pressure distributions along the mixing tube were measured, as shown in fig. 4 and fig. 5, and also the diffuser inlet temperature T_{12} was observed by means of a thermocouple.

The method of the experiment was similar to the one described in the first report[2] and the second report. Thus, the present experiments were performed

1) L. J. Kastner, J. R. Spooner, Proc. Inst. Mech. Engrs., vol. 162, 1950, pp. 149/159

2) I. Watanabe, T. Watanabe, S. Iso, T. Kawahito, This Proceedings, vol. 7 no. 26, 1954, pp. 51/60

by varying the weight flow of the secondary air G_2 under constant driving pressure p_1. Denoting p_{12}: the absolute pressure at the diffuser inlet, T_{12}: the absolute temperature of the air at the diffuser inlet, F_{12}: the cross-sectional area at the diffuser inlet, w_{12}: the mean air velocity at the section F_{12}, and further, putting p_e: the absolute pressure at the diffuser exit section, F_e:cross-sectional area at the diffuser exit section, w_e:mean air velocity at the diffuser exit section, we have the following relation if the losses in the diffusion process were absent.

$$\frac{w_{12}^2}{2g}-\frac{w_e^2}{2g}=\frac{w_{12}^2}{2g}\left\{1-\left(\frac{F_{12}}{F_e}\right)^2\left(\frac{p_{12}}{p_{ei}}\right)^{\frac{2}{k}}\right\}=\frac{k}{k-1}RT_{12}\left\{\left(\frac{p_{ei}}{p_{12}}\right)^{\frac{k-1}{k}}-1\right\} \tag{1}$$

where p_{ei} represents the absolute pressure at the diffuser outlet section accompanying with no losses in the diffusion process. In the above expression, w_{12}, p_{12}, T_{12}, F_{12} and F_e are the known quantities either by direct measurement or by dimensions of the ejector, and so the numerical value of p_{ei} may be obtained by solving eq. (1) graphically. Then, the diffuser efficiency may be expressed as follows.

$$\eta_d=\frac{\left(\frac{p_e}{p_{12}}\right)^{\frac{k-1}{k}}-1}{\left(\frac{p_{ei}}{p_{12}}\right)^{\frac{k-1}{k}}-1} \tag{2}$$

The ejector efficiency, on the other hand, may be calculable from the following equation.

$$\eta=\frac{G_2\frac{k}{k-1}RT_2\left\{\left(\frac{p_e}{p_2}\right)^{\frac{k-1}{k}}-1\right\}+G_2\frac{w_e^2}{2g}}{G_1\frac{k}{k-1}RT_o\left\{\left(\frac{p_1}{p_0}\right)^{\frac{k-1}{k}}-1\right\}}\doteqdot\frac{G_2T_2\{(p_e/p_2)^{\frac{k-1}{k}}-1\}}{G_1T_0\{(p_1/p_0)^{\frac{k-1}{k}}-1\}} \tag{3}$$

where G_1 : weight flow of the driving fluid, G_2 : weight flow of the secondary air, p_0, T_0 : the pressure and the absolute temperature of the surrounding atmosphere, p_1 : driving pressure, p_2, T_2 : the pressure and the absolute temperature at the secondary stream (vacuum side).

2・2 The Experimental Results and the Considerations.

The relations between the pressure ratio p_2/p_0 and weight flow ratio G_2/G_1 when the driving pressure (gauge) p_1=150mmHg, 250mmHg and 450mmHg in case of the diffuser length L_d=60.2mm, 160.2mm and 260.2mm are shown in fig. 1. As is observed from this figure, the vacuum obtained by the ejector shows little improvement as the diffuser length increases.

Fig. 2 shows the relations between the diffuser efficiency η_d, enumerated by eqs. (1) and (2), and the weight flow ratio G_2/G_1, for L_d=60.2mm, 160.2mm and 260.2mm.

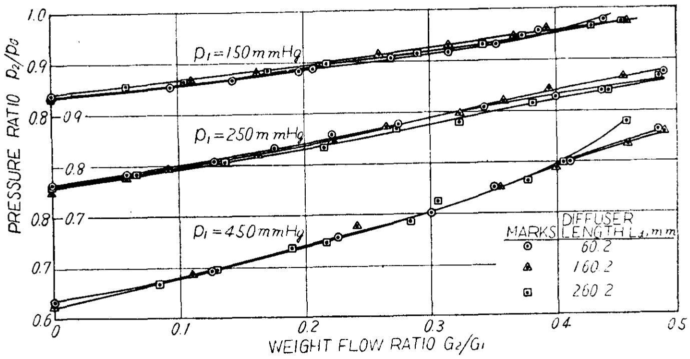


Fig. 1. Pressure Ratio p_2/p_0 vs. Weight Flow Ratio G_2/G_1

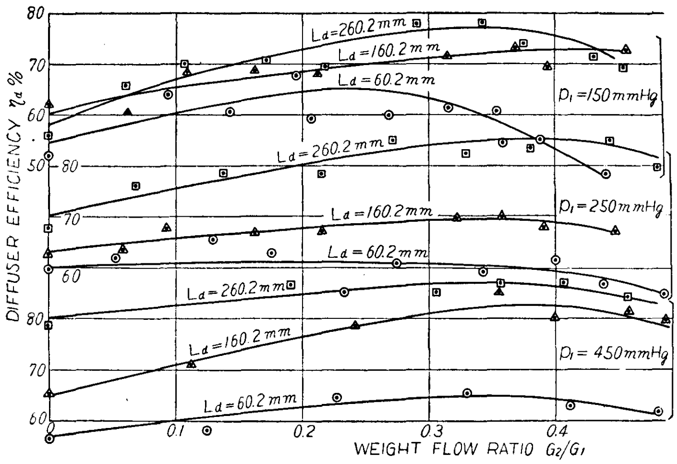


Fig. 2. Diffuser Efficiency η_d vs. Weight Flow Ratio G_2/G_1

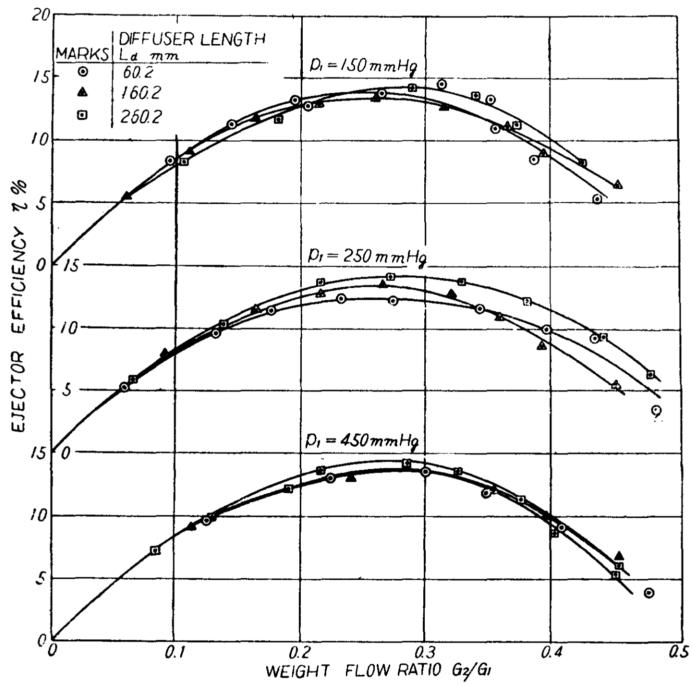


Fig. 3. Ejector Effiiciency η vs. Weight Flow Ratio G_2/G_1

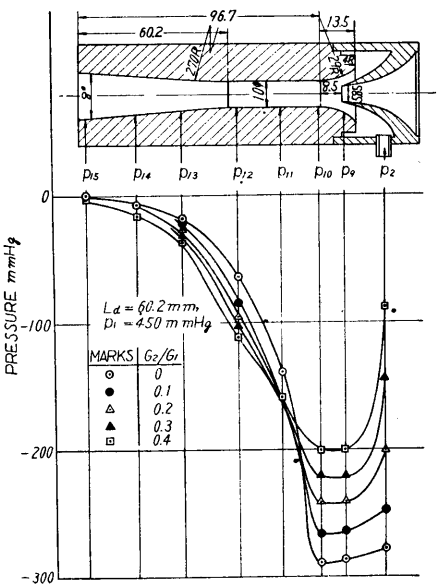
The diffuser efficiency improves considerably as L_a increases, which is considered to be due to the decrease of the exit velocity w_c.

The relations between the ejector efficiency η and the weight flow ratio G_2/G_1 for various driving pressure p_1 and the diffuser length $L_d=60.2$mm, 160.2mm, and 260.2mm are plotted in fig. 3. Little improvement is observed at $p_1=450$mmHg as the diffuser length increases. This tendency is also observed both for $p_1=250$mmHg and 150mmHg, but improvements of the ejector efficiencies are quite small with those of η_d. It is conceivable to us that, the function of the ejector comprises of the traction of the secondary stream by means of the viscosity of the driving fluid at the mixing chamber and the parallel part, and so, the flow in the diffuser or the downstream flow condition will not affect considerably the flow within the mixing chamber and the parallel part, in other words, the function of the ejector. Thus, it is clear that the function of the diffuser is a less important factor in the ejector design.

From the bibliographies already published on ejectors, little information is available about the diffuser length, and most of them are concerned with the diverging angle of the diffuser. Prof. K. Hayami states in his paper,[3] that the divergence angle of the diffuser should be selected within the ranges of 2~6 degrees for steam ejectors. L. J. Kastner and J. R. Spooner[4] conducted an experimental study of the pneumatic ejector with area ratio $m=2.25$, $l/d=3.33$ (l and d denoting the length and the diameter of the parallel part respectively) and the diffuser length 12 d. In these researches, the distance from the nozzle exit section to the entrance of the parallel part was kept optimum, and the divergence angles were used as follows, 5°, 10° and 15° for driving pressures (gauge) 0.49kg/cm^2, 0.98kg/cm^2 and 1.41kg/cm^2. The conclusions are as follows: (1) As the driving pressure becomes lower, it is more necessary to use the diffuser with the correct diverging angle. (2) The optimum diverging angle of the diffuser tends to increase as the driving pressure becomes larger, but we cannot say this positively for want of the available data. Further, R. Royds and E. Johnson[5] made a conclusion. That is, according to their experiments on the steam ejectors, the diffuser length was less important. It is observed that the conclusion of the present survey is near to those conclusions shown in the above mentioned bibliographies.

The pressure distributions along the mixing tube, when $p_1=450$mmHg gauge, are shown in figs. 4 and 5. Fig. 4 represents the case for $L_d=60.2$mm, while fig. 5 shows those for $L_d=160.2$mm and 260.2mm. In these graphs, the pressures $p_{14}\sim p_{18}$ for $L_d=160.2$mm as well as the pressures $p_{15}\sim p_{21}$ for $L_d=260.2$mm are plotted in mm Aq instead of mmHg, because of the small difference of pressures from

3) K. Hayami, Trans. Japan Soc. Mech. Engrs., vol. 7 no. 28, 1941/8, pp.II−11 and vol, 8 no. 31, 1942/5 pp. II−25/32.
4) loc. cit. 1).
5) R. Royds, E. Johnson, Proc. Inst. Mech. Engrs., vol. 145, pp. 193/209



the atmospheric pressure. It is easily known from these diagrams that the pressures rise smoothly along the diffuser length and further, that the pressures in the mixing chamber p_9 and p_{10} at the ranges $G_2/G_1=0\sim0.4$ for the nozzle type inlet mixing chamber show smaller differences than those for the conical inlet type.[6] In other words, the mixing process in the nozzle inlet mixing chamber may be treated as constant pressure mixing.

Fig. 4. Pressure Distributions along the Mixing Tube (L_d=60.2mm, p_1=450mmHg)

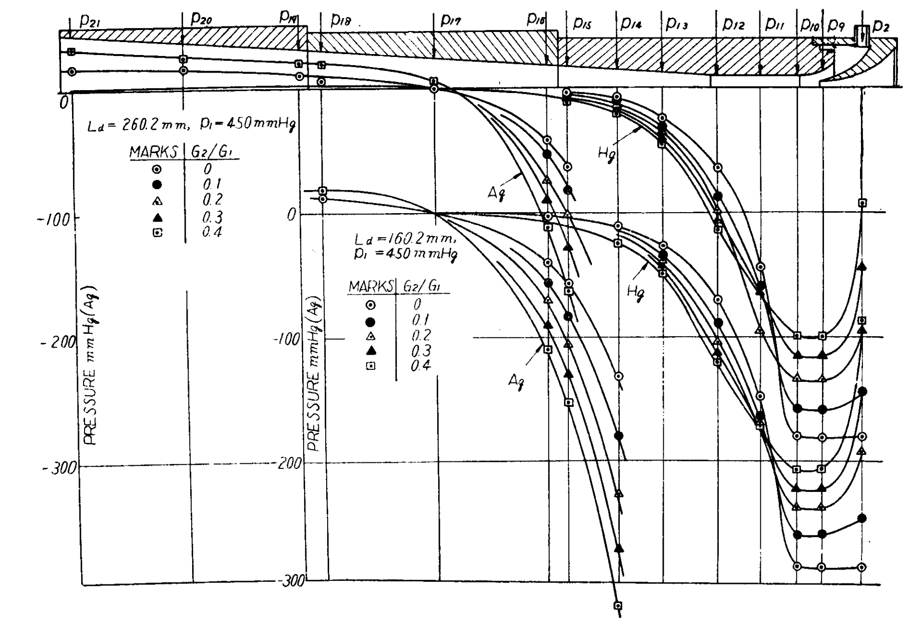


Fig. 5. Pressure Distributions along the Mixing Tube (L_d=160.2mm and 260.2mmHg, p_1=450mmHg)

III. The Experiments on the Effect of the Length of the Parallel Part of the Mixing Tube.

3・1 The Set and the Method of the Experiment.

6) Compare with fig. 10 in reference 2), and figs. 7~10 in the 2nd report.

The set and the method of the experiment are similar to those shown in the first and the second report. The length l[7] of the parallel part of the mixing tube was as follows: —— $l=9$mm, 34mm, 59mm, 84mm, 109mm and 134mm, by means of the five parallel part interpieces as shown in fig. 6. As the inner diameter e of the parallel part is $e=10$mm, the values l/d amount to 0.9, 3.4, 5.9, 8.4, 10.9 and 13.4 respectively. Fig. 7 shows the arrangement of the elongated parallel part ($l=84$mm) by means of the three parallel part interpieces. The pressures at the entry to the mixing tube were measured at the three points shown in the figure. Temperature measurements were also taken by thermocouples set up in the two holes of 2mm diameters shown below the diffuser part.

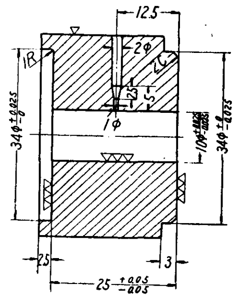


Fig. 6. Parallel Part Interpieces

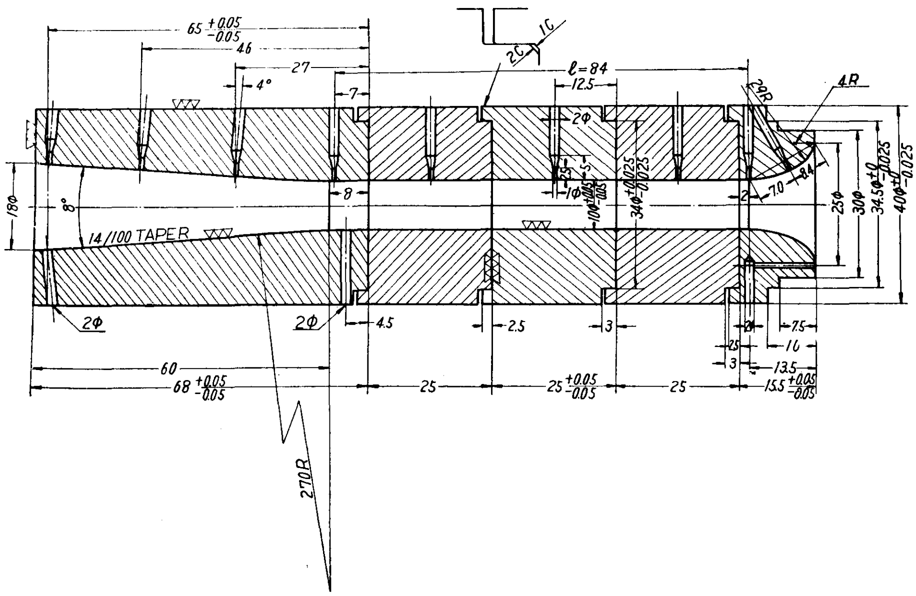


Fig. 7. The Elongated Parallel Part by means of three Parallel Part Interpieces

3・2 The Experimental Results and the Considerations.

The relations between the pressure ratio p_2/p_0 and the weight flow ratio G_2/G_1 in case of driving pressure $p_1=50$mmHg, 150mmHg, 250mmHg, 350mmHg and 450mm Hg are shown in fig. 8. The replotting of the pressure ratio p_2/p_0 versus l/e from the figure yields to that shown in fig. 9. As the function of the ejector is considered to be

7) In this case, the length l means the distance from the inlet of the parallel part to the pressure measuring hole just ahead of the diffuser inlet.

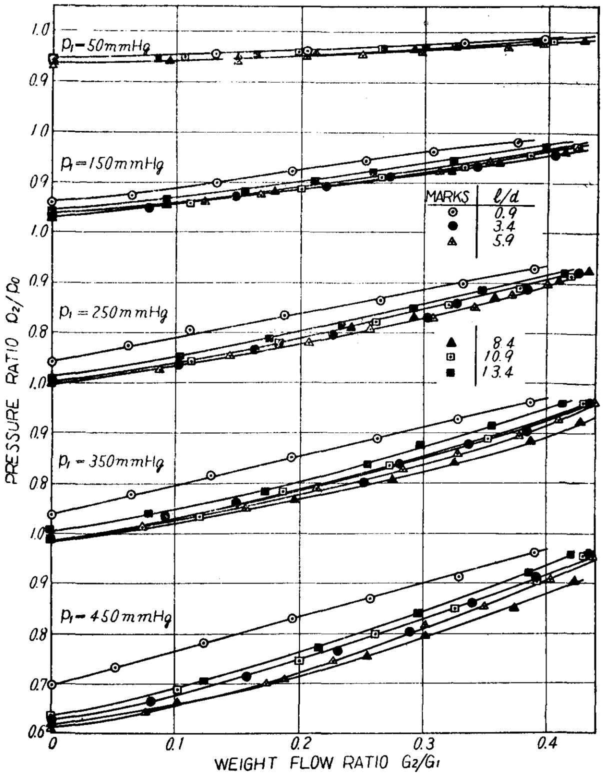


Fig. 8. Pressure Ratio p_2/p_0 vs. Weight Flow Ratio G_2/G_1

due to the traction phenomenon of the secondary stream by the viscosity of the driving fluid, it is probable, that, the longer length l is required, when the velocities of the driving stream as well as the weight flow ratios G_2/G_1 become larger. From fig. 9, it is observed that, while the optimum l/e which renders the pressure ratio to a minimum is about 8.5 for $G_2/G_1=0.4$, 0.3 and 0.2, the optimum value decreases to about 7 as G_2/G_1 decreases to 0.1. When $p_1=350$mmHg, this optimum values l/e yield to 8.5 $\sim$8 for $G_2/G_1=0.4$, 0.3, 0.2 and about 6 for $G_2/G_1=0.1$. In the case when $p_1=150$mmHg, the optimum values are 8.5 for $G_2/G_1=0.4$, 0.3 and about 6 for $G_2/G_1=0.2$ and 0.1. These figures abovementioned show that the above-mentioned concepts are satisfactory on the whole.

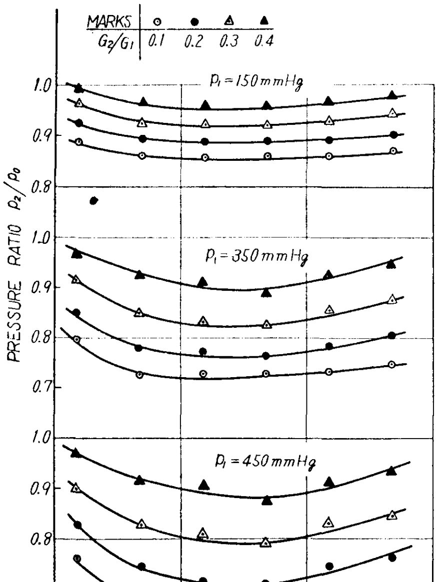


Fig. 9. Pressure Ratio p_2/p_0 vs. l/e

The ejector efficiencies, evaluated by eq. (3), plotted against the weight flow ratio G_2/G_1 are shown in fig. 10. Replotting the efficiencies versus l/e for $G_2/G_1=0.1\sim0.4$, we got fig. 11. The curves in fig. 11 seem to be rather complicated, on account of complications of the curves in fig. 10. The optimum l/e, in this case, however, coincides near exactly with those just mentioned above. That is, the optimum values l/e for η when $p_1=450$ mmHg amount to 8.5 in case of $G_2/G_1=0.4$, 0.3 and 0.2, and about 7$\sim$6 in case of $G_2/G_1=0.1$. When $p_1=350$mmHg, these optimum values are 8.5 for $G_2/G_1=0.4$, 0.3 and 0.2 and

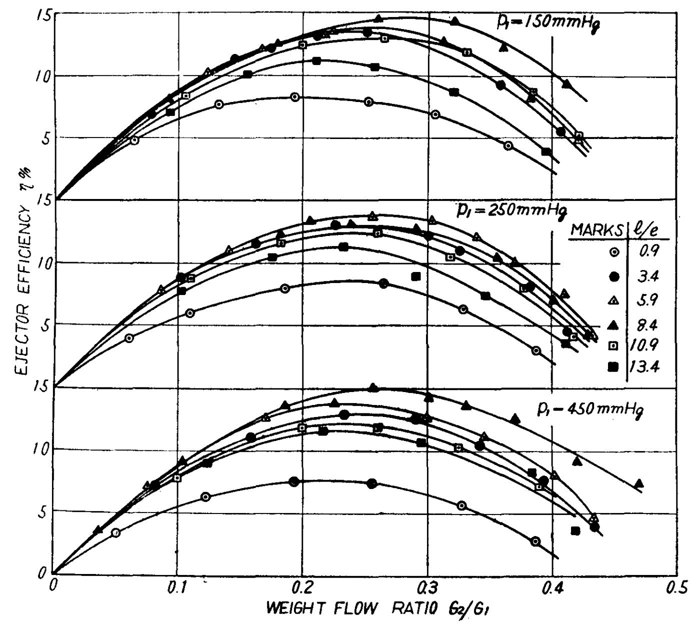


Fig. 10. Ejector Efficiency η vs. Weight Flow ratio G_2/G_1

5 for $G_2/G_1=0.1$. Further, as $p_1=50$mmHg, $l/e=6$ for $G_2/G_1=0.4$, 0.3, $l/e=5.5$ for $G_2/G_1=0.2$ and $l/e\doteqdot 5$ for $G_2/G_1=0.1$.

The pressure distributions along the mixing tube are shown in fig. 12 ($l/e=0.9$, $G_2/G_1=0.4$) and fig. 13 ($l/e=13.4$, $G_2/G_1=0.4$). Fig. 12 corresponds to the case when the length of the parallel part is the shortest, and the pressures increase gradually in the parallel part. On the other hand, fig. 13 corresponds to the case when the length l is the longest. In the latter case, the pressure curves have the maxima at the observation point 11", and then, after lowering along the axis, the pressures increase finally in the diffuser. This phenomenon may also be explained by W. Tollmien's theory, as described in the second report. The boundary η^*, which bisects the driving stream and the secondary stream, expands gradually, as explained

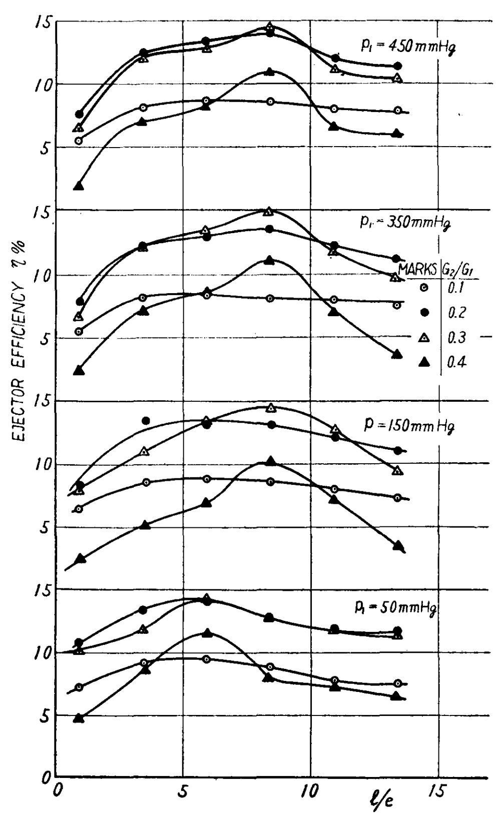


Fig. 11. Ejector Efficiency η vs. l/e

in the second report, and thus, using $\eta^*/a = -0.1855$ (W. Tollmien's solution) and $a = 0.0845$, we have $\eta^* = -0.01568$. As the difference in $e = 10$mm and $d_n = 5.85$ in radius amounts to $4.15/2 = 2.075$mm, the evaluation of the point, at which the boundary η^* just impinges on the inner wall of the parallel part, yields to $x = 132.2$mm, because $\eta^* = y/x = -2.075/x = -0.01568$, where x and y being the axis for axial and radial direction respectively. This value $x = 132.2$mm corresponds to the intermediate point beween 11'''' and 12 in fig. 13. As we mentioned in the second report, W.Tollmien assumed that the secondary stream was initially at rest, and further, the driving fluid as well as the secondary fluid had the same densities. If the secondary stream has velocity component parallel to the axis of the driving stream initially, the line η^* tends to spread more widely, as we said in the second report. Further, if the density of the driving stream is larger than that of the secondary stream, as in the present case, it is conceivable that the tendency above-mentioned will be more emphasized. Thus, the result is that, the position, at which the line η^* impinges the inner wall of the parallel part, will reach farther upstream with respect to the point 11'''' or nearer

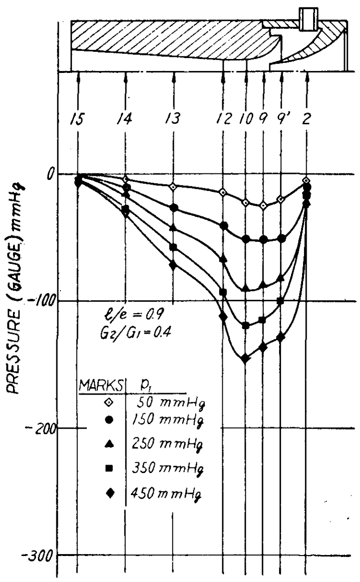


Fig. 12. Pressure Distributions along the Mixing Tube ($l/e = 0.9$, $G_2/G_1 = 0.4$)

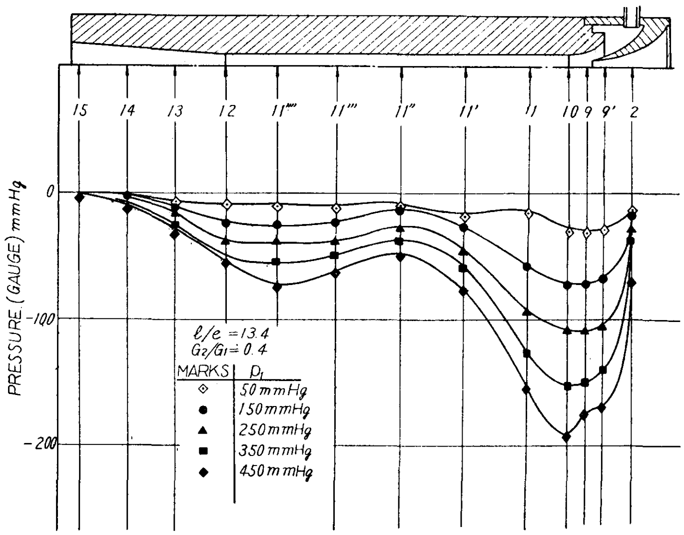


Fig. 13. Pressure Distributions along the Mixing Tube ($l/e = 13.4$, $G_2/G_1 = 0.4$)

to the point 11. The fact, that there exists the maximum at point 11", is considered due to the above-mentioned point of view. From this point afterwards, it is conceivable that the flow expands to lower the pressure, up to the diffuser, where the stream will be compressed again. According to these considerations, it is preferable that the parallel part of the mixing tube should be cut out at the point 11". As the point 11" is situated at the distance $l=64.5$mm, the cut out at this point implies that $l/e=6.45$, which, roughly speaking, is of the same magnitude as the optimum values discussed in figs. 12 and 13.

The bibliographies hitherto published show us the following data. That is, F. R. B. Watson states, in his paper concerning the steam ejector, that, although the length l of the parallel part does not affect so severely as the distance a' from the nozzle exit to the entrance to the parallel part, it affects on the generated vacuum in some respects. Figs. 3 and 5 in his paper, show us that, higher vacuum is attained with $l=38.1$mm than that with $l=1.6$mm.[8]

Prof. K. Hayami [9] shows, in his paper on steam ejectors, that, though the length l has little effect on the maximum suction vaccum obtained, the minimum exhaust vacuum is lowered as the length l increases, and finally, the operation of the ejector will become unsatisfactory like the case in which the length is extremely short. He also suggests that the length l should be selected as follows: —— $l=(3\sim6)\ e$ in the design procedure, where e denotes the inner diameter of the parallel part. The value 6 e, above-mentioned, is a figure close to that obtained in the present experiment.

L. J. Kastner and J. R. Spooner,[10] states, in their paper on the pneumatic ejector, that the value l/e is a significant factor which should be selected as follows: —— $l/e=7\sim8$. These conclusions are in close agreement with those of the present authors. L. J. Kastner and J. R. Spooner states, further, that, though the compensation of the distance l by the distance a' when too small value of l/e is selected, is possible, the optimum performance of the ejector would not be attained any more.

IV. Conclusions

From the above-mentioned results, the conclusions are as follow.: ——

(1) Although the increment of the diffuser length improves the diffuser efficiency η_a to some extent, it gives little improvement both on the vaccum obtained and the ejector efficiency η. The function of the ejector is conceived to be the traction phenomenon of the secondary stream by the viscosity of the driving fluid, and from this concept, the fact that the flow in the diffuser gives little effect on the ejector vacuum and the efficiency, is quite clear.

8) F. R. B. Watson, Proc. Inst. Mech. Engrs., vol. 124, 1933/2, pp. 231/261.

9) Loc. cit. 3)

10) Loc. cit. 1)

(2) The experimental results by varying the length l of the parallel part shows us that, the higher the driving pressure is, in other words, the higher the exit velocity from the nozzle is, and also, the larger the weight flow ratio G_2/G_1 is, the longer l should be employed. These fact may also be explained by the concept just described in (1). The optimum value l/e varies in the ranges 8.5∼5, and the optimum value, both for the pressure ratio and the ejector efficiency, coincides with each other.

(3) This observation of the pressure distributions along the parallel part shows us that, the pressure curves have maxima at some point intermediate of the parallel part, if long parallel part is employed. The cut off of the parallel part beyond this point, yields to $l/e=6.45$, which coincides approximately with the figures shown in (2).

(4) L. J. Kastner and J. R. Spooner states, in their paper concerning the pneumatic ejector, that l should be selected in such a way as the condition $l/e=7\sim8$ would be fulfilled. The figures also coincide closely with those obtained by the present authors.

The authors are indebted for the efforts made by Teturo Nakata and Hirosi Nakagawa, and wish to express their heartful thanks to them.