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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<sup>rd</sup> and 4<sup>th</sup> 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 ( $e=10\text{mm}$ ). 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  $l/e=8.5\sim 5$ , 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  $a' = 8.5\text{mm}$ , was employed.

The divergence angle, 8 degrees, of the diffuser was put to use, and the length of the diffuser was varied viz.  $L_d = 60.2\text{mm}$ ,  $160.2\text{mm}$  and  $260.2\text{mm}$ . 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.  $9\text{mm}$ ,  $34\text{mm}$ ,  $59\text{mm}$ ,  $84\text{mm}$ ,  $109\text{mm}$  and  $134\text{mm}$ . As the inner diameter of the parallel part  $e = 10\text{mm}$ , 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  $l/e = 8.5 \sim 5$ . These figures coincide with those obtained by L. J. Kastner and J. R. Spooner<sup>1)</sup> 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  $5.85\text{mm}$ . The mixing tube used was provided with a nozzle-shaped entrance with the inner diameter of the parallel part  $10\text{mm}$ , 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 ( $a' = 8.5\text{mm}$ ). In order to change the total length of the diffuser, the original diffuser with the divergence angle 8 degrees and the length  $60.2\text{mm}$ , 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  $160.2\text{mm}$  and  $260.2\text{mm}$  respectively as shown in fig. 5. Namely, the length of the diffuser was varied as follows;  $L_d = 60.2\text{mm}$ ,  $160.2\text{mm}$  and  $260.2\text{mm}$ . 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<sup>2)</sup> 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.

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

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.

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

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

$$\eta = \frac{G_2 \frac{k}{k-1} RT_2 \left\{ \left( \frac{p_e}{p_2} \right)^{\frac{k-1}{k}} - 1 \right\} + G_2 \frac{w_e^2}{2g} \frac{G_2 T_2 \left\{ \left( \frac{p_e}{p_2} \right)^{\frac{k-1}{k}} - 1 \right\}}{G_1 \frac{k}{k-1} RT_0 \left\{ \left( \frac{p_1}{p_0} \right)^{\frac{k-1}{k}} - 1 \right\}}}{G_1 \frac{k}{k-1} RT_0 \left\{ \left( \frac{p_1}{p_0} \right)^{\frac{k-1}{k}} - 1 \right\}} \quad (3)$$

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)  $p_1 = 150\text{mmHg}$ ,  $250\text{mmHg}$  and  $450\text{mmHg}$  in case of the diffuser length  $L_d = 60.2\text{mm}$ ,  $160.2\text{mm}$  and  $260.2\text{mm}$  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  $\eta_d$ , enumerated by eqs. (1) and (2), and the weight flow ratio  $G_2/G_1$ , for  $L_d = 60.2\text{mm}$ ,  $160.2\text{mm}$  and  $260.2\text{mm}$ .

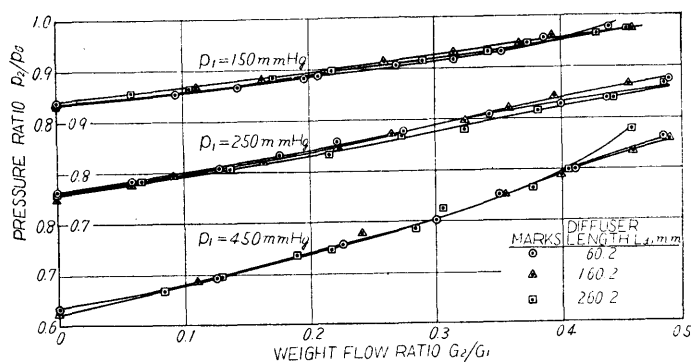


Fig. 1. Pressure Ratio  $p_2/p_0$  vs. Weight Flow Ratio  $G_2/G_1$

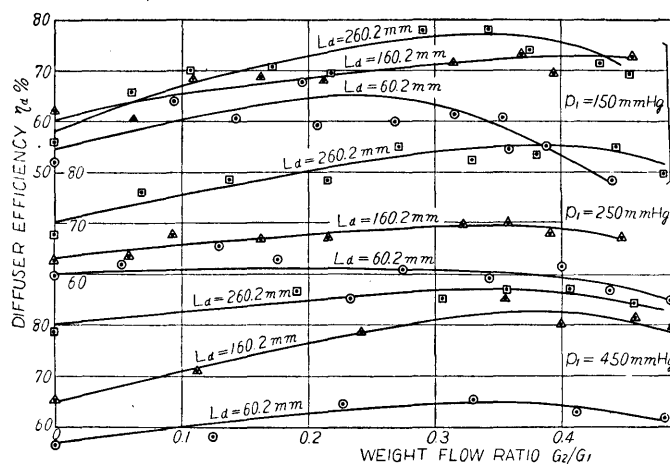


Fig. 2. Diffuser Efficiency  $\eta_d$  vs. Weight Flow Ratio  $G_2/G_1$

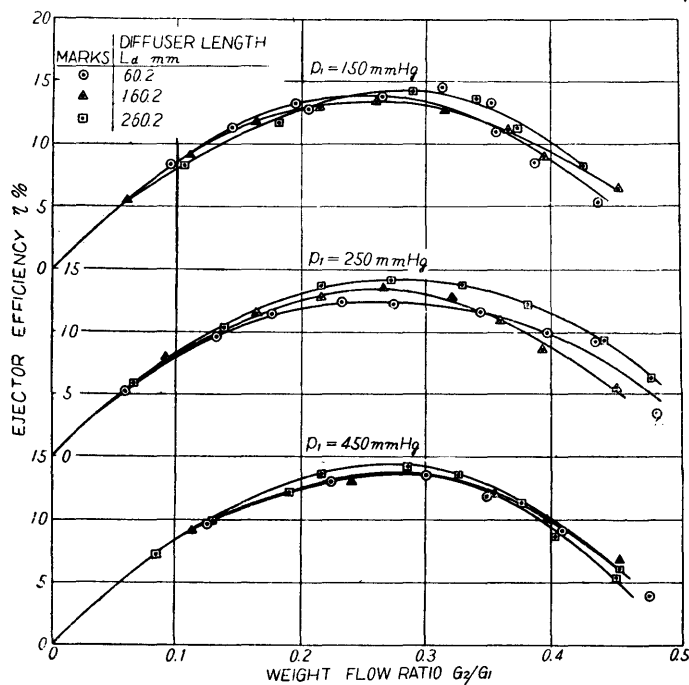


Fig. 3. Ejector Efficiency  $\eta$  vs. Weight Flow Ratio  $G_2/G_1$

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

The relations between the ejector efficiency  $\eta$  and the weight flow ratio  $G_2/G_1$  for various driving pressure  $p_1$  and the diffuser length  $L_d=60.2\text{mm}$ ,  $160.2\text{mm}$ , and  $260.2\text{mm}$  are plotted in fig. 3. Little improvement is observed at  $p_1=450\text{mmHg}$  as the diffuser length increases. This tendency is also observed both for  $p_1=250\text{mmHg}$  and  $150\text{mmHg}$ , but improvements of the ejector efficiencies are quite small with those of  $\eta_d$ . 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.

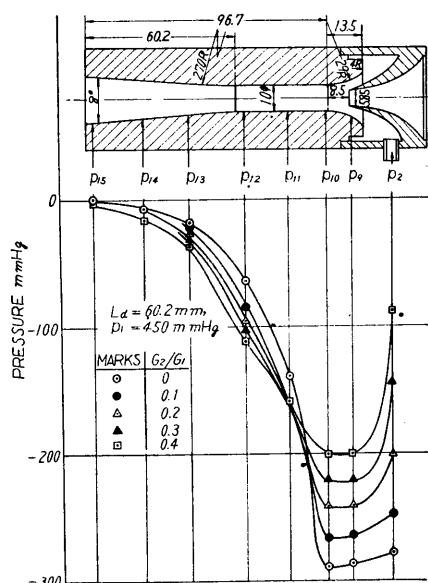
From the bibliographies already published on ejectors, little information is available about the diffuser length, and most of them are concerned with the diverging angle of the diffuser. Prof. K. Hayami states in his paper,<sup>3)</sup> that the divergence angle of the diffuser should be selected within the ranges of  $2\sim 6$  degrees for steam ejectors. L. J. Kastner and J. R. Spooner<sup>4)</sup> 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  $12d$ . 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,  $5^\circ$ ,  $10^\circ$  and  $15^\circ$  for driving pressures (gauge)  $0.49\text{kg/cm}^2$ ,  $0.98\text{kg/cm}^2$  and  $1.41\text{kg/cm}^2$ . 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<sup>5)</sup> 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  $p_1=450\text{mmHg}$  gauge, are shown in figs. 4 and 5. Fig. 4 represents the case for  $L_d=60.2\text{mm}$ , while fig. 5 shows those for  $L_d=160.2\text{mm}$  and  $260.2\text{mm}$ . In these graphs, the pressures  $p_{14}\sim p_{18}$  for  $L_d=160.2\text{mm}$  as well as the pressures  $p_{15}\sim p_{21}$  for  $L_d=260.2\text{mm}$  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  $G_2/G_1 = 0 \sim 0.4$  for the nozzle type inlet mixing chamber show smaller differences than those for the conical inlet type.<sup>6)</sup> 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 ( $L_d = 60.2\text{mm}$ ,  $p_1 = 450\text{mmHg}$ )

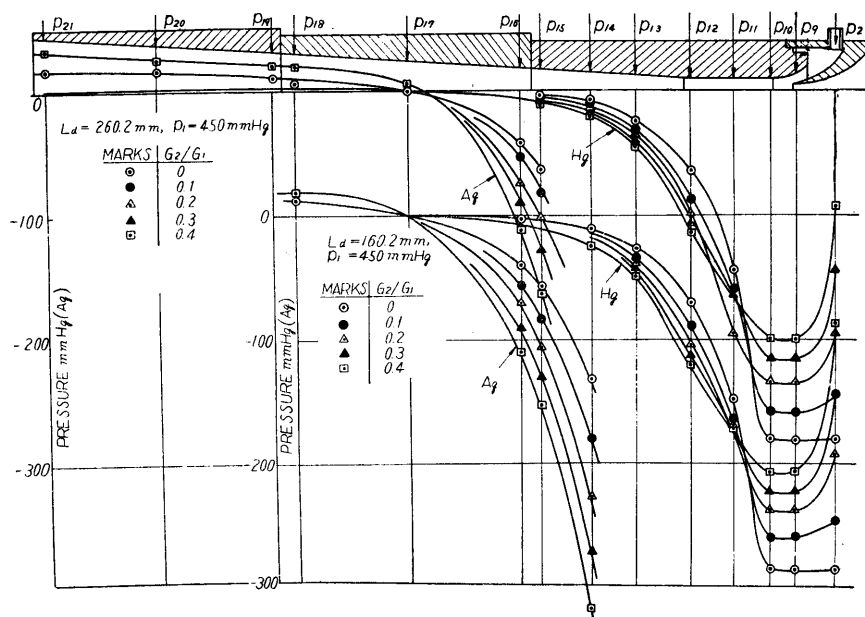


Fig. 5. Pressure Distributions along the Mixing Tube ( $L_d = 160.2\text{mm}$  and  $260.2\text{mmHg}$ ,  $p_1 = 450\text{mmHg}$ )

### 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.

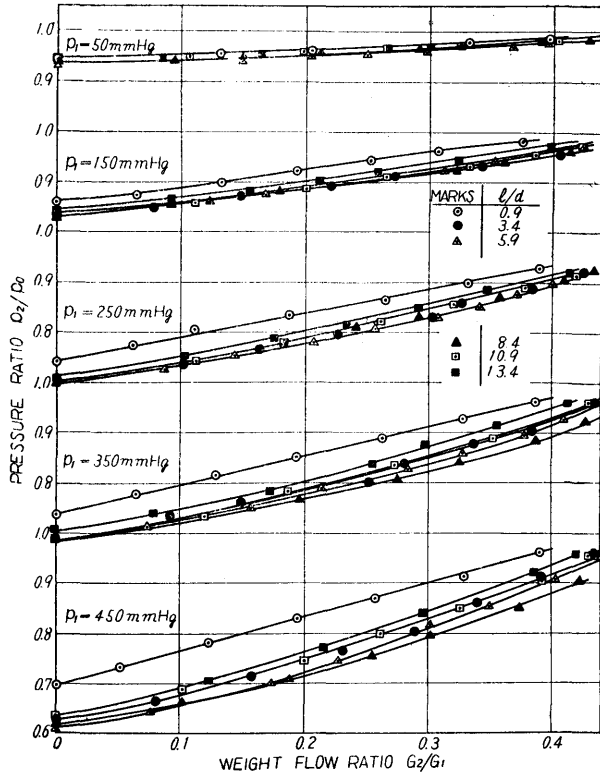
- 6) Compare with fig. 10 in reference 2), and figs. 7~10 in the 2nd report.

[illegible]

### 3.2 The Experimental Results and the Considerations.

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.





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

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

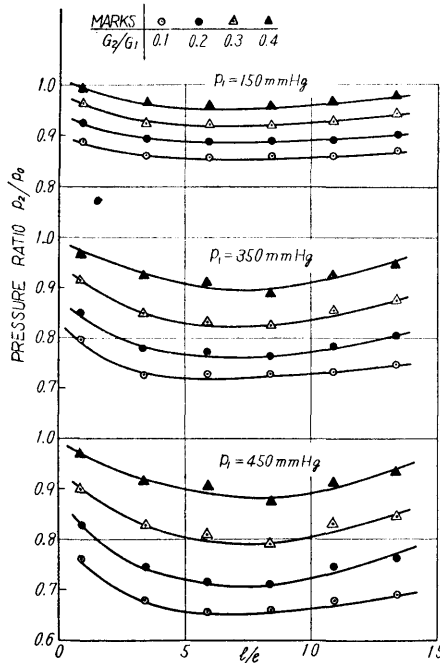
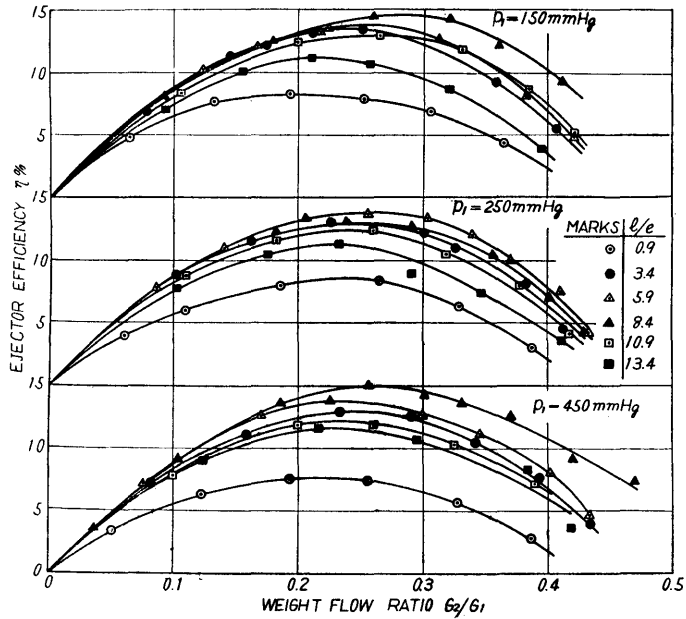


Fig. 9. Pressure Ratio  $p_2/p_0$  vs.  $l/e$

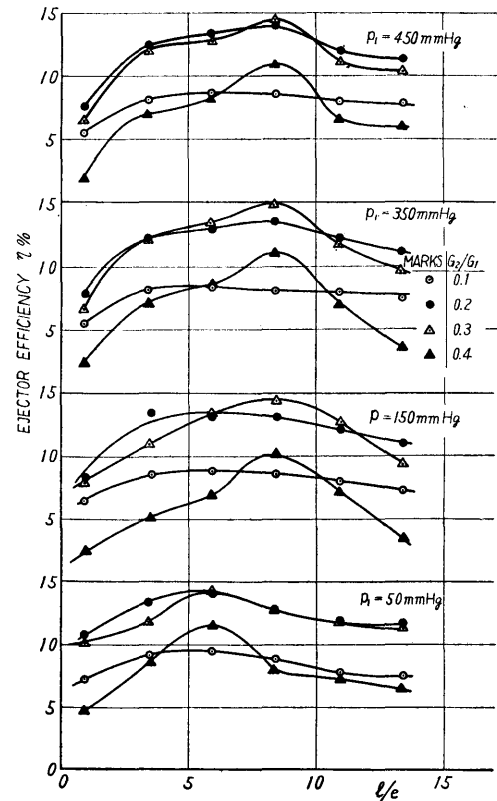
case when  $p_1=150$  mmHg, the optimum values are 8.5 for  $G_2/G_1=0.4, 0.3$  and about 6 for  $G_2/G_1=0.2$  and  $0.1$ . These figures above-mentioned show that the above-mentioned concepts are satisfactory on the whole.

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  $G_2/G_1=0.1 \sim 0.4$ , 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  $\eta$  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

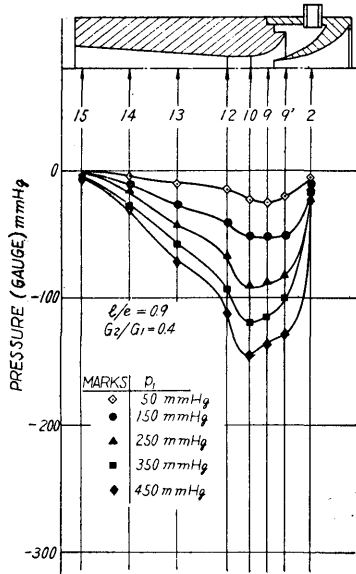
Fig. 10. Ejector Efficiency  $\eta$  vs. Weight Flow ratio  $G_2/G_1$ 

5 for  $G_2/G_1=0.1$ . Further, as  $p_1=50\text{mmHg}$ ,  $l/e=6$  for  $G_2/G_1=0.4, 0.3$ ,  $l/e=5.5$  for  $G_2/G_1=0.2$  and  $l/e=5$  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  $\eta^*$ , which bisects the driving stream and the secondary stream, expands gradually, as explained

Fig. 11. Ejector Efficiency  $\eta$  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  $e = 10\text{mm}$  and  $d_n = 5.85$  in radius amounts to  $4.15/2 = 2.075\text{mm}$ , the evaluation of the point, at which the bound-



ary  $\eta^*$  just impinges on the inner wall of the parallel part, yields to  $x = 132.2\text{mm}$ , because  $\eta^* = y/x = -2.075/x = -0.01568$ , where  $x$  and  $y$  being the axis for axial and radial direction respectively. This value  $x = 132.2\text{mm}$  corresponds to the intermediate point between 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  $\eta^*$  tends to spread more widely, as we said in the second report. Further, if the density of the driving

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

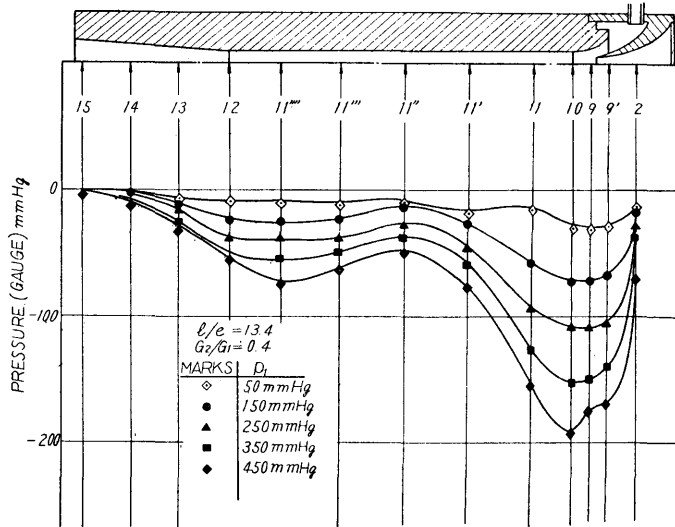


Fig. 13. Pressure Distributions along the Mixing Tube  
( $l/e = 13.4$ ,  $G_2/G_1 = 0.4$ )

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  $\eta^*$  impinges the inner wall of the parallel part, will reach farther upstream with respect to the point 11''' or nearer

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  $l=64.5\text{mm}$ , 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  $l=38.1\text{mm}$  than that with  $l=1.6\text{mm}$ .<sup>8)</sup>

Prof. K. Hayami<sup>9)</sup> shows, in his paper on steam ejectors, that, though the length  $l$  has little effect on the maximum suction vacuum 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  $6e$ , above-mentioned, is a figure close to that obtained in the present experiment.

L. J. Kastner and J. R. Spooner,<sup>10)</sup> 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  $\eta_a$  to some extent, it gives little improvement both on the vacuum obtained and the ejector efficiency  $\eta$ . 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.

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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  $l/e=7\sim 8$  would be fulfilled. The figures also coincide closely with those obtained by the present authors.

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