THE EFFECTS OF PRIMARY TEMPERATURE AND DISCHARGE PRESSURE
ON THE PERFORMANCE OF AN AIR EJECTOR

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David I-Jaw Wang
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LIST OF SYMBOLS

English Letters

A  Cross-sectional area, sq. in. or sq. ft.
b  A complex expression, defined on p. 11.
c  A complex expression, defined on p. 11.
cp  Specific heat at constant pressure, for air
    \[ c_p = 0.241 \text{ B.T.U./lb.-}^\circ\text{F.}, \text{ or } 188 \text{ ft.-} \text{lb.}/\text{lb.-}^\circ\text{F.} \]
D  Diameter, in. or ft.
e  Diffuser efficiency, per cent.
g  Gravitational acceleration, 32.2 ft./sec.\(^2\)
h  Enthalpy per unit mass, B.T.U./lb.
J  Mechanical equivalent of heat, 778 ft.-lb./B.T.U.
k  Ratio of specific heat at constant pressure to specific heat at
    constant volume, for air \( k = 1.400 \).
L  Length of constant area throat of secondary nozzle, inches.
M  Mach Number, stream velocity divided by local acoustic velocity,
    dimensionless.
\( p \)  Pressure, lb./sq. in. or lb./sq. ft.
R  Gas constant, for air \( R = 53.6 \text{ ft.-} \text{lb.}/\text{lb.-}^\circ\text{F.} \)
s  Entropy per unit mass, B.T.U./lb.-\(^\circ\text{F.} \)
T  Absolute temperature, \(^\circ\text{R.} \)
t  Temperature, \(^\circ\text{F.} \)
V  Velocity, ft./sec.
v  Specific volume, cu. ft./lb.
Mass rate of flow, lb./sec.

Distance between throat of primary nozzle and beginning of constant area throat of secondary nozzle, inches.

Greek Letters

\( \rho \quad \) Density, lb./cu. ft.

\( \tau \quad \) Ratio of stagnation temperatures, secondary fluid to primary fluid, dimensionless.

\( \omega \quad \) Ratio of secondary to primary flow.

Subscripts

1 \quad \) Refers to section 1, Fig. 1.
2 \quad \) Refers to section 2, Fig. 1.
3 \quad \) Refers to section 3, Fig. 1.
i \quad \) Refers to section 1, Fig. 1.
o \quad \) Refers to section o, Fig. 1.
t \quad \) Refers to section t, Fig. 1.
x \quad \) Refers to section x, Fig. 1.

Superscripts:

\(^*\quad \) Refers to primary fluid.

\( ^" \quad \) Refers to secondary fluid.
SUMMARY

The purpose of this investigation was to study the effects of primary air temperature and discharge pressure on the performance of an air ejector.

Originally, the design and analysis of the ejector system was based on reversible adiabatic flow in the primary nozzle, a one-dimensional constant pressure process of mixing, a transverse compression shock in the constant area throat of the secondary nozzle, and an isentropic compression in the diffuser.

The ejector was to operate at high discharge pressure and low flow ratio. Thus the ratio of the throat areas of the secondary nozzle to the primary nozzle was designed to be 1.90, while most of the existing ejectors now being used in industry have area ratios from twenty to two-hundred.

The results of the experiment indicate, however, that for such a small area ratio, the flow in the primary nozzle is not reversible adiabatic, but one containing a shock phenomenon. This necessitated the operation of the ejector at a primary pressure of approximately thirty p.s.i.a., far below the design value of one hundred p.s.i.a. Thus, it seems that the analysis must be refined in order to arrive at a more rational design of ejectors at small area ratios.
Otherwise, the test results indicate that for every primary pressure, the performance of the ejector is at an optimum at some particular primary temperature; that the flow ratio decreases, almost linearly at first, with increasing discharge pressure, and very sharply later, when the discharge pressure is further increased. Comparison between theoretical calculations based on a shock process in the primary nozzle and the test values show that the actual $p_3/p_1$ is approximately ninety per cent of the calculated value, indicating that the ejector is quite satisfactory in the subsonic range.
CHAPTER I

INTRODUCTION

Early classical study of the ejector was made by Stodola,\(^1\) and later by Bosnjakovic\(^2\) and Flugel.\(^3\) The work of these men dealt mainly with a one-dimensional theoretical analysis, based on the assumption of uniform and one-directional velocity at certain sections and of adiabatic flow throughout the ejector. Due to these assumptions, equations of conservation of energy and momentum were written and solved simultaneously. Certain predictions were made with regard to the flow ratio, i.e., the ratio of the secondary flow to primary flow, as affected by the geometric configurations of the ejector and by the thermodynamic properties of the fluids.

Several of the assumptions made in the one-dimensional theoretical analysis of the ejector were somewhat questionable, and it was felt that this analysis could not lead to a rationalized design of ejectors. Some investigations of the ejector based on a two-dimensional analysis were therefore made, notably by Goff and Coogan.\(^4\)

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In recent years, more elaborate research regarding the theory and practice of the ejector has been made by Keenan and Nuemann\textsuperscript{5} who applied a one-dimensional analysis to a central type ejector, Fig. 1.

This analysis covered two cases, (a) mixing at constant pressure, (b) mixing at constant area. Test data were obtained for constant area of mixing using large ratios of mixing tube area to primary nozzle area and relatively small primary air pressures. A later paper by Keenan, Nuemann and Lustwerk\textsuperscript{6} greatly extended the range of variables. Assumptions other than constant pressure or constant area mixing yielded no satisfactory analysis.\textsuperscript{7}

The greater portion of these recent investigations, however, dealt with ejectors having a high flow ratio and operating at moderate or low pressures and temperatures. Information regarding ejectors with low flow ratios or ones operating at high temperature is extremely scarce.


\textsuperscript{7} Ibid., pp. A-299--A-309.
The object of this study was to make a preliminary design and an experimental investigation of a central type air ejector operating at high temperature, in the neighborhood of 1000 degrees Fahrenheit, and having a low flow ratio, in the order of 1:10.

In the light of the very limited amount of research already done on an ejector of the low flow ratio and high operating temperature type, the design and analysis of such an ejector used in this investigation are based mainly upon the practice and recommendations of those whose investigations dealt with ejectors of the moderate operating temperature and high flow ratio type.

Due to time limitations the scope of the theoretical investigations was limited to a one-dimensional analysis and design as well as some preliminary study of the effects of the shock wave phenomenon in the primary and secondary nozzles on the performance of an ejector; while the experimental investigations consisted of observation of the effects of primary air pressure, primary air temperature and discharge pressure on the ratio of secondary air flow to primary air flow. All the experimental work was carried out on an ejector of a fixed geometrical configuration.
CHAPTER II

APPARATUS

The general arrangement of the experimental apparatus is shown in a schematic layout in Fig. 17. Primary air was supplied by a reciprocating compressor. This air was heated by ten 1000-watt high temperature electrical heating elements placed within five heater tubes. A pressure regulator was installed in the primary air line to maintain a constant pressure; an oil strainer was also incorporated in the primary air line to keep the primary air as oil-free as possible.

Secondary air was drawn into the induction chamber from the atmosphere. Mixing of the primary and secondary air took place in the secondary nozzle section. The mixed stream was discharged to the atmosphere through the discharge valve.

Static pressures were measured by pressure gauges at the approach of the primary nozzle and at the exit of the secondary nozzle. Temperatures were recorded by thermocouples installed at the approach of the primary nozzle and at the exit of the secondary nozzle. Ambient dry bulb and wet bulb temperatures as well as barometric pressure were also recorded.

The flow of primary air was metered by a thin plate orifice, 42/64 inch in diameter, installed at the approach of the heater. The orifice plate was located concentrically between two pipe flanges in a 1-1/4 inch pipe line. Vena Contracta taps, drilled 1-3/4 inch upstream and 5/8 inch downstream from the orifice face, were used to measure the differential pressure across the orifice. The flow of the secondary air was metered
using an orifice, 0.190 inch in diameter, installed in a 3/4 inch pipe line leading to the induction chamber. The construction and the installation of all gauges and meters were made in accordance with specifications outlined in the American Society of Mechanical Engineers Test Code.8

The distance between the primary nozzle and the secondary nozzle was made adjustable by means of an indexed metal bellows. The discharge pressure was varied by means of the discharge valve.

Fig. 18 shows the electrical circuit connections; the amount of heat added to the compressed primary air was controlled by varying the electrical output of the heating coils.

Fig. 19 gives the details of the primary and secondary nozzles. Both nozzles were machined from a length of stainless steel stock. Throat dimensions of both nozzles were held within ± 0.001 inch; inner surfaces of both nozzles were finished.

Due to the high temperatures encountered, all pipes in the ejector system were of the extrastrong schedule 80 class, and most of the piping connections were arc-welded. To prevent any appreciable heat loss, the heater tubes as well as the pipes were covered with high-temperature insulation.

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CHAPTER III

PROCEDURE

Upon the completion of the apparatus, and prior to the actual experimental tests, all pressure gauges, orifices, and thermocouples were calibrated in routine fashion. Some general preliminary runs were then made on the apparatus, covering almost the entire range of operating conditions, to determine the optimum $X/Z_1$ ratio (defined on p. vi). This was not included anywhere in this thesis since it is felt that the purpose of the thesis, as indicated in the title, is the "effects of primary temperature and discharge pressure," and not the $X/Z_1$ ratio, on the performance of an air ejector.

During the preliminary runs, it was discovered that the ejector would not operate at all around 100 p.s.i.a. as was originally intended, but that it would operate only in the range of 20 to 40 p.s.i.a. Thus, all the experimental runs were made in this range; an explanation will follow later.

Altogether, five sets of experimental runs were made on the apparatus with primary temperature varying from 95 degrees F. up to approximately 750 degrees F. During each set, that is, at each primary temperature, three or four subsets of tests were made by varying the primary pressure from 20 to 40 p.s.i.a. Finally, in each subset, that is, at every particular combination of primary temperature and pressure, three to five test points were taken, the variable this time being the discharge pressure.
The exact procedure of the experimental runs was as follows: The primary air was first heated to approximately the desired temperature; then it was maintained constant by means of the powerstat and the individual heater switches, see Fig. 18. After the primary temperature had reached a steady value, the primary pressure was then roughly adjusted by means of the pressure regulator, see Fig. 17. After this a test run was made.

First, the primary pressure was adjusted by means of the primary valve to the exact desired value. With the discharge valve wide open, the pressure and temperature at the primary orifice, the primary pressure and temperature, the discharge pressure and temperature, the primary and secondary orifice manometer difference and the room dry bulb and wet bulb temperatures were recorded. The discharge valve was then partially closed to build up a certain back pressure; and when flow had once more become steady, the above instruments were read again. This procedure was continued until the secondary manometer gauge difference had become zero or negligible. Thus, a "subset" of test runs was completed.

In proceeding to the next "subset," the primary pressure was readjusted as before, keeping the primary temperature constant in the meantime, and the process was repeated until a "set" of test runs was completed. The primary temperature was then adjusted to commence the next "set" of tests.
CHAPTER IV

DISCUSSION

Preliminary Design Considerations.—The simplicity of construction and assembly led to the choice of a central type ejector for this investigation. With considerations towards the capacity of existing air compressors in the Mechanical Engineering laboratory, as well as the high cost of heating the compressed air to approximately 1000 degrees Fahrenheit, a relatively small ejector system was designed, having a rate of flow of approximately 0.02 pounds of primary air per second.

Two general methods of mixing for ejectors have been studied, constant pressure mixing, and constant area mixing. Both one-dimensional and two-dimensional flow analysis have been investigated. Theoretical considerations as well as experimental results seem to indicate that the one-dimensional analysis has many serious limitations. The two-dimensional analysis, on the other hand, presents so many complex problems that it has so far not produced a satisfactory method of solution.

The design of the primary and secondary nozzles used in this thesis was based on a constant pressure mixing and a one-dimensional flow analysis.

12 See Appendix, pp. 38-44.
The actual design of the nozzles followed methods and recommendations suggested by Elrod, and by Keenan, Nuemann and Lustwerk.

Design of Primary Nozzle.---Referring to Fig. 1, repeated, the primary nozzle was assumed to be an ideal nozzle and was designed as follows:

![Diagram of Central Type Ejector](image)

Fig. 1, Repeated. Central Type Ejector.

from the ideal nozzle relations

\[ V_t = 224 \sqrt{h_i - h_t} \]

\[ V_x' = 224 \sqrt{h_i - h_x'} \] (1)

and the continuity equations

\[ A_t = \frac{w_t V_t}{V_t} \]

\[ A_x' = \frac{w_x' V_x'}{V_x'} \] (2)

The throat and exit dimensions of the primary nozzle, \( A_t \) and \( A_x' \), may thus be calculated from equations 1 and 2.

---


14Keenan, Nuemann and Lustwerk, op. cit., pp. A-300--A-309,
Secondary Nozzle Analysis.—The analysis of the secondary nozzle was based on a one-dimensional, constant pressure process.

The basic assumptions under this analysis were that there was no external heat transfer; that there was no wall friction; that all fluid characteristics were uniform across section 2, i.e., mixing was complete at that section; that a transverse compression shock may precede section 2 if the mixed stream was supersonic at section 1, and finally, that pressure was uniform and constant between sections x and 1.

For flow through the secondary nozzle, the following equations of constraint may be written:

continuity

\[ \Sigma \left( \frac{VA}{V} \right)_a = \Sigma \left( \frac{VA}{V} \right)_b \]  \hspace{1cm} (3)

conservation of energy

\[ \Sigma \left[ w(h + \frac{V^2}{2g}) \right]_a = \Sigma \left[ w(h + \frac{V^2}{2g}) \right]_b \]  \hspace{1cm} (4)

conservation of momentum

\[ \Sigma \left( \frac{W}{g} \right)_a - \Sigma \left( \frac{W}{g} \right)_b + (p_a - p_b)A_b = 0 \]  \hspace{1cm} (5)

In addition, primary and secondary fluids were both assumed to be perfect gases, so that

the perfect gas relations

\[ py = kT \]  \hspace{1cm} (6)

\[ h = c_pT \]  \hspace{1cm} (7)

An ideal secondary diffuser was assumed, so that

\[ \frac{P_3}{P_2} = (\frac{k-1}{2} M_2^2 + 1)^{\frac{k}{k-1}} \]  \hspace{1cm} (8)
When supersonic flow existed at section 1 and a transverse compression shock preceded section 2, the pressure rise between sections 1 and 2 may be computed by combining equations 3, 4, 5, 6 and 7 between sections 1 and 2

\[
\frac{p_2}{p_1} = \frac{2k}{k+1} M_1^2 - \frac{k-1}{k+1}
\]  

(9)

Keenan, Nuemann and Lustwerk\(^\text{15}\) combined equations 3, 4, 5, 6 and 7, along with the assumption that processes \(\omega\)-\(x\) and \(1\)-\(x\) were reversible adiabatic, to obtain the following equation

\[
v_1 = -\frac{b}{2} \left(1 + \sqrt{1 - \frac{4c}{b^2}}\right)
\]

(10)

where

\[
b = \frac{-2g \rho (T_x - T_o) + \frac{2gk}{k-1} \frac{p_x A_x}{w} V_x^N}{V_x^N - V_x^N + \frac{2gk}{k-1} \frac{p_x A_x}{w}}
\]

(11)

and

\[
c = \frac{2g \rho (T_1 V_x^N - T_0 V_x^N)}{V_x^N - V_x^N + \frac{2gk}{k-1} \frac{p_x A_x}{w}}
\]

(12)

Applying equation 5, conservation of momentum, between sections \(x\) and 1, along with the constant pressure mixing assumption, the following simple relation may be derived

\[
\omega V_x^N + V_x^N = (1 + \omega) V_1
\]

(13)

\(^\text{15}\)Keenan, Nuemann and Lustwerk, op. cit., p. A-301.
Design of Secondary Nozzle.--From the above flow analysis, the secondary nozzle throat may be designed by a method of trial and error.

Knowing the stagnation conditions of the primary and secondary fluids, as well as $p_x$ and $w^1$ from the design of the primary nozzle, $V_1$ may be solved from equation 10 by assuming a value of $A_1$.

$T_1$ may be solved by applying equations 4 and 7 between the initial states of the primary and secondary fluids and section 1.

Pressure after the transverse compression shock, $p_2$, may be calculated from equation 9.

$V_2$ and $T_2$ could be evaluated by applying equations 3, 4, 5, 6 and 7 between sections 1 and 2.

Finally, $p_3$ may be calculated from equation 8.

This process is repeated, by assuming different $A_1$'s, until the value of $p_3$ or $w$ corresponds to the desired discharge pressure or flow ratio, respectively.

Up to the present time, there seems to be no rational means for designing the shape of the passage walls between sections $x$ and 1.\(^{16}\) Consequently, the geometric configuration of the section $x-1$ for the ejector used in this thesis was arbitrarily designed as shown in Fig. 19.

Since it was assumed in the analysis that mixing was complete at section 2, the entrance to the diffuser, it was then necessary to determine the length required for complete mixing.

Experimental observations made by Keenan, Nueemann and Lustwerk\(^ {17}\) indicated that the length required between the junction of the two streams,


section x and the entrance to the diffuser, section 2, depended upon both the mixing process and the shock phenomena. When the mixed fluid stream was subsonic, the requirement of mixing dominated; when the mixed flow was supersonic, the requirement of shock was predominant.

Schlieren photographs by Keenan, Nuemann and Lustwerk\textsuperscript{18} gave some qualitative information regarding the effect of the shock in the mixing tube. In the afore-mentioned photographs it was shown that when the exhaust pressure $p_3$ was raised sufficiently, a transverse shock appeared at the downstream end of the constant area throat section of the secondary nozzle. When the exhaust pressure was further increased, the shock would penetrate to the secondary inlet section x-1. These photographs seemed to suggest that some definite length was required for a shock.

Other experimental works of Keenan, Nuemann and Lustwerk, as well as those of Flugel,\textsuperscript{19} seemed to indicate that in the case of supersonic mixed stream, the shock took place in the constant area throat when the secondary inlet, section x-1, was short. On the other hand, when the section x-1 was sufficiently long, the shock would appear in the downstream section of x-1, as well as in the constant area throat. In that case, pressure would begin to rise in the downstream section of x-1, and the mixing process along x-1 would no longer be one of constant pressure as assumed in this analysis.

Hence, from this observation Keenan, Nuemann and Lustwerk concluded that regardless of the position of the primary nozzle relative to section

\textsuperscript{18}Ibid., p. A-305.

\textsuperscript{19}G. Flugel, "Berechnung von Strahlapparaten," No. 395.
The total length from primary nozzle exit to section 2 for best performance was nearly a constant when the mixed stream was supersonic.

The ejector employed in this thesis was designed such that at section \( x \), the Mach Number of the primary fluid was \( M_x \approx 2 \); the mixed streams emerged from section 1, the beginning of the constant area throat, with a Mach Number slightly less than \( M_x \), but still very close to 2.\(^2\) Thus, somewhere between sections \( x \) and 2 a transverse compression shock would take place.

From the conclusions of Keenan, Nussmann and Lustwerk mentioned above, it seemed that there was no definite optimum value for the length of the constant area throat. They found, however, that over an extremely wide range of operating conditions, the value \( L/D_1 = 8 \) seemed to give most consistently the better test results. Based upon this recommendation, the constant area section of the ejector employed in this thesis was designed to have a \( L/D_1 \) ratio of 8, or a constant area throat length of 1.30 inches.

**Theoretical Performance and Efficiency.** A common application of the ejector is found in the vacuum refrigeration system where usually the primary fluid is steam at a pressure in the order of 100 p.s.i.a., and the secondary fluid is also steam but at a very low pressure, around one-tenth of one p.s.i.a. The discharge pressure of such a system is in the order of one p.s.i.a.

When an ejector system is used as a pump, the discharge contains both the actuating, or primary, fluid and the pumped, or secondary, fluid.

\(^2\)This is true because the flow ratio employed in this thesis is quite low, in the order of 0.1. This very small amount of secondary fluid did not slow down the primary stream appreciably. See Appendix, pp. 38-42.
The two streams of fluid are thoroughly mixed and have practically homogeneous thermodynamic and physical properties.

In such an ejector system, the work of compression, or the output, may be defined as the change in enthalpy, at constant entropy, of the secondary fluid, $\Delta h_2$, multiplied by the flow ratio $\omega$. A logical expression for the input would be the corresponding isentropic change in enthalpy, $\Delta h_1$, of the primary fluid. See Fig. 2. However, the expression for efficiency obtained from the above definitions of input and output

$$\eta_j = \frac{\omega \Delta h_2}{\Delta h_1}$$

would not represent either the mechanical or thermal efficiency of the apparatus. In order to obtain an appropriate expression for the thermal efficiency, it is necessary to evaluate the heat added to the primary fluid, which cannot be readily found since the primary fluid in general does not execute a cycle. In view of the lack of a definite expression for efficiency, the criterion of performance commonly accepted is the ratio of the secondary flow to primary flow, or the flow ratio. 21

---

In the case of most heat engines, there exists a standard comparison of performance easily evaluable from the thermodynamic states of the working fluid as it executes a cycle and from the physical proportions of the engine. For example, ideal efficiency of a spark ignition engine can be evaluated from the theoretical Otto Cycle; that of the steam engine or turbine, from the Rankine Cycle.

The ideal flow ratio of the ejector, however, cannot be evaluated very readily from the operating conditions and the geometric configurations of the apparatus. Since the process of mixing between the primary and the secondary fluids is not very well defined and its mechanisms not even well understood, the best one can attempt to do is to obtain an approximate prediction of performance.

Method of Solution for Calculation of Theoretical Flow Ratio.—In general, the calculation of the theoretical flow ratio makes use of all the equations derived previously, equations 1 through 13, inclusive.

For the ejector employed in this investigation, $T_1$, $T_0$, $A_x$, $w$, $p_1$ and $p_0$ are all given quantities for any particular run. The values of $p_x$ and $V_x$ can be evaluated by simultaneously solving the following three equations, assuming the perfect gas relations,
equation 2, continuity:

$$w = \frac{A_x V_x}{V_x}$$
equation 1, ideal nozzle relations:

$$V_x = 224 \sqrt{h_1 - h_x} = 14.9 \sqrt{p_1 V_1 - p_x V_x}$$
and the isentropic relations between sections i and x:

\[ p_i v_i^k = p_x v_x^k \]  

(15)

Assuming adiabatic reversible relations between sections o and x, \( v_x'' \) may now be calculated from \( p_o, v_o \) and \( p_x \).

After having evaluated \( p_x, v_x', v_x'' \), \( v_1 \) may be obtained from equations 10, 11 and 12:

Hence, from equation 18, \( \omega \) is evaluated.

Next, it is desired to determine the theoretical discharge pressure, \( p_3 \). Before this can be done, however, \( p_2 \) must be evaluated.

Applying the equation of conservation of energy between the initial states of the primary and secondary fluids and the state at section 1, assuming also the perfect gas relations, the following equation is derived

\[ c_p(T_1 + \omega T_o) = (1 + \omega) \left( c_p T_1 + \frac{V_1^2}{2k} \right) \]  

(16)

In equation 16, since \( T_1 \) and \( T_o \) are both given quantities and since \( V_1 \) has already been evaluated, \( T_1 \) may be calculated.

The remaining steps in the calculation of \( p_3 \) are the same as those employed in the design of the secondary nozzle shown on p. 12.
CHAPTER V

RESULTS

The test data for the ejector are given in Tables 1 through 7. The results of the experimental runs are shown in a series of curves of a summary nature, Figures 3 through 10.

Figures 3, 5 and 7 show the relations between flow ratio \( \omega \) and primary temperature \( T_1 \), with discharge pressure as the parameter, for average primary pressures of 33.2, 28.9 and 24.8 p.s.i.a., respectively. Figure 3 reveals that for any discharge pressure, the flow ratio increases monotonically with the primary temperature; although there seems to be a tendency for the slopes of the curves to flatten out at the higher primary temperatures. Thus it is believed that if the primary temperature had been raised sufficiently, say to 1500 degrees R., all the \( \omega \) versus \( T_1 \) plots would have reached a maximum. In the cases of primary pressures of 28.9 and 24.8 p.s.i.a., all the \( \omega \) versus \( T_1 \) plots did reach a maximum value. Where the primary pressure is 28.9 p.s.i.a., the maximum \( \omega \) for all values of discharge pressures occur at around 850 degrees R., see Fig. 5; while for average primary pressures of 24.8 p.s.i.a., the maximum flow ratios are found to be at approximately 800 degrees R., see Fig. 7.

In Figures 4, 6 and 8, flow ratio is plotted against the discharge pressure with primary temperature being the parameter, and again for average primary pressures of 33.2, 28.9 and 24.8 p.s.i.a., respectively. In all three cases, the monotonically decreasing trend of the flow ratio with increasing discharge pressure at all primary temperatures is clearly
indicated. The general shape and trend of these plots are quite consistent with the analytical curves by Keenan, Neuwan and Lustwerk for the case of a no-shock diffuser. This is more or less expected since due to the shocks in the primary nozzle, the flow in the secondary nozzle during the test runs was predominantly subsonic, resulting in a shock-less diffuser.

Based on the assumption of reversible adiabatic flow and not foreseeing any shock phenomenon, the primary nozzle was designed for full expansion; and the secondary nozzle was designed on the basis of constant pressure mixing followed by a transverse compression shock in the constant-area throat. Thus, a method for calculating the performance analytically was introduced, see pp. 16 through 17. The dimensions of the primary nozzle were designed to that the ratio of the exit pressure $p_x$ to initial $p_1$, $p_x/p_1$, was a constant equal to 0.14. During the preliminary runs it was discovered that for primary pressures above 40 p.s.i.a., the receiver pressure was always above atmosphere with primary temperatures ranging from 100 degrees F. up to 1200 degrees F. This definitely established the fact that a shock was present in the primary nozzle. The shock phenomenon necessitated, not only in the operation of the test apparatus at primary pressures far below the design value of 100 p.s.i.a., but also in a modification of the method of solution, originally derived for the theoretical calculation of performance, to account for the effect on the primary exit velocity due to the shock. On pages 43 and 44, in the Appendix, is shown the revised method of solution. Unfortunately, even this still leaves the analysis somewhat questionable, since the shock in the primary nozzle not only affects the primary exit velocity and the induction chamber pressure, but also due to its spreading and reflecting characteristics downstream.
from the primary nozzle exit, it renders the mixing process one of extreme complexity. Just how closely the actual mixing process is approximated by the simple one-dimensional constant pressure analysis merits serious consideration.

In Fig. 11 is shown a comparison between calculated performance based on the modified method of solution on pages 43 and 44 and the actual test performance for primary temperatures of 680 and 880 degrees Rankine. It is seen that at a given value of flow ratio, the test value of $p_3/p_1$ is approximately 90 per cent of the calculated value. This seems to indicate that the ejector is quite satisfactory as a subsonic fluid compressor, even though it did not operate in the supersonic range.

Figure 10 shows the relationship between the discharge pressure at zero secondary flow and the primary temperature, with primary pressure being the parameter. Here it is seen that at any particular primary temperature, the highest primary pressure gives the highest value of discharge pressure at no secondary flow. The two curves for which the average primary pressures equal 24.8 and 28.9 p.s.i.a. also display definite maxima points, while the one at the average primary pressure of 33.2 p.s.i.a. seems to approach a maximum value slightly beyond the upper limit of the primary temperature tested in this investigation.

Figure 9 is quite similar to Figure 10 except, in this case, the flow ratio at zero discharge pressure is plotted against primary temperature, again with primary pressure being the parameter. As before, definite maximum values of flow ratios are shown by the curves for which the average primary pressures are 24.8 and 28.9 p.s.i.a., and a decreasing slope for the one with average primary pressure equal to 33.2 p.s.i.a.
In Figure 12, the primary stagnation temperature is plotted against the primary exit temperature calculated from equation 18 for all three average values of primary pressures. For maximum thermal reversibility the primary exit temperature should be equal to the secondary temperature at the beginning of the mixing zone which is approximately 550 degrees R. The three values of primary stagnation temperature for which the exit temperature is 550 degrees R. for average primary pressures of 24.8, 28.9 and 33.2 p.s.i.a. are 585, 610 and 630 degrees R., respectively. Although these temperatures do not correspond to the test values for optimum performance closely at all, it nevertheless substantiates the test trend that for best performance, the primary temperature goes up with primary pressure.
Figure 3

\( V_s \) vs. \( T_f \) - Average \( P_c = 33.2 \) P.S.I.A.
Figure 4

$\omega$ Vs. $P_3$

Average $P = 33.2$ P.S.I.A.
Figure 5
W Vs. \( T_i \)
Average \( P_i = 28.9 \text{PSI.A.} \)
Figure 6

$\omega$ Vs. $P_3$

Average $P_i = 28.9$ PSIA.
Figure 7, \( W \) vs. \( T_i \)

Average \( P_i = 24.8 \) P.S.I.A.
Figure 8, \( W \) Vs. \( P_3 \)

Average \( P_c = 24.8 \) P.S.I.A.
Fig. 10. Discharge Pressure At Zero Secondary Flow Vs. Primary Temperature

$P_3$ in 1 P.S.I.A.

$T_i$ in $10^2$ Degrees R.

Av. $P_i = 33.2$ P.S.I.A.
Fig. 11. Theoretical and Actual Performance

$T_i = 680^\circ R.$

$T_i = 880^\circ R.$
FIG. 12. PRIMARY EXIT TEMPERATURE
FOR MAXIMUM REVERSIBILITY
CONCLUSIONS AND RECOMMENDATIONS

As summarized in Chapter V, the test results clearly indicate the limitations and shortcomings of the method of analysis employed in the design of the ejector. So far, with an analysis based on one-dimensional flow, pressure and velocity distributions are assumed to be uniform across section x of the primary nozzle and across section 1 of the secondary nozzle. In an exact analysis, the spreading of a jet is a complex problem involving a two-dimensional analysis. Golstein has shown that, while the pressure at the mouth of the primary nozzle may be assumed to be uniform, the velocity distribution is not at all uniform. For both the laminar and turbulent cases, velocity distribution and the mass flux of a jet mixing with a relatively still stream of fluid have been solved analytically by Golstein and Pai employing a method very similar to the Blasius solution of the boundary layer theory.

The primary nozzle was designed to operate at a receiver pressure corresponding to the fully expanded conditions at the nozzle exit. This receiver pressure, see state point x1 in Fig. 13, is unique for a given

---


24 Ibid., pp. 592-599.


primary nozzle and a particular set of stagnation primary fluid conditions.

For any other receiver pressures, flow through the nozzle assumes various conditions. With receiver pressure between 1 and \( x_1 \), flow is subsonic throughout; at \( x_1 \) flow is sonic at the throat and subsonic throughout other sections of the nozzle; between \( x_1 \) and \( x_4 \) the flow becomes quite complex, being sonic at the throat. Due to its inertia, the fluid would assume supersonic velocities beyond the throat at first; but at a certain place in the diverging part of the nozzle a shock front intervenes, the gas is compressed and slowed down. The position and strength of the shock front are automatically adjusted so that the end pressure at the exit becomes the receiver pressure. These shock fronts, in the simplest description, would be curved discs across the nozzle perpendicular to the nozzle wall. Actually the shock front is oblique and consequently changes the direction of the flow abruptly, leading to jet detachment.\(^{27}\) The situation is represented in Fig. 14.

When the receiver pressure is lowered from \( x_1 \) to \( x_2 \), the shock front moves from the throat towards the exit. At \( x_2 \) the place of detachment moves to the rim of the nozzle and remains there as the receiver pressure becomes less than \( x_2 \) while the shock front leaving the rim becomes longer, see Fig. 15.
If the receiver pressure is decreased to exactly $x_r$, the strength of the shock leaving the rim becomes zero and fully expanded flow prevails. When the receiver pressure is further decreased below $x_r$, a new set of extremely complex phenomena begins, but since this condition was not encountered any time during this investigation, an explanation is omitted here.

The experimental data show that on all the runs made on the test apparatus, the receiver pressure was between the limits $x_1$ and $x_3$ where a shock wave always occurred somewhere in the diverging section of the primary nozzle. Never once was fully expanded flow obtained. At primary pressures above 40 p.s.i.a. the ejector would not operate at all simply because the receiver pressure would rise above atmospheric and cause part of the primary fluid to escape through the secondary inlet. Now the questions arise as to whether the receiver pressure could be controlled at all and, secondly, what caused it to assume the values it did during the test?

In answer to the first question the receiver pressure is determined by the flow through the secondary nozzle, and there is no mechanical means of adjusting this pressure. As for the second question, since the secondary nozzle throat employed in this thesis was only slightly larger than the primary nozzle throat, and since the proportions of the induction chamber were very much larger compared to the dimensions of the nozzles, there is a tendency for the primary fluid to reach stagnant conditions, and consequently a high pressure, in the chamber and re-expand through the secondary nozzle, the action being somewhat similar to the conditions in certain sections of a wind tunnel. Yet the secondary nozzle was designed on a constant pressure analysis suggested by Keenan, Nuemann and Lustwerk. This again points out some of the inadequacies of the one-dimensional analysis.
Experiments by the afore-mentioned authors on actual ejectors of relatively large secondary nozzle throat to primary nozzle throat ratios the design of which were based on the one-dimensional constant pressure analysis have produced results as high as eighty-five per cent of the theoretical calculations. Evidently the one-dimensional constant pressure analysis is only valid for relatively large secondary nozzle throats. Due to the very complex shock waves that form in the primary nozzle and in the mixing zone, or the converging section, of the secondary nozzle in the ejector employed in this investigation, a more elaborate, two-dimensional analysis must be formulated in order to obtain a reasonably rational method of solution.

Until such a satisfactory analysis has been formulated, the only recommendations that the author can suggest at the present time are:

1. Reconsider the construction of the test apparatus; in particular, instrumentation and finish for both the induction chamber and the nozzle sections should be improved.

2. Build various secondary nozzles with larger throat areas and thus determine the degree of agreement between analysis and performance by test.

3. Study the characteristics of the ejector at even higher primary temperatures, say up to 1800 degrees Rankine.
Design Calculations

Fig. 1, Repeated. Central Type Ejector.

The initial conditions of the primary and secondary air are given as follows:

\[ P_1 = 100 \text{ p.s.i.a.} \quad P_0 = 14.2 \text{ p.s.i.a.} \]
\[ T_1 = 1460^\circ \text{ R.} \quad T_0 = 540^\circ \text{ R.} \]

A primary nozzle is to be designed, having an exit pressure \( P_x \) of 14.0 p.s.i.a.

Assuming a perfect nozzle and reversible adiabatic expansion, throat velocity may be found by equation 1

\[ V_t = 224 \sqrt{h_{1} - h_t} \]

where \( h_t \) is the enthalpy of air corresponding to \( P_t \) and \( u_t \), and where

\[ P_t = 0.53 \cdot P_1 \]

From gas tables, \( h_1 = 274.0 \text{ B.T.U./lb.} \), \( h_t = 214.5 \text{ B.T.U./lb.} \),
\[ v_t = 8.65 \text{ cu. ft./lb.}, \text{ and } T_t = 1230^\circ \text{ R.} \]

From equation 1

\[ V_t = 224 \sqrt{274.0 - 214.5} = 1720 \text{ ft./sec.} \]
A primary throat diameter of 1/8 inch is desired; equation 2 gives

\[ A_t = \frac{w^t v_t}{V_t} \]

\[ \frac{\pi (\frac{d}{8})^2}{4 \times 144} = \frac{v^t(3.65)}{1720} \]

\[ w^t = 0.0172 \text{ lb./sec.} \]

To calculate velocity at exit, again using equation

\[ v_x^t = 224 \sqrt{\frac{274.6 - h_x^t}{s_x^t}} \]

where \( h_x^t \) is the enthalpy corresponding to \( p_x^t \) and \( s_x^t \).

From gas tables, \( h_x^t = 124 \) B.T.U./lb., \( v_x^t = 22.1 \) cu. ft./lb. and \( T_x^t = 855^\circ \text{R.} \)

\[ v_x^t = 224 \sqrt{274.6 - 124} = 2780 \text{ ft./sec.} \]

Again from equation 2

\[ A_x^t = \frac{w^t v_x^t}{V_x^t} = \frac{0.0172(22.1)(144)}{2780} = 2.01 \times 10^{-2} \text{ sq. in.} \]

or

\[ D_x^t = \frac{4A_x^t}{\pi} = \frac{4(2.01 \times 10^{-2})}{3.14} = 0.169 \text{ in.} \]

The diverging section t-x is designed to have an included angle of 7 degrees.

Length of section t-x = \( \frac{D_x^t - D_t}{\tan 3.5^\circ} \)
\[ t-x = \frac{0.169 - 0.125}{0.061} = 0.288 \text{ in.} \]

For the design of the secondary nozzle throat, in addition to all the given quantities carried over from the design of the primary nozzle, it is further stipulated that a flow ratio of approximately 0.1 is desired.

Following the procedure outlined in Chapter IV, a secondary throat diameter of \(11/64\) inch is assumed.

\[ A_1 = A_2 = \left(\frac{11}{64}\right)^2 \frac{\pi}{4} = 0.0233 \text{ sq. in.} \]

\(V_x''\) may be determined from equation 1 and 15 applied between sections 0 and x.

**equation 15**

\[ V_x'' = V_0 \left(\frac{P_0}{P_x}\right)^{\frac{1}{k}} \]

\[ V_x'' = 14\left(\frac{14.2}{14.0}\right)^{\frac{1}{k}} = 14.14 \text{ cu. ft./lb.} \]

**equation 1**

\[ V_x'' = 14.9 \sqrt{\frac{P_0 V_0}{P_x V_x''}} \]

\[ V_x'' = 14.9 \sqrt{\frac{144(14.2)(14.0)}{144(14.0)(14.14)}} \]

\[ V_x'' = 160 \text{ ft./sec.} \]

Now \(V_1\) may be determined from equations 10, 11 and 12.

**equation 10**

\[ V_1 = \frac{b}{2} \left(1 + \sqrt{1 - \frac{4c}{b^2}}\right) \]
\[
\begin{align*}
\frac{b}{2780 - 160 + \frac{64.4(1.4)}{1.4 - 1} \frac{14(0.0233)}{0.0172}} &= -1675 \\
b &= \frac{64.4(187)(1460 - 540) + 64.4(1.4) \frac{14(0.0233)}{0.0172} 160}{2780 - 160 + \frac{64.4(1.4)}{1.4 - 1} \frac{14(0.0233)}{0.0172}} \\
\frac{c}{2780 - 160 + \frac{64.4(1.4)}{1.4 - 1} \frac{14(0.0233)}{0.0172}} &= -2.21 \times 10^6 \\
V_1 &= \frac{1675}{2} \left[ 1 + \sqrt{1 + \frac{4(2.21 \times 10^6)}{(1675)^2}} \right] = 2360 \text{ ft./sec.}
\end{align*}
\]

\( \omega \) may now be solved from equation 13.

\[
\omega = \frac{2780 - 2560}{2560 - 160} = 0.092
\]

This value of \( \omega = 0.092 \) is quite close to 0.1; therefore, the assumed throat area is satisfactory.

\( T_1 \) may be determined from equation 15.

\[
T_1 = 1460 - 0.092(540) - \frac{(2560)^2}{64.4(187)} = 865^\circ \text{ R.}
\]

\[
M_1 = \frac{V_1}{\sqrt{2gR_T}} = \frac{2560}{\sqrt{1.4(32.2)(53.3)(865)}} = 1.775
\]

The pressure after shock, \( P_2 \), is evaluated from equation 9.

\[
P_2 = 14 \left[ \frac{2.8}{2.4} (1.775)^2 - \frac{0.4}{2.4} \right] = 49.2 \text{ p.s.i.a.}
\]

Applying equation 5, conservation of momentum between sections 1 and 2, \( V_2 \) may be solved.
\[ V_2 = V_1 \frac{(p_2 - p_1)A_2g}{w^2(1 + \omega^2)} \]

\[ V_2 = 2560 - \frac{(49.2 - 14)(0.0233)(32.2)}{0.6172(1 + 0.092)} = 1155 \text{ ft./sec.} \]

By combining equations 3 and 6, continuity and perfect gas relation, respectively, between sections 1 and 2, \( T_2 \) may be expressed as

\[ T_2 = \frac{p_2V_2T_1}{p_1V_1} \]

\[ T_2 = \frac{49.2(1155)(86)}{14(2560)} = 1370^\circ \text{ R.} \]

Hence,

\[ M_2 = \frac{V_2}{\sqrt{\gamma R T_2}} = \frac{1155}{\sqrt{1.4(32.2)(33.3)(1370)}} = 0.636 \]

Finally, \( p_3 \), the discharge pressure, may be determined from equation 8.

\[ p_3 = 49.2 \left[ \frac{1.4}{2} - \frac{1}{2}(0.638)^2 + 1 \right]^{\frac{1.4}{1.4 - 1}} \]

\[ p_3 = 64.7 \text{ p.s.i.a.} \]
Revised Method of Solution

When a compression shock wave prevails within the primary nozzle, the method of solution for calculation of theoretical flow ratio must be modified since the exit velocity of the primary fluid is now no longer determined by the completely reversible adiabatic process.

![Diagram of shock processes in primary nozzle](image)

Fig. 16. Shock Process in Primary Nozzle.

In Fig. 16, let state b be the one at which the shock begins, and let state e be the one at which it ends. The processes 1-b and e-x are assumed to be isentropic.

When the exit pressure $p_x$ lies between the limits $x_1$ and $x_3$, the shock is within the diverging portion of the nozzle, and the actual cross-sectional area of the jet leaving the nozzle may be considered as equal to the exit area of the nozzle. However, when the exit pressure is below $x_3$, the actual cross-sectional area may no longer be regarded as the same as the exit area of the nozzle due to jet detachment. For air the ratio

---

\( P_{x3}/P_1 \) is approximately 0.43.

In the test runs made on the ejector employed in this thesis, the exit, or chamber, pressure did not vary appreciably at all, the average value being approximately 14.0 p.s.i.a. The highest primary pressure was 33.2 p.s.i.a., giving a \( P_{x3}/P_1 \) ratio of 0.423 which is not much below the limiting ratio of 0.43. Thus the exit velocity of the primary nozzle can be calculated on a fairly simple basis.

By combining the expression for critical flow in a converging diverging nozzle

\[
\nu' = A' \frac{2}{(k + 1)} \frac{1}{k - 1} \sqrt{\frac{2gk}{R T_1(k + 1)}} \tag{17}
\]

with equation 3, continuity, the following equation is obtained:

\[
A' \frac{2}{(k + 1)} \frac{1}{k - 1} \sqrt{\frac{2gk}{R T_1(k + 1)}} = \frac{A_x' \nu_x' \nu_x}{R \left( T_1 - \frac{\nu_x^2}{2g c_p} \right)} \tag{18}
\]

Since the initial condition of the primary fluid as well as the exit pressure are known, \( \nu_x' \) may be determined from the above equation.

From here on the procedure is essentially the same as outlined in Chapter III since the mixing process is still assumed to be one-dimensional constant pressure. The only modification necessary is due to the subsonic flow in the secondary nozzle. There is no longer a transverse compression shock between sections 1 and 2 and the two sections may be considered to coincide.
Sample Calculation of Performance Based on the Modified Method of Solution

The following is the calculated performance for primary pressure of 24.8 p.s.i.a., primary temperature of 680 degrees Rankine and secondary stagnation temperature of 555 degrees Rankine.

Equation 17 gives the mass rate of primary flow

\[ w^* = \frac{A_1 p_1 \left( \frac{2}{k+1} \right)^{k-1}}{\sqrt{\frac{2gk}{\gamma(\gamma+1)}}} \]

\[ w^* = 0.0122 \times 24.8 \frac{1}{1.4 + 1} \frac{1}{1.4 - 1} \frac{1}{1.4 + 1} \frac{-2}{\sqrt{\frac{2}{3} \times 32.2 \times 1.4}} \frac{1}{\sqrt{53.3 \times 680(1.4 + 1)}} \]

\[ w^* = 6.45 \times 10^{-3} \text{ lb./sec.} \]

whereby equation 18 may be rearranged to solve for \( V_x \)

\[ V_x^* = \frac{-A_x \gamma P_x \sqrt{(A_x \beta_x)^2 + \frac{4(v' R_x)^2 \gamma T_x}{2g_p}}} \]

\[ V_x^* = \frac{-0.0201 \times 14.0 + \sqrt{(0.0201 \times 14)^2 + \frac{4(6.45 \times 10^{-3} \times 53.3)^2 \times 580}{2 \times 32.2 \times 187}}} \]

\[ V_x^* = 780 \text{ ft./sec.} \]

During the tests it was found that the induction chamber pressure did not vary appreciably from 14 p.s.i.a.; thus the chamber pressure is to be taken as 14.0 p.s.i.a.
Hereafter, $V_1$, $\omega$, $M_1$ and $p_3$ are calculated in exactly the same manner as shown on pp. 40 through 42; consequently, they are omitted here.
EXPERIMENTAL DATA

Barometric pressure: 29.02 in. Hg  Ambient dry bulb temperature: 95° F.

X/D₁ ratio: 8.7  Ambient wet bulb temperature: 77° F.

Table 1. Discharge Pressure-Flow Ratio Survey Data, T₁ = 555° R.

<table>
<thead>
<tr>
<th>P₁</th>
<th>P₃</th>
<th>T₃</th>
<th>w'</th>
<th>w''</th>
<th>ω</th>
</tr>
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<tr>
<td>p.s.i.a.</td>
<td>p.s.i.g.</td>
<td>°R.</td>
<td>10⁻³ lb./sec.</td>
<td>10⁻³ lb./sec.</td>
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<td>0.00</td>
<td>555</td>
<td>10.00</td>
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<td>0.0361</td>
</tr>
<tr>
<td>32.1</td>
<td>0.25</td>
<td>555</td>
<td>10.00</td>
<td>0.160</td>
<td>0.0160</td>
</tr>
<tr>
<td>29.3</td>
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<td>9.07</td>
<td>0.976</td>
<td>0.108</td>
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<td>9.07</td>
<td>0.814</td>
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</tr>
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Table 2. Discharge Pressure-Flow Ratio Survey Data, $T_1 = 680^\circ$ R.

<table>
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<th>$P_3$</th>
<th>$T_3$</th>
<th>$w^1$</th>
<th>$w^2$</th>
<th>$\omega$</th>
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<td>p.s.i.a.</td>
<td>p.s.i.g.</td>
<td>°R</td>
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<td>$10^{-3}$ lb./sec.</td>
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</table>
Table 3. Discharge Pressure-Flow Ratio Survey Data, $T_1 = 800^\circ$ R.

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<tr>
<th>$P_1$</th>
<th>$P_3$</th>
<th>$T_3$</th>
<th>$w'$</th>
<th>$w''$</th>
<th>$\omega$</th>
</tr>
</thead>
<tbody>
<tr>
<td>p.s.i.a.</td>
<td>p.s.i.g.</td>
<td>$^\circ$R.</td>
<td>$10^{-3}$ lb./sec.</td>
<td>$10^{-3}$ lb./sec.</td>
<td></td>
</tr>
<tr>
<td>28.5</td>
<td>0.00</td>
<td>750</td>
<td>6.05</td>
<td>1.84</td>
<td>0.304</td>
</tr>
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<td>755</td>
<td>6.05</td>
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<td>0.0821</td>
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</tr>
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<td>750</td>
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</table>
Table 4: Discharge Pressure-Flow Ratio Survey Data, $T_1 = 885^\circ R$.

<table>
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<tr>
<th>$P_1$ (p.s.i.a.)</th>
<th>$P_3$ (p.s.i.g.)</th>
<th>$T_3$ ($^\circ R$)</th>
<th>$w'$ ($10^{-3}$ lb./sec)</th>
<th>$w''$ ($10^{-3}$ lb./sec)</th>
<th>$\omega$</th>
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<tbody>
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<td>0.133</td>
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<td>0.638</td>
<td>0.0950</td>
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<tr>
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<td>1.10</td>
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<td>6.65</td>
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<td>0.0553</td>
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<td>845</td>
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<td>0</td>
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<td>0.00</td>
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<td>0.304</td>
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<td>1120</td>
<td>0</td>
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<tr>
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<td>0.005</td>
<td>99.7</td>
<td>0</td>
<td>1120</td>
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<tr>
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<td>99.7</td>
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<td>1120</td>
<td>0</td>
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<tr>
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<td>1120</td>
<td>0</td>
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<td>0.15</td>
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<td>0.3</td>
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</tr>
</tbody>
</table>

**Table 2.** Discharge Pressure Flow Ratio Survey Data, \( P = 9900 \) m.
Table 6. Discharge Pressure-Flow Ratio Survey Data, $T_1 = 1085^\circ$ R.

<table>
<thead>
<tr>
<th>$P_1$</th>
<th>$P_3$</th>
<th>$T_3$</th>
<th>$w'$</th>
<th>$w''$</th>
<th>$\omega$</th>
</tr>
</thead>
<tbody>
<tr>
<td>p.s.i.a.</td>
<td>p.s.i.g.</td>
<td>°R.</td>
<td>$10^{-3}$ lb./sec.</td>
<td>$10^{-3}$ lb./sec.</td>
<td></td>
</tr>
<tr>
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<td>0.00</td>
<td>979</td>
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<td>1.43</td>
<td>0.249</td>
</tr>
<tr>
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<td>1.00</td>
<td>975</td>
<td>5.74</td>
<td>0.997</td>
<td>0.174</td>
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<td>1.75</td>
<td>980</td>
<td>5.88</td>
<td>0.521</td>
<td>0.0886</td>
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<td>2.00</td>
<td>985</td>
<td>5.88</td>
<td>0.155</td>
<td>0.0264</td>
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<tr>
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<td>0.00</td>
<td>985</td>
<td>5.20</td>
<td>0.968</td>
<td>0.186</td>
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<tr>
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<td>990</td>
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<td>990</td>
<td>5.12</td>
<td>0.0945</td>
<td>0.0185</td>
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<td>1.54</td>
<td>0.358</td>
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<td>985</td>
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<td>0.920</td>
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<td>990</td>
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</table>
Table 7. Discharge Pressure-Flow Ratio Survey Data, $T_1 = 1170^\circ$ R.

<table>
<thead>
<tr>
<th>$p_1$ (p.s.i.a.)</th>
<th>$p_3$ (p.s.i.g.)</th>
<th>$T_3$ ($^\circ$R)</th>
<th>$w^*$ ($10^{-3}$ lb./sec.)</th>
<th>$w''$ ($10^{-3}$ lb./sec.)</th>
<th>$\omega$</th>
</tr>
</thead>
<tbody>
<tr>
<td>33.0</td>
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<tr>
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<td>5.44</td>
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<td>0.490</td>
<td>0.108</td>
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</tbody>
</table>
Figure 17: Schematic Layout of Apparatus

Georgia Institute of Technology
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Figure 18: Electrical Circuit Connections
Figure 19
Primary and Secondary Nozzles

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Drawn by: John Doe, July 9, 1952
BIBLIOGRAPHY


