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HIGH SUBSONIC FLOW TESTS OF A PARALLEL PIPE FOLLOWED BY A LARGE AREA RATIO DIFFUSER

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HIGH SUBSONIC FLOW TESTS OF A PARALLEL PIPE FOLLOWED BY A LARGE AREA RATIO DIFFUSER

By

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SUMMARY

Experiments were performed on a pilot model duct system in order to explore its aerodynamic characteristics. The model was scaled from a design projected for the high speed operation mode of the Aircraft Noise Reduction Laboratory located at NASA Langley Research Center. The test results show that the model performed satisfactorily and therefore the projected design will most likely meet the specifications. The experiments were conducted in the Aerodynamic Laboratory of Old Dominion University.

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LIST OF SYMBOLS

A = sectional area, m^2

a_o = throat area of Venturi, m^2

c_p = diffuser pressure recovery factor

c_d = coefficient of discharge

D = diameter of the parallel pipe, m

$\overline{f}$ = average friction factor along parallel pipe

g = gravitational acceleration, m/sec^2

K = specific heat ratio

ΔL = length of pipe section under consideration

M = Mach number measured along parallel pipe and diffuser

p = absolute pressure measured along parallel pipe and diffuser, kg/m

Δp = pressure difference in Venturi, kgf/m^2

Q = volumetric flow rate, m^3/sec

R = gas constant, joule/kg °R

T = stagnation temperature, °R

V = velocity, m/sec

$\dot{W}$ = flow rate, kg/sec

x = distance measured along parallel pipe and diffuser

y = distance across the flow, m

γ = specific weight, kgf/m^3

θ = static temperature

ρ = fluid density, kg/m^3

ρ_o = fluid density at isentropic stagnation conditions, kg/m^3

Subscripts

$_1$ = inlet to a section

$_2$ = outlet

INTRODUCTION

The experiments presented in this report were planned to explore the flow behavior of an air duct system scaled down to approximately a one-tenth* size pilot model of the prototype. The prototype itself is a duct system projected for the Acoustic Noise Reduction Laboratory at NASA Langley Research Center. The three main components of this system are an air intake, a parallel pipe, and a diffuser. The flow entering the air intake is to be accelerated in transit to a Mach number of about 0.5 and is to be further accelerated along the parallel pipe to unity (sonic condition) at pipe exit where the flow is permitted to enter the diffuser for pressure recovery. A large centrifugal blower is projected to perform the necessary flow induction, and it is anticipated that the model operation will furnish satisfactory data for estimating the performance of the prototype.

Preliminary studies indicate that there arises no particular problem with the intake and a sensible design will satisfy the need for a uniform distribution of flow to the parallel pipe. The requirement of attaining a Mach number 0.7 along the pipe, and preferably being able to increase it to unity at pipe exit, calls for attention considering the variation of friction factor along a parallel pipe anticipated under highly accelerating flow conditions. Furthermore, the unconventional design of the diffuser with a proposed area ratio of 16 to 1 appeared highly problematical from a recovery point of view. In addition, the effects on the flow of an instrument-mounting employed inside the parallel pipe appears theoretically unpredictable. Finally, all of these effects combined presents the problem of matching the pressure changes and flow rates required from the system against the actual quantities being available by the induction under operational conditions.

TEST EQUIPMENT

The test equipment essentially consisted of a parallel pipe which was provided with a well-rounded entry at its inlet and was connected to a diffuser at its outlet.

* The actual scaling factor was 1:9.6.

Located upstream from the entry a venturi meter of standard design was employed for measuring the flow rate. The general arrangement is shown in figure 1a.

The parallel pipe was made of extruded acrylic tubing with approximately .00476 m wall thickness and was fitted with flanges at each end. To establish the exact location of the section where diffusion started, the end of the pipe was slightly tapered from about one inch distance from its exit plane. Pressure tappings were distributed at regular spacing along the pipe but the distance between tappings was decreased towards the pipe exit where large pressure gradients were anticipated. Some sections were provided with a single tapping while others had three taps which were interconnected, thus allowing for the average of three pressures to be recorded in any one plane of measurement.

While there was only one pipe employed throughout the experiments, there were two diffusers tested. The first diffuser was made of plastic material and had straight walls with a taper angle of 3 degrees, while the second diffuser was made of sheet metal. This started with a lower taper angle of 2.4 degrees which changed to a higher angle of 4.76 degrees halfway along the diffuser length, as shown in figure 1b. Pressure tappings were distributed along the diffusers at regular spacing.

Tests were conducted under two different operational modes: (a) under suction, and (b) under pressure, and the test arrangement was changed accordingly. Under suction, the flow was "inducted" and operation was maintained with the aid of a vacuum system consisting of an air ejector to which the primary air was supplied by a compressor. The ejector was specially designed for these tests to reduce pressure to about 22.86 cm mercury below atmosphere inside two tanks each of 28.317 m^3 capacity. Evacuation time to the desired vacuum lasted for about 15 minutes. Under suction operation all pressures along the tube and diffuser were measured with a multitube manometer, while the suction pressure in the vacuum line was read on a single tube manometer, both manometers using mercury as indicating fluid (see fig. 2a). Under pressure, flow through the system was maintained by a powerful centrifugal blower which discharged into a settling chamber where the pressure could be varied from .0508 m to .254 m water gage. Fluctuations of the airstream were damped out with honeycomb screens. From the side of the settling chamber the flow entered the parallel pipe through the inlet section and was finally discharged into the atmosphere through the diffuser (see fig. 2b). Under the pressure operation, velocity traverses across various cross sections of the pipe and the diffusers were obtained with a standard pitot static tube of .001587 m diameter.

For velocity traverses and for measuring the flow rate, inclined tube manometers were employed using alcohol as indicating fluid. Atmospheric temperature and pressure were read from a standard barometer located near the test equipment.

For the boundary layer suction experiments in the second diffuser a small blower unit was used and attempts at sucking the layer were made both at diffuser inlet and at the halfway section. The flow rate was measured with a small Venturi meter built into the suction line.

PROCEDURE

The experiments were performed under either quasi-steady or fully steady flow conditions. More particularly, under suction the induced airflow into the tanks through the pipe was greater than the evacuating volume flow, because the compressor was unable to maintain an adequate supply of primary air to the ejector under decreasing vacuum. However, when the flow attained sonic speed at the pipe exit the pressure distribution along the pipe remained unchanged for some short period of time, long enough for it to be considered steady, while under subsonic flow conditions at pipe exit the pressures were steadily rising. In either case, the pressure distribution was instantaneously recorded by photographing the manometer board. On the other hand, fully steady operation was maintained when operating under the pressure mode.

For the suction mode experiments the following routine applied. Air was withdrawn from the closed tank until a maximum vacuum of about 23.368 cm Hg was attained. Each test was started by quickly opening the admission valve to the tank and immediately observing the manometer board. When the desired flow conditions were attained the board was photographed and simultaneously the suction pressure and the pressure differential of the Venturi were recorded. The tests were subsequently repeated for several different suction pressures. Immediately after each test the admission valve was closed and remained closed until maximum vacuum in the tank was restored.

Several test runs were performed with unrestricted flow through the pipe and one run was performed with a protruberance that restricted the passage at a particular cross section. It was located .2794 m from the pipe entrance. The protruberance was shaped after an instrument holding device and it simulated partial blockage.

During the pressure mode experiments, velocity traverses were made at the relevant points along the flow--at the inlet to the parallel pipe, at the inlet to the diffuser, at the half-way section of the diffuser, and at the exit. These tests were subsequently repeated with boundary layer suction which was applied to either the inlet to the diffuser or the half-way section of the diffuser. During these experiments the effects resulting from varying the suction flow rate were also studied. During the tests the atmospheric temperature and pressure were recorded.

EVALUATION OF TEST RESULTS

For calculating the Mach number M from the mass flow rate, stagnation temperature and static pressure, the Fanno equation was used

$$\frac{\dot{W}\sqrt{T_o}}{pA} = \sqrt{\frac{Kg}{R}}\; M\sqrt{1 + \frac{K-1}{2}M^2} \quad . . . \qquad (1)$$

Details of the procedure for calculating M are given in reference 1.

For the calculation of the recovery in the diffuser, the recommended expression was employed (ref. 2).

$$c_p = \frac{p_2 - p_1}{\frac{1}{2}\rho_1 V_1{}^2} \quad . . . \qquad (2)$$

where $V_1 = M_1\sqrt{KgR\theta_1}$ and $\theta_1 = T_o/\left(1 + \frac{K-1}{2}M^2\right)$. The value of ρ_1 can be found from the equation of state $\rho_1 = p_1/gR\theta_1$.

It is noted that the entry condition to the diffuser (subscript 1) was assumed to be identical to the exit condition from the parallel pipe.

For calculating the local skin friction along the parallel pipe it was assumed that the distances between consecutive pressure tappings were small enough for the factor $\overline{f}$ to be considered constant over those distances. Then

$$\overline{f} = \frac{D}{4\Delta L}\left[\frac{1}{K}(\phi_{M_1} - \phi_{M_2}) - \frac{K+1}{2K}\ln\frac{\phi_{M_1}}{\phi_{M_2}}\right] \qquad (3)$$

where

$$\phi_{M_1} = \frac{1 + \frac{K-1}{2} M_1^2}{M_1^2}$$

and

$$\phi_{M_2} = \frac{1 + \frac{K-1}{2} M_2^2}{M_2^2}$$

Here the subscripts 1 and 2 refer to the inlet and outlet section respectively of the pipe length (ΔL) under consideration.

The mass flow rate of air through the Venturi was calculated with the expression

$$\dot{W} = C_d \; a_o \sqrt{2g\gamma\Delta p} \tag{4}$$

where the pressure differential ΔP was measured on the inclined manometer connected to the throat section. The other limb of the manometer was open to the atmosphere.

DISCUSSION OF THE RESULTS

The results of the experiments are presented in figures 3 to 8. In figure 3, the distribution of Mach number along the pipe and diffuser is shown for four different suction pressures from 12.7 cm to 21.1 cm Hg vacuum. In figure 4 the pressure recovery of the test diffusers is plotted against inlet Mach number, while in figure 5 the variation of the friction factor along the pipe is shown for the specific case when $M = 1$ at exit. In figures 6 and 7 the effects of boundary layer suction at various locations are shown and in figure 8 the velocity distribution across the inlet plane to the parallel pipe is presented.

Figure 3a shows that the Mach number at inlet to the parallel pipe was about 0.62 as against 0.58 calculated with an assumed constant friction factor of 0.02. About halfway downstream from inlet the flow attained $M = 0.7$, and from $x/d = 20$

onwards there appears a rapid increase in M. The gradient dM/dx is highest at the exit, where M becomes just slightly greater than unity*. In going downstream a rapid decrease in M is experienced in the inlet region of the diffuser and ultimately the flow exits with $M \approx 0.05$. For suction pressures lower than 18 cm Hg the pressure distributions change, resulting in lower Mach numbers as shown in figures 3c and 3d. While at pipe exit the Mach number falls substantially below unity the curves remain rather similar in appearance. After the insertion of the protruberance some tests were repeated to study the effect of partial blockage. The changes were found negligible and presentation of these results has been omitted.

The pressure recovery factor C_p for the original diffuser was found to be about 0.8 (fig. 4) and was slightly improving with increasing inlet Mach number. This figure is in good agreement with results obtained by other investigators (refs. 3 and 4). The pressure recovery for the proposed new diffuser was found about 4 percent higher and the experiments show that most of the recovery is already accomplished at halfway along the diffuser.

The local friction factor varies considerably along the pipe (fig. 5). It decreases first, then increases and finally markedly decreases near the exit. The scatter of the experimental points suggests that under unsteady flow the pressure measurements obtained by the mercury-in-glass piezometers have rather limited accuracy. It becomes clear, however, that the assumption of constant friction factor in a highly accelerating gas flow does <u>not</u> hold.

Velocity traverses obtained at various cross sections, namely at the inlet (X), outlet (Z), and mid-section (Y) of the second half of the new diffuser show the beneficial effects of boundary layer control with suction. The suction was applied at mid-section X only and shows little (if any) effect on the profile at X (fig. 6a). The effect, however, is more noticeable at section Y (fig. 6b) and is quite marked at the exit section Z. The gradual improvement of profiles with increasing boundary layer suction is shown in figure 7.

Finally, the velocity distribution obtained at the inlet to the parallel pipe (fig. 8) shows the effectiveness of the contraction. While the velocity distribution is fairly uniform across the center portion of the pipe, it appears to be slightly higher on both sides due to curvature of the contraction walls. This effect is

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* The increase of M above unity at the exit plane is attributed to the streamlines slightly diverging over a short distance ahead of the inlet to the diffuser.

considered negligible and usually disappears shortly after the flow enters the parallel pipe.

From the measured local friction factor an overall can be established and with this value assumed constant along the pipe, a variation of the inlet with the exit Mach number may be predicted as shown in figure 9. It appears that at lower Mach numbers (say, up to 0.4) the flow may be considered as incompressible.

CONCLUSION

The general conclusion that may be drawn from the experiments appears favorable and the projected duct design should meet the requirements. In other words, by scaling up the experimental results it may be predicted that the operational demand from the duct appears to match the performance of the induction system. Relevant details of calculations are presented in Appendix A.

More particularly it was found that:

1. A Mach number equal to unity at the exit from the parallel pipe may be anticipated in the range of induction pressures 18.03 to 21.1 cm Hg (vacuum) corresponding to 7888 kgf/m^2 and 7473.6 kgf/m^2, respectively. Since the effect of stream line curvature on the exit Mach number is rather difficult to estimate accurately, a conservative value that may be recommended for the induction pressure would be about 20.32 cm Hg corresponding to about 7593 kgf/m^2.

2. The pressure recovery in both of the diffusers may be considered satisfactory. However, the velocity distribution at the exit of the new proposed diffuser appears more attractive especially when boundary layer suction is applied at half distance along its length. Flow oscillations, which always accompany separated flow, thus may be reduced or even eliminated because the risk of flow separation is minimized with "full" velocity profiles. Relevant details of boundary layer suction are presented in Appendix B.

3. Variation of the friction factor along the pipe is considerable and as a result, the experimentally found Mach number at pipe inlet was found higher than the calculated figure based on constant friction.

4. The design of the contraction (air intake) appears satisfactory as the velocity distribution across the pipe inlet is fairly uniform.

5. A comparison drawn between pressure distributions along the pipe with unrestricted flow and restricted passage due to the protruberance indicated no substantial difference. Thus the partial blockage, due to the instrument panel, may be predicted to be negligible.

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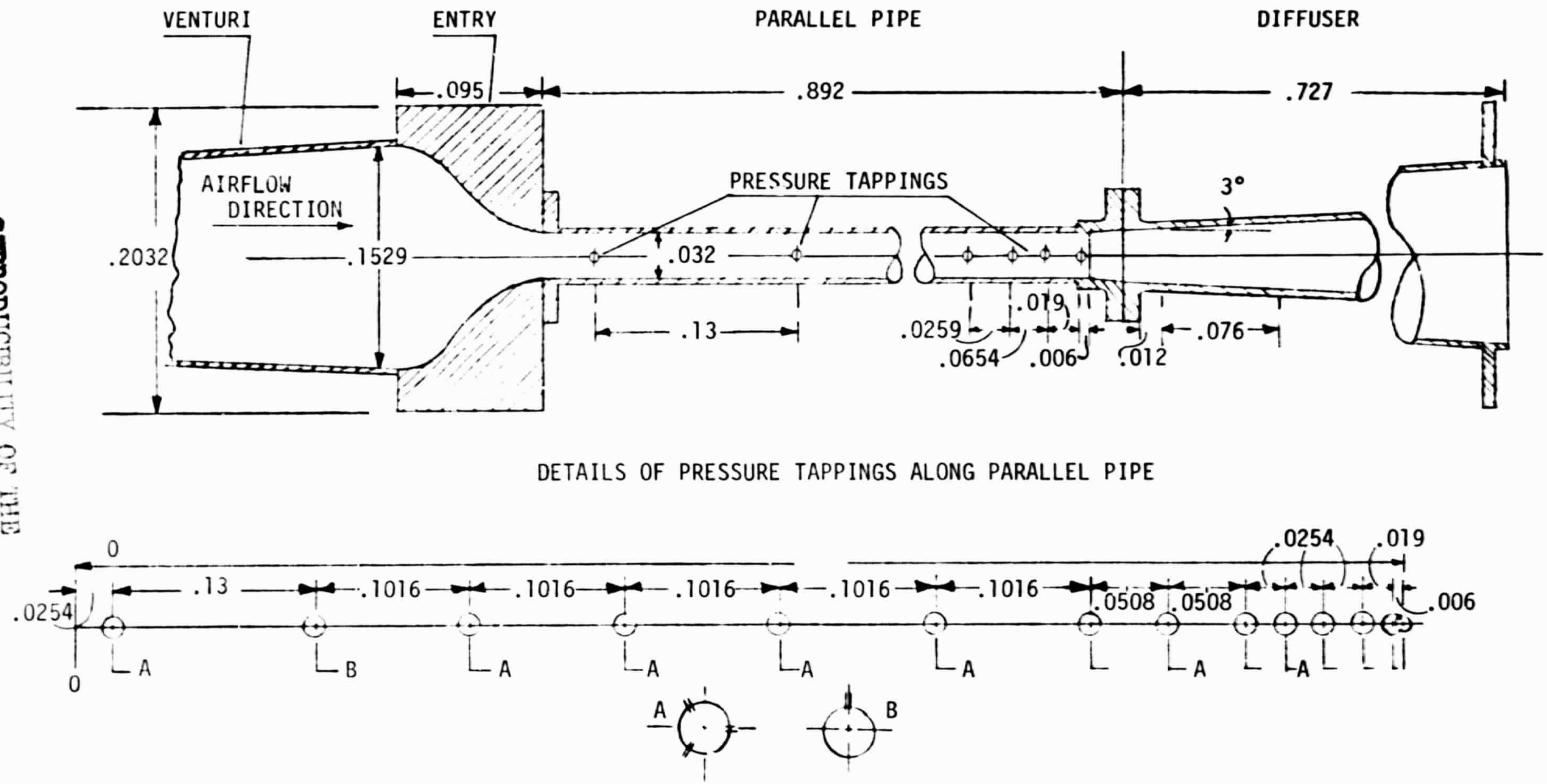
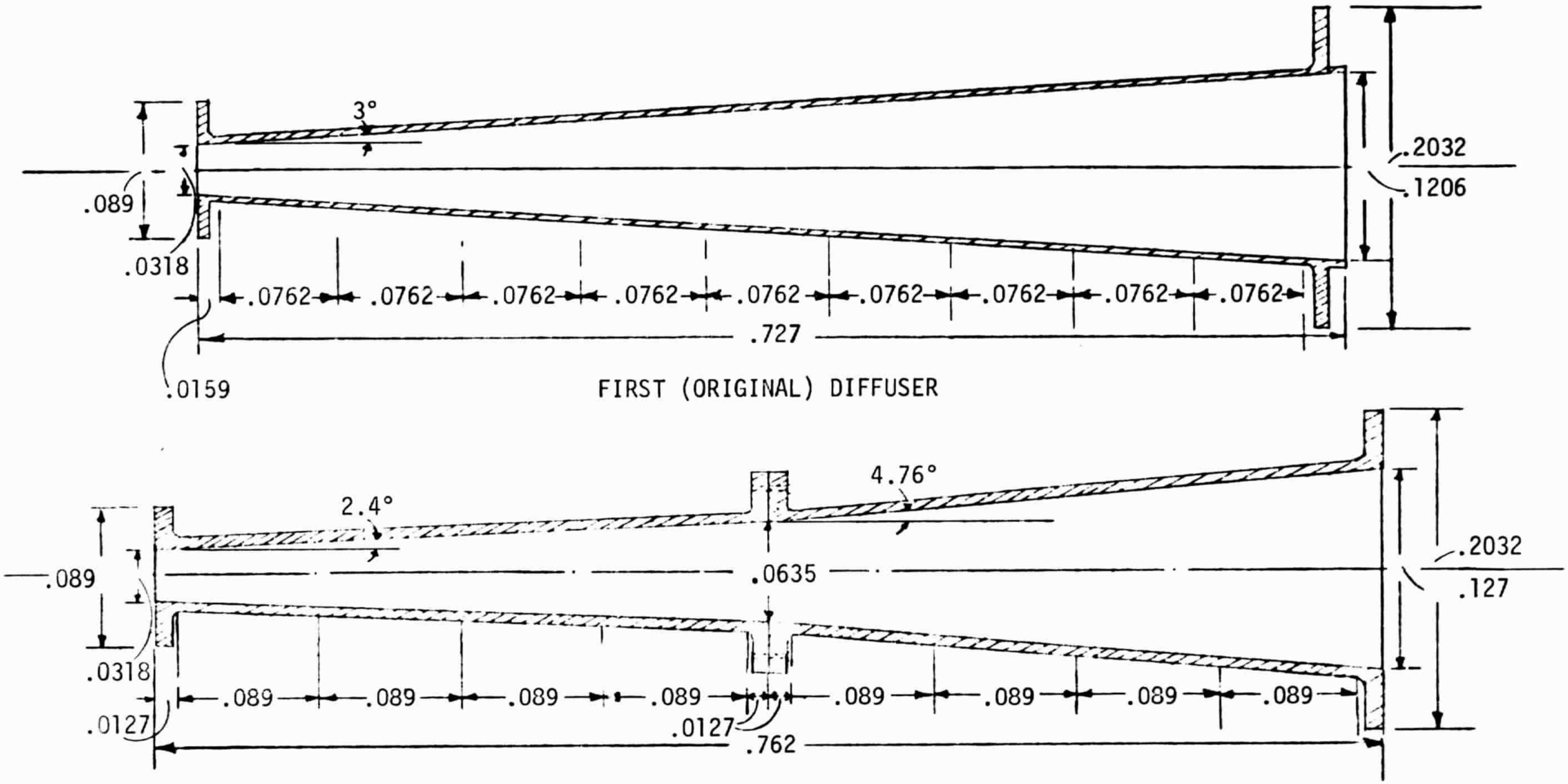


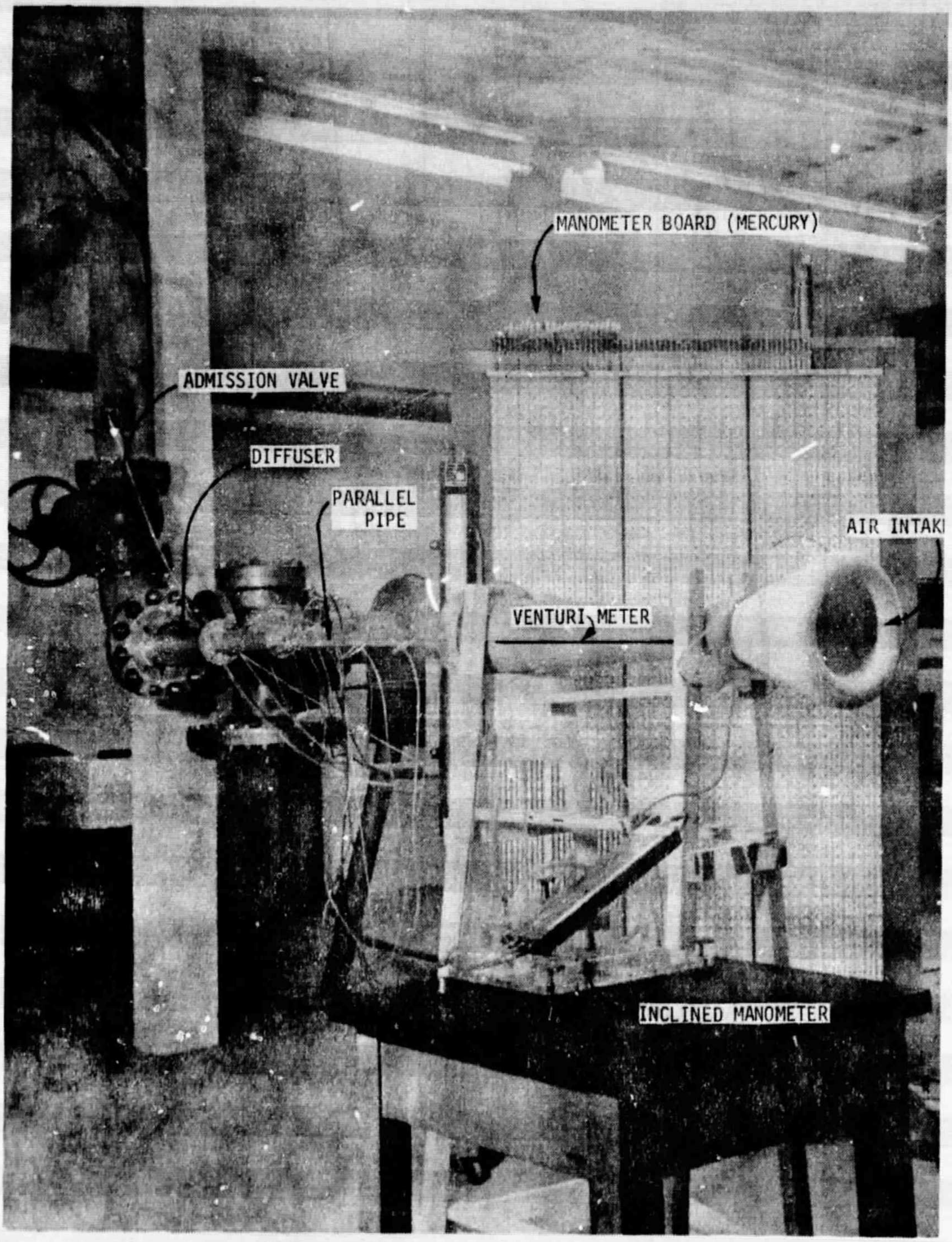
(a) Parallel pipe (all dimensions in meters).

Figure 1. General Arrangement of Test Duct-systems.



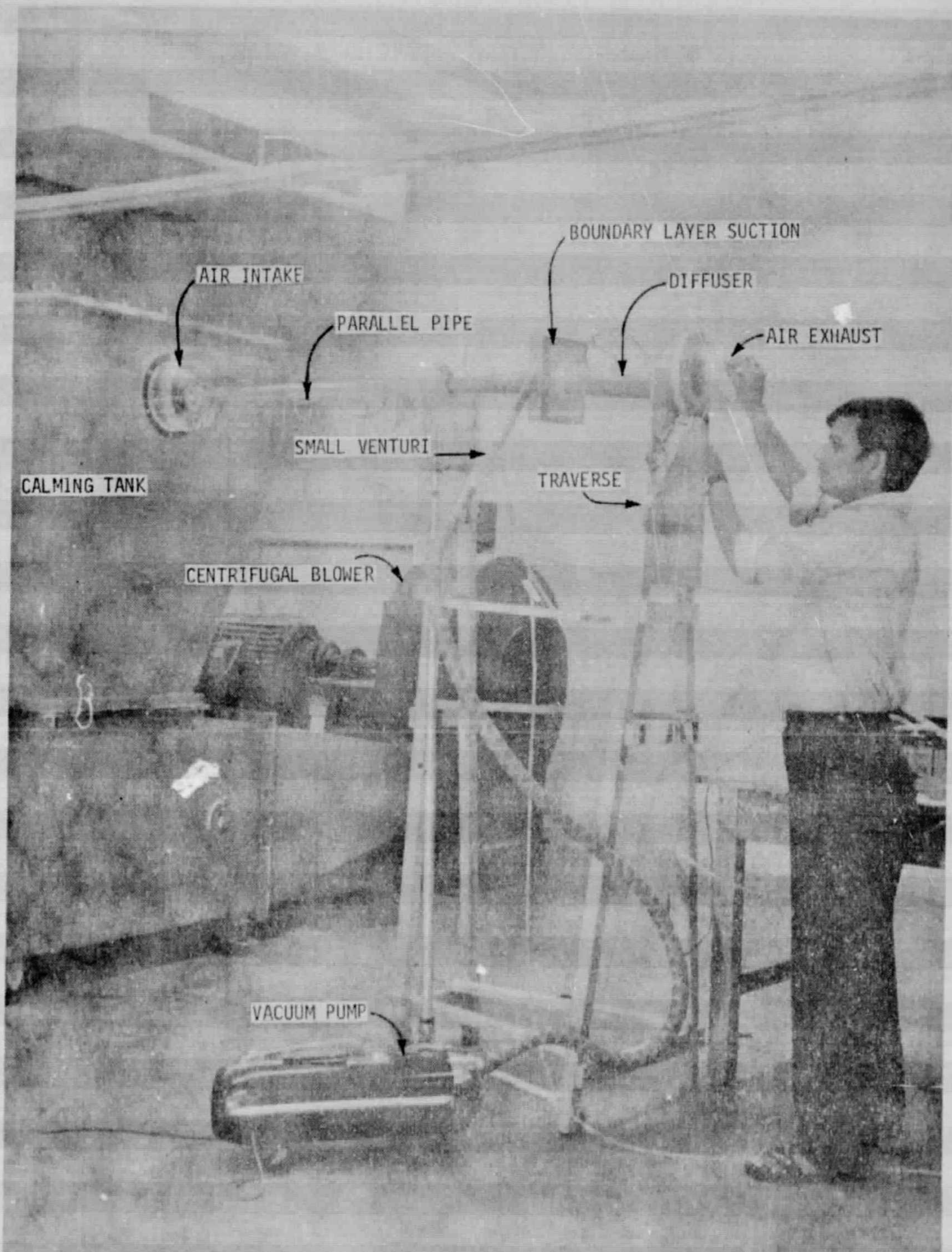
(b) Design details of test diffusers showing location of pressure tappings (all dimensions in meters).

Figure 1. Concluded.



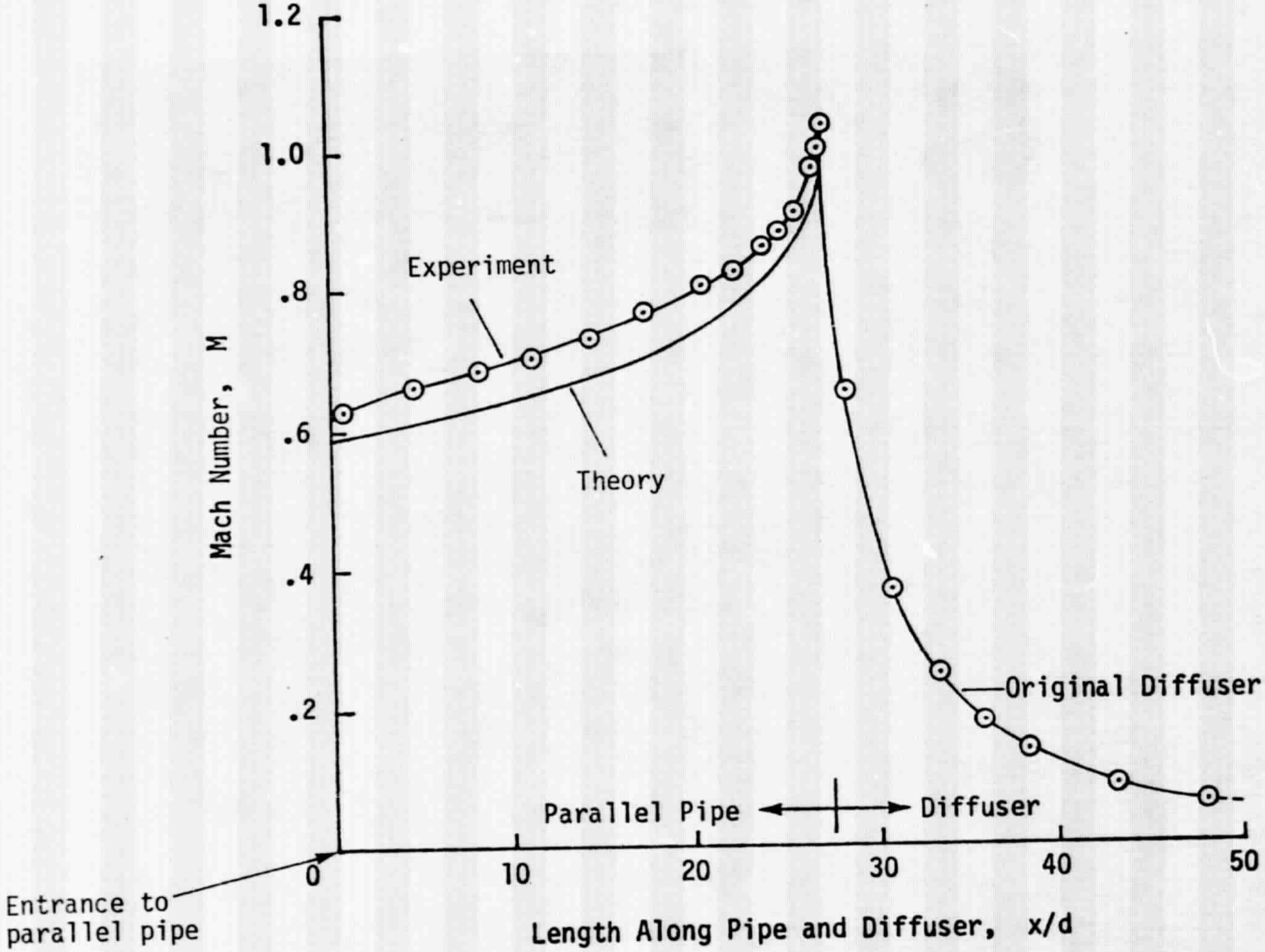
(a) Set-up for suction mode operation.

Figure 2. Photograph of experimental apparatus.



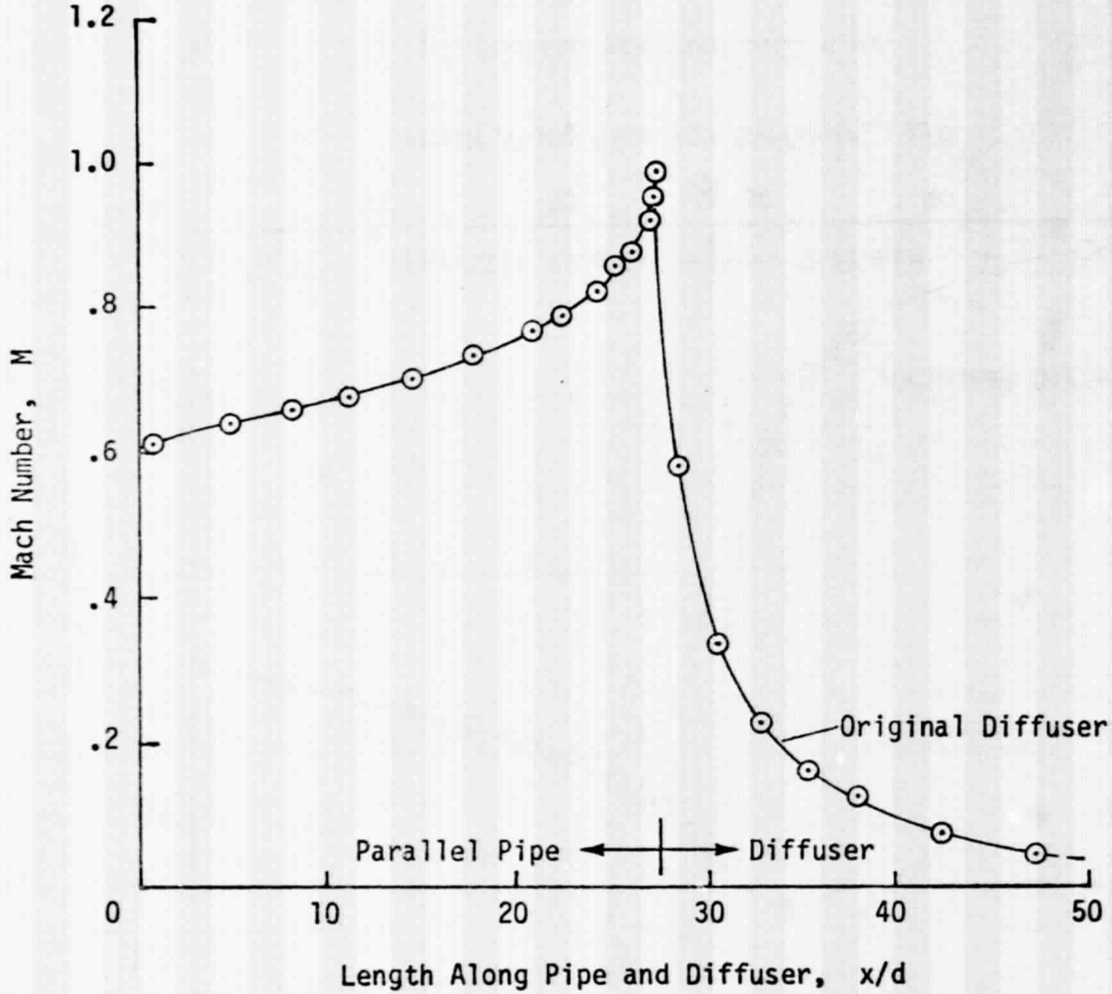
(b) Set-up for pressure mode operation.

Figure 2. Concluded.



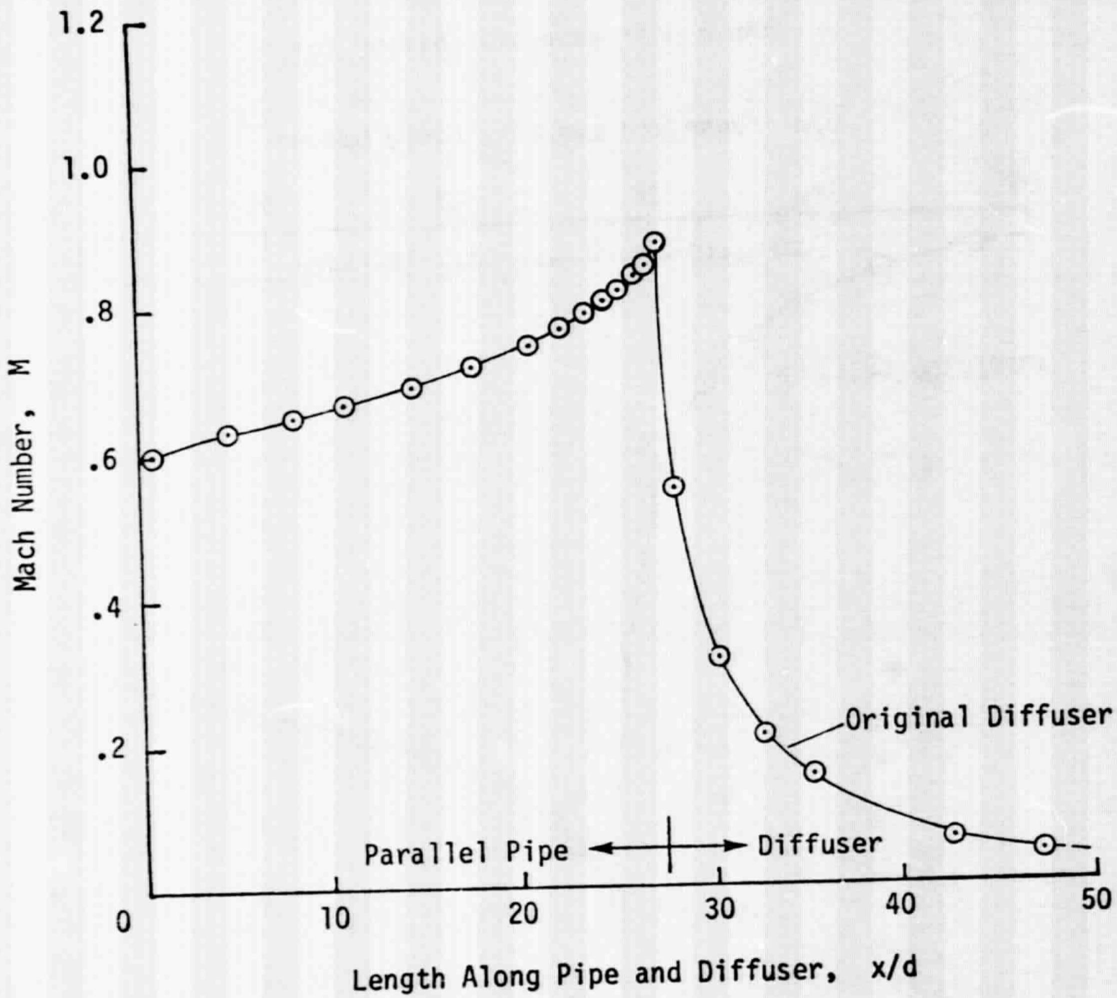
(a) Suction pressure .211 m Hg.

Figure 3. Variation of Mach Number along Parallel Pipe.



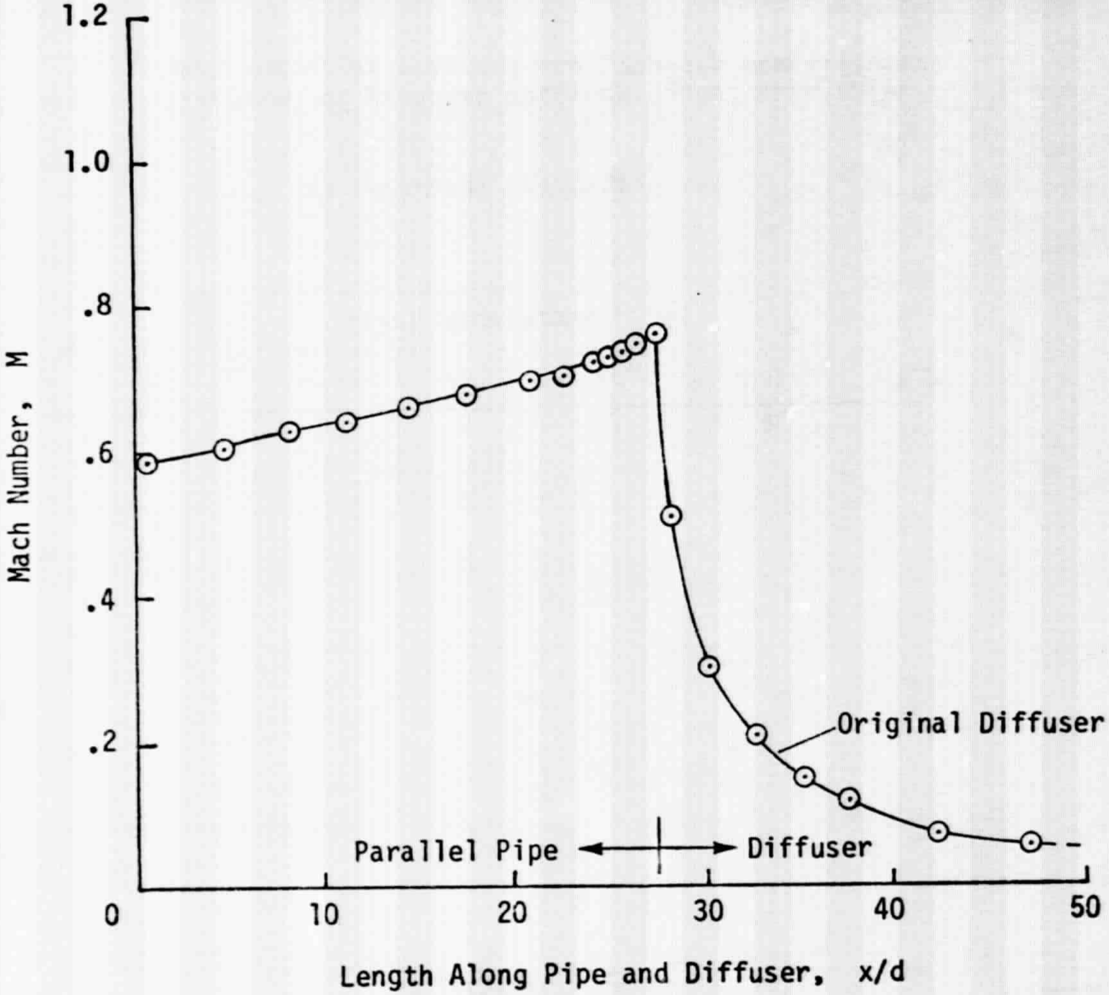
(b) Suction pressure .180 m Hg.

Figure 3. Continued.



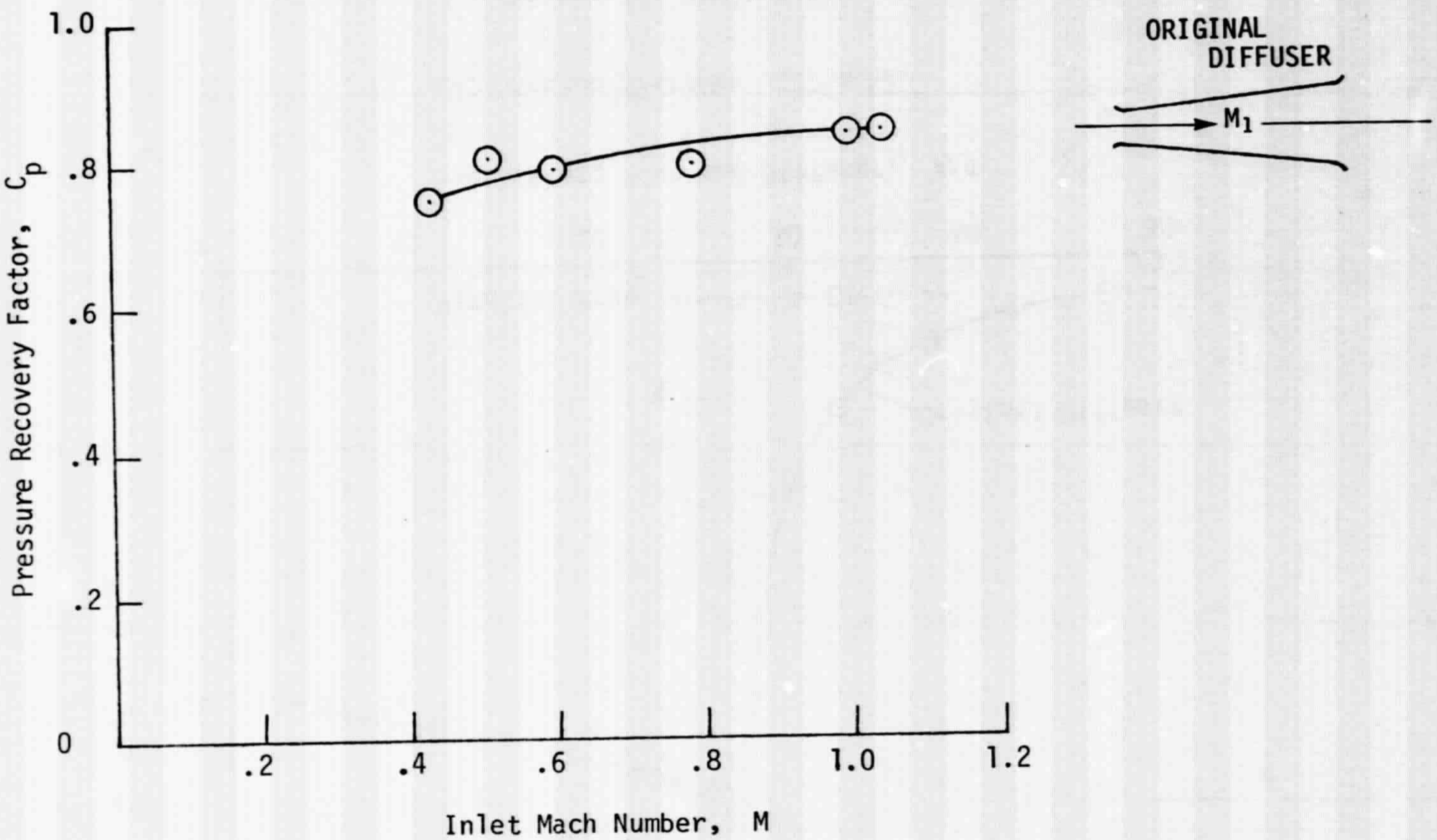
(c) Suction pressure .152 m Hg.

Figure 3. Continued.



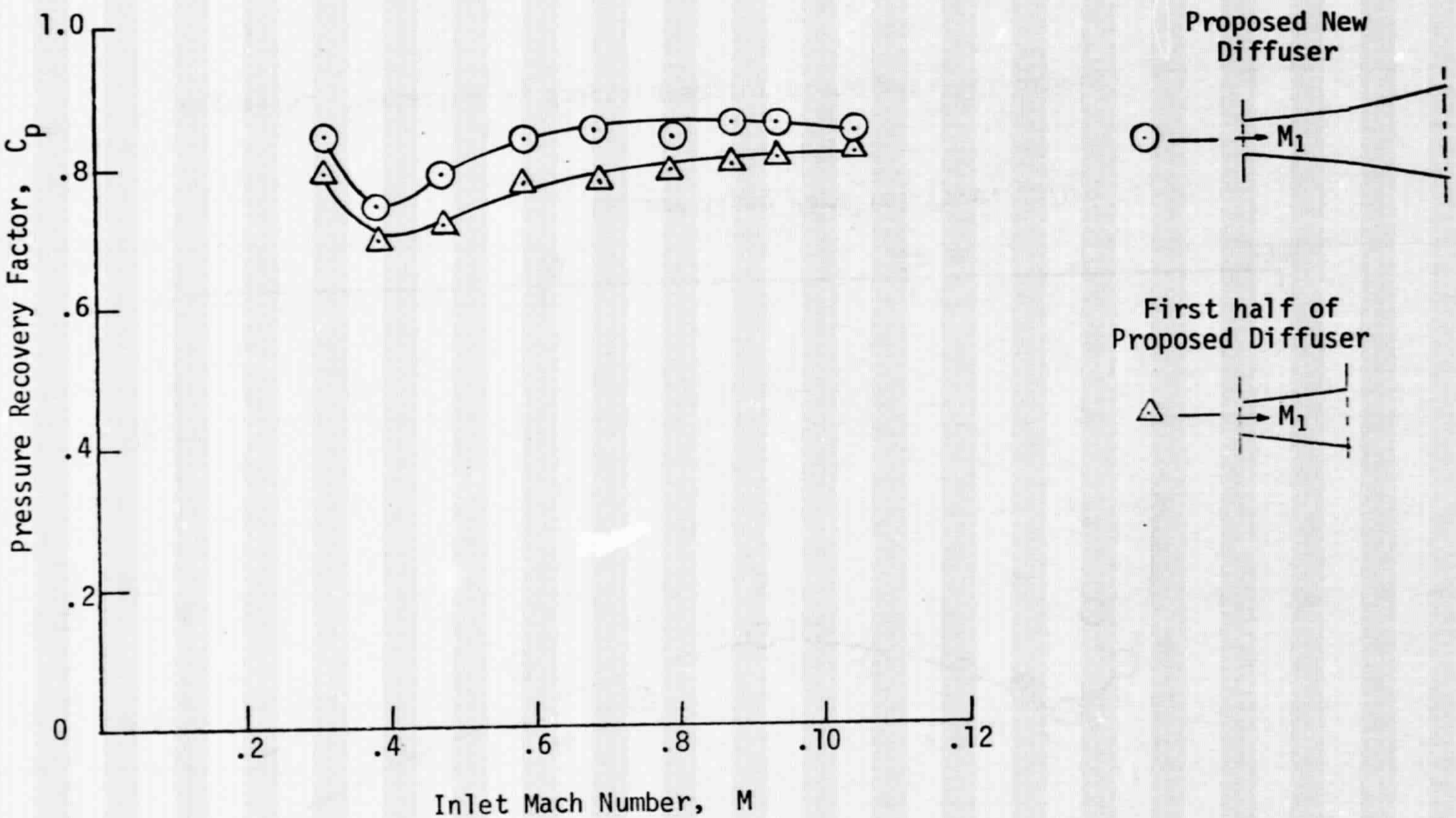
(d) Suction pressure .127 m Hg.

Figure 3. Concluded.



(a) Original Diffuser.

Figure 4. Variation of pressure recovery factor with inlet Mach number of original and proposed new diffuser.



(b) Proposed new diffuser.

Figure 4. Concluded.

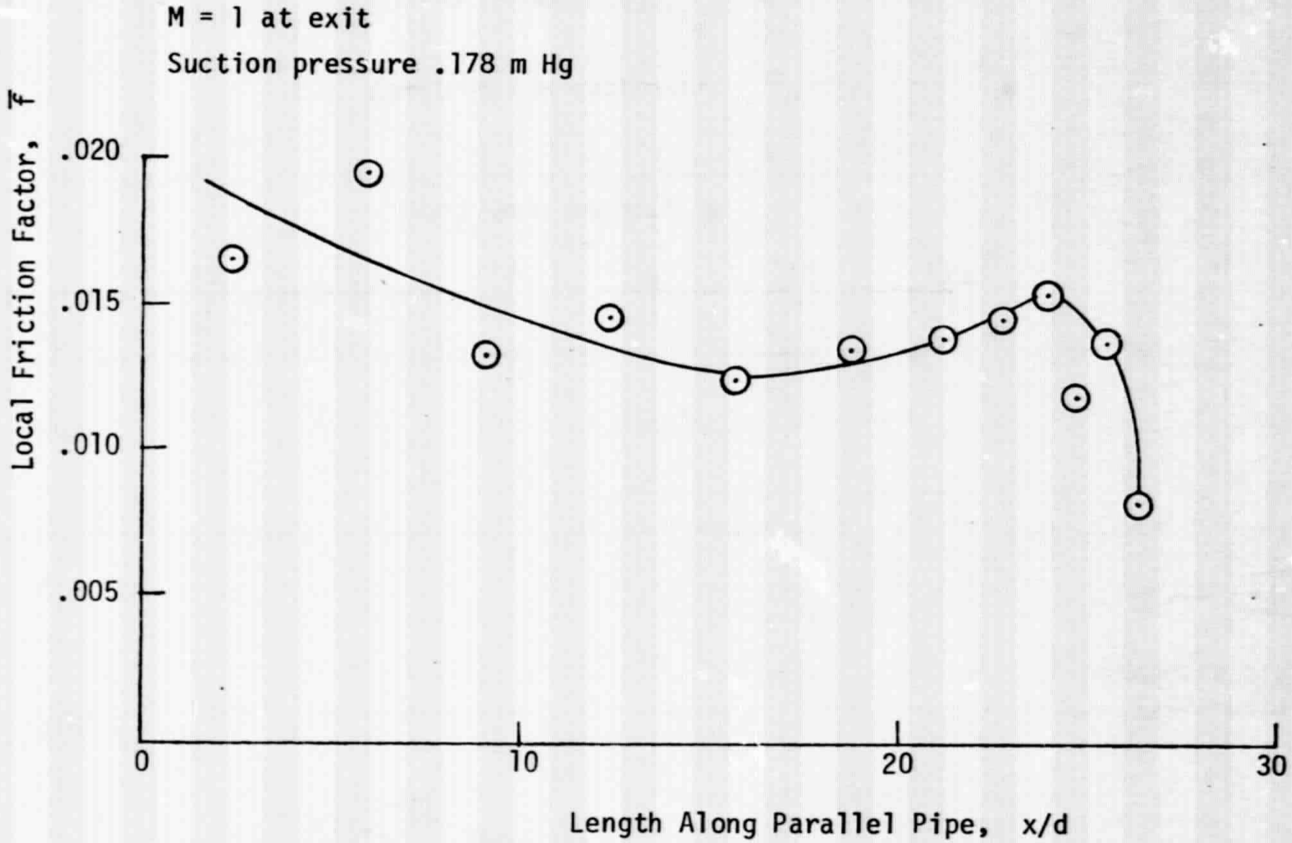
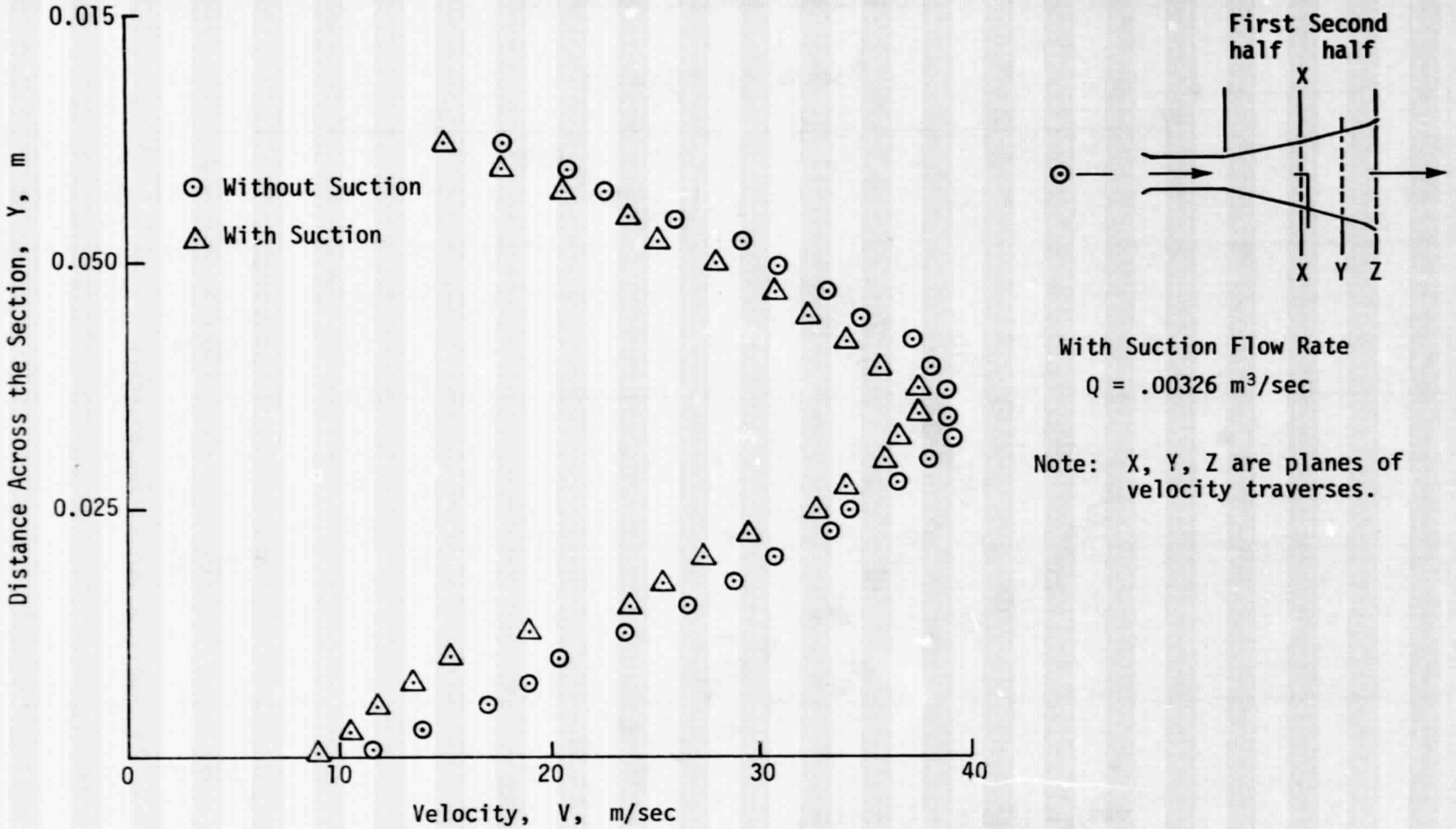
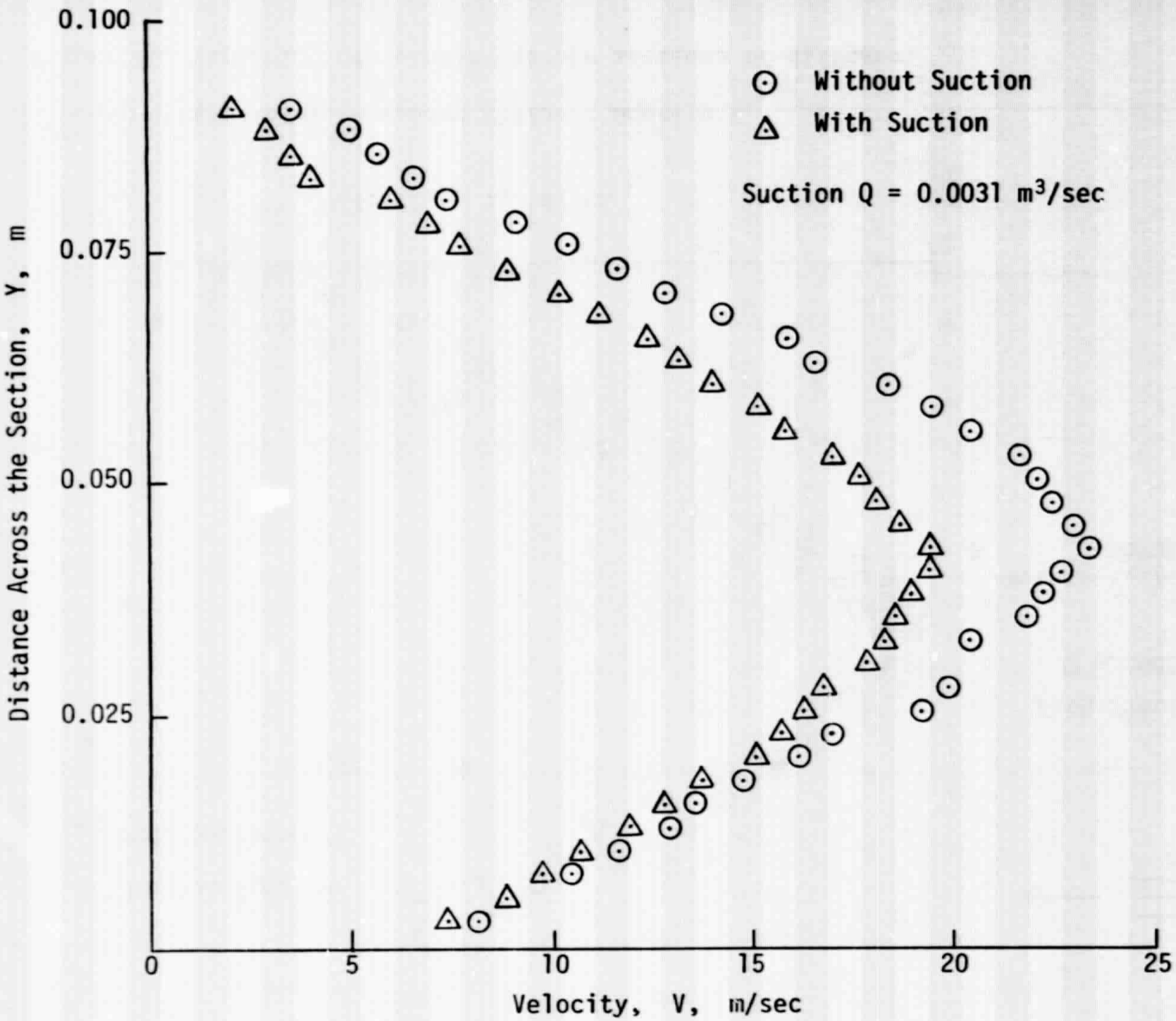


Figure 5. Variation of friction factor along parallel pipe.



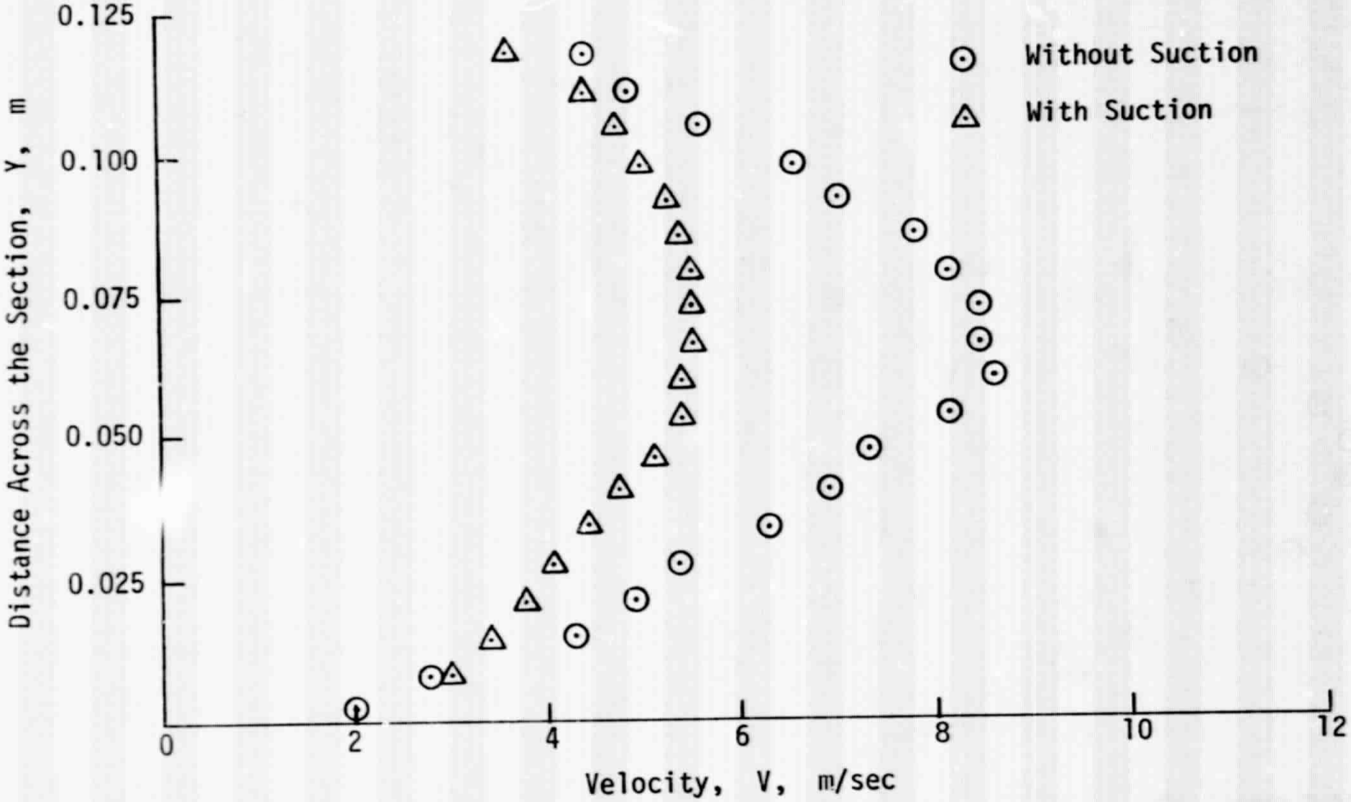
a) Inlet to second diffuser Section X.

Figure 6. Variation of velocity across sections of diffuser.



(b) Second diffuser, Section Y.

Figure 6. Continued.



(c) Exit Plane, Section Z.

Figure 6. Concluded.

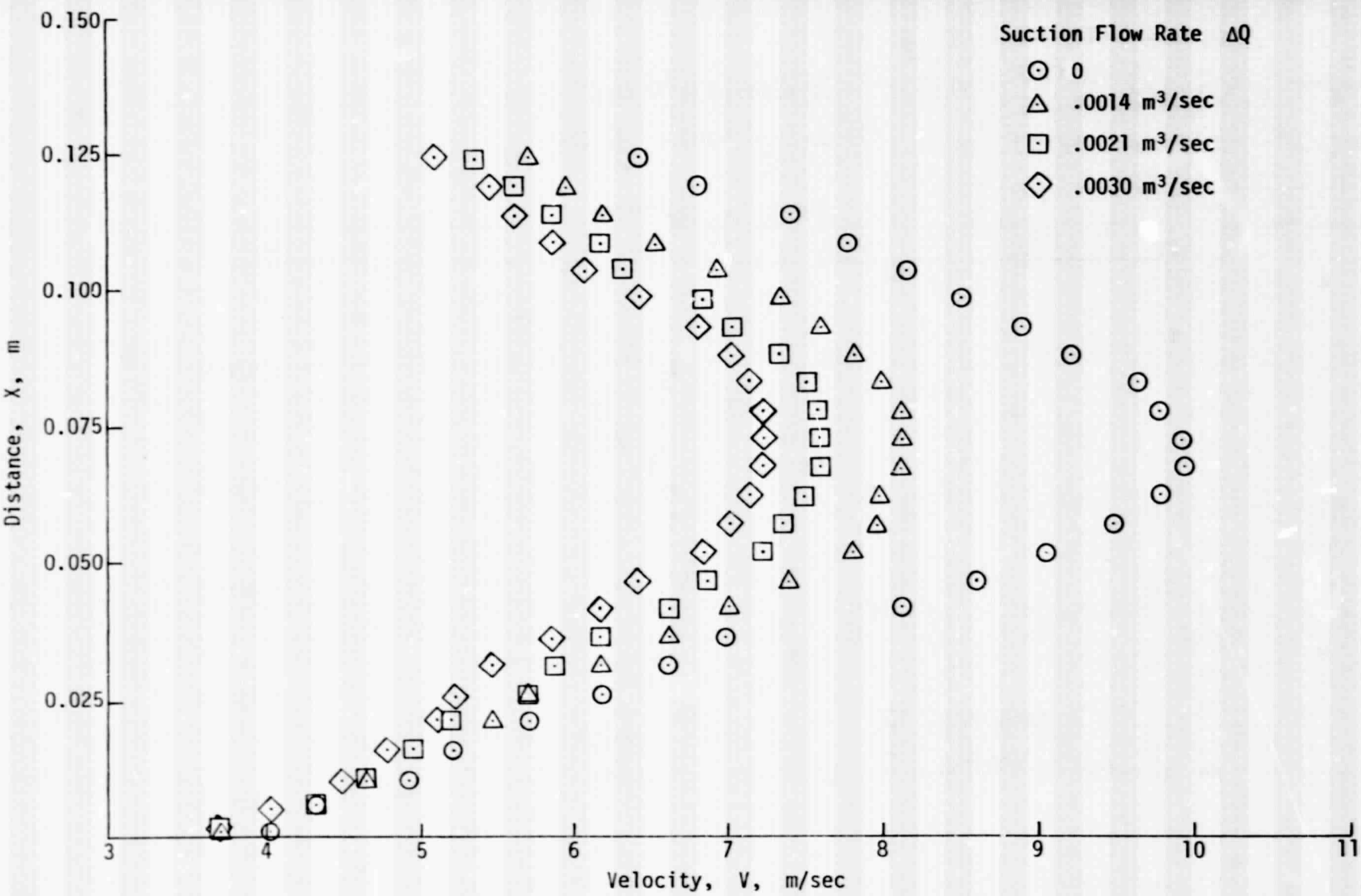


Figure 7. Variation of velocity across exit plane (Z) for different suctions.

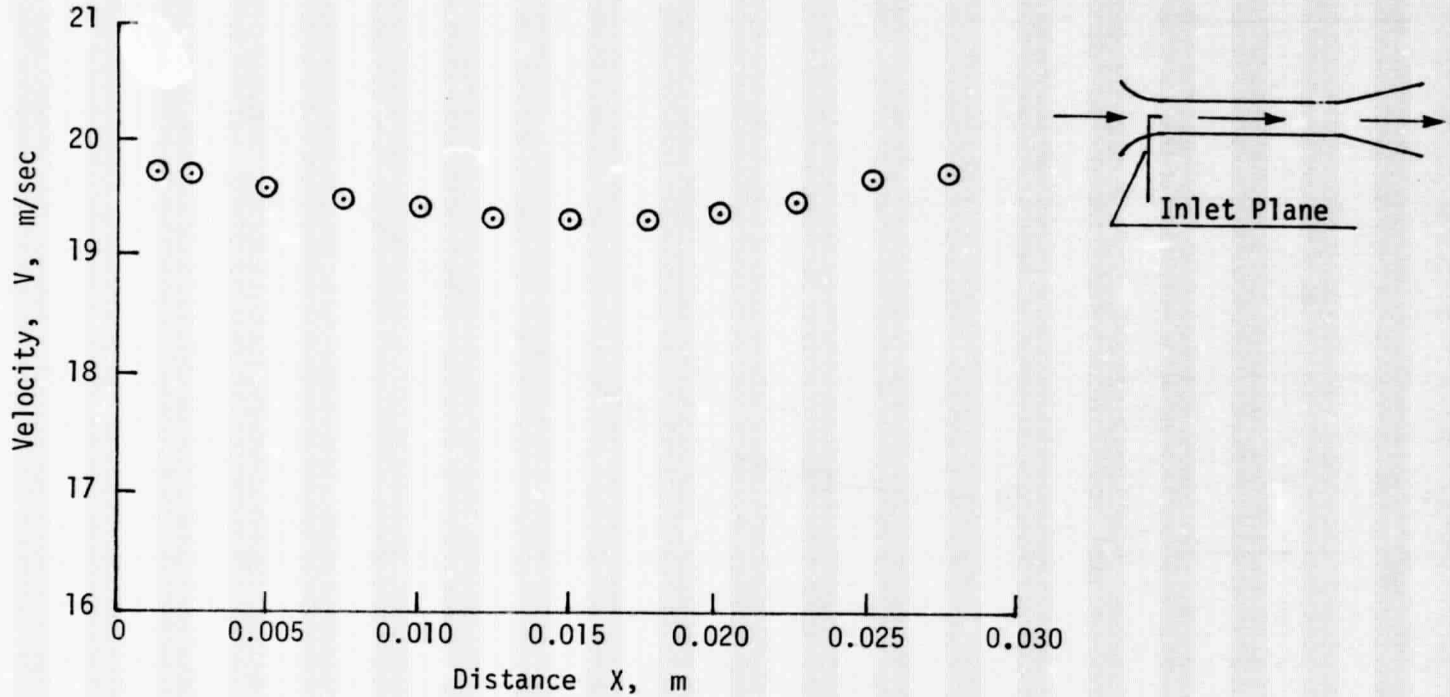


Figure 8. Variation of velocity across inlet plane to the parallel pipe.

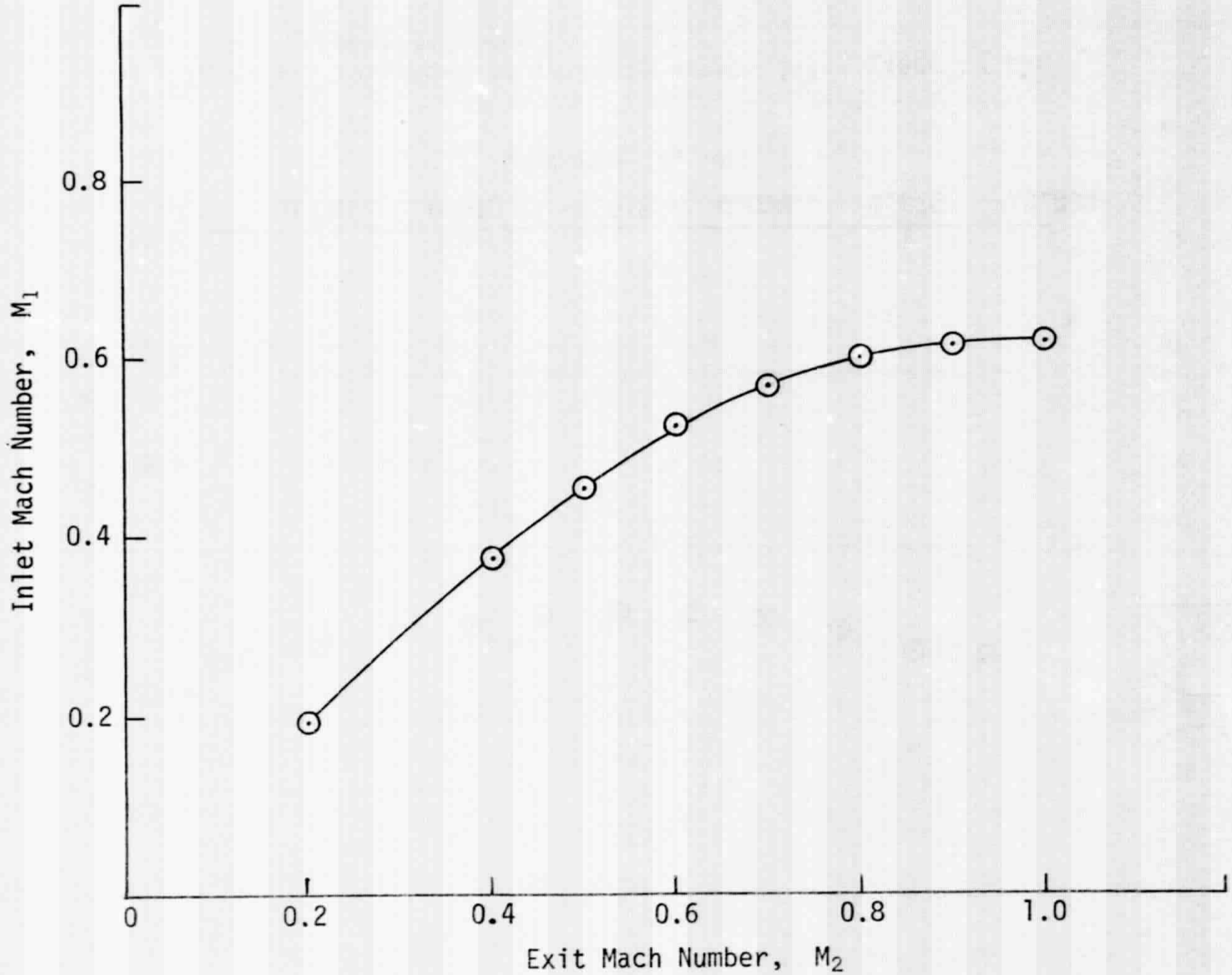


Figure 9. Predicted variation of inlet Mach number M_1 with exit Mach number M_2 for the parallel pipe assuming an overall friction factor $\overline{f} = 0.0146$.

APPENDIX A

CALCULATION OF THE DESIGN POINT FOR COMPRESSOR

From the experiments performed on the model, the design operational condition representing the anticipated mass flow rate and diffuser exit-pressures can be estimated for the full scale duct.

Assuming isentropic flow, the stagnation pressure at inlet to the parallel pipe would be atmospheric, with the experimentally found $M_1 = 0.62$ at inlet, the static pressure

$$p_1 = \frac{p_o}{\left(1 + \frac{K-1}{2} M_1^2\right)^{\frac{K}{K-1}}} = \frac{(14.7)(6894.75)}{(9.80665)\left[1 + \frac{K-1}{2} M_1^2\right]^{\frac{K}{K-1}}}$$

$$= \frac{(14.7)(6894.75)}{(9.80665)\left[1 + .2(.62)^2\right]^{3.5}} = 7975.25 \text{ kgf/m}^2$$

Since the flow is frictional and $M_2 = 1$ at pipe exit, from the Fanno equation

$$p_1 M_1 \sqrt{1 + \frac{K-1}{2} M_1^2} = p_2 M_2 \sqrt{1 + \frac{K-1}{2} M_2^2}$$

hence

$$p_2 = \frac{(7975.25)(.62)\sqrt{1 + (.2)(.62)^2}}{\sqrt{1.2}} = 4686 \text{ kgf/m}^2$$

The static temperature at pipe exit at $t_o = 70° \text{ F}$,

$$\theta_2 = \frac{T_o}{(K+1)/2} = \frac{530}{1.2} = 441.7° \text{ R} \quad .$$

By the equation of state $p/\gamma = R\theta$,

$$\gamma_2 = \frac{p_2}{R\theta_2}$$

$$R = \frac{(53.3)(4.448)(.3048)}{(9.80665)(.45359)} \quad \frac{kgf - m}{kg\ ^\circ R}$$

$$\gamma_2 = \frac{(4686)(9.80665)(.45359)}{(53.3)(4.448)(.3048)(441.7)}$$

$$\gamma_2 = 0.653\ kg/m^3 \quad .$$

Velocity of air at sonic condition for $\gamma = 1.4$,

$$R = \frac{(53.3)(4.448)(.3048)}{(9.80665)(.45359)} = 16.245\ \frac{kgf - m}{kg\ ^\circ R}$$

$$g = (32.2)(.3048) = 9.81456\ m/sec^2$$

$$V_2 = \sqrt{(1.4)(9.81456)(16.245)\theta} = 314.02\ m/sec$$

Hence the massflow rate, with pipe area

$$A = \Big[(1.25)(.0254)\Big]^2 \frac{\pi}{A} = .007917\ m^2$$

and without taking the boundary layer into consideration

$$\dot{W} = \gamma AV = (.653)(.007917)(314.02) = 0.16234\ kg/sec.$$

Experimentally, mass flow rate of the model found

$$(.3515)(.45359) = .159437\ kg/sec \quad .$$

Without the boundary layer thickness the sealed up flow rate in the duct would be $\dot{W} = (.653)(.785)(.092903)(314.2) = 14.95$ kg/sec; with a scaled boundary layer displacement, this reduces to an actual flow rate

$$\dot{W}_a = 14.817\ kg/sec \quad .$$

Since sonic flow was attained at about .203 m mercury, the exit pressure from the diffuser was found

$$p = \frac{(14.7)(6894.75)}{9.80665} \quad \frac{(36 - 8)}{30} = 7579.08 \text{ kgf/m}^2 \quad .$$

The point of operation is shown on the compressor performance chart (fig. A1) as OP.

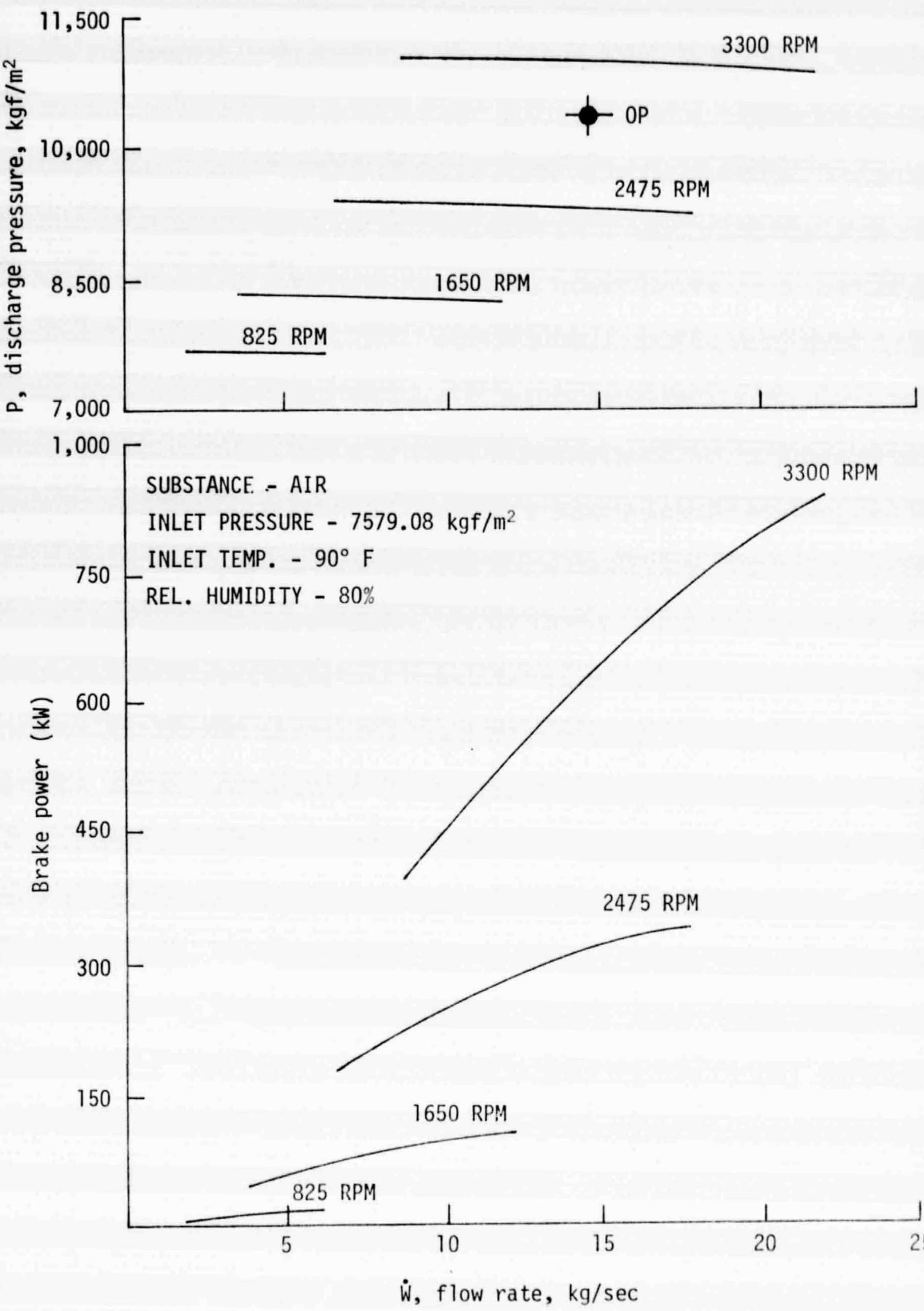


Figure A1. Performance characteristics for ANRL duct system.

APPENDIX B

For the estimation of boundary layer suction power the following approximate calculation was employed.

For the maximum volume suction for the model $\Delta Q = (.105)(.0283168) = .00297\ m^3/sec$ there corresponds a $\Delta Q = (92.16)(.00297) = .274016\ m^3/sec$ in the duct. Experiments show that the pressure at halfway section was $7733.76\ kgf/m^2$, hence the difference between atmospheric pressure $\Delta p = 10335.11 - 7733.76 = 2601.35\ kgf/m^2$.

The theoretical power required to remove this air is

$$p = (\Delta\phi)(\Delta P) = (.274016)(2601.35) = 712.81\ kgf - m/sec = 6.995\ kW.$$

To match this requirement a 11.60 kW motor would be recommended.