ON THE USE OF JET DRIVES FOR WIND TUNNELS OF HIGH VELOCITY

(ÜBER DIE VERWENDUNG DES STRAHLANTRIEBES BEI WINDKANÄLEN HOHER GESCHWINDIGKEIT)

by

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ABSTRACT:

Jet drives have been used successfully many times in foreign countries for small wind-tunnel installations of high velocity. The question is treated in the following report as to whether such a drive is essentially economical from the viewpoint of energy consumption, and further as to how the ratio of weight of driving material to weight of induced air changes with the pressure of the available driving material. These relations are calculated on the supposition that the speed of sound will be attained in the test section, and they are compared with experimental results.

Outline:

I. INTRODUCTION

The power requirement of a wind tunnel installation increases, as is well known, with the third and on approaching the speed of sound, with a still higher power of the velocity required in the test section: For example, a power of 40,000 to 50,000 H.P. is required to produce a sonic jet of one square meter cross section exhausting into the atmosphere. Although part of this power can be recovered in a wind tunnel with return passage, a continuous-flow installation of this type can hardly be considered. A significant reduction in the power requirement is possible if the free jet is discarded and the closed test section is used, and further if the wind tunnel is made with return passage and the entire installation is evacuated. Another way to decrease the building cost of a high-velocity wind tunnel to a reasonable amount is to store up the energy in some form and then use a high-velocity test section with short period drive. An installation of this type was constructed for the first time in Göttingen to the plans of Professor Prandtl (Reference 1). In this case the air from the atmosphere was sucked into an evacuated vessel through a normal nozzle or through a Laval nozzle by interconnection to an enlarged test chamber. A further possibility of driving a high-speed test section for a short time consists in the use of jet propulsion. The driving material, air or steam, is stored in a pressure vessel which, compared with a vacuum vessel, possesses no disadvantages to provide proportionately large quantities of energy in a small space. Small wind tunnels of this type are in use in foreign countries, as for example the American NACA tunnel with a test section of 280 mm diameter (Reference 2), and the English NPL high-speed tunnel with a test section of 300 mm diameter (Reference 3). The arrangement of these installations is essentially as follows: Air from the atmosphere is sucked in through a closed, slightly-enlarged test section. The suction tube is surrounded by a ring-shaped chamber out of which the driving material is conducted through a nozzle to the mixing cross section. The mixing follows in the diffusor directly connected to this cross-section. Compressed air is used as driving material for these installations.

In general, the effectiveness of a jet drive is fairly low as soon as a considerable work performance is required, that is the furtherance of a fixed quantity of air by overcoming a greater pressure difference. The ratios are so favorable for a jet-driven wind tunnel with closed test section, since the flow resistances to be overcome are proportionately less, that the energy requirements of a jet drive in this case lie within entirely reasonable limits. The inevitable sources of losses which occur in such an installation will be treated more fully in the following. For simplicity the arrangement shown in Figure 1 will be considered in which the mixing process takes place in a cylindrical tube to which a diffusor with a relatively large inlet cross section is attached.

Until now no successful accurate advance calculations have been made on the mixing phenomena of two air streams of different total pressures in a tube. In addition, the existing experimental information hardly suffices to answer certain questions of principles. For example, what length of tube is required to provide for the mixing of two flows of different total pressures into an approximately uniform flow, etc. If, however, the assumption is made that a sufficiently long tube is available for the mixing, then the law of momentum renders the statement feasible as to whether loss of the capacity of the air for doing work is connected with this mixing as regards equalization of velocity, so long as the loss of energy caused by friction of the tube is neglected. There is to be added to this mixing loss, in the case of the jet drive of Figure 1, the friction loss in the suction tube length, the drag of the model installed in the test section, the important energy loss in the diffusor, and finally the exit loss. It is customary in the case of ordinary wind tunnels to judge the excellence of the arrangement from the ratio of the fan drive energy to the jet energy, $(\frac{\rho}{2} F v^3)$. This efficiency factor, as is well known, amounts to about 0.5 to 0.6 for open-jet wind tunnels, and from 0.25 to 0.35 for tunnels with closed test sections. The most important sources of loss are the mixing loss in the free jet, the loss in the corners, in the expansions, and in the fan itself. Corresponding numerical results are lacking for jet-driven wind tunnels up to the present time. It may therefore be explained that the excellence of a jet-driven wind tunnel may be judged from the ratio of actual consumption of driving material to the theoretically determined required amount, obtained by calculations from the approximate sources of loss.

The problem is usually so stated in the design of a jet drive that a definite velocity is required in a given test cross section, which means that a certain weight of air must be sucked through this cross section. The value of the required weight of driving material is determined by the velocity with which this weight of air or steam is exhausted from the driving nozzle, therefore primarily by the magnitude of the pressure in the ring chamber preceding the mixing jet. The theoretical required weight ratios for a jet drive are calculated with simplified assumptions in Section III as a function of the pressure in the nozzle chamber under the supposition that sonic velocity is attained in the suction cross section, and that compressed air is used as driving material. Experiments are reported in Section IV which were carried out on a wind tunnel model driven by a large rotary compressor. The results of these measurements are finally compared with the theoretically obtained values.

The relationships of a short period jet drive are such that a certain weight of air or steam is stored at high pressure in a suitable vessel connected to the driving nozzle through a reducing valve. This valve throttles the pressure in the vessel to such an extent as is necessary for the velocity required in the test section. As approximately carried out in Section IV, it is not necessary to hold the pressure exactly constant in the ring chamber because, with proper arrangement of the test section, the induced velocity is practically independent of this pressure as long as it does not drop below a certain minimum value. On the other hand it would be very uneconomical to forego all pressure reduction since the weight of air exhausting from the driving nozzle would be unnecessarily large at first, leading to a decrease in the useable measuring time.

II. List of Symbols

$p_o \; T_o \; \rho_o \; i_o$	Pressure, temperature, density, and heat content of the atmosphere.
a^*	Critical velocity of sound.
$p_a \; T_a \; \rho_a \; i_a$	Pressure, temperature, density, and heat content of the air in the ring chamber of the driving nozzle.
F_1	Exit cross-section of the driving nozzle.
F_1'	Narrowest cross-section of the driving nozzle.
F_2	Suction tube cross section.
w_1, w_2	Velocities in F_1, F_2.
p_k	Static pressure in the mixing plane.
$p(p_{ü}\; p_{u})\; w(w_{ü}\; w_{u})\; \rho(\rho_{ü}\, \rho_{u})$	Pressure, velocity, and density after the mixing. (without regard to compression shock).
$\beta = \frac{F_1}{F_2}$	
p_e^*	Pressure at the end of the diffusor: $= p_o$ with proper arrangement of p_a and β.
G_1	Weight of the conducted driving material.
G_2	Weight of the quantity of air sucked through F_2.
$A = \frac{1}{427} \times \frac{kg\ cal}{m\ kg}$	

k = Efficiency factor = $\frac{\text{Drive energy}}{\text{Jet energy}}$

III. Theoretical Portion

The mixing is considered first of two air streams with the velocities w_1 and w_2 ($w_1 > w_2$) in a cylindrical tube as shown in Figure 1. Mixing is to start in cross-section AA and to finish in cross-section BB so that a uniform velocity w exists. (1) The static pressure p_k and p before and after the mixing are constant over the cross-section. It is further assumed that the velocities w_1 in the ring area F_1, and w_2 in the area F_2 are far below the speed of sound and thus can be considered without density changes, ($\rho = \rho_o$). The law of momentum then gives, in combination with the continuity equation, an increase in pressure between the cross-section AA and BB (neglecting the very comprehensive friction losses) of the value:

$$p - p_k = \rho_o (w_1 - w_2)^2 \cdot \frac{F_1 F_2}{(F_1 + F_2)^2} \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots \quad (1)$$

The method of operation of an injector pump is thus already principally explained. Also with a deficient diffusor ($p = p_o$) the value $p_k < p_o$; that is, air is sucked through F_2 in connection with which the quantity is so put in as to satisfy equation (1). The effect is correspondingly increased by placing a diffusor next to the mixing tube. A loss of the capacity of the air for doing work is connected with the mixing, of the value:

$$\Delta E = \frac{\rho_o}{2}\left[F_1 w_1^3 + F_2 w_2^3\right] + (p_k - p)(w_1 F_1 + w_2 F_2) - \frac{\rho_o}{2} \cdot (F_1 w_1 + F_2 w_2)^3 \frac{1}{(F_1 + F_2)^2} .$$

(1) A short time ago a very practical method for calculating jet apparatus was published by G. Flügel (VDI Forschungsheft Nr. 395). He proceeded on the supposition that the mixing occured in a very small and cylindrical tube. The shape of the conical part is so imagined that the mixing there proceeds with constant pressure, and the pressure increase occurs first in the cylindrical tube. The narrowest cross section is so dimensioned that the weight ratios attain a favorable value. The question as to whether these considerations maintain their validity for the case of high driving material velocity as well as of high suction velocity is not treated further. A comparison of the theoretical results with the measurements is kept for another project.

The turbulent mixing thus causes a part of the existing kinetic energy to be transformed into heat.

If the velocity of sound is approached in the suction tube together with the correspondingly higher exit velocities from the driving nozzle (Laval nozzle), then the simplified assumption of constant density no longer holds. On the contrary the assumption should be maintained that p_k = a constant over $(F_1 + F_2)$, by suitable dimensioning of the driving nozzle. Neglecting the flow resistances in the suction tube, p_k is obtained for a required velocity w_2 with adiabatic expansion, from:

$$\left(\frac{p_k}{p_o}\right)^{\frac{\chi-1}{\chi}} = 1 - \frac{\chi-1}{\chi+1}\left(\frac{w_2}{a^*}\right)^2 .$$

$$a^* = \sqrt{\frac{2\chi}{\chi+1} \quad \frac{p_o}{\rho_o}} = \text{Velocity of sound for the critical state.}$$

For the case: $w_2 = a^*$, which is to be investigated here, the pressure ratio when $\chi = 1.4$ is:

$$\frac{p_k}{p_o} = 0.528$$

and the density ratio:

$$\frac{\rho_2}{\rho_o} = 0.63$$

The supposition that $p_{st} = p_k$ in the exit of the driving nozzle should be fulfilled by suitable choice of the dimensions of this nozzle. Each state of the air in the ring chamber of the driving nozzle (p_a, ρ_a, T_a) is then associated with a definite exit velocity w_1 and air density ρ_1 at the mixing location. The mixing to a uniform velocity w shall again take place in a cylindrical tube. The energy equation must also be applied to the determination of the final state of the air after mixing (p, w, ρ) in addition to the continuity equation and the law of momentum.

Continuity equation:

$$F_1\rho_1 w_1 + F_2 w_2 \rho_2 = (F_1+F_2) w\rho \quad \ldots\ldots (2)$$

Law of momentum:

$$(F_1+F_2)p_k + F_1\rho_1 w_1^2 + F_2\rho_2 w_2^2 = (F_1+F_2)(\rho w^2 + p). \qquad (3)$$

Energy equation:

$$F_1 w_1 \rho_1 i_a + F_2 w_2 \rho_2 i_o = (F_1+F_2) w\rho \frac{A}{g}\left[\frac{x}{x-1}\frac{p}{\rho} + \frac{w^2}{2}\right] \ldots\ldots (4)$$

$$i = \frac{x}{x-1}\,\frac{A}{g}\,\frac{p}{\rho}$$ Heat content of the air after the mixing.

It is assumed, as may be seen, in setting up the energy equation that no notable losses occur either in the suction tube or in the driving nozzle, and further that the velocity is negligibly small in the nozzle ring chamber, and that no heat exchange takes place between the mixing tube and the outer air.

The further process of calculation is as follows: For an exit condition of the air in the nozzle ring chamber (p_a, T_a) the velocity w_1 and the density ρ_1 are calculated for adiabatic expansion to the pressure p_k. The solution of equations (2), (3) and (4) gives the still unknown values, p, w, ρ, so far as a deliberate assumption is also still made for the area ratio, $\frac{F_1}{F_2} = \beta$. Finally the end pressure is determined at the diffusor exit on the basis of an estimate of the efficiency of a diffusor attached to the mixing section. Since $\frac{F_1}{F_2}$ was assumed as desired, the calculated end pressure will not agree with the external pressure. Repetition of these calculations will finally give the correct area ratio for which the determined pressure requirement is fulfilled, and thereby a connection between p_a and β . An analytical solution of the system of equations with the result $p_a = f(\beta)$ is not possible from consideration of the form of the equations.

Next some dimensionless coefficients are introduced:

$$\left.\begin{array}{l}\varphi = \dfrac{w_1}{w_2} \\[2ex] \psi = \dfrac{\rho_1}{\rho_2}\end{array}\right\}$$ determined from assumptions of p_a and T_a.

$$p_k^* = \frac{p_k}{\rho_2/2 \; w_2^2},$$

further the values:

$$C_1 = w\rho = w_2 \rho_2 \frac{\beta\varphi\psi + 1}{\beta + 1}$$

$$C_2 = p + \rho w^2 = \rho_2 w_2^2 \left[\frac{p_k^*}{2} + \frac{\beta\varphi^2\psi + 1}{\beta + 1}\right]$$

$$C_3 = \frac{x}{x-1} \cdot \frac{p}{\rho} + \frac{w^2}{2} = \frac{\beta\varphi\psi i_a + i_o}{\beta\varphi\psi + 1} \; \frac{g}{A}$$

or, in case $i_a = i_o$:

$$C_3 = i_o \frac{g}{A} = \frac{x+1}{2(x-1)} (a^*)^2.$$

The pressure after the mixing follows from equations (2) to (4), using the foregoing values:

$$p = C_2 \left[\frac{1}{x+1} \pm \sqrt{\frac{1}{(x+1)^2} - 2\,\frac{x-1}{x+1}\left[C_3 \left(\frac{C_1}{C_2}\right)^2 - \frac{1}{2}\right]}\right], \quad \ldots\ldots\ldots\ldots (5)$$

or with $x = 1.40$

$$p = C_2 \left[0.417 \pm \sqrt{0.340 - 0.333\, C_3 \left(\frac{C_1}{C_2}\right)^2}\right], \quad \ldots\ldots\ldots\ldots (5a)$$

and the velocity after the mixing:

$$w = \frac{C_2 - p}{C_1}.$$

Equation (5) gives two values for p (p_u, p_u''), which are coordinated with two velocities (w_u, w_u'') according to equation (6). A simpler connection can be shown between w_u and w_u'' if the heat contents i_e and i_a are set equal to each other in the expression for C_3. The supposition is made, in other words, that the same temperature exists in the ring chamber as in the atmosphere, which was approximately the case in later experiments. A simple calculation of (5) and (6) gives:

$$w_u'' \cdot w_u = a^{*2},$$

a relationship which always holds, as is well known, if a supersonic flow changes to a subsonic flow through a vertical compression shock. (Reference 4). This means in the foregoing case that the mixing of the sonic velocity w_2 in F_2 with the supersonic velocity w_1 in F_1 can proceed in such a manner that the average velocity after mixing $w_u'' > a^*$, and that this flow as for every supersonic flow with a compression shock can change to a subsonic flow with $w_u = \frac{a^{*2}}{w_u''}$. It can also be shown that, as opposed to incompressible flows, mixing to an average supersonic velocity is associated with a pressure decrease so that $p_u'' < p_k$, while a pressure increase to $p_u > p_k$ occurs after the compression shock.

Without going deeper into the essential relationships of a jet drive, the possibility should be studied at least theoretically, of the transition from supersonic flow in a supersonic diffusor (tube contracting) to a sonic velocity, and finally a transition to subsonic flow in a normal diffusor without finite compression shock. On the other hand, by foregoing the supersonic part of the diffusor, a mixing process could be expected at the beginning of which, through a decreasing pressure, the slow stream in the center of the tube is accelerated. The supersonic flow thus produced in the mixing tube or in the attached diffusor changes to a subsonic flow through a normal shock. It is anticipated from the results of the measurements on the previously-mentioned wind-tunnel model, that such a progression of pressure could not be observed, so that experiments were foregone with a contracting diffusor. A steep pressure increase exists immediately behind the mixing location, for which no satisfactory explanation can be given at present. There corresponds to the theoretically possible known types of flow, however, the observed progression of pressure of those in which the mixing is connected with a compression shock in the mixing tube. In these cases a pressure p_u is attained that follows from equation (5) with respect to the positive sign. Consideration has already been given in the calculation thus made of the pressure p_u to the reduction of the capacity of the air for doing work caused only by the turbulent mixing and the compression shock. A separate calculation of these losses is saved if the comparison of the test results with the calculations is made by opposition of the theoretically required and the actually used weights of driving material. The end pressure which can be attained through adiabatic compression in a large expansion attached to the cylindrical mixing section, amounts to:

$$p_e = p_u \left[\frac{x-1}{x} \frac{\frac{\rho_u}{2} w_u^2}{p_u} + 1 \right]^{\frac{x}{x-1}}$$

The actual pressure increase will be smaller than the theoretical value ($p_e - p_u$). Let η_{Di} designate the efficiency of the pressure conversion, then the attainable end pressure:

$$p_e^* = p_u + \eta_{Di} (p_e - p_u),$$

or, since $\rho_u w_u^2 = C_2 - p_u$:

$$p_e^* = p_u \left\{ 1 + \eta_{Di} \left[\left(0.143 \frac{C_2}{p_u} + 0.587\right)^{3.5} - 1 \right] \right\}.$$

This end pressure must agree with the external pressure p_o with the correct arrangement of p_a and β.

The efficiency of a diffusor for incompressible flow, that is, the ratio of the actually attained pressure increase to that theoretically possible, as given by the Bernoulli equation, amounts to about 0.85, or to 0.90 for very good expansions. The experimental values are not evident for compressible flows; however, it may be assumed that their ratios are unfavorable so that in the foregoing case in which the exit loss must be taken into account in the efficiency, η_{Di} probably approximates 0.70 to 0.75, which was also to be confirmed by advance experiments. Using these figures for η_{Di}, the connection between p_a and the area ratio β was calculated according to the previously described process, for an induced velocity $w_2 = a^*$; ($p_0 = 10{,}000\ kg/m^2$; $T_0 = T_a = 293°$; $\rho_0 = 0.119$, $a^* = 313\ m/s$, $i_0 = i_a$). The desired weight ratios then are:

$$\frac{G_1}{G_2} = \frac{F_1 \rho_1 w_1}{F_2 \rho_2 w_2} = \beta \varphi \psi.$$

Figure 2 illustrates these relationships from which it may be seen that the curves for η_{Di} = 0.70 and 0.75 do not differ much from each other. The required pressure p_a decreases in a hyperbolic manner with increase in area ratio, while the weight ratio increases approximately linearly. This ratio increases from about 0.14 to 0.32 in the practical range of β (0.05 to 0.20). (1) Figure 3 shows the weight ratios as functions of p_a. The significance of the test points plotted in Figure 2 will be treated further in Section IV.

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(1) The question as to whether the process calculated for the continuous case still holds for very small pressures in the ring chamber, will not be further investigated here since these small pressures need not be considered because of their unfavorable weight ratios.

IV. EXPERIMENTS ON A MODEL TUNNEL. (1)

A rotary compressor used for another purpose was available for the present investigation. This compressor had a suction volume of $5700 m^3$ per hour, and a maximum pressure of 4.3 (at) (atmospheres). The dimensions of the model tunnel were so chosen that a continuous drive was possible with this compressor, in order not to hinder the experiments through too short a measuring time. Figure 4 shows the test section with the main dimensions. The narrowest diameter was 148 mm ($F_2 = 0.0172 m^2$) to which was attached a very easy expansion to 149.3 mm. The end of the suction section was expanded to 154.3 mm because of structural reasons. This shape of the test section was found to be favorable from previous experiments. The dimensions of the drive nozzle (D_1, D_1') could be changed by removable sections so that the area ratio could be varied from $\beta = \frac{F_1}{F_2} = 0.063$, 0.110, to 0.156. The ring-shaped nozzle exit cross section F_1 is thus immediately obtainable. The narrowest cross section F_1 of the drive nozzle is adjusted according to the value of the pressure in the ring chamber, which is taken from Figure 2 for $\eta_{Di} = 0.70$. F_1' is therefore determined on the assumption of no-loss adiabatic expansion to $p_k = 0.528\, p_o$. (Reference 5). The following table contains the most important characteristics of the three drive nozzles used.

Nozzle	$\beta = \frac{F_1}{F_2}$	F_1 (m^2)	p_a (at)	$\frac{p_k}{p_a}$	$\frac{F_1'}{F_1}$	F_1' (m^2)
I	0.063	0.00108	4.2	0.125	0.580	0.00063
II	0.110	0.00189	2.6	0.203	0.745	0.00140
III	0.156	0.00268	2.1	0.252	0.820	0.00220

(1) The tests were carried out by B. Bünger and P.v.d. Knesebeck.

The accuracy of set-up with respect to the areas amounted to about two or three percent.

The wind tunnel model was arranged so that it could be tested with a cylindrical mixing tube about 0.8 m long and attached diffusor, as well as without this tube. The length of the diffusor was about 3.0 m, the expansion angle about 8°, the end cross section 0.292m² $\approx$ 17.0 F_2. The entire length of the diffusor was subdivided so that tests could also be made with half its length. The weight of driving material G_1, was measured by a diaphragm built into the air line, while the induced weight G_2 was determined from a pressure orifice p_1 in the test section. (See Figure 4).

Figure 5 shows the ratio of induced velocity to critical velocity of sound as a function of the ring chamber pressure for the various nozzles for the arrangement without mixing tube. For each jet and with a definite pressure, the maximum velocity was attained which did not change further with further increase of pressure, at least within the available test range. The fact that the attainable maximum velocity was somewhat above the velocity of sound with these arrangements is probably due to the shape of the entrance cone. The lowest pressure required to produce the highest velocity is easily seen from the plot in Figure 5 to be least for the larger area ratios. This figure also shows the weight ratios that were obtained from the measurements. The actual lowest pressure and the corresponding weight ratio are plotted also in Figure 2, and show good agreement with the calculated curve. The actual weight ratios are therefore not unfavorable, as they could be expected to be from considerations of mixing loss, shock loss, and normal pressure transition loss in the diffusor.

The good agreement of the test results with the previous calculations is no proof, of course, that the actual flow processes of the mixing are not of a very complicated type. It should be noted in this connection that, for purposes of comparison, it was assumed and also carried out in the experiments, that a diffusor was attached immediately following the mixing plane. It was found that, after installation of the cylindrical mixing tube in front of the diffusor, a pressure increase of about 0.5 (at) was required to produce the maximum velocity with nozzle II. The maximum velocity is therefore somewhat less than without this intermediate tube. The basis for this decreased performance is obviously to be found in the relatively large friction losses in the cylindrical tube. Figure 6 gives the comparison of these two arrangements, as well as the results of the tests with the half length of diffusor. It therefore follows that the first arrangement is the most favorable, and that the shortening of the diffusor because of space requirements causes no appreciable disadvantages. The induced velocity without a diffusor was, as expected, very low.

The pressure distribution determined from the wall orifices is plotted in Figure 7 ($p_a = 3$ at) for the arrangements without and with an intermediate tube in front of the diffusor . The test locations p_1 to p_4 are along the suction tube, p_5 is at the narrowest cross section of the drive nozzle, p_6 is at the mixing plane, and the remaining orifices are along the cylindrical tube and the diffusor. The pressure in the suction section is fairly uniform. A steep pressure increase occurs in all cases immediately behind the mixing plane. A pressure which is only slightly lower than atmospheric is quickly attained when the intermediate tube is omitted; a pressure in the mixing tube is soon attained that changes only slightly up to the beginning of the expansion. This pressure is somewhat higher than the pressure p_u calculated from equation (5a). The supposition made in the calculation that the pressure at the exit of the drive nozzle is equal to p_k is at least approximately fulfilled.

The power required for continuous drive was calculated in HP per cm² of cross section in order to obtain a clear picture of the power consumption of a wind tunnel installation with jet drive. The drive power required to produce the compressed air of pressure p_a according to Figure 2 (η_{Di} = 0.7) was calculated on the basis of an isothermal compressor efficiency of η_{is} = 0.6 and plotted in Figure 8 against the area ratio. The power required amounts on the average to about 0.9 HP per cm². Since the theoretical values of Figure 2 correspond fairly well to the test values, these power estimates can therefore serve as a basis for the design of a jet drive. Finally, the efficiency factor k useful for estimating normal wind tunnel installations, is plotted on the same diagram, (Figure 8). This value amounts on the average to 0.55, therefore about half as good as for wind tunnel installations with closed test section and fan drive, in so far as comparative values are available for such installations of low velocities.

It is not difficult to estimate a suitable area ratio $\frac{F_1}{F_2}$ to $\frac{F_1'}{F_2}$ for an intermittent jet drive with a pressure vessel, on the basis of the foregoing results. There can be calculated on the one hand the actual volume relationships for the pressure vessel, and on the other hand the highest power for the compressors, and finally the useful test time.

It should be pointed out that an induced velocity $w_2 < a^*$ can be produced in the test section in the customary manner if the cross section in front of the mixing plane is contracted by an adjustable nozzle, (easily performed by using a rectangular cross section). The velocity in the test section is then independent of a definite minimum pressure from that in the ring chamber.

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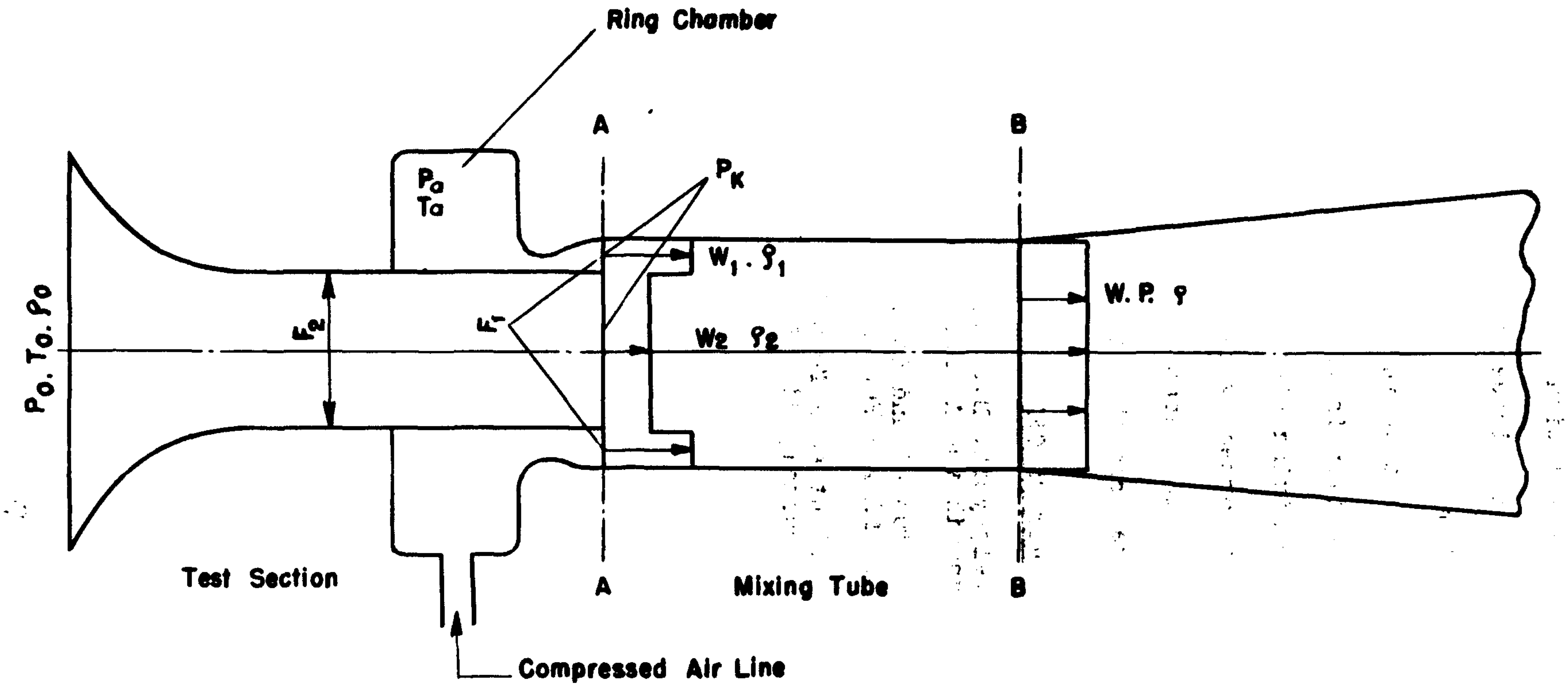

Figure 1 - High Velocity Test Section with Jet Drive
(Schematic Diagram)

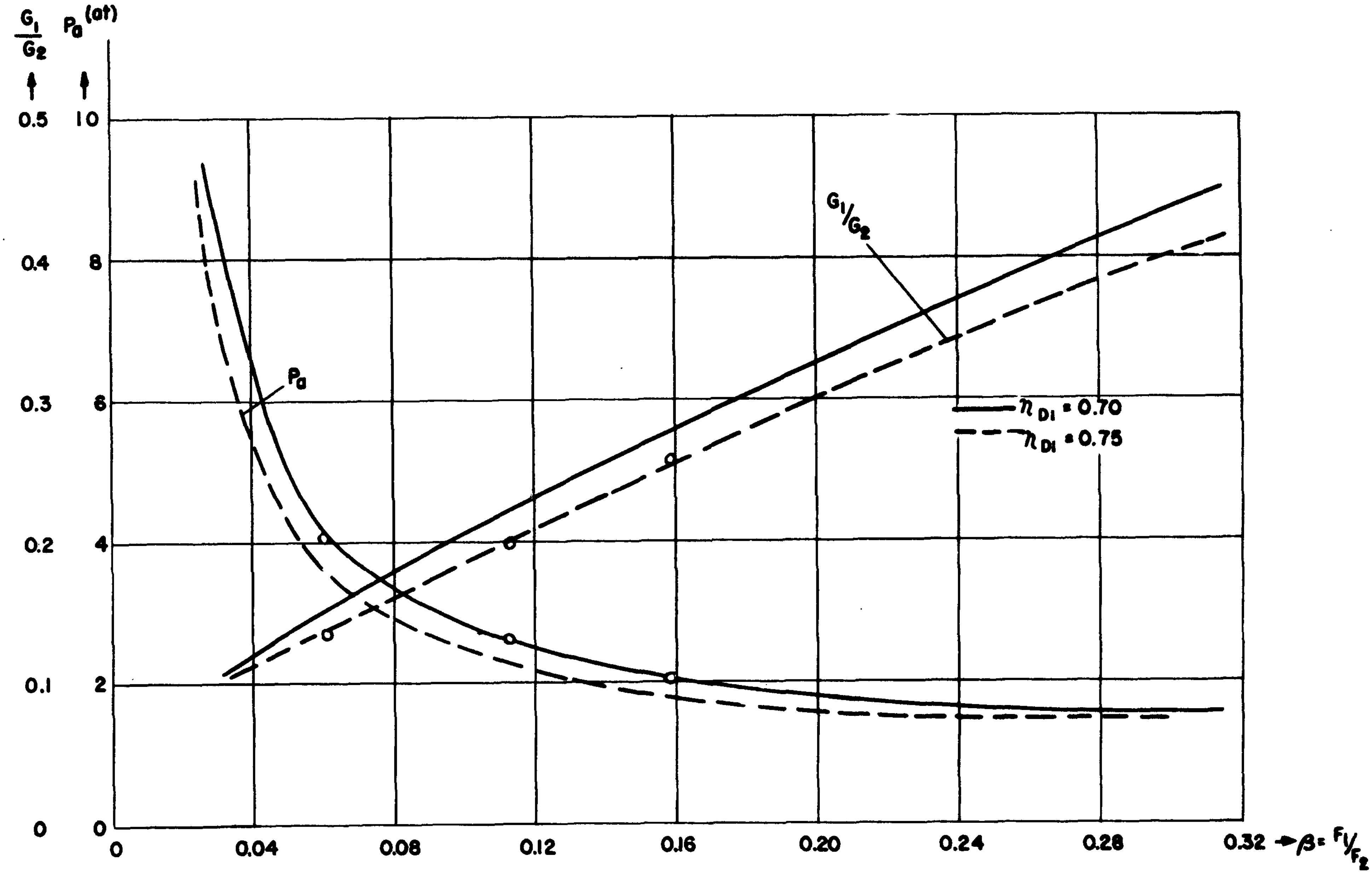

Figure 2 – Ring Chamber Pressure P_a and Weight Ratio $\frac{G_1}{G_2}$ as a Function of the Cross-Section Ratio $\frac{F_1}{F_2}$, Calculated for $W_2 = a^*$, ($P_0 = 10{,}000$ kg/m^2, $T_0 = T_a = 293°$, $a^* = 313$ m/s). The Plotted Test Points are for the Arrangement without Mixing Tube and with Long Diffusor, (L = 3.0 m)

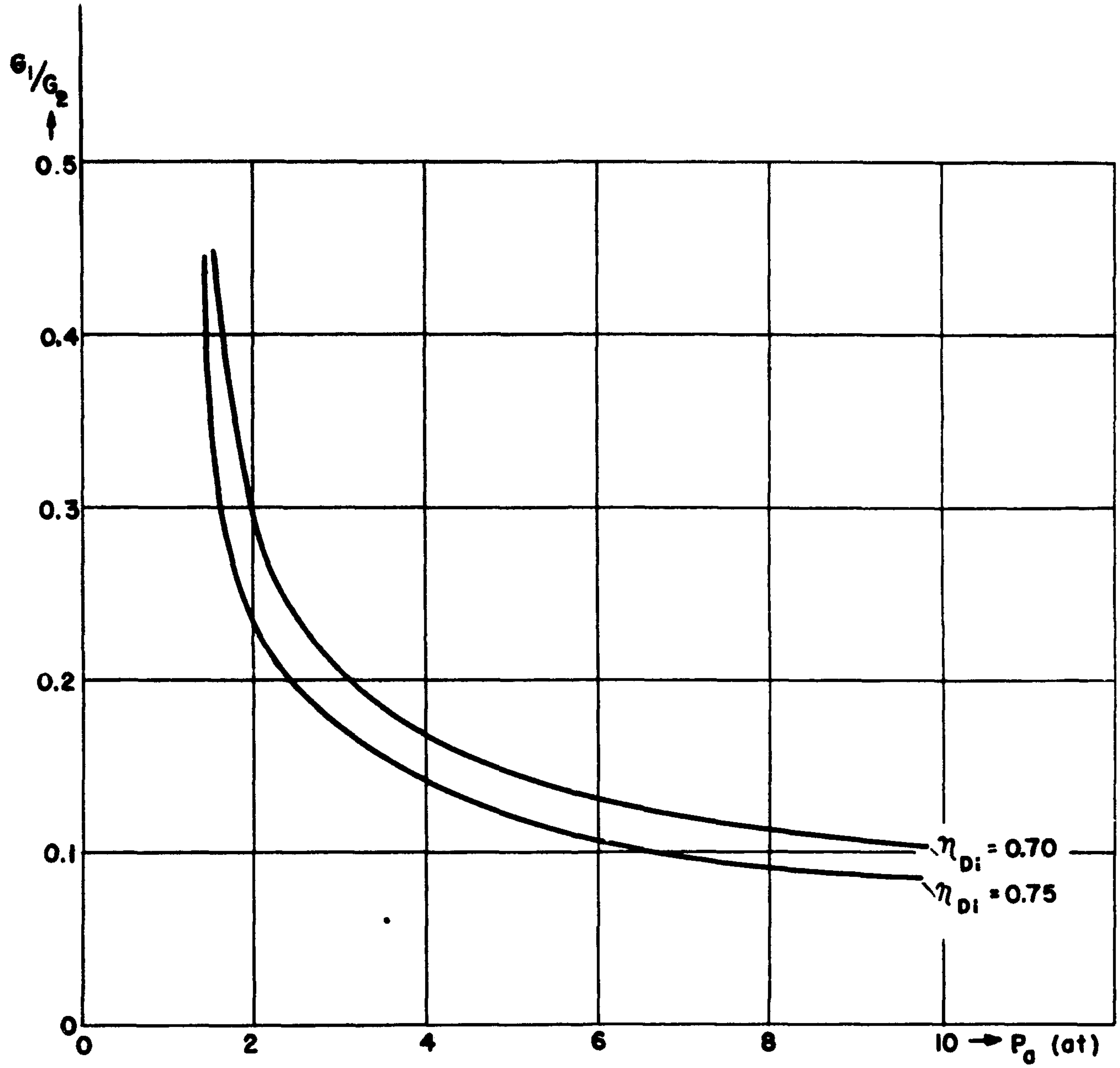

Figure 3 – Weight Ratio $\frac{G_1}{G_2}$ as a Function of the Ring Chamber Pressure P_a.

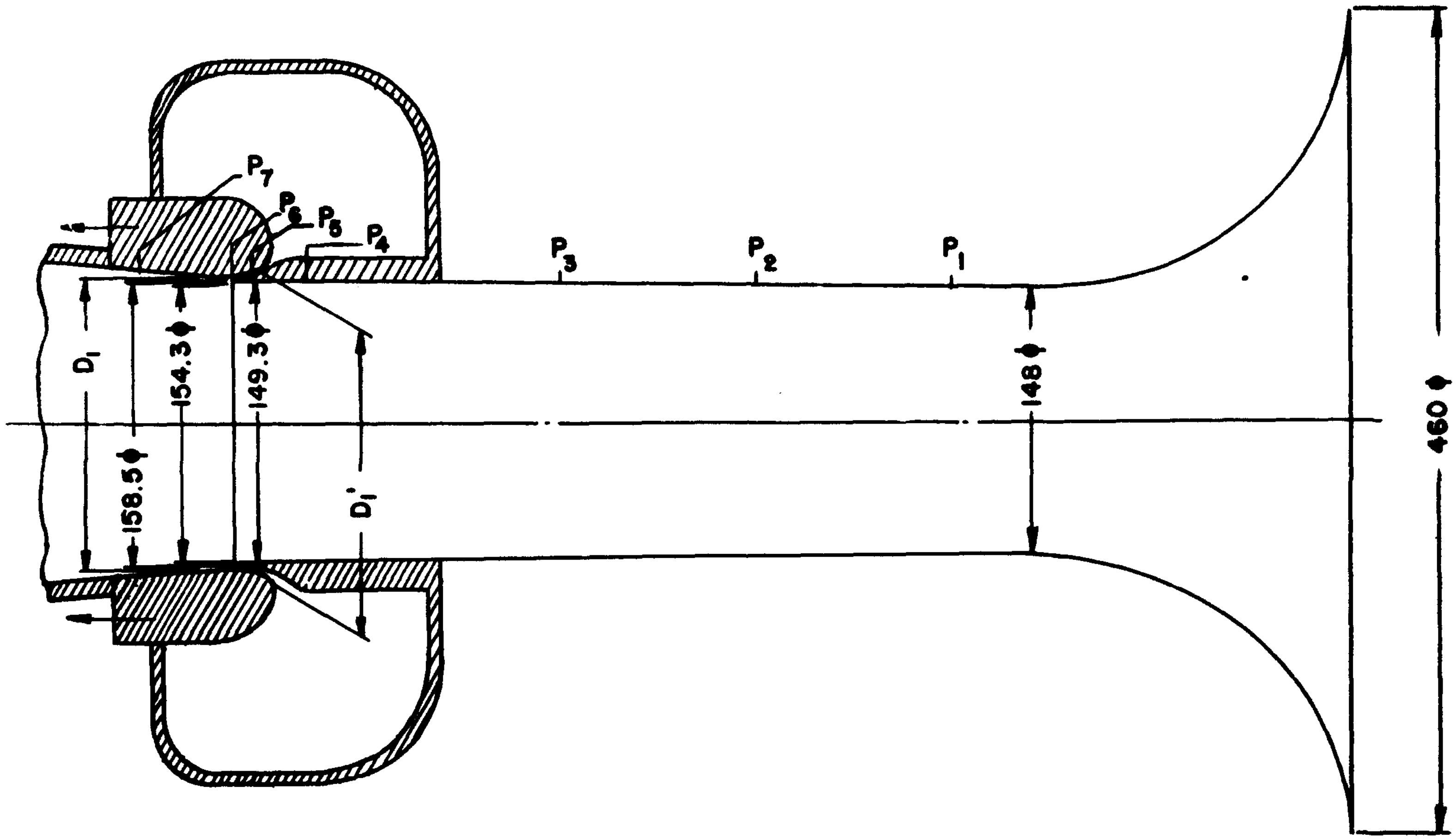

Figure 4 – Arrangement of the Test Section with the Drive Nozzle

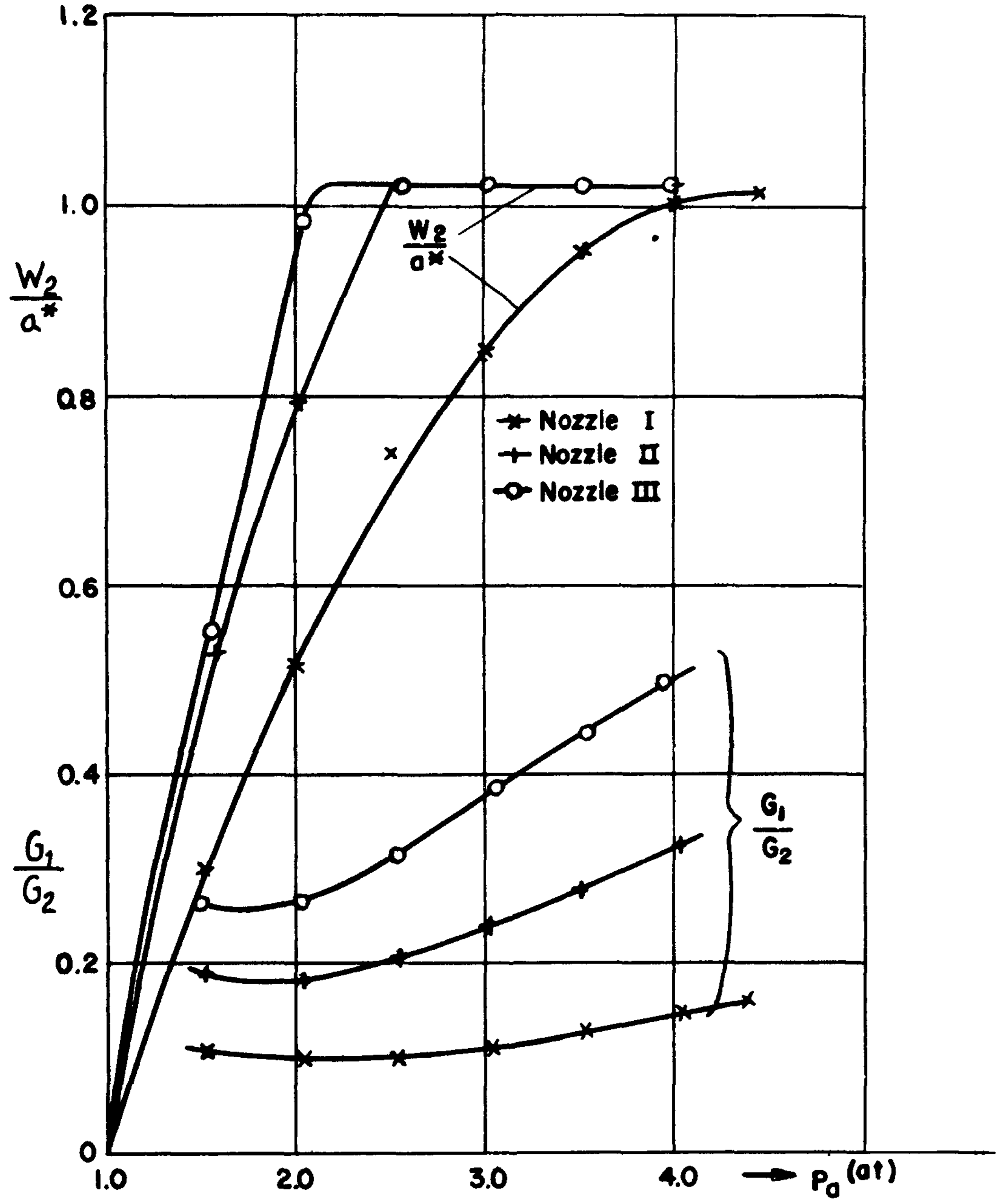

Figure 5 – Velocity in the Test Section, $\frac{W_2}{a^*}$, and Weight Ratio $\frac{G_1}{G_2}$ for Nozzles I, II, and III as a Function of the Ring Chamber Pressure P_a, without Mixing Tube.

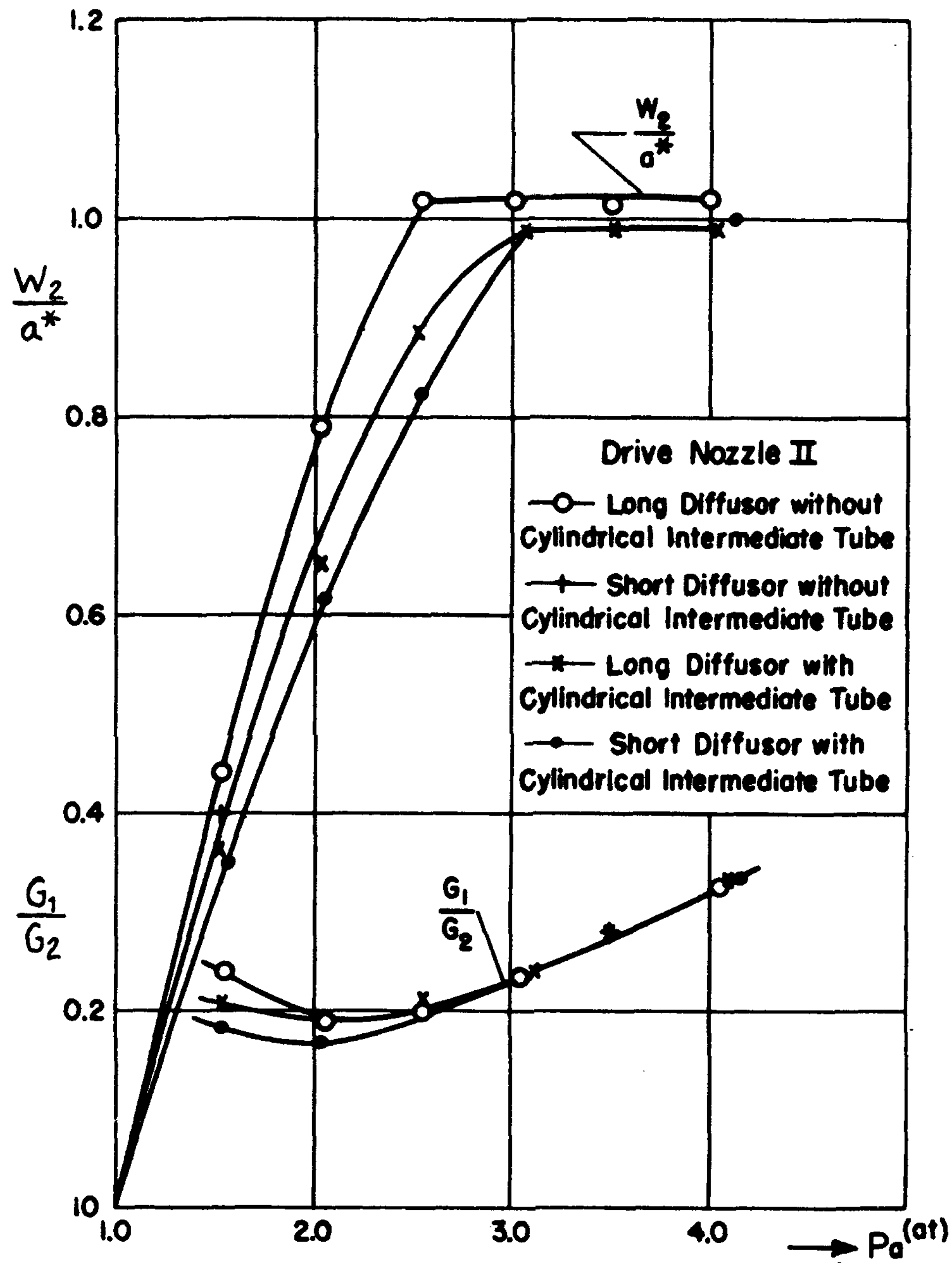

Figure 6-Velocity in the Test Section, $\frac{W_2}{a^*}$, for Nozzle II without and with Mixing Tube, Whole and Half Length Diffusor.

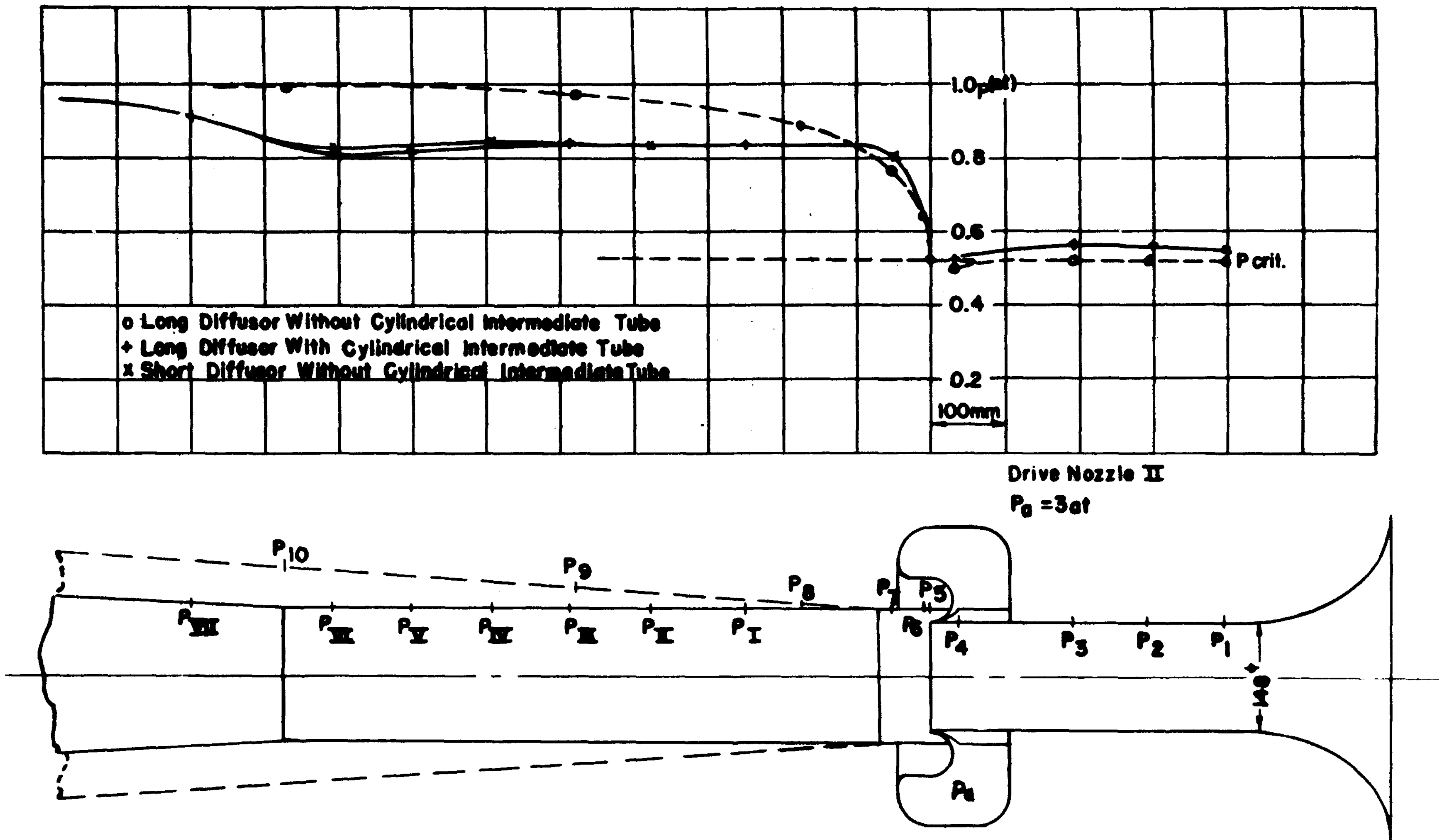

Figure 7— Pressure Distribution Determined from Wall Orifices for Various Arrangements

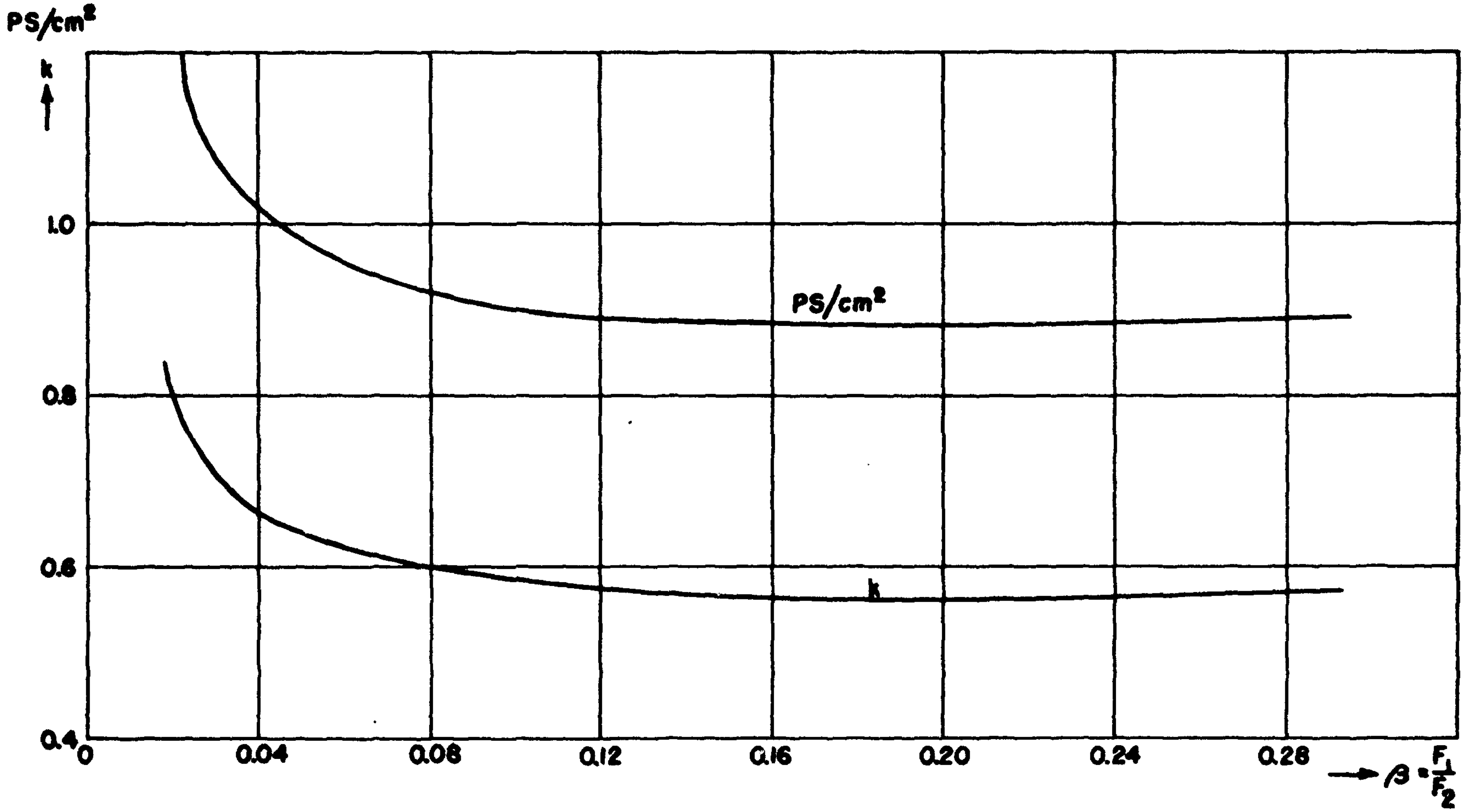

Figure 8 – Power Consumption of the Jet Drive per cm^2 of Nozzle Area, and the Efficiency Factor of the Wind Tunnel.

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