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The Prediction of the Optimum Performance of Venturi-Type Wind-Energy Concentrators

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THE PREDICTION OF THE OPTIMUM PERFORMANCE OF VENTURI-TYPE WIND-ENERGY CONCENTRATORS

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Abstract

A simple theoretical analysis is presented which allows the optimum performance to be predicted for venturi-type wind-energy concentrators. The results of the analysis are presented in convenient parametric form in such a manner that, for example, the influence on optimum performance of reducing the outlet area of the exit diffuser, an obvious means of reducing the bulk and cost of a venturi-type concentrator, can be established directly from the curves. It is shown that the most important parameters affecting concentrator performance are: the effectiveness of the exit diffuser, the area ratio of that diffuser and the magnitude of the base-drag coefficient applicable at the diffuser exit plane. It was found that base-drag has the beneficial effect of lowering, below that of the surroundings, the static pressure prevailing at the outlet end of a concentrator.

1. INTRODUCTION

It has been shown by Betz (1) that an ideal, non-ducted, wind-turbine can extract only 16/27 (or 59.3%) of the kinetic energy of the wind based on a flow area equal to the cross-sectional area of the turbine when projected onto a plane normal to the wind direction. The 16/27 factor arises due to the slowing down of the flow which occurs as fluid passes through the rotor of a non-ducted machine which, due to the absence of a shroud duct, cannot sustain a drop in static pressure a-cross the rotor. The implication is, therefore, that for maximum power production in a wind of prescribed velocity, the cross-sectional area, well upstream of the rotor, of the stream-tube passing through the rotor is less than the cross-sectional area of the rotor. Furthermore, the flow leaving the rotor carries with it residual kinetic energy. In fact Betz (1) showed that the ideal leaving velocity, downstream of the rotor, is 1/3 of the velocity of the wind.

Without doing anything to increase the kinetic energy available, per unit cross-sectional area, in the undisturbed flow approaching a wind-turbine it is possible to increase the output, above the Betz limit (1), per unit projected area of turbine rotor by "focussing" or concentrating the oncoming flow into a smaller cross-sectional area than that of the undisturbed stream. A device for performing this task is generally known as a concentrator. In a ventrui-type concentrator a specially designed turbine is mounted at the throat of a venturi-like duct as shown in Fig. 1. The ducted turbine is of a form which can sustain a drop in static pressure equal to the drop in stagnation pressure which occurs due to energy extraction. The ducted turbine differs, therefore, from a machine suitable for non-ducted operation.

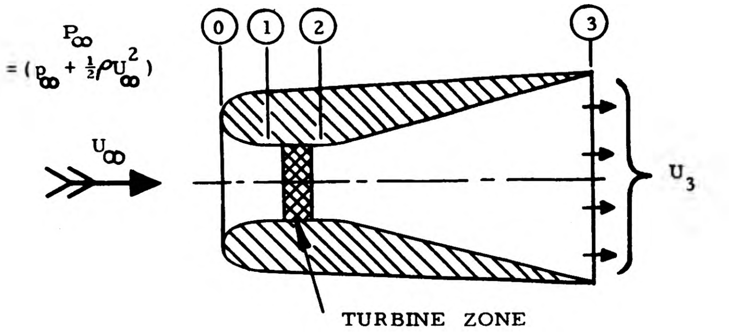

FIG. 1 TYPICAL VENTURI-TYPE WIND-ENERGY CONTRATOR

The diffuser of a venturi-type concentrator serves the very important function of slowing down, as reversibly as possible, the comparatively high velocity flow leaving the turbine. This results in the recovery of the greatest possible proportion of the kinetic energy of the turbine exhaust stream. Consequently the static pressure at the exit plane of the diffuser is thus sufficient for discharge to occur into the atmosphere downstream of the concentrator. Due to a suction, or base drag, effect caused by the relatively fast moving external flow passing over the outside of the concentrator the static pressure at the diffuser exit plane is normally lower than the static pressure of the surroundings.

Because of the acceleration of the flow occurring in the concentrator the static pressure at station 1 is lower, and due to the turbine pressure drop that at station 2 is generally substantially lower, than the static pressure of the flow surrounding the concentrator. Since the cross-sectional areas of stations 1 and 2 are equal, the axial components of velocity at these stations are also equal: the flow is, of course, assumed to be incompressible.

Figure 2 shows a more sophisticated form of venturi-type concentrator in which the boundary layer in the diffuser is energised by means of an annular jet-flow drawn from the surroundings. This usually results, for a prescribed diffuser area ratio, in a higher diffuser effectiveness and a shorter unit than for the configuration of Fig. 1. Igra (2) and other workers (3,4) have carried out tests on concentrators of the types shown in Fig. 1 and

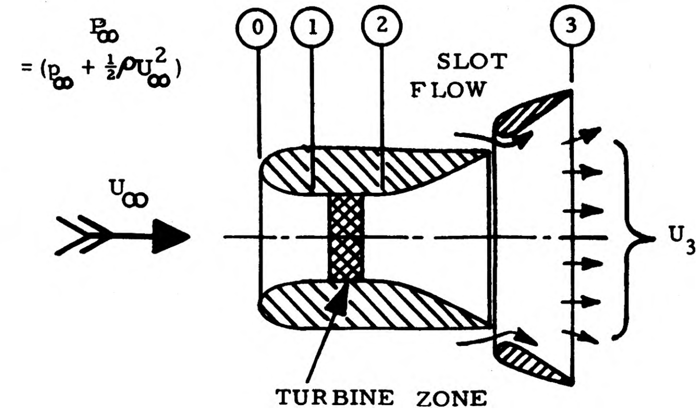

FIG. 2 SHORT VENTURI-TYPE CONCENTRATOR WITH BOUNDARY LAYER CONTROL JET IN DIFFUSER

2. For test purposes Igra (2) used, in his model concentrators, screens to generate pressure drops corresponding to the presence of appropriate turbines. The objective of the present work is to present a theoretical analysis of the venturi-concentrator problem defining, in parametric form, the optimum performances of a range of configurations.

2. THEORY

The derivation of equations follows which allows the performances of venturi-type concentrators to be expressed in terms of appropriate variables.

2.1 NOTATION

Symbol	Definition
A	cross-sectional area
C_D	turbine pressure drop coefficient $\equiv (p_1-p_2)/\frac{1}{2}\rho U_t^2$
C_{DB}	base drag coefficient $\equiv (p_\infty-p_3)/\frac{1}{2}\rho U_{REL}^2$
m	dimensionless velocity at diffuser exit $\equiv U_3/U_\infty$
p	static pressure
P	stagnation, or total, pressure
r	power ratio: ratio of output of ideal turbine in concentrator to that of Betz output of non-ducted turbine of equal size
U	flow velocity
U_{REL}	relative flow velocity $\equiv (U_\infty-U_3)$
η_D	diffuser effectiveness $\equiv \dfrac{p_3 - p_2}{\rho/2\{U_t^2 - (mU_\infty)^2\}}$

Symbol	Definition
ρ	density of air (incompressible flow)
Subscripts	
1,2,3	stations within concentrator (see Fig. 1 and 2)
(MAX)	Maximum value of the subscripted variable
t	throat zone of venturi (Note: $A_1 = A_t = A_2$)
∞	pertaining to surroundings conditions
Superscript	
*	superscripted variable corresponds to the conditions for $r_{(MAX)}$

2.2 ANALYSIS

The following assumptions apply to the analysis of the flow through a venturi-type concentrator:

a) the flow can be regarded as incompressible,

b) flow is reversible, i.e. losses are absent, within the contracting flow passage between 0 and 1,

c) flows at the venturi throat-stations 1 and 2 are axial. Strictly this condition implies the presence of flow-straightening guide vanes downstream of the turbine rotor,

d) flow is one-dimensional.

2.2.1 Power Ratio r

From the definition of power ratio r:

$$r = \frac{U_t A_t (P_1 - P_2)}{\frac{16}{27} \cdot \frac{1}{2} \rho U_\infty^3 A_t} \tag{1}$$

Application of Bernoulli's equation at stations 1 and 2 gives

$$P_1 = p_1 + \tfrac{1}{2}\rho U_1^2$$

$$P_2 = p_2 + \tfrac{1}{2}\rho U_2^2$$

or since, from continuity:

$$U_1 = U_t = U_2$$

hence:

$$P_1 - P_2 = p_1 - p_2 \tag{2}$$

Substitution in equation (1) from (2), and from the definition of C_D, leads to the result:

$$r = \frac{27}{16} \left(\frac{U_t}{U_\infty}\right)^3 C_D \tag{3}$$

This is the equation for power ratio r given by Igra (2). Obviously equation (3) can only be applied when U_t/U_∞ and C_D are both known. An alternative expression for r is required for analytical purposes which allows the maximum value of r, $r_{(MAX)}$, to be evaluated in terms of η_D, C_{DB} and m.

2.2.2 Maximum Power Ratio $r_{(MAX)}$

Consider the general case of a concentrator for which the internal performance of the diffuser is expressed in terms of a diffuser effectiveness, η_D, and for which the influence of base drag occurs at the exit of the diffuser thereby resulting in a suction at station 3 giving, at that station, a static pressure lower than that of the surroundings. From the definitions of η_D and the base drag coefficient C_{DB}:

$$\eta_D = \frac{p_\infty - p_2 - C_{DB}\tfrac{1}{2}\rho(1-m)^2 U_\infty^2}{\tfrac{1}{2}\rho\{U_t^2 - (mU_\infty)^2\}} \tag{4}$$

and from the definition of η_D:

$$p_3 = p_2 + \eta_D \tfrac{1}{2}\rho\{U_t^2 - (mU_\infty)^2\} \tag{5}$$

also from the definitions of C_{DB}, U_{REL} and m:

$$p_3 = p_\infty - C_{DB}\tfrac{1}{2}\rho\{1-m\}^2 U_\infty^2 \tag{6}$$

From subtraction of equation (5) from equation (6):

$$p_2 = p_\infty - C_{DB}\tfrac{1}{2}\rho\{1-m\}^2 U_\infty^2 - \eta_D\tfrac{1}{2}\rho\{U_t^2 - (mU_\infty)^2\} \tag{7}$$

Also application of Bernoulli's equation to loss-free flow in the inlet gives:

$$p_1 = p_\infty + \tfrac{1}{2}\rho U_\infty^2 - \tfrac{1}{2}\rho U_t^2 \tag{8}$$

thus from equations (7) and (8):

$$p_1 - p_2 = \tfrac{1}{2}\rho U_\infty^2 [1 + C_{DB}(1-m)^2 - \eta_D m^2] - \tfrac{1}{2}\rho U_t^2 [1 - \eta_D] \tag{9}$$

Thus substitution for $P_1 - P_2$ in equation (1) from equation (2) and subsequently substituting for $p_1 - p_2$ from equation (9) gives the following result after simplification:

$$r = \frac{27}{16}\left\{[1 + C_{DB}(1-m)^2 - \eta_D m^2]\frac{U_t}{U_\infty} - [1-\eta_D]\left(\frac{U_t}{U_\infty}\right)^3\right\} \tag{10}$$

Differentiating equation (10) with respect to U_t/U_∞ to establish a maximum leads to the

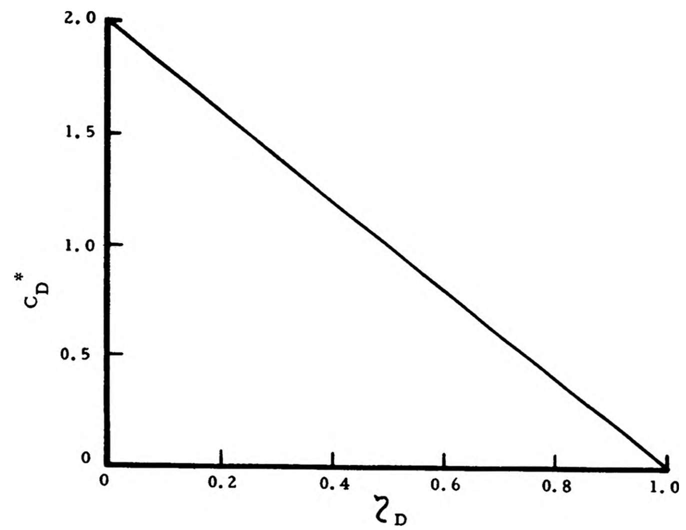

FIG. 3 PREDICTED TURBINE PRESSURE DROP COEFFICIENT C_D^*, CORRESPONDING TO THE MAXIMUM POWER RATIO, VERSUS DIFFUSER EFFECTIVENESS η_D.

result, for invariant η_D, C_{DB} and m:

$$\frac{dr}{d(U_t/U_\infty)} = 0 = [1+C_{DB}(1-m)^2-\eta_D m^2]-3[1-\eta_D](\frac{U_t}{U_\infty})^2$$

a second differentiation confirms that this condition describes a maximum of $dr/d(U_t/U_\infty)$.

Hence, after invoking the definition of the * superscript:

$$\frac{U_t^*}{U_\infty} = \sqrt{\frac{1 + C_{DB}(1-m)^2 - \eta_D m^2}{3(1-\eta_D)}} \qquad (11)$$

Substitution of (U_t^*/U_∞) for (U_t/U_∞) in equation (10) and subsequently substituting for (U_t^*/U_∞) from equation (11) gives, after simplification, the final result:

$$r_{(MAX)} = \frac{9}{8}\sqrt{\frac{1+C_{DB}(1-m)^2-\eta_D m^2}{3(1-\eta_D)}}\{1+C_{DB}(1-m)^2- \eta_D m^2\} \qquad (12)$$

It now remains to establish corresponding expressions for C_D^* and A_3^*/A_2.

Rewriting equation (3) for the condition for $r_{(MAX)}$ gives:

$$r_{(MAX)} = \frac{27}{16}\left(\frac{U_t^*}{U_\infty}\right)^3 C_D^* \qquad (13)$$

It can be shown from comparison of equations (12) and (13) that:

$$C_D^* = 2(1-\eta_D) \qquad (14)$$

It is interesting to note that equation (14) is independent of both C_{DB} and m.

The diffuser area ratio corresponding to $r_{(MAX)}$ can be established from an application of the continuity condition between the throat of the concentrator and the diffuser exit, station 3, thus:

$$\rho U_t^* A_t = \rho m U_\infty A_3^*$$

hence, since $A_t = A_2$:

$$\frac{A_3^*}{A_2} = \frac{1}{m}\left(\frac{U_t^*}{U_\infty}\right) \qquad (15)$$

Equations (11), (12), (14) and (15) allow the performance of optimised venturi-type concentrators to be established for prescribed values of η_D, C_{DB} and m. The results of parametric studies based on these equations are presented in Fig. 3, 4, 5 and 6.

2.2.3 Limiting Diffuser Area Ratios

It is apparent, after substitution for (U_t^*/U_∞) in equation (15) from equation (11), that within the range $0 \leq \eta_D \leq 1.0$ A_3^*/A_2 is

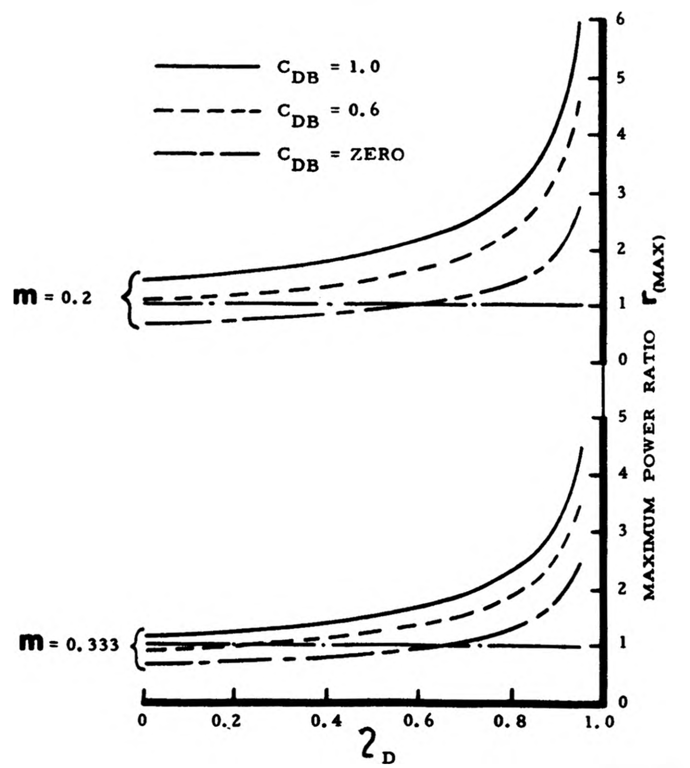

FIG. 4 PREDICTED MAXIMUM POWER RATIO, $r_{(MAX)}$, VERSUS DIFFUSER EFFECTIVENESS, η_D, FOR VARIOUS VALUES OF THE BASE DRAG COEFFICIENT C_{DB}. (DIFFUSER EXIT VELOCITY RATIO, m, = 0.333 and 0.2)

infinite when m is zero. This represents the upper bound limit of A_3^*/A_2 and the lower limit of m. Curves displaying the relationship

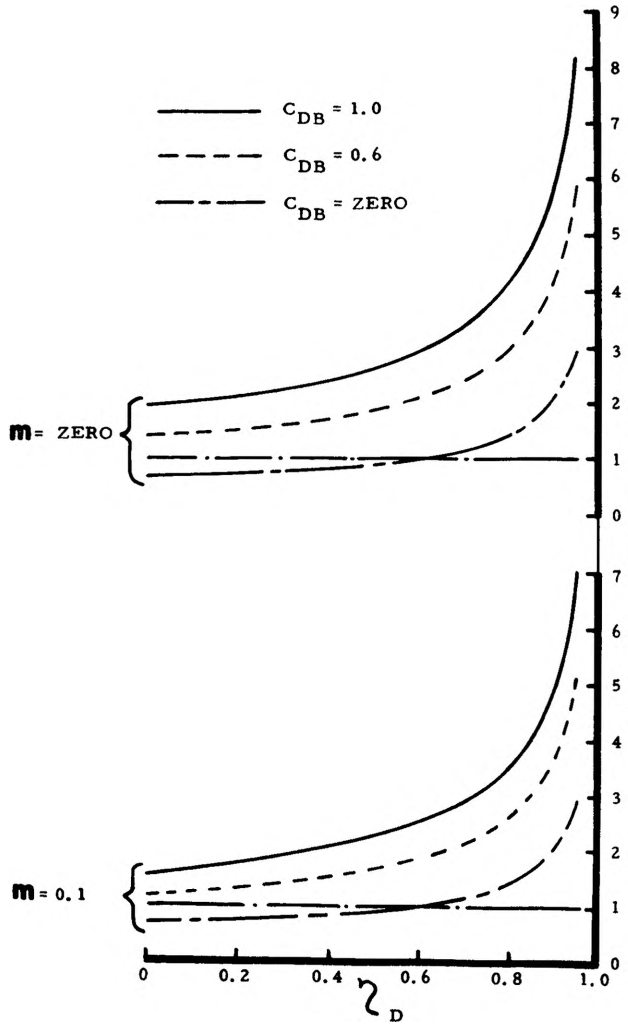

FIG. 5 PREDICTED MAXIMUM POWER RATIO, $r_{(MAX)}$, VERSUS DIFFUSER EFFECTIVENESS, η_D, FOR VARIOUS VALUES OF THE BASE DRAG COEFFICIENT C_{DB}. (DIFFUSER EXIT VELOCITY RATIO, m, = 0.1 AND ZERO)

between $r_{(MAX)}$ and η_D are presented, with parameters of C_{DB}, in Fig. 5 for the condition $m = 0$.

Another case of some practical interest is that corresponding to the absence of a diffuser. This case is modelled by setting $\eta_D=0$ and $A_3^*/A_2 = 1$. Since from equations (11) and (15):

$$\frac{A_3^*}{A_2} = \frac{1}{m}\sqrt{\frac{1+C_{DB}(1-m)^2-\eta_D m^2}{3(1-\eta_D)}} \tag{16}$$

hence when $A_3^*/A_2 = 1$, the lower limiting diffuser area ratio, and $\eta_D = 0$ equation (16) leads to a result defining, in terms of C_{DB}, what can be shown to be the maximum physically meaningful value of m. This value of m has been identified as $m_{(MAX)}$:

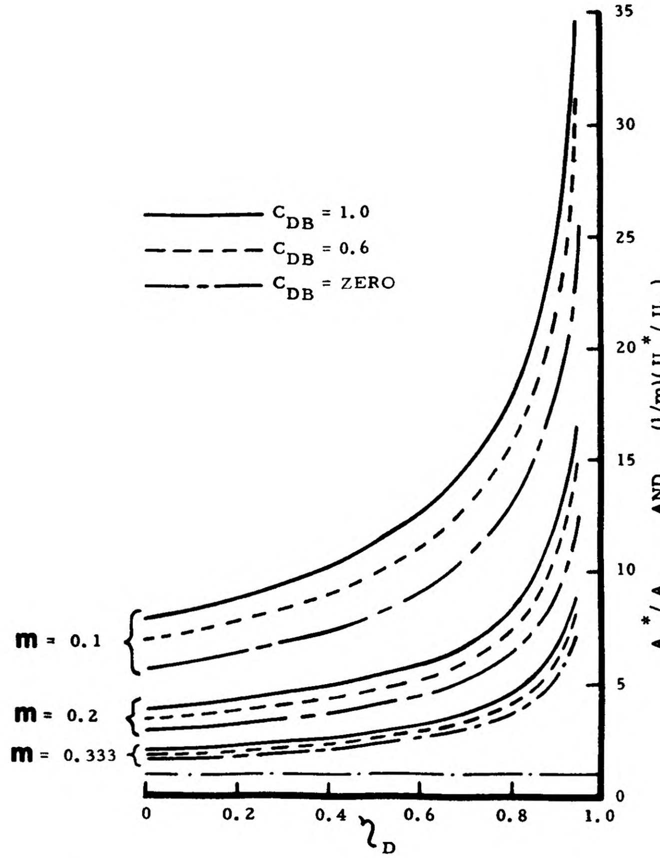

FIG. 6 PREDICTED DIFFUSER AREA RATIO A_3^*/A_2 CORRESPONDING TO $r_{(MAX)}$, VERSUS DIFFUSER EFFECTIVENSS η_D FOR VARIOUS VALUES OF C_{DB} AND m.

$$1 = \frac{1}{m_{(MAX)}}\sqrt{\frac{1 + C_{DB}(1-m_{(MAX)})^2}{3}} \tag{17}$$

The values of $r_{(MAX)}$ corresponding to those of $m_{(MAX)}$ can be established, for prescribed C_{DB}, from equation (12) when $\eta_D = 0$ and m is replaced by $m_{(MAX)}$ obtained from equation (17). The results of such an evaluation are presented graphically in Fig. 7.

2.2.4 Diffuser Exit-Plane Pressure Coefficient

The concentrator diffuser exit-plane stagnation pressure can be expressed, conveniently, in terms of a pressure coefficient $(P_3-p_\infty)\rho\frac{1}{2}U^2$. This definition is such that a diffuser exit-plane pressure coefficient of unity implies the extraction of zero energy from the flow passing through the concentrator. Negative values of the coefficient indicate a diffuser exit stagnation pressure lower than the static pressure, p_∞, of the surroundings.

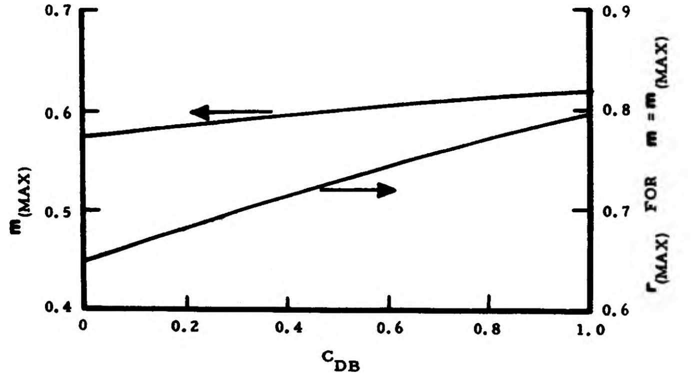

FIG. 7 PREDICTED MAXIMUM VALUE OF DIFFUSER EXIT VELOCITY RATIO $m_{(MAX)}$, AND CORRESPONDING $r_{(MAX)}$, VERSUS BASE DRAG COEFFICIENT C_{DB}.

The exit-plane pressure coefficient can be evaluated easily when it is expressed in terms of m and C_{DB}. From Bernoulli's equation applied at station 3:

$$P_3 = p_3 + \tfrac{1}{2}\rho m^2 U_\infty^2$$

or, after invoking the definitions of C_{DB} and m and hence expressing p_3 in terms of p_∞, m and C_{DB}:

$$P_3 = p_\infty + \tfrac{1}{2}\rho U_\infty^2\{m^2 - C_{DB}(1-m)^2\}$$

hence:

$$(P_3 - p_\infty)/\tfrac{1}{2}\rho U_\infty^2 = \{m^2 - C_{DB}(1-m)^2\} \qquad (18)$$

An evaluation, from equation (18), of the diffuser exit-plane pressure coefficient is presented graphically, versus C_{DB} with parameters of m, in Fig. 8.

3. RESULTS

Figure 3 presents a graphical representation of the turbine pressure drop coefficient, C_D^*, corresponding to $r_{(MAX)}$. The linear relationship is in accordance with equation (14). The interesting feature is that C_D^* is a unique function of η_D and is independent of m and C_{DB}. Physically this independence arises due to the definition of C_D^* which incorporates in its denominator $(U_t^*)^2$. Hence since, from equation (11), it can be seen that $(U_t^*/U_\infty)^2$ is proportional to both m and C_{DB} and is inversely proportional to $(1-\eta_D)$ it is apparent that C_D^* is inherently compensated for variations in m and C_{DB}. It is also clear from Fig. 3 that the lower the diffuser effectiveness the higher C_D^*. The condition $C_D^* = 0$ when

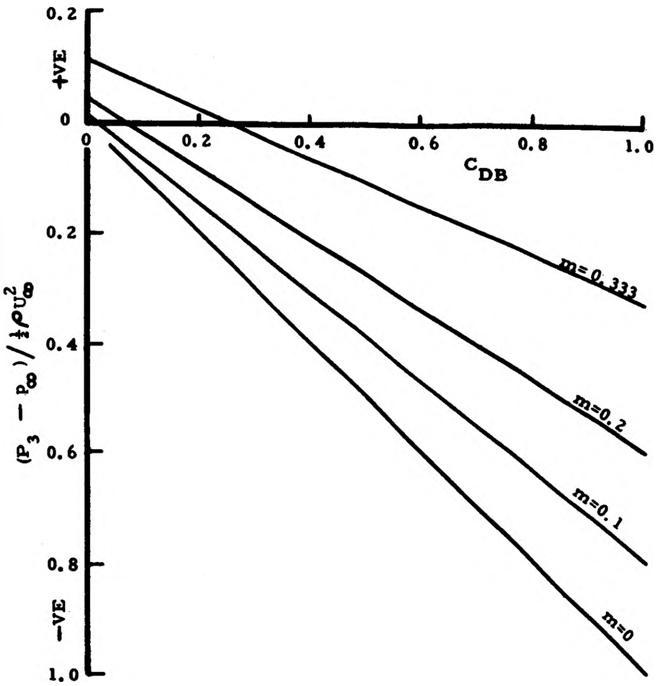

FIG. 8 DIFFUSER EXIT-PLANE DYNAMIC PRESSURE-COEFFICIENT VERSUS BASE DRAG COEFFICIENT C_{DB}.

$\eta_D = 1$ is consistent, as can be seen from equation (11), with an infinite value of (U_t^*/U_∞) corresponding to the use of an, hypothetical, ideal diffuser.

Figures 4 and 5 present plots of $r_{(MAX)}$ versus η_D for four values of m ranging from zero to 0.333. For each value of m three values of the base drag coefficient, C_{DB}, were considered. The curves emphasise the importance, in order to obtain a high value of $r_{(MAX)}$, of achieving the highest possible values of η_D and C_{DB} and the lowest possible value of m. Generally high η_D and C_{DB} appear, on the basis of experimental work of others (2,3,4), to be consistent with the use of slotted diffusers of the kind shown in Fig. 2. However a low value of m at high η_D implies a very large diffuser area ratio, not a configuration consistent with low capital costs.

Diffuser area ratios, corresponding to $r_{(MAX)}$, are presented as functions of η_D in Fig. 6 for values of m of 0.1, 0.2 and 0.333. The relatively enormous increases in diffuser area ratio associated with comparatively small improvements in η_D, and reductions in m, are apparent, for efficient diffusers, from the right-hand portion of Fig. 6. It should be noted that the area ratios presented in Fig. 6 do not include the flow areas associated with inflow through any slot, or slots, (see Fig. 2) provided to control the boundary layer in the diffuser.

The optimum performances of concentrators for which the diffuser area ratio is unity, with $\eta_D = 0$, are shown in Fig. 7. The $(A_3^*/A_2) = 1$ condition gives the highest physically meaningful value of m possible; this has been labelled $m_{(MAX)}$ in Fig. 7. It is apparent that even with a base drag coefficient of unity the attainable $r_{(MAX)}$ is slightly less than 0.8. This result confirms the supreme importance of the diffuser of a venturi-type concentrator and implies that a very simplified form of concentrator, in which reliance is placed on base-drag in the absence of a diffuser to generate a power ratio greater than unity, is not likely to be successful.

The diffuser exit-plane pressure coefficient is displayed, versus C_{DB}, in Fig. 8. It can be seen that for quite realistic operating conditions, for example when $C_{DB} = 0.6$ and $m = 0.2$, it is possible for the stagnation pressure of the flow at station 3 to be less than the static pressure, p_∞, of the surroundings. This situation can only occur, as can be deduced from Fig. 8, because of the beneficial effect of base-drag.

4. COMPARISON WITH EXPERIMENT

Experiments, using small (88 mm throat diameter) model concentrators with mesh screens as substitutes for turbines, have been carried out be Igra (2). Igra's tests involved a number of configurations of both elementary form, as represented by Fig. 1, and also with slotted diffusers on the lines of the configuration shown in Fig. 2.

The results of Igra's tests (2), for which the concentrators were not yawed to the flow, lay within the approximate range: $r_{(MAX)} \approx 1.7$, $C_D^* \approx 0.6$ to $r_{(MAX)} \approx 2.6$, $C_D^* \approx 0.4$. These values of C_D^* imply, as can be seen from Fig. 3, $0.7 \leq \eta_D \leq 0.8$. The range of diffuser area ratios of Igra's apparatus was approximately $6 \leq A_3^*/A_2 \leq 8$ when allowance is made for the additional cross-sectional area required for the slot flow in the diffusers of the most

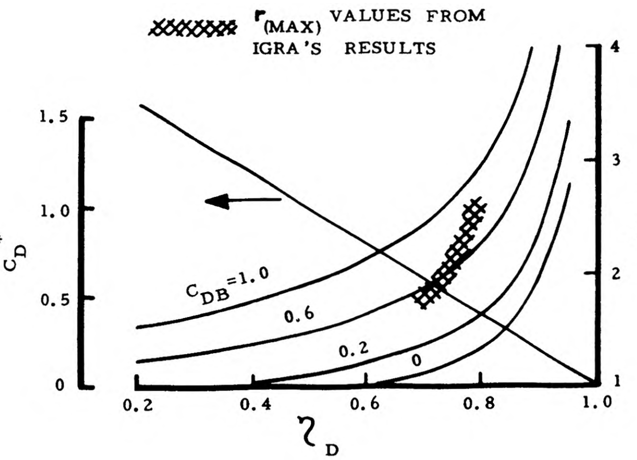

FIG. 9 PREDICTED MAXIMUM POWER RATIO $r_{(MAX)}$, AND CORRESPONDING TURBINE PRESSURE DROP COEFFICIENT C_D^*, VERSUS DIFFUSER EFFECTIVENESS η_D: $m = 0.2$. IGRA'S TEST RESULTS (2) REPRESENTED BY CROSS-HATCHING.

sophisticated configurations. These values of A_3^*/A_2 imply a value of m of approximately 0.2. Figure 8, a plot of C_D^*, and $r_{(MAX)}$ for $m = 0.2$, versus η_D, has a cross-hatched area added to it representing the approximate region in which Igra's results lie. Generally the results for the more sophisticated configurations, with slotted diffusers, resulted in the achievement of the highest values of both η_D and C_{DB}. Figure 10 shows a similarly cross-hatched region representing Igra's results in terms of diffuser area ratio versus η_D.

An interesting observation based on Igra's tests was that concentrators with non-slotted diffusers (see Fig. 1) tended to produce their greatest power ratio when yawed to the free stream at approximately $\pm$ 25°. It may be conjectured that yaw results in a tendency to sweep-off, on the lee side, the boundary layer forming over the outer surface of the concentrator thereby giving a thinner boundary layer over the outer surface in the region of the diffuser exit. This results in an increase in C_{DB} compared with that obtainable without yaw. It is well known that the base drag-coefficient, of truncated bodies, is often correlated with the forebody drag: generally the lower the forebody drag the higher the base drag (5). Yaw did not produce a beneficial effect with concentrators having slotted diffusers. Presumably, the operation of the slots is interferred with when such a

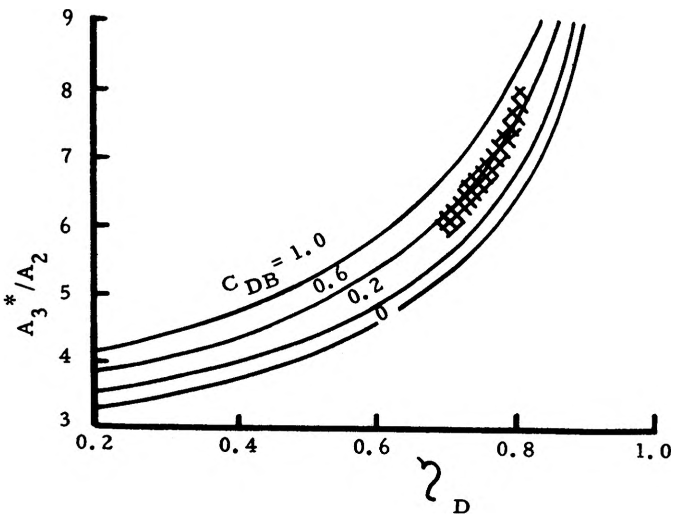

FIG. 10 PREDICTED DIFFUSER AREA RATIO A_3^*/A_2, CORRESPONDING TO $r_{(MAX)}$, VERSUS DIFFUSER EFFECTIVENESS η_D: m = 0.2. IGRA'S TEST RESULTS (2) REPRESENTED BY CROSS-HATCHING.

concentrator is yawed to the oncoming stream.

Other workers have tested venturi-type concentrators the configurations of which were not optimised. Gilbert, Oman and Foreman (3) obtained a power ratio of 1.89 with a screen equipped, small scale, model using a multiply-slotted diffuser of area ratio 2.78. Later, Gilbert and Foreman (4) claimed a power ratio of 2.7 using a large scale (high Reynolds number), turbine-equipped, model also provided with a multiply-slotted diffuser having an area ratio of only 2.78.

5. CONCLUSIONS

A simple analysis is presented which permits the optimum performance of venturi-type concentrators to be predicted within the limitations imposed by the assumption of one-dimensional flow. An approximate comparison with the optimised experimental results of Igra (2) shows that the analysis is consistent with Igra's results. Both the results of Igra (2) and those of other workers (4) indicate that power ratios in the region of 2.5 are achievable with current technology.

It would appear that if the unlimited supply of fluid surrounding a concentrator can be made to serve as a source for boundary-layer-blowing jets in a more advanced form of diffuser than current designs (2,3,4) it may be possible to achieve values of $r_{(MAX)}$ as high as 3.5 or 4. This implies a combination of very high values of η_D in the region of 0.8 to 0.9, a base-drag coefficient of 0.7 to 0.8 and, relative to current practise, an increase of diffuser area ratio with a consequent reduction in m. It seems, therefore, that a target for $r_{(MAX)}$ of 3.5 to 4 is only likely to be achievable after a protracted period of development of a suitable diffuser.

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