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A STUDY OF THE EFFICIENCY OF A RADIAL DIFFUSER WITH BOUNDARY LAYER CONTROL AT THE THROAT

by

J. S. de Krasinski and A. Aziz

SUMMARY

A radial diffuser has the advantage of extreme shortness and possibility of noise abatement. The upstream disc is provided with a bell-shaped entry where the separation of the boundary layer occurs when the gap between the discs is large. To improve the performance and the stability of such a diffuser, three methods of boundary layer control were tried at the entry where the adverse pressure gradients are the most critical, i) Rotation of the entry bell improving the boundary layer profile by means of centrifugal forces, ii) Blowing of the boundary layer at the bell by means of a series of jets, iii) Vortex generators situated upstream of the bell to energise the boundary layer entering the bell. The energy balance is drawn and an optimization of the diffuser is discussed in terms of experimental and theoretical data obtained in this study. It was also found that applying boundary layer control the stability of the diffuser is greatly improved when the flow is disturbed and uneven at the entry, what often occurs in reality.

J. S. de Krasinski, Associate Professor, Dept. of Mechanical Engineering, University of Calgary, Calgary, Canada.
A. Aziz, Graduate Student, Dept. of Mechanical Engineering, University of Calgary, Calgary, Canada.

INTRODUCTION

Classical subsonic diffusers have been well studied in the past and their performance is close to an optimum (1). In many engineering applications like jet engines, hovercrafts, wind tunnels, etc., severe space limitations are often imposed and such diffusers may be inapplicable. Also in big installations high costs follow exorbitant space requirements.

A long term research programme has been initiated at The University of Calgary (Alberta, Canada) to develop an unorthodox radial diffuser of extreme shortness (Fig. 1). It consists of a supply pipe (diam. d_1), a bell-shaped entry and two disks (diam. D) separated by a gap h. It follows that the area ratio $A_2/A_1 = AR = 4(D/d_1)\cdot(h/d_1)$. For a fixed geometry the area ratio can be changed by varying the width of the gap h.

The efficiency of the diffuser is defined as

$$\eta_0 = \frac{p_2 - p_1}{\frac{1}{2}\rho U_1^2\left[1 - \left[\frac{A_1}{A_2}\right]^2\right]} = 1 - \frac{p_{01} - p_{02}}{\frac{1}{2}\rho U_1^2\left[1 - \left[\frac{A_1}{A_2}\right]^2\right]} \tag{1}$$

the subscripts 1 and 2 refer to the entry and exit; p and p_0 refer to static and pitot pressures respectively.

The original design of a radial diffuser goes back as far as 1931 and is due to A. Hofmann (2), it was later developed by P. Moller (3). In both cases the curvature of the entry piece was constant, an arc being used. A typical efficiency curve using Moller's design is shown in Fig. 2. One observes a maximum close to $h/d = 0.15$. The sharpness of its peak suggests a quasi-stability at optimum performance. This is not desirable if the flow is disturbed at the entry which is the usual case in industrial applications.

THE POTENTIAL FLOW SOLUTION

The first step of the study was to find an "exact solution" for non-viscous fluids (4). The Laplace equation was solved for axially symmetrical co-ordinate system using the velocity potential φ

$$\frac{\partial^2\varphi}{\partial\xi^2} + \frac{1}{\xi}\frac{\partial\varphi}{\partial\xi} + \frac{\partial^2\varphi}{\partial y^2} = 0 \tag{2}$$

At the front disk the stagnation point flow velocity potential function was used and between the discs the properties of the line source of constant strength were employed whose velocity potential is also known. To match the two regions within the prescribed boundaries a "perturbation potential" was required. The shape of the streamlines approaching the front disc determines the shape of the entry bell. Each gap between the discs was related to a different geometry of the entry bell. Configurations A, B, C, and D were constructed (Fig. 3). Apparently the ideal fluid solution contributes nothing to the efficiency of a diffuser as p_0 remains constant and Eq. (1) becomes trivial. Nevertheless the shape of the entry piece developed in this way increased surprisingly the efficiency as shown in Fig. 2.

EFFECTS OF $\frac{h}{d}$, BLEEDING AND VORTEX GENERATORS ON THE EFFICIENCY

Flow visualization indicated that the boundary layer was at the point of separation at the end of the entry bell, when the efficiency was at its maximum value. Further study confirmed this assumption (6); the shape factor H reached in this region the value of 2.1 which is usually associated with a turbulent separating boundary layer. The sharp maximum of the efficiency curve plotted against h/d_1 can be explained as follows: When $H/d_1 < 0.15$ the viscous stresses, particularly in the throat region become predominant because of high velocity gradients and the efficiency drops very strongly. When $h/d_1 > 0.15$ the area ratio increases, the flow separates and the efficiency drops again. Another peculiarity of this diffuser is that a part of the flow recompresses isentropically in the region of the stagnation point at the front disk. The recompression efficiency of that fraction of the flow rate is practically unity. At high subsonic speeds this air could be used for special purposes like driving some auxiliary equipment, or could be fed back into the diffuser for boundary layer control. Figure 4 shows the effect of increasing the bleeding area on the efficiency. As expected the maximum efficiency peak moved to the left, because for smaller gaps viscous stresses were also smaller. Otherwise no significant effects were noted upon $\eta_{D\ max}$.

To delay incipient separation with all its unfavourable consequences energising the boundary layer was studied. The simplest method was to use vortex generators (6). Their height and position was optimised. It was found that for a turbulent boundary layer at the entry δ = 0.365" their optimum height was 0.125" and optimum position 7" upstream of the bell entry piece. It was found that such vortex generators could increase the efficiency by a further 4% to 5%.

EFFECTS OF ROTATION ON EFFICIENCY

The next method was to use centrifugal forces on the boundary layer by rotating the entry bell (5, 6). Some results are shown in Fig. 5 when vortex generators and rotation were applied. The efficiency reached about 80% and the steepness of the efficiency curve $\eta_D = f(h/d_1)$ decreased showing a better stability. When the vortex generators were installed on the rotating bell itself their name was changed into "vanes" because of the different role played in such conditions. It should be noted that with rotation the second important dynamical similarity parameter (besides Re) is $\omega d_1/2U_1$. The results are summarised in Fig. 6. One observes that the steepest rate of growth of $\eta_{D\ max}$ occurs with rotating vanes. Some tests were carried out for high values of $\omega d_1/2U_1$ and in these conditions $\eta_D > 1$. This is not surprising when one considers Eq. (1). Feeding extra energy into the system increases p_{02} and the efficiency can reach any value. In such cases one should think of η_D in terms of "apparent efficiency." To study this effect the rotating bell was mounted on an air bearing and driven by a series of compressed air jets which acted on externally mounted cups. The system was optimized by using the correct number of jets, correct jet velocities and the appropriate jet diameters.

THE ENERGY BALANCE

To discuss the energy balance imagine that a diffuser is located downstream of the working section of a wind tunnel where the velocity is U_1. The fan pumping power for a low diffuser efficiency is represented by the cubic curve marked 0-2 (Fig. 7) and for a high diffuser efficiency by the curve 0'-1. To simplify the argument this can be approximately represented by

$$\dot{P}_p = C(1 - \eta_D)\ d_1^2\ U_1^3 \tag{3}$$

where C is a constant.

The reduction in power obtained from the more efficient diffuser is at U = constant.

$$\Delta\dot{P}_p = C(\eta_{D_1} - \eta_{D_2})d_1^2\ U_1^3 \tag{4}$$

The jet power required to rotate the bell is

$$\dot{P}_j = \frac{1}{2} A_j\ V_j{}^3 \tag{5}$$

A_j = the total jet area.

Calling ε the ratio of the jet velocity to the peripheral velocity and λ the ratio of the exit to the inlet bell radii one obtains for the jet power

$$\dot{P}_j = \frac{1}{2}\ \rho\ A_j \varepsilon^3 \lambda^3 \left(\frac{\omega d_1}{2U_1}\right)^3 \tag{6}$$

Figure 6 indicates an almost linear relation between the increase of the diffuser efficiency and the parameter $(\omega d_1/2U_1)$. It follows that

$$\eta_{D_1} = \eta_{D_2} + m\left(\frac{\omega d_1}{2U_1}\right) \tag{7}$$

where m is the slope of the efficiency increase. Substituting (7) into (4) the expression for $\Delta\dot{P}_p$ becomes

$$\Delta P_p = C\ m \left(\frac{\omega d_1}{2U_1}\right) d_1^2 U_1^3 \tag{8}$$

In order to increase the diffuser efficiency without consuming extra power $\dot{P}_j < \Delta\dot{P}_p$. For $\dot{P}_j = \Delta\dot{P}_p$ the apparent power gain $\Delta\dot{P}_p$, or the real gain in velocity $(U_1 - U_0)$, at constant power is offset by the losses required for boundary layer control. For this extreme case one may argue that for a real installation the extra power is not easily available for the fan. Also an improvement of the flow quality in the diffuser will make it less prone to stall for poor entry conditions. Often an additional power supply is available from another source as for example a compressed air installation.

Substituting (6) and (8) into $\dot{P}_j \leq \Delta\dot{P}_p$ one obtains an important relation for the similarity parameter

$$\left(\frac{\omega d_1}{2U_1}\right) \leq \sqrt{\frac{C\ m}{\frac{1}{2}\rho A_j \varepsilon^3 \lambda^3}}\ d_1 \qquad (9)$$

The net gain in power is zero for no rotation, it is also zero for $\dot{P}_j = \Delta\dot{P}_p$ therefore the function of $(\Delta\dot{P}_p - \dot{P}_j) = f(\omega d_1/2U_1)$ must pass through a maximum.

It follows that the net reduction in power

$$\dot{P}_r = \Delta\dot{P}_p - \dot{P}_j = C\ m\ d_1^2 U_1^3 \left(\frac{\omega d_1}{2U_1}\right) - \frac{1}{2}\rho\ A_j\ \varepsilon^3 \lambda^3 U_1^3 \left(\frac{\omega d_1}{2U_1}\right)^3 \qquad (10)$$

by differentiating this expression with respect to $\omega d_1/2U_1$ for a constant value of U_1 and equating to zero one obtains

$$\left(\frac{\omega d_1}{2U_1}\right)_{opt.} = 0.577 \sqrt{\frac{C\ m}{\frac{1}{2}\rho A_j \varepsilon^3 \lambda^3}}\ d_1 \qquad (11)$$

For small installations this optimum value is insignificant because small $\omega d_1/2U_1$ yield small gains of η_D. One may argue, however, that as this optimum value is proportional to d_1 big installations in which the jet diameter and the cup size would be kept the same as on the model would operate on big values of $\omega d_1/2U_1$ which is of practical interest. Say for equal values of U_1 for the model and the prototype, high periferal velocities of the prototype would gain the required power from high jet velocities. These in all the cases would have to be double the periferal velocity for optimum efficiency.

CONCLUSIONS

In spite of various restrictive assumptions the above argument is of practical interest. The same reasoning can be extended to another solution, i.e., energizing the boundary layer directly by blowing instead of rotating the bell. Figure 8a, b shows the comparison between the classical and the radial diffusers. One observes its remarkable shortness and comparatively good efficiency.

ACKNOWLEDGEMENT

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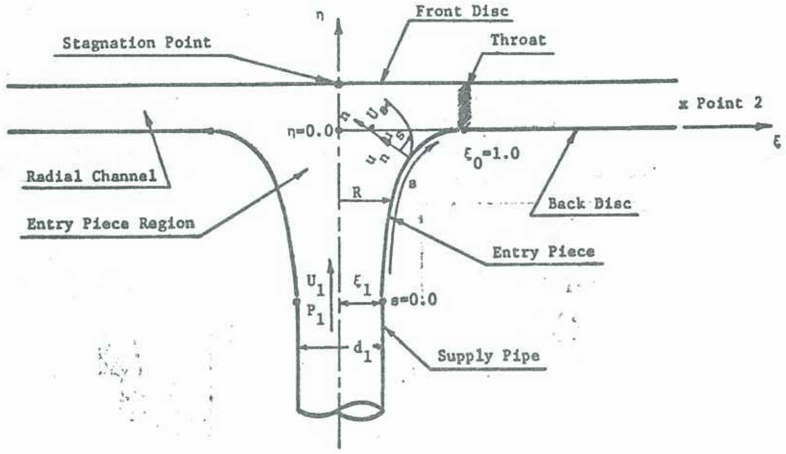

Fig. 1 Schematic outline of the radial diffuser

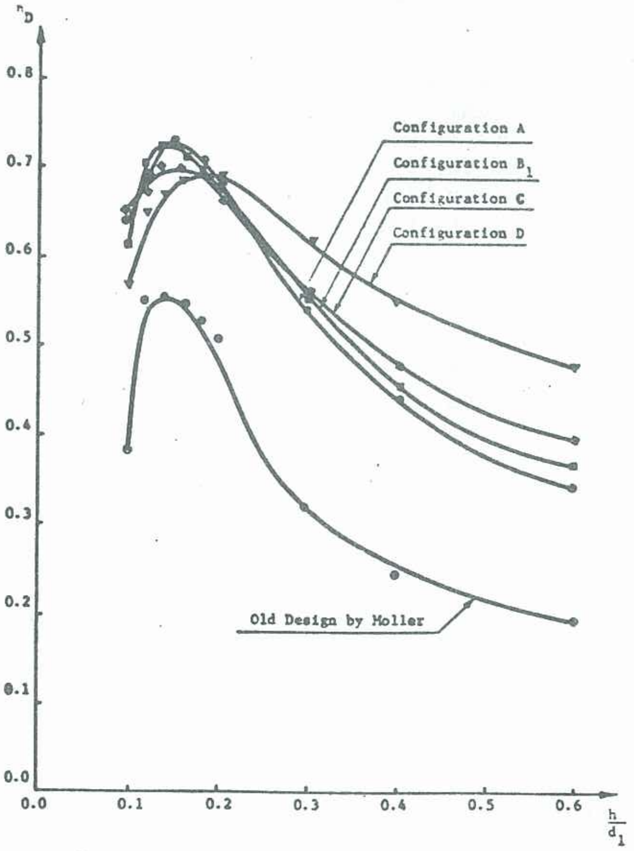

Fig. 2 The effect of the new design on the efficiency

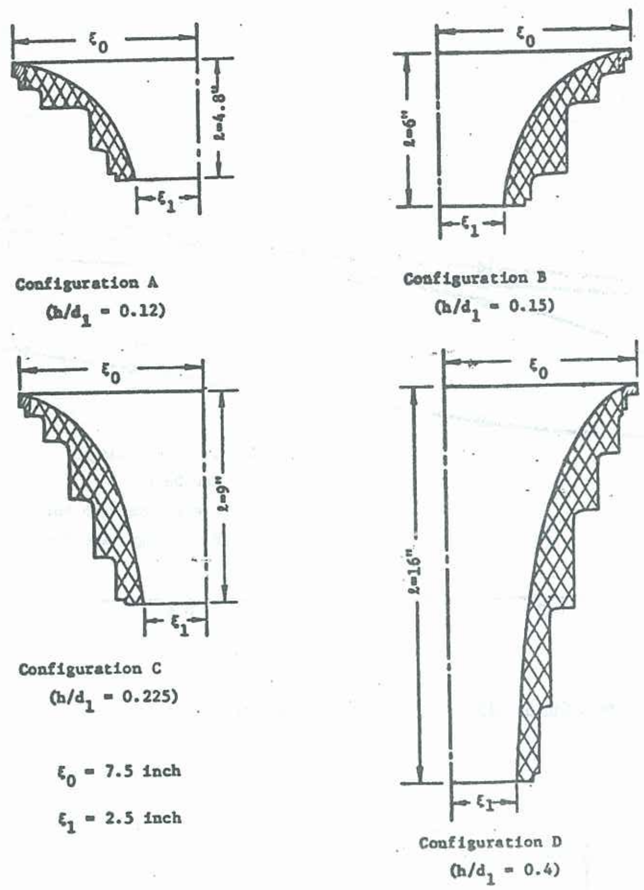

Fig. 3 Four configurations of the bell-shaped entry

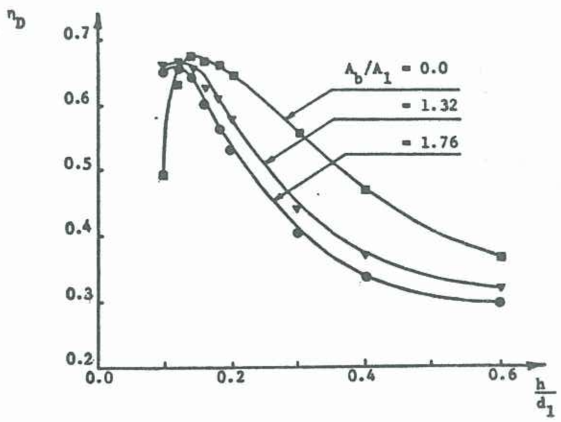

Fig. 4 Effect of increasing the bleeding area (A_b) on the diffuser efficiency

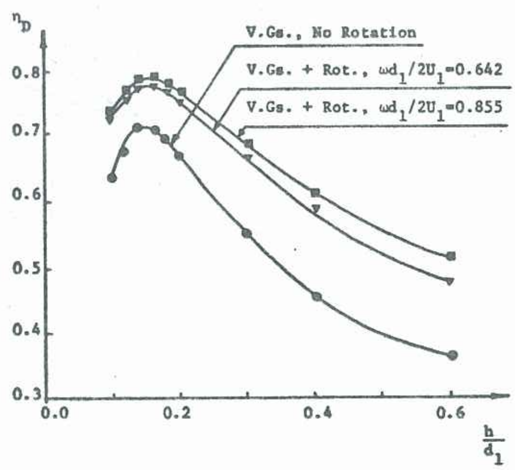

Fig. 5 Effect of the vortex generators and rotation on the diffuser efficiency

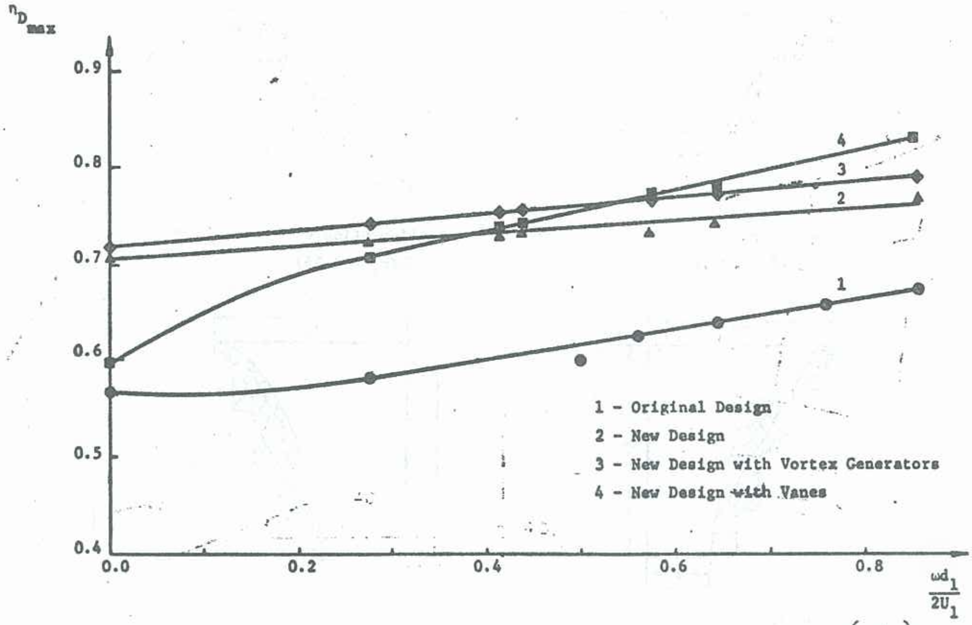

Fig. 6 The maximum diffuser efficiency as function of $\left(\frac{\omega d_1}{2U_1}\right)$

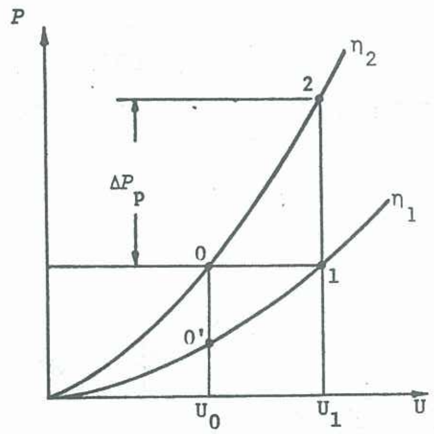

Fig. 7 The pumping power required to increase the flow velocity

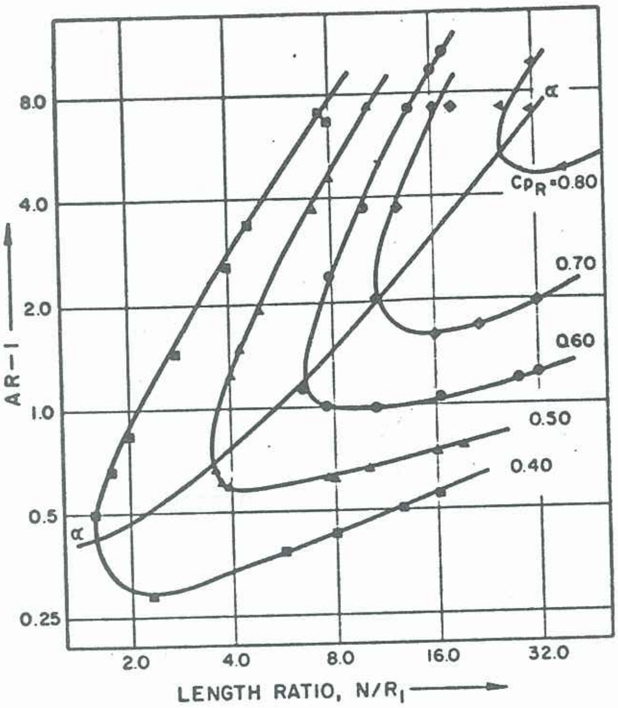

Fig. 8a Typical contours of constant performance coefficient C_{pR} for conical diffusers (1)

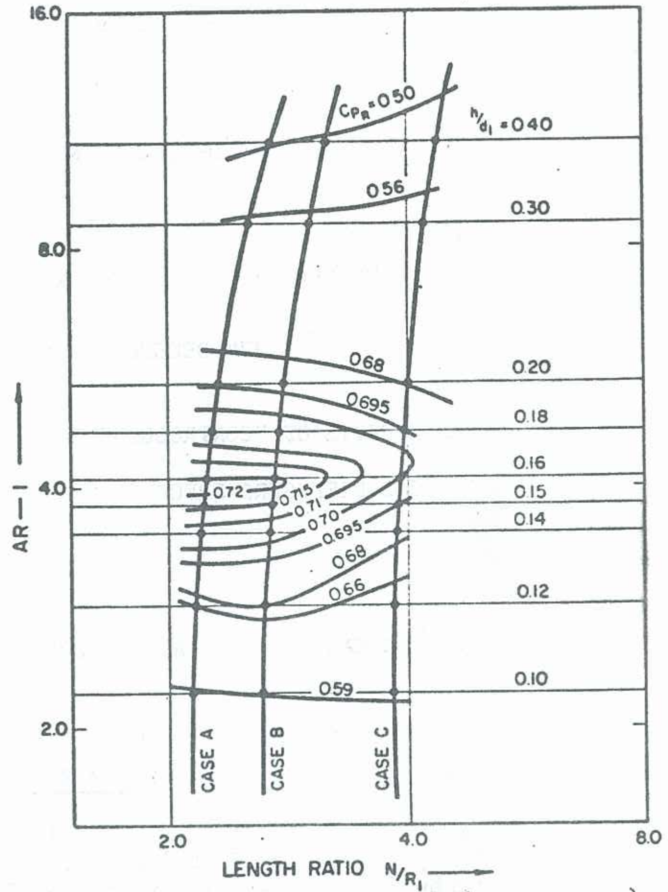

Fig. 8b The effect of rotation $\left(\frac{\omega d_1}{2U_1} = 0.435\right)$ on the contours of constant performance coefficient C_{pR} for a radial diffuser (configuration B)