Griffith Diffusers[1]

A. KLEIN.[2] It has been known for some considerable time that, in the absence of special means of flow control, optimum diffusers have bell-shaped walls. Contours of this type follow from a variety of boundary-layer optimization principles, notably from the postulation that the flow should produce vanishing wall shear stress (for example Fernholz, reference [1] and Hackeschmidt, reference [2]), from a modification of this concept attributable to Huo (reference [3]) and from application of a special deceleration parameter proposed by Senoo and Nishi, reference [4]. In spite of the deficiencies of the boundary-layer methods used in these analyses, the neglect of the effect of turbulence level on flow behavior and of the effects of wall curvature on turbulence structure, the overall conclusion of these investigations is certainly correct: bell-shaped walls are optimum.

If one considers the gain in performance of such contoured-wall diffusers relative to rectilinear configurations, however, the result is disappointing. Although experiments conducted in reference [2] were claimed to have shown that a 50 percent shorter contoured-wall design would provide the same performance as an optimum conical diffuser, this was not confirmed in more recent investigations. The misleading statement in reference [2] is obviously due to insufficient knowledge at that time of optimum straight-wall diffuser performance. In fact, the results of references [3], [4], and [5] indicate such small performance gains of optimum vs. straight-wall configurations, that wall shaping must be considered an unnecessary design complication.

Bell-shaped walls, in order to be attractive in practical applications, must therefore be used together with reasonable flow control methods. Out of these, "passive" flow control devices such as screens are known to permit considerable reductions of unseparated-flow diffuser length for a given area ratio, and to produce a high degree of flow uniformity at the diffuser exit, but at the expense of very low performances (reference [6]). The high effectivenesses obtained by Yang and Nelson by "active" flow control at comparatively small suction rates are therefore worthy of note. The actual merits of their design can, however, only be appreciated when compared with the results for corresponding straight-wall diffusers and, in a more comprehensive way than the authors did, with the performance of Adkins' configurations (Yang and Nelson's reference [4]). Such a comparison has been attempted by this writer. It is presented in Table 1 for tubular configurations with Miller's (reference [7]) results for an optimum conical diffuser of the same non-dimensional length, and in Table 2 for annular and two-dimensional configurations with Sovran and Klomp's (reference [8]) and Reneau, et al.'s (reference [9]) results for corresponding straight-wall diffusers. Adkins' correlations were used to deduce effectivenesses and bleed requirements to be expected if the authors' geometries (either the same area ratio or the same non-dimensional length) were built as Adkins instead of as Griffith diffusers.

In establishing the data, a number of difficulties were encountered:

1. For the results of tubular and annular diffusers the authors cite reference [5]. However, they are in fact not available there. They were found in the additional reference [10], and the entrance dimensions required for applying Adkins' correlations have been taken from there.
2. Although the values for η_3 given by the authors coincide with those in reference [10] (if the last digit is rounded off), the definition of η_3 in reference [10] differs from that of the authors in that $\kappa_i = \kappa_e = 1$.
3. C_p-values when calculated from η_3 as defined by the authors with κ_i and κ_e as stated by them, are so high that effectivenesses $\eta_1 > 100$ percent result, and this is even true with $\kappa_i = \kappa_e = 1$. On the other hand, the authors claim values of $\eta_1 = 98$ percent to 99 percent.
4. $\kappa_i = 1.025$, stated by the authors, appears to be too high for the very short entrance length (and small inlet blockage) used. A more likely value $\kappa_i = 1.010$ has been chosen for determining the performance of the equivalent Adkins diffusers.

A comment by the authors on these remarks would be appreciated. In order to circumvent the difficulties stated, in the tables two differently obtained values of C_p and, in addition to the authors' η_3-data, two more values for η_3 have been given for the Griffith diffusers. For all the other diffusers η_3 is for $\kappa_i = \kappa_e = 1$; for diffusers without suction $\eta_3 = \eta_1$.

The data of Table 1 and 2 show the superiority of the Griffith diffusers over diffusers without flow control. Although this superiority is hardly apparent when effectiveness is compared, it is clearly demonstrated by the much higher pressure recoveries, a consequence of the considerably larger area ratios to be achieved at a given non-dimensional length. In addition to this, credit must also be given to the good uniformity of the exit flow, while on the other hand the energy required for suction should be considered.

Comparison with Adkins' diffusers is of particular interest. It can be seen that the effectiveness of only 80 percent for an area ratio 3.2, quoted by the authors, is somewhat misleading. In fact, Griffith and Adkins diffusers appear to have approximately equal performance when the entrance conditions are comparable. Exit flow uniformity cannot be assessed since Adkins does not mention it. On the other hand, contrary to the authors, he also presents performances measured with thicker inlet boundary layers, as would be common in practical applications. Such additional information for Griffith diffusers would be very valuable.

[1] By Tah-teh Yang and C. D. Nelson, published in the December, 1979, issue of the ASME JOURNAL OF FLUIDS ENGINEERING, Vol. 101, No. 4, p. 473.

[2] Motoren- und Turbinen-Union (MTU) München GmbH, Postfach 50 06 40, 8000 Munich, W. Germany.

Additional References

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4 Senoo, Y., and Nishi, M., "Deceleration Rate Parameter and Algebraic Prediction of Turbulent Boundary Layer," ASME JOURNAL OF FLUIDS ENGINEERING, Vol. 99, 1977, No. 2, pp. 390–394.
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7 Miller, D., "Performance of Straight Diffusers," "Internal Flow. A Guide to Losses in Pipe and Duct Systems," Part II. B.H.R.A.-Fluid Engineering, Cranfield/Bedford, 1971.
8 Sovran, G., and Klomp, E., "Experimentally Determined Optimum Geometries for Rectilinear Diffusers with Rectangular, Conical or Annular Cross Section," In: *Fluid Mechanics of Internal Flow* (ed.: G. Sovran), Elsevier Publishing Company, Amsterdam - London - New York, 1967.

Table 1

	Yang and Nelson	Adkins (reference [4] of paper)	Hacke-schmidt (reference [2])	Miller (reference [7])
L/R_{in}	3.5	2.7 3.5	3.5	3.5
AR	3.0	3.0 3.75	1.9	2.1
$2\,\delta^*/R_{in}$	0.012- 0.016	0.012	0.012	$\approx$0.010
Q_S/Q	6.1 % 1)	5.3 % 6.5 %	0	0
C_p	0.88 2) 0.90 3)	0.86 0.88	0.70	0.67
η_3	0.92 4) 0.90 2) 0.936 5)	0.90 0.88	0.97	0.87

Table 2

	Annular			Two-dimensional	
	Yang and Nelson	Sovran and Klomp (reference [8])	Adkins (reference [4] of paper)	Yang and Nelson	Reneau, et al. (reference [9])
L/h_{in}	3.9	3.9	3.7	4.0	4.0
AR	3.0	1.8	3.0	3.0	2.0
$2\,\delta^*/h_{in}$	0.012- 0.016	0.02	0.012	?	0.007
Q_S/Q	9.0 % 1)	0	8.0 %	14 %	0
C_p	0.88 2) 0.91 3)	0.55	0.86	0.88	0.63
η_3	0.88 4) 0.86 2) 0.909 5)	0.80	0.88	0.83	0.84

Table 1 and 2: Comparison of diffuser performances
Table 1: Tubular diffusers, Table 2: Annular and two-dimensional diffusers

1) from reference [10]
2) from $\eta_1 = 0.99$ with $\kappa_i = 1.025$; $\kappa_e = 1.011$
3) from η_3 with $\kappa_i = \kappa_e = 1$
4) from $\eta_1 = 0.99$ with $\kappa_i = \kappa_e = 1$
5) as given in paper
h_{in} is the diffuser height at the inlet

9 Reneau, L. R., Johnston, J. P., and Kline, S. J., "Performance and Design of Straight, Two-Dimensional Diffusers," ASME *Journal of Basic Engineering*, Vol. 89, 1967, No. 1, pp. 141-150.

10 Nelson, C. D., Jr., Hudson, W. G., and Yang, T., "The Design and Performance of Axially Symmetrical Contoured Wall Diffusers Employing Suction Boundary Layer Control," ASME-Paper 74-GT-152, 1974.

P. S. BARNA.[3] To those who have been puzzled by the mysteries of boundary layer control in diffusers, it is refreshing news to learn of such a successful approach to both diffuser design and flow control as the Griffith diffuser. To this reviewer it was even a gratifying experience to personally observe the flow attachment to the walls of a test diffuser shown in an impressive movie film prepared by Professor T. Yang and his associates and presented at the meeting of the Colorado Joint Symposium of Fluid Machinery in June 1978.

It is now a well-established fact that diffuser performance depends on both the design proportions of the diffuser and the flow conditions at the inlet to the diffuser. This means that for a given specified geometry, performance entirely depends on the inlet flow conditions, so that manipulations for varying the velocity distribution (and Reynolds number) *at inlet* lead to results which are found to be more satisfactory if the distribution is uniform and the blockage is small. Tests have shown that this can be achieved successfully by employing boundary layer control at inlet, by removing *small* amounts of fluid relative to the total flow.

To employ boundary layer control *downstream* from the inlet is another matter, and the question then arises how far downstream should the control be applied and how much fluid needs to be removed? If, for example, one employs suction halfway along the length of a diffuser with straight walls, the experiments conducted by this reviewer show high rate requirements with only a limited amount of improvement in performance. To the best of my knowledge the question pertaining to straight diffusers apparently remains unanswered – where should control be applied to the best advantage?

It is this question which may be answered by employing curved walls as is the case of the Griffith diffuser. Although it is specific only for a certain curved wall geometry, nevertheless the theory locates the suction slot by using the "in-

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verse" method that specifies the velocity distribution. Assuming incompressible and irrotational flow theory as the basis for the calculations, the method developed in the paper under review furnishes an answer to the wall geometry. Additional attraction is the shorter length and high area ratio attainable.

While the reviewer finds the paper commendable and thinks that Griffith diffusers may well have an application in the future, much of their acceptance will depend on practical considerations, such as the willingness to expend energy for boundary layer control. This will especially apply in large diffusers where the machinery required to provide the necessary suction may add to the already higher costs of producing the curved walls as compared to the simpler straight-walled diffuser design. In addition, there is some evidence that the suction rates in practice were found higher than predicted from theory and successful operation was limited to a narrow range of inlet conditions.

To the designer of diffusers it would come in handy to have a *Griffith Diffuser Data Book,* including charts for various area ratios and blockage factors, similar to that now existing for straight-walled diffuser design.

Authors' Closure

I would like to thank Professor Barna for his kind words about our work and agree with him on the comments that the required suction rates were often found to be higher than predicted. I believe there are two reasons to account for these differences.

First, we do not often predict the boundary layer profile accurately enough. It is the profile, not the momentum thickness or displacmeent thickness, that was used in Taylor's criterion to predict the minimum suction rate. The second source contributing to the difference in suction rate is that in actual flow systems there is always some disturbance, and the boundary layer can be momentarily and locally thickened to require higher overall suction rate to meet the local requirement. Flow separation could take place if the diffuser is operated with a minimum suction rate predicted by Taylor's criterion. Therefore, in practice one must provide a suction rate somewhat higher in order to have a margin for stability.

As far as a *Griffith Diffuser Data Book,* we have no plan for one at this time. However, I will welcome the opportunity to publish such a data book as Professor Barna suggested when circumstances permit.

I would like to thank Dr. A. Klein for his interest in our work. I appreciate his providing two tables to compare performances among diffusers, particularly between the Griffith type and the ones using vortex-fences. (Dr. Adkins' work at Cranfield.)

Because of Dr. Klein's comments relative to the values of K_i used in our paper, I have recomputed values for several test runs of our study. These of course depend upon the inlet velocity profiles. Unfortunately, our recorded profile data do not include measurements near the diffuser wall which would reflect the boundary layer development (or indicate inlet length.) Consequently we assumed a few additional non-dimensional velocity values near the wall to complete the profile for each K_i computation. Our results show that K_i vary from 1.001 to 1.01 instead of 1.025 as stated in NASA CR 2209 and several other publications. Apparently it was a computational error that led to our misquoting of a significant flow parameter; I wish to apologize for the error. On the other hand, we stated that "It is apparent that the gain from enhancement of pressure recovery of the diffuser may be eroded rapidly when the required suction flow rate is high. . . . . it is equally apparent that a Griffith type diffuser is not the choice when the upstream already has a thick boundary layer." Therefore the authors' feeling about the applicability of such diffusers to long inlet sections should be apparent; stating $K_i = 1.025$ by no means implies a long inlet was involved in our test set-up.

It also should be pointed out that the deviation of K_i value from 1.0 can be attributed to the nonuniformity of the potential core due to wall curvature as well as to the boundary layer, and inlet length is not the only contributing factor on the value of K_i.

Now I wish to respond to Dr. Klein's specific comments No. 1 and 2. ". . . . Table 1 of reference [5] . . . ." of the last paragraph, 1st column on page 476 should read ". . . . Table 1 of reference [6] . . . ." The definitions of η_2 and η_3 of "Griffith Diffusers" are identical to those used in reference [6]. With regard to both of Dr. Klein's comments 1 and 2, a typographical error in reference number has caused this confusion. With regard to his comments 3 and 4. I recomputed ϵ and η values of table 1 of NASA CR 2209 with $K_i = 1.01$ and $K_e = 1.0$. (K_e is only of secondary importance to the accuracy of ϵ and η.) We stated in CR 2209 that

$$\epsilon = \frac{(1.0-FS)\,(P_{s,e}-P_{s,i})}{P_{d,i}\left[\kappa_i-\kappa_e\left(\frac{1.0-FS}{AR}\right)^2\right]}$$

and

$$\eta = \frac{(P_{s,e}-P_{s,i})}{P_{d,i}\left[\kappa_i-\kappa_e\left(\frac{1.0-FS}{AR}\right)^2\right]}$$

With $K_i = 1.01$ and $K_e = 1.0$, our η values ranged from 0.97 to 1.01. Allowing 1 percent error in η, one may state that η ranged from 0.96 to 1.00 instead of 0.97 to 0.99. The η used in CR 2209 is η_3 of the December 1979 paper entitled "Griffith Diffuser." It should be evident from the above expressions of ϵ and η that a smaller value of K_i will lead to a higher value of ϵ and, therefore, to a higher value of η. Taking K_i to be one would lead to a result of $\eta = 1.02$ for test run No. 12 of Table 1 CR 2209. I believe this value of η suggests that $K_i = 1$ is not a reasonable value for the measured values of suction flow rate and Δp. Cp values of Table 1 were computed from Δp. If boundary layer is the sole cause of K_i being greater than 1, then it follows that $K_i = 1$ yields zero boundary layer thickness and no suction requirement, certainly not the case that we investigated.

I have stated in the introduction that limited publications will be discussed for the purpose of suggesting that "perhaps the combined effect of using curved walls to control the pressure and boundary layer suction to maintain the designed pressure distribution is a more effective approach in achieving a high degree of pressure recovery than by either effect alone." It was along these lines that our discussion on several types of diffusers was made. Again the paper states that the peformances were reported in *different forms,* and they are not directly comparable. As long as we are discussing the difficulties in comparing various sets of diffuser data, perhaps it should be pointed out that the fine work reported by Adkins was with a diffuser havng a constant area section following the last pressure tap. Generally a downstream constant area section has a stabilizing effect over the upstream diffuser, and one should expect the same diffuser may have a slightly lower performance when the downstream constant area section is removed.