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ANNULAR EXHAUST DIFFUSERS FOR AXIAL FLOW FANS

by

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S U M M A R Y

The difficulties in optimizing designs of annular exhaust diffusers for axial flow fans are examined. The unreliable nature of commonly used prediction methods is attributed to their failure to consider all the important variables. The influence of wall angle combination was investigated and advantages of divergent centrebody arrangements are indicated. Three divergent centrebody diffusers are tested downstream of an axial flow fan. Experimental and theoretical results indicate that certain divergent centrebody arrangements are optimum where either minimum length for a given performance, or maximum pressure recovery for a given length is sought, and are less susceptible to the adverse effects of inlet swirl than convergent centrebody diffusers. The results of several experimental programmes, including the present one, are predicted within experimental accuracy, by a method presented in the literature.

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NOTATION

A	plane cross-sectional area
C_p	pressure recovery coefficient $(\Delta p/\tfrac{1}{2}\rho\overline{U}_1^2)$
C_{pc}	exhaust efficiency $(\Delta p/\phi_1\tfrac{1}{2}\rho\overline{U}_1^2)$
h	diffuser hub ratio at inlet (r_1/R_1)
L	diffuser axial length
p	fluid static pressure
p_{atm}	ambient static pressure
Δp	$(p_{atm} - p_1)$
Q	total volume flow rate
R	radius of outer wall
Re	Reynolds Number $(2\overline{U}_1\Delta R_1/\nu)$
r	radius of inner wall
ΔR	annulus width $(R - r)$
S	surface area of diffuser
T	turbulence level $(\overline{u'^2}/\overline{u}^2)^{\frac{1}{2}}$
T_m	mean turbulence level $(1/Q \int_A T\overline{u}_x dA)$
t	time
$\overline{U}$	mean axial velocity in cross section $(1/A \int_A \overline{u} \times dA = Q/A)$
$\overline{u}$	velocity (temporal mean)
u'	fluctuating component of velocity
x	axial distance from diffuser inlet
α	swirl angle
β	flow parameter $(\overline{U}/u_{max})$
ϕ	energy coefficient $(1/A \int_A (\overline{u}/\overline{U})^3 \cos\alpha dA)$
δ	diffuser wall angle
μ	diffuser area ratio $(R_2^2-r_2^2)/(R_1^2-r_1^2)$
ρ	fluid density
λ	diffuser length ratio (L/R_1)
ν	kinematic viscosity of fluid

Subscripts

1	diffuser inlet
2	diffuser exit
o	outer wall
i	inner wall
x	axial
y,z	normal to axial

1. INTRODUCTION

The exhaust diffuser is a relatively simple component of a turbomachine or fluid transport system in the geometric sense, yet the difficulties in estimating the performance of diffusers have plagued designers for decades. Whatever the design task, the process of optimization is severely limited because the methods of performance prediction commonly used, are unreliable near optimal conditions.

For annular diffusers, the large number of important variables involved is a major obstacle in formulating a design procedure. Figure 1 illustrates that four parameters are required to define the form of a straight walled annular diffuser. Ref.1 shows that predictions of diffuser performance require mean turbulence level T_{m1}, velocity profile parameters β_1 and ϕ_1, and Reynolds number Re_1 at the diffuser inlet, to be known; and that optimum geometry is a function of Tm_1 and β_1.

Where an exhaust fan operates over a range of loads, the diffuser is required to perform efficiently over a range of inlet conditions. For designs based on the common practice of using convergent centrebodies, this requirement has led to extremely long diffusers for large installations such as primary mine ventilators.

The work described in this paper was aimed at developing fan exhaust diffuser designs which are shorter than current practise and which exhibit uniformly high performance over a range of inlet conditions. The importance of wall angle combination was investigated; advantages of divergent centrebody arrangements over those with convergent centrebodies were indicated and an optimum wall angle combination was predicted. Three divergent centrebody diffusers of wall angle combination near and at the optimum, were tested downstream of an axial flow fan. An analogy between conical and annular diffusers and a mathematical model of conical diffuser performance, were used to predict the performance of the test diffusers and other published experimental results. Design recommendations based on the investigation, are made.

2. PERFORMANCE PREDICTION AND DESIGN METHODS

The published experimental results for annular exhaust diffusers [2],[3],[4],[5] indicate that hub ratio and wall angle combination influence performance and the optimum area ratio for a given length. The use of higher area ratios for divergent or cylindrical centrebodies, relative to those for convergent centrebodies, is recommended [4]. However, the breadth of the published data does not permit optimization in general.

Prediction methods based on the boundary-layer model [5],[6],[7] are inappropriate for general design because the model is only valid where ideal boundary-layer type conditions of flow exist; this rarely occurs in diffusers, particularly those operating near peak recovery.

Reference 8 postulates that for any annular diffuser, there exists an analogous conical diffuser geometry which would suffer the same losses, given equivalent inlet flow parameters β_1,

ϕ_1, Tm_1 and Re_1. Annular and conical geometries are considered analogous when their area ratios are equal and wall angles are related by

$$\tan\delta = \frac{\tan\delta_o - h\tan\delta_i}{1+h} . \qquad \ldots(1)$$

The analogy is compared with experiment [8] for 9 convergent centrebody diffusers, using the mathematical model described in Ref.1 to predict the performances of analogous conical diffusers. Predicted and measured values of C_{pc} agree within ±0.03. However, the greatest value of the analogy is that it offers a simple indication of the influence of wall angle combination and hub ratio.

3. THE INFLUENCE OF WALL ANGLE COMBINATION

The analogy [8] is manipulated in Appendix A to indicate the wall angle combination which gives the shortest possible diffuser for a given performance (i.e. a given analogous conical diffuser). The solution is given by

$$\tan\delta_i = h\tan\delta_o , \qquad \ldots(2)$$

which defines a family of diffusers with apexes of inner and outer cones coincident. This is also the combination giving the most conservative diffuser for a given area ratio and length.

The lines in Fig.2 show optimum area ratios at given length ratio (L/R_1), for diffusers of hub ratio 0.7 with centrebodies closing at the exit (type 1), cylindrical centrebodies (type 2) and those with wall angles satisfying equation (2) (type 3), predicted by the analogy and the conical diffuser performance map of Ref.9. Although only valid for the inlet conditions of the experiments in Ref.9, the lines indicate the influence of wall angle combination. Predicted performances of the optimum geometries for the 3 types is equal for a given area ratio; therefore the optimum geometries of type 3 diffusers are considerably shorter than type 1 for a given predicted performance.

4. EXPERIMENTAL PROGRAMME

The published data on divergent centrebody diffusers, notably Ref.3, was not appropriate for the flow conditions expected from axial flow fans (β = 0.85 to 0.90). Therefore, an experimental programme on divergent centrebody diffusers downstream of an axial flow fan was undertaken to test the analogy and to provide useful experimental data on fan exhaust diffusers.

Equipment

A schematic diagram of the test rig is shown in Fig.3. The conical inlet, constructed in accordance with Ref.10, provided a measure of total flow. Fan load was varied by placing mesh screens against the flow straightener. Fig.4 shows the Fan Total Pressure characteristic. Table 1 gives details of the 3 diffusers tested; their geometries are denoted on Fig.2 by numbers 1, 2 and 3.

TABLE 1

Geometry of test diffusers

radius of outer shell at inlet R_1 = 0.610m, hub ratio h = 0.70				
diffuser No.	outer wall angle δ_o (deg)	inner wall angle δ_i (deg)	length ratio L/R_1	area ratio μ
1	8.3	5.8	6	3.50
2	8.3	4.0	6	4.36
3	10.4	5.8	4	3.46

Diffuser Inlet Conditions

Detailed measurement of the flow was made at station 1, 0.140m upstream of the fan diffuser interface, without a diffuser attached. This allowed circumferential traverses using a system which pivoted about the rig axis.

Distributions of velocity, swirl angle, static pressure and turbulence level were determined at 5 different radii, over 2 representative straightener vane spaces, for 3 fan loads. A small 3 hole yaw tube was calibrated as in Ref.11 and traversed at fixed orientation to the rig axis to measure velocity, swirl and static pressure. A hot-wire probe was aligned to measure axial and circumferential components of turbulence.

At 3.5 m^3s^{-1} flow, swirl was 5^o and -3^o at the outer and inner walls respectively. At 5.1 m^3s^{-1}, swirl was about -2^o across the annulus. Radial distributions of circumferentially averaged velocity are shown in Fig.5.

Variation of the inlet flow parameters ϕ, β and T_m with fan load, is shown in Fig.6. These parameters were evaluated by integrating the data from the traverse measurements. The mean axial velocity calculated from the conical inlet reading, agreed with that obtained by integration of the traverse data within 0.5% in every case.

Diffuser Measurements

Diffuser performance was measured at several fan loads for each diffuser, using

$$\Delta p = p_{atm} - p_1 \qquad \ldots(3)$$

where p_1 is piezometer ring pressure at station 1, and p_{atm} is atmospheric pressure. Figure 7 shows the variation of $C_{pc}(\Delta p/\phi_1 \tfrac{1}{2}\rho \bar{U}_1{}^2)$ with fan load. An experimental accuracy of ±0.05 is claimed for measured values of C_{pc} although differences of ±0.01 are significant for relative comparisons between the diffusers tested, because the results could be repeated to that accuracy.

Diffuser number 1 showed little sensitivity to fan load over the range tested. Observation at the diffuser exit revealed no reversed flow at any of the test loads. The fan stator wakes persisted to the diffuser exit at the higher flows, but were not distinguishable at 3.5 m^3s^{-1}. Diffuser number 2 gave a similar result up to 5.5 m^3s^{-1}, exhibiting bi-stable behaviour above this. At about 6.0 m^3s^{-1}, the diffuser could operate for an indefinite period in either an unstalled mode or a less efficient stalled mode, the stalled mode becoming predominant at about 6.5 m^3s^{-1}. Diffuser number 3 exhibited a similar bi-stable characteristic from about 5.0 m^3s^{-1} upward. Where stall occurred in diffusers 2 and 3, it started from a straightener vane wake and progressed circumferentially, rather than radially from a wall surface.

5. DISCUSSION OF EXPERIMENTAL RESULTS

Performance measurements of the three test diffusers demonstrate that a divergent centrebody allows a shorter diffuser for a given performance, than a convergent centrebody. All the test diffusers are considerably shorter than the optima recommended by Ref.4 for closing centrebodies, yet exhibit performances which compare well with those of Ref.12, which are optimal designs for closing centrebodies.

The inherent flow stability observed in the test diffusers, particularly number 1, is related to insensitivity to radial nonuniformities of inlet flow. All the test diffusers exhibited constantly high recovery up to a certain flow, where stall occurred suddenly. The observed flow in stalled diffusers indicated that circumferential, and not radial, velocity gradients were the critical factor in precipitating stall.

6. DESIGN ASPECTS - FAN EXHAUST DIFFUSERS

The predicted results of the current work, using the analogy of [8] and the mathematical model described in Refs. [1] and [13], are shown in Fig.7; they are within experimental accuracy (±0.05) of the measured results. Reference 14 gives experimentally determined performances of four fan exhaust diffusers with closing and cylindrical centrebodies (area ratios from 1.6 to 2.1, L/R_1 of 4.2 and 3.7). Predictions agreed with the measurements of pressure recovery C_p, within ±0.03 over a wide range of fan loads. The predictions deviate from measurement where inlet swirl exceeded 6^o.

While the analogy is a gross simplification of the physical problem, its predictions are supported by experiment for type 1 [8],[14], type 2 [14] and type 3 diffusers; it is therefore concluded that the indications of Appendix A are close to the truth.

Ref.7 shows that swirl can have a marked effect on the performance of diffusers. Streamline convergence along tapering centrebodies causes swirl velocities to increase by conservation of angular momentum, creating a low pressure area around the centrebody and promoting reversed flow at the diffuser exit. The boundary layer, thickening because of the reducing centrebody perimeter, encourages separation further. Even where no separation occurs, the pressure recovery can be limited by the high exit kinetic energy due to swirl. The divergent centrebody will suffer far less from these effects because of the increasing radius of the annular passage.

The design recommendations of this work can be summarized as follows :

(a) The wall angle combination of type 3 diffusers allows the maximum pressure recovery for a given length or the minimum length for a specified performance, since it permits the effective use of higher area ratios than other combinations. The optimum area ratio can be chosen from curves such as those of Fig.2, constructed using the analogy [8] and conical diffuser data for appropriate inlet conditions. Having chosen area ratio and length, the wall angles of type 3 diffusers are given by

$$\tan \delta_o = \frac{\sqrt{\mu} - 1}{\lambda}, \text{ and} \qquad \ldots(4)$$

$$\tan \delta_i = h \tan \delta_o \ . \qquad \ldots(5)$$

(b) Type 3 diffusers will suffer less from the adverse effects of inlet swirl than types 1 or 2.

(c) For performance prediction, the analogy [8] is recommended in conjunction with reliable data on conical diffuser performance. The mathematical model of Refs.1 and 13 or the experimental data of Ref.2 is recommended.

7. CONCLUSIONS

Experimental results and observations from several sources, including the present work, indicate that in addition to the commonly used variables of area ratio and length ratio, wall angle combination and hub ratio are important geometric parameters in the design of straight-walled annular diffusers. The velocity profile and turbulence level at inlet, influence performance and optimum geometry of diffusers.

For exhaust applications, divergent centrebodies allow a shorter diffuser than convergent centrebodies for a given performance, and they are less susceptible to the adverse effects of inlet swirl. A wall angle combination such that the apexes of inner and outer cones are coincident is recommended for minimum length at a specified performance or best performance for a given length. Given reliable conical diffuser performance data, the analogy described in Ref.8 predicts performance of straight walled annular diffusers to an acceptable accuracy for engineering design, provided inlet swirl is moderate. The analogy is a valuable tool for engineering design because of its simplicity.

APPENDIX A

DERIVATION OF OPTIMUM WALL ANGLE COMBINATION FOR SHORT DIFFUSERS

Given a straight walled annular diffuser of hub ratio h, area ratio μ, and wall angles δ_o, δ_i; according to Ref.8, the analogous conical diffuser has the same area ratio and wall angle δ given by

$$\tan \delta = \frac{\tan \delta_o - h \tan \delta_i}{1 + h} \qquad \ldots(A1)$$

For a straight walled annular diffuser, the area ratio

$$\mu = \frac{(1 + \lambda \tan \delta_o)^2 - (h + \lambda \tan \delta_i)^2}{1 - h^2} \qquad \ldots(A2)$$

where $\lambda = L/R_1$ for the annular diffuser, and $\tan \delta_i > -h/\lambda$ (centrebodies not closing before exit).

Therefore,

$$\mu(1-h^2) = (\tan^2\delta_o - \tan^2\delta_i)\lambda^2 + 2(\tan \delta_o - h\tan \delta_i)\lambda + 1 - h^2 \qquad \ldots(A3)$$

From (A1), $\tan \delta_o - h \tan \delta_i = (1 + h)\tan \delta$, therefore

$$z \lambda^2 + 2C_1\lambda - C_2 = 0 \qquad \ldots(A4)$$

where

$$z = \tan^2\delta_o - \tan^2\delta_i \qquad \ldots(A5)$$

$$C_1 = (1 + h)\tan \delta \qquad \ldots(A6)$$

$$C_2 = (\mu - 1)(1 - h^2) \text{ , and} \qquad \ldots(A7)$$

$$\lambda = \frac{-C_1 \pm \sqrt{C_1^2 + C_2 z}}{z} \qquad \ldots(A8)$$

To find the shortest analogous annular diffuser for a given conical diffuser, stationary values of λ are sought taking $\tan \delta_i$ as the independent variable, thus

$$\frac{d\lambda}{d(\tan \delta_i)} = \frac{d\lambda}{dz} \cdot \frac{dz}{d(\tan \delta_i)} \qquad \text{...(A9)}$$

It can be shown that $d\lambda/dz$ has no zero solution and the stationary value of λ at $dz/d(\tan \delta_i) = 0$ is a minimum satisfied by

$$\tan \delta_i = h \tan \delta_o \qquad \text{...(A10)}$$

The wall angle combination (A10) defines a family of diffusers with common apexes of inner and outer cones. It can be shown that this combination also corresponds to the longest analogous conical diffuser for a given h, μ and λ.

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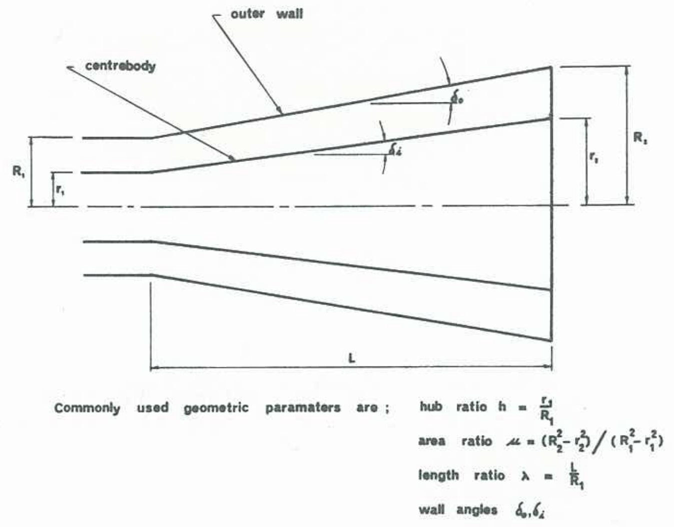

Fig.1. Geometric parameters - annular diffusers.

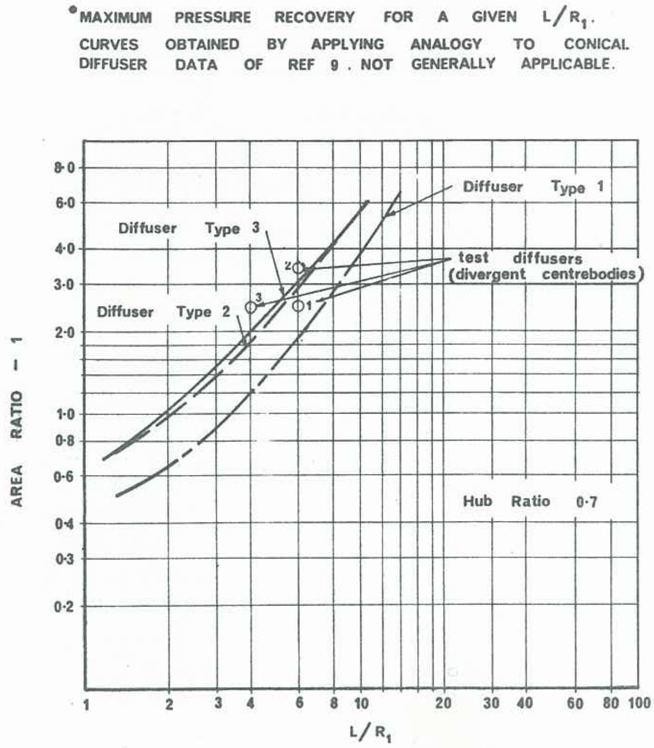

Fig.2. Influence of wall angle combination on optimum geometry.

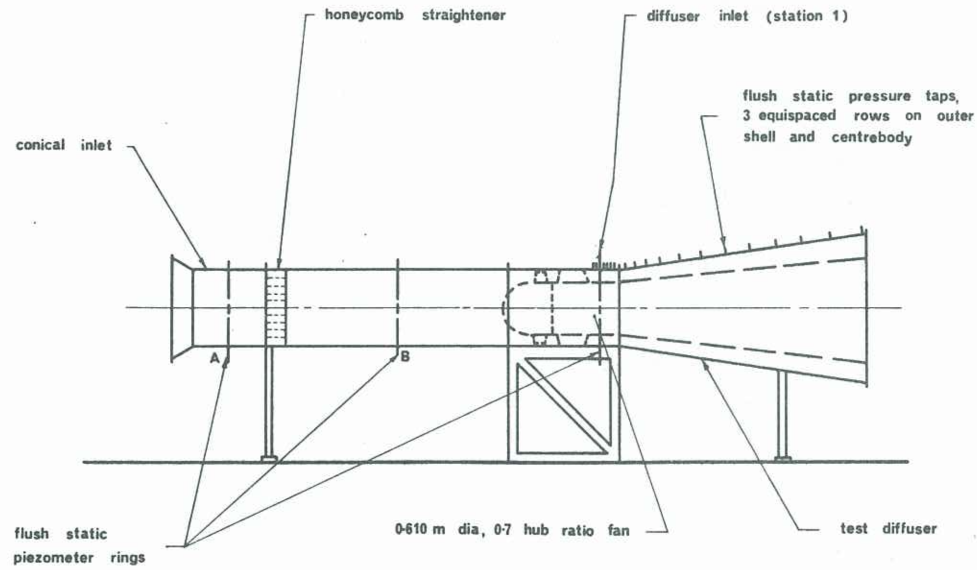

Fig.3. Test rig.

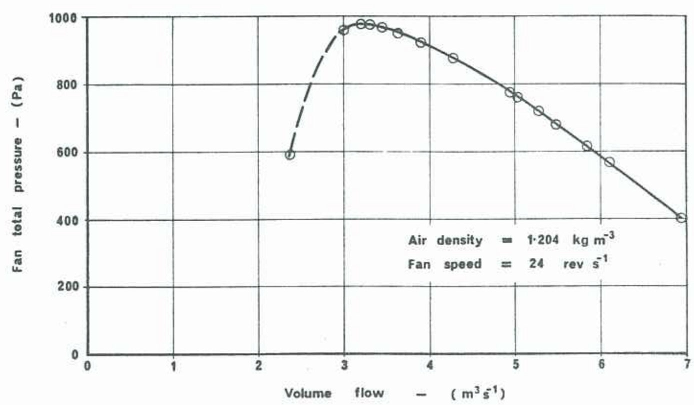

Fig.4. Fan Total Pressure characteristic.

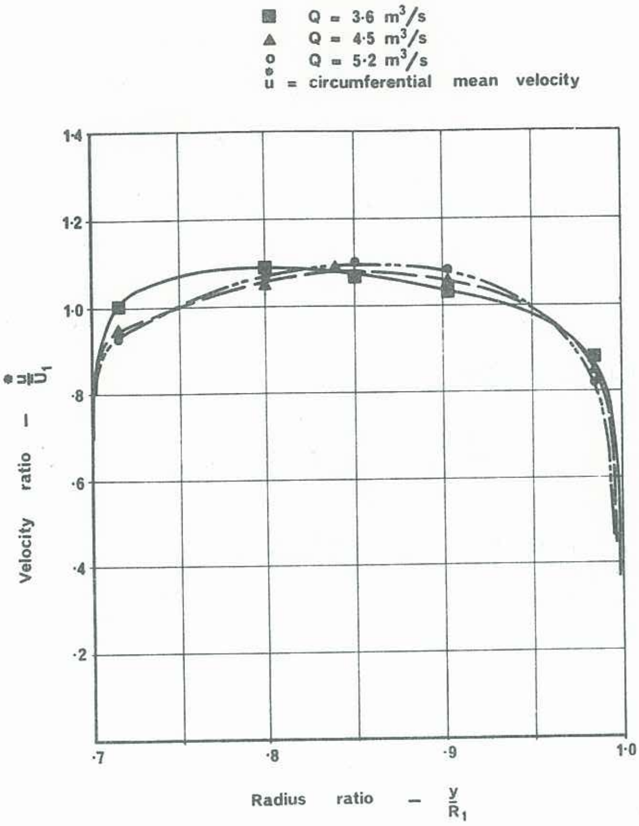

Fig.5. Radial distribution of velocity at fan exit.

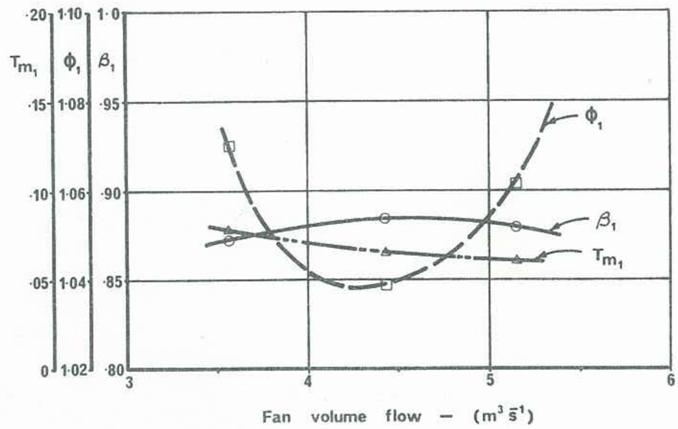

Fig.6. Variation of inlet flow parameters with fan load.

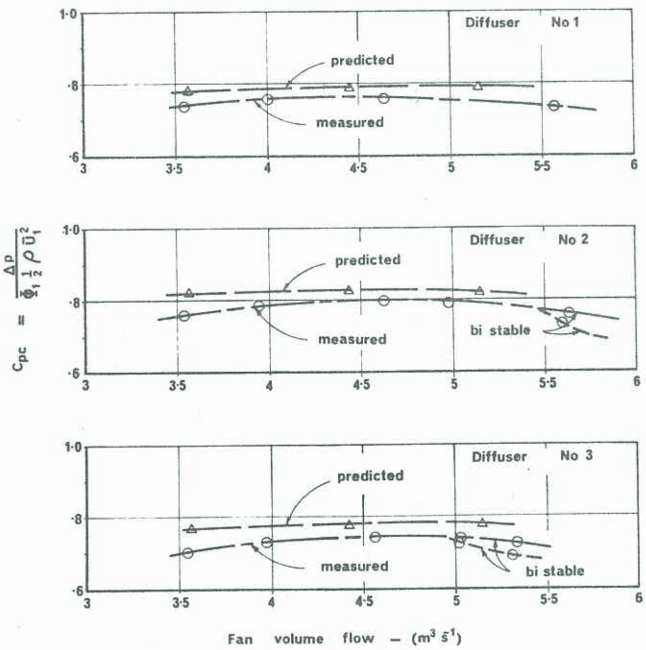

Fig.7. Variation of diffuser performance with fan load.