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OPTIMIZATION OF VANE - ISLAND DIFFUSERS AT HIGH SWIRL

BY

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ABSTRACT

An experimental investigation on the performance of Vane-island diffusers at highly distorted inlet flow is presented in this paper. The flow field at diffuser inlet is characterized by a high swirl number ($\lambda = 9$) and unsteadiness. The design of the vanes is optimized starting from the available data in the literature for plane diffuser. The optimum pressure recovery is optimized with regard to the geometry of the interaction between impeller exit and vane leading edge (diffuser inlet). Three different vane setting angle were investigated and their results were reported.

The effect of free stream turbulence on the performance is also studied. The turbulence is generated by a set of cylinderical rods perpindicular to the flow and parallel to the diverging walls of the diffuser.

The experimental results show that the vane setting angle must be considered in the performance optimization process of the Vane-island diffusers. The increment of the vane setting angle by 5° over the design setting angle (based on the absolute velocity angle at exit) improves the pressure recovery by about 7 %. Also it was found that decreasing the setting angle deteriorates the diffuser performance.

The experimental results has also shown that the increasing of turbulence level at diffuser inlet improves the pressure recovery.

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INTRODUCTION

The radial diffuser is one of the basic components of fluid machinary as blowers and compressors. Optimizing these machines requires a proper impeller-diffuser matching. Small turbomachinery usually running at high swirl which resulted in highly distorted velocity profiles at diffuser inlet. This will act in the account of provoking an early separation. Due to this nature of flow, precise diffuser recovery prediction is required. This is still an unsolved problem, which explains why the majority of diffuser researchs has been experimentally.

An understanding of basic mechanisms which control diffuser performance will improve the design, and hence optimizing the efficiency. In vanned diffusers four different regions of flow are present [1]; interaction impeller-vaneless diffuser, vaneless diffuser, interaction vaneless diffuser-vaned diffuser, and vanned diffuser. The experiments of Eckardt [2] has shown that jet-wake flow pattern at impeller exit is mixed at a distance approximately equals1.15 R_{im}.

In the literature there is a large amount of research work devoted to the prediction of static pressure recovery in plane diffusers. This extensive research work has primerly considered the cases where the inlet velocity profile is steady and uniform [3,4,5]. These researchs reveal that some stall is present in high performance diffusers. The results of these research work are today very often used for diffuser design. The experimental results of Kaiser and McDonald [6] show that diffusers with distorted inlet velocity profiles exhibit stall behaviour quite different from that found in diffusers with uniform inlet profiles. The actual inlet velocity profiles in the radial diffusers of compressors and blowers are far from being uniform and steady. It is controlled by the mechanism of flow at impeller exit.

The researchs [1,2,7] on vanned diffusers show that their performances are largely dependant on the flow mechanisms in the region of interaction between vaneless and vanned diffuser. In order to improve the mixing process in the plane diffuser entry region, Hoffman [8] has increased the turbulence intensity level at the entry region by fixing vertical cylinderical rods at the entry. The results show that the diffuser's pressure recovery coefficient was increased by about 24 %. Also it was shown that the eddy axis orientation in the direction perpendicular to the flow and parallel to the diverging walls of the diffuser, apparently more effectively transmits turbulent energy to the diffuser walls. The application of this technique is attractive in radial diffuser, since the problem is the non-unifomity of the flow pattern. The increment of turbulence intensity level will act in the account of accelerating the mixing process.

The objective of this investigation is to study the effect of vane setting angle and free stream turbulence on the performance of Van-island diffusers.

EXPERIMENTAL APPARATUS

A schematic diagram of the test rig is shown in Fig.(1) and Fig.(2). The test rig consists of a single stage blower, the design speed is 15000 rpm, and the design flow rate is 136 m^3/hr. The blower has a radial impeller of 12 cm diameter and 8 mm hight, equipped with five blades curved backword at 148°. The diffuser was made of two 24 cm. diameter parallel disks of plexyglass. The plexyglass was chosen to permit flow visualization.

A number of twenty vanes were made of plexyglass, the detailed dimensions of the vane are shown in Fig. (3). The design of the vanes were performed according to the data of references [1,3 and 4]. The diffuser data are given in Table I. After Barclina et al [4] the diffuser is laying in the region of inception and transitory stall. This was expected since stall is presented in high performance diffusers. It is important to note that a rotating stall which is a characteristics of vaneless diffuser is appeared in the present investigation. This explains the necessity of increasing the static pressure taps at the diffuser inlet at different locations in order to avoid reporting the readings of a stalled diffuser channel. Tweleve static pressure taps are used, three of them to check only the repeatability and stall characteristics in the vaneless space. The total pressure at impeller exit was measured by Kiel probe. The temperatures at impeller exit and diffuser exit were measured by thermocouples. There were no significant difference in temperatures. The static pressure taps were connected to water manometer. The flow rate was measured by a set of ASME thin orifice plates placed at the inlet of plenum chamber. For each run the overall pressure differential was obtained from a static pressure reaching at the plenum chamber with reference to the ambient.

The vanes were set at three different angles corresponding to the design flow angle based on the theoritical velocity triangle. This value is increased by 5° to include the expected deviation according to real flow pattern at impeller exit. The second angle is smaller by 5° than the first while the third angle is greater by 5° than the first one. The selection of these two setting angles was made to permit the study of the influence of increasing and decreasing the setting angle. Figures (4) and (5) show the arrangement of the diffuser vanes.

A set of cylenderical rods of 3 mm diameter were fixed at the diffuser inlet, the rods were set at a distance equal to 1.2 R_{1m}. At this stage of experiment, only one rod was constructed at the inlet of each diffuser channel.

The calculation of the static pressure rise coefficient C_p, is performed from the following expression;

$$C_p = (P_r - P_i) / (\frac{1}{2} \rho C_i) \qquad ....(1)$$

The diffuser effectiveness is defined as:

$$\eta_e = (C_p) / (C_{p\ ideal}) \qquad ....(2)$$

where, $C_{p\ ideal}$ is the diffuser recovery based on the diffuser area ratio.
The swirl coefficient is calculated using the following expression;

$$\lambda = C_u / U \qquad ....(3)$$

where; C_u is calculated from the theoritical velocity triangle.
The flow coefficient is calculated as follows;

$$\phi = C_r / U \qquad ....(4)$$

where, C_r is the radial component of the absolute velocity calculated from the mass balance.
The diffuser total pressure loss coefficient was calculated from the following relation;

$$\xi = (P_{oi} - P_{oe}) / (\frac{1}{2} \rho C_i^2) \qquad ....(5)$$

An error analysis was carried out according to standard procedures. It was found that the estimated uncertainty of the experimental measurements is ± 2 mm in the differential U tube manometer connected to the plenum chamber, and ± 1 mm in the static pressure readings of pressure distribution along the diffuser channel. This will result in a relative error less than ± 1 % in discharge measurements and ± 3 % error in the static pressure rise coefficient and total pressure loss coefficient.

RESULTS

The diffuser performances are presented in Fig. (6), (7), and (8). The results were presented as the static pressure coefficient C_p versus the dimensionless distance R/R_{im}, along the diffuser. The values of C used in the calculation of the non-dimension static pressure rise coefficient is based on the mean flow velocity at impeller exit. Due to the nature of flow which exhibits a nonuniformity and unsteadiness, the measured values of C were highly flactuated. These results were shown also by the measurements of Krain [7] which has shown a highly distorted, unsteady flow character with a large variation in the local flow angle. For these reasons the mean flow velocity based on the mass balance presents a mean value of the kinetic energy in the dominator of the C_p expression. Despite this convenience, the real flow velocity may be larger than this mean value, which may result in larger C_p values than the actual. This effect should not alter conclusions since these are based on a qualificative comparisons. The diffuser data is presented in Table I. The area ratio of the diffuser was kept approximately constant at 2.2 and the diffuser angle also was kept constant at $2\theta = 10^\circ$.

In each figure, two representative results at two different swirl coefficient are presented. The upper values of the awirl coefficient are 10, and the lower values are approximately 6. The dashed curves illustrate the experimental results with a higher turbulence level created by a set of vertical cylinderical rods. The dimensions of rods and setting angle were taken from reference [8] taking into consideration the nature of the flow. The rods were fixed at a distance equal 1.2 R_{im} to assure complete mixing.

The largest static pressure rise coefficient of 0.73 was obtained at the third setting angle. The diffuser effectiveness versus the flow coefficient ϕ, is presented in Fig. (9). The total pressure loss coefficient versus the flow coefficient, along with data of Rayan et al [1] and Klassen et al [9] are shown in Fig. (10). The present data shows a significance improvement of Clemson's data [1].

DISCUSSIONS

The results of the static pressure recovery coefficient presented in figures (6), (7), and (8) show clearly the high performance achieved. These results show that the data of Runstadler et al, and Bordina et al for plane straight channel diffuser present a good basis for the radial diffuser design. In figures (6) and (8) the pressure recovery distribution along the radial distance is generally uniform. In Fig. (7), corresponding to case II, the pressure recovery distribution exhibits a non-uniformity which may be attributed to stall and reverse flow. The pressure recovery in the entery region in this case is nearly constant independent of the swirl coefficient to a distance 1.25 R_{im}. The pressure recovery in the vaneless area is about 15 % of the total pressure recovery. This ratio reachs 45 % in cases I and 30 % in case III.

The setting angle I is corresponding to the theoretical flow exit angle plus 5° of estimated deviation. There is no uniform idea about the deviation angle, which is the direct result of the jet-wake flow pattern at the impeller exit, it ranges from 3° to 10° and recently the experiments of Krain [7] show that it may reach 15° in severe deformation. The results of the present investigation show clearly that increasing the setting angle by 5° over the assumed deviation of 5° improves the static pressure recovery by 10 %. The increment of 5° is in fact an increment in the deviation angle to reach 10°

which seems reasonable regarding the data of Eckardt et al [2] and Krain [7].

The vaneless space is an important factor in determining the overall pressure recovery coefficient. In fact, the length of the vaneless space is connected to the mechanism of flow at impeller exit, it was shown by Eckardt [2], that the mixing is completed at a distance of 1.15 R_{im}. In the present investigation this distance was taken approximately equal 1.2 R_{im}. This distance changes slightly with the change in vane setting angle. It is clear from figures (6), and (8) that the pressure recovery in the vaneless space presents 25 to 45 % of the total pressure recovery.

Increasing the turbulence level may improve and accelerate the mixing process. This technique has been used successfully in straight wall diffuser [8], the use of this technique seems to be promising in radial diffuser. From figures (6), (7), and (8), when cylinderical rods were used to increase the turbulence, the performance of the diffuser become uniform. The pressure recovery in the vaneless space is less dependant on the vane setting angle and swirl. In the diffuser channel itself an increase of 5 to 7 % in the static pressure coefficient is achieved. A detailed investigation of this effect is necessary to classify this technique.

The diffuser effectiveness show that set I, and III gave the best diffuser effectiveness with relatively wide range of the flow coefficient ϕ as shown in Fig. (9).

In Fig. (10) the present results were compared with previous results obtained by Clemson university turbomachinery laboratory [1], and NASA results obtained by Klassen et al [9]. The total pressure loss coefficient is improved regarding Clemson's results, it is slightly higher than NASA data, this may be attributed to the large swirl and low flow coefficient.

CONCLUSIONS

Based on this investigation, the following conclusions and recommendations are offered:

1- Increasing the vane setting angle improves the diffuser performance, an improvement as high as 10 % is achieved.
2- Increasing the turbulence level at diffuser inlet improves the diffuser performance, also it increases the static pressure recovery in the diffuser.
3- The increasing of the turbulence level at inlet rends the recovery in the vaneless space constant at approximately 30% independent of the vane setting angle.

Some of the above conclusions are tentative and require further investigations.

ACKNOWLEDGEMENTS

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REFERENCES

1 Rayan, M.A. and Yang, T.T. "An Investigation of Vane-Island Diffusers at High Swirl" ASME Paper 80-GT. 148, March 1980.

2 Ekardt, D.,"Jet-Wake Mixing in the Diffuser Entery Region of High-Speed Centerifugal Compressor",Proceedings of Joint Symposium on Design and Operation of Fluid Machinery, June 1978, Color. State University.

3 Runstadler, P.W., Jr., Dolan, F.X., and Dean, R.C., Jr., Diffuser Data Book Creare. TN-186, 1975.

4 Bardina, J., Lyrio, A., Kline, S.J., Ferziger, J.H.,and Johnston, J.P., "A Prediction Method for Planar Diffuser Flows" Trans. AMSE, J. Fluids Engineering, Vol. 103/315 PP. 315-321, June 1981.

5 Ashjaee, J., and Johnston, J.P., "Straight - Walled, Two-Dimensional Diffusers Transitory Stall and Peak Recovery" Trans.AMSE, J.Fluid Engineering, Vol.102/275, PP. 275-282, Sept. 1980.

6 Kaiser, K.F., and McDonald, M.T., "Effect of Wake-Type Non-uniform Inlet Velocity Profiles on First Appreciable Stall in Plane-Wall Diffusers", Trans.ASME, Vol. 102, PP.283-289, Sept. 1980.

7 Krain, H. "A Study on Centrifugal Impeller and Diffuser Flow", Trans. ASME, J. Engineering for Power, Vol. 103, PP. 688-697, Oct. 1981.

8 Hoffman,J.A. "Effects of Free-Stream Turbulence on Diffuser Performance", ASME, J. Fluid Engineering, Vol. 103, PP.385-390, Sept. 1981.

9 Klassen, H.A., Wood, J.R., and Schumann,L.F., "Experimental Performance of a 13.65 Centimeters Tip Diameter Tandem Bladed Swept Back Centrifugal Compressor Designed for Pressure Ratio of 6", NASA, Technical Paper 1091, Nov. 1977.

NOMENCLATURE

A area
B diffuser
C absolute velocity
D impeller diameter
L diffuser length
P pressure
R radius
U impeller tip velocity
W diffuser width
AR diffuser area ratio
AS diffuser aspect ratio
C_p static pressure rise coefficient

Greek letters

α vane settin angle
2θ diffuser diverging angle
ϕ flow coefficient
λ swirl coefficient based on the ideal velocity tringle at impeller exit
ξ total pressure loss coefficient
ρ density
η diffuser effectiveness

Subcripts

e diffuser exit
i diffuser inlet
im impeller exit
o total pressure
r radial component
t diffuser throat
u tangential component

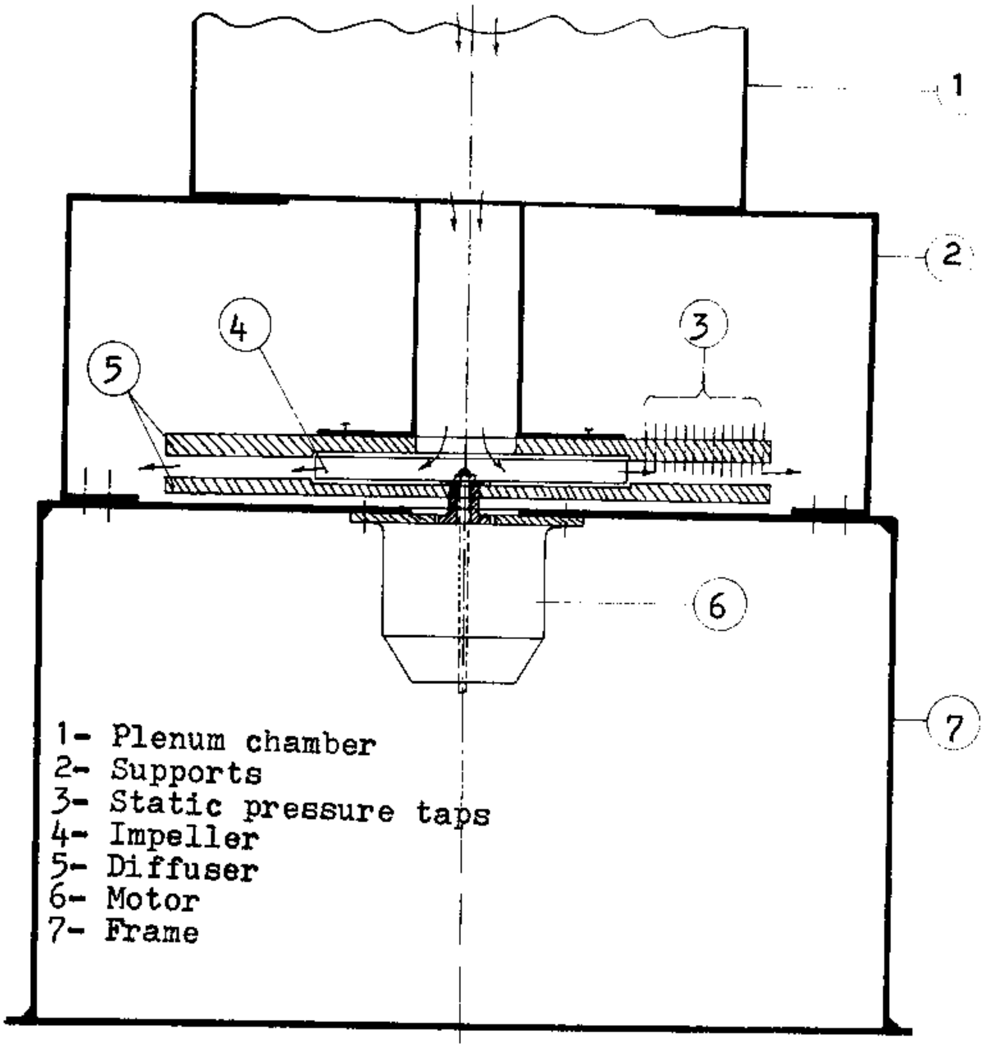

Fig.1. Schematic diagram of test rig.

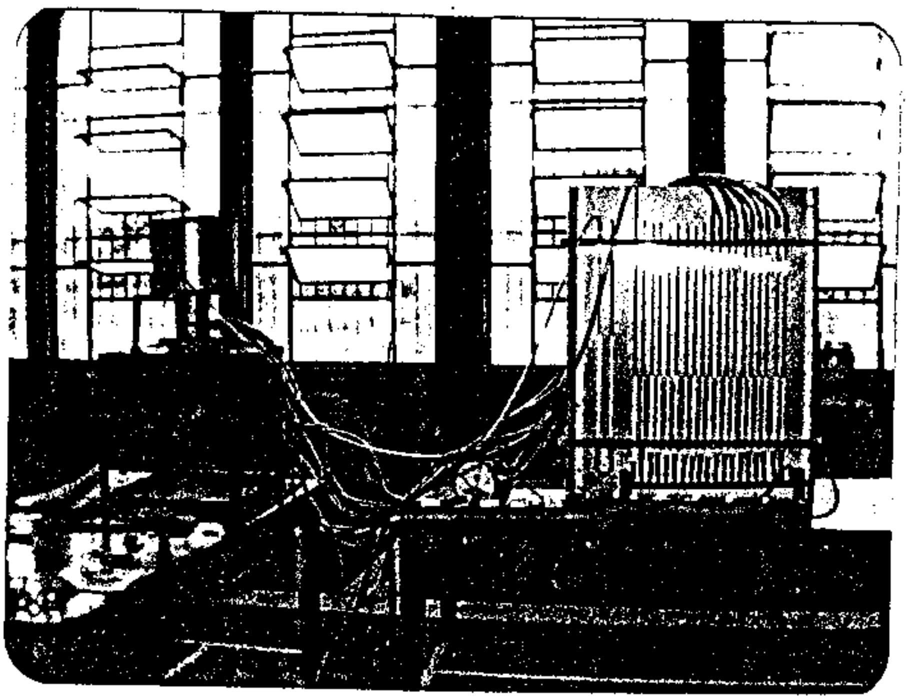

Fig.2. Test rig.

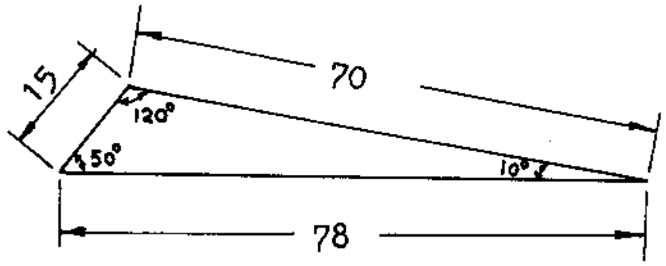

Fig.3. Vane dimensions (mm)

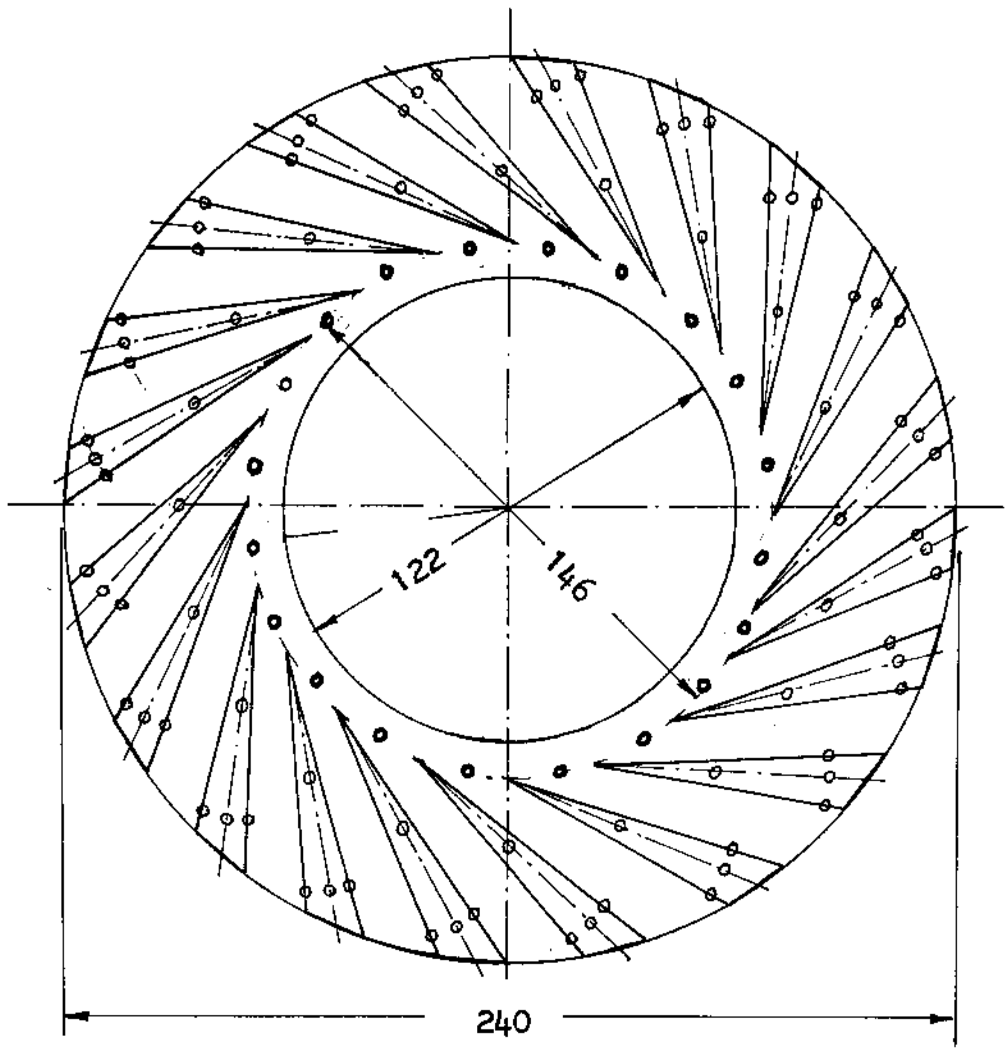

Fig.4. Arrangement of diffuser vanes

Table I

Vane setting angle	R_t/R_{im}	AR	AS	2θ	L/W	A_e/A_{im}	ξ_{best}
I	1.2	2.2	0.89	10°	7.56	1.06	0.10
II	1.13	2.5	1.14	10°	9.50	0.96	0.14
III	1.23	2.2	0.80	10°	7.00	1.16	0.10

Diffuser height = 0.8 cm
No. of Vane Island = 20

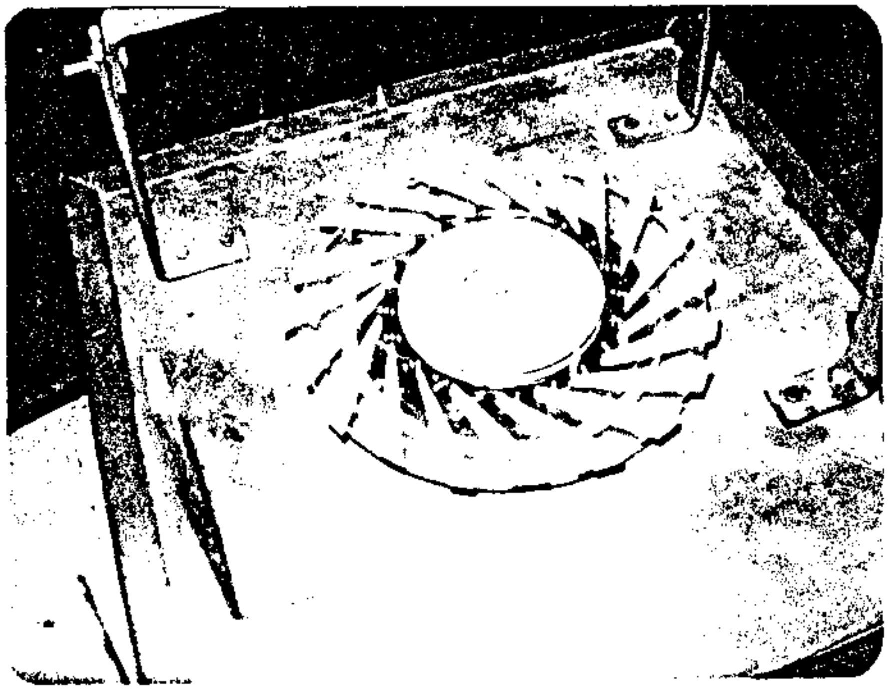

Fig.5. Diffuser vanes

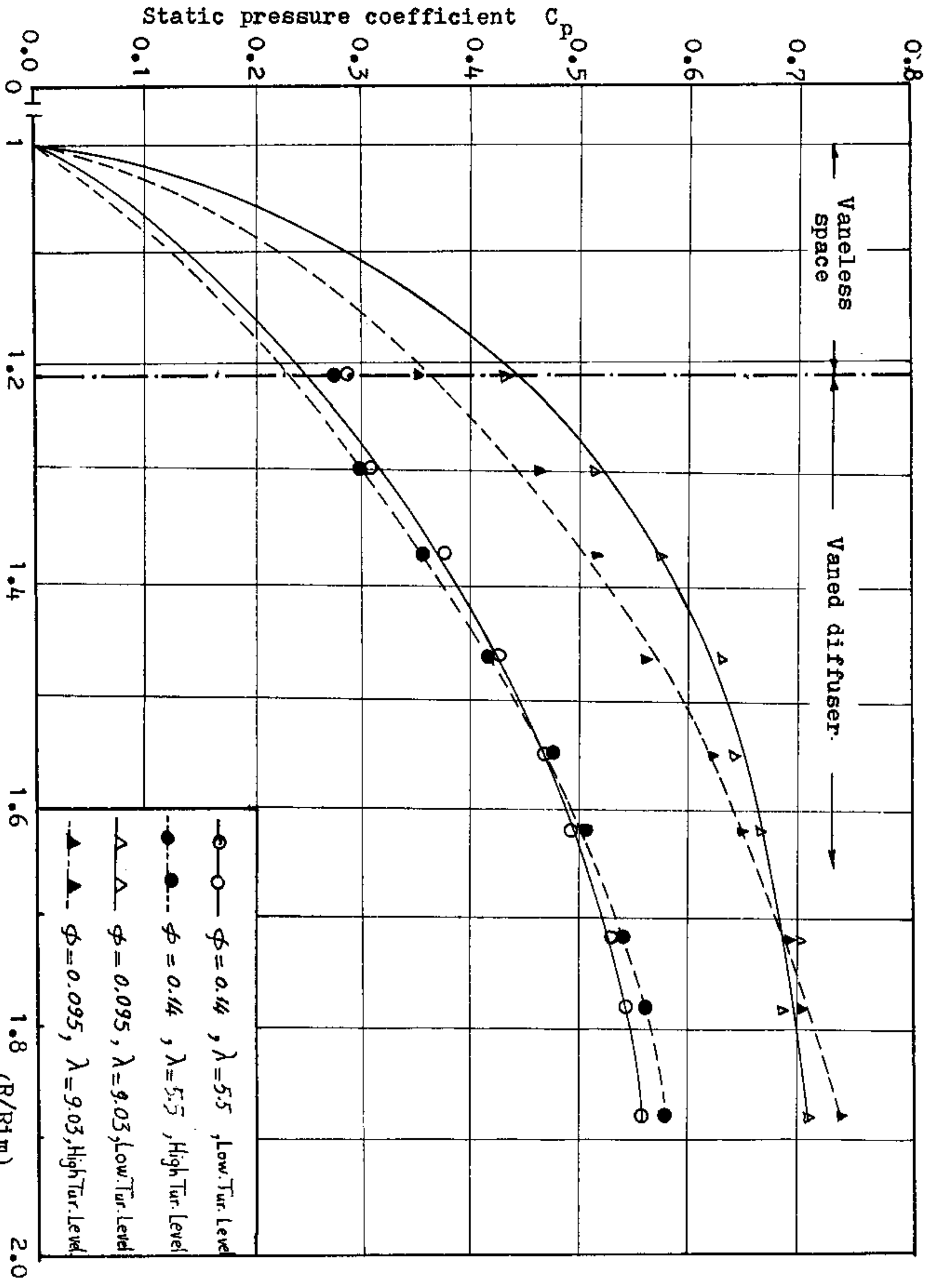

Fig. 6. The static pressure coefficient C_p versus the dimensionless distance R/R_{im}

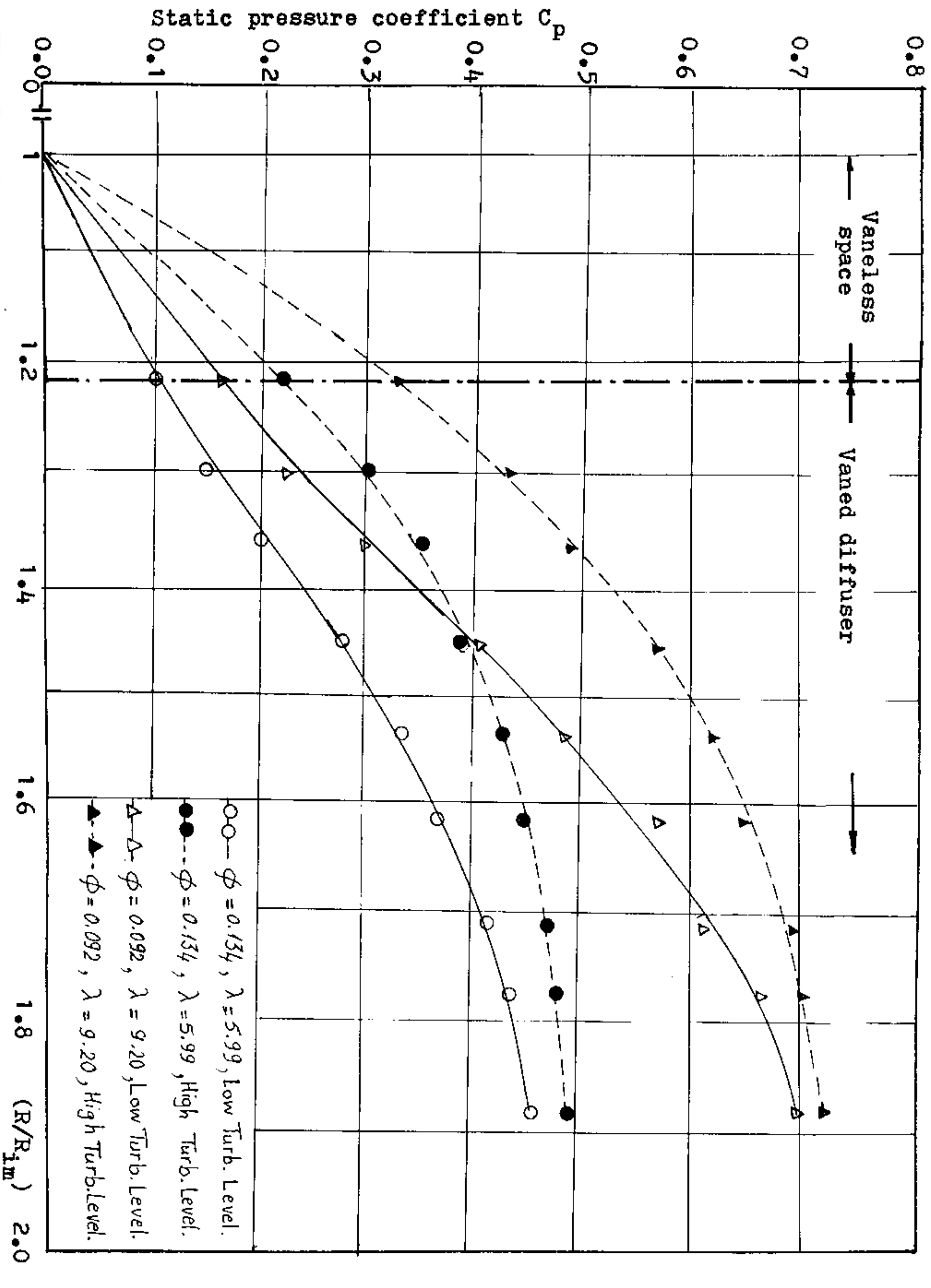

Fig. 7 ; The static pressure coefficient C_p versus the dimensionless distance R/R_{im}

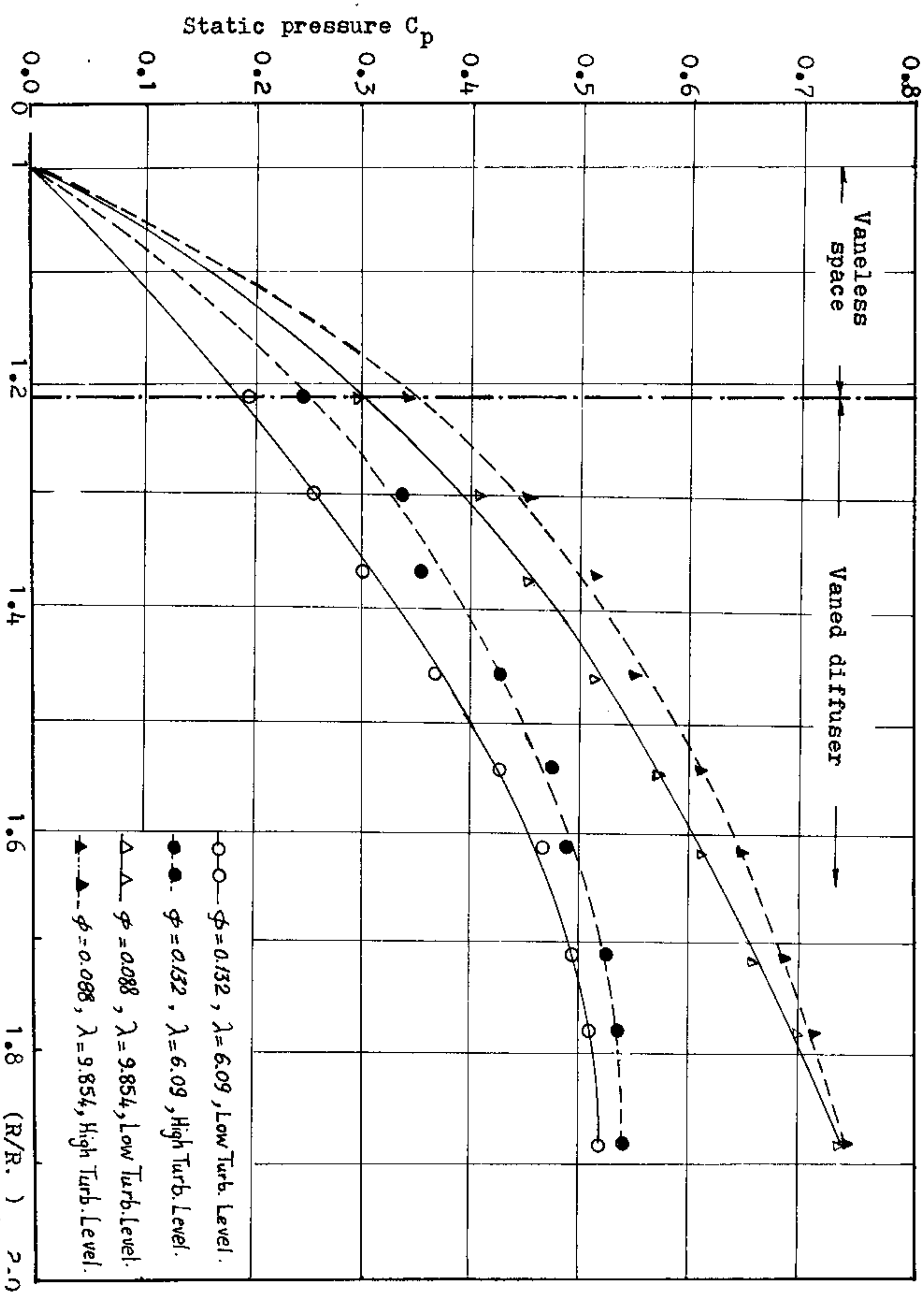

Fig. 8 . The static pressure coefficient C_p versus the dimensionless distance R/R_{im}

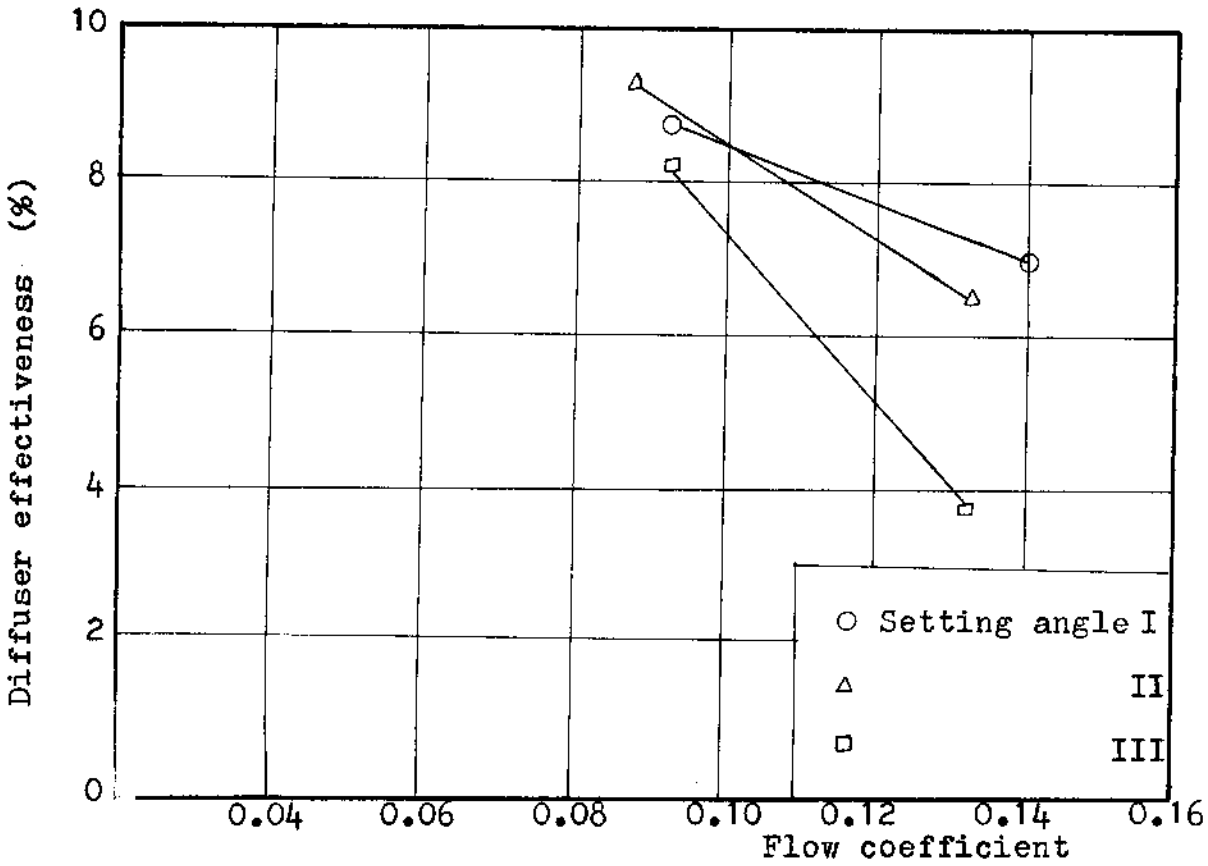

Fig.9. Diffuser effectiveness versus the flow coefficient

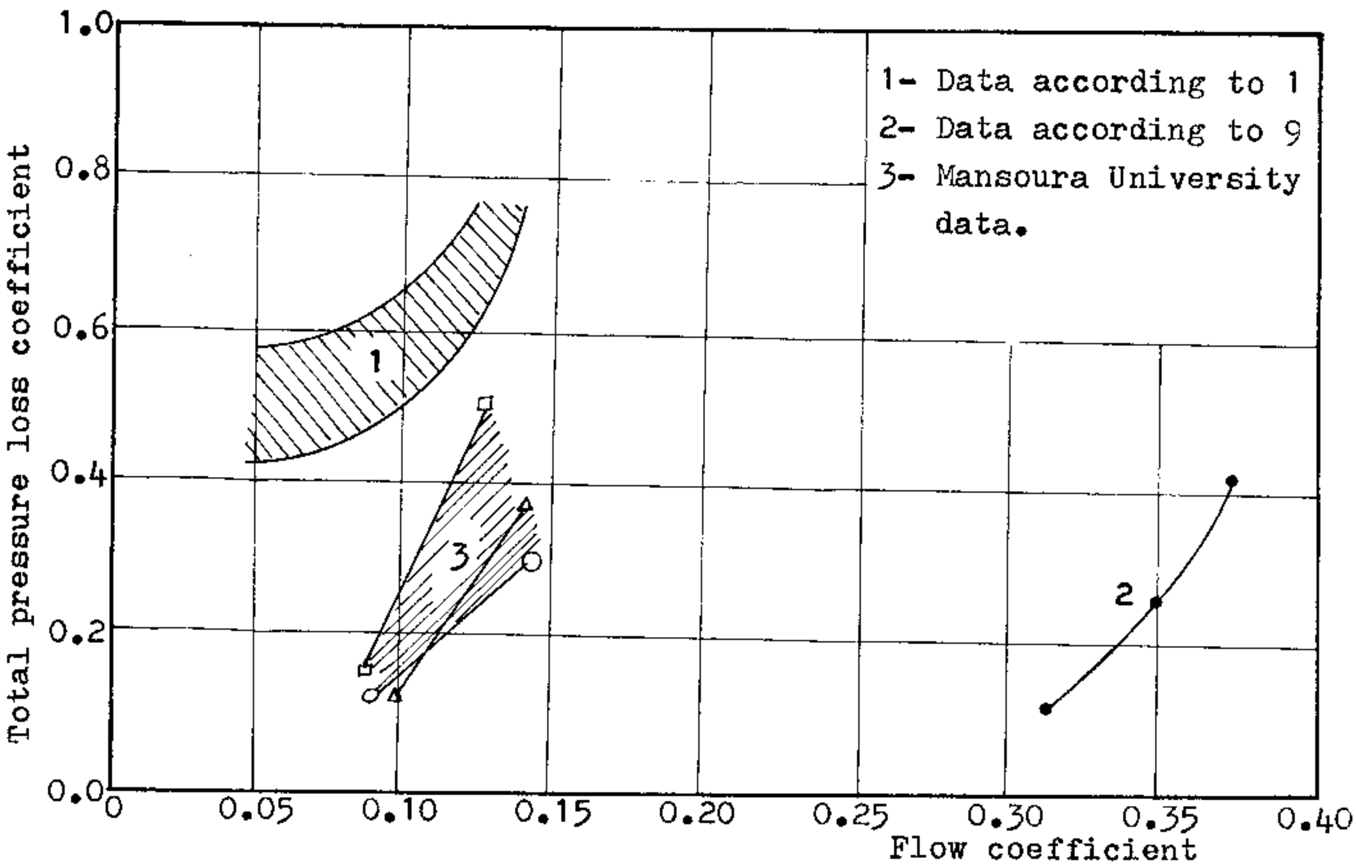

Fig.10. Total pressure loss coefficient versus flow coefficient