FULL LENGTH PAPER

EFFECT OF INLET VANE ANGLE OF A PARTIAL VANE DIFFUSER ON THE PERFORMANCE AND FLOW FIELD OF A CENTRIFUGAL COMPRESSOR

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Abstract

The paper presents experimental results on the effect of inlet vane angle on the performance and flow field of Partial Vane Diffuser (PVD) of a low speed centrifugal compressor. Performance tests are conducted for eight configurations of PVDs. From these tests, an optimum configuration with 5° increase in inlet angle of partial diffuser vanes on shroud (PVDHS5) is selected for further measurements i.e. static pressure on the diffuser hub and shroud and diffuser passage flow field at four flow coefficients. From passage flow measurements flow parameters are obtained within the diffuser flow passage. Flow parameters are axially averaged and their contours are presented. Radial variation of these flow parameters is also presented. PVDHS5 gives 2.3% increase in peak energy coefficient and 4.9% increase in efficiency compared to that of vane less diffuser. The flow at the exit of PVDHS5 is more uniform.

Keywords:*Centrifugal compressor; Partial vane diffuser; Inlet vane angle; Experimental studies*

Nomenclature

C_D = Discharge velocity (m/s)
C_m = Meridional velocity (m/s)
C_S = Suction velocity (m/s)
D_2 = Rotor tip diameter (m)
D_S = Inlet duct diameter (m)
N = Speed in rpm
N_C = Coupling power (Watt)
U_2 = Blade tip speed = $\pi D_2 N/60$
P = Static pressure (N/m^2)
P_D = Average static pressure at delivery side (N/m^2)
P_O = Total pressure (N/m^2)
P_S = Average static pressure at suction side (N/m^2)
R = Radius ratio = r/r_2
r = Radius (m)
S = Blade spacing (m)
V = Volume flow (m^3/s)
W = Specific work (m^2/s^2) = $(P_D-P_S)/\rho + (C_D{}^2-C_S{}^2)/2+g\Delta Z$
ΔZ = Geodetic level difference between delivery and suction flanges (m)
ϕ = Flow coefficient = $V/\pi D_2 b_2 U_2$
γ = Power coefficient = $8N\varepsilon/(\rho\, \pi\, D_S^2\, U_2^3)$
η = Efficiency = $\rho VW/\eta_m N_m$

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ρ = Density of air (kg/m^3)

ψ = Energy coefficient = $2W/U_2^2$

ψ_o = Total pressure coefficient = $2P_o/\rho\ U_2^2$

ψ_s = Static pressure coefficient =$2P_s/\rho\ U_2^2$

Superscripts

- = Axially averaged value

= = Mass averaged value

Abbreviations

H = Hub

S = Shroud

VD = Vane diffuser

VLD = Vaneless diffuser

PVDHS = Partial vane diffuser of height equal to 30% of diffuser passage width and with 11 vanes on hub and shroud staggered at 50% of vane spacing

Introduction, Motivation and Objective

Centrifugal compressors are designed for low mass flow rates at high pressure ratios and with wide operating range. They find a wide range of applications especially for power plants in small aircraft, helicopters, turbochargers, in process industries, compression of gases and vapours, refrigeration etc. In a centrifugal compressor energy is transferred to the fluid by the rotating rotor. The flow leaves the rotor with a high kinetic energy. This kinetic energy is converted to static pressure in the diffuser of the centrifugal compressor. Hence diffuser is an important component of centrifugal compressor. Different types of diffusers are in use, mainly based on application. If the required pressure ratio is not high and cost is a prime concern then a vaneless diffuser is normally used. It is easy to manufacture and has a wide operating range. If high pressure ratio and efficiency are main concerns, as in the case of turbochargers, then a vane diffuser is normally used. However it has limited operating range and its efficiency falls rapidly at off-design flow coefficients. Hence various new types of diffusers were proposed so that both efficiency and operating range are good. The first among these is low solidity vane diffusers (LSVD) proposed by Senoo in 1978 [1]. Extensive work is done on low solidity vane diffusers [2-4] of low speed and subsonic centrifugal compressors. Another new type of diffuser is partial vane diffuser, characterized by the height of the vane being less than the flow passage width. It was first introduced by Yoshinaga et al. in 1987 [5]. Its efficiency and pressure rise are higher than that of a vaneless diffuser but lower than that of a vane diffuser. It has better operating range than a vane diffuser. Different types of diffuser arrangements are shown in Fig.1. The present investigation is aimed to further improve the performance of centrifugal compressor with partial vane diffuser. This is planned to achieve by finding out an optimum inlet angle for the diffuser. It may be mentioned that variable diffuser angle is used by some manufacturers [6], but these diffusers have vanes with height equal to the flow passage width. A recent computational investigation had shown that diffuser with variable inlet angle improves the performance of centrifugal compressors used in turbocharger [7]. These studies are also carried out on full vane diffusers. The present investigation is a continuation of our previous work aimed at improving diffuser performance in centrifugal compressors in general with the possibility of applying these concepts to turbocharger compressors.

If the flow angle at the inlet of the diffuser is different from the vane diffuser inlet angle, the flow is forced to change its direction from that of vane itself just inside the passage. This causes a reduction in the velocity at the diffuser inlet. The loss due to this change of direction of the inlet velocity may be assumed to be proportional to the kinetic energy represented by the reduction in velocity. In the present experimental investigation, the inlet angle of the diffuser vane is varied to obtain different configurations and performance test is conducted for each of these configurations. From these tests, an optimum configuration is selected for further analysis.

Experimental Facility and Program

Experimental Facility

Experiments are conducted in a low speed centrifugal compressor available in the Thermal Turbomachines Laboratory, Department of Mechanical Engineering, Indian Institute of Technology Madras. This facility is used by Sitaram et al. [8] for systematic experimental investigation of partial vane and other types of diffusers. However for the sake of completeness, brief details of the facility are provided below.

A schematic lay out of the experimental set up is shown in Fig.2a and the meridional view of the centrifugal compressor is shown in Fig.2b. The set up mainly consists of a single stage centrifugal compressor, which is directly coupled to an A.C. motor having specifications of 10 HP, 50 Hz, 13.5A with a rated speed of 3,000 rpm. The motor is controlled by a variable frequency drive, so that the speed can be controlled within ±1 rpm. The major design details of the compressor and partial vane diffuser are presented in Tables-1 and 2 respectively. Although the design speed of the compressor is 4,500 rpm, earlier performance measurements (with a different drive so that the compressor can be run upto a speed of 4500 rpm) had shown that the effects of Reynolds number were negligible above a speed of 3,000 rpm. For the present investigation partial vane diffuser fixed to both hub and shroud is used. The blades are designed as per the method suggested by Spraker and Young [9]. The blade height is 6 mm and has a thickness of 3 mm. The inlet angle of blade is 15° and exit angle is 25° with respect to tangential direction.

Instrumentation

For the measurement of pressure a micro manometer (Model: FCO12; range: 2000 mm WC; accuracy is ±1 mm of water column) manufactured by M/s Furness Control Ltd., Bexhill, U.K and a scanning box (FCO 91-3) is used. Scanning box has twenty channels. Speed is measured with a non contact digital type tachometer. The range of tachometer is 0-9999 and its accuracy is ±1 rpm. The power is measured directly from the control panel of the motor.

For measurement of static pressure, holes of 1.1 mm diameter are drilled on the diffuser plate. One end of stainless steel tube of 1 mm diameter and 10 mm length, is bent 90 degree at mid length is inserted in these holes. A polyethylene tube of 1 mm diameter is inserted on the other end of stainless steel tube. Openings are provided in the casing to take out these tubes. There are 357 holes at 17 radial locations. The location of static pressure taps in two vane passages (or one partial vane passage) is shown in Fig.3a. The radial locations of the static pressure taps are also shown in the inset of this figure. At each radial location, there are 21 nos. of equally spaced static pressure taps covering two vane passages or one partial vane passage.

The flow in the diffuser passages is measured with a pre calibrated three hole probe. The traverses are carried in the holes that are located on circumferential arcs, from the rotor exit to the diffuser exit, at six radius ratios (1.071, 1.125, 1.176, 1.226, 1.303 and 1.379) at circumferential intervals of 25% as shown in Fig.3b. The probe traverse passages and passages containing the static pressure taps are situated far away from the volute tongue, (See inset of

Table-1 : Design Details of Centrifugal Compressor [Ref.8]

Pressure ratio, P_{02}/P_{01}	1.08	Design speed, n	4500 rpm
Mass flow, m	0.84 kg/s	Shape Number, $N_{sh} = n\sqrt{V}/W^{3/4}$	0.0843
Inducer hub diameter, D_{ih}	0.110 m	Vane angle at inducer hub, β_{ih}	45°
Inducer tip diameter, D_{it}	0.225 m	Vane angle at inducer tip, β_{it}	29°
Impeller exit diameter, D_2	0.393 m	Vane angle impeller exit, β_2	90°
Number of impeller vanes, Z	20	Vane diffuser L.E. diameter, D_3	0.432 m
Diffuser width, b_3	0.020 m	Vaneless diffuser exit diameter, D_5	0.600 m
Reynolds number based on impeller blade exit width = $U_2\, b_3/\nu_1$	0.82×10^5	Reynolds number based on diffuser chord = $C_3\, C_{h3}/\nu_1$	3.2×10^5 to 3.5×10^5
All angles are measured with w.r.t. tangential direction			

Table-2 : Details of Partial Vane Diffuser [Ref.8]

Solidity, σ	No. of Vanes	R_3	R_4	Chord, Ch (mm)	α_3 (Deg)	α_4 (Deg)	Camber Line Radius, (mm)
0.7	11+11	1.1	1.2514	86.07	15	25	485.5
PVDHS Partial vane diffuser with vane height of 0.3 times the diffuser passage width 11 number of partial vanes fixed on hub and shroud and staggered at half the vane spacing							

Fig.3b), to minimise the influence of volute tongue on the measured flow. The probe traverse holes, when not in use, are covered with brass plugs. A probe traverse mechanism is used for the axial and circumferential movement of the probe. It contains an axial traverse mechanism, which is mounted on a circumferential traverse mechanism. The mechanism is fixed to the back of the casing of the compressor. It has a protractor to measure the angle by which the probe is being rotated, and a linear scale to know the axial distance. The accuracy of the angular measurement is 1 degree and accuracy for linear movement is 0.1 mm.

Experimental Program and Procedure

The eight configurations that are tested for performance are given below:

VLD	vaneless diffuser
PVDHS	partial vane diffuser of height equal to 30% of diffuser passage width and with 11 vanes on hub and shroud staggered at 50% of vane spacing [Ref.11]
PVDHS5	similar to PVDHS, but inlet angle of vanes on shroud is increased by 5°
PVDHS-5	similar to PVDHS, but inlet angle of vanes on shroud is decreased by 5°
PVDH5S	similar to PVDHS, but inlet angle of vanes on hub is increased by 5°
PVDH-5S	similar to PVDHS, but inlet angle of vanes on hub is decreased by 5°
PVDHS10	similar to PVDHS, but inlet angle of vanes on shroud is increased by 10°
PVDHS5S5	similar to PVDHS, but inlet angle of vanes on hub and shroud is increased by 5°

All PVDs have a height of 6 mm i.e. 30% of diffuser width.

Performance tests are conducted at a constant speed of 3,000 rpm. The volume flow through the compressor is varied from maximum to zero. At each volume flow rate, suction pressure across nozzle throat, discharge pressure, speed and power input to the motor are noted. From these data, performance characteristics, i.e. energy coefficient (ψ), power coefficient (γ) and efficiency (η) are then calculated and are plotted against the flow coefficient (ϕ). Static pressure is measured on the diffuser hub and shroud at four flow coefficients, viz. ϕ=0.23 (below design flow coefficient), ϕ=0.34 (near design flow coefficient), ϕ=0.45 (above design flow coefficient) and ϕ=0.60 (near maximum flow coefficient for vane diffuser). Flow field measurements are carried out at four flow coefficients using a pre calibrated three hole probe. From the probe measurements, static pressure, total pressure, flow angle and flow velocity are calculated. The measurements are taken at twelve axial locations, at six radial locations and at seven circumferential locations at each radius. At each flow coefficient, the steady flow measurements are taken for twelve axial planes from shroud to hub. At each location, probe pressures are noted. Also position of the probe from the protractor reading is recorded. The velocities obtained from steady state probe measurements are nondimensionalised by rotor tip speed. The pressures are nondimensionalised with respect to the dynamic head based on the impeller tip speed.

Results and Discussion

The performance characteristics are shown in Fig.4. Variation of energy coefficient with flow coefficient is shown in Fig.4a. A comparison of energy coefficients of various configurations is given in Table-3. The VLD shows lowest peak energy coefficient. The peak energy coefficient is 1.265 and occurs at a flow coefficient which is close to the design flow coefficient and the operating range is 0.713. Energy coefficient curves for all partial vane diffuser configurations are slightly above the VLD curve. The partial vane diffuser without change in inlet angle (PVDHS) is having a maximum peak energy coefficient of 1.31 at a flow coefficient of 0.36. The maximum flow coefficient for PVDHs is 0.986, and operating range for this configuration is 0.626. The maximum flow coefficient for all PVDs is slightly lower than that of VLD due to the blockage effect of PVDs. The blockage changes slightly with the PVD inlet angle. Hence the maximum flow coefficient changes for PVDs with different inlet angles.

When the inlet angle of partial vanes on the shroud side is increased by five degrees (PVDHS5) it gives a better energy coefficient than VLD. Its peak energy coefficient is 1.313 and it is 3.7% more than the peak energy coefficient of VLD. Flow coefficient at peak energy coefficient (ϕ at ψ_{max}) is 0.281 which is less than ϕ at ψ_{max} of VLD. This gives more operating range towards the left of design flow coefficient for PVDHS5. The maximum flow coefficient for this arrangement is 1.034 which is close to that

Table-3 : Comparison of Performance of Different Diffuser Configurations

Configuration	ψ_{max}	ϕ_{max}	ψ_d	$\Delta\psi_{max}$	ϕ at ψ_{max}	ψ at ϕ_{max}	$\Delta\phi_{max}$	ϕ_{op}
VLD	1.265	1.036	1.275	0.000	0.323	0.426	0.000	0.713
PVDHS	1.310	0.986	1.306	0.045	0.360	0.479	0.048	0.626
PVDHS5	1.313	1.034	1.312	0.048	0.281	0.427	0.002	0.753
PVDHS-5	1.318	1.028	1.317	0.053	0.350	0.460	0.008	0.678
PVDH5S	1.315	0.995	1.312	0.050	0.350	0.472	0.040	0.645
PVDH5S	1.318	0.998	1.317	0.053	0.336	0.484	0.037	0.662
PVDH5S5	1.318	1.000	1.317	0.053	0.360	0.470	0.035	0,640
PVDHS10	1.298	1.000	1.290	0.033	0.366	0.442	0.035	0.634

Table-4 : Comparison of Maximum Efficiencies of Different Diffuser Configurations

Configuration	Maximum Efficiencies (η_{max})	Increase in η_{max} Compared to VLD (in %)
VLD	0.670	0.0
PVDHS	0.716	6.9
PVDHS5	0.703	4.9
PVDHS-5	0.715	6.7
PVDH5S	0.714	6.6
PVDH-5S	0.684	2.1
PVDH5S5	0.730	8.9
PVDHS10	0.722	7.8

of VLD, hence this configuration is having a good operating range (ϕ_{op}=0.753). The peak energy coefficient for all PVDs is higher than that of VLD due to the pressure recovery of the vanes. The pressure recovery changes slightly with the PVD inlet angle. Hence the peak energy coefficient is slightly different for PVDs with different inlet angles. It can be emphasized that performance of PVD falls in between VLD and VD as explained in Sitaram et al. [10 and 11].

The peak energy coefficient for PVDHS-5, PVDH-5S and PVDH5S5 are 1.318 whereas peak energy coefficient for PVDH5S is 1.315. The operating range for PVDHS-5 is 0.678 and for PVDH-5S is 0.662. Operating range for PVDH5S and PVDH5S5 is 0.645 and 0.64 respectively.

All partial vane configurations show almost similar peak energy coefficient, but the flow coefficient at which this peak energy coefficient occurs is slightly higher in configurations other than PVDHS5. Also their maximum flow coefficient is slightly less than that of the PVDHS5. This limits their operating range to a value less than that of PVDHS5. Hence for further flow field analysis this configuration is selected. As the experiments are conducted at a constant speed of 3000 RPM the power coefficient remain same for all diffuser vane configurations.

Variation of efficiency with flow coefficient is shown in Fig.4b. Comparison of maximum efficiency is presented in Table- 4. The efficiency is calculated as follows:

Efficiency of the Compressor, $\eta = \rho\ V\ W / \eta_m\ N_m$

Where

W : Specific work $(m^2/s^2) = (P_D-P_S)/\rho+(C_D^2-C_S^2)/2+g\Delta Z$

and includes energy change due to pressure, kinetic and potential energy.

N_m : Motor input power obtained from measured input voltage and current and power factor given by the manufacturer

η_m : Motor efficiency given by the manufacturer

The calculated efficiency includes both hydraulic and mechanical losses. The mechanical losses can be assumed to remain the same with PVD and VLD as the speed remains constant. Although the efficiency includes both mechanical and hydraulic losses, same trend can be expected for the hydraulic efficiency. No attempt is made to calculate mechanical losses and determine hydraulic efficiency as the accuracy of motor input power and output power is limited. However the trends can be expected as shown in the figure.

Efficiency of partial vane diffuser is greater than that of vaneless diffuser (VLD). The maximum efficiency occurred for partial vane diffuser with inlet angle varied by five degree at both hub and shroud (PVDH5S5). The efficiency of PVDH5S5 is 8.9% higher than that of VLD. It occurs at slightly lower flow coefficient than that of VLD. At higher flow coefficient, efficiencies of all the configurations are more or less similar. Maximum efficiency of PVDHS is 6.9% higher than that of vaneless diffuser. The efficiency of PVDHS5, PVDHS-5 and PVDH5S is almost similar to that of PVDHS; however PVDHS10 shows an increase in efficiency of 7.8%.

Static pressure coefficient distributions of PVDHS5 on diffuser shroud are shown and are compared with PVDHS in Fig.5. The comparison has done at two different flow coefficients below and above design point (ϕ=0.23, close to surge and ϕ=0.60, close to the maximum volume flow for compressor with vane diffuser respectively). The data for PVDHS are obtained from Sitaram et al. [8]. This configuration is extensively tested and flow field measurements in the passage of PVDHS were presented in Sitaram et al. [10 and 11]. Unlike the conventional vane diffuser, there is no semi-vaneless region in PVDs. Hence most of the static pressure rise occurs in the initial vaneless space and within the diffuser passage. Outside the vane passage stall cells are observed at both off-design conditions. In a partial vane diffuser, it is not possible for the flow to mix effectively in the circumferential direction on its way to vane exit. As the effective flow area is large, the flow guidance is much lower compared to a conventional vane diffuser.

Axially averaged static pressure coefficient of PVDHS5 are compared with PVDHS [9] at two flow coefficients (i.e. ϕ=0.23 and ϕ=0.60) and are shown in Fig.6. Axial averaging of any flow parameter, q is done as follows,

$$\overline{q} = \int_0^b q c_m \, dy \Big/ \int_0^b c_m \, dy$$

A steady, circumferential uniform rise in static pressure is observed from inlet to the exit of the diffuser, except near the trailing edge. At ϕ=0.23, the incoming flow to the diffuser vane is more tangential and the flow will have a tendency to separate from the lower concave surface of the diffuser vane. This is visible from the distorted contours near the trailing edge of the diffuser vane. The contours are more crowded near the diffuser inlet portion indicating that the pressure recovery is high at the initial vaneless region of the diffuser.

At ϕ=0.60 the static pressure rise comes down. As mentioned earlier this operating point lies far away from the design point. It is quite intriguing to note that, at ϕ=0.60 most of the static pressure rise occurs inside vane passage aft the leading edge. The rate of increase of static pressure is found to be higher after the mid-chord region.

Distribution of axially averaged total pressure coefficient at flow coefficients ϕ=0.23 and ϕ=0.60 is shown in Fig.7. The distribution is found to be quite dense after the vane trailing edge at ϕ=0.23, giving indications that the losses are higher in that region. At higher flow coefficient, i.e. at ϕ=0.60, much of the constant total pressure lines are radially outwards which indicates that total pressure varies in circumferential direction of diffuser flow passage. Apart from this the loss regions are clearly seen from the islands in the distribution of total pressure coefficient. The distribution for PVDHS5 and PVDHS are almost same. The only difference lies in the number of curves used for interpolation.

The variation of mass averaged flow parameters from diffuser inlet to outlet are shown in Fig.8. Mass averaged value of any flow parameter, q is defined as follows:

$$\overline{\overline{q}} = \int_0^b \int_0^s q \, c_m \, dy \, dx \Big/ \int_0^b \int_0^s c_m \, dy \, dx$$

Along the radial direction the drop in the total pressure mass averaged total pressure coefficient will give the amount of losses in the diffuser passage. It is interesting to note that the total pressure at the inlet of diffuser is varied with the diffuser vane inlet angle.

It suggests that the back pressure from the diffuser may have some impact on the mass averaged value of total pressure at the diffuser inlet. However this aspect is need to be explored further. Static pressure coefficient is higher for PVDHS5 compared to PVDHS at all flow coefficients. The variation in the mass averaged values of static pressure and velocity are complimentary to each other. The increase in the static pressure recovery for PVDHS5 is quite evident from the plots. The flow angle does not show much variation in diffuser passage.

Conclusions

The following major conclusions are drawn from the present experimental investigation on the effect of vane angle of partial vane diffuser on the performance and flow field of a centrifugal compressor:

- Performance tests show that all configurations of partial vane diffusers have a better energy coefficient and efficiency compared to that of vaneless diffuser (VLD). VLD shows the lowest peak energy coefficient. However, it has a wider operating range than partial vane diffusers.
- The energy coefficients for all partial vane diffuser configurations are almost similar, but the flow coefficient at which peak energy coefficient occurs is slightly higher in configuration other than partial vane diffuser with inlet angle increased by five degree at shroud (PVDHS5). Also their maximum flow coefficient is slightly less than that of PVDHS5. This indicates operating range of PVDHS5 is higher than that of other configurations. PVDHS5 has a peak energy coefficient which is three percent more than that of VLD. Maximum efficiency of PVDHS5 is nearly seven percent higher than that of VLD.
- The performance of compressor with PVDHS5 is not improved up to the level of expectation, because the compressor used for experimental analysis is a low specific speed compressor. It is expected that this configuration will give much better performance in high specific speed compressor.
- Distribution of static pressure coefficient shows that flow is nearly uniform in the circumferential direction. Static pressure distribution for a flow coefficient of 0.23 shows a slightly higher value than other flow coefficients. This is because near the surge point energy coefficient is high.
- At ϕ=0.23 much of the static pressure occurs in the vaneless space before leading edge but at ϕ=0.60 most of the static pressure rise occurs inside vane passage aft the leading edge.
- Radial variation of mass averaged flow parameters shows that static pressure increases with radius and total pressure decreases with radius.

The present investigation has shown that variable inlet angle of partial vane is an additional technique to improve the performance of centrifugal compressor (particularly turbocharger compressors for low emission Diesel engines, where good off-design efficiency and broad operating range are major requirements) in addition to other techniques such as variable inlet guide vanes and casing treatment. Further variable height vane diffusers in partial vane diffuser configuration (with vanes alternatively placed on the diffuser hub and shroud) [12] can be used as another technique to improve the performance of centrifugal compressor.

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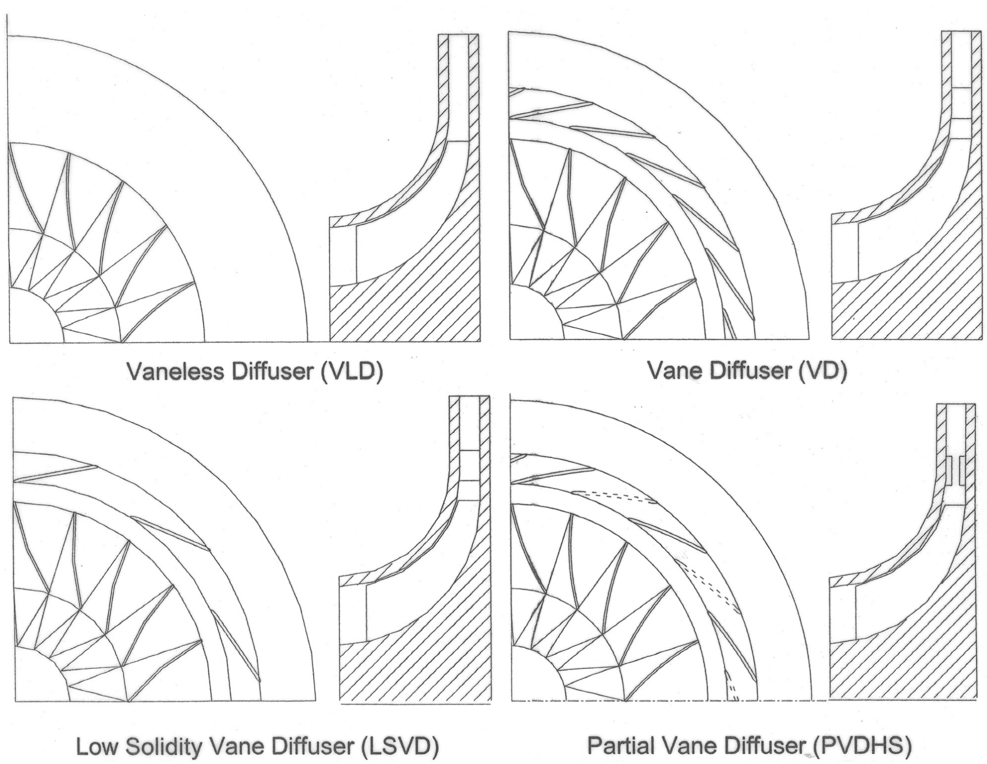


Fig.1 Schematic Views of Centrifugal Compressor with Different Types of Diffuser [Ref.8]

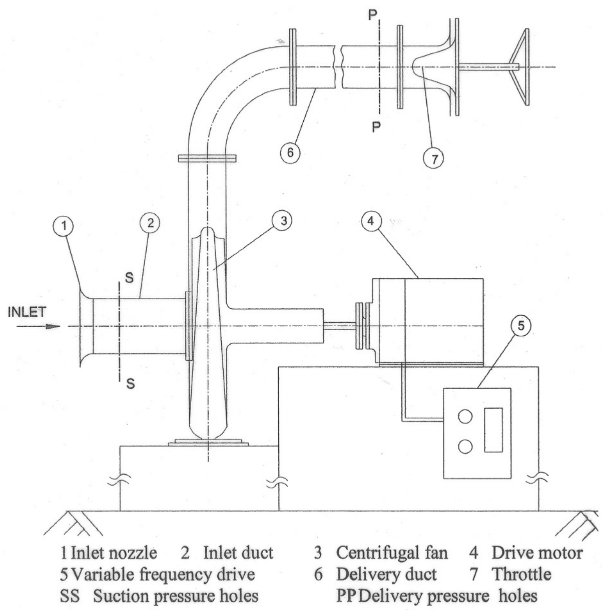


Fig.2a Schematic Layout of the Experimental Setup [Ref.8]

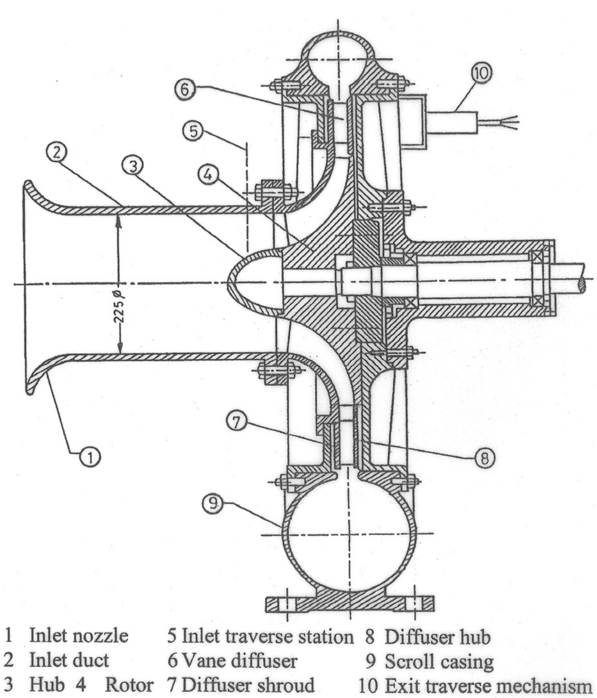


Fig.2b Meridional View of the Compressor Showing Rotor and Vane Diffuser [Ref.8]

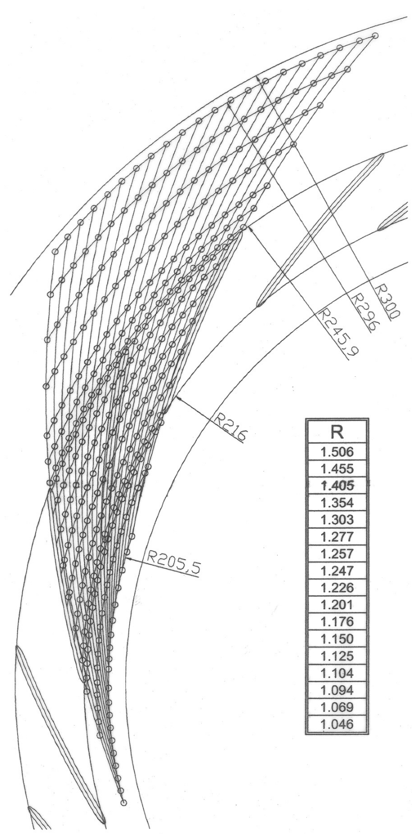


Fig.3a Location of Static Pressure Holes on Diffuser Hub and Shroud for Two Vane Diffuser Passages (or One Partial Vane Passage)

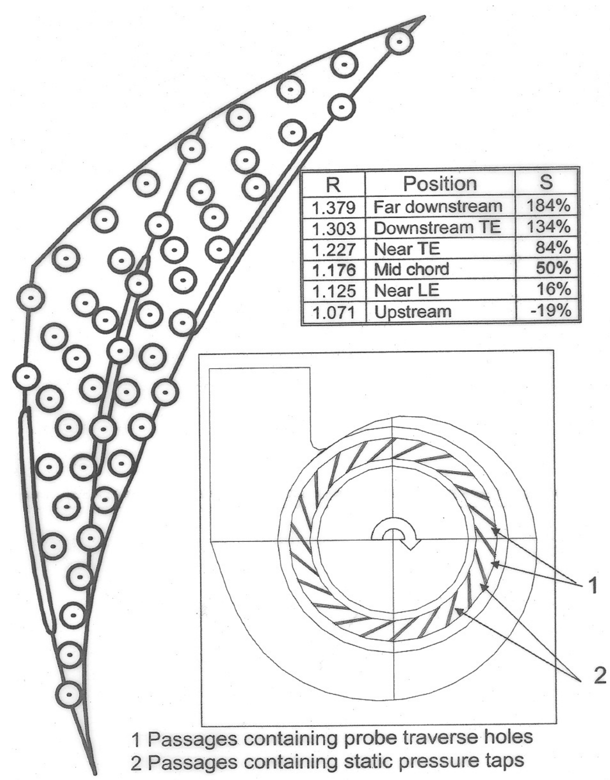


Fig.3b Location of Probe Traverse Hole on Diffuser Hub for Two Vane Diffuser Passages (or One Partial Vane Passage)

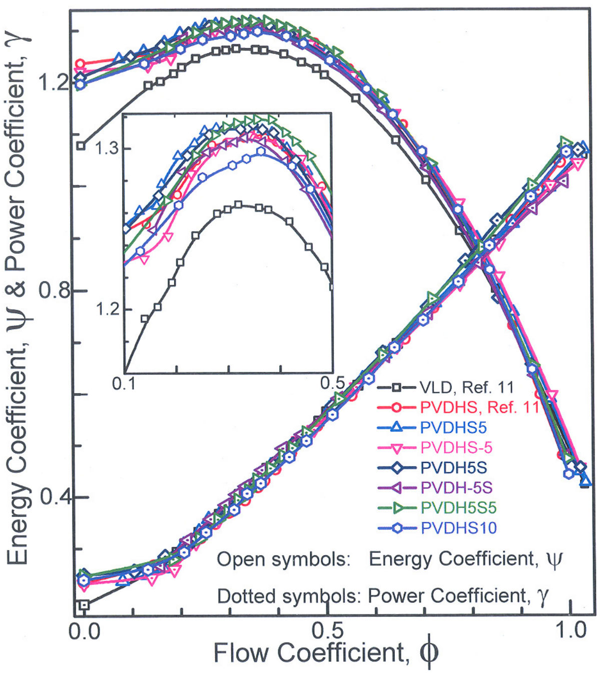


Fig.4a Variation of Energy and Power Coefficients with Flow Coefficient

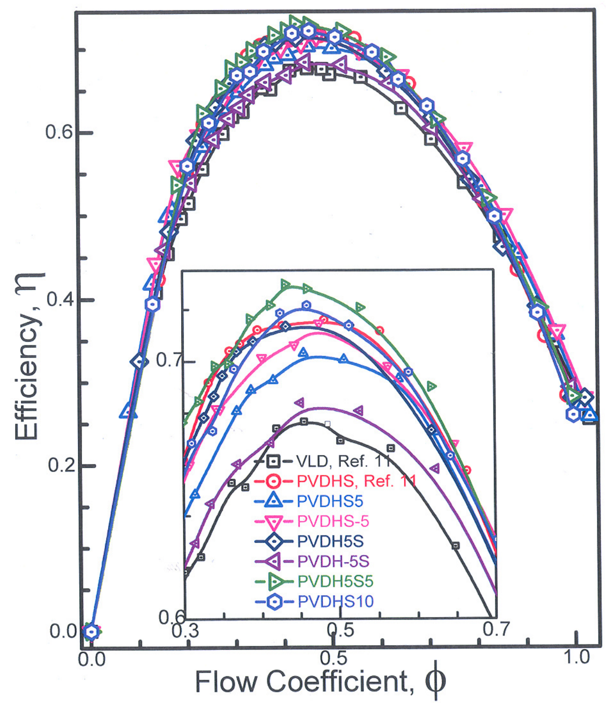


Fig.4b Variation of Efficiency Power Coefficient with Flow Coefficient

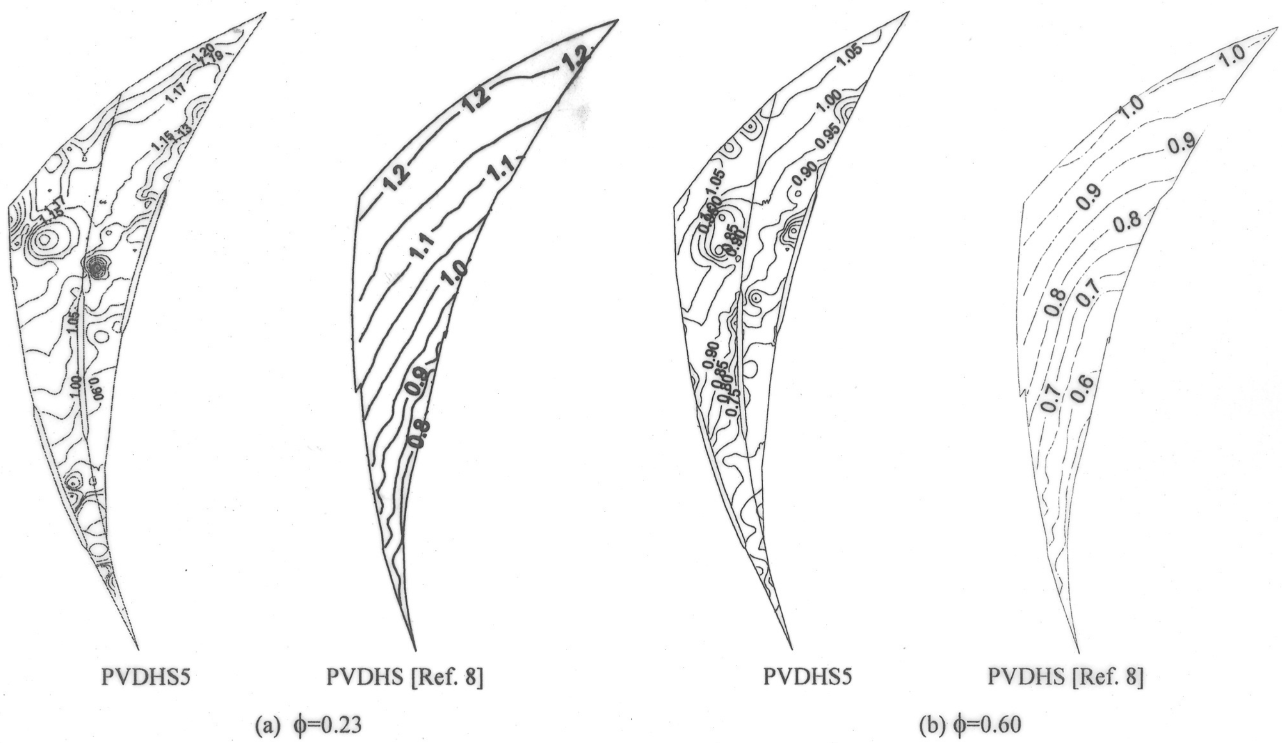
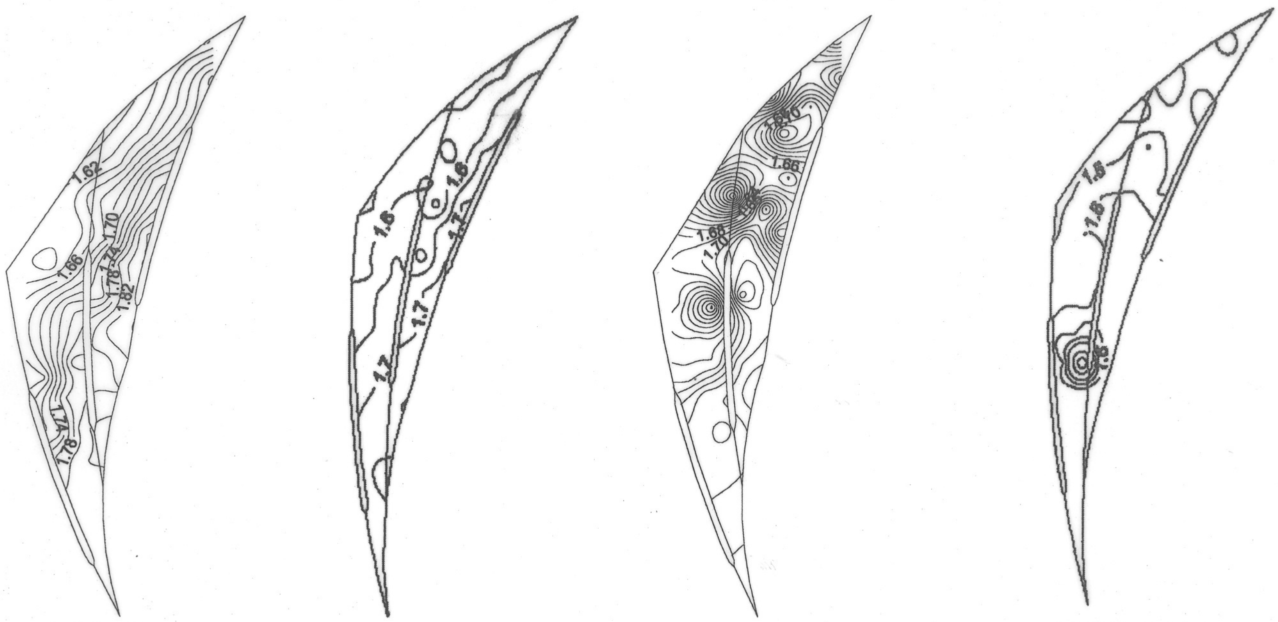


Fig.5 Contours of Static Pressure Coefficient on the Diffuser Shroud

PVDHS5 PVDHS [Ref. 11] PVDHS5 PVDHS [Ref. 11]

(a) ϕ=0.23 (b) ϕ=0.60

Fig.6 Contours of Axially Averaged Static Pressure Coefficient in the Diffuser Passage



PVDHS5 PVDHS [Ref. 11] PVDHS5 PVDHS [Ref. 11]

(a) ϕ=0.23 (b) ϕ=0.60

Fig.7 Contours of Axially Averaged Total Pressure Coefficient in the Diffuser Passage

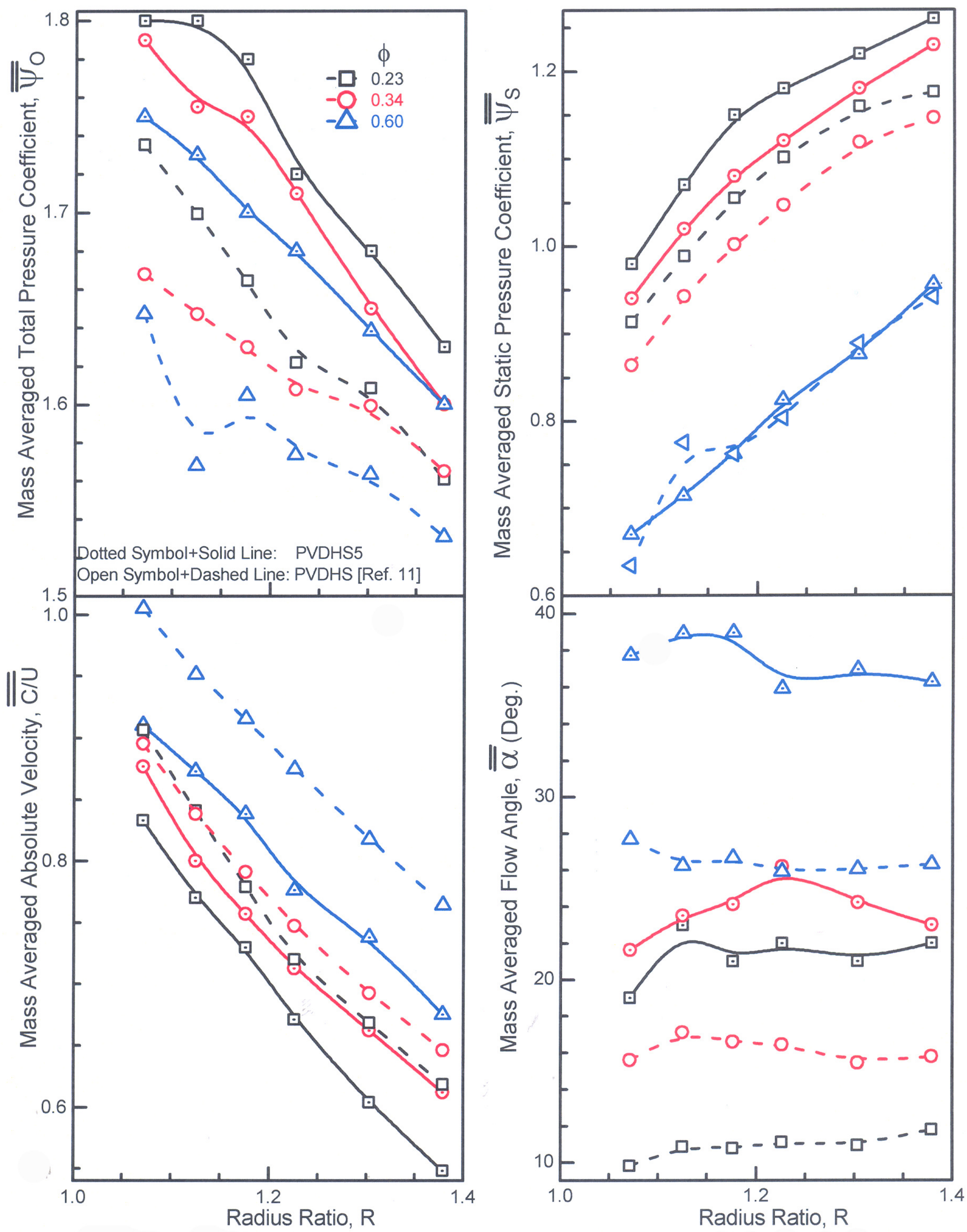


Fig.8 Variation of Mass Averaged Flow Parameters with Radius Ratio