# Effect of diffuser vane height and position on the performance of a centrifugal compressor

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**Abstract:** The present paper reports experimental investigations on the effect of diffuser vane height and position on the performance of a low-speed centrifugal compressor. The diffuser vane height is systematically varied from 0.2 to 0.9 times the diffuser width. In addition, the effect of vane position is examined by fixing the partial vanes to the hub, shroud, or hub and shroud. The compressor performance is determined with these vanes in the vane and low solidity vane diffuser configurations. It is found that there is an optimum height for the diffuser vane height. In the present investigation, it is found to be 0.3 times the diffuser width. The effect of position of the partial vanes on the hub or shroud on the compressor performance is found to be negligible. However, when partial vanes are fixed on the hub and shroud staggered at half spacing, the compressor performance is improved substantially.

**Keywords:** centrifugal compressor, vaneless diffuser, vane diffuser, low solidity vane diffuser, partial vane diffuser, vane height, vane position, performance

## NOTATION

| | |
|---|---|
| $b$ | diffuser width (m) |
| $c_d$ | velocity in delivery duct (m/s) |
| $c_s$ | velocity in suction duct (m/s) |
| $h$ | diffuser vane height (m) |
| $p_d$ | static pressure on delivery duct (N/m$^2$) |
| $p_s$ | static pressure on suction duct (N/m$^2$) |
| $r_2$ | impeller tip radius (m) |
| $R$ | radius ratio $= r/r_2$ |
| $U_2$ | impeller tip speed (m/s) |
| $V$ | volume flow (m$^3$/s) |
| $W$ | specific work (m$^2$/s$^2$) $= (p_d - p_s)/\rho + (c_d^2 - c_s^2)/2$ |
| $\alpha$ | vane angle (°) |
| $\Delta\phi_{max}$ | $\phi_{max\,VLD} - \phi_{max}$ |
| $\Delta\psi_{peak}$ | $(\psi_{peak} - \psi_{peak\,VLD})/\psi_{peak\,VLD}$ |
| $\phi$ | flow coefficient $= V/\pi D_2 b_2 U_2$ |
| $\phi_{max}$ | maximum flow coefficient |
| $\phi_{op}$ | operating range $= (\phi_{max} - \phi$ at $\psi_{max})$ |
| $\eta$ | efficiency |
| $\rho$ | density (kg/m$^3$) |
| $\sigma$ | solidity = chord/pitch |
| $\psi$ | energy coefficient $= 2W/U_2^2$ |
| $\psi_d$ | energy coefficient at design flow coefficient, $\phi_d = 0.34$ |
| $\psi_{peak}$ | peak energy coefficient |

### Abbreviations

| | |
|---|---|
| LSVD | low solidity vane diffuser |
| PVD | partial vane diffuser |
| VD | vane diffuser |
| VLD | vaneless diffuser |

## 1 INTRODUCTION

In a centrifugal compressor, flow leaves the impeller with high velocity and inclined at a large angle to the radial direction. The role of the diffuser is to decelerate flow while it is passing through a divergent passage. Kinetic energy of flow is thereby transformed to pressure energy. Centrifugal compressor diffusers can be broadly classified into two types: (1) vane diffuser (VD) and (2) vaneless diffuser (VLD). In a centrifugal compressor, it is well established that the conventional vane diffusers exhibit a higher performance (i.e., efficiency and static pressure rise vs. mass flow) than the vaneless diffuser, but with the compromise of reduced operating range. The factor favouring the vaneless diffuser is that of low cost. In addition, it can

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accept a wider range of inlet flow variations without a severe performance impact. The use of conventional vane diffusers in process applications carries a greater risk with respect to performance. Senoo [**1**] reported a new type of diffuser vane called low solidity vane diffuser (LSVD). The major advantage of the low solidity vane diffuser is that it does not have a throat between the vanes. Hence, the diffuser passage is not choked. The low solidity vane diffusers provide a higher performance than the vaneless diffusers and a larger flow range than the vane diffusers. Yoshinaga *et al.* [**2**] reported improved performance of a centrifugal compressor when diffuser vanes with height less than the passage width were fixed to the shroud. This diffuser is named the partial vane diffuser (PVD). However, no systematic investigations on the comparative merits of these diffusers are reported in the literature. Hence the present investigation has been undertaken, where the height of the partial vane is systematically varied. In addition, the effect of partial vane position, namely on the hub, shroud, and hub and shroud, on compressor performance is investigated.

## 2 EXPERIMENTAL FACILITY, INSTRUMENTATION AND PROGRAMME

### 2.1 Experimental facility

A low-speed single-stage centrifugal compressor is used for the present experimental investigations. The details of the compressor along with a meridional view are given in reference [**3**]. The design details of vanes for the vane and low solidity vane diffusers are also given in the same reference. For the partial vane diffusers, the same vanes are used except the vane height is reduced.

### 2.2 Instrumentation

The compressor performance with different diffusers is determined from average wall static pressures from the inlet duct and exit duct. All the pressure tapings are connected to a scanning box (FCO 91-3) and measured with a micro manometer (FCO12 Mode14, range $\pm 1999$ mm of WC, accuracy $\pm 0.1$ per cent of full-scale reading). Both are manufactured by Furness Control Ltd., Bexhill, UK.

### 2.3 Experimental programme

The compressor is tested with a vane diffuser having partial vanes of *h*/*b* ratio of 0.2, 0.3, 0.4, 0.5, 0.7, and 0.9. The partial vanes are fixed to the hub or shroud. The compressor is also tested with low solidity vane diffusers. In addition, a low solidity vane diffuser of 11 partial vanes on the hub and shroud is also tested. The vanes on the hub and shroud are staggered at one-half of the vane spacing. Some of the tested diffuser configurations are shown in Fig. 1 and the details are given in Table 1. The abbreviation used to identify the partial vane is given at the end of Table 1. All the performance measurements are carried out with the

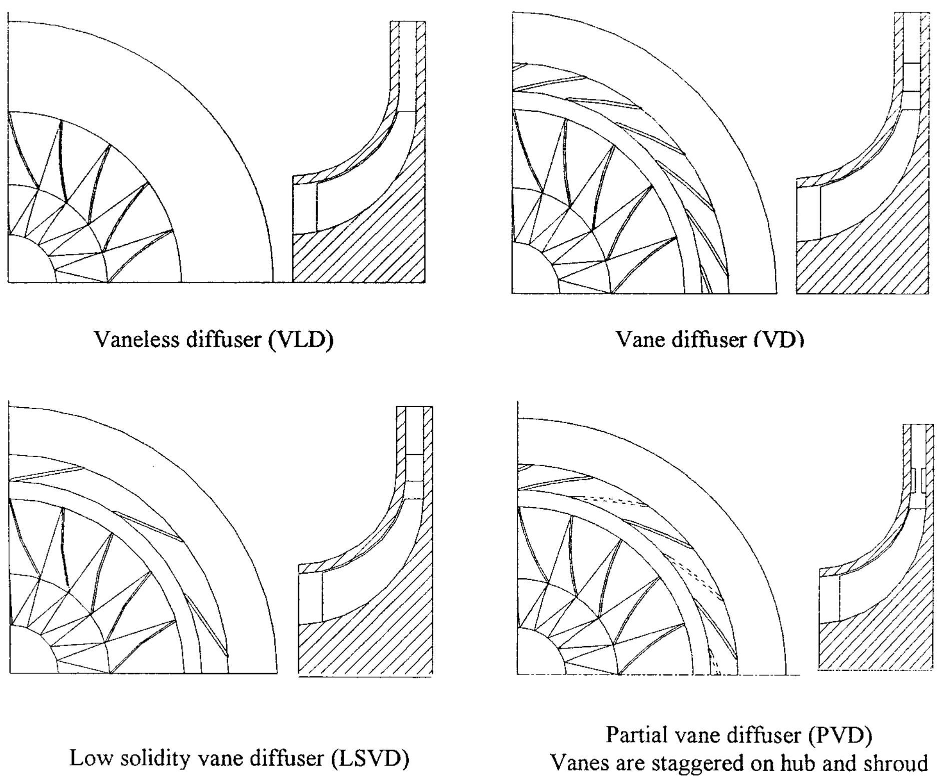


**Fig. 1** Schematic view of centrifugal compressor with vaneless, vane, low solidity vane, and partial vane diffusers

 

**Table 1** Geometric details of diffusers tested

| Diffuser type | Vaneless diffuser | Vane diffuser | Law solidity vane diffuser |
|---|---|---|---|
| Solidity, $\sigma$ | – | 1.4 | 0.7 |
| No. of vanes | – | 22 | 11 |
| $R_3$ | 1.0 | 1.1 | 1.1 |
| $R_4$ | 1.5267 | 1.2514 | 1.2514 |
| Chord, Ch (mm) | – | 86.07 | 86.07 |
| $\alpha_3$ | – | 15 | 15 |
| $\alpha_4$ | – | 25 | 25 |

$R_3$ and $R_4$: vane inlet and exit radius ratios; $\alpha_3$ and $\alpha_4$: vane inlet and exit angles in degrees with respect to tangential direction. 11PVD3HS: First two numbers give number of vanes (11 for LSVD configuration and 22 for VD configuration); PVD means partial vane diffuser; next number gives ten times ratio of vane height/diffuser width; next letter(s) gives position of vanes; (H – vanes fixed to hub, S – vanes fixed to shroud, and HS – vanes fixed to both hub and shroud).

compressor running at a speed of 3000 $\pm$ 1 rpm. Because of the large amount of data, only typical performance characteristics are presented below for the sake of brevity.

## 3 RESULTS AND DISCUSSION

### 3.1 Effect of diffuser vane height

The variation of energy coefficient vs. flow coefficient characteristic showing the effect of vane height is shown in Figs 2 to 4. The performance of the partial vane diffusers is compared with that of the vaneless (VLD), vane (VD), and low solidity vane (LSVD) diffusers. From Fig. 2 (corresponding to the diffuser of 22 partial vanes of heights 0.2 to 0.9 fixed to the hub), it is observed that the maximum flow coefficient increases continuously from 0.62 to 0.94, as the partial vane height is decreased. Also, the operating range increases.

When the vane height is decreased, the peak energy coefficient occurs at a higher flow coefficient. However, the peak energy coefficient increases as the vane height is decreased from $h/b = 0.9$ to 0.5. Further decrease in the vane height results in a decrease of the energy coefficient. A similar trend is observed for the diffuser with 11 partial vanes fixed to the hub (not shown here). However, the minimum value of peak energy coefficient occurs for a vane height of 0.4 rather than 0.5. Similar observations are made for other cases, that is when 22 partial vanes are fixed to the shroud, 11 partial vanes are fixed to the shroud (Fig. 3), and 11 partial vanes are fixed to both hubs and shroud but staggered at half spacing (Fig. 4). It may be concluded that the partial vanes of $h/b = 0.2$ to 0.3 are beneficial in increasing the maximum flow coefficient and operating range compared to the vane diffuser or low solidity vane diffuser. Although the peak energy coefficient of this partial vane diffuser is lower than that of the vane diffuser, it is substantially higher than that of the vaneless diffuser.

The variation of efficiency with flow coefficient, showing the effect of the vane height, is shown in Figs 5–7.

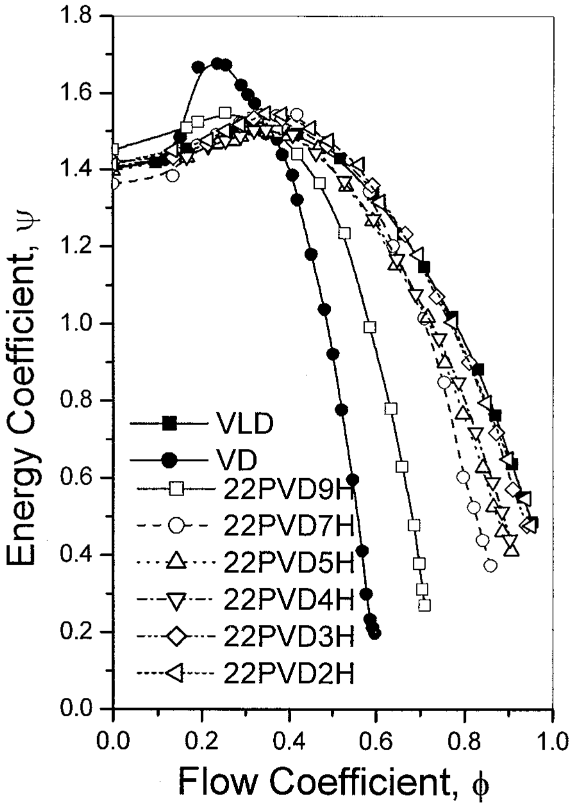


**Fig. 2** Effect of vane height on energy coefficient of the compressor (vane diffuser configuration with partial vanes on hub)

Efficiency ($\eta$) is defined as

$$\eta = \frac{\rho V W}{N_C}$$

where $N_C$ is coupling power and is given by

$$N_C = \eta_M (V_A I_A + V_F I_F)$$

where $V_A$ and $V_F$ are armature and field voltages, $I_A$ and $I_F$ are armature and field currents, $\eta_M$ is appropriately assumed combined efficiency of the drive motor and gear motor, and $\eta_M$ is taken as 0.75 and assumed constant over the entire range of operation.

The efficiencies are compared with the efficiencies of the VLD, VD, and LSVD. The efficiency of the partial vane diffusers is higher when compared with that of the vane and vaneless diffusers. The peak efficiency point is shifted to a higher flow coefficient when compared with that of the vane diffuser. Diffuser 22PVD2H shows higher efficiency than the vaneless diffuser through all its operating range, but the vane diffuser shows higher efficiency in the lower flow range. As partial vane height increases, the peak efficiency and the flow range also decrease. In the higher flow range, diffuser 22PVD7S shows higher efficiency than the vane and vaneless diffusers in the flow coefficient

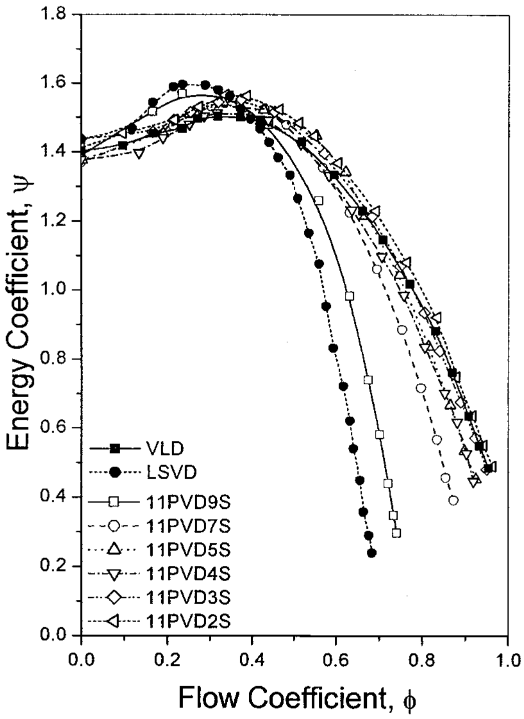


**Fig. 3** Effect of vane height on energy coefficient of the compressor (low solidity vane diffuser configuration with partial vanes on shroud)

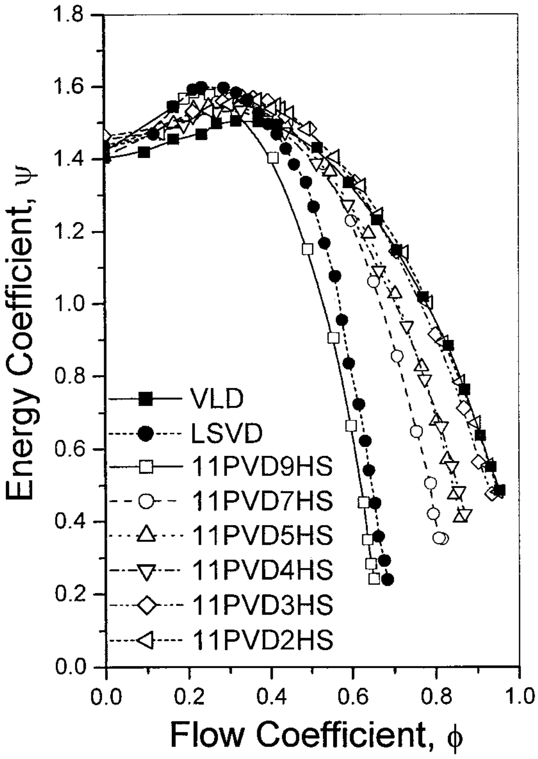


**Fig. 4** Effect of vane height on energy coefficient of the compressor (low solidity vane diffuser configuration with partial vanes on hub and shroud)

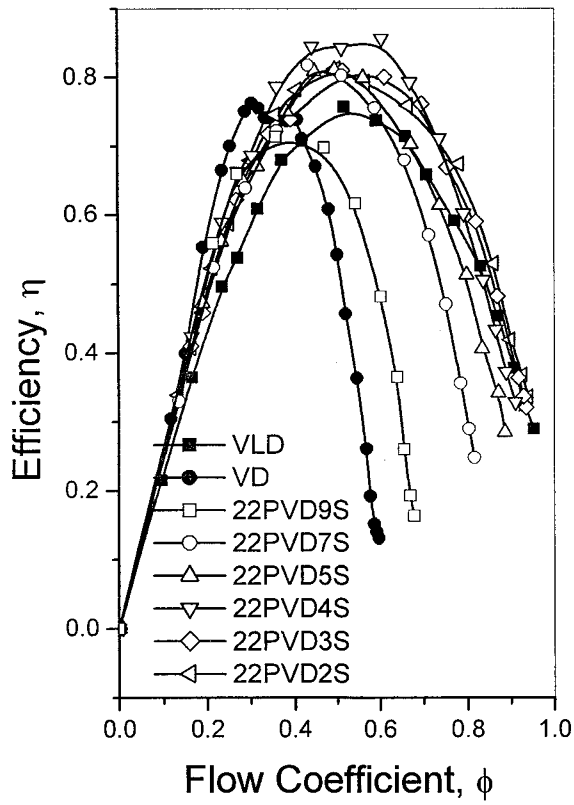


**Fig. 5** Effect of vane height on efficiency of the compressor (vane diffuser configuration with partial vanes on shroud)

range of 0.2 to 0.7 (Fig. 5). However, the peak efficiency of the partial vane diffuser is slightly less than that of the low solidity vane diffuser. In the lower flow range, the partial vane diffusers show lower efficiency compared with the low solidity vane diffuser. However, in the higher flow range, the drop in efficiency is slower for the partial vane diffusers. Similar observations are made for the partial vane diffusers in the low solidity vane diffuser configuration (Figs 6 and 7).

### 3.2 Effect of diffuser vane position

The effect of the vane position is compared in Figs 8 to 13 by carrying out performance tests with the partial vane diffuser fixed in three positions, such as on the hub, on the shroud, in both vane and low solidity vane diffuser configurations and by fixing 11 vanes on both the hub and shroud. These vanes on the hub and shroud are circumferentially staggered by one-half vane spacing. The energy coefficient of the partial vane diffusers at different positions is compared with that of the VLD, VD, and LSVD in Figs 8 to 10.

The effect of vane position (for a vane height of $h/b = 0.3$) on the energy coefficient of the compressor is shown in Fig. 8. From the figure, it is evident that the effect of vane position is very small for this vane height. However, for the vane height of $h/b = 0.5$ (Fig. 9), the

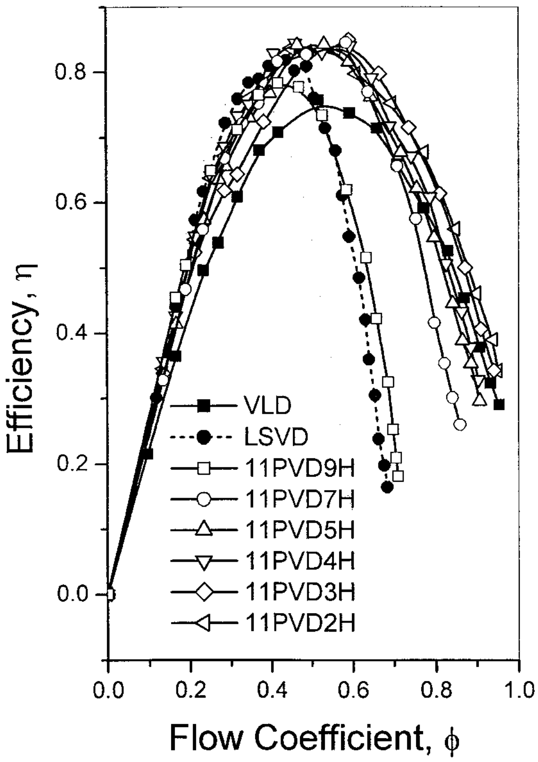


**Fig. 6** Effect of vane height on efficiency of the compressor (low solidity vane diffuser configuration with partial vanes on hub)

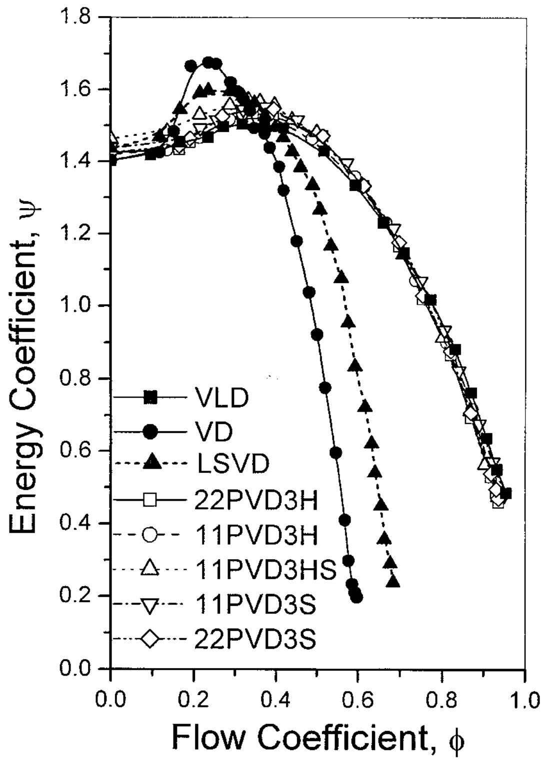


**Fig. 8** Effect of vane position on energy coefficient of the compressor (vane height $h/b = 0.3$)

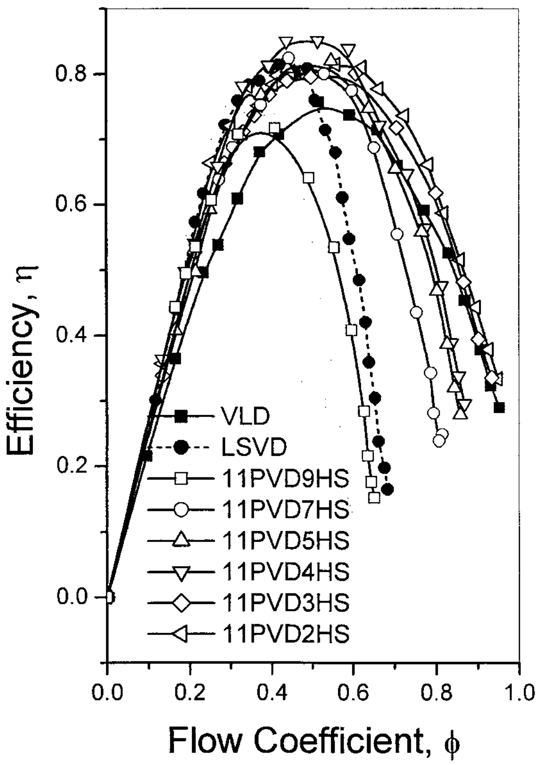


**Fig. 7** Effect of vane position on energy coefficient of the compressor (low solidity vane diffuser configuration with partial vanes on hub and shroud)

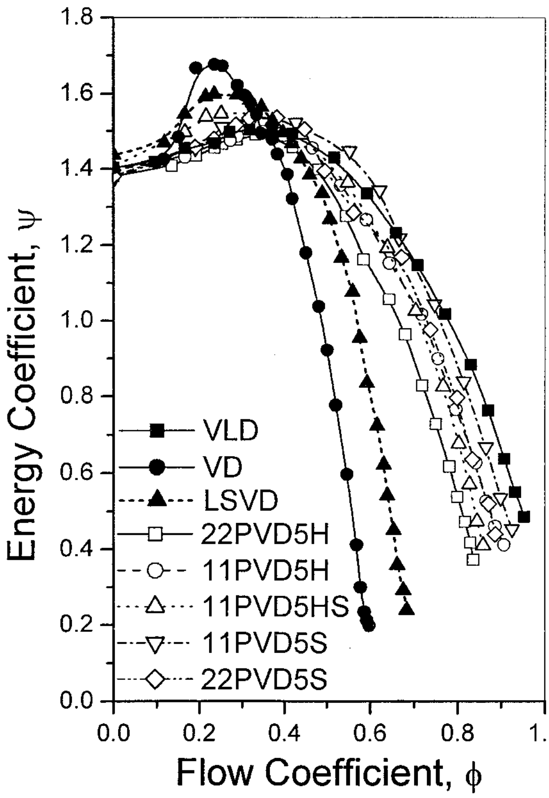


**Fig. 9** Effect of vane position on energy coefficient of the compressor (vane height $h/b = 0.5$)

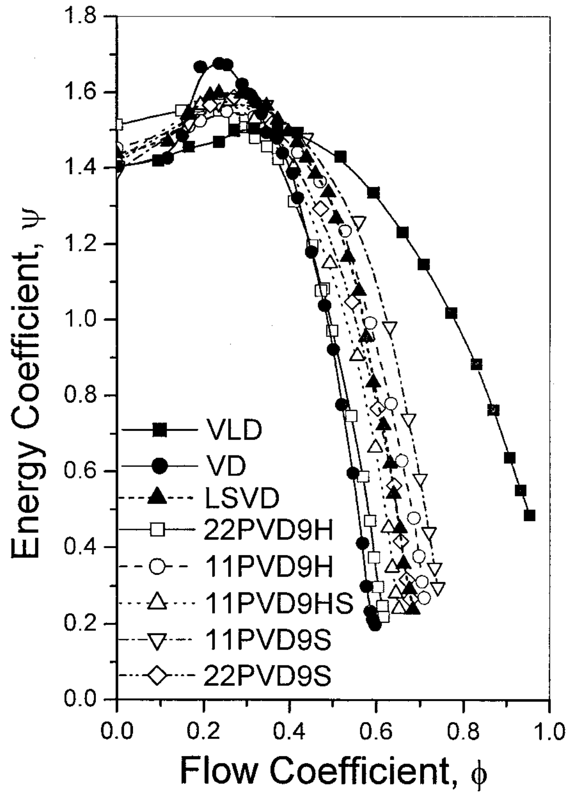


**Fig. 10** Effect of vane position on energy coefficient of the compressor (vane height $h/b = 0.9$)

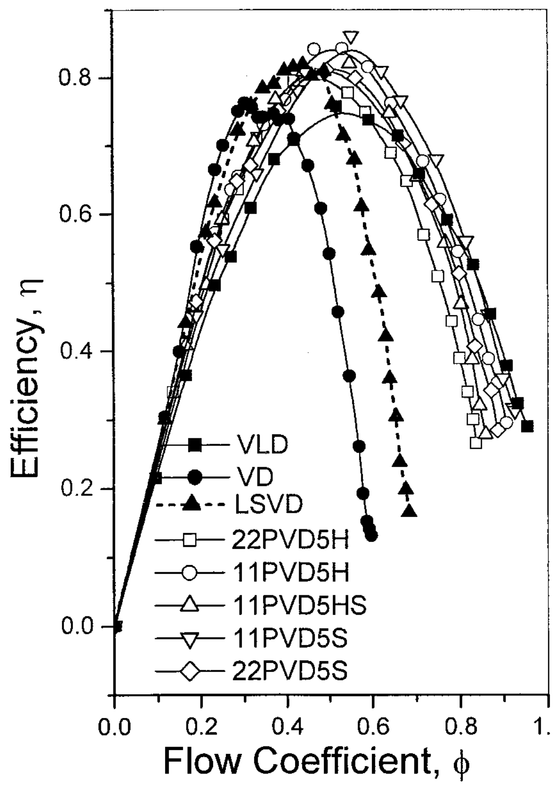


**Fig. 12** Effect of vane position on efficiency of the compressor (vane height $h/b = 0.5$)

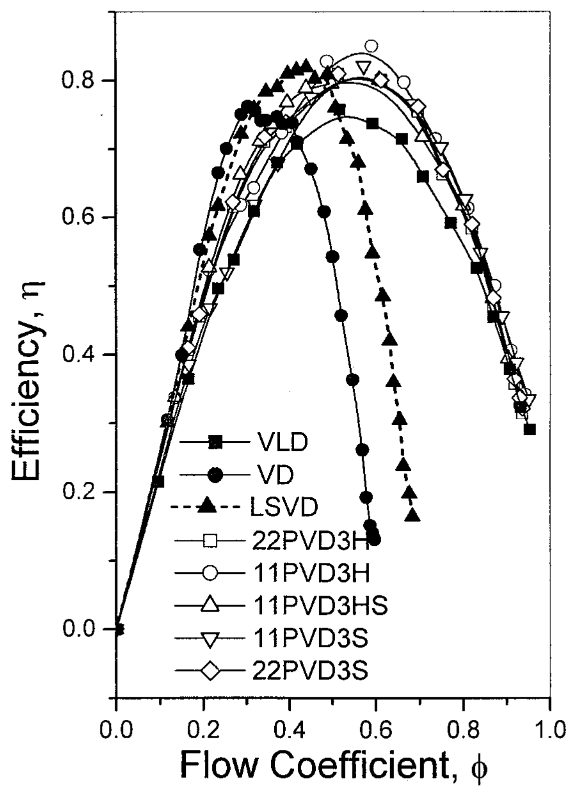


**Fig. 11** Effect of vane position on efficiency of the compressor (vane height $h/b = 0.3$)

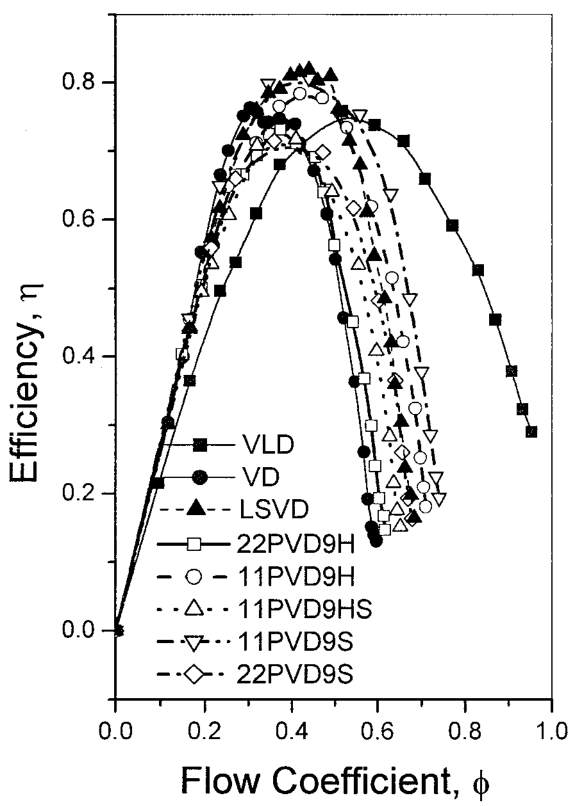


**Fig. 13** Effect of vane position on efficiency of the compressor (vane height $h/b = 0.9$)

 

effect of vane position on the energy coefficient of the compressor is substantial. The flow at the exit of the centrifugal impeller or at the inlet of the diffuser is highly complex. The flow exhibits jet wake pattern on the suction surface shroud corner. Tip leakage flow influences the flow in the shroud region, whereas strong secondary flows influence the flow in the hub region. Hence the flow leaves the impeller with large variations both in the axial and circumferential directions. However, the performance variations in the compressor fitted with partial vanes on the hub or the shroud are small. The most probable reason may be that increased spanwise mixing of the flow occurs when the partial vane diffuser is used. Another reason may be that the compressor tested in the present investigation has

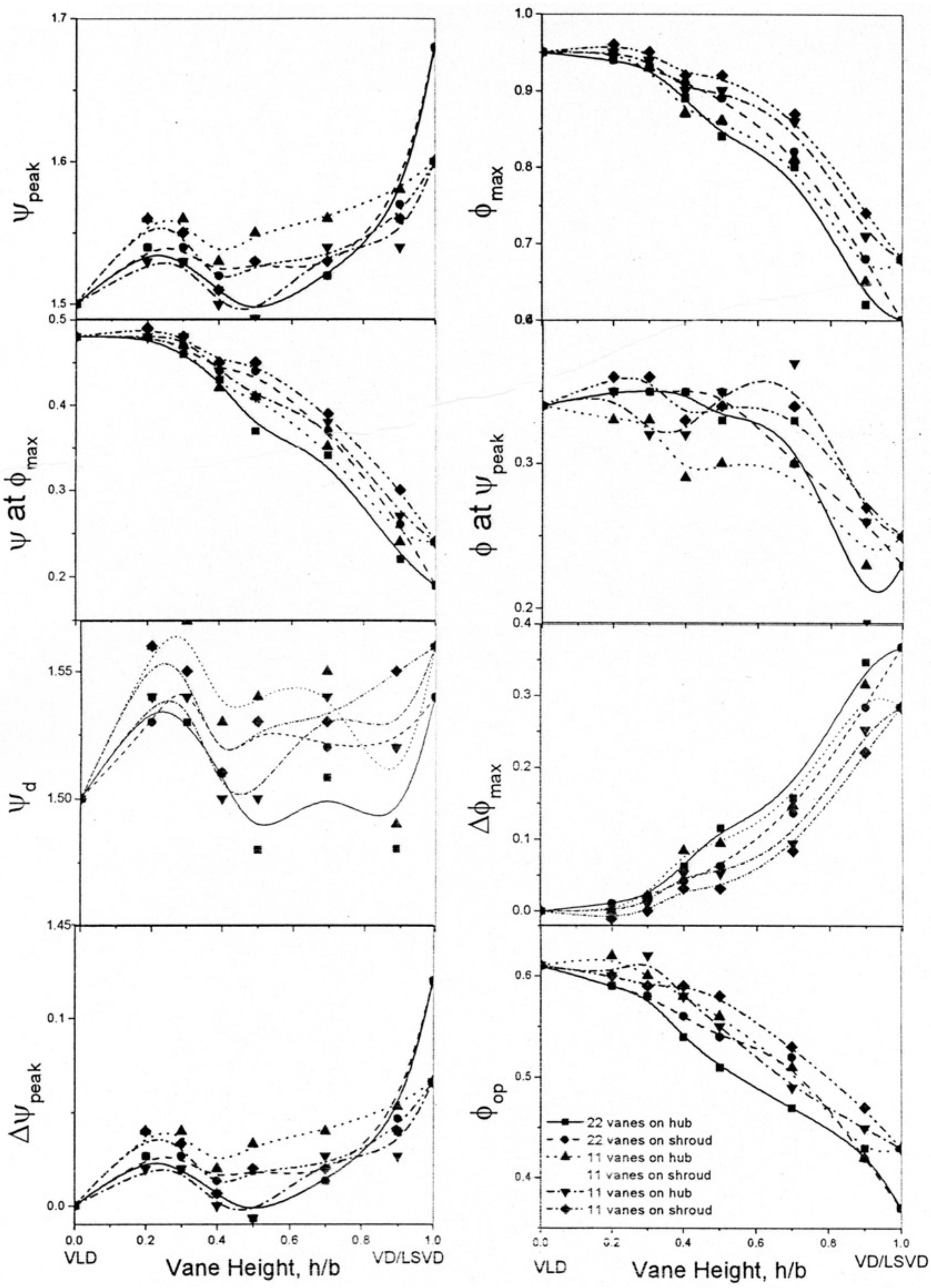


**Fig. 14** Variation of flow parameters with vane height and vane position

low specific speed, so the impeller exit flow is more uniform compared to the flow at the exit of a high specific speed centrifugal compressor. Yoshinaga *et al.* [**2**] had reported substantial improvement in the performance of a high specific speed centrifugal compressor when partial vanes are fixed to the shroud. Further measurements such as static pressure measurements on the diffuser hub and shroud and flow traverses in the diffuser passage should reveal the reasons for the small difference in the performance of the compressor with PVDs with vanes on the hub or on the shroud.

The maximum flow coefficient decreases in the following order: VLD, 11PVD5S, 11PVD5H, 22PVD5S, 11PVD5HS, 22PVD5H, LSVD, and VD. However, the peak energy coefficient of the compressor with 11PVD5HS is higher compared with the VLD and other PVDs. As the partial vane height is further increased to $h/b = 0.9$, the maximum flow coefficient is reduced (Fig. 10). However, the peak energy coefficient is increased with the peak energy coefficient, occurring at lower flow coefficient, compared to that of the VLD and LSVD. However, for the partial vane diffuser (11PVD3HS), the peak energy coefficient is visibly higher compared to that of other partial vane and vaneless diffusers.

The efficiency curves of the compressor with different diffusers (VLD, VD, LSVD, PVD on hub, shroud, and hub and shroud) are shown in Fig. 11 for a vane height of $h/b = 0.3$. The efficiency of the compressor with PVDs is higher compared to that of the compressor with VLD, VD, and LSVD. However, the peak efficiency occurs at a higher flow coefficient compared to VD and LSVD. The effect of vane position on the efficiency is very small. However, partial vanes fixed to the hub show slightly higher efficiency among all PVDs having the same $h/b$ (Figs 11–13).

### 3.3 Summary of results for all diffusers

Figure 14 shows variation of various performance parameters with vane height. The parameters are:

- $\psi_{peak}$: peak energy coefficient
- $\psi$ at $\phi_{max}$: energy coefficient at maximum flow coefficient
- $\psi_d$: energy coefficient at design flow coefficient
- $\Delta\psi_{peak}$: $(\psi_{peak} - \psi_{peak\,VLD})/\psi_{peak\,VLD}$
- $\phi_{max}$: maximum flow coefficient
- $\phi$ at $\psi_{peak}$: flow coefficient at which peak energy coefficient occurs
- $\Delta\phi_{max}$: $\phi_{max\,VLD} - \phi_{max}$
- $\phi_{op}$: operating range ($\phi_{max} - \phi$ at $\psi_{peak}$)

In the figure, $h/b = 0$ represents the vaneless diffuser (VLD) and $h/b = 1$ represents the vane (VD) or low solidity vane diffuser (LSVD). As the vane height increases, peak energy coefficient increases with a minimum around $h/b = 0.5$. Both maximum flow coefficient and flow coefficient at peak energy coefficient decrease as the vane height increases. Operating range decreases with decrease in vane height. At design flow coefficient, the partial vane diffusers in LSVD configuration show higher energy coefficient. A partial vane diffuser of $h/b = 0.5$ shows a lower peak energy coefficient than the vaneless diffuser. The partial vane diffuser with vanes fixed to both diffuser hub and shroud shows higher energy coefficient than partial vanes fixed at other positions. Both maximum flow coefficient and flow coefficient at peak energy coefficient decrease as $h/b$ decreases. For partial vanes on both hub and shroud, these have a minimum value at $h/b = 0.9$ rather than $h/b = 1$ in LSVD configurations. This is to be expected as this partial vane diffuser offers more blockage.

## 4 CONCLUSIONS

From the present experimental investigations on the effect of diffuser vane height and position on the performance of low-speed centrifugal compressor, the following conclusions are drawn. It is found that there is an optimum height for the diffuser vane height. In the present investigation, it is found to be 0.3 times the diffuser width. The effect of placement of the partial vanes on the hub or shroud on the compressor performance is found to be negligible. However, when partial vanes are fixed on the hub and shroud in staggered spacing, the compressor performance is improved substantially.

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