TECHNICAL NOTE

Measurements in the vaneless diffuser of a radial flow compressor

P. M. Ligrani*, R. Van Den Braembussche*, and M. Roustan†

Results from an experimental study of flow behaviour at the inlet of a vaneless diffuser of a centrifugal compressor are presented. Measurements from a crossed hot-wire probe are given for operating points having inlet flow coefficients ranging from 0.006 to 0.019 at different Reynolds numbers. Instantaneous, time-averaged, and phase-averaged absolute velocity and flow angle at the diffuser inlet are deduced from the hot-wire signals after correction for mean density variations. These results show how flow behaviour varies in stable, rotating stall and surge regimes of compressor operation

Key words: *compressors, diffusers, unsteady flow, rotating stall*

Flow instabilities in radial flow compressors, such as rotating stall and surge, result in reduced efficiencies and pressure ratios and possible vibrational excitation of the machine. These vibrations may limit the range of operation of a compressor, particularly in reinjection compressors used in chemical processing plants which operate at high pressure levels. The low volumetric flow rates and moderate rotational speeds typical of these types of compressors result in low specific speed impellers. The flow behaviour at the diffuser inlet of such a radial compressor was investigated in this study.

The earliest studies of flow in diffusers of radial flow compressors generally considered the high frequency unsteadiness at the diffuser inlet due to the jet/wake pattern of flow leaving the impeller[1–4]. Theoretical models have also been developed to relate rotating stall to local flow reversals on diffuser walls[5–7]. The agreement of these theoretical results with experimental data of Ref 8 allowed the development of a correlation relating the onset of diffuser rotating stall to non-dimensional diffuser inlet width, and inlet flow angle. Another recent study relates rotating stall onset to the unsteady interaction between flows in the impeller and diffuser[9]. Generally, however, these studies provide only a limited description of the unsteady flow in vaneless diffusers of radial flow compressors. This study was motivated by the need for additional information about the unsteady behaviour of such flows. In particular, attention is focussed on the onset of rotating stall, and how fluid characteristics change before and after onset, and as the compressor surge limit is approached.

Test facility and instrumentation

The facility used for these tests is the R-6 compressor test stand of the von Karman Institute[10]. It was designed to test the effect of Reynolds number on overall compressor performance particularly for low volumetric flow rates. The test section consists of an inlet channel, rotor, diffuser, and return channel which provide a flow path similar to one stage of a multistage industrial compressor. The compressor is contained within a cylindrical casing which may be pressurized between 1 and 4 bars. The impeller has seventeen blades, each of which has a 4.75 mm height and an outer exit radius of 114.2 mm. The volumetric flow rate in these tests is approximately 0.1 m^3/s with a maximum pressure ratio of about 1.15. The impeller rotational speed for all measurement points is 14 000 r/min. The density, and hence the Reynolds number, may be varied by changing the pressure in the test section.

A crossed hot-wire sensor, in the location shown in Fig 1 with respect to the compressor rotor, was used for measurement of flow direction and magnitude. Each wire on the probe is 9.0 μm diameter platinum plated tungsten with a sensing length of approximately 1.8 mm. The probe may be rotated about an axis perpendicular to the plane of the wires using a carriage to orient the hot-wires for optimal sensitivity for flow angle measurements. Alignment of the angular position with respect to the carriage is to within ±0.5 degrees. The probe was placed in the centre of the 3.5 mm wide diffuser

* von Karman Institute for Fluid Dynamics, Rhode-St-Genese, Belgium. Present address for P. M. Ligrani: Department of Aeronautics, Imperial College of Science and Technology, Prince Consort Road, SW7 2BY, London, England
† Société Creusot-Loire, Le Creusot, France
Based on a paper presented at the ASME 27th International Gas Turbine Conference and Exhibit, London, April 18–22, 1982. Revised manuscript received on 4th January 1983 and accepted for publication on 22nd February 1983

passage with an accuracy of ±0.5 mm such that the plane of the two wires were perpendicular to the axis of rotation of the compressor impeller (Fig 1). The signals from the hot-wire bridges were digitized and then sent to a PDP 11/34 computer to be stored for later data reduction, including hot-wire signal linearization. Additional details on the measurement and data reduction procedures are provided elsewhere[10].

Experimental results

Some understanding of the different types of flow behaviour which exist during stable, rotating stall, and surge operating conditions may be obtained by referring to Fig 2. Results are given for four different values of the inlet flow coefficient ϕ_1 at $Re_{b_3} = 1.7 \times 10^4$. The results of Fig 2 qualitatively similar to those obtained at $Re_{b_3} = 4.5 \times 10^4$. These Reynolds numbers were calculated using the flow conditions which exist at the onset of rotating stall, and the diffuser inlet width as the length scale.

In Fig 2(a), α_3 and V signal traces are presented during stable operation, when no rotating stall cells or surge are present. Unsteadiness due to wakes from impeller blades, which are expected to pass every 0.25 ms, is evident on the signal traces.

The traces shown in Fig 2(b) (operating point 94) were obtained during stall, at values of ϕ_1 near the onset of rotating stall. The magnitudes of fluctuations are larger than those in Fig 2(a). Also, from Figs 2(a) and (b), it is evident that the average flow angle $\bar{\alpha}_3$ increases, and the average flow velocity $\bar{V}$ decreases as ϕ_1 becomes smaller, a trend which may be observed for all traces when $\phi_1 > 0.010$. Ensemble averaged spectra of V signals obtained during stall, determined using fast Fourier transforms, show larger energy levels at frequencies below 200–300 Hz than were present during stable flow operation. The energy contribution to spectra from the presence of two stall cells appears at 110 Hz for operating point 94. An energy peak at 600 Hz also appears in all spectra taken when stall cells were present, but not otherwise. The passage frequency of impeller blade wakes, 4000 Hz, was also identifiable in spectra as an energy peak, except in cases where this energy level was not significantly greater than energy levels from broad-band turbulence existing near the same frequency.

Signals from two hot-film probes placed at two

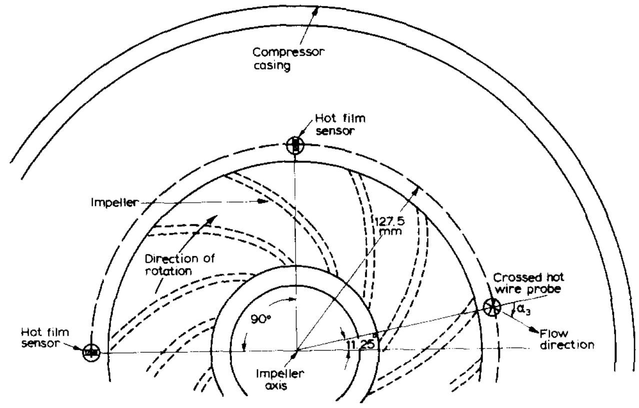

Fig 1 Compressor impeller and casing showing position of hot-wire probe

different circumferential positions at the diffuser inlet (Fig 1) indicated the presence of two stall cells, each rotating at approximately 55 rotations per second[10]. This corresponds to about 23% of the rotor speed which is lower than values normally observed in axial flow compressors.

Periodic variations of α_3 and V, due to the presence of fully developed rotating stall cells, are shown in Fig 2(c) for $\phi_1 = 0.0089$. The low frequency periodic unsteadiness contains greater energy than broad-band turbulence existing when stable conditions are present. The velocity and angle traces are out of phase as shown in Fig 2(c), and in Fig 3 where the phase-averaged velocity and flow angle traces are presented for one stall cell period. These phase averaged traces were constructed using 18 sequential portions of the data sample, each having the same time period (8.80 ms). In the traces of Fig 3, the phase lag between velocity and angle seems to be approximately one quater of one period, and the peak of $\hat{\alpha}_3$ occurs about 2.4 ms after the maximum of the phase-averaged velocity. The flow angle passes through a maximum (the flow direction is more tangential) as the velocity decreases. The shape of the velocity seems to be somewhat like a saw-tooth, whereas the variation of $\hat{\alpha}_3$ is more sinusoidal with time. Such behaviour may occur because the shear layers developing along the walls of the diffuser affect the flow at the hot-wire location. This variation is consistent with an increase of the radial component of the velocity which usually occurs as wall layers produce more blockage. Such behaviour, however, would not normally be expected in

Nomenclature

b_3	Inlet diffuser width
Re_{b_3}	$\rho \dfrac{\bar{V} b_3}{\mu}$
t	Time
V	Instantaneous absolute velocity
V'	Instantaneous absolute velocity fluctuation
α_3	Instantaneous flow angle at diffuser inlet relative to radial direction
α'_3	Flow angle fluctuation
ϕ_1	Inlet flow coefficient
ρ	Density
μ	Absolute viscosity

Subscripts

c	Critical value at onset of rotating stall

Superscripts

$\bar{}$	Mean (time-averaged) value
$\hat{}$	Phase-averaged value

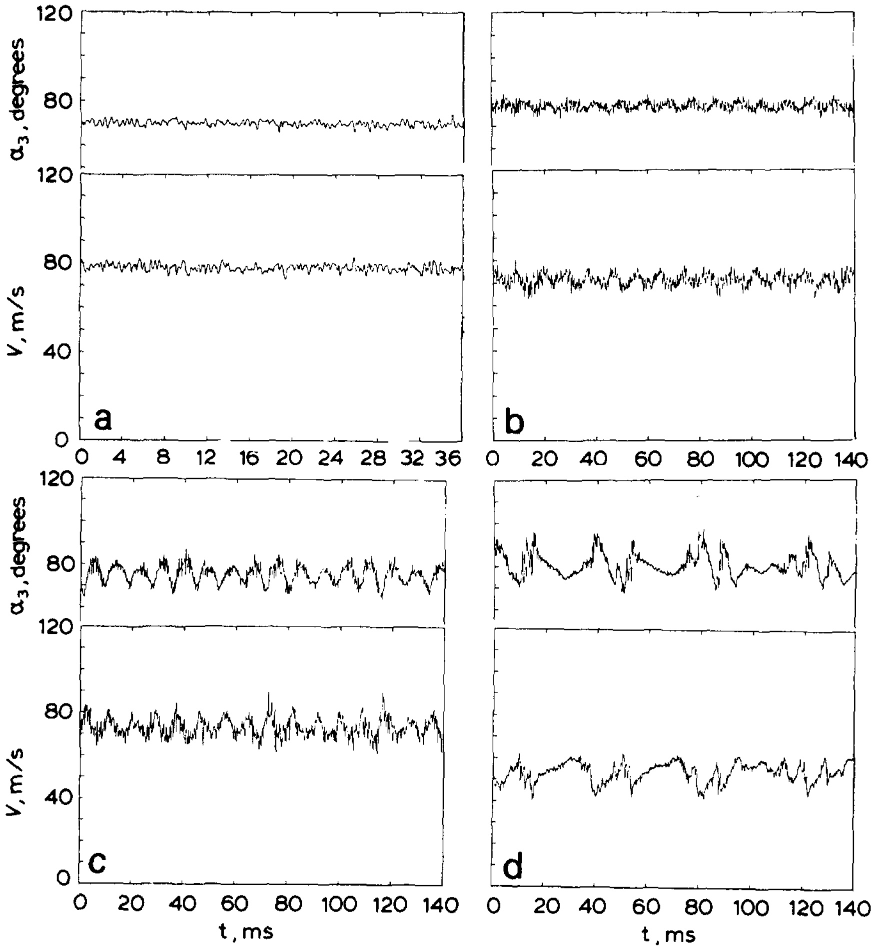

Fig 2 Crossed hot-wire velocity and angle traces, $Re_{b_3} = 1.7 \times 10^4$ *(a) operating point 90;* $\phi_1 = 0.0158$ *(b) operating point 94;* $\phi_1 = 0.0096$*, (c) operating point 96;* $\phi_1 = 0.0089$ *(d) operating point 97;* $\phi_1 = 0.0068$

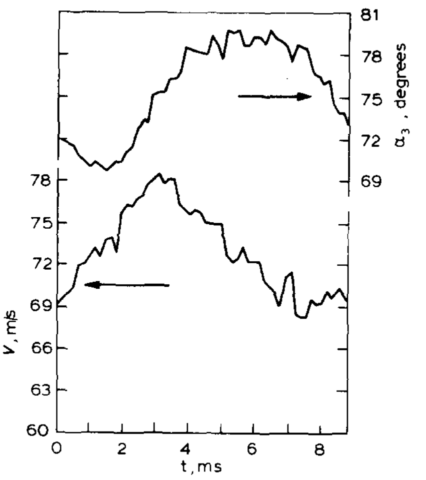

Fig 3 Phase-averaged flow angle and velocity, operating point 96, $Re_{b_3} = 1.7 \times 10^4$*,* $\phi_1 = 0.0089$

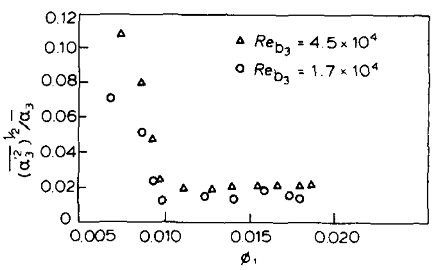

Fig 4 Normalized flow angle fluctuation versus inlet flow coefficient, $Re_{b_3} = 4.5 \times 10^4$*,* $Re_{b_3} = 1.7 \times 10^4$

diffusers if flow instabilities are caused by circumferential non-uniformity of relative flow from backward leaning blades. In this case, velocity variations would be expected to be in phase with angle variations such that maximum V occur at the same times as maximum α_3.

The results in Fig 2(d) were obtained when the compressor was operating in surge. The velocity and angle traces have larger fluctuations and longer time scales than those observed during rotating stall. The times when V decreases and increases seem to correspond closely with the times when α_3 increases and decreases. Thus, during surge, no phase lag occurs between the α_3 peaks and the troughs in V. The frequency at which the largest amplitude fluctuations occur is about 25 Hz. In contrast to flow disturbances from rotating stall, which periodically pass a given circumferential position. the large scale flow oscillations due to surge affect the entire diffuser passage. Such phenomena due to surge seem to occur in the R-6 test stand after rotating stall has developed in the impeller as well as the diffuser.

Normalised time-averaged magnitudes of the fluctuating component of α_3 are shown as they vary with the inlet flow coefficient in Fig 4 for two values of Re_{b_3}. In the figure, the normalised quantity does not vary significantly in magnitude at a given Re_{b_3} when stable flow conditions exist in the inlet of the vaneless diffuser and $\phi_1 > 0.010$. As ϕ_1 becomes less than 0.010, $(\overline{\alpha_3'^2})^{1/2}/\bar{\alpha}_3$ increases abruptly to become greater than the value for stable flow conditions due to the fluid oscillations created by rotating stall cell development. The abrupt change in the trends of the normalised fluctuating angle with ϕ_1 thus allows one to determine that $\phi_1 = 0.010$ at the onset of rotating stall in this machine. Similar conclusions may be made regarding the variation of $(\overline{V'^2})^{1/2}/\bar{V}$ with ϕ_1, and are important since abrupt changes in the normalised components of fluctuating velocity or angle may be used in other centrifugal machines to determine rotating stall onset.

Time averaged values of α_3 for $Re_{b_3} = 4.5 \times 10^4$ are shown in Fig 5 as a function of inlet flow coefficient. Maximum and minimum values of α_3 are also shown and indicate distinctly different trends for the three observed types of flow behaviour. During stable operation, the maximum–minimum difference is nearly constant and the mean value of α_3 changes almost linearly with ϕ_1. When rotating stall is present, these trends change. The difference between maximum and minimum flow angle becomes greater and the mean value decreases with decreasing ϕ_1.

Maximum values of α_3 during rotating stall seem to be limited to the maximum values at the onset of rotating stall. This limit disappears during surge at lower values of ϕ_1, when fluctuations become very large and α_3 sometimes exceeds 90°, indicating that the radial velocity component becomes negative. Thus, one difference between flow behaviour during surge and rotating stall is the presence of a limit on the maximum α_3, which exists during rotating stall but not when the compressor surges.

The magnitudes of mean flow angles at the onset of rotating stall, $\overline{\alpha_{3_c}}$, were calculated for the flow conditions and compressor geometry of the present tests using the empirical methods given elsewhere[8]. The result from one such calculation is shown in Fig 5, and is quantitatively consistent with the measured values of $\bar{\alpha}_3$ near the onset of rotating stall. This agreement seems to support the hypothesis that $\bar{\alpha}_3$ is an important quantity in determining stall onset along with Re_{b_3} and compressor diffuser geometry.

Conclusions

During stable operation, the average absolute flow angle at the diffuser inlet, $\bar{\alpha}_3$, increases almost linearly with decreasing ϕ_1, and the magnitude of $(\overline{\alpha_3'^2})^{1/2}/\bar{\alpha}_3$ is approximately constant as ϕ_1 varies. The onset of rotating stall is clearly defined by the abrupt way in which the normalized time-averaged component of the fluctuating angle changes with the inlet flow coefficient. The periodic variations of velocity and angle which occur during stall are a consequence of two cells of retarded flow, each rotating at about 23% of the rotor rotational speed. The periodic variations of velocity and angle, deduced from phase-averaging, indicate that these two quantities are out-of-phase, where a phase lag of approximately one quarter of one period exists at $\phi_1 = 0.0089$. As ϕ_1 decreases and the radial compressor approaches the surge limit, $\bar{\alpha}_3$ decreases and the magnitudes of the fluctuating components of velocity and angle increase. Instantaneous magnitudes of α_3 seem to be limited to a maximum value during rotating stall corresponding to the α_3 maximum at stall onset. This maximum is exceeded as ϕ_1 decreases and the compressor surges. During surge, velocity and angle fluctuations are largest and occur over the entire impeller contour at a given time, such that maximum α_3 occur at the same times as minimum V.

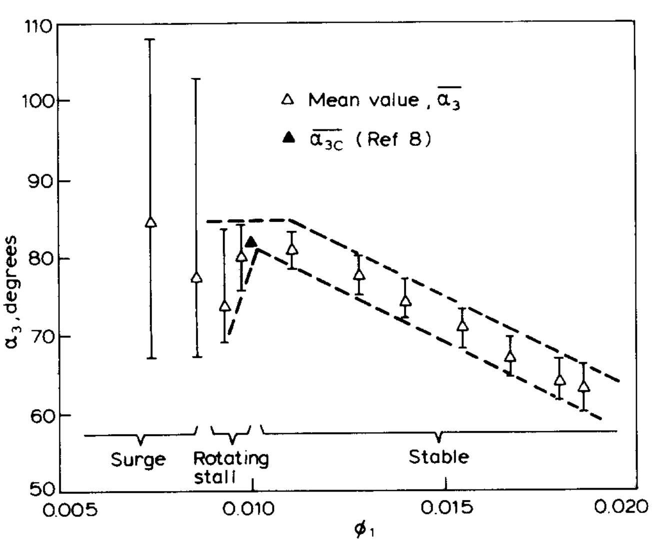

Fig 5 Variation of flow angle α_3 with inlet flow coefficient, $Re_{b_3} = 4.5 \times 10^4$

Acknowledgements

The work described in this paper is part of a research project carried out at the von Karman Institute under sponsorship from the Société Creusot-Loire. The authors express their gratitude to Société Creusot-Loire for permission to publish this material.

References

1. **Johnston J. P. and Dean R. C.** Losses in vaneless diffusers of centrifugal compressors and pumps; analysis, experiment and design. *Trans. ASME, Series A: J. Engrg. Power, 1966* 88(1) *49–62*
2. **Dean R. C. and Senoo Y.** Rotating wakes in vaneless diffusers. *Trans. ASME., Series D: J. Basic Engrg., 1960* 82(2), *563–574*
3. **McDonald G. B., Lenneman E. and Howard J. H. G.** Measured and predicted flow near the exit of a radial-flow impeller. *Trans. ASME., Series A: J. Engrg. Power, 1971* 93(4), *441–446*
4. **Eckhardt D.** Instantaneous measurements in the jet-wake discharge flow of a centrifugal compressor impeller. *Trans. ASME., Series A: J. Engineering for Power*, 1975, **97**(3), *337–346*
5. **Jansen W.** Rotating stall in a radial vaneless diffuser. *Trans. ASME., Series D: J. Basic Engrg., 1964*, **86**(4), *750–758*
6. **Senoo Y., Kinoshita Y. and Ishida M.** Asymmetric flow in vaneless diffusers of centrifugal blowers. Centrifugal Compressor and Pump Stability, Stall and Surge. *The Fluids Machinery and Turbomachinery Committee of the Fluids and Gas Turbine Division, ASME., 1976*
7. **Senoo Y. and Kinoshita Y.** Limits of rotating stall and surge in vaneless diffusers of centrifugal compressors. *ASME Paper 78-GT-19, 1978*
8. **Van Den Braembussche R., Frigne P. and Roustan, M.** Rotating non-uniform flow in radial compressors. *AGARD CP282, 1980*
9. **Abdelhamid A. N.** Analysis of rotating stall in vaneless diffusers of centrifugal compressors. *ASME Paper 80-GT-184, 1980*
10. **Ligrani P. M., Van Den Braembussche R. and Roustan M.** Rotating stall measurements in the vaneless diffuser of a radial flow compressor. *ASME Paper 82-GT-257, ASME Gas Turbine Conference and Exhibit, London, April 18–22, 1982*