EXPERIMENTAL INVESTIGATION OF UNSTEADY FLOW PHENOMENA IN A CENTRIFUGAL COMPRESSOR VANEY DIFFUSER OF VARIABLE GEOMETRY

Friedrich Justen, Kai U. Ziegler and Heinz E. Gallus
Rheinisch-Westfälische Technische Hochschule Aachen
Institut für Strahlantriebe und Turboarbeitsmaschinen
D-52056 Aachen, Germany

ABSTRACT

The behaviour of vaney radial diffusers is generally considered to be due to the flow phenomena in the vaneless and the semi-vaney space in the diffuser inlet region. Even considering unsteady aspects, the adjacent diffuser channel is regarded as less important. The flat wedge vane y diffuser of the investigated centrifugal compressor stage allows an independent continuous adjustment of the diffuser vane angle and the radial gap between impeller outlet and diffuser vane inlet, so that information about the importance of these geometric parameters can be obtained.

The time dependent pressure distribution on the diffuser front wall and on suction and pressure surface of the diffuser vanes reveal that in the semi-vaney space mainly the region near the vane suction side is influenced by the unsteady impeller-diffuser-interaction. Downstream in the diffuser channel the unsteadiness does not decay. Here, pressure fluctuations are appearing which are distinctly higher than the pressure fluctuations in the vaneless space.

An estimation of the influence of the unsteadiness on the operating performance of the centrifugal compressor stage is made by measurements at choke and surge limit for different diffuser geometries.

NOMENCLATURE

<table>
<thead>
<tr>
<th>Symbol</th>
<th>Description</th>
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<tbody>
<tr>
<td>AR</td>
<td>area-ratio acc. Kline et al. (1959)</td>
</tr>
<tr>
<td>AS</td>
<td>aspect-ratio acc. Kline et al. (1959)</td>
</tr>
<tr>
<td>F</td>
<td>fluctuation (Eq. 1)</td>
</tr>
<tr>
<td>L</td>
<td>length of diffuser channel (Fig. 3)</td>
</tr>
<tr>
<td>LWR</td>
<td>length-width-ratio acc. Kline et al. (1959)</td>
</tr>
<tr>
<td>c</td>
<td>absolute velocity</td>
</tr>
<tr>
<td>f</td>
<td>frequency of pressure fluctuation</td>
</tr>
<tr>
<td>f0</td>
<td>blade passing frequency</td>
</tr>
<tr>
<td>h</td>
<td>diffuser width (Fig. 3)</td>
</tr>
<tr>
<td>m</td>
<td>mass flow</td>
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<tr>
<td>n</td>
<td>speed</td>
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<td>p</td>
<td>pressure</td>
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<tr>
<td>r</td>
<td>radius</td>
</tr>
<tr>
<td>s</td>
<td>arc length</td>
</tr>
<tr>
<td>t</td>
<td>time, impeller blade pitch</td>
</tr>
<tr>
<td>x/l</td>
<td>dimensionless channel coordinate (Fig. 6)</td>
</tr>
<tr>
<td>α45S</td>
<td>diffuser vane suction side angle (Fig. 3)</td>
</tr>
<tr>
<td>ηt</td>
<td>total isentropic efficiency</td>
</tr>
<tr>
<td>ηd</td>
<td>diffuser channel divergence angle (Fig. 3)</td>
</tr>
<tr>
<td>π</td>
<td>pressure ratio</td>
</tr>
</tbody>
</table>

Subscripts

I, II, III, IV, V operating points close to surge limit
0 nominal speed
1 impeller inlet
2 impeller outlet
4 diffuser inlet
5 diffuser outlet
6 diffuser throat
N reference geometry
PS pressure side
SS suction side
max period maximum value of time dependent signal
min period minimum value of time dependent signal
red corrected to ISA inlet conditions (p = 1.013 bar, T = 288.15 K)
t total

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INTRODUCTION

In the last years the development of centrifugal compressor stages has reached a standard that requires, for further optimization, a better understanding of the unsteady flow aspects, especially if vaned diffusers are considered. Unsteady flow phenomena have significant influence on the loading, the efficiency, the pressure ratio, and the noise emission of centrifugal compressor stages. Modern impellers reach absolute discharge Mach numbers between 0.9 and 1.3, so that at least transonic diffuser inlet conditions are expected. Owing to the highly distorted impeller discharge flow, the flow field in the diffuser inlet region is determined by strong velocity and flow angle fluctuations, as well in circumferential as in axial direction (Eckardt, 1975; Krain, 1980; Krain et al., 1995). With increasing radius in the vaneless space, the flow mixes out comparatively fast in circumferential direction due to intensive exchange of momentum between jet and wake flow, whereas the non-uniformity in axial direction remains longer (e.g. Haß and Rautenberg, 1977).

Early after the introduction of the “jet-wake” model by Dean and Senoo (1960), numerous experimental as well as theoretical approaches to model the impeller discharge flow were published (e.g. Johnston and Dean, 1966; Eckardt, 1973; Senoo and Ishida, 1975). Progress of unsteady measurement techniques allows for a longer time a quite precise prediction of the flow patterns in centrifugal compressor stages with vaneless diffusers (Japikse, 1987). Comparable results for vaned diffusers could not be achieved as yet, because the region between impeller exit and diffuser inlet is determined by phenomena that are difficult to model theoretically and to obtain experimentally:

- Unsteady flow characteristic:
  Starting from a simplified model, Dean and Senoo (1960) developed a method describing the mixing process approximatively. In the rotating frame of reference the flow is regarded as steady and two-dimensional. The blade pitch is divided into two zones with different, but in each case constant velocities and equal flow angles. As a consequence of the non-uniformity of the relative flow, the absolute flow must be unsteady.

- Impeller-diffuser-interaction:
  The diffuser receives the inhomogeneous impeller discharge flow as an unsteady inlet flow, while the presence of the diffuser vanes means an unsteady disturbance for the impeller. Unsteady impeller-diffuser-interactions were investigated early by Krain (1981), Fisher and Inoue (1981) and Arndt et al. (1982). However, coincident results do not exist yet.

- Increase of boundary layer thickness and shock boundary layer interaction:
  The development of boundary layers in the viscous flow in the vaneless and semi-vaned space is determined substantially by the pressure gradient in flow direction. The boundary layer is turbulent, three-dimensional, and involved in secondary flow effects.

The flow phenomena mentioned above are not independent from each other concerning their development and extent. An enlargement of the radial gap, for instance, leads to a reduction of impeller-diffuser-interaction and to a more uniform diffuser inlet flow, but also to an increased growth of boundary layer thickness.

The transfer of experimental results to different types of centrifugal compressors in order to derive a generally valid design method for vaned diffusers is very problematic. Impeller, diffuser, and other compressor components can not be regarded as isolated parts. On the contrary, losses and operating performance of the stage are determined by the interaction of all components. I.e. empirical methods derived from component studies only have a limited universal validity to describe the real flow characteristics in the stage.

Further problems arise, attempting to predict the operating performance close to the limits of the compressor characteristics. Especially stages with vaned diffusers and transonic flow conditions have a limited operating range. As the point with maximum efficiency is usually located close to surge limit, it is necessary to expand the range of control up to the limits of the particular characteristic in order to enable an efficient operation. Several publications are concerned with the causal mechanisms triggering the instability of a centrifugal compressor. For instance, Elder and Gill (1984) state different diffuser variables influencing the location of the stability limit, e.g. the number of impeller and diffuser blades, the pressure recovery in the semi-vaned space or the sensitivity of the diffuser vanes to incidence.

EXPERIMENTAL FACILITY

The experimental investigations were carried out on a centrifugal compressor stage with a vaned diffuser of variable geometry. A detailed description of the development, the technical realization, and the measurement of compressor maps is given by Rothstein (1993).

Test Bed

The compressor stage is run in a closed loop using air and allowing a variation of the pressure level (Fig. 1). The components of the test bed are: compressor stage (1), suction pipe with honeycomb (2), settling chamber with air filter (3), throttle arrangement (6, 7, 9, 11), heat exchanger (8), flow nozzle in conformity with DIN 1952 (10), 500 kW d.c. motor with continuous speed control (4), gearbox with speed increasing ratio of 25 (5).

Centrifugal compressor stage

The stage consists of an unshrouded impeller with 15 backswpt blades (38° backswep from radial) and a diffuser with 23 wedge vanes. The design of suction pipe and inlet contour and omission of inlet guide vanes ensure axial flow at impeller inlet. Figure 2 shows a view of the stage, with the diffuser front wall
The aerodynamic design of the wedge vaned diffuser is based on the characteristic parameters for flat diffusers of Runstadler et al. (1975). The construction of the diffuser allows an independent continuous adjustment of the diffuser vane angle, indicated by the vane suction side angle $\alpha_{4SS}$, and the radial gap between impeller outlet and diffuser vanes inlet, indicated by the radius ratio $r_4/r_2$ (Fig. 3 and Fig. 4). The most important data of the stage for nominal speed and diffuser reference geometry is collected in Tab. 1.

### Tab. 1 Stage data for nominal speed and diffuser reference geometry

<table>
<thead>
<tr>
<th>Number of impeller blades</th>
<th>$z_1$</th>
<th>15</th>
</tr>
</thead>
<tbody>
<tr>
<td>Blade angle impeller exit</td>
<td>$\beta_2$</td>
<td>128°</td>
</tr>
<tr>
<td>Impeller tip radius</td>
<td>$r_2$</td>
<td>135 mm</td>
</tr>
<tr>
<td>Impeller tip speed</td>
<td>$u_{2g}$</td>
<td>498 m/s</td>
</tr>
<tr>
<td>Relative tip Mach number</td>
<td>$M_{a_{2g}}$</td>
<td>0.95</td>
</tr>
<tr>
<td>Absolute Mach number</td>
<td>$M_{a_2}$</td>
<td>0.94</td>
</tr>
<tr>
<td>Shaft speed</td>
<td>$n_o$</td>
<td>35200 1/min</td>
</tr>
<tr>
<td>Number of diffuser vanes</td>
<td>$N_d$</td>
<td>23</td>
</tr>
<tr>
<td>Meridional diffuser height</td>
<td>$x_d$</td>
<td>11 mm</td>
</tr>
<tr>
<td>Radial gap</td>
<td>$r_{4/r_2}$</td>
<td>1.10</td>
</tr>
<tr>
<td>Diffuser vane angle</td>
<td>$\alpha_{4SS}$</td>
<td>16.5°</td>
</tr>
<tr>
<td>Diffuser channel divergence angle</td>
<td>$\theta$</td>
<td>9°</td>
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</table>

#### Test program

The following detailed measurements were carried out to estimate the influence of radial gap and diffuser vane angle on unsteady flow phenomena in the vaned radial diffuser:
- Unsteady pressure measurements on the diffuser front wall,
- Unsteady pressure measurements on suction and pressure surface of the diffuser vanes at midspan,
- Unsteady pressure measurements at the stability limit,
- Unsteady schlieren observations in the diffuser inlet region.

### Tab. 2 Diffuser channel characteristic numbers for the investigated diffuser geometries

<table>
<thead>
<tr>
<th>$r_{4/r_2}$</th>
<th>$\alpha_{4SS}$</th>
<th>AS</th>
<th>AR</th>
<th>LWR</th>
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<tbody>
<tr>
<td>1.06</td>
<td>16.5°</td>
<td>0.78</td>
<td>2.60</td>
<td>10.19</td>
</tr>
<tr>
<td>1.08</td>
<td>16.5°</td>
<td>0.73</td>
<td>2.48</td>
<td>9.42</td>
</tr>
<tr>
<td>1.10</td>
<td>16.5°</td>
<td>0.70</td>
<td>2.42</td>
<td>9.00</td>
</tr>
</tbody>
</table>

For four different diffuser geometries (Tab. 2) the measurements were carried out at three different speeds in each case ($n_{red} = 60\%, 70\%, 80\%$), and at each speed in three different operating points (choke limit, middle region of characteristic, close to surge limit). The investigations at the stability limit are an exception. They were carried out for two different radial gaps ($r_{4/r_2} = 1.10$ and 1.06) both combined with the same vane angle ($\alpha_{4SS} = 16.5°$). The compressor maps of the investigated diffuser geometries are shown in Fig. 5.
Measurement techniques

The experimental investigation of high loaded centrifugal compressors makes high demands on the used measurement technique. The achievable stage pressure ratios cause a corresponding thermal loading of all components in direct or indirect contact with the fluid. Probes positioned directly after impeller exit are exposed to an extreme dynamic loading due to the highly unsteady flow. In addition, owing to the very sensible transonic flow in the mostly narrow flow channels, the use of probes even of smallest dimensions is restricted or impossible. Although laser measurement techniques are most suitable under these conditions, their utilization is not efficient for first basic studies due to the large expenditure of installation.

For measuring the endwall pressure distribution, there are 120 measuring points available on the diffuser front wall (Fig. 6). To acquire the vane pressure distribution, 21 measuring positions on each side of the vane at midspan and one additional on the vane leading edge are used (Fig. 6). Because of the good rotational symmetry (Rothstein, 1993) the measuring points can be distributed on three measuring vanes used simultaneously. With the adjustment mechanisms shown in Fig. 7 the measuring vanes can be adapted to different diffuser geometries. The periodic pressure fluctuations are measured with semiconductor pressure transducers of the type Kulite XCS-093 (reference pressure type). Their natural frequency is 200 kHz, and the membrane diameter is 0.96 mm. Owing to the not reproducible zero displacement, they are used only to measure the fluctuating portion of the signal. For recording the time dependent signals, a transient recorder with eight channels, 65536 words memory per channel, and a maximum sampling frequency of 10 MHz is used. Because of the large measurement expenditure and the considerable amount of data the unsteady wall pressure measurements are limited to the first half of the diffuser channel. To adjust every measuring point, a Kulite is fitted in a multiple eccentrical adjustment mechanism (Fig. 8).

The limitation in space and financial aspects (43 measuring points) do not allow an application of Kulites flush to the surfaces of the diffuser vanes, so that a separation of measuring point and pressure transducer is necessary. By that, the pressure signal is falsified, because amplitude damping and phase shift are occurring. A thorough investigation and calibration of the dynamic systems behaviour, as described by Justen (1993), becomes necessary.

For visualization and qualitative analysis of the flow, the well-known schlieren method according to Toepler is applied. The stroboscopic exposure is realized with a high frequency spark lamp allowing a flash duration of less than 20 ns. Through an adjustable delay circuit a spark can be triggered at different rotor positions.

EXPERIMENTAL INVESTIGATIONS

Unsteady diffuser flow

In the following figures showing the unsteady pressure distribution on the diffuser front wall "s" represents the coordinate in circumferential direction and "t" is the arc length of an impeller blade pitch. The specified values of the isobars are the local static pressures normalized by the pressure in the settling chamber (Fig. 1, number 3). Starting from s/t = 0, the figures show from left to
right and from top to bottom instantaneous pressure distributions for eight equidistant circumferential impeller positions.

Figure 9 shows the unsteady pressure contours for an operating point close to surge limit at 80% nominal speed. At the impeller position $s/t = 0$ an impairment of the pressure rise can be recognized in the diffuser throat decaying not before the impeller has rotated to $s/t = 0.375$. At the same time the isobar $\pi = 2.17$ moves downstream, and raises the distance to the isobar $\pi = 2.12$, which moves hardly. This reveals a diminishing pressure rise shortly after the diffuser throat. The isobars in the rear part of the diffuser channel demonstrate a reverse tendency. Here, close to the pressure side at $s/t = 0$ a zone with distinctly higher pressure between two isobars of equal value can be identified, which expands up to the suction side after a rotation to $s/t = 0.125$. The pressure contours upstream of this region do not change their position in this time. From $s/t = 0.375$ on, the isobars with higher pressure level travel towards diffuser exit. Simultaneously the isobars situated before reduce their distance, and move in the direction of the throat. As a result of this, two isobars with the same pressure value occur in the diffuser channel despite divergent flow path ($\pi = 2.34$ at $s/t = 0.625$). At $s/t = 0.5$ an impairment of the pressure rise before the throat becomes visible causing a diminished pressure gradient in the semi-vaned space. At $s/t = 0.875$ the growing high pressure region close to the pressure side can clearly be identified already. The beginning enhancement of the pressure rise in the semi-vaned space can be recognized as well (compare $s/t = 0$).

As revealed by Fig. 10, a reduced diffuser vane angle does not change the basic structure of the time dependent diffuser flow. The distinctly weaker disturbances in the diffuser throat region are remarkable. At $s/t = 0$ the high pressure zone in the rear part of the diffuser channel already ranges over the entire flow path. The impairment of the pressure rise in the diffuser throat has travelled downstream. Here as well, the isobars with higher pressure level travel towards diffuser exit from $s/t = 0.375$ on. Likewise, the slight drop of the pressure gradient in the semi-vaned space between $s/t = 0.5$ and $s/t = 0.75$ can be detected.

Observing the pressure signals of the single measuring points during the passing by of an impeller blade pitch, a consequence for the diffuser flow can be shown (Fig. 11, 13, and 15). By plotting the local normalized pressure fluctuations according to Eq. 1

$$F(x/l) = \frac{P_{\text{max}}(x/l) - P_{\text{min}}(x/l)}{2P(x/l)} \quad (1)$$

versus the dimensionless channel coordinate (Fig. 6), zones of high and low pressure fluctuations can clearly be distinguished. After the diffuser throat suction and pressure side show similar trends. In operating points close to the surge limit the shape of the
fluctuations within the diffuser channel are comparable to a standing wave, because the reference pressure perpendicular to the diffuser channel is constant due to the proper channel flow, and the relative maxima and minima of the fluctuations of suction and pressure side occur at the same x/l-position. Interference phenomena of at least two pressure waves are the cause of the standing wave. Due to a reflection at the exit of the diffuser blading the upstream and the downstream running pressure waves are interfering. The superposition of their instantaneous values results in the total fluctuation value for each position. Hence, the unsteady effects do not decay along the diffuser channel, but maintain the fluctuation level inspite of the increasing local reference pressure.

Figure 11 shows such a diagram described above. Here, pressure fluctuations from measurements on suction and pressure surface of a diffuser vane at midspan are plotted versus the dimensionless channel coordinate for the same diffuser geometry and operating point as in Fig. 10. The measurements on the vane suction surface reveal pronounced relative maxima and minima in the semi-vaned space. In this region, the steep increase of the fluctuations shortly before the diffuser throat after a slight decrease is remarkable. This is due to intensified reflection phenomena and herewith to intensified impeller-diffuser-interactions. In the diffuser throat the fluctuations first decrease slightly, but then increase to their maximum. Within the diffuser channel, suction side and pressure side show very similar courses. Due to their common positions of relative maxima and minima they indicate a node-antinode structure and herewith a standing wave.

An enlargement of the radial gap leads to a decrease of unsteadiness due to the momentum exchange process, so that a distinctly reduced fluctuation level results at the diffuser vanes leading edges. As an example, Fig. 12 shows the time dependent pressure contours for an enlarged radial gap, but otherwise comparable conditions as in Fig. 9. The smooth structure of all instantaneous pressure distributions is remarkable. If the isobar $\pi = 2.22$ was not plotted, an any time homogeneous channel flow could be assumed. The actual movement of the isobars occurs downstream. The distinct motion in this region depending on the rotor position reveals noticeable pressure fluctuations within the diffuser channel.

Besides the generally diminished level of the pressure fluctuations, the enlargement of the radial gap changes as well the course of the fluctuations along the channel center line, especially in the semi-vaned space. For an enlarged radial gap, but otherwise comparable conditions as in Fig. 11, Fig. 13 gives an example. In the semi-vaned space the fluctuations grow nearly linearly at first. Thereby a relative maximum is reached nearly coinciding with the relative minimum of the smaller radial gap, concerning as well the position as the value. In the sequel, after a distinct drop before the throat, here as well the absolute maximum of the suction-sided fluctuations is reached not before the diffuser throat. In the adjacent diffuser channel the enlargement of the radius ratio leads to an elongation of the oscillation antinodes, so that fewer relative

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**Fig. 11** Pressure fluctuations on vane surfaces ($r_4/r_2 = 1.06, \alpha_{ass} = 14.5^\circ, \eta_{red}/\eta_{red,0} = 0.8$, close to surge limit)

**Fig. 12** Time dependent pressure contours ($r_4/r_2 = 1.10, \alpha_{ass} = 16.5^\circ, \eta_{red}/\eta_{red,0} = 0.8$, close to surge limit)

**Fig. 13** Pressure fluctuations on vane surfaces ($r_4/r_2 = 1.10, \alpha_{ass} = 14.5^\circ, \eta_{red}/\eta_{red,0} = 0.8$, close to surge limit)
80% nominal speed characteristic is displayed. At \( s/l = 0 \) a low pressure zone \((n = 1.85)\) can be detected extending to the suction side in the course of a rotation through half of an impeller pitch. In this manner the flow is accelerated after passing the throat, so that the pressure rise sets in downstream. While the impeller rotates on, the low pressure zone decreases and moves back to the pressure side. The distortion of the pressure contours in the semi-vaned space reveals the upstream effect of the described throat flow on the inlet region of the diffuser. In addition, the impairment of the pressure rise in the throat region causes small zones of higher pressure downstream in the diffuser channel.

The incidence of the diffuser vanes, caused by increased mass flow, as well influences significantly the pressure fluctuations within the diffuser channel. For the same operating point as in Fig. 14, in Fig. 15 the local fluctuations on the diffuser front wall are plotted. In general, the fluctuations have moved towards the suction side. The considerable rise of the suction-sided fluctuations in the throat region indicates an increased interaction between alternating incidence of the diffuser vanes and reflection phenomena.

A reduction of shaft speed does not change the basic structure of the results presented above (see Justen, 1993). For that reason, a figure showing these results shall be renounced here.

In conclusion, a further preparation of the experimental data is presented in Fig. 16. The local time dependent portions of the signal from the vane pressure side are plotted versus the dimensionless channel coordinate for several equidistant time intervals within a rotation through one impeller pitch. The resulting node-antinode structure reveals clearly a standing wave in the diffuser channel. Though it must be mentioned that this diagram cannot be compared with the figures above, since here only the time dependent portions of the pressure signals are plotted without considering the increasing steady mean pressure.

Regarding Figures 11, 13, 15, and 16 it should be noticed that the position \( x/l = 1.0 \) is located in the middle region of the diffuser channel and not at the end. As shown in Fig. 6 the last measuring point on the vane pressure side is found at \( x/l = 1.0 \).
Unsteady investigations close to choke and surge limit

**Choke limit.** Running the compressor at the choke limit, the steady schlieren photos of the diffuser flow show the shocks as bright broad stripes (Justen, 1993). In contrast to this, the use of the high frequency spark lamp allows a detailed observation of the shock oscillation depending on impeller position. By exposing with only one flash, instantaneous shock configurations, as shown in Fig. 17, can be visualized. The two schlieren photos in the upper part of Fig. 17 represent two different rotor positions in an operating point, where the flow is still choked. The left picture reveals clearly that the strong shock is perpendicular in the middle of the channel only. At the leading edge the distinct pressure-sided low pressure region caused by high incidence can be identified. The right photo shows that the shock has moved upstream due to a rotation of the impeller through few degrees. The stronger curvature of the strong shock at the vane suction side compared with the pressure side is obviously due to an additional shock wave attached to the vane leading edge. This shock configuration is described in detail by Bühler and Suter (1986). The lower part of Fig. 17 shows again two stroboscopic schlieren photos at different impeller positions in an operating point slightly more throttled. The raising of the back pressure leads to a lower incidence, so that the additional shock described above does not appear any more. The strong shock has moved upstream, and now is perpendicular to the channel center line. The operating point shown here is the real choke limit, because a further little raise of the back pressure leads to a complete unchoking of the diffuser, and herewith the compressors range of control begins.

**Surge limit.** In each case one semiconductor pressure transducer mounted flush in the diffuser front wall at impeller inlet, at impeller exit, at the diffuser throat, and at diffuser exit (numbers 1, 2, 5, and 6 in Fig. 18) allows an investigation of unsteady flow phenomena shortly before and during surge. For an estimation of the mechanical loading during surge, the axial shaft movement is recorded additionally. First, the compressor is pre-throttled with a pressure-sided slide valve (number 7 in Fig. 1). To exclude the influence of a changed characteristic of the closed loop, always the same end position of the slide valve is adjusted. The further throttling up to the surge limit is done by a slow stepwise closing of the pressure-sided butterfly valve (number 6 in Fig. 1). The fast-opening-mechanism of this valve is connected with the transient recorder, which allows the recording of data in a certain space of time before the release of the fast-opening-mechanism, depending on the chosen pre-trigger. Surge is identified by the "acoustic method", because it is accompanied by an audible sound comparable to a heavy hammer stroke on a pipe. The reason for pre-throttling with a slide valve is to keep the mechanical loading of the butterfly valve low in order to avoid the danger of jamming.

![Fig. 18 Positions of unsteady pressure transducers for measurements at surge limit](http://gasturbinespower.asmedigitalcollection.asme.org/GT/proceedings-pdf/GT1998/78620/V001T01A095/2410255/v001t01a095-98-gt-368.pdf)

Figure 19 shows the courses of the unsteady pressure signals at 80% nominal speed. Before the first surge cycle impeller exit (2) and diffuser throat (5) reveal already a distinct alteration of the pressure signal, while impeller inlet (1) and diffuser exit (6) do not show any noticeable changes. The beginning of reverse flow is characterized by a slight pressure drop at diffuser exit, whereas impeller inlet and exit and diffuser throat show a steep positive pressure rise. Simultaneously, the rotor moves abruptly towards the shroud, what is leading to a heavy loading of the thrust bearing and involving the danger of an impeller shroud contact. During the reverse flow operation as well impeller exit as diffuser throat have strong pressure oscillations decreasing to the normal level shortly before reaching the pressure minimum. The following normalization first leads to an overswing of the pressure signals in the diffuser, whereas in the impeller a stable state is reached relatively fast. The restart of the regular flow can be identified clearly by the backward shift of the rotor. In the time between two surge cycles the flow can be characterized as stable.
The influence of a smaller radial gap is described with the help of Fig. 22. Here, the impeller exit shows a distinct alteration of the pressure signal before the first surge cycle, whereas in the diffuser throat only a slightly increased amplitude can be detected. With a lower radial gap the instability now is triggered obviously by the impeller. Here as well, a further analysis of the signal does not reveal frequencies typical for rotating stall.

In Fig. 21 the frequency spectra during the approach to the stability limit are plotted. In a moderate operating point relatively far from the stability limit (number I in Fig. 21) impeller exit, diffuser throat, and diffuser exit show very similar spectra. Impeller and diffuser exit have a little more pronounced second harmonic. Throttling the compressor, the first harmonics in the diffuser, at first especially in the diffuser throat, increase reaching their maximum close to the stability limit (number V in Fig. 21). (The phenomenon of the not decaying pressure fluctuations along the diffuser channel was discussed above.) Reducing the mass flow, the first harmonic at impeller exit decreases. This indicates a more uniform relative flow field across the blade pitch close to the shroud in highly throttled operating points, what is typical of backswept impellers (Rhone and Baumann, 1988). The increased amplitudes in the lower frequency range (right part of Fig. 21) occur in all operating points, and can be interpreted as the natural frequency of the diffuser channel. It is remarkable that distinct amplitudes of this frequency appear at impeller exit as well, as a result of the upstream effect of the vane diffuser.

The instability of the system is obviously triggered by a local degradation of the flow in the diffuser throat causing increased oscillation amplitudes in this region and leading to a pressure rise at impeller inlet and exit. A further signal analysis of the section with increased amplitudes does not reveal frequencies typical for rotating stall.

Fig. 20 shows an extended time scale. It can be seen that there is a pressure rise in the impeller accompanied by a pressure drop in the diffuser throat, before the pressure begins to decrease in the diffuser exit (not yet visible in Fig. 20). The instability of the system is obviously triggered by a local degradation of the flow in the diffuser throat causing increased oscillation amplitudes in this region and leading to a pressure rise at impeller inlet and exit. A further signal analysis of the section with increased amplitudes does not reveal frequencies typical for rotating stall.

Fig. 19 Real-time signals during surge \((r_4/r_2 = 1.10, \sigma_{455} = 16.5°, n_{red}/n_{red,0} = 0.8)\)

Fig. 20 Real-time signals shortly before the first surge cycle \((r_4/r_2 = 1.10, \sigma_{455} = 16.5°, n_{red}/n_{red,0} = 0.8)\)
Fig. 22  Real-time signals shortly before the first surge cycle ($r_4/r_2 = 1.06, \sigma_{4SS} = 16.5^\circ, n_{red}/n_{red,0} = 0.8$)

Fig. 23  Real-time signals shortly before the first surge cycle ($r_4/r_2 = 1.10, \sigma_{4SS} = 16.5^\circ, n_{red}/n_{red,0} = 0.6$)

Fig. 24  Real-time signals shortly before the first surge cycle ($r_4/r_2 = 1.10, \sigma_{4SS} = 16.5^\circ, n_{red}/n_{red,0} = 0.7$)

Fig. 25  Frequency spectra at impeller inlet during approach to stability limit ($r_4/r_2 = 1.10, \sigma_{4SS} = 16.5^\circ$)

For two additional shaft speeds (60% and 70% nominal speed) the courses of the unsteady pressure signals up to the first surge cycle are given in Fig. 23 and Fig. 24. At $n_{red}/n_{red,0} = 0.6$ the impeller exit shows the first remarkable alterations, whereas at $n_{red}/n_{red,0} = 0.7$ it can not be definitely clarified, if the diffuser throat or the impeller exit triggers the instability.

In the literature it is often reported on an inducer stall triggering the instability of the system (e.g. Jansen et al., 1980; Kosuge et al., 1982). For an analysis of this, in Fig. 25 the frequency spectra at impeller inlet during the approach to the stability limit are plotted. In all operating points the first harmonic is dominating the higher harmonics. The signal components of lower frequency in the right part of Fig. 25 do not give any hints at rotating stall. The variations of the amplitudes can be explained by stochastic events, because they vanish through an averaging of the signal and a following Fourier transformation.

The region between impeller exit and diffuser throat is sure to be critical in regard of the triggering of instability. In general, here the first remarkable alterations are appearing, which trigger the instability of the system due to an interactive building-up process. The radial gap determines the amplitudes of the fundamental pressure fluctuations already in the stable regime, and thus influences first of all the "sharpness" of the surge limit. As the results show likewise, the unsteady measurements do not allow direct conclusions about the position of the operating point relative to the surge limit. The first remarkable alterations do not appear before reaching the surge limit.
CONCLUSIONS
Unsteady flow phenomena in a vaned radial diffuser have been investigated with the help of unsteady pressure measurements on the diffuser front wall and on suction and pressure surface of the diffuser vanes at midspan, considering four different diffuser geometries and different operating points. The presented wall pressure distributions allow an observation of the static pressure rise depending on impeller position, and offer vivid instantaneous views of the pressure field. It is shown that for all operating points and diffuser geometries the unsteadiness of the impeller discharge flow influences not only the time dependent pressure rise in the vaned and semi-vaned space, but also the configuration in the adjacent diffuser channel to a high degree. The pressure fluctuations occurring there are caused by a periodic stimulation by the jet-wake flow. An analysis of the amplitudes of the pressure fluctuations reveals that in the semi-vaned space especially the region close to the vane suction side is affected by unsteady impeller-diffuser-interactions. Downstream in the diffuser channel the unsteadiness does not decay, even with an enlarged radial gap. Here, pressure fluctuations are appearing which can be distinctly higher than the pressure fluctuations in the vaned space, depending on the operating point. The amplification of a pressure wave without additional energy input is possible through the interference with other pressure waves, in this case caused by reflections at the exit of the diffuser blading and at the side-walls. In operating points close to the surge limit the shape of the local pressure fluctuations within the diffuser channel is comparable to the structure of a standing wave.

At the choke limit the diffuser throat region acts as a convergent/divergent nozzle, owing to the steep inlet flow angle and the resulting separated region on the vane pressure side. Consequently, operating points close to the choke limit are determined by supersonic flow and shocks in the diffuser channel, despite subsonic diffuser inlet flow.

After overstepping the stability limit, the beginning reverse flow is characterized by a slight pressure drop at diffuser exit, whereas impeller inlet and exit and diffuser throat show a steep positive pressure rise. Simultaneously, the rotor moves abruptly towards the shroud. During the reverse flow operation as well impeller exit as diffuser throat have strong pressure oscillations decreasing to the normal level shortly before reaching the pressure minimum. A preceding rotating stall or inducer stall cannot be detected. A single component of the stage which triggers the instability of the system at all shaft speeds and diffuser geometries cannot be identified. Neither, the unsteady measurements do not allow direct conclusions about the position of the operating point relative to the surge limit.

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