Steady and Unsteady Flow Phenomena in a Channel Diffuser of a Centrifugal Compressor*

Jeong-Seek KANG** and Shin-Hyoung KANG***

The aim of this paper is to understand the time averaged pressure distributions and unsteady pressure patterns in a channel diffuser of a centrifugal compressor. Pressure distributions from the impeller exit to the channel diffuser exit are measured and discussed for various flow conditions. And unsteady pressure signals from six fast-response sensors in the channel diffuser are analyzed by decomposition method. The strong non-uniformity in the pressure distribution is obtained over the diffuser shroud wall caused by the impeller-diffuser interaction. As the flow rate increases, flow separation near the throat, due to large incidence angle, increases aerodynamic blockage and reduces the aerodynamic flow area downstream. Thus the minimum pressure location occurs downstream of the geometric throat, and it is named as the aerodynamic throat. And at choke condition, normal shock occurs downstream of this aerodynamic throat. The variation in the location of the aerodynamic throat is discussed. The pressure ratio waveforms by blade passing show regular oscillation not only for the normal but also for the surge conditions and the high frequency fluctuations are superposed on the oscillating pressure waveform as the flow rate increases. Periodic unsteadiness by blade passing does not decay in the diffuser channel. It depends on the operating point and is generally larger in the channel than in the vanless space. Aperiodic unsteadiness rapidly decrease downstream of diffuser channel.

Key Words: Centrifugal Compressor, Channel Diffuser, Pressure Distribution, Decomposition of Instantaneous Pressure

1. Introduction

Channel diffuser has high pressure-recovery characteristics in centrifugal compressor and, therefore, it is widely used in high efficiency centrifugal compressors. Extensive experimental data in single diffuser channel have been published by Runstadler et al. (1969, 1973, 1975). However, knowledge about the influence of blockage or inlet velocity distribution for single channel diffuser is not directly applicable to centrifugal compressor (Filipenko et al., 1998). Because the three-dimensional flow from the impeller becomes more complex due to impeller and vaned diffuser interaction and the high-level unsteadiness from the interaction affects the performance of compressors. Furthermore the large incidence angle at the diffuser leading edge in off-design conditions causes the flow phenomena in the diffuser more complex. Hence it is necessary to investigate the flow phenomena in the channel diffuser and in the vanless space in various operating conditions.

Detailed flow phenomena through an impeller and a diffuser was measured by Krain (1981, 1984) using a L2F. Dawes (1995) investigated the unsteady interaction between the centrifugal impeller and vaned diffuser using unsteady numerical simulation. How-
ever, the flow phenomena at off-design flow conditions also require further study. Kano et al. (1982) measured static pressure distributions in a channel diffuser at several flow rate conditions. Justen et al. (1998) measured unsteady pressure field in a channel diffuser in design and near surge flow rates using fast-response sensors and visualized the instantaneous shock configurations at the choke limit.

The purpose of this paper is to understand the static pressure distribution and the unsteady pressure characteristics in the vaned diffuser over the entire flow condition from the near surge to choke condition. The distributions of static pressure in the vaneless space and in the channel diffuser were measured over the entire flow condition from the near surge to choke condition. And, at the same time, unsteady pressures were also measured at six locations at the diffuser inlet and along the channel center. The unsteady characteristics and the variation of pressure waveform by blade passing were investigated using a decomposition method modified from Suryavamshi et al. (1997, 1998).

2. Nomenclature

\[ A_{SN} \] : diffuser throat aspect ratio  
\[ L \] : length of channel diffuser  
\[ N_b \] : number of impeller blades  
\[ N_{rev} \] : number of revolutions of data acquired  
\[ N_{data,\text{window}} \] : number of data samples in a window  
\[ N_{data} \] : number of total data samples acquired  
\[ \bar{P} \] : time averaged pressure  
\[ P_{inlet} \] : inlet chamber total pressure  
\[ Pr \] : static pressure ratio \((= P/P_{inlet})\)  
\[ Pa \] : atmospheric pressure  
\[ \bar{Pr} \] : time averaged pressure ratio  
\[ Pr_{a} \] : instantaneous pressure ratio  
\[ Pr_{ua} \] : shaft unresolved pressure ratio component  
\[ (Pr_{ua})_{ab} \] : shaft resolved pressure ratio component  
\[ (Pr_{ua})_{bp} \] : revolution periodic pressure ratio component  
\[ (Pr_{ua})_{ka} \] : revolution aperiodic pressure ratio component  
\[ RMS \] : root mean square value  
\[ R_{c} \] : impeller tip radius  
\[ R_{i} \] : channel diffuser inlet radius  
\[ TLE \] : diffuser vane leading edge thickness  
\[ W_t \] : diffuser throat width  
\[ ZD \] : number of diffuser vanes  
\[ \alpha \] : vane angle at diffuser inlet (from radial)  
\[ 2\theta \] : divergence angle of diffuser  
\[ 2\phi \] : divergence angle of diffuser vane

Subscripts

\( i \) : index of revolution  
\( j \) : index of blade passage  
\( k \) : index of window in a blade passage

3. Experimental Facility and Instrumentation

The measurements were made in a single stage centrifugal compressor with a channel diffuser. The test rig is composed of an impeller, a diffuser, a large collector, a inlet settling chamber, an air turbine, and an instrumentation system (Kang et al., 1998). The meridional view of test section is shown in Fig. 1. The radial air turbine is driven by compressed air. The flow rate is measured with the pre-calibrated nozzle at the settling chamber prior to the compressor inlet and the flow rate is controlled by the throttling valve at the compressor exit. The inlet and outlet total temperatures are measured using T-type thermocouples installed at the settling chamber and the collector. The rotating speed of the impeller is measured by an induced electro-magnetic force signal between the coil and a permanent magnet fixed on the turbine shaft.

The test impeller and channel diffuser is shown in Fig. 2. The specifications of the impeller and diffuser are as follows:

- Impeller diameter: 110 mm
- Inducer tip diameter: 63.4 mm
- Inducer hub diameter: 20.4 mm
- Inducer tip blade angle: -60.0 deg.
- Inducer hub blade angle: -29.2 deg.

![Fig. 1 Meridional view of experimental facility](image-url)
Figure 3 shows a schematic view of the channel diffuser. The locations of six fast-response pressure sensors over the shroud wall of diffuser are shown in Fig. 4. Sensors $a$, $b$, and $c$ are located at the radial position of $R/R_2 = 1.05$. Sensors $d$, $e$, and $f$ are located at $R/R_2 = 1.23$, 1.50, and 1.91 respectively along the center line of the channel respectively. The sampling rate at each sensor is 39,060 Hz. The measured signals are amplified and digitized. All the sensors are statically calibrated. The steady pressure distributions in the diffuser were measured at 121 pressure taps of diameter 0.5 mm as shown in Fig. 5. The measured uncertainties are 0.51% in mass flow rate and 0.35% in static pressure ratio. A comparison of the time averaged pressure from fast-response sensor to the time averaged pressure from steady pressure sensor at the $d$ sensor at design flow condition is compared. The difference is 0.18% which is within the error bound of the sensor written above.

4. Performance and Pressure Distributions

4.1 Performance

Steady compressor performance was obtained for full range of flow rate from the choking to surging conditions using time average of static pressures at the channel exit and impeller exit. Figure 6 shows the
variations of static pressure ratio of the compressor system. Points AA and A are the choking condition, I is the design (normal) condition, and P and Q are the surge condition. The pressure variation at the diffuser exit shows a positive slope at the surge condition \((P - Q)\). The static pressure variation at the impeller exit shows a negative slope for all the corresponding flow rates. The pressure recovery through the diffuser shows its maximum value near the design condition \((I)\). It is interesting to note the small increases in the impeller exit pressure ratio in the choking flow rates (A and AA) which is related with normal shock in the diffuser. The diffuser exit pressure becomes larger than impeller exit pressure. The diffuser no longer works as diffuser but as a convergent-divergent nozzle.

4.2 Pressure distributions

Figure 7 shows the mean static pressure distributions. At the normal flow condition \((I)\) shown in Fig. 7 \((e)\), the pressure increases through the diffuser and the pressure recovery gradient is larger in the vaneless and semi-vaneless spaces than that through the channel. The flow stagnates at the vane leading edge, and a circumferentially non-uniform pressure distribution is observed in the vaneless space. The pressure is low at the mid-vane location and high upstream of the vanes location. For near surge flow rate \((O)\), the pressure recovery in the vaneless and semi-vaneless space is lower than that for the normal flow condition \((I)\) as shown in Fig. 7 \((f)\) and vane leading edge stagnation pressure strongly influence the pressure field in the vaneless and semi-vaneless space.

As the flow rate increases from I to E and onto B, the flow stagnates at the vane suction side in the semi-vaneless space, so high pressure zone is developed there. On the other hand, the flow separation at the vane leading edge, due to large incidence angle, increases aerodynamic blockage and reduces the aerodynamic flow area downstream. Thus the minimum pressure location occurs downstream of the geometric throat as marked on Fig. 7 \((c)\) and \((d)\). The minimum pressure location is named as the aerodynamic throat. The pressure at the aerodynamic throat decreases as the flow rate increases.

As the flow rates increase further (A and AA), the flow turns rapidly towards the throat, accelerates and becomes uniform over the throat. The flow continues to accelerate and the pressure rapidly decreases.

The location of aerodynamic throat is not fixed. Figure 8 shows the minimum pressure locations nondimensionalized by the width of the geometric throat. The distance from the geometric throat locations with the flow rate. Mach numbers at the minimum pressure location are approximated from the other measured parameters. For the case of B, the value of Mach number is slightly less than 1.0 at the minimum pressure location. However, the values become greater than 1.0 for the cases of A and AA. If the vaned diffuser acts as a convergent-divergent nozzle and shock occurs in the channel, then the location of unit Mach number is the aerodynamic throat. A shock should occur at the location of minimum pressure if the flow is supersonic. The minimum pressure is about 0.6 and Mach number is 1.3 for the choking case of AA. Therefore the minimum pressure locations at B, C, D, and E conditions are the aerodynamic throat, and the minimum pressure locations of A and AA conditions are shock location. The aerodynamic throat at A and AA conditions is located between the geometric throat and shock location. The pressure rapidly decreases from the geometric throat to the shock location and recovers downstream. The minimum pressure contour line extends to the suction side and the shock strength is not constant across the channel; weak on the pressure side and middle of channel, and strong on the suction side.

5. Decomposition of Instantaneous Pressure

5.1 Decomposition relations

Suryavamshi et al. (1997, 1998) suggested a method to decompose instantaneous pressure signals. Each discrete measurement of the pressure is nondimensionalized to have instantaneous pressure ratio.

\[ P_{\text{ratio}} = P_{\text{ratio}}/P_{\text{ratio}} \]

(1)

Here subscripts \(i\), \(j\) and \(k\) represent indices in the ensemble averaging process \((i\) indicates the index of revolution, \(j\) the index of the blade and \(k\) the index of the window in the blade passage\). The number of windows in the blade passage is 30 in the present
Fig. 7  Measured distributions of static pressure ratio in the diffuser for various flow rates

study. The instantaneous pressure ratio is decomposed into shaft resolved \((Pr_{sh})\) and unresolved \((Pr_{sh})\) components.

\[ Pr_{sh} = (Pr_{sh}) + Pr_{sh} \]  

where the shaft resolved component is obtained by ensemble averaging the instantaneous pressure ratio.
data set.

\[
\left( Pr_{ib} \right)_a = \frac{\sum \left( Pr_{ib} \right)/N_{rev}}{N_{data}} \quad (3)
\]

\[
\left( Pr_{ib} \right)_s = \frac{\sum \left( Pr_{ib} \right)/N_{rev,ib}}{N_{data}} \quad (3.a)
\]

\[
Pr_{ib} = Pr_{ib} - \left( Pr_{ib} \right)_a
\]

Several decades of data are usually measured between the blade to blade to obtain the resolved component. However the sampling rate is not enough to measure appropriate numbers of data needed for the decomposition of instantaneous pressure in high speed impeller. The blade passage frequency is 18,000 Hz for an impeller with 18 blades. If the data were measured at 39,010 Hz, then only 2 or 3 samples are obtained between two adjacent blades. Such value is not unusual for high speed turbo-machines. So, Eq.(3a) is used instead of Eq.(3) to avoid this difficulty. Every blade to blade space is divided into appropriate number of equally spaced windows (here 30) and each data measured in a sensor is assigned with revolution (i), blade (j) and window (k). Care must be taken in assigning i, j, and k exactly for a high speed compressor. The signal from the magnetic speed pick-up is used for accurate assigning. One revolution of impeller generates a sinusoidal induced current from the coil and the permanent magnet fixed on the turbine shaft. Each sinusoidal current is divided into 18 space (for blades) and a space is divided further into 30 windows (equally spaced) between every blade to blade space. Then, each measured data is assigned with revolution (i), blade (j) and window (k).

The shaft resolved component is further decomposed into time average \( Pr \), revolution periodic \( \left( Pr_{ib} \right)_{av} \), and revolution aperiodic \( \left( Pr_{ib} \right)_{as} \) components.

\[
\left( Pr_{ib} \right)_s = Pr + \left( Pr_{ib} \right)_{av} + \left( Pr_{ib} \right)_{as}
\]

where

\[
Pr = \sum_{\text{i=1}}^{N_{data}} Pr_{\text{i}}/N_{data}
\]

\[
\left( Pr_{ib} \right)_{av} = \left( \sum_{\text{i=1}}^{N_{data}} \left( Pr_{ib} \right)_{s} - Pr \right)/N_{data}
\]

\[
\left( Pr_{ib} \right)_{as} = \left( Pr_{ib} \right)_{s} - Pr - \left( Pr_{ib} \right)_{av}
\]

RMS values of the unsteady components are calculated as follows:

\[
\text{RMS}((Pr_{ib})) = \sqrt{\sum_{\text{i=1}}^{N_{data}} \sum_{\text{r=1}}^{N_{rev}} \left( Pr_{ib} \right)^2/(N_{r} \times N_{data})}
\]

\[
\text{RMS}((Pr_{ib})) = \sqrt{\sum_{\text{i=1}}^{N_{data}} \left( Pr_{ib} \right)^2/N_{data}}/Pr
\]

5.2 Decomposed result

Figure 9 shows the decomposition results of the instantaneous pressure ratio at the normal (i) and surge (Q) conditions measured from the d sensor near the throat. Figure 9(a) shows the variation of the instantaneous pressure ratio at the surge flow condition (Q), where large-scaled surge waves occur periodically. Variations of the ensemble average of the pressure ratio are shown in Fig.9(b) and the revolution periodic component of pressure ratio are shown in Fig.9(c). Their waveforms are regular not only for the normal condition but also for the surge condition. Figure 10(a) shows the pressure waveforms (revolution periodic) from the d sensor. The waveforms are sinusoidal for low flow rates; however, high frequency fluctuations appear as the flow rate increases. As the flow rate increases further, the peak forms becomes regular again for the choked flow rate.

The large-scaled surge waves with wavelength comparable to 1,500 - 1,700 times that of blade passing waves appear for the surge conditions P and Q, however the local variation shows the blade passing waveform. And also in the choking condition (AA), the revolution periodic component shows waveform by the blade passing.

Figure 10(b) shows the revolution periodic component from the a sensor in the vanesless space. The signals obtained in the vanesless space have waveforms with higher frequency components which are due to the strong interaction of highly non-uniform and turbulent impeller exit flow with the diffuser. The amplitudes of waveform variations are smaller than those in the channel. Quite distorted, but more regular waveforms are obtained from b and c sensors. The location of the a sensor is near the suction side, where the impeller and diffuser interaction is stronger. This result agrees well with the measured result of Justen et al. (1998). Figure 11 shows the waveform comparisc
sons of revolution periodic components from the six sensors at AA, I, and Q conditions. The waveforms of revolution periodic component from the sensors $d$, $e$ and $f$ in the diffuser channel also show similar variations. However the waveforms from the $a$, $b$ and $c$ sensors over the vaneless space are irregular and different each other. The difference of waveform from sensors in the vaneless space is due to strong impeller-vaned diffuser interaction.

5.3 Unsteadiness

Of some interest in the diffuser is the extent to which the unsteady flow in the diffuser entry zone propagates downstream. Inoue (1980), Krain (1984) concluded by measuring velocity fluctuations in the diffuser that flow unsteadiness at blade passing scale in the vaned diffuser was confined largely to the entry zone and diminished rapidly in the diffuser channel. And Dawes (1995) came to similar conclusion by unsteady numerical simulation. This is plausible conclusion. However, the measured result of Justen et al. (1998) was contrary to them. They measured unsteady pressure fluctuations in the diffuser and concluded that downstream in the diffuser channel the unsteadiness did not decay, even with an enlarged radial gap. He also showed that pressure fluctuations were appearing which could be distinctly higher than the pressure fluctuations in the vaneless space, depending on the operating point.

The contradictory results on the decay of unsteadiness in the diffuser seem to be solved by decomposing unsteadiness into two components: periodic unsteadiness and aperiodic unsteadiness. Blade scale periodic unsteadiness is defined as root mean square (rms) value of the revolution periodic compo-
The root mean square (rms) value of shaft unresolved component (Eq. (9)) and aperiodic unsteadiness is defined as rms value of shaft unresolved component (Eq. (10)).

Figure 12 shows the rms values of the revolution periodic component and the results are interesting. The values in the channel are generally higher than those in the vaneless space. The values do not necessarily decrease downstream of diffuser channel. And they show large values for both small and large flow rates. This results support that periodic unsteadiness does not decay in the diffuser channel, that it depends on the operating point, and that it is generally larger in the channel than in the vaneless space.

The rms value of shaft unresolved component which is a good index of aperiodic unsteadiness is shown in Fig. 13. It is higher in the vaneless space than in the channel and decreases downstream of diffuser channel in all flow conditions which means
that aperiodic unsteadiness rapidly decrease downstream of diffuser channel. In the diffuser channel, the rms value is relatively low near the normal flow condition, however, it increases at the off-design conditions due to large incidence angle in the diffuser vane leading edge. In the vaneless space, the rms values are not strongly dependent on the flow rate. However, they increase near the surge condition because of large-scaled surge waves. Under high flow conditions in the vaneless space, they increase due to the impeller-vaned diffuser interaction. At choke conditions (A and AA) in the vaneless space, they suddenly decrease since the flow accelerates. The variation of rms values of the d sensor with the flow rate is dramatic. The values near the normal flow conditions are lower than those in the vaneless space. However, they show rapid increases for the low and high flow rates, since the flows become complex due to large incidence angle.

6. Conclusion

Distributions of static pressure over the channel diffuser and the variations of unsteady pressure were measured for a full range of operating conditions in the present study. The non-uniformity of static pressure and unsteady characteristics were investigated. The results obtained through the study are as follows:

(1) A strong non-uniform pressure distribution is obtained over the diffuser wall due to the impeller and diffuser interaction. The location of minimum static pressure, so called aerodynamic throat, occurs downstream of the geometric throat. The pressure rapidly decreases from the geometric throat to the shock location and recovers along the downstream of the shock.

(2) The waveforms of the revolution periodic component of instantaneous pressure show regular oscillation not only for the normal but also for the surge conditions. The high frequency fluctuations are superposed on the oscillating waveform as the flow rate increases due to impeller-vane diffuser interaction.

(3) Periodic unsteadiness by blade passing does not decay in the diffuser channel. It depends on the operating point and is generally larger in the channel than in the vaneless space. However, the aperiodic unsteadiness rapidly decreases downstream of diffuser channel.

References


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