Clearance Effects on the Onset of Instability in a Centrifugal Compressor

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This report intends to shed an insight into the effect of large relative tip clearances on the onset of instability in a highly loaded centrifugal compressor. Time-resolved pressure measurements have been performed along the casing of a scaled-up model of a small compressor for two clearances at a wide range of operating conditions. Based on these time-resolved measurements, the pressure distribution along the meridional length and the blade loading distribution are calculated for each operating condition. In addition, the phase locked pressure fluctuation and its deviation are computed. The results show the behavior of each subcomponent of the compressor at different flow conditions and explain the role of the relative tip clearance on the onset of instability. For high mass-flow rates, the steady pressure distribution along the casing reveals that the inducer acts as an accelerating nozzle. Pressure is only built up in the radial part due to the centrifugal forces and in the subsequent diffuser due to area change. For off-design conditions, incidence effects are seen in the blade loading distribution at the leading edge while the inducer is unloaded. A region of high pressure deviation originates at the leading edge of the main blade and conveys downstream. This feature is interpreted as the trajectory of the leakage vortex. The trajectory of these vortices is strongly affected by the mass-flow coefficient. If the mass-flow rate is sufficiently small, the trajectory of the leakage vortex becomes perpendicular to the axis of rotation, the leakage vortex interacts with the adjacent blade, and inlet tip recirculation is triggered. If the flow rate is further reduced, the leakage vortex vanishes and rotating stall is initiated in the diffuser. For larger clearances, stronger vortices are formed, stall is triggered at higher flow rates, and the overall compressor performance deteriorates. [DOI: 10.1115/1.2776956]

Introduction

Small-scale centrifugal compressors are widely used in high volume applications such as automotive turbocharging or in distributed power applications. For these applications, typically unshrouded impellers with splitter blades are used. The blades are designed using ruled surfaces and show little rake or lean. The performance of such blading is hampered by the pressure loss over the tip gap and the formation of secondary flows. In addition, the effect of the tip clearance flows is very strong as the manufacturing accuracy determines a minimum clearance and thus the relative tip clearance ratio is large.

Only little work has been done in centrifugal compressors on the interaction of the tip clearance flow with the passage flow and its influence on the stability. Based on measurements in several centrifugal impellers, Senoo and Ishida [1] described the deterioration of the compressor performance due to tip clearance. For all compressors, an increased tip clearance width results in increased leakage ratios and higher losses. A general description of the flow interaction within a centrifugal compressor was presented by Eckardt [2]. In his work, Eckardt presented a schematic model of the secondary flow pattern, which included effects due to tip clearance flows. Hong et al. [3] showed experimental results on the influence of the relative tip clearance on the flow structure and efficiency in radial compressors. A complete and generalized overview on the stability of compressing systems is given by Denton [4]. The stability of the present compressor system was discussed based on the information gained at the inlet of the diffuser (Schleer et al. [5,6]). It was found that for increased tip clearance ratio, stall occurs at higher flow coefficients while the stage pressure ratio was substantially reduced. In an accompanying paper by the authors [7], the influence of the tip clearance ratio on the evolution and mixing of the flow in the diffuser is studied in detail.

In the current work, the influence of the relative tip clearance width on the onset of instability is studied by analyzing time-resolved measurements of the static pressures along the casing for a wide range of flow coefficients. A dense array of flush-mounted pressure sensors is used along the flow path to measure the static pressure fluctuation and its deviation in order to detect clearance vortices and stall precursors. The method is similar to the method used by Yoon et al. [8] in axial compressors. Based on these measurements, the blade loading distribution and its sensitivity to clearance ratio and operating condition are examined. Further, the sequence of events leading to rotating stall in the diffuser is investigated. Measurements at different clearance ratios show a relation between the strength of the interaction of the leakage jet and the passage flow and the compressor performance.

Experimental Setup and Data Evaluation

The experimental investigations have been performed on the “Rigi” research facility in the Turbomachinery Laboratory of the Swiss Federal Institute of Technology in Zurich, Switzerland. The compressor facility is a scaled-up model of a one-stage centrifugal compressor used in small-scale distributed power generation or automotive turbocharging. It was designed to match the main design criteria and nondimensional parameters typical for these small devices and a comparable stability and operational behavior were achieved.
The Rigi facility consists of a single stage, vaneless centrifugal compressor system operating in a closed loop, as shown in Fig. 1. It is designed for air and delivers a design volume flow rate of $\dot{V}=3.5$ m$^3$/s at a design pressure ratio of $\pi=2.8$. The centrifugal impeller has an outer diameter $D_2=400$ mm and is designed with seven full and splitter blades. At impeller exit, the blades are swept 30 deg backward from the radial directions while the rake and lean angle are nearly 0 deg, respectively. The rotor is followed by a parallel vaneless diffuser with an exit diameter of $D_3=580$ mm. A flow straightener mounted in the suction pipe ensures axial flow at the stage inlet. A large toroidal collecting chamber follows the radial diffuser. This arrangement ensures a virtually uniform circumferential pressure distribution at the impeller exit under all through-flow conditions.

A 440 kW dc motor coupled to a two-stage gearbox drives the facility. As the facility is a closed loop type, the compressed air has to be cooled by a counterflow air-water heat exchanger. The mass flow is measured in the backflow duct with a standard orifice assuming steady state. The system inlet pressure can be varied between 16 kPa and 125 kPa. The impeller load can be varied with a throttle downstream of the cooler. The pressure is reduced back to inlet conditions by the throttle and controlled by a pressure regulation system. The closed loop arrangement facilitates measurements under repeatable and constant conditions without depending on the atmospheric conditions.

Experiments have been conducted at two different relative clearances. The base line tip clearance ratio at impeller exit was set to a value of $CR=12.7\%$. This very large tip clearance ratio is common for small-scale applications as the axial displacement in the bearing is large compared to the diffuser width. In the Rigi facility, it is possible to reduce the tip clearance width to 0.7 mm by changing shim plates between the inlet assembly and the casing. The reduced gap configuration corresponds to a clearance ratio of $CR=4.5\%$ and is equivalent to a well-designed industrial scale compressor. The actual tip gap configuration is sketched for both cases in Fig. 2.

In Fig. 3, a comparison of the operating maps and stability behavior of the compressor is shown for the base line and reduced clearance cases. The stability of the compressor system is determined by investigating the pressure fluctuation in the inlet of the diffuser (Schleer and Abhari [6]).

The operating mode of the compressor is assessed by analyzing the frequency content of a pressure reading using a fast Fourier transformation (FFT). To determine if stall or surge are present, the relative magnitude of the dominant instability frequency is compared to the magnitude related to the blade passing frequency. Rotating stall shows a rather wide and modulated peak at more than one frequency in the FFT spectrum. The frequency of rotating stall depends on the rotational speed of the shaft, the number of stall cells, the direction, and the speed of propagation. Surge is seen in the FFT as a distinct frequency, which depends only on the geometry of the compression system and the arrangement of the piping and is thus not affected by impeller speed. Classic surge is seen in the FFT as a coupled mode where the rotating stall signature is superimposed and modulated by the surge frequency.

In the comparison, an improved characteristic is seen for the reduced clearance case. The pressure ratio is improved by 6% for the same impeller speed. The choke and the system resistance limit remain unchanged. Toward low mass-flow rates, the stability limit is not affected for high impeller speeds. For part speed, an improvement is found for the reduced tip clearance case. At high impeller speed, the surge margin remains unchanged as surge depends mainly on the overall setup of the compression system and is not affected by changes of the local flow pattern at impeller exit due to tip gap variations. The gain in stability margin for the reduced clearance case at part speed is due to a shift of the onset of rotating stall inception toward lower mass flows. It is postulated that flow features, which depend on the tip clearance, affect the onset of rotating stall. In this work, the detailed measurements show the mechanism how the onset of rotating stall is affected by the tip clearance flow.

### Investigated Operating Conditions

To understand the influence of the relative tip clearance on the onset of stability, an experimental investigation was performed with two relative tip clearances for a rotational speed of $\mu=0.6$ (9949 rpm). In Fig. 4, the investigated operating conditions are shown in terms of isentropic head and flow coefficients for both tip clearance ratios.

The letters in Fig. 4 mark the operating conditions where detailed pressure measurements have been obtained by a combination of static wall pressure taps and time-resolved high frequency response pressure plugs along the casing. The onset of instability is marked for each tip clearance case by a black line. The shift of
the instability line toward lower mass flow described above is also seen in Fig. 4. With increasing tip clearance, pressure rise is reduced and stall inception is shifted toward higher mass flows. This shift of the stall limit at different clearances agrees with previous findings in axial and radial compressors.

Data Acquisition

The unsteady pressure fluctuations have been measured using flush-mounted pressure sensors with a high temporal resolution. The time-resolved pressure plugs were developed in-house and successfully applied in the present study. The inserts are assembled using a 1 bar differential piezoresistive pressure transducer. The reference pressure side of the transducer can be connected to static pressure taps along the flow path. The signals are amplified with an amplifier with a cutoff frequency of 25 kHz and are sampled at 100 kHz using a multichannel analog-to-digital (AD) data acquisition system. The complete measurement chain is calibrated for offset and gain before the measurements and is estimated to be accurate within 100 Pa.

A total of 30 high frequency response sensors is located in the inlet pipe, along the shroud wall, and the diffuser. Figure 5 shows the locations of the pressure sensors. For each measurement location, a time series of 300,000 pressure readings has been obtained at a sampling rate of 100 kHz. The steady reference pressure for each sensor and the steady pressure along the casing are acquired with a multichannel pressure scanner system at 32 locations along the meridional flow path. The accuracy of the PSI 9016 pressure scanner is given with ±0.05% of the full scale range of 1 bar.

Using this experimental setup, the time history of the static wall pressure can be reconstructed as the sum of the steady pressure $\bar{p}$, the periodic pressure fluctuation $p(t)$, and the random pressure $p'$.\[ p(t) = \bar{p} + p(t) + p'(t) \] (1)

The steady pressure $\bar{p}$ is acquired by the steady pressure measurement system. The time related pressure fluctuation $p(t) + p'(t)$ is given by the time-resolved measurements. By performing an ensemble average on the time-resolved pressure reading, the periodic fluctuation $p(t)$ due to the blade passing can be separated from the random part $p'(t)$. The ensemble averaging is performed based on the impeller position delivered by a pickup on the shaft, and all seven full-blade channels are treated similarly. By calculating for each sensor the phase locked periodic pressure fluctuation $\bar{p}(t)$ relative to the position of the blades, the blade loading distribution throughout the flow path is obtained. One full-blade passage is discretized into 103 equal parts representing an angular discretization of 0.5 deg. In each of these discrete angular positions or so-called “boxes,” more than 2500 samples are acquired and averaged to obtain a good statistical accuracy.

To allow for a better comparability of different measurements, the pressure values are normalized with the inlet static pressure $p_0$ and the normalized phase locked pressure coefficient is obtained:

\[ \bar{C}_p(box) = \frac{\bar{p}(box)}{p_0} \] (2)

The randomness of the pressure fluctuation $p'(t)$ is described by the root-mean-square (rms) or standard deviation of the pressure samples in each individual angular box. The phase locked deviation is computed as

\[ \sigma_{p(box)} = \sqrt{\frac{1}{N} \sum_{i=1}^{N} (p_i(box) - \bar{p} - \bar{p}(box))^2} \] (3)

where $N$ is the number of samples and $p_i(box)$ are the individual pressure values. The local value of the pressure deviation is interpreted as the level of unsteadiness of the flow due to vortices and separation at this location. It can be used to resolve the trajectory of tip vortices or the extent of separated zones.

Steady Pressure Distribution

In Fig. 6, the steady wall pressure distribution along the meridional length is plotted for the base line tip clearance case “STR
17.2. The measured pressure has been normalized using the inlet static pressure. Each individual line in the plots represents a different mass-flow condition between the choke and the stall limit. Stalled conditions are displayed in magenta, while stable conditions are plotted in blue.

The overall pressure rise in the machine is mostly dependent on the pressure rise within the radial section of the impeller. A similar pressure rise in the radial section of the impeller and the diffuser is observed for all mass-flow conditions. The slope of the pressure rise in the diffuser is nearly independent of the flow rate. For high mass-flow conditions, a decompression takes place in the inducer section of the impeller before pressure is built up in the radial section and the diffuser. At these conditions, the inducer behaves as an accelerating nozzle providing the inflow condition into the radial section. Unlike axial compressors, the pressure rise in the radial section is mostly due to the centrifuge effect instead of turning by the blades. The pressure rise in the diffuser is mainly due to diffusion as a consequence of the increase in flow area. Toward stall, no decompression is observed in the inducer and the pressure is monotonously rising throughout the machine. For unstable operating conditions, a deterioration of the diffuser performance is seen. The deterioration occurs due to the increased blockage in the diffuser by the rotating stall pattern. For stalled operating conditions, a pressure rise in front of the leading edge is encountered. This pressure rise is a sign that energy has been imparted into the fluid. Since this is seen before the impeller leading edge, it indicates the occurrence of a local flow recirculation, which transports energy upstream of the impeller. Gyarmathy et al. [9] concluded from their similar measurements that a ring-shaped separation bubble exits at the impeller leading edge in which fluid is driven around the annulus at blade speed. In this separated zone, energized fluid is able to migrate upstream and is seen as raised pressure in front of the impeller.

**Blade Passing Related Periodic Pressure Fluctuation**

The normalized local periodic pressure fluctuation \( \tilde{C}_{p_{(box)}} \) caused by the blades and the deviation \( \tilde{C}_{\delta(\text{box})} \) of the periodic pressure fluctuation are calculated for each sensor along the flow path. The results are plotted in the representation shown in Fig. 7. Displayed is an unwrapped development of the casing surface for one blade pitch. The arclength and the meridional length are plotted on the ordinate and abscissa, respectively. The inlet, impeller, and diffuser sections are labeled. The positions of the full and splitter blade tips are marked, and the dotted lines give the positions of the individual pressure sensors. The normalized phase locked pressure coefficient \( \tilde{C}_{p_{(box)}} \) and the phase locked deviation \( \tilde{C}_{\delta(\text{box})} \) are interpolated along the blade line and are plotted as a color surface on top of this representation.

![Fig. 6 Measured steady wall pressure distribution for the base line clearance ratio (CR=12.7%), normalized by the inlet static pressure](image6)

In Fig. 8, the phase locked pressure coefficient \( \tilde{C}_{p} \) is shown at four mass-flow conditions for the base line tip clearance case STR 17.2. The color scale is identical for all operating conditions and the specific flow coefficient is labeled in each subplot. Positive values of the phase locked pressure coefficient are seen on the pressure side, while negative values are found on the suction side. The difference across the blade can be interpreted as blade loading. The integral value of the blade loading is related to the overall power consumption of the compressor. As the mass-flow rate is much higher for the case near choke (a), the power consumption is also higher even though the pressure rise is lower. Therefore, despite the lower pressure ratio, the highest variations of the pres...
ure coefficient \( \bar{C}_p \) are seen near choke conditions while the smallest variation of the pressure coefficient is seen for the operating condition near stall \((g)\).

In the high mass-flow cases \((a)\) and \((b)\), the pressure difference across the blades is maintained in the diffuser. These pressure traces can be interpreted as the trajectory of the jet-wake flow leaving the impeller. If the mass-flow rate is throttled, the jet-wake pattern becomes weaker and the trajectory moves toward the tangential direction. At near stall condition \((g)\), the flow in the diffuser is disturbed and no clear jet-wake pattern can be identified in the diffuser. At all mass-flow rates, the highest variation of the phase locked pressure coefficient \( \bar{C}_p \) is seen at about 70% of the meridional length in the radial section of the impeller. For the near stall condition \((g)\), the axial inducer section is barely loaded while in the design incidence condition \((b)\), a more even distribution of the blade loading along the meridional blade length is obtained. The unloading in the inducer section is consistent with the inlet tip recirculation postulated above for low mass-flow conditions. In condition \((d)\), a locally high loading is seen at the main blade leading edge. This locally high loading at the leading edge is caused by positive incidence at part load.

A more quantitative description of the flow can be given if the blade loading along the meridional length of the blade is calculated. The blade loading is determined for each meridional location as the difference between the periodic pressure fluctuation value at the pressure and suction sides. The local pressure values are interpolated at circumferential positions, which are shifted by 3 deg from the blade chamber line. These locations for the interpolation of the blade loading value are marked in the left subplot of Fig. 9 by thin lines beside the positions of the blades. The resultant blade loading coefficient

\[
\bar{C}_p = \frac{\bar{p}}{p_1}
\]

is plotted in the right subplot. The loading on the full blade is shown by solid lines while the loading of the splitter blade is plotted using dashed lines.

In Fig. 10, the normalized blade loading distribution is plotted for the base line clearance case \((CR=12.7\%)\) at four operating conditions. The small perturbations of the curves are due to the limited meridional resolution of the pressure measurements as only 30 sensors have been placed along the flow path.

For all conditions, the blade loading is the highest at 70–80% of the meridional length, which agrees with the location of the highest loading calculated in the design phase. Due to the higher mass-flow rate, the absolute blade loading is elevated at near choke condition \((a)\). In the design incidence case \((b)\), a steady distribution of the blade loading is seen. In the case near choke, a negative loading is seen as a result of the negative incidence at the leading edge of the full blade. The opposite trend is seen in case \((d)\) where due to positive incidence, a locally high loading is seen at the leading edge. In the case near stall, only little loading is seen due to the recirculation pattern and no incidence effects can be identified in the inducer section.

**Random Pressure Fluctuation**

More insight into the stability behavior and the sequence of the events is gained if the deviation of the phase locked pressure fluctuation is investigated. The deviation is calculated from the periodic pressure fluctuation using Eq. (3). High values of the pressure deviation indicate regions where the temporal fluctuations for the phase locked pressure value are large and thus indicate regions of high unsteadiness. For example, high unsteadiness can result if a vortex is shed at a frequency different from the blade passing frequency or if a stall cell is present in the diffuser. In Fig. 11, the plots of the deviation are given for eight mass-flow conditions at a clearance ratio of \(CR=12.7\%\) (STR 17.2). Subplot \((a)\) is acquired at a mass-flow rate close to the system resistance line, while subplot \((h)\) represents a condition within rotating stall. As a dominant feature, a strong pressure deviation is found at the location of the blades. The highest values of the phase locked deviation are found for the highest mass flow, as in this condition, the pressure difference between the pressure and the suction sides is the largest (Fig. 8). For lower mass-flow coefficients, the pressure deviation across the blades in the inducer section is diminished.

The clearance flows are seen as high levels of flow unsteadiness beside the blades. Toward low flow coefficients, the pattern is dominated by the unsteady phenomena such as rotating stall and inlet tip recirculation. In subplots \((a)-(e)\) of Fig. 11, a pattern with high levels of random pressure deviation is seen departing at the leading edge of the main blade. This pattern of high pressure deviation is marked by a dashed line and is interpreted as the trajectory of the leakage vortex. The vortex is caused by the interaction and collision of the tip clearance jet and the passage flow. Under the action of the pressure difference across the blade, a leakage jet is formed and penetrates into the passage. After a certain distance, the leakage jet is stopped by the main flow, which separates the jet from the wall, turns it backward, and leads to the formation of the tip leakage vortex (Song and Martinez-Sanchez [10]). The leakage vortex trajectory starts in all cases at the leading edge tip of the full blade and convects with the flow. For smaller mass-flow rates, the leakage vortex trajectory moves...
away from the suction side and migrates toward the adjacent pressure side. A similar behavior of the leakage vortex is observed in axial compressors. In axial compressors, Inoue et al. [11] and Yoon et al. [8] found a strengthening and upward movement of the leakage vortex at smaller mass-flow coefficients. This was also postulated theoretically by Chen et al. [12] who linked the upward migration of the vortex at decreasing mass-flow rates with a strengthened leakage jet and longer convection times.

At design incidence, the leakage vortex of the previous full blade interacts with the splitter blade leading edge resulting in a lift-off of the splitter leakage vortex. If the flow is throttled further, the leakage vortex hits the leading edge of the adjacent blade (Figs. 11(c) and 11(f)) and a strong increase of the pressure deviation in the inlet region of the compressor is seen.

This sudden increase of the pressure deviation at the inlet occurs at the mass-flow condition where the steady wall pressure distribution shows an inlet tip recirculation zone. From this coincidence, it is concluded that inlet tip recirculation is triggered as soon as the leakage vortex trajectory becomes perpendicular to the machine axis and approaches the leading edge of the adjacent blade. Due to the inlet tip recirculation, the overall level of pressure unsteadiness increases in the entire impeller passage. If the compressor is throttled further to a flow coefficient below phi =0.022, rotating stall is initiated in the vaneless diffuser (Figs. 11(g) and 11(h)). As a result of the rotating pattern, large pressure deviations are recorded in the diffuser section. During rotating stall, the inlet tip recirculation zone recovers and low levels of pressure deviations are seen in the inlet. Thus, rotating stall becomes the dominant feature and suppresses the tip recirculation in the inducer of the impeller.

Near the suction side of the splitter blade, a strong feature can be identified in subplots (A)–(D) of Fig. 11. The feature is the strongest and closest to the suction side for the highest flow coefficient and weakens at smaller mass-flow rates. For smaller mass-flow rates, it lifts off the blade surface while at choked condition, it appears directly at the suction side. The feature could not be observed for operating conditions in the unstable regime (subplots (F)–(H)). This structure is associated with the leakage vortex forming at the leading edge of the splitter blade. In the measurements, a strong, but localized, interaction is seen at the splitter leading edge. However, the downstream convection of the splitter blade leakage vortex is not clear. This is a result of the different interaction mechanisms of the splitter blade leakage vortex. In the case of the main blade, the leakage vortex interacts with the non-rotational freestream approaching the compressor. Behind the splitter blade, the vortex interacts with a skewed and rotating channel flow. As no downstream convection of the leakage vortex is seen, it is postulated that due to the interaction with the skewed channel flow, the leakage vortex is entrained in the channel and lifted off the casing. This hypothesis is supported by the time-resolved flow measurements presented in an accompanying paper by the authors [7] where the area, which is affected by the clearance flows across the splitter blade, shifts toward the midheight of the diffuser channel.
Fig. 11 Deviation of the measured periodic pressure fluctuation for the base line clearance case (CR=12.7%) at different mass-flow rates
Effect of Reduced Clearance on Steady Pressure Distribution and Blade Loading

In Fig. 12, the normalized steady wall pressure along the meridional length is plotted for the base line tip clearance case CR = 12.7% and the reduced clearance ratio CR = 4.5%. The measured pressures are normalized with the inlet static pressure. For each configuration, three mass-flow rates between choke and the stall limit have been plotted. The measurements in the base line configuration are displayed using a solid line, while the reduced clearance case is plotted using a dashed line. For all mass-flow rates, no differences in the steady wall pressure distributions are seen for the inlet and the inducer section. In the radial part of the impeller, a significantly better performance is seen for the reduced clearance case. The improved performance persists throughout the diffuser and is seen as an improved pressure ratio at the stage exit (Fig. 3).

In Fig. 13, the normalized blade loading distributions on the main blade and the splitter blade are plotted for the reduced clearance case CR = 4.5%. The mass-flow rates are the same as shown in Fig. 10 for the base line case.

At similar mass-flow rates, the blade loading is higher for the reduced clearance case. This is consistent with the increased pressure rise for the reduced clearance case seen in Figs. 3 and 12. As in the base line clearance case, a slightly smaller loading is seen on the splitter blade for small mass-flow rates while toward choke, a high loading of the splitter leading edge is found. The same incidence effect is seen for both clearance cases. Due to the positive incidence in the midrange condition (d), a high loading is seen on the main blade leading edge while near choke (a), the negative incidence causes a negative loading. Also, for the condition near stall (b), a very low loading is seen for both clearance cases in the inducer section while the radial section is still loaded.

Comparison of the Pressure Deviation at Base Line and Reduced Clearance

In Fig. 14, the deviation of the periodic pressure is plotted for the reduced clearance case. In all stable operating conditions, a comparison with Fig. 11 reveals generally lower levels of pressure deviations for the reduced clearance case. Especially in the radial
Fig. 14 Deviation of the measured periodic pressure fluctuation for the reduced clearance case (CR=4.5%) at different mass-flow rates
section, where the tip clearance has been changed the most by the shim, the deviation of the phase locked pressure fluctuations is strongly elevated for the larger clearance case. This indicates a stronger interaction and disturbance of the passage flow by the tip clearance flows for the wide clearance case. This increased flow interaction would lead to increased mixing losses and a deteriorated compressor performance.

In the inlet region, a similar migration and trajectory of the leakage vortex are found for both clearance cases. For reduced tip clearance, the leakage vortex core appears weaker than in the base line configuration. A similar upstream movement of the main blade leakage vortex trajectory is found for both clearance cases. Inlet tip recirculation is provoked for both clearance cases as soon as the leakage vortex becomes perpendicular and approaches the adjacent blade leading edge (Fig. 14(e)). The splitter blade leakage vortex behaves similarly in both cases but is weaker for the reduced clearance case. A similar lift-off from the splitter blade suction side is observed for positive incidence conditions. Differences are seen in the operating condition at phi=0.022 (Fig. 14(f)) where rotating stall in the diffuser is present for the larger base line clearance case while for the reduced clearance configuration, only little deviations are found in the diffuser. This complies with the stability analysis shown in Fig. 4. A shift of the stall inception point toward lower mass-flow rates was found for the reduced clearance case. In the condition at phi=0.020 (Fig. 14(h)), rotating stall is present for both clearances and no differences in the stability behavior are found. For the enlarged tip clearance, stronger interactions of the leakage and passage flows are concluded. This complies with the observed reduction in stall margin and pressure rise for the increased tip clearance configuration.

Conclusion

The influence of the relative tip clearance width on the onset of instability has been studied by analyzing time-resolved measurements of the static pressures along the shroud casing of a centrifugal compressor. The distribution of steady pressure along the casing shows that for high mass-flow rates, the inducer acts as an accelerating nozzle, which sets the in-flow condition for the radial section. In the radial section, pressure is built up due to centrifugal forces. In the subsequent diffuser, the pressure rise is due to area change and recovery of the dynamic pressure.

Based on the experimental results, the mechanism leading to rotating stall and deterioration of stage performance can be summarized. For highest mass-flow condition, and evenly distributed blade loading dominates the flow pattern. Highest values are found as predicted by the design calculations at around 75% meridional length of the impeller. At off-design conditions, the effects of incidence are seen as high loading at the full-blade leading edge. In the deviation of the phase locked pressure fluctuation, the trajectory of the clearance vortex could be identified on the main blade. This clearance vortex is caused by the interaction of the leakage jet and the passage flow. The passage flow stops the leakage jet, separates it from the wall, and turns it backward. This leads to the formation of the tip leakage vortex next to the suction side of the main blade. On the splitter blade, an immediate interaction between the leakage flow and the skewed channel flow is seen but no trajectory of the leakage vortex could be identified.

Measurements at different clearance ratios reveal a close relation between the strength of the leakage jet and the compressor performance. Even though no significant differences have been found in the blade loading distribution for large and reduced relative tip clearance, a significant deterioration of the steady pressure rise is seen in the radial section. For larger clearance, the level of the random pressure deviations reveals higher flow disturbances. The increased levels of pressure disturbance are caused by the increased intensity of the secondary flows and result in increased losses.

The sequence of events leading to rotating stall in the diffuser is seen in the measured random pressure fluctuation. For high mass-flow rates, a leakage vortex is formed as a result of the interaction between the flow through the clearance space and the passage flow. Starting on the leading edge, the leakage vortex convects with the flow and migrates toward the adjacent pressure side. Similar to axial compressors, the trajectory of the tip leakage vortex is related to the mass-flow coefficient. If the flow coefficient is reduced, the apparent stagger angle of the leakage vortex increases and the leakage vortex trajectory moves upstream. If the mass-flow rate is sufficiently small, the leakage vortex trajectory becomes perpendicular to the axis of the machine and interacts with the adjacent leading edge. A stability analysis revealed that at this condition where the leakage vortex approaches the adjacent blade, inlet tip recirculation is triggered. This leads to a sudden increase of random pressure fluctuation in the inducer section and a change in the loading distribution. If the flow is throttled further, rotating stall is initiated in the diffuser while the inlet recovers. For increased tip clearance, the onset of rotating stall is shifted to higher mass flows while the onset of tip recirculation is not affected by a change in tip clearance width.

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Nomenclature

- \( b \) = axial diffuser width (m)
- \( D \) = diameter (m)
- \( L_m \) = meridional length (m)
- \( p \) = pressure (kPa)
- \( t \) = tip gap width (m)
- \( U \) = circumferential speed (m/s)
- \( \dot{V} \) = volume flow rate (m\(^3\)/s)
- \( \rho \) = density (kg/m\(^3\))
- \( \sigma \) = pressure deviation (kPa)
- \( \Delta h_{ts} \) = total-static enthalpy rise (kJ/kg)

Dimensionless Numbers

- \( \phi = \frac{V_0}{D_2^2 U_2} \) = specific flow coefficient (phi)
- \( \psi = \frac{\Delta h_{ts}}{U} \) = isentropic head coefficient
- \( \text{Mu} \) = Impeller Mach number
- \( \text{C}_p \) = total-static pressure ratio
- \( \text{CR} = rac{1}{b} \) = tip clearance ratio

Subscripts

- 0 = stage inlet
- 1 = impeller inlet
- 2 = impeller outlet
- 3 = diffuser outlet

References


