Investigation on Inlet Recirculation Characteristics of Double Suction Centrifugal Compressor with Unsymmetrical Inlet Postprint

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Abstract

This paper focuses on the flow characteristic and local resistance of non-Newtonian power law fluid in a curved 90 bend pipe with circular cross-sections, which are widely used in industrial applications. By employing numerical simulation and theoretical analysis the properties of the flow and local resistance of power law fluid under different working conditions are obtained. To explore the change rule the experiment is carried out by changing the Reynolds number, the wall roughness and different diameter ratio of elbow pipe. The variation of the local resistance coefficient with the Reynolds number, the diameter ratio and the wall roughness is presented comprehensively in the paper. The results show that the local resistance force coefficient hardly changes with Reynolds number of the power law fluid; the wall roughness has a significant impact on the local resistance coefficient. As the pipe wall roughness increasing, the coefficient of local resistance force will increase. The main reason of the influence of the roughness on the local resistance coefficient is the increase of the eddy current region in the power law fluid flow, which increases the kinetic energy dissipation of the main flow. This paper provides theoretical and numerical methods to understand the local resistance property of non-Newtonian power law fluid in elbow pipes.

Full Text

Investigation on Inlet Recirculation Characteristics of Double Suction Centrifugal Compressor with Unsymmetrical Inlet

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Abstract

The inlet recirculation characteristics of a double suction centrifugal compressor with unsymmetrical inlet structures were investigated using numerical methods, focusing on three main issues: the magnitude and differences of inlet recirculation under various operating conditions, the circumferential non-uniform distributions of inlet recirculation, and the recirculation velocity distributions upstream of the rear impeller slot. The results demonstrate significant differences between the recirculation patterns of the front and rear impellers across all operating conditions. At design speed, the recirculation flow rate of the rear impeller exceeds that of the front impeller in the large flow range, while the opposite occurs in the small flow range. Under different operating conditions, the recirculation velocity distributions of both front and rear impellers exhibit circumferential non-uniformity with substantially different extents. The circumferential non-uniformity of recirculation velocity varies with operating condition changes. For the front impeller, the circumferential non-uniform extent and distribution of recirculation velocity are determined by the static pressure distribution of the front impeller itself. For the rear impeller, these characteristics are governed by the coupled effects of inlet flow distortion, the circumferential unsymmetrical distribution of the upstream slot, and the asymmetric structure of the volute. Under design flow and small flow conditions, the recirculation velocities at different circumferential positions along the mean line of the rear impeller's upstream slot cross-section show considerable variation, with different distribution patterns on either side of the mean line. The recirculation velocity distributions in the upstream slot cross-section depend on the static pressure distributions within the intake duct.

Keywords: asymmetric intake, double suction centrifugal compressor, inlet recirculation, recirculation flow, circumferential distribution

Introduction

Compared with conventional compressors, the double suction centrifugal compressor offers advantages of a wide flow range and relatively compact size [1]. Turbochargers equipped with double suction centrifugal compressors can improve the matching between the compressor and turbine, enabling the turbine to operate closer to its optimal speed ratio. This represents an effective method for further improving turbocharger performance. Previous studies have demonstrated that turbochargers with double suction centrifugal compressors achieve better matching with engines, highlighting the advantages of this configuration [2].

The double suction centrifugal compressor typically adopts a back-to-back structure with a common back plate separating both sides of the impeller. In turbocharger applications, the impeller side away from the intermediate section uses a straight inlet duct to ensure axial intake; this side is designated the front impeller. The opposite side, adjacent to the intermediate section, features intake along the radial direction through a complex-shaped bent duct, designated the rear impeller. Due to this bent inlet duct, flow losses are inevitable and a distorted flow field forms at the rear impeller inlet. These different inlet flow fields on both sides create distinctive operating characteristics for the double suction centrifugal compressor. Studies have found that the front impeller flow rate consistently exceeds that of the rear impeller throughout the stable operating range. As total flow rate decreases, the flow rate difference between front and rear impellers increases. When the compressor total flow rate reaches a certain threshold, extensive flow separation appears in the rear impeller passages, reducing the rear impeller flow rate to very low values. At this point, the front-to-rear impeller flow ratio exceeds 2, the rear impeller ceases to impart work to the airflow, and enters an inert state. Consequently, the double suction compressor operating mode transitions from parallel operation of both impeller sides to single-side operation of the front impeller alone [3, 4].

The recirculation device is an effective method for broadening the stable operating range of centrifugal compressors and is widely employed in such applications. This device is typically arranged on the casing wall, consisting of an upstream slot, a bleed slot, and an annular cavity connecting both slots. The slot located upstream of the impeller is called the upstream slot, while the slot positioned downstream of the blade leading edge is termed the bleed slot. Under large flow conditions, the pressure in the bleed slot is lower than the external pressure, causing air to flow through the recirculation slot and cavity into the blade passage. The resulting increase in compressor flow rate depends on the recirculation flow from the bleed slot, effectively increasing the impeller inlet flow area [5]. Under small flow conditions, the pressure in the bleed slot exceeds the external pressure, causing airflow to exit the blade passage through the bleed slot, flow through the recirculation cavity and upstream slot, and create a recirculation region at the impeller inlet. This increases the actual flow rate at the blade leading edge, reduces the incidence angle at the inlet tip region, and significantly decreases flow fluctuation amplitude near stall conditions [6, 7].

Previous research on recirculation devices includes Harley et al. [8], who improved the mean-line model of inlet recirculation by considering its direct effects, enabling more accurate simulation of centrifugal compressor performance with inlet recirculation. Tamaki [9] investigated the effects of upstream and bleed slots on blade passage flow fields, demonstrating that the bleed slot reduces blade loading at the impeller tip region and decreases tip leakage flow compared to configurations without recirculation devices. Additionally, the bleed slot alters the position of shocks downstream of the bleed slot. Chen [10] and Tamaki [11] studied performance changes in centrifugal compressors with guide vanes installed in the recirculation cavity. Guide vanes can remove pre-swirl from the port flow to minimize mixing losses with the main flow, and a properly designed guide vane can further reduce the surge mass flow rate. Tamaki et al. [12] used a one-dimensional flow method to study the effects of inlet recirculation on turbocharger centrifugal compressor performance, proposing a method using small fins in the compressor inlet duct to control flow in the recirculation zone near the blade leading edge tip under small flow conditions. Experimental results indicated this method can further reduce surge flow rate at low compressor speeds. Hunziker et al. [13] conducted optimization design studies on inlet recirculation geometry structures. Hu et al. [14] designed a double bleed slot system where different recirculation slots close depending on operating conditions, effectively solving the efficiency drop caused by inlet recirculation. Gancedo [15] measured the flow field near the inlet recirculation of a centrifugal compressor using PIV, showing that flow field structures are similar with and without recirculation devices under large flow and design flow conditions, but differ significantly near stall conditions. These studies primarily focused on the mechanism of inlet recirculation for range extension, flow fields in compressors with recirculation devices, geometric determination of recirculation devices, and mathematical performance calculation models incorporating recirculation effects.

The flow field inside centrifugal compressors is asymmetric along the circumferential direction, with different flow rates in each blade passage, meaning each passage operates under different conditions. This asymmetric flow field also creates non-uniform circumferential distributions of recirculation flow. Tamaki et al. [16] proposed a bleed slot with non-axisymmetric circumferential arrangement, with experimental results showing that compressor characteristics vary with changes in the circumferential non-axisymmetric distribution of the bleed slot, exhibiting a decreasing trend in stall flow rates. The asymmetric flow field in centrifugal compressors is caused by volute structure. Studies on both conventional and double suction centrifugal compressors have found that static pressure distribution in the volute is circumferentially asymmetric. Under small flow conditions, static pressure increases from the volute tongue to the outlet along the streamline direction, with the volute acting similarly to a diffuser. Under large flow conditions, this static pressure distribution is reversed and the volute acts similarly to a nozzle [17-20]. The circumferential static pressure gradient is maximum under large flow conditions and decreases with reducing flow rate. The circumferential static pressure distribution in the diffuser follows the same pattern as in the volute. For a given flow condition, the non-uniform extent of static pressure distributions at different radii in the diffuser is essentially the same [20].

For the rear impeller of a double suction centrifugal compressor, the internal non-axisymmetric flow field is affected by inlet distortion in addition to volute geometry effects. Therefore, compared with conventional centrifugal compressors, the non-axisymmetric flow field in the rear impeller forms under the coupled effects of inlet distortion and the non-axisymmetric volute structure [20]. Additionally, flow distortion caused by the rear impeller inlet bent duct increases flow losses in the diffuser. Reference [21] studied flow losses in a double suction centrifugal impeller, finding that additional flow losses are produced in the diffuser due to total pressure differences between the front and rear impeller inlets, with these losses increasing as the total pressure difference grows. Under large flow conditions, the primary diffuser flow loss comes from flow mixing between both impeller sides. Under small flow conditions, separation vortices at the diffuser wall surface intensify, with separation vortex losses significantly exceeding the flow mixing losses between both impeller sides, and diffuser losses increase as the inlet difference between both sides strengthens. Due to the reduced work capacity of the rear impeller under small flow conditions, the double suction centrifugal compressor operating mode transitions from both sides working together to front impeller working alone. Jing [22] proposed a method to enhance rear impeller work capacity by increasing its radius, thereby delaying rear impeller stall and widening the double suction compressor operating range. Results showed an optimum rear impeller radius exists to match front and rear impeller operation within a stable range, with the stable operating range narrowing when the rear impeller radius is either insufficiently or excessively increased. In summary, previous studies on double suction centrifugal compressors focused primarily on the operating mode conversion process and mechanism, flow rate distribution between front and rear impellers under different conditions, circumferential non-uniform flow field distribution phenomena and formation mechanisms, with limited attention to inlet recirculation in double suction centrifugal compressors with asymmetric inlets.

Significant differences exist in internal flow field structures between asymmetric inlet double suction centrifugal compressors and conventional centrifugal compressors. For the rear impeller of double suction compressors, special flow phenomena arise due to circumferential flow field distribution characteristics affected by both the non-symmetric volute and bent inlet duct. Meanwhile, the rear impeller recirculation device differs significantly from that of conventional compressors. Under the coupled effects of these factors, the recirculation flow of the rear impeller differs from that of single-sided impellers. This paper focuses on the recirculation flow rate distribution characteristics of front and rear impellers, quantitatively describing differences in recirculation flow rates between both sides under various operating conditions. The circumferential distribution patterns of flow rates for both impeller sides under typical operating conditions are presented and their formation mechanisms analyzed. The recirculation flow rate differences between the two inlet branch ducts upstream of the rear impeller slot are compared. The circumferential flow rate distribution characteristics of the rear impeller recirculation flow at the two upstream slot positions are obtained by analyzing the circumferential distribution of flow parameters in the rear impeller upstream slot on the inlet cross-section.

Model Description

A double suction centrifugal compressor was adopted for this study. The main region of the impeller is shown in Fig. 1 [Figure 1: see original paper], including the inlet ducts, double suction impeller, inlet recirculation devices on both sides, shared vaneless diffuser, volute, and outlet ducts. Due to installation space limitations imposed by engine structural arrangements, the inlets on both impeller sides are supplied from a common inlet duct. The inlet forms for the double suction impeller differ significantly, with the front impeller using axial intake and the rear impeller achieving intake through a bent duct. After entering the import duct, the airflow divides into two parts: one portion flows through the straight duct into the front impeller, while the other flows to the rear impeller through a branch duct that subsequently divides into left and right inlet ducts before entering the rear impeller passage. The gas is pressurized by both impeller sides, then flows into the shared diffuser and is finally collected by the volute.

The inlet recirculation structures of the double suction centrifugal compressor are visible in Fig. 1. The recirculation devices of the front and rear impellers differ significantly. Fig. 2 [Figure 2: see original paper] shows an enlarged view of the recirculation device region. The front impeller inlet recirculation device is identical to that of traditional centrifugal compressors, with the upstream slot located upstream of the impeller and the bleed slot positioned at the impeller inlet axial section, both featuring annular structures. For the rear impeller, the recirculation device remains connected to an annular cavity, but the inlet duct is divided into two intake branches due to structural constraints of the double suction compressor. The upstream slot of the rear impeller is configured as a portion of an annular cavity, as shown in Fig. 3 [Figure 3: see original paper]. Fig. 4 [Figure 4: see original paper] presents a schematic diagram of the rear impeller upstream slot relative to the impeller circumferential position, with the arc position and three selected circumferential positions of the upstream slot marked in the figure. In the following discussion, the recirculation flow velocity distributions at these positions will be analyzed.

Numerical Calculation Method

The FINE/Turbo software was employed to solve the steady Reynolds-Averaged Navier-Stokes (RANS) equations. In the numerical calculations, the Spalart-Allmaras (S-A) turbulence model and finite volume center discrete method were used. Spatial discretization employed a central scheme, while the temporal term used an explicit four-stage Runge-Kutta scheme for iterative solutions. Multigrid and residual smoothing methods were adopted to accelerate numerical convergence. Ideal gas was assumed in the computation process, with all solid surfaces treated as adiabatic and non-slip boundaries. The compressor inlet flow direction was axial. Uniform total pressure and total temperature were imposed at the inlet, while static pressure was specified at the outlet, with static pressures gradually increased to simulate different operating conditions. When flow reduced to a critical value, mass flow was specified as the outlet condition instead of static pressure. For the rotor-stator interface, the "frozen-rotor" approach was selected because the calculation model used integer-cycle impeller passage grids and the rear impeller inlet recirculation featured a non-annular structure. Four operating speed conditions were calculated, with the compressor design speed being 97,000 r/min.

The relative flow rate is defined as the ratio of actual flow to design flow. The performance characteristics of the double suction compressor are shown in Fig. 7 [Figure 7: see original paper], where Fig. 7(a) presents the efficiency-flow characteristic curves and Fig. 7(b) shows the pressure ratio-flow characteristic curves. In the calculation process, the relative circumferential positions of the rotor and volute differed, and results at different circumferential positions were compared and analyzed. The results indicate that differences in internal flow structure and performance are minimal.

Calculation Grids

To ensure the calculation domain accurately matches the actual flow field of the double suction compressor, the selected domain includes the front and rear inlet ducts, rear impeller intake bent duct, inlet recirculation devices, impeller, volute, and outlet duct. The entire double suction compressor calculation domain is complex, divisible into the impeller rotor part and stator parts of other components. The grids for stator parts and front/rear impellers were generated separately and then assembled into the complete model using the IGG module. During meshing, adjacent blocks in the same direction were assigned appropriate node numbers, with wall surface y+ values maintained below 10. The grids for stator parts such as the volute and inlet pipe were manually meshed using the IGG module, while grids for symmetric impeller structures were automatically generated by the Autogrid module.

Due to the enormous grid count of the complete double suction centrifugal compressor model, validating grid independence for the entire unit is challenging. Therefore, the complete computational domain was divided into several regions using a modularization approach, with regional grids validated sequentially. First, grid independence was conducted for the double impeller without inlet and outlet ducts. Then, impeller grids satisfying independence requirements were matched with different volute grids to determine volute grid independence. Inlet and outlet duct grids were validated separately and subsequently connected with the volute and compressor. The final grid system includes: front and rear inlet ducts with 0.85 million grids, front impeller and inlet recirculation structure with 7.60 million grids, rear impeller and inlet recirculation structure with 7.0 million grids, and volute with 1.55 million grids, totaling approximately 17 million grids for the complete double suction centrifugal compressor model. The grids are shown in Fig. 5 [Figure 5: see original paper] and Fig. 6 [Figure 6: see original paper].

Results and Discussion

Recirculation Flow Distributions and Differences of Front and Rear Impeller

The flow rate through the recirculation channel defines the recirculation flow. Under large flow conditions, pressure in the bleed slot is lower than impeller inlet pressure, causing air to flow into the impeller passage through the bleed slot, resulting in negative recirculation flow. Under small flow rate conditions, bleed slot pressure exceeds impeller inlet pressure, causing airflow to exit the impeller passage through the bleed slot, producing positive recirculation flow. Fig. 8 Figure 8: see original paper shows recirculation flow changes under four different speed conditions, with recirculation flows at seven monitor points marked A through G. The recirculation flow transitions from negative to positive as back pressure increases and compressor flow rate decreases, reaching maximum recirculation at condition B. From condition B to A, recirculation flow rate decreases slightly due to reduced bleed slot pressure.

Fig. 8 Figure 8: see original paper shows the relative recirculation flow rate, defined as the ratio of recirculation flow rate to impeller outlet flow rate. The front impeller recirculation flow can reach 15.9% of its impeller flow rate under the four speed conditions. At constant speed, relative recirculation flow rate increases as compressor flow decreases from condition E to B. From condition B to A, relative recirculation flow rate decreases slightly. Although total compressor flow decreases, flow through the front impeller increases, causing relative recirculation flow rate to decrease. Under small flow conditions, the pressure difference between the upstream slot and bleed slot determines the recirculation flow.

Fig. 9 Figure 9: see original paper shows the rear impeller recirculation flow rate changes across four speed lines. The rear impeller recirculation flow rate transitions from negative to positive as compressor flow decreases, reaching maximum at condition A. In contrast, the front impeller maximum recirculation occurs at condition B and decreases at condition A. Fig. 9 Figure 9: see original paper indicates that under small flow conditions, the rear impeller recirculation flow rate reaches approximately 35.2% of the total rear impeller flow rate, substantially larger than the front impeller maximum. This occurs because rear impeller flow decreases while its recirculation flow increases in the small flow range, significantly increasing the relative recirculation flow rate. Additionally, the rear impeller recirculation flow at condition A shows a decreasing trend across the four speeds.

Fig. 10 [Figure 10: see original paper] presents the absolute recirculation flow rate curves for front and rear impellers versus total compressor flow at design speed. Three typical operating conditions are marked, with point H representing the boundary between large and small flow regions. To the right of point H is the large flow region, while the left side represents the small flow region. The recirculation flow rates of front and rear impellers differ significantly. On the right side of point H, negative recirculation flow rate indicates air flowing into impeller passages through the recirculation device, with the rear impeller recirculation flow rate exceeding that of the front impeller. In the small flow region left of point H, positive recirculation flow rate indicates air flowing from impeller passages to the inlet, with the front impeller recirculation flow rate exceeding that of the rear impeller. These results show that the recirculation flow rate transition from negative to positive occurs at different compressor flow rates for front and rear impellers, with the front impeller transition point occurring at a higher flow rate than that of the rear impeller.

The Circumferential Distribution of the Recirculation Flow

The recirculation flow differs between both impeller sides at stable operating conditions and the design point. For the front impeller, the radial velocity at the bleed slot represents the recirculation flow. The static pressure and radial velocity at the middle position of the bleed slot were analyzed, with ten values normalized by their circumferential average values (rV/V and P/P). Fig. 11 [Figure 11: see original paper] presents results at three operating points, where represents the circumferential angle referenced in Fig. 4. Both radial velocity and static pressure are non-uniform along the circumferential direction. The non-axisymmetric volute structure leads to circumferential static pressure distribution variations [17, 19], with Yang [20] providing detailed studies showing that circumferential distribution changes with operating conditions. Under large flow conditions, static pressure circumferential distribution exhibits large non-uniformity that diminishes with decreasing flow rate.

At large flow conditions, the circumferential distribution characteristics of radial velocity at the bleed slot are opposite to those of static pressure, as shown in Fig. 11(a). Since air flows into the impeller passage, static pressure acts as a back pressure with blocking effects, meaning lower static pressure corresponds to higher radial velocity. However, at the design point, air flows from the impeller to the inlet, as shown in Fig. 11(b), and static pressure in the bleed slot drives airflow through the duct, with higher static pressure areas corresponding to higher radial velocity. The small operating point shows a similar variation trend to the design point, as illustrated in Fig. 11(c). The radial velocity distribution at the three points demonstrates that static pressure distribution at the bleed slot determines the front impeller recirculation flow.

Table 1 compares the circumferential non-uniform extent of radial velocity at three operating conditions, showing that large flow rate values exceed those at other points, indicating less uniformity. The circumferential non-uniform extent of static pressure gradually decreases with reducing mass flow. These phenomena illustrate that the circumferferential non-uniform extent of static pressure determines the recirculation velocity distribution as flow decreases. The recirculation device reduces the circumferential non-uniform extent of the flow field at large flow rates. Higher static pressure areas at the impeller outlet correspond to lower inlet mass flow, and flow from the slot also decreases. Since the total flow is the sum of inlet flow and slot flow, the slot aggravates the uneven flow field distribution under large flow conditions.

Fig. 12 [Figure 12: see original paper] plots the static pressure and radial velocity distributions of the rear impeller at three operating points. The variation law is similar to the front impeller, with similar correspondence between static pressure and radial velocity on both impeller sides, but different variation patterns around the slot position. The bent duct installed at approximately 240°–300° at the rear impeller creates a low static pressure area, and flow field distortion leads to altered circumferential static pressure distribution at the bleed slot. At three operating points, normalized pressure and velocity show roughly the same trend but do not match perfectly, indicating that rear impeller bleed slot recirculation flow is also affected by the non-annular upstream slot and inlet bent duct.

The circumferential non-uniform extent of the rear impeller, shown in Table 2 , is obviously greater than that of the front impeller at three operating points, and the circumferential non-uniform extent of radial velocity does not decrease with flow reduction. Therefore, the uneven circumferential distribution of radial velocity at the rear impeller is determined by inlet flow field distortion, the non-circular upstream slot, and the non-axisymmetric volute structure.

The Flow Allocation at Branch Ducts of the Rear Impeller

Fig. 13 [Figure 13: see original paper] describes the flow allocation in the rear branch ducts versus total flow. At large flow, left duct flow exceeds right duct flow, while small flow conditions show the opposite tendency. These phenomena can be interpreted by the circumferential static pressure distribution inside the compressor. The non-symmetrical volute structure determines the uneven static pressure distribution, with each passage having different mass flow. Impeller passages corresponding to higher static pressure in the diffuser and volute have lower mass flow, with circumferential uneven distribution changing at various operating points.

The uneven distribution inside the compressor not only influences passage flow but also affects rear impeller branch duct flow. The right branch has lower flow due to higher static pressure, but this pattern reverses under small flow conditions.

Fig. 14 [Figure 14: see original paper] describes the flow difference between both upstream slots of the rear impeller. The left upstream slot has higher flow than the right slot at large flow, but small flow conditions show inverse results. The increasing rate of recirculation flow shows a slowing trend as flow decreases within the small flow range.

The Circumferential Distribution of the Recirculation Velocity and Recirculation Flow Rate of the Rear Impeller

The circumferential distribution of recirculation flow in the rear impeller requires further analysis due to different recirculation flows in branch ducts at various points. The different recirculation flows in both side ducts upstream of the slot represent the uneven circumferential allocation of recirculation flow, plotted using axial velocity Vz. Fig. 15 [Figure 15: see original paper] describes the axial velocity contour at three operating points. The static pressure distribution along the duct cross-section is also analyzed to elaborate its influence on recirculation velocity Vz, where velocity Vz has negative values when air flows into the impeller at large flow rates. Fig. 15(a) validates these results, showing positive values in the upper trace area. Both side upstream slots have the same axial velocity distribution, but circumferential distributions differ.

Without wall boundary effects, the lower area axial velocity exceeds the upper area due to higher static pressure in the lower area. The axial velocity distribution differs between both upstream slots and within the same slot at design speed, as shown in Fig. 15(b). The upper area has higher axial velocity than the lower area, with the left slot maximum less than the right slot. Corresponding to static pressure distribution, the upper area and left slot have lower static pressure. Under small flow conditions, the upper area has higher axial velocity in both slots, as shown in Fig. 15(c). The lower static pressure area leads to higher axial velocity at the left slot. The above analysis shows that the circumferential distribution of axial velocity in both upstream slots differs at three operating points, with upper axial velocity exceeding lower area values. This non-uniformity is influenced by static pressure at the inlet duct, and the duct shape determines static pressure around the slot. Therefore, static pressure can be modified by adjusting duct shape, consequently changing axial velocity at the upstream slot.

The difference in axial velocity circumferential distribution representing upstream slot recirculation flow is determined from Fig. 15. The quantitative axial velocity circumferential distribution along the mean line in the upstream slot is plotted in Fig. 16 [Figure 16: see original paper], with the mean line shown in Fig. 4. Because air flows into the impeller passage, axial velocity is negative. The axial velocity distribution shows slight differences between both side slots in Fig. 16(a), but the pattern is basically similar. The axial velocity degradation rate differs across the entire arc range, with axial velocity rapidly reducing in the upper slot area. This change process is determined by static pressure distribution in the inlet ducts. Based on absolute Vz values, right side recirculation flow exceeds left side flow, a result confirmed by Fig. 14. The axial velocity distribution at both side upstream slots shows significant differences in Fig. 16(b) at the design point. The right line has positive values across the entire arc range, indicating flow is pumped out from the impeller at the right slot, while air flows into the impeller in the 0-50% arc range and is pumped out to the inlet duct in the 50%-100% arc range at the left slot, with right recirculation flow exceeding left slot flow. Upper axial velocity exceeds lower upstream slot values on both sides, as shown in Fig. 16(c). The above analysis demonstrates that the circumferential distribution of axial velocity at the rear impeller upstream slot is determined by static pressure distribution in the inlet duct.

Recirculation velocity differs between the upstream slot and bleed slot due to velocity changes within the cavity. Fig. 17 [Figure 17: see original paper] describes the radial velocity distribution at the front impeller upstream slot, showing the same tendency as the volute static pressure distribution at three operating points, but with a certain phase shift due to circumferential position changes of the impellers during static pressure back-propagation in the volute. The radial velocity distribution is more uniform at the upstream slot compared with the rear impeller.

Fig. 18 [Figure 18: see original paper] shows the axial velocity distribution at different circumferential positions of the upstream slot within the rear impeller under three operating conditions, with Fig. 4 describing the detailed circumferential positions. The axial velocity trend lines on both sides at the same circumferential positions remain broadly consistent under large flow conditions, as shown in Fig. 18(a). Axial velocity decreases continuously from inside (near impeller) to outside (away from impeller), corresponding to maximum and minimum values determined by flow direction inside the duct. At large flow rates, air flows into blade passages through the branch duct, moving from outside the upstream slot to inside the inlet branch duct. Due to inertia effects, larger velocity is not generated outside the upstream slot, and velocity gradually increases along the flow direction. Therefore, velocity increases from outside to inside the upstream slot. However, axial velocity distribution differs between both sides at design flow rate, as shown in Fig. 18(b). The axial velocity at L3 is positive with maximum located at the middle slot, while axial velocities at L1 and L2 are negative and increase after initial decrease at the left slot. Velocity distributions of the three lines are roughly identical at the right slot, being symmetrical in the middle slot with maximum velocity, similar to flow between two parallel plates. With increasing recirculation flow, velocity distributions on both sides become symmetrical at the middle slot under small flow rates, as shown in Fig. 18(c). The above analysis indicates that velocity distribution along the slot width tends toward symmetry, with higher speed gradients within the boundary layer and maximum axial velocity in the middle slot.

Conclusions

This paper analyzes the recirculation flow characteristics of a double suction centrifugal compressor with non-axisymmetric recirculation devices under stable operating conditions, comparing recirculation flow differences between both impeller sides at design speed. The circumferential non-uniform distribution of recirculation velocity is presented and its formation mechanisms explained. Finally, differences and circumferential non-uniform distributions of recirculation flow between branch ducts at the rear impeller are described. The main findings are summarized as follows:

- The recirculation flow of the front impeller occupies 15.9% of front impeller flow. In the small flow range, after the front impeller recirculation peak, both recirculation flow and relative values at four speeds decrease with flow reduction, indicating the front impeller is moving away from stall boundary. However, the rear impeller recirculation flow continuously increases, occupying 35.2% of rear impeller flow and exceeding the front impeller relative recirculation peak. At four speeds, front impeller recirculation flow slightly increases with decreasing speed, while rear impeller recirculation flow decreases. At design speed, rear impeller recirculation flow exceeds front impeller flow at large flow rates, while front impeller recirculation flow exceeds rear impeller flow at small flow rates.
- Circumferential non-uniform distribution of recirculation velocity always exists at three classical operating points for the front impeller. The non-uniform extent at large flow rates is obviously greater than at other points, with small flow having slight extent. The circumferential non-uniform extent of static pressure determines the recirculation velocity distribution. For the rear impeller, the circumferential non-uniform extent of recirculation velocity is significantly greater than for the front impeller under the same operating condition. The circumferential distribution of recirculation velocity changes within the cavity, but the circumferential non-uniform extent does not reduce at the rear recirculation cavity with decreasing compressor flow rate. Three elements determine the circumferential non-uniform extent of rear impeller recirculation velocity: rear impeller flow distortion, circumferential non-symmetrical distribution of the upstream slot, and non-symmetrical volute structure. The circumferential distribution of static pressure inside the impeller dominates the circumferential distribution of bleed slot recirculation velocity.
- Under design flow and small flow conditions, the recirculation velocity at different circumferential positions along the mean line inside the upstream slot cross-section of the two branch ducts shows significant differences, with major differences even within the same recirculation slot. The static pressure distribution of the inlet duct determines the recirculation velocity distribution at the upstream slot cross-section.

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Note: Figure translations are in progress. See original paper for figures.