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Quasi-Dynamic Experimental Investigation of the Internal Flow Field in the Stationary Components of a High-Flow-Coefficient Centrifugal Compressor

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Abstract

Stationary components of centrifugal compressors (the vaneless diffuser, U-bend, return channel and L-bend) account for approximately 5%–10% of stage losses, yet experimental data on their internal flow fields in high-flow-coefficient machines remain scarce because rotating test rigs are costly and offer limited probe access. In this study, a quasi-dynamic test rig integrated with a wind tunnel was developed. By rotating the stationary guide vane (SGV) through one blade pitch in 3° steps to eight circumferential positions, the circumferential migration of the impeller wake of a high-flow-coefficient ($\varphi = 0.2$, $Mu = 0.8$) centrifugal compressor was simulated for the first time on a stationary rig. High-resolution five-hole probe measurements at four planes captured the complete evolution of the wake from generation and mixing to attenuation and disappearance. The results show that wake migration is reproduced one-to-one with the rotation angle at the diffuser outlet; the U-bend attenuates the peak-to-valley non-uniformity of the spanwise averaged total pressure from about 5.2% to about 0.5% (about 90%), while the return channel vanes induce a new 20°-period fluctuation through the potential-flow mechanism. At the stage outlet, the flow is circumferentially uniform but retains significant spanwise gradients. The losses of the U-bend and of the return channel + L-bend vary complementarily while the overall stage loss remains essentially constant ($< 2\%$), indicating that the total loss of the return passage is determined by the inlet area-averaged flow conditions and is independent of the circumferential position of the wake.

Keywords Centrifugal compressor · Stationary components · Quasi-dynamic test rig · Five-hole probe · Flow field measurement · Wake migration

1 Introduction

Centrifugal compressors are widely used in the petrochemical industry, compressed-air energy storage, industrial transport and aerospace propulsion owing to their high single-stage pressure ratio, wide operating range and compact structure. As industrial applications demand ever higher energy efficiency and compactness, centrifugal compressors continue to evolve towards high flow coefficients ($\varphi \geq 0.15$), high pressure ratios and high reliability. The aerodynamic design of centrifugal impellers is already approaching its efficiency limit, with single-stage efficiencies reaching 96% [1]; improving the performance of the stationary components—including the vaneless diffuser, U-bend, return channel and L-bend—has therefore become the key to further increasing stage efficiency. Previous studies show that stationary-component losses account for approximately 5%–10% of stage losses [2] and have an important influence on the stability margin [3–5]. A thorough understanding of the complex flow structures and loss-generation mechanisms inside the stationary components is therefore of both academic value and engineering significance.

Experimental measurement remains the fundamental means of revealing flow mechanisms in turbomachinery and validating numerical simulations. According to the measurement principle, experimental techniques can be classified as intrusive or non-intrusive [6]: intrusive techniques such as pneumatic probes, fast-response aerodynamic probes (FRAP) and hot-wire anemometry [7, 8] provide quantitative characterization of velocity fields, pressure distributions and turbulence intensity, whereas non-intrusive optical methods such as particle image velocimetry (PIV) and laser Doppler velocimetry (LDV) [9, 10] provide undisturbed instantaneous flow field data. These techniques have been widely applied in centrifugal compressor research over the past decades.

Numerous experimental studies have been reported for the stationary components of centrifugal compressors. Pinarbasi et al. [11] investigated the unsteady flow characteristics in a vaneless diffuser using hot-wire anemometry and found that the impeller wake is concentrated near the casing side of the diffuser. Roduner et al. [12] measured the unsteady flow field of a centrifugal compressor with FRAP and established a link between diffuser rotating stall and the impeller secondary-flow structures. Ferrara et al. [13–15] systematically investigated the influence of vaneless diffuser geometry on rotating stall inception. The wall static pressure distributions in a U-bend and a return channel measured by Rube et al. [16] agreed well with numerical simulations. Ubben et al. [17] found that increasing the diffuser vane clearance significantly reduces the total pressure ratio and isentropic efficiency. Rake-probe measurements by Rossbach et al. [18] showed that, at high flow rates, the large flow angles in the vaneless diffuser increase the negative incidence at the return channel inlet, inducing strong secondary flows.

Although rotating test rigs have provided the above insights, their inherent limitations are equally significant: they are expensive to build and operate, high-speed rotating components pose safety risks, and the thick casing severely restricts probe access. Measurements therefore usually cover only a few

inlet and outlet planes, and the detailed flow field inside the stationary components is difficult to obtain [19]. To overcome these difficulties, stationary-component test rigs in which stationary guide vanes (SGVs) replace the rotating impeller have been developed, with the inflow supplied by a wind tunnel or a compressed-air source [20, 21].

Fister et al. [22] pioneered a stationary test rig using water as the working medium to study the flow characteristics in U-bend passages. Kanemoto et al. [23] developed a single-channel stationary rig equipped with pre-swirl vanes and demonstrated that endwall secondary flows significantly interfere with blade-surface separation. Simpson et al. [24] built a sector cascade rig containing three return channel vanes, simulated the impeller-exit vortex system with an upstream swirl generator, and demonstrated that steady air injection effectively reduces losses at low flow rates; related separation-control strategies validated in linear compressor cascades achieve similar benefits [25]. Aalburg et al. [26] further extended the stator vane upstream across the 180° bend, improving the flow in the return channel. On this basis, Aalburg et al. [2] and Dolle et al. [20, 27] developed full-annulus stationary rigs with replaceable SGV sets to reproduce impeller-exit flow fields under different operating conditions, and investigated the effects of U-bend geometric parameters and inlet turbulence intensity on aerodynamic losses. In addition, Ellis [28], using a wooden passage equipped with pre-swirl vane arrays, found that a novel three-dimensional return channel design reduced the total pressure loss by about 50%. Reddy et al. [29] adjusted the swirl vane angle to simulate different operating conditions and verified the positive effect of an optimized blade leading-edge loading design.

However, the above stationary rigs were mostly developed for low-flow-coefficient centrifugal compressors ($\varphi \leq 0.12$). High-flow-coefficient machines ($\varphi \geq 0.15$, $\text{Mu} \geq 0.8$) differ markedly from low-flow-coefficient machines in meridional passage geometry, blade loading distribution and secondary-flow structures, and the flow separation, wake development and mixing processes inside their stationary components are also more intense. To date, experimental studies targeting the stationary components of high-flow-coefficient centrifugal compressors are almost absent from the open literature.

Moreover, a key limitation of existing stationary rigs is that the SGVs are usually fixed at a single circumferential position. In an operating compressor, the relative position between the rotating impeller and the downstream diffuser varies continuously with time, and this rotor–stator interaction significantly affects the flow field and aerodynamic losses inside the diffuser. Experimental methods capable of simulating the circumferential migration of impeller wakes are therefore urgently needed to clarify the quasi-dynamic flow characteristics inside the stationary components of high-flow-coefficient centrifugal compressors.

To fill these gaps, the present study builds on a previously developed stationary-component test rig for a high-flow-coefficient ($\varphi = 0.2$) centrifugal compressor [30] and proposes a quasi-dynamic measurement method in which the SGV is rotated in discrete angular increments. The main contributions of this work are as follows: (1) the SGV is rotated through one complete blade pitch in increments of about 3° (eight circumferential positions), reproducing, for the first time on a stationary rig, the circumferential migration of the impeller wake of a high-flow-coefficient centrifugal compressor and providing an economical and safe experimental means for studying rotor–stator interaction; (2) high-resolution five-hole probe measurements are performed at four measurement planes, yielding detailed distributions of total pressure, static pressure, Mach number, yaw angle and pitch angle; (3) the complete evolution of the impeller wake—from generation (MP1), through mixing and migration (MP2) and attenuation (MP3), to near dissipation (MP4)—is tracked, the homogenizing effect of the U-bend on the periodic wake is quantified under the present test conditions, and the potential-flow feedback of the return channel vanes on the upstream flow field is revealed; and (4) through component-level performance evaluation, a complementary loss pattern between the U-bend and the return channel assembly is revealed—their losses vary in opposite directions with SGV rotation angle while the overall stage loss remains essentially constant—providing a simplified performance prediction model for the design optimization of stationary components.

2 Quasi-Dynamic Test Rig and Measurement Methodology

This section describes the overall configuration of the quasi-dynamic test rig, the design principle of the SGV, the measurement planes and point arrangements, the instrumentation, the experimental procedure and data processing methods, and concludes with an uncertainty analysis.

2.1 Overall Configuration of the Test Rig

The quasi-dynamic test rig is integrated into an existing open-circuit wind tunnel, as schematically shown in Fig. 1. The tunnel is driven by a 350 kW variable-frequency fan, with a design maximum flow rate of 12.5 m³/s and a maximum absolute pressure of 1.2 bar. The inlet flow rate of the test section is precisely controlled by three independent regulating valves. To ensure the uniformity of the incoming flow and break up large-scale vortical structures, a honeycomb and two damping screens are installed in sequence in the settling chamber. The flow is accelerated through the contraction into the test section and, after leaving the test section, is discharged to the atmosphere through the diffuser section and the silencer.

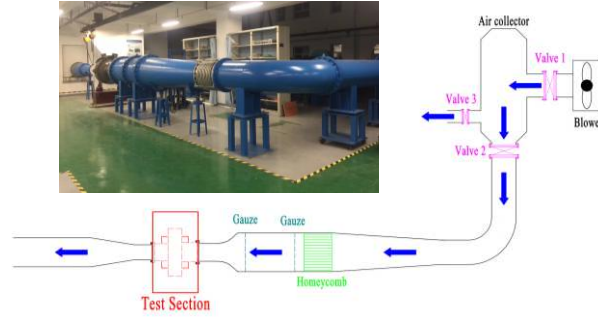


Fig.1 Schematic diagram of the quasi-dynamic test rig integrated with the wind tunnel system

The test section houses the stationary components of the first stage of a centrifugal compressor, including the stationary guide vane (SGV), the vaneless diffuser, the 180° U-bend, the return channel and the L-bend. All components are precision-machined from alloy materials to meet the requirements of structural strength and aerodynamic surface quality, with machining accuracy consistent with rotating-rig standards. High-temperature-resistant rubber sealing rings are installed at all component joints to prevent leakage, keeping the leakage rate below 0.3% of the test-section flow rate. Photographs and a structural schematic of the test section are shown in Fig. 2, and the main geometric parameters and design conditions of the prototype compressor are listed in Table 1.

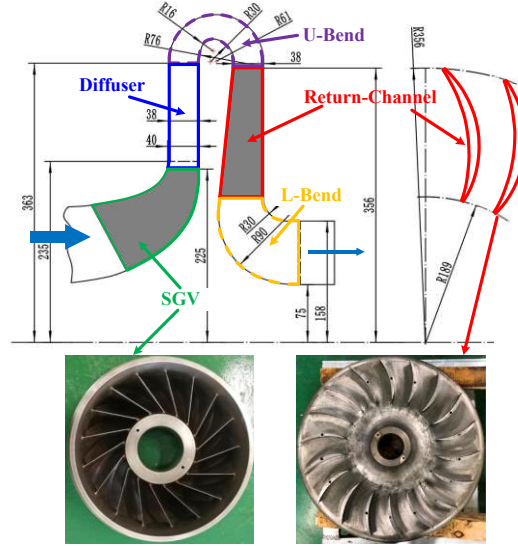


Fig.2 Photographs and structural schematic of the test section

Table 1 Main geometric parameters and design conditions of the centrifugal compressor

Parameter	Symbol	Value
Design flow coefficient	φ	0.2
Design machine Mach number	Mu	0.8
Design speed	n (r/min)	11650
Number of impeller blades	Z_{lm}	17
Number of return channel vanes	Z_{Re}	18
Impeller outlet diameter	D_2 (mm)	450
Impeller outlet width	b_2 (mm)	40
Impeller outlet blade geometric angle	$\beta_{s,2}$ (°)	65
Return channel vane inlet geometric angle	$\beta_{s,5}$ (°)	45
Return channel vane outlet geometric angle	$\beta_{s,6}$ (°)	100

2.2 Design Principle and Validation of the SGV

The SGV replaces the rotating impeller, and its core design objective is to match the spanwise distributions of the SGV-exit flow field (including Mach number, yaw angle, meridional velocity and circumferential velocity) to those at the exit of the actual rotating impeller at the

design condition. Keeping the meridional passage geometry and the blade count (17) unchanged, this objective is achieved by iteratively optimizing the blade angle distribution (the wrap-angle distribution from inlet to outlet).

A CFD-assisted optimization procedure is adopted: first, the velocity triangles and flow angle distributions at the impeller exit plane (MP1), extracted from three-dimensional viscous flow computations of the rotating impeller, are taken as the target; a parametric SGV model is then built, and the camber-line angle distribution is adjusted to minimize the root-mean-square error between the spanwise distributions of the SGV-exit flow field and the target. After multiple rounds of iterative optimization, the final SGV blade angle distribution reproduces the design spanwise distribution of the exit yaw angle in the main flow region (Span 0.1–0.85) with errors within $\pm 2^\circ$.

Fig. 3 compares the design spanwise distributions of the flow parameters at the SGV outlet and the rotating impeller outlet. The two agree well overall in the main flow region but deviate to some extent near the shroud (Span > 0.85). This deviation has a clear physical origin: the rotating impeller transfers energy to the gas through blade work, and its shroud-side boundary layer, in which separation is suppressed by the strong centrifugal pressure gradient, can sustain a large yaw angle. The SGV, in contrast, does no work and deflects the flow solely through blade curvature, so that a small blade curvature radius is required near the shroud to achieve the required exit flow angle, which inevitably induces local flow separation and forms a low-velocity, low-angle region. The deviation is therefore an inherent limitation of the SGV approach, but its influence is mainly confined to the near-wall region on the shroud side (Span > 0.9) and has limited impact on the main flow structure.

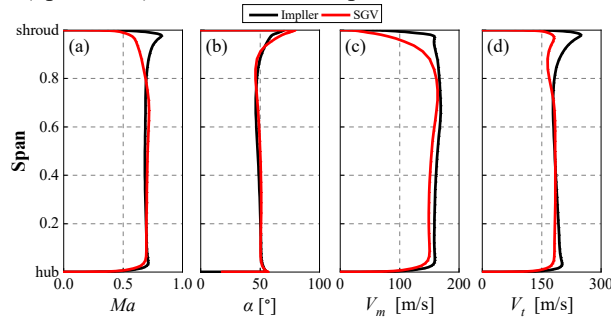


Fig.3 Comparison of the design spanwise distributions of flow parameters between the SGV outlet and the rotating impeller outlet

Fig.4 compares the three-dimensional geometric models of the rotating impeller and the SGV. The two share the same meridional passage contour and blade count, but the blade thickness distribution and the hub/tip profiles of the SGV are specially optimized to accommodate the flow characteristics under no-work conditions.

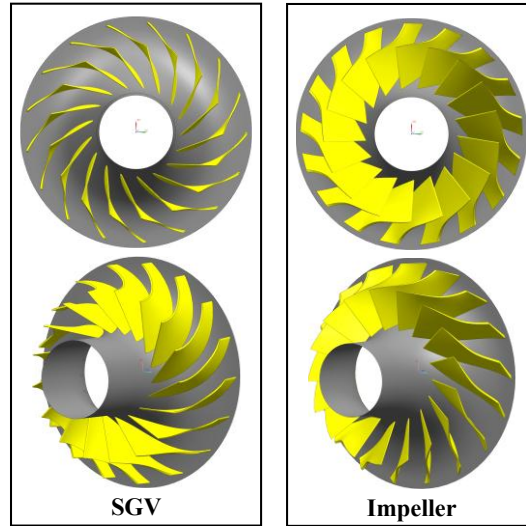


Fig.4 Comparison of the three-dimensional geometric models of the stationary guide vane (left) and the rotating impeller (right)

2.3 Measurement Planes and Point Arrangement

Five measurement planes (MP0–MP4) are arranged in the test section; their locations and functions are as follows (see Fig. 5):

MP0: located in the inlet duct of the test section, used to measure the inlet flow uniformity and determine the inlet reference parameters. Four static pressure taps (90° apart) are evenly distributed around the circumference of the pipe wall, and total pressure and total temperature probes are installed between adjacent taps. The inlet non-uniformity (total pressure non-uniformity coefficient) at MP0 is kept below 1.5%, meeting the

requirements of aerodynamic performance testing.

MP1: located at the SGV outlet / vaneless diffuser inlet. A five-hole probe is installed with two degrees of freedom of movement in the spanwise and circumferential directions. The circumferential measurement range is 24° , slightly larger than one complete SGV pitch ($360^\circ/17 \approx 21.18^\circ$), so that one blade passage is completely covered to fully capture the periodic wake structures.

MP2: located at the vaneless diffuser outlet / U-bend inlet. The point arrangement is identical to that at MP1, facilitating the tracking of wake evolution inside the diffuser.

MP3: located at the U-bend outlet / return channel inlet. The point arrangement is identical to that at MP2 and is used to evaluate the homogenizing effect of the U-bend on the upstream wake. The circumferential measurement range at MP3 is also 24° , corresponding to about 1.2 return-channel vane pitches (vane pitch = $360^\circ/18 = 20^\circ$), so that one complete period of the vane-induced fluctuation is covered.

MP4: located at the stage outlet (L-bend outlet). The flow passage cross-section in this region is irregular, and the circumferential measurement range is also 24° . In addition, four static pressure probes are evenly distributed along the circumference at 0.5 outlet heights downstream of MP4 as the outlet static pressure reference.

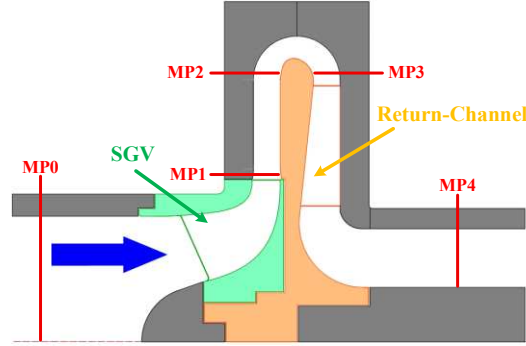


Fig.5 Layout of the measurement planes (MP0–MP4) in the stationary-component test section

In the validation phase, a dense point arrangement was adopted to ensure data accuracy and spatial resolution: 19 (spanwise) $\times$ 49 (circumferential, 0.5° interval) = 931 points at each of MP1–MP3 and 49 (circumferential) $\times$ 27 (spanwise) = 1323 points at MP4, totaling about 4100 points per complete survey (Fig. 6). In the subsequent detailed analysis, a sparse point arrangement was used to balance accuracy and measurement efficiency: 17 spanwise points at MP1–MP3 and 21 at MP4 (covering 0.1 – 0.9 of the blade height), with 25 circumferential points (one per degree over 24°), i.e. about 43.7% of the validation-phase total.

In the variable-angle experiments, to simulate the circumferential migration of the impeller wake, eight SGV relative angles ($0^\circ, 3^\circ, 6^\circ, 9^\circ, 12^\circ, 15^\circ, 18^\circ$ and 21°) were selected within one blade pitch ($360^\circ/17 \approx 21.18^\circ$) at a step of about 3° , covering one complete blade pitch.

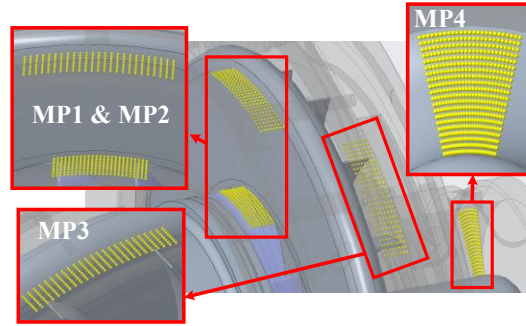


Fig.6 Schematic of the measurement point arrangement at the MP1–MP4 planes

2.4 Instrumentation and Measurement Accuracy

To accurately measure the inlet flow uniformity at MP0, a dedicated measurement device providing two degrees of freedom—circumferential rotation and radial displacement—was designed. Except for the stepper motor, all parts were manufactured by 3D printing (nylon, accuracy ± 0.1 mm), as shown in Fig. 7. The circumferential rotation mechanism consists of four rings: the middle two form the rotating frame, and the other two are a large gear disc and a sealing plate. The stepper motor drives the gear disc through a bevel gear, achieving circumferential rotation over a range of 190° . Multi-stage sealing teeth on the sealing plate reduce clearance leakage, with the measured leakage below 0.5% of the flow rate. The radial displacement is realized by a precision translation stage (stroke 200 mm, repeatability ± 0.02 mm), and a rotary stage adjusts the five-hole probe head so that it faces the incoming flow precisely.

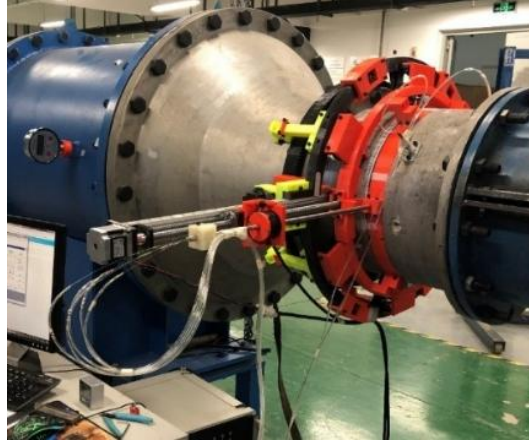


Fig.7 Measurement device for the inlet flow field uniformity at the MP0 plane

For the MP1–MP4 planes, the probe must be able to move in both the circumferential and spanwise directions. The circumferential motion is realized by the cooperation of a stepper motor, a gear set and a precision gear disc: the gear disc is mounted on the upper casing cover concentrically with the impeller, and the probe and its fixture are installed on the disc. The spanwise motion is provided by precision translation stages mounted on the disc. The aerodynamic parameter measurement device is shown in Fig. 8. MP1 and MP2 share one translation stage, while MP3 and MP4 each have an independent stage. A follow-up linkage mechanism is mounted on each translation stage, which allows the probe to move freely along the span while keeping the sealing plate moving synchronously with the probe, effectively suppressing clearance leakage.

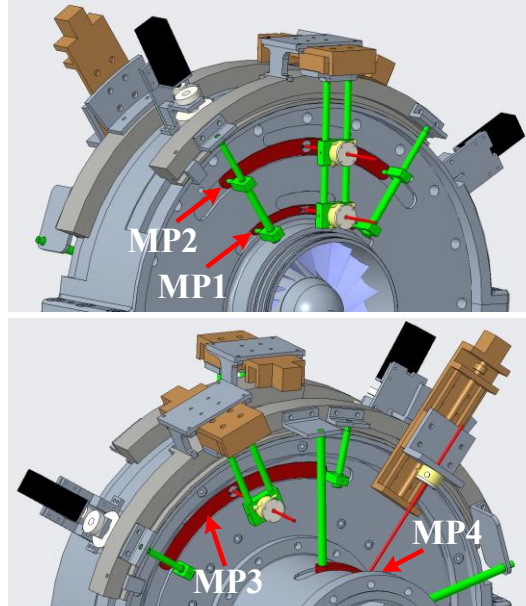


Fig.8 Five-hole probe measurement device for the MP1–MP4 planes

All pressure data are acquired by a DSA3217 electronic pressure scanner equipped with 16 silicon piezoresistive differential pressure sensors (range ± 5 psi, i.e. ± 34.47 kPa; accuracy $\pm 0.05\%$ FS; sampling rate 500 Hz per channel; 16-bit resolution). Total temperature is measured with Pt100 resistance thermometers (range -50 – 300 °C); after accounting for sensor heat-transfer errors and systematic errors, the effective measurement accuracy is within ± 1 °C, and a four-wire connection is used to eliminate the influence of lead resistance.

An L-shaped bundled five-hole probe is used in this experiment; the bundled section is 2.1 mm in diameter, the stem is 6 mm in diameter, and the five pressure holes at the head are each 0.5 mm in diameter, making the probe suitable for three-dimensional flow measurements within $\pm 30^\circ$. The probe was calibrated in a standard calibration section of a wind tunnel covering Mach numbers of 0.1–0.9, yaw angles of $\pm 30^\circ$ and pitch angles of $\pm 30^\circ$, with the calibration process covering all quadrants (Fig. 9). The measured Mach numbers at the four planes range from 0.30 to 0.75, within the calibrated Mach number range; local yaw angles in the MP1 wake region approached the calibration limit ($\pm 30^\circ$), and such points were flagged during data processing and excluded when out of range (less than 0.4% of all points). All remaining points lie within the valid calibration range.



Fig.9 The calibration wind tunnel

The probe head diameter (2.1 mm) is about 3% of the local passage width, so its blockage effect on the flow field is negligible. The probe stiffness was verified to meet the measurement requirements, with deformation below 0.01 mm under the maximum dynamic pressure. The calibrated measurement accuracy of the five-hole probe is as follows: the flow angle calibration error is within $\pm 0.5^\circ$, and the overall angular error in the experimental measurements is within $\pm 1^\circ$ after accounting for installation and referencing errors; the calibration errors of total and static pressure are $\pm 0.05\%$, and that of Mach number is ± 0.008 .

2.5 Experimental Procedure and Data Processing

Before each operating-condition measurement, the wind tunnel was run steadily for no less than one hour to ensure that the inlet parameters reached thermodynamic and aerodynamic stability. The formal measurement proceeded as follows: (1) coordinate referencing—the circumferential origin (0°) was defined at the trailing edge of a selected blade, and the spanwise coordinate was defined with Span = 0 at the hub wall and Span = 1.0 at the shroud wall; (2) baseline measurement—with the SGV rotated to the 0° position, MP0–MP4 were surveyed in sequence; (3) variable-angle measurement—the SGV was rotated successively to 0° , 3° , 6° , 9° , 12° , 15° , 18° and 21° , and MP1–MP4 were surveyed at each position; and (4) repeatability check—three repeated measurements were performed at the 0° and 12° positions to verify data repeatability.

With about 4100 points across the four planes, a complete dense-grid survey in the validation phase took about 9 hours (about 8 hours of data acquisition). In the sparse-grid and variable-angle measurements, the duration was reduced to about 4 hours by optimizing the measurement strategy (e.g. surveying MP1 and MP2 simultaneously and reducing the dwell time per point to 3 s).

The operating conditions are summarized in Table 2. The tests were conducted on the same full-scale geometry ($D_2 = 450$ mm) as in the previous work [30]. By adjusting the wind tunnel flow rate, the test flow coefficient was matched to the design value $\phi = 0.2$, and the area-averaged Mach number at MP1 was comparable to the impeller-exit Mach number at the design point, ensuring Mach number similarity. The characteristic Reynolds number based on D_2 and the average velocity at MP1 is about 2.6×10^6 , close to the self-similar regime; previous stationary-rig studies [20, 24] have shown that, at this Reynolds number level, Reynolds number differences have limited influence on the time-averaged flow structure and loss distribution of stationary components. The present test data are therefore representative of the true flow characteristics at the design condition in the time-averaged sense. The influence of the Reynolds number difference from the real machine on the transition location and near-wall loss details is further discussed in Section 5.1.

Table 2 Test operating conditions

Parameter	Value
Mass flow rate (kg/s)	7.8
Area-averaged Mach number at MP1	0.62
Characteristic Reynolds number	2.6×10^6

The five-hole probe data are processed with a non-nulling interpolation method: first, the calibration coefficients $K\alpha$ and $K\beta$ are computed from the five hole pressures; the local yaw angle α , pitch angle β , total pressure P_t and static pressure P are then obtained by interpolating the calibration surfaces; finally, the local Mach number is calculated from the isentropic relations.

During the measurements, the atmospheric pressure and temperature drifted slightly (the inlet total pressure fluctuated by about ± 200 Pa on average). In the loss calculations, this was compensated by real-time monitoring of the inlet parameters and non-dimensionalization, introducing an error of about $\pm 0.5\%$.

The aerodynamic performance of each stationary component is quantitatively evaluated by the total pressure loss coefficient and the static pressure recovery coefficient, defined as follows:

Static pressure recovery coefficient:

$$c_{ps} = \frac{P_{Out} - P_{In}}{P_{t,In} - P_{In}} \quad (1)$$

Total pressure loss coefficient:

$$c_{pt} = \frac{P_{t,In} - P_{t,Out}}{P_{t,In} - P_{In}} \quad (2)$$

where P is the static pressure, P_t is the total pressure, and the subscripts In and Out denote the component inlet and outlet planes, respectively. The static pressure recovery coefficient characterizes the ability of the component to convert kinetic energy into pressure energy (ideal value 1.0), whereas the total pressure loss coefficient represents the proportion of the irreversible total pressure loss caused by the component relative to the inlet dynamic pressure (ideal value 0).

2.6 Experimental Uncertainty Analysis

Various uncertainty factors exist in the experimental process and introduce errors into the measurement results. The uncertainty sources include the inherent accuracy of the instruments, calibration errors, environmental drift, repeatability errors and data processing errors. The Type A (repeatability) uncertainty is obtained from $n = 3$ repeated measurements at the 0° and 12° positions, corrected by the t-distribution at the 95% confidence level. The uncertainty components of the directly measured quantities are summarized in Table 3.

Table 3 Uncertainty budget for direct measurement parameters

Parameter	P	P_t	T	α	β	Ma
Type A uncertainty (repeatability, $n = 3$)	$\pm 0.1\%$	$\pm 0.15\%$	$\pm 0.05^\circ \text{C}$	$\pm 0.2^\circ$	$\pm 0.3^\circ$	± 0.005
Type B uncertainty (instrument accuracy)	$\pm 0.05\% \text{ FS}$	$\pm 0.05\% \text{ FS}$	$\pm 1.0^\circ \text{C}$	$\pm 1.0^\circ$	$\pm 1.0^\circ$	± 0.008
Combined standard uncertainty	$\pm 0.11\%$	$\pm 0.16\%$	$\pm 1.0^\circ \text{C}$	$\pm 1.02^\circ$	$\pm 1.04^\circ$	± 0.009
Expanded uncertainty ($k = 2$)	$\pm 0.22\%$	$\pm 0.32\%$	$\pm 2.0^\circ \text{C}$	$\pm 2.04^\circ$	$\pm 2.09^\circ$	± 0.018

According to the law of error propagation, the combined standard uncertainty of the indirectly measured quantities is: $q = f(x_1, x_2, \dots, x_n)$

$$u_c(q) = \sqrt{\sum_{i=1}^n \left(\frac{\partial f}{\partial x_i} \right)^2 u^2(x_i)} \quad (3)$$

Based on Eq. (3), the uncertainties of the key derived quantities are evaluated as follows:

(1) Uncertainty of the total pressure loss coefficient c_{pt}

From Eq. (2), the combined standard uncertainty is $u(c_{pt}) = \pm 0.008$ and the expanded uncertainty ($k = 2$, 95% confidence level) is $U(c_{pt}) = \pm 0.016$; taking a typical total pressure loss coefficient of $c_{pt} \approx 0.25$, the relative expanded uncertainty is about $\pm 6.4\%$. The relative uncertainties for the individual components are listed in Table 4.

(2) Uncertainty of the static pressure recovery coefficient c_{ps}

From Eq. (1), the combined standard uncertainty is $u(c_{ps}) = \pm 0.007$ and the expanded uncertainty ($k = 2$, 95% confidence level) is $U(c_{ps}) = \pm 0.014$.

(3) Uncertainty of mass-flow-weighted averaged quantities

The uncertainties of area-averaged and mass-averaged quantities depend on the single-point measurement uncertainty and the number of measurement points. For the area-averaged total pressure at the MP1–MP3 planes: with a single-point combined standard uncertainty of $\pm 0.16\%$ and 931 points, the random component of the area-averaged total pressure uncertainty is $\pm 0.005\%$, and about $\pm 0.15\%$ after including the systematic error component.

Based on the above evaluation, the main uncertainty sources of this experiment, ranked by importance, are as follows: (1) the angular measurement accuracy of the five-hole probe (within $\pm 1^\circ$ under test conditions)—the dominant source, which directly affects the measurement accuracy of the yaw and pitch angles and hence the calculation accuracy of the velocity components and mass flow rate, with a more pronounced influence in high-angle regions such as the MP1 wake region; (2) the zero and temperature drift of the pressure scanner ($\pm 0.05\% \text{ FS}$)—over continuous surveys of 4–9 hours, temperature drift may cause a systematic error of $\pm 0.3\%$, which can be controlled within $\pm 0.15\%$ by inserting reference pressure points into the measurement sequence for real-time correction; (3) the slow drift of the inlet total pressure and total temperature during long-duration operation (about $\pm 200 \text{ Pa}$ and $\pm 0.5^\circ \text{C}$)—with real-time monitoring at MP0 and non-dimensionalization, the influence of this error on the loss coefficients is about $\pm 0.5\%$; (4) probe positioning error—the repeatability of the stepper motor and translation stages is $\pm 0.02 \text{ mm}$ (spanwise) and $\pm 0.05^\circ$ (circumferential), negligible relative to the local passage scale; and (5) data interpolation error—downsampling from the 19×49 grid to the 17×25 grid at MP1–MP3 and from the 49×27 grid to the 25×21 grid at MP4 in the sparse point arrangement introduces an interpolation error of about $\pm 0.3\%$ (evaluated by comparing the validation measurements of the dense and sparse arrangements).

It should be noted that, when comparing the performance coefficients at the eight SGV rotation angles in the following sections, the criterion for the “relative variation” is the Type A (repeatability) uncertainty and the scatter of the three repeated measurements, rather than the expanded

uncertainty containing systematic components. Since the eight angles are measured within the same measurement sequence, systematic errors affect the data at all angles essentially equally and do not affect the assessment of angle-to-angle relative differences.

The validity of the quasi-dynamic simulation method has been verified in previous work through CFD comparison [30]. The verification results show that the spanwise distribution of the SGV-exit flow field agrees well overall with that of the rotating impeller exit, confirming the data reliability of the test rig. The variable-angle measurements in this paper extend this validated baseline, providing complete flow field evolution data under quasi-dynamic conditions through systematic surveys at eight circumferential positions.

3 Detailed Flow Field Analysis of the Stationary Components

This section systematically analyzes the flow field characteristics at the four measurement planes (MP1–MP4). The analyzed parameters include the yaw angle (α), pitch angle (β), Mach number (Ma), total pressure (P_t) and static pressure (P). By comparing the measurement results at different SGV rotation angles, the complete evolution of the impeller wake along the stationary components—from generation and development through mixing and attenuation to near disappearance—is revealed. For cross-plane comparison, identical color scales are used for the same parameter at different planes; in the averaged plots, the eight curves correspond to the eight SGV rotation angles (0° – 21° in 3° steps).

3.1 Flow Field at the SGV Outlet (MP1)

Fig. 10 presents the parameter contours at MP1, including the yaw angle, pitch angle, Mach number, total pressure and static pressure.

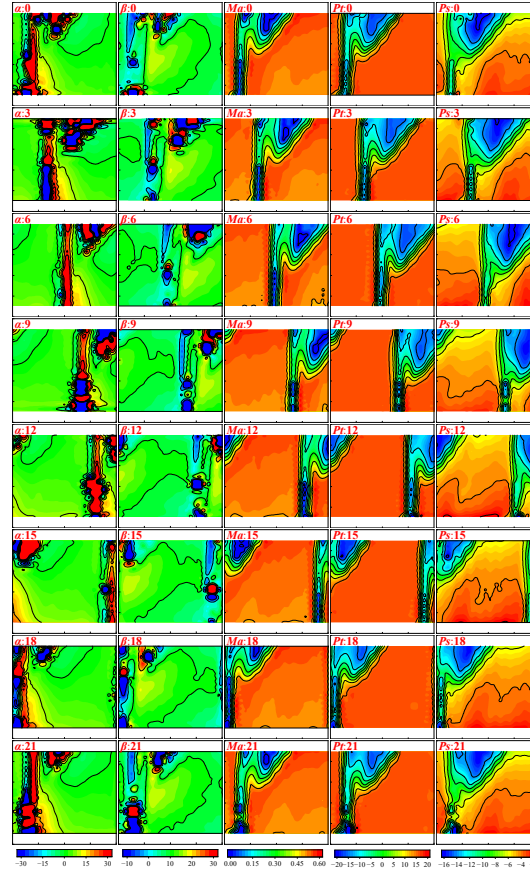


Fig.10 Contours of flow yaw angle (α), pitch angle (β), Mach number (Ma), total pressure (P_t), and static pressure (P) at the SGV outlet for eight SGV rotation angles

The contours show that the 3° SGV rotation step resolves the migration of the wake with circumferential position. The circumferential point spacing of 1° is comparable to the characteristic scales of the blade trailing-edge thickness and the shroud-side separation zone; consequently, point-to-point interpolation oscillations exist in these local regions and the contour continuity is somewhat reduced. This apparent waviness is caused by point discretization and interpolation smoothing and does not affect the interpretation of the parameter distributions in the main flow region.

The contour patterns of Mach number, total pressure and static pressure are highly consistent, and their low-value regions correspond well to the high-loss regions; after the point density is reduced, the interpolation smoothing effect visually enlarges the low-value regions in the contours. The contours also show that, at the impeller exit of the high-flow-coefficient compressor (simulated by the SGV), a large low-velocity separation

zone exists on the suction side near the shroud. The contours qualitatively reflect the parameter variations downstream of the blade trailing edge, whereas quantitative trends with rotation angle require averaging. Fig. 11 presents the circumferentially averaged (left) and spanwise averaged (right) results. The circumferentially averaged spanwise distribution of the yaw angle shows that, over 0.1–0.9 of the blade height and excluding the wake and shroud-side separation regions, the yaw angle is about 48°, in good agreement with the SGV design value; the spanwise averaged circumferential distribution clearly exhibits the periodic fluctuation within one pitch.

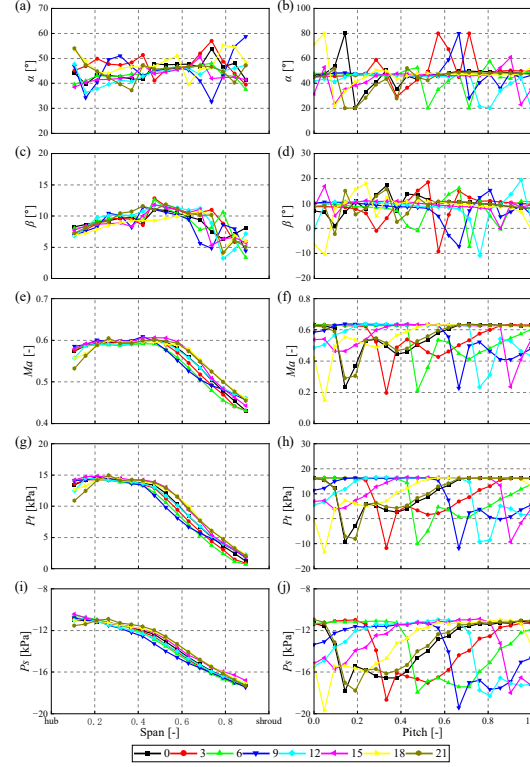


Fig.11 Circumferentially averaged spanwise profiles (left) and spanwise averaged circumferential profiles (right) of the flow parameters at MP1

The circumferential averages show that, except for the yaw angle, all parameters follow essentially the same trend with SGV rotation angle. The yaw angle is the most sensitive to the rotation angle: its circumferential average is generally flat, but pronounced fluctuations appear near the shroud owing to suction-side separation, and the change in the relative position between the SGV and the diffuser further amplifies these fluctuations; the hub-side fluctuation amplitudes are smaller and are related to the chamfer discontinuity at the passage joint, a feature visible in all contours. The pitch angle varies similarly to the yaw angle but with weaker fluctuations. The circumferential averages of Mach number, total pressure and static pressure are nearly insensitive to the rotation angle, indicating that SGV rotation mainly alters the flow angle distribution rather than the overall energy level.

The spanwise averages show pronounced periodic variations of all parameters within one pitch, reflecting the combined influence of the blade trailing-edge vortices and the shroud-side shedding vortices. The flow angles fluctuate most strongly, reflecting the formation and shedding of swirl; the Mach number, total pressure and static pressure exhibit periodic fluctuations of the energy level, in which the dominant period corresponds to the action of the blade trailing edges and the superimposed smaller-scale fluctuations originate from the shroud-side vortex shedding.

In summary, a 3° SGV rotation step achieves a reasonable balance between measurement accuracy and experimental effort: it resolves the periodic migration of the wake and meets the experimental requirements of quasi-dynamic simulation.

3.2 Flow Field at the Vaneless Diffuser Outlet (MP2)

Fig.12 presents the parameter contours at MP2 for the eight rotation angles. Compared with MP1, the flow field changes significantly after the deceleration and diffusion process in the vaneless diffuser.

Compared with MP1, the MP2 flow field is considerably more uniform: all flow angles lie within the probe calibration range, the regions where the yaw angle exceeded the calibration limits—caused by the blade trailing edges and the shroud-side separation at MP1—have disappeared completely, and the uniformity of the central main flow region over the whole pitch is markedly improved. The large shroud-side separation zone has vanished, although its residual traces remain visible in the contours, while the narrower trailing-edge separation zone has been fully mixed out. Meanwhile, the hub-side yaw angle increases and the trailing-edge region widens accordingly, which is associated with the combined effect of wall friction and local separation.

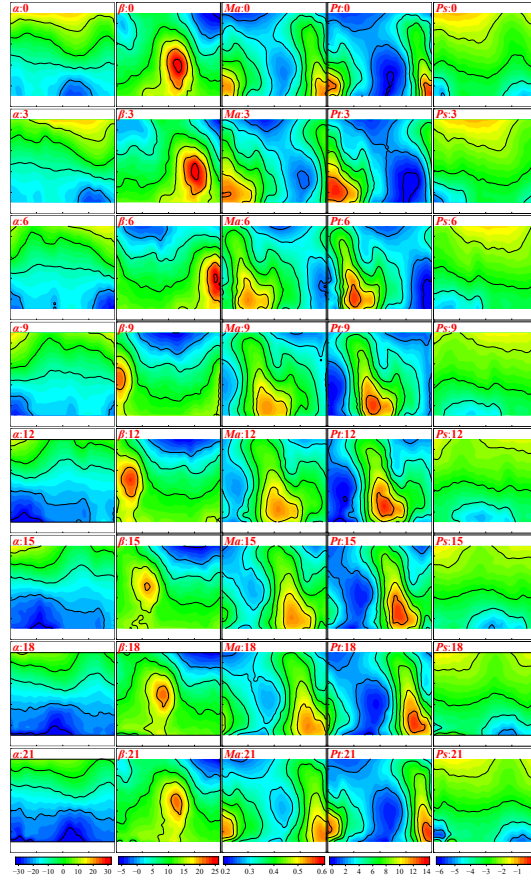


Fig.12 Contours of flow yaw angle (α), pitch angle (β), Mach number (Ma), total pressure (Pt), and static pressure (P) at the vaneless diffuser outlet (MP2) for eight SGV rotation angles

The pitch angle (β) evolves differently from the yaw angle. At MP1, the sharp variations of the yaw and pitch angles are concentrated in the blade trailing-edge region and the shroud-side separation zone; at MP2, the pitch angle forms two distinct high and low regions: the low-pitch region on the shroud side is inherited from the MP1 separation zone, whereas the high-pitch regions on the hub side and in the main flow originate from the central main flow at MP1 and develop further during the diffusion and deflection process. The average pitch angle at MP1 is about 10° , whereas at MP2 it is significantly higher on the hub side; this difference produces the high–low contrast visible in the contours.

The Mach number and total pressure contours at MP2 retain patterns similar to those at MP1, but the static pressure distribution differs markedly from that at MP1. At MP1, the SGV wake is distributed almost as a straight line, with the hub-side and shroud-side wakes at the same circumferential position; at MP2, the low-energy regions of the wake and of the separation vortex become circumferentially displaced: the hub-side wake low-energy region keeps its shape but widens circumferentially, while the shroud-side low-energy region shifts from the right of the trailing edge to the left. This displacement is mainly caused by the change in flow angle during the diffusion process.

Fig. 13 presents the circumferentially averaged and spanwise averaged parameters at MP2.

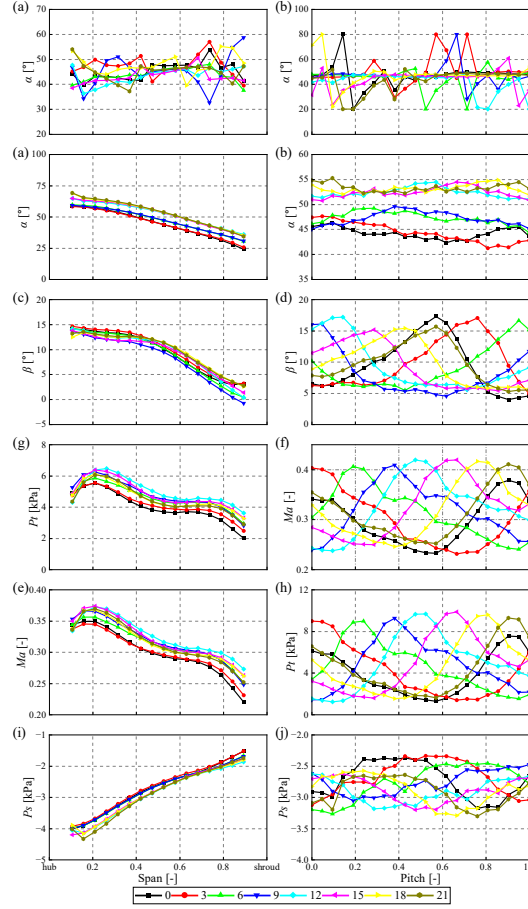


Fig.13 Circumferentially averaged spanwise profiles (left) and spanwise averaged circumferential profiles (right) of the flow parameters at MP2

The circumferential averages on the left of Fig. 13 show that, after the vaneless diffuser, the spanwise distributions of all parameters become more consistent. The yaw angle decreases monotonically from hub to shroud—the hub-side flow is more radial and the shroud-side flow more axial—forming a step-like distribution, which also explains why the shroud-side low-energy fluid region is offset to the left relative to the wake. The pitch angle varies similarly to the yaw angle but changes more drastically near the shroud. The underlying reason is the flow-guiding character of the SGV: doing no work and only guiding the flow deflection, the SGV has an exit flow dominated by axial velocity, so the fluid migrates towards the hub side inside the vaneless diffuser and the shroud-side velocity decreases significantly. When the pitch angle drops to negative values, local separation occurs; this is consistent with the total pressure contours, which are higher on the hub side and lower on the shroud side.

The spanwise averages on the right of Fig. 13 show that the periodic variation brought about by SGV rotation is still visible at MP2, with the yaw angle being an exception. Comparing MP1 and MP2, the yaw angle at MP1 is about 48° after excluding the wake and separation regions, whereas at MP2—although the variation process is still visible in the contours—the spanwise averaged curve no longer shows clear periodicity. The other parameters still exhibit significant periodic variation: low-energy and high-energy fluids are distributed alternately in the circumferential direction after mixing in the vaneless passage, appearing as alternating peaks and valleys in the spanwise averaged curves. Measured by the ratio of the peak-to-valley difference of the spanwise averaged total pressure within one SGV pitch to its mean value, the circumferential non-uniformity at MP2 is about 5.2%.

Comparing the contours and averaged curves at MP1 and MP2, it can be concluded that the 3° SGV rotation step reproduces the time-averaged process of wake migration with impeller rotation angle; the interpolation error introduced by grid sparsening (about $\pm 0.3\%$, see Section 2.6) does not affect this interpretation. The MP2 results clearly demonstrate the mixing process inside the vaneless diffuser and its homogenizing effect. This multi-angle flow field characterization method can provide a reference for subsequent studies on how the U-bend and the return channel respond to the periodic flow field when the wake is at different circumferential positions.

3.3 Flow Field at the U-bend Outlet (MP3)

The preceding analysis shows that the flow becomes more uniform after the vaneless diffuser, and the periodic migration of the wake is also

clearly discernible. This section further analyzes the MP3 flow field to clarify the flow evolution inside the U-bend. Fig. 14 presents the parameter contours at MP3.

Compared with MP2, the wake-induced periodic fluctuations have almost disappeared at MP3. As shown by the spanwise averaged parameters in Fig. 15, the periodic fluctuations of all parameters except total pressure are no longer evident, and the fluctuation intensity of total pressure is significantly weaker than at MP2. Using the same metric as in Section 3.2 (the peak-to-valley non-uniformity of the spanwise averaged total pressure within one SGV pitch), the value at MP3 is about 0.5%, an attenuation of about 90% from the 5.2% at MP2. These observations demonstrate that the U-bend plays a significant role in eliminating the upstream periodic wake and homogenizing the flow; the underlying mechanism is discussed in detail in Section 5.2.

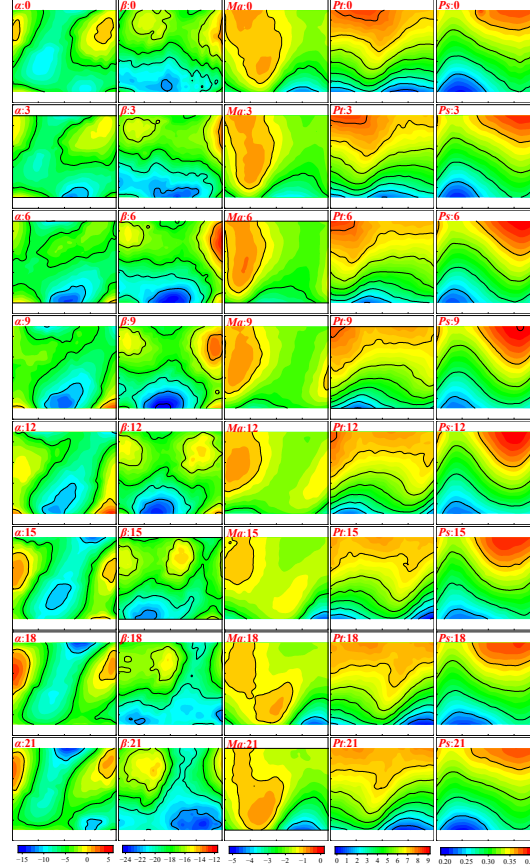


Fig.14 Contours of flow yaw angle (α), pitch angle (β), Mach number (Ma), total pressure (Pt), and static pressure (P) at the U-bend outlet (MP3) for eight SGV rotation angles

The contours in Fig. 14 show that weak periodic traces remain visible in total pressure and Mach number, but their intensity is significantly attenuated; the tilt direction and amplitude of the wake-induced yaw angle are consistent with those at MP2, and the pitch angle also exhibits some periodicity. These residual fluctuations, however, are barely discernible in the spanwise averaged curves on the right of Fig. 15.

Unlike MP1 and MP2, the distribution patterns of Mach number, total pressure and static pressure change distinctly at MP3. There are no blades in the diffuser and the U-bend upstream of MP3, and the return channel vanes downstream of the plane are relatively fixed in position; the MP3 flow structure is therefore determined mainly by the inlet conditions homogenized by the U-bend. Fig. 14 shows that the MP3 contour structures are essentially fixed at different SGV rotation angles, indicating that the return channel inlet flow field barely changes with the circumferential position of the upstream wake—i.e. the return channel vane row exerts a strong constraint on the upstream flow field. This is consistent with the velocity distribution determined by the SGV flow-guiding characteristics described in Section 2.2.

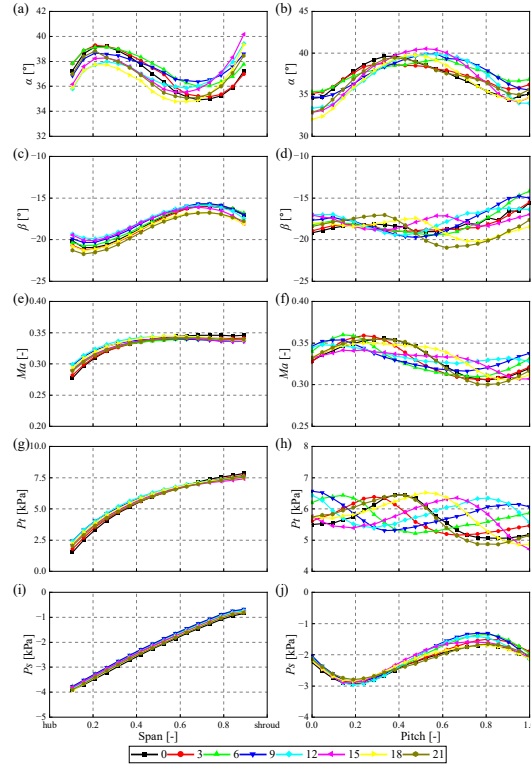


Fig.15 Circumferentially averaged spanwise profiles (left) and spanwise averaged circumferential profiles (right) of the flow parameters at MP3

The circumferential averages on the left of Fig. 15 further confirm the homogenizing effect of the U-bend: after the 180° turn, the spanwise consistency of all parameters improves markedly. The curves of Mach number, total pressure and static pressure are similar in shape and all increase from hub to shroud—the fluid migrates towards the shroud side through the U-bend, so that the shroud-side total pressure and velocity are higher than those on the hub side. Meanwhile, the spread among the curves at different SGV rotation angles is significantly reduced, and the pressure and Mach number distributions become smoother, indicating that the bend further weakens the differences in inlet conditions.

The spanwise distributions of the flow angles behave differently. If the trend of Section 3.2 were extrapolated, the flow angles would be expected to vary monotonically after the U-bend, like the pressure and velocity parameters; however, the measured yaw and pitch angles both display an inverted-S distribution: their increasing and decreasing trends are opposite and mirror each other, approximately symmetric about 50% blade height.

The spanwise averaged results are shown on the right of Fig. 15. Unlike MP2, the variations of all parameters over one SGV pitch at MP3 converge, and the periodic fluctuations almost completely disappear (only a weak fluctuation remains in total pressure). Notably, within the measurement range, the spanwise averaged curves exhibit a new fluctuation with a period of about 20° , consistent with the return channel vane pitch ($360^\circ/18 = 20^\circ$); moreover, the yaw angle, Mach number and static pressure curves at the eight SGV rotation angles almost completely overlap, a phenomenon not observed at the two upstream planes. This fluctuation with a period equal to the return channel vane pitch is induced by the downstream return channel vanes.

Fig. 16 shows the relative positions between the return channel vanes and the circumferential measurement angles.

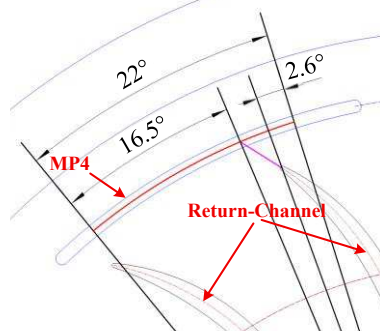


Fig.16 Relative positions between the return channel vanes and the circumferential measurement angles at the MP3 plane

As shown in Fig. 16, the measured circumferential range (24°) covers one complete return channel vane pitch (20°) plus a margin of about 2° on each side. The offset of the return channel vanes relative to the measurement window only changes the phase of the observed fluctuation and does not affect the identification of the 20° period. As shown by the spanwise averaged yaw angle in Fig. 15, the maximum is reached slightly left of the center of the measurement range, while the boundary points on both sides exhibit local minima, following an approximately sinusoidal distribution. This is because the flow near the vane leading edge is hindered by the blockage effect and the yaw angle decreases, whereas the flow at the passage center is less constrained and continues into the vane passage, so that the fluid converges towards the passage center under the action of the return channel vanes.

The pitch angle responds weakly to the vane disturbance and remains essentially stable. Total pressure is the only parameter at MP3 that retains a weak periodicity, but it is generally smooth, indicating that the periodicity of the upstream wake has been significantly attenuated. The static pressure responds most directly to the vane position and with the largest amplitude: its maxima correspond to the vane leading-edge positions and its minima to the main-flow regions of the vane passages, corroborating the yaw angle distribution. Meanwhile, with the total pressure essentially unchanged and the static pressure fluctuating periodically, the Mach number exhibits a periodic fluctuation in antiphase with the static pressure.

In summary, after passing through the U-bend, the upstream periodic wake is significantly homogenized, and the periodic fluctuation within one SGV pitch essentially disappears; however, a new fluctuation with a period of 20° —the return channel vane pitch—appears within the measurement range. This fluctuation results from disturbances propagating upstream from the return channel vanes through the potential-flow mechanism, and the vane row exerts significant feedback on the yaw angle, Mach number and static pressure ahead of its leading edges. To the authors' knowledge, this is the first quantitative experimental observation of this potential-flow feedback phenomenon on a stationary-component test rig, further demonstrating the value of stationary rigs as a complement to rotating rigs.

3.4 Flow Field at the Stage Outlet (MP4)

Fig. 17 presents the parameter contours at MP4. The contours at the eight SGV rotation angles are almost identical, indicating that, after the return channel and the L-bend, the wake simulated by SGV rotation has been completely homogenized in the circumferential direction. In the yaw angle contours, the hub-side and shroud-side angles are close to each other and contrast with the central main-flow region; the pitch angle contours contain a fixed-position high-angle region; and the Mach number and pressure vary monotonically from hub to shroud. These features are further quantified below using the averaged profiles.

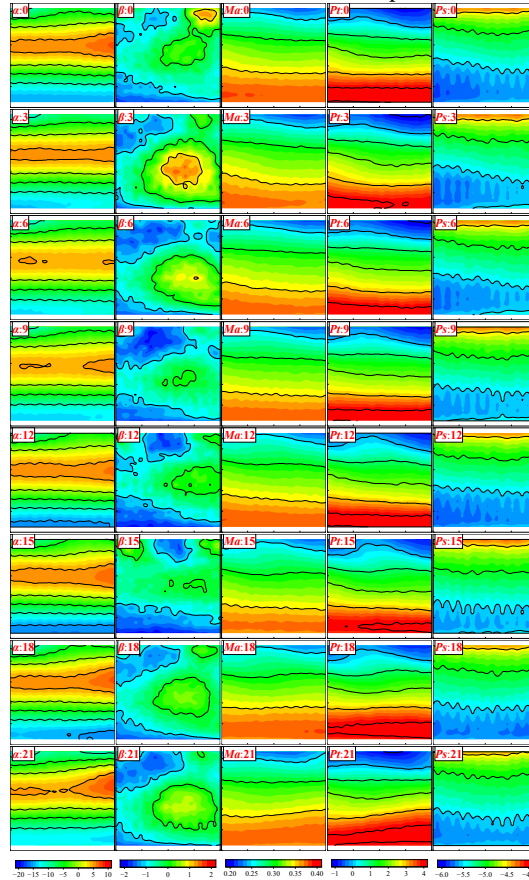


Fig.17 Contours of flow yaw angle (α), pitch angle (β), Mach number (Ma), total pressure (P_t), and static pressure (P) at the stage outlet (MP4) for eight SGV rotation angles

Fig. 18 presents the averaged parameter profiles at MP4. The results show that the averaged curves at the eight SGV rotation angles essentially overlap, and the weak periodicity remaining at MP3 has completely disappeared at MP4. Circumferentially, the distributions of all parameters are nearly horizontal, indicating good flow field uniformity; spanwise, however, significant gradients exist: affected by local reverse flow in the near-wall regions, the flow angle there deviates from the main flow by up to about 20°; the Mach number, total pressure and static pressure all decrease approximately linearly from hub to shroud, with the highest values on the hub side.

In summary, at the stage outlet, the periodic wake simulated by SGV rotation has completely disappeared, the exit flow fields at different rotation angles are highly consistent, and the circumferential uniformity is good; however, the spanwise non-uniformity remains significant: the yaw angle deviation between the near-wall regions and the main flow can reach about 20°, and the Mach number and pressure decrease monotonically from hub to shroud. Such spanwise distortion increases the dispersion of the downstream impeller inlet incidence along the blade height, which is detrimental to power output and operating stability; it is therefore necessary to control the stage outlet flow field through structural optimization.

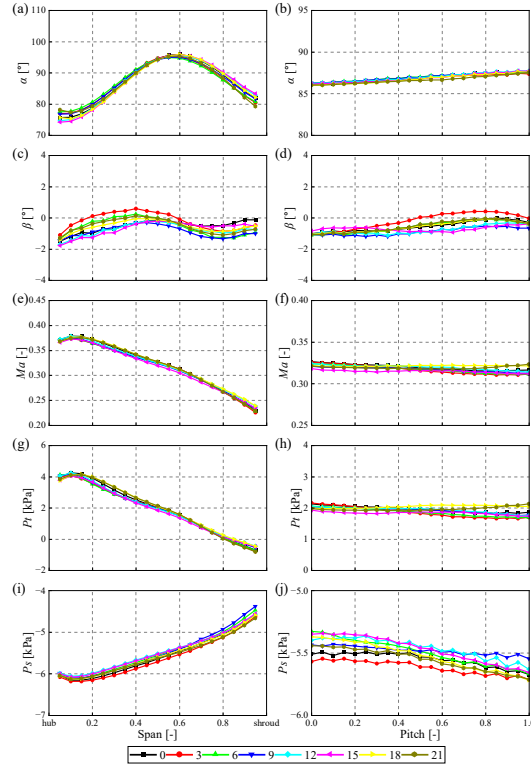


Fig.18 Circumferentially averaged spanwise profiles (left) and spanwise averaged circumferential profiles (right) of the flow parameters at MP4

4 Aerodynamic Performance Evaluation of the Stationary Components

This section quantitatively evaluates the aerodynamic performance of the stationary components using the total pressure loss coefficient (c_{pt}) and the static pressure recovery coefficient (c_{ps}). Fig. 19 presents the variations of the performance parameters of each component and of the whole stage at the eight SGV rotation angles.

As shown in Fig. 19(a) and (c), the static pressure recovery coefficient and total pressure loss coefficient of the vaneless diffuser (MP1–MP2) are essentially identical at the eight angles, with angle-to-angle relative variations below 3%. The “relative variation” is assessed against the Type A (repeatability) uncertainty and the scatter of the three repeated measurements (see Section 2.6); since the eight angles are measured within the same sequence, systematic errors affect all angles consistently. Together with Section 3.2, this indicates that the periodic wake generated by SGV rotation is reproduced and develops inside the vaneless diffuser, but the area-averaged diffuser performance barely changes with the circumferential position of the wake.

The static pressure recovery coefficient of the diffuser is about 0.47–0.49, and the total pressure loss coefficient is about 0.13–0.15. The static pressure recovery arises from the conversion of kinetic energy into static pressure through cross-sectional area expansion, while wall friction and shear mixing between the wake and the main flow constitute the irreversible total pressure loss. Note that the performance coefficient of each component is normalized by the dynamic pressure at its own inlet plane, so the values cannot be added directly; cross-plane comparison should be based on the cumulative coefficients computed with a unified reference, as shown in Fig. 19(b) and (d).

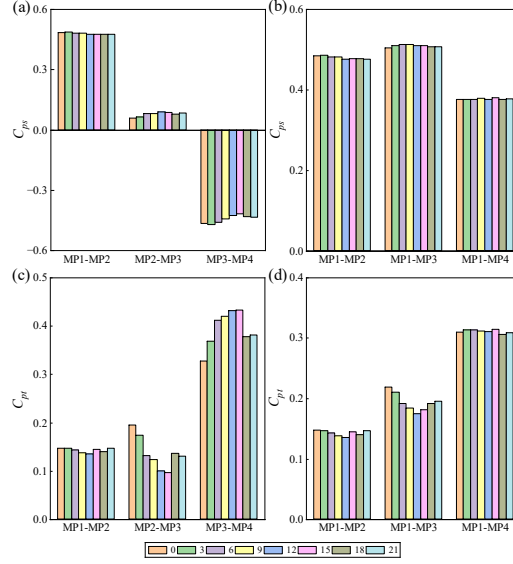


Fig.19 Performance comparison of the stationary components at the eight SGV rotation angles: (a) component static pressure recovery coefficients; (b) cumulative static pressure recovery coefficients; (c) component total pressure loss coefficients; (d) cumulative total pressure loss coefficients

Unlike the diffuser, the performance coefficients of the U-bend (MP2–MP3) and of the return channel + L-bend (MP3–MP4) vary non-monotonically with SGV rotation angle, and the two vary in antiphase: at angles where the U-bend loss is large, the return channel + L-bend loss is small, and the static pressure recovery coefficients follow the same opposite relationship. This correspondence is consistent with the variation of the MP3 flow field with rotation angle in Fig. 15, and both originate from the potential-flow feedback of the return channel vanes and the differences in mixing intensity within the bend.

The total pressure loss coefficient of the U-bend is about 0.10–0.20, with a peak-to-valley difference of about 0.1 among the eight angles, i.e. the maximum is about twice the minimum; its static pressure recovery coefficient does not exceed 0.09, indicating that the static pressure rise brought by the 180° turn is almost entirely offset by friction losses and intense mixing losses. Physically, the circumferential position at which the upstream wake enters the bend changes the strength and structure of the secondary flow (Dean vortices) in the bend, and hence the mixing loss in the bend.

The total pressure loss coefficient of the return channel + L-bend (MP3–MP4) is about 0.32–0.43, varying oppositely to that of the U-bend. The reason is that the two components are strongly coupled through the MP3 flow field: the more complete the mixing—and thus the larger the loss—in the U-bend, the more uniform the velocity distribution at MP3, the more favorable the incidence distribution at the return channel inlet, and the smaller the additional loss in the return channel; conversely, the residual non-uniformity at the bend outlet must be eliminated through additional mixing loss in the return channel.

The loss complementarity can be summarized as follows: at the eight SGV rotation angles, the losses of the U-bend and of the return channel + L-bend trade off against each other, while the total loss of the return passage (MP2–MP4) remains essentially unchanged. The circumferential position of the periodic wake only changes the distribution of the loss among the components of the return passage, without changing its total value.

The cumulative performance coefficients in Fig. 19(b) and (d) further confirm this finding: the cumulative total pressure loss coefficient and static pressure recovery coefficient of the whole stage (MP1–MP4) are essentially invariant with angle, and their variation is comparable to the angle-to-angle scatter of the diffuser (MP1–MP2) itself.

Quantitatively, the cumulative total pressure loss coefficient (MP1–MP4) is about 0.31 and the cumulative static pressure recovery coefficient is about 0.38; the angle-to-angle relative difference of the overall stage loss is below 2%, within the angle-to-angle repeatability criterion. In terms of component magnitude, the return channel + L-bend has the largest loss (c_{pt} about 0.32–0.43), while the vaneless diffuser (0.13–0.15) and the U-bend (0.10–0.20) are lower and of comparable order; the loss of the return passage system is much higher than that of the vaneless diffuser, with the return channel + L-bend being the dominant loss component. These results indicate that the total loss of the return passage is determined by the inlet area-averaged flow conditions and is independent of the specific circumferential position of the upstream wake, providing a simplified model for the performance prediction of stationary components.

Based on the uncertainty analysis in Section 2.6, the expanded uncertainties ($k = 2$) of the performance coefficients are summarized in Table 4.

Table 4 Expanded uncertainties ($k = 2$) of the performance coefficients

Component	Static pressure recovery coefficient c_{ps}	Total pressure loss coefficient c_{pt}
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Diffuser (MP1–MP2)	± 0.014 ($\pm 2.9\%$ – 3.0%)	± 0.016 ($\pm 10.7\%$ – 12.3%)
U-bend (MP2–MP3)	± 0.014 ($\geq \pm 15.6\%$, based on $\text{cps} \leq 0.09$)	± 0.016 ($\pm 8\%$ – 16%)
Return channel + L-bend (MP3–MP4)	± 0.014 ($\pm 9.3\%$)	± 0.016 ($\pm 3.7\%$ – 5.0%)

5 Discussion

This section discusses the effectiveness of the quasi-dynamic method, the mechanism of wake attenuation in the U-bend, the potential-flow feedback of the return channel vanes, the loss complementarity and the spanwise non-uniformity at the stage outlet; all the analyses are based on the measured data presented in Sections 3 and 4.

5.1 Effectiveness Assessment of the Quasi-Dynamic Simulation Method

The data in Sections 3 and 4 support the effectiveness of the quasi-dynamic method in three respects. First, the parameters at MP1 vary very little with SGV rotation angle (relative variation $< 1\%$), showing that rotation only changes the circumferential phase of the wake and does not change the flow field of the SGV itself. Second, MP2 exhibits periodic fluctuations corresponding one-to-one to the rotation angle, proving that the wake migration is successfully reproduced and transmitted to the diffuser outlet. Third, the periodicity within one SGV pitch essentially vanishes at MP3 (peak-to-valley non-uniformity reduced from about 5.2% to about 0.5%) and disappears completely at MP4, consistent with the time-averaged trend of wake attenuation along the stationary components in real machines. The limitations of the method are that it cannot reproduce the instantaneous wake shedding, tip leakage vortices or unsteady rotor–stator coupling, and that the test Reynolds number (about 2.6×10^6) is lower than that of the real machine at full pressure. The conclusions of this paper are therefore restricted to time-averaged flow structures and the loss distribution among components.

5.2 Mechanism of Periodic Wake Elimination in the U-bend

Section 3.3 shows that the peak-to-valley non-uniformity of the spanwise averaged total pressure, about 5.2% at MP2, drops to about 0.5% at MP3 after the U-bend—an attenuation of about 90%. This homogenization results from the combined action of three effects. First, the 180° turn induces strong Dean-vortex secondary flows; estimated from the Dean number definition $De = Re \cdot (D_h/2R_c)0.5$, the Dean number under the present experimental conditions is about 1.5×10^4 , in the strong secondary-flow regime, which can drive large-scale fluid exchange between the hub and shroud sides. Second, the strong shear in the bend raises the turbulent dissipation rate; estimated from $\varepsilon \propto u^3/l$, a two- to three-fold increase of the turbulent fluctuating velocity corresponds to an approximately 8- to 27-fold increase of the dissipation rate, so that the wake structures are rapidly dissipated. Third, the geometric asymmetry—radial inflow and axial outflow—disrupts the original circumferential phase relationship of the wake. Compared with previous low-flow-coefficient stationary-rig studies, in which the homogenizing effect of the bend was only observed qualitatively, the present work provides, for the first time, a quantitative attenuation rate of about 90% under high-flow-coefficient conditions ($\varphi = 0.2$). It should be emphasized that the “elimination” by the U-bend acts only on the upstream wake; a new fluctuation with a period of 20° (the return channel vane pitch) is still observed within the MP3 measurement range.

5.3 Potential-Flow Feedback of the Return Channel Vanes on the Upstream Flow Field

Section 3.3 further shows that the static pressure at MP3 is highly sensitive to the vane position, following an approximately sinusoidal distribution with a period of 20° , whereas the total pressure responds only weakly. This combination of features is consistent with a potential-flow disturbance mechanism: the stagnation region (high pressure) at the vane leading edge and the acceleration region (low pressure) in the passage form a periodic static pressure field upstream of the vane row, while the total pressure remains essentially unaffected because potential-flow disturbances themselves do not directly produce dissipation. These conclusions are based on a 24° circumferential measurement range (about 1.2 vane pitches), and the full-annulus consistency still requires further measurement or numerical verification. From an engineering perspective, the leading-edge shape, setting angle and circumferential position of the return channel vanes modify the MP3 static pressure distribution through this potential-flow coupling, thereby affecting separation and loss in the U-bend; the return channel and the bend should therefore be designed as a coupled system.

5.4 Physical Origin of the Loss Complementarity in the Return Passage

The key finding of Section 4 is that the losses of the U-bend and of the return channel + L-bend vary in opposite directions with SGV rotation angle, while the total loss of the return passage and of the whole stage remains constant (angle-to-angle relative difference $< 2\%$). The reason is that the area-averaged total pressures at MP2 and MP4 do not vary significantly with rotation angle, so the upstream and downstream boundary conditions for the total loss are fixed; the rotation angle only changes the distribution of the loss among the components—when the mixing in the bend is thorough and the bend loss is large, the MP3 flow field is more uniform and the additional loss in the return channel is smaller, and vice versa. Because the SGVs of previous stationary rigs were mostly fixed at a single position, this loss complementarity could not be observed;

the present results show that, for performance prediction of the return passage, the inlet area-averaged flow conditions can be used as the boundary without tracking the specific circumferential phase of the wake.

5.5 Engineering Implications of the Spanwise Non-uniformity at the Stage Outlet and Remedial Measures

The stage outlet (MP4) flow is circumferentially uniform but exhibits significant spanwise distortion: the yaw angle deviation between the near-wall regions and the main flow can reach about 20° , and the Mach number decreases approximately linearly from hub to shroud. Possible remedial measures include installing guide vanes or a flow-straightening grid at the L-bend outlet, optimizing the meridional profile of the L-bend to weaken the curvature discontinuity, adopting asymmetric return channel vanes with spanwise-varying setting angle or camber, and carrying out coordinated inter-stage matching design. Similar optimization-oriented approaches have proven effective for S-shaped compressor transition ducts [31] and for one-dimensional turbomachinery aerodynamic design [32].

6 Conclusions

Using a quasi-dynamic test rig designed and built in-house, the internal flow field of the stationary components of a high-flow-coefficient ($\varphi = 0.2$, $\mu = 0.8$) centrifugal compressor was systematically measured and analyzed with a high-precision five-hole probe. In the feasibility validation phase, a dense point arrangement was used: 19 (spanwise) $\times$ 49 (circumferential, 0.5° interval) = 931 points at each of MP1–MP3 and 49 (circumferential) $\times$ 27 (spanwise) = 1323 points at MP4, totaling about 4100 points per complete survey; in the subsequent detailed analysis, the number of measurement points was reduced to $17/21$ spanwise points and 25 circumferential points, about 43.7% of the validation phase. By rotating the SGV through one blade pitch in steps of about 3° (eight circumferential positions), the circumferential migration of the impeller wake was simulated on a stationary rig for a high-flow-coefficient centrifugal compressor, and high-resolution flow field data were obtained at four key planes. The main conclusions are as follows:

(1) The effectiveness of the quasi-dynamic simulation method is experimentally verified. Rotating the SGV in steps of about 3° over one complete blade pitch successfully simulates the circumferential migration of the impeller wake. The parameters at MP1 vary very little with SGV rotation angle (relative variation $< 1\%$), proving that rotation does not change the flow field structure of the SGV itself; MP2 exhibits pronounced periodic fluctuations whose peak/valley positions correspond one-to-one to the rotation angle, confirming the successful simulation of the circumferential wake migration; the periodic fluctuation within one pitch essentially disappears at MP3 (except for a weak total pressure fluctuation) and disappears completely at MP4.

(2) The complete evolution of the wake along the stationary components is revealed. The impeller wake undergoes four stages along the stationary components—generation (MP1), mixing and migration (MP2), attenuation (MP3) and disappearance (MP4). At MP1, the yaw angle is about 48° after excluding the wake and separation regions; after the vaneless diffuser, the trailing-edge separation zone has been completely mixed out, while the hub-side yaw angle increases; after the U-bend, the yaw and pitch angles exhibit an inverted-S distribution and mirror each other.

(3) The homogenizing effect of the U-bend and the potential-flow feedback mechanism of the return channel vanes are quantitatively revealed. Measured by the peak-to-valley non-uniformity of the spanwise averaged total pressure within one SGV pitch, the U-bend attenuates the non-uniformity of the upstream periodic wake from about 5.2% to about 0.5% (an attenuation of about 90%), demonstrating a significant homogenizing effect. Meanwhile, the inlet flow structure of the return channel barely changes with the circumferential position of the upstream wake; within the measurement range, a new fluctuation with a period of 20° (the vane pitch), propagating upstream from the return channel vanes through the potential-flow mechanism, is observed.

(4) The loss complementarity and the constant-total-loss behavior in the return passage are identified. The loss of the return passage system is much higher than that of the vaneless diffuser: the vaneless diffuser (c_{pt} about 0.13 – 0.15) and the U-bend (about 0.10 – 0.20) have lower losses of comparable magnitude, while the return channel + L-bend is the dominant loss component (about 0.32 – 0.43). The losses of the U-bend and of the return channel + L-bend vary in opposite directions with SGV rotation angle, but the cumulative total pressure loss coefficient of the whole stage (MP1–MP4) is about 0.31 and the cumulative static pressure recovery coefficient is about 0.38 , with angle-to-angle relative differences below 2% . The total loss of the return passage is determined by the inlet area-averaged flow conditions; the periodic wake only changes the distribution of the loss among the components without changing the total loss.

(5) The spanwise non-uniformity at the stage outlet is quantitatively evaluated and remedial measures are proposed. The stage outlet (MP4) flow is circumferentially uniform but still exhibits significant spanwise gradients: the yaw angle deviation between the near-wall regions and the main flow can reach about 20° , and the Mach number decreases approximately linearly from hub to shroud. Remedial measures such as installing guide vanes at the L-bend outlet, optimizing the meridional profile, or adopting asymmetric return channel vane designs are recommended.

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Data availability Data are available from the corresponding author upon reasonable request.

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Declarations

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