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Experimental investigation of surge and stall in a turbocharger centrifugal compressor with a vaned diffuser

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ABSTRACT

The stable flow range of a compressor is dominantly limited by flow instability, known as surge and stall. In this paper, surge and stall in a turbocharger centrifugal compressor with a vaned diffuser are investigated by experiments. The transitional process from stable to unstable conditions is quite different at different rotating speeds. With decreasing mass flow rate, the compressor successively experiences stable conditions and deep surge at low speed, while the compressor experiences stable conditions, mild surge and deep surge at high speed. During experiments, deep surge occurs at all tested rotating speeds with different characteristics and behavior, and a deep surge cycle includes three periods, named recovery period, oscillation period and breakdown period. The dynamic experimental signal shows that pressure behavior in the oscillation period differs at different rotating speeds: high frequency fluctuation occurs at low speed, while mild surge appears at high speed. Mild surge occurs independently or appears within a deep surge cycle with a frequency close to the Helmholtz resonant frequency of the compression system. The behavior of mild surge at the vaned diffuser inlet is different along the circumferential direction, which is a subversion of the conventional view that surge is an axisymmetric phenomenon. It is highly possible that the non-axisymmetric behavior of mild surge is induced by the volute. Because the volute affects the flow field structure significantly, a new volute design method considering the circumferential non-uniformity of the flow field may be established and developed to improve the compressor stability.

1. Introduction

Energy conservation and emission reduction are two global problems for this new century, and the transport sector shares a large proportion of energy consumption and emission. According to Davis et al. [1], nearly two-thirds of transport-related emissions are from road transport, whose power is mainly from oil, and road transport emissions are projected to grow rapidly over the next several decades because of the driving effect of developing economies. Therefore, increasing the efficiency and reducing the emissions of road transport engines (internal combustion engine is most widely used) will be of great significance. One of the most potent technologies to achieve this goal is engine downsizing by turbocharging [2], and the centrifugal compressor is the core component of a turbocharger.

A turbocharger centrifugal compressor applied in an internal combustion engine must not only meet pressure ratio and efficiency requirements but also achieve a wide, stable flow range to ensure that it works well throughout the entire operating envelope [3]. The main limitation to obtaining a wide, stable flow range in compressors is the phenomenon of aerodynamic instability, known as stall and surge. When operating at a reduced mass flow, the compressor suffers stall or surge; at the same time, the stable operating boundary is drawn because the compressor cannot work normally at a smaller mass flow.

Surge is a global instability of the entire compression system, and an overall oscillation of annulus-averaged flow in the system can be observed during surge; sometimes, even backflow appears, as seen in the experimental results by Greitzer [4]. Usually, surge can be divided into two types according to the occurrence of backflow: mild surge without backflow and deep surge with backflow. Mild surge often results in minor flow oscillation and reduced efficiency, while deep surge often leads to drastic fluctuations in pressure, flow rate and noise, and catastrophic failure of the whole system can occur [5].

Stall is different from surge; it is an internal compressor instability related to flow separation, and it often occurs earlier than surge, although no evidence shows that stall triggers surge in some way. In stall, one or several stall cells usually arise locally and

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Nomenclature

A	equivalent cross-sectional area of the duct
a	speed of sound
α	flow angle
B	a non-dimensional parameter on Greitzer's theory
c	absolute velocity
d	diameter
f	frequency
f_H	Helmholtz frequency
l	equivalent length of the duct
p_r	reference dynamic pressure
T	temperature
u	blade speed
V	volume of compressed air
ρ	air density
γ	specific heat ratio

Subscripts

0	ambient parameter
1	impeller inlet
2	impeller outlet, diffuser inlet
θ	circumferential component
ave	averaged value
rev	reference value

propagate circumferentially with a lower speed than that of the rotor. It is recognized that there are two different types of stall inceptions: modal type and so-called spike type. These two stall inceptions differ in both timescale and length scale [6] and can be distinguished according to the amplitude of the oscillation and the rotating speed of the stall cells. Both analytical [7,8] and experimental [9] investigations show that modal-type stall occurs at or slightly beyond the peak of the stagnation-to-static pressure rise of the compressor [10]. Other works about modal-type stall inception have been carried out by Haynes et al. [11] and Paduano et al. [12], and their investigations show that the amplitude is relatively small and stall cells rotate with 20–50% of the impeller speed in the circumferential direction. Regarding spike-type stall inception, usually, an upward spike emerges in the transient pressure curve and a downward spike occurs in the velocity curve [6], and the rotating speed of the stall cells is faster than that of the modal type [13]. According to Vo et al. [10], the spike rotates at approximately 50–70% of the rotor speed. Vo et al. [10] also note that spike occurs at conditions in which the stagnation-to-static pressure rise characteristic has a negative slope.

Simplified theories have been developed to help explain the instability phenomenon and predict the instability boundary. Emmons et al. [14] give a conventional explanation for the rotation mechanism of stall cells. Greitzer [15] and Stenning [16] developed one-dimensional models to investigation compressor stability, and an important non-dimensional parameter B is developed, which determines whether the system will enter rotating stall and surge. Moore et al. [7,17,18] established a minor-disturbance model, which deduces that high-order propagation wave disturbance causes the rotating stall. Moore's model was improved by Spakovszky [19], who added subcomponent modules to the model and used it to predict the system resonances. Recently, Zheng and Liu [20] introduced a new half-quantitative "mass-spring-damping" model based on the work of Stenning [16], and this model explains the diversity of surge or stall patterns at different operating conditions.

Experimental and numerical investigations are also carried out on compressor stabilities. Research by Vo et al. [10,21] found that backflow and spillage flow leaking from the tip clearance and impinging on the trailing edge on the pressure side are the main cause of the spike-type disturbance. Lei et al. [22] established a criterion for "hub-corner stall". Toyama et al. [23] experimentally studied the instability of a centrifugal compressor with a vaned diffuser, and pressure variation during surge was measured at different locations inside the compressor. Everitt et al. [24] captured the spike-type stall inception in the vaned diffuser by setting an isolated simulation model. Kammer and Rautenberg [25] found that low-frequency rotating stall tends to occur in the diffuser when a centrifugal compressor with a vaneless diffuser operates at high speed. Zheng and Liu [20] demonstrated a panoramic map of instability evolution across all typical operations. They also found a special instability behavior of a high-pressure ratio centrifugal compressor, called a "two-regime-surge", and both the phenomenon and mechanism have been investigated in detail [26].

According to excellent analytical and experimental works, significant improvements have been achieved in understanding the instability phenomenon and the instability mechanism of compressors. However, most works are focused on rotating stall inceptions, and there have been relatively few experimental studies of surge in centrifugal compressors, especially those with vaned diffusers. In addition, few investigations focus on the circumferential difference caused by the non-axisymmetric volute during surge. In this paper, a turbocharger centrifugal compressor with a vaned diffuser is investigated by experiment. The different instability behavior at different rotating speeds are presented, and the transient process from stable points to surge is described. Meanwhile, the circumferential difference in pressure oscillation at the vaned diffuser inlet during surge is also captured and discussed.

2. Experimental setup

The schematic of the test rig facility is shown in Fig. 1. The compression system composes of the centrifugal compressor and the inlet and outlet pipes. During the experiment, pressurized air is heated in a combustion chamber in front of the turbine, and the turbine extracts energy from the high pressure and high temperature gas and drives the compressor with a required rotating speed.

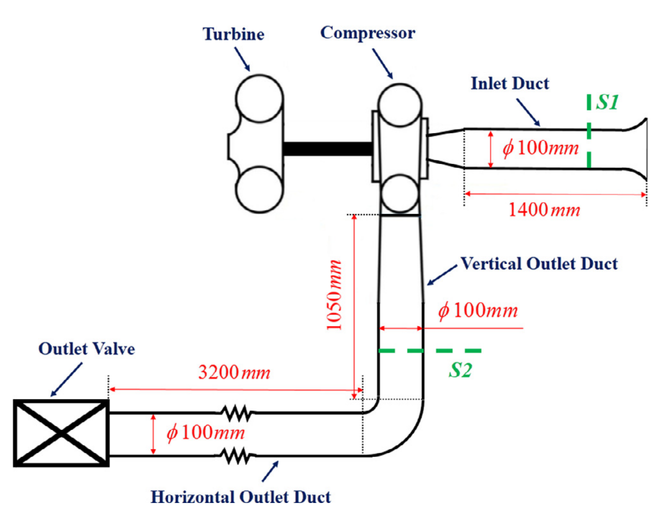

Fig. 1. Schematic of experimental test rig.

Flow parameters are measured at both inlet and outlet of the compressor. Low-frequency-response probes are installed at section *S1* and section *S2* to measure the mass flow rate, the total/static pressure and temperature. In addition, the rotating speed, the ambient pressure and temperature are also obtained. The measuring error of the temperature probes at both inlet and outlet is ±0.25 K, while the measuring error of the pressure probes is 0.05% of the measuring range (−30–30 kPa at section *S1*, and 0–700 kPa at section *S2*). All of the steady data acquired above are solved to identify the operating conditions. Meanwhile, the compressor performance map is also obtained.

The investigated compressor is a turbocharger centrifugal compressor with a pressure ratio over 3.0. The detailed specifications and the components photos are shown in Table 1 and Fig. 2, respectively.

For the purpose of capturing the instability phenomenon during surge or stall, sixteen high-frequency-response pressure probes are mounted into the compressor casing, as shown in the photo in Fig. 2. These probes are *XTE-140 (M) SERIES PRESSURE TRANSDUCERS* of Kulite Semiconductor Products, Inc. The sensing principle of the probe is "Fully Active Four Arm Wheatstone Bridge Dielectrically Isolated Silicon on Silicon", and it uses a standard miniature silicon diaphragm to obtain extremely high natural frequencies in the smallest thread mount available. The probe's natural frequency is 240 kHz (pressure range of 25 psi), 300 kHz (pressure range of 50 psi) and 380 kHz (pressure range of 100 psi). In addition, a protective grid is used in each probe to protect the sensing part. The layout of these high-frequency-response pressure probes is shown in Fig. 3. Three positions—the impeller inlet (marked as "A"), the diffuser inlet (marked as "B"), and the diffuser outlet (marked as "C")—are selected as measurement sections. Circumferential locations of probes at each measurement section are shown in Fig. 3. Four 25 psi probes are placed at 30, 150, 210 and 270 deg and numbered from A1 to A4, respectively (0 deg is defined at the section antiparallel to the axis of the volute exit). Six 50 psi probes numbered from B1 to B6 are installed at 30, 75, 120, 142.5, 187.5 and 277.5 deg to the diffuser inlet, respectively. Similarly, six 100 psi probes numbered from C1 to C6 are installed at 60, 105, 150, 172.5 and 307.5 deg to the diffuser outlet, respectively. Probes numbered with the same Arabic number at sections B and C share the same diffuser passage. Therefore, the probes at sections B and C measure the pressure at *passages 4, 6, 8, 9, 11 &15* (the flow passage whose outlet interacts with the x-axis is defined as *passage 1*, as shown in Fig. 3). The probe distribution in the circumferential direction is not uniform, and most probes are placed close to the volute tongue (150 deg) because more data are desired to investigate the influence of the volute on instability.

Two high-response thermocouples numbered T1 and T2 are installed at the impeller inlet (section A). The signals from the thermocouples are used to help determine the surge point, and detailed information is given by Liu and Zheng [27]. In addition, a microphone is placed in front of the entrance of the inlet duct to capture the sound signal during surge, and the sound data are used to help determine the surge patterns.

Table 1
Main parameters of the compressor.

Parameters	Value (unit)
Impeller blade number	16
Inlet shroud diameter	62.70 (mm)
Impeller outlet diameter	100.00 (mm)
Impeller backsweep angle	34.50 (°)
Diffuser vane number	16
Diffuser inlet diameter	117.60 (mm)
Diffuser outlet diameter	144.40 (mm)
Diffuser width	3.98 (mm)
Throat area of volute	3300 (mm^2)

3. Compressor performance and stability

The compressor performance map obtained from experimental data is shown in Fig. 4. Five specific rotating speeds, from 40,000 r/min to 90,000 r/min, are tested. The dash line lying in the left side of the figure shows the boundary beyond which the compressor will experience deep surge conditions. According to Liu and Zheng [27], the temperature standard deviations (SD) at stable points are relatively low yet will rise notably once deep surge occurs. The temperature SD values of different operating points at each rotating speed are presented in Fig. 5, by which deep surge points of each specific speed line are recognized. In the performance map in Fig. 4, these deep surge points are marked by square (the mass flow rate of these deep surge points in the MAP is not real because the mass flow rate fluctuates drastically when deep surge occurs).

The mild surge point (marked by a diamond in Fig. 4) at 90,000 r/min is recognized by the sound and the pressure signals (detailed analysis of the pressure signal will be carried out later), but no mild surge points are captured at other tested rotating speeds. The pressure oscillations obtained by high-frequency-response probes are used to recognize stall points at each rotating speed, and these stall points are marked by triangles in Fig. 4. All stall points occur with large-amplitude pressure oscillations at the impeller inlet and/or diffuser inlet; unfortunately, no sufficient evidence proves the occurrence of rotating stall during the experiment.

According to Fig. 4, the transient process from stable condition to deep surge condition is clear. At low speed, with decreasing mass flow rate, the compressor experiences stable points, stall points, and then the deep surge point. While at high speed, the compressor first goes into mild surge from stable points and then suffers deep surge.

4. Surge and stall characteristics and behavior

In this section, deep surge and mild surge characteristics and behavior are displayed and analyzed in detail. Pressure oscillations and temperature fluctuations during deep surge conditions are compared between rotating speeds, and the influence of the rotating speed on the compressor instability is investigated. Meanwhile, deep surge cycles are also analyzed in detail to describe the surge process more clearly. A comparison of pressure oscillations between different mass flow rates is conducted to analyze the evolution process from stable condition to unstable condition. The propagation of the instability is studied by comparing pressure oscillations in different streamwise locations. Finally, behavior of mild surge and a non-axisymmetric phenomenon during mild surge condition are introduced.

4.1. Behavior of deep surge at different rotating speeds

When the compressor operates at deep surge points of different rotating speeds, pressure oscillations at different locations and temperature fluctuations at the impeller inlet are shown in Fig. 6. Each subplot includes pressure or temperature information at deep surge points of five tested speeds, corresponding to points marked by square in the performance map in Fig. 4. Fig. 6(a–c) illustrate pressure oscillations at different measurement sections, while Fig. 6(d) presents temperature fluctuations at the impeller inlet. For each rotating speed, the ordinate scales in Fig. 6(a–c) are of the same magnitude of reference dynamic pressure (defined as (1)):

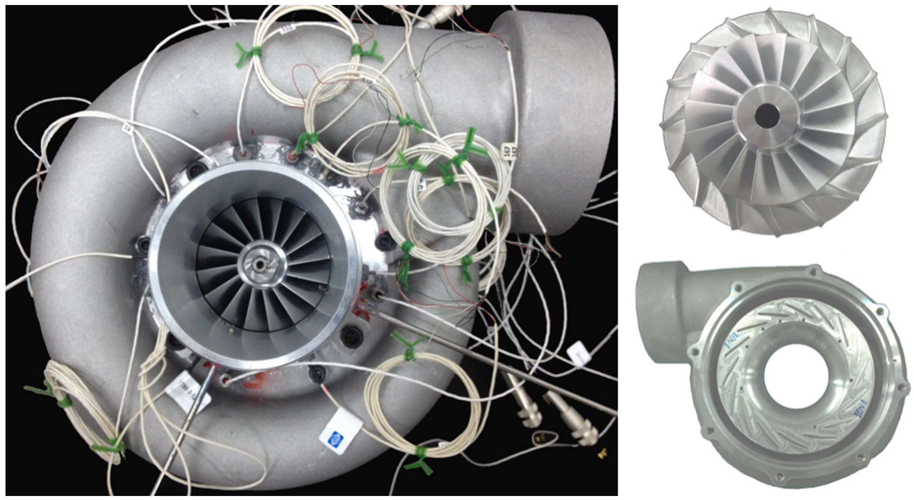

Fig. 2. The tested compressor with vaned diffuser.

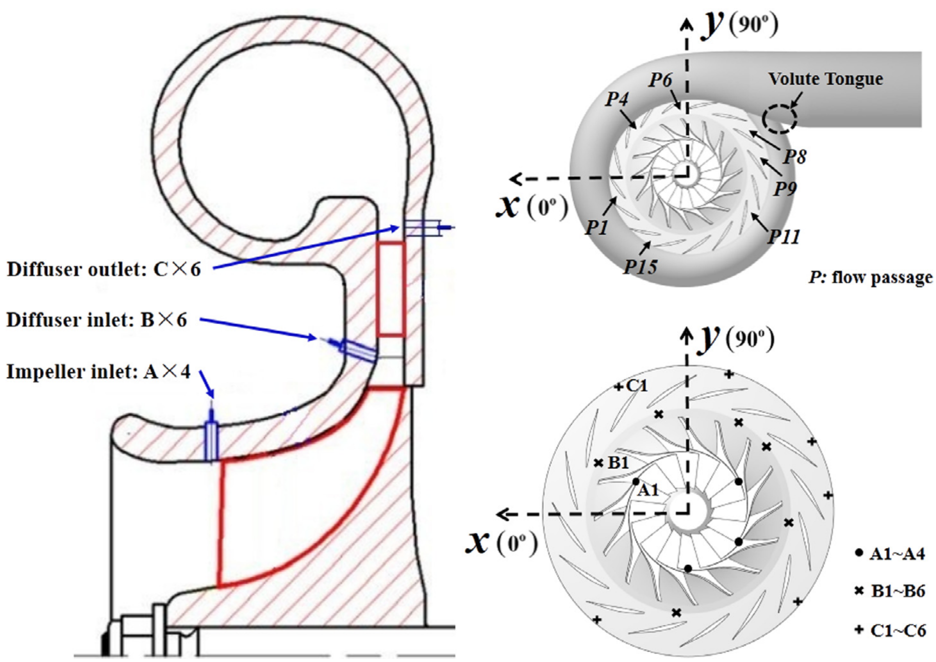

Fig. 3. The layout of the high-response probes and the diffuser passage numbers.

$$p_r = \frac{1}{2} \times \rho_0 \cdot u_2^2 \tag{1}$$

where ρ_0 is the air density of the atmosphere, approximately 1.205 kg/m^3, and u_2 is the tangential velocity of the impeller outlet.

A deep surge cycle (marked as "DS" in Fig. 6a–c)) includes three periods. As shown in Fig. 7, following the end of the last deep surge cycle is the "recovery period", during which the pressure fluctuates with minor amplitude and the static pressure at the diffuser outlet recovers to a high level quickly. The succeeding period is named the "oscillation period". The pressure oscillates with remarkable amplitude in this period, and the static pressure no longer increases. After the oscillation period, the whole compressor suddenly falls into an excessive oscillation with increased noise. The pressure at the impeller inlet and vaned diffuser inlet remains at a relatively constant level after a sudden and sharp spike and then drops to a low level and fluctuates. The pressure at the vaned diffuser outlet drops quickly to a low level. This period is named the "breakdown period". During both the recovery period and oscillation period, the temperature at the impeller inlet decreases gently, and it increases quickly once the breakdown period begins, as shown in Fig. 6(d). Because there is no high-frequency and large-amplitude oscillation in the temperature signal, it is easy to recognize and is used to capture the frequency of deep surge correctly or determine the beginning and ending of the breakdown period.

According to Fig. 6, deep surge is different at different rotating speeds, especially the oscillation period. With rotating speed changes, the pressure oscillation presents different characteristic and behavior. Regarding the tested centrifugal compressor, when operating at low and middle speeds (from 40,000 to 78,000 r/min), it suffers high-frequency and large-amplitude pressure oscillation (stall) during the oscillation period. Nevertheless, when the rotating speed is high (90,000 r/min), mild surge cycles occur during the oscillation period (as marked by dash line circles in

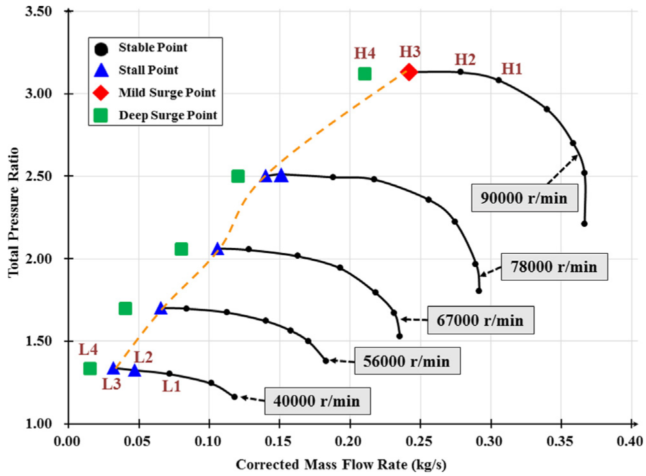

Fig. 4. Performance MAP of the tested compressor.

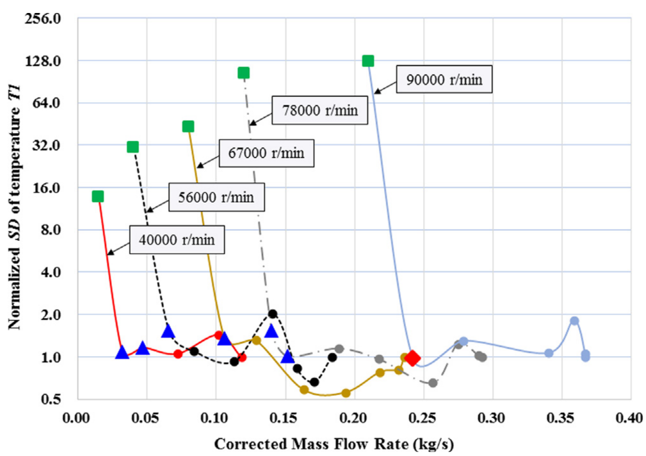

Fig. 5. SD of temperature at different operating points.

Fig. 6). In other words, a deep surge cycle could include several mild surge cycles at high rotating speed. Fig. 6 also shows that the duration of the oscillation period changes at different rotating speeds, and this is the main cause of the change in deep surge frequency. In addition, the duration of the oscillation period can also change at different deep surge cycles of a tested rotating speed (rotating speed varies within a small range as deep surge brings strong disturbance to the system). Particularly, the duration of the oscillation period at 67,000 and 78,000 r/min can be long or short (Fig. 6); thus, the cycle period of deep surge changes with time. If the duration of oscillation is very short, the breakdown period of the next surge cycle can successively occur shortly after the last deep surge cycle.

4.2. Development and propagation of the pressure oscillation

The previous section discussed the surge phenomenon, and the results show that the characteristics and behavior of deep surge are quite different at different rotating speeds, especially during the oscillation period. In addition, it is found that a deep surge cycle includes several mild surge cycles in the oscillation period at high speed. In this section, the development of the pressure oscillation from stable status to deep surge will be discussed, and attention will be paid to the streamwise propagation of pressure oscillations at surge points. The rotating speeds of 40,000 r/min and 90,000 r/min will be taken as analyzed objects to reflect pressure propagations at low speed and high speed.

When the compressor operates at 40,000 r/min, the impeller inlet pressure oscillations measured at different operating conditions are displayed in Fig. 8. The pressure oscillation is slight when the mass flow rate is 0.072 kg/s. As the mass flow rate decreases to 0.047 kg/s, the amplitude of pressure oscillation increases significantly, indicating that the compressor begins to suffer instability. When the mass flow rate is further reduced to 0.032 kg/s, the amplitude of the pressure oscillation decreases unexpectedly, whereas the frequency of the pressure oscillation slightly increases. To judge whether rotating stall occurs, pressure oscillations obtained by probes A2, A3 and A4 are enlarged and presented in the top-right side of Fig. 8. Unfortunately, no evidence of rotating stall—for example, the oscillation phase lag—is shown in the enlarged view. Works to search for rotating stall were also performed at other operation conditions and other measurement sections, and none of them provide sufficient evidence of rotating stall.

As the mass flow rate continues to decrease, the deep surge occurs with a much more drastic pressure oscillation, as shown in the bottom of Fig. 8. The "self-similarity" introduced by Zheng and Liu [20] is found in the deep surge cycle of the tested compressor. "Self-similarity" indicates a phenomenon occurring in compressors in which the transient process from stable status to unstable status is replayed in a deep surge cycle. According to the analysis above, when operating at low speed (40,000 r/min), the tested compressor first experiences large-amplitude pressure oscillation (mass flow rate equals 0.047 kg/s), then experiences smaller amplitude but higher frequency pressure oscillation (mass flow rate equals 0.047 kg/s), and finally enters deep surge. As shown in the bottom of Fig. 8, during the oscillation period of a deep surge cycle, large-amplitude pressure oscillation and small-amplitude pressure oscillation occur successively, and then dramatic and violent pressure oscillation with much larger amplitude but much lower frequency occurs during the breakdown period. Therefore, during a deep surge cycle, the compressor experiences the transient process from stable to unstable conditions. In other words, if the process from stable to unstable conditions is clear, the deep surge behavior can be predicted based on the "self-similarity" principle.

Fig. 9 shows the pressure oscillation in different measurement sections inside the compressor when deep surge occurs at low speed. During the oscillation period, noticeable oscillations occur first at the impeller inlet; at the same time, minor fluctuations appear at the diffuser inlet. At the end of the oscillation period, several spikes in pressure are detected at the diffuser inlet. The compressor enters the breakdown period, during which both the impeller inlet and diffuser inlet suffer large-amplitude oscillation, but the frequency of the oscillation at the diffuser inlet seems higher than that at the impeller inlet.

The pressure oscillation at different sections provides the evidence that the impeller inlet more easily becomes unstable at low speed, because larger oscillation occurs here first. Considering this result, the performance of the impeller inlet has a significant effect on the stability of the whole system. When attempting to enhance the stability of the compressor, especially at low rotating speeds, flow control methods at the impeller inlet, such as the impeller casing treatment, should be considered first without hesitation.

At 90,000 r/min, pressure oscillations at different measurement sections are presented in Fig. 10; it is noted that pressure is relatively stable at the impeller inlet and diffuser outlet during the recovery period, while pressure at the diffuser inlet oscillates with high frequency during the whole surge cycle, including the recovery period. Details are shown in the enlarged views at the bottom

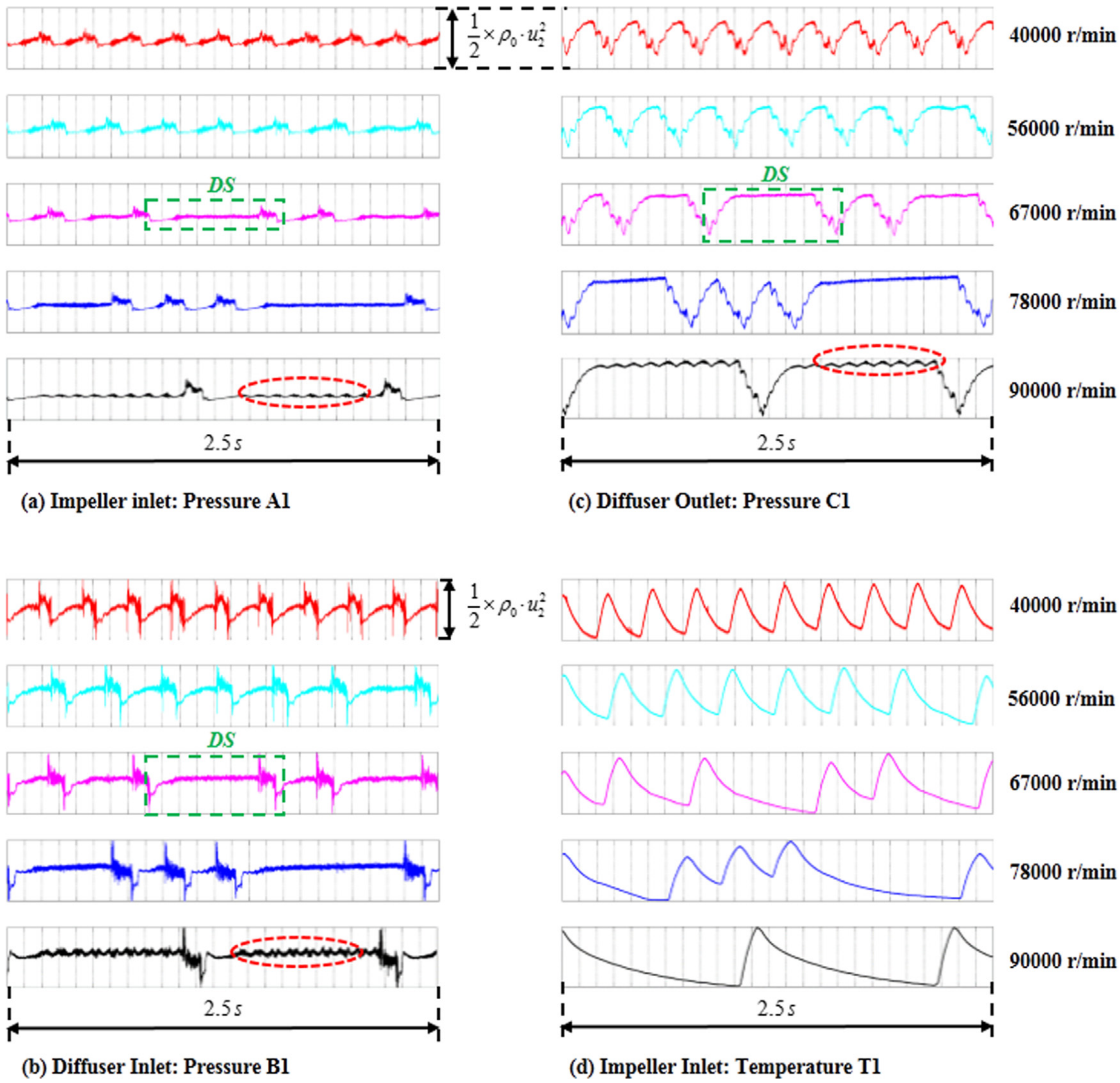

Fig. 6. Pressure oscillation and temperature fluctuation at surge points of different rotating speeds.

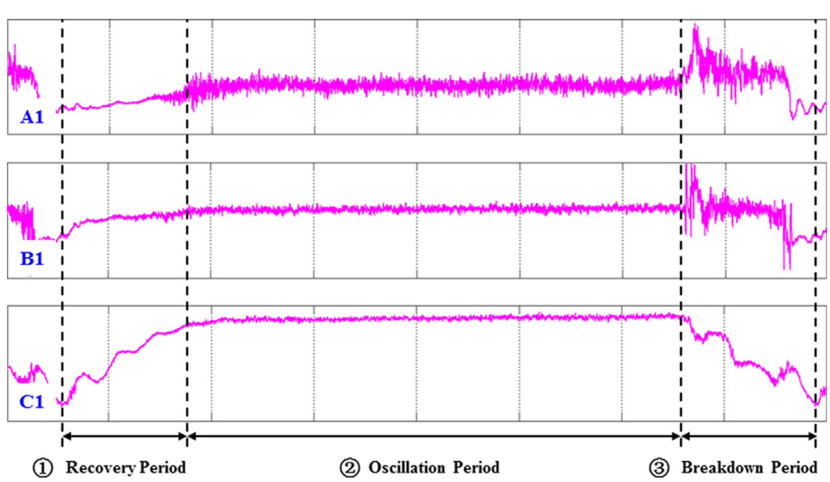

Fig. 7. Enlarged view of a deep surge cycle marked as "DS" in Fig. 6.

of Fig. 10. Therefore, pressure oscillation occurs first at the diffuser inlet according to the "self-similarity" principle, which indicates that the location where instability occurs first changes from the impeller inlet to the diffuser inlet when the compressor operates at high speed. Actually, both experimental works [3,19,28,29] and analytical work [30] indicate that the instability of a centrifu-

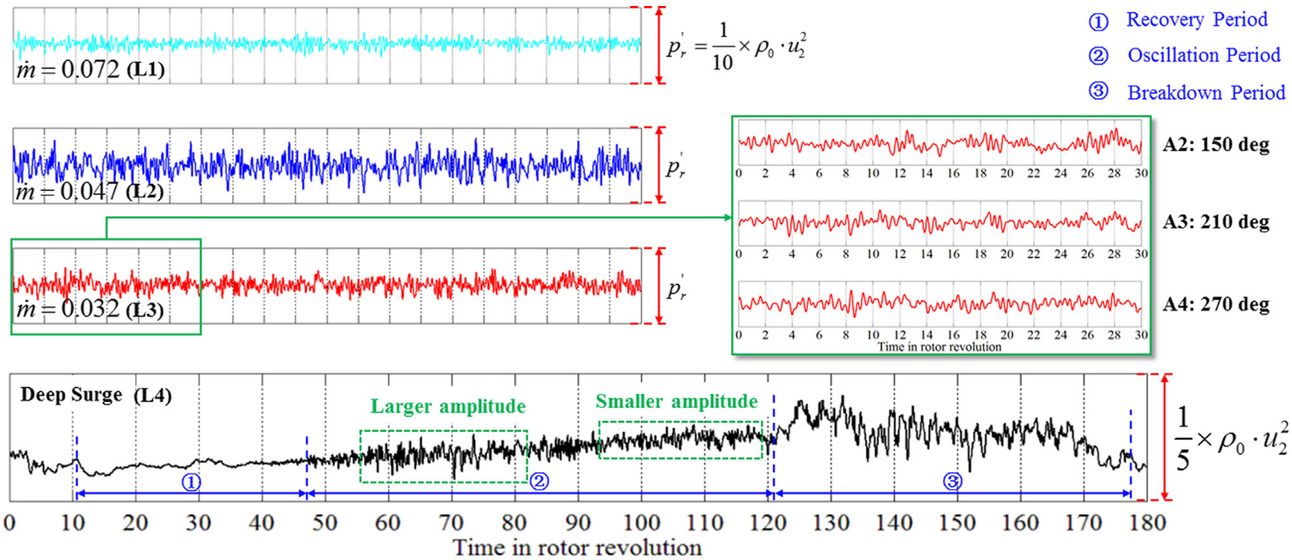

Fig. 8. Impeller inlet pressure oscillation at conditions with different mass flow rate (marked by "L1~L4" in Fig.4 @ 40,000 r/min).

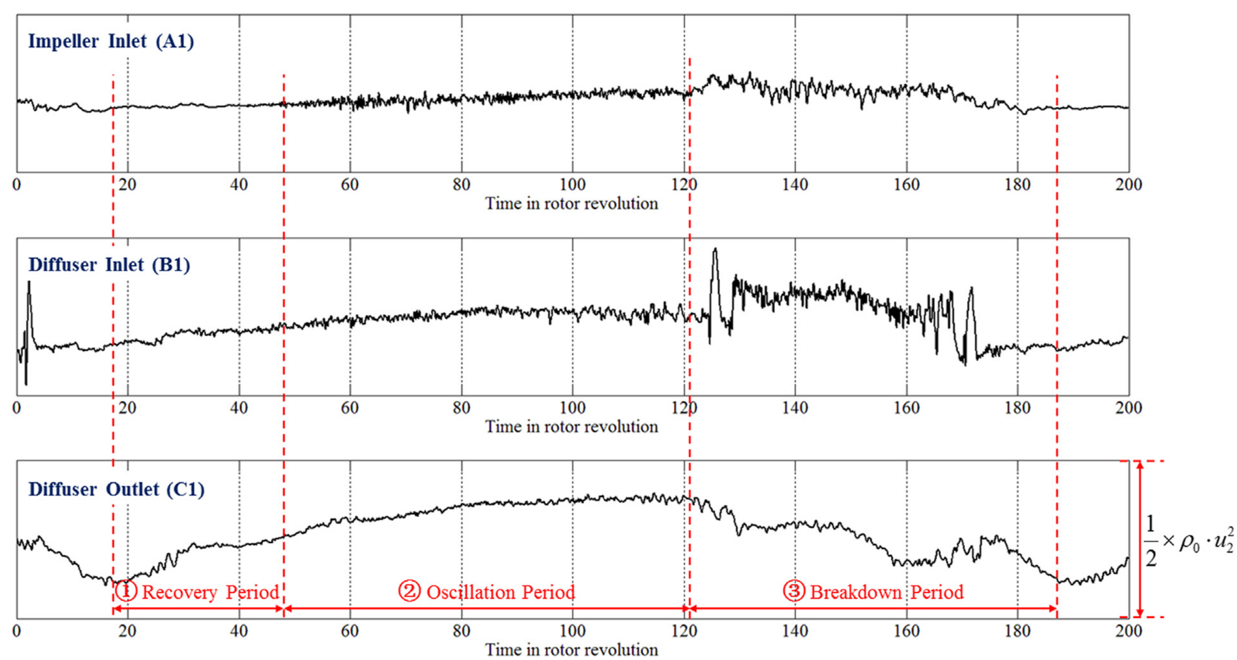

Fig. 9. Pressure oscillation propagation in streamwise direction at deep surge point (40,000 r/min).

gal compressor with a vaned diffuser initiates in the diffuser inlet region when operating at high rotating speeds. Thus, it is appropriate to pay more attention to the improvement of the diffuser inlet region when enhancing the stability of a high-speed centrifugal compressor with a vaned diffuser.

Fig. 11 shows the pressure oscillation at the diffuser inlet at different operation conditions. When the mass flow rate decreases from 0.306 kg/s to 0.279 kg/s, the pressure oscillation occurs with a high frequency and large amplitude, and mild surge occurs successively when the mass flow rate further decreases to 0.242 kg/s. According to Fig. 11, the period of the mild surge cycle is approximately 150 rotor revolutions (the rotor frequency is 1500 Hz when the rotating speed is 90,000 r/min), so the frequency of the mild surge is approximately 10 Hz. Actually, the microphone installed at the entrance of the inlet duct also captures the signal of the mild surge. Fast Fourier Transform (FFT) analysis of the sound signal shows that the frequency is 9.8 Hz, which is close to the calculated frequency of the pressure signal. As the mass flow rate continues to decrease, the compressor operates from mild surge to deep surge; high-frequency pressure oscillation appears in the recovery period, and mild surge cycles compose the oscillation period. Thus, the "self-similarity" appears during a deep surge cycle when the compressor operates at high speed, just as it does at low speed.

The enlarged view of the recovery period and oscillation period of a deep surge cycle at 90,000 r/min is displayed in Fig. 12. According to pressure oscillations in different measurement sections, the mild surge wave propagates to all components of the compressor stage, and the oscillations in each measurement section share the same period and the same frequency. It could be counted that there are 7 mild surge cycles within a 0.7-s duration, so the frequency of the mild surge occurring in the oscillation period of a

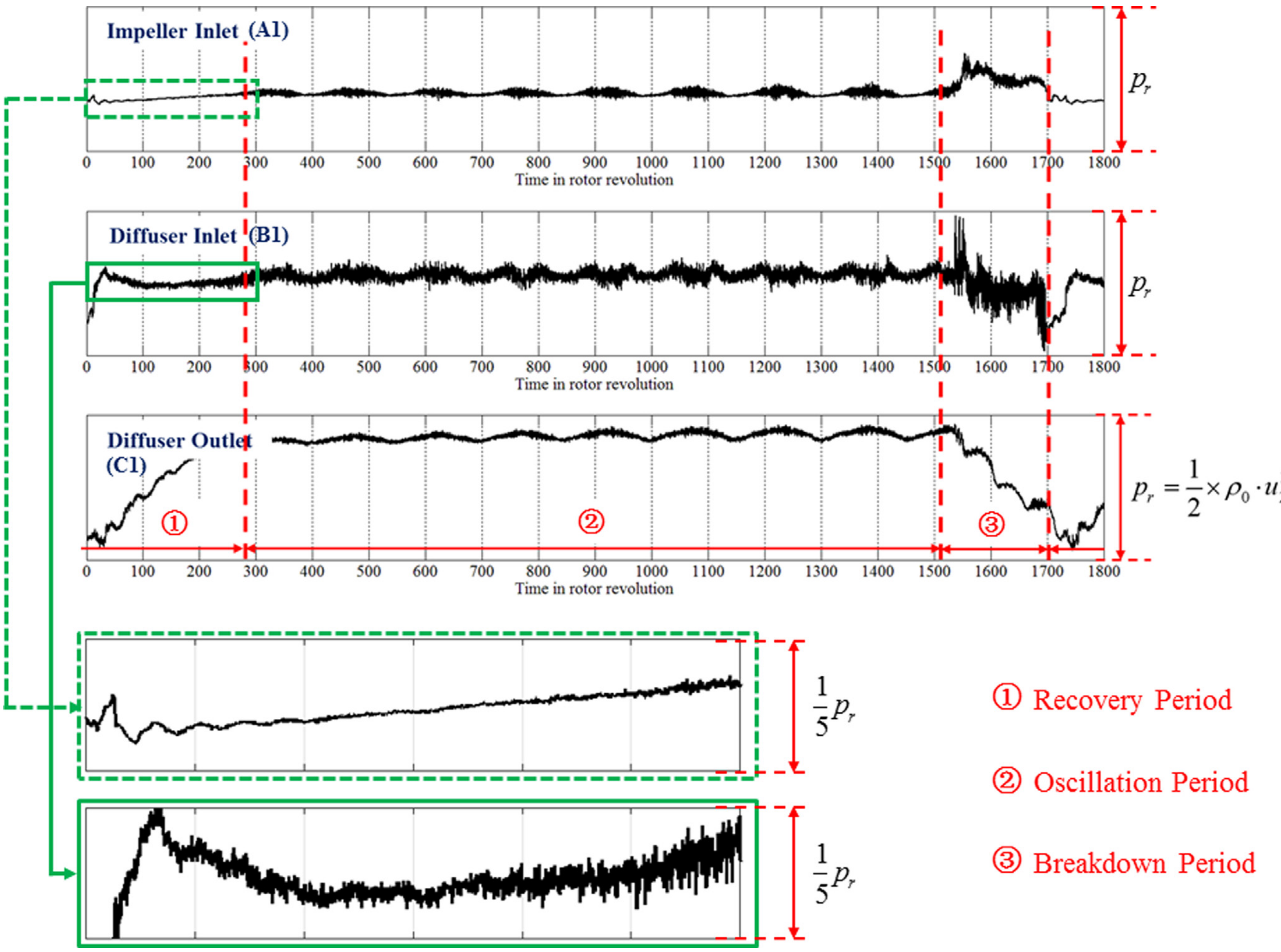

Fig. 10. Pressure oscillation at different measurement sections during a deep surge cycle (90,000 r/min).

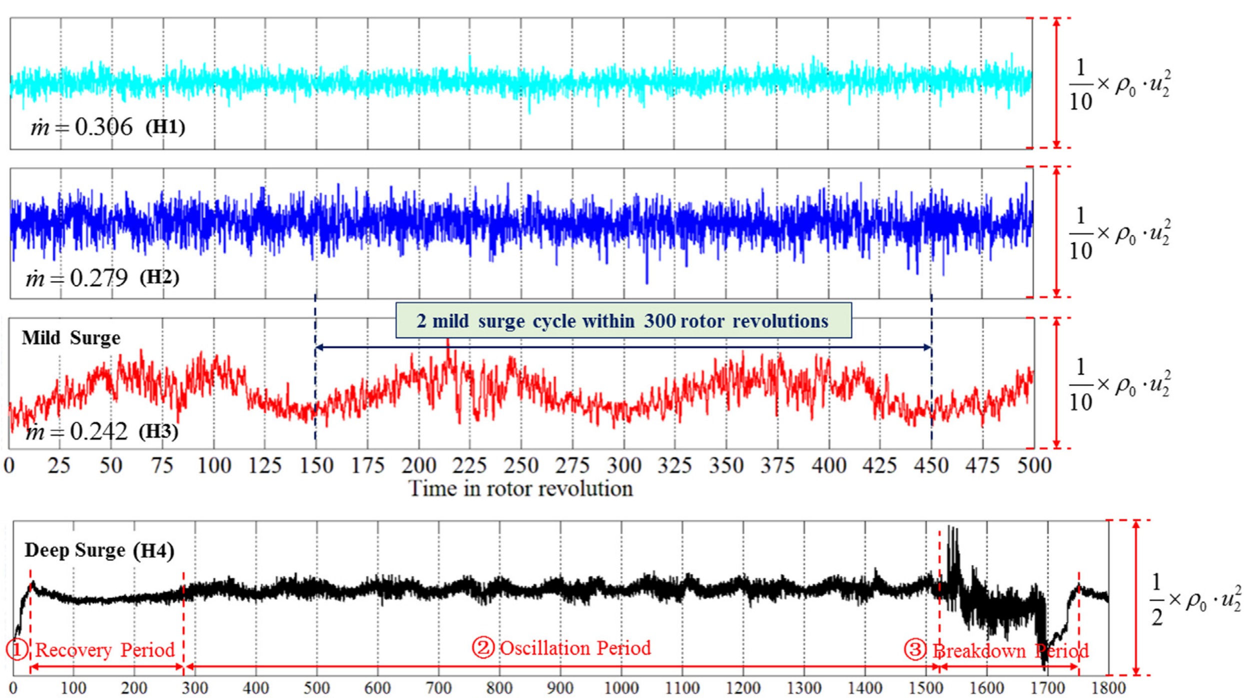

Fig. 11. Pressure oscillation at diffuser inlet of conditions with different mass flow rate (marked by "H1~H4" in Fig.4 @ 90,000 r/min).

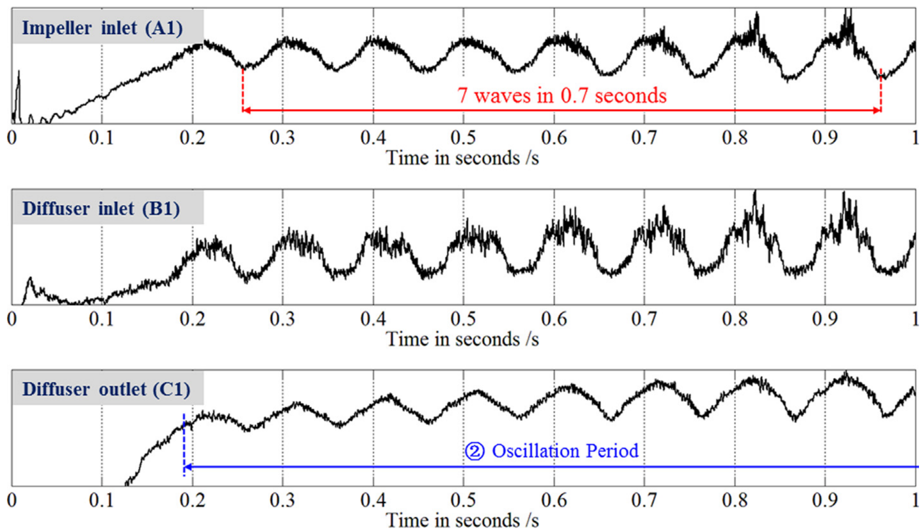

Fig. 12. Enlarged view of recovery period and oscillation period of a deep surge cycle at 90,000 r/min.

deep surge cycle is also 10 Hz. As time goes by, the mild surge becomes increasingly serious, and the fluctuations in each wave crest become increasingly violent; finally, the mild surge disappears and the breakdown period begins.

The discussion of the mild surge cycle shows that although the mild surge occurs independently (e.g., when mass flow rate equals 0.242 kg/s in Fig. 11) or appears during a deep surge cycle, the frequency is always 10 Hz if the rotating speed does not change significantly (in this investigation, mild surge appears only at 90,000 r/min). This is because the frequency of mild surge is determined by the compressor system. According to Zheng and Liu [26], the frequency of mild surge approximately equals to the Helmholtz resonance frequency of the whole compression system. The Helmholtz resonance frequency definition from the one-dimensional Duct–Plenum Model of Greitzer [15] is shown as (2)

$$f_H = \frac{a}{2\pi} \cdot \sqrt{\frac{A}{l \cdot V}} \tag{2}$$

where a is the averaged sound speed of the compression system, and l, A, and V are the equivalent length of the duct, the equivalent cross-sectional area of the duct, and the compressed air volume, respectively. All mentioned geometric properties of the tested compression system are shown in Table 2.

Taking the averaged value of the total temperature at the impeller inlet and outlet, 375 K, as the equivalent temperature of the whole compression system, the speed of sound is

$$a = \sqrt{\gamma RT} \approx \sqrt{1.4 \times 286.7 \times 375} \approx 388.0 \text{ m/s}$$

According to (2) and the geometric properties in Table 2, the Helmholtz frequency should be

$$f_H = \frac{388.0}{2\pi} \cdot \sqrt{\frac{1}{719.3786 \times 0.033380}} \approx 12.6 \text{ Hz}$$

which is close to the frequency of the mild surge (10 Hz). Thus, the calculation of the resonance frequency and the measurement of the mild surge frequency indicate that these two frequencies are closely related, and the characteristics of the mild surge are dominantly determined by the compression system. Previous works draw a quite similar conclusion that system characteristics influence surge frequency. Galindo et al. [31] found that the main frequency of surge in turbocharger compressors is between 5 Hz and 15 Hz; they also noted that the volume downstream of the compressor is directly related to the surge frequency. In addition, the length of the downstream duct could affect the instability model of compressors. Galindo et al. [32] showed that surge operation may change from low-frequency deep surge to a high-frequency mild surge by changing the duct length.

The transient process from stable status to unstable status is different between low and high speeds. A non-dimensional parameter $\boldsymbol{B}$ proposed by Greitzer [15] is accepted to determine the kind of instability mode (stall, mild surge and deep surge) the compressor experiences, and the definition of $\boldsymbol{B}$ is given in (3):

$$B = \frac{u_{rev}}{2a} \cdot \sqrt{\frac{V}{A \cdot l}} \tag{3}$$

where u_{rev} is the reference velocity of the compressor; in this investigation, the tangential velocity at impeller outlet u_2 is taken as the reference velocity. Obviously, $\boldsymbol{B}$ is proportional to $\boldsymbol{u_{rev}/a}$. and there is a positive correlation between $\boldsymbol{a}$ and $\boldsymbol{(u_{rev}c_0)^{1/2}}$, so approximate estimation shows a positive correlation between $\boldsymbol{B}$ and $\boldsymbol{(u_{rev}/c_0)^{1/2}}$, indicating a positive correlation between $\boldsymbol{B}$ and rotating speed $\boldsymbol{N}$. Therefore, $\boldsymbol{B}$ is small at low speed and large at high speed. According to Greitzer's theory, the systematic resonance frequency will be suppressed when $\boldsymbol{B}$ is small, and the system will enter stall; this reflects the characteristic at low speed. However, when $\boldsymbol{B}$ is large, the system exhibits resonance in the form of mild surge. If $\boldsymbol{B}$ is large

Table 2
Geometric specifications of the system.

Components	l (mm)	A (mm^2)	l/A (m^{-1})	V (m^3)
Inlet Duct	1400	7854	178.2531	–
Outlet Duct-V	1050	7854	133.6898	0.008247
Outlet Duct-H	3200	7854	407.4357	0.025133
Total	–	–	719.3786	0.033380

enough, the resonance will induce the system into surge directly without stall, which reflects the characteristic of higher rotating speeds. Therefore, with the fact that $\boldsymbol{B}$ increases with increasing rotating speed, the system will experience stall before deep surge occurs at low speed, while mild surge will occur with the resonant frequency, and then the system enters deep surge at high speed. These conclusions are in good agreement with the results obtained from the experiment.

4.3. Volute influence on the behavior of mild surge

It is inappropriate to neglect the influence of the volute on the stability of a centrifugal compressor because the volute induces strong circumferential non-uniformity throughout the compressor [33], and the role of the volute on the stability inception process is worthy investigating [24]. Zheng et al. [34] found that the volute significantly reduced the stable flow range of a turbocharger centrifugal compressor by steady CFD, and strong non-uniformity in the flow field is found. In this paper, the influence of the volute on pressure oscillations at mild surge condition is investigated by experiment, and obvious circumferential difference is captured at the diffuser inlet.

At mild surge condition (marked by the diamond in Fig. 4), the pressure oscillation at different circumferential positions is shown in Fig. 13. Subplot (a) displays signals obtained by probes A1–A4, and subplot (b) displays signals obtained by probes C1–C6. It is obvious that all pressure oscillations share the same period and the same phase, and no circumferential difference exists at either the impeller inlet or the diffuser outlet. More surprisingly, there is no phase lag or period difference between the impeller inlet and diffuser outlet, indicating that pressure oscillations at the impeller inlet and diffuser outlet are determined not by component properties but by systematic properties. Nevertheless, the shape of the pressure oscillation could change in different circumferential positions, especially at the diffuser outlet. According to Fig. 13(b), pressure oscillates much more drastically downstream of the volute tongue (C3, C4 & C5). Similar conclusions could be drawn when analyzing pressure oscillations during the oscillation period of a deep surge cycle at high speed, as shown in Fig. 14. Thus, the volute influences the shape of pressure oscillation, but does not affect its phase and period at the impeller inlet and diffuser outlet.

The circumferential difference is captured in the pressure oscillation at the diffuser inlet. As shown in Fig. 15, when the

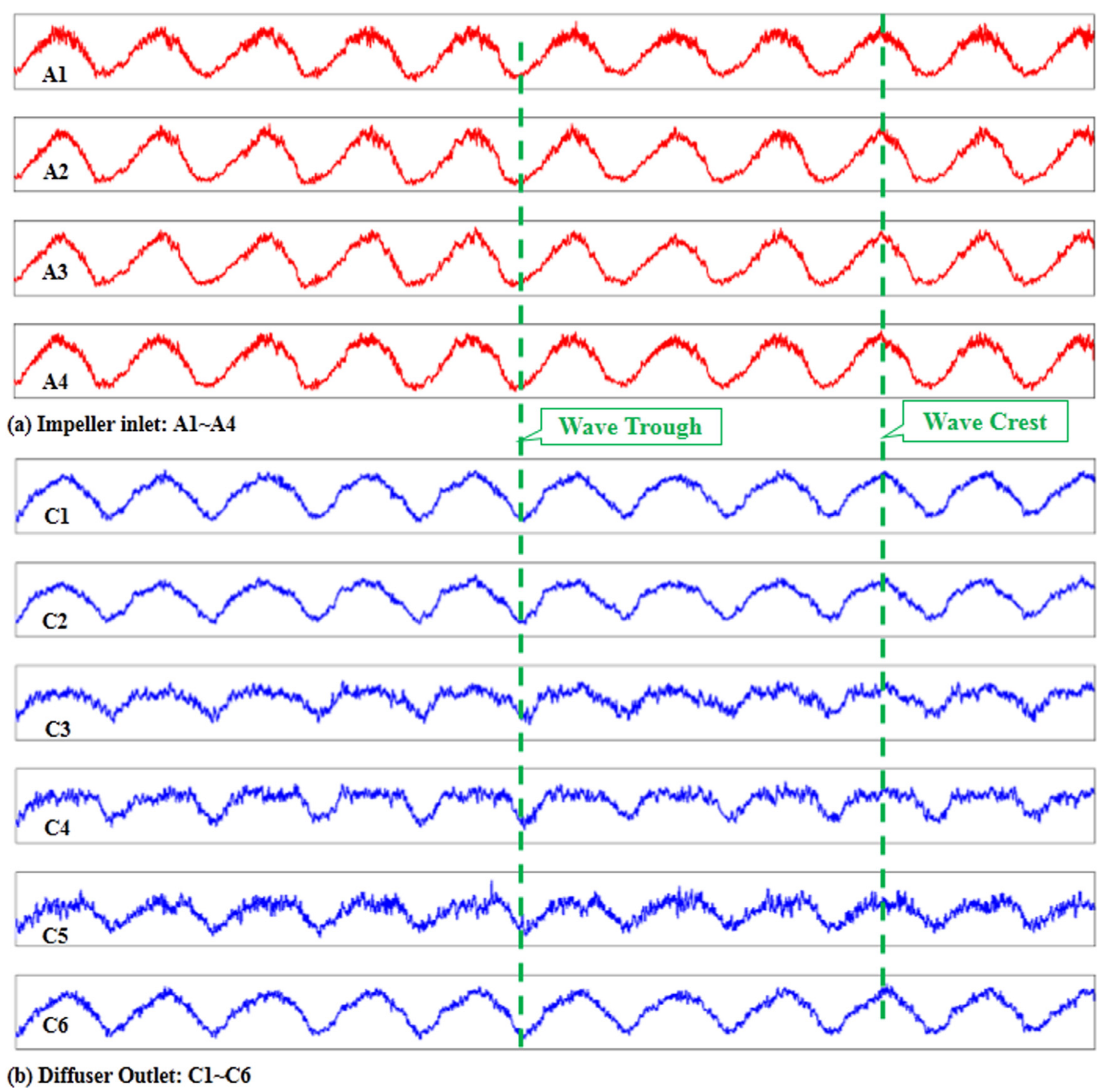

Fig. 13. Pressure oscillation at different circumferential positions at the impeller inlet (section A) and diffuser outlet (section C) when mild surge occurs.

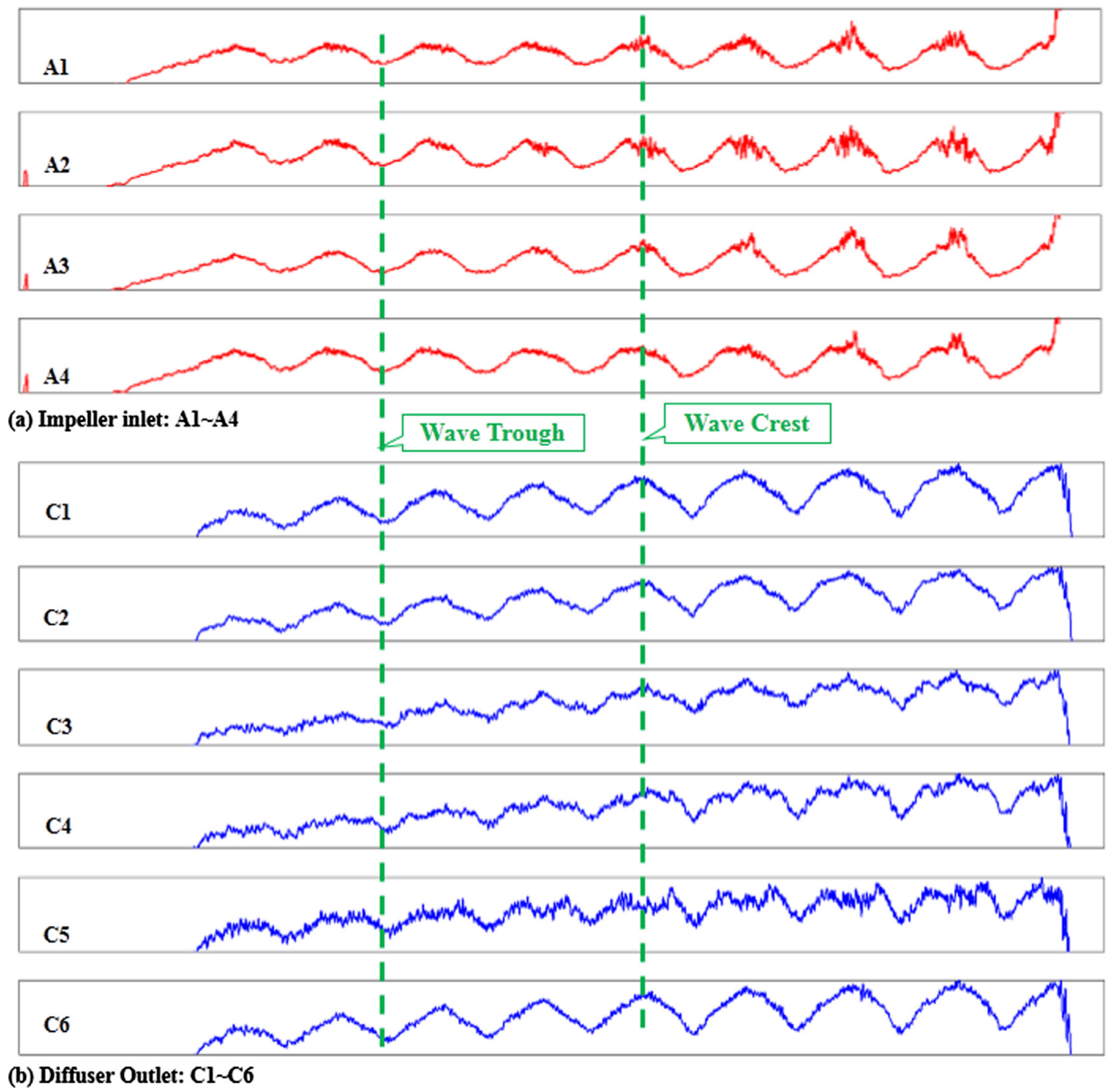

Fig. 14. Pressure oscillation at different circumferential positions at the impeller inlet (section A) and diffuser outlet (section C) during the oscillation period of a deep surge cycle at 90,000 r/min.

compressor operates at mild surge condition (marked by the diamond in Fig. 4), pressure oscillations at *passage 4* (probe B1) and *passage 6* (probe B2) have an antiphase compared with the pressure oscillation at the impeller inlet (A1), although the period of each mild surge cycle does not change. A change of frequency appears at *passage 8* (probe B3) and *passage 11* (probe B5), as displayed in Fig. 15(b) and (c). Within a mild surge cycle, two wave peaks in the pressure oscillation appear; in other words, the frequency of the oscillation doubles in some way. As a result, one peak shows a cophase while the other shows an antiphase (see pressure oscillation in B5). Considering the geometric relationship that the volute tongue is near the outlet of *passage 8*, it is obvious that the antiphase phenomenon of pressure oscillation occurs at passages upstream of the volute tongue, while frequency-doubling of pressure oscillation occurs at passages downstream of the volute tongue.

Considering that volute is the only component that could geometrically induce a circumferential difference and the volute tongue is near the outlet of *passage 8*, it is evident that at the inlet of the diffuser passages upstream of the volute tongue, the pressure oscillation shares an antiphase with that at the impeller inlet, while at the inlet of downstream passages of the volute tongue, pressure oscillation doubles its frequency by forming two wave peaks during a mild surge cycle.

A similar difference in phase and frequency is also captured when mild surge occurs during a deep surge cycle, and the development process is obtained by analyzing the deep surge cycle, as shown in Fig. 16. During the recovery period, a declining tendency of the pressure at *passage 4* (B1) and *passage 6* (B2) is detected, while at other locations where the pressure oscillation shares a cophase, the pressure shows a rising tendency. In this way, the oscillation period begins with a low level (wave trough) at *passage 4* (B1) and *passage 6* (B2) but achieves a high level (wave crest) at other locations, and this is the beginning of the antiphase phenomenon. When the compressor operates from the recovery period to the breakdown period, the frequency doubling phenomenon is increasingly clearer, especially at *passage 9* (B4). Namely, when the compressor becomes increasingly more unstable, the frequency of pressure oscillation is more likely to double, and frequency doubling can reflect the stability of the compressor to some extent. According to Figs. 15 and 16, the frequency-doubling phenomenon is more likely to occur at diffuser passages downstream of the volute tongue, so it can be deduced that the instability initiates from regions downstream of the volute tongue.

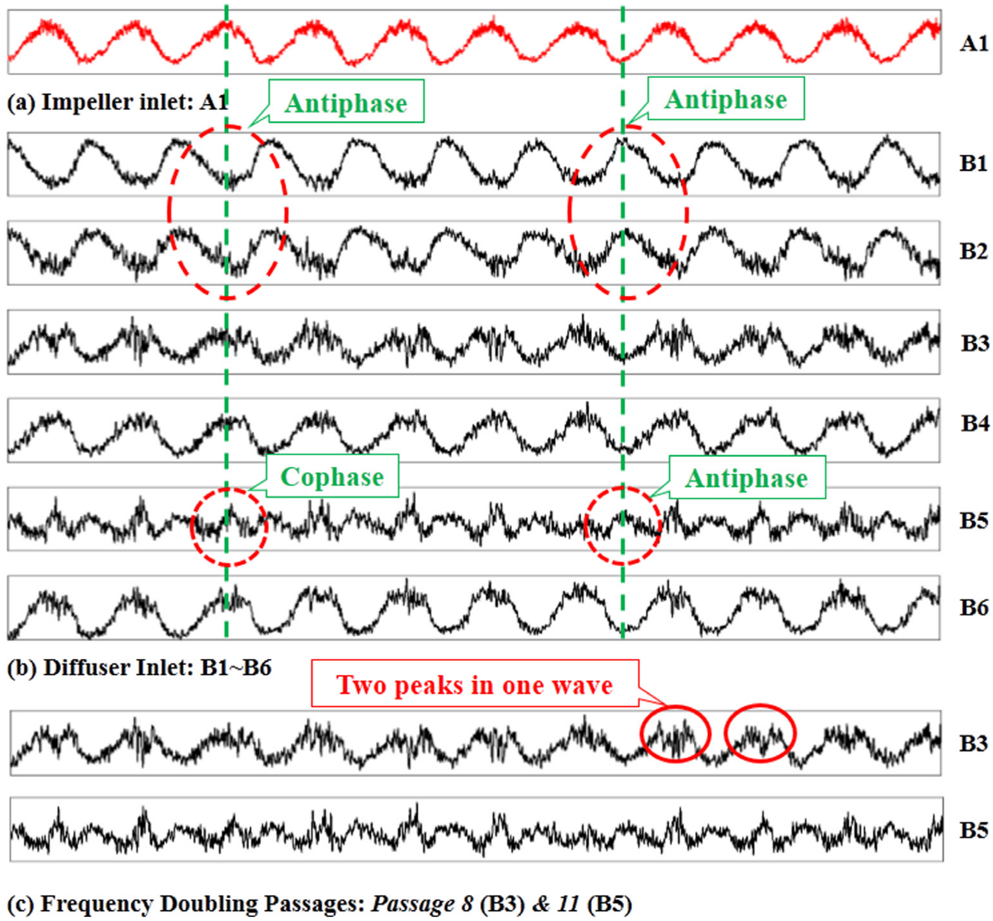

Fig. 15. Pressure oscillation at different circumferential positions at the diffuser inlet (section B) when mild surge occurs.

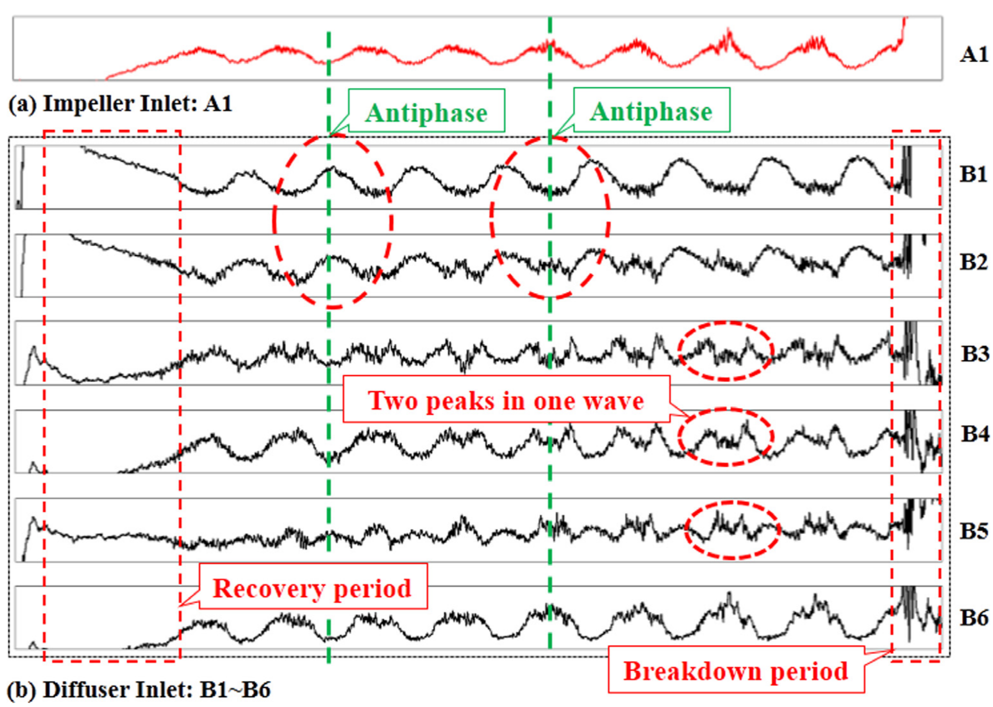

Fig. 16. Pressure oscillation at different circumferential positions at the diffuser inlet (section B) during the oscillation period of a deep surge cycle at 90,000 r/min.

In fact, the behavior difference at different circumferential locations reflects the non-uniformity of the flow field, and obviously a non-axisymmetric flow field is formed. Thus, instability will initiate from certain local locations, and then the stability of the whole system will deteriorate. Regarding a turbocharger compressor, volute is the only component that could geometrically induce a circumferential difference, so it is natural to associate the volute and the circumferential difference in pressure oscillation in mild surge, and it is highly possible that volute is one of the main causes of the antiphase and frequency-doubling phenomenon. Therefore, this investigation also proves the significant influence of volute on compressor stability. However, the conventional design methodology of volute fails to consider circumferential difference, which seems to be inappropriate. Thus, it is recommended to design the volute considering the non-uniformity of the flow field, considering the circumferential difference when attempting to enhance the stability of a turbocharger compressor.

5. Conclusions

An experiment has been carried out to investigate the instability behavior of a turbocharger centrifugal compressor with a vaned diffuser. Data obtained by both low-frequency-response and high-frequency-response probes have been analyzed in detail to provide a clear insight into the surge and stall phenomena. Some conclusions are drawn as follows.

1. When operating at low speed, the centrifugal compressor successively experiences stable status, stall and deep surge with decreasing mass flow rate. In addition, instability initiates at the impeller inlet, and the flow control method is recommended to focus on this region when attempting to enhance the compressor stability at low speed.
2. At high speed, the compressor operates from stable status to mild surge and then to deep surge with decreasing mass flow rate. The instability process initiates at the diffuser inlet region in the form of drastic pressure oscillations. To improve the stability of the compressor at high speed, the diffuser inlet region deserves more considerations and efforts.
3. Deep surge occurs at all tested rotating speeds with quite different characteristics and behaviors. A deep surge cycle includes three periods: the recovery period with minor pressure fluctuation, the oscillation period with remarkable pressure oscillation, and the breakdown period with drastic pressure oscillation. The oscillation periods at different rotating speeds show different behavior: high-frequency pressure oscillation occurs at low speed, but at high speed, this oscillation disappears and mild surge occurs, indicating that a deep surge cycle includes several mild surge cycles. In addition, the duration of oscillation changes at different rotating speeds, resulting in a change in deep surge frequency.
4. Mild surge occurs independently or appears during the oscillation period of a deep surge cycle at high rotating speed with a frequency close to the Helmholtz resonant frequency of the compression system. The behavior of mild surge varies along the circumferential direction at the diffuser inlet, and it is possible that the volute induces this non-axisymmetric behavior. When mild surge occurs, a circumferential difference in the phase and frequency of the pressure oscillation is captured, and the circumferential locations where the difference appears are near the volute tongue. At the inlet of the diffuser passages upstream of the volute tongue, an antiphase phenomenon in pressure oscillation is detected, while at the inlet of diffuser passages downstream of the volute tongue, the frequency-doubling phenomenon of the pressure oscillation is captured.
5. Because the volute affects the behavior of mild surge significantly and causes circumferential difference, its influences should be considered carefully when attempting to enhance the compressor stabilities. Considering the fact that the volute essentially causes a non-axisymmetric flow field structure in the compressor, new volute design methodology considering the non-uniformity flow field and circumferential difference may be established and developed to improve the compressor stability.

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