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Prediction and Active Control of Surge Inception of a Centrifugal Compressor

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ABSTRACT

This paper presents a method of predicting surge inception in a centrifugal compressor, which is done by investigating the change in attractor behaviors expressed in phase portraits. It also presents the results obtained from experiments when we applied this method to actively control the surge in a centrifugal compressor.

INTRODUCTION

As compression systems including centrifugal compressors are damaged by surge, their occurrence has to be prevented in industrial plants. The maximum efficiency of a centrifugal compressor is often obtained at an operating point close to the surge line. Therefore, suppressing the occurrence of surge is beneficial in industrial applications. Moore and Greitzer (1986) suggested that small amplitude perturbation develops into large amplitude perturbation, instabilities known as surge and rotating stall in a compression system. Much work has been done, based to this approach, on suppressing surge while the amplitude of surge disturbances is small, e.g., McDougall (1990) and Garnier (1991). Methods of predicting the occurrence of surge in a compression system have been described by many authors, e.g., Breugelmans and Palomba (1995). Active surge control in centrifugal compressors with a flow control valve was demonstrated experimentally by Pinsley et al (1991) and Blanchini (2002).

This paper discusses a method of predicting surge inception in centrifugal compressors, and it presents the results of experimental investigations into applying it to actively control surge in a centrifugal compressor.

We will first describe the predicting method of surge. The small, periodic fluctuations in pressure that occur in flow through the compressor at surge inception are often buried in the flow turbulence, so that it is occasionally difficult to detect these with commonly used FFT techniques. Further, when surge frequency is low, its detection with FFT usually requires a great deal of time. We present a method of predicting the inception of surge in a centrifugal compressor that involves investigating the change in attractor behaviors expressed in phase portraits. The unsteady pressure time traces measured at the compressor exit plenum are reconstructed on the phase portraits and the deformation in the attractor shapes occurred preceding surge is detected. This enables surge to be predicted in its initial stages. We measured surge in a centrifugal compressor by applying this method and we also provide experimental confirmation of its effectiveness.

We will now discuss an active method of suppressing centrifugal compressor surge with a control valve that is provided independent of the flow control valve. The use of a small actuator enables fast response for feedback signals. The surge control valve in the compressor exit plenum is sufficiently small to operate at high speed and it operates with plenum pressure as the feedback signal. We carried out experiments on actively controlling surge in a centrifugal compressor and the results revealed that surge can successfully be suppressed with our method.

We applied both methods of predicting and controlling surge inception to a compression system, and to a centrifugal compressor.

Our measurements of surge and experimental results indicate that surge can successfully be suppressed at its initial stages of inception.

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NOMENCLATURE

A_C	Compressor inlet area	
a	Speed of sound	
B	Non-dimensional parameter B	$= \dfrac{U}{2a}\sqrt{\dfrac{V_P}{A_C\,\ell_C}}$
D	Embedding dimension	
G	Throttle coefficient when surge control valve open	
K	Throttle coefficient	
ℓ_C	Effective compressor duct length	
N_C	Compressor rotational speed	
P_C	Compressor pressure rise	
ΔP_P	Fluctuations of pressure in plenum	
R	Impeller tip radius	
R_A	Numerical value to predict surge	
T	Period of pressure fluctuations	
t	Time	
t_d	Delay time of delayed feedback control	
t_e	Embedded time	
U	Impeller tip speed	
V	Axial velocity	
V_P	Plenum volume	
η	Pressure amplitude ratio	$= \dfrac{\overline{\Delta\Psi_2{}^*}}{\overline{\Delta\Psi_1{}^*}}$
λ	Period ratio	$= \dfrac{\overline{T_2}}{\overline{T_1}}$
ρ_a	Density of air	
σ	Feedback valued of delayed feedback control	

Φ	Flow coefficient	$= \dfrac{V}{U}$
Φ_{CV}	Flow coefficient of surge control valve	
Φ_T	Flow coefficient of throttle vale	
Ψ	Pressure coefficient	$= \dfrac{P_c}{\rho U^2}$
$\Delta \Psi_P$	Fluctuations of pressure coefficient in plenum	$= \dfrac{\Delta P_P}{\rho U^2}$
ω_H	Helmholtz frequency	

Superscript

*	Normalization
—	Time average

Subscript

1	With control
2	Without control
m	Max. pressure point

EXPERIMENTAL APPARATUS AND PROCEDURE

Figure 1 has a block diagram for the surge prediction and active control system, including the centrifugal compressor . The centrifugal compressor used in the experiment consists of an impeller with 6 blades and a vaned diffuser. The impeller blades are backswept and its tip diameter is 110mm. The design rotational speed of the compressor is 35000 rpm and the rotational speed is controlled by varying the voltage of power supplied to the motor with a slide regulator. A cylindrical plenum was installed at the compressor exit and its internal diameter was 592 mm and its length was 900 mm. The compressor mass flow was adjusted with a throttle valve that was installed at the plenum exit. This was a gate valve type that leaked little air. The compressor inlet flow rate was measured with a Venturi nozzle at the compressor intake duct. The total pressure ratio of the compressor was measured with Kiel-type total pressure probes installed at the compressor intake and exit ducts. To extend the compressor's lifetime, we carried out the experiments at 18,000 and 20,000 rpm, which represented about 51% and 57% of the design rotational speed. Surge occurred in the compression system at these speeds. The compressor rotational speed was measured with an optics-type speed indicator installed at the axial center of the duct 30 mm upstream from the compressor inlet. The dynamic pressure in the plenum was measured as unsteady. A pressure transducer was installed at the axial center of the plenum wall, so that the dynamic pressure measured there was not influenced by noise originating from flow turbulence in the compressor. These dynamic pressures were used as the feedback signals to actively control surge. The sensor head of the pressure transducer was installed so that it was on the same surface as the plenum inner wall.

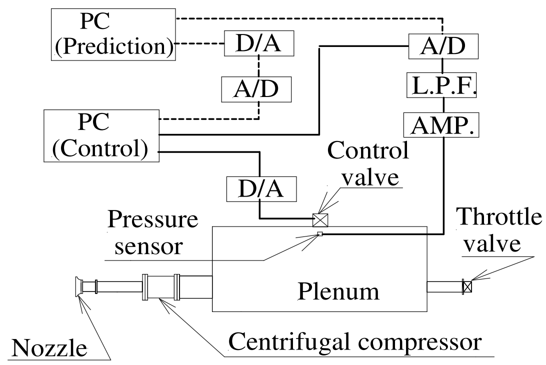


Fig. 1 Prediction and control system for surge.

Table. 1 System parameters.

Effective comp. length ℓ_C (m)	0.600
Comp. inlet area A_C (m^2)	0.0165
Plenum volume V_P (m^3)	0.203
Rotational speed N_C (rpm)	20000
Helmholtz frequency ω_H (rad/s)	88.5
Impeller tip speed U (m/s)	115
B parameter	1.08
$B*$ parameter	21.2

If we had used a single computer to both predict and control surge at the same time, the time from prediction to control would have been too long. Therefore, we used two separate systems/PCs for prediction and control. The dashed lines in Fig. 1delineate the system for predicting surge. The data measured by the pressure transducer are input into the predicting computer through an A/D converter. The measurements data with pressure transducer are fluctuation components. The computer then calculates when the inception of surge can be detected as will be discussed later. When the predicting system has detected surge inception, the signal to start control is sent to the control computer.

The solid lines in Fig. 1 delineate the active system to control surge. A bleed valve is used as the surge control valve. A flow rate adjustment valve is installed at the control valve exit to regulate the control valve flow rate. The operation delay times of this surge control valve after receiving the control signal are 19 ms (open)

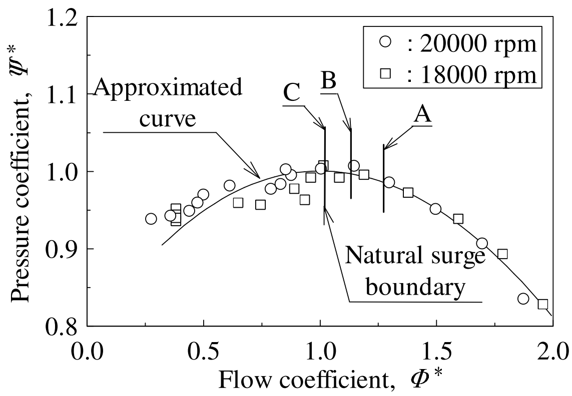


Fig. 2 Compressor characteristics.

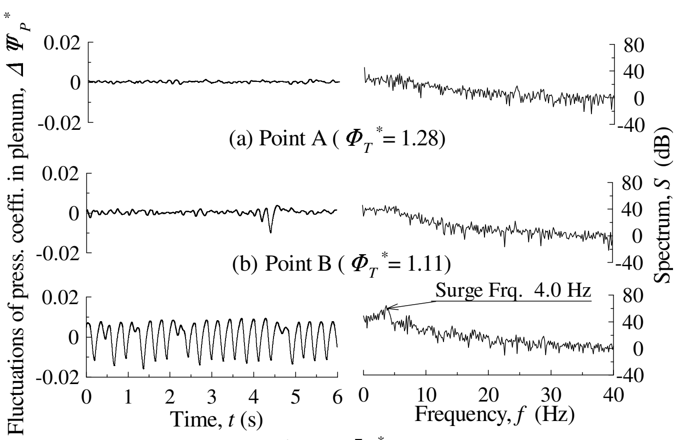


Fig. 3 Time traces for dynamic pressure in plenum.

and 16 ms (closed). The surge control valve was installed at the axial center of the plenum wall, because the effectiveness of control would not be spoiled by the position of the control valve in this compression system. A low pass filter was installed between the amplifier and the A/D converter to prevent false operation caused by noise. The cut off frequency was 40 Hz. The frequency of the surge in this compression system was less than 20 Hz. Therefore, it was possible to measure the main components of the surge wave at a cut off frequency of 40 Hz. The operating point of the compressor was determined by the throttle valve.

RESULTS AND DISCUSSIONS

Surge behaviors in the centrifugal compressor

The measured steady characteristics of the centrifugal compressor are shown in Fig. 2, using non-dimensional flow coefficient Φ and pressure coefficient Ψ . The compressor characteristics expressed by Φ and Ψ almost coincide regardless of the rotational speed. The maximum pressure coefficient was Ψ=0.610 at Φ=0.0311. In Fig. 2, the flow and the pressure coefficients are normalized by the value at the maximum pressure point as defined by the following equations. Normalized B is defined by Eq. (2), and normalized K is defined by Eq. (3). Non-dimensional parameter B was suggested by Greitzer (1976), and it determines the characteristic of the instability phenomena encountered in a compressor.

$$\Phi^* = \frac{\Phi}{\Phi_m} \quad , \quad \Psi^* = \frac{\Psi}{\Psi_m} \qquad \cdots\cdots (1)$$

$$K^* = K \times \frac{\Phi_m^{\ 2}}{\Psi_m} \qquad \cdots\cdots (2)$$

$$B^* = B \times \frac{\Psi_m}{\Phi_m} \qquad \cdots\cdots (3)$$

The solid line in Fig.2 is the approximation curve calculated from the measured values indicated by the squares and circles. Points A and B show the stable operating points of this compressor, which are at the right of the maximum pressure point. Point C is the surge inception boundary and almost coincides with the maximum pressure point. Surge occurred to the left of point C. The specifications for the compression system at 20,000 rpm that we used for the measurements are listed in Table.1.

The measured time traces of the dynamic pressure coefficient in the plenum are shown in Fig. 3, where $\Delta \Psi^*$ is normalized by the pressure coefficient at the maximum pressure point. Pressure fluctuations did not occur at point A, which lies to the right of the pressure peak in terms of characteristics. We could not find a frequency component for the surge in the FFT analysis results. As the throttle flow rate decreased, small amplitude pressure fluctuations occurred at point B (Φ_T^* =1.11) near the maximum pressure point. FFT analysis revealed that a small peak appeared in the neighborhood of the frequency component for the surge. The reasons for this are that these fluctuations indicate periodic oscillations; however, their amplitude is small, and disturbance is large. Therefore, it is difficult to detect periodic fluctuations with the FFT technique. Further, as the throttle flow rate decreases, large periodic fluctuations at a frequency of 4.0Hz appeared at point C(Φ_T^* = 1.02).

The time traces for dynamic pressure in the plenum are in Fig. 4, while the operating point of the compressor was moving from point A to C. As soon as the operating point passed point B, the amplitude of fluctuations of the pressure coefficient began to increase. Thus, Fig. 4 shows that small amplitude periodic fluctuations appear prior to the surge, and develop into surge with large amplitude fluctuations. If small amplitude periodic fluctuations appearing prior to surge could be detected, it would be possible to suppress successive surge inceptions using a limited amount of energy.

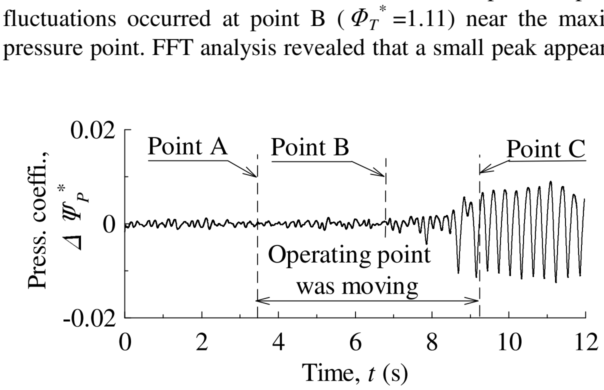


Fig. 4 Dynamic pressure in plenum during surge inception.

Prediction of surge using attractor behaviors

The prediction of the surge inception would be indispensable for the active control at the period of the mild surge, when the amplitude of perturbation is small. Takens (1981) has suggested that full motion in phase space can be reconstructed by measuring the variable of single degree of freedom. To reconstruct the surge, we adopted $\mathbf{x}_{(t)}$ from Eq. (4) which is obtained from the dynamic pressure in the plenum (Hagino, 2002).

$$\mathbf{x}_{(t)} = \left(\Delta \Psi_{P(t)}^{\ *} \ , \Delta \Psi_{P(t-te)}^{\ *} \right) \qquad \cdots\cdots (4)$$

where t_e is the embedded time. Figure 5 shows two-dimensional phase portraits that are reconstructed from the pressure time traces in Fig. 4. Choosing t_e is important in reconstructing the phase portraits from the pressure time traces. If t_e is smaller than the period of the surge, it is difficult to predict periodic fluctuations. If t_e is longer than the period of the surge, the response time to active control for surge oscillations would be long, so that the surge might be not controlled.

In our experiment, we adopted a t_e of 0.05 s, which was approximately 20% of the surge period. For this t_e , the autocorrelation coefficient of dynamic pressure at point A became zero. During period I succeeding point A (see top of Fig. 5), in which the operating point moved from A to B, visible fluctuations in pressure occurred, but, as the amplitudes of fluctuations were small, the patterns on the phase portraits concentrated on the center. These pressure fluctuations were a random wave, so that $\mathbf{x}_{(t)}$ plotted a complicated trajectory on the phase portrait. During period II, the amplitude of pressure fluctuations gradually increased and large periodic oscillations appeared. Figure 5 (c) shows that the attractor

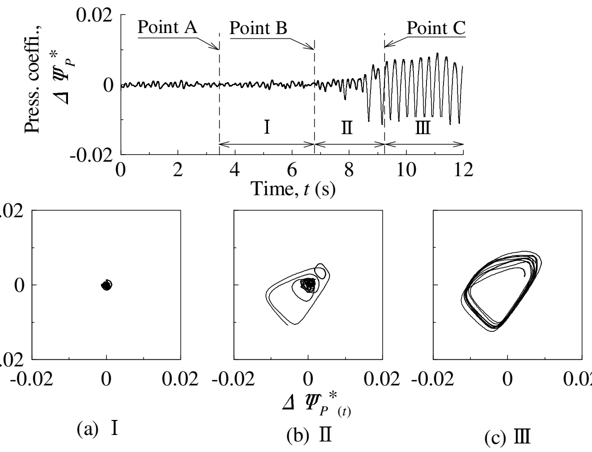


Fig. 5 Phase portraits in surge inception.

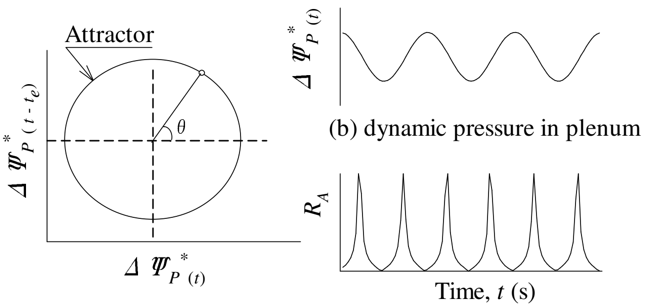
on the phase portrait leaves the center, and it is oval in shape. During period Ⅲ, the amplitude of fluctuations becomes almost constant, and the attractor describes an elliptic orbit.

We presented a method of predicting surge inception in a centrifugal compressor by investigating changes in attractor behaviors (Hagino, 2002). It uses the slope of the attractor to detect changes in its behavior in the phase portrait. However, there was a case where a fatal calculation error occurred. Numerical calculation to predict the surge was done using discrete data that was obtained through the pressure transducer. As a result, when the amplitude was small, the calculated results were affected by noise. In addition, if the magnitude of pressure fluctuations was less than the resolution of the A/D converter, the calculation errors were large. Because the reason is that the slope of the attractor is calculated by numerical differentiation.

As a result of evaluating the prediction method, which has little sensitivity to noise at the initial stages of surge inception, we attempted to use the numerical value of R_A which is defined by the following equation.

$$R_A = \left|\tan(\theta)\right| = \left|\frac{\Delta \Psi_{P\ (t-te)}^{*}}{\Delta \Psi_{P\ (t)}^{*}}\right| \quad \text{..................................} (5)$$

where θ is indicated in Fig.6 (a). If small periodic fluctuations occur in the pressure in the plenum, the change in the attractor's shape will be detected as shown by Fig. 6 (c). The influence of noise for R_A is small. The reason is that the calculation method of R_A is not used the numerical differentiation. Figure 7 is a printout of the display of the computer to predict surge. To confirm validity, we measured surge at operating point B. The sampling frequency



(a) Phase portrait

(c) Numerical value for surge prediction

Fig. 6 Numerical value for surge inception reconstructed from dynamic pressure in plenum.

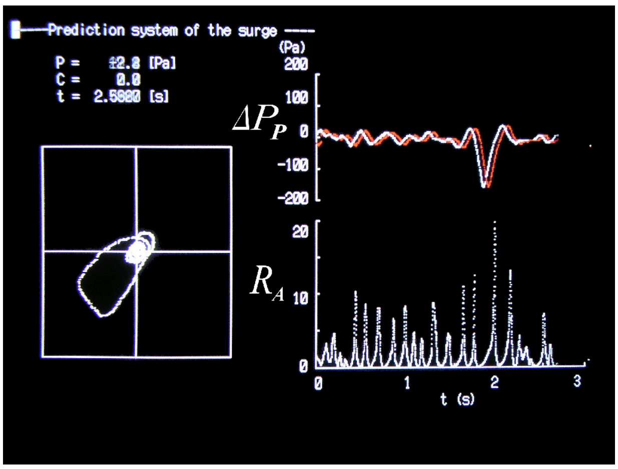


Fig. 7 Printout of display on predicting computer.

was 1 kHz. Smoothing on the calculated value was done through moving averages. Ten data items were used for smoothing. Figure 7 shows the results of the experiment. The top right shows the time traces of dynamic pressure in the plenum, the bottom of the same figure shows the absolute the numerical value for surge prediction.

We can see in Fig.7 that, when small oscillations in surge inception occur, there is a pulse-shaped wave pattern of R_A in the figure. When the amplitude of pressure fluctuations exceeded 20% of the surge amplitude, R_A was larger than 10.

Using this method, the start signal for surge was sent to the control computer. The threshold value of R_A was 10. Figure 8 shows that the starting signal for control is sent to the control computer before the surge amplitude increases.

Surge control

Various methods to actively control surge with the flow control valve have been presented by many researchers (e.g., Pinsely, 1991). These methods have required a lot of energy to operate the large flow control valve. The present paper discusses a method that actively controls surge through the surge control valve, which is separate from the flow control valve, and this is applied to the compression system, including the centrifugal compressor (Hagino, 2003). Surge control is operated by the feedback signal, and we chose plenum pressure for this (Fig. 1). The expected advantages of using a surge control valve that is separate from the flow control valve are a miniaturized control actuator resulting in fast actuator response. We used the bleed valve as the surge control valve.

According to Eq.(6), the bleed valve is operated. When $\Delta \Psi_P^*$ becomes larger than threshold level $\Delta \Psi_{TH}^*$, the surge control valve is opened. In the following experiment, threshold level $\Delta \Psi_{TH}^*$ was zero. The value of $\Delta \Psi_P^*$ is fluctuation components of plenum pressure, so that the surge control valve do not always open. When the controller is switched on, $\Delta \Psi^*$ fluctuates with high frequency

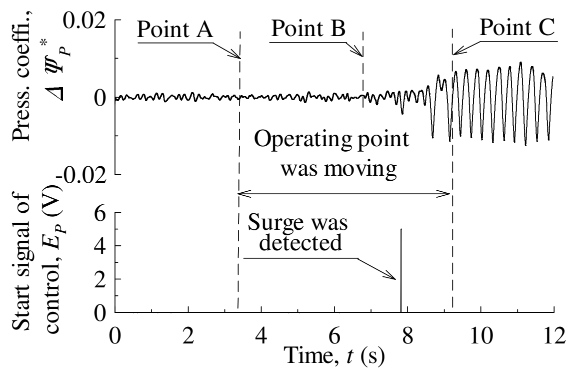
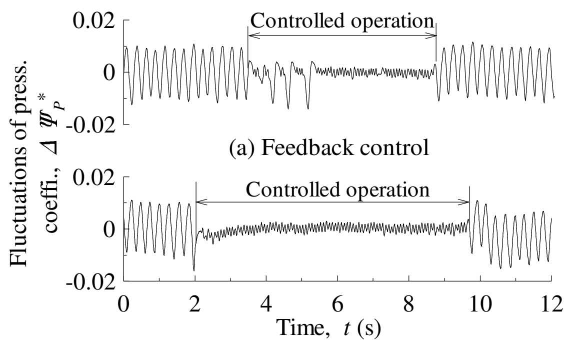


Fig. 8 Prediction of surge inception.



(b) Delayed feedback control

Fig. 9 Effect of the active control of surge at point C and amplitude become small (Hagino,2002).

$$K_{CV}^* = \begin{cases} G & \Delta\Psi_{P}^*{}_{(t)} > \Delta\Psi_{TH}^* \\ \infty & \Delta\Psi_{P}^*{}_{(t)} \leqq \Delta\Psi_{TH}^* \end{cases} \quad \cdots(6)$$

Figure 9(a) shows the time traces for the dynamic pressure coefficients that appeared when the controller was switched on at operating point C where surge had already occurred. As can be seen from Fig. 9(a), there are still large amplitude fluctuations under the control procedure in Eq. (6). Ott et al. (1990) presented the OGY method to actively control chaos vibration, and Pyragas (1992) suggested chaos vibration could be stabilized through delayed feedback. The advantages of delayed feedback control are as follows;

(1) Delayed feedback control is more robust against noise than the OGY method.
(2) It is not necessary to prepare a Poincare map from unsteady measurements

In the following, we apply delayed feedback control to actively control surge in a compression system with a centrifugal compressor.

Figure 10 shows a block diagram for delayed feedback control. The delayed element is installed in the feedback path. The input value for the nonlinear system is determined by the difference between the current pressure coefficient and it's value before t_d. If the value of input $u_{(t)}$ is 0, the outputs are periodic oscillations with period t_d. Valve operation was based on Eq. (7), where σ is larger than threshold level σ_{TH} and, the surge control valve is open. In the following experiment, the threshold level σ_{TH} was zero.

$$K_{CV}^* = \begin{cases} G & \sigma > \sigma_{TH} \\ \infty & \sigma \leqq \sigma_{TH} \end{cases} \quad \cdots\cdots\cdots(7)$$

$$\sigma = \Delta\Psi_P^*{}_{(t)} - \Delta\Psi_P^*{}_{(t-te)}$$

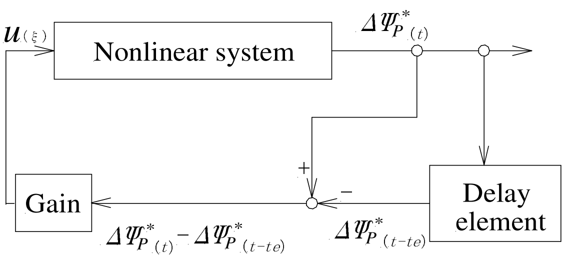


Fig. 10 Block diagram of delayed feedback control.

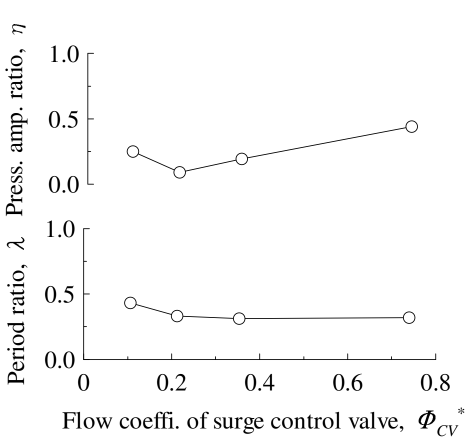


Fig. 11 Effect of flow coefficient of surge control valve.

Applying delayed feedback, we actively controlled surge in the compression system. The specifications of the compression system are in Table.1. The experimental results for surge control with delayed feedback are in Fig. 9 (b), which shows pressure oscillations of surge at point C. The delay time of delayed feedback control is 0.05 s. The flow coefficient for the control valve Φ_{CV}* is 0.354. The flow rate through the surge control valve could not be measured precisely because the valve moved too rapidly. We assumed that the flow rate for the surge control valve was approximated by the difference in the value of flow rate under control conditions and under conditions without control. The throttle coefficient of the control valve, K_{CV}^* was 16.0 at Φ_{CV}* =0.354. As can be seen from Fig. 9 (b), the large amplitude fluctuations in pressure that appeared immediately after control started in Fig. 9 (a) disappeared under delayed feedback control.

Figure 11 shows the influence of the flow coefficient of the control valve on control effectiveness. Pressure amplitude ratio η and period ratio λ were non-dimensionalized by each value under conditions without control. As the flow rate of the surge control valve decreased to Φ_{CV}*=0.212, the pressure amplitude ratio deceased, but, with Φ_{CV}* decreasing further, the pressure amplitude increased. The reason for this increase in pressure amplitude is that the flow rate for the surge control valve was too small to suppress surge.

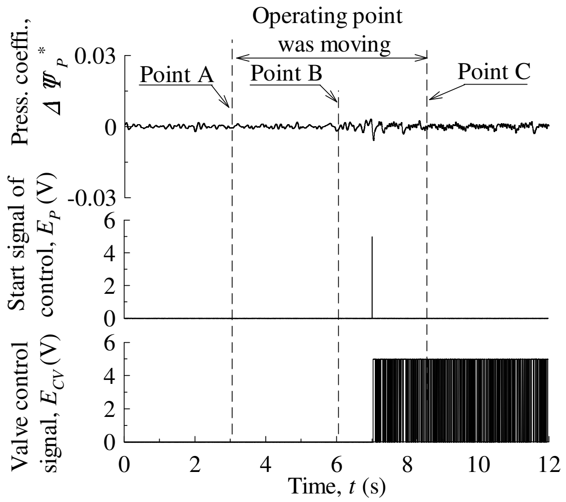


Fig. 12 Prediction and active control of surge inception

(N_C=20000 rpm, B^*=21.2).

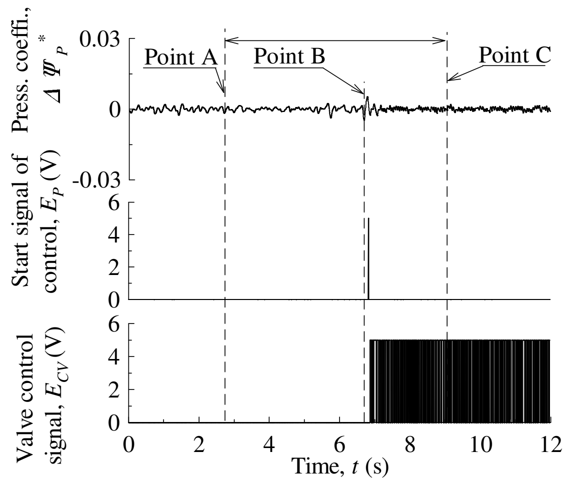


Fig. 13 Prediction and active control of surge inception

(N_C=18000 rpm, B^*=19.1).

Prediction and active control of surge inception

We described the prediction method and the active control method in the preceding section. Both of these have been applied to the compression system including the centrifugal compressor. The experimental results at *Nc* =20,000 rpm are shown in Fig. 12. The top, middle, and bottom show the dynamic pressure, the starting signal for active control of surge, and the control signal for the surge control valve. Without control, small fluctuations in pressure in the plenum develop into large amplitude oscillations, while the operating point moves from B to C. With prediction and control, surge control starts immediately after periodic fluctuations in pressure are detected in the plenum. The delay time from the starting signal of control to the actual control is 28 ms (Fig. 12). This delay time is 10% of the period of surge occurring in the compression system. The fluctuations of pressure are transformed to the small oscillations which have period approximately equal to the delay time of delayed feedback control (Hagino, 2002), so that $\Delta\Psi^*$ and E_{CV} fluctuates with high frequency after the start signal (Fig.12). The flow coefficient for surge control valve $\Phi_{CV}{}^*$ is 0.106. It is possible to suppress surge oscillations with a smaller control valve flow rate of case that control was started after surge oscillations grew up large. It seems that the stall of the compressor impeller can be improved by the surge control at the initial stages of surge inception. As a result, even if the operating point of the compressor moved across the surge line, pressure fluctuations would not develop into large amplitude oscillations for surge. Figure 13 shows the experimental result at *Nc* =18,000 rpm, which shows a similar effect on suppressing surge.

CONCLUSION

We presented a method of predicting and actively controlling surge inception in compression systems that have a centrifugal compressor and we confirmed its effectiveness through experiments. The results revealed that surge could be suppressed at the initial stages of inception and what follows is a summary.

(1) Without control, the attractor reconstructed in the phase portrait from plenum pressure was elliptical at surge inception, and large amplitude oscillations of plenum pressure occurred.
(2) The initial stages of surge inception could be predicted by changes of numerical values R_A, which were calculated from the attractor.
(3) Delayed feedback control successfully suppressed the pressure fluctuations that occurred during surge.
(4) When prediction and active control of surge were near the surge line, surge oscillations could be suppressed through a smaller control valve flow rate of case that control was started after surge oscillations grew up large.
(5) The experimental results indicated that surge could be suppressed at the initial stages of its inception.

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