A Study on Surge and Rotating Stall in Axial Compressors*

(A Summary of the Measurement and Fundamental Analysis Method)

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In multistage axial compressors used for gas turbine engines, it is important to understand unsteady behaviors such as surge and rotating stall. The purpose of this study is to develop a practical analytical method for both surge and rotating stall in axial compressors. In this paper, firstly, some characteristics of the unsteady phenomena in axial compressors derived from the measurements of test compressors are summarized. Secondly, a numerical method for the unsteady phenomena is described, assuming incompressible flow. A feature of the method is solving the fluid equations by Galerkin's method with the circumferential flow distortion expressed in the form of a high-order Fourier series. Finally, we examine the usefulness of the method, using some results from parameter-vary computations and comparisons with the measured results in a 3-stage test compressor.

Key Words: Axial Compressor, Surge, Rotating Stall, Galerkin's Method, Simulation Computation

1. Introduction

In multistage axial compressors used for gas turbines, it is important to understand unsteady behaviors such as surge and rotating stall. These behaviors bring about not only poor performance of a compressor but also excessive vibration to its blades and shaft, which may result in the forced stop of the gas turbine or the failure of the blades in some cases. Main targets for clarification of the unsteady behaviors include the prediction of those onset points and the quantitative evaluation of the dynamic of those onset points, and of dynamic flow velocity and pressure inside the compressor during these occurrences. The former is important in evaluating stall margins such as surge and rotating stall margins in unsteady performance. The latter is essential to the estimation of blade vibration during rotating stall, especially during low speed operation. The authors have shown

some features of surge and rotating stall based on the measurements in multistage axial test compressors, and also the possibility of a relationship between surge and rotating stall⁽¹⁾⁽²⁾. Most researches of unsteady phenomena of compressors have considered surge and rotating stall separately⁽³⁾. Moore and Greitzer proposed a basic numerical method which can deal with both surge and rotating stall in the post stall transient of an axial compression system. This method also showed several patterns of the simultaneous occurrence by numerical simulations as well as the relationship of the patterns to compressor parameters⁽⁴⁾⁽⁵⁾. However, it is not always sufficient for applications describing the behaviors observed in real machines.

The purpose of this study is to establish a numerical method which can deal with both surge and rotating stall in real axial compressors. As a first step, in this report we begin by summarizing the features of surge and rotating stall of multistage axial compressors based on test results of the past. Secondly, expanding on Moore's method, assuming inlet distortion, we propose an analytical method for both surge and rotating stall, which involves solving the equations by

* Received 20 th February, 1989. Paper No. 87-1250 A

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Galerkin's method. In it, the circumferential flow velocity distortion for rotating stall is expressed in the form of a high-order Fourier series. Finally, the usefulness of this method is examined by parameter-vary computations and comparisons with test results in a 3-stage compressor.

Nomenclature

- A : amplitude of angular velocity potential
 a_n, b_n : Fourier coefficient of angular velocity potential
 A_c : compressor duct area
 a : reciprocal time lag parameter of blade passage, $= R/(N\tau U)$
 a_s : speed of sound
 B : B parameter, equation(13)
 F : pressure-rise coefficient in blade passage
 g : disturbance of axial velocity coefficient
 H : semiheight of cubic axisymmetric characteristic, Fig. 6
 h : circumferential velocity coefficient of disturbance
 K_T : throttle coefficient
 l_c : nondimensional total length of compressor and ducts
 L_E, L_I, L_T : length of exit duct, entrance duct, throttle duct
 l_E, l_I, l_T : nondimensional "
 m : compressor-duct flow parameter
 N : number of stages
 P : static pressure
 P_T : total pressure ahead of entrance and following the throttle duct
 q : disturbance velocity
 R : mean wheel radius
 r : time-dependent phase angle
 t : time
 U : wheel speed at mean diameter
 u : circumferential velocity of disturbance
 V : average axial velocity
 V_p : volume of plenum
 v : axial velocity of disturbance
 W : semiwidth of cubic axisymmetric characteristic, Fig. 6
 γ : stagger angle of blade
 η : axial velocity of inlet distortion
 θ : axial distance measured in wheel radii
 ξ : time for wheel to rotate one radian, $= Ut/R$
 ρ : density of fluid
 τ : coefficient of pressure-rise lag
 $\tilde{\theta}$: disturbance velocity potential
 Φ : average axial flow velocity in compressor
 Φ_T : average axial flow velocity in throttle duct
 ϕ : local axial flow velocity

- Ψ : total unsteady pressure-rise coefficient
 Ψ_c : axisymmetric pressure-rise coefficient of compressor
 Ψ_{co} : shut-off value of axisymmetric characteristic
 Ω : vorticity

2. Summary of Measurements

2.1 3-stage compressor

Figure 1 shows the cross section of the 3-stage test compressor. This compressor is composed of 3 stages of rotor blade rows and stator blade rows, inlet guide vanes (IGV), and exit guide vanes (EGV). At the design point, the rotational speed is 13 200 rpm, the mass flow is 21 kg/s, and the pressure ratio is 2.2. In the tests, dynamic pressure was measured by means of pressure transducers installed in the casing wall, and blade vibration was measured by means of strain gauges attached to the root parts of the blades⁽¹⁾.

Figure 2 shows a time history of dynamic pressure during surge obtained by reducing mass flow. The time scale of the lower waveforms is expanded against that of the upper waveforms. As seen in the upper waveforms, dynamic pressure exhibits periodic pulsations peculiar to surge, the period of which is about 0.6 s. The dynamic pressure is also composed of several sharp peaks during a pulsation. The period of the peaks corresponds to 2.3 rotations, according to the lower waveforms. Judging from its period, this is due to rotating stall with one stall cell propagating

Fig. 1 3-stage test compressor and test facility

circumferentially at 0.43 times the rotational speed of the compressor. In this test, it was found that rotating stall occurs simultaneously with surge.

2.2 17-stage compressor

Figure 3 shows the cross section of the 17-stage test compressor. This compressor is composed of 17 stages of rotor blade rows and stator blade rows, IGV and EGV. At the design point, the rotational speed is 10 800 rpm, the mass flow is 36.2 kg/s, and the pressure ratio is 14.7. Dynamic pressure and blade vibration were measured in the same way as in the 3-stage compressor⁽²⁾. The pressure ratio was so high at high speeds that the pressure at the compressor inlet was reduced during surge tests. Figure 4 shows a time history of dynamic pressure during a surge test at 50% design speed. The waves have periodic pulsations due to surge. Also, the hump part of a pulsation is composed of perturbations with shorter periods, which correspond to a pulsation period to rotation period ratio of about 0.45. Judging from the period, the perturbations are caused by rotating stall with one stall cell propagating at about 0.45 times the

Fig. 2 Time history of dynamic pressure during surge and rotating stall in 3-stage compressor

Fig. 3 17-stage test compressor

rotational speed. It is also thought that rotating stall occurs during all the stages with its cell expanded parallel to the compressor shaft, as there is no difference in the phase of pressure waves among all the stages. It is found that rotating stall occurs simultaneously with surge in this test, as was observed in the 3-stage compressor.

Figure 5 shows a time history of dynamic pressure during a surge test in the reduced pressure operation. One of the main features of the waveforms is that a sudden pressure change occurs close to the start

Fig. 4 Time history of dynamic pressure during surge and rotating stall in 17-stage compressor (50 % rotational speed)

Fig. 5 Time history of dynamic pressure during surge and rotating stall in 17-stage compressor (100% rotational speed/reduced pressure operation)

and then after a while, fluctuations with a short period start increasing in amplitude gradually, eventually forcing the stop of the compressor. As the compressor is operated in surge conditions, the sudden pressure change is caused by the onset of surge. However, this surge is different from regular surges in the sense that it is accompanied by fluctuations with a short period instead of continuous pulsation cycles peculiar to surge. Judging from the fact that the period of the fluctuation is about half the ratio of pulsation period to rotation period, the fluctuation is considered to be caused by rotating stall with one stall cell propagating at half the rotational speed. It is also observed that the fluctuations in the 8th-stage begin earlier than those in the 1st- and the 4th-stages. The rotating stall is thought to originate at the blade rows in the downstream, expanding to that in the upstream. It can be concluded that rotating stall occurs due to surge in this case.

The features of surge and rotating stall in multi-stage axial compressors, based on the test results in the 3-stage and 17-stage compressors, are summarized as follows:

(I) It is possible that surge and rotating stall occur simultaneously;

(II) It is very likely that rotating stall due to surge occurs during all stages.

Surge and rotating stall can be said to be related to each other in the sense that both phenomena are concerned with blade stall and that the onset limits of both phenomena are in proximity to each other, although the phenomena themselves are quite different. Rotating stall is mainly the circumferential propagation of pressure and flow velocity, whereas surge is the axial propagation of them. As most conventional numerical methods deal with either surge or rotating stall, they cannot show the interrelation between the two phenomena. In the design of axial compressors, numerical methods which can evaluate the characteristics of both surge and rotating stall simultaneously are required.

3. Analysis Method

3.1 Theoretical model

Our method is based on the theoretical model by Moore and Greitzer⁽⁴⁾ and Moore⁽⁵⁾. The outline of our model is described below. As shown in Fig. 6, a compressor system is divided into several elements: an inlet duct, an IGV, compressor blade rows, an exit duct, a plenum, a throttle, and a throttle duct. It is assumed that the inlet and exit ducts have two-dimensional incompressible flow in the axial and circumferential directions, that the IGV and blade rows have one-dimensional incompressible flow in the blade stag-

ger direction, and that the exit duct has one-dimensional incompressible flow in the axial direction. Also, the plenum is assumed to have compressible flow with an adiabatic process, as the velocity is low, and the pressure is almost constant inside the plenum. In other words, the plenum is considered to act as an air spring displaying surge oscillation. The flow velocity at the IGV entrance is expressed by the axial component ϕ and the circumferential component h , which are both transformed to nondimensional values with the wheel speed at the mean diameter. The axial flow velocity ϕ is also divided into an average component Φ and a circumferentially distorted component g . The components ϕ , g and h are functions of time ξ and angle θ ; Φ is only a function of time.

$$\text{axial direction: } \phi(\xi, \theta) = \Phi(\xi) + g(\xi, \theta) \quad (1)$$

$$\text{circumferential direction: } h(\xi, \theta) = h(\xi, \theta) \quad (2)$$

By definition, the circumferential averages of g and h are zero from the continuity condition, as there is no charge and discharge of flow in the circumferential direction. The component Φ describes flow oscillation by surge, and the components g and h describe circumferential flow distortion by rotating stall. Circumferential inlet distortion ϵ , which appears in front of the inlet duct, is given as input data. Also, the characteristics of the compressor (the relationship between flow velocity Φ and pressure rise Ψ) are given as a 3rd-order function, and the characteristics of the throttle are given as a 2nd-order function.

3.2 Equations of theoretical model

Firstly, the equations of motion are constituted for each element from the inlet duct to the exit duct to

Fig. 6 Theoretical model of compression system

obtain the total pressure rise in the compression system from its pressure gradient. As the circumferential flow distortion is assumed to be a 1 st-order sine wave in Moore's model, there are certain omissions made in deriving the equations for pressure rise. In our method, generalized equations without such omissions will be derived to consider the effect of high-order Fourier effects of circumferential flow distortion. As flow in the inlet duct is vortex flow, the following equation is used as a basic equation.

$$\nabla \left(\frac{P}{\rho} + \frac{1}{2} \bar{q}^2 \right) + \frac{\partial q}{\partial t} - \bar{q} \times \bar{\Omega} = 0 \quad (3)$$

By integrating Eq. (3) in the axial direction while neglecting the circumferential component out of vortex velocity, the following equation are obtained:

$$\int_0^\infty \bar{q} \left(\frac{P}{\rho} + \frac{1}{2} \bar{q}^2 \right) + \int_\infty^0 \frac{\partial v}{\partial t} d\eta + \int_\infty^0 u \frac{1}{R} \frac{\partial v_\infty}{\partial t} d\eta = 0 \quad (4)$$

In Eq. (4), U is the axial velocity of distortion upstream of the inlet duct. Here, we introduce the assumption is introduced here that the velocity components with circumferential distortion vary exponentially with axial distance in the inlet duct, as the following equations indicate:

$$u = u_0 e^{\eta/R}, \quad v = v_0 e^{\eta/R} \quad (5)$$

If integrating equation (4) with the assumption that the inlet duct is long compared with the mean radius, the following equation can be derived.

$$\frac{1}{\rho} (P_\infty - P_0) = \frac{1}{2} (q_0^2 - q_\infty^2) + L_1 \left(\frac{\partial V_0}{\partial t} + \frac{\partial v_\infty}{\partial t} \right) + \left(\frac{\partial \bar{\Phi}}{\partial t} \right)_0 + u_0 \frac{\partial v_\infty}{\partial t} \quad (6)$$

Here, $\bar{\Phi}$ is the velocity potential with circumferential distortion defined as follows:

$$\left(\frac{\partial \bar{\Phi}}{\partial \eta} \right)_0 = v_0, \quad \left(\frac{\partial \bar{\Phi}}{\partial \theta} \right)_0 = u_0 \quad (7)$$

Using total pressure at the entrance of the inlet duct, the pressure rise in the inlet duct can be obtained by the following equation in the nondimensional form:

$$\frac{P_0 - P_T}{\rho U^2} = -\frac{1}{2} (\phi^2 + h^2) - l_1 \left(\frac{\partial \Phi}{\partial \xi} + \frac{\partial \varepsilon}{\partial \xi} \right) - \left(\frac{\partial \bar{\Phi}}{\partial \xi} \right)_0 - \frac{\partial \bar{\Phi}}{\partial \theta} \frac{\partial \varepsilon}{\partial \theta} \quad (8)$$

Using the same procedure as in the inlet duct, the pressure rise in the exit duct can thus be obtained:

$$\frac{P_S - P_E}{\rho U^2} = -l_E \left(\frac{\partial \Phi}{\partial \xi} + \frac{\partial \varepsilon}{\partial \xi} \right) - (m-1) \left(\frac{\partial \bar{\Phi}}{\partial \xi} \right)_0 \quad (9)$$

The parameter m represents the effect of damping for circumferentially distorted velocities, where $m=2$ would refer to a very long exit duct and $m=1$ would refer to the situation without an exit duct. Usually, m is selected between 1 and 2.

Applying the equations obtained by Moore's numerical model for other elements, the total pressure rise from the inlet duct to the exit duct can be expressed

in the following equation:

$$\begin{aligned} \Psi = & \Psi_c(\phi) - l_c \frac{\partial \Phi}{\partial \xi} - m \left(\frac{\partial \bar{\Phi}}{\partial \xi} \right)_0 - \frac{1}{2a} \left(\frac{\partial^2 \bar{\Phi}}{\partial \varepsilon \partial \eta} + \frac{\partial^2 \bar{\Phi}}{\partial \theta \partial \eta} \right) \\ & - l_c \frac{\partial \varepsilon}{\partial \xi} - \left(\frac{1}{2a} + \frac{\partial \bar{\Phi}}{\partial \theta} \right) \frac{\partial \varepsilon}{\partial \theta} + \Phi \varepsilon + \frac{1}{2} \varepsilon^2 \\ & - \frac{1}{4\pi} \int_0^{2\pi} \varepsilon^2 d\theta \end{aligned} \quad (10)$$

Secondly, the pressure drop in the throttle duct and the throttle can be obtained by the following equation from the one-dimensional equation of motion in the axial direction:

$$\Psi = \frac{\Delta P_T}{\rho U^2} + \frac{P_E - P_T}{\rho U^2} = \frac{1}{2} K' r \Phi_T^2 + l_T \frac{\partial \Phi_T}{\partial \xi} \quad (11)$$

Finally, applying the continuity equation for the inlet and the exit of the plenum and substituting density for pressure by using the equation of the adiabatic process, the following equation can be derived:

$$l_c \frac{\partial \Psi}{\partial \xi} = \frac{1}{4B^2} [\Phi(\xi) - \phi_T(\xi)] \quad (12)$$

The parameter B is defined as follows:

$$B = \frac{U}{2a_s} \sqrt{\frac{V_p}{A_c L_c}} \quad (13)$$

3.3 Numerical solution by Galerkin's method

The derived equations (10), (11), and (12) are the functions of time, ξ , and angle, θ . In solving them, we use the upstream differential method for the variation of time, and Galerkin's method for the angle variable.

The velocity potential with circumferential distortion $\bar{\Phi}$ satisfies Laplace's equation, and its average about the circumferential direction should be zero. Then, it can be expressed as in the following equation:

$$\bar{\Phi} = \sum_{n=1}^{\infty} \frac{1}{n} e^{n\eta/R} (a_n \sin n\theta + b_n \cos n\theta) \quad (14)$$

Substituting the solution by Eq. (14) into Eq. (10) to formulate a residual of $\sin(K\theta)$, we get:

$$\begin{aligned} \frac{da_k}{d\xi} = & \frac{1}{2} \left(\frac{m}{k} + \frac{1}{a} \right) \left\{ \frac{1}{2\pi} \Psi_c(\Phi + \sum_{n=1}^{\infty} \right. \\ & \times (a_n \sin n\theta + b_n \cos n\theta) + \varepsilon) \sin k\theta d\theta \\ & + \frac{k}{4a} b_k - \frac{1}{2\pi} \int_0^{2\pi} \sum_{n=1}^{\infty} (a_n \cos n\theta - b_n \sin n\theta) \\ & \times \frac{\partial \varepsilon}{\partial \theta} \sin k\theta d\theta - \frac{1}{2\pi} \frac{1}{2a} \int_0^{2\pi} \frac{\partial \varepsilon}{\partial \theta} \sin k\theta d\theta \left. \right\} \end{aligned} \quad (15)$$

and for a residual of $\cos(K\theta)$, we get:

$$\begin{aligned} \frac{db_k}{d\xi} = & \frac{1}{2} \left(\frac{m}{k} + \frac{1}{a} \right) \left\{ \frac{1}{2\pi} \int_0^{2\pi} \Psi_c(\Phi + \sum_{n=1}^{\infty} \right. \\ & \times (a_n \sin n\theta + b_n \cos n\theta) + \varepsilon) \cos k\theta d\theta \\ & - \frac{k}{4a} a_k - \frac{1}{2\pi} \int_0^{2\pi} \sum_{n=1}^{\infty} (a_n \cos n\theta - b_n \sin n\theta) \\ & \times \frac{\partial \varepsilon}{\partial \theta} \cos k\theta d\theta - \frac{1}{2\pi} \frac{1}{2a} \int_0^{2\pi} \frac{\partial \varepsilon}{\partial \theta} \cos k\theta d\theta \left. \right\} \end{aligned}$$

$$(k=1\sim\infty) \quad (16)$$

On the other hand, substituting the solution of Eq. (14) into Eq. (10) to integrate in the circumferential direction, we get:

$$\Psi = \frac{1}{2\pi} \int_0^{2\pi} \Psi_c(\Phi) d\theta - l_c \frac{\partial \Phi}{\partial \xi} - \frac{1}{2\pi} \int_0^{2\pi} \frac{\partial \tilde{\Phi}}{\partial \theta} \frac{\partial \varepsilon}{\partial \theta} d\theta \quad (17)$$

As Eqs. (11), (12), and (15)~(17) give the deviation of time for unknowns, their time histories can be evaluated with the upstream differential method.

It is important to know the propagating speed of stall cells during rotating stall. This can be obtained from the velocity potential with circumferential distortion as follows:

$$\begin{aligned} \left(\frac{\partial \tilde{\Phi}}{\partial \eta} \right)_0 &= \sum_{n=1}^{\infty} (a_n \sin n\theta + b_n \cos n\theta) \\ &= \sum_{n=1}^{\infty} A_n \sin(n\theta - \gamma_n) \\ \frac{a\gamma_n}{d\xi} &= \frac{-a_n \frac{ab_n}{d\xi} + b_n \frac{da_n}{d\xi}}{a_n^2 + b_n^2} \end{aligned} \quad (18)$$

4. Numerical Result

4.1 Circumferential flow distortion

Firstly, we shall determine how many orders of Fourier series should be taken when calculating the circumferential flow distortion, based on comparisons of numerical results. Figure 7 shows a comparison of three cases for the approximation of the circumferential distortion of axial velocity, which are: 1st-order only, from 1st- to 6th-order, and from 1st- to 12th-order. The flow distortion at the entrance of the inlet duct, which is given by initial input data, is assumed to be a 1st order sine wave. The solutions with higher orders tend to yield trapezoid-shaped distortions as shown in Fig. 7. This is due to the nonlinearity of the equations. The 1st-order approximation cannot

describe this effect. However, the 1st-order approximation is reasonable for a rough estimation in spite of the limitation for multiple stall cells. This is because the amplitude of the distortion by the 1st-order approximation does not differ much from that of the higher-order one. Moreover, the shape and the amplitude of the distortion of the 6th-order and 12th-order approximations are almost the same. This means that the solution will almost converge when taking the 1st to 6th order of Fourier series approximations. The following calculations deal with 1st- to 6th-order Fourier series for circumferential flow distortion, while also considering computation time. The amplitude of flow distortion at the duct inlet, which initializes the unsteady behaviors, is defined as $\varepsilon=0.01$, as it seldom affects the results of circumferential distortion, g and h after rotating stall.

4.2 Unsteady behaviors and compressor parameter

Figure 8 shows three representative cases of unsteady behaviors with changing parameters, B and Ψ_{co} . The upper parts of the figures represent the trajectory of pressure Ψ against flow velocity Φ , which are both averaged circumferentially, and the lower parts represent the time history of local flow

Fig. 7 Circumferential distortion of axial velocity

Fig. 8 Representative examples of unsteady behavior modes against compressor parameters

velocity ϕ at a selected circumferential position.

In condition I), the trajectory repeats a cycle in which ϕ decreases rapidly after a stall point to enter the reversed flow region, and then increases back to the original point. The trajectory figure follows the steady characteristics of contour on both of its sloping sides, i.e. during pressure increase and decrease. The time history is repetition of a trapezoid-shaped wave, which is peculiar to surge. Thus, it is found that surge occurs in condition I). In condition II), with the same ψ_{co} as in condition I) and smaller B , the trajectory looks like a coil after the stall point and traces a small loop around a point on the throttle characteristic curve. The time history is a trapezoid-shaped wave with a short period, which is peculiar to rotating stall. Rotating stall therefore occurs in condition II). Condition III) has a larger value of ψ_{co} and the same B parameters as in condition II). As parameters H and W (see Fig. 6) for the steady characteristics of the compressor are constant in all conditions, the pressure rise per stage in condition III) is smaller than that in condition II) by the proportional relationship between ψ_{co} and the number of stages of the compressor. In condition III), the trajectory shows a smaller loop than that in condition I) in the region where ψ is over ψ_{co} , and does not follow the steady compressor characteristics during decrease in pressure. The time history is a large trapezoid-shaped wave followed by several trapezoid-shaped waves with short periods. In this condition, it is found that surge and rotating stall occur simultaneously. Judging from the three cases mentioned above, it can be deduced, firstly, that whether a surge loop is to be constructed or not depends on the compressor parameters. The loop amplitude of surge also depends on the compressor parameters. Secondly, the final mode of unsteady behaviors is determined by whether the trajectory passes the rotating stall region (negative slope region of compressor characteristics) or not. If a surge loop is not formed, the final destination of the trajectory falls on the rotating stall region.

Figure 9 is a summary of the relationship between compressor parameters and modes of unsteady behaviors. The figure includes the result by a 1st-order approximation of Fourier series for reference. The occurrence boundary between surge and rotating stall depends on parameter B ; that is, surge tends to occur as B parameter increases, and rotating stall tends to occur as B parameter decreases. When the value of ψ_{co} is relatively large, it shows that there is a coexistence zone of surge and rotating stall between each occurrence region. It is also found that occurrence boundaries obtained by 1st-order approximation are quite different from those obtained by 1st- to 6th-

order approximation. The 1st-order approximation tends to give results in which the boundary line between surge and rotating stall moves toward the smaller value of the B parameter and the coexistence zone becomes wider. So, it can be said that 1st-order approximation is not always sufficient for prediction of modes of unsteady behaviors. It is concluded from the above results that the proposed method can deal with surge, rotating stall and coexistence of both phenomena, and that it can quantitatively clarify the relationship between compressor parameters and modes of unsteady behaviors.

4.3 Comparison with measurements in the 3-stage compressor

The usefulness of the method is verified below by comparisons with the result of surge tests using the 3-stage test compressor. Calculations were made by the method under a condition where surge and rotat-

Fig. 9 Relationship between compressor parameters and unsteady behavior modes

Fig. 10 ϕ - ψ trajectory in 3-stage compressor

Fig. 11 Comparison of time histories in 3 stage compressor

ing stall were observed simultaneously in the measurements. Figure 10 shows ϕ - ψ trajectories and Fig. 11 shows time histories. A 1st-order approximation solution and a high-order one are compared in the figures. Under this condition, the 1st-order solution gives a ϕ - ψ trajectory with a big loop and a time history peculiar to surge, while the high-order one gives a ϕ - ψ trajectory with a smaller loop than the 1st one and a time history of simultaneous occurrence of surge and rotating stall. This means the high order solution describes the same phenomena as observed in the measurement, however the 1st order one gives a different result. By this method, the number of cells is one and its propagation to rotation period ratio is about 0.4. This agrees well with the measured result (the number of cells is one, its propagation to rotation period ratio is 0.43).

From the above qualitative comparison with the results of the 3-stage test compressor, it can be said that the high-order approximation from 1st- to 6th-order Fourier series, used for the circumferential flow distortion, is likely to describe phenomena in real machines accurately, although the 1st-order approximation does not necessarily give the same results as those observed in actual tests. The usefulness of the method has been verified for the application to real machines, although further research needs to be done for quantitative improvements, including the consideration of compressibility.

5. Conclusions

- (1) The features of surge and rotating stall in

multistage axial compressors are summarized, based on the measurements in 3-stage and 17-stage compressors, as follows:

- I) It is possible that simultaneous occurrence of surge and rotating stall exists in axial compressors;
- II) It is very probable that rotating stall caused by surge occurs during all compressor stages.

(2) A practical analytical method for handling both surge and rotating stall in axial compressors is proposed. Its usefulness has been verified by parameter-vary computations and by comparisons with measurements in a 3-stage axial compressor, featured as follows:

- I) The method can express simultaneous occurrence of surge and rotating stall, and clarify the quantitative relationship between modes of unsteady behaviours and compressor parameters;
- II) An approximation with 1st- to 6th-order Fourier series as circumferential flow distortion can describe phenomena in real machines better than an approximation with 1st-order only, which does not necessarily correspond with observed results.

Acknowledgement

Finally, we acknowledge Prof. Moore of Cornell University for his informative advice on the development of this research.

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