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VIBRATIONAL RESPONSE ANALYSIS OF MISTUNED BLADED DISK SYSTEM OF GROUPED BLADES

Y. Kaneko and K. Mori
Takasago R&D Center
Mitsubishi Heavy Industries, Ltd.
Takasago, Japan

H. Ohyama and E. Watanabe
Takasago Machinery Works
Mitsubishi Heavy Industries, Ltd.
Takasago, Japan

ABSTRACT

For the purpose of the efficient analysis of a mistuned bladed disk system, a new analysis method which applies the substructure synthesis method and the modal analysis method is proposed. Using the proposed method, the vibrational characteristics of the grouped blades structure are studied. From the results, it is found that the grouped blades structure is very sensitive to the mistuning. It is also found that the mixed grouped blades structure (a bladed disk system consisting of some different types of grouped blades relating to the number of blades contained) has an undesirable effect on the forced response. Moreover, by comparing the vibrational characteristics of the integral shroud blades (ISB) structure with those of the grouped blades structure, it is clarified that the reliability of the ISB structure is superior to other structures also from the viewpoint of the mistuning.

INTRODUCTION

Some bladed disks of steam turbines consist of grouped blades in which several blades are connected into one group by elements such as a shroud or stub. When carrying out a vibration response analysis for such a bladed disk of grouped blades, the effect of the disk is usually neglected under the assumption that the stiffness of the disk is sufficiently larger than that of the blade, in order to avoid the complicated calculation. And even if taking the disk into account, all blades on a disk are assumed to be identical (tuned system). In a real bladed disk, however, the vibration of grouped blades is coupled with each other through the disk and furthermore, the vibrational characteristics of blades on a disk vary slightly due to manufacturing and material tolerances (mistuned system). Therefore, in the forced response of a mistuned system by a harmonic excitation, Eq.(1), which is a conditional equation of resonance for a tuned system (e.g., Wagner and Griffin 1996), is no longer effective because of the split of natural frequencies or the distortion of vibrational modes, and a difference in the amplitudes among individual blades appears, causing certain blades to respond remarkably. Moreover, blades become responsive over a wider excitation frequency range than the tuned system.

$$H\Omega = \omega, \quad H \pm n = kN_G \qquad (1)$$

H : Excitation harmonic number (engine order)
Ω : Rotating speed of rotor
ω : Natural frequency of bladed disk
n : Number of nodal diameters of vibrational mode
N_G: Number of blade groups on a disk
k : Arbitrary integer

Although such mistuning phenomena have been systematically studied since the 1980s (e.g., Griffin and Hoosac, 1984), they are mostly related to the analysis of the free-standing blades structure and the mistuning phenomena of the grouped blades structure have hardly been clarified yet. Especially, any vibrational characteristics of a mixed grouped blades structure, in which the grouping number of each group on a disk is not identical, have not been studied.

In this paper, therefore, an effective method of analyzing the vibrational response of a mistuned grouped blades is proposed, applying the substructure synthesis method and the modal analysis method. Then, the vibration response analysis of a mistuned grouped blades is carried out using the proposed method and the vibrational characteristics is clarified. In addition, in order to research the effective technique for decreasing the mistuning effect, vibrational responses of grouped blades are compared to those of integral shroud blades, which are continuously coupled by the blade untwist due to the centrifugal force.

ANALYSIS METHOD

Eigenvalue Analysis

Because the periodic symmetry of a bladed disk is lost in a mistuned system, the cyclic symmetry method (e.g., Nagamatsu, 1985) cannot be applied to the eigenvalue analysis of a mistuned bladed disk system. As far as a simple analysis model, it may be possible to carry out the direct analysis of the whole bladed disk system owing to the recent remarkable improvement of the computer capability. However, in the vibration response analysis with a detailed model, it is impractical to directly calculate the whole bladed disk due to an excessive computational cost. Therefore, a new analysis method has been

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developed applying the substructure synthesis method in order to efficiently carry out the vibration response analysis of a mistuned bladed disk system even if a detailed analysis model is used. In this analytical approach, it is possible to perform the eigenvalue analysis of the whole system with the reduced freedoms after the blade-part and disk-part having been analyzed separately and integrated into the whole system by use of the calculated results of each parts. By applying the substructure synthesis method, the eigenvalue analysis and vibration response analysis can be carried out very effectively for the mixed grouped blades structure (a bladed disk system consisting of some differeni types of grouped blades relating to the number of blades contained) which is the mistuned system peculiar to the grouped blades structure. The outline of the analysis method is described in the following.

Consider that a bladed disk system consists of one substructure (disk part D) and a number of substructure groups (grouped blades part B_i, $i=1 \sim N_G$) as shown in Fig.1. Furthermore, it is assumed that each substructure has been modeled respectively using finite elements. Displacements of the nodal points in the substructures are divided into two types: one is displacements of the parts connected with other substructure (connecting region) $\{U_n\}$ and another is displacements of remaining parts (inner regions) $\{U_m\}$. According to this assumption, the potential energy of the substructure E is expressed as:

$$2E=\left\{\{U_m\}^T\{U_n\}^T\right\}\begin{bmatrix}[K_{mm}] & [K_{mn}]\\ [K_{nm}] & [K_{nn}]\end{bmatrix}\begin{Bmatrix}\{U_m\}\\ \{U_n\}\end{Bmatrix} \tag{2}$$

Similarly, the kinetic energy of the substructure T is expressed as:

$$2T=\left\{\{\dot{U}_m\}^T\{\dot{U}_n\}^T\right\}\begin{bmatrix}[M_{mm}] & [0]\\ [0] & [M_{nn}]\end{bmatrix}\begin{Bmatrix}\{\dot{U}_m\}\\ \{\dot{U}_n\}\end{Bmatrix} \tag{3}$$

where, [K] and [M] are the stiffness matrix and mass matrix, respectively. Superimposing constrained modes $\{\phi_i\}$, which are calculated by fixing the connecting points, and the static displacements of inner points caused by the forced displacements of the connecting points $[T_{mn}]\{U_n\}$, displacements of the inner points $\{U_m\}$ can be expressed by Eq.(4).

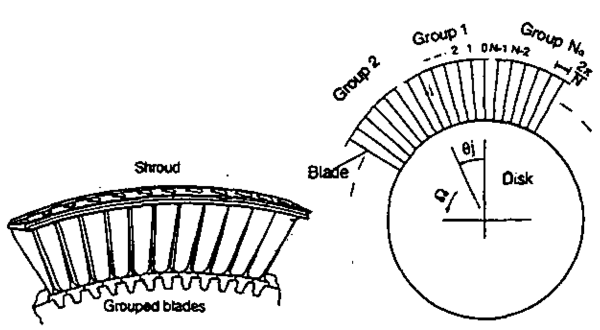


Fig.1 Analysis model of grouped bladed disk.

$$\{U_m\}=\sum_{i=1}^{N}\{\phi_i\}q_i+[T_{mn}]\{U_n\} \tag{4}$$

where, q_i is modal coordinates and $[T_{mn}]$ is a matrix defined by Eq.(5).

$$[T_{mn}]=-[K_{mm}]^{-1}[K_{mn}] \tag{5}$$

From Eq.(2) through Eq.(5), the potential energy and kinetic energy of the substructure are expressed as:

$$2E=\left\{\{q\}^T\{U_n\}^T\right\}\begin{bmatrix}[\tilde{K}_{qq}] & [0]\\ [0] & [\tilde{K}_{nn}]\end{bmatrix}\begin{Bmatrix}\{q\}\\ \{U_n\}\end{Bmatrix} \tag{6}$$

$$2T=\left\{\{\dot{q}\}^T\{\dot{U}_n\}^T\right\}\begin{bmatrix}[\bar{M}_{qq}] & [\bar{M}_{qn}]\\ [\bar{M}_{nq}] & [\bar{M}_{nn}]\end{bmatrix}\begin{Bmatrix}\{\dot{q}\}\\ \{\dot{U}_n\}\end{Bmatrix} \tag{7}$$

where,

$$\left.\begin{aligned}[\tilde{K}_{qq}] &=[\Phi]^T[K_{mm}][\phi]\\ [\tilde{K}_{nn}] &=[K_{nn}]-[T_{mn}]^T[K_{mn}]\\ [\bar{M}_{qq}] &=[\Phi]^T[M_{mm}][\Phi]\\ [\bar{M}_{qn}] &=[\Phi]^T[M_{mm}]^T[T_{mn}]\\ [\bar{M}_{nn}] &=[M_{nn}]+[T_{mn}]^T[M_{mm}][T_{mn}]\end{aligned}\right\} \tag{8}$$

$$[\Phi]=[\{\phi_1\}\{\phi_2\}\cdots\{\phi_N\}] \tag{9}$$

The stiffness matrix and mass matrix of the substructure can be obtained from Eq.(6) to Eq.(9) reducing the degrees of freedom, and those of the whole hladed disk can be obtained by superimposing the matrix of the substructure over the wbole system. After constructing the stiffness matrix and mass matrix of the whole system in the form of the reduced degrees of freedom, q_i and $\{U_n\}$ in Eq.(4) can be obtained by carrying out the eigenvalue analysis through Eq.(6) to Eq.(9). As for the substructure of the disk, it can be considered to be cyclically symmetric. Therefore, in calculating $[T_{mn}]$ and constrained modes of the disk, the cyclic symmetry method can be applied.

Response Analysis

In the case where a bladed disk is rotating in a flow field with a circumferential nonuniformity, the equation of motion of the whole bladed disk system can be expressed by Eq.(10) by neglecting the damping term.

$$[M_T]\{\ddot{U}_T\}+[K_T]\{U_T\}=\{F_T\}. \tag{10}$$

where, $[M_T]$, $[K_T]$ and $\{U_T\}$ are the mass matrix, stiffness matrix and displacement vector of the whole system respectively, which are expressed in the physical coordinate. In general, these matrices of a mistuned system are not cyclic matrices. The excitation force on a blade generated from the circumferential variation in the flow field $\{F_T\}$, which has the same magnitude and a certain phase difference for each blade, is expressed as:

$$\{F_T\}=\left\{\{F\}\cos\omega t\ \{F\}\cos(\omega t+\alpha_H)\cdots\{F\}\cos(\omega t+(N-1)\alpha_H)\right\}^T \tag{11}$$

where, $\{F\}$ is the excitation force vector acting on one blade, ω is the excitation frequency, α_H is the interblade phase angle caused by the rotation. The excitation frequency and the interblade phase angle can be written as:

$$\omega=H\Omega,\ \alpha_H=\frac{2\pi H}{N} \tag{12}$$

where, H is the engine order of the excitation, Ω is the rotating speed, and N is the number of blades on the disk. In order to carry out the frequency response analysis of Eq.(10) by the modal analysis method, $\{U_T\}$ is expanded as shown in Eq.(13), using the vibrational modes of the whole bladed disk $\{\phi^n\}$, which are calculated according to the procedure explained in the previous section.

$$\{U_T\}=\sum_{n=1}^{m}q_n(t)\{\phi^n\} \tag{13}$$

By substituting Eq.(13) into Eq.(10) and incorporating the modal damping, the equation of motion can be obtained as shown in Eq.(14) regarding the n-th vibrational mode.

$$m_n\left(\ddot{q}_n+2\zeta_n\omega_n\dot{q}_n+\omega_n^2q_n\right)=f_c^n\cos\omega t+f_s^n\sin\omega t \tag{14}$$

where, m_n, ζ_n and ω_n are the modal mass, modal damping ratio and natural frequency of the n-th vibrational mode, respectively. Meanwhile, f_c^n and f_s^n are the modal forces, which can be calculated as the scalar product of the excitation force vector $\{F_T\}$ in Eq.(11) and the vibrational mode of the whole bladed disk system $\{\phi^n\}$. They can be expressed by Eq.(15) for the analysis model of the bladed disk shown in Fig.1, for instance.

$$f_c^n=\sum_{j=1}^{J}F_j\phi_j^n\cos H\theta_j,\quad f_s^n=\sum_{j=1}^{J}F_j\phi_j^n\sin H\theta_j \tag{15}$$

where, ϕ_j^n is the displacement of the j-th nodal point in the n-th vibrational mode, F_j is the amplitude of the excitation force on the j-th nodal point and θ_j is angular coordinate of the j-th nodal point in θ direction, respectively. By solving Eq.(14), the vibrational response of the n-th mode can be obtained from Eq.(16) and Eq.(17).

$$q_n(t)=A_n\cos\omega t+B_n\sin\omega t \tag{16}$$

$$\begin{Bmatrix}A_n\\B_n\end{Bmatrix}=\frac{1}{m_n\omega_n^2}\begin{bmatrix}1-\left(\frac{\omega}{\omega_n}\right)^2 & \frac{2\zeta_n\omega}{\omega_n}\\ -\frac{2\zeta_n\omega}{\omega_n} & 1-\left(\frac{\omega}{\omega_n}\right)^2\end{bmatrix}^{-1}\begin{Bmatrix}f_c^n\\f_s^n\end{Bmatrix} \tag{17}$$

Moreover, the vibrational response of the whole bladed disk system can be obtained by substituting Eq.(16) into Eq.(13). The analysis method mentioned above can be applied naturally to a tuned system. However, because the matrix $[M_T]$ and $[K_T]$ in Eq.(10) are so-called cyclic matrices in the case of the tuned system, the cyclic symmetry method can be applied to the eigenvalue analysis. Furthermore, because the displacement vector $\{U_T\}$ and the excitation force vector $\{F_T\}$ also show cyclic symmetry, the vibration response analysis can be carried out efficiently using the vibrational modes of one segment (one blade group) by utilizing the feature of the cyclic symmetry (e.g., Wagner and Griffin, 1996).

RESULTS OF ANALYSIS

Analysis Model

In order to investigate systematically the mistuning characteristics of the grouped blades structure, vibration response analyses were carried out for three typical models described below.

(1) Model A: A tuned bladed disk system consisting of 26 groups of grouped six blades.

(2) Model B: A mistuned bladed disk system consisting of 24 groups of grouped six blades and two groups of grouped seven blades (Group 1 and Group 14 are grouped seven blades).

(3) Model C: A mistuned bladed disk system consisting of 26 groups of grouped six blades. The stiffness of one group (Group 1) is smaller than that of other 25 groups by 10% (equivalent to 5% of the natural frequency).

Model B (mixed grouped blades structure) is a mistuned system peculiar to the grouped blades structure. The purpose of Model C is to investigate the effect of the existence of a group with lower frequencies

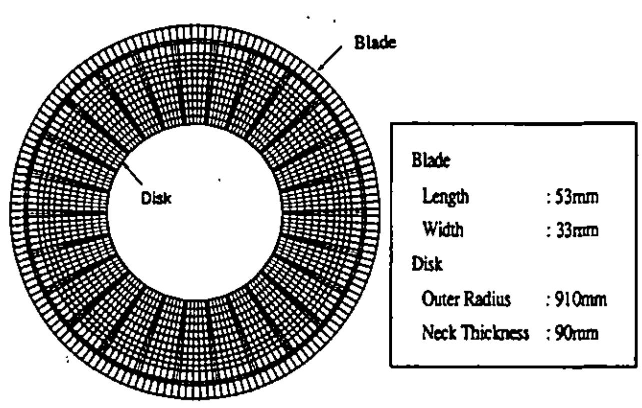


Fig.2 Finite elements model for mistuning analysis.

on the vibrational response of the whole bladed disk system. The mixed grouped blades structure like Mode B has been used in steam turbines and sometimes experienced failures caused by the vibration even though the nearly same bladed disk consisting of the grouped blades with the ideotical grouping number like Model A has experienced few failures. These three models were chosen to understand such phenomena experienced in actual machines. In carrying out the eigenvalue analyses of the substructures, the finite element package NASTRAN was used and the beam elements (blades) and shell elements (disk) were utilized as shown in Fig.2.

Verification of Substructure Synthesis Method

In order to confirm the validity of the substructure synthesis method for the vibration response analysis of a mistuned bladed disk, eigenvalue analyses were carried out by both of the substructure synthesis method and the direct analysis method, and both results were compared with each other. In substructure synthesis method, 10 constrained modes of grouped blades and 42 constrained modes of a disk were used. Fig.3 shows the comparison results of the natural frequencies of Model B. As shown in Fig.3, it is verified that the calculated results by both methods are coincident with each other within accuracy of 3%. Moreover, the comparison regarding the calculated results of the vibrational modes and response analyses of the whole bladed disk system were also performed and ii was clarified that both results show a good agreement with the accuracy applicable in practice (accuracy of 5%). The typical example of calculated modes by both methods is shown in Fig.8. From these results, it was verified that the vibration response analysis can be carried out effectively by the analysis method proposed here.

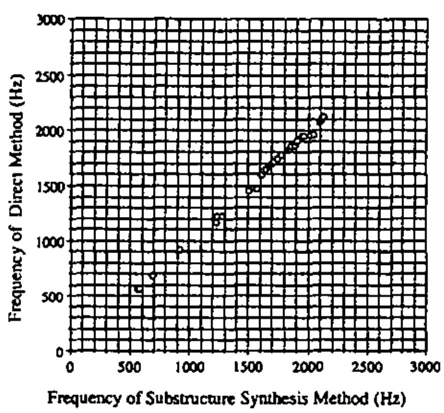


Fig.3 Comparison of calculated frequencies of Model B by substructure synthesis method and direct method.

Mistuning Characteristics of Grouped Blades Structure

First, in order to investigate the effect of the mistuned grouped blades on the natural frequencies and vibrational modes of the whole bladed disk system, eigenvalue analyses of each model were carried out. Fig.4 shows the typical vibrational modes of Model A (tuned system) and Fig.5 shows the natural frequencies of the whole bladed disk system of each model. Because Model A is a tuned system, its eigenvalues are multiple roots except for the vibrational modes of 0 nodal diameter (in-phase modes) and 13 ($N_G/2$) nodal diameters. On the other hand, in the case of Model B and Model C a pair of multiple roots split into two simple roots whose frequencies are slightly different due to the mistuning effects. As shown in Fig.5, the difference

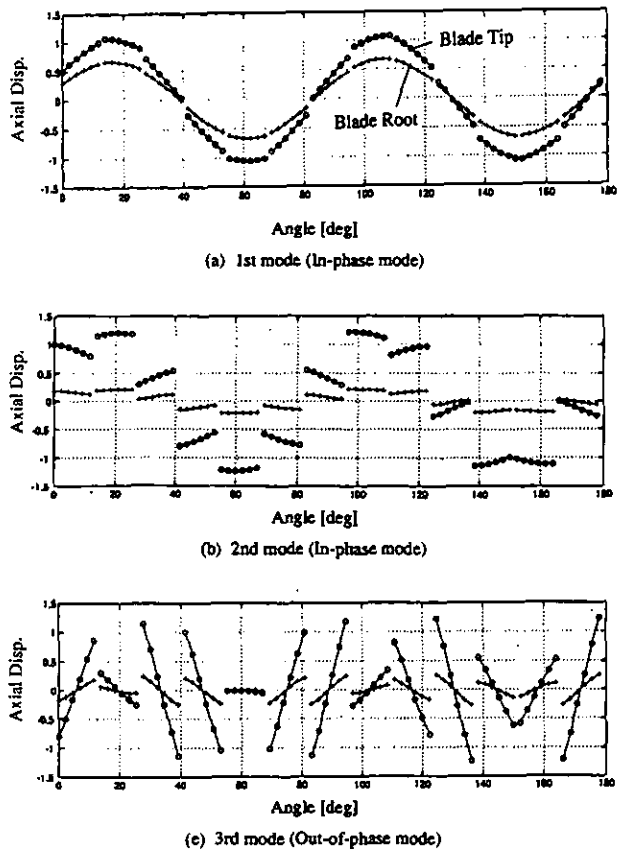


Fig.4 Vibrational modes of tuned bladed disk. (Model A, 4 nodal diameter mode)

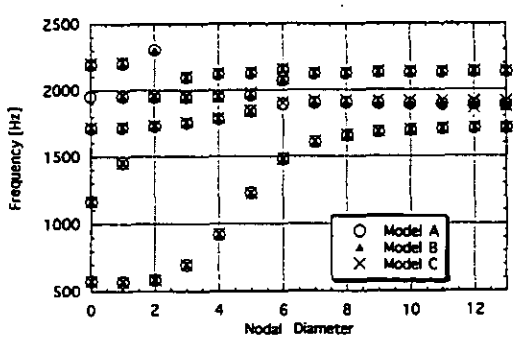


Fig.5 Natural frequencies of bladed disk.

of natural frequencies between the tuned system (Model A) and the mistuned system (Model B and Model C) is very small but the difference of the vibrational modes is becoming remarkable because the feature of the cyclic symmetry is lost in the mistuned system. Figure 6 shows Fourier analysis results of the axial displacements of the shrouds for the vibrational modes of 4 nodal diameters. Since the vibrational modes of Model A is cyclically symmetric, one Fourier component (number of nodal diameters) is dominant. On the other hand, in the case of Model B and Model C two or more Fourier components are dominant. Especially in the mistuned system, the higher the mode order, the more Fourier components are dominant. It can be explained that because the frequency change due to the increase of the number of nodal diameters becomes less in the higher vibrational modes as shown in Fig.5, a change of mode shapes of the whole bladed disk is easier to occur even in a small mistuning such as a slight frequency change of one group. For this reason, a lot of Fourier components are dominant in the vibrational modes of the mistuned system and Eq.(1) (conditional equation of resonance for a tuned system) is no longer effective. Consequently, many peaks appear in the frequency response of the mistuned system. It is important to note that the abscissa in Fig.5 is defined as the number of nodal diameters for convenience but in the strict meaning, it is equivalent

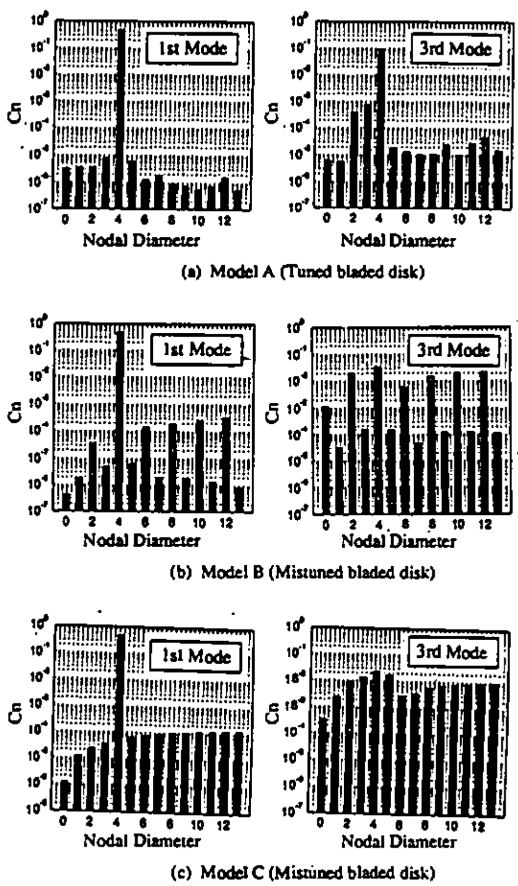


Fig.6 Fourier component of vibrational mode, C_n.

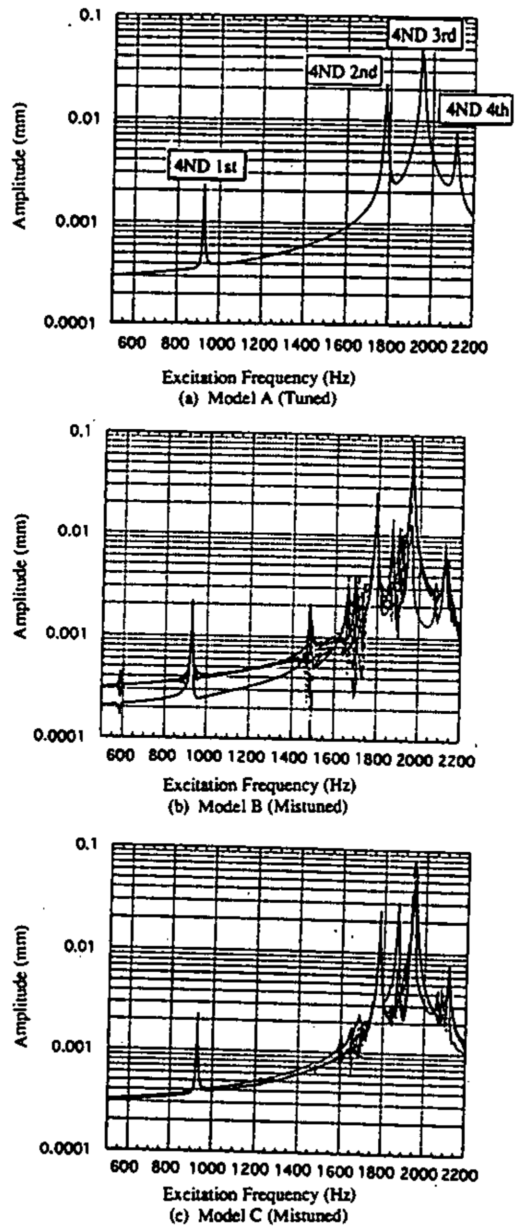


Fig.7 Frequency response analysis results of bladed disk. (H=30, Log. decrement 0.01 for all modes)

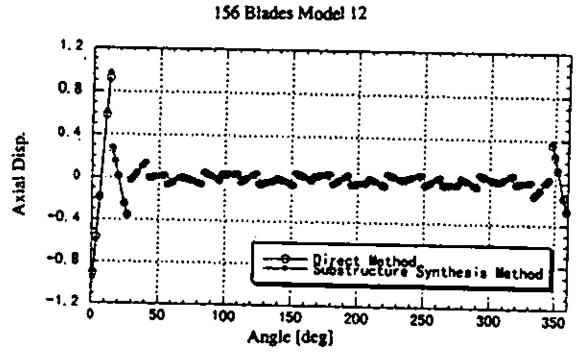
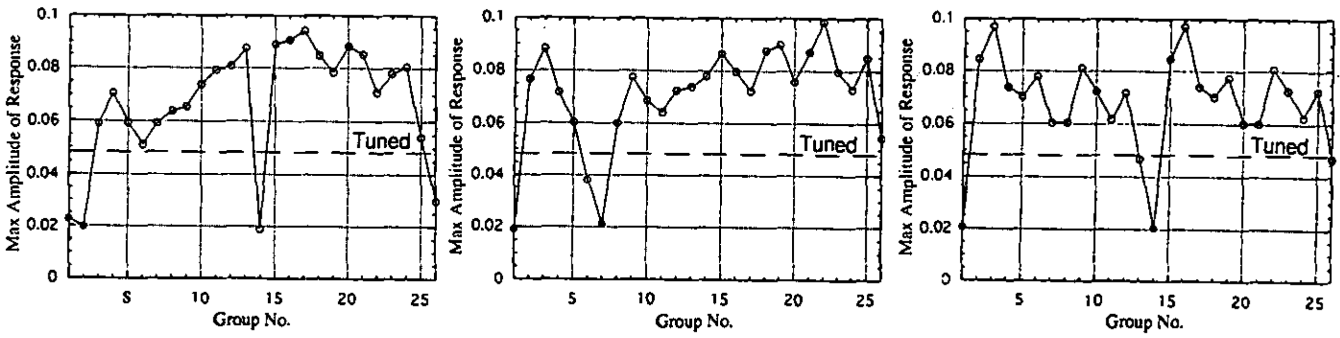


Fig.8 Example of mode localization. (model C, Vibrational mode near 1850Hz)



(a) No.1. 2 (7blades group) (b) No.1. 7 (7blades group) (c) No.1. 14 (7blades group)

Fig.9 Maximum amplitude of mistuned bladed disk in frequency response.

to the harmonic number of the vibrational mode of the cyclic symmetry method (Nagamatsu, 1985). The harmonic number of the cyclic symmetry method and the number of nodal diameters in the physical meaning are equal in the in-phase modes in which the leading blade and the trailing blade of a group vibrate in the same direction. However, as for the out-of-phase modes in which the leading blade and the trailing blade of a group vibrate in opposite direction, "number of groups on a disk - harmonic number" is equal to the number of nodal diameters. Therefore, 3rd mode in Fig.4(c) is 22(26-4) nodal diameter mode in the physical meaning, for example.

Figure 7 shows the results of the frequency response analysis of three models for 30th excitation harmonics (H=30). The ordinate in Fig.7 is the amplitude of the blade tip and the vibrational responses of all trailing blades of 26 groups are superposed in Fig.7. It can be seen from Fig.7 that the vibrational modes of Model A are excited only when the resonant condition (Eq.(1)) is satisfied, that is, only vibrational modes of 4 nodal diameters (30-26=4) are excited and the resonant amplitudes of all groups on a disk are identical. On the other hand, in the responses of Model B and Model C, resonant amplitudes of lower modes (in-phase 1st mode near 950Hz) are nearly equal to that of Model A. However, many peaks appear in the frequency region beyond 1500Hz and the resonant amplitudes are remarkably different among groups. And the maximum amplitude of the whole bladed disk is greater than that of the tuned system (Model A) and in the case of maximum mistuning effect, it can reach almost twice as large as that of the tuned system. Namely, from these results it is clarified that the grouped blades structure is very sensitive to the mistuning similarly to the free-standing blades structure and the resonant amplitude can be about twice as large as that of the tuned system. Moreover, it can be seen in the frequency response of Model C that a peak frequency, at which only one group with lower frequencies (Group 1) has a remarkable amplitude, appear near 1850Hz. It is because that the amplitude of Group 1 becomes extremely greater than that of other groups in one of the vibrational modes near 1850Hz due to the mistuning effect, as shown in Fig.8. As is well known, such a phenomenon is called the localization of vibrational modes and this localization is easier to occur in the free-standing blades structure (Wei and Pierre, 1988). But it is clarified first by this study that the localization can occur also in the grouped blades structure.

Figure 9 shows the maximum amplitudes in the frequency response analysis of the mixed grouped blades structure consisting of 158 blades (24 groups of grouped six blades and two groups of grouped seven blades). In this analysis, in order to investigate the mistuning effect on the mixed grouped blades structure, changing the location of grouped seven blades the frequency response analyses of the whole bladed disk for 30th excitation harmonics were carried out and the maximum amplitude of each group was examined. From Fig.9, it can be seen that the maximum amplitude of grouped seven blades is always smaller than that of grouped six blades and the maximum amplitude of grouped six blades can reach about twice as large as that of the tuned system (26 groups of grouped six blades). In other words, although the location of the group with the maximum amplitude can be changed by altering the location of grouped seven blades, the maximum amplitude of the mixed grouped blades structure is kept always about twice as large as that of the tuned system. Therefore, it is clarified that the mixed grouped blades structure has an undesirable effect on the forced response through an increase in the maximum amplitude experienced by some blades.

Comparison of Mistuning Characteristics Between Grouped Blade Structure and ISB Structure

Recently, in order to improve the reliability of blades, the integral shroud blades (ISB) structure, in which mechanical damping is increased by bringing adjacent shrouds into contact with each other by means of the blade untwist caused by the centrifugal force as shown in Fig.10, has been adopted (Kaneko, 1997). The ISB structure can be expected to be insensitive to the mistuning because all blades on a disk are connected continuously by shrouds and consequently the coupling in the circumferential direction is stronger than that of the free-standing blades structure and the grouped blades structure. In order to examine what advantage the ISB structure has to the mistuning comparing with the grouped blades structure, frequency response analyses of both structures for 12th excitation harmonic were carried out under the condition in Table 1. The ISB structure in Table 1 is a blade used in an actual low pressure steam turbine. Meanwhile, the grouped blades structure is comprised of grouped 6 blades, and the blade and shroud are the same as those of the ISB structure in order to make the comparison easy to understand. The frequency response analysis of the mistuned system was carried out by changing the natural frequency of only one blade on a disk (No.1 blade for the ISB structure, while the trailing blade of Group 1 for the grouped blades structure) by 5% from the tuned system. Because the purpose of this study is to compare the sensitivity to the mistuning of both structures, the most simple case was chosen so as to understand the results easily. (If the mistuning effect is small, the response of bladed disks with any mistuning can be estimated by superimposing these results.) Figure 11 shows the calculated natural frequencies of the ISB structure and the grouped blade structure and its abscissa is the number of nodal diameters of vibrational modes in the physical meaning. In Fig.11, the number of nodal diameters was determined by Fourier analysis results of the axial displacement of the blade tip. And also, the calculated results of the mistuned ISB structure is omitted because the frequency difference between the tuned and mistuned system of the ISB structure is too small to be distinguished. Figure 12 and Fig.13 show the results of the frequency response analyses of the ISB structure and the grouped blades structure, respectively. In Fig.12 and Fig.13, vibrational responses of every 6th blade (trailing blade) of the grouped hlade structure and all blades of the ISB structure are plotted. Figure 14 shows the maximum amplitudes of both structures at the resonances near 400Hz and its ordinate is normalized by the maximum amplitude of the tuned system. It can be seen from Fig.11, the vibrational characteristics of the ISB structure is remarkably simplified comparing to the grouped blades structure because the ISB structure is equivalent to the ring-type blades structure. Although the change of the natural frequencies of the whole system due to the mistuning hardly appear for both structures as shown in Fig.11, remarkable differences can be observed regarding the change of the resonant amplitudes as shown in Fig.12, Fig.13 and Fig.14. That is, in the mistuned system of the ISB structure, a blade with increased amplitude and a blade with decreased amplitude periodically appear and form 24 waves (twice the number of excitation harmonics) as shown in Fig.14. Moreover, the difference of the resonant amplitudes among all blades is small and the maximum amplitude of the whole bladed disk system is increased by only about 20% comparing to that of the tuned system. On the other hand, in the mistuned system of the grouped hlades structure, significant variations in resonant amplitudes occur for different groups and the maximum amplitude of the whole bladed disk becomes larger by about 50% than thai of the tuned system. As is well known, the damping effect of the ISB structure can be expected to be about five to ten times compared with the grouped hlades structure because it utilize the friction damping between shrouds. In the response analysis of the ISB structure and the grouped blades structure, the same logarithmic damping is assumed so as to make the comparison easier. If the analysis of the ISB structure should be carried out using the large damping, the mistuning effect would be much less than the results in Fig.12 and Fig.14. Namely, it can be concluded that the ISB structure is far more reliable compared with the grouped blades structure and the free-standing blades structure taking the following features into account.

(1) Vibrational characteristics can be simplified.
(2) Damping effect can be increased.
(3) Mistuning effect can be reduced.

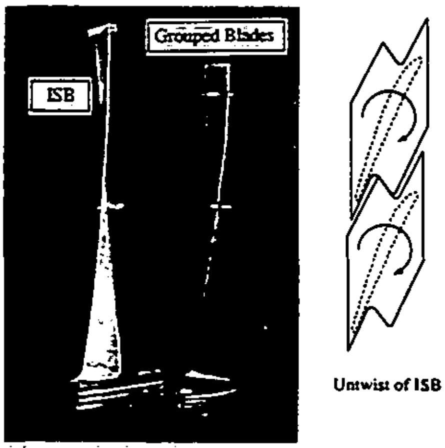


Fig.10 ISB and grouped blade of steam turbine.

Table 1 Condition of frequency response analysis.

Item \ Structure	Grouped Blades	ISB
Blades / Row, N	120	120
Groups / Row, N_G	20	-
Blades / Group	6	-
Log. Decrement for All Modes	0.01	0.01
Excitation Engine Order, H	12	12

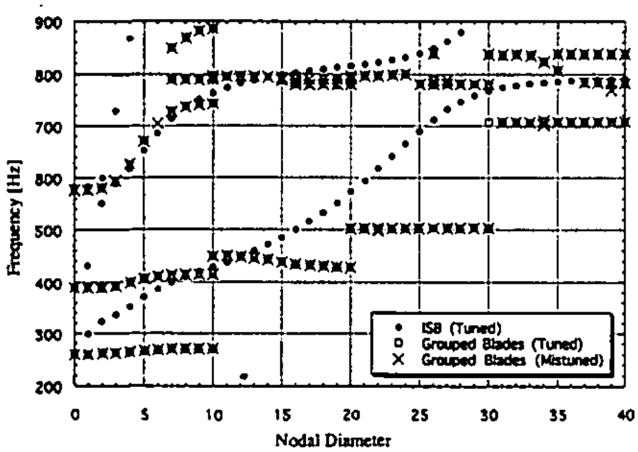
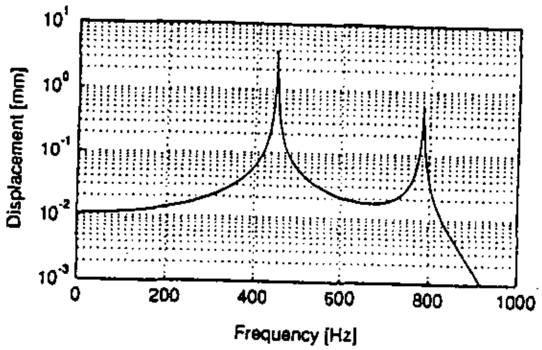
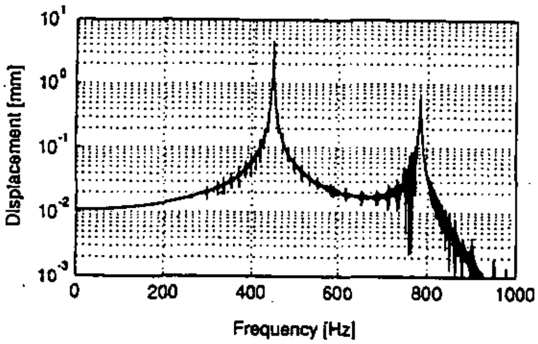


Fig.11 Natural frequencies of ISB and grouped blades.

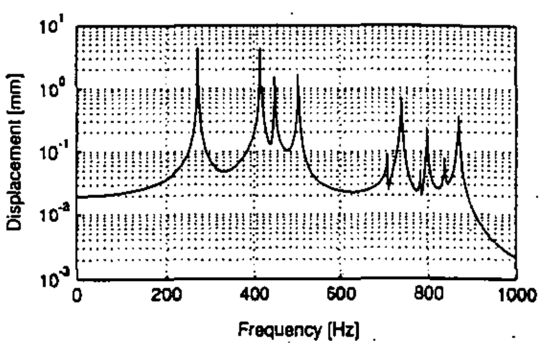


(a) Tuned system

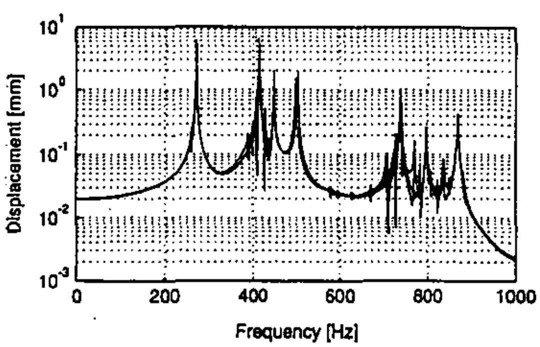


(b) Mistuned system

Fig.12 Frequency response analysis result of ISB.



(a) Tuned system



(b) Mistuned system

Fig.13 Frequency response analysis results of grouped blades.

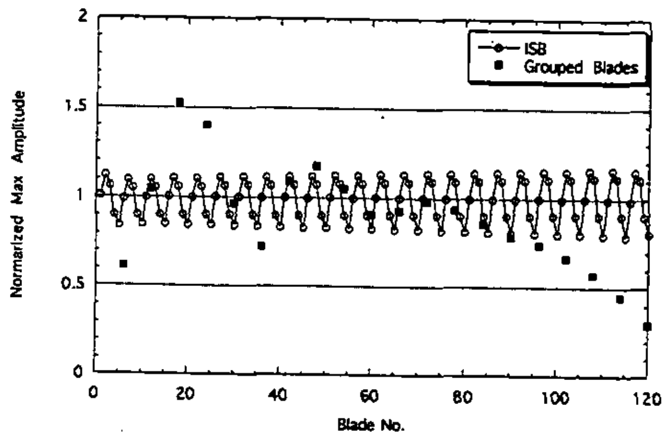


Fig.14 Maximum amplitude of mistuned bladed disk appeared near 400Hz.

CONCLUSION

In this study, first, in order to carry out the vibrational response analysis of a mistuned bladed disk, an effective analysis method which applies the substructure synthesis method and the modal analysis method was proposed. Next, using proposed method, vibrational characteristics of the grouped blades structure was studied. From these results, it was found that the grouped blades structure is very sensitive to the mistuning and the mixed grouped blades structure has an undesirable effect on the forced response. In addition, by comparing the mistuning phenomenon of the ISB structure with that of the grouped blades structure, it was clarified that the reliability of the ISB structure is superior to other structures also from the viewpoint of the mistuning.

REFERENCE

Griffin, J. H., and Hoosac, T. M., 1984, "Model Development and Statistical Investigation of Turbine Blade Mistuning," ASME *Journal of Vibration, Acoustics, Stress, and Reliability in Design*, Vol. 106, pp.204-210.

Kaneko, Y., Mase, M., and Watanabe, E., "Vibration Analysis of Integral Shroud Blade for Steam Turbine," 1997, *Proceedings of International Conference on Power Engineering-97 (Icope-97)*, Tokyo, pp.455-460.

Nagamatsu, A., "Modal Analysis (in Japanese)," 1985, Baifukan, Tokyo, pp.236-241.

Wagner, L. F., and Griffin, J. H., 1996, "Forced Harmonic Response of Grouped Blade Systems: Part-1 Discrete Theory," ASME, *Journal of Engineering for Gas Turbines and Power*, Vol. 118, pp.130-136.

Wei, S. T., and Pierre, C., 1988, "Localization Phenomena in Mistuned Assemblies with Cyclic Symmetry Part I: Free Vibration," ASME, *Journal of Vibration, Acoustics, Stress, and Reliability in Design*, Vol. 110, pp.429-438.