Research Journal of Applied Sciences 9 (11): 800-805, 2014
ISSN: 1815-932X

The Influence of a Blade Vibrations Connectivity on a Degree of Disturbance of Turbine Wheels Rotation Symmetry

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Abstract: The study discusses the vibrations of rotor wheels with non-identical blades. Due to disturbance of a rotational symmetry the eigenmodes of wheels are always characterized by distortion of the harmonic distribution law for the displacements along the circumference (the distortion of eigenmodes). It has been established that the value of the distortion and hence the degree of disturbance of rotational symmetry of a wheel depends not only on how much different blades from one another are but also on connectivity ofvibrations. The process on distortion of eigenmodes by rotor wheels with disturbance of rotational symmetry with decreasing of connectivity of blade vibrations to complete disappearance of the symmetry is studied.

Key words: Turbine wheel, disturbance of rotation symmetry, distortion of eigenvibrations, a connectivity of vibrations, blades

INTRODUCTION

Turbine wheel cannot be made terrain with absolute precision so that its shape completely matched to the nominal CAD Model. Availability of even small manufacturing tolerances always leads to occurrence of difference of wheels blades from each other. Non-identity of blades entails loss by the rotor wheel of rotation symmetry (Castanier and Pierre, 2006; Pierre and Murthy, 1992). It is conventional to consider that disturbance of rotation symmetry in turbine wheels is small. Such, disturbance is accompanied by a segregation of the multiple eigen frequencies and distortion of paired eigenmodes (Ivanov, 1969; Tobias and Arnold, 1957). According to the existing concepts the less one blade differ from another, the less significant are a segregation of the multiple frequencies and a degree of distortion of paired modes (Klauke *et al*., 2009).

Resonant vibrations of a rotor wheel with a small deviation from rotation symmetry represent a superposition of its forced vibrations under paired modes with the segregated multiple frequencies (Feiner and Griffin, 2002). Availability of such superposition leads to that the peak dynamic stresses on various blades differ by their value and are achieved in some resonance band Δf on various angular velocities of rotor spinning. It is the principal distinctive feature of resonant vibrations of such systems. The maximum width of a resonance band Δf in which the identified superposition can be realized depends on value of damping and does not exceed value 2-3 Hz in real wheels (Murthy and Pierre, 1992).

Forced vibrations of rotor wheels occur as a result of influence of exciting harmonicson them. The vibration frequency of any harmonic is defined by Eq. 1:

$$f_{ex} = m_{ex} n_s \tag{1}$$

Where:

f_{ex} = A vibration frequency of the exciting harmonic having the number m_{ex}

n_s = A rotor speed (rpm)

Equation 1 allows a frequency range of rotor spinning in rev/min to define in which the resonance band is realized. For $\Delta f = 3$ Hz and $m_{ex} \geq 4$ it makes the value which is not exceeding 45 rev min^{-1}.

Experimental researches of engines have shown that the range may be essentially more wide for rotor wheels with the significant detuning of blades. In this case, super position of vibrations for paired modes has absolutely another character which is not peculiar to systems with a small deviation from rotation symmetry. If the deviation from rotation symmetry is not small then how much strongly paired modes are distorted? In the case of a strong distortion of eigen modes any of them with modal diameters is no longer orthogonal to exciting harmonics which numbers are distinct from the number m (Rivas-Guerra and Mignolet, 2003). It essentially changes, the character of forced vibrations of rotor wheels in a

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resonance band especially in the case when eigen frequencies of a set of eigen vibrations are close. In connection with noted important issue there is a problem: what determines the distortion degree for the eigen forms of the rotor wheel? Simple arguments suggest that they depend not only from value of a detuning for partial frequencies of blades. When connectivity of vibrations in a wheel are absent its eigen modes for which it is valid, actually turn to single vibrations of one blade on a nonde formable disk. Eigen modes of the rotor wheel differing inthat the small number of blades participates in vibrations in them and their base number has inappreciable displacements are named in the studyas localized. In the case of the full localization the eigen mode of the wheel turns to vibrations of asingle blade. If to transfer blades of the wheel in which the weak connectivity of vibrations occurs, to another disk where such connectivity will be strong the localized modes will no longer be the same. They will turn to modes with nodal diameters. Unconditionally, distortion of a harmonicity of circumferential distribution of displacements in these modes will be saved but its value will be considerably reduced. Thus, the connectivity of vibrations should influence on distortion of paired modes and hence, on a degree of a deviation of the rotor wheel from rotation symmetry. The present study is also devoted to this problem.

MATERIALS AND METHODS

The procedure: Investigations were executed for a model and full-scale rotor wheel by means of computer system ANSYS. Computations were carried on the super computer of Samara State Space University "Sergey Korolyov". The model wheel has been made of a round plate by width of h = 7 mm. It has 16 blades. Its sizes are shown in Fig. 1.

In the computational and experimental studies the wheel was rigidly clamped at its centre by the are alimited by a round in diameter of 42 mm. Figure 2 shows the fragment of a spectrum of eigen frequencies of the model wheel for modes without nodal rounds. The spectrum is defined by computation and experimentally.

Table 1 shows partial frequencies of blades at vibrations under the first bending mode. They are defined at rigid fixation of the disk bythe entire area by means of thick plates and large vices.

Non identityof blades was simulated by arranging of point wise massesover their rim. It was possible to detect experimentally a separation of the multiple frequencies

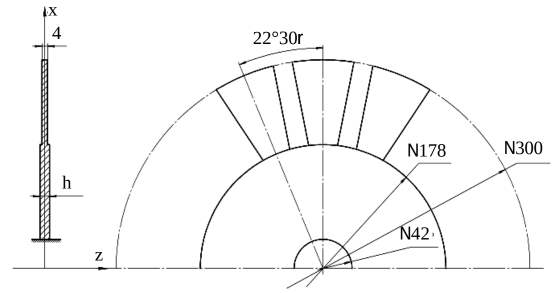

Fig. 1: The model rotor wheel design

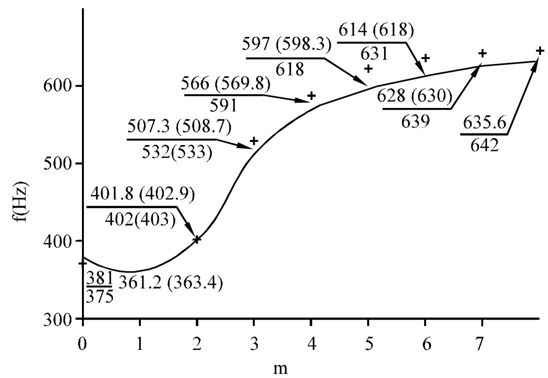

Fig. 2: The spectrum of eigenfrequencies of an initial design; —: Computation; +: experiment; $f_m^{(1)}(f_m^{(2)})/f_{me}^{(1)}(f_{me}^{(2)})$: the eigen frequencies of paired modes defined by computation ($f_m^{(1)}(f_m^{(2)})$) and experimentally ($f_{me}^{(1)}(f_{me}^{(2)})$)

Table 1: Partial eigenfrequencies of blades of the model rotor wheel

The blade number	f (Hz)	The blade number	f (Hz)
1	675	9	684
2	670	10	675
3	684	11	670
4	663	12	679
5	676	13	686
6	688	14	687
7	687	15	684
8	684	16	673

only for modes with two and three nodal diameters. Figure 3 shows computed and experimental distribution of displacements of blades over the rim for paired modes with two nodal diameters. Experimental displacements are defined by interferograms.

The degree of disturbance of rotation symmetry in the rotor wheel was evaluated by means of coefficient λ_{nm} which was evaluated by Eq. 2:

$$\lambda_{nm} = \frac{\overline{W_N}-\overline{W_R}}{\overline{W_N}-\overline{W_O}} \qquad (2)$$

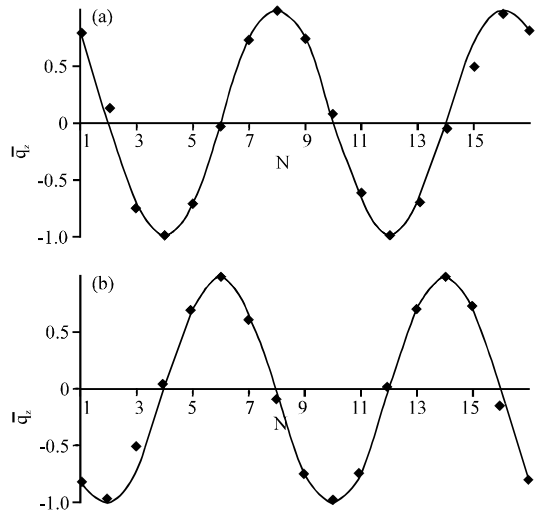

Fig. 3: Distribution of displacements of peripheral points of blades for vibrations of the initial model under paired modes with m = 2; —: Computation, ◆: experiment

Where:

- λ_{nm} = Coefficient of rotation symmetry disturbance with for the mode n with nodal rounds and m in nodal diameters
- $\overline{W_N}$ = A kinetic energy of the blade row of the rotor wheel having the nominal geometry and strict rotation symmetry
- $\overline{W_R}$ = A kinetic energy of the blade row of the rotor wheel with real geometry
- $\overline{W_O}$ = A kinetic energy of the blade row at the full lack of vibrations connectivity (a kinetic energy of the single blade installed on an absolutely inelastic disk)

Kinetic energies $\overline{W_N}$, $\overline{W_R}$, $\overline{W_O}$ were defined for the normalized modes of vibrations. The norming was carried out by the maximum spatial amplitude of the linear displacements. This amplitude is equal to unity inthe normalized mode. The coefficient λ_{nm} depends on a connectivity of vibrations of blades and in this connection differs for different eigen modes of the wheel. In the case of strict rotation symmetry it is equal to zero. In case of lack of a connectivity of vibrations its value makes 1. The initial model wheel has a small deviation from rotation symmetry. It is well visible in Fig. 4 on which dependence of coefficient λ_{nm} on number of nodal diameters for eigen modes which frequencies are presented in Fig. 2 is shown.

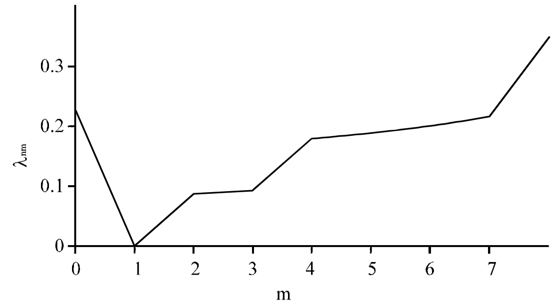

Fig. 4: Dependence of rotation symmetry disturbance coefficient on number of strain wavesin an initial design of the model

It is visible from the represented data that for an initial design of the model wheel the computed results will well agree with experimental data.

When holding studies the change of a connectivity of blade vibrations in the model wheel was carried out by means of increasing of the disk width.

RESULTS AND DISCUSSION

The smooth change of a disk width and accordingly a connectivity of blade vibrations in a model wheel has allowed to trace process of distortion of eigen modes from the beginning of disturbance of an circumferential harmonicity of displacement distribution up to occurrence of full localization. Fragments of model wheel eigen modes distortion processare shown in Fig. 5-7. Digit 1 in them means the initial mode of vibrations, digits 2, 3 and 4 are various stages of its distortion. Captions in these figure show values of coefficient.

Table 2 shows numbers of the blades having the peak amplitude of vibrations for each considered modes. It follows from the analysis of the table that in the case of decreasing a connectivity of vibrations a blade in which a particular mode of a model wheel is finally localized is not selected immediately.

Distortion of a spectrum for the nominal model wheel eigen frequencies in the case of increasing of its disk thickness is shown in Fig. 8. Figure 9-11 show the same eigen modes as in Fig. 5-7 but in the form expanded into a Fourier series. It is clear that the localized modes in the capacity of components contain practically all the harmonics resolved by a degree of symmetry which amplitudes are comparable by value.

As an example, Fig. 12 and 13 show eigen vibrations with four nodal diameters for two rotor wheels of the

Table 2: Numbers of the blades having the peak vibration amplitude

		m								
$h \times 10^3$ (m)	f (Hz)	0	1	2	3	4	5	6	7	8
7	$f_m^{(1)}$	11	3	8	2	4	5	12	3	7
	$f_m^{(2)}$	-	15	14	11	11	11	6	14	-
14	$f_m^{(1)}$	11	4	8	2	10	16	3	9	7
	$f_m^{(2)}$	-	10	14	11	11	10	15	7	-
28	$f_m^{(1)}$	4	2	16	5	12	15	9	8	6
	$f_m^{(2)}$	-	11	10	1	8	3	13	14	-
38	$f_m^{(1)}$	4	2	16	1	12	15	9	7	6
	$f_m^{(2)}$	-	11	10	5	8	3	13	14	-

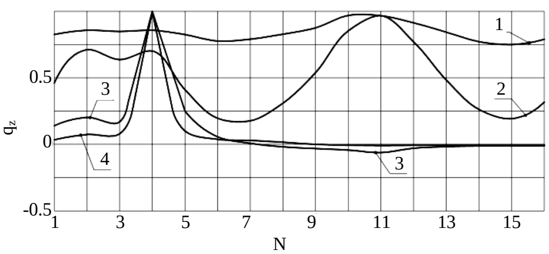

Fig. 5: Change of vibration modes with m = 0 at increasing of thickness of the model rotor wheel disk; 1: $\lambda_{nm} = 0.29$; 2: $\lambda_{nm} = 0.75$; 3: $\lambda_{nm} = 0.99$ and 4: $\lambda_{nm} = 0.998$

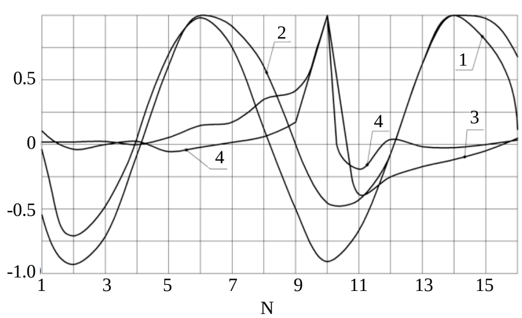

Fig. 6: Change of vibration modes with m = 2 at increasing of thickness of the model rotor wheel disk; 1: $\lambda_{nm} = 0.08$; 2: $\lambda_{nm} = 0.27$; 3: $\lambda_{nm} = 0.86$ and 4: $\lambda_{nm} = 0.96$

engine NK-12. One of them is in appreciably distorted and another is localized. Digits in figures match to values of partial frequencies of blades above which numbers they are plotted. It is necessary to pay attention that not only blades with equal or close partial frequencies have significant displacements in the localized mode. For example, the blade under the number 13 has the same partial frequency as epy blade with the maximum vibration amplitude, however, its displacements in vibrations are close to null.

The studies carried out allow us to make the conclusion on that the degree of distortion of full scale rotor wheels eigen vibrations depends not only on that

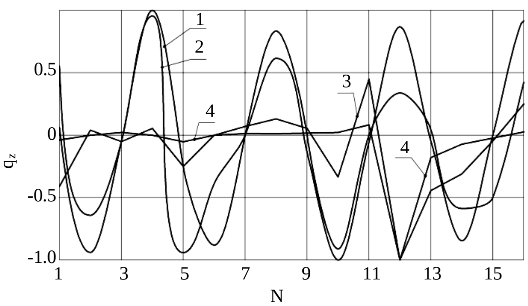

Fig. 7: Change of vibration modes with m = 4 at increasing of thickness of the model rotor wheel disk; 1: $\lambda_{nm} = 0.16$; 2: $\lambda_{nm} = 0.41$; 3: $\lambda_{nm} = 0.87$ and 4: $\lambda_{nm} = 0.99$

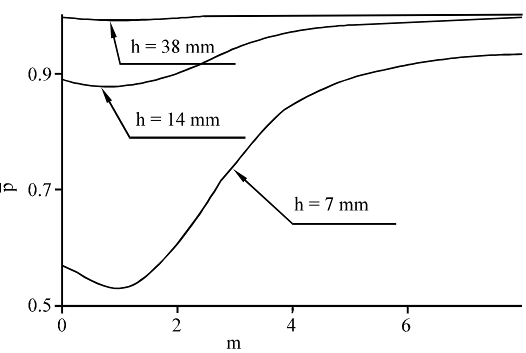

Fig. 8: Distortion of a spectrum of eigen frequencies of the model in the capacity of arotation symmetrical system at increasing the disk thickness

how much their blades are nonidentical but also on a connectivity of vibrations. The less is the last, the more is circumferential distortion of eigen modes. For any design of a wheel decreasing in a connectivity of vibrations leads to qualitatively the same process of distortion of eigen modes. It is possible to select two stages in it. In the course of the first there is a build-up of a mode distortion that expresses in the relative decreasing in vibration amplitudes of the majority of blades. In such a case, the maximum amplitude may appear in one or another blade. The stage comes to an end when the mode accepts the

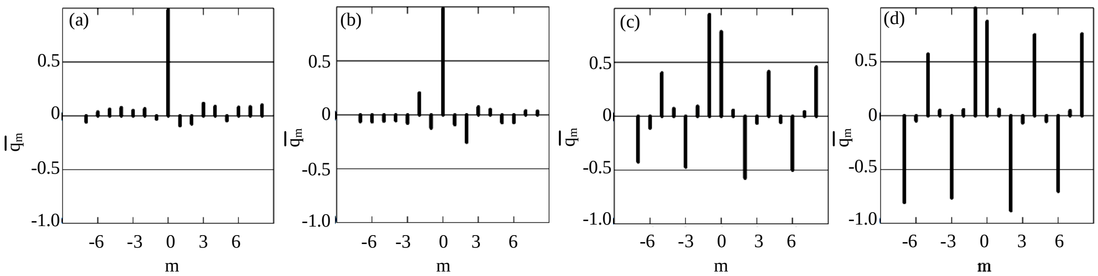

Fig. 9: A change of a harmonic composition of the vibrations mode with m = 0 at increasing of disk thickness of the model rotor wheel; a) h = 7 mm; b) h = 14 mm; c) h = 24 mm and d) h = 38 mm

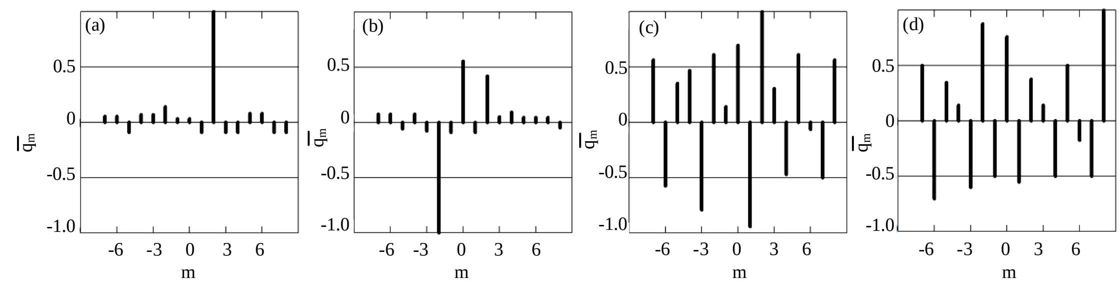

Fig. 10: Change of a harmonic composition of the vibration mode with m = 2 at a increasing of thickness of a disk of the model rotor wheel: a) h = 7 mm; b) h = 14 mm; c) h = 24 mm and d) h = 38 mm

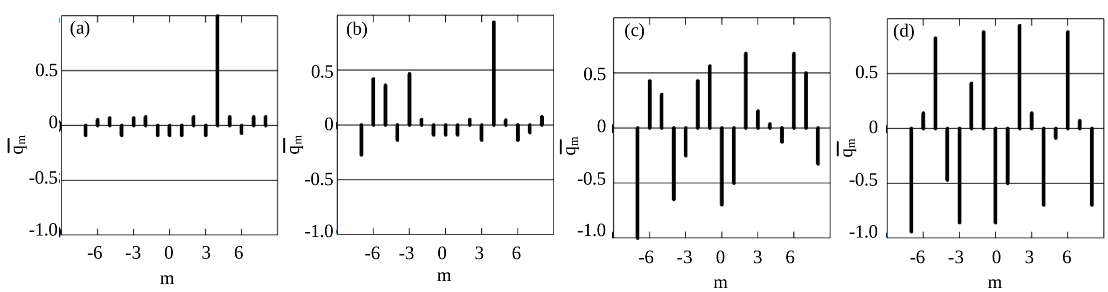

Fig. 11: Change of a harmonic composition of the vibration mode with m = 4 at increasing of disk thickness of the model rotor wheel: a) h = 7 mm; b) h = 14 mm; c) h = 24 mm and d) h = 38 mm

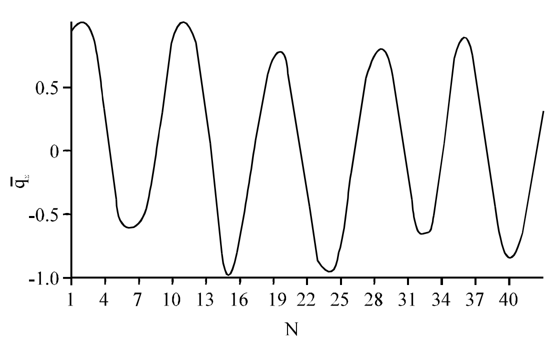

Fig. 12: The vibrations mode with m = 4 of the engine NK-12 compressor 2nd stage rotor wheel

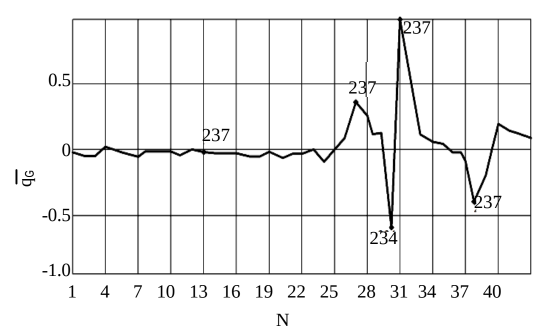

Fig. 13: The vibrations mode of the engine NK-12 compressor 4th stage rotor wheel

strongly pronounced localized form. Under vibrations according to this mode the rotor wheel cannot be considered as having small deviation from rotation symmetry. In the second stage of distortion localization isincreased. Vibrations start to be focused on one blade and finally turn to its single displacements. At this stage, the maximum amplitude is always saved at the same blade.

CONCLUSION

It is necessary to note that it is senseless to characterize the localized modes by the number of nodal diameters. The modes are not self-equilibrated and consequently can be an excitation source for vibrations of a rotor.

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