INVESTIGATION OF DYNAMIC CHARACTERISTICS AND DAMPING CAPACITY OF BIMETALLIC GAS TURBINE BLISKS

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Abstract. The investigation of bimetallic blisk's usage produced by HIP process from powder and cast alloys, is perspective for development of gas turbine engines. But absent of construction damping in blades-disk root connection can lead to increase dynamic stresses. This problem may lead to limitation of blisk usage.

The main task for blisk design is to compose its elements in such a ways, that high resistance to vibrations might be achieved. One of the perspective directions for this task fulfillment is new designing of dry damping devices, inserted under lower blade shrouds. It is possible if blade design includes long shank of a necessary size and the damper's parameters of mass and deformation ability have been rightly chosen.

The tests of bimetallic blisk, manufactured by HIP process from powder and cast Ni-base super alloys, in condition of resonance vibration excitation are carried out. Then some structural procedures were providing in purpose to increase compliance of blades and damping. The optimal damping devices mass is determinate for the blisk.

Introduce. Increasing aviation engine parameters requirements leads to necessary of using new materials, new designs. Usage of heat resistant single crystal materials for gas turbine blades build-in powder disk (bimetallic blisk) by hot isostatic pressing (HIP) method is having prospects for aviation gas turbine engine next generation. This structure makes it possible to eliminate complicated mechanical (flange and root) joints, welding and brazing. Previous investigations [1,2] are show that it leads to increase the service time and cycle life, margins of safety, decrease the weight of bimetallic wheels, etc.

However absent of construction damping in blades-disk root connection can lead to increase dynamic stresses. This problem may lead to limitation of blisk usage. Optimal design of blisk subject to the requests of durability and processing limits is necessary for increasing of structure damping and therefore decreasing vibratory stress.

One of the perspective directions for this task fulfillment is new designing of dry damping devices, installed under lower blade shrouds. It is possible if blade design includes long shank of a necessary size and the damper's parameters of mass and deformation ability have been rightly chosen.

Investigation of dynamic characteristics and damping capacity of bimetallic difference configuration gas turbine blisks by calculation and tests are presented in this article.

Calculation investigations of difference configuration wheel dynamic characteristics.

Some calculations and tests of bimetallic blisk: a rootless construction produced by joining blades of single crystal alloy and a disk of powder alloy by means of hipping (HIP technique) are carried out early in condition of resonance vibration excitation. [3]. Blades had difference configuration part under platform (types A, B and C).

Type A. Blade footing is made in a shape of a root block. The material-joining surface runs along a blade's root and lower shroud (fig. 1a). The shanks were absent; clearances between shrouds were absent too.

Type B. Bimetallic blisk had blades with small shanks (fig. 1b). The shanks were been make by burning through by electric spark method. The clearances between shrouds were making.

Type C. New design of blisk was been make from previous structure by burning through radial gaps into disk between shanks by electric spark method (fig. 1c). This operation provides elongation of shanks.

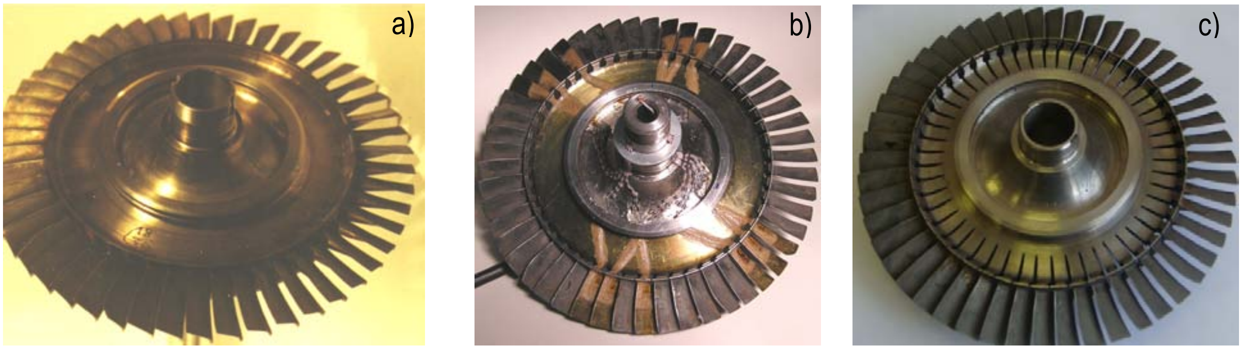

Figure 1. Bimetallic blisks. The blades have: a) without shanks, b) with short shanks, c) with long shanks.

Some entry-level schemes of vibration are consider. Vibrations of blades with inflexible disk, inflexible blades with flexible disk and combined vibration of system “disk-blades” is investigated with help 3D models by finite-element method (ANSYS program). Speed rotation was about 52000 rpm. Temperature field was corresponded work state.

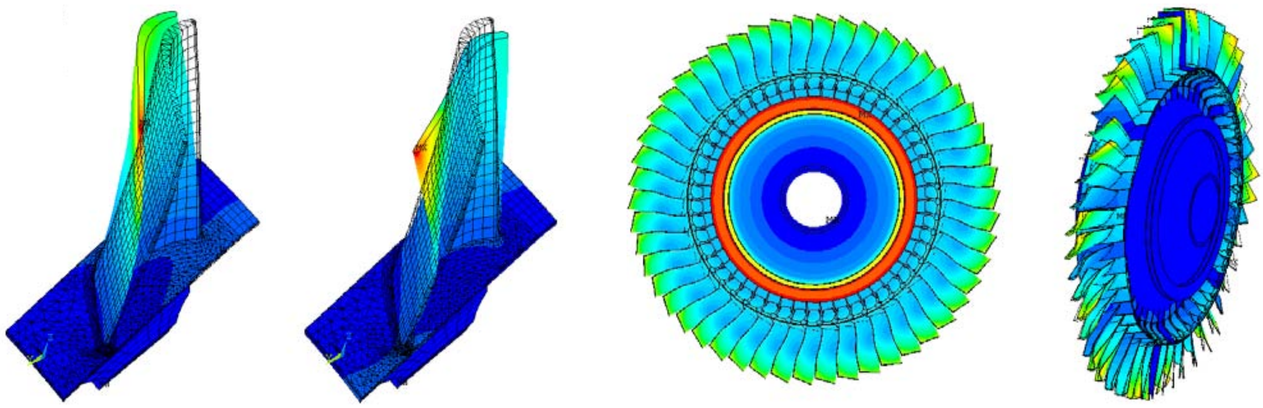

Figure 2. 1^{th} and 2^{nd} natural frequency modes of blade with inflexible disk (on left), 6^{th} natural frequency mode of blisk without nodal line (on center) and 2^{nd} natural frequency mode of blisk (m=7).

Calculations of natural frequencies and mode shapes are carried out for three blisk structures and assembly wheel same size which has blades of SC alloy and a disk of PM joined with the fir-tree root connections (root wheel). First four natural frequencies of wheels by nodal lines (m=0…15) were determinate.

Some vibration modes of blade and blisk are shown at fig. 2.

First (f1) and second (f2) natural frequencies with nodal lines (m=0…7) of four wheels are given at fig.3.

General first mode shape transition (from disk-shaped to blade-type) is occurred when m=3. Second mode is become transformed when m=4. Disk may be consider as inflexible after m=5.

First natural frequencies (m= 0…2) of investigational wheels are differ few because it are depended from disk rigidity generally. Natural frequencies of blisks with large quantity of nodal line are changed considerably subject to wheel structural features. All natural frequencies of types “A” and “B” blisks are more than one of “root wheel” because solid support free sum length of blade with shank upper is smaller. Certain free displacement exists into blade-root connection. Availability of long radial gaps into disk (type “C”) lead to decreasing of natural frequencies and approaching to “root wheel” ones.

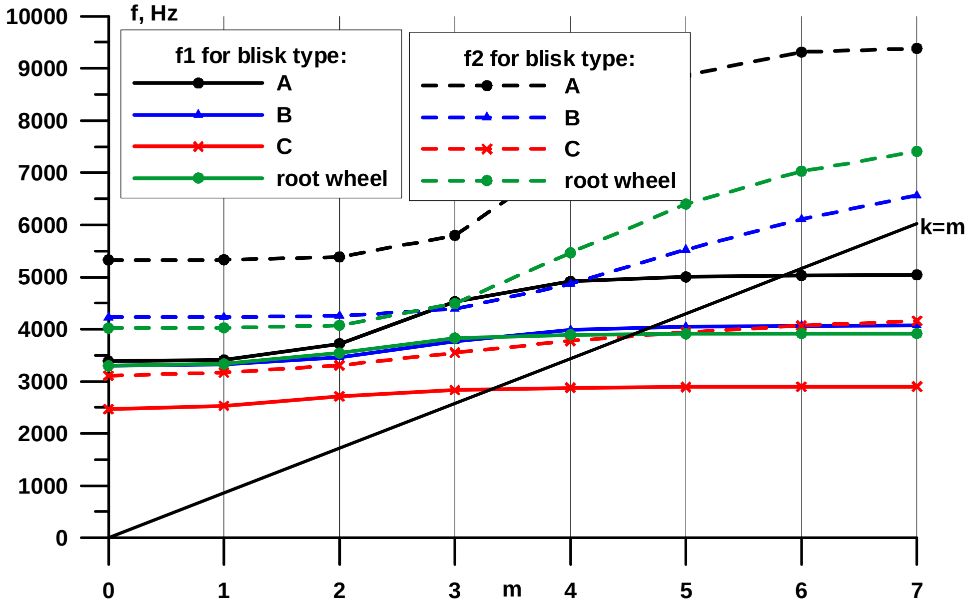

Figure 3. Dispersion diagram of four wheels. First and second natural frequencies with different nodal lines.

The decreasing of natural frequencies leads to changing of stimulation harmonic of oscillations (k) witch can excite resonance oscillations of blisk by first mode with nodal lines m=k.

All above mentioned is necessary take into account during design of such structure.

The problem of strength vibration of wheel blades.

Often the decision of strength vibration problem of wheel blades is including special damping device to wheel structure for absorption of energy of stimulant forces as the result of friction forces works on contact facets of ‘blade-damping’. The same effect is realized into root looks blades - disk connections.

Absent of structural damping into root looks blades - disk connections lead to the increasing of dynamic stresses of blades. The problem of structural damping of blisks is especially actuality.

The blades with shanks and dampers are using into structure of turbine wheels. These dampers are type of insertion setting under tract shrouds and pressing to him by centrifugal forces.

The damping power of blades with insertion dampers is formed by works of friction forces on contact facets of damper in during displacement of the dampers relatively shrouds when neighboring blades are made vibration. The press force of insertion depends on speed rotation and damper mass. The optimal damper mass for ensuring of maximal damping may be select for each blade and given speed rotation by calculation and experimental methods.

There is method of the forced nonlinear vibration calculation of blade with dry friction damper models for estimation of damper using efficiently and determination of rational value of his parameters with account influence of geometrical characteristics of blades [4].

Test investigations of dynamic characteristics and damping problem are made into this paper.

Test equipment and parameters for investigations of difference configuration wheel dynamic characteristics.

Previous test dynamic investigations of “A” and “B” type’s bimetallic blisks [5] are show that some probe-tested structure decisions allow to decrease maximal resonance vibration level on ~ 20% and increase logarithmic decrement by 1.5-2 times. The vibration excitation of rotating blisk is making by air jet nozzles.

Test of type “C” blisk were carried out on special equipment (fig. 4).

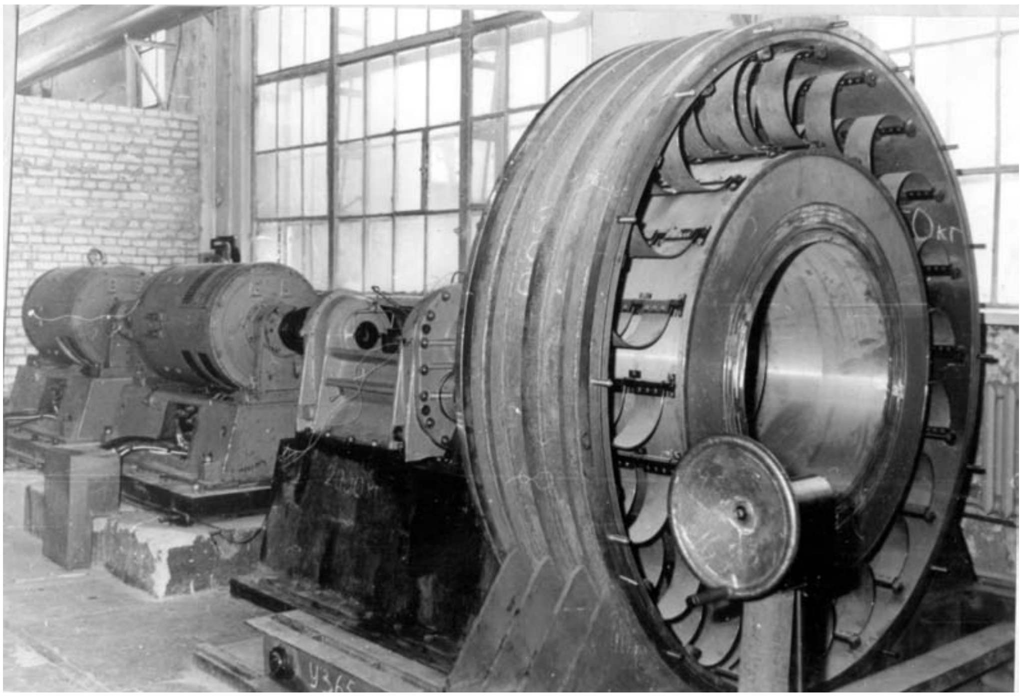

Figure 4. The equipment for tests of wheels, investigation of dynamic characteristics and optimization of structural damping of turbine blades.

The equipment is using for test of turbine wheel in centrifugal field for investigate of the stress-strain state on cyclic forces, frequency and modes of vibration, vibration stresses, damping of blades and disks.

The dynamic behavior of bimetallic blisk in condition force air excitation of vibration critical frequencies was investigated. The excitation vibration system of blisk blades was made with help 40 air jet nozzles (fig.5). The collection and processing system of test information was made. Speed rotation and input/output traffic control was performed by special program generated on base HP VEE 5.0 for work with VXI (VMEbus eXtention for Instrumentation) by firm Agilent Technologies. Initial and final speed rotations were setting by experimenter subject to anticipated value of blade first natural frequencies. When resonance speed rotation is determinate, the speed rotation range about this value and changing step are setting to recode. The speed rotation at every step is last set time. Speed rotation is changing in the interior of the range from lower range value to upper value and backwards. The signal recording was performed during this process.

Actuating signals from resistive-strain sensors installed on blades with using signal amplifier ware recoded by recorder-analyzer MIC-300M using program MR-300.

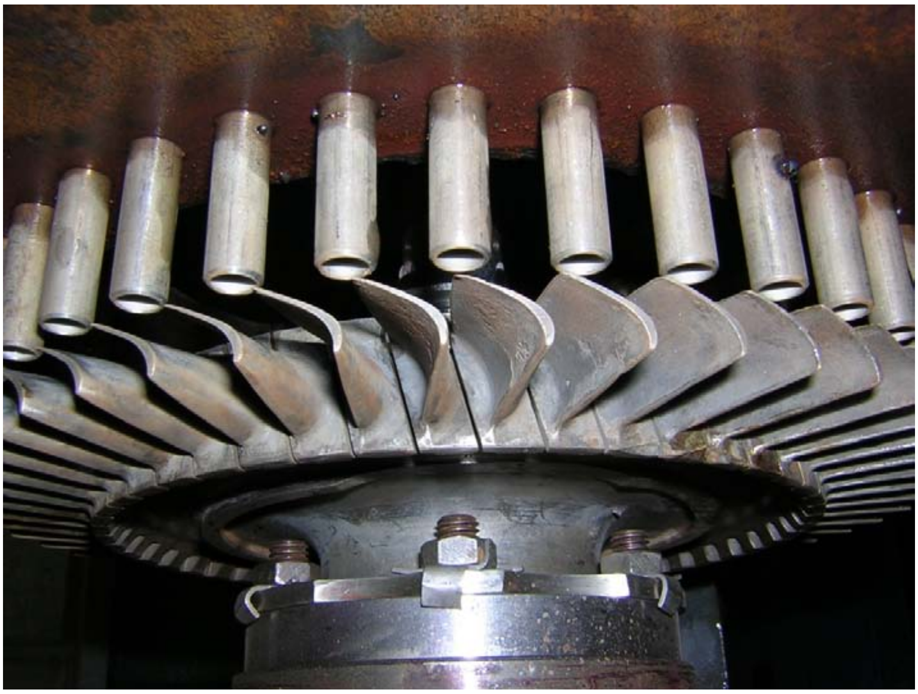

Figure 5. The blisk with air jet nozzles set in the test equipment.

Signal processing was performed by virtual device generated with help software environment “LabView4”. The production performance is next. The selection length on channel is 16384 points; sampling rate is 27 kHz, and Hanning filter.

First natural frequencies of ten blades of wheel were investigated. The blades cast from single crystal material had different directions crystallographic axis <001> with blade radial axis. This parameter was influenced to value of first natural frequencies of blisk blades. The frequency spread of nine blades of “C” blisk determination by calculation (f1) was from 2811 to 3067 Hz. So speed rotation for oscillatory impulse of first natural frequencies of nine blades was changed in the range 4050…44000 rpm. All blades were excited oscillations during tests.

Investigations were carried out for blisk without damping devices and with damping devices of different mass: 0.66g, 0.80g, 0.98g, 1.22 g. Damping devices are made tube-shaped which are installed between blade shanks (fig. 6).

Figure 6. The blisk of type “C” with damping devices.

Test results were compared with previous blisk structures (with short shanks and without shanks) too.

Results of test investigations

Typical test results of blisks without damping are shown at figure 7 for one from blades. The time changing speed rotation (RPM - red line), quadratic mean value of vibration amplitude (RMS - blue line) and first natural frequency (f - green line) into range of first natural frequency vibration excitation are shown at fig. 7a. The amplitude-frequency characteristic in compliance with value vibration amplitude (Max_Spectr) is shown at fig. 7b. The dependence RMS-f is shown at fig. 7c.

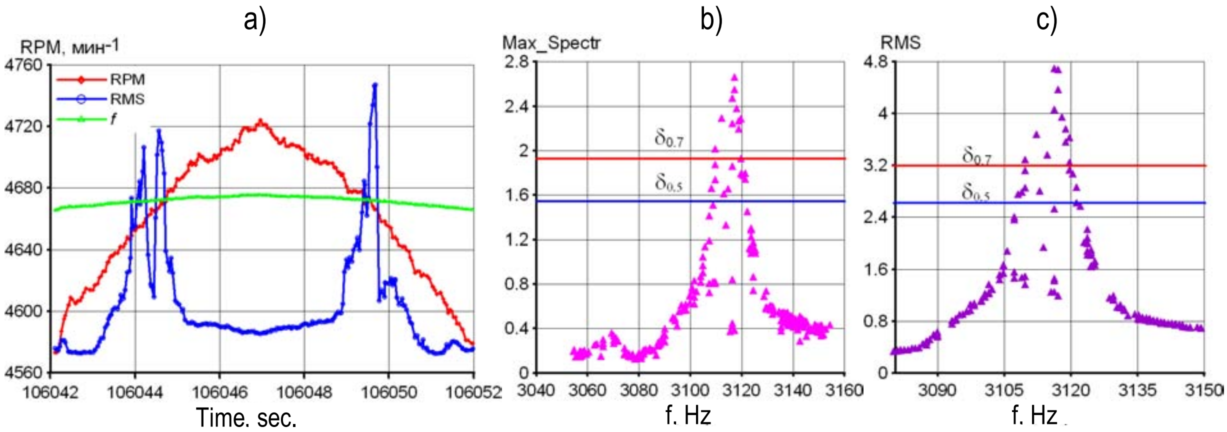

Figure 7. a) The time changing RPM, RMS, f; b) The amplitude-frequency characteristic; c) The dependence RMS-f.

Damping decrements are determinate by formulas:

$$\delta_{0.5} = \pi \frac{\Delta f_{0.5}}{\sqrt{3} \cdot f} \cdot 100\%, \qquad \delta_{0.7} = \pi \cdot \frac{\Delta f_{0.7}}{f} \cdot 100\ \%,$$

Where $\Delta f_{0.5}$, $\Delta f_{0.7}$ - width of amplitude curve up-to-date 0.577 and 0.707 from maximal amplitude into first frequency band (blue and red lines at fig. 7b,c).

Calculate and test first natural frequency of some single crystal blades with difference crystallographic orientations and difference damper mass too are given in table 1.

Table 1. Calculate and test first natural frequency of some single crystal blades with difference crystallographic orientations.

			Calculate	f, Hz Tests				
Blade №	α, °	E, kg/mm²	f, Hz	without damping	Damper mass, g			
					0.66	0.8	1.22	0.98
2	11.3	14605	2842	2973	3002	3026	3160.8	4879
3	10.2	14439	2837	2916	2947	2999	3149	4573
17	41.9	23121	3067	2958	3356	3398	3404	4372
19	0	13717	2811	2846	2855	2969	3082	4584
25	-16.7	15672	2877	2986	2999	3079	3114	4523
27	36.8	22089	3036	3117	3195	3229	3274	4601
42	-31	20198	2996	2933	2999	3015	3179	4462
43	38.6	22536	3045	3013	3264	3141	3167	4727
45	31	20197	2998	2823	2857	2905	3148	4497
Average				2952	3053	3085	3186	4580
%				100	103.4	104.50	107.95	155.16

The α is angle between directions of blade radial axis and crystallographic axis <001>. These angles are difference into investigational blades so longitudinal elastic modulus of each blade is differing and its first natural frequencies is differing too.

Calculate value of blade first natural frequency is coming to agreement with test one.

Average values of blisk blade first natural frequencies are differing negligible (on 3.5, 4.5, 8% with respect to damper absent) with increasing damper mass on 0.66, 0.8, 1.22 g on the average. Damping devices of mass=0.98 g were setting with tightness. So length of blade vibration part is decreasing. The first natural frequencies are increasing on 55% and approaching to blisk without shank (type "A").

Arithmetical mean value of damping decrement δ % between $\delta_{0.5}$ and $\delta_{0.7}$ for some blade with difference dampers mass are given in table 2. Damping decrements of difference blades are differ. The average damping decrement of all investigation blades are given in table 2 too. The ratio of average damping decrement (for each damper mass) to one without damping (k_m) for blisk "C" is given in penultimate line of table 2.

Table 2. Damping decrement (δ, %) of blisks "B" and "C" with difference dampers mass.

Blade №	Blisk "B"			Blisk "C"				
	without damping	Damper mass, g		without damping	Damper mass, g			
		0.8	1.22		0.66	0.8	1.22	0.98
2	1.092	1.296	1.080	0.955	3.811	6.450	4.063	0.934
3	0.654	0.689	0.830	0.711	4.206	4.673	-	3.068
17	0.928	0.918	0.514	–	4.301	5.694	3.363	3.266
19	0.695	0.514	0.580	0.585	2.362	4.545	5.060	2.959
25	0.646	1.391	1.344	0.543	4.132	4.940	6.524	2.779
27	0.649	1.068	1.790	0.961	2.985	3.956	8.548	1.645
42	0.324	0.498	1.125	0.483	3.020	5.185	6.630	1.146
43	-	-	-	1.147	3.239	6.973	4.852	1.685
45	-	-	-	0.678	2.845	7.279	4.368	2.214
Average	0.713	0.911	1.038	0.758	3.433	5.522	5.426	2.188
k_m				1	4.5	7.3	7.2	2.9
K_l	1	1	1	1.063	-	6.06	5.27	-

The optimal damper mass for this blisk "C" is 0.8 or 1.22 g for difference blades.

Damping effect for damping devices inserted with tightness is decreasing so length of blade vibration part is decreasing and dry friction effect is decreasing.

Damping decrements of blisks "B" and "C" without damping devices and with difference mass (0.8, 1.22 g) dampers are compared. The ratio of damping decrements average value for each damper mass of blisk "C" to one of blisk "B" (k_l) are given in last line of table 2.

Damping decrements of blisks "B" and "C" without damping devices are differ few. Using of damping devices is increasing vibration damping effect by 5-6 times for blisk "C" in comparison with blisk "B" having same mass damping devices.

Comparative analysis is carried out for four blades with different crystallographic orientations of blisks types "A", "B" and "C".

First natural frequencies of under test objects are shown at diagram (fig. 8a). Dynamic parameters of blade No 2 (blisk "C") were investigated. Dependences first natural frequencies, damping decrement and vibration stress from damper mass installed to blisk "C" are shown at diagram (fig.8b).

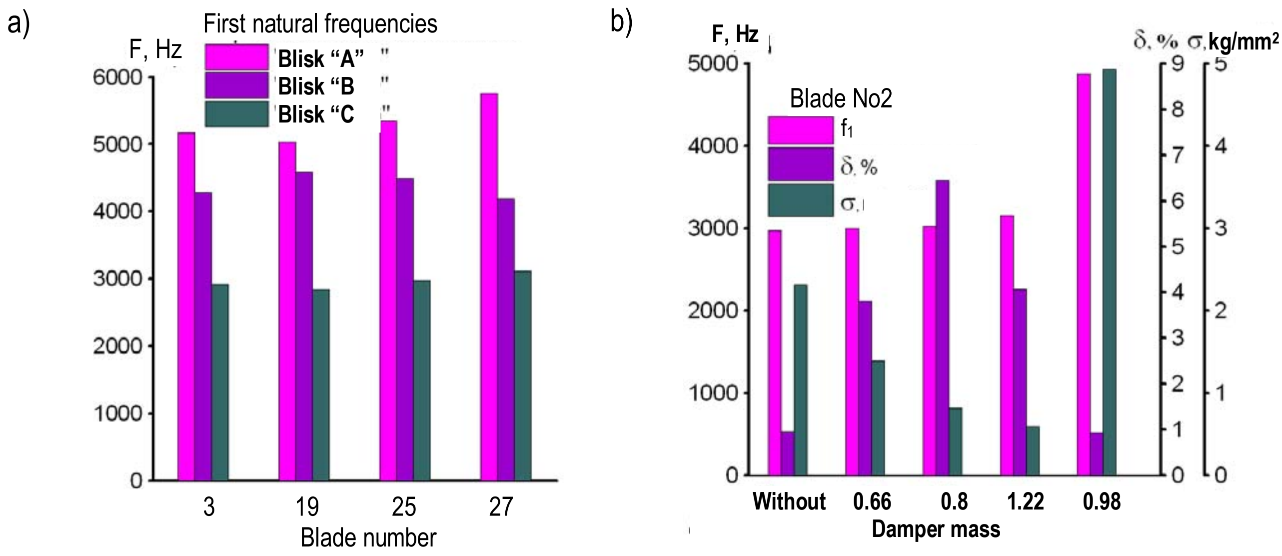

Figure 8. a) First natural frequencies of blisks "A", "B", "C"; b) Dynamic parameters of blade No 2 (blisk "C").

Conclusions.

Application of blisk blade shanks is decreasing value of first natural frequency by 1.2…1.8 times with increasing of blade vibration part by 1.12….1.56 times.

Setting of damping devices is increasing damping decrement of blade vibrations by ~ 4…7 times. The optimal damper mass for this blisk "C" (with long vibration blade parts) is 0.8 -1.22g. These damping devices are increasing damping decrement of blade vibrations by 7 times.

The level of vibration stress is decreasing by 4…7 times subject to damper mass in comparison with damping devices absent.

These dynamic characteristics are necessary to take into account for design process of blisks.

These investigations will help to decide of blisk vibration problems.

Reference.

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