УДК 629.7.036.34

INVESTIGATION OF INFLUNCE OF ROTOR SUPPORTS ELASTIC ELEMENTS CHARACTERISTICS ON GAS TURBINE ENGINES

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The article considers task of determination of stiffness characteristics of elastic elements - squirrel cage and ring elastic elements set in gas turbine engines supports. Significant dependence of elastic elements characteristics on the engine operating mode and their non-linearity is noticed. Results of dynamic analysis of the rotor system of the real gas-turbine engine calculated at linear and non-linear statement are compared.

Key words: rotor dynamics, DYNAMICS R4, elastic elements, nonlinear analysis.

Generalities

One of the most effective means to decrease general vibrations and dynamic stress in the gas turbine engine (GTE) units is elastic-damping supports. Designs of these supports are different, but irrespective of this they perform two main functions:

1. They decrease the rotor support stiffness that leads to change in characteristics of the elastic dynamic system of the engine. At the same time the system natural frequencies decrease, resonances at operating modes are avoided.
2. They absorb vibration energy of the engine rotor system, converting it to heat that does not allow developing of high vibrations amplitudes, dynamic loads and stress in the engine details.

At the present time mainly two elastic elements are used in the rotor supports of aviation GTE –squirrel cages and ring elastic elements (Fig. 1).

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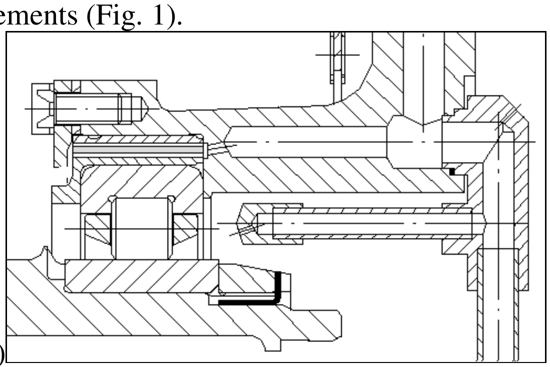

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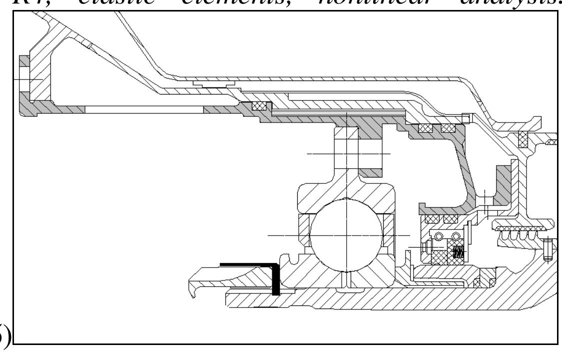


Fig. 1 – Elastic-damping supports
a)example of support design with ring elastic element; b) example of support design with squirrel cage

Squirrel cages have stable enough stiffness characteristics and they are used to carry out frequency detuning of rotor critical speeds from operating mode. They may be mounted together with the hydrodynamic dampers. Ring elastic elements are used both for frequency retuning and for creating of damping cavity.

Squirrel cages may be mounted in supports both with angular-contact ball bearings and radial roller bearings. In the first case they transmit radial and axial loads to the case, in the second case – only radial. One of the main characteristics of squirrel cages is their radial flexibility determining elastic characteristics of the rotor system.

As a rule, supports with ring elastic elements together with squirrel cages are used for the engines mounted on the aircrafts that bear high evolutional overloads.

Squirrel cages

Only flexibility under radial force is traditionally taken into consideration at estimation of elastic characteristics. Influence of some other factors is not taken into account.

Those factors are deviations of elastic beams dimensions (it is shown in work [1] that deviation of squirrel cage flexibility in two orthogonally related directions may be up to 20-30%), influence of axial force transmitted through the bush to the case, change in elasticity modulus of elastic elements material from their temperature, impact of additional evolutionary loads.

Axial load exceeding radial loads significantly may achieve several tones at some regimes of the gas turbine engines work and may change the squirrel cage flexibility considerably. Nonlinear change in axial load at regimes leads to nonlinearity of squirrel cage flexibility.

Let us consider the elastic element in the support of the high pressure compressor (HPC) of the AL-31F engine designed as squirrel cage (Fig. 2).

Fig. 2 – Elastic element "squirrel cage" in HPC support

Table 1 shows geometry and characteristics of squirrel cage material.

Table 1

Parameter	Value
Beam width a, mm	5,2
Beam thickness b, mm	2,6
Beam length l, mm	34
Beams number n	64
Squirrel cage material	Titan alloy
Elasticity modulus E, N/m^2	$1{,}1 \cdot 10^{11}$
Poisson ratio, μ	0,3
Material density ρ, kg/m^3	4500

Analytical model to determine elastic element flexibility is the following [2]:

$$\delta = \frac{nEab(a^2 + kb^2)^3}{2l^3},$$

where

$$k = \frac{1}{(1 + \frac{2\sqrt{ab}}{l})^3}$$ – correction factor depending on the elastic support geometry. It should be noticed that in this equation the squirrel cage flexibility is determined only by elastic beams flexibility, other units flexibility is not taken into consideration. Calculated in this equation the squirrel cage flexibility δ accounts for $2{,}68 \cdot 10^{-5}$ mm/N.

Joint influence of radial load (weight force) and axial force on the squirrel cage flexibility is investigated on the model built in the ANSYS program. Fig.3 shows the finite-element model of the squirrel cage.

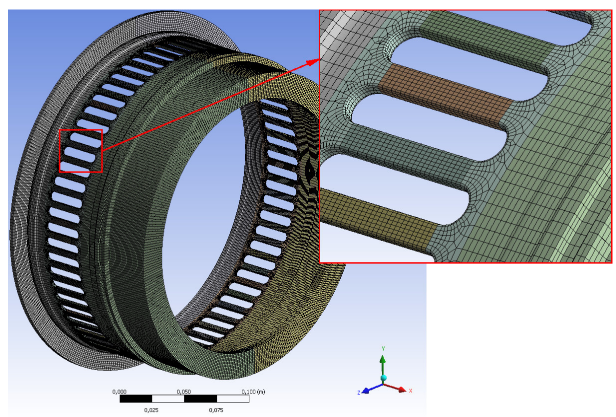


Fig. 3 – Finite-element model of squirrel cage

To estimate axial load influence of the squirrel cage stiffness, FEM-model is loaded by axial load whose value changes at operation from 0 to 60000 N, along with the fixed radial load of 1000 N that imitates weight force. Fig.4 gives the calculation results.

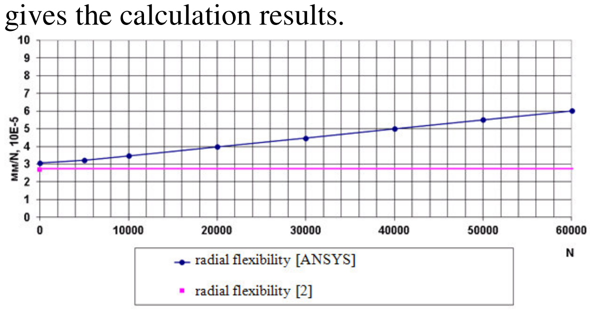


Fig. 4 – Axial force influence on squirrel cage flexibility

As calculation shows, radial displacements of the rotor support, where the squirrel cage is designed, depend on axial load taken by the support. At the same time if radial flexibility virtually correspond to the one calculated analytically (~12% difference) at zero axial load, then radial flexibility increases virtually in two times: from $3{,}07 \cdot 10^{-5}$ to $6{,}01 \cdot 10^{-5}$ mm/N at the axial force of about 60000 N.

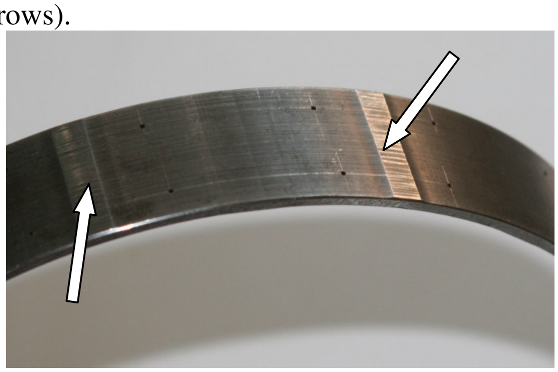
It is also obvious that the squirrel cage transmit not only radial and axial load but also moments. It means that at calculation of whole rotor systems including cases and suspension, it is necessary to create complete flexibility (stiffness) matrixes. Fig. 5 shows an example of the complete stiffness matrix of the squirrel cage obtained in the DYNAMICS R4 program system.

		Fx	Fy	Fz	Mx	My	Mz
		N	N	N	N m	N m	N m
ut_x	m	3.25e+007	126000	119000	8010	1.31e+006	-25700
ut_y	m		3.07e+007	38800	-1.31e+006	39100	-5550
ut_z	m			1.52e+009	73600	133000	-41500
ur_x	rad				1.25e+007	-9470	-573
ur_y	rad					1.25e+007	-566
ur_z	rad	symm					592000

Fig. 5 – Complete stiffness matrix of squirrel cage

Ring elastic element

Analytical solutions determine the ring flexibility by solving of the task about a curved beam supported along the edges [2]. At the same time, practice shows presence of sliding traces on the different ring surfaces that suggests the more complicated ring loading under the rotor precession. For instance, work [3] also notices that the ring works with breakaway of inner and outer lugs from contact surfaces. Fig. 6 shows surfaces of the ring elastic element with marks of sliding (shown by arrows).

a) 

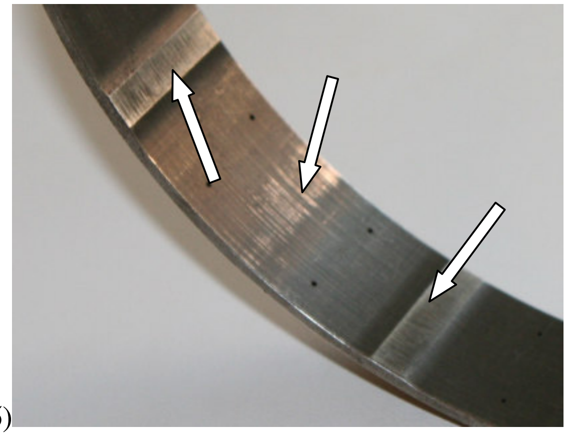
б) 

Fig. 6 – Ring elastic element with marks of sliding a) at outer lugs; b) at inner lugs and inner ring surface between lugs

Work [4] shows that stiffness of ring elastic elements set in elastic-damper supports is nonlinear and depends on fit of the ring elastic element, tolerances on sizes and on the value of radial forces acting in supports and transferred through ring elastic elements. The value of the ring flexibility may change in 2-3 times. During operation sliding of the ring lugs relative to cases may take place. Fig.7 gives an example of the elastic ring stiffness depending on changing radial load.

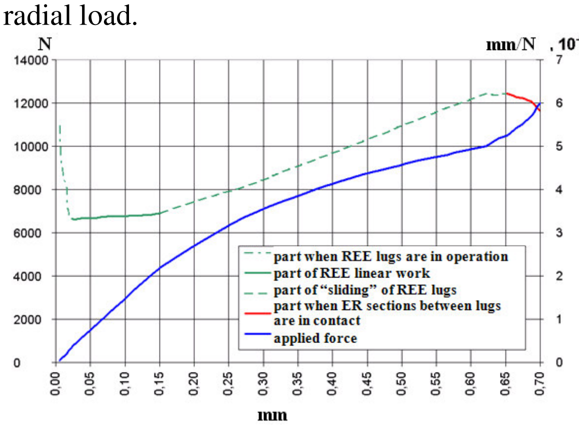


Fig. 7 – Dependence of ring elastic element (REE) flexibility on radial load in support

As investigation results show, the ring elastic element stiffness is relatively linear only in limited range of radial loads and start changing significantly when these loads are increasing. At the same time it should be noticed that the ring elastic element flexibility is rather higher than the one calculated analytically considering that only the ring sector between two lugs is taken into account, and possibility of deformation, contact and sliding in other elements of vibration package is neglected.

Analysis of AL-31F rotor system

Let us consider influence of nonlinear characteristics of elastic supports on dynamics of the rotor system of the AL31-F engine. Investigation took place in the DYNAMICS R4 program system. The rotor system of the AL-31F engine (see Fig. 8) includes high and low pressure rotors (HPR and LPR). HPR has two supports, LPR – four supports, elastic-damping elements are included into construction of three supports – front support of the low pressure compressor (the squirrel cage), the high pressure compressor (the squirrel cage) and the low pressure turbine (the ring elastic element) shown in Figure by “S1”, “S2” and “R”, correspondingly.

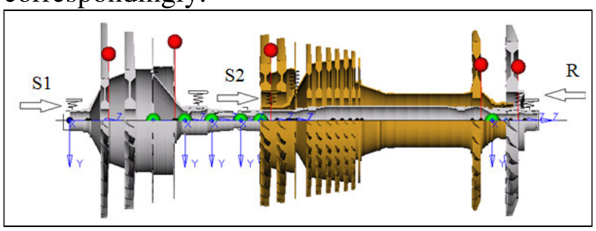


Fig. 8 – Rotor system of AL-31F engine

As it has been already mentioned, flexibility of supports, including elastic elements, depends on the regime and loads acting on them. Fig. 9 and 10 show graphs of change in axial loads and flexibilities of the supports with the squirrel cages “S1” and “S2” depending on the engine operating mode.

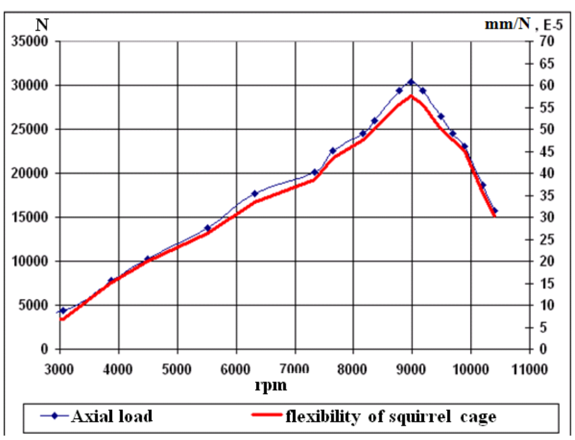


Fig. 9 – Dependence of axial load and flexibility of “S1” support on LPR rotating speed

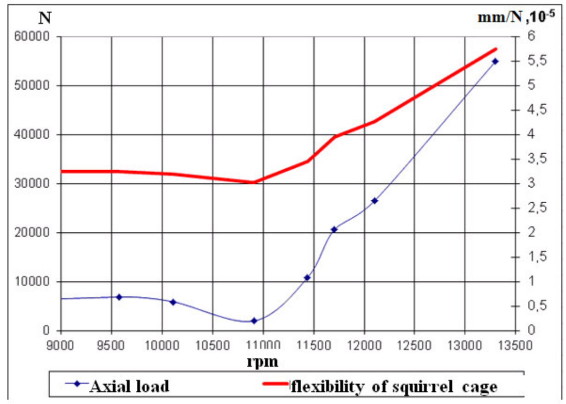


Fig. 10 – Dependence of axial load and flexibility of “S2” support on HPR rotating speed

For the LPT support flexibility depends on radial load due to the rotor weight and unbalance. Fig.11 shows the graph of radial force dependence in “R” support on the engine operating mode (rotating speed) and the corresponding change in the support flexibility.

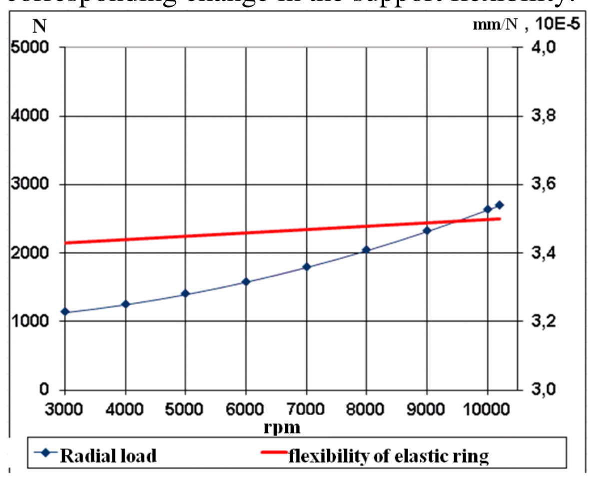


Fig. 11 – Dependence of radial load and flexibility of “R” support on LPR rotating speed

Table 2 shows factors of the supports flexibilities used in the rotor system model.

Table 2

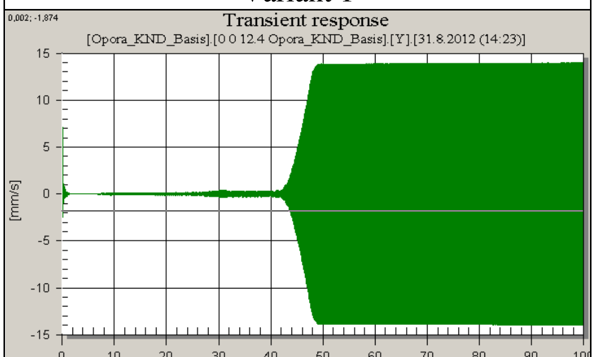
Support	Radial flexibility, mm/N·10^{-5}	
	Variant № 1	Variant № 2
LPC front support («S1»)	6,8	6,8…56,3
LPC rear support	1,0	1,0
HPC support («S2»)	2,68	2,68…5,9
LPT front support	1,0	1,0
LPT rear support («R»)	3,3	3,3…3,52

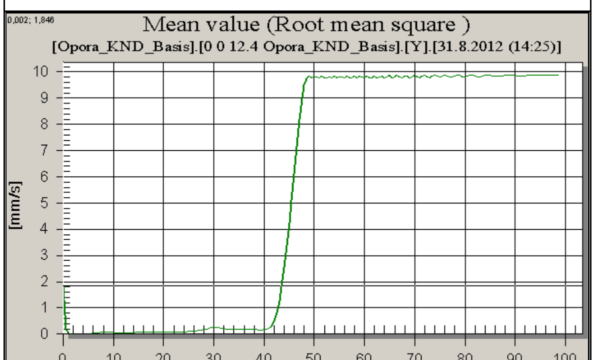
HPT support	0,5	0,5

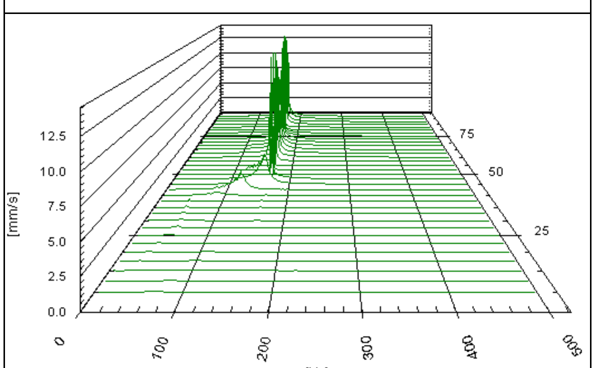
1st variant of the model was analyzed at linear statement, 2nd variant – at nonlinear statement. Table 3 gives the results for linear and non-linear rotor system correspondingly obtained in transient response (by example of LPC front support ("S1" support)). Amplitude-time characteristics, mean value of time signal, waterfall diagram of vibration spectra are presented. Table 4 shows the rotor natural frequencies at the operating mode. Fig.12-21 present mode shapes.

Table 3

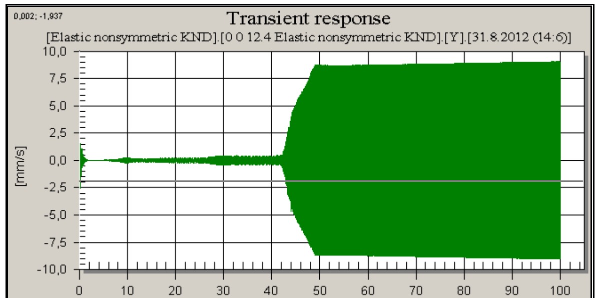
Variant 1

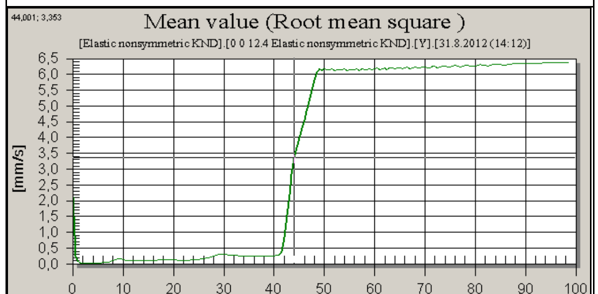






Variant 2





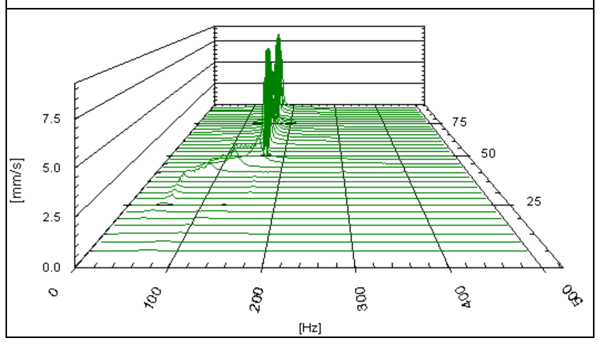


Table 4

Shape number	Frequency, Hz	
	Variant № 1	Variant № 2
1	54,6	59,0
2	107,9	73,8
3	141,5	94,1
4	266,9	267,9
5	294,1	295,5

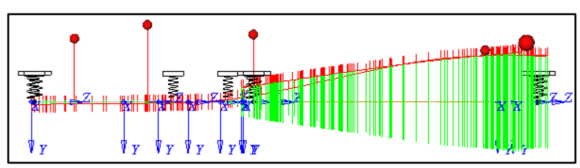


Fig.12 – Mode shapes of rotors at first natural frequency 54,6 Hz (variant № 1)

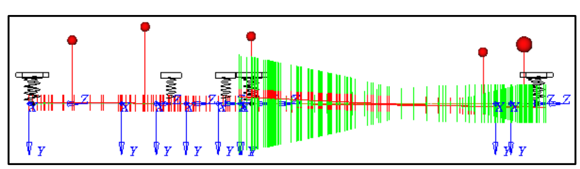


Fig.13 – Mode shape of rotors at second natural frequency 107,9 Hz (variant № 1)

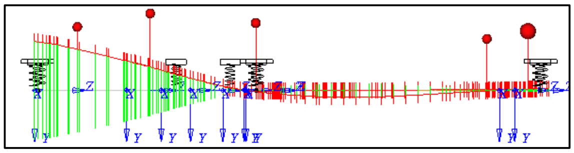


Fig.14 – Mode shape of rotors at third natural frequency 141,5 Hz (variant № 1)

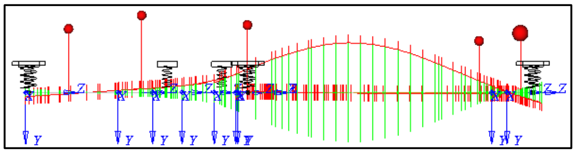


Fig.15 – Mode shape of rotors at fourth natural frequency 266,9 Hz (variant № 1)

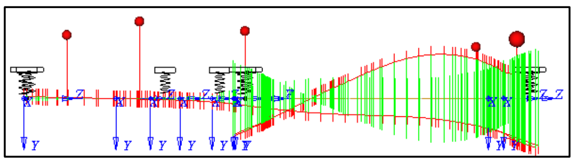


Fig.16 – Mode shape of rotors at fifth natural frequency 294,1 Hz (variant № 1)

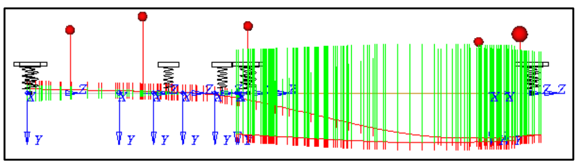


Fig.17 – Mode shape of rotors at first natural frequency 59,0 Hz (variant № 2)

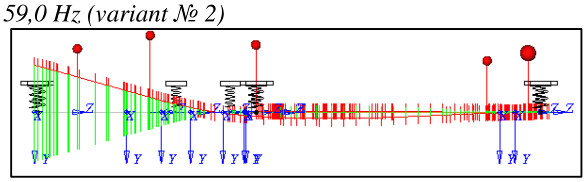


Fig.18 – Mode shape of rotors at second natural frequency 73,8 Hz (variant № 2)

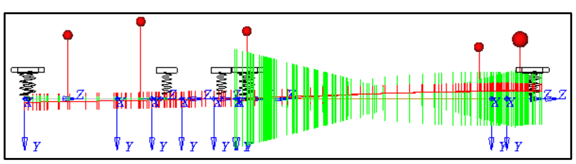


Fig.19 – Mode shape of rotors at third natural frequency 94,1 Hz (variant № 2)

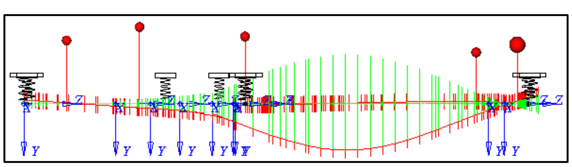


Fig.20 – Mode shape of rotors at fourth natural frequency 267,9 Hz (variant № 2)

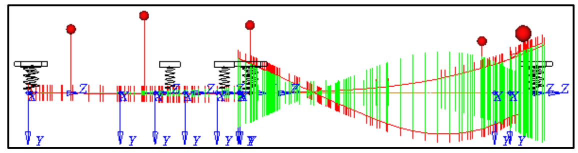


Fig.21 – Mode shape of rotors at fifth natural frequency 295,5 Hz (variant № 2)

Conclusion

The presented results show necessity of more exact estimation of flexibility of elastic elements used in supports of GTE rotors. Their value may change in several times depending on the engine operating mode.

Natural frequencies of the rotor systems change significantly at operating modes taking into consideration influence of exploitation loads; sequence of mode shapes may change.

General dynamics of the GTE rotor systems must be analyzed at nonlinear and unstable statement considering variety of factors noticed above and changing according to the regimes.

Literature

1. Леонтьев М.К., Фомина О.Н. Активное управление жесткостью опорных узлов роторов. Конструкция и статический анализ, Журнал «Вестник Московского авиационного института» т.14 № 4, 2007 г., стр. 57-62.
2. Леонтьев М.К. Конструкция и расчет демпферных опор роторов ГТД: Учебное пособие. - М.: Изд-во МАИ, 1998. – 44 с.:ил.
3. Лобанов В.К., Хрусталев А.Б. Оценка демпфирующих свойств одного типа упругих опор ГТД, Сб. ”Вибрационная прочность и надежность двигателей и систем летательных аппаратов”. КуАИ, 1977 г., с.91-96.
4. Леонтьев М.К., Терешко А.Г. Исследование характеристик упругих колец в опорах роторов газотурбинных двигателей, Вестник МАИ. 2011. Т.18 № 3. С. 135-146.