RESEARCH OF CHARACTERISTICS OF ELASTIC RINGS MOUNTED IN ROTOR BEARINGS OF GAS-TURBINE ENGINES

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Annotation

The problem of determination of an elastic ring stiffness mounted in gas-turbine engines bearings is considered. Analytical solutions give flexibility of an elastic ring as a curved beam with pinned-end supports. At once practice shows occurrence of abrasion on ring ledges during exploitation. It means the response of a ring in rotor precession conditions is more complicated comparing to the beam bending only. The FEM model of an elastic ring placed between two stiff rings is prepared. Results of analysis show the large difference between flexibilities obtained by analytical solution and FEM statement when rings have contact interactions. The flexibility of elastic rings also strongly depends on assembling conditions.

Keywords

Gas-turbine engines, rotor bearings, elastic rings, flexibility coefficient

Introduction

One of the most effective methods to decrease the general vibrations level and dynamic stress in the engine units is elastic damper supports. [1]. Design of such supports varies, but irrespective of this they perform two main functions:

1. They lower stiffness of the rotor supports which leads to change in the characteristics of the elastic dynamic engine system. Meanwhile, the systems natural frequencies decrease, resonances at operating modes are removed.

2. They absorb the energy of oscillations of the engine dynamic system turning it into heat. This prevents the great oscillations amplitudes, dynamic loads and stress in all engine details from their development.

Elastic bushings of various design ("the squirrel cages") and thin-walled elastic rings are used as elastic elements. Elastic bushings have sufficiently stable stiffness characteristics and are applied to conduct the rotor critical speed detuning from the operating mode. They can be mounted together with hydrodynamic dampers. Elastic rings are used both for the purpose of detuning and for the organisation of the damping cave. In the latter case they are mounted together with the elastic bushings transferring axial and radial load from the rotor part.

Supports with elastic rings are usually used for the engines set at the airplanes exposed to great evolutionary overloading. Fig.1 shows a version of the elastic damper support of the low-pressure turbine rotor of one of the engines.

a) b)

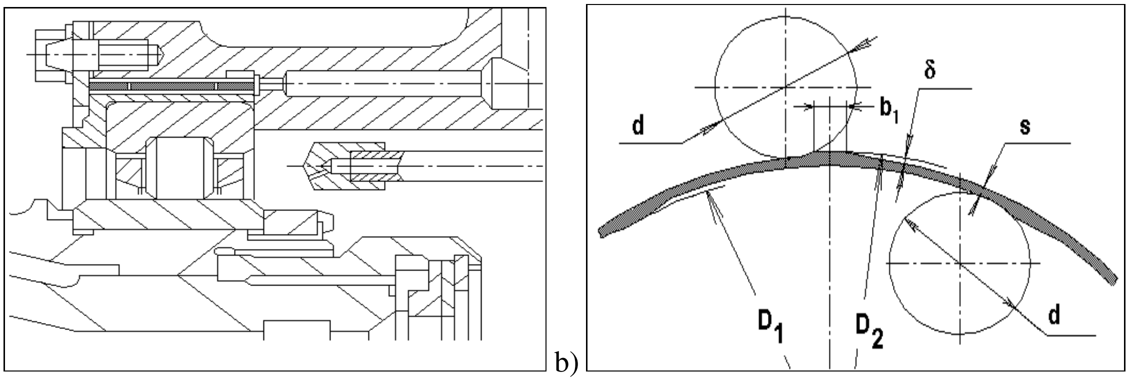


Fig.1. The damper support with the elastic ring
a) an example of construction design; b) base dimensions of the elastic ring

The main elastic element of a support is a thin-walled ring mounted into clearance between a housing and a bearing outer ring. The ring has projections equispaced on a circle alternate at outer and inner surfaces. Projections together with the other details of the support form hydraulic cavities where oil is constantly supplied. The number of projections and all geometric characteristics of the ring are regulated by the branch standard 1.10779-72. Projections height is usually 0.15...0.3 m, it determines allowable in terms of stability value of the ring buckling. The ring width is obtained by calculation.

Under radial loading elastic parts of the ring warp and elastic reaction to the bearing appears. Oil is pressed from one cavity into another one through butt slits and metering holes in the ring, therefore damping effect arises. The rings are mounted on inner package ring either at transition fit or at tightness on inner package ring.

Table 1 shows the examples of elastic rings mounted in different types of gas turbine engines.

Table 1

Gas turbine engine	D1, mm	D2, mm	Projections number	Projections length b1, mm	Ring working stroke, mm δ	Instrument diameter, mm d	Material
AI-24	96	93	6	6	0.12	30	Steel 60Si2
AI-25	137	134	10	6	0.15	30	Steel 60Si2
TV3-117	136	133.2	12	5	0.2	30	Steel 40CrNiMo4

The advantages of damper supports with elastic rings are: small dimensions and mass of the elastic part of the support; a linear stiffness characteristic; a rotor is centered relative to an engine axis; high enough damping qualities; presence of the branch standard on elastic rings which allows to select ring sizes easily and which also includes methodologies for calculation of the ring compliance and stiffness. The disadvantages are the necessity to provide with high accuracy of all dimensions and seats; absence of accurate mathematical models for calculation of damping capability; necessity of development in an engine.

The ring compliance is obtained analytically, by solving the task about a curvilinear beam supported on edges [2]. In addition to that, practice shows presence of sliding traces on different ring surfaces which means more complex ring loading at rotor precession movement. In work [3] it is also mentioned that the ring works with loss of touch between inner, outer projections and contact surfaces. Fig.2 presents surfaces of the elastic ring with touch traces (shown by arrows).

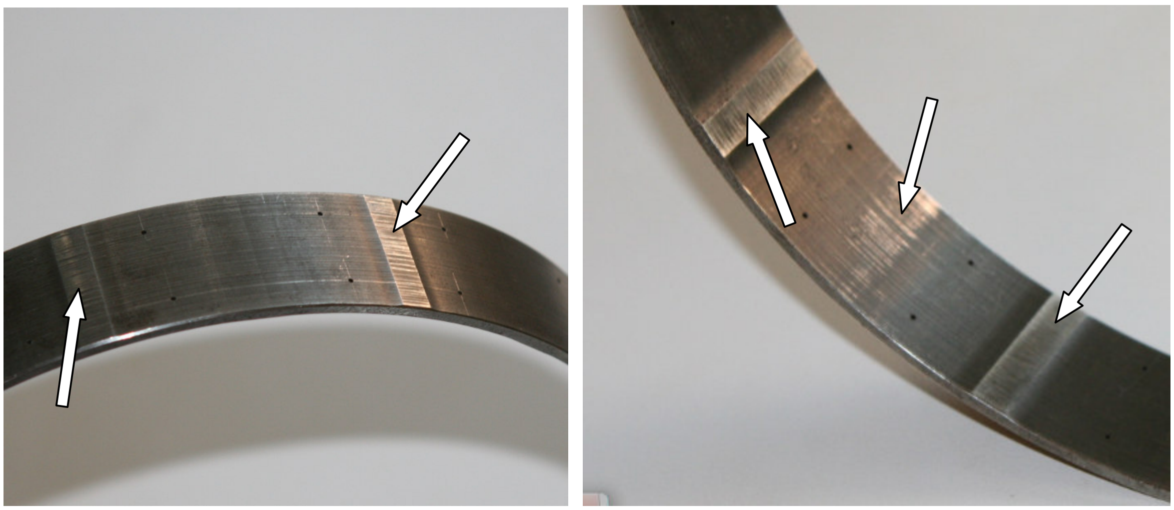


Fig.2. The elastic ring with sliding traces
a) on outer projections; b) on inner projections and inner ring surface between projections

The task of the present research is to determine elastic rings behaviour while working and to obtain their stiffness characteristics in different conditions of assembling and loading.

Initial data

Elastic qualities of the ring of elastic damper support of high pressure compressor of the prospective engine are investigated. Damping support package is simulated by three rings – inner, outer and elastic ones having the following characteristics, Table 2.

Table 2

Parameter	**Value**
Inner diameter D1, mm	$149{,}6^{(+0{,}04)}$
Outer ring diameter D2, mm	$155{,}6_{(-0{,}04)}$
Ring width b, mm	$23{,}8_{(-0{,}0033)}$
Ring projection length b1, mm	5±1
Ring thickness s, mm	1,25±0,03
Outer projection height δ_1, mm	0,47±0,03
Inner projection height δ_2, mm	1,23
Projections number *n*	16
Mill diameter for inner projections d_2, mm	16±2
Mill diameter for outer projections d_1, mm	6±1
Rings material	steel
Modulus of elasticity *E*, N/m^2	$1{,}93\cdot10^{11}$

Poisson ratio, μ	0,3
Material density ρ, kg/m^3	7750

Analytical model

The equation [2] calculates elastic rings compliance:

$$\Pi = \frac{(D_{cp} - 0{,}3b_1 n)^3}{0{,}129 bEn^4 S^3}\left[1 - (1 - \frac{S^3}{S^3_{выст}})(1{,}45A - 0{,}9A^2 + 0{,}2A^3)\right]$$

where $D_{cp} = \frac{D_1 + D_2}{2}$ – pitch diameter of elastic ring;

$S = \frac{D_1 - D_2}{2} - 2\delta$ – elastic ring thickness;

$S_{выст} = S + \delta$ – thickness of elastic ring projection;

$\delta = (\delta_1 + \delta_2)/2$ – working stroke of elastic ring;

$A = \frac{(b_1 + \sqrt{d\delta})n}{D_{cp}}$ – correction factor for the case of equal mills diameter – d, by the help of which the elastic rings projections are turned, fig.2.

$A = \frac{(b_1 + \sqrt{d_{cp}\delta})n}{D_{cp}}$ – correction factor for the case of different mill diameters (d_1, d_2); $d_{cp} = (d_1 + d_2)/2$ – mean diameter of two mills.

Rated compliance of elastic ring according to analytical formulas accounted for $1{,}62 \cdot 10^{-8}$ m/N. Meanwhile, dispersion of rated value in blueprint tolerance limits is $1{,}38 \ldots 1{,}95 \cdot 10^{-8}$ m/N.

Finite-element model of elastic ring

Simulation and investigation of an elastic ring behaviour was carried out in the ANSYS program. Fig.3 shows the finite-element model of the elastic ring.

a) b)

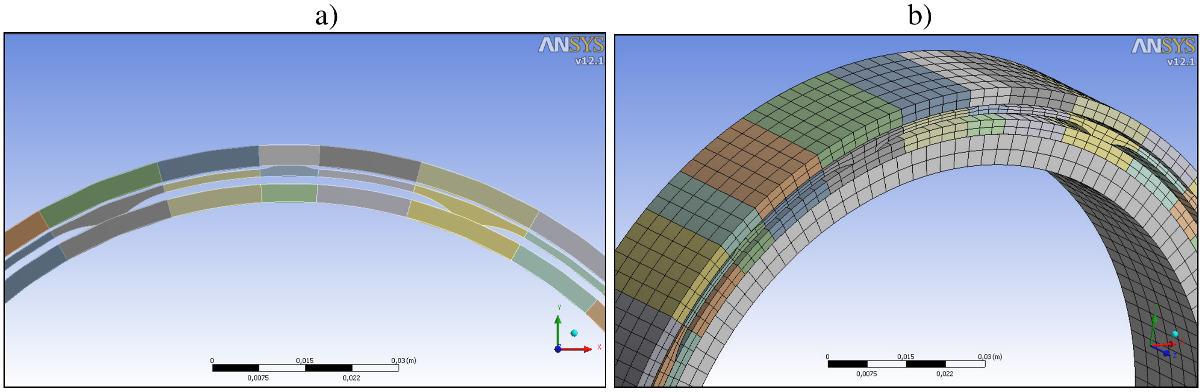


Fig. 3 Finite-element method model of the elastic ring

In the general case the model is created for solution of the contact task about the elastic ring behaviour in case of precession motion of the rotor loading the ring with unbalanced force. So possibility of sliding and loss of contact between projections of the inner elastic ring of the vibropackage and surfaces of the outer rings was considered. For the convenience of results presentation the ring projections are numbered, Fig.4.

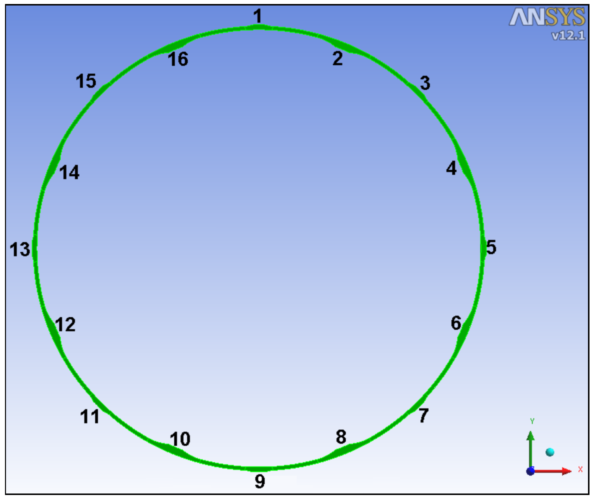


Fig.4. Numbering of projections in the elastic ring

At the finite-element model creation eight-unit HEX elements were used. Before the finite-element model building initial volumes were divided into sectors the number of which is equal to the number of contact pairs (16 sectors for a ring, Fig.3). So the contact surfaces assignment takes place between the sectors surfaces, but not the whole rings. The accuracy of geometry building in UNIGRAPHICS accounted for $1 \cdot 10^{-17}$ мm that, however, is not enough for ANSYS – it is desirable to have no divergence between rings geometry.

To obtain proper results while solving contact tasks in ANSYS the following parameters are varied:

- normal stiffness factor – 1,0 on default (at this value the model works only under the condition of ideal geometry), under restrictions the parameter should be reduced to 0.1 to receive convergence;

- type of contact – at secure contact continuity the “frictional” contact may be replaced by the “no separation” one – in this case a friction coefficient will be also used, but loss of contact isn't implied – only sliding. This also improves calculation convergence. Under absence of clearance contact may be simulated using the parameter “symmetric”, when «target» and «contact» surfaces are equivalent. If there is geometric clearance (in our case in contact pairs in spans between projections) it is necessary to simulate contact with the “asymmetric” parameter and to use the “adjust to tough” calculation type. Such contact works only after appearance of deformations leading to clearance adjustment.

- pinball region – the parameter works between "program controlled" and "automatic detection value". In different calculations the result convergence may change depending on this parameter value.

The rings interaction is simulated by 25 contact pairs, Fig.5. Eight contacts simulate interaction between outer projections of the elastic ring and inner surface of the outer ring (white arrows), the other eight contacts simulate interaction between inner projections of the elastic ring with an outer surface of the inner ring (dark arrows) and one contact pair simulates interaction between inner ring with the bearing ring. Eight contact pairs are also added to simulate contact between projections on inner and outer sides of elastic rings.

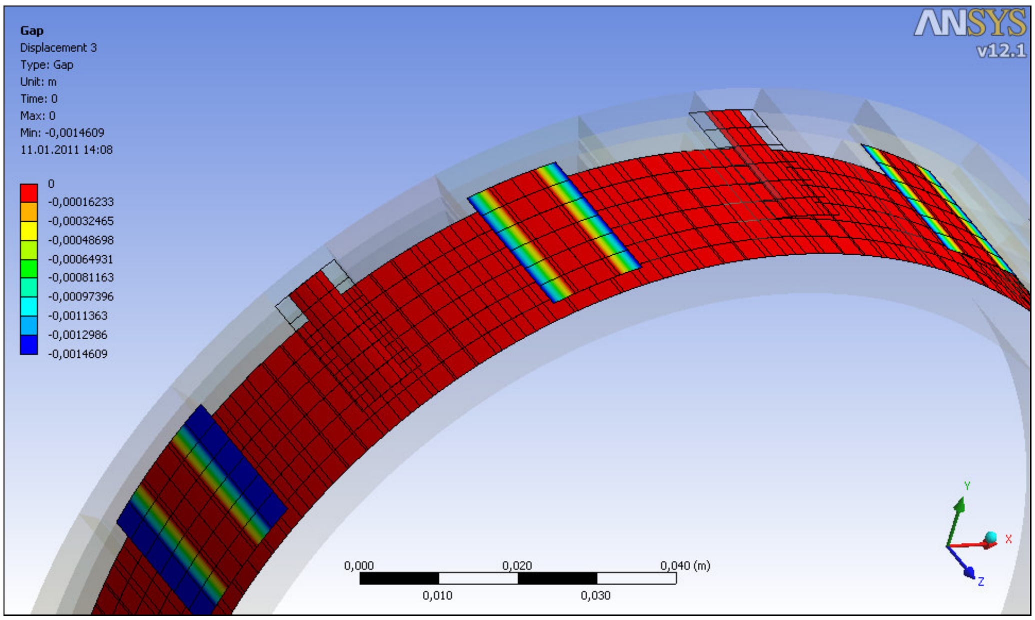


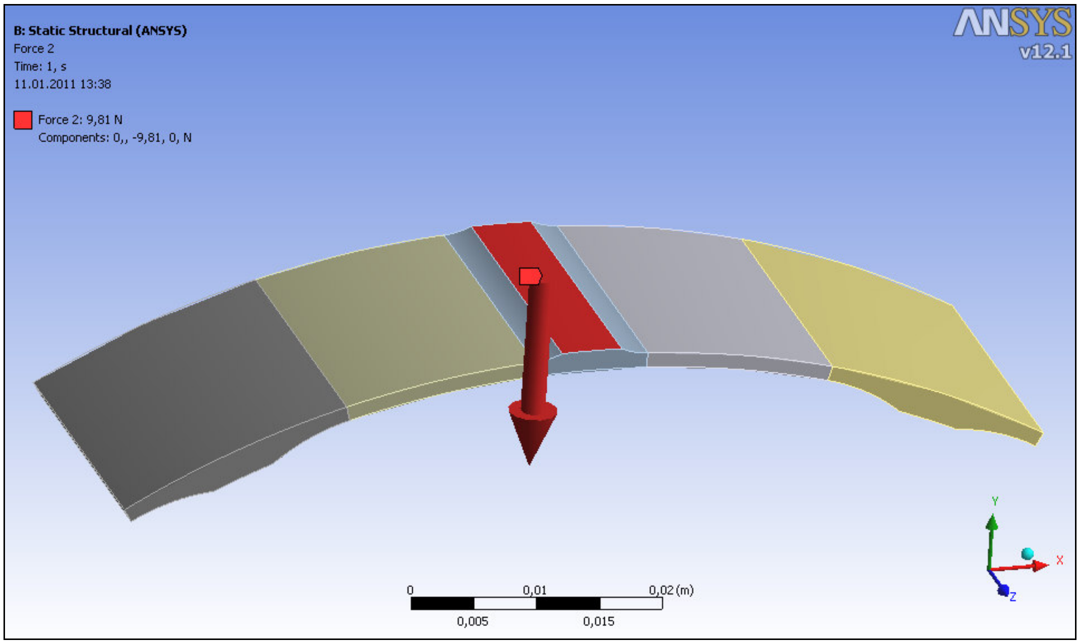
Fig. 5 Contact pairs of the vibropackage model

Contacts are simulated by the Frictional type with friction coefficient 0,15 (steel on steel). For contacts on elastic rings projections zero tightness was chosen, because all contacts have clearance fit according to the drawing. For the contact between the bearing outer ring with the vibropackage inner ring the tightness of 0,01 mm is simulated.

Calculation of the ring sector compliance

Under the assumption of absence of the inner ring sliding calculation of compliance of an elastic ring sector between two projections was carried out, Fig.6 a). This task corresponds to the problem statement with analytical solution [2] with the assumption that in case of the rotor precession motion only one sector of the elastic ring works. In compliance with this statement the ring sector is loaded with the radial unit force. Outer and inner vibropackage rings are fixed rigidly on outer surfaces and they do not deform during ring loading. Sliding and loss of contact between projections and outer rings are not simulated. Fig.6 b) shows the results of displacements under load of 9.81 H.

а)



б)

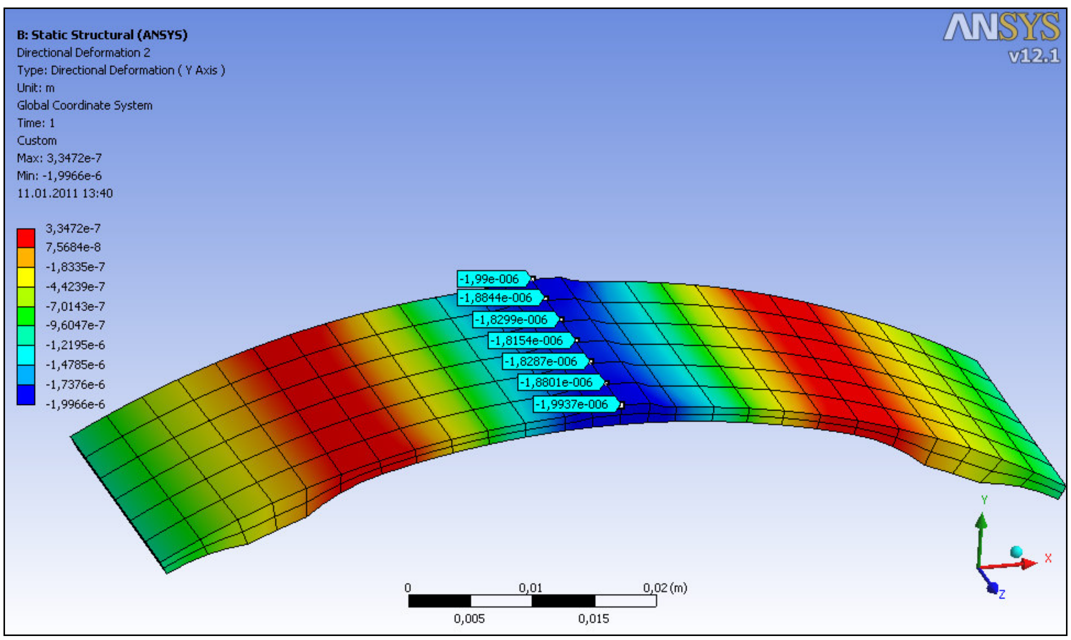


Fig.6 Model and results of calculation of compliance of the ring sector between two projections under absence of sliding
a) loading of elastic ring sector by force of 10 H; b) displacement of elastic ring, m

The ring displacement accounted for 0, 0018…0,002 mm that corresponds to compliance of ~$1{,}93 \cdot 10^{-8}$ m/N. Divergence with analytical calculation accounts for ~15%. Meanwhile, the obtained value is kept within dispersion conditioned by blueprint tolerances.

Analysis of the ring behaviour depending on the rotor weight

During calculation the behaviour of the elastic ring placed between two undeformed rings depending on the rotor weight was simulated. The rotor weight accounted for 1099 H. Elastic and inner rings are fixed at the Y-axis (from rotation). The elastic ring between outer and inner ring is mounted between the outer and inner rings at transition fit. In calculation clearance is accepted as equal to 0.

Calculations showed that ring contact pairs are in different conditions, Fig. 6. Seven upper projections formed clearance with the inner surface of the outer ring. Nine lower projections are in condition of sliding on the inner ring, and one projection is in condition of stationary contact.

For convenience of the further analysis the following notations of the contact pairs statement were introduced: **near** – disclosed contact, **sticking** – stationary contact, **sliding** – sliding without contact opening.

Fig. 7 presents condition of the contact pairs depending on the rotor weight. Meanwhile, the maximal displacements in vibropackage account for 0, 0273 mm that corresponds to compliance of $2.44 \cdot 10^{-8}$ m/N.

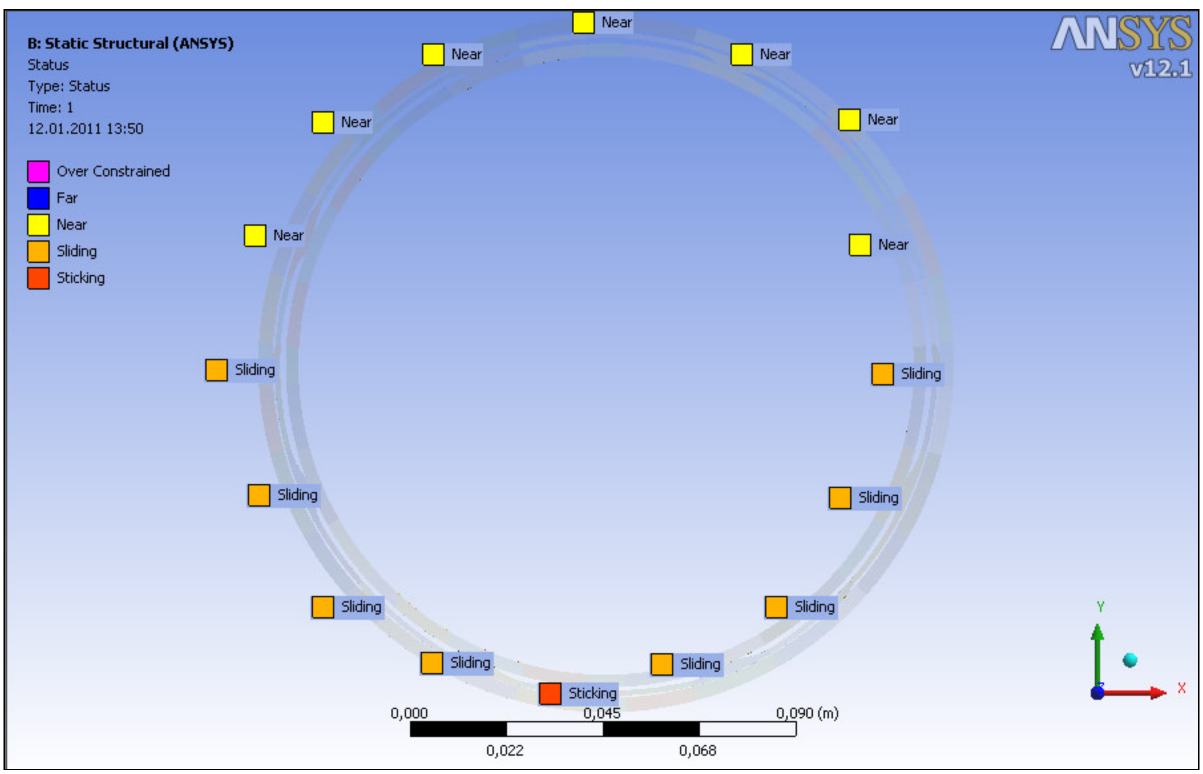
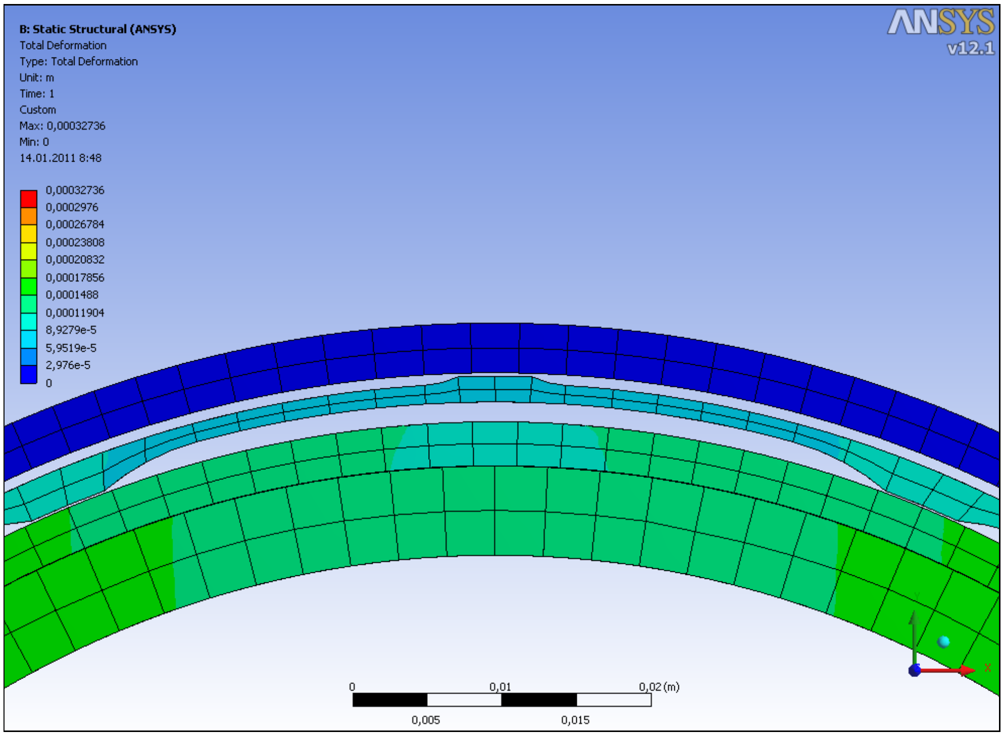


Fig.7 Condition of the contact pairs of the elastic ring depending on load simulating the rotor weight

Determination of the elastic ring stiffness

It follows from the antecedent results that the ring stiffness is a non-linear characteristic. This conclusion is important because at evolutionary loads the significant gyroscopic moment causing big displacements acts on the rotor supports. Besides, the appearance of thermal unbalance is possible. Meanwhile, its value may be high enough. Load applied at Y-axis changes in the range from 0 to 12000 H. Fig.8 shows the picture of vibropackage displacements depending on the static force. Fig.9 presents compliance dependence on displacements.

a)



в)

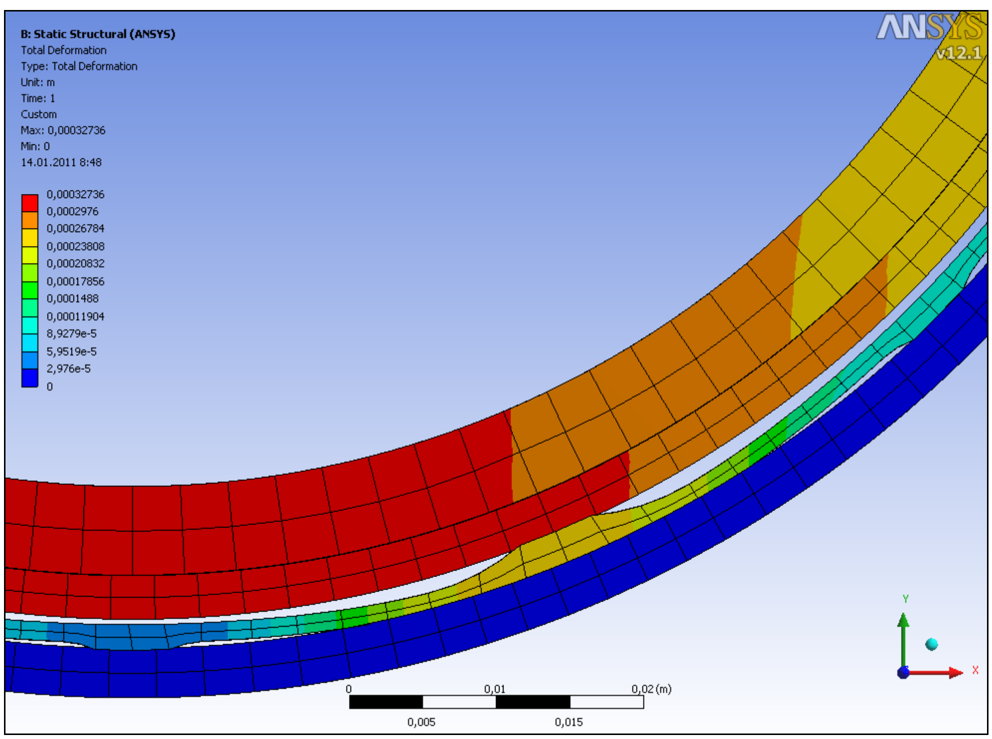


Fig.8. Picture of the vibropackage displacements depending on the rotor weight
a) the sector between the projections 16 and 2, contacts on projections are absent; b) the sector between projections 7 and 9, there are contacts on the projections and the ring middle part between the outer projections

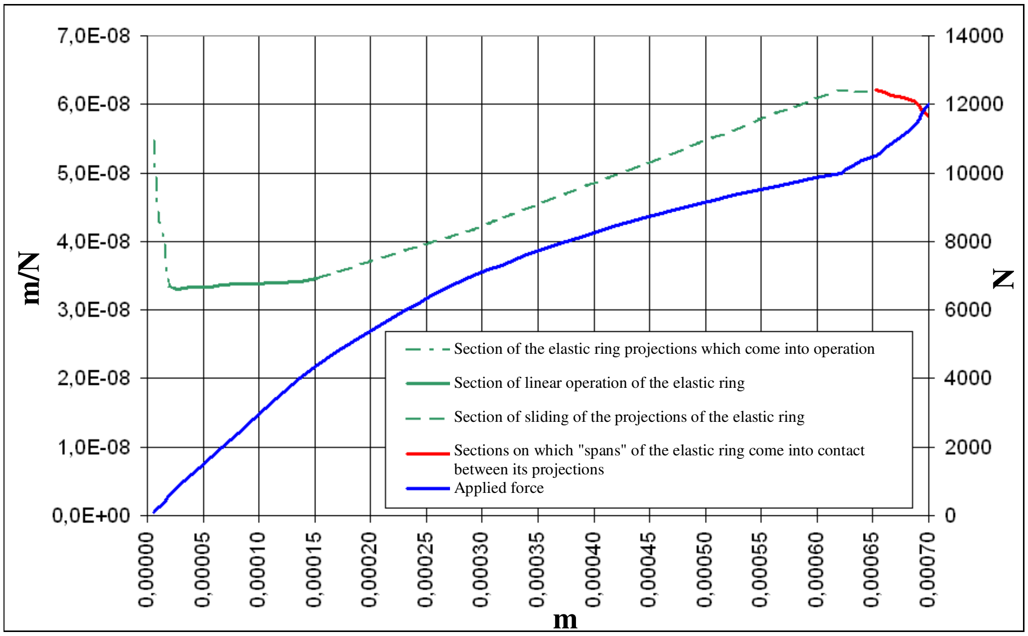


Fig.9 The elastic ring stiffness

As the plot shows, the characteristic has four strongly marked sections:

I. Section of small deformations (displacements up to 0,02 mm) – on this section clearances are taken up, and contact pairs on the elastic ring projections come into operation.

II. Section of linear characteristic (displacements 0,02-0,14 mm) – displacements in the elastic ring are determined by its sector stiffness between two projections.

III. Section of projections “sliding” (displacements 0,14-0,62 mm) – when load increases (in our case >4000 H) sliding of side projections of the elastic ring happens, because applied load exceeds friction force on projections.

IV. Section on which at significant force (>10000) «spans” of the elastic ring between its projections come into contact (displacements 0,62 mm and more). The elastic ring squeezing takes place and new contact pairs appear.

The important conclusion is that elastic ring compliance depending on static load may almost double.

Clearances influence on ring compliance

The previous calculations were carried out in the absence of clearance in the vibropackage. According to the drawing clearances on the inner projections may account for up to 0,05 mm, on the outer ones – 0,043 mm. Analysis of the ring behaviour at its loading by the rotor weight force (1099N) showed that condition of contact pairs has not virtually changed. However, maximal displacements increased and accounted for 0,327 m that corresponds to compliance of $2{,}98 \cdot 10^{-8}$ м/Н. It means that if the ring mounted with clearances the ring compliance increases.

Compliance of ring mounted with tightness

Calculation was carried out with tightness of 0,05 mm on all projections of the elastic ring. Static load applied on the Y-axis changed in range from 0 to 17000N. Fig.10 shows the ring stiffness dependence on the displacements.

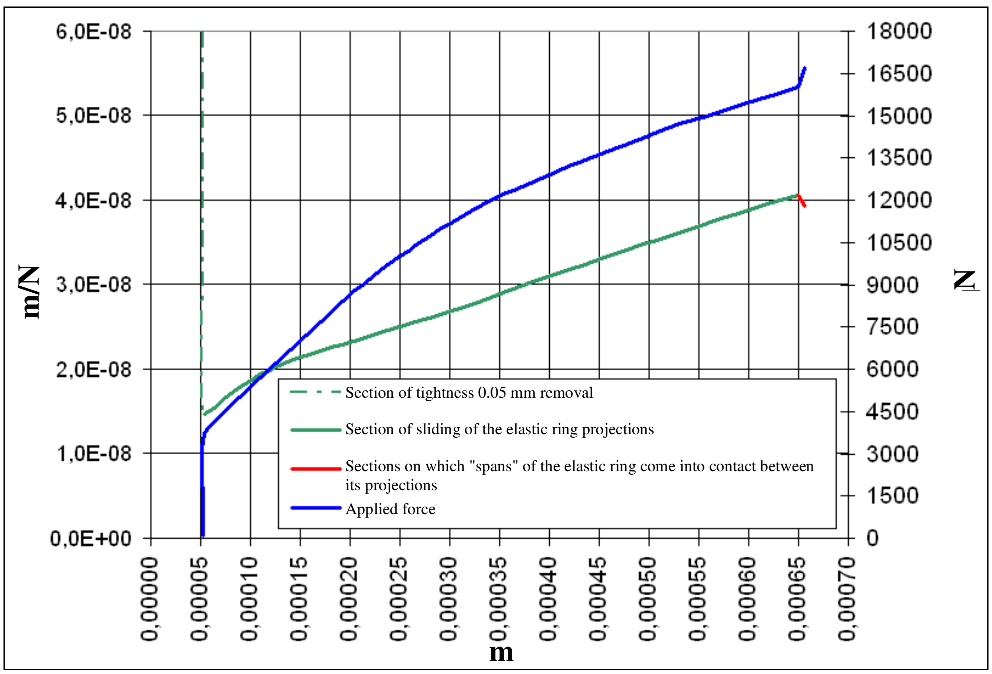


Fig.10 Stiffness of the elastic ring mounted with tightness of 0,05 mm

Table 3 presents condition of the contact pairs depending on the static load.

Table 3

Force, N	Displacement, mm	Projection number															
		1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16
2000	0,00	+	+	+	+	C	+	+	+	+	+	+	C	C	+	C	+
4000	0,01	C	+	C	C	C	C	+	+	+	+	+	C	C	C	C	+
6000	0,07	-	C	C	C	C	C	+	+	+	+	C	C	C	C	C	C
8000	0,13	-	C	C	C	C	C	C	C	+	C	C	C	C	C	C	C
10000	0,20	-	-	C	C	C	C	C	C	+	C	C	C	C	C	C	-
12000	0,29	-	-	C	C	C	C	C	+	+	+	C	C	C	C	C	-
14000	0,43	-	-	C	+	C	C	+	+	+	+	+	+	C	+	C	-
16000	0,60	-	-	C	+	C	C	C	+	+	+	C	+	+	+	+	-
17000	0,62	-	-	-	+	C	+	C	+	+	+	+	+	+	+	+	-

The following notations are introduced: **- -** clearance; C **-** sliding; **+ -** full contact.

By analogy with the previous analysis the characteristic of the elastic ring mounted with tightness has three marked sections:

I. Section of small deformations (displacements up to 0,05 mm) – at this section removal of fitting tightness on the ring projections takes place – in our case it is 0,05 mm. Force necessary for fitting tightness removal accounted for ~3700 N.

II. Section of projections "sliding" (displacements 0,05-0,63 mm) – at loading increase sliding of side projections of the elastic ring takes place, because applied load exceeds friction force on the projections. Meanwhile, it should be noticed that initial curve inclination is bigger because it corresponds to the moment of tightness removal and, as consequence, to abrupt displacements increase.

III. Section on which at significant force (>16000 N) "spans" of the elastic ring come into contact between its projections (displacements 0,63 mm and more). The elastic ring "squeezing" takes place and new contact pairs appear.

Compliance of the elastic ring mounted with tightness may increase in more than 2.5 times depending on reaction operating in the support.

Analysing stiffness of rings mounted with clearance and tightness it may be noticed that compliance of a ring mounted with tightness approximately twice lower than compliance of a ring mounted with clearance in all loads range.

Analysis of characteristics taking into account the rotor weight and unbalances

We will examine the ring characteristics under weight and unbalanced force appearing from residual unbalance of 25 gcm (according to the engineering specification of the drawing) at the rotor running speed of 10000 rpm. Table 4 shows the results of calculation at different phase of unbalanced force in relation to the gravity force vector.

Table 4

Phase angle	Maximal displacements, mm	Contact number															
		1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16
0°	0,2672	-	-	-	-	C	C	C	C	+	+	C	C	C	-	-	-
45°	0,2692	-	-	-	-	+	+	+	+	+	C	C	C	C	-	-	-
90°	0,2737	-	-	-	-	C	+	+	+	+	C	C	C	C	-	-	-
135°	0,2774	-	-	-	-	C	C	C	+	+	+	+	+	C	-	-	-
180°	0,2788	-	-	-	-	C	C	C	+	+	+	+	+	+	-	-	-
225°	0,2783	-	-	-	-	C	C	C	+	+	+	+	+	+	-	-	-
247,5°	0,2774	-	-	-	-	C	C	C	C	+	+	+	+	+	-	-	-
270°	0,2755	-	-	-	-	+	+	+	C	+	C	+	+	+	-	-	-
315°	0,2731	-	-	-	-	-	+	+	+	+	+	+	+	+	-	-	-

As the table shows, upper projections of the elastic ring (1, 2, 3, 4, 14, 15 and 16) are not in contact at any direction of unbalanced force; projection number 9 placed at low point is always clamped. This is conditioned by direction of the rotor weight force, and contacts at other projections change their condition from open to clamped depending on direction of unbalanced force vector. Fig. 11 presents change in overall force and maximal displacements along the Y-axis depending on a phase angle of unbalanced force.

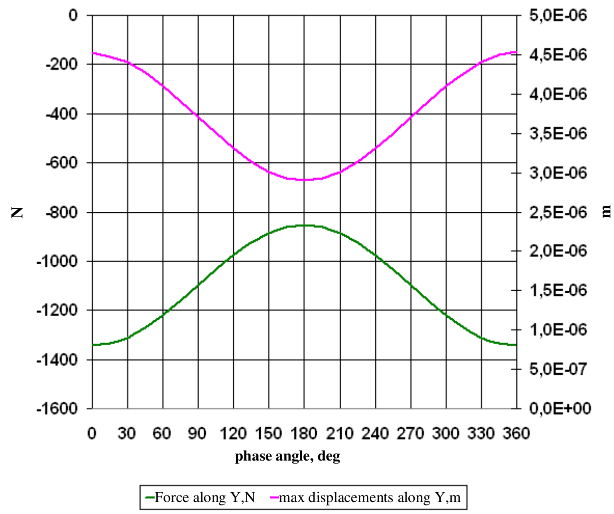


Fig. 11 Change in overall force and maximal displacements along the Y-axis

It may be noticed that in this case unbalanced force from unbalance of 25 gcm does not virtually change the mean value of the ring stiffness. The crucial factor is the rotor weight force falling on the support. However, actual unbalanced forces may be considerably greater. For example:

- after operation time of several hundreds of hours the rotor residual unbalance may increase in two or three times;
- as a result of torsional and bending deformations of the engine cases during operation unbalances may exceed 1000 gcm.
- thermal unbalances may also account for from 1000 gcm and more.

At such unbalances inertial forces exceed weight force, and rotor in clearances of bearings and elastic rings goes into whirling motion. This effect is already achieved at unbalance of about 100 gcm. The ring compliance will be determined mainly by unbalance load with minimal influence of weight force. In this case for estimation of ring compliance the characteristics obtained from static loads such as those shown in Fig.9 or 10 may be used.

The methodology of the elastic ring characteristics determination

Finally the obtained results about ring compliance in different combinations of static load, coupling sizes and unbalance are presented in Table 5.

Table 5

Ring compliance, m/N						
Calculation using analytical formulas [2]	Finite-element method calculations					
	Compliance of ring sector between two projections	Ring compliance (zero clearance)	Ring compliance (with clearance)	Ring compliance in range of static load from 1000 N to 12000 N (zero clearance)	Ring compliance in range of static load from 1000 to 17000 N (set with tightness)	Ring compliance from weight force and unbalance of 25 gcm (zero clearance, running speed)
$(1.36…1.93)\cdot10^{-8}$	$1.93\cdot10^{-8}$	$2.48\cdot10^{-8}$	$2.97\cdot10^{-8}$	$(3.3…6.2)\cdot10^{-8}$	$(1.5…4.1)\cdot10^{-8}$	$(3.35…3.4)\cdot10^{-8}$

On the ground of the conducted calculations and analysis of the obtained results the methodology of preparation of data about elastic ring compliance may be proposed for their further use in nonlinear and unsteady analysis of a whole rotor system.

The following tasks are the stages of the supposed methodology:

1. Calculation of possible loads acting on the reference node – a rotor weight force, evolutionary loads from an aircraft, unbalanced forces (from residual unbalances, working unbalances, thermal unbalances) and range from possible change.
2. Determination of a non-linear ring characteristic (compliance-displacement) taking into account weight force, working and thermal unbalances, housings deflections, clearances depending on an engine mode of operation. Unbalanced force acts on a support with tan elastic ring.
3. Estimation of dynamic characteristics of a rotor system taking into account non-linear characteristics of reference nodes with elastic rings and estimation of impact of the above mentioned factors on dynamic characteristics of a rotor system.

Conclusions

1. The mathematical model in the ANSYS program was developed, elastic-contact task about the ring stiffness determination was solved, the methodology of the model preparation and the ring elastic qualities analysis in elastic-damper supports was developed.
2. Stiffness characteristics of the elastic rings mounted in elastic-damper supports are nonlinear and depend on elastic rings seats, dimensional tolerances and value of forces acting in supports and transmitted through elastic rings. Compliance value may change in 2-3 times comparing to the obtained one according to analytical formulas. Slidings of the ring projection relative to housings are possible during operation.
3. At a rotor system designing with elastic rings in supports non-linear stiffness characteristics should be taken into account, so the analysis of a rotor system should be conducted in non-linear and unstable setting in consideration of all factors, the above mentioned and changing according regimes.

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