Clearances and Rubbings

DYNAMICS R4

Научно-технический центр “Альфа-Транзит”

TRANZIT

Doosan Heavy Industry&Construction

http://www.doosanheavy.com/en/products/energy/service.do

Steam Turbines

- power unit turbine accident 225 Mw station "Gallatin" (United States), June 19, 1974;

- accident at a thermal power plant in the United States (Tennessee, 1974);
-
- accident in Russia - Kashirskaya Termal Power Plant of 300 Mw (October 2002);
-
- accident in Ukraine on one of the power units Pridneprovskaya TPP (2007);
-
- accident in Russia OAO ZSMK (2010).

Steam Turbines

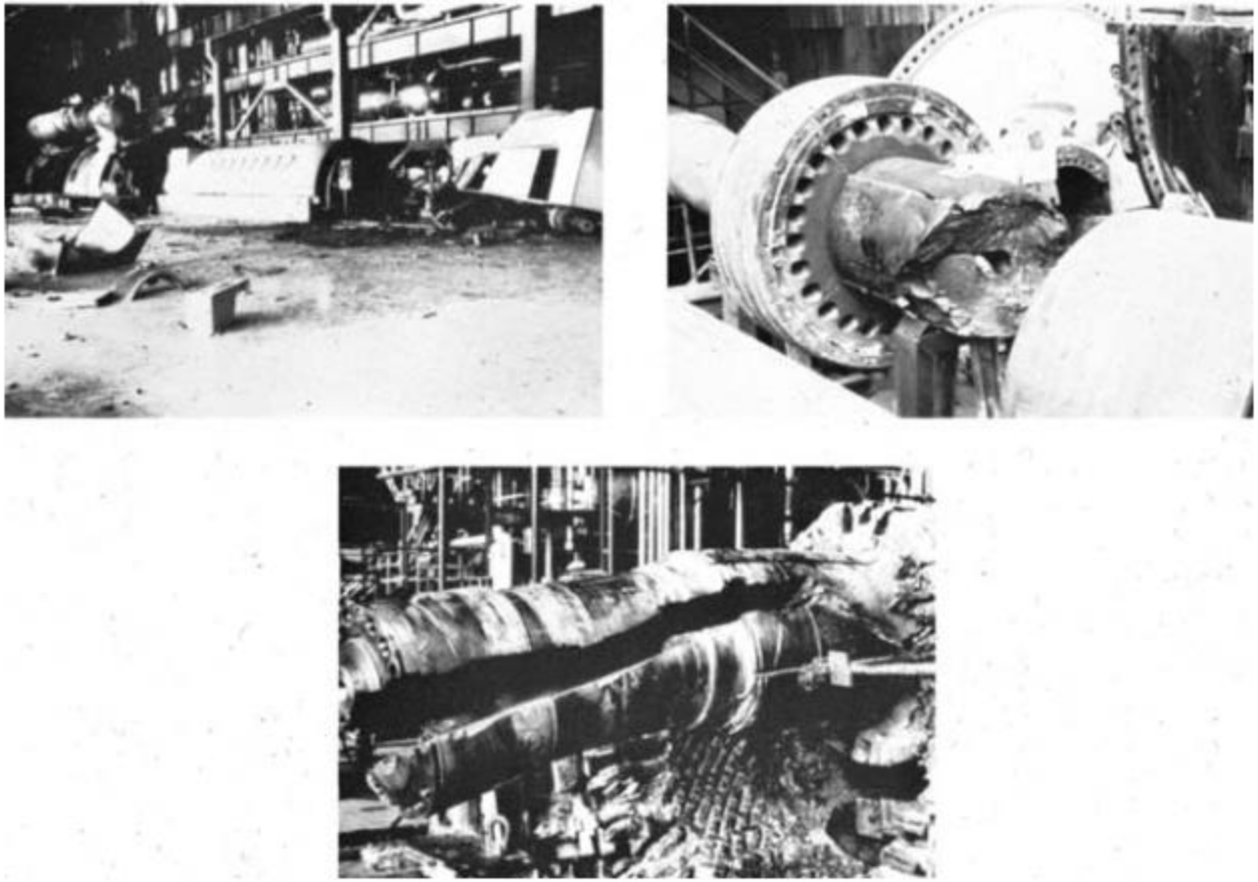
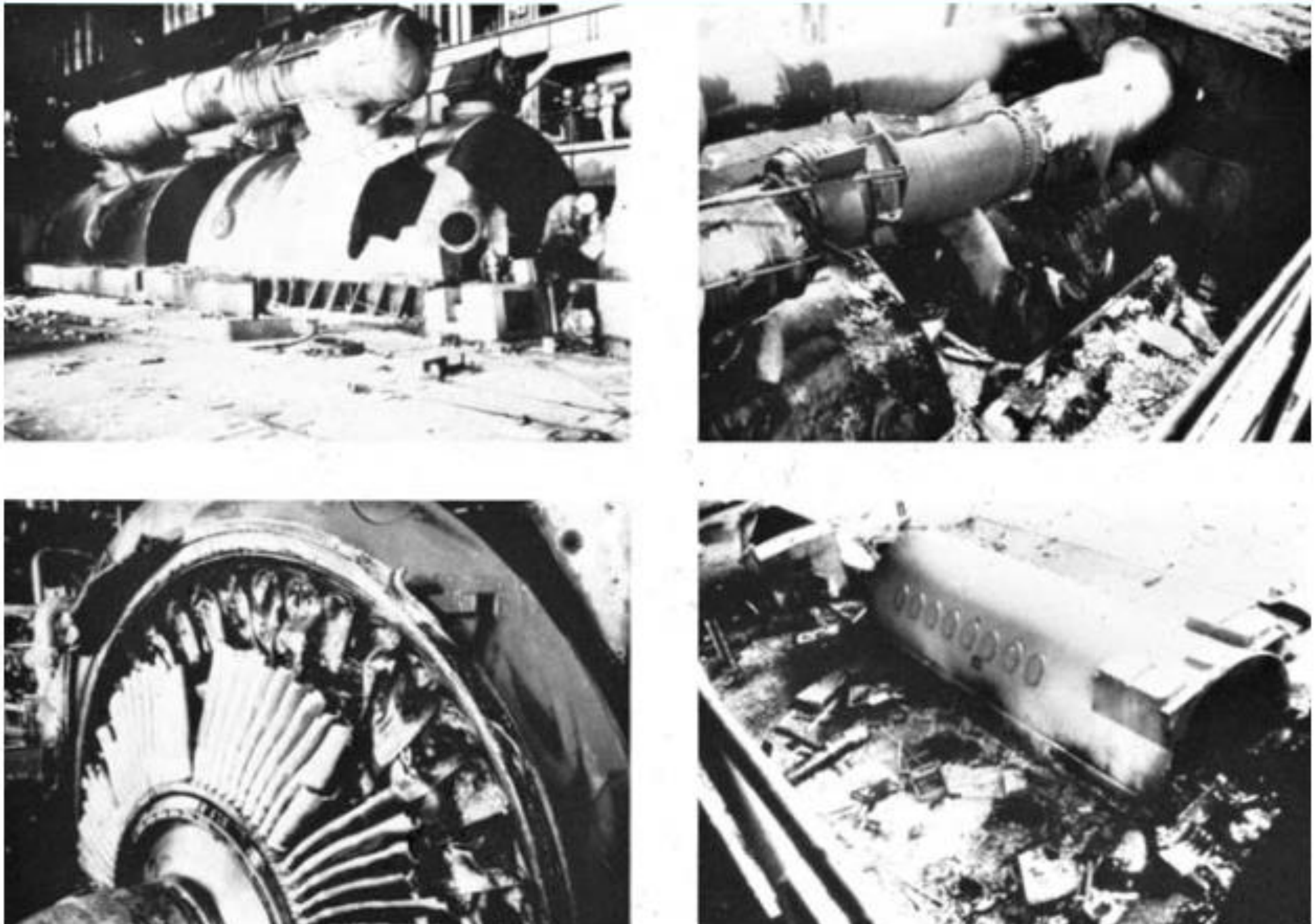
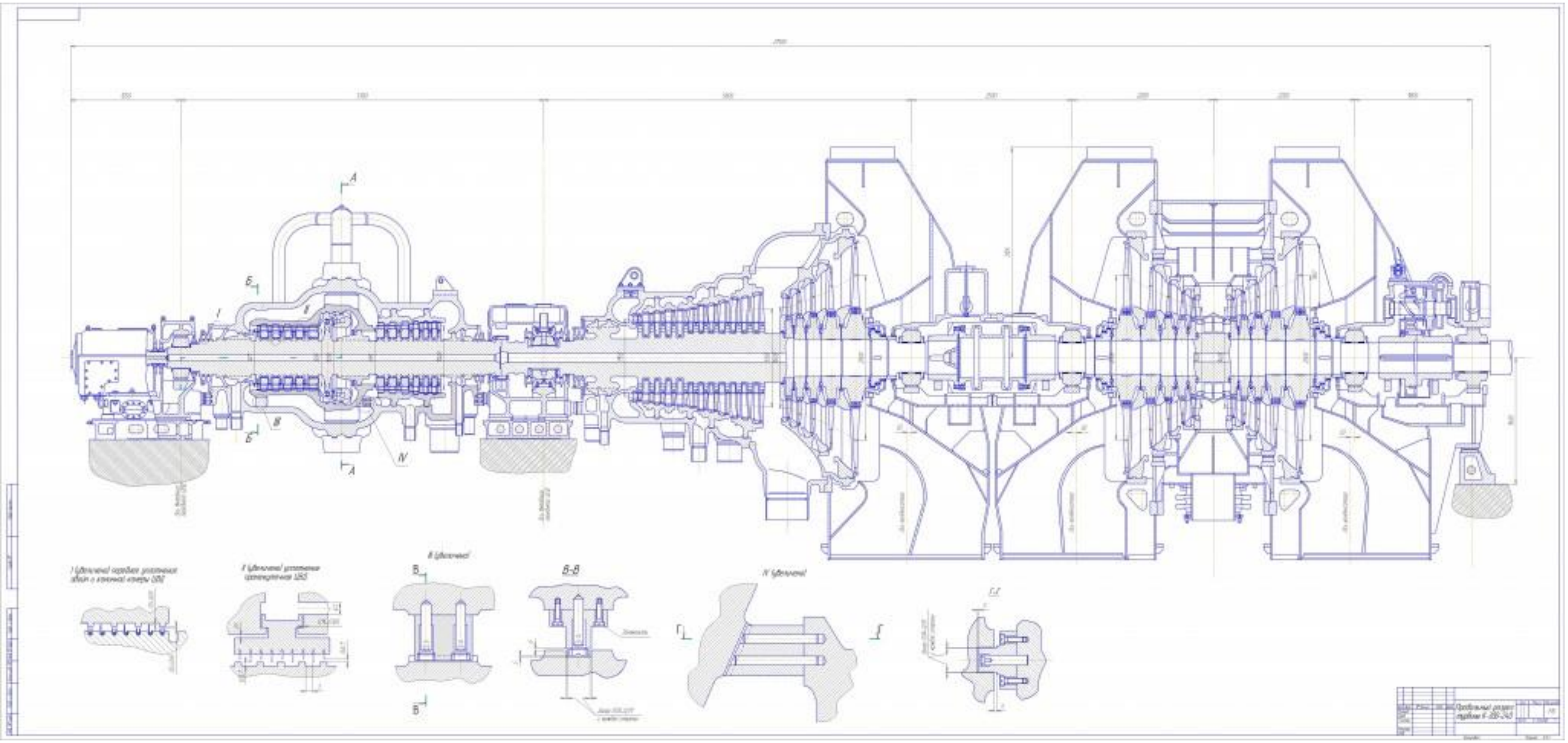


FIGURE 14 Photographs from two catastrophic failures in the 1970s of large 600 MW steam turbine-generator sets. Using nonlinear rotor dynamic response computations, failures could be potentially traced to the large unbalance from loss of one or more large low-pressure turbine blades at running speed, coupled with the behavior of fixed-arc journal bearings during large unbalance.

Steam Turbines disasters



Case Study of Steam Turbine К-300-240-1

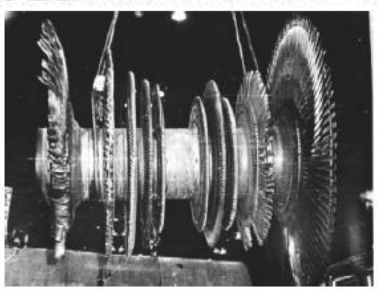


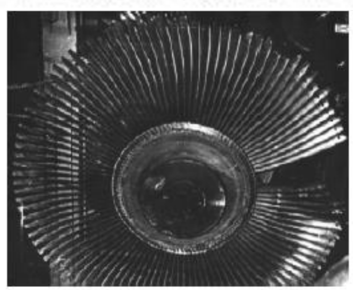
Destruction of steam turbine K- 300 MW

The accident happened on the complete destruction of the turbine and generator

Accident of Kashirskaya TPP – 2002

At the time of the accident the steam turbine had generation 228.5 thous. н. The appointed individual resource 250 thousand. h.
Number of accumulated launches-190.
Turbine generator was put into operation in 1968, and during this period was repaired 14 times.
Steam parameters, vibration and other indicators were within normal limits.


а)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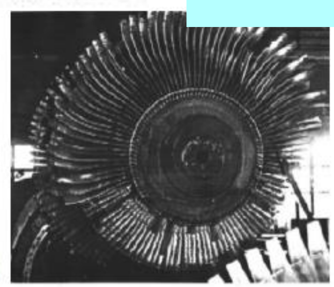
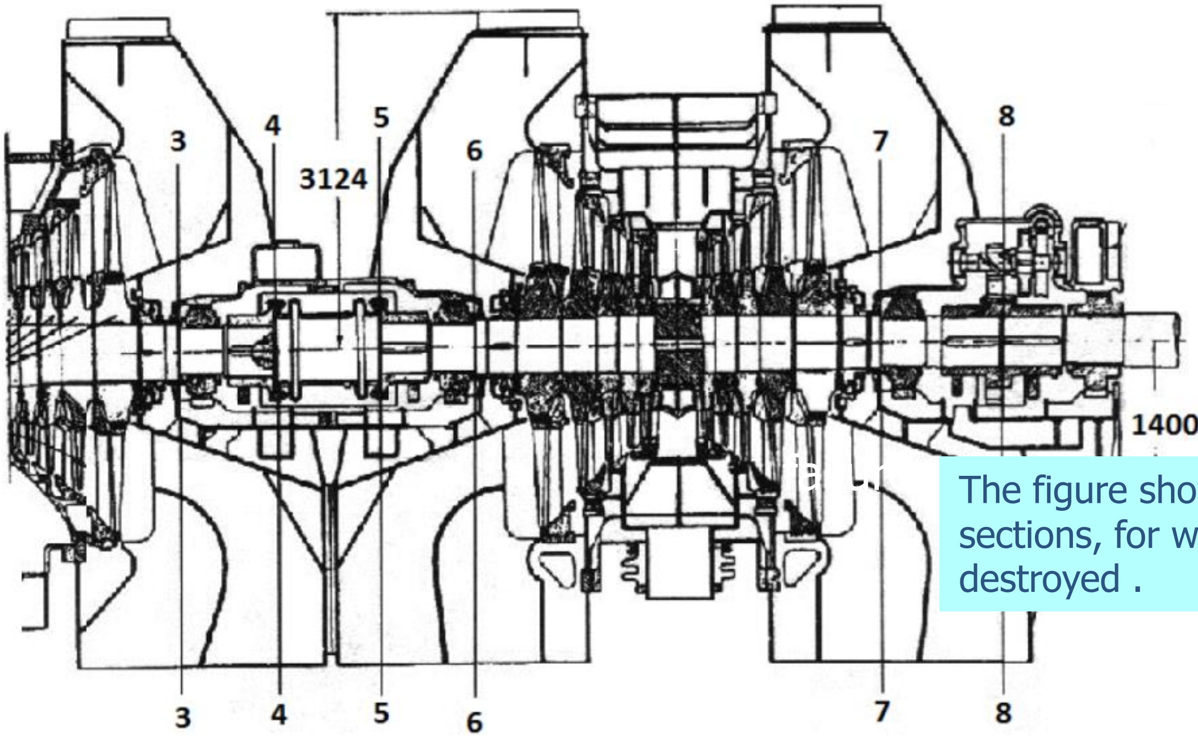

в)


г)

Destruction of steam turbine K- 300 MW



The figure shows the turbine and marked sections, for which the shafting was destroyed .

All the fracture turbine shafting have a power character twisting with or without proportion flexural component.

No indication of fatigue damage has been detected. This reflects an important dynamic twisting caused by rotor braking speed with large negative acceleration.

Bending component is probably the consequence of large shaft precession with big deflection

Destruction of steam turbine and generator

Destruction of generator shaft

Heavy damages of blades and disks

The reasons of heavy destructions enumerating

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•Breakage blades or group of blades, leading to large imbalances and contact the shaft with the stator

• Small gaps between the crowns of impellers and stators, resulting in contacts of shaft and stator

•Bending and torsion vibrations, equal to the working mode of the turbine, imbalances in the work process

•Contact the turbo generator rotor windings on the stator

•Short circuit in turbo generator

•Violation of stiffness characteristics of the shafts train leading to a shift of resonance modes

•Incorrect design

•Drop the rotor supported AMB on safety bearings

•Shafting cracks due to low cycle fatigue, resulting in contacts of shaft and stator, etc

These terms and conditions do not always lead to such serious consequences, even for the same type of turbines.

This means that the design of turbines should take account of the possible consequences of any irregularities in the work of turbine unit

Destruction of steam turbine K- 300 MW

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But despite their reason difference, destruction of most of them has the same process:

• Contact shafts and housings of gas turbines shafts train

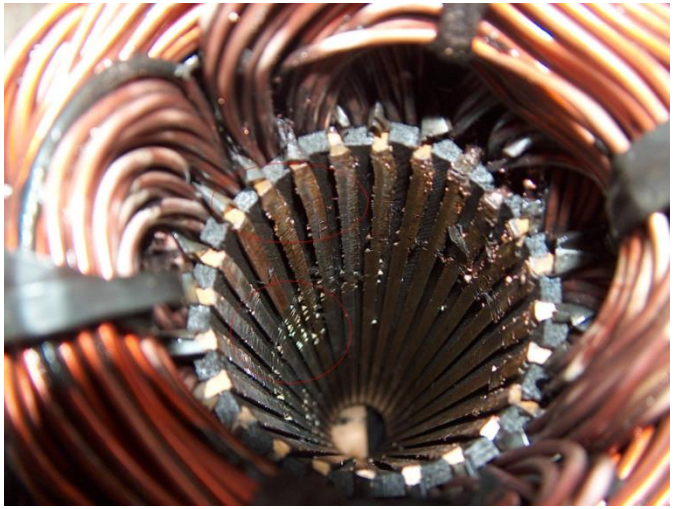
• Sharp braking individual sections of shafts train up full stop with liberation of vast quantities of energy at the time of braking

• Twisting of shafts train sections relative to each other and finally, the complete destruction of the turbines and perhaps the whole energy complex - turbine halls and buildings

Destruction of rotating machines

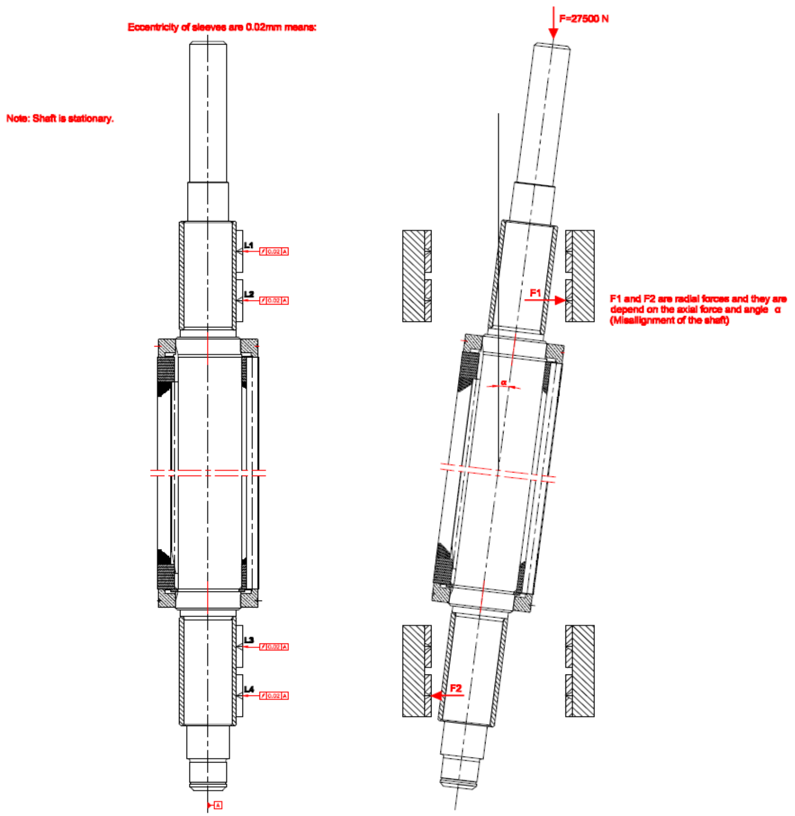
Let us explain the damage and destruction mechanism of rotating machinery on submersible motor example

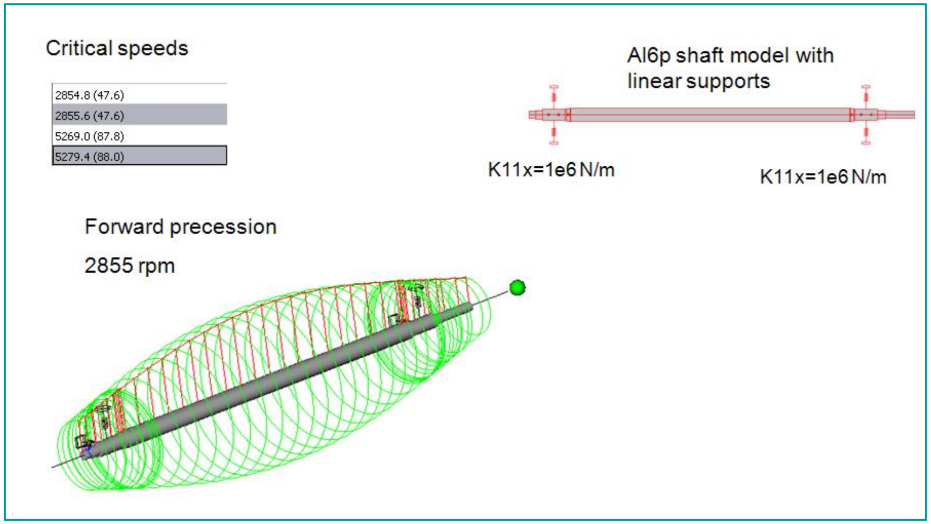
Destruction of Submercible Motor

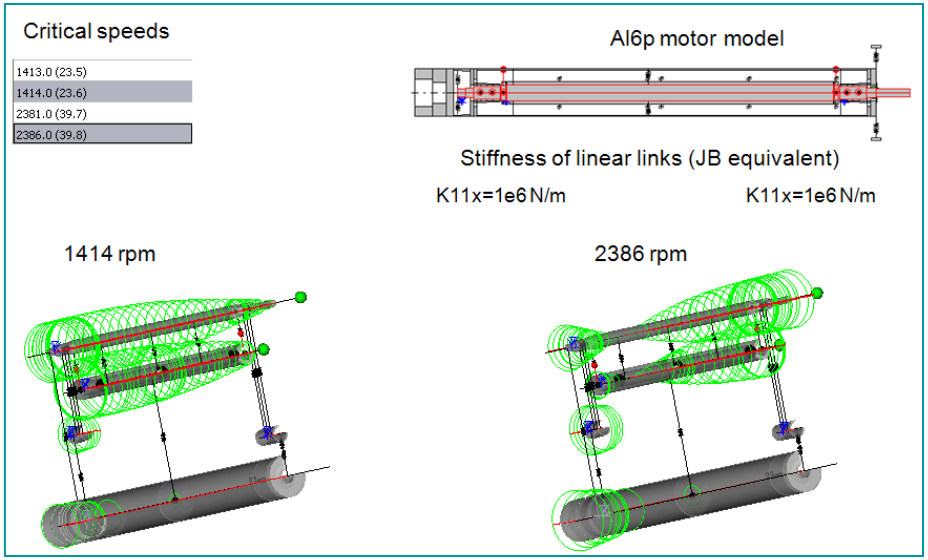


Destruction of motor journal bearings shaft

Submersible Motor Wrong Design







Destruction of Submercible Motor

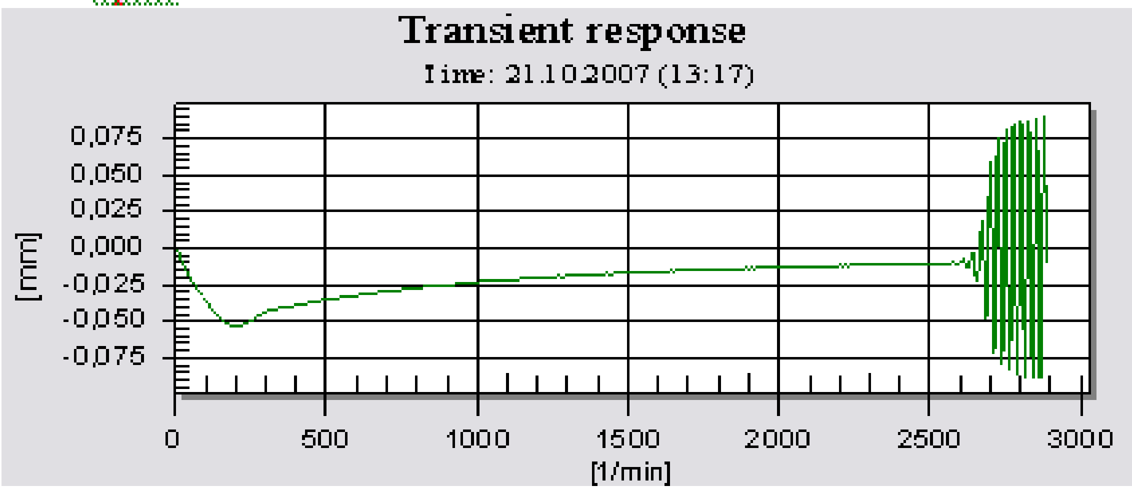
Additional statical moment on the shaft due axial force 10 Nm

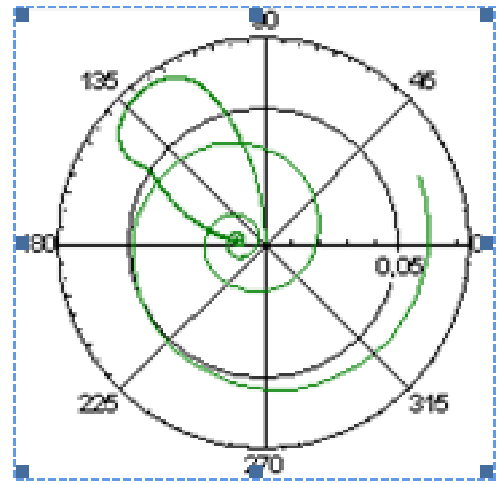
Time 0-0.4 - no moment

Time 0.4…… - applied moment

1 st left bearing

Start up 0… 6000





Instability ~ 2500 rpm

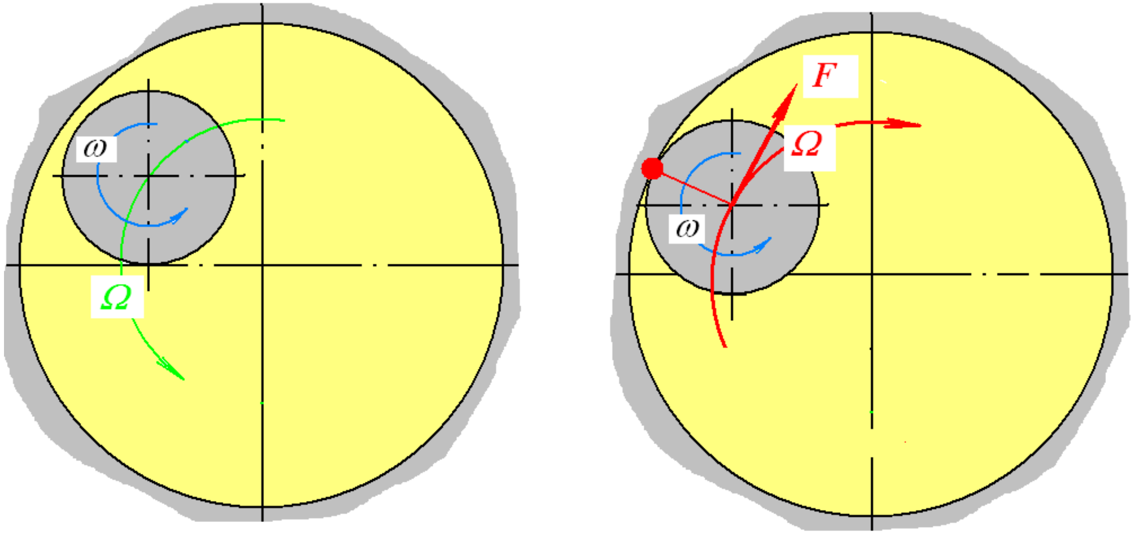
Loss of stability due incorrect choose of rotor structure and journal bearings/Operating speed 3000 rpm

Damage mechanism

1. Critical speed in the work range
2. Clearance adjustment
3.Rugh rubbing
4. Shaft braking

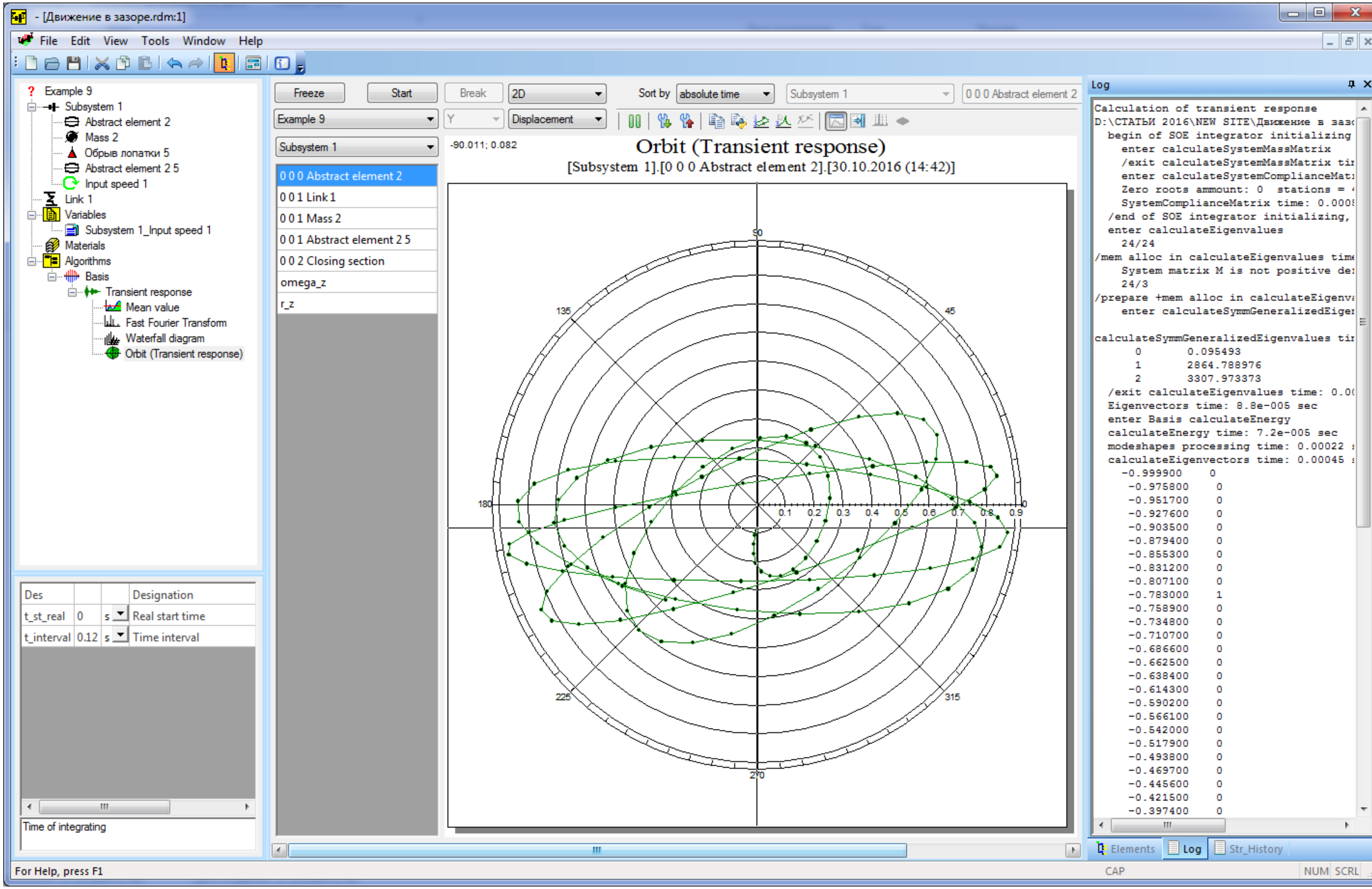
Fast rotor moving to zero speed and appearance of backward precession

Forward precession of rotor – normal work

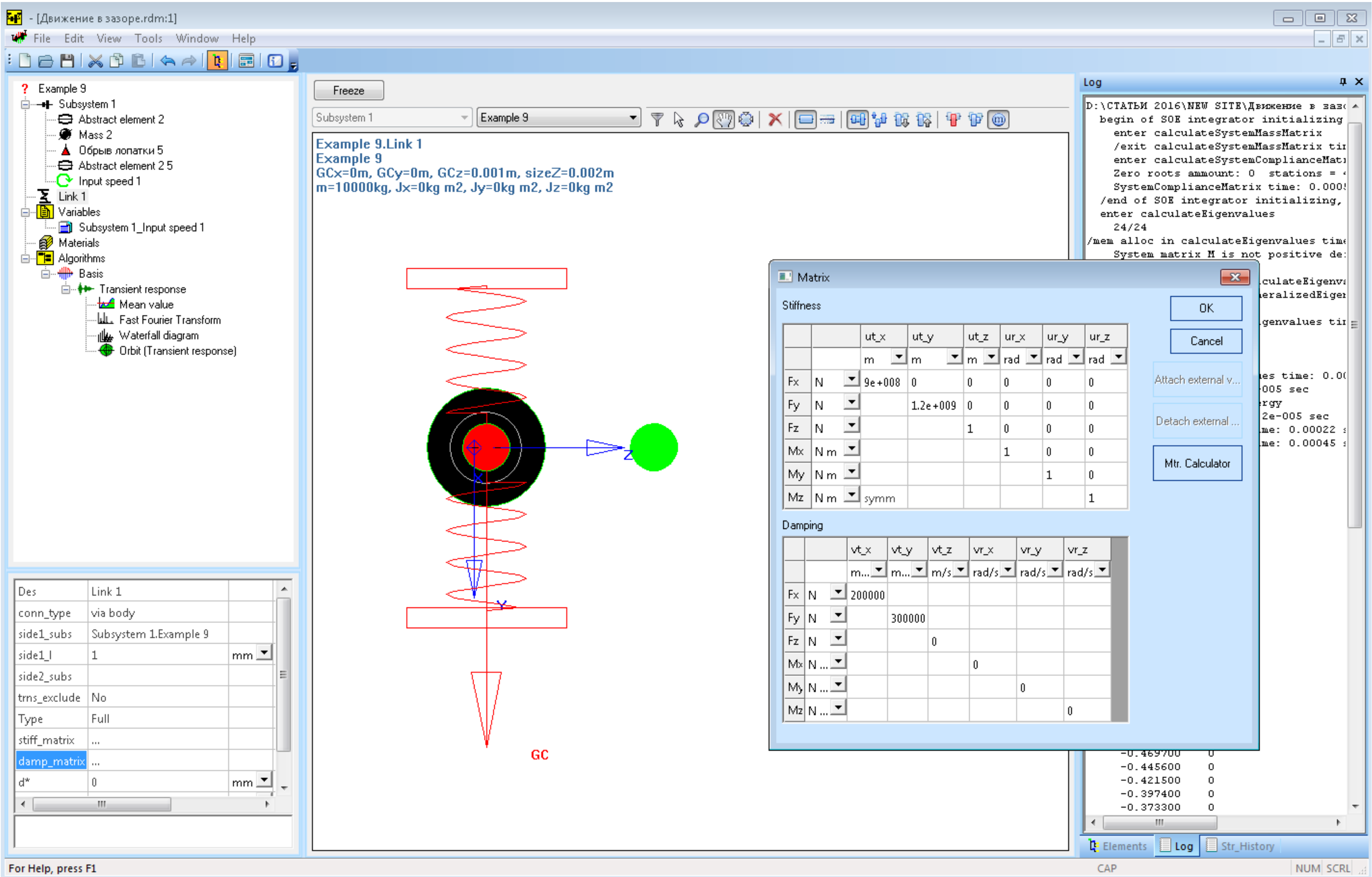


The reason of destruction is wrong design

DYNAMICS R4 – powerful tools for design and dynamic improvement of rotating machinery



No rubbings? Instability…



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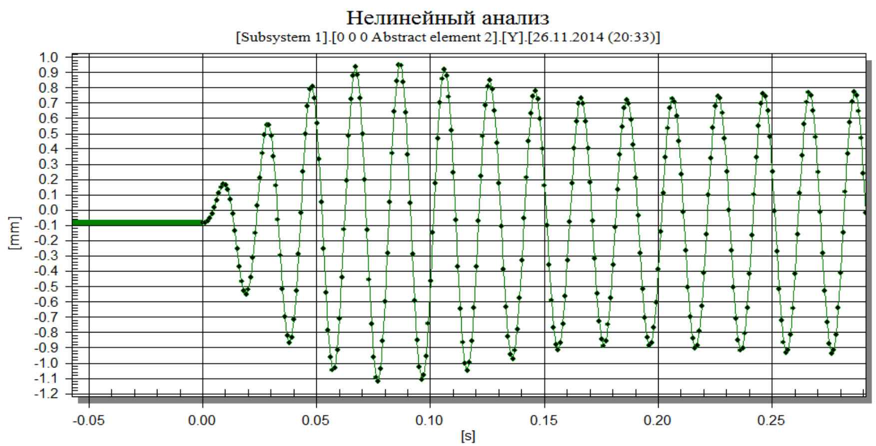
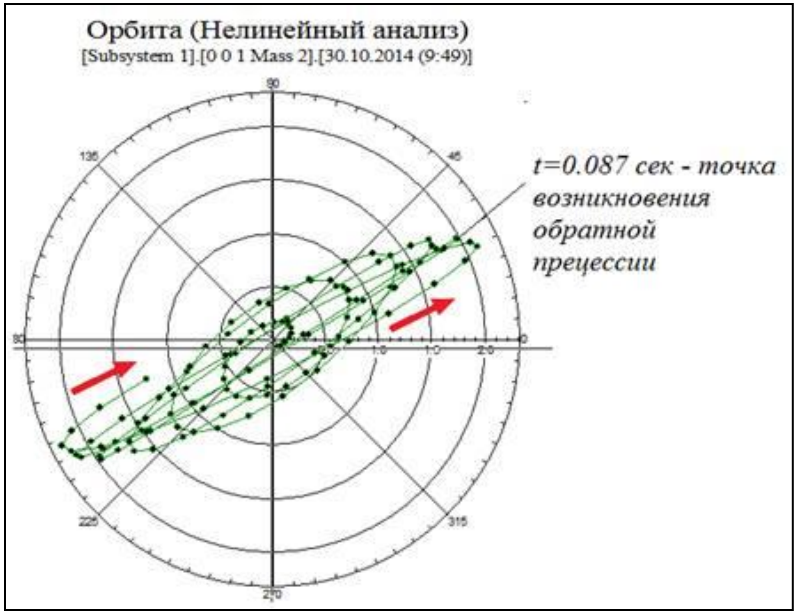
No rubbings? Instability…

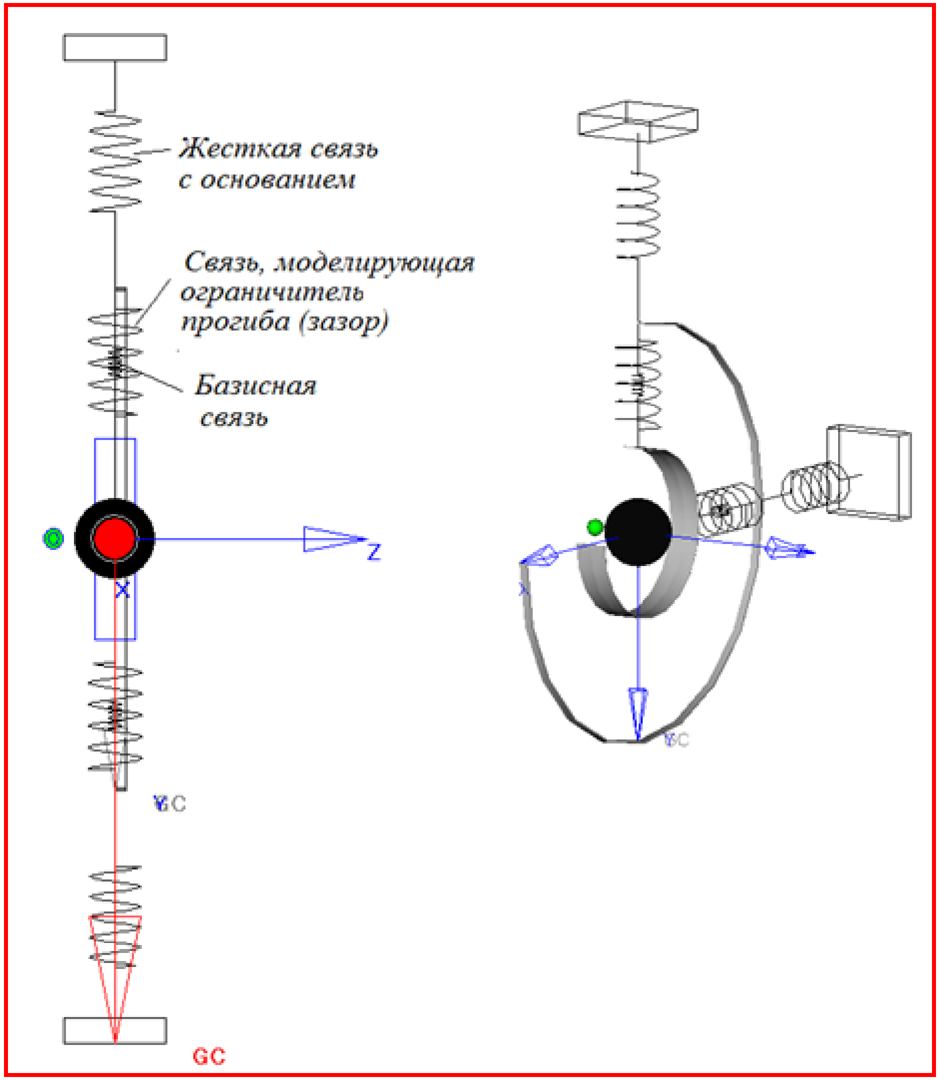




The point rotor with an orthotropic support was taken as an example how to simulate the rotor's motion in clearance without contact. Dependence of the rotor's motion on the position of the rotor's rotating speed ω at the unbalancing moment in relation to the natural frequencies p_1 and p_2 is shown.

Going to backward precession takes place if rotating speed ω, which sudden unbalance happens at, is between the first natural frequency p_1 and the second one p_2.

Influence of case flexibility on rotor instability

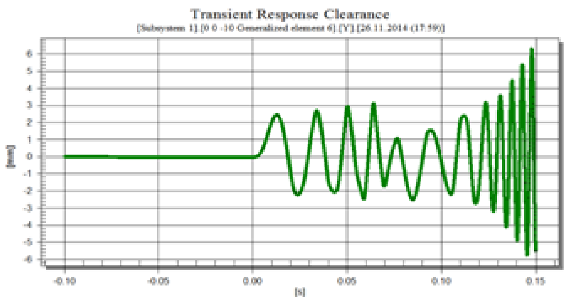


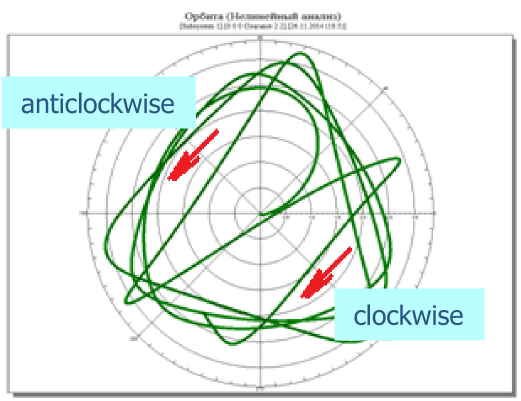
The point rotor having two freedom degrees was taken as an example to simulate the rotor's motion in clearance.

It is investigated how flexibility in the contact point influences precession type. It is shown that increase in flexibility in the contact point of the rotor and the stator decreases probability of the rotor's going to backward precession.

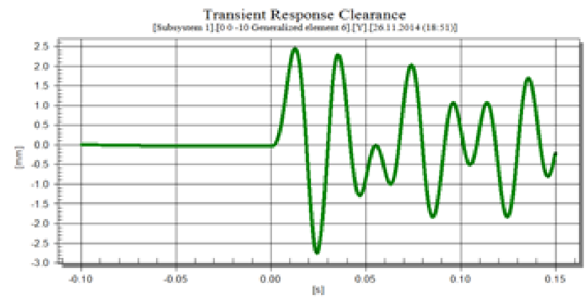
Flexibility influence in contact on rotor motion

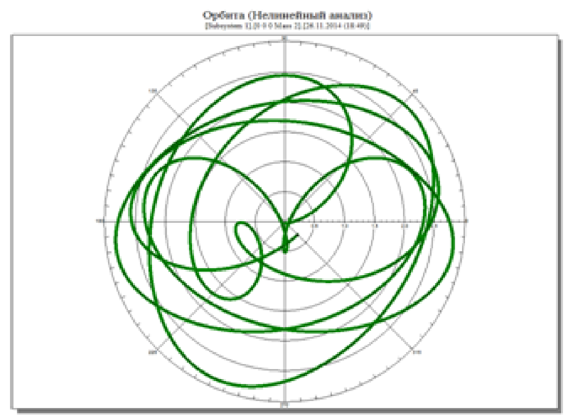
$K_1=K_2=1e11$ Н/м



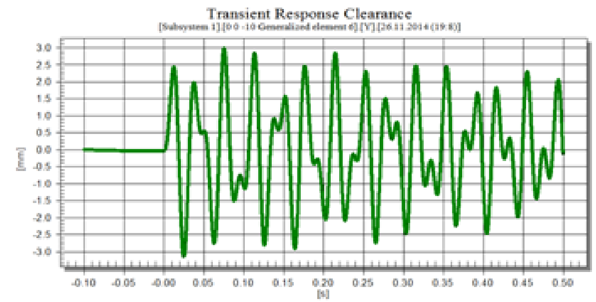


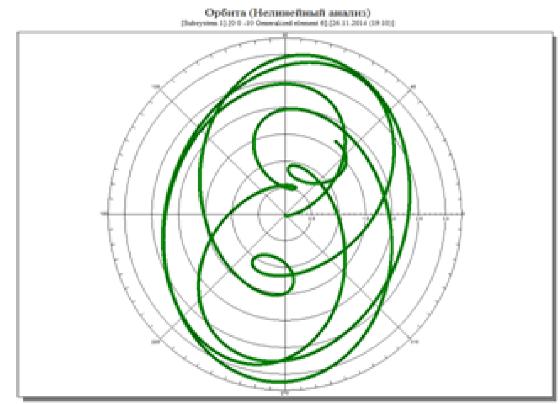
$K_1=K_2=1e10$ Н/м



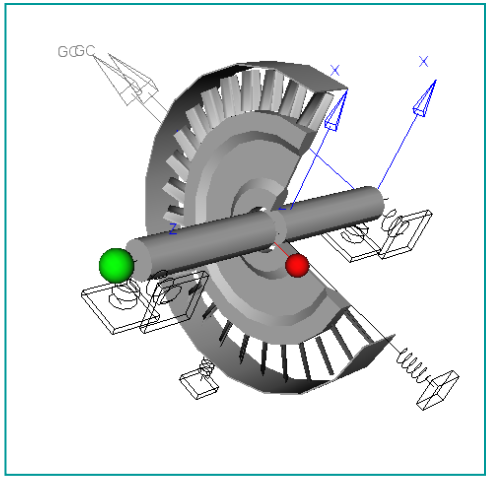


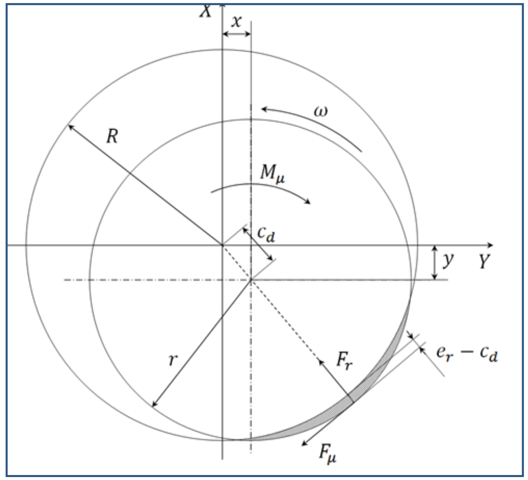
$K_1=K_2=1e09$ Н/м

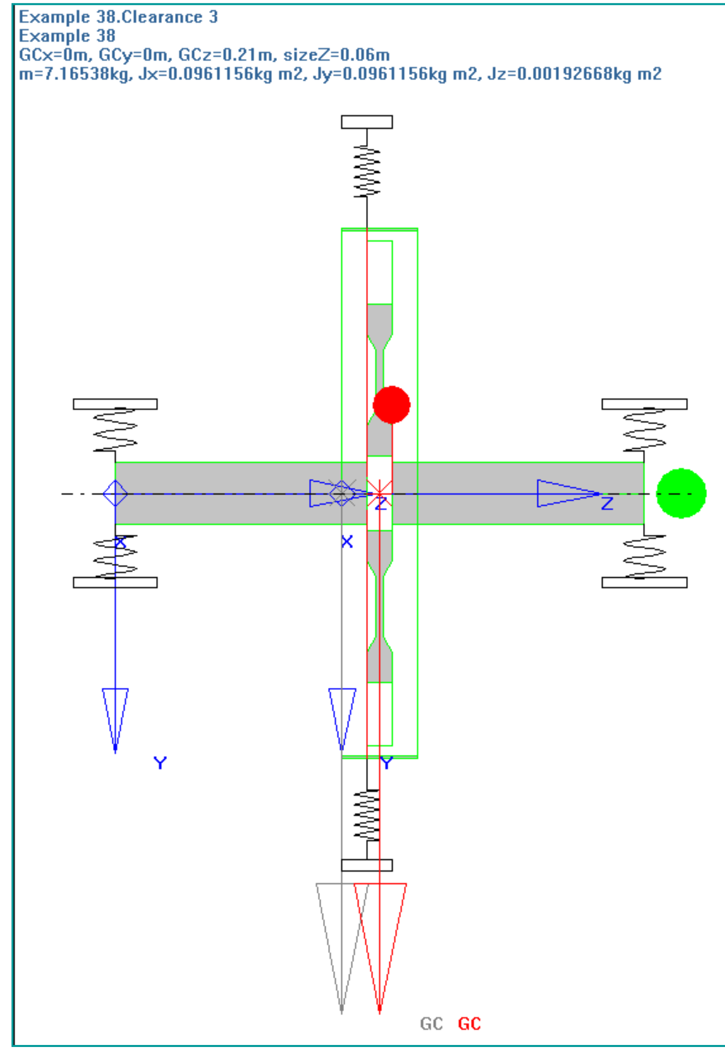




Rubbing. [Dry bush]







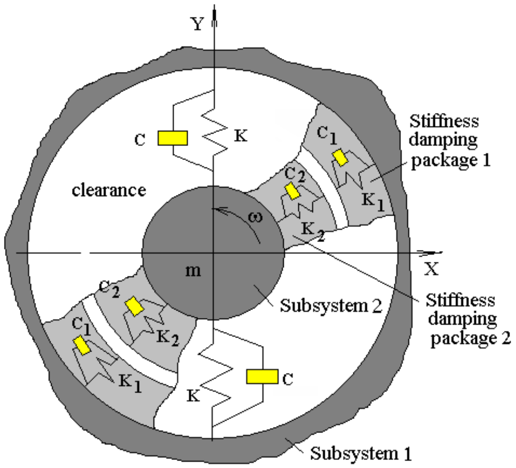
$$F_\mu = \begin{cases} \mu * Fr, \text{при } V > 0 \\ -\mu * Fr, \text{при } V < 0 \\ \mu s * Fr, \text{при } \le Vsmin \end{cases}$$

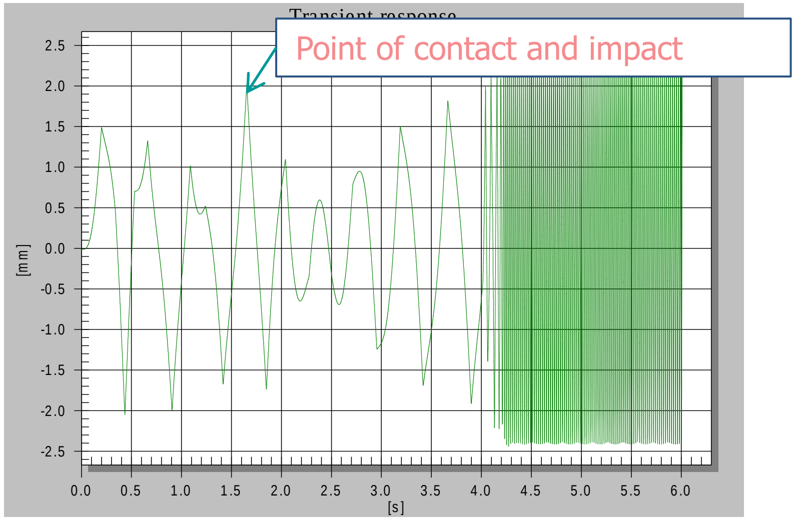
$$M\mu = R * F\mu$$

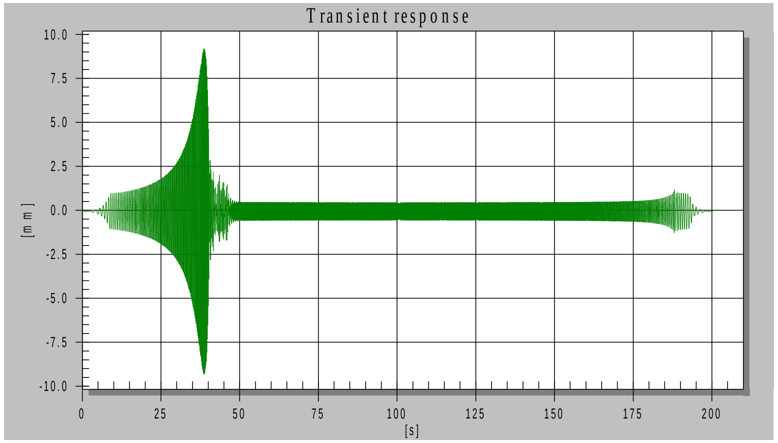
$$Vs = \Omega * \delta + (\omega 1 * r - \omega 2 * R)$$

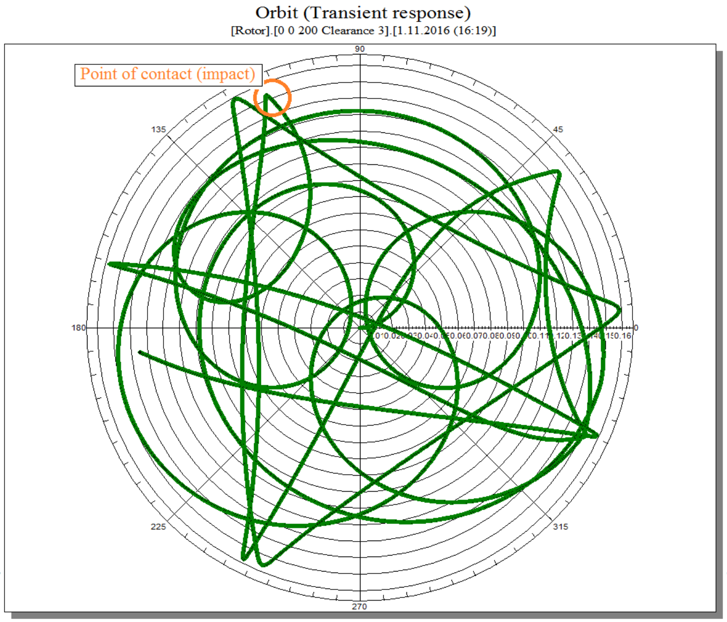
$$|Vs| \le Vsmin$$

Rubbing . [Clearance]









Short circuit simulation [Torque load]

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The [Torque load] element represents an external dynamic load applied to the shaft station. It is used for simulation of the dynamical variations of the torque load. For example, harmonic excitations can appear as a result of 2-phase, 3-phase short circuit in the electrical machines. Torque load is defined as a harmonic series. In order to simulate various behavior of dynamic torsion moment, a user can assign amplitude values for torque components, time constants and phases.
The following harmonic series is used to define the mathematical model of torque moment variations.

$$T = \begin{cases} T_{rated}[T_o e^{-a_0 t} + T_1 e^{-a_1 t}\sin(\omega t + \varphi_1) + T_2 e^{-a_2 t}\sin(2\omega t + \varphi_2)], t > t1 \\ T_{rated}, \quad t \le t1 \end{cases}$$

where T_{rated}, T_o, T_1, T_2, - the engine torque moment and coefficients for amplitude components of torque load variations., a_0, a_1, a_2 – time constants for decay process definition, φ_1, φ_2 – phase angles, ω – grid frequency.

Des	Torque Load		Designation
z1	0	mm	Start coordinate
R*	0	mm	Fictive radius
t_st_type	real		Start time definition
t_st_real	0.1	s	Real start time
Trated	31831	N m	Steady state driving torque
T0	0.837		Aperiodic torque component
T1	9.234		First order harmonic torque component
T2	0.435		Second order harmonic torque component
a0	14.7		Time constant of aperiodic torque component
a1	13.8		Time constant of first order harmonic torque component
a2	13.8		Time constant of first order harmonic torque component
f	3600	1/min	Grid frequency
phase1	0	deg	Phase angle 1st harmonic torque component
phase2	0	deg	Phase angle 2nd harmonic torque component
Visibility	0		Element Visibility