VIBRATION ANALYSIS OF ELECTRIC GENERATOR ROTOR SUPPORTED ON JOURNAL BEARINGS

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Annotation

The present paper shows the calculation results of the separate generator “BRUSH” being the part of the system “the power turbine, the transmission and the generator”, with the aim of its stability thresholds determination. Several variant investigations are carried out.

The task of the first research is to determine its stability thresholds on plain journal bearings in order to check two methodologies of determination of the rotor generator stability boundaries.

The task of the second research is to determine the stability thresholds of the rotor on the elliptical journal bearings. Modeling and linear analysis of the generator rotor system was carried out previously [1].

Transient response of generator rotor on plain journal bearings

In the Dynamics R4 software there is the “plain journal bearing” element. It simulates two main types of a plain bearing. The first type is short (without any ring seals). The second one is long (with ring seals). The thresholds conditions for the fluid film – [π-film] (half coverage) and [2π-film] (full coverage). The element can be used in the transient analysis for any rotor system modes of operation: both stationary and transient. Oil inlet pressure is taken into consideration. The element can be included in the rotor system model. In this case it becomes nonlinear, i.e. the hydraulic forces arising in the bearing are the functions of displacement and the rotor journals speed in the bearing clearance.

Fig. 1 shows the model of the given rotor generator. In more details it is described in [1]. Two elastic supports are changed for the identical plain journal bearings.

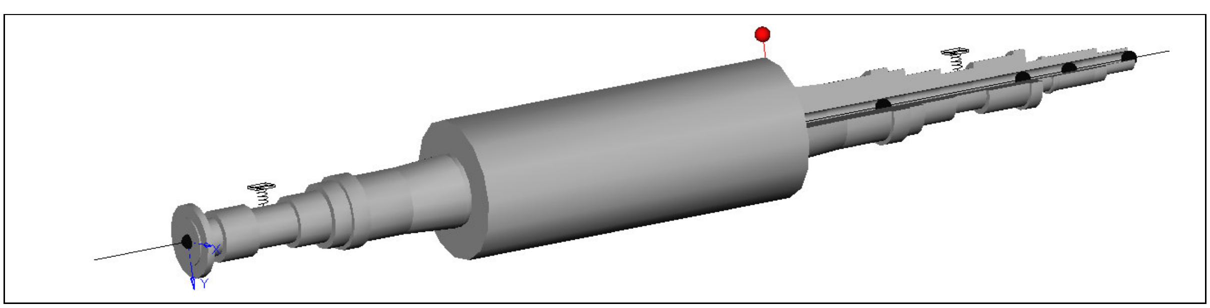


Fig. 1 3D generator rotor model

Calculations of the generator as a nonlinear system are carried out in the transient analysis.

According to the manufacturer data the minimum bearing clearance is in the range from 0.230 to 0.314 mm. For calculation the mean value of 0.272 mm is chosen. Fig. 2 presents the initial data on the plain bearings.

Des	Plain Journal Bearing support 3 3		Designation
conn_type	via body		Type of connection
side1_subs	Generator rotor		Side1 subsystem
side1_l	470	mm	side1 offset
side2_subs			Side2 subsystem
Type	short		Design type
R	114.5	mm	Bearing radius
film	pi-film		Cavitation film model
Lr	152	mm	Bearing length
delta	0.136	mm	Radial clearance
mue	14.38	cp	Lubricant absolute viscosity
P0	30000	Pa	Supply pressure to the bearing

Fig. 2 Plain journal bearing parameters

Fig. 3 presents time signal, obtained in the transient calculation of the generator on the nonlinear bearings, the output point is the left (front) bearing from the part of the power train. Time signal shows that the rotor loses its stability at 5500…6000 rpm.

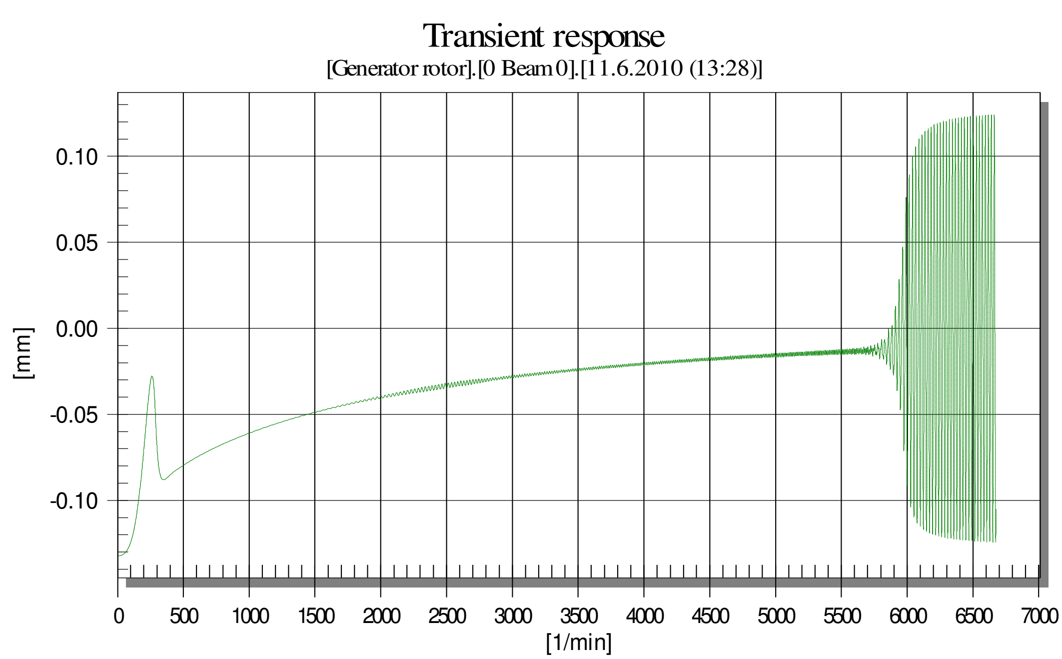


Fig. 3 Transient rotor calculation on plain bearings

To adjust the stability thresholds the method using the impulsive load at certain frequency was implemented to check the system response on it. If vibrations provoked by impulse do not subside, this generator operating regime is beyond the stability threshold.

The time signal received while motion equation integrating after exposure on the rotor by impulsive load of 10000 N at rotating speed of 4000 rpm leads to the displacement of the visible stability thresholds to 4000…4500 rpm, Fig. 4.

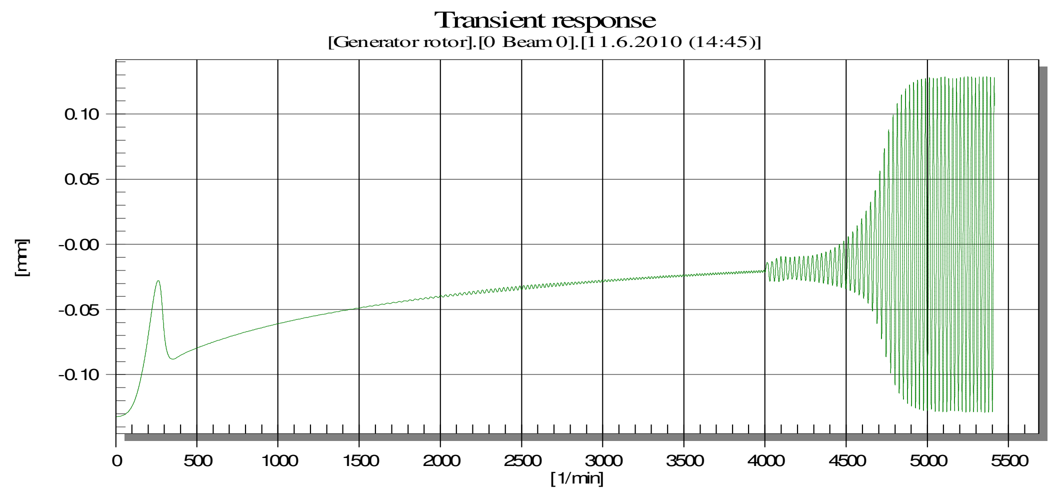


Fig 4 Transient calculation, impulse 10 000 N per 4000 rpm

Impulsive load at 3000 rpm does not lead to stability loss of the rotor, Fig. 5. An oscillation subsides, and after that rotor behaves similarly to time signal at Fig. 3. It means that stability threshold is significantly higher than 3000 rpm.

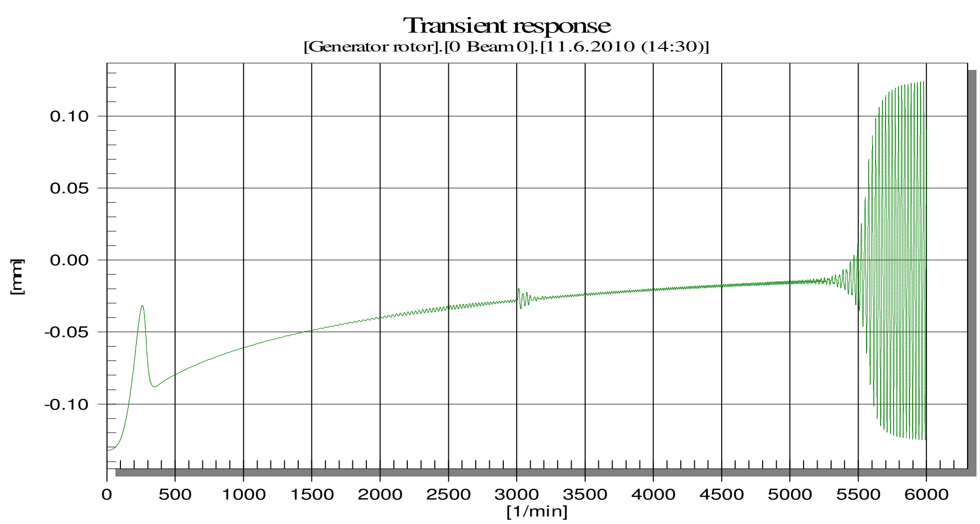


Fig. 5 Transient calculation, impulse 10000 N per 3000 rpm

To determine the stability thresholds some calculations at steady regimes were carried out. The stability of motion about an equilibrium position is determined by observing the motion of the linear system after giving it small perturbation about an equilibrium position. If this motion dies out with time and the system returns to its original position, the system is said to be stable; on the other hand, if this motion grows with time, the system is said to be unstable.

Fig. 6 shows the rotor calculation at constant rotating speed of 3500 rpm with impulsive load at 10000 N. Vibrations aroused by impulse subside rapidly.

The system also behaves similarly at frequency of 4000 rpm (Fig. 7). Oscillations excited by impulse subside. It means that the rotor does not pass stability thresholds yet. At the regime of 4200 rpm vibrations aroused by impulse do not subside, they start increasing in time, Fig. 8. It brings us to the conclusion that the rotor stability thresholds is within 4000 and 4200 rpm.

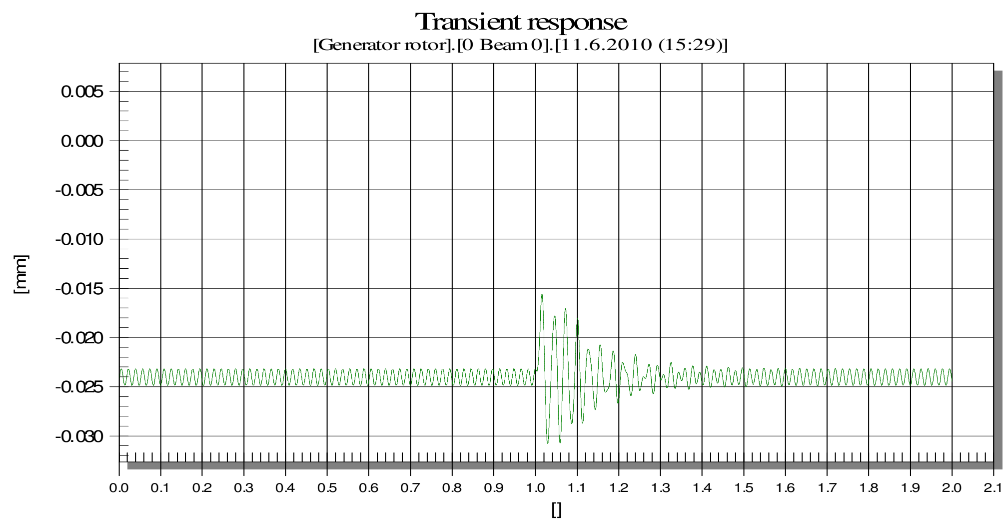


Fig. 6 Impulse on rotating speed of 3500 rpm

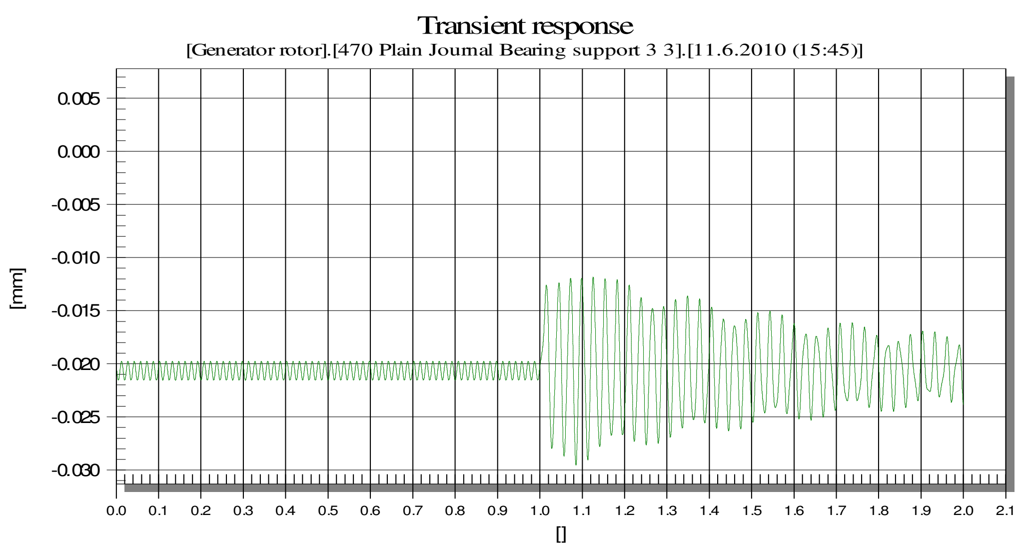


Fig. 7 Impulse on rotating speed of 4000 rpm

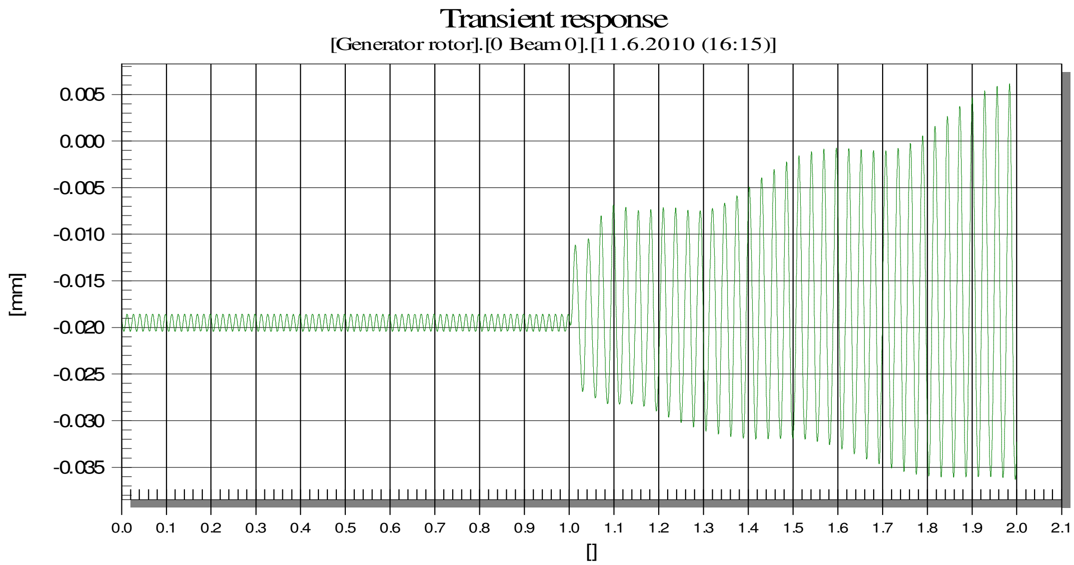


Fig. 8 Impulse on rotating speed of 4200 rpm

1.1 Quasi-nonlinear calculation of rotor on plain journal bearings

To confirm the obtained results quasi-nonlinear calculation and analysis of the rotor system on plain bearings with the adjusted characteristics is carried out.

The rotor generator is mounted on the elliptical rolling bearings. The bearing model used serves only for plain journal bearings calculation. It has a number of limitations concerning including the geometry and the operating parameters of such bearings. In particular what is not included is oil flow turbulence, its inertia, existence of grooves for oil inlet, temperature conditions, change in oil viscosity depending on operating mode, etc.

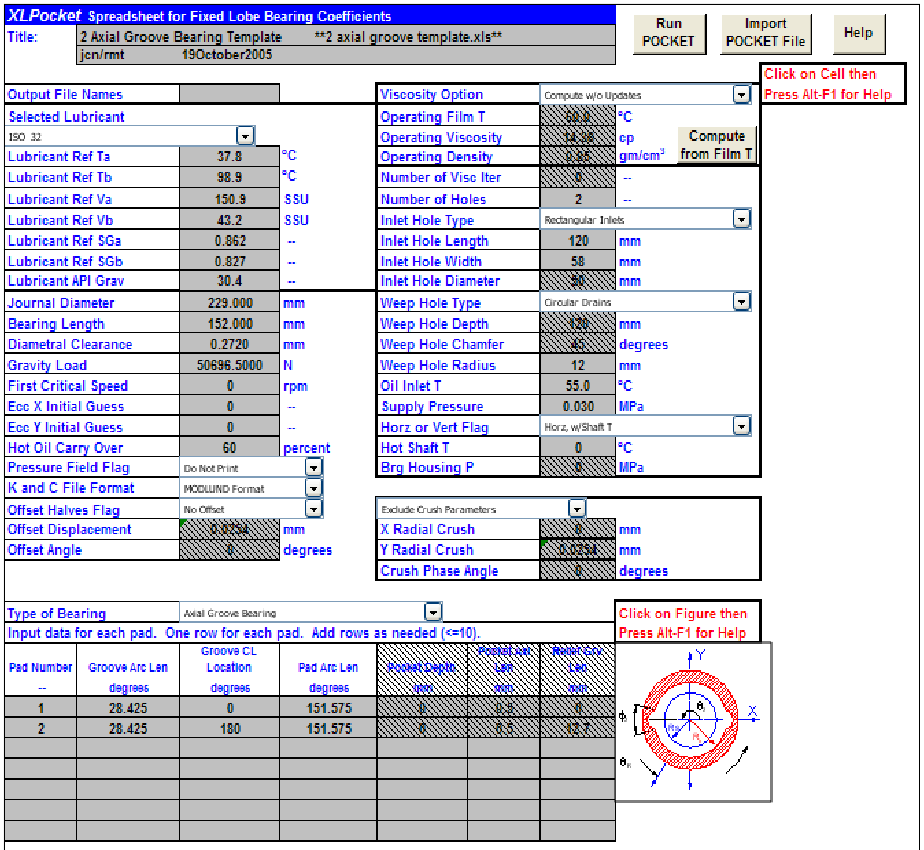
More complex algorithms in a transient analysis provoke great loss in time and it is difficult to use the model in practice during direct integration of the motion equations of the rotor system. That's why for the rotor generator calculation a quasi-nonlinear approach is used. It implies preliminary calculation of the stiffness and damping matrixes at the stationary operating regimes of the generator in the certain frequencies range. Then the received data on coefficients are used only for linear calculation of the natural frequencies map and the stability maps.

To determine the stiffness and damping coefficients the XLPocket software is used. It was developed by Dr. John Nicholas of RMT Inc. in Wellsville, New York. This software calculates stiffness and damping characteristics of journal bearings on stationary operating modes. The calculation results are presented as nonsymmetrical stiffness and damping matrixes. The received stiffness and damping matrixes are included into linear links between rotor and stator in the generator model.

In further modeling and analysis the following operating peculiarities of journal bearings were taken into account:

- oil flow turbulence ;
- oil inertia;
- temperature and pressure of oil supplied the journal bearing;
- change of oil's viscosity and density with the regime change;
- percent hot oil carry over from pad-to-pad;
- the number and dimensions of the oil inlet opening.

Fig. 9. presents the general view of the software interface.



X Ext Load N	Y Ext Load N	Speed rpm	Kxx N/m	Kxy N/m	Kyx N/m	Kyy N/m	Cxx N-s/m	Cxy N-s/m	Cyx N-s/m	Cyy N-s/m
		500	591.49E+6	-264.33E+6	-1.93E+9	3.40E+9	9.23E+6	-16.75E+6	-23.16E+6	75.86E+6
		1 000	574.53E+6	-79.34E+6	-1.62E+9	2.22E+9	5.17E+6	-7.89E+6	-10.41E+6	31.73E+6
		1 500	591.57E+6	28.76E+6	-1.53E+9	1.75E+9	3.91E+6	-5.18E+6	-6.72E+6	19.44E+6
		2 000	593.63E+6	85.13E+6	-1.46E+9	1.50E+9	3.18E+6	-3.87E+6	-4.87E+6	13.88E+6
		2 500	593.28E+6	122.79E+6	-1.42E+9	1.35E+9	2.65E+6	-3.04E+6	-3.73E+6	10.71E+6
		3 000	587.19E+6	145.99E+6	-1.39E+9	1.24E+9	2.27E+6	-2.48E+6	-2.98E+6	8.68E+6
		3 500	598.30E+6	182.68E+6	-1.38E+9	1.15E+9	2.06E+6	-2.15E+6	-2.52E+6	7.34E+6
		4 000	604.28E+6	211.23E+6	-1.37E+9	1.08E+9	1.89E+6	-1.89E+6	-2.17E+6	6.35E+6
		4 500	600.36E+6	225.19E+6	-1.36E+9	1.02E+9	1.70E+6	-1.63E+6	-1.86E+6	5.56E+6
		5 000	607.85E+6	253.36E+6	-1.36E+9	969.92E+6	1.60E+6	-1.49E+6	-1.66E+6	4.98E+6
		5 500	601.88E+6	263.50E+6	-1.35E+9	932.24E+6	1.47E+6	-1.31E+6	-1.46E+6	4.48E+6
		6 000	609.20E+6	289.28E+6	-1.35E+9	889.13E+6	1.41E+6	-1.22E+6	-1.33E+6	4.10E+6
		6 500	605.11E+6	300.09E+6	-1.34E+9	857.41E+6	1.32E+6	-1.11E+6	-1.20E+6	3.76E+6
		7 000	608.34E+6	320.02E+6	-1.34E+9	823.41E+6	1.27E+6	-1.04E+6	-1.10E+6	3.49E+6
		7 500	605.25E+6	333.70E+6	-1.34E+9	794.56E+6	1.20E+6	-953.94E+3	-1.00E+6	3.25E+6
		8 000	602.00E+6	347.83E+6	-1.34E+9	767.71E+6	1.15E+6	-881.21E+3	-920.07E+3	3.04E+6
		8 500	600.77E+6	364.19E+6	-1.34E+9	741.12E+6	1.10E+6	-820.32E+3	-850.50E+3	2.86E+6
		9 000	596.91E+6	377.17E+6	-1.34E+9	717.94E+6	1.06E+6	-763.74E+3	-786.14E+3	2.71E+6
		9 500	592.72E+6	390.47E+6	-1.34E+9	696.05E+6	1.02E+6	-713.66E+3	-729.07E+3	2.57E+6
		10 000	590.00E+6	404.53E+6	-1.35E+9	675.13E+6	985.79E+3	-672.07E+3	-680.37E+3	2.45E+6

Fig. 9 General view of the software XLPocket interface

Data for calculation of bearing characteristics

The generator operating mode is 3000 rpm (50 Hz). The total generator rotor mass is 10725 kg. The lubrication type - ISO VG 32. The oil temperature at the bearing entry is 55 °C. The oil inlet pressure accounts for 2 bar. In the regime of 3000 rpm pressure is decreased to 0.3 bars. The oil inlet is accomplished through symmetrical horizontal square apertures of 120x58 mm. The oil grade ISO VG 32 parameters are taken from the existing XL pocket data base and are presented on the Fig. 10. Table 1 shows the oil parameters deciphering.

Selected Lubricant		
ISO 32		
Lubricant Ref Ta	37.8	°C
Lubricant Ref Tb	98.9	°C
Lubricant Ref Va	150.9	SSU
Lubricant Ref Vb	43.2	SSU
Lubricant Ref SGa	0.862	--
Lubricant Ref SGb	0.827	--
Lubricant API Grav	30.4	--

Fig. 10

Table 1

Parameter	Value	Dimension
Viscosity at the temperature of 37.8 °C	150.9	SSU
Viscosity at the temperature of 98.9 °C	43.2	SSU
Specific weight at 40 °C	0.862	kg/sm^3
Specific weight at 98.9 °C	0.827	kg/sm^3
Density in degrees API	30.4	-

Fig. 11, 12, 13 and Tables 2, 3 present geometry dimensions and operating bearing parameters.

Journal Diameter	229.000	mm
Bearing Length	152.000	mm
Diametral Clearance	0.2720	mm
Gravity Load	50696.5000	N
First Critical Speed	0	rpm
Ecc X Initial Guess	0	--
Ecc Y Initial Guess	0	--
Hot Oil Carry Over	60	percent
Pressure Field Flag	Do Not Print	
K and C File Format	MODLUND Format	
Offset Halves Flag	No Offset	
Offset Displacement	0.0254	mm
Offset Angle	0	degrees

Journal Diameter	229.000	mm
Bearing Length	152.000	mm
Diametral Clearance	0.2720	mm
Gravity Load	54479.821	N
First Critical Speed	0	rpm
Ecc X Initial Guess	0	--
Ecc Y Initial Guess	0	--
Hot Oil Carry Over	60	percent
Pressure Field Flag	Do Not Print	
K and C File Format	MODLUND Format	
Offset Halves Flag	No Offset	
Offset Displacement	0.0254	mm
Offset Angle	0	degrees

Fig. 11 Parameters of the generator left and right bearings correspondingly

Type of Bearing: Axial Groove Bearing

Input data for each pad. One row for each pad. Add rows as needed (<=10).

Pad Number --	Groove Arc Len degrees	Groove CL Location degrees	Pad Arc Len degrees	Pocket Depth mm	Pocket Axl Len mm	Relief Grv Len mm
1	28.425	0	151.575	0	0.5	0
2	28.425	180	151.575	0	0.5	12.7

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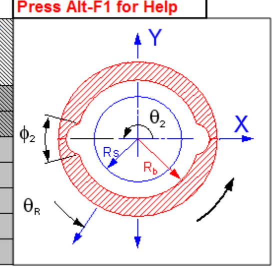

Fig. 12 Oil inlet grooves arrangement

Table 2

Parameter	Value	Dimension
Diameter	229	mm
Diametral clearance	0.272	mm
Effective bearing length	152	mm
Load on the left bearing	50696.5	N
Load on the right bearing	54479.8	N
The number of oil inlet apertures	2	-
Oil inlet apertures length	120	mm
Oil inlet apertures width	58	mm
Oil temperature at the bearing entry	55	^{o}C
Oil inlet pressure before the operating regime	2	bar
Oil inlet pressure at the operating regime	0.3	bar

The angular groove length are chosen in compliance with the oil input apertures width and equal to 28.425^{O}. Fig. 13 shows the bearing design.

Table 3

Parameter	Value		Dimension
	1-st groove	2-nd groove	
Angular groove length (Angular extent of axial oil supply or feed grooves)	28.425	28.425	degree
Groove arrangement angle (Locates centerline (CL) of the axial oil supply grooves with rotation from +X axis)	0	180	degree
Arc length of lobe	151.575	151.575	degree

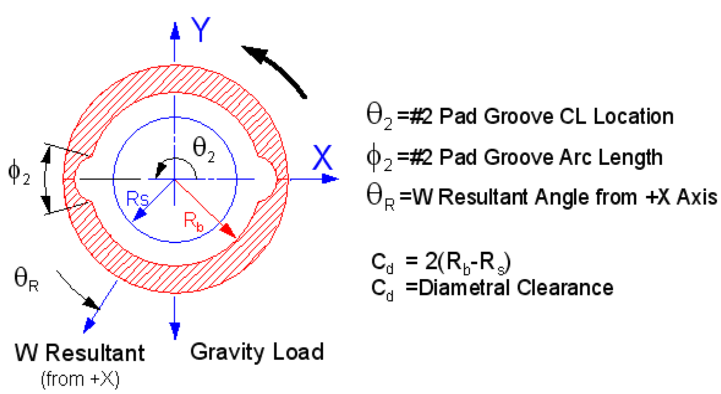

Fig. 13 Plain bearing design

Results of generator calculations on plain bearings

The matrixes of the stiffness and damping coefficients are calculated for oil pressure 0.3 bar and the regimes range from 500 to 10000 rpm, tables 4, 5.

Table 4

Stiffness and damping coefficients for left bearing

Speed rpm	Kxx N/m	Kxy N/m	Kyx N/m	Kyy N/m	Cxx N-s/m	Cxy N-s/m	Cyx N-s/m	Cyy N-s/m
500	591.49E+6	-264.33E+6	-1.93E+9	3.40E+9	9.23E+6	-16.75E+6	-23.16E+6	75.86E+6
1 000	574.53E+6	-79.34E+6	-1.62E+9	2.22E+9	5.17E+6	-7.89E+6	-10.41E+6	31.73E+6
1 500	591.57E+6	28.76E+6	-1.53E+9	1.75E+9	3.91E+6	-5.18E+6	-6.72E+6	19.44E+6
2 000	593.63E+6	85.13E+6	-1.46E+9	1.50E+9	3.18E+6	-3.87E+6	-4.87E+6	13.88E+6
2 500	593.28E+6	122.79E+6	-1.42E+9	1.35E+9	2.65E+6	-3.04E+6	-3.73E+6	10.71E+6
3 000	587.19E+6	145.99E+6	-1.39E+9	1.24E+9	2.27E+6	-2.48E+6	-2.98E+6	8.68E+6
3 500	598.30E+6	182.68E+6	-1.38E+9	1.15E+9	2.06E+6	-2.15E+6	-2.52E+6	7.34E+6
4 000	604.28E+6	211.23E+6	-1.37E+9	1.08E+9	1.89E+6	-1.89E+6	-2.17E+6	6.35E+6
4 500	600.36E+6	225.19E+6	-1.36E+9	1.02E+9	1.70E+6	-1.63E+6	-1.86E+6	5.56E+6
5 000	607.85E+6	253.36E+6	-1.36E+9	969.92E+6	1.60E+6	-1.49E+6	-1.66E+6	4.98E+6
5 500	601.88E+6	263.50E+6	-1.35E+9	932.24E+6	1.47E+6	-1.31E+6	-1.46E+6	4.48E+6
6 000	609.20E+6	289.28E+6	-1.35E+9	889.13E+6	1.41E+6	-1.22E+6	-1.33E+6	4.10E+6
6 500	605.11E+6	300.09E+6	-1.34E+9	857.41E+6	1.32E+6	-1.11E+6	-1.20E+6	3.76E+6
7 000	608.34E+6	320.02E+6	-1.34E+9	823.41E+6	1.27E+6	-1.04E+6	-1.10E+6	3.49E+6
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8 500	600.77E+6	364.19E+6	-1.34E+9	741.12E+6	1.10E+6	-820.32E+3	-850.50E+3	2.86E+6
9 000	596.91E+6	377.17E+6	-1.34E+9	717.94E+6	1.06E+6	-763.74E+3	-786.14E+3	2.71E+6
9 500	592.72E+6	390.47E+6	-1.34E+9	696.05E+6	1.02E+6	-713.66E+3	-729.07E+3	2.57E+6
10 000	590.00E+6	404.53E+6	-1.35E+9	675.13E+6	985.79E+3	-672.07E+3	-680.37E+3	2.45E+6

Table 5

Stiffness and damping coefficients for right bearing

Speed rpm	Kxx N/m	Kxy N/m	Kyx N/m	Kyy N/m	Cxx N-s/m	Cxy N-s/m	Cyx N-s/m	Cyy N-s/m
500	663.56E+6	-298.80E+6	-2.17E+9	3.85E+9	9.69E+6	-18.05E+6	-25.11E+6	83.50E+6
1 000	640.69E+6	-90.75E+6	-1.81E+9	2.50E+9	5.75E+6	-8.56E+6	-11.77E+6	34.91E+6
1 500	615.96E+6	-13.30E+6	-1.65E+9	2.01E+9	3.94E+6	-5.45E+6	-7.12E+6	21.13E+6
2 000	617.67E+6	48.69E+6	-1.57E+9	1.72E+9	3.17E+6	-4.03E+6	-5.13E+6	15.01E+6
2 500	629.68E+6	103.70E+6	-1.54E+9	1.53E+9	2.74E+6	-3.24E+6	-4.03E+6	11.62E+6
3 000	639.63E+6	147.81E+6	-1.52E+9	1.39E+9	2.44E+6	-2.74E+6	-3.31E+6	9.49E+6
3 500	641.68E+6	176.28E+6	-1.50E+9	1.29E+9	2.16E+6	-2.31E+6	-2.75E+6	7.95E+6
4 000	641.44E+6	198.34E+6	-1.48E+9	1.21E+9	1.94E+6	-2.00E+6	-2.34E+6	6.83E+6
4 500	649.03E+6	228.97E+6	-1.47E+9	1.14E+9	1.81E+6	-1.79E+6	-2.06E+6	6.02E+6
5 000	643.63E+6	240.83E+6	-1.46E+9	1.09E+9	1.64E+6	-1.57E+6	-1.79E+6	5.34E+6
5 500	651.47E+6	269.94E+6	-1.45E+9	1.04E+9	1.56E+6	-1.44E+6	-1.61E+6	4.83E+6
6 000	645.50E+6	279.42E+6	-1.44E+9	1.00E+9	1.44E+6	-1.29E+6	-1.43E+6	4.38E+6
6 500	652.94E+6	304.99E+6	-1.44E+9	955.74E+6	1.40E+6	-1.22E+6	-1.32E+6	4.04E+6
7 000	649.76E+6	319.20E+6	-1.44E+9	920.37E+6	1.31E+6	-1.11E+6	-1.19E+6	3.73E+6
7 500	648.29E+6	335.08E+6	-1.43E+9	886.90E+6	1.25E+6	-1.03E+6	-1.09E+6	3.46E+6
8 000	647.60E+6	352.92E+6	-1.43E+9	854.20E+6	1.20E+6	-954.36E+3	-1.01E+6	3.24E+6
8 500	642.18E+6	365.15E+6	-1.43E+9	826.50E+6	1.14E+6	-878.97E+3	-924.54E+3	3.04E+6
9 000	642.35E+6	383.83E+6	-1.43E+9	797.41E+6	1.11E+6	-832.39E+3	-863.16E+3	2.88E+6
9 500	638.40E+6	397.12E+6	-1.43E+9	772.33E+6	1.06E+6	-775.07E+3	-799.54E+3	2.73E+6
10 000	634.78E+6	411.31E+6	-1.43E+9	748.83E+6	1.03E+6	-729.37E+3	-745.00E+3	2.60E+6

The received radial stiffness bearing coefficients Kxx and Kyy (are highlighted in blue), at 3000 rpm are somewhat different from the data given by the generator manufacturer (Kxx=500e6 N/m; Kyy=1000e6 N/m).

Fig. 14, 15, 16 show the calculation results of the generator on the elastic supports with the obtained stiffness and damping matrixes. Fig. 14 presents the generator natural frequencies map.

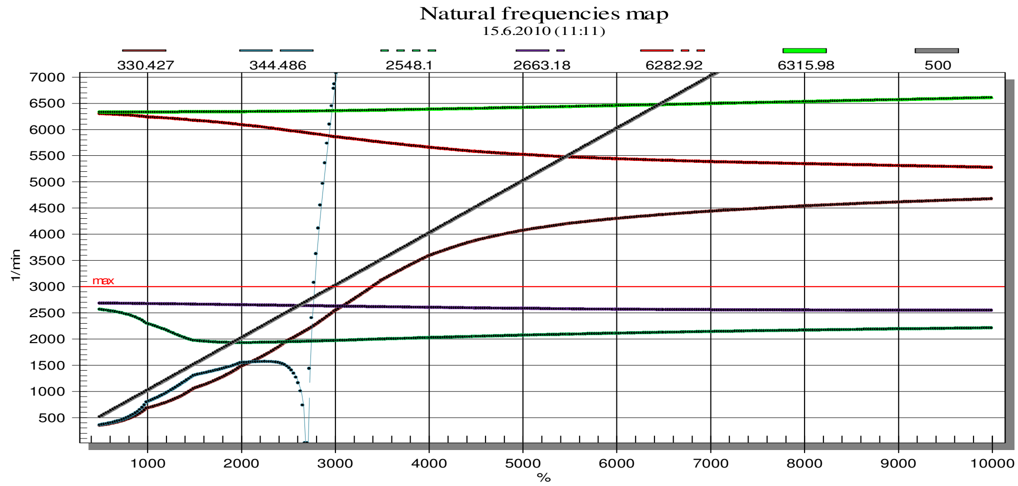


Fig. 14 Natural frequencies map of rotor gas generator on plain bearings

Fig. 15 shows the generator stability map - the damping ratio coefficient versus the rotor rotating speed. The stability maps indicate that at the 3825 rpm frequency - the damping ratio coefficient changes its sign into the minus one which proves that the rotor loses its stability at this speed.

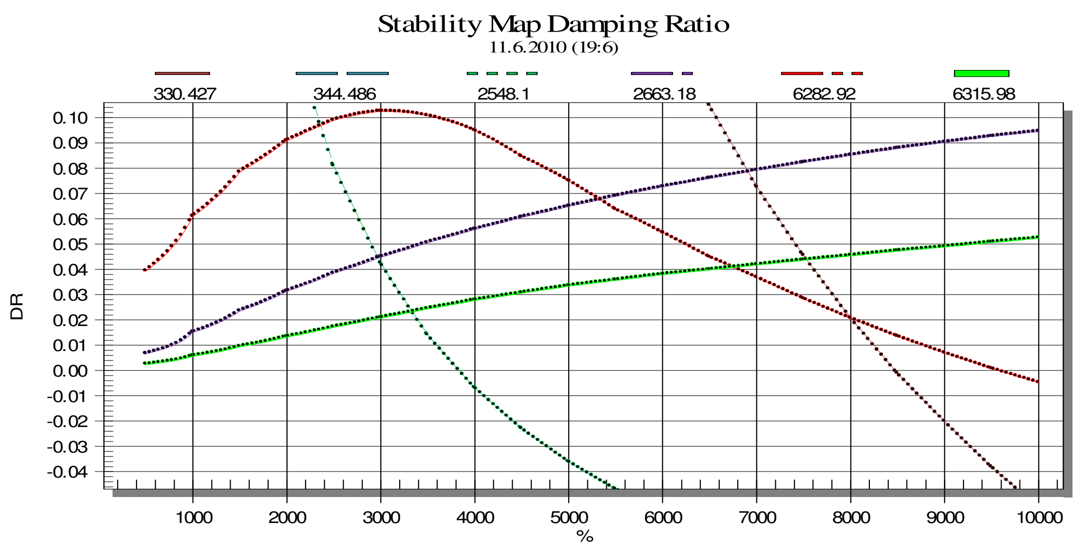


Fig. 15 Map of relative damping coefficients of different mode shapes versus rotor rotating speed

Fig. 16 presents the logarithmic decrement versus the natural oscillations frequencies of the rotor system. It allows estimating the frequency and the corresponding mode shape, at which stability loss happens.

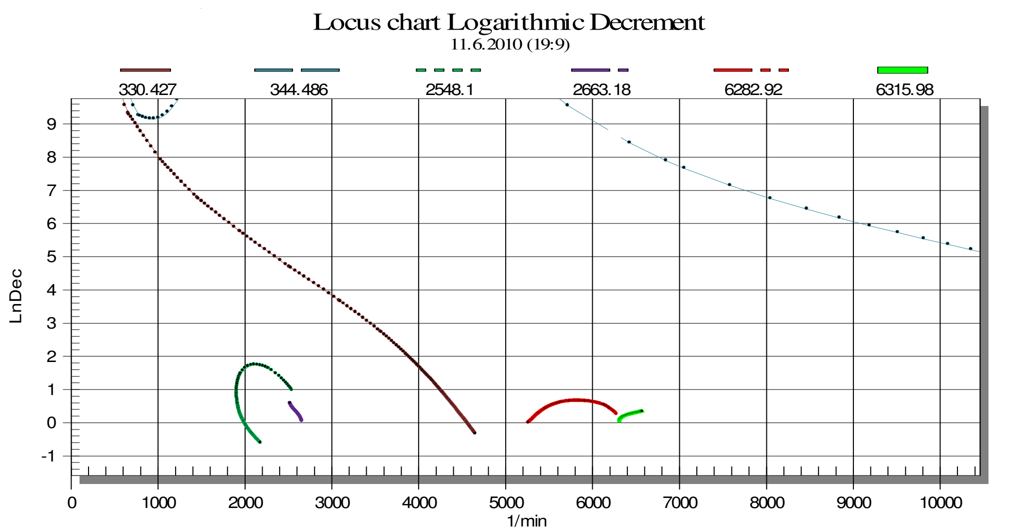


Fig. 16. Logarithmic decrement versus rotor natural oscillations frequencies

Fig. 17 shows the natural frequencies, calculated at the 3825 rpm rotor rotating speed . The frequency and the corresponding mode shape at which stability loss happens are highlighted. This frequency may also be defined at Fig 16.

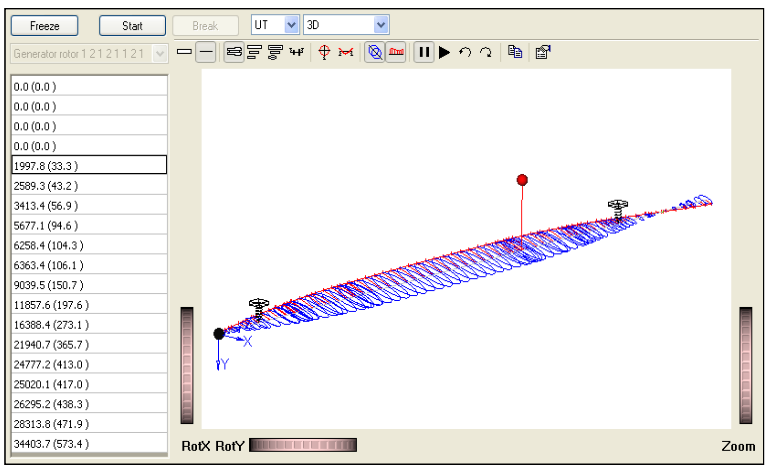


Fig. 17 Rotor generator natural frequencies at 3825 rpm.

At 3825 rpm rotor generator loses its stability, the regime is close to the operating mode of 3000 rpm. Dispersion in the clearance, oil inlet pressure and temperature may even more approximate the rotor stability loss thresholds to the operating mode.

It is impossible to use the generator on the journal bearings for the generator given with current efficiency of 60 Hz (i.e. the rotor rotating speed is 3600 rpm). It is evident that these facts forced the generator developers to use elliptical bearings here. In comparison with plain bearings elliptical ones significantly increase the rotor stability thresholds other things being equal.

Ggenerator rotor on elliptical bearing

As opposed to a plain bearing an elliptical one has an irregular circular clearance. This irregularity is set by the “Preload” parameter. The value of this parameter is equal to 0.56 (given by the manufacturer). All the other data – on the bearing geometry, oil temperature, regimes – coincide with the plain bearing. Fig. 18 and Table 6 present the bearings geometrical parameters.

Type of Bearing: Mutli Lobe or Elliptical Bearing

Input data for each pad. One row for each pad. Add rows as needed (<=10).

Pad Number --	Groove Arc Len degrees	Groove CL Location degrees	Lobe Arc Len degrees	Lobe Preload --	Lobe Offset --	Relief Grv Len mm
1	28.425	0	151.575	0.56	0.5	0
2	28.425	180	151.575	0.56	0.5	12.7

Click on Figure then Press Alt-F1 for Help

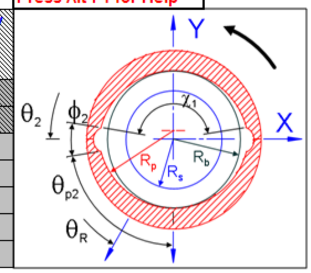


Fig. 18 Oil inlet grooves arrangement

Table 6

Parameter	Value		Dimension
	1-st groove	2-nd groove	
Angular groove length (Angular extent of axial oil supply or feed grooves)	28.425	28.425	degree
Groove arrangement angle (Locates centerline (CL) of the axial oil supply grooves with rotation from +X axis)	0	180	degree
Arc length of lobe	151.575	151.575	degree
Preload of lobe	0.56	0.56	
Offset of lobe	0.5	0.5	

Fig 19. presents the elliptical bearing design and its parameters.

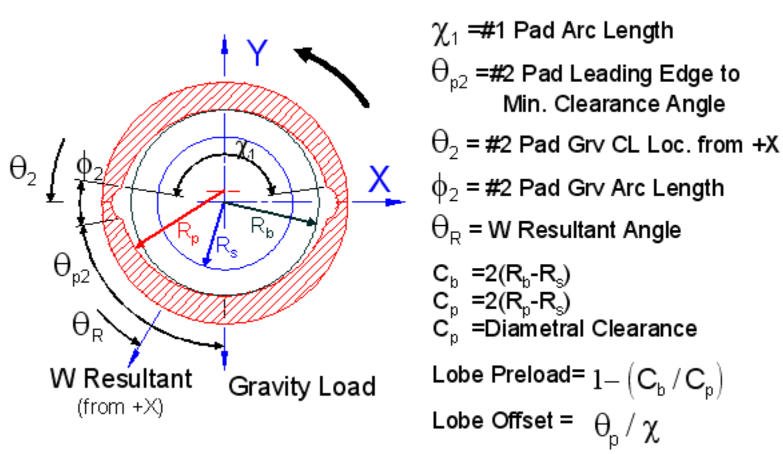


Fig 19 Elliptical bearing design and parameters

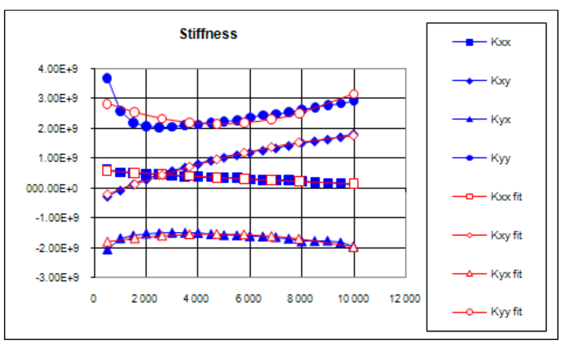
Journal bearing stiffness and damping characteristics

The stiffness and damping matrixes were calculated at the operating range from 500 to 10000 rpm. Fig. 7 and Fig. 8 present the coefficients of these matrixes. Fig. 20, 21 show these coefficients change according to the regimes.

Table 7

Stiffness and damping coefficients for left (front) bearing

Speed rpm	Kxx N/m	Kxy N/m	Kyx N/m	Kyy N/m	Cxx N-s/m	Cxy N-s/m	Cyx N-s/m	Cyy N-s/m
500	621.54E+6	-268.71E+6	-2.06E+9	3.67E+9	9.23E+6	-16.38E+6	-23.19E+6	78.66E+6
1 000	527.01E+6	-62.95E+6	-1.68E+9	2.59E+9	5.83E+6	-6.88E+6	-9.97E+6	34.02E+6
1 500	510.31E+6	135.17E+6	-1.59E+9	2.20E+9	4.76E+6	-3.32E+6	-5.57E+6	21.72E+6
2 000	471.51E+6	298.15E+6	-1.52E+9	2.08E+9	4.18E+6	-1.58E+6	-3.36E+6	16.47E+6
2 500	458.75E+6	458.95E+6	-1.51E+9	2.03E+9	3.80E+6	-604.23E+3	-2.15E+6	13.60E+6
3 000	424.99E+6	589.56E+6	-1.49E+9	2.05E+9	3.45E+6	108.07E+3	-1.27E+6	11.76E+6
3 500	386.15E+6	709.37E+6	-1.49E+9	2.10E+9	3.16E+6	649.57E+3	-600.69E+3	10.49E+6
4 000	380.17E+6	818.31E+6	-1.50E+9	2.13E+9	2.93E+6	859.22E+3	-314.21E+3	9.52E+6
4 500	361.15E+6	931.31E+6	-1.54E+9	2.18E+9	2.80E+6	1.06E+6	-35.37E+3	8.92E+6
5 000	347.74E+6	1.03E+9	-1.58E+9	2.23E+9	2.68E+6	1.18E+6	152.58E+3	8.42E+6
5 500	345.84E+6	1.12E+9	-1.58E+9	2.27E+9	2.50E+6	1.21E+6	250.70E+3	7.87E+6
6 000	291.38E+6	1.20E+9	-1.61E+9	2.37E+9	2.37E+6	1.46E+6	526.62E+3	7.46E+6
6 500	260.93E+6	1.27E+9	-1.62E+9	2.44E+9	2.25E+6	1.55E+6	660.44E+3	7.08E+6
7 000	268.09E+6	1.35E+9	-1.64E+9	2.48E+9	2.15E+6	1.52E+6	669.46E+3	6.77E+6
7 500	257.06E+6	1.43E+9	-1.70E+9	2.54E+9	2.08E+6	1.53E+6	714.06E+3	6.59E+6
8 000	218.97E+6	1.53E+9	-1.79E+9	2.63E+9	2.08E+6	1.63E+6	839.38E+3	6.57E+6
8 500	193.00E+6	1.58E+9	-1.77E+9	2.70E+9	1.96E+6	1.68E+6	916.69E+3	6.21E+6
9 000	175.69E+6	1.64E+9	-1.76E+9	2.77E+9	1.86E+6	1.70E+6	957.00E+3	5.95E+6
9 500	165.85E+6	1.72E+9	-1.83E+9	2.83E+9	1.82E+6	1.69E+6	972.85E+3	5.85E+6
10 000	126.22E+6	1.82E+9	-1.95E+9	2.94E+9	1.84E+6	1.78E+6	1.08E+6	5.96E+6



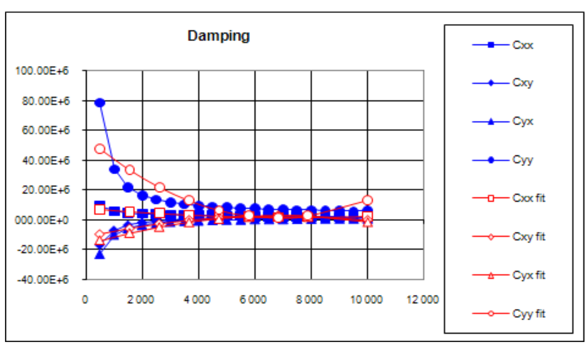


Fig. 20 Diagrams of stiffness and damping change

The curves with the subscript “fit” are the approximating second-order ones

Table 8

Stiffness and damping coefficients for right (rear) bearing

Speed rpm	Kxx N/m	Kxy N/m	Kyx N/m	Kyy N/m	Cxx N-s/m	Cxy N-s/m	Cyx N-s/m	Cyy N-s/m
500	694.19E+6	-306.97E+6	-2.31E+9	4.16E+9	9.72E+6	-17.87E+6	-25.32E+6	86.82E+6
1 000	593.41E+6	-100.45E+6	-1.87E+9	2.90E+9	5.88E+6	-7.65E+6	-10.87E+6	37.10E+6
1 500	567.89E+6	99.34E+6	-1.75E+9	2.43E+9	4.97E+6	-4.14E+6	-6.50E+6	23.67E+6
2 000	511.00E+6	243.14E+6	-1.64E+9	2.28E+9	4.18E+6	-2.07E+6	-3.87E+6	17.45E+6
2 500	503.22E+6	419.30E+6	-1.64E+9	2.19E+9	3.90E+6	-995.49E+3	-2.60E+6	14.37E+6
3 000	477.69E+6	563.77E+6	-1.63E+9	2.17E+9	3.61E+6	-253.59E+3	-1.69E+6	12.39E+6
3 500	454.51E+6	673.00E+6	-1.60E+9	2.20E+9	3.25E+6	217.76E+3	-1.06E+6	10.89E+6
4 000	405.47E+6	790.29E+6	-1.61E+9	2.26E+9	3.04E+6	715.02E+3	-454.33E+3	9.90E+6
4 500	404.58E+6	887.60E+6	-1.61E+9	2.28E+9	2.81E+6	854.89E+3	-259.77E+3	9.05E+6
5 000	385.68E+6	996.24E+6	-1.64E+9	2.33E+9	2.70E+6	1.02E+6	-20.42E+3	8.53E+6
5 500	367.03E+6	1.10E+9	-1.69E+9	2.38E+9	2.61E+6	1.16E+6	168.38E+3	8.14E+6
6 000	371.28E+6	1.18E+9	-1.69E+9	2.41E+9	2.46E+6	1.16E+6	228.13E+3	7.64E+6
6 500	340.32E+6	1.26E+9	-1.72E+9	2.48E+9	2.35E+6	1.28E+6	388.76E+3	7.29E+6
7 000	277.51E+6	1.34E+9	-1.74E+9	2.59E+9	2.23E+6	1.53E+6	663.59E+3	6.95E+6
7 500	277.77E+6	1.41E+9	-1.75E+9	2.63E+9	2.13E+6	1.50E+6	672.00E+3	6.66E+6
8 000	280.57E+6	1.49E+9	-1.78E+9	2.67E+9	2.05E+6	1.48E+6	679.46E+3	6.42E+6
8 500	247.65E+6	1.59E+9	-1.87E+9	2.75E+9	2.04E+6	1.57E+6	792.29E+3	6.41E+6
9 000	224.88E+6	1.67E+9	-1.93E+9	2.83E+9	2.01E+6	1.60E+6	849.50E+3	6.29E+6
9 500	193.97E+6	1.72E+9	-1.90E+9	2.92E+9	1.88E+6	1.68E+6	961.92E+3	5.94E+6
10 000	182.00E+6	1.79E+9	-1.90E+9	2.98E+9	1.83E+6	1.67E+6	957.51E+3	5.77E+6

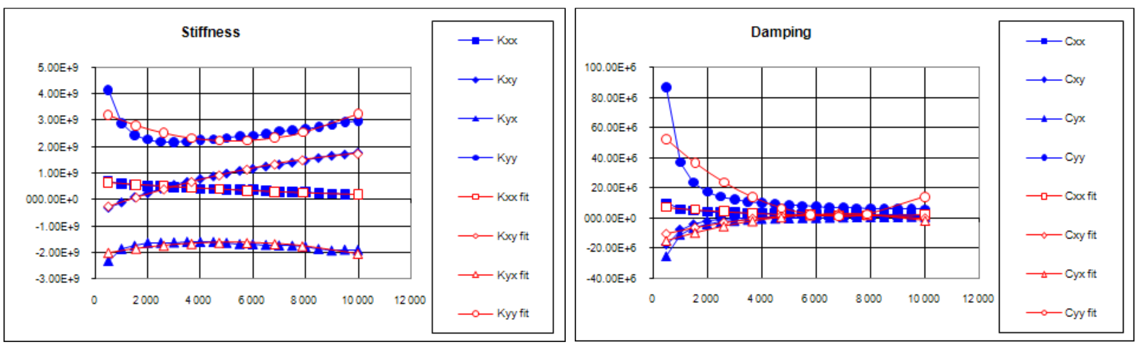


Fig. 21 Diagrams of stiffness and damping change
The curves with the subscript "fit" are the approximating second-order ones

Generator rotor analysis results

Fig. 22 presents the natural frequencies map of generator rotor calculated up to 10000 rpm.

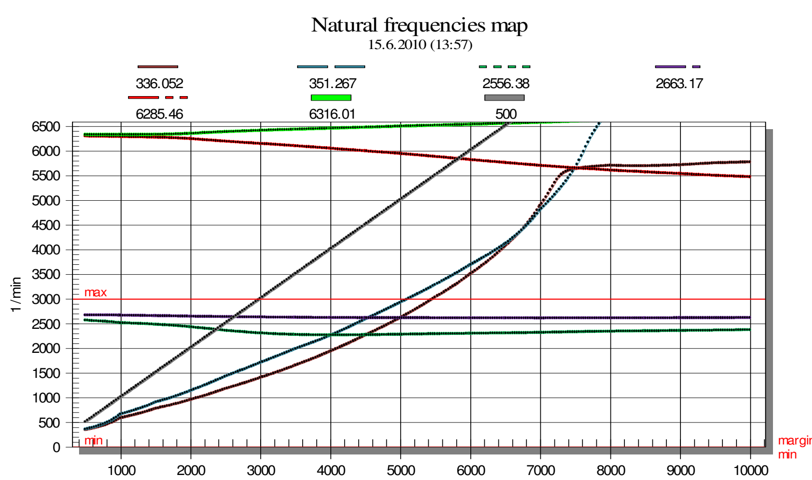


Fig. 22 Natural frequencies map of rotor on elliptical bearings

The map in Fig. 23 shows that the rotor generator loses its stability at the frequency of 5915 rpm (intersection of the damping ratio curve with the X-axis).

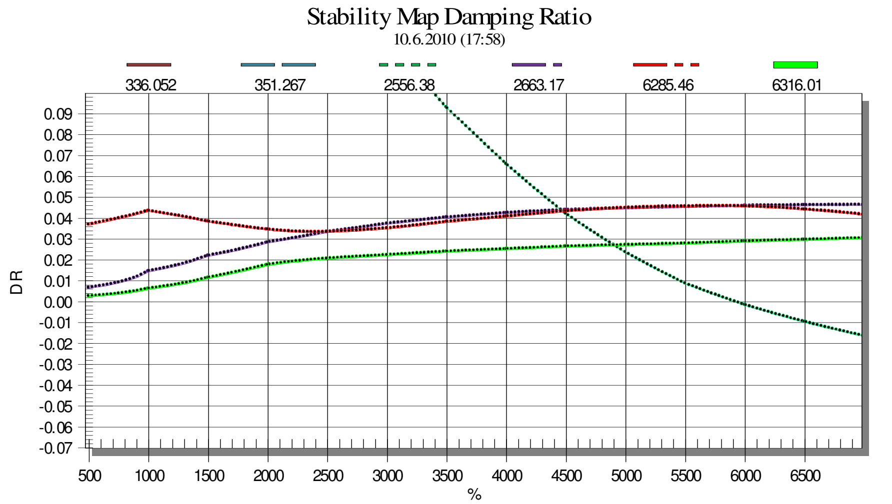


Fig. 23 Damping ratios versus rotating speed

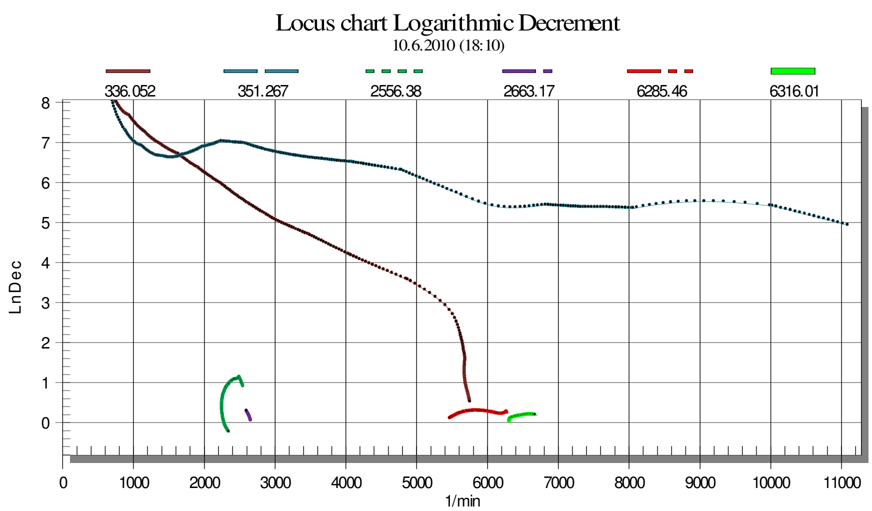


Fig. 24 Dependence map of logarithmic decrements on rotor natural frequencies

Fig. 25 presents the natural frequencies calculated at this regime. It highlights the frequency and mode shape of oscillations where the stability loss happens. The same frequency could be determined on the map at Fig. 24.

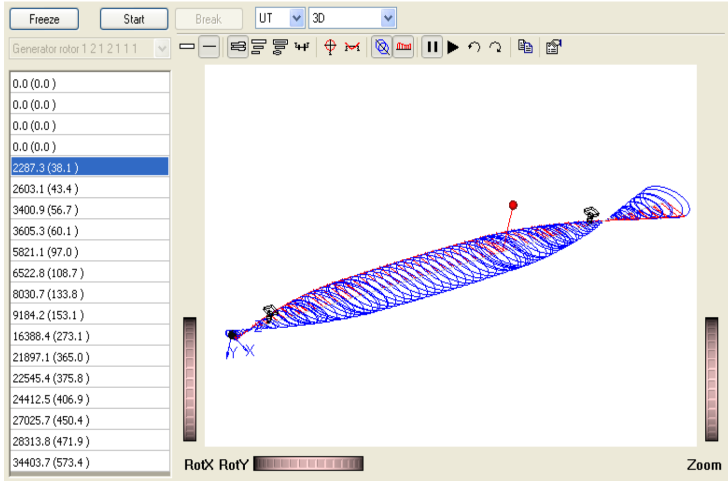


Fig. 25 Natural frequencies at 5915 rpm and mode shape

Oil inlet pressure influence on journal bearings

If changing oil inlet pressure from 2 bar to 0.3 bar at the operating mode the stiffness and damping bearing characteristics are not virtually changed, Table 9, 10.

Table 9

Operating mode, pressure 2 bar

X Ext Load	Y Ext Load	Speed	Kxx	Kxy	Kyx	Kyy	Cxx	Cxy	Cyx	Cyy
N	N	rpm	N/m	N/m	N/m	N/m	N-s/m	N-s/m	N-s/m	N-s/m
		3 000	504.47E+6	33.40E+6	-1.30E+9	1.42E+9	1.71E+6	-2.14E+6	-2.76E+6	8.11E+6

Table 10

Operating mode, pressure 0.3 bar

X Ext Load	Y Ext Load	Speed	Kxx	Kxy	Kyx	Kyy	Cxx	Cxy	Cyx	Cyy
N	N	rpm	N/m	N/m	N/m	N/m	N-s/m	N-s/m	N-s/m	N-s/m
		3 000	505.02E+6	33.32E+6	-1.30E+9	1.42E+9	1.71E+6	-2.14E+6	-2.77E+6	8.11E+6

The slight change in the damping coefficients does not result in the significant change in the rotor generator stability thresholds.

Conclusions

1. Calculated in the nonlinear transient analysis the stability threshold of the certain generator on the plain bearings is between 4000 rpm and 4200 rpm.

2. Calculated in the quasi-nonlinear analysis according to the adjusted journal bearings model the stability map of certain generator is at the rotor rotating speed at 3895 rpm. The stability loss happens at the rotor mode shape of 1997 rpm.

3. Calculated in the quasi-nonlinear analysis according to the adjusted journal bearings model the stability map of certain generator on the elliptical bearings is at the rotor rotating speed of 5915 rpm. The stability loss happens at the rotor mode shape of 2287 rpm.

Reference

1. Modeling and vibration analysis of the “GT25000 Power turbine – BRUSH generator” power train. Technical report. Prepared for Orenda Aerospace Corporation by Engineering & Consulting Centre for Dynamic Problems in Rotating Machinery “Alfa-Transit Co., Ltd”. CONSULTING AGREEMENT № 1/2010, Stage 1, 2010