



High-Speed Hermetic Turbogenerator with a Hybrid Bearing System

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Abstract

Purpose In the power industry, the most commonly used hydrostatic bearings are usually oil lubricated. On the contrary, for a distributed combined heat and power production, in the working machine—a hermetic high-speed turbogenerator—an oil-free bearing system is required.

Method For this purpose, hydrostatic bearings lubricated with an organic, oil-free working fluid have been designed. Characterized by its limited lift force, hydrostatic bearing system can be used in small-power turbomachinery. To expand the functionality of the application, a new type of bearing—a hybrid bearing—was designed, built, and tested. In this new bearing, both hydrostatic and hydrodynamic effects are combined. In addition, a magnetic thrust bearing has been designed, so that the whole hybrid bearing system is characterized by hydrostatic, hydrodynamic, and magnetic effects.

Results To confirm the good stability of rotor dynamics, numerical calculations and experimental tests have been conducted. The presented design resulted in good bearing lift and reliability of the bearing system.

Keywords Dynamics · Steam turbines · Rotor vibration · Oil-free bearings

Introduction

The most common way to produce electricity is by burning fossil fuels and then to convert the energy to the thermal energy of steam to finally let the steam expand in a steam turbine. Fossil fuels are of great importance, because they can produce significant amounts of energy per unit weight. However, nowadays special interest in the energy production has been paid on the environmental aspects such as reduction of the greenhouse gas emissions. Practical energy efficient solution can be combined heat and power (CHP) cogeneration. In distributed power generation dedicated to CHP systems, microturbines of a power output ranging between 1 and 100 kW and rotational speeds of approximately 15,000–40,000 are applied. A small-power system can be based on the organic rankine cycle (ORC), where the working medium is a low boiling fluid [1]. To maintain the purity of the cycle, we are searching for hermetic machines equipped with bearings lubricated with the working medium.

For microturbines of electrical power output of about 1–3 kW, special gas bearings can be used, [2, 3]. In this study case, the electrical power output of the turbogenerator is up to 80 kW, and for the machine of such size and loading gas, bearings reach performance limits. The bearing system needs to be oil-free liquid lubricated.

General Specifications

The turbogenerator is a hermetic construction, where the generator is integrated with the turbine shaft. Main parameters of the designed machine are as follows:

Inlet pressure at $T_{in} = 248\text{ °C}$, $P_{in} = 7\text{ bar}$.

Outlet pressure, $P_{out} = 0.17\text{ bar}$.

Flow rate, $\dot{m} = 1.57\text{ kg/s}$.

Rotational speed, $\omega = 20000\text{ rpm}$.

Maximum electrical power output, $P_e = 80\text{ kW}$.

Bearing system: hermetic, oil-free.

The turbine working medium is an organic fluid, the siloksan MDM, the properties of which are listed in Table 1.

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Table 1 MDM properties in different temperatures

Temperature [°C]	60	80	90
Density [kg/m ³]	780.86	759.68	749.09
Viscosity [Pa s]	5.24E-04	4.465E-04	4.072E-04

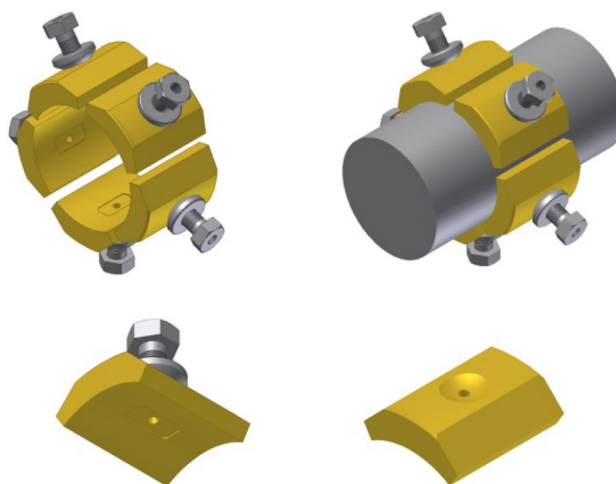
Numerical Calculations

One of the basic problems connected to a practical application of a high-speed machine equipped with non-conventional gas or liquid-lubricated bearings is the machine operational reliability under various working conditions. This requires an adjustment of the machine design in terms of: turbine blade rim, bearing system, and rotor dynamics. The first stage of the machine design should include a mathematical analysis to understand its behavior [4]. This means that a solution to the set of equations needs to be found. The closed form (analytical) solution is difficult or impossible to find for complex or non-linear equations. That is why, numerical methods need to be used to solve them. It is only an approximation, but it can be a very good approximation under certain circumstances.

Bearing System

For the project needs, a hybrid bearing system is designed—two hybrid liquid-lubricated bearings [5] and a thrust magnetic bearing. Two designed journal bearings are not equally loaded and so the drive end (DE or turbine side) bearing and the non-drive end (NDE) bearing are distinguished.

Hybrid liquid-lubricated bearings take maximum advantage of hydrostatic and hydrodynamic effects. These are actually five pads, tilting pads bearings with supply orifices in each pad (see Fig. 1). Five recess are supplied with a liquid lubricant, each through a separate restrictor. The dissipation of energy in the restrictor causes the recess pressure to be lower than the supply pressure. When an external load is applied to the shaft, the thin clearance is reduced in the direction of the applied load. The pressure in the corresponding recess is now higher than in an opposite one. To calculate the magnitude of the applied load which can be supported, it is necessary to establish the pressure in the bearing and then to sum the product of pressure times area over the bearing surface. These and other calculations, [5], determine hydrostatic parameters of bearings (see Tables 2, 3). When the shaft is rotating, the viscosity effects appear in the bearing clearance, where the hydrodynamic pressure field is generated (see Fig. 2).

**Fig. 1** 3D model of the bearing construction**Table 2** Static parameters of the non-drive end (NDE) hybrid bearing

Static load (W_{\max})	32	[N]
Number of pads	5	[-]
Bearing length (L)	50	[mm]
Bearing diameter (D)	58	[mm]
Eccentricity (e)	0.094	[-]
Dynamic stiffness ($K_{XX} \approx K_{YY}$)	6.70E06	[N/m]
Required supply pressure (P_S)	1.52E05	[Pa]
Total flow rate (q_0)	4.84	[l/min]

Table 3 Static parameters of the drive end (DE) hybrid bearing

Static load (W_{\max})	277	[N]
Number of pads	5	[-]
Bearing length (L)	50	[mm]
Bearing diameter (D)	70	[mm]
Eccentricity (e)	0.190	[-]
Dynamic stiffness ($K_{XX} \approx K_{YY}$)	2.35 E07	[N/m]
Required supply pressure (P_S)	4.83 E05	[Pa]
Total flow rate (q_0)	13.12	[l/min]

The variable pressure distribution produces a bearing aerodynamic force, which acts normally to the shaft and it balances the applied load. For a given eccentricity of the journal, the pressure force is dependent on speed, viscosity, and bearing area. The design offers a zero start-up war, good bearing film stiffness at zero speed, and high overload capacity when operating at full speed.

Dynamic properties of hybrid bearings, according to the linear theory, can be characterized by a set of eight coupled dynamic coefficients, determined for a static equilibrium position of the bearing [6]. These coefficients are K_{ij} and B_{ij} ,

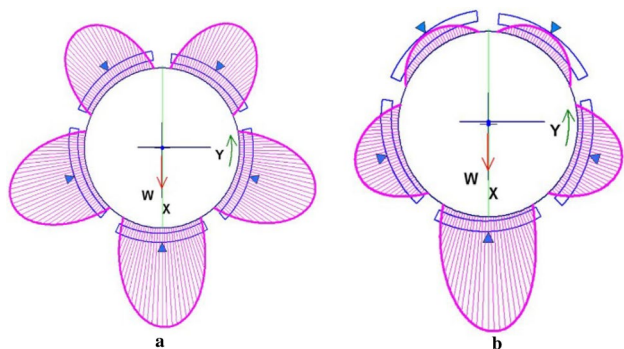


Fig. 2 Pressure distribution in hybrid bearings. **a** NDE bearing and **b** DE bearing

where i and j indicate directions in global coordinate system. They represent a convenient form for the dynamic analysis of the rotor-bearings system, provided that we limit the scale of excitation forces. This is because the linearization is only possible if the basic condition of small displacements around

the equilibrium point is fulfilled. Dynamic coefficients calculated for NDE and DE bearing are given in Eqs. (1) and (2), respectively.

A thrust magnetic bearing has been designed for the maximum lift force of 2500 N. The active electromagnetic system allows adjustable blade tips clearance. The magnetic bearing parts are shown in Fig. 3:

$$K = \begin{bmatrix} 5.171 \times 10^3 & 0 \\ 0 & 5.325 \times 10^3 \end{bmatrix} \text{ N/mm},$$

$$B = \begin{bmatrix} 3.280 & 0 \\ 0 & 3.330 \end{bmatrix} \text{ Ns/mm}$$
(1)

$$K = \begin{bmatrix} 1.001 \times 10^4 & 0 \\ 0 & 1.777 \times 10^4 \end{bmatrix} \text{ N/mm},$$

$$B = \begin{bmatrix} 5.716 & 0 \\ 0 & 7.925 \end{bmatrix} \text{ Ns/mm}.$$
(2)



Fig. 3 Thrust magnetic bearing

Rotor Dynamics

In Fig. 4, the schematic representation of the one-stage, supersonic, impulse turbine rotor is shown. The great advantage of the centrifugal stage is relatively small axial force. The finite-element model used for rotor dynamics' calculations is shown in Fig. 5 and its physical parameters are listed in Table 4.

Based on the model, calculated support reactions for DE and NDE bearing, equal to 277 and 31.5 N, respectively, served to calculate dynamic properties of hybrid bearings.

Fig. 4 Schematic representation of the turbine rotor

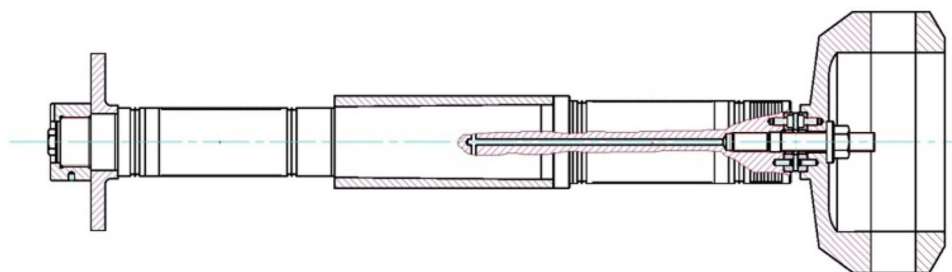


Fig. 5 Numerical model of the turbine rotor

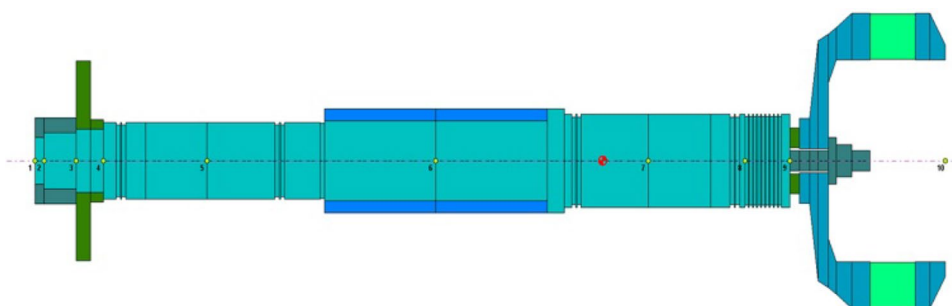


Table 4 Physical parameters of the rotor

Length, [mm]	Masse, [kg]	Gravity center, [mm]	Moment of inertia, [kg m ²]	Moment of inertia (polar), [kg m ²]	Bearings distance, [mm]
762	31.4	475	1.843	0.112	369.5

Then, the dynamic properties, namely stiffness coefficients, have been used in the rotor dynamics model.

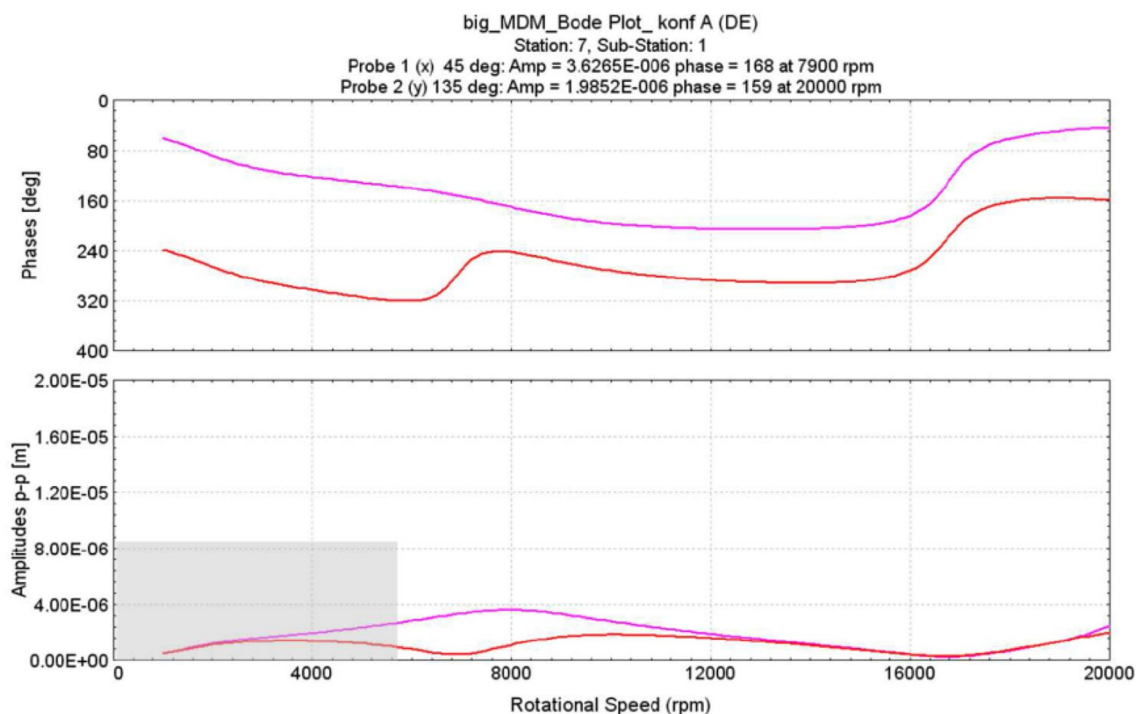
In Figs. 6 and 7, the theoretical Bode plots of the investigated rotating system under the imposed dynamic load are shown. By means of numerical calculations, three modes of vibrations and critical speeds of the rotating system have been identified. The theoretical first critical speed is 7180 rpm. The evolution of relative vibrations in drive end bearing (Fig. 6) and non-drive end bearing (Fig. 7) has been calculated for a maximum unbalance in class G 2.5 (43 g) added near to the gravity center of the analyzed rotor. Gray color indicates experimentally verified rotational range. This is about 30% of maximum admissible rotational speeds and it corresponds to the expected level of vibrations during preliminary experimental tests on the turbogenerator described in the next section.

Experimental Tests of Rotor Dynamics

As it is well known, the particular role in rotor dynamics is played by bearings. Their dynamic properties, damping and stiffness, decide on vibrations amplitudes and modes of rotating system critical frequencies. For the numerical simulations needs, the simplification of small displacements of the shaft center in the closest vicinity of the equilibrium position has been made. Otherwise, the non-linear modeling of bearing dynamic properties would have to be undertaken. Therefore, to confirm the good stability of the rotating system with hybrid bearings, experimental tests have been conducted.

Preliminary tests have been conducted at ambient temperature, and in Fig. 8, the test bench and the hybrid bearing prototype are shown.

The experimental Bode plot obtained during rotor dynamics tests confirms the stability of the machine rotating system during the start-up up to 5300 rpm rotational speed (see Figs. 9, 10). At the drive end (DE), the X component of amplitude of vibrations was about 6 μm and on non-drive end (NDE)—no greater than 16 μm . The measurements have shown that the total volumetric flow of the incoming low-viscosity siloxane MDM through a single hybrid bearing is about 8 lpm when the supply pressure is less than 6 bar.

**Fig. 6** Theoretical Bode plot for DE of the ORC turbogenerator rotor

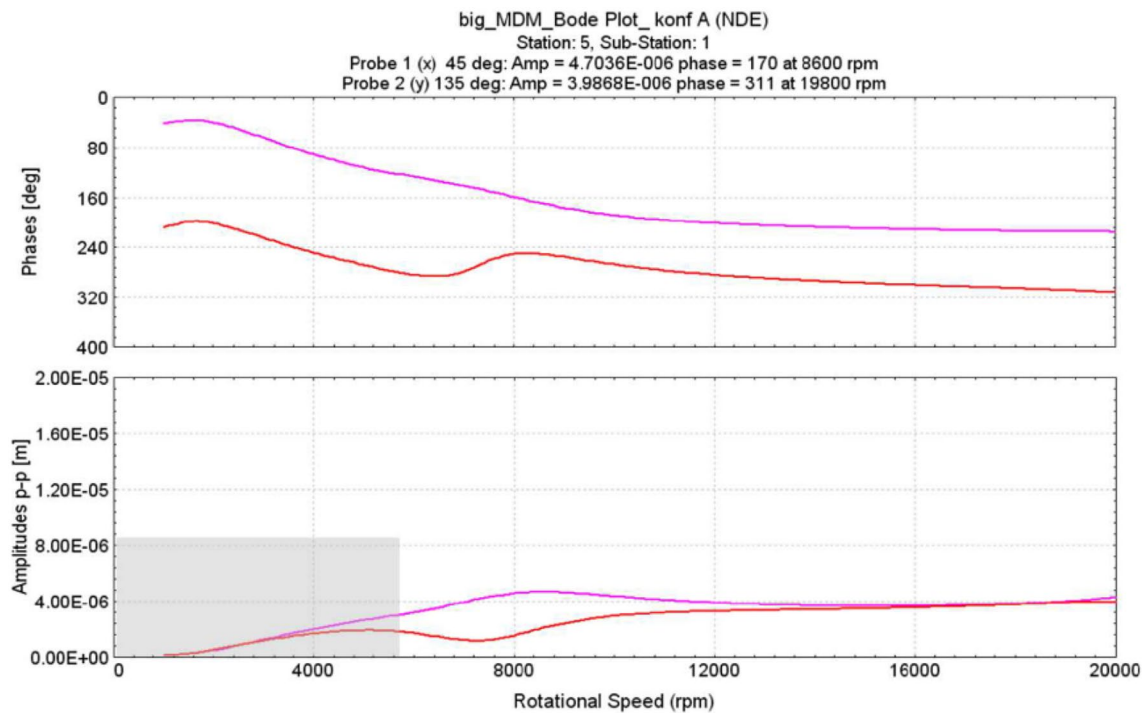


Fig. 7 Theoretical Bode plot for NDE of the ORC turbogenerator rotor

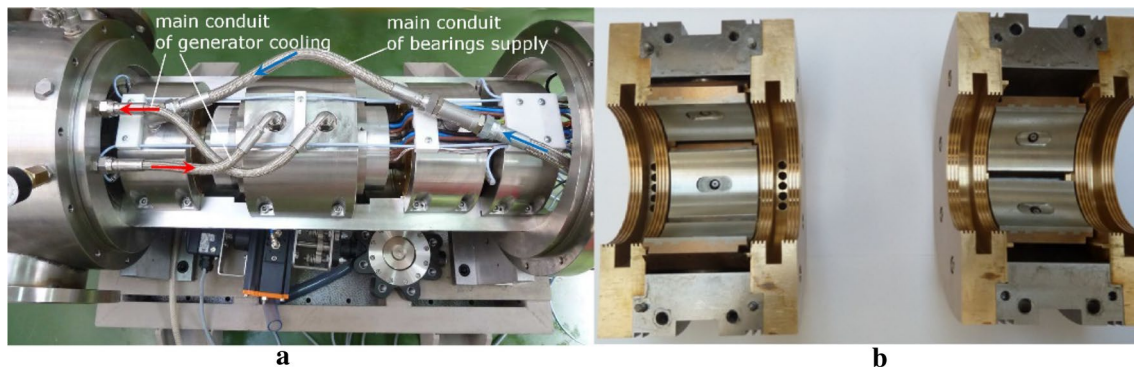


Fig. 8 Turbogenerator test bench: **a** bearing supply system and **b** hybrid bearing surface state after conducted tests

Conclusions

In the study, the rotating system of the ORC turbogenerator equipped with hybrid bearings lubricated with the liquid siloxane MDM has been built and tested under ambient temperature. Achieved results allow drawing the following conclusions and comments.

The machine was tested up to 5370 rpm. The rotational speed was limited by the rate of delivery of compressed air system and not by hybrid bearings performance. More

precisely, the bearing system was designed to support the nominal speed of 20,000 rpm.

Measurements of relative vibrations of the bearings as well as other measurements, namely the phase of vibration or the journal equilibrium position, confirm the stable work in the analyzed range of speeds and loads. The results of the tests are positive, and allow to choose the design concept as a technical solution for the machine construction and to approve the machine for farther tests on the target installation.

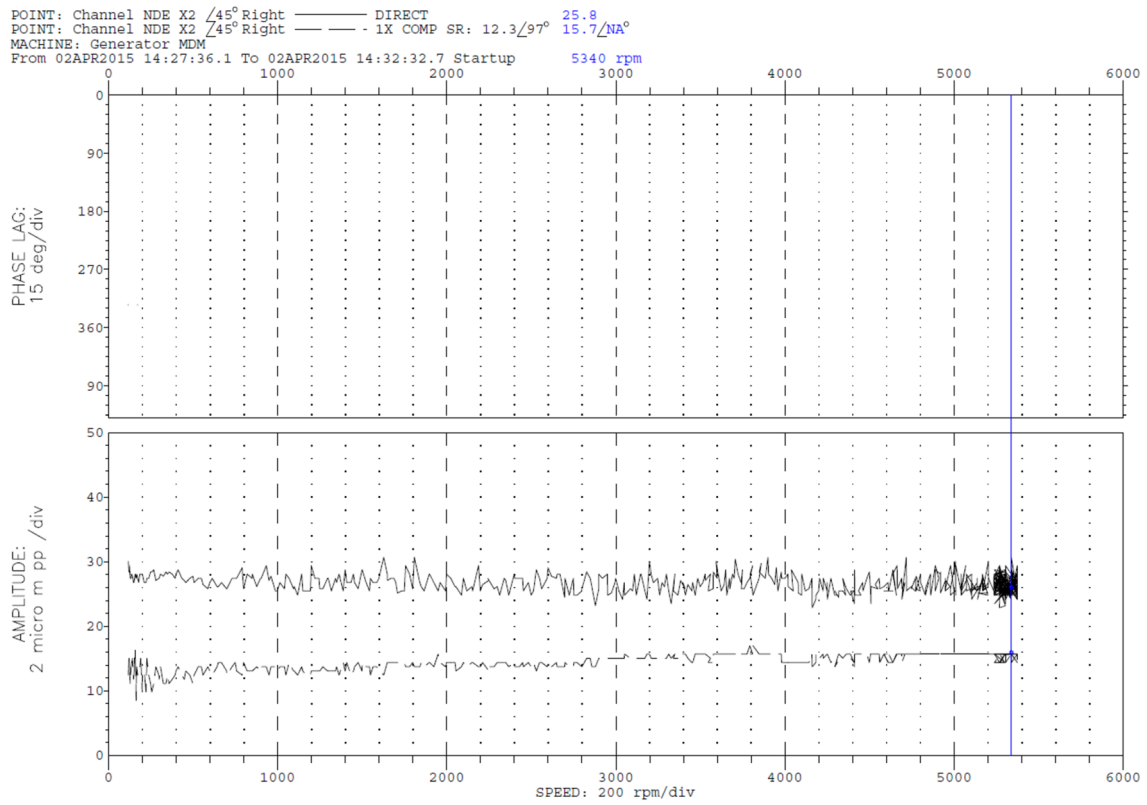


Fig. 9 Amplitude of vibration on NDE bearing during the start-up of the turbogenerator

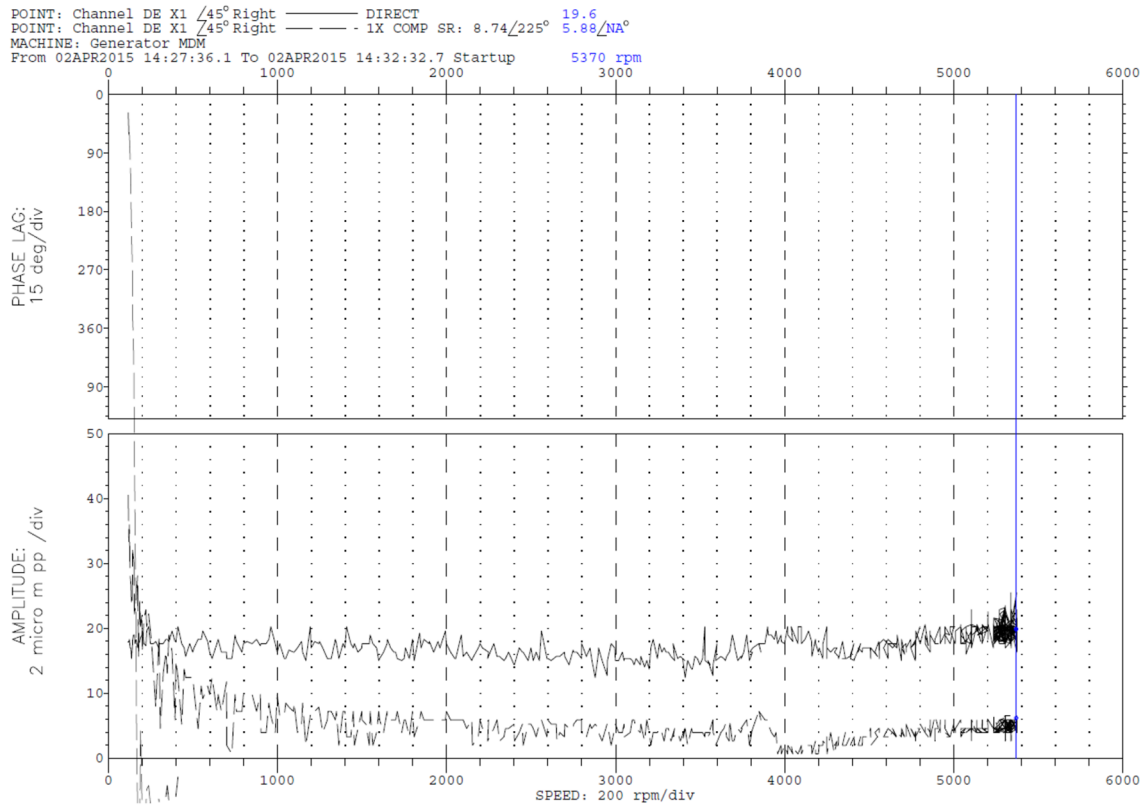


Fig. 10 Amplitude of vibration on DE bearing during the start-up of the turbogenerator

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