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TEXAS A&M UNIVERSITY

TURBOMACHINERY LABORATORY

TEXAS A&M ENGINEERING EXPERIMENT STATION

Successful 2 planes balancing confirm by rotordynamics calculation

Guillaume Christin
Nicolas Peton

Bently Nevada
a Baker Hughes business

Presenter/Author Bios

Guillaume Christin is Machinery Diagnostic Services (MDS) Technical Leader for Southern Europe for BHGE Bently Nevada , located near Lyon in France.. He began his career with GE Power at the heavy gas turbine engineering department in Belfort, France, as mechanical engineer. In 2010, Guillaume joined Bently Nevada as MDS engineer where he acquired deep knowledge in condition monitoring and machinery diagnostics on various machines. He has twelve years of expertise with vibration analysis, balancing, predictive maintenance/monitoring and mechanical system design and analysis. Guillaume holds a degree « Diplôme d’ingénieur » in mechanical engineering with a major in vibration and acoustics from the University of Technology of Compiègne, France.

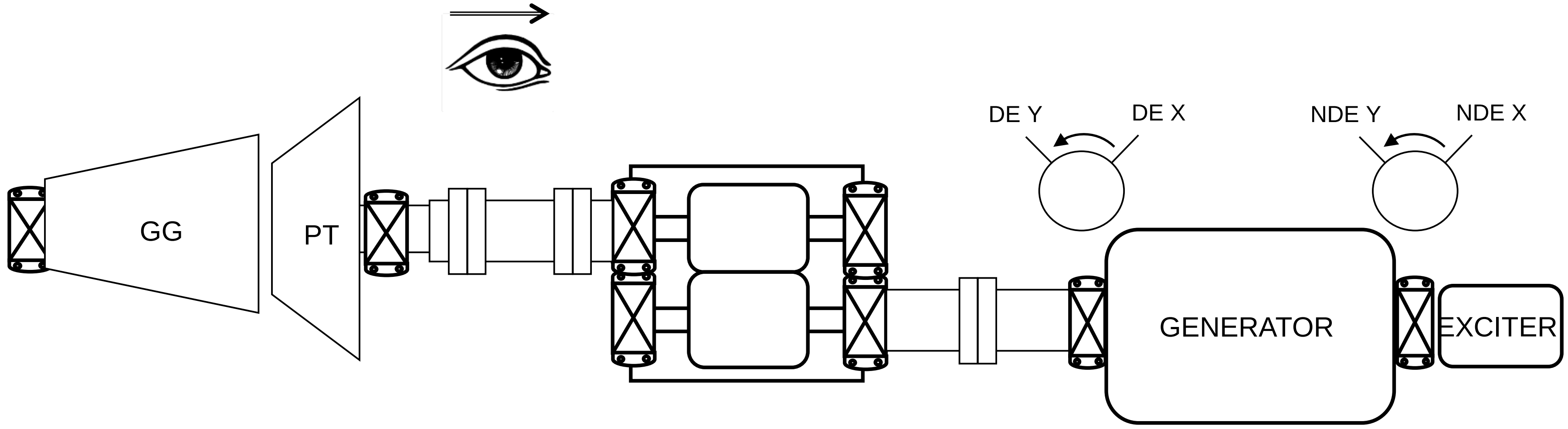
Nicolas Péton joined GE in 2006 in the Machinery Diagnostic Services (MDS) group. He has spent the last 13 years with the MDS team progressing to the Global Director position. Previously he worked for two different manufacturers (Alstom Steam turbine and Cryostar expander/compressor) where he was in charge of onsite startup activities worldwide.

He also worked as an operation and maintenance engineer in the chemical industry (PPG industry, USA) and as Free Lance for startup activities worldwide. He has been also a mechanical/acoustical research engineer in research institutes (Technion, Haifa and TU Berlin). He has a « Diplôme d’ingénieur » from the University of Technology of Compiègne, (1997), France and is a European Professional Engineer (Eur-Ing).

Abstract

This historical case is focus on the vibration behavior of a generator. After a short circuit issue on the grid, the machine train trip. At this time the level of vibration of the generator was acceptable and well below the level of alarm. However, 10 days after the restart of the unit the level of vibration started to increase. The analysis of the data pointed out an increase of the 1X component. There were no signs of rub or other malfunctions. The analysis of the data concluded on an increase of unbalance or a decrease of the dynamic stiffness. Even if, It is unusual to observe such increase of vibration on a generator, the full inspection of the generator was rejected. In order to operate, even if the root cause was not identified, it was decided to balance the generator. The balancing activity was a success using a modal method. Since the root cause was not determined and in order to be sure that the dynamic behavior of the generator was as expected compare to its design, a lateral analysis was done on site using basic information available on site. Comparing the vibration data recording on site with the result of the lateral analysis, it was concluded that the dynamic stiffness of the machine was almost as expected. The increase of vibration could have been due to a real modification of the unbalance or a potential increase of the bearing clearances. The customer could be confident to continue to operate the unit until the next overall.

Machine train diagram



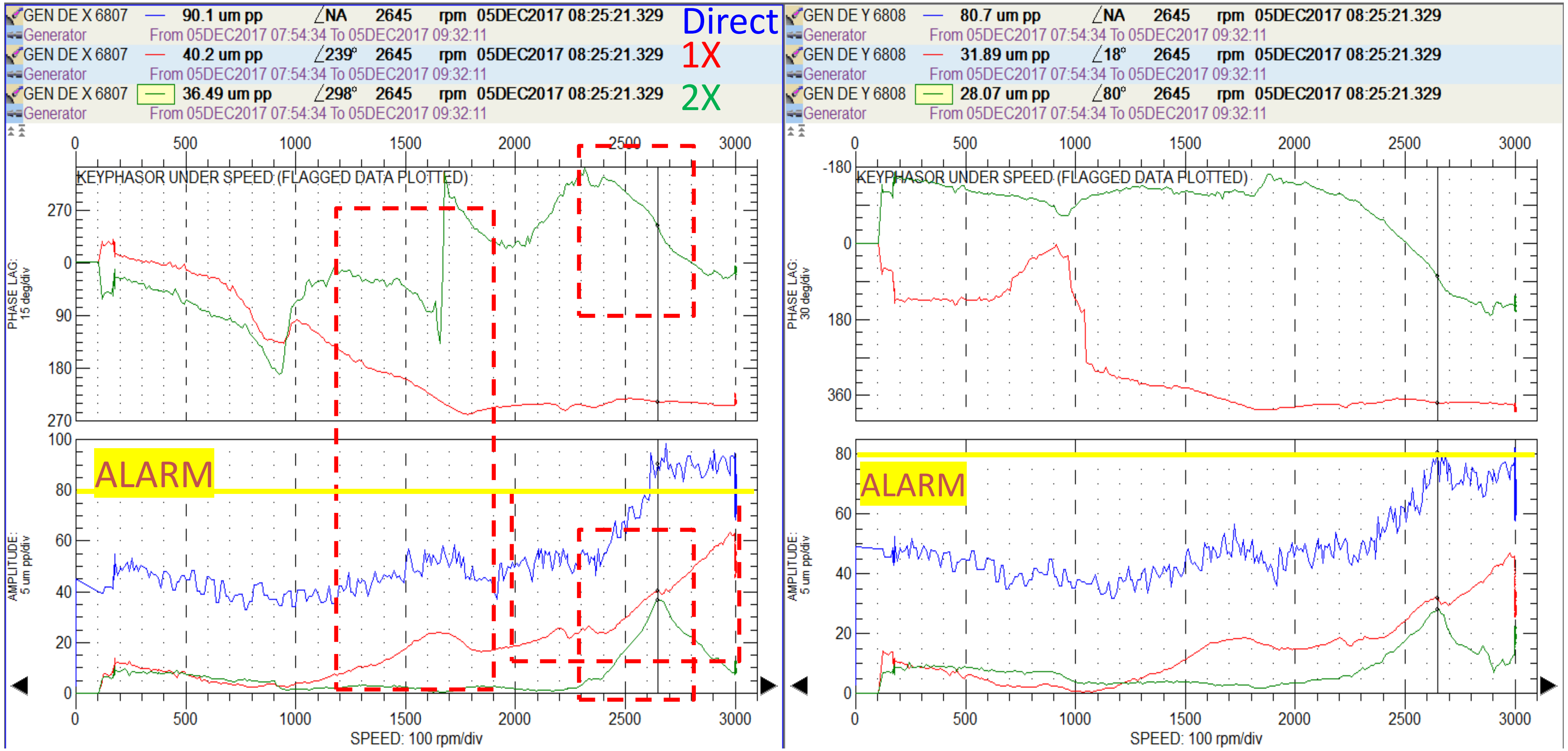
Machine train

- Aero gas turbine
- Gearbox
- Generator (48MW)
- 3000rpm

Historical backround

- short circuit issue on the grid and the unit trip
- **10 days later (after restart) the level of vibration of the generator reached the level of Alarm**
- Alignment was checked and found as expected

Analysis of one start up / Bode plots



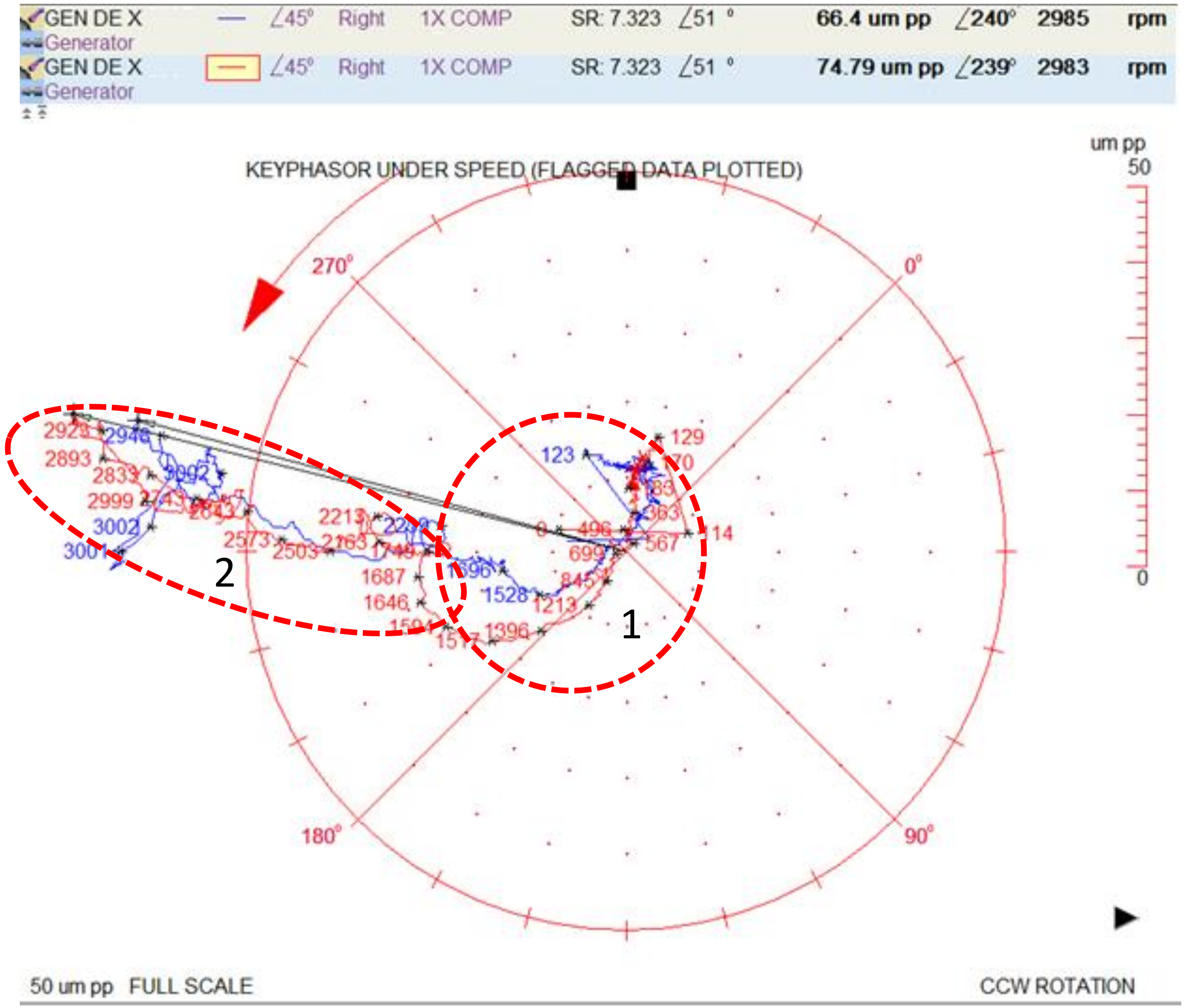
Several run were recorded

- Similar dynamic behavior
- Repeatable behavior
- Thermal transient repeatable
- Shaft movement inside the bearing as expected

Transient information :

- A first critical speed seems to be present around 1700 rpm
- From 2000rpm to 3000rpm the level of 1X increase.
- The 2X response is amplified around 2600rpm indicating that a mode at 5200 cpm may exist.

Analysis of a start up & Shutdown

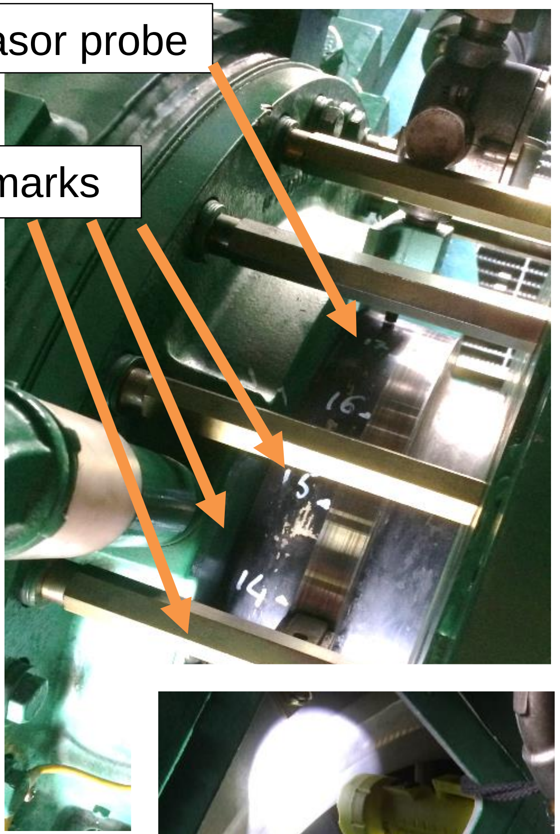


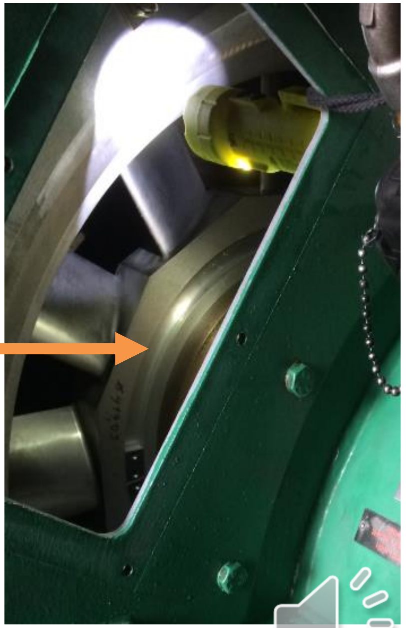
1 - First critical speed

2- Second critical speed ?

Steady State data

- The Vibration are mainly 1X with forward precession
- The orbits are elliptical
- The 1X amplitude increases with the “square” of the speed
- The Dynamic behavior is repeatable from one run to another run
- There is no hysteresis between startup and shutdown
- The 1X vectors of DE and NDE sides are opposite in phase. It most probably due to the influence of a second mode.





The level of vibration is most probably due to an unexpected unbalance.

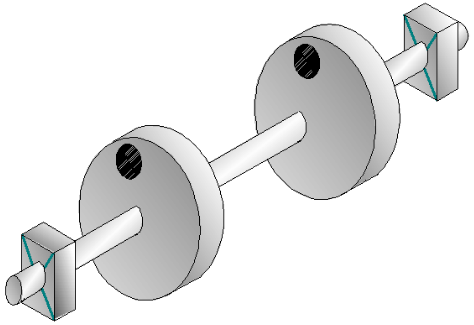
Trim Balance / Modal Method

Basic about modal balancing method (Static-Couple)

To Affect the 1st Mode Only

Place equal weights at the same angular location into the balance plane at each end of the rotor

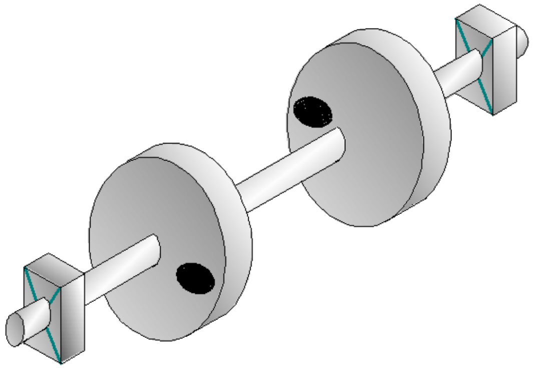
(TRANSLATIONAL)

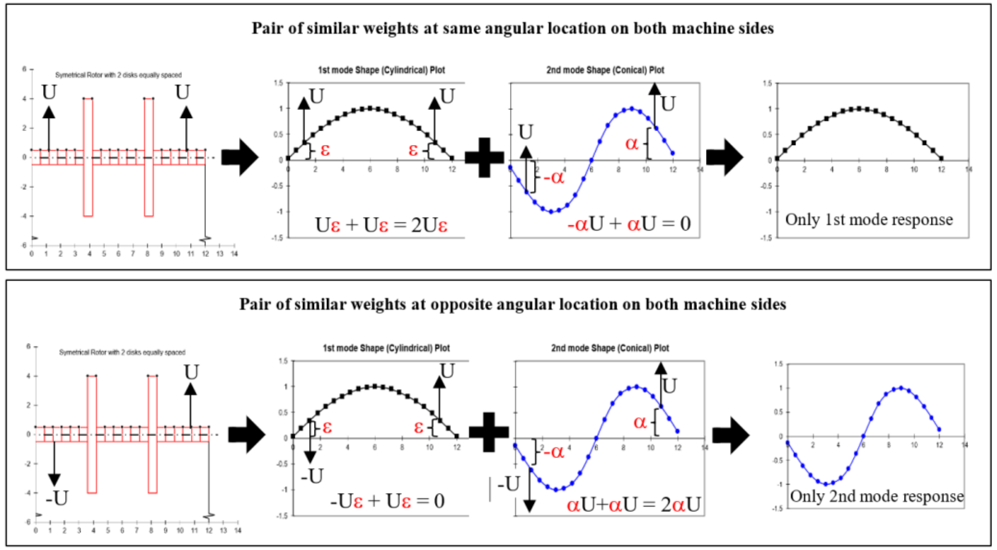


To Affect the 2nd Mode Only

placing a couple (equal weights located 180 degrees apart) into the balance planes at each end of the machine.

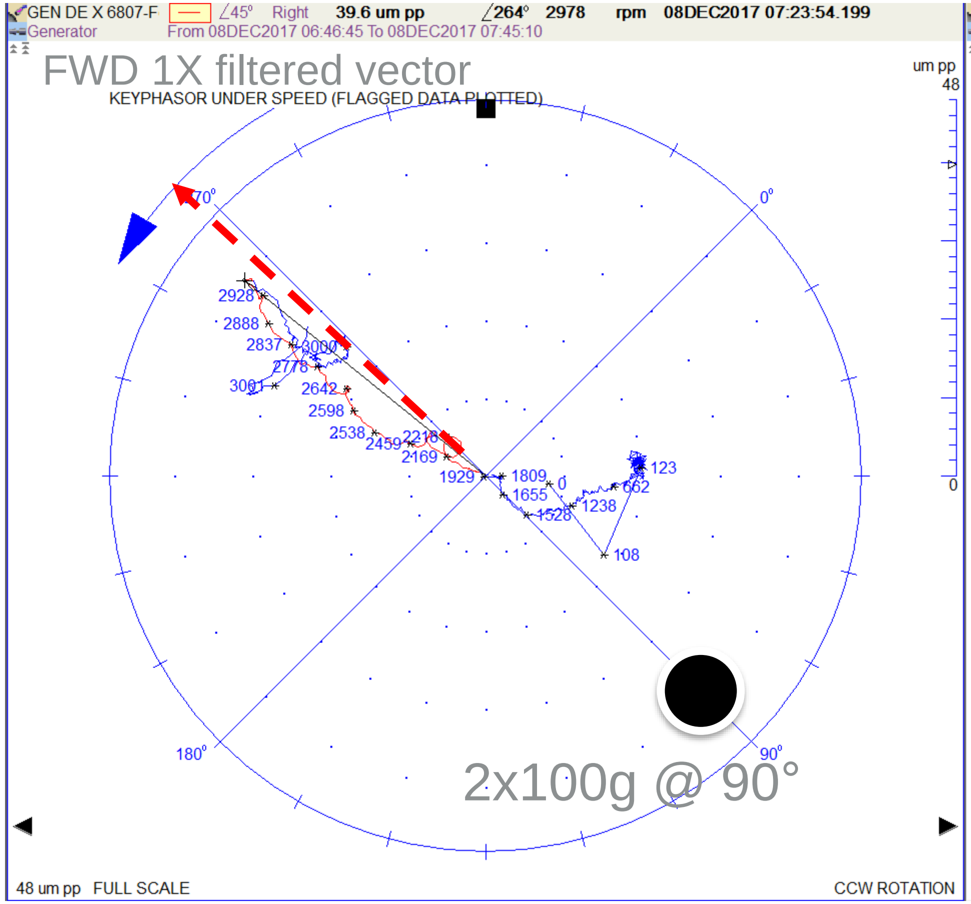
(PIVOTAL)

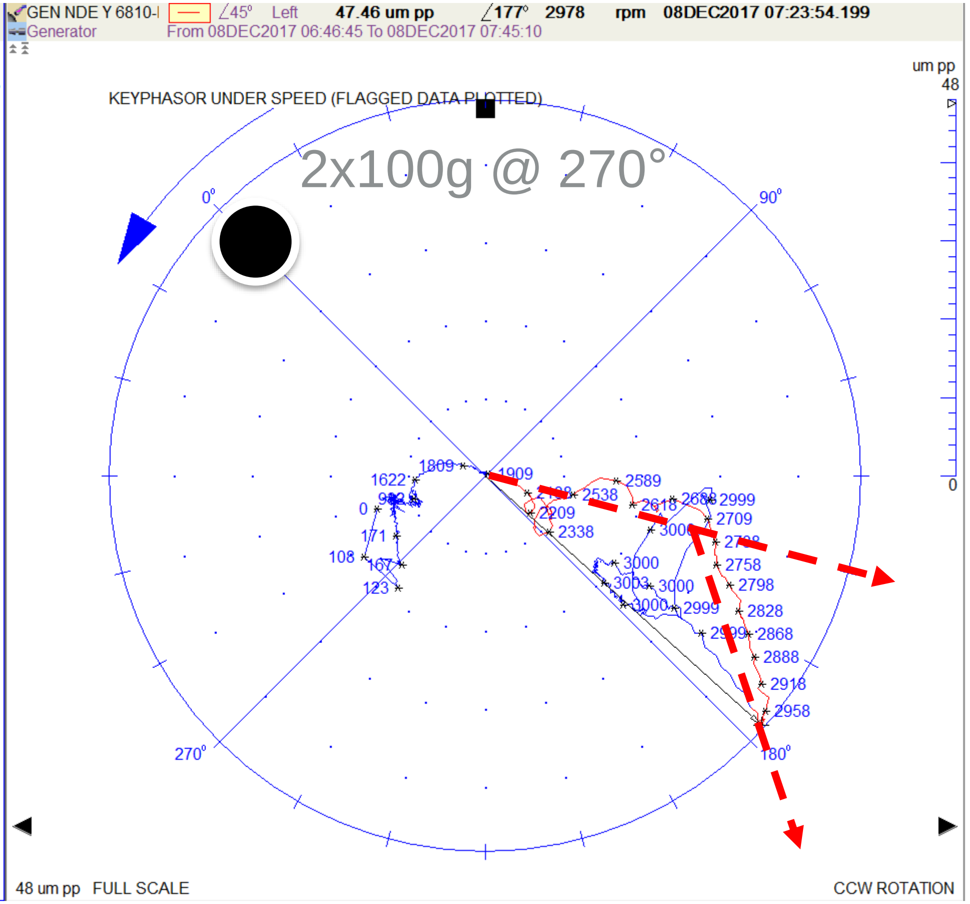




Modes are orthogonal and they can be balanced one by one

Trim Balance / Modal Method





Rotor mass = 14T

Weight radius = 250mm

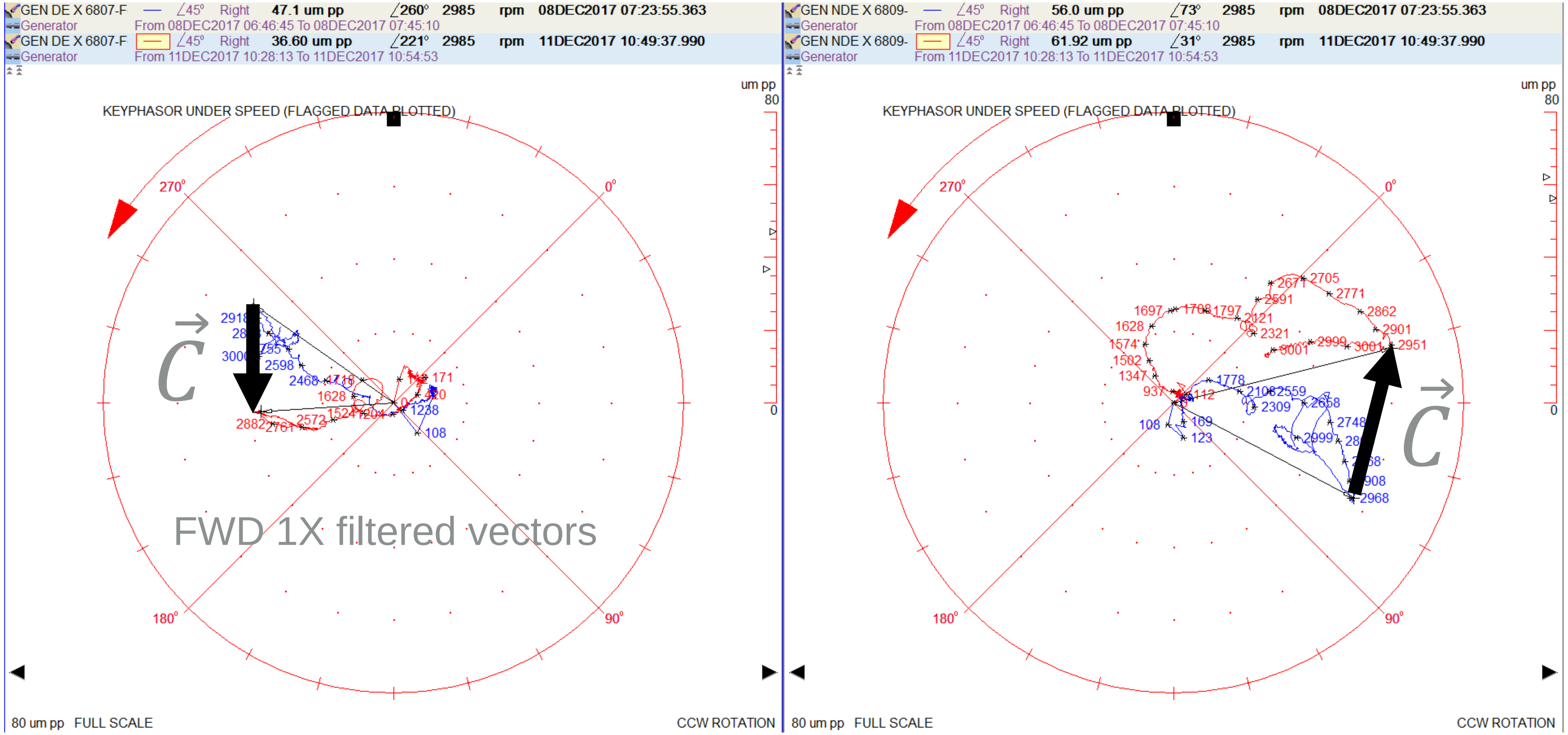
Thumb rule ~10%

Trial weight ~ 550g

Where to install the weights ? Well below a resonance heavy spot and high spot are in phase

Do you think it will be balanced in ONE run ?

Trim Balance / Modal Method

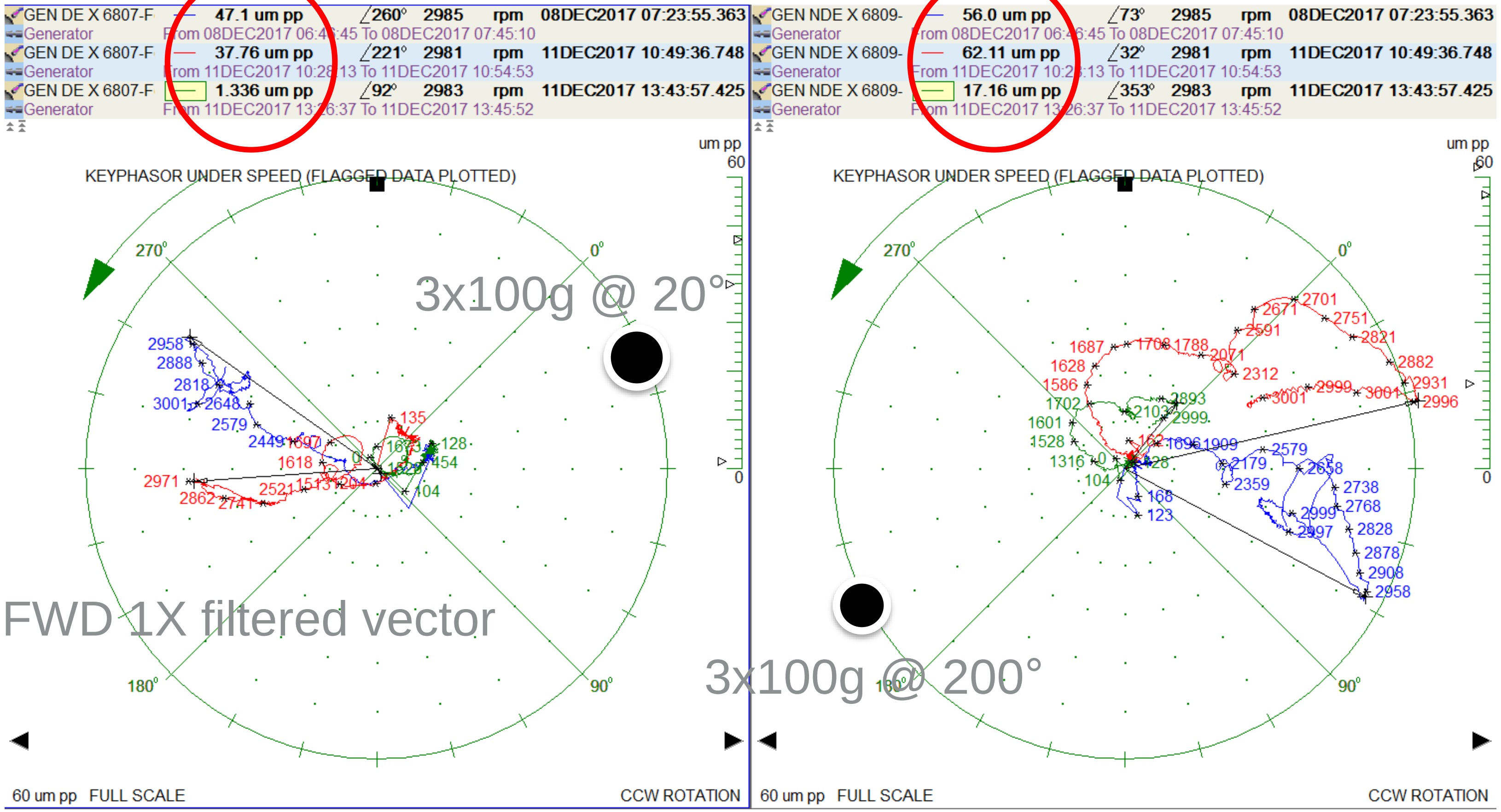


Reference run

Trial Run

$\vec{C}$ vectors is useable but not oriented as expected. It is not opposed at the original vector $\vec{O}$

Trim Balance / Modal Method



Reference run

Trial Run

Final Run

That's was a success !

1X amplitude decrease by more than 70%

However what was "strange" in this balancing job compare to theory ??

Trim Balance

Why it was balanced in TWO runs and not ONE ?

Theory:

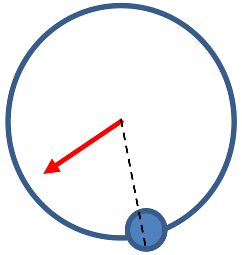
- Well below a resonance the heavy spot and the high spot are in phase.
- At the resonance, the Heavy spot leads by 90° the high spot.

Measurement :

The vibration increased in one direction without an evident phase shift. So it was assumed that at 3000rpm the generator was well below the 2nd critical speed.

Important trim balance remark :

The C vector lags the trial weight by 65°



Theoretically it means that at 3000rpm the generator is running close to a resonance

Should we have balanced this unit ?

Lateral Analysis

Obtain the Rotor geometry

GENERAL PARAMETERS

GENERATOR FRAME SIZE .... BDAX 71-290 ER, BDAX 72-290 ER & BDAX 75-290 ER
EXCITER FRAME SIZE .... BX 10-20
PILOT EXCITER FRAME SIZE .... MX 44-07
SPEED / FREQUENCY .... 3600 RPM / 60 Hz OR 3000 RPM / 50Hz
ROTOR SLOTTING DETAILS TO DRAWING NO. B3162045

ROTOR BODY PARAMETERS

MASS INCLUDING WINDINGS .... = 3414 kg/m LENGTH
2ND MOMENT OF AREA .... = 1.429E+10 mm^4
POLAR MASS MOMENT OF INERTIA .... = 252.1 kg m^2 /m

INERTIAS (Mk^2)

PILOT EXCITER, MAIN EXCITER, DIODE CARRIER ASSEMBLY AND EXCITER SHAFT = 6 kg m^2
MAIN ROTOR INCLUDING WINDINGS, ENDCAPS, COUPLING ETC .... = 973 kg m^2
COMPLETE GENERATOR ROTOR (SUM OF THE ABOVE) .... = 979 kg m^2
NOTE: Mk^2 IS THE POLAR MASS MOMENT OF INERTIA WHERE k IS THE RADIUS OF GYRATION

MASSES

GENERATOR ROTOR ONLY .... = 14054 kg
EXCITER ARMATURE .... = 354 kg
COMPLETE GENERATOR ROTOR (SUM OF THE ABOVE) = 14408 kg

NOTE
1. BC
CO
2. NC

BEARING DETAILS

TYPE.................. = ELLIPTICAL BORE
DIAMETER.................. = 254 mm
DIAMETRAL CLEARANCE = 0.254/0.331 mm
EFFECTIVE LENGTH....... = 180 mm

Basic Bearings data

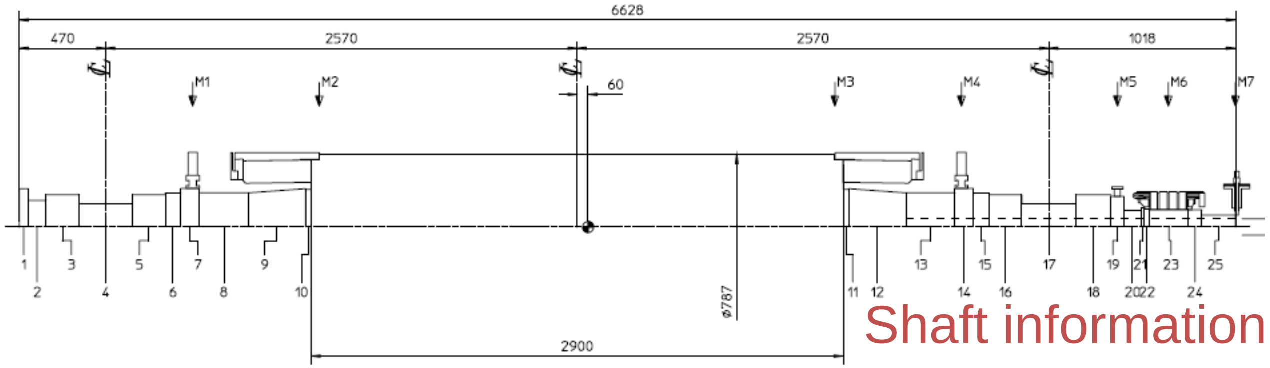
SUPPORT STIFFNESS

HORIZONTAL BEARING SUPPORT STIFFNESS = 500 N/µm
VERTICAL BEARING SUPPORT STIFFNESS ... = 1000 N/µm

ADDITIONAL MASSES AND POLAR INERTIAS

M1: FAN .... MASS = 22 kg INERTIA = 1.6 kg m^2
M2: ROTOR CAP AND ENDWINDING .... MASS = 839 kg INERTIA = 98.1 kg m^2
M3: ROTOR CAP AND ENDWINDING .... MASS = 839 kg INERTIA = 98.1 kg m^2
M4: FAN .... MASS = 22 kg INERTIA = 1.6 kg m^2
M5: PILOT EXCITER ARMATURE .... MASS = 10.5 kg INERTIA = 0.42 kg m^2
M6: EXCITER ARMATURE .... MASS = 147 kg INERTIA = 3.59 kg m^2
M7: DIODE CARRIER ASSEMBLY .... MASS = 38 kg INERTIA = 1.18 kg m^2

Local Masses & Inertias

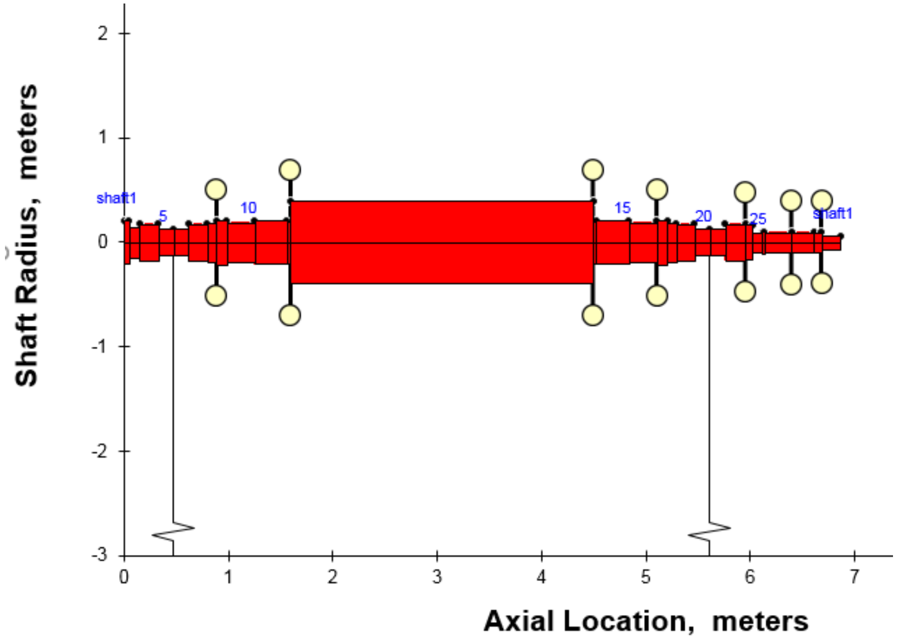


Shaft information

NO.	1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	17	18	19	20	21	22	23	24	25
LENGTH (mm)	48	98	178	292	178	86	100	268	312	30	30	312	268	100	86	178	292	194	70	95	18	26	212	74	183
O.D. (mm)	416	290	350	254	350	370	420	372	416	408	408	416	372	420	370	350	254	350	332	180	210	194	190	186	126

TAPERED TAPER TO (segments 9 and 12)

Create XLTRC² model

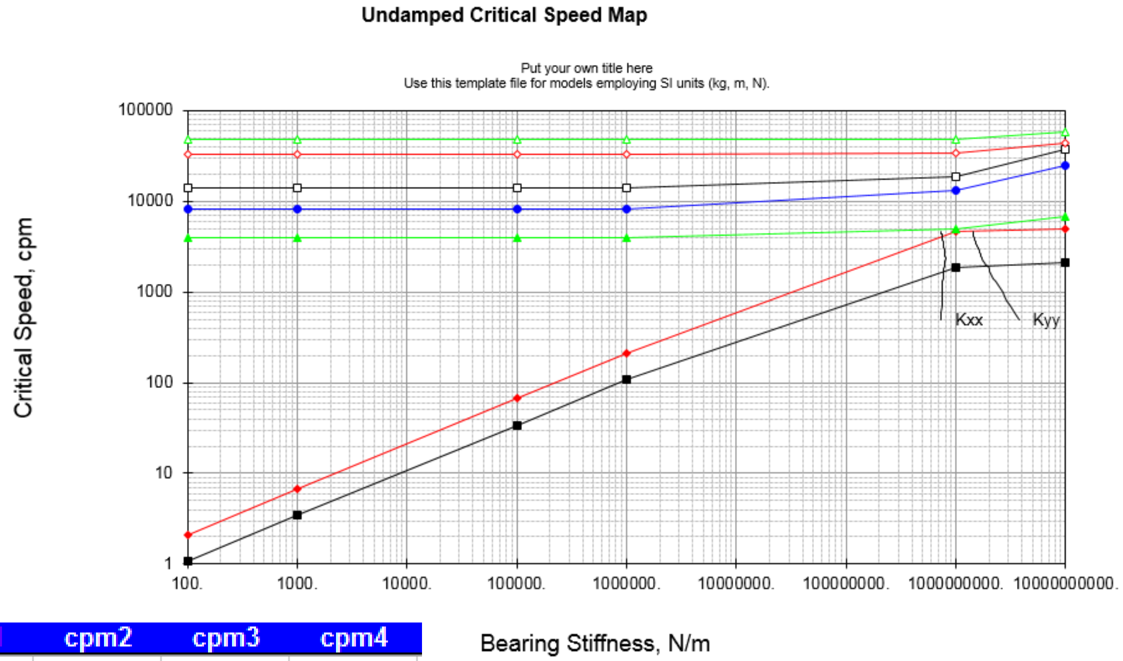


Simple rotor model

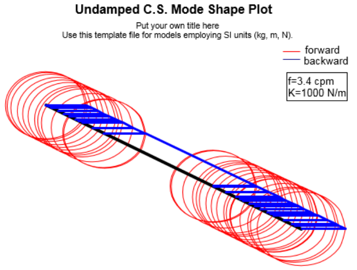
Almost all information are available in this drawing

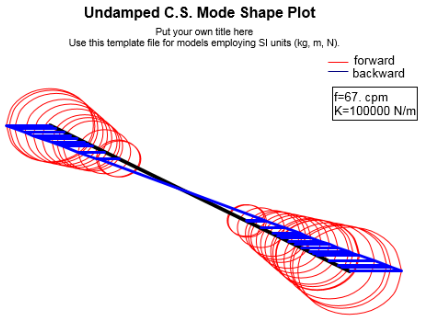
Lateral Analysis

Undamped Critical Speed Map

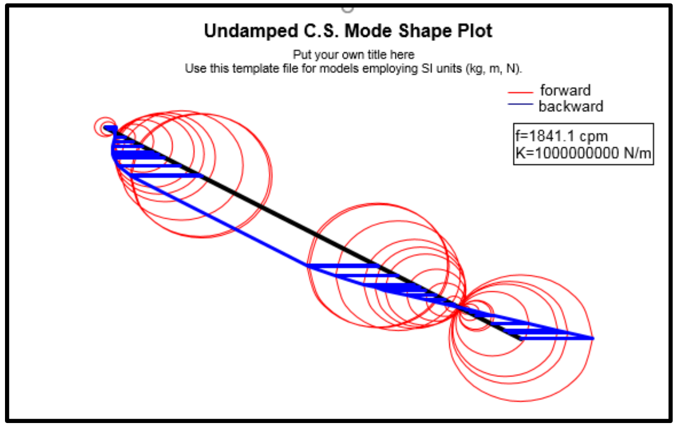


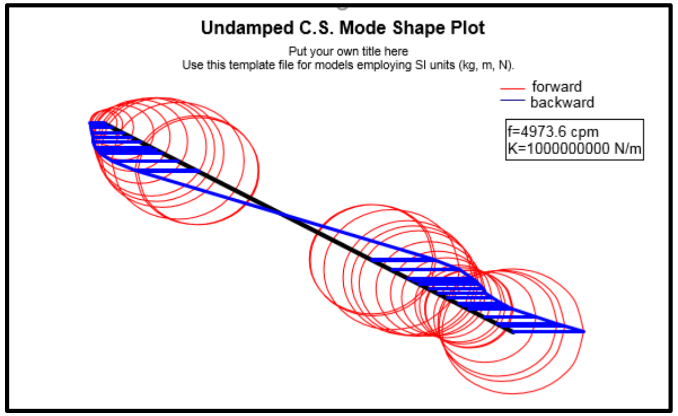
Stiffness	cpm1	cpm2	cpm3	cpm4
1000.	3.4	6.7	4014.7	8123.6





Stiffness	cpm1	cpm2	cpm3	cpm4
1000000000.	1841.1	4693.2	4973.6	12950.8

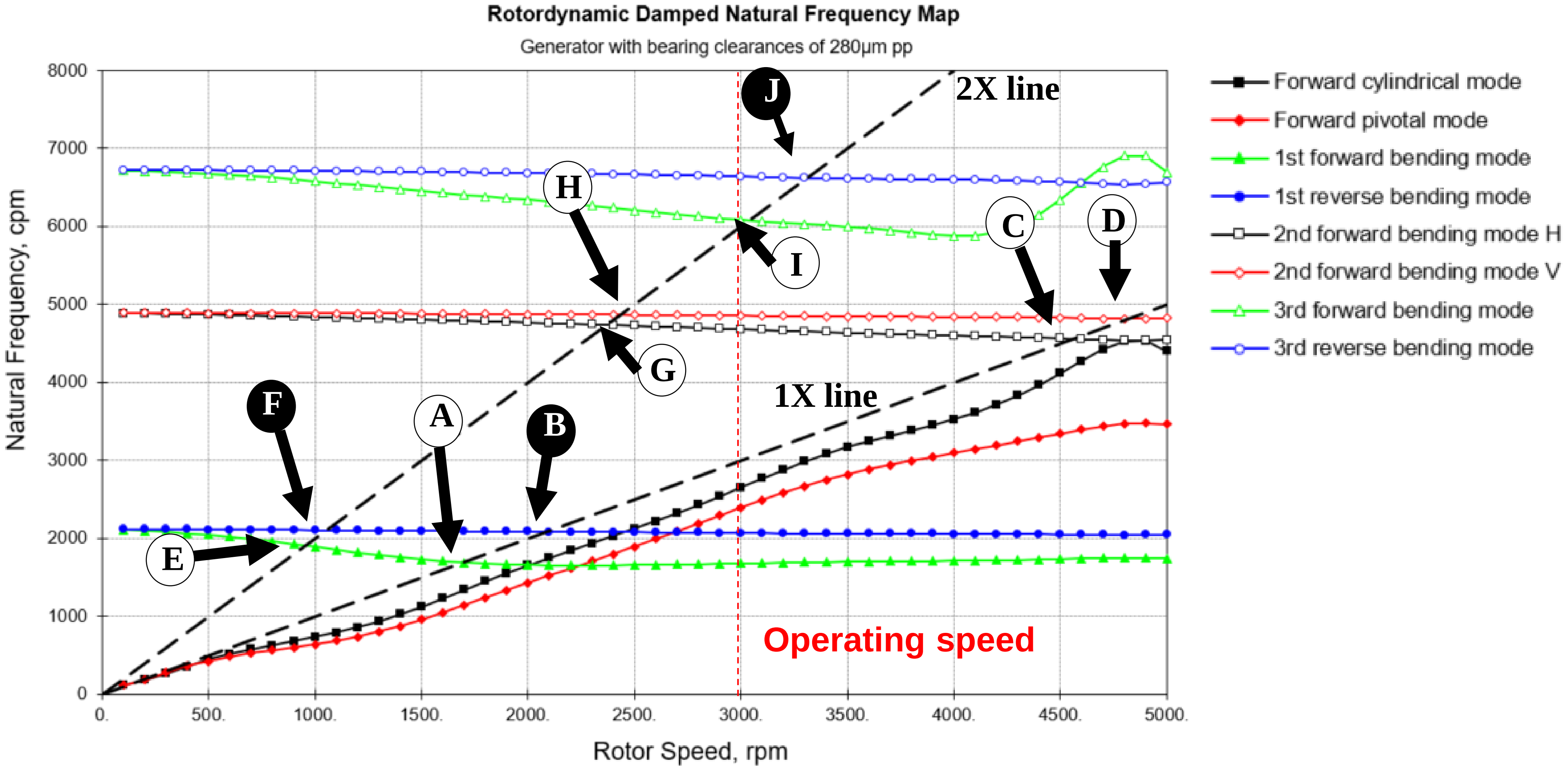




Pivotal mode identified around 4700cpm ….
Relatively far from 3000cpm

Lateral Analysis

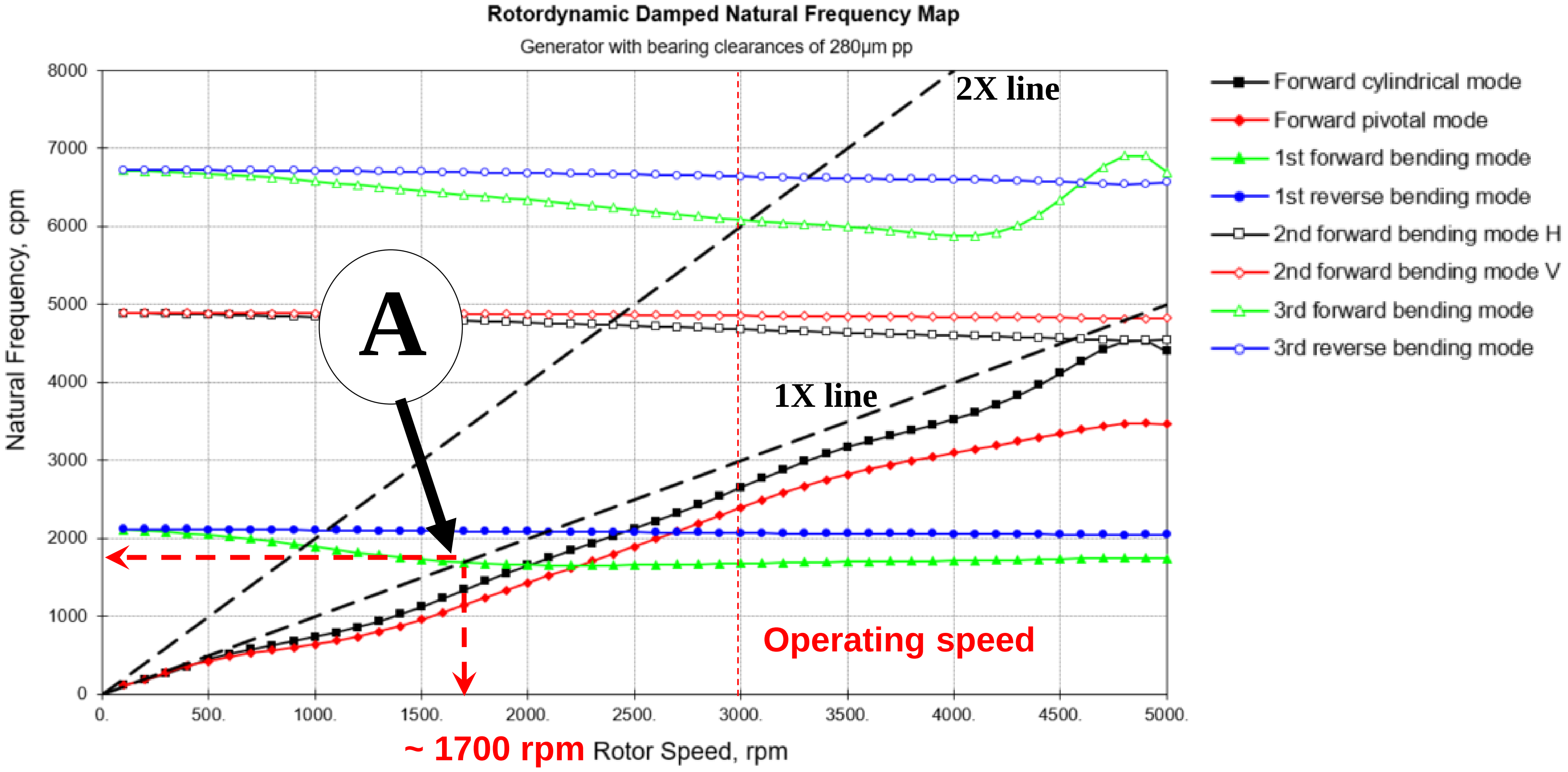
Damped Critical Speed Map



Do those intersection points match with the measurements ?

Lateral Analysis

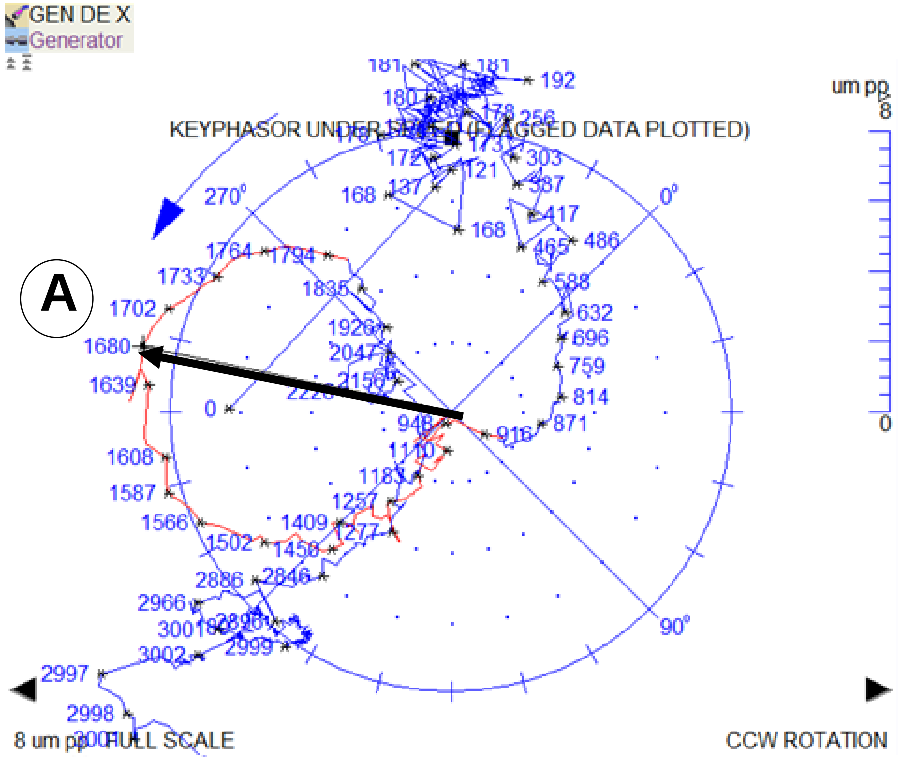
Damped Critical Speed Map

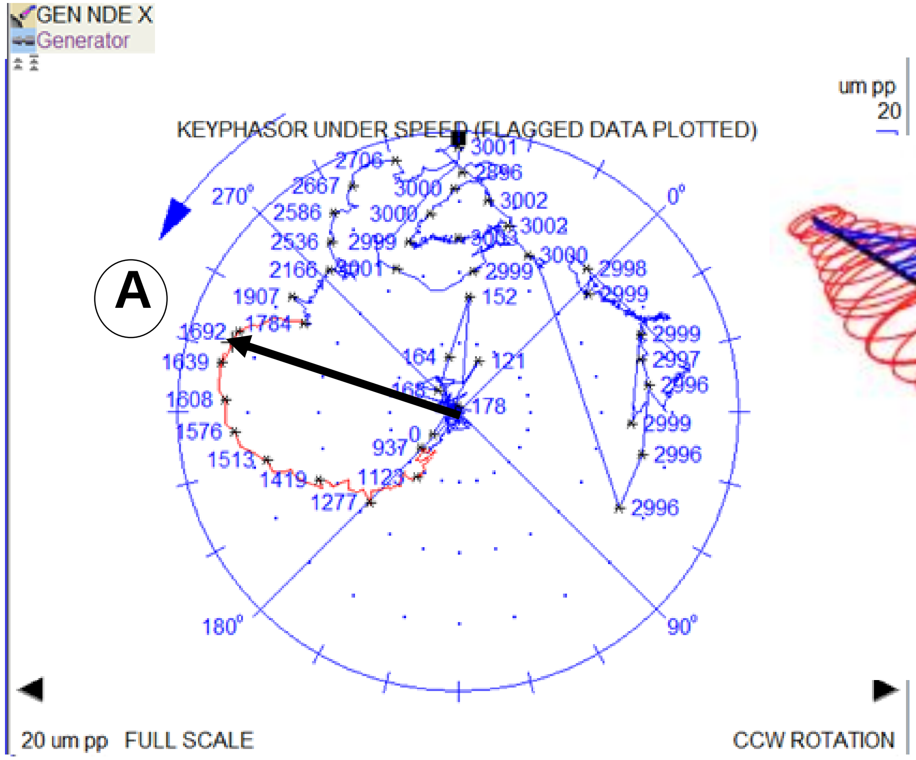


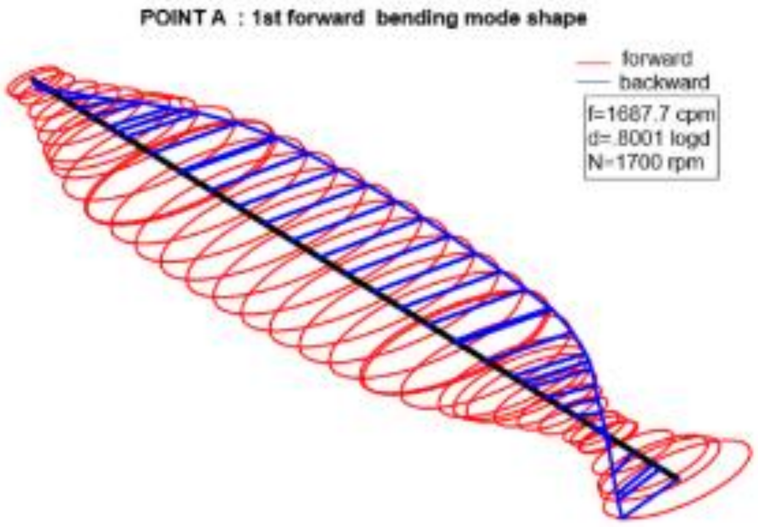
Lateral Analysis

Damped Critical Speed Map

1X FWD polar plots for DE and NDE generator side



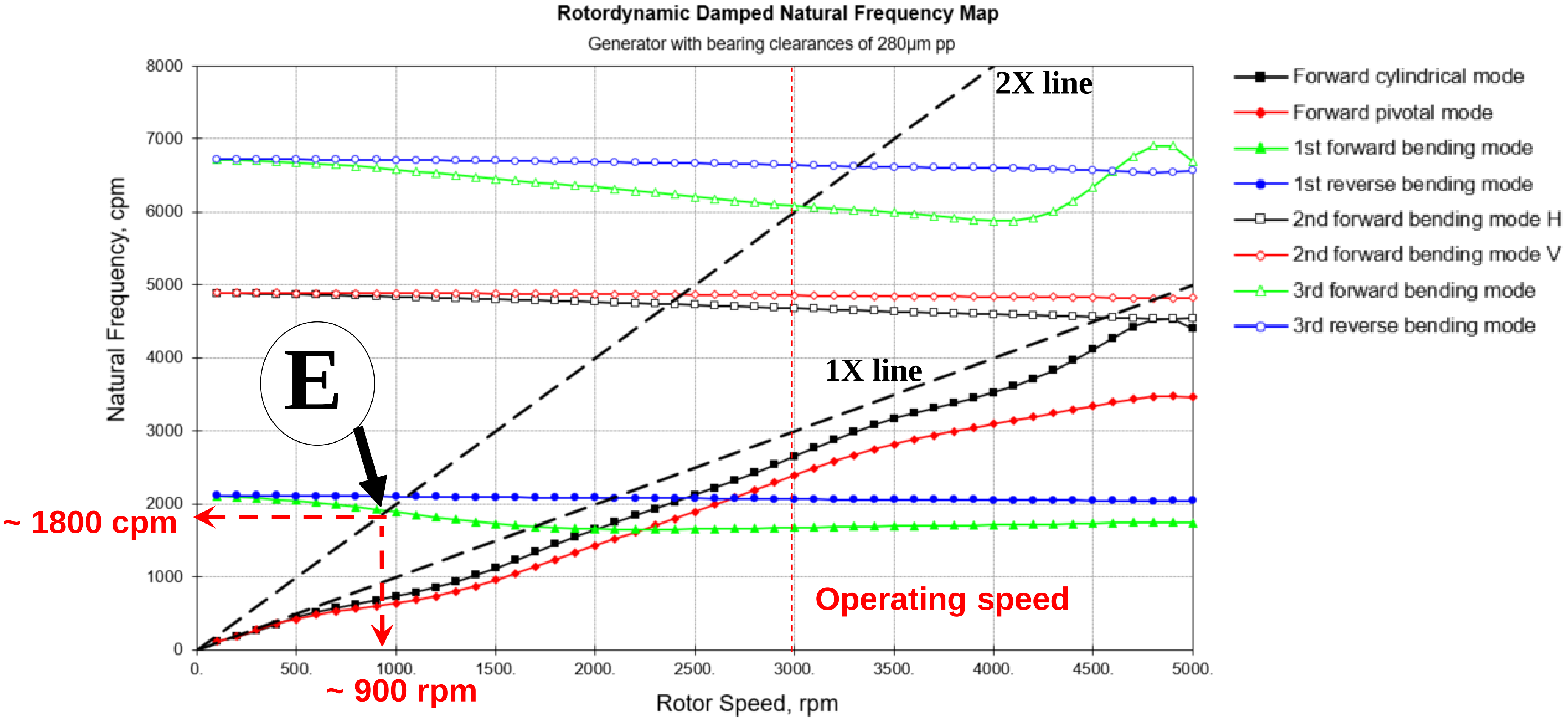




1X DE and 1X NDE vectors are in phase and resonance is around 1680rpm, in accordance with calculation

Lateral Analysis

Damped Critical Speed Map

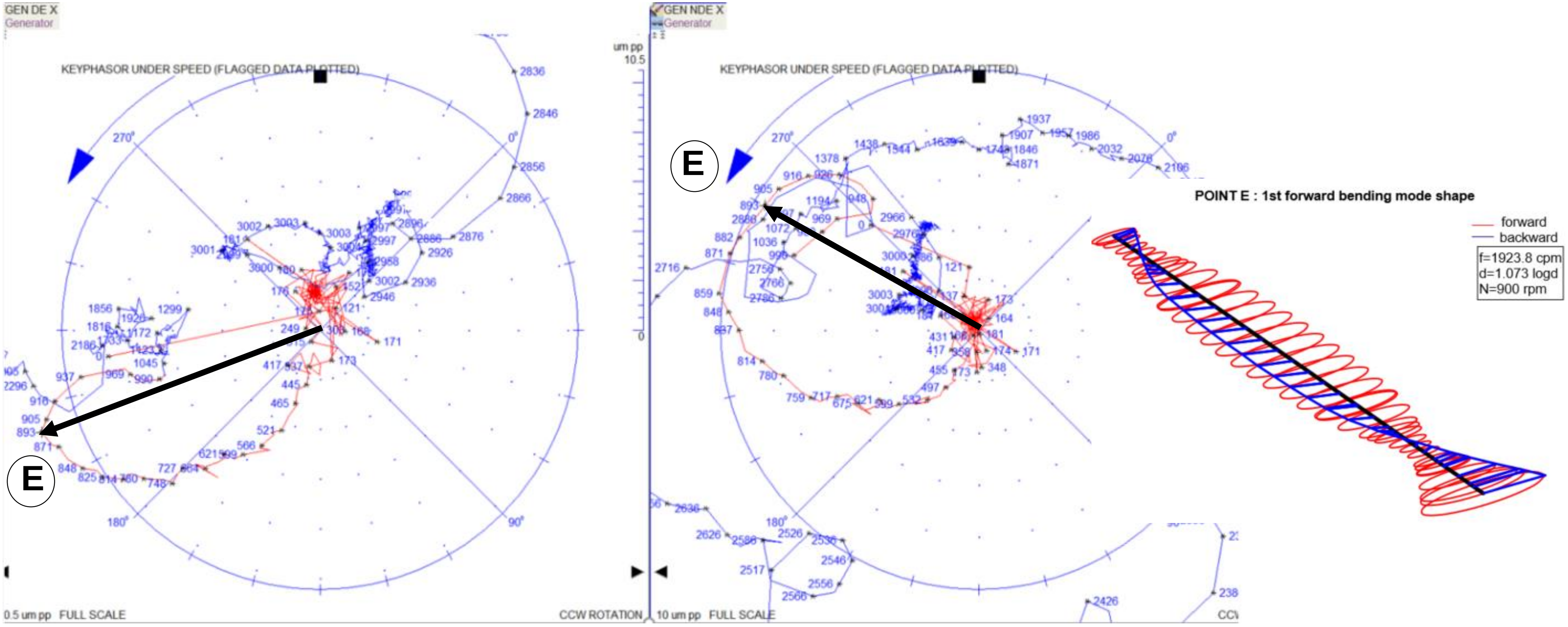


Lateral Analysis

E

Damped Critical Speed Map

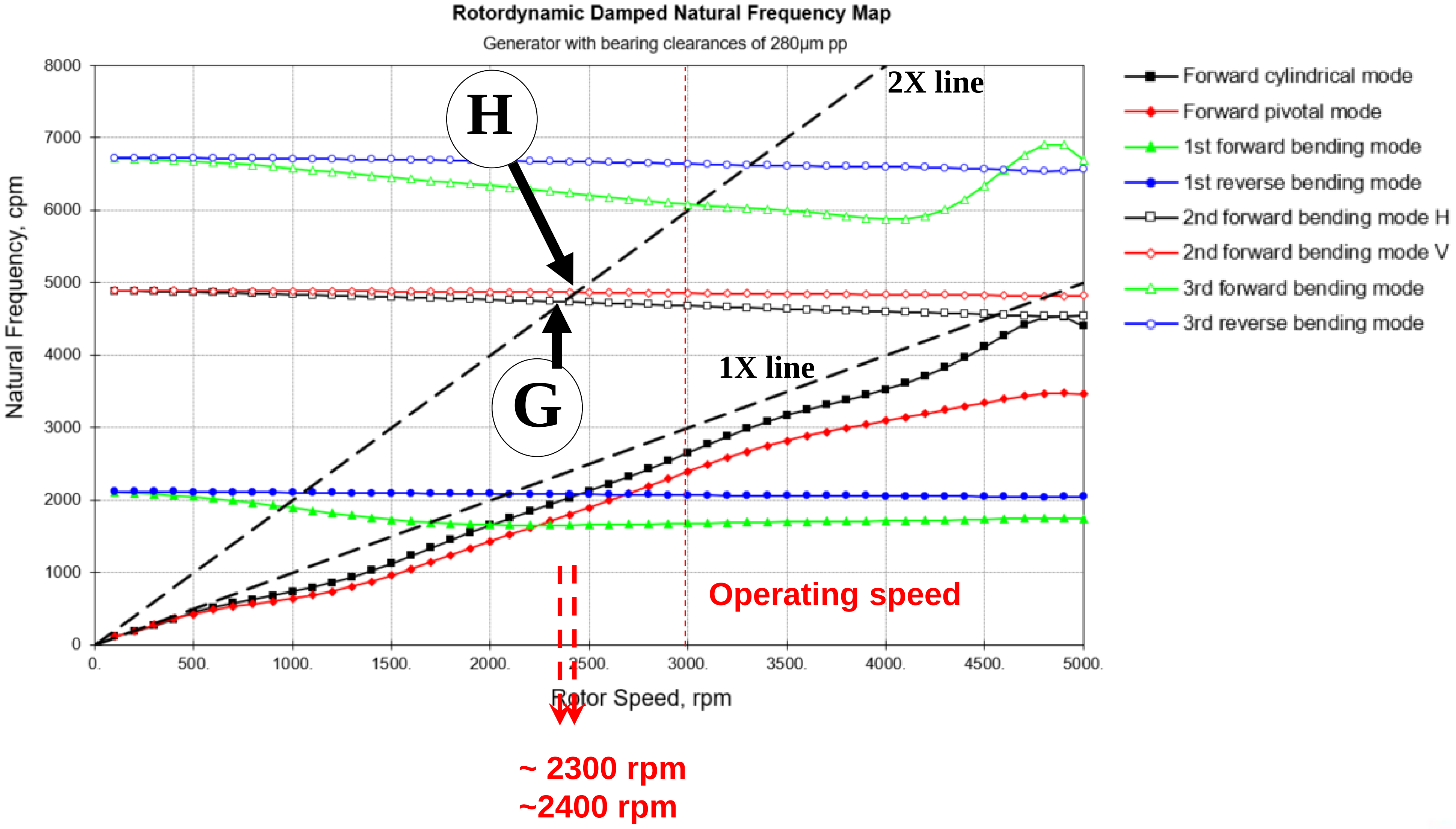
2X FWD polar plot for DE and NDE generator side



2X DE and 2X NDE vectors are almost in phase and resonance is around 900rpm, in accordance with calculation

Lateral Analysis

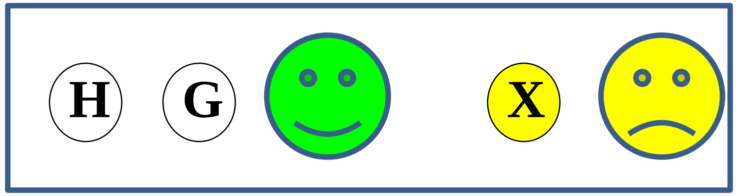
Damped Critical Speed Map

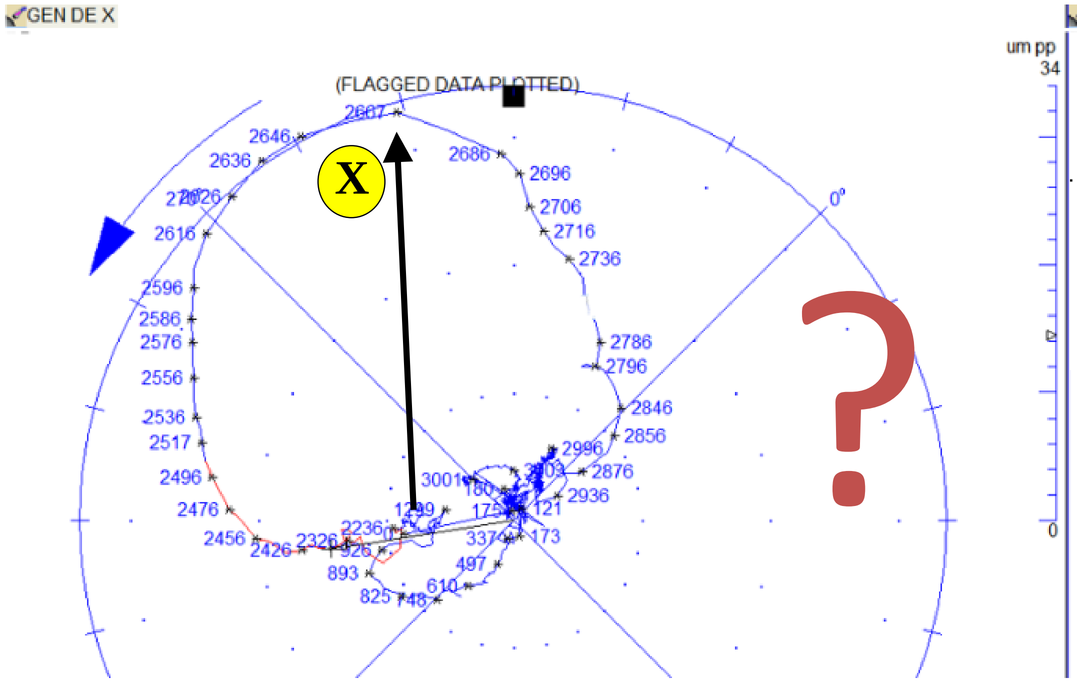


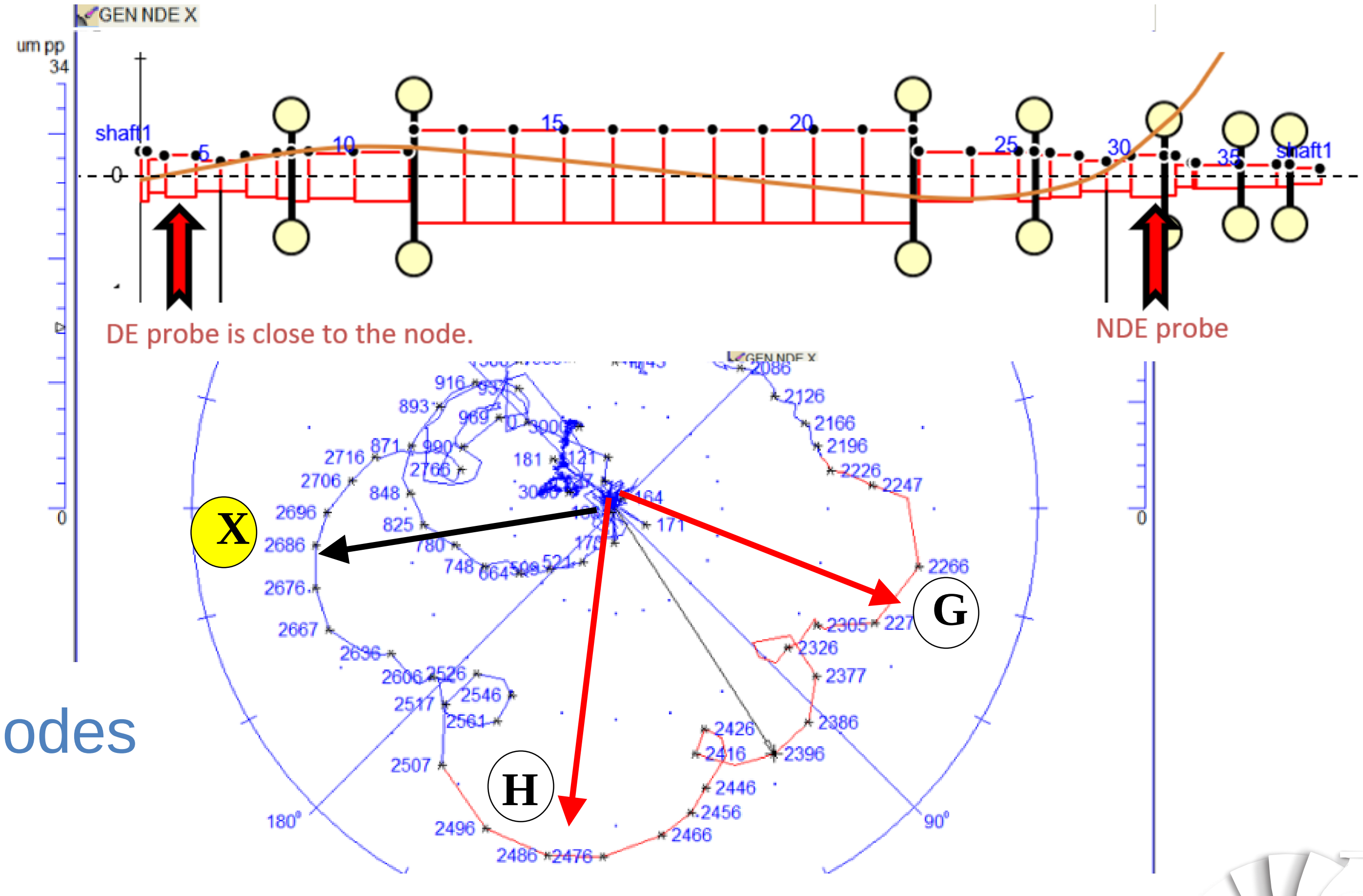
Lateral Analysis

Damped Critical Speed Map

2X FWD polar plot for DE and NDE generator side





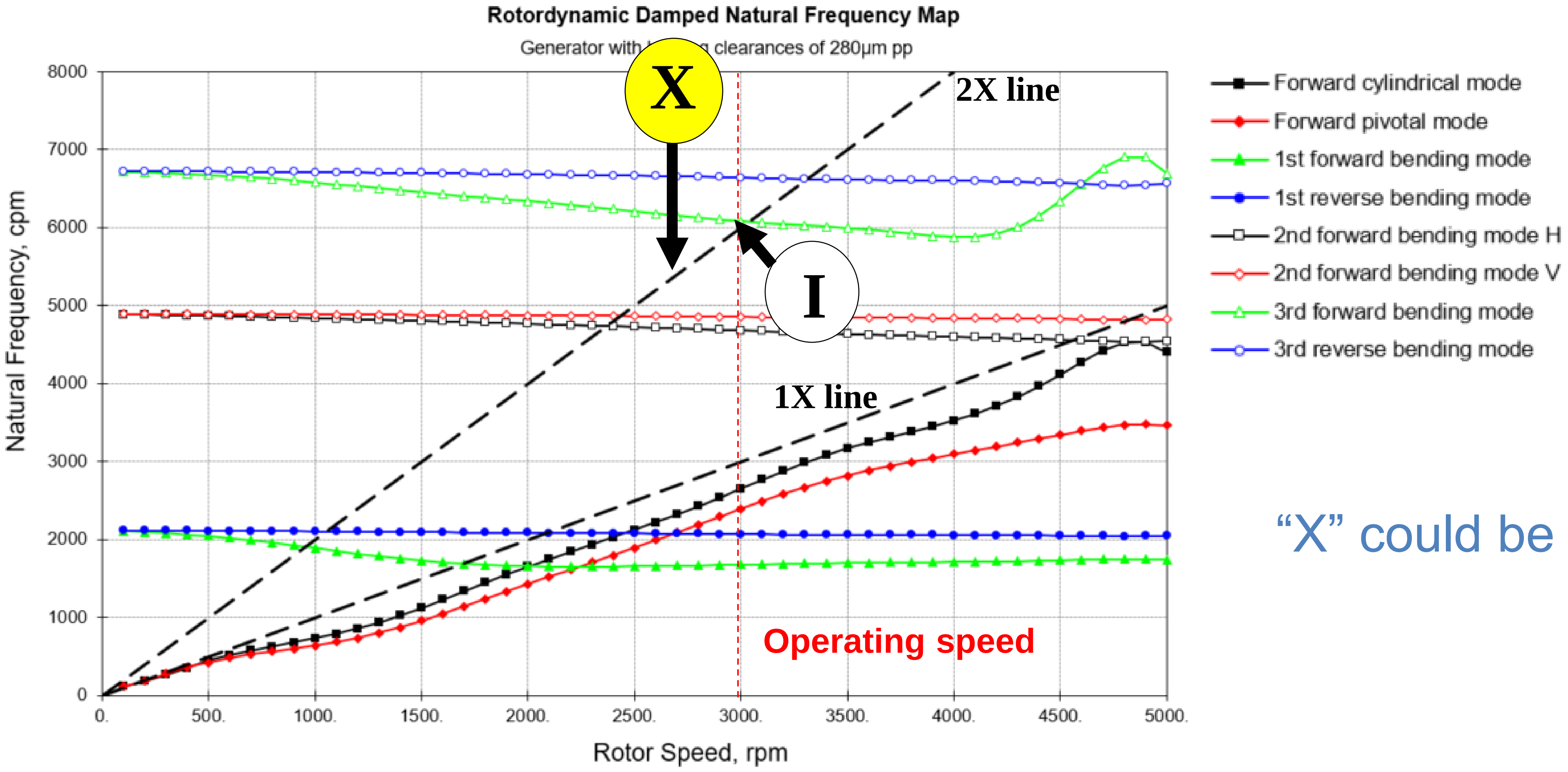


DE side probes are located at the nodes so “H” & “G” are not visible.

“X” was not on the DCS map ???

Lateral Analysis

Damped Critical Speed Map



"X" could be "I" …

Lateral Analysis

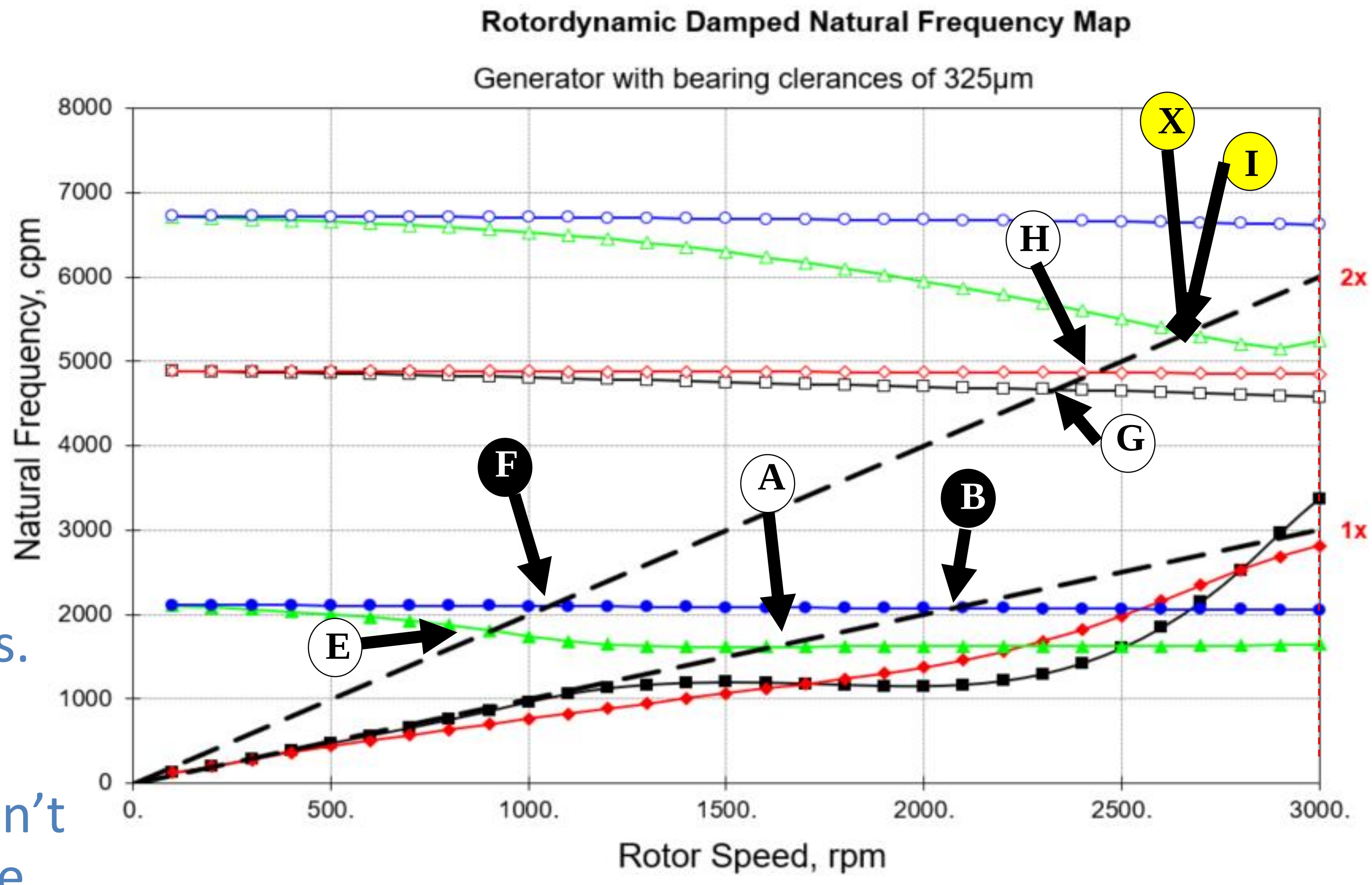
XLTRC² model / Damped Critical Speed Map

Let’s increase bearing clearances from 280µm to 325 µm ?

X and I points merge ….

Given clearances were not correct ?

Bearings worn ?



In any case, the model and measurement match for several points.

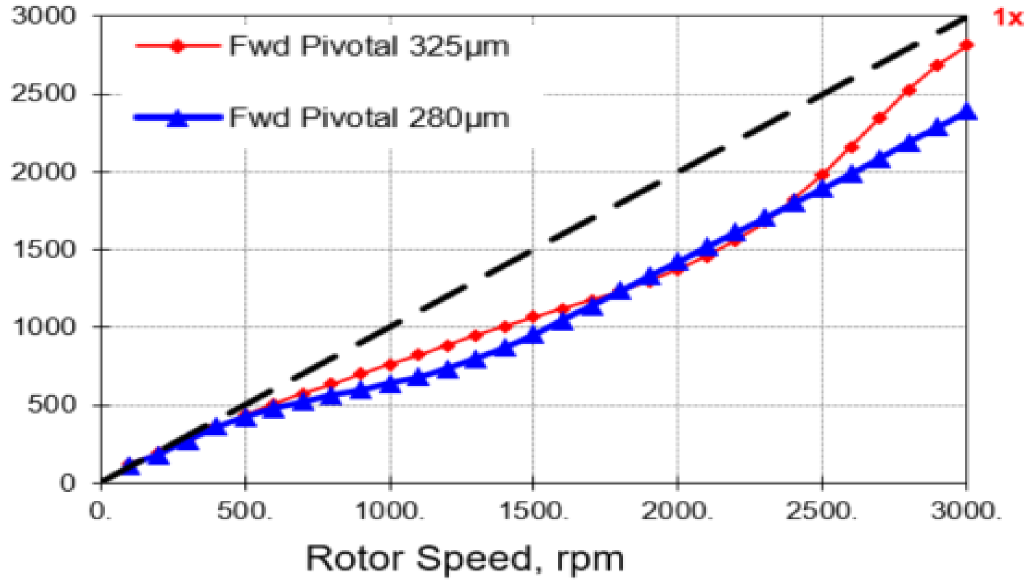
It seems that the increase of vibration after the grid short circuit incident didn’t induced major mechanical issue on the generator

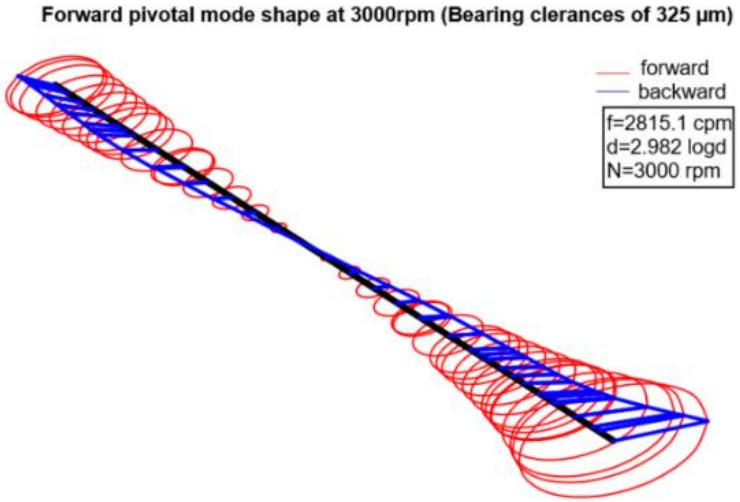
Lateral Analysis

Flash back to the trim balancing

Important trim balance remark : The C vector lags the trial weight by 65°

Theoretically it means that at 3000rpm the generator is running close to a resonance.

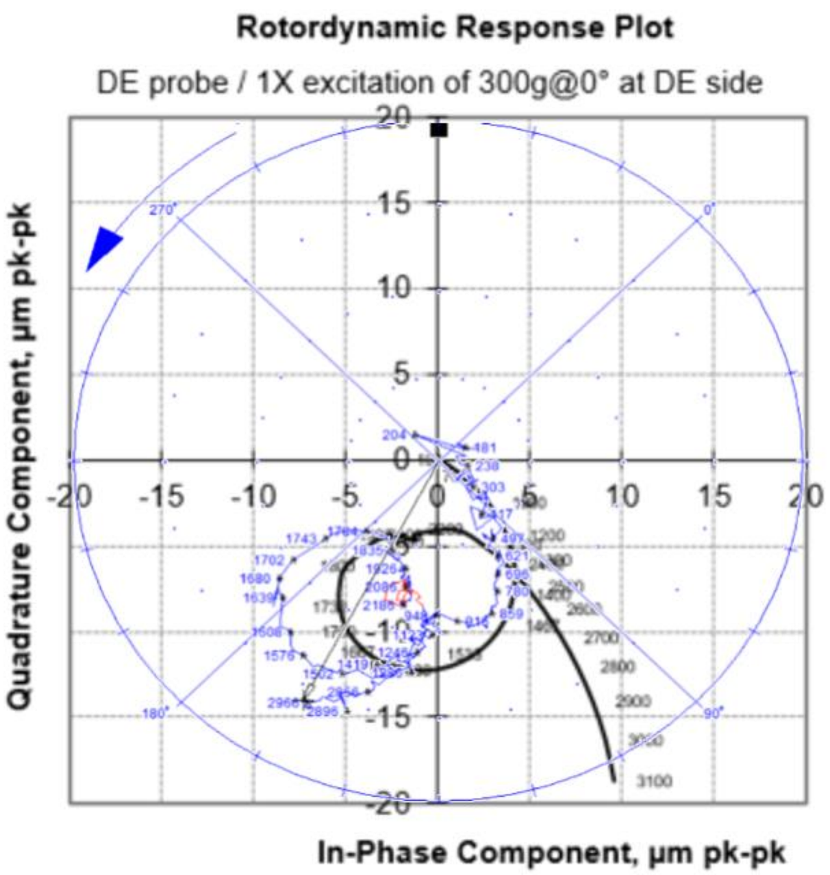


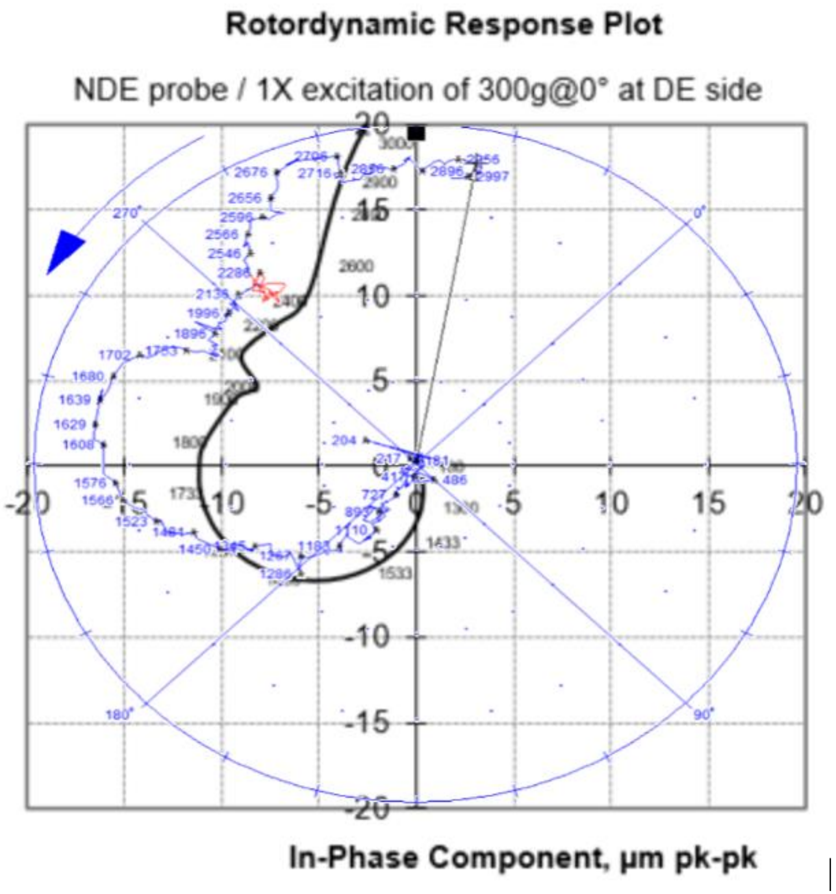


The lateral analysis seems to confirm this assumption ! The pivotal mode frequency follow the 1X line and it is really close to 1X line at operating speed (3000rpm).

Lateral Analysis

Imbalance response / 300g at DE side





Measurement (BLUE) and calculated 1X response (BLACK) are really close

Conclusion

- This unit is still running after 3 years with a low level of vibration
- A boroscopic inspection of the generator was done by OEM. Everything was as expected (no sign of moving part)
- The bearings were not inspected yet.
- Even a simple generator shaft can have a complex dynamic behaviour
- A simple rotor model can really help to confirm that the state of the machine is acceptable on SITE.