# Decrease axial vibration generator with balancing

Guillaume Christin[1], Nicolas Péton[2]

[1]GE Measurement & Control, 68 chemin des Ormeaux, 69760 Limonest, France
[2]GE Measurement & Control, 14 rue de la Haltinière, 44303 Nantes, France

## Abstract

GE's Machinery Diagnostic Services team was invited to investigate vibration issue on generator installed on a power plant. Machine train consists of a generator and a gas turbine. The unit was under commissioning since 2 years and there was not any vibration issue until axial vibration was measured with a temporary accelerometer at NDE generator bearing. The axial vibrations at the bearing location were not monitored for this machine. Vibration testing was carried out using ADRE 408 DSPi (Dynamic Signal Processing Instrument) to hook up vibration data from the existing Bently Nevada's 3500 Series vibration monitor system. Relative vibration data were collected from the Bently Nevada's Rack 3500 monitor buffered outputs. Each of the Machine Train's bearings is monitored by Bently Nevada XY proximity probe pair and one vertical absolute velocity transducer fixed on the bearing housing. Absolute vibration data were collected from the velocity transducer buffered outputs in the Mark VI cabinet. Moreover, a temporary axial accelerometer was installed on the NDE generator bearing housing. The shaft rotates counter clockwise, when viewed from the driver (Gas Turbine) to the driven (Generator).

A periodic vibration measurement, on a route based monitoring schedule, allowed the customer to observe an increasing trend of casing vibration in the radial and axial directions on both of generator bearings. There was a significant increase in vibration amplitudes in the axial direction of generator NDE bearing in situations similar to a full load conditions (i.e. base load of 75 MW) during the months of July, August and September. A first axial vibration analysis was carried out in order to characterize the phenomenon. An axial structural resonance of the NDE generator bearing housing was highlighted. An inspection of the NDE generator bearing support and pedestal inspection was done. Everything was correct and respects the specifications. From experience, the fact to balance a unit with radial vibration measurement can decrease the axial vibration because of the cross effect between radial and axial dynamic stiffness. The generator disposed of two balancing planes. First, the balance trim was carried out following the method of the influence vectors. In order to optimize the balancing and since the vibration were mainly out of phase between both bearings, the influence vector method was linked to the couple-static method. The axial vibrations were decreased by more than 60%

# 1 Discusion

## 1.1 Machine train

Machine train as shown in Fig. 1 consists of a gas turbine driven a Generator via a gear. Each of the machine train's bearing is monitored by one vertical seismic probe and displacement transducers mounted on the machine casing in a plane (XY) perpendicular to the rotor axis of the machine to observe radial motion of the shaft. The XY pairs of non-contacting proximity probes are mounted at 45-degrees left (Y-probe) and 45-degree right (X-probe). One axial seismic probe was temporary mounted on NDE generator bearing support. The machine train diagram is shown in Fig 1. The shaft rotates counter clockwise, when viewed from the driver to the driven.

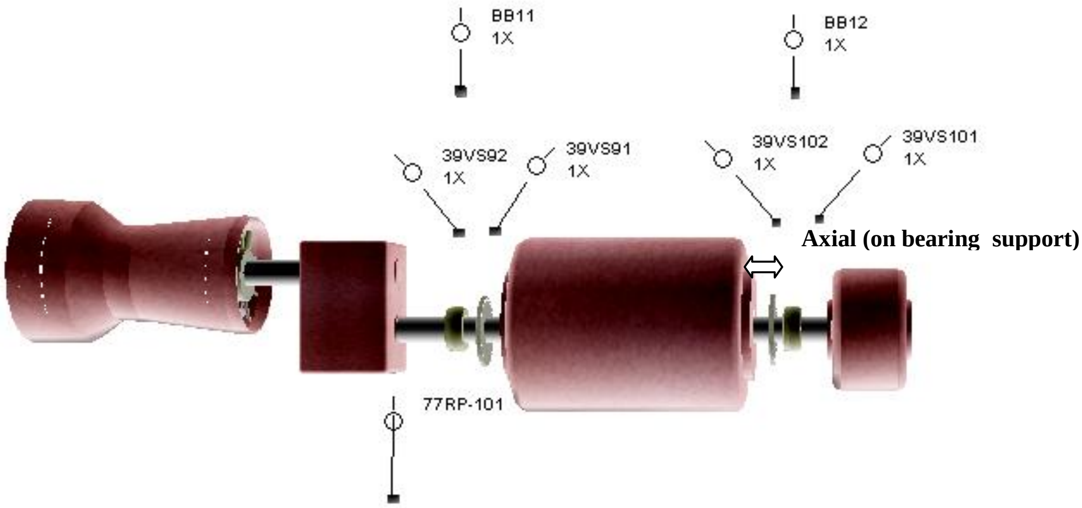



Figure 1: Machine train layout

## 1.2 Description of the issue

The unit had been under commissioning for 2 years and there was no vibration issue until axial vibration was measured with a temporary accelerometer at NDE generator bearing. The axial vibration at bearing location was not monitored for this machine. There is no limitation and obligation in any of the international standard to take care of the axial vibration.

A periodic vibration measurement was carried out by the customer to monitor the casing vibration in radial and axial directions on both of generator bearings. There was significant increase in vibration amplitude in axial direction of generator NDE bearing in similar load conditions i.e. base load i.e. 75 MW in the months of July, August and September. The level of axial vibration reached around 0.4 in/s rms. An inspection of the NDE generator bearing support and pedestal inspection was done. Everything was correct and as per the specifications. Consequently, it was decided to carry on a first axial vibration analyzed to characterize the axial vibration behavior on the NDE generator bearing support. The machine was run at full speed and steady state condition. An axial accelerometer was fitted with magnetic support at different locations (See Fig2) on the NDE generator bearing support while the machine was running at identical condition. From this test, an axial structural resonance of the NDE generator bearing housing was highlighted. The fact that the vibration level was decreasing from 12 mm/s rms to 2 mm/s rms by reduction of only 300 rpm, confirmed the presence of a resonance in axial direction

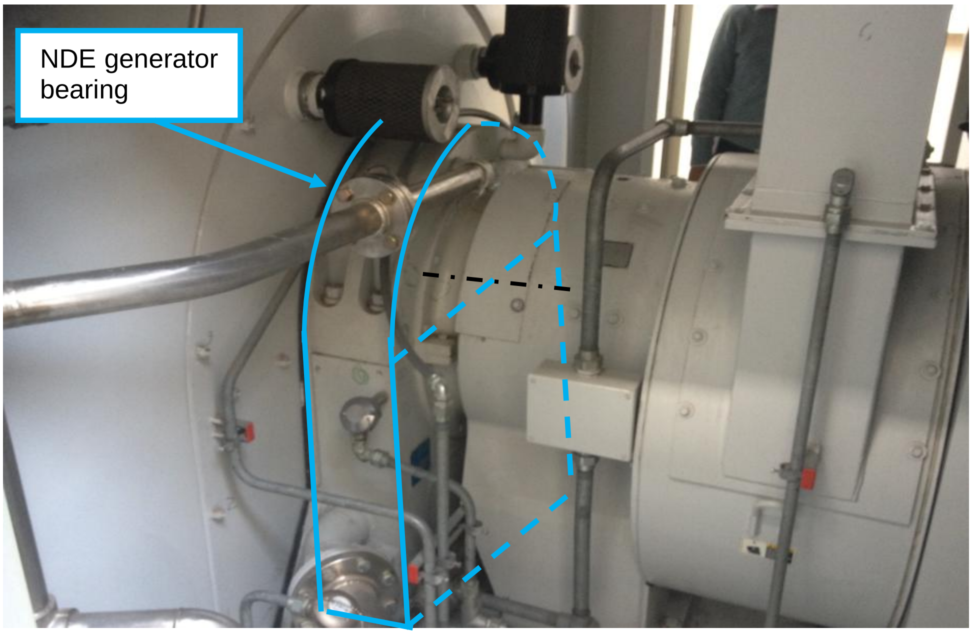


Figure 2: Picture of the NDE generator bearing support

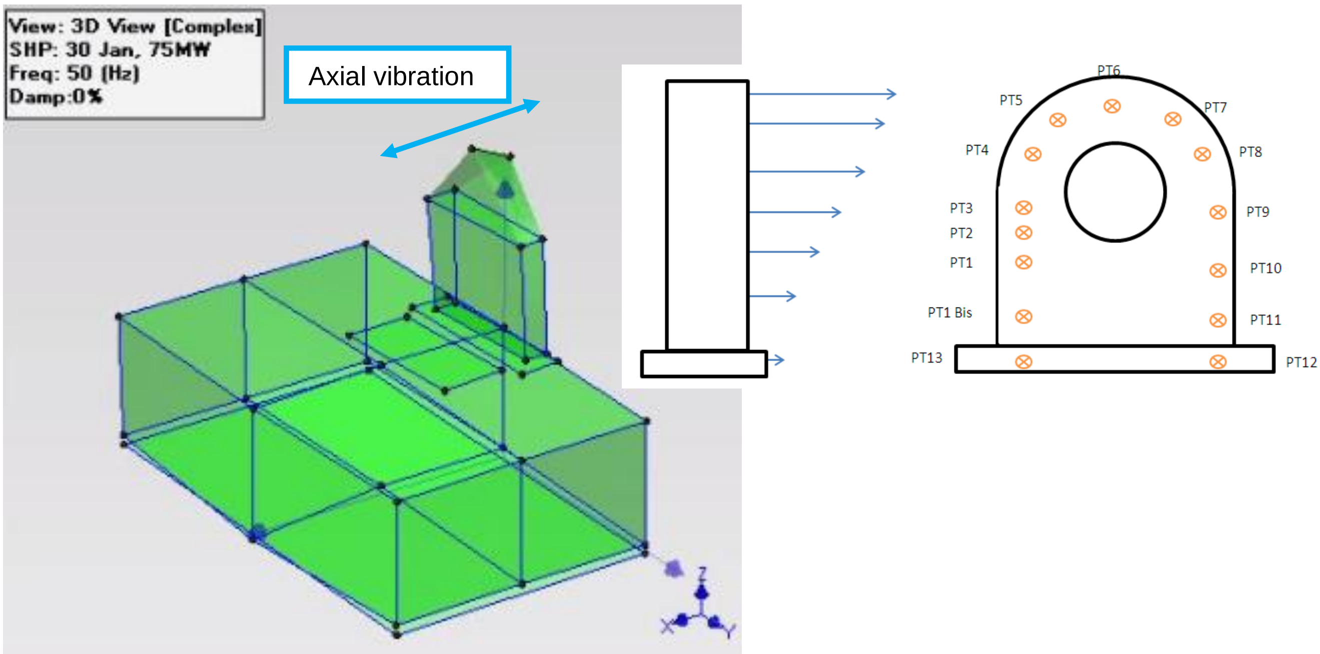



Figure 3: Map of the axial measurement location at the NDE generator bearing support

It was established that a structural resonance was the cause of those high axial vibration. All structural resonances frequencies are basically defined by $\omega_n = \sqrt{\frac{K_n}{M_n}}$ .with $n$ being the number of the mode. To decrease the level of vibration when a structure is excited at a frequency close to one of its own natural resonances there are two main solutions. The first one is to change the design of the structure in order to move away the structural resonance. The modification of the design should change the ratio between the modal stiffness ( $K_n$ ) and the mass stiffness ( $M_n$ ) so the natural resonance frequency $\omega_n$ could be moved away from the actual one

In our case, the first solution was not possible to be implemented. However, since the deterioration of a fixation can usually decrease the stiffness of a structure and so decrease the natural resonance of this structure, all the fixations of the support were inspected. No abnormal fitting or looseness was observed. Since the option to change the natural resonance of the NDE bearing support was not possible, the only way to decrease the vibration was to drop down the force acting on the structure

In order to reduce axial vibration it was decided to balance the unit. Effectively, the vibration is an oscillating movement compare to a reference due to a force. The main force acting is the synchronous one so the unbalance. The synchronous response (the 1X vibration) is the force divided by the Synchronous dynamic stiffness. Among the forces acting, the unbalance was, in that case, the most important one so by reducing this one, the response should go down. From our experience it is known that the unbalance force which is acting on the radial direction can have an impact on the vibration in the axial direction. This characteristic is due to the fact that in a machine there are some mechanical paths that transform a radial force in axial forces. In numerical model, this type of behavior may be visible in the matrix and it can be assimilate to the cross-factors. It means that when a force is acting in one direction it is possible to have a movement in an orthogonal direction

The commissioning team checked twice the assembly and the quality of the foundation but proven ineffective

## 2 Balancing the unit using the influence vector method

The vibration signal was mainly due to synchronous component (1X) and the 1X component was increasing with the speed. The behaviour of the machine was the same during start up and shutdown which means that the behaviour of this phenomenon was repeatable. The behaviour of the machine seemed to be repeatable between a startup and the shutdown associated. It was observed that the orbit shape was elliptical with a forward precession. Since the unbalance is a forward force the fact to reduce this force should decrease the forward vibration. From all the previous observations, it was decided to try to balance the unit

Selecting a trial weight can be a dangerous proposition. In many cases, the vibration on the machine is already high, and adding an excessive trial weight could increase the vibration amplitudes. So a good practice is to at least get a correct trial weight location to be opposite at the actual residual unbalance location and add enough weight to get some response. This selection is not easy unless a reasonable amount of information is known about the machine. Adequate trial weight would be considered to be enough weight to produce a 10% change in vibration vector based on the original vibration

Before presenting the trim balance of this unit, it is important to review the influence vectors method. Basically, this method consists in installing a trial weight in a balance plane and in observing what is the influence of this mass on all the vibration response. This action has to be repeated for all the balance planes available on the machine. Then software like Bently Balance will propose an optimal solution to install one correction weight in one location for each balance plane

Below is an explanation of what the influence vector $\vec{H}$ is and how it can be calculated

First, it is necessary to obtain the original vibration vector $\vec{O}$ due to the original unbalance of the unit. Then a calibration weight $\overrightarrow{W_{cal}}$ is installed on the machine and a vibration vector $\overrightarrow{O+C}$, which is the response of the original unbalance and of the trial weight, is obtained. The response vector $\vec{C}$ due to only the calibration weight is calculated and finally, the influence vector $\vec{H}$ can be obtained by the following equation

$$\vec{H} = \frac{\vec{C}}{\overrightarrow{W_{cal}}}$$

With

$\vec{H}$ the influence vetor,

$\vec{C}$ the response due to the calibration weight

$\overrightarrow{W_{cal}}$ the calibration weight.

The influence vector $\vec{H}$ is independent of the calibration weight and it represents how the machine response to a synchronous excitation. It is important to note that an influence vector is calculated at one specified speed

When the influence vector is obtained, it is necessary to resolve the following equation to determine the amount of the correction weight and its location

$$\overrightarrow{W_{cor}} = \frac{\vec{N}}{\vec{H}}$$

With

$\vec{H}$ the influence vetor,

$\vec{N}$ the vector that we want to create thanks to the correction weight. $\vec{N} = -\vec{O}$

$\overrightarrow{W_{cor}}$ the correction weight.

The objective is to create a vector $\vec{N}$ equal in amplitude and out of phase of the Original vector $\vec{O}$. Theoretically, if the correction weight is installed then the vibration should be equal to zero. Nevertheless, the assumption that our system is linear is not true. There is always a part of non-linearity

Usually, on a machine there are several sensors and balance planes. $\vec{H}$ has to be calculated for each vibration sensors of the machine and for each balancing planes. Then it is necessary to use software like Bently Balance to help us

When a trim balance is carrying out with the support of a software like Bently Balance, the number of influence vector ( $\overrightarrow{H_{ij}}$ ) is equal to $i \times j$ with $i$ the number of sensors and $j$ the number of balancing planes. In this case, the following system has to be solved

$$\left\{\overrightarrow{O_i}\right\} + \left[\overrightarrow{H_{ij}}\right] \times \left\{Wcor_j\right\} = \{0\}$$

The above equation is an over determined systems of linear equations since there are more equations than variables. In order to solve this equation system, the software uses a least mean square method to decrease the level of vibration for all the sensors. Effectively there is not one perfect solution to solve this system

The trim balance of this generator unit was done using this influence vector methodology and Bently Balance software

A first run, named the REF RUN, was done in order to get cold reference data (One start up (SU) and one run at Full Speed No Load (FSNL) during 2-5 minutes and then a shutdown (SD)). Then another run (RUN1) was done at Full Speed Full Load (FSFL) during 1 hours in order to check if any thermal phenomenon had an influence on the vibration behavior. No influence was observed between FSFL and FSNL

Since the vibration behaviour of the generator was similar between FSFL and FSNL it was decided to balance the unit at FSNL. This method saved time and cost to balance the unit. The generator was balanced using the influence vector method

Two balance planes were available on the generator. It is important to remember that even if there are two or more planes available it is not necessary to use all of them. The more the number of used balance plane is important the more the price of the trim balance will be expensive (time cost, gas cost, loss of production…etc. ...). The fact to use one more balance plane is equivalent to do one more run. This need has to be justified to the customer

In order to determine if two planes are needed, the first thing to do is to compare the phase relationship between DE 1X vectors and NDE 1X vectors. Obviously, this comparison has to be done for probes which are installed in the same angular position. The objective is to understand which lateral rotor mode dominates the dynamic behaviour of the rotor

If a machine is mainly influenced by its first mode then the 1X vectors should be in phase. So, wherever the weight is installed along the rotor (in opposite position of the heavy spot) the vibration should decrease. This last sentence is true only if the second mode of the rotor is far from the operating speed and do not have any influence at the operating speed

From our data, it appears that the generator was mainly influenced by its second mode since the 1X vectors were out of phase from one bearing (DE) to another bearing (NDE). It was then decided to use both balance planes

## 2.1 First trial run

A first trial weight of 392g was installed at 30° CCW from the keyphasor notch (when looking from the generator to the turbine) on the balance plane at NDE generator side. and the machine was run to FSNL during 2 minutes

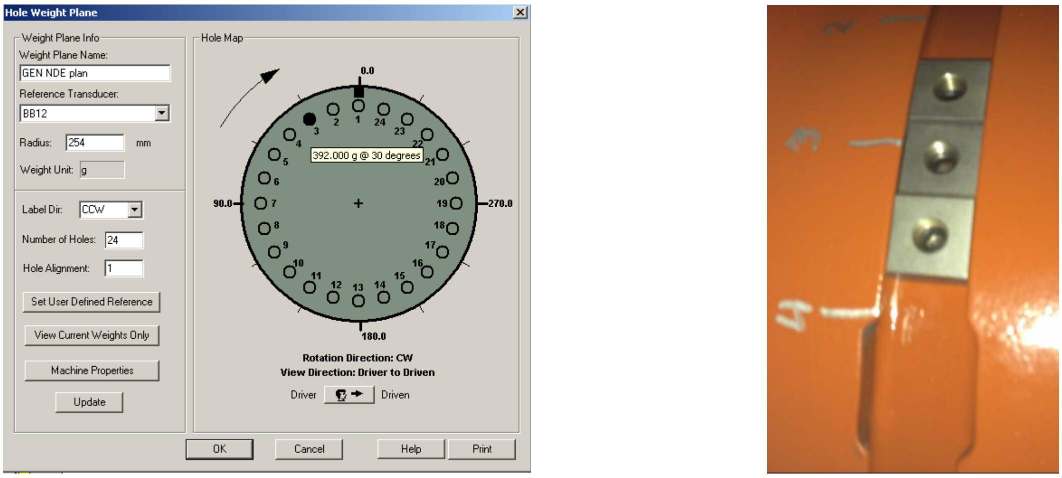



Figure 4: Details of the bearing #8 balance plane and picture of the installed trial weights

The polar plots (Fig 5 & Fig 6) show that the weight had an influence on the vibration since the 1X amplitude and the 1X angle changed for all the probes. The vibration units (in/s pk) are from the American system. Since the goal of this balance activity was to reduce the (axial) seismic vibration at NDE end, the plots of the velocity probes will be presented in this article

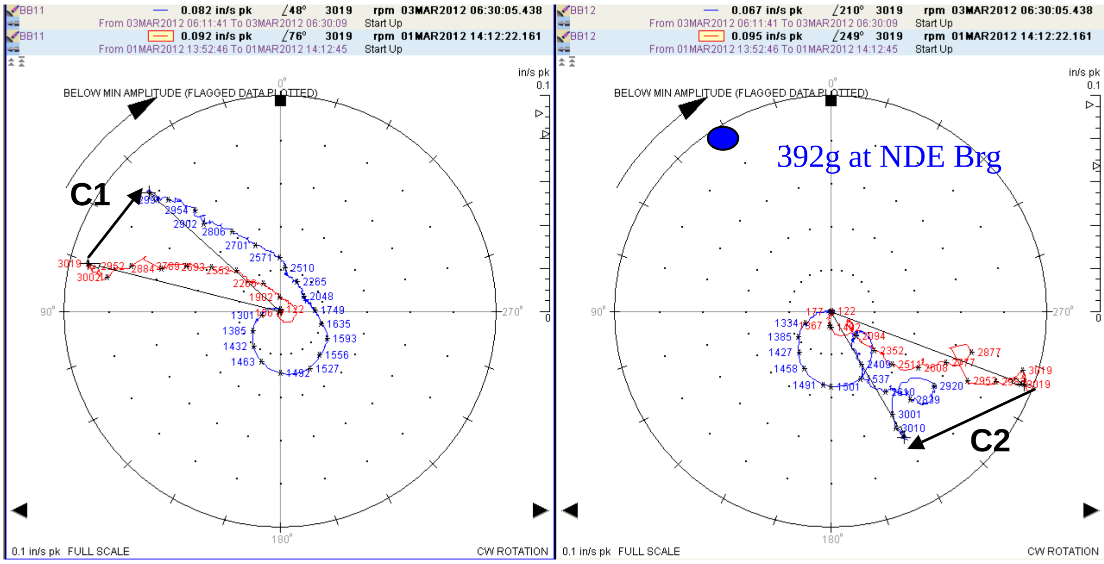



Figure 5: 1X polar plots for BB11 and BB12. The red curve is the reference run (Ref RUN) data.

The blue curve is the trial run (RUN2) data. It can be observed that the trial weight had an influence on the 1X vector. The vibrations are more important at operating speed. It can be observed that the first resonance is around 1500rpm

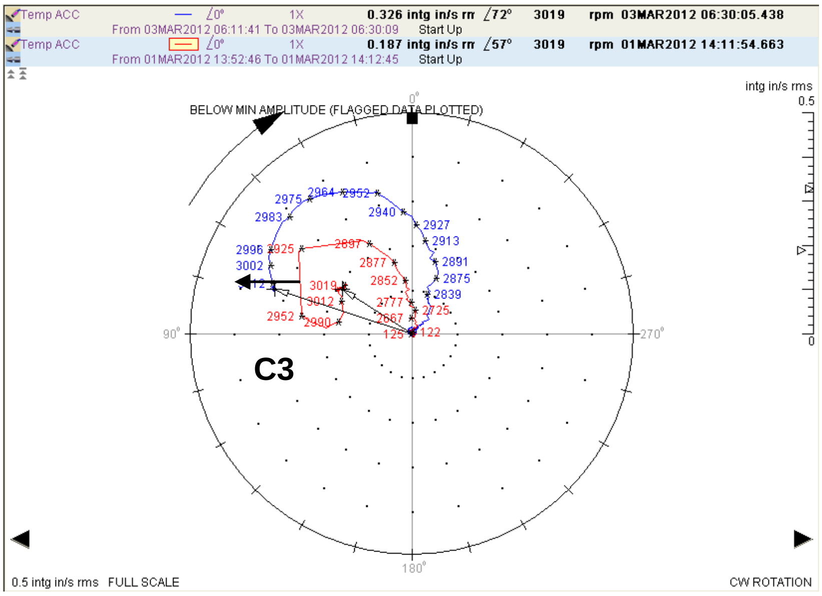



Figure 6: 1X polar plots for the temporary axial accelerometer at bearing #8.

The red curve is the Reference run (REF RUN) data. The blue curve is the trial run (RUN2) data. It can be observed that the trial weight increased the 1X vector

## 2.2 Second trial run

The previous trial weight at NDE generator side was removed. A trial weight of 392g was installed at 240° CCW from the keyphasor notch (when looking from the generator to the turbine) on the balance plane close to DE generator side and the machine was run to FSNL during 2 minutes

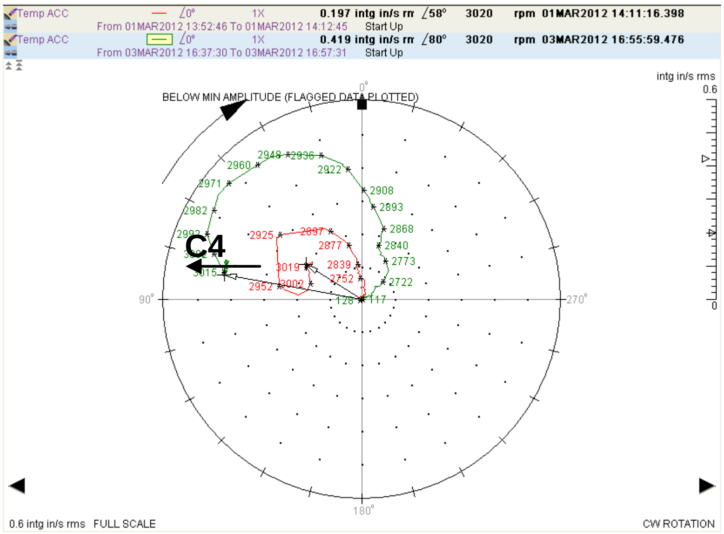



Figure 7: 1X polar plots for the temporary axial accelerometer at bearing #8.

The red curve is the Reference run (REF RUN) data. The green curve is the trial run (RUN3) data. It can be observed that the trial weight increased the 1X vector

## 2.3 Calculation of final weight with multiple method

The calculation to find the location of the final weights was done with Bently Balance software. Bently Balance software uses the influence vector method and a least mean square algorithm to propose the best solution to decrease the level of vibration for all the sensors. The next picture is an extract from Bently

Balance software. In the green area, Bently Balance advices to add one mass of **899g@322° in NDE GEN plane** and another mass of **205g@252° in DE GEN plane**. In the blue area, predicted results of the final vibration are shown if the proposed solution is applied. In the orange area, the original vibration vectors are shown

| Weight Plane | Solution Weight | Solution Angle (deg) | User Sol Angle(deg) | Ist Split Weight | Ist Split Angle |
|---|---|---|---|---|---|
| GEN NDE plan | 898.80 g | 322 Deg | | | |
| GEN DE | 205.25 g | 252 Deg | | | |

| Predicted Results | | | | (Uncomp) | |
|---|---|---|---|---|---|
| **Channels** | **Sample** | **Speed** | **Load** | **1X Amp** | **1X Phs** |
| 39VS91 | N/A | 3019 rpm | 0.00 MW | 0.21 mil | 355 Deg |
| 39VS92 | N/A | 3019 rpm | 0.00 MW | 0.33 mil | 37 Deg |
| 39VS101 | N/A | 3019 rpm | 0.00 MW | 0.73 mil | 135 Deg |
| 39VS102 | N/A | 3019 rpm | 0.00 MW | 0.13 mil | 177 Deg |
| BB11 | N/A | 3019 rpm | 0.00 MW | 0.03 in/s | 3 Deg |
| BB12 | N/A | 3019 rpm | 0.00 MW | 0.04 in/s | 36 Deg |

| Ref Data | | | | (Uncomp) | |
|---|---|---|---|---|---|
| **Channels** | **Sample** | **Speed** | **Load** | **1X Amp** | **1X Phs** |
| 39VS91 | 4 | 3019 rpm | 0.00 MW | 0.76 mil | 114 Deg |
| 39VS92 | 4 | 3019 rpm | 0.00 MW | 2.25 mil | 22 Deg |
| 39VS101 | 4 | 3019 rpm | 0.00 MW | 0.18 mil | 305 Deg |
| 39VS102 | 4 | 3019 rpm | 0.00 MW | 1.30 mil | 186 Deg |
| BB11 | 4 | 3019 rpm | 0.00 MW | 0.09 in/s | 78 Deg |
| BB12 | 4 | 3019 rpm | 0.00 MW | 0.10 in/s | 250 Deg |

Figure 8: Results extracted from Bently Balance software. Multi plane method results.

The solution proposed by the software was not possible to implement since the weight was too heavy. The maximum weight available was around 330g. The algorithm used in Bently Balance doesn't have any weight limitation. It proposes a solution to obtain the best result

The next step could be to use BentlyBalance and simulate several solutions. A first solution that could be tested is to replace the maximum solution weight (840g) by the maximum available weight (330g). The value of the second weight (205g) was changed to 75g (*205x330/899*) in order to keep the proportionality with the first solution. The next table shows that the predicted results are not really good compare to the original solution

| | | | Weight Plane | Solution Weight | Solution Angle (deg) | User Sol Angle(deg) | Alternate Weight | Alternate Angle (deg) |
|---|---|---|---|---|---|---|---|---|
| | | | GEN NDE plan | 898.80 g | 322 Deg | | 330.00 | 322 |
| | | | GEN DE | 205.25 g | 252 Deg | | 75.00 | 252 |
| **Predicted Results** | | | | (Uncomp) | | | | |
| **Channels** | **Sample** | **Speed** | **Load** | **1X Amp** | **1X Phs** | | **Alt 1X Amp** | **Alt 1X Phs** |
| 39VS91 | N/A | 3019 rpm | 0.00 MW | 0.21 mil | 355 Deg | | 0.45 mil | 105 Deg |
| 39VS92 | N/A | 3019 rpm | 0.00 MW | 0.33 mil | 37 Deg | | 1.54 mil | 24 Deg |
| 39VS101 | N/A | 3019 rpm | 0.00 MW | 0.73 mil | 135 Deg | | 0.15 mil | 141 Deg |
| 39VS102 | N/A | 3019 rpm | 0.00 MW | 0.13 mil | 177 Deg | | 0.87 mil | 186 Deg |
| BB11 | N/A | 3019 rpm | 0.00 MW | 0.03 in/s | 3 Deg | | 0.06 in/s | 69 Deg |
| BB12 | N/A | 3019 rpm | 0.00 MW | 0.04 in/s | 36 Deg | | 0.05 in/s | 261 Deg |

Figure 9: Results extracted from Bently Balance software. Alternative calculation.

Infinity of other solutions could be tested but it is not really scientific. So it was decided to associate the influence vector method and the Static-Couple method in order to guide us to the best final solution

# 3 Optimise the trim balance with Static/Couple method

## 3.1 Static/Couple method review

Let's review first the static-couple method. In this method, vectors at two planes are decomposed into in-phase (static) and out-of-phase (couple) components. The static component can be due to the first mode or third mode. The couple component is typically due to the second mode. Static weight is defined as the weights placed at two ends with the same orientation (in-phase), whereas couple weight is defined as the weights placed at two ends with the opposite orientation (out-of-phase)

.

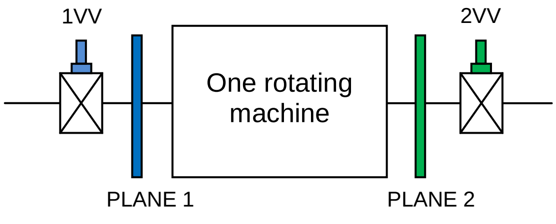



Figure 10: A rotating machinery monitored by two probes 1VV and 2VV positioned at each and of the machine. Two balance planes are also available at each end of the machine.

The vibration vectors $\vec{O}_{1VV}$ and $\vec{O}_{2VV}$ are measured during a first reference run. These two vibration vectors are represented on the same polar plot and decomposed in a static vector $\vec{S}$ and a couple vector $\vec{C}$ (or $-\vec{C}$). So $\vec{O}_{1VV} = (\vec{S}+\vec{C})$ and $\vec{O}_{2VV} = (\vec{S}-\vec{C})$. The idea consists to observe which part of the vector is dominant. In our example, it can be noticed that the vector $\vec{C}$ has higher amplitude than the vector $\vec{S}$ .Consequently; it will be more interesting to balance those couple vectors before the static's.

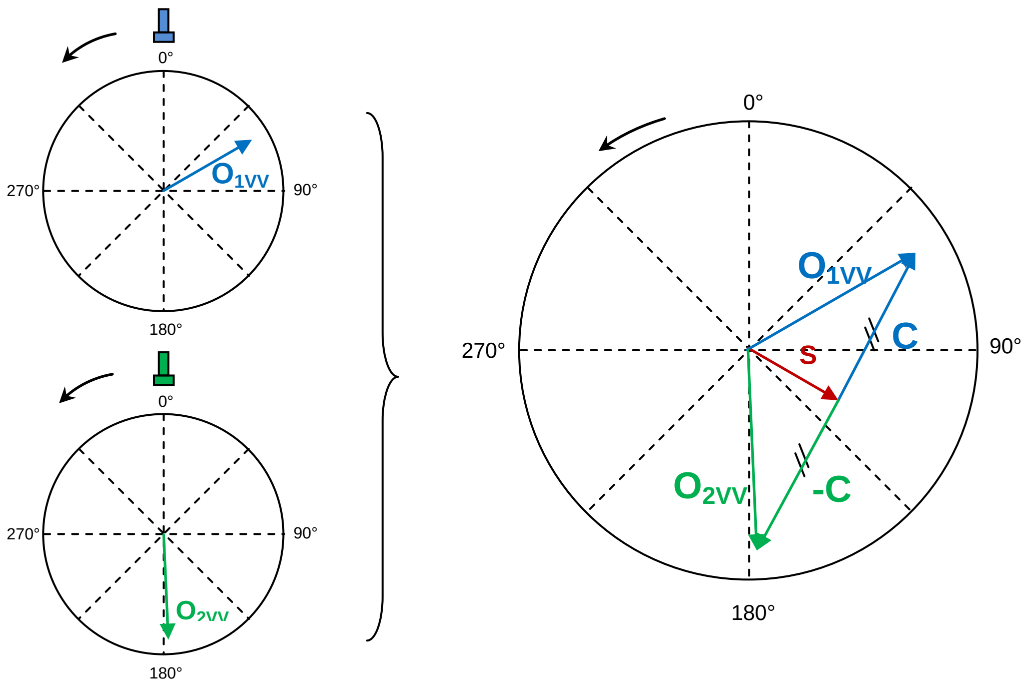
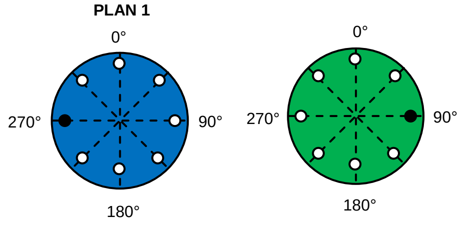
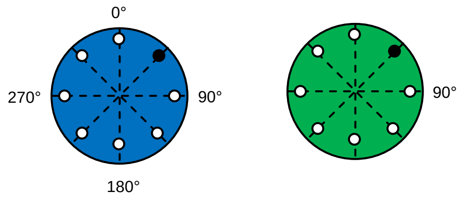



Figure 11: Polar plots showing the static and couple part of probes.

The trial weights should be installed in opposition of phase in each plane as illustrated in the following picture. The new vibrations vectors should be measured and the influence vectors calculated for both sensors




**TRIAL Weights locations (out of phase) to balance Couple part**




**TRIAL Weights locations (In phase) to balance Static part**

To balance the static vector, two trial weights have to be installed at the same angular orientation in each plan. The new vibration vectors should be measured and the influence vectors calculated for both sensors

This method can be really efficient on symmetrical machine like generator and allows balancing separately the first mode and the second mode. The influence vector obtained for this example are named $\vec{H}_{SS}$ , $\vec{H}_{SC}$ $\vec{H}_{CS}$ and $\vec{H}_{CC}$ .

Then, the following system has to be solved to determine $Ws$ (the static weight) and $Wc$ (the couple weight).

$$\begin{Bmatrix} S \\ C \end{Bmatrix} + \begin{bmatrix} \vec{H}_{SS} & \vec{H}sc \\ \vec{H}_{CS} & \vec{H}_{CC} \end{bmatrix} \begin{Bmatrix} Ws \\ Wc \end{Bmatrix} = \begin{Bmatrix} 0 \\ 0 \end{Bmatrix}$$

The influence vectors obtained for the static-couple method are different from the influence vectors use with the multi plane method. However, it is proven that the influence vectors of one method can be used to calculate the influence vectors of the other method

Sometimes for an asymmetric rotor, or due to influence via couplings from adjacent rotors, the first/third mode shape (in-phase) may not be very symmetric and the second mode shape (out-of-phase) may not be very anti-symmetric. In this case, cross-effects may need to be included

If both direct and cross-effect influence coefficients are included, then theoretically the rotor can be balanced perfectly. The new format of influence data for the static-couple model is comparable to that for the multi-plane balance model

Sometimes one needs to know individual probe influence due to static or couple weights only. The static weight influence to probes near planes 1 and 2 can be given by $h_{1,S}$ and $h_{2,S}$ , while the couple weight influence to probes near planes 1 and 2 can be given by $h_{1,C}$ and $h_{2,C}$

Conversion equations can be developed to obtain direct and cross-effect static-couple influence coefficients from the four influence coefficients of the multi-plane balance model or to obtain the four influence coefficients of the multi-plane balance model from direct and cross-effect static-couple influence coefficients

Concerning our historical case, direct and cross-effect static-couple influence coefficients were calculated from the four influence coefficients of the multi-plane balance model. The next formula developed in the literature [1] was used

$$\vec{H}_{SS} = \frac{1}{2}\left(\vec{h}_{11} + \vec{h}_{22} + \vec{h}_{12} + \vec{h}_{21}\right)$$

$$\vec{H}_{CS} = \frac{1}{2}\left(\vec{h}_{11} - \vec{h}_{22} + \vec{h}_{12} - \vec{h}_{21}\right)$$

$$\vec{H}_{SC} = \frac{1}{2}\left(\vec{h}_{11} - \vec{h}_{22} - \vec{h}_{12} + \vec{h}_{21}\right)$$

$$\vec{H}_{CC} = \frac{1}{2}\left(\vec{h}_{11} + \vec{h}_{22} - \vec{h}_{12} - \vec{h}_{21}\right)$$

## 3.2 Balance the generator using both method static/couple and multi plane

A complete excel file was developed to easily obtained the Static-Couple influence vector from the multi-plane influence vector method. Using this tools all the static-couple influence vectors were obtained for the generator

In order to details the procedure, the example for proximity probes 39VS91, 39VS92, 39VS101 and 39VS102 will be presented. First it is necessary to come back to the values of the original vectors

| **DE Plane** | | | **NDE Plane** | | |
|---|---|---|---|---|---|
| | | $\vec{O}$ **vectors** | | | $\vec{O}$ **vectors** |
| **39VS91** | **45°R** | 0.38@137° | **39VS101** | **45°R** | 0.309@306° |
| **39VS92** | **45°L** | 2.25@23° | **39VS102** | **45°L** | 1.298@192° |

Then, in order to eliminate the effect of the anisotropic stiffness, the forward and reverse component for both pair of probes 39VS91/92 and 39VS101 /102 were calculated

| **DE Plane 39VS91/92** | | **NDE Plane 39VS101/102** | |
|---|---|---|---|
| Forward | 1.30@116° | Forward | 0.79@287° |
| Reverse | 0.95@288° | Reverse | 0.51@95° |

It can be observed that the reverse component was not negligible compare to the forward component. Then the static and couple components were calculated for this pair of 1X forward vector

| | |
|---|---|
| Static | 0.27@129° |
| Couple | 1.04@113° |

The couple component is much more import than the static. So it was decided to balance only the couple component. To do it, it is necessary to have the $\vec{H}_{CC}$ influence vector and the original couple part vector that was calculated above (1.04@113°).

After using the previous equation, the following Static-Couple influence vectors matrix was obtained

$$\begin{bmatrix} \vec{H}_{SS} & \vec{H}_{SC} \\ \vec{H}_{CS} & \vec{H}_{CC} \end{bmatrix} = \begin{bmatrix} 0{,}000429239@33° & 0{,}000179051@175° \\ 0{,}000347825@139° & 0{,}003135887@112° \end{bmatrix} \text{mil pp /g}$$

The influence vector for the couple part $\vec{H}_{CC}$ is equal to $0{,}003135887@112°$. The next equation has to be solved to obtain the amount and the locations of the couple weights to balance the unit

$$\overrightarrow{Wc_{cor}} = \frac{\vec{N}}{\overrightarrow{H_{CC}}}$$

With $\overrightarrow{H_{CC}}$ the influence vector of the couple component,

$\vec{N}$ the vector that we want to create thanks to the correction weight. $\vec{N} = -\vec{C}$ (with $\vec{C}$ the couple component).

$\overrightarrow{Wc_{cor}}$ the Couple correction weight.

Finally, this calculation proposed a solution of 331g@0° on NDE generator and 331°@180° on DE generator. The amount of weight proposed by this solution was possible to implement on the generator. Before to implement this solution, this result was tested in Bently Balance to check the predicted results

Balance Summary | Alternate Prediction | Reference Data | Influence Vectors | Balance Report | Final Results

| | | Weight Plane | Solution Weight | Solution Angle (deg) | User Sol Angle(deg) | Alternate Weight | Alternate Angle (deg) |
|---|---|---|---|---|---|---|---|
| | | GEN NDE plan | 898.80 g | 322 Deg | | 331.00 | 0 |
| | | GEN DE | 205.25 g | 252 Deg | | 331.00 | 180 |
| | | | | | | | |
| | | | | | | | |
| | | | | | | | |
| Channels | Speed | Load | (Uncomp) 1X Amp | 1X Phs | | Alt 1X Amp | Alt 1X Phs |
| 39VS91 | 3019 rpm | 0.00 MW | 0.21 mil | 355 Deg | | 0.25 mil | 79 Deg |
| 39VS92 | 3019 rpm | 0.00 MW | 0.33 mil | 37 Deg | | 1.01 mil | 349 Deg |
| 39VS101 | 3019 rpm | 0.00 MW | 0.73 mil | 135 Deg | | 0.62 mil | 176 Deg |
| 39VS102 | 3019 rpm | 0.00 MW | 0.13 mil | 177 Deg | | 1.20 mil | 140 Deg |
| BB11 | 3019 rpm | 0.00 MW | 0.03 in/s | 3 Deg | | 0.07 in/s | 11 Deg |
| BB12 | 3019 rpm | 0.00 MW | 0.04 in/s | 36 Deg | | 0.03 in/s | 175 Deg |

Figure 12: Results extracted from Bently Balance software. Alternative calculation for static/couple method.

In the blue area, it can see that this alternative solution works but could be better. Several other alternative simulations were tested in Bently Balance close to this one. The best and final results were obtained for **one weight of 331g@330°CCW at NDE generator and one weight of 331g@150°CCW at DE generator.** The fact to shift the solution of 30° increases a lot the predictive results (see next table)

Balance Summary | Alternate Prediction | Reference Data | Influence Vectors | Balance Report | Final Results

| | | Weight Plane | Solution Weight | Solution Angle (deg) | User Sol Angle(deg) | Alternate Weight | Alternate Angle (deg) |
|---|---|---|---|---|---|---|---|
| | | GEN NDE plan | 898.80 g | 322 Deg | | 331.00 | 330 |
| | | GEN DE | 205.25 g | 252 Deg | | 331.00 | 150 |
| | | | | | | | |
| | | | | | | | |
| | | | | | | | |
| Channels | Speed | Load | (Uncomp) 1X Amp | 1X Phs | | Alt 1X Amp | Alt 1X Phs |
| 39VS91 | 3019 rpm | 0.00 MW | 0.21 mil | 355 Deg | | 0.25 mil | 149 Deg |
| 39VS92 | 3019 rpm | 0.00 MW | 0.33 mil | 37 Deg | | 0.79 mil | 39 Deg |
| 39VS101 | 3019 rpm | 0.00 MW | 0.73 mil | 135 Deg | | 0.57 mil | 138 Deg |
| 39VS102 | 3019 rpm | 0.00 MW | 0.13 mil | 177 Deg | | 0.69 mil | 137 Deg |
| BB11 | 3019 rpm | 0.00 MW | 0.03 in/s | 3 Deg | | 0.02 in/s | 358 Deg |
| BB12 | 3019 rpm | 0.00 MW | 0.04 in/s | 36 Deg | | 0.02 in/s | 325 Deg |

Figure 13: Results extracted from Bently Balance software. Alternative calculation for final solution obtained with the combination of multi plan method and the static couple method.

## 3.3 Final run with solution from static/couple method & multiplane method

The precedent trial weight at DE generator was removed. One weight of 331g was installed at 150° CCW from the keyphasor notch (viewing from Generator to turbine) at DE generator balance plane. Another one of 331g was installed at 330° CCW from the keyphasor notch (viewing from generator to turbine) at NDE generator balance plane

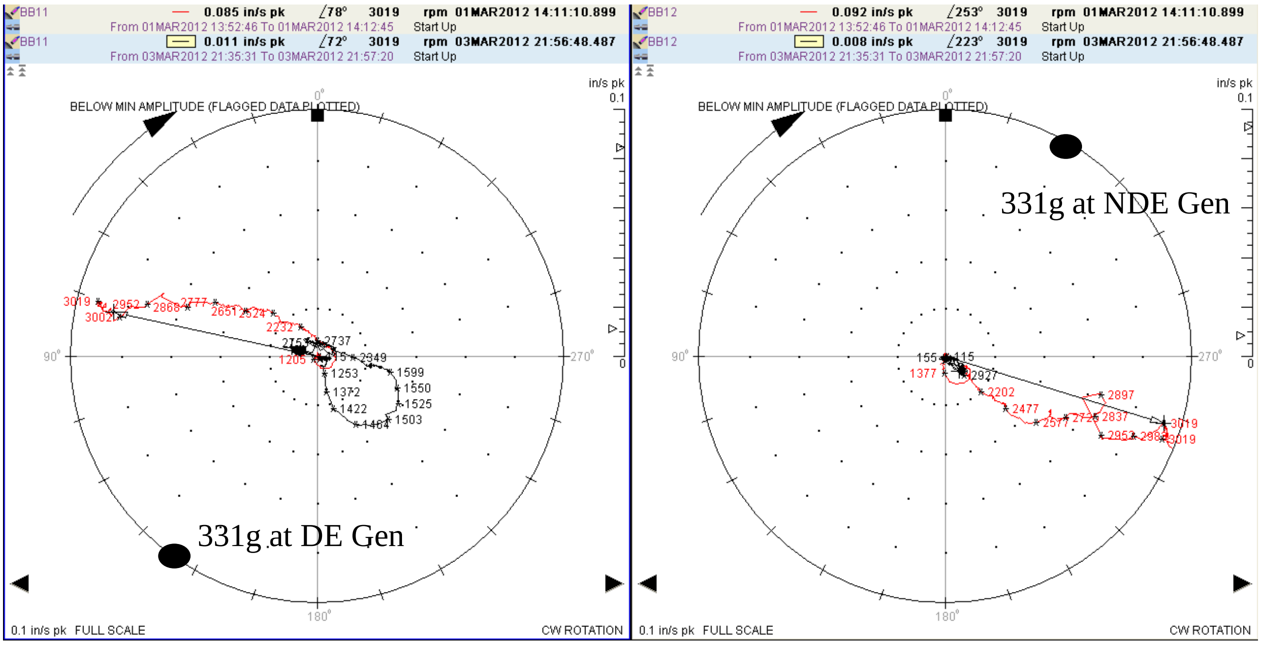



Figure 14: 1X polar plots for BB11 and BB12

The red curve is the Reference run (REF RUN) data. The black curve is the final run (RUN4) data. It can be observed that the final weight decreased the 1X vectors. BB11 from 0.085 in/s pk to 0.011 in/s pk. BB12 from 0.092 in/s pk to 0.008 in/s pk

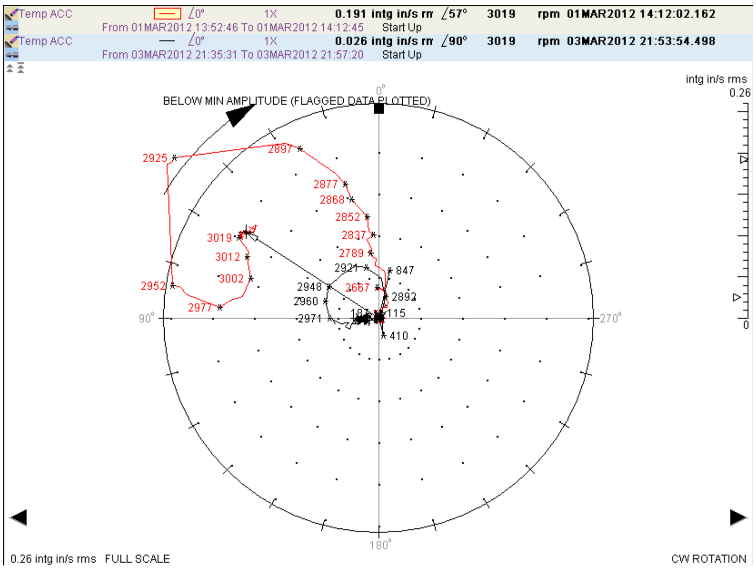



Figure 15: 1X polar plot for the temporary axial accelerometer at NDE generator plane

The red curve is the Reference run (REF RUN) data. The black curve is the final run (RUN4) data. It can be observed that the trial weight decreased the 1X vector from 0.191 in/s rms to 0.026 in/s rms

From the polar plots ( Fig 13 and Fig 14) it can be concluded that the installation of the final weights decreased considerably the vibration for all the probes. The balance job was a success. In order to check the vibration level at FSFL, the data were stored for a last run (**RUN 5**). The machine ran at FSFL during several hours in order to be sure that the vibrations were stable

The table below shows that the level of vibration at FSFL for the entire generator probes decreased by around 60%

| RUN | 39VS91 mil pk-pk | 39VS92 mil pk-pk | 39VS101 mil pk-pk | 39VS102 mil pk-pk | Temp Axial in/s rms | BB11 in/s pk | BB12 in/s pk |
|---|---|---|---|---|---|---|---|
| **RUN REF FSNL** | **0.81 @ 117°** | **2.31@23°** | **0,247@311°** | **1,33@191°** | **0.195@60°** | **0.085@78°** | **0.092@253°** |
| RUN FINAL FSNL | 0.198@128° | 0.661@25° | 0.072@43° | 0,173@154° | 0.029@90° | 0.005@60° | 0.010@234° |
| **Decrease X%** | **75%** | **71%** | **70%** | **86%** | **85%** | **94%** | **89%** |
| RUN FSFL Before | 0,513@140° | 2,825@18° | 0,473@141° | 1,872@166° | 0,494@46° | 0,111@75° | 0,088@246° |
| RUN FINAL FSFL | 0.282@168° | 0.777@9° | 0.259@105° | 0.591@78° | 0.193@316° | 0.042@197° | 0.030@219° |
| **Decrease X %** | **45%** | **72%** | **45%** | **68%** | **60%** | **62%** | **65%** |

Figure 16: Tabular list of the 1X vibration before and after balancing at FSNL and FSFL.

Previously in this report, it was explained that the axial vibration could be decreased by reducing the radial vibration. This remark can be confirmed since the radial excitation was reduced and the axial vibrations also

# 4 Conclusion

Prior to attempting any balance corrections, a proper vibration analysis should be done to determine the likelihood that the machine is in fact out of balance. Making balance corrections to a machine with some other fault can in many cases reduce the vibration amplitudes. However, if balance corrections are made to a machine that is not out of balance to start with, the forces generated by the fault will still exist even though balance corrections may reduce the amplitude at some measurement points

In our example, the root cause of the important axial vibration was due to the coincidence between the operating speed frequency of the generator with an axial structural resonance frequency. A temporary solution to decrease the stress in the generator support bearing was to balance the unit in order to decrease the axial vibration. The paths of vibration are really complex and it is difficult to predict the impact of the residual unbalance on axial vibration. This example confirms that in some cases the fact to decrease radial vibration can have an important impact on axial vibration

The method to balance this unit using static-couple method combined to influence vector method is atypical but really powerful. If a trim balance is done starting with one of both method and the result is not as well as expected, it exists a way to pass directly to the other method

Table 1: Algorithm's performance and separation quality

Figure 2: The logo of the SURVEILLANCE7 International Conference

All equations must be numbered consecutively. Their numbers must parenthesised at the end of the corresponding line as in the following example:

$$\Phi\left(Ax + n\right) \tag{1}$$

# References


[1] John J Yu. (2009) Relationship of influence coefficients between Static-Couple and Multiplane Methods on two plane balancing. Volume 29 N°1 ORBIT.

[2] Donald E.Bently with Charles T.Hatch. Fundamentals of Rotating Machinery Diagnostics Edited by Bob Grissom.