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Dynamic Balancing of the Axial Compressor of a Gas Turbine using the Influence Coefficient Method

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Abstract

In the present case, the axial compressor rotor of a gas turbine was dynamically balanced in-situ and in two-planes. For this procedure two accelerometers were mounted directly into the bearing journals case of the compressor, and one phase reference sensor was mounted observing the rotating shaft. A data acquisition systems was used to gather all the information, and to obtain the time domain graph, vibration spectrum and filtered vibration vectors. SpectraQuest's balancing software: BalanceQuest, was used to obtain the correction weights magnitude and angular position for both planes, with the influence coefficient method.

Figure 1: Axial Compressor of the Gas Turbine

1. Vibration Analysis

The studied machine is a gas turbine (figure 1), in which de gas generators nominal velocity at full charge is 10,739 RPM. Two accelerometers were mounted in each of the two journal bearing of the compressor, in the positions C1R and C2R (figure 2). For the phase angle a infrared reference sensor was mounted vertically, observing a mark in the shaft.

The measured global vibration amplitudes and 1X filtered vibration amplitudes were high for this type of machine. The vibration spectra for the suction side of the compressor journal bearing, taken in point C1R and at 30^o from the horizontal, are shown in figure 3. In this frequency domain graph appears a very high 1X vibration peak with synchronic component (2X, 3X, ...). The global vibration measured in this point was 28.9 $mm/s\ rms$, and the 1X filtered vibration vector was 27.2 mm/seg @260^o.

In the time domain graph (figure 4) appears a sinusoidal pattern, with a period that equals the rotation velocity of the shaft, characteristic of an imbalance problem.

The vibration measurements made in the discharge side journal bearing, C2R with the same angle of measurements made in bearing C1R, were very similar in shape to the time domain of figure 4, but with less amplitude. The global vibration measured in this point was 6.1 $mm/s\ rms$, and the 1X filtered vibration vector was 5.8 mm/seg @212^o.

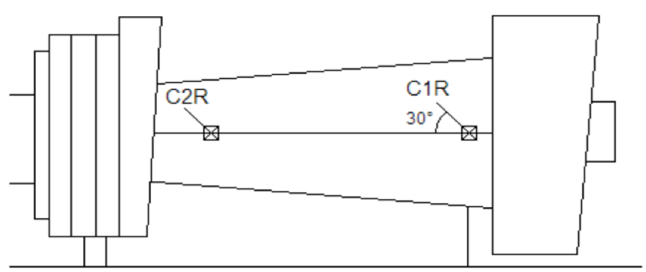

Figure 2: Accelerometers location

This 1X vibration vectors were used as the initial data for performing the balancing procedure.

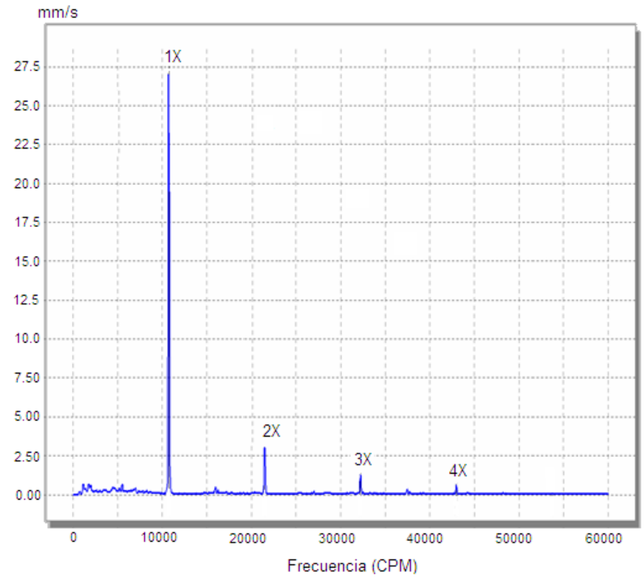

Figure 3: Vibration spectrum for C1R

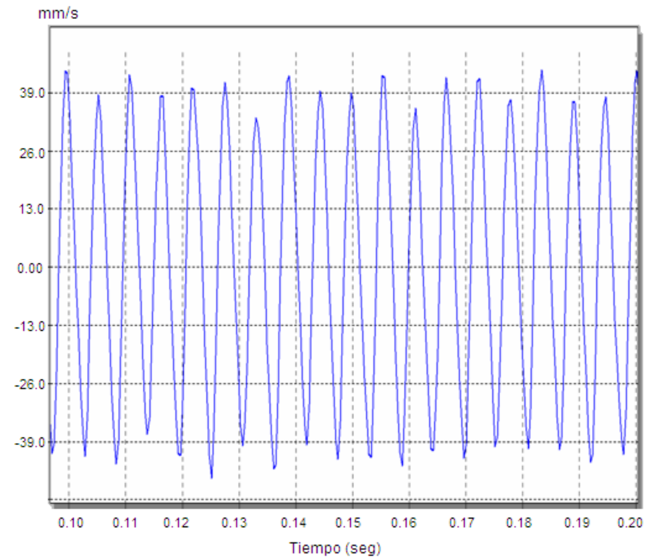

Figure 4: Time domain graph for C1R

2. Balancing Procedure

Thus the axial compressor rotor of the gas turbine operates above its first critical velocity, it must be balanced in-situ and at operational conditions. The influence coefficient method was used to calculate the magnitude and angular location of the corrective weights to be distributed in the two balancing planes.

In the second run, a trial weight of $7.5\ grams$ was located at 270^o in the suction side balancing plane (1); and in the third run a trial weight of $14.5\ grams$ @ 150^o was mounted in the balancing planes at the discharge side of the compressor (2). Table 1, shows the vibration vectors measured in the tree experiments.

Table 1: 1X filtered vibration vectors

	Bearing 1	Bearing 2
Run 1	$27.2\ mm/s$ @260^o	$5.8\ mm/s$ @212^o
Run 2	$29.7\ mm/s$ @276^o	$6.3\ mm/s$ @187^o
Run 3	$23.6\ mm/s$ @276^o	$6.1\ mm/s$ @242^o

This data was used with the influence coefficient method to obtain the correction weights vectors for each balancing plane. Sensibility vectors were obtained from the following equations:

$$\vec{S_{11}} = \left[\frac{\vec{W_{P1}}}{\vec{B_{11}} - \vec{O_1}}\right] \tag{1}$$

$$\vec{S_{21}} = \left[\frac{\vec{W_{P1}}}{\vec{B_{21}} - \vec{O_2}}\right] \tag{2}$$

$$\vec{S_{12}} = \left[\frac{\vec{W_{P2}}}{\vec{B_{12}} - \vec{O_1}}\right] \tag{3}$$

$$\vec{S_{22}} = \left[\frac{\vec{W_{P2}}}{\vec{B_{22}} - \vec{O_2}}\right] \tag{4}$$

where,
S_{md}: Sensibility vectors
m: measurement plane
d: correction plane
W_{P1}: Trial weight vector - Plane 1
W_{P2}: Trial weight vector - Plane 2

The imbalance vectors in the two correction planes ($W_{C1}\ y\ W_{C2}$), were obtained using equations (5) and (6).

$$\vec{W_{C1}} = \frac{(\vec{S_{12}} \times \vec{O_1}) - (\vec{S_{22}} \times \vec{O_2})}{\left(\frac{\vec{S_{12}}}{\vec{S_{11}}}\right) - \left(\frac{\vec{S_{22}}}{\vec{S_{21}}}\right)} \tag{5}$$

$$\vec{W_{C2}} = \frac{(\vec{S_{21}} \times \vec{O_2}) - (\vec{S_{11}} \times \vec{O_1})}{\left(\frac{\vec{S_{21}}}{\vec{S_{22}}}\right) - \left(\frac{\vec{S_{11}}}{\vec{S_{12}}}\right)} \tag{6}$$

Solving equations (5) and (6), yields the correction weights vectors for each balancing plane.

$$W_{C1} = 7.8grs\ @\ 209.90^o$$

$$W_{C2} = 38.8grs \ @ \ 79.81^o$$

After mounting the correction weight in the calculated positions in the rotor, vibration levels dropped to normal values for this machione again. Figure 5 shows the vibration spectra before and after the balancing procedure.

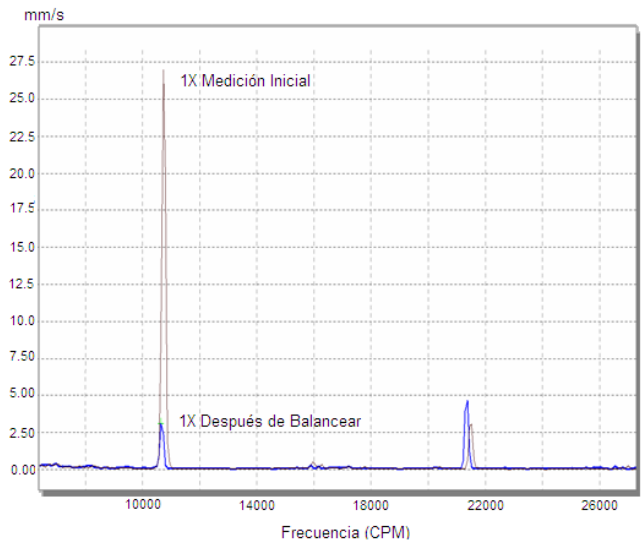

Figure 5: Vibration Spectra in C1R, before and after balancing

3. Using BalanceQuest Software

For simplifying calculations, BalanceQuest Software, by SpectraQuest, Inc was used.

Selecting a 2-Planes balancing and entering the data in the respective fields, yields to the correction weights vectors for the two balancing planes. In this case $7.82grs \ @ \ 209.90^o$ and $38.83grs \ @ \ 79.81^o$ as shown in figure 6.

4. Results

After mounting the obtained correction weights of $7.8grs \ @ \ 210^o$ and $38.8grs \ @ \ 80^o$ in each plane respectively, the vibration amplitude values for this machine dropped to normal levels. Figure 5, shows how the 1X filtered vibration component produced by the imbalance dropped from $27mm/s$ to $2.6mm/s$ after balancing.

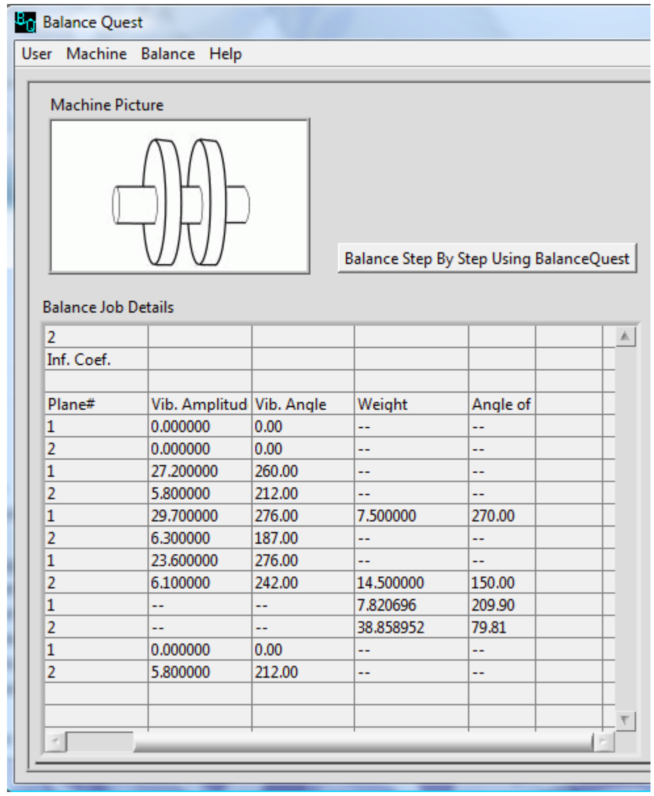

Figure 6: SpectraQuest's BalanceQuest software: Influence Coefficient Method 2-Planes Balancing, results window

References

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