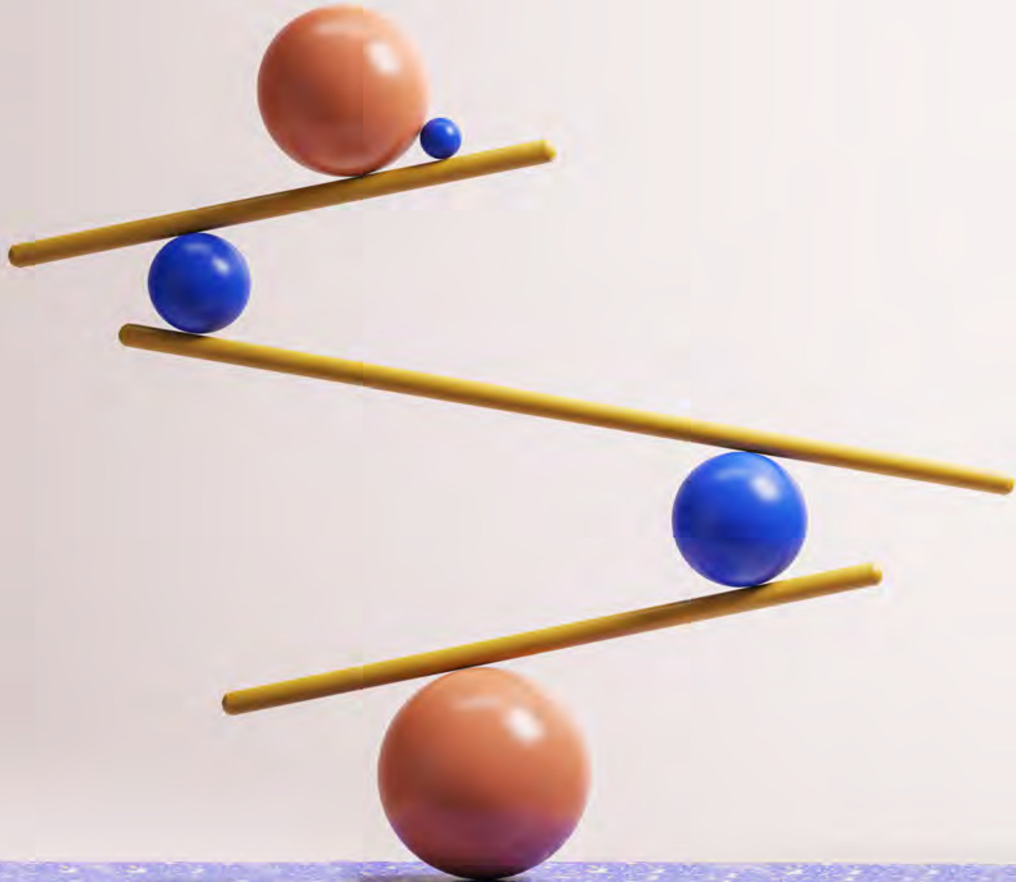
HIGH-SPEED BALANCING



Brian Hantz, Qingyu Wang and Brian Pettinato, Ebara Elliott Energy, consider API acceptance criteria for the high-speed balancing of turbomachinery rotors.

During planned or unplanned maintenance, turbomachinery rotors are typically balanced at low speed before reinstallation. However, rotor repairs or replacement of rotor parts can cause changes to the rotor dynamics that are not detected during low-speed balance. At operating speed, dynamic unbalances can cause excessive rotor vibration. High-speed balancing can minimise rotor vibration throughout the entire speed range and relieve residual stresses introduced during the repair process. Operation in a high-speed balance facility can also be used to verify the unbalance response analysis, similar to a mechanical test.

Acceptance criteria for high-speed (i.e., at-speed or operating speed) balancing of turbomachinery rotors, as specified in API standards, are based on either pedestal velocity or shaft displacement.

During testing, rotor response is measured during acceleration to maximum speed and deceleration to minimum speed. Values are plotted on the same coordinates as for the rotor response analysis. The plot of shaft vibration and phase angle of unbalance vs shaft speed is called a Bode plot. Bode plots indicate the location of critical speeds, the change of shaft vibration with speed, and the phase angle of unbalance at any speed.

API balancing acceptance criteria

Table 1 summarises the high-speed balancing criteria from API standards. Abbreviations are as follows:

- API: maximum allowable low-speed residual unbalance specified as:
  - SI units: $U = 6350\ W/N$ (1)
  - USCS units: $U = 4\ W/N$ (2)

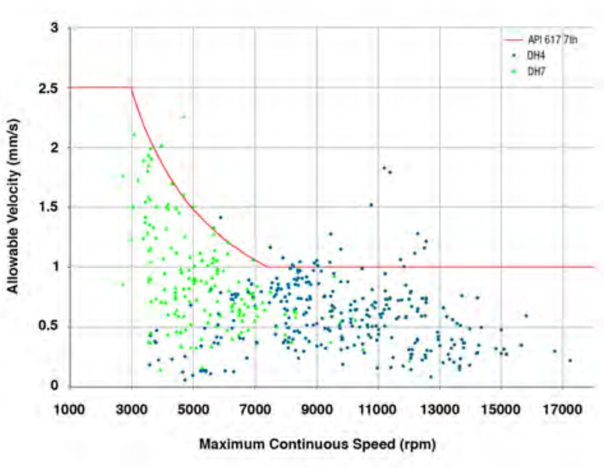


Figure 1. Maximum continuous speed vs allowable velocity (shop order data).

Table 1. API balancing criteria summary				
			Acceptance criteria	
API No.	**Application**	**Edition**	**High speed**	**Low speed**
API 611	General purpose steam turbines	5th (2008)	MA	API
API 612	General purpose steam turbines	6th (2005) 7th (2014)	MA V1, V2, D1	API
API 613	Special purpose gear	5th (2003)	None	API
API 616	Gas turbines	5th (2011)	V3	API
API 617	Axial and centrifugal compressors and expander compressors	7th (2002) 8th (2014)	V1 MA	API
API 672	Packaged, integrally geared centrifugal air compressors	4th (2004)	OEM	OEM
API 684	Rotordynamics tutorial	2nd (2005)	V2	API
API 687	Rotor repair	1st (2001)	V1	API

Where:

U = the residual unbalance measured in units of mass and distance (g-mm [oz-in]).

W = the bearing static load in kgf (lbf).

N = the maximum continuous speed in rpm.

- D1: the maximum allowable shaft vibration (1x filtered and runout compensated) shall not exceed 25.4 µm peak-to-peak at any response or 12.7 µm peak-to-peak over operating speed range for probes near the bearings.
- MA: mutually agreed.
- OEM: manufacturer's standard balancing procedure.
- V1: for all speeds at or less than 3000 rpm, the pedestal vibration shall not exceed 2.5 mm/s root mean square (RMS). For speeds above 3000 rpm, the pedestal vibration shall not exceed the calculated value of (7400/N) mm/s or 1 mm/s RMS, whichever is the greater, where N is the maximum continuous speed in rpm. The criterion applies to the major axis velocity.
- V2: velocity calculated such that the maximum allowable unbalance force at any journal at maximum continuous speed shall not exceed 10% of the static loading of that journal.
- V3: the acceptance criterion, only used in API 616, 5th edition, is a combination of residual unbalance and pedestal vibration.

In current low-speed balancing standards, Equation 1 (g-mm) or Equation 2 (oz-in) are predominant. ISO standards use Grade, which limits the velocity of the centre of gravity (cg) of the rotor, and they are essentially the same as the API standards. The limit of the unbalance amount (or eccentricity of cg) assumes that the rotor can be simplified as a single mass.

For high-speed balancing, limiting the unbalance amount cannot be used directly since the rotor cannot be simplified as a single mass. An alternative, the V2 method, limits the force induced by the unbalance by 10% of the static weight. The V1 method for high-speed balancing is only related to the operating speed, and may be significantly different as compared to the V2 method.

Figure 1 illustrates the relationship between V1 and V2 based on data from 723 Ebara Elliott Energy shop orders where the red line is the V1 method, and all the dots are calculated based on the V2 method (0.2 'g', i.e. 10% static load per pedestal) using maximum continuous speed, rotor weight and pedestal stiffness. The stiffness values for the DH4 and DH7 pedestals were provided by the vendor: 560 N/µm for DH4 and 1334 N/µm for DH7.

Although both velocity and displacement measurements are available in balancing facilities, the velocity measurement is usually used for balancing and balancing criteria because of the relative stable situation: the velocimeters are built-in within the pedestals so the quality of measurement stays the same regardless the rotor being balanced. The eddy-current probes might be shifted depending on the rotor and bearing combination. Sometimes the probes might not even be at a burnished area.

If the probe location at the balancing facility is different from the actual machine location due to the bearing housing configuration, a multiplier can be used to adjust the acceptance criteria as suggested in ISO 11342, Section 8.2.5. The value of the multiplier can be derived through comparison of the vibration amplitudes at the two locations by performing rotordynamic analysis.

Pedestal dynamics

A pedestal model is generally needed for comparing the measured unbalance response to the predicted response. There are different ways to characterise pedestal dynamics. A simple way is to use mass and stiffness. Usually the original pedestal manufacturer (vendor) provides the values, and these values are used in the balancing criteria and rotordynamic analysis. Another way is to use frequency response functions (FRFs). Since it appears that using the pedestal FRFs would improve the rotordynamic predictions, Ebara Elliott Energy initiated a project to acquire accurate FRFs for the pedestals in the company's balancing facility.

To obtain improved pedestal transfer functions, the company contracted with a consultant to perform modal tests. However, after repeating the tests with Ebara Elliott Energy equipment, the pedestal responses did not agree with expectations.

To determine the root cause of the discrepancies in the FRFs, the company performed a series of tests by moving pedestals, testing the rails, lifting and dropping the pedestals and retesting in the same location, adjusting pedestal bed bolt torque, and adjusting bearing cap bolt torque.

While there may be other contributing factors for inconsistent measurements, results indicated that the most important parameter for a consistent result is the torque of the bed bolts. Based on these results, the company conducted further tests and balancing using a pneumatic torque wrench with a bed bolt torque of 600 ft-lb (813 N-m) for all pedestals. Test models included the single degree of freedom (SDOF) curve-fit model, the multiple degree of freedom (MDOF) curve-fit model, the vendor model, and the plug (added mass) model, where a known mass is added to the pedestal during measurement to characterise the dynamics.
An example plot showing the amplitude of the measured FRFs is provided in Figure 2. An example plot showing the measured FRFs and the identified models is provided in Figure 3.

From all the measured FRFs, the following observations can be made:

- The dynamics of different pedestals are largely different.
- The dynamics of the horizontal and vertical directions are different both in terms of peak locations and magnitude.
- The cross-coupling dynamics are at least a magnitude smaller than the principal dynamics in this instance.

For the models, the following observations can be made:

- The SDOF model, the vendor model, and the plug model are relatively close to each other.
- Different models usually have closer values in the Y (vertical) directions (2 - 27% from the vendor model stiffness). The discrepancies in the X (horizontal) direction are usually larger (14 - 88% from the vendor model stiffness).

Unbalance verification

After obtaining more accurate pedestal transfer functions as described above, Ebara Elliott Energy performed a rotordynamic analysis of an Elliott® 46MB rotor using the different types of pedestal test models. The rotor was balanced in the company's balancing facility, and residual unbalance subtraction was used to perform the unbalance verification. The unbalance verification tests were performed with DH7 pedestals (stiffening on). The rotor was later assembled into the compressor and passed all tests on the test floor before being shipped to the field where it is running successfully.

The rotor, as shown in Figure 4, was approximately 1650 kg with 6 x 3 in. bearings. The company used standard bearing models (measured bearing clearances and oil inlet temperatures were used, but the oil lift grooves were not considered). The company analysed the system using different pedestal models

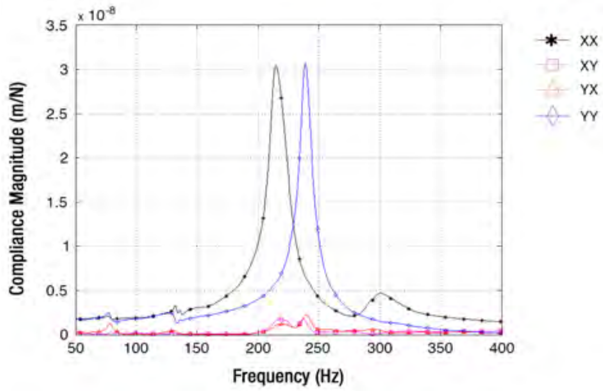


Figure 2. Measured FRFs for DH7 – pedestal 1 – stiffening off.

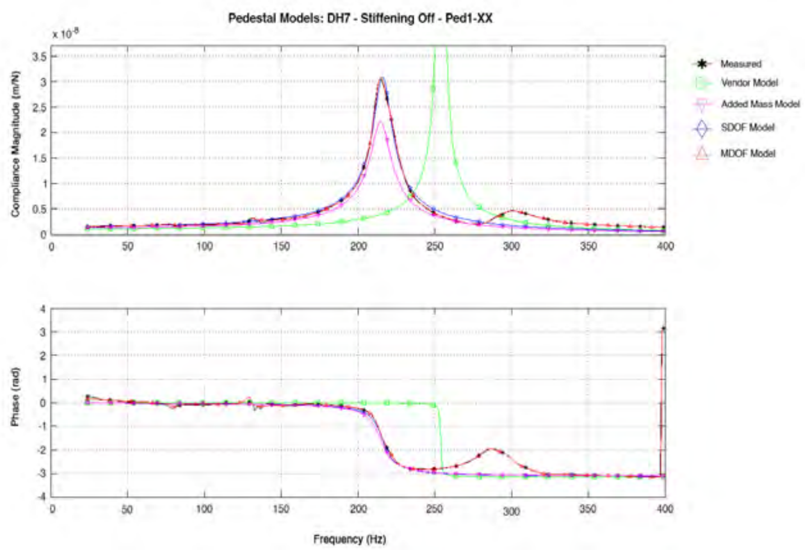


Figure 3. Measured vs identified models for DH7 – pedestal 1 – stiffening off - XX.

and included the mass of the bearing and bearing adapter in the analysis.

Results show that there are no significant differences in unbalance response between different pedestal models. Additionally, all models predict higher first critical speed than the measured value (~200 - 300 rpm higher, 7 - 20% above the measured first critical speed), and lower than the rigid support (~100 rpm lower) (Figure 5). The fact that there are no significant differences in unbalance response between the different pedestal models is to be expected since the stiffness of the pedestals from all models is more than 3.5 times the maximum bearing stiffness for DH7 stiff in both X and Y directions. API 617 considers the supports to be essentially rigid when it is more than 3.5 times the maximum bearing stiffness. Therefore, the effect of the differences between models is not manifested in this particular case. However, for other cases where the bearing stiffness is higher relative to pedestal stiffness, large differences will show up.

The conflicting results between measurement and prediction in the balancing facility may be attributed to one measurement that was not taken previously: the actual bearing centreline locations after rotor installation. Unlike job bearing housings where the bearing locations are known, the bearing locations in the balancing facility are set by manually moving each pedestal with a hand crank in an attempt to line up the proximity proves with the burnished areas. This does not lend to high accuracy, and axial deviations of a few cm can be expected.

Although the actual bearing locations in the balancing facility are impossible to know now that the rotor has since been removed, the rotordynamic models can be modified to move the bearing locations. For the Elliott 46MB compressor rotor balanced in the DH7 pedestals, moving the bearings outboard as much as possible results in agreement within 5% between the predicted and measured critical speed as shown in Figure 6.

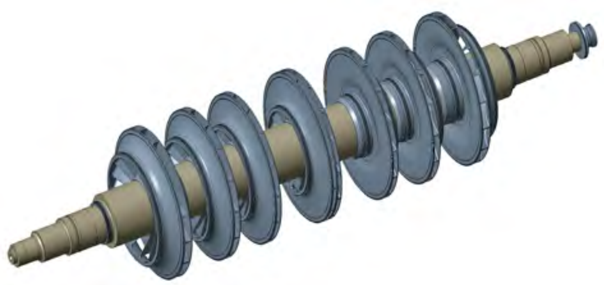


Figure 4. Rotordynamic analysis model - 46MB rotor.

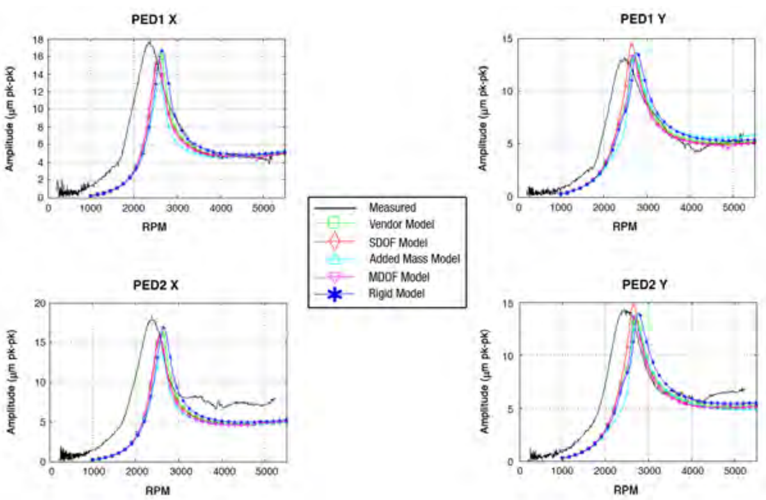


Figure 5. Balancing facility measurement vs prediction.

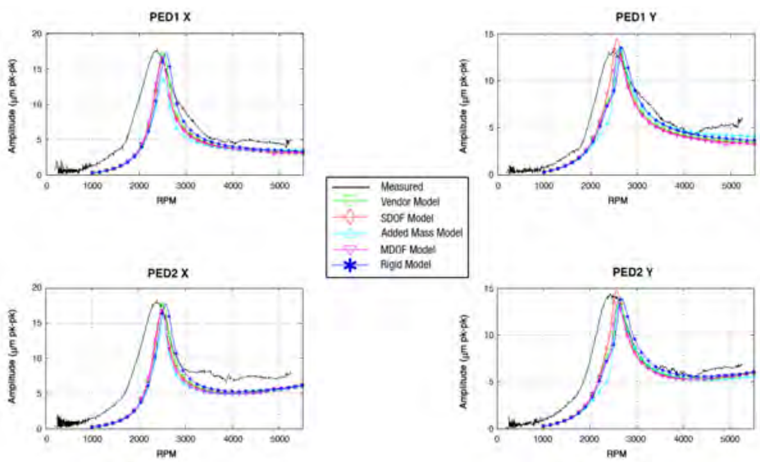


Figure 6. DH7 benchmark case with bearings shifted outboard in model.

Recommendations

In conclusion, balancing in a high-speed balance facility provides a better balance than low-speed balancing.

Operation in a high-speed balance facility also provides the opportunity for unbalance response verification, which is not available from low-speed operation.

When performing unbalance response verification in a high-speed facility, the following should be considered:

- All bolts, including bed bolts and bearing cap bolts, should be tightened to proper values to provide consistent pedestal characteristics.
- Further refinement of the pedestal model would not provide a benefit unless the pedestal stiffness is below 3.5 times the maximum bearing stiffness. The vendor pedestal model was sufficient in this case, as all models used in the analysis yielded similar results.
- The relative vacuum conditions in the balancing facility have no discernable impact on the bearing dynamics and rotordynamic performance. Tests with and without vacuum conditions yielded nearly identical vibration plots.
- The relative inaccuracy inherent with rotor installation in a balancing facility can result in varying bearing spans and probe locations, which must be recorded and reflected in the rotor model.

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