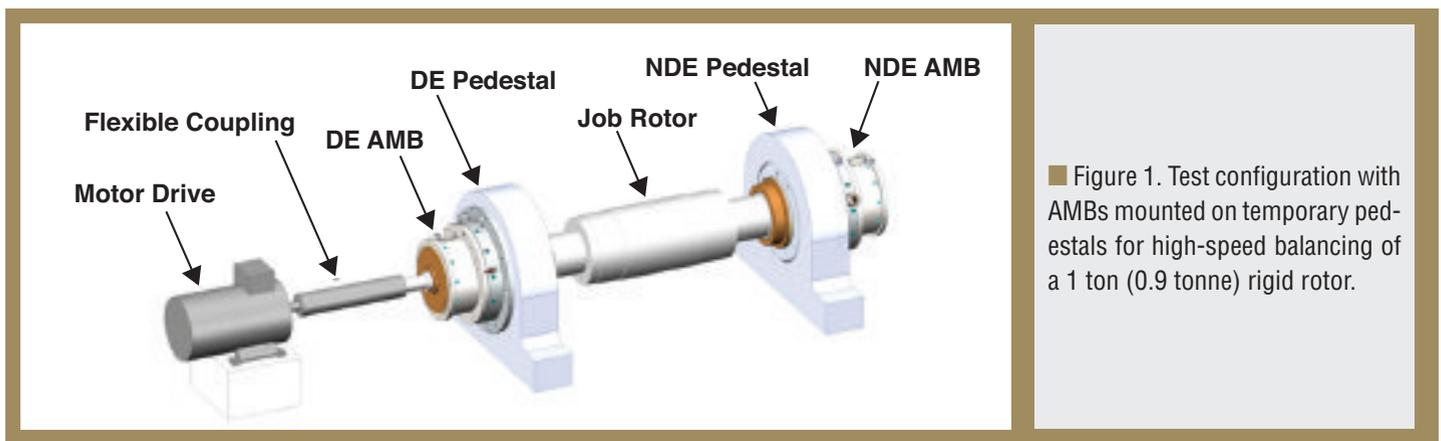


Efficient Balancing Of Active Magnetic Bearing-Supported Rotors

> Considerations, precautions needed to ensure proper techniques

BY ANDREA MASALA



■ Figure 1. Test configuration with AMBs mounted on temporary pedestals for high-speed balancing of a 1 ton (0.9 tonne) rigid rotor.

Shafts of high-performing rotating equipment experience a certain mass unbalance distribution resulting from manufacturing tolerances, component assembly and material nonhomogeneities. The residual unbalance generates uneven forces on the rotor as it spins, resulting in vibrations and the transfer of dynamic loads to the bearings and supporting structure.

To limit the negative effects of rotor vibration, ISO and API standards set balancing criteria and acceptance levels for residual rotor unbalance. Whether magnetic or fluid film bearings are used in the machine, there is a prerequisite for spinning a rotor to its rated speed. The rotor must be balanced to meet an acceptable residual unbalance level. The ability of an active magnetic bearing (AMB) system to tolerate high levels of unbalance on the rotor by applying unbalance force rejection control algorithms does not alter the balancing requirements. Rotors supported by AMBs and rotors supported by fluid film bearings are subject to the same balancing requirements.

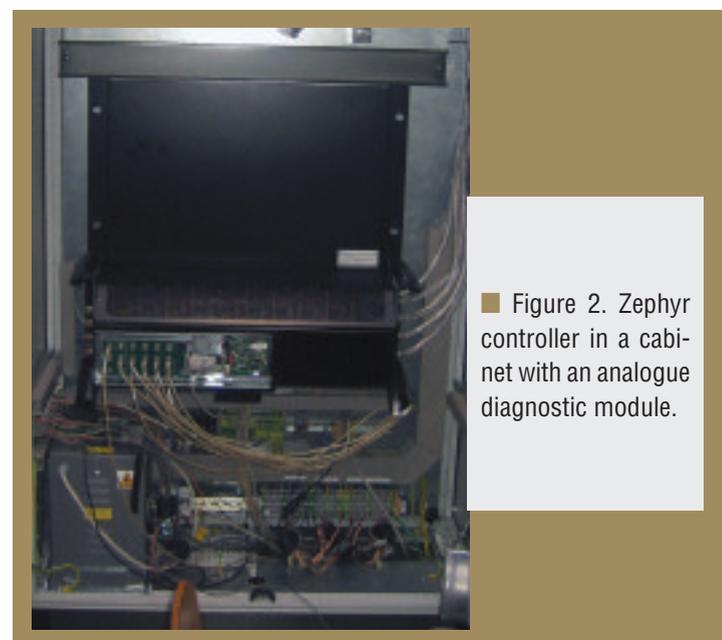
Low- and high-speed balancing options

AMB-supported rotors can be balanced with conventional

low- and high-speed balancing techniques, but some additional considerations and precautions are typically required.

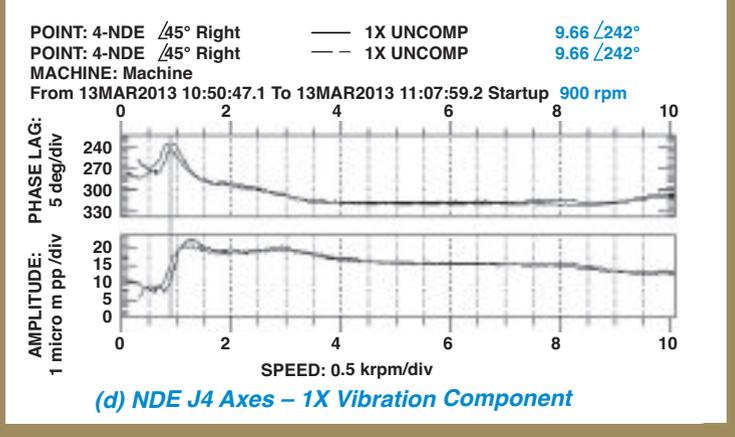
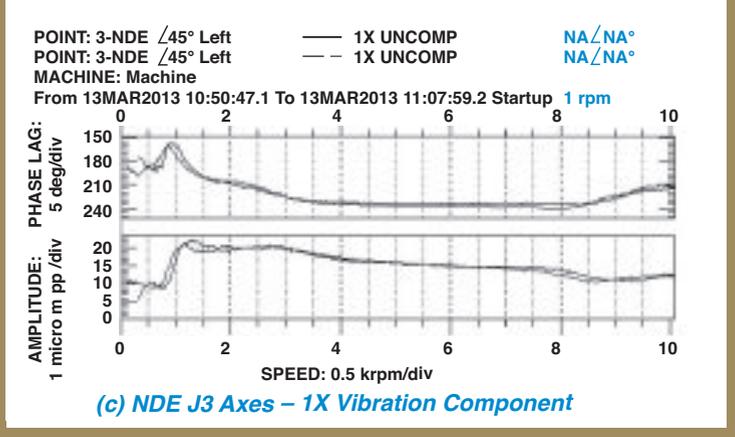
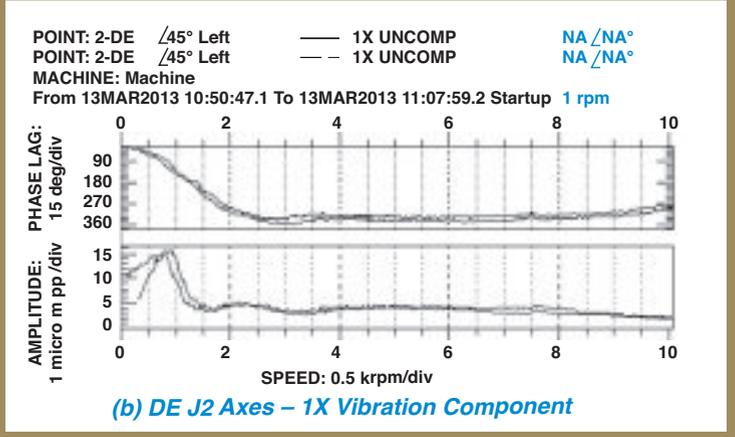
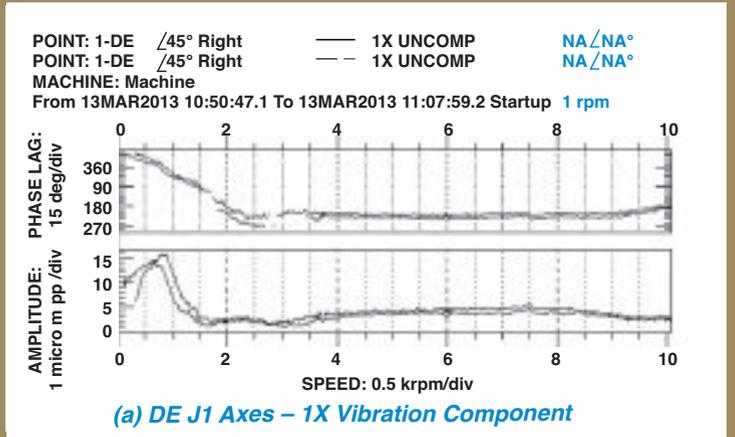
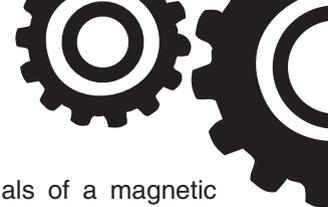
Low-speed balancing performed on rollers requires dedicated areas to support the rotor, which need to be provided for during the design phase. Supporting and spinning the

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■ Figure 2. Zephyr controller in a cabinet with an analogue diagnostic module.

Andrea Masala is the systems engineering team leader in the mechanics and dynamics department at Waukesha Magnetic Bearings. Contact him at: amasala@waukbearing.com.



■ Figure 3. Vibration amplitudes and phases captured by the customer test equipment during the final balancing check.

rotor on the soft lamination materials of a magnetic bearing could damage the laminations and result in higher eddy current losses. The damage is even more detrimental where sensor target areas are involved. Scratches or irregularities on the sensor target surface could result in high-frequency dynamic current harmonics when the machine is rotating and residual dynamic loads on the bearings. Temporary bearing surfaces — support sleeves — can protect the laminations and sensor target surfaces during balancing.

Because the position sensor target identifies the center of the rotor during rotation, the temporary bearing surfaces must be centered about the radial position sensor within a few microns. Any runout between the balancing center and the sensor will result in an unbalance that is proportional to the runout distance and the rotor mass. To minimize runout error between the sensor target and support sleeve, the machining of the sensor and sleeve should be done at the same time.

When high-speed balancing is required, as in the case of high-speed motors or flexible rotors on multistage compressors, there are several available techniques that can be used for both low- and high-speed balancing.

A common choice is to balance the rotor by running it on fluid-film bearings in a high-speed balancing bunker. Tests conducted on several rotors and materials have





demonstrated that, in this balancing technique, the rotor can be supported on the main rotor laminations without detrimental effects on the magnetic properties of the lamination. Although the fluid-film bearing stiffness and damping properties may differ from the magnetic bearing stiffness and damping, high-speed balancing on fluid-film bearings will balance the rotor to acceptable limits. That occurs only if the rotor mode shapes with the fluid-film bearings remain the same with the magnetic bearings. A rotor-dynamic analysis prior to selecting the balancing strategy can ascertain whether this condition is met.

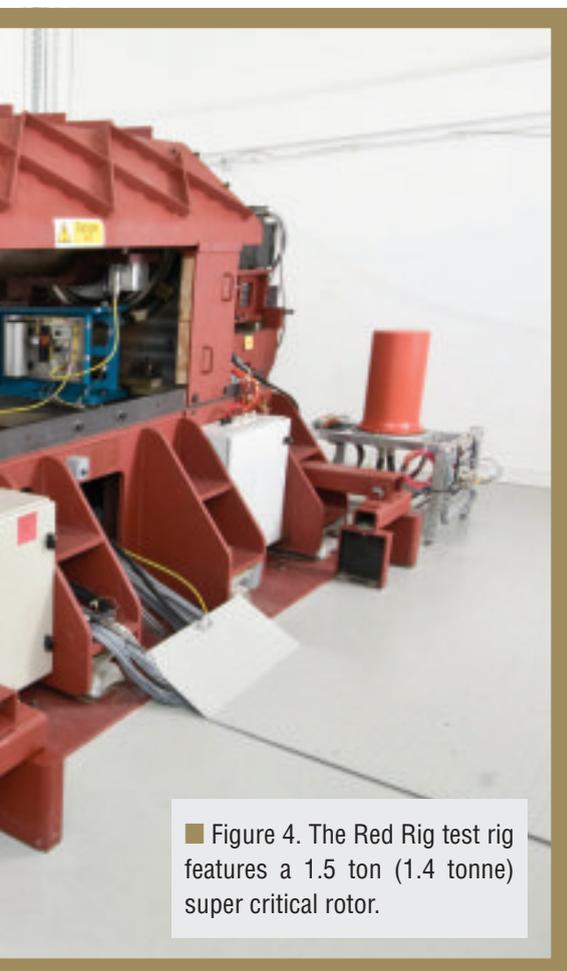
When significant differences between rotor mode shapes with fluid-film bearings and magnetic bearings are predicted, high-speed balancing should be performed with the job AMBs. This approach is also advisable when a high-peripheral speed and low specific loads on the rotor laminations are not compatible with fluid-film

bearings and instability on fluid-film bearings is expected. Practical issues related to the availability of fluid-film bearings for testing may be a further reason to use this approach.

In some cases, a hybrid balancing procedure may be adopted, with low-speed balancing performed on fluid-film bearings and high-speed balancing performed on magnetic bearings.

Rotor balancing on magnetic bearings has the advantage of achieving a better balancing with the fewest balancing runs. The unbalance force rejection algorithms typically adopted on magnetic bearings can be activated on the spinning rotor before it is balanced and across the balancing trial runs, allowing the rotor

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■ Figure 4. The Red Rig test rig features a 1.5 ton (1.4 tonne) super critical rotor.

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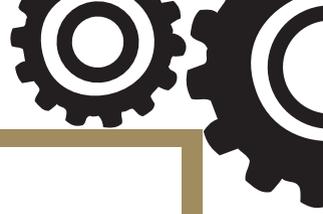
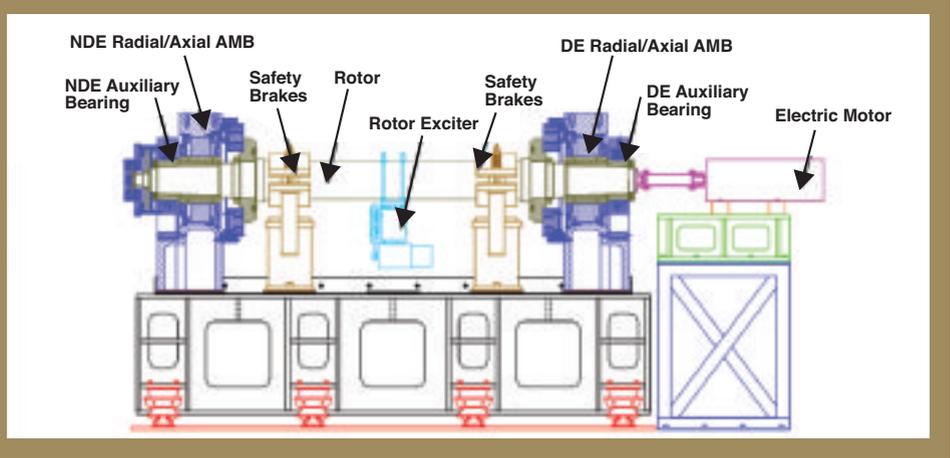


Figure 5. Radial/axial active magnetic bearings were balanced using the Red Rig test rig at speeds up to 7500 rpm.

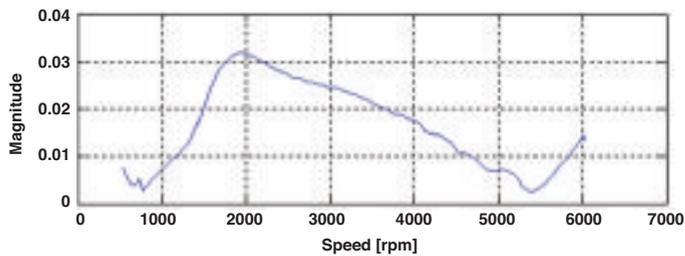


to reach high speed during each trial run, even with large amounts of unbalance. This achieves a better final balancing in the operating speed range with a limited number of balancing runs.

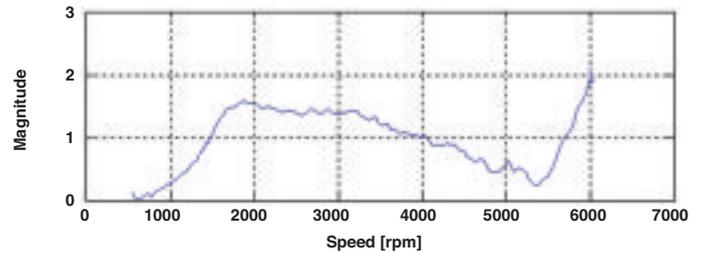
On high-speed machines where the balancing planes are located close to the mid-span of the rotor, the ability to perform balancing on the assembled machine is limited, as major disassembly would be required for each trial and balancing run. A practical solution is to build temporary pedestals on which to

mount the magnetic bearings and support the rotor. In addition to providing ease of access, this solution has the benefit of allowing the rotor balancing and the initial AMB system commissioning to occur simultaneously, dramatically reducing the time required to commission the AMB system in the machine.

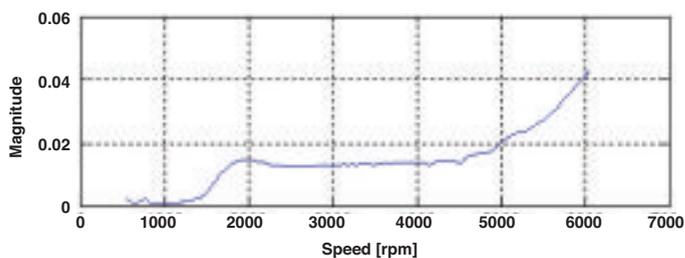
The efficiency of using AMB system software functionalities to perform high-speed rotor balancing is illustrated by two cases, one featuring a rigid rotor in an electric motor and one featuring a high-speed flexible rotor.



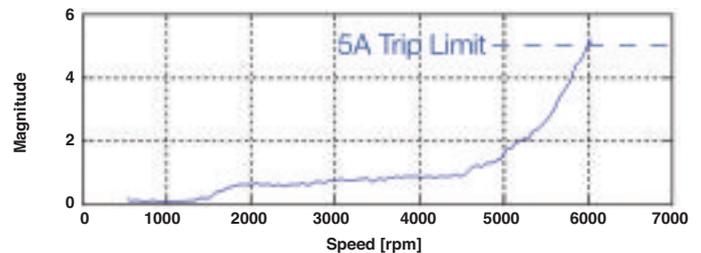
(a) DE Vibration Magnitude, mm 0-pk



(c) DE Currents Magnitude, mm 0-pk



(b) NDE Vibration Magnitude, mm 0-pk



(d) NDE Currents Magnitude, mm 0-pk

Figure 6. Radial AMB vibrations and currents after low-speed balancing.



Case 1: Pedestal-mounted AMB balancing of a rigid rotor

Rotor balancing on AMBs allows for varying levels of integration with customer resources and diagnostic platforms. In the case of a high-speed 1.5 MW electric motor rotor, the AMB system could be connected to customer test equipment. Two radial AMBs, including position sensors and auxiliary bearings, were provided by Waukesha Magnetic Bearings and mounted on temporary pedestals provided by the customer. The 1 ton (0.9 tonne) rotor was driven by a shop motor via the high-speed job coupling (Figure 1), supplied by the customer. For safety reasons, the rotor was tested in a restricted-access area of the customer workshop, surrounded by heavy concrete blocks.

The job controller, a Zephyr digital controller from Waukesha Magnetic Bearings, was used to measure the radial vibration and speed based on signals from the position sensors and keyphasor. The vibration measurements and keyphasor signals were redirected from the controller to an external signal analyzer through an analogue diagnostic module (Figure 2). The signal analyzer was used to extract the 1x revolution vibration amplitude and phase signals required to carry out the balancing procedure. The AMB controller was also connected to customer test equipment to cross-check the vibration measurements and phases across each run.

In addition to performing typical AMB commissioning activities, Waukesha Magnetic Bearings provided engineering support during the balancing process to determine the position of trial and correction masses across each run. An influence coefficient method combined with a least square approach was used to determine the correction masses of the rotor when running at the maximum speed of 10,000 rpm. Due to the rigid rotor configuration, only three balancing planes were used: one on the midspan and two close to the radial AMBs.

The vibration outputs from the analogue diagnostic module and those recorded by the customer test equipment on the final balancing check run (Figure 3) show the synchronous (1x) vibration measured at each AMB sensor. The final synchronous peak-to-peak vibrations were less than 20 μm , putting the residual unbalance well within the ISO G2.5 limit for this rotor.

Case 2: Pedestal-mounted AMB balancing of a flexible rotor

The commissioning of the Waukesha Magnetic Bearings Red Rig (Figure 4), a test rig with a 1.5 ton (1.4 tonne) supercritical rotor, used AMB controller functionalities to streamline the balancing process. The Red Rig rotor, supported by two radial and two thrust AMBs mounted on pedestals (Figure 5), runs to a maximum speed of 8000 rpm and presents a first bending mode at 6800 rpm. For safety reasons dictated by the job configuration, the rig was evaluated at a maximum speed of 7500 rpm during commissioning.

The main challenges in balancing the Red Rig rotor were related to the presence of a flexible mode in the operating speed range and potential thermal effects on the rotor resulting from limited cooling flow availability. Built-in harmonic capture functionalities in the controller software allowed the extraction of the 1x revolution amplitude and phase information from the vibrations and current signals without external equipment and in real time.

The rotor presented four balancing planes, two on the in-board thrust bearing discs located at each side of the rotor and two on the auxiliary bearing discs.

The first stage of the rotor balancing process was aimed at correcting the static and dynamic unbalance of the first two rigid modes by running the rotor up to 4000 rpm with different combinations of trial masses and calculating the correction masses accordingly. The run-up vibration and AMB current measurements on the drive end (DE) and nondrive end (NDE) axes after the low-speed balancing are shown in Figure 6. While the rigid modes were balanced satisfactorily with the low-speed balancing, the rotor still presented a significant residual unbalance to excite the first bending mode at 6800 rpm. The increasing level of vibration and current as the rotor approached the first bending mode limited the running speed to 6000 rpm, when the

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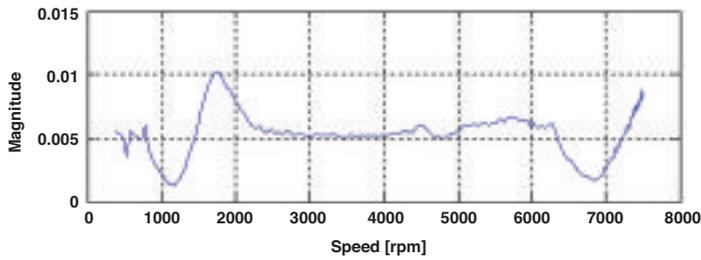
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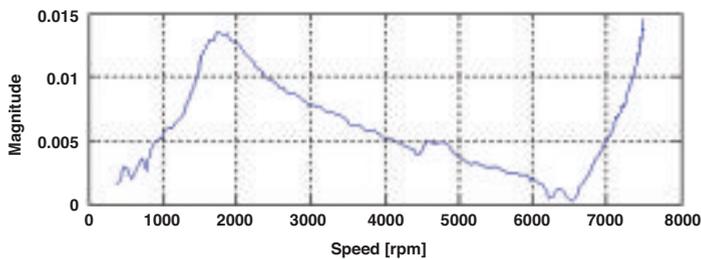
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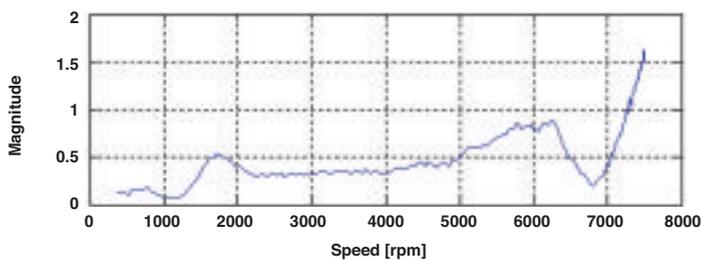
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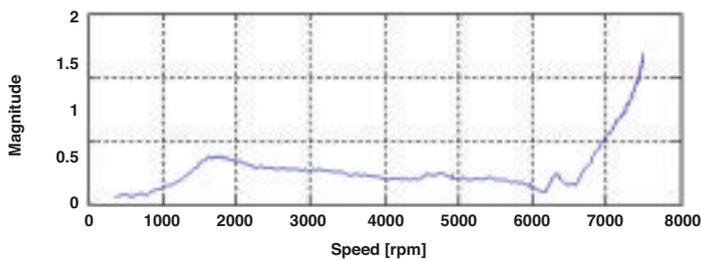
(a) DE Vibration Magnitude, mm 0-pk



(b) NDE Vibration Magnitude, mm 0-pk



(c) DE Currents Magnitude, mm 0-pk



(d) NDE Currents Magnitude, mm 0-pk

■ Figure 7. Radial AMB vibrations and currents after final unbalance correction.

conservative dynamic current trip threshold of 5A was exceeded at the NDE bearing.

To achieve a higher rotating speed, high-speed rotor balancing was also performed. Due to the limitation on the maximum achievable speed, the measurement speed range for this second stage of the rotor balancing was bounded from 4000 and 6000 rpm.

Influence coefficients for the high-speed balancing were calculated using a least square approach, with measurements weighted according to the rotor speed. Because the measurements presented repeatable and smooth amplitude and phase trends, the stochastic and optimization capabilities offered by the balancing software were not used. An additive mass approach was adopted across the trial runs to reduce the time for trial weight placements.

The final balancing check run measurements with the added correction masses (Figure 7) confirmed a dramatic reduction of the vibration and current magnitudes as the rotor ran across the first bending mode (6800 rpm). Note the difference in scale for the magnitudes in Figure 7 compared to Figure 6. Although some of the predicted thermal effects increased the vibration levels above 7000 rpm, the balancing procedure confirmed the effectiveness of using built-in controller software combined with balancing algorithms to address a challenging balancing problem.

Summary: Planning ahead for efficient balancing

Active magnetic bearings offer advanced capabilities to perform rotor balancing, especially on challenging high-speed machines. Built-in instrumentation and software reduce requirements for external instrumentation and software. The availability of special control algorithms, commonly known as unbalance force rejection control, also allow high-speed trial runs even with initially high levels of residual unbalances. Customers can also integrate their own diagnostic platforms into the balancing for result verification. Further integration of diagnostics and balancing algorithms are envisaged for the near future, which will further simplify the balancing process for OEMs and end users.

Efficient balancing of an AMB-supported rotor does not start with the test runs and application of algorithms, however. It starts during the machine design phase, with identification of the appropriate balancing strategy and the required equipment and rotor features. With these provisions determined early, customers can avoid delays in the commissioning of the equipment and may be able to package balancing and commissioning services. CT2