AVOIDING NONLINEAR RESPONSES OF TURBINE-GENERATOR ROTORS IN PRACTICE

Z. Racic[1] and M. Racic[1]

[1]Z-R Consulting, 53214 West Allis, WI, USA
E-mail: *mzracic@gmail.com*

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Abstract: Many instances of vibration that appear as nonlinear turbine-generator rotor behavior in industry are misdiagnosed and unsuccessfully resolved, especially when dealing with rotor systems with "significant" mass eccentricities. This is due in part to the oversimplification or neglecting in standard models of the full effects of distributed eccentricity and torque. Using a new, and thus far qualitative understanding of these effects, excellent success has been seen in resolving otherwise "intractable" turbine-generator vibration problems in power plants.

Introduction

Standard power generation industry practices of inspection, machining, balancing, assembly and alignment have been developed and streamlined by OEMs for newly manufactured rotors, and implicitly assume that rotors are within ideal dimensional specifications. However, applying these standard practices in the service industry can fall short for rotors with eccentricities outside such specifications, and by design do not catch the errors that cause true dynamic problems, because they are assumed not to exist or their effects are fully unrecognized within "traditional industry practices". These standard practices are based on theory and modeling that represent eccentricities as an equivalent unbalanced disk, sometimes even with a massless shaft. However, this representation oversimplifies and inaccurately predicts the real life behavior of "significantly" eccentric rotors, and disconnects the true interaction between the forces/motion originating on the rotor and the associated reaction forces in the bearings. More importantly, because of the assumptions and simplifications required for current models, the solutions suggested by the standard models to remedy high vibrations are not effective in practice when dealing with significantly eccentric rotors.

The mathematical representation of nonlinear rotor responses results from the problem setup and assumptions used, which define the rotor-bearing system as a closed system, with the equations of motion derived as time independent, without any particular phenomena dependent on acceleration. In the real world, turbine rotors behave as a continuous open system with continuous energy input (as torque), and the full description of a rotor behavior (especially the phase shift) requires dependence on the presence of acceleration of the rotor's angular velocity. When dealing with eccentric rotors in particular, this continuous energy input is converted from power into "torque moment" forces (1-Racic, 2014) generated through rotation (best viewed in a rotating reference frame), which are then coupled and transferred to kinetic and potential energy as linear reaction forces in the bearings (best observed in a non rotating reference frame). This arises from the unique condition of forced, non-centroidal rotation, where the rotor's true mass axis not coincident to the axis of torque input, nor concentric to the axis of rotation (up to the first critical). With the mathematical manipulation of standard models, the effect of torque on eccentric masses is ignored and true dynamics problems are viewed as "nonlinear" rotor behavior. The true dynamics of rotating bodies viewed from an inertial mechanics point of view reveals that mass eccentricities are a primary cause of "nonlinear responses" in rotors, with a relatively straightforward solution.

The current method of dealing with nonlinear rotor response (vibrations) in industry practice is by modal balancing, (a reduction of the modal deflections of rotors at the various critical speeds seen within their operating speed range).

Displacement amplitudes of modal deflections are observed from the reaction forces at bearings, measured from sensors located at a global coordinate system reference (based on the static rotor position at standstill), which implies the equivalence or appropriate representation of rotor motion as the mathematical combination of two orthogonal linear motions. In reality, forces creating “vibrations” originate in the non-inertial or rotating frame of reference, and need to be modeled within this rotational framework to suggest the truly effective solution to resolving the vibration behavior in practice.

In order to avoid nonlinear rotor response in the non rotating reference frame, distributed eccentricities inherent to rotors must be distinguished from equivalent or local unbalance, and the "rigid mode" responses arising from distributed eccentricities must be distinguished from modal responses arising from unbalance. In balancing, the rigid-mode effects of distributed eccentricities must be compensated first (shifting the rotor's mass axis to become coincident to the geometric axis of the rotor journals and couplings, with sufficient axial distribution of balance weights to avoid any induced distortion), avoiding the creation and effect of torque moments, before subsequently dealing with the rotor's residual modal deflections resulting from centrifugal force and resonant amplification at a particular resonant speed (2-Racic, 2014).

This view on rotor nonlinear responses has a practical application during turbine-generator major outages as well. With current industry practices in the service shop, insufficient attention is placed on mass eccentricities of the rotor body or of the rigid couplings. Rotor body eccentricity leads to a rotor (uncoupled) in the balancing facility being balanced inadvertently around its mass axis above the first critical, although the rotor in the field will be constrained to its geometric axis. If the couplings are eccentric (radially or axially), mass eccentricities are induced in the total rotor train between coupled rotors. Both of these conditions create the situation in which "well-balanced" rotors in the shop create nonlinear responses (vibrations) in the field after reassembly. The new economic requirements in power plants cannot afford the costs associated with correcting those “vibrations” in the field after an outage, and demand a more rigorous approach in dealing with rotor and coupling eccentricities in the shop, where they can be corrected in advance as proactive avoidance of rotor "nonlinear responses" in operation, and bring any outage to a “successful” restart.

A “successful” start up following a planned outage means that there will be no need for field balancing following the restart of the unit. There are two key processes that must be incorporated into an outage scope to ensure success. The first is to collect and evaluate (1x and 2x) total indicator runout (TIR) readings to identify any excessive bow or distributed eccentricity in the rotor body between the journals, and to verify and machine all rotor couplings and journals to within the criteria of ISO 1940. Any excessive off-squareness of the coupling faces or any taper or runout of the journals *must* be corrected by machining. The second key process is utilizing a new balancing method if the rotor body exceeds runout limits of ~0.002" eccentricity. Any such bowed or eccentric rotor *must* be balanced in a minimum of 3 balancing planes (or more accurately, 2N+1planes, where N is the highest mode reached by the rotor within its operating speed range). By applying these methods in the shop to resolve and prevent nonlinear rotor responses, and following OEM alignment procedures, a successful restart of a turbine-generator unit without field balancing can be assured.

References

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