

The Effect of Applied High Speed Balancing Method On Flexible Generator Rotor Response in Operation

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Biography

Mr. Zlatan Racic, Engineering Consultant was Manager, Vibration Analysis and Rotating Machinery Diagnostics in the Product Service Division of Siemens in the USA. In this capacity he was responsible for Fact Finding and Trouble-shooting of Turbine – Generator sets delivered and installed by Siemens. Before joining Siemens, Mr. Racic had extensive experience in Marine Diesel Engines Operations. He received his Bachelor of Science degree in Mechanical Engineering in 1979 from Milwaukee School of Engineering, and MBA degree from Nova University in Fort Lauderdale, Florida.

Mr. Juan Hidalgo is currently Dynamic Analyst and Balance Engineer at ReGENco LLC responsible for trouble-shooting turbine generator vibration problems and running ReGENco's balance bunker. He received his B. Sc. degree in Mechanical Engineer in 2000 from Universidad Nacional de San Juan, Argentina, and a M. Sc. degree in Mechanical Engineering in 2003 from University of Wisconsin Milwaukee.

ABSTRACT

Two very flexible generator rotors of identical design were refurbished and high speed balanced by a turbine-generator manufacturer other than the OEM. All shop tests and the balancing results were very good, showing no anomalies. After placing the first rotor in operation it exhibited load dependent vibration with a "thermal vector" proportional to load. A second rotor went through the same process by the same non OEM. This rotor also exhibited a "thermal vector" when reinstalled, but at one half of the magnitude of the first rotor. A major observation was that both rotors had large residual body eccentricity, with first rotor having the eccentricity twice of the second rotor. This paper deals with a Root Cause analysis of the rotors behavior described above and solution.

Keywords: Vibrations, Balancing Flexible Rotors, Eccentricities, FE Modeling

INTRODUCTION

High speed balancing of rotors in bunkers was in the past an exclusive domain of turbo-generator manufacturers. Each OEM had developed their own method of correcting unbalance based on engineering philosophy, cost and market requirements [1-5]; these philosophies were later emulated by the industry. In most cases the final results are the same as far as vibration amplitude indications at measuring locations. Measured points were typically at bearing locations.

The unique problems may arise when balancing flexible rotors which body eccentricities are above ISO 1940 limits. Typically, these generator rotors are also inescapably operating at or very close to their 3rd mode. In these cases the applied method of balancing becomes crucial in leaving rotors balanced by bringing their mass center axis coincidental with rotational axis, or bringing only the sum of modal masses in line with rotational axis.

In the first case more of the unbalance forces and moment are resolved to zero. In the second case the remaining moments from centrifugal forces are resolved by deflecting sections of the rotor in some distorted contour. The resulting deformation point can in some cases become a point of thermal sensitivity. The result of thermal distortion proportional to heat input is increased vibration. The magnitude of thermal sensitivity of the eccentric rotor depends strictly on the residual body eccentricity of the rotor prior to balancing and the applied method of balancing.

CASE HISTORY

The Unit 1 generator rotor #542 had experienced load dependant vibration change since its installation and placement in service after refurbishment and high speed balance. Rotor #777 which was removed had never exhibited “thermal” sensitivity. Following initial discussions and review of the operating problems experienced, an investigation of the rotor behavior was initiated in an effort to determine the likely root cause and potential resolution of the vibration issues of the rotor #542. It is important to note that during thermal transients, except on the exciter, there was no significant vibration response on the rest of turbine train.

The scope of analysis included the following:

- On-site data collection of rotor response for correlation with theoretical modeling work.
- FEA model of the rotor based on data provide by the Client.
- Simulation of unbalance response to approximate the current vibration behavior of the rotor.
- Evaluation of unbalance response in relation to the thermal (load dependant) behavior of the rotor in order to characterize location and magnitude of the load dependent change in vibration.
- Provide conclusions and recommendations based on analysis.

History of Rotor

A spare generator rotor #542 was purchased by the utility for a change-out / upgrade program. The spare rotor, a duplicate of the existing generator rotors, was originally manufactured in the late 1970’s for a unit that was never assembled or placed in service. The essentially new rotor had been in storage for ~25 years before being purchased for use at the station.

There are three rotors involved at the station. To avoid confusion, they are identified as follows:

Identification	ID Number
Unit 1 Original Rotor	777
Unit 2 Original Rotor	778
Spare Rotor	542

Table 1: Rotor identification numbers

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The specific rotors involved in this study are the spare rotor #542 which after refurbishing was installed and briefly operated in Unit#1. After experiencing “thermal” sensitivity, rotor #542 was removed and a refurbished rotor #777 was placed back in Unit#1.

The rotor #542 was inspected at the shop of another OEM prior to its installation in the Unit 1 generator. The rotor retaining rings were removed in order to carry out inspections and recommended upgrades. Several non-magnetic rotor slot wedges were removed and replaced with magnetic steel replacements by others. During this inspection, the rotor winding was not removed. The rotor was reassembled, electrically tested, high speed balanced and over-spiced. Running electrical tests and a “heat run” were conducted. The electrical tests and balance condition were reported to be satisfactory. The rotor balance and heat run were conducted in a vacuum bunker.

Upon delivery of the rotor #542 to the station, it was installed in the Unit #1 generator for operation during the spring of 2005. After 1 month initial operation at part load, and field balancing to correct LP vibration, the #542 rotor experienced a rise in vibration amplitude and shifting vibration phase angle when reaching 3600 rpm, after each restart. Rotor also experienced vibration changes with increase in load from each new position at 3600 rpm. The maximum generator reactive load was limited when vibration reached 0.008 mils peak to peak.

After an attempt to “shift” a thermal vector by balance correction in July, 2005, the machine was over-spiced to 110% of rated speed. Following the over-speed, the machine was held at 3600 rpm for 2 hours with no load. During the 2 hour period, the rotor vibration phase shifted 180°, with a ~6 mil total vector change. After the event, start-up behavior of the rotor had been consistent and very low. But vibration level and phase angle changes under load continued, with a ~8+ mil thermal vector observed at both generator bearings, acting in essentially the same direction.

Following the vibration problems experienced with the #542 rotor, the utility learned that several rotor slot wedges had been replaced during inspection work on the rotor. Several non-magnetic steel wedges had been removed and replaced with magnetic steel wedges. No other observations regarding the rotor inspection that could be related to the vibration behavior were identified.

SITE DATA – VIBRATION CHARACTERIZATION

Vibration response data was collected on-site to observe and compare the response of the rotor to thermal load influences and to collect shut-down data as comparison and validation of the FEA model and analysis.

Load change data for the Rotor #542:

Data Point	Load		Field Amps	#9 Vib. mils p-p
	MW	MVAR		
1	887	0	4166	6
2	881	0	4174	6
3	882	210	4705	8
4	585	0	3260	1
5	589	273	4217	5
6	28	70	2597	0.5

Table 2: Generator load points

Data noted above represents the steady state condition observed before load change and after vibration had stabilized. In all cases the vibration stabilized within 3-4 hours after initiating a change in load. Response at each bearing was essentially uniform and symmetric. The absolute vibration (1X uncompensated) at the #9 bearing is used as representative of thermal response.

Rotor vibration changed with load in response to rotor heating. The rotor is primarily heated by rotor winding resistance (I^2R) losses. There is an additional heating effect from other stator losses. The rotor is hydrogen cooled and thus affected by those variable losses dissipated by the hydrogen. The following table shows the approximate loss distribution for a generator and estimates the variable losses involved. For a water cooled winding, only a portion of the variable stator losses are dissipated by the hydrogen.

Approximate Generator Loss Distribution:

Stray Loss	10%	Proportional to MW output - dissipated by H ₂	10%	34%
Stator I ² R Loss	14%	Proportional to MW output - primarily dissipation - cooling water	2%	
Rotor I ² R Loss	23%	Proportional to field current ² - dissipated by H ₂	23%	66%
Core Loss	13%	Constant at rated voltage		
Friction Loss	23%	Constant at rated speed		
Windage Loss	17%	Constant at rated speed		

H₂ Dissipated Variable Losses (% of Total): 35%

Total Losses:	100%
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% of Variable Losses Dissipated by H₂: 100%

Table 3: Generator heat losses

By calculating the percentage change in load related heating of the rotor and comparing this to the change in vibration, proportionality of response can be evaluated. The following graph (Fig.1.) illustrates the result of the load change data evaluation. The change in rotor vibration observed is in general agreement with load related heating affecting the rotor (Fig. 2 and 3)

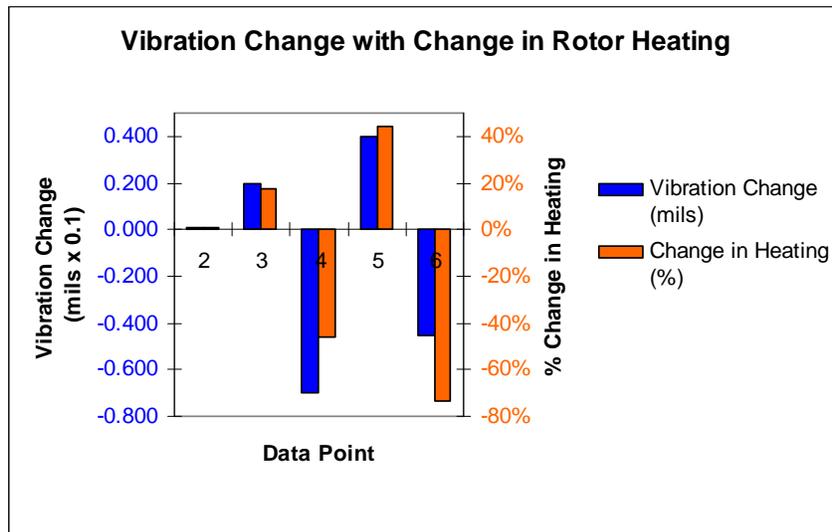


Figure 1: Vibration load dependency

The polar plots of the load test clearly illustrate the thermal response of the Rotor #542.

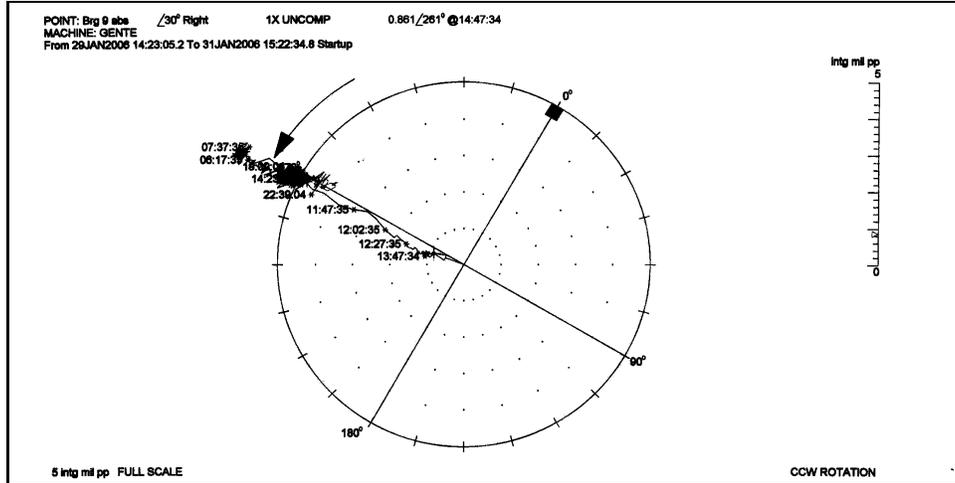


Figure 2: Bearing 9 Abs. – Thermal Response of Rotor #542

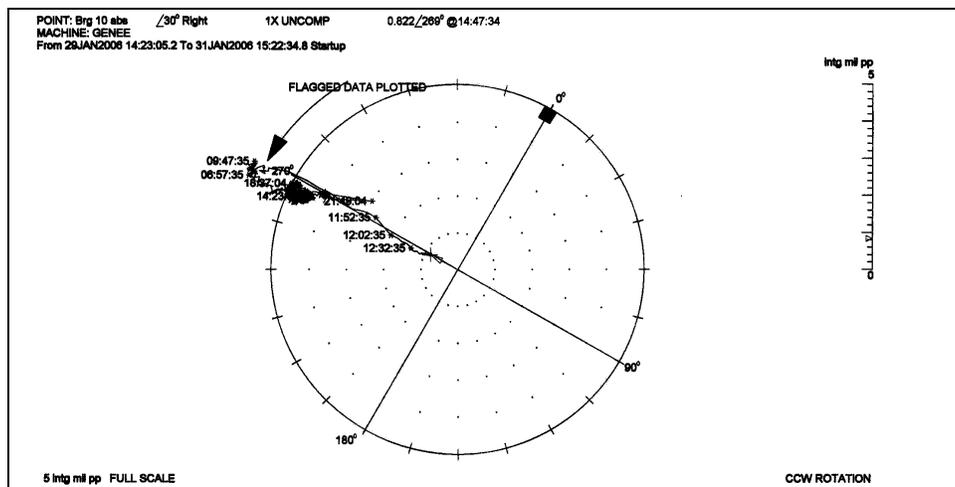


Figure 3: Bearing 10 Abs. – Thermal Response of Rotor #542

Following the load test, shut-down mechanical data was taken. After the generator was taken off line, rotor speed was increase to ~3700 RPM in order to observe the 3rd mode response of the rotor. The polar plot of vibration shows a phase angle change and peaking vibration at ~3650 rpm. As speed was reduced further another peak is observed in ~3300 rpm range. The double loops likely result from the asymmetric rotor/bearing stiffness in the vertical and horizontal axis (Fig. 4 and 5).

The polar plots of the shut-down appear to confirm the proximity of the 3rd critical to operating speed and the residual bow eccentricity effect on vibration response of Rotor #542.

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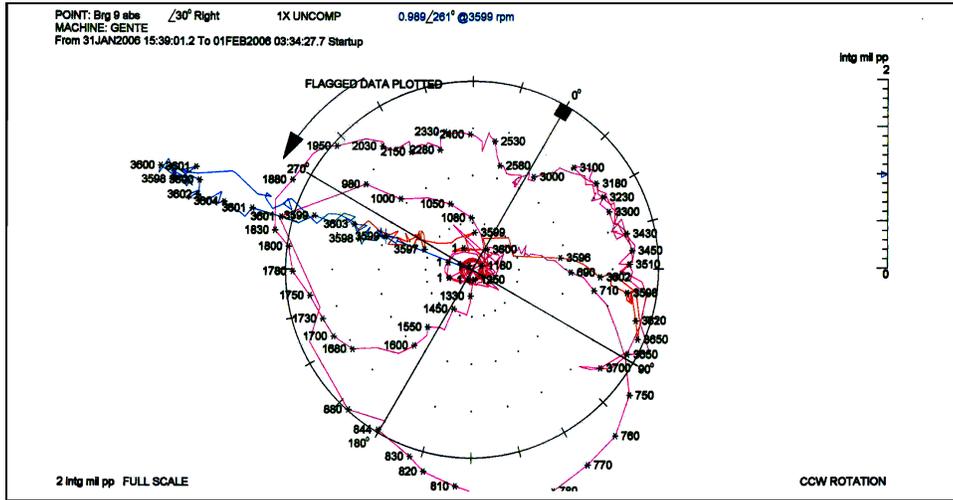


Figure 4: Bearing #9 Abs. – Mechanical Shut-Down Response of Rotor #542

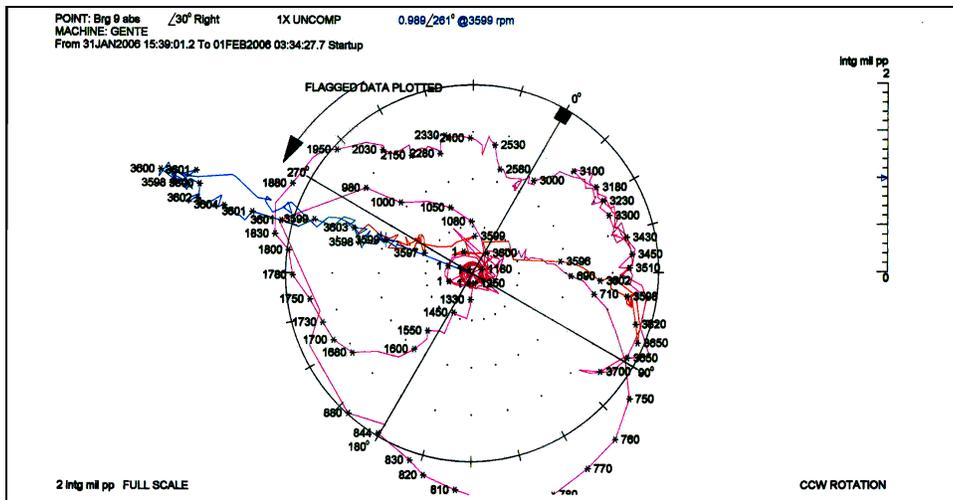


Figure 5: Bearing #10 Abs. – Mechanical Shut-Down Response of Rotor #542

COMPARATIVE BALANCE DATA EVALUATION

Rotor #542 / Rotor #777

Data was provided by the Client for both the #542 and #777 rotors. Body balance weight distributions (as-received and after high-speed balance) and rotor runout measurements were provided for both rotors. Comparative evaluations were carried out using this information.

Body Weight Evaluation	Rotor ID #542		Rotor ID #777	
	As-Received	Final	As-Received	Final
Equivalent Effective Weight Total:	13.5 lbs	13.8 lbs	7.5 lbs	8.0 lbs
Effective Angle with respect to Pole 1 Center:	51°	29°	338°	347°
Equivalent Eccentricity based on Effective Weight Total:	0.0018 in	0.0019 in	0.0010 in	0.0011 in
CF Effect of Equivalent Effective Weight:	110,315 lbs	112,411 lbs	61,333 lbs	65,196 lbs
CF as % of Total Rotor Weight:	67%	68%	37%	39%

Table 4: Rotr weight body evaluation

Based on the evaluation above, there is a significant difference in the weight placement and amounts of weight needed between the two rotors.

The equivalent CF effect of the weight, in the case of the #542 rotor at 68% of the rotor weight, is significant. In general, when the CF effect of balance weights approaches 50% of the total rotor weight, the weight is generally required to correct for a significant eccentricity of the body with respect to the rotational axis of the journal centers or other mass asymmetry. Machining tolerances not being met or bowing of the rotor are usual reasons for this condition.

Plots (Fig. 6-11) are of the “as-received” and “final” condition effective weight vectors, for each body balance plane. Only the # 542 rotor is presented here for illustration purposes.

Rotor #542 Body Weight Evaluation Plots – As-Received Condition

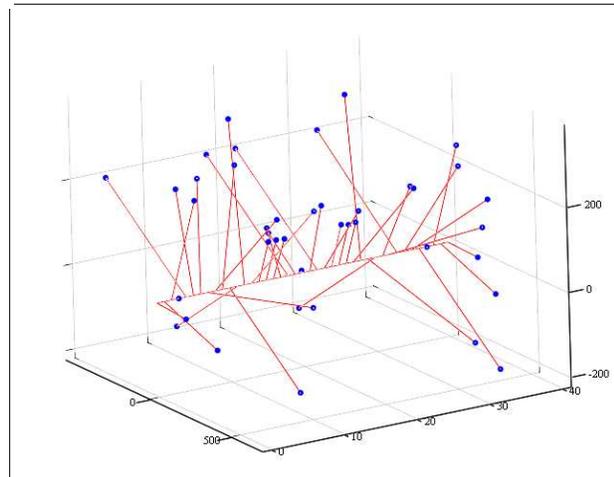
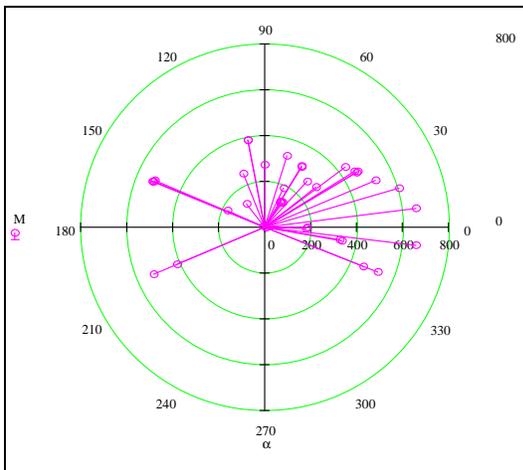


Figure 6: Polar Plot of Weight Effect across Rotor Body

Figure 7: 3D Plot of Effective Weight across Rotor Body

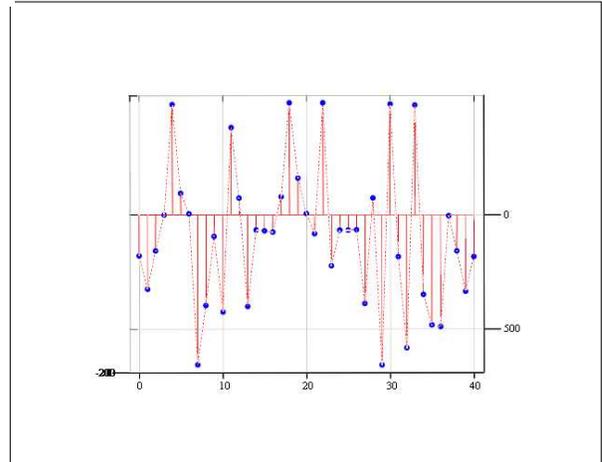
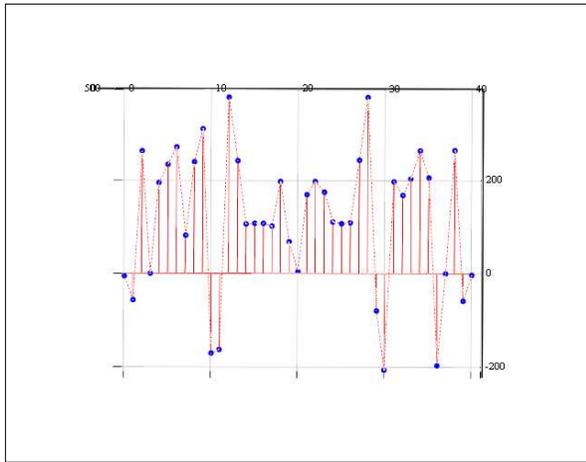


Figure 8: Rotated 3D View Looking Horizontally at Pole axis (0°) and Looking Down at Quadrature axis (90°)

Rotor #542 Body Weight Evaluation Plots – Final Condition

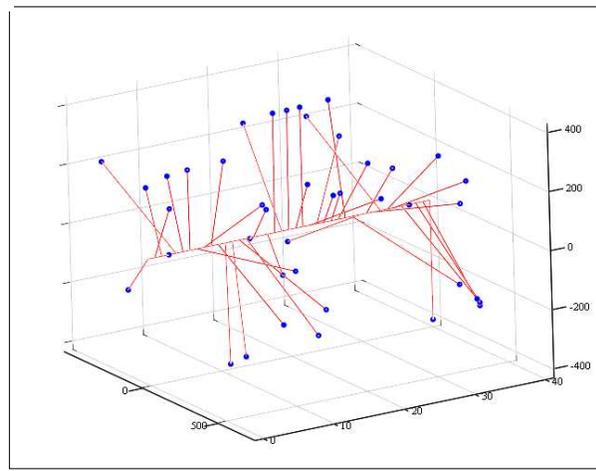
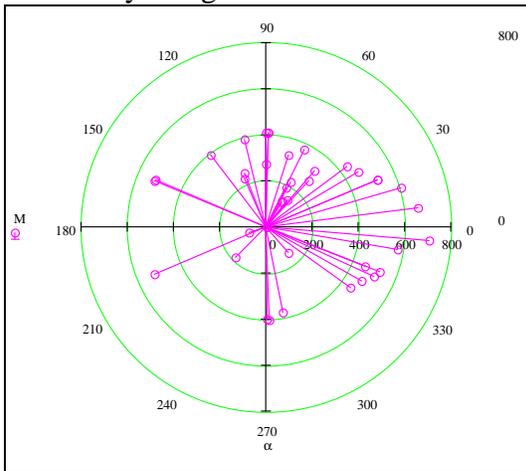


Figure 9: Polar Plot of Weight Effect across Rotor Body Figure 10: 3D Plot of Effective Weight across Rotor Body

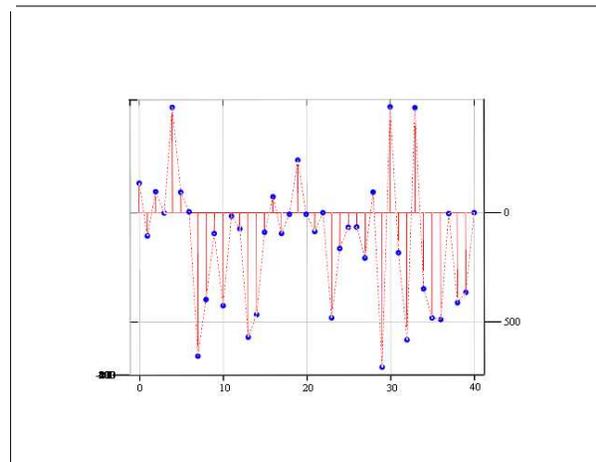
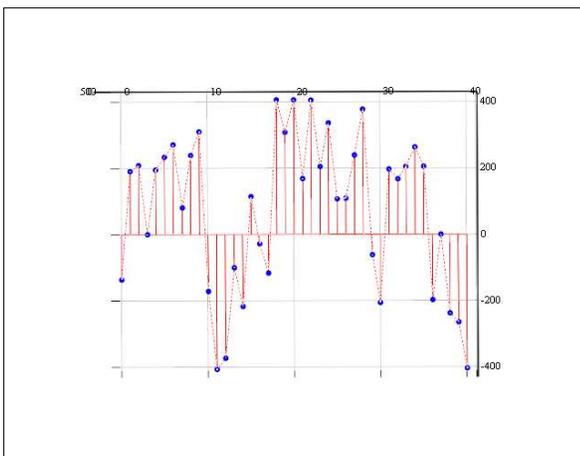


Figure 11: Rotated 3D View Looking Horizontally at Pole axis (0°) and Looking Down at Quadrature axis (90°)

Comparison of the final balance condition of body total weight placements shows a significant difference to rotor's "as received" condition. The effective weight for the #542 rotor is concentrated

primarily on one side of the rotor, but the weights axial distribution differs greatly between “as received” and “final” after balancing, indicating a significant weight asymmetry or eccentricity in the rotor body. The #777 rotor shows approximately half the influence compared with the #542 rotor using the same evaluation method.

Runout - Evaluated Eccentricity

Runout measurements for the two rotors, provided by the Client, were analyzed. Evaluated eccentricity was calculated from the runout data provided. Runout measurement data was mathematically evaluated at each axial measuring point.

It is important when analyzing 2-pole rotors to segregate the influence of the pole vs. quadrature axis stiffness, represented by the 2X term. The remaining (1X) eccentricity is a measure of mass displacement from the true centerline of rotation defined by the journal centers and has a significant influence on rotor balance condition. Other components of TIR (Total Indicated Runout) like lobe and ovality are also segregated by the software processing.

The evaluated 1X eccentricity conditions, as shown in the following plots, were noted. Again only the #542 rotor evaluated eccentricity is presented here (Fig. 12)

Rotor # 542 Runout Evaluation – Data from Balance Bunker Containing Body Runout Values

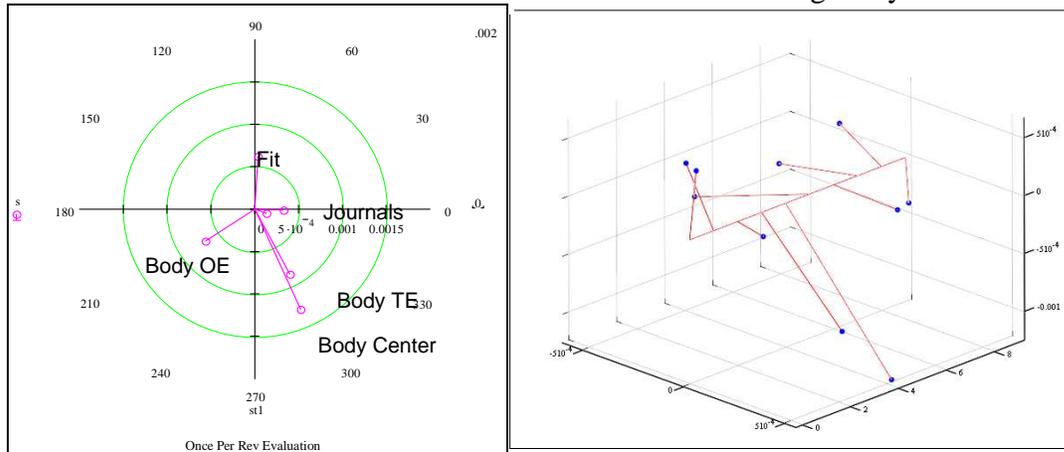


Figure 12: Polar and 3D plots of evaluated eccentricities at selected body locations

Note:

- 1) Axial lengths between points in the plot are equally spaced and not to actual length between measuring points on rotor.
- 2) Plot includes points on shaft ends and body, not including retaining rings.
- 3) 0° Reference not known.

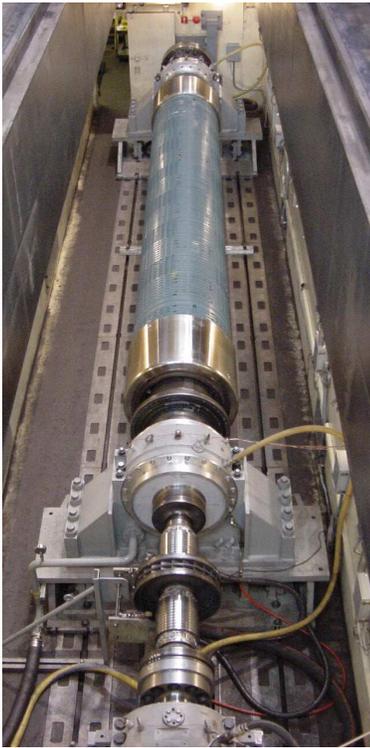
The maximum eccentricity of the rotor body with respect to the journals is approximately:

Rotor #542: ~0.0013” vs. 0.0018” - 0.0019” (equivalent based on balance weights)

Rotor #777: ~0.0004” vs. 0.0010” - 0.0011” (equivalent based on balance weights)

The maximum body eccentricity for each rotor roughly correlates with the equivalent eccentricity calculated from the body balance weight placements.

ROTOR FEA MODELING AND ANALYSIS



Based on the above analysis of the weight placement distribution and data supplied by the client an FEA model of the rotor was developed in an effort to explain the two observed different behaviors in operation of rotors #542 and #777 before and after refurbishing. It should be noted that couplings stiffness springs were added to better simulate coupled rotor in the field.(Figure 13).

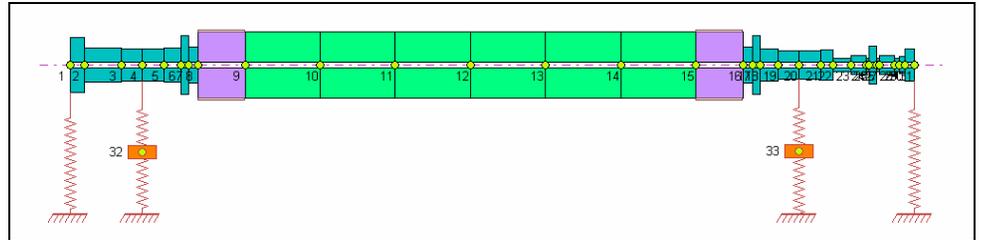


Figure 13: FEA Model of Generator 542 Rotor

The undamped critical speeds and corresponding mode shapes were calculated from the developed model. The following illustrations show the results of the calculation (Fig. 14)

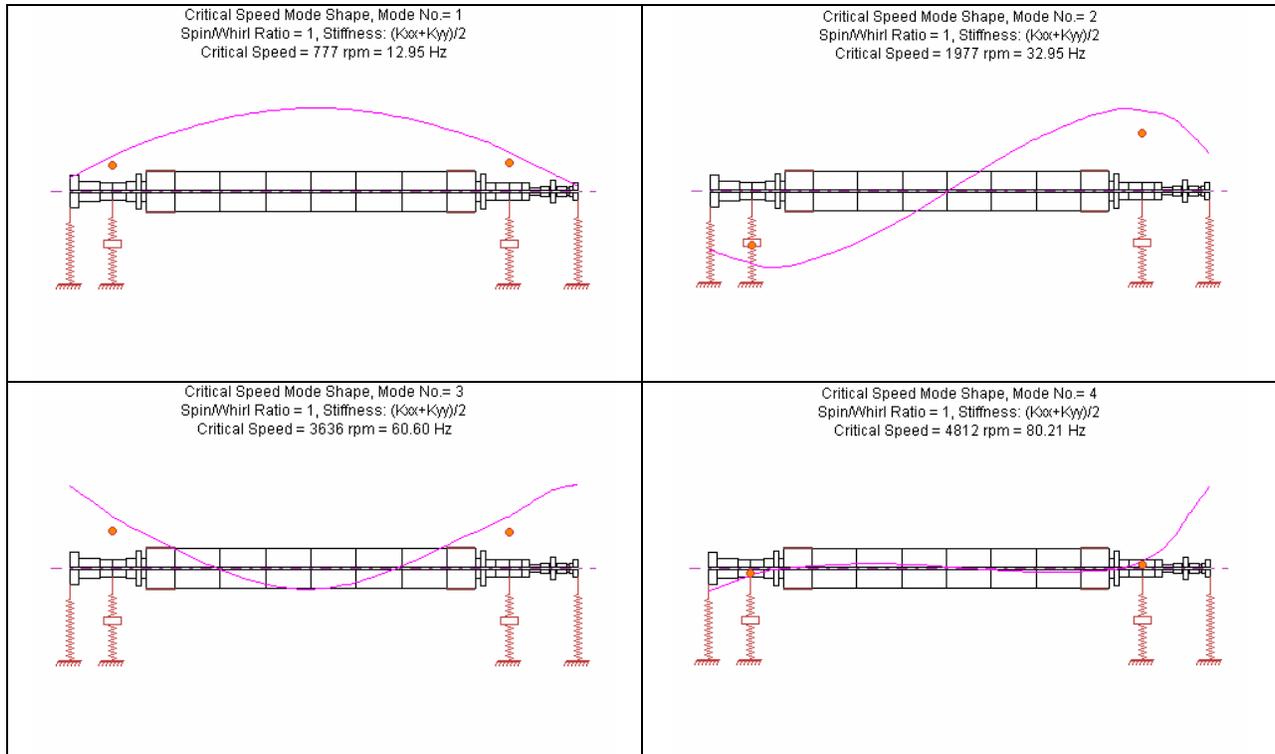


Figure 14: Rotor undamped mode shapes

Critical speeds for the rotor match the response of the rotor as observed during shut-down of the machine. It is significant to note that the rotor runs nearly on top of its 3rd mode and assumes the 3rd mode shape at operating speed. This rotor will be sensitive to unbalance due to the proximity of its operating speed to this critical speed.

The large changes in phase angle after reaching 3600 rpm, experienced initially between subsequent start ups are likely the result of the initial locking and releasing of some wedges and magnified resonant response to unbalance conditions due to proximity of “critical”. Thermal changes that result in deformation of the rotor and influence mass eccentricity of the rotor will act to excite the 3rd mode resonance at load.

The damped whirl modes were also calculated and are illustrated as follows (Fig. 15)

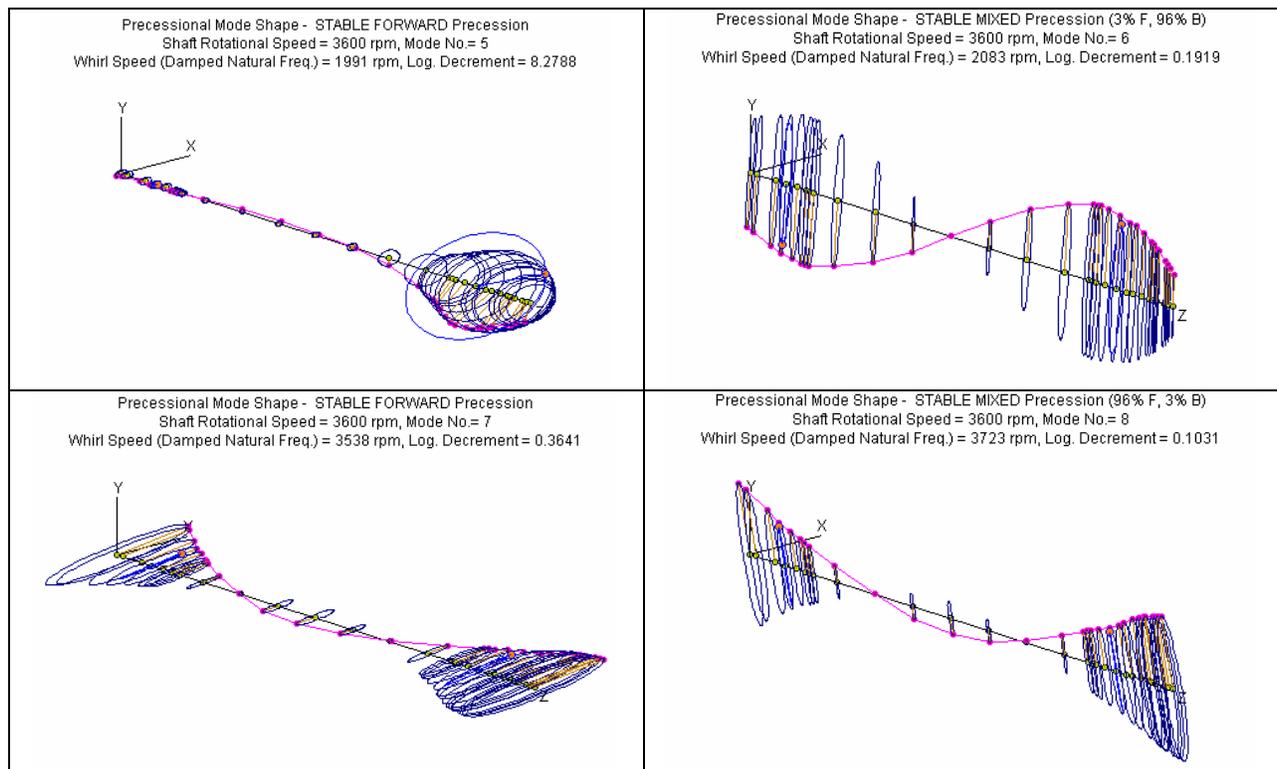


Figure 15: Rotor damped critical speeds mode shapes

The damped whirl modes further illustrate the 3rd mode configuration assumed by the rotor at operating speed.

The 3rd mode operating mode shape of the rotor has points of inflection $\sim 1/6^{\text{th}}$ of the body length inboard of each retaining ring. The central $\sim 2/3^{\text{rd}}$ of body length acts on one side of the axis of rotation.

Unbalance Response of Rotors to Weight Placements

The weight effect at each body weight plane for both the “as-received” and “final” condition effective weight vectors for each rotor were applied to the FEA rotor model to compare the effect of the weight placement on ideal rotor at the operating speed of the rotor. The following unbalance response plots resulted as seen on Fig. 16a, 16b, 17a and 17b.

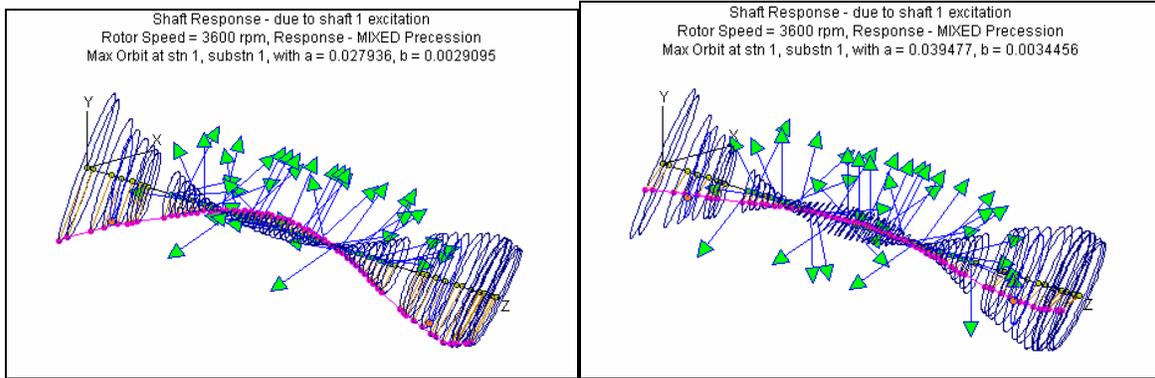


Figure 16: Rotor #542 – a) As-Received Weight Configuration and b) As Balanced

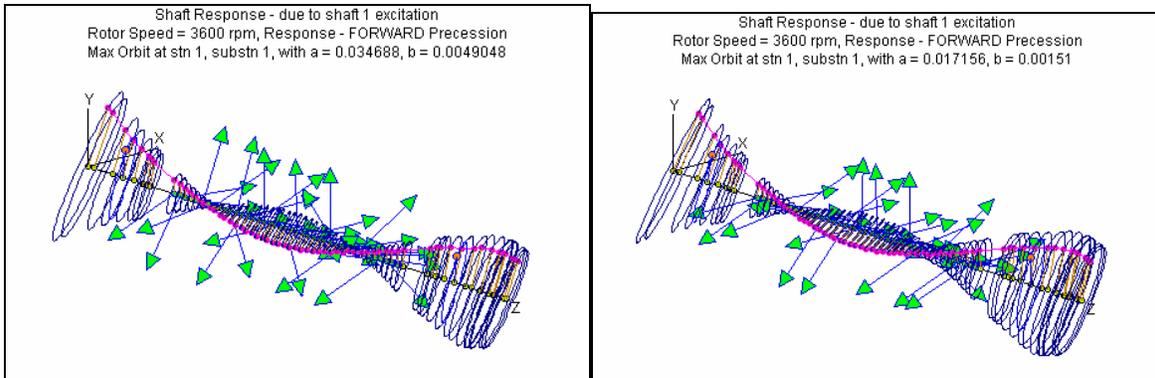


Figure 17: Rotor #777 – a) As-Received Weight Configuration and b) As Balanced

We note from the maximum orbit response ellipse (semi-axes a & b values) that the “final” weight configuration evaluated response is ~80% greater for the rotor #542 and ~50% greater for the Rotor #777 than the “as-received” weight configuration evaluated response for each. In addition, the “final” weight configuration for Rotor #542 produces a change from a circular to a flat orbit at the rotor mid body with greater shaft centerline displacements resulting from the CF effect of “final” concentrated weights.

From a balance point of view we can say that rotors in both cases are well balanced, as measured in terms of shaft vibration at the bearings, but with profoundly different consequences.

When a rotor moves in a circular orbit pattern the amount of bending work done by the rotor is minimal; on the other hand when it does it in a narrow and elongated orbit the amount of bending work done by the rotor is maximized. This rotor bending has to be superimposed onto the natural gravity sag of the rotor. The rotor goes into a twice per revolution bending as it travels from a peak of the orbit thru

a valley, a second peak and a second valley and back to the original peak. This bending work generates a substantial amount of heat that raises the rotor temperature in a localized manner depending upon each individual local orbit along the rotor length. This is a slow but steady phenomenon and probably the source of thermal sensitivity, with increasing response to general heat input, proportional to load.

The following orbit plots illustrate the influence of the balance weight distribution, as described above, specifically for the Rotor #542 (Fig. 18a and 18b).

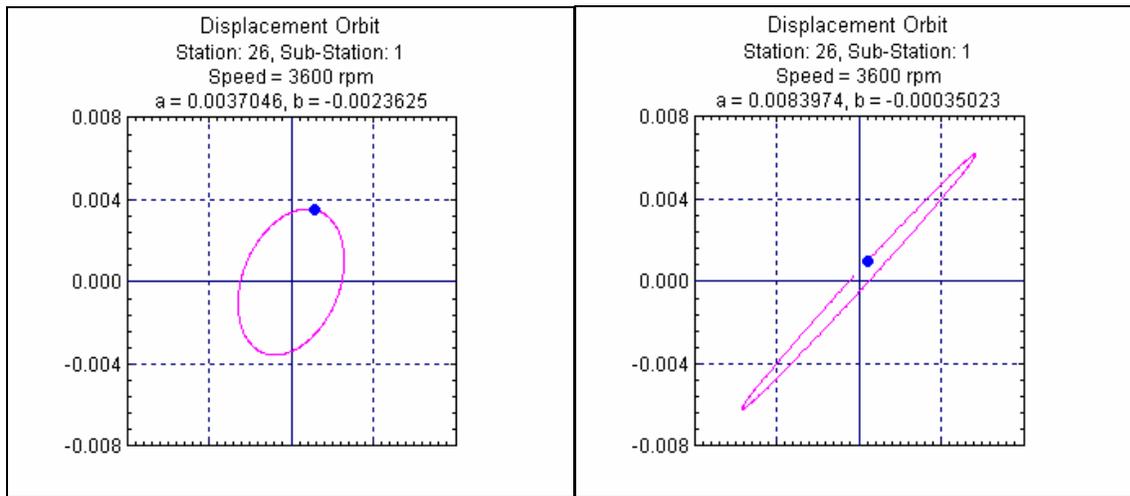


Figure 18: a) As-Received Weight Distribution b) Final Weight Distribution - orbit plot from unbalance response analysis (~ Rotor Center)

SUMMARY OF OBSERVATION

The rotor operates in proximity to the 3rd critical and assumes the associated 3rd mode shape. The proximity of the rotor critical to operating speed makes the rotor sensitive to small unbalance effects.

- Pedestal or bearing case resonance may be excited by rotor unbalance response at operating speed.
- The Rotor #542 responds to load related thermal heating.
 - Vibration change occurs over a 3-4 hour period until steady state conditions are achieved at the new load point.
 - Thermal vibration response is essentially uniform and symmetric at both ends (bearings) of the rotor.
 - Vibration changes appear to be repeatable and correlate to rotor heating.
- The rotor is flexible and the influence of balance weight distribution across the rotor body can affect the shaft centerline displacement resulting from the CF effect of concentrated weights.
 - The “as-received” and “final” balance weight distributions across the rotor body differed.
 - The “final” weight configuration was more concentrated and arranged in a pattern reflecting a 3rd mode correction.
 - The “as-received” weight configuration was more evenly distributed across the rotor body, in relation to the body eccentricity present in the rotor.
 - Unbalance response of the Rotor #542 based on the as-received and final weight distributions indicate maximum response orbits ~80% greater for the “final” weigh distribution compared to the “as-received”. In addition to the increased orbit size, the shape of the orbit changed from circular to a flattened, extended orbit as a result of weight concentration.

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- Body eccentricity of approximately 0.0020" (~0.005" TIR) exists in the body of Rotor #542 based on the evaluation of runout data. This eccentricity in the main portion of the rotor accounts for the lower mode weight correction required.
- Significant changes in phase angle and mechanical response of the rotor upon shut-down indicate a residual thermal bow, with the bow exciting the 3rd critical.
- Changes in vibration and phase angle that occurred after the initial mechanical start-ups of Rotor #542 indicate the presence of mechanical influences that changed with subsequent operation of the rotor. This behavior eventually stabilized and potentially contributed to the thermal response of the rotor.
- When the Rotor # 542 was removed from the generator in January, 2006, an inspection was carried out by the Client. The following findings were subsequently reported:
 - Borescopic inspection of all rotor cooling vent openings revealed no obstructions or blockage.
 - Wedges that had been removed and replaced by others were found to be "extremely tight" in relation to other wedges in the rotor.
 - The "tight" wedges were asymmetrically arranged (each end of the rotor and all in pole 1 winding slots).
 - No electrical faults were detected.

CONCLUSIONS

The balance weight distribution, with concentrations of weight, is likely a root cause of the thermal problem. Concentrations of weight, particularly on a very flexible rotor and with significant body eccentricity, can create CF loads that distort the rotor centerline in order to achieve a balanced condition. As observed from the unbalance response evaluation of the weight distributions, concentrated weights result in an increase in orbits and changes to the orbit shape, both of which can have negative effects also on rotor wedged conditions, particularly at points of inflection in the operating mode shape. This condition could further constrain free expansion of the rotor, particularly with enlarged and distorted orbits.

The thermal response of the rotor being uniform at both bearings and relatively slow in time response indicates symmetric effect acting at the center portion of the rotor body. The slow time response also indicates an effect pointing to other than asymmetric winding heating or strong restriction to the axial expansion of the winding coils end turns.

Rotors of this design are normally wound with "loose" fit wedges. The "tight" fit replacement wedges found in the rotor #542 are a contributor to the vibration problems experienced. "Tight" fit wedges, with "loose" fit wedges on either side will constrain the winding tight in the slot under the wedge. The winding on either side will move under centrifugal load, essentially following the "loose" fit wedges. The winding will deform around the "tight" fit wedge creating a potential pinch point. The initial influence of the deformation may be seen in mechanical operation, but will ultimately create an expansion restriction and add to deformation of the already distorted rotor thus affecting vibration behavior.

When the refurbished generator rotor #777 was placed back into service it also exhibited "thermal response", which it did not have before, and rotor #777 did not have "tight" fit wedges.

The magnitude of eccentricity on this rotor was about half of the one on #542 rotor and at 800MW the corresponding magnitude of the thermal response was also about half of that from rotor #542:

	HpF	HpR	IpF	IpR	Lp1F	Lp1R	Lp2F	Lp2R	GenF	GenR	ExcF	ExcR
Jrnl	#1	#2	#3	#4	#5	#6	#7	#8	#9	#10	#11	#12
Amp	1.1	1.7	2.2	1.9	3.2	0.8	3.4	0.3	4.0	4.4	6.3	2.3
Ø	154°	158°	187°	126°	327°	247°	106°	286°	127°	144°	218°	6°

With this realization the role of wedges on rotor(s) behavior was minimized. From the Author's previous experience in a similar case [6] and based on preceding conclusion of the effect of concentrated weights correction for lowest mode, an attempt was made to "unbalance", or, overcompensate the rotor's first mode and diminish the effect of the weight distribution at high speed by "redistributing" the weights in two additional outboard planes.

This field balance attempt was successful in its attempt to eliminate the thermal response, at the cost of slightly higher response at first critical speed of the rotor. Final vibration readings in mils peak to peak through the load range were constant and at 800 MW were:

	HpF	HpR	IpF	IpR	Lp1F	Lp1R	Lp2F	Lp2R	GenF	GenR	ExcF	ExcR
Jrnl	#1	#2	#3	#4	#5	#6	#7	#8	#9	#10	#11	#12
Amp	1.3	2.1	2.2	2.4	5.2	0.4	4.0	1.2	1.8	0.6	3.9	0.8
Ø	162°	168°	195°	112°	327°	214°	70°	254°	82°	143°	277°	122°

This result confirmed the previous conclusion of a dramatic effect of the applied balancing method on very flexible rotors with body eccentricity, in a high speed balancing facility.

ACKNOWLEDGMENT

The Authors thank to ReGENco, LLC for giving numerous opportunities to acquire vibration data during balancing processes in the high speed balancing bunker (BOB). Every rotor destined for balancing at ReGENco goes through TIR measurements and eccentricity evaluation. Using such procedure it makes it easy to observe the relation between rotors eccentricities and the ability to achieve a minimum residual unbalance, measured as journal displacement and bearing velocities (forces), based on applied method of balancing.

REFERENCES

1. Bishop R.E.D. , Parkinson A.G., "On the Use of Balancing Machine for Flexible Rotors", (1972) Journal of Engineering for Industry, pp 561-572
2. Kellenberger W., "Should a Flexible Rotor be Balanced in N or N+2 Planes?"(1972) Trans. ASME Journal of Engineering for Industry 94, No 2.
3. Federn K., " Fundamental of Systematic Vibration Elimination from Rotors with Elastic Shaft" (1957), VDI Ber., 24
4. Zorzi, "Balancing Flexible Rotors at Low Speed", (1979) NASA Tech Briefs MFS-28132
5. Rieger N.F. and Zhou S., "Comparison of Effectiveness of Several Balancing Methods for Flexible Rotors," (1982) Proceedings, IFToMM Conference on Rotordynamic Applications in Power Plants, Rome, Italy.
6. Racic Z., "A comprehensive Analysis of a Rotordynamic Problem", (2005) ISCORMA-3, Cleveland, Ohio.