Vibration Measurements and FE Modeling Tools in Root Cause Analysis (I.)

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ABSTRACT
This paper covers a case history of a generator rotor which went through extensive repair and modifications. Despite a great effort in following the work processes and quality control in the shop and with successful completion of repairs and high speed balancing (solo) in the bunker, the rotor exhibited unusual behavior when installed in the field. Vibration of any machine is only a symptom which in a traditional approach is treated by "balancing", very often without ever discovering the root cause of the vibration. Treating the vibration process as a symptom, and diagnosing it for it's root cause, is the topic of this paper.

Keywords:
Vibrations, eccentricities, modal balancing.

INTRODUCTION
The generator rotor of an 800MW unit was replaced with a spare generator rotor from another station. This rotor was fully modified and upgraded to the latest design changes. The rotor was also balanced at high speed in the OEM high speed balancing facility.

Upon the rotor installation, start-up vibration data was not recorded, but by operator's observation on the monitors start-up was relatively smooth. Shortly after the rotor was on line, vibration of the rotor started to increase. Vibrations had reached approximately 15 mils pk-pk when Z-R Consulting was asked for assistance in determining the cause of vibration changes.

DISCUSSION

First Shutdown
Since the start-up data was not available, the machine was shut down in order to record vibrations. Only Front Right and Rear Right shaft absolute vibration is presented here. (Figures 1 and 2). One can see that vectors are in phase since the rotor is operating practi-

cally on top of "third critical". There are no 1st or 2nd criticals visible, which means that the rotor was mechanically well balanced.

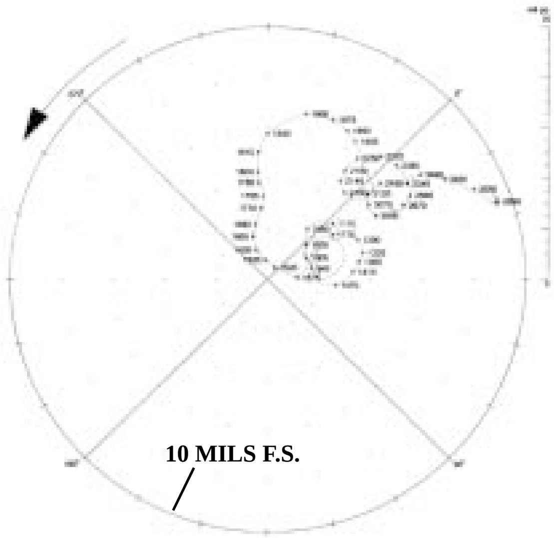


Figure 1: Original Shut Down (GENFR)

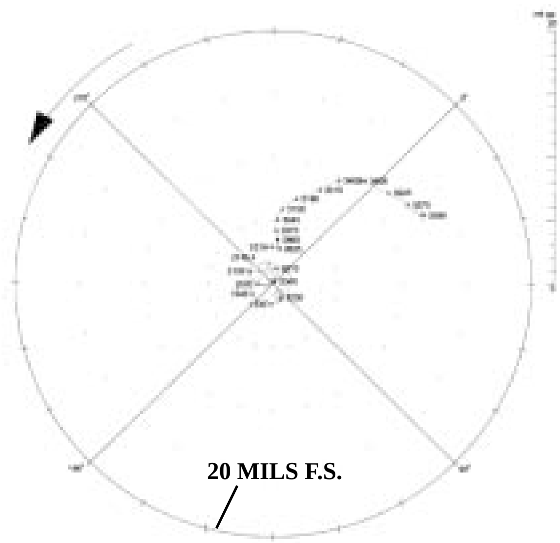


Figure 2: Original Shut Down (GENRR)

First Balance Trial

If we consider that the response vector at 3600 load condition is purely "thermal" unbalance response and it is repeatable, then a trial run based on the historical sensitivity of the rotor would require ~800 gr. at each end of the rotor, i.e. a static weight placement affecting primarily the 3rd mode, i.e., operating speed deflection shape.

The after balancing results were surprising because the total vector change was double than which was expected (Figures 3 and 4). Again, there is no evidence of 1st or 2nd critical response, and all the change (increase) in amplitudes occurs between 2500 to 3600 RPM.

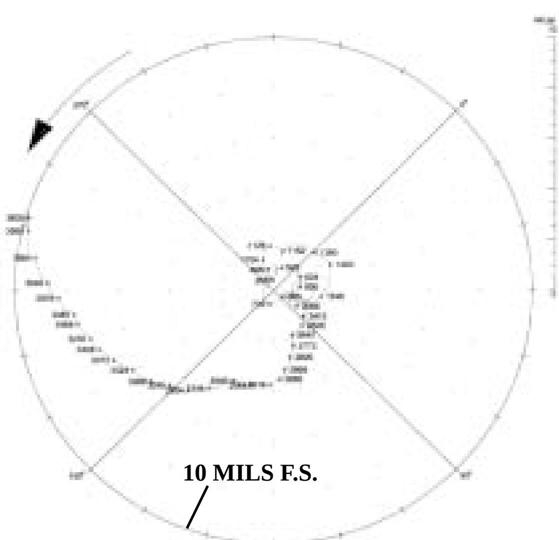


Figure 3: Shut Down with First Trial Run– 760 gr. @ 180º st. (GENFR)

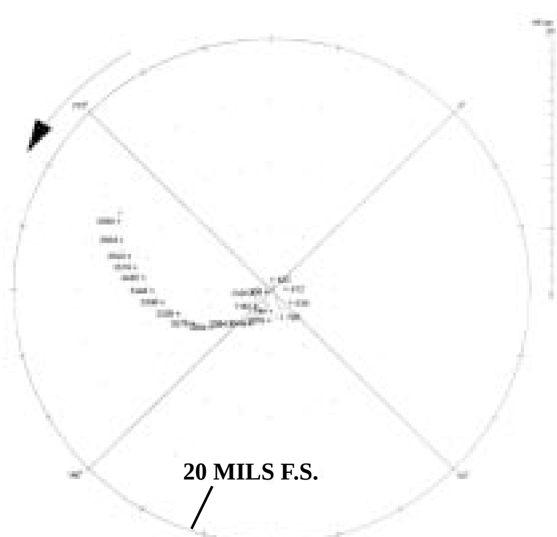
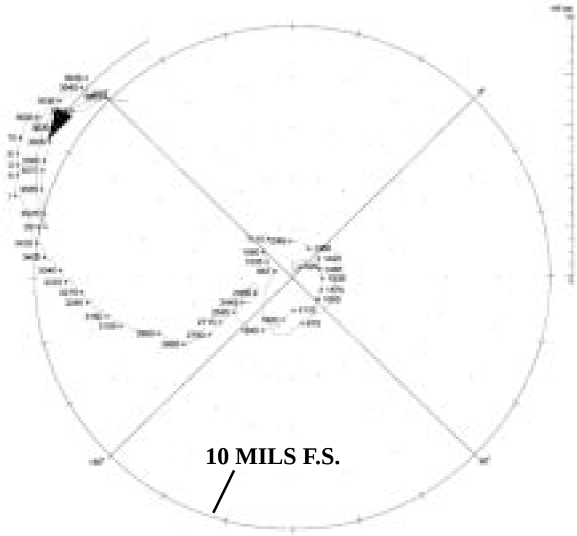


Figure 4: Start Up with First Trial Run– 760 gr. @ 180º st. (GENRR)



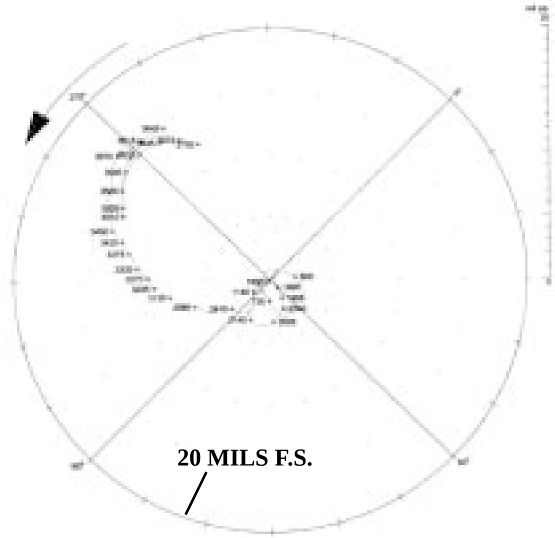


Figure 5: Overspeed and Verification of "Critical" Peak (GENFR)

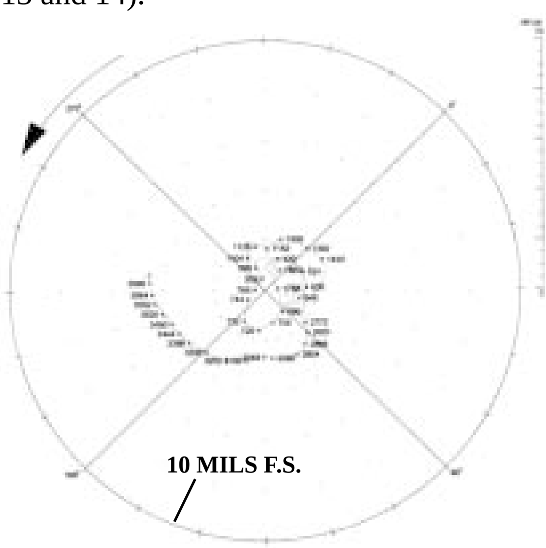
Figure 6: Overspeed and Verification of "Critical" Peak (GENRR)

For the sake of confirmation of the resonance at 3600 RPM, the speed of the rotor was increased to 3700 RPM prior to shutdown. Now it becomes evident that the 3rd critical peak is practically at 3600 RPM. (Figures 5 and 6).

Second Balance Trial

The amount of weights were halved and we obtained the starting position for the "thermal" vector where we wanted (Figures 7 and 8).

As soon as the Field Breaker, and 3 minutes later, the Main Breaker were closed, vibration vector "took off" by almost 10 mils (Figures 9, 10 and 11). It seems that vibrations settled at that point, but during the continued load ascending, the vibrations at first went lower, and then went back high, extending the original "thermal" vector jump (Figures 12, 13 and 14).



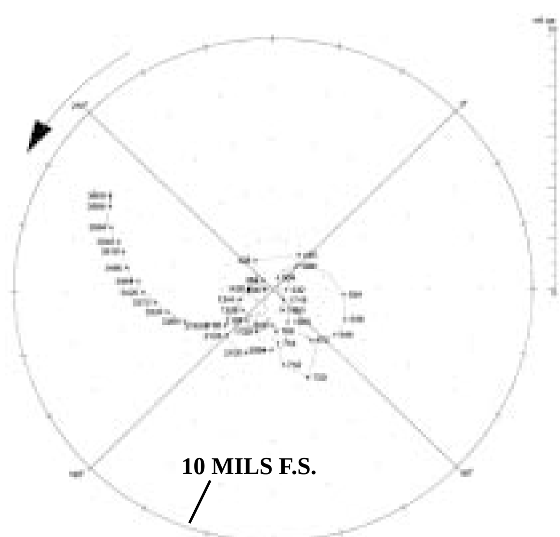


Figure 7: Start Up with Reduced Weight (GENFR)

Figure 8: Start Up with Reduced Weight (GENRR)

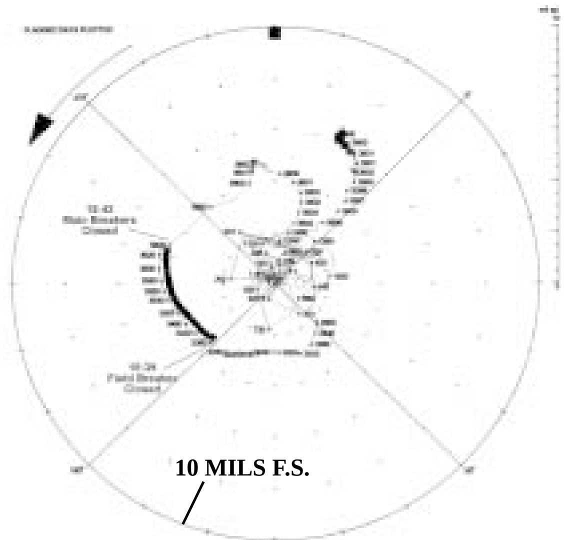


Figure 9: "Thermal Vector" Change During Field Excitation (GENFR)

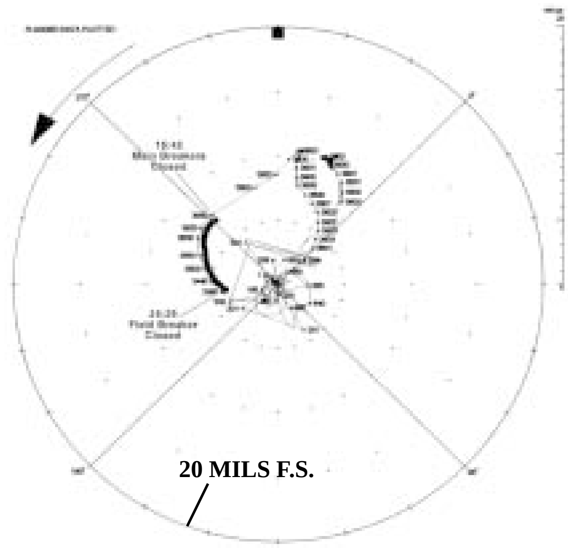


Figure 10: "Thermal Vector" Change During Field Excitation (GENRR)

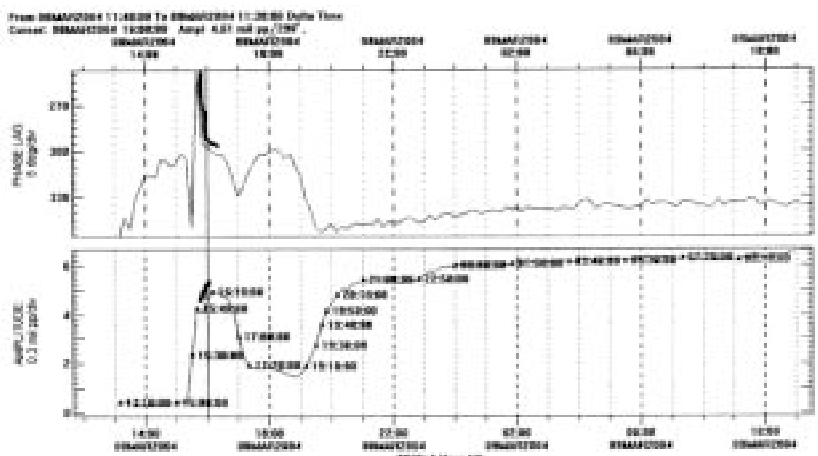


Figure 11: "Thermal Vector" Change During Field Excitation (GENFR)

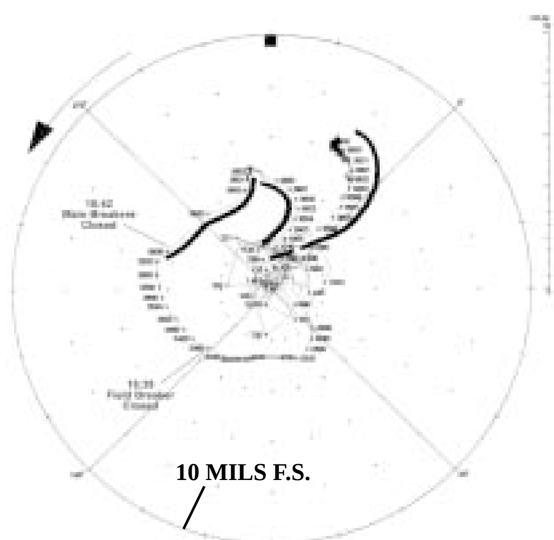


Figure 12: "Thermal Vector" Wandering with Load Increasing (GENFR)

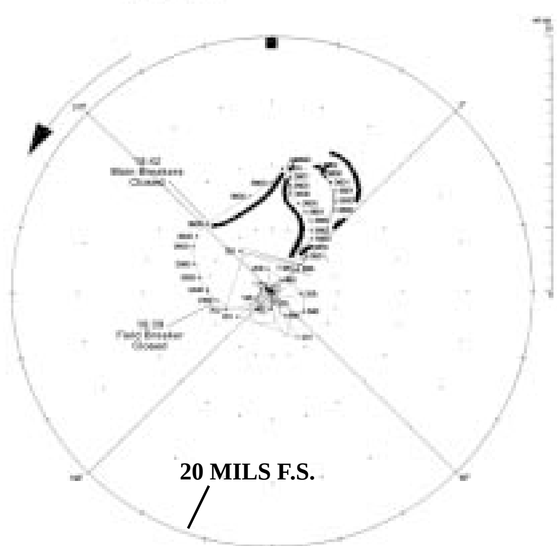


Figure 13: "Thermal Vector" Wandering with Load Increase (GENRR)

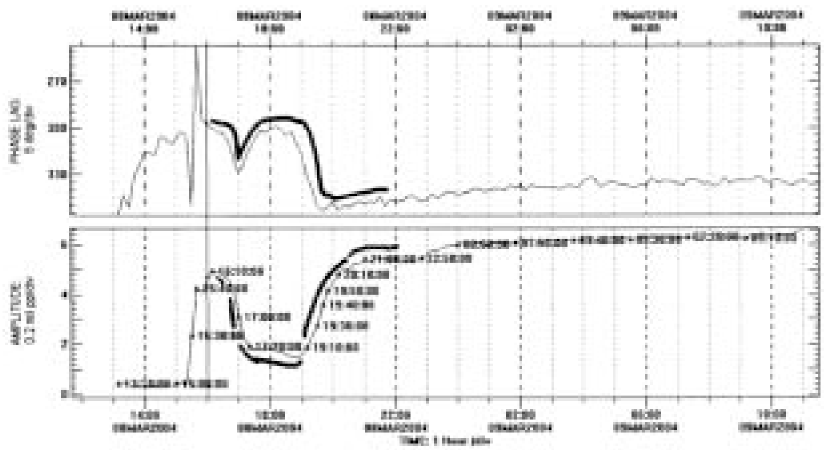


Figure 14: “Thermal Vector” Wandering wioth Load Increase (GENFR)

Third Balance Trial

After the second balance trial, it became obvious that weights have an affect on the rotor from pure mechanical aspect, but the rotor behavior was controlled by some other events.

Even though, the term “thermal” is used, it could not be associated as proportional to field current input. There was too much of hysteretic behavior to link the occurrences to mechanical elastic dynamic effects. One more attempt was made to move the mechanical unbalance to location where the mid-point of the thermal vector will intersect the origin of the vector coordinates. The balance weights were carefully scaled and rotated to new locations on the rotor.

Surprisingly after the start up vibrations, instead of being 6-7 mils as expected, they were in the range of 1-3 mils. They stayed there through the entire load range, without ever observing the “thermal” jump.

A minor change was observed during the first load swing from high of ~740 MW to low of 500 MW and back. This vibration change was attributed to lube oil temperature swing and the specific characteristics of this bearing assembly.

Lube oil temperature set point was changed to a range at which it can be maintained constant. No more vibration changes where observed regardless of load or field current changes.

ANALYSIS

The events and processes from the time of rotor receipt in the repair shop through machining, rewind, assembly and balancing, were reviewed to find anything that could explain the rotor behavior at the plant. Since, from the author’s past experience, the cause of ANY synchronous vibration is the result of the existing rotor mass eccentricities, their magnitude and distribution, the author had developed a computer program which is used to thoroughly analyze indicated rotor runouts. These runouts are then entered into the FE program Dyrobes® as local eccentricities, and a rotor anticipated modal response was simulated on the model. Several sets of rotor TIR readings were evaluated.

Total Indicator Runout (TIR) (Table 1)

TIR is a set of readings which contains key information of the physical condition of the rotor (Gen) assembly from these to decide the absolutely necessary corrective steps to be completed in the shop. Typically these have to be within the range of machining tolerances.

The TIR must be evaluated mathematically, i.e., objectively and quantitavely to accom-

SURFACE	Typical Acceptance Criteria	Author's Acceptance Criteria
Journal	TIR <.0005" Lobe <.0005" Taper <.001"/Ft Finish 16µ RMS	Evaluated Eccentricity <.0004" Out of Round <5% x C_{min} Taper <10% x C_{min}/L_j Finish 16µ RMS
Coupling	Concentric <.001" Axial Runout <.0005" Face Flat <.0005"	Evaluated Eccentricity <.0004" Axial Runout <.0005" Tight or Loose
Rabbet Fit	TIR <.0005" Roundness <.00056" Concentric <.0005" Fit .001–.003" Interference	– Evaluated Eccentricity <.004" Fit– Depends Upon Manufacturer Tight or Loose (OEM Design)
Oil Seal Surfaces	TIR <.0010"	Evaluated Eccentricity <5%xC_{min}
Rotor Body	TIR <.0040" –	Evaluated Eccentricity <.002" Or Within Weight Tolerances (ISO 1940)

TABLE 1: Rotor Indication (TIR) and Journal Evaluation Criteria

plish that. To do this, one must have the right tools:

- – Mathematical derivation of values, using computer program.
- – A reference against which to compare those values.
- – Objective and agreed upon acceptance limits of deviations.

The elements of TIR to be extracted:

- – Eccentricity 1x (the most important)
- – 2x eccentricity
- – Ovality
- – Taper

The other elements of inspections while performing TIR are:

- – Surface quality (journals, seals)
- – Surface damage (dings, grooves, etc.)

Evaluation of TIR– Tools

- – Mathematical Program (RUKO2)
- – Reference standards (Basic ISO 1940)
- – Raw TIR data taken along rotor and body at ~20-30 planes with 8-12 readings per plane minimum.

- – Acceptance criteria for:
- – Eccentricity (per plane absolute)
- – Ovality
- – 2x swing
- – Taper and (cocking) R-R and Fan Hubs
- – Eccentricity (between planes relative)

TIR Data Evaluation

Three sets of TIRs were evaluated.

1) Initial as Received (Figure 15)

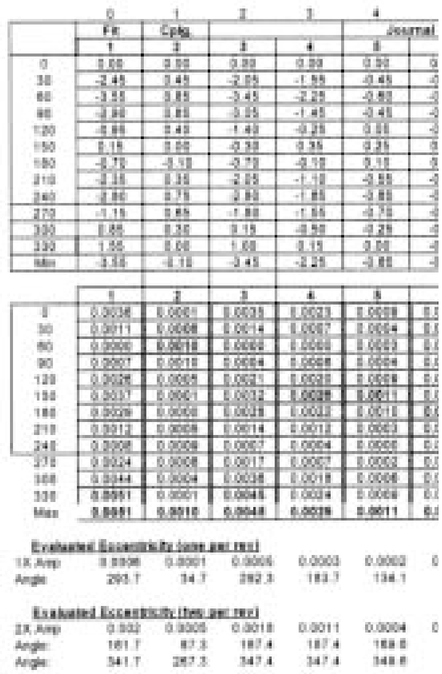
These readings were taken to assess the rotor condition as it had arrived from the plant to the repair shop.

Total non-evaluated maximum TIR readings were high at many locations, but they are not an absolute indication that something is wrong with the rotor. This raw TIR data, after the analysis, yields the most important sets of information:

1 x eccentricity and phase, and
2 x eccentricity and phase.

From this information the key points exceeding the acceptable 1x eccentricity limit set by the author can easily be seen:

TE face .... 0.0014" at 207º
EE rim .... 0.0018" at 197º
Rotor Body:
TE .... 0.0023" at 275º
Mid .... 0.0033" at 106º
EE .... 0.0017" at 357º



	0	1	2	3	4
	FE	Cplg.			Journal
	1	2	3	4	5
0	0.00	0.00	0.00	0.00	0.00
30	-2.45	3.45	-2.05	-1.55	-0.45
60	-3.55	0.85	-3.45	-2.25	-0.80
90	-2.90	0.85	-3.05	-1.45	-0.45
120	-0.85	0.40	-1.40	-0.25	0.05
150	0.15	0.00	-0.30	0.35	0.25
180	-0.70	-0.10	-0.70	-0.10	0.10
210	-2.35	0.35	-2.05	-1.10	-0.55
240	-2.80	0.75	-2.80	-1.85	-0.85
270	-1.15	0.85	-1.80	-1.55	-0.70
300	0.85	0.30	0.15	-0.50	-0.25
330	1.55	0.00	1.60	0.15	0.00
Min	-3.55	-0.10	-3.45	-2.25	-0.80

	1	2	3	4	5
0	0.0036	0.0001	0.0035	0.0023	0.0008
30	0.0011	0.0008	0.0014	0.0007	0.0004
60	0.0000	0.0010	0.0002	0.0000	0.0002
90	0.0007	0.0010	0.0004	0.0008	0.0004
120	0.0026	0.0005	0.0021	0.0020	0.0008
150	0.0037	0.0001	0.0032	0.0026	0.0011
180	0.0029	0.0002	0.0028	0.0022	0.0010
210	0.0012	0.0008	0.0014	0.0012	0.0003
240	0.0008	0.0009	0.0007	0.0004	0.0000
270	0.0024	0.0008	0.0017	0.0007	0.0002
300	0.0044	0.0004	0.0036	0.0018	0.0006
330	0.0051	0.0001	0.0045	0.0024	0.0008
Max	0.0051	0.0010	0.0048	0.0026	0.0011

Evaluated Eccentricity (one per rev)

1X Amp	0.0036	0.0001	0.0006	0.0003	0.0002
Angle	293.7	34.7	282.3	183.7	134.1

Evaluated Eccentricity (two per rev)

2X Amp	0.002	0.0005	0.0018	0.0011	0.0004
Angle	161.7	87.3	167.4	187.4	169.6
Angle	341.7	267.3	347.4	347.4	349.6

DETAIL

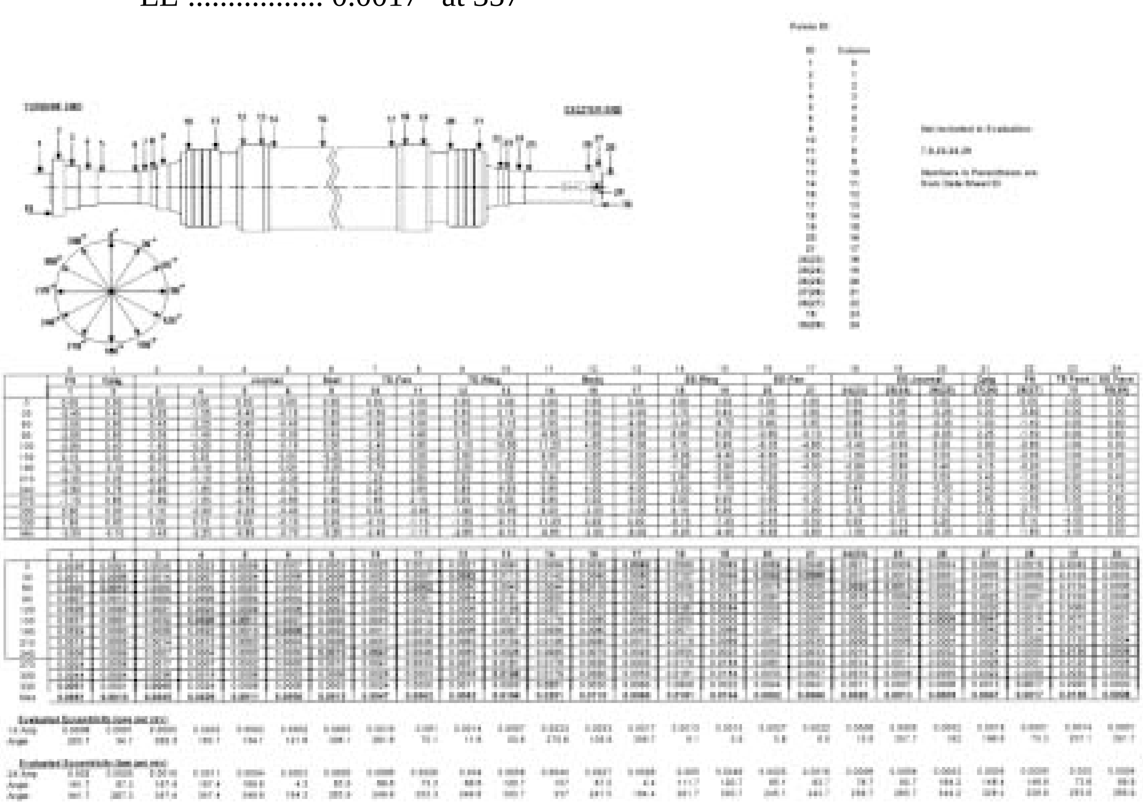


Figure 15: Generator Rotor Eccentricity Evaluation– Initial as Received in Shop

2) Before Balancing (Figure 16)

After forging was stripped, some machining repair by the shop was performed on the rotor to clean grooves and other surfaces' imperfection. Upon completion of the rewind, sliding in the slot wedges, re-installing the retaining rings and fan hubs, another set of TIR readings were taken.

Some additional machining was done, based on as found recorded dimensional data, most likely to correct for deformation of TE coupling overhangs, after shrink fitting the TE fan hub. The key points used for comparison and an evaluation are:

TE coupling spigot ... 0.0024" at 90º
Rotor Body:
TE .... 0.0033" at 142º
MID .... 0.0025" at 42º
EE .... 0.0017" at 157º

The TE coupling spigot was left eccentric as reference to see if any change on the attached components (retaining rings and fan hubs) has shifted when exposed to high centrifugal forces during the overspeed test in the bunker. From the tabulated data above, we can see that the rotor body eccentricity readings, in comparison to "initial" readings, had shifted 0.003"–0.005", after journals and seals were machined to correct rotor free TE ends extension deformation.

	0	1	2	3	4
	Fit	Cplg			Journal
	1	2	3	4	5
0	0.00	0.00	0.00	0.00	0.00
30	-0.50	-1.45	-0.80	-0.80	-0.05
60	-0.90	-2.80	-1.30	-1.30	-0.35
90	-0.35	-2.40	-1.40	-1.40	-0.40
120	0.50	-0.85	-0.85	-0.85	-0.35
150	0.35	-0.40	-0.15	-0.15	-0.15
180	-1.50	0.00	0.05	0.05	-0.15
210	-4.00	-1.70	-0.45	-0.45	-0.30
240	-5.50	-2.95	-1.25	-1.25	-0.50
270	-4.35	-2.65	-1.35	-1.35	-0.55
300	-2.00	-1.00	-0.80	-0.80	-0.50
330	-0.35	0.35	-1.10	-1.10	-0.20
Min	-5.50	-2.95	-1.40	-1.40	-0.55

	1	2	3	[illegible]	5
0	0.0055	0.0030	0.0014	0.0014	0.0006
30	0.0050	0.0015	0.0008	0.0008	0.0005
60	0.0048	0.0004	0.0001	0.0001	0.0002
90	0.0052	0.0008	0.0000	0.0000	0.0002
120	0.0060	0.0021	0.0008	0.0008	0.0002
150	0.0058	0.0028	0.0013	0.0013	0.0004
180	0.0040	0.0030	0.0015	0.0015	0.0004
210	0.0015	0.0013	0.0010	0.0010	0.0003
240	0.0000	0.0000	0.0002	0.0002	0.0001
270	0.0012	0.0003	0.0000	0.0000	0.0000
300	0.0035	0.0020	0.0008	0.0008	0.0001
330	0.0052	0.0033	0.0003	0.0003	0.0004
Max	0.0060	0.0033	0.0015	0.0015	0.0006

Evaluated Eccentricity (one per rev)

1X Amp	0.0024	0.0002	0.0003	0.0003	0.0001
Angle	90.4	138	191.8	191.8	80.7

Evaluated Eccentricity (two per rev)

2X Amp	0.0015	0.0015	0.0006	0.0006	0.0002
Angle	169.1	176	7.1	7.1	15.4
Angle	349.1	356	187.1	187.1	195.4

DETAIL

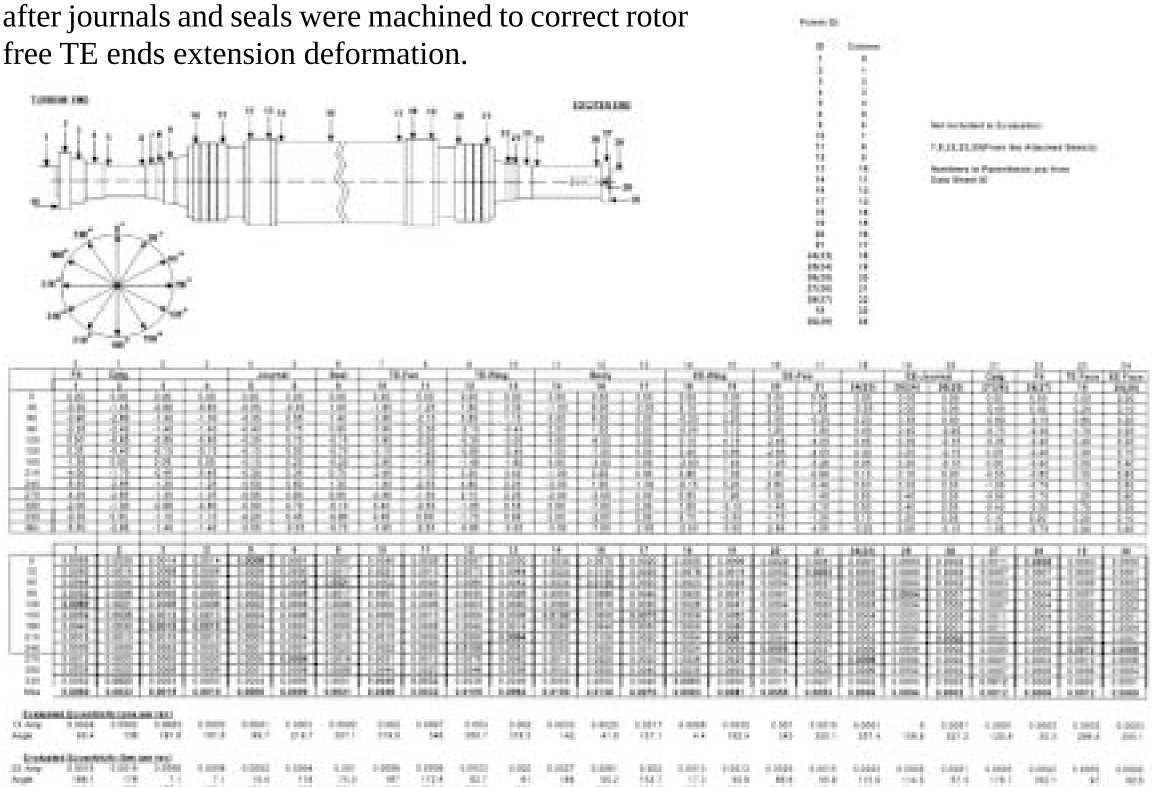


Figure 16: Generator Rotor Eccentricity Evaluation– After Assembly and Before Balancing

This type of deformation is possible when the shrunk-on component, like fan hub, is not cooled uniformly during the cooldown process.

It is the author's opinion, when such deformation is observed, that it is better to again reheat and cooldown the rotor and hub uniformly (preferably slowly rotating), rather than to machine out the deformation.

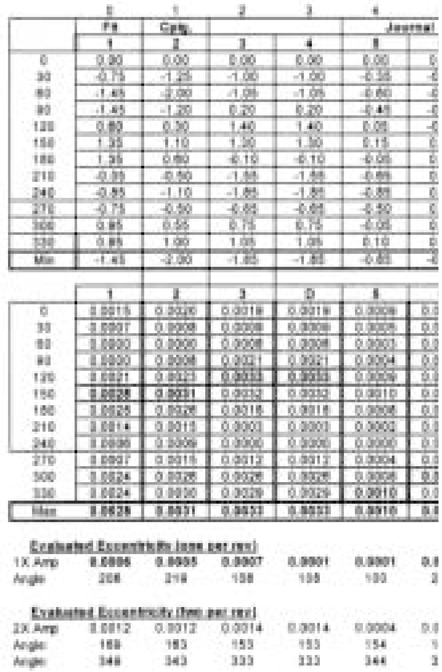
3) After Balancing (Figure 17)

After the rotor was assembled and machined for the second time, it was placed in the bunker for balancing and the overspeed testing.

There was no major change in TE coupling center shift during balancing, and the spigot was machined to match the centers of the coupling, journals and seals, to facilitate the alignment in the field. The key readings after balancing were:

Rotor Body:
TE................... 0.0029" at 107º
MID............... 0.001" at 122º
EE................... 0.0009" at 357º

All other readings were good and acceptable by the recommended criteria. It is noted though, that body eccentricity readings were somewhat reduced in comparison to readings before balancing.



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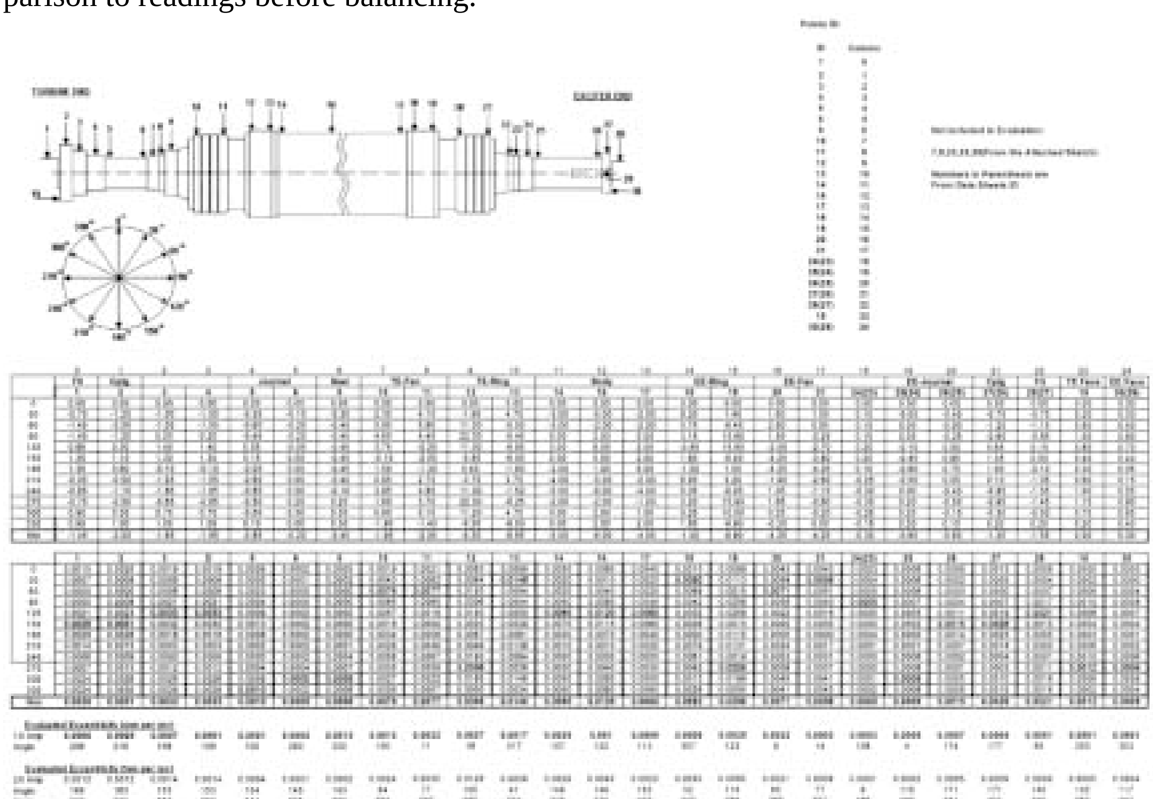


Figure 17: Generator Rotor Eccentricity Evaluation– After Balancing

Balancing in the Bunker

The final balancing results in the bunker were very good. Three things were never-the-less somewhat peculiar:

- First critical speed on this rotor at 720 RPM was practically not visible.
- Second critical at 2000 RPM was also suppressed.
- Third critical at 3860 RPM was not fully balanced. (Figures 18 and19)

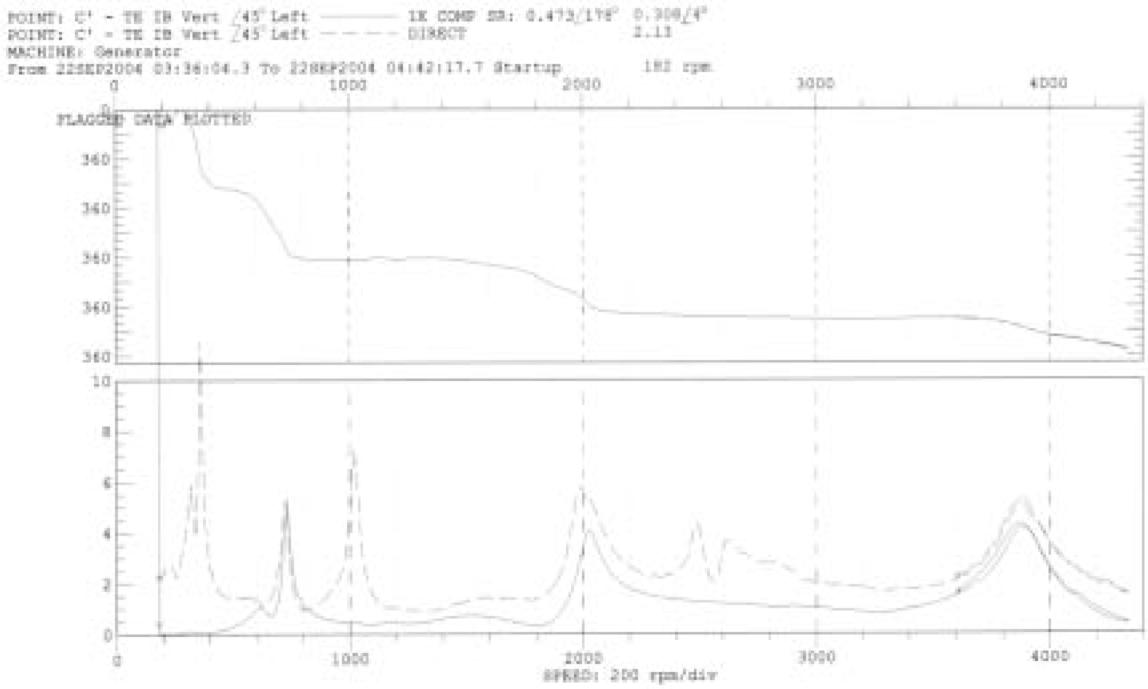


Figure 18: Third Critical– Not Balanced (GEN-FRONT)

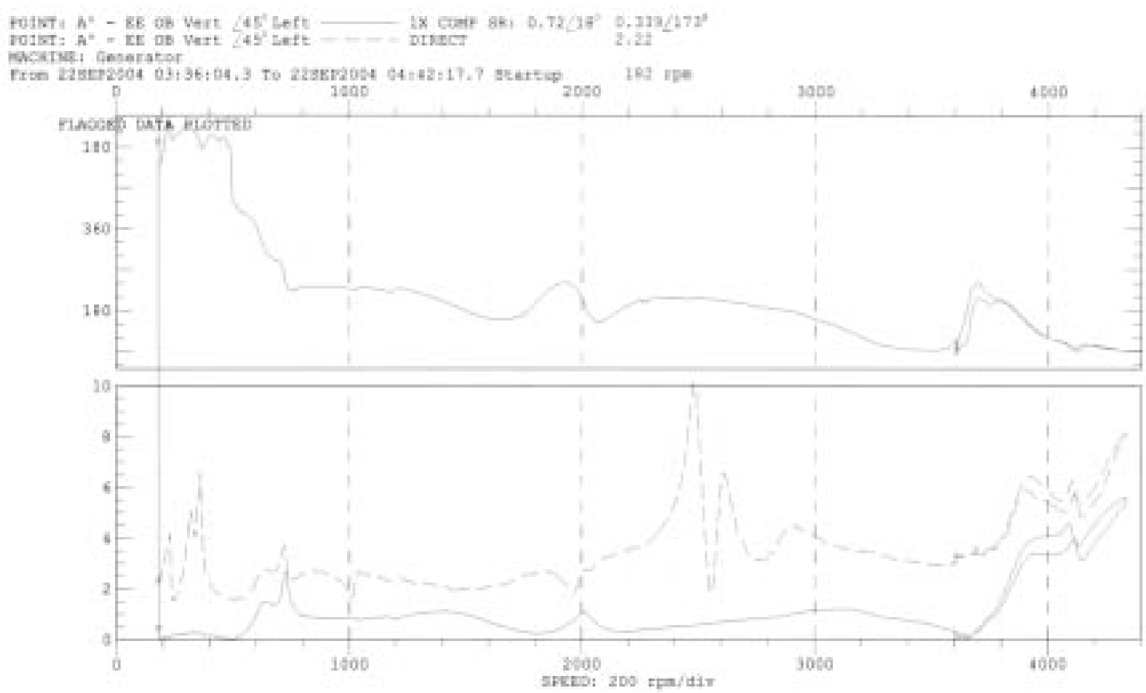


Figure 19: Third Critical– Not Fully Balanced (GEN-REAR)

The calculated mode shapes of the rotor using DYROBES®, closely match those in the bunker (Figures 19a, b ,c, d, e).

Figure 19A: Generator Rotor Cross Section

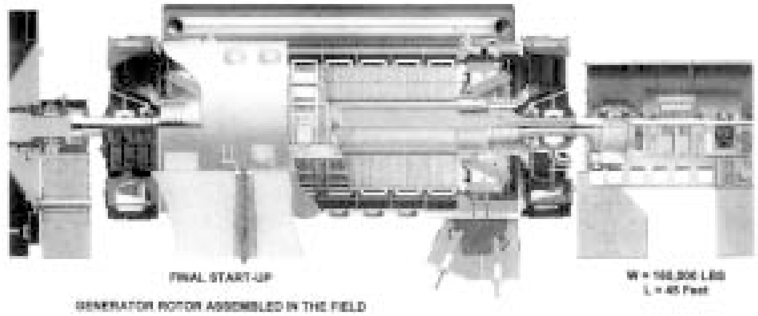


Figure 19B: Generator Rotor FE Model and its First Mode

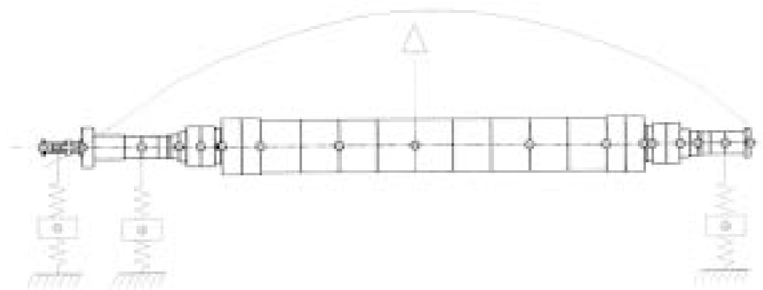


Figure 19C: Rotor Calculated 2nd Mode

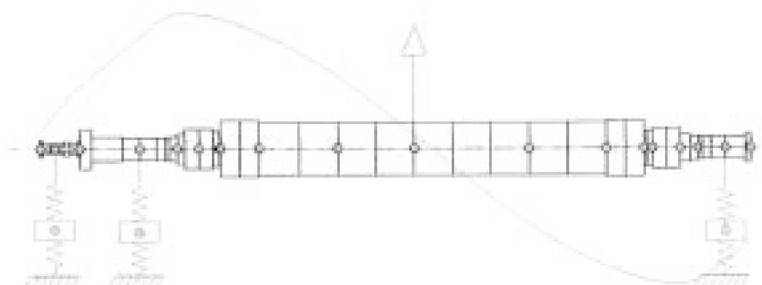


Figure 19D: Rotor Calculated Undamped 2nd Mode

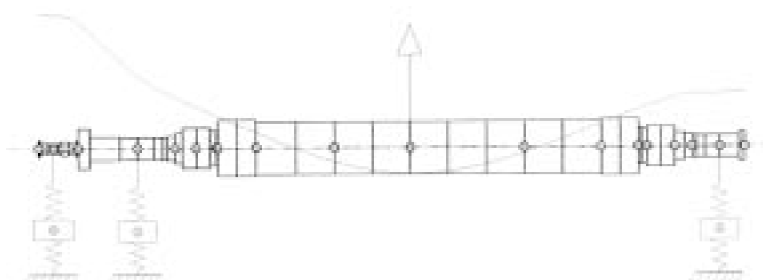
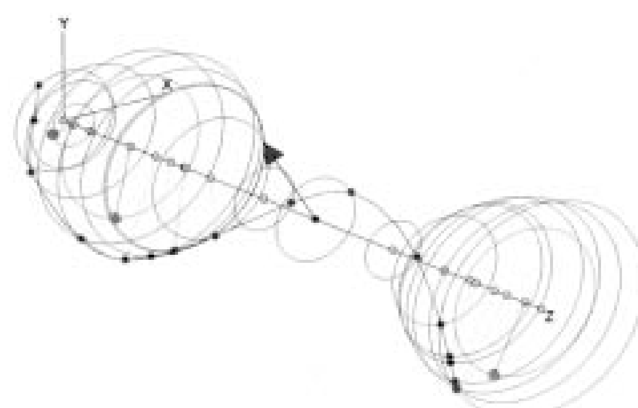


Figure 19E: Rotor Calculated Damped Deflection Shape at 3600 RPM



Never-the-less, the rotor “fulfilled” the contractual criteria and was released for shipment to the plant. (Figures 20 and 21)

The one thing that stands out up to this point was the amount and distribution of balancing weight in the rotor body. There was ~24 lbs of weight used for correction of the first critical speed, distributed across the full length of the rotor body. (Figure 22)

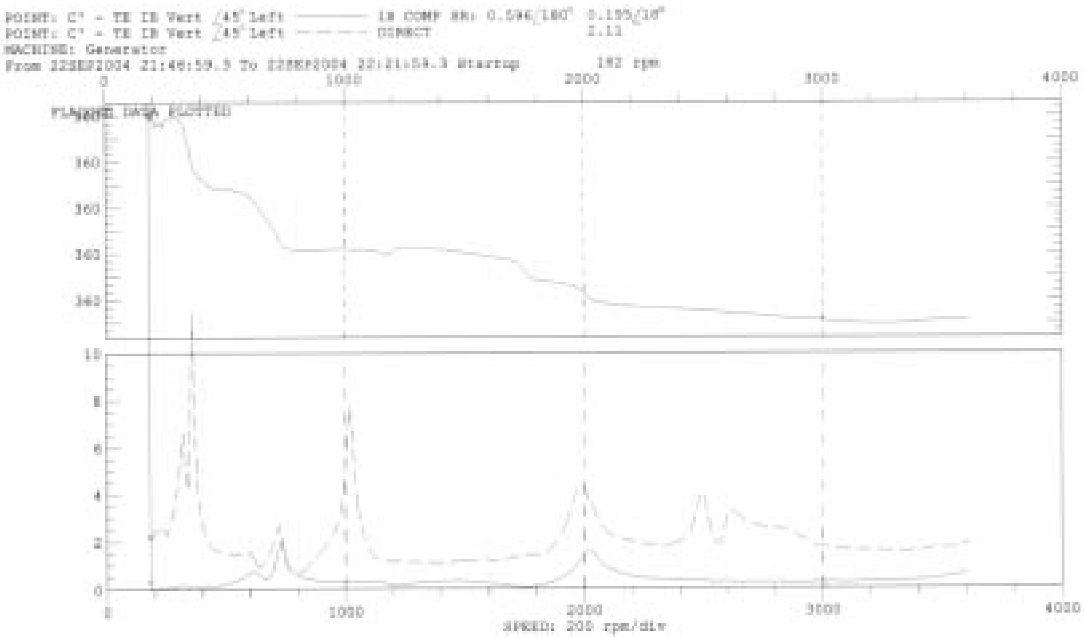


Figure 20: “Final Balance” in Bunker (GEN FRONT)

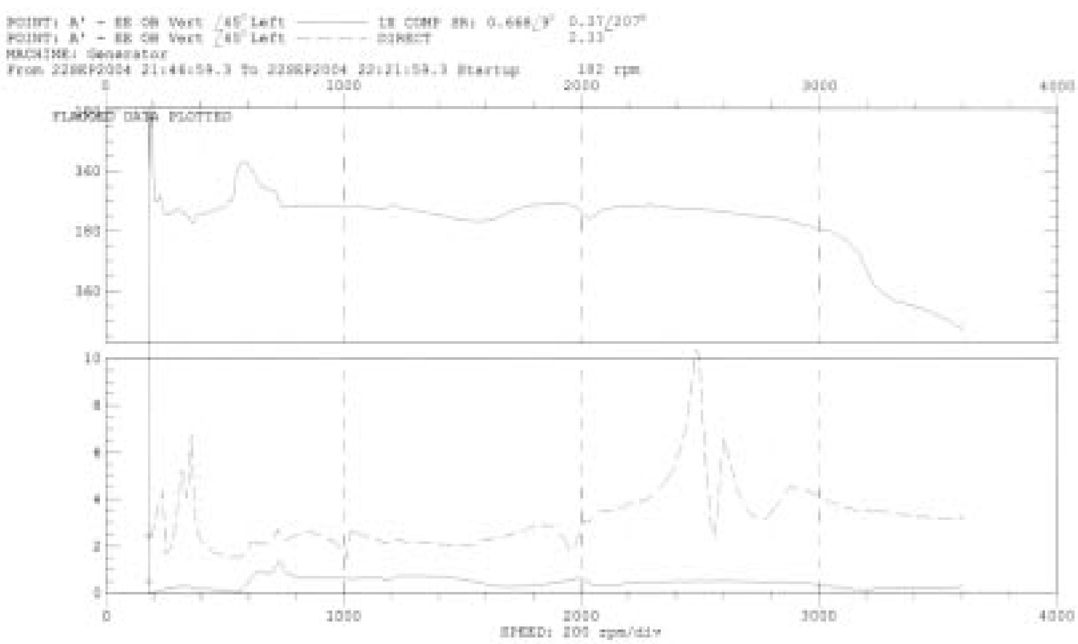


Figure 21: “Final Balance” in Bunker (GEN-REAR)

Figure 22: Final Distribution of Weights in “Body” (~24 lbs.)

CONCLUSION

After the third balance trial in the field, the rotor response was surprisingly good. From the initial roll-up to speed and then loading to 750 MW vibrations were very good and most importantly, “the thermal sensitivity” was gone.

The problem observed with this particular rotor can be traced back to its extreme flexibility and operating above third critical mode. Actually its operating deflection mode is at exactly the third critical.

In such cases it is extremely crucial to minimize the rotor body eccentricity. From the author’s experience, Body eccentricity should be less than 0.002”. Balancing should be done by N+2 method, or the third critical, which appears in the bunker at 3860 RPM should have been balanced much better than it was. That way the correction for the first mode should not be able to excite the third mode.

In reality what happened in this case is that the first mode balance correction weights distorted the body to bring the total sum of rotor mass into rotational centerline.

The distorted rotor balanced condition formed a bow in the middle of the rotor body.

At speed above 2000 RPM, the inertia of the rotor switched the rotation from journal centerline to rotor mass centerline, and the weight for correcting first mode became the unbalance for the third mode. (This phenomena of switching the axis of rotation was studied by very few, (Dr. Ehrich, Dr. Gunter).

There is another factor from the assembly that comes into play affecting rotor operating behavior. The slot wedges which secure the copper bars in the slots, when installed with tight fit, may “lock” rotor forging in a bowed shape of the particular mode. If the wedge friction is larger than the rotor elastic restoration force, the rotor may stay locked in that position. For better illustration of the events, one has to imagine a slender rotor with an eccentric body to be balanced by deforming the rotor into “W” shape.

This shape is also a natural third mode rotor shape. When cold, the rotor may be balanced to low amplitudes measured at the journals as in the bunker. When the rotor is exposed to heat (excitation current when in operation), it elongates. But the locked bow can only grow by increasing the bow, and thus increasing the eccentricity, unbalance and subsequently, the vibration. All this is occurring at rotor resonance, so responses are magnified by the rotor’s inherent amplification factor.

What had finally, inadvertently, happened is that the third balance correction in the field had reduced the effect of the original first critical balance correction in the bunker, and reduced the bow of the rotor. In turn, that prevented the wedge(s) from “locking up”.

This can be seen by observing the increased 1st critical response during shutdown, and correcting the third mode operating speed.

POST COMMENTS

This case clearly illustrates that "vibration" of the machine in the field is not just a balancing issue. Obviously the "problem" could have been "solved" by balancing without ever knowing the root cause. But by utilizing the method of troubleshooting and diagnosing the root cause, similar events can be avoided in the future. By addressing total repair process and adhering to and validating all acceptance criteria during the repair process, surely the "unknown" problems can be avoided. The whole issue may be, fine tuning some work processes, and transferring some work process costs to the right place, where they belong.

This case history also confirmed that a diagnostic approach to solving vibration problems does not increase the total cost vs. the traditional approach to "balancing". It only removes some of the "black magic" from "balancing" the traditional way.

REFERENCES:

Ehrich, F.F., 1992 *"Handbook of Rotordynamics"*, McGraw-Hill, Inc. New York, N.Y.
Gunter, E.J., PhD., and Chen, W.J., PhD., *"Dyrobes Dynamics of Rotor Bearing Systems"* software. (Ver 8.0).
E.J.Gunter, Jr. and Ronald L. Eshleman, *"Selected Papers on Rotordynamics"*, Part I and II, Vibration institute, Clarendon hills, IL 1983.
Bently-Nevada Corporation, *"Advanced Machinery Diagnostics & Dynamics Seminar Proceedings"*, Minden, Nevada, 1989.
Vibration Institute, *"Rotordynamics and Balancing Course"*, Syria, Virginia, 1991
Bently-nevada Corporation, "ADRE, Vibration Measurements System", (ver.5.1)

ADDENDUM

First Shutdown (Figures 23–26)

Final Start-up (Figures 27–30)

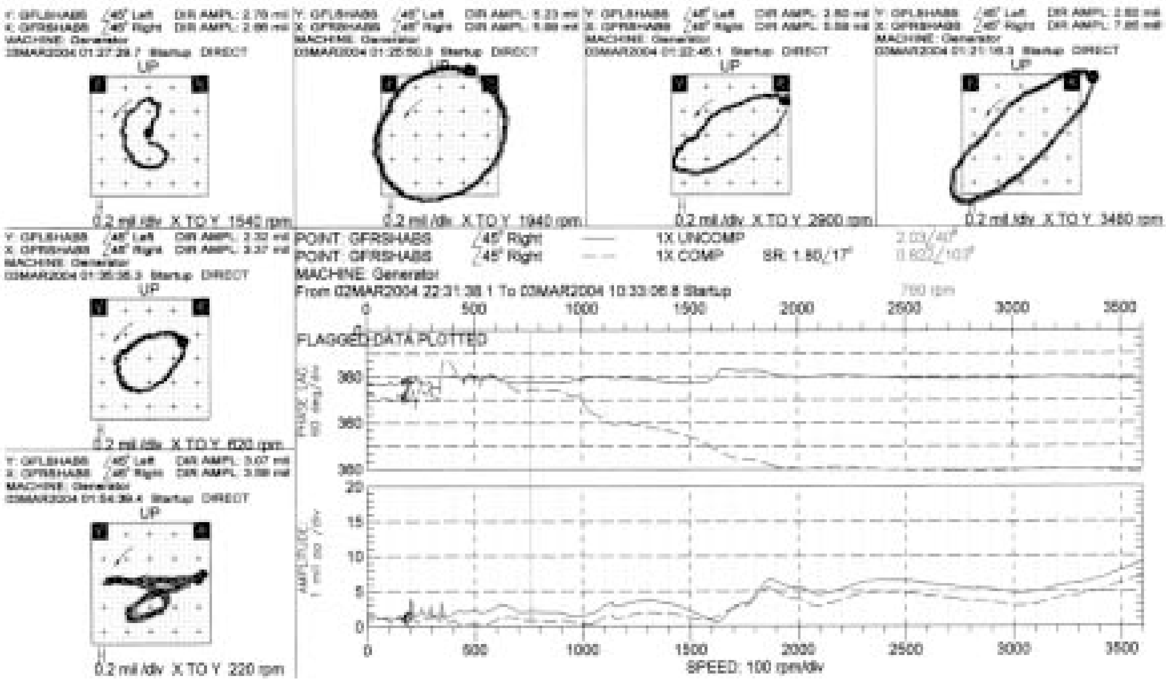


Figure 23: Field Balancing– First Shut Down

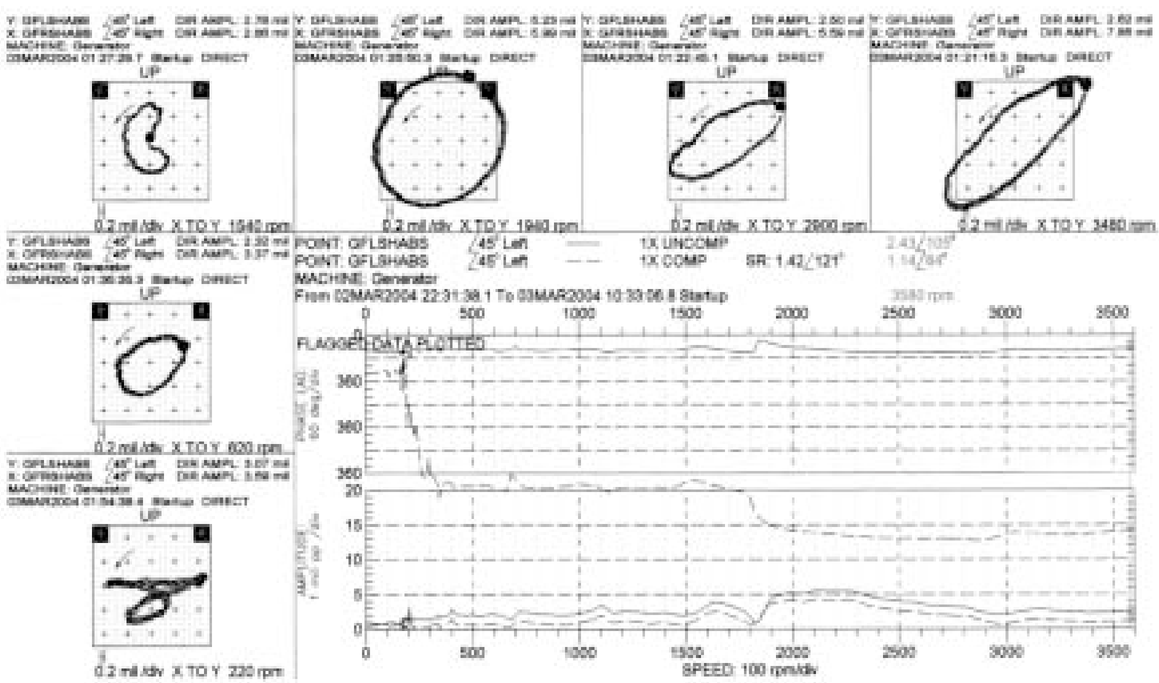


Figure 24: Field Balancing– First Shut Down

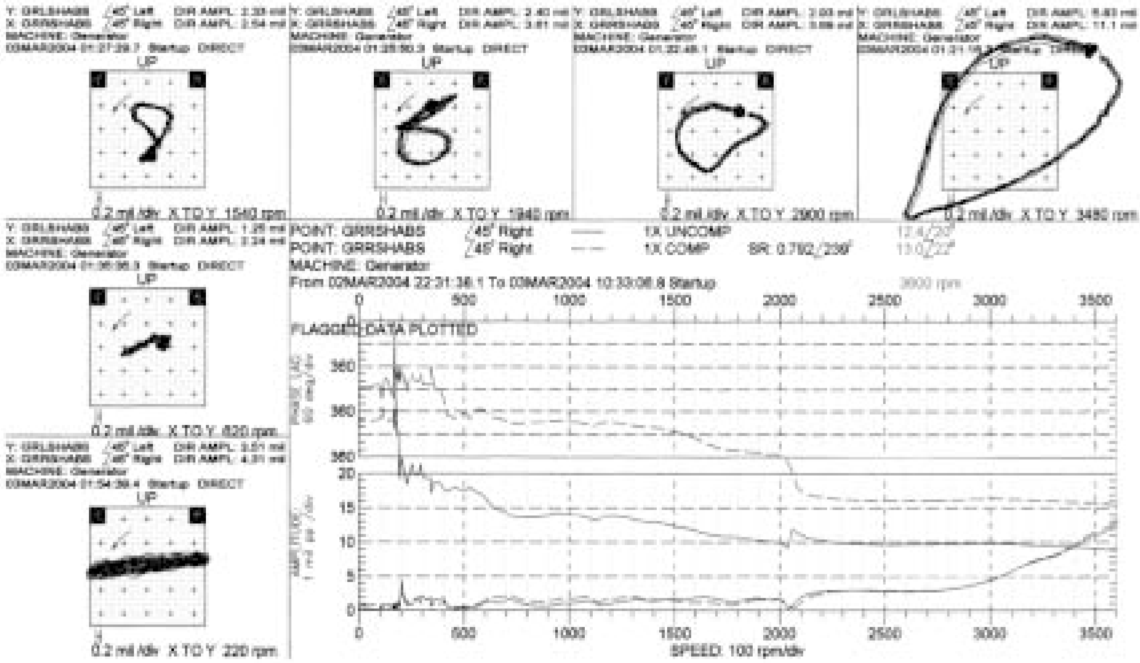


Figure 25: Field Balancing– First Shut Down

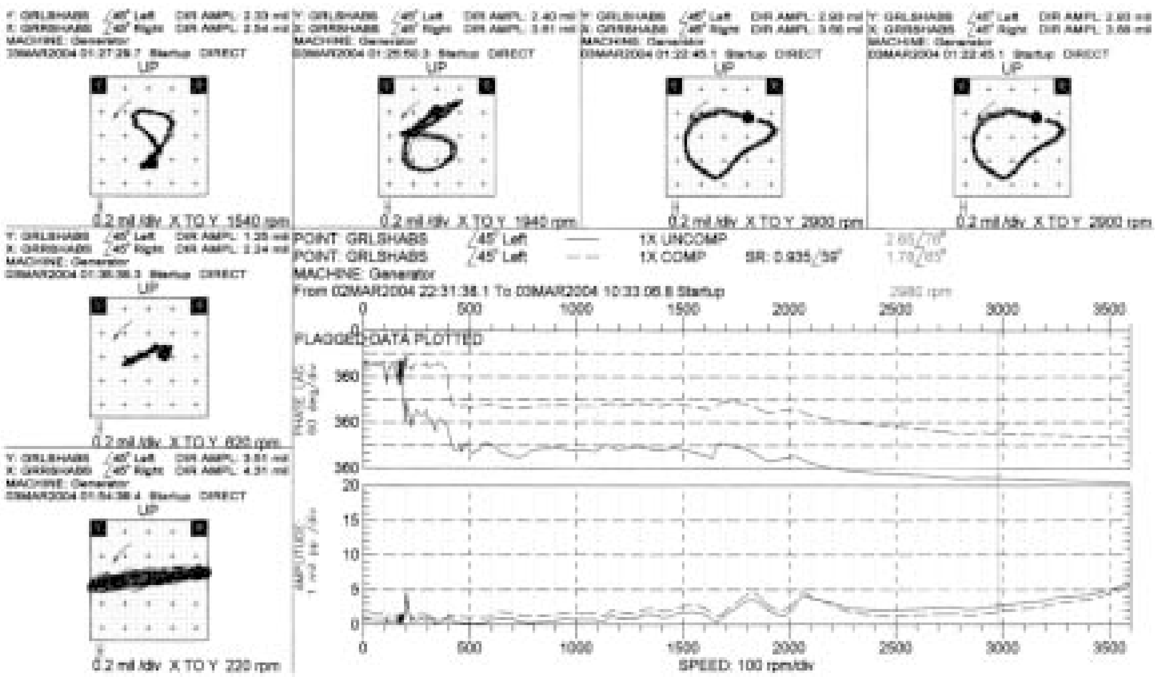


Figure 26: Field Balancing– First Shut Down

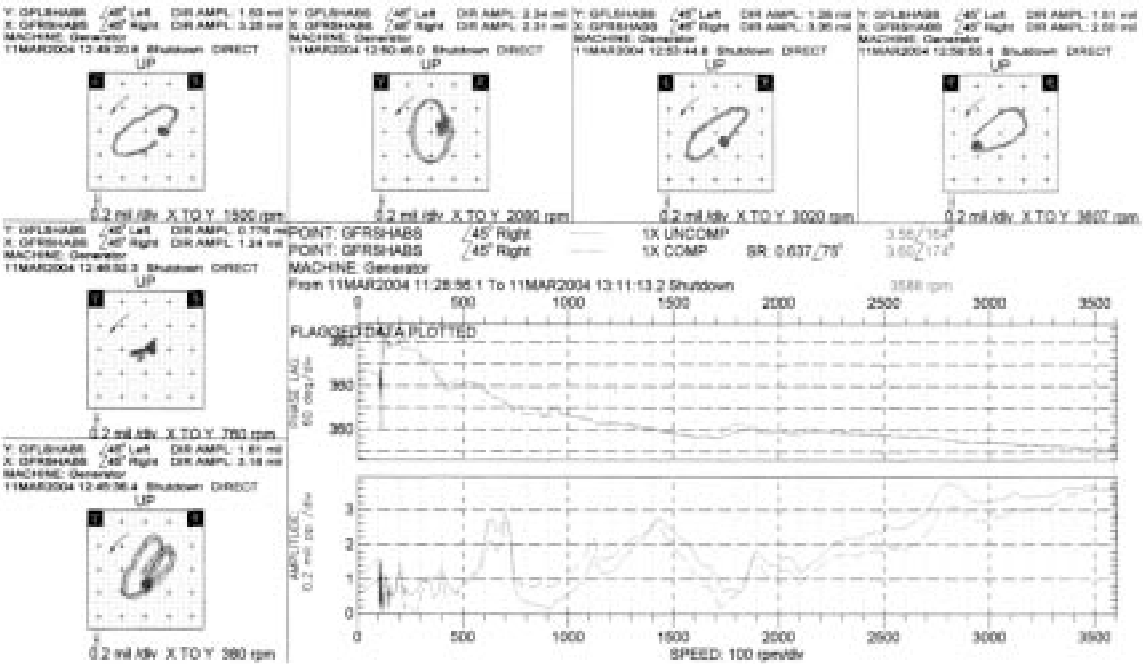


Figure 27: Field Balancing– Final Start Up

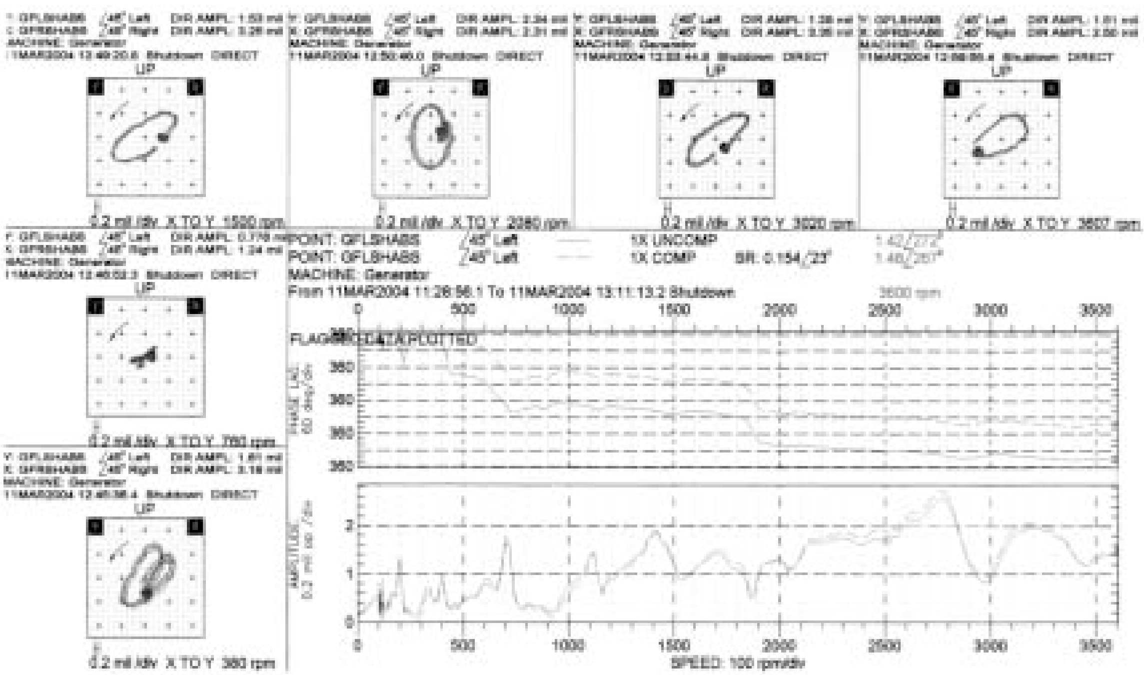


Figure 28: Field Balancing– Final Start Up

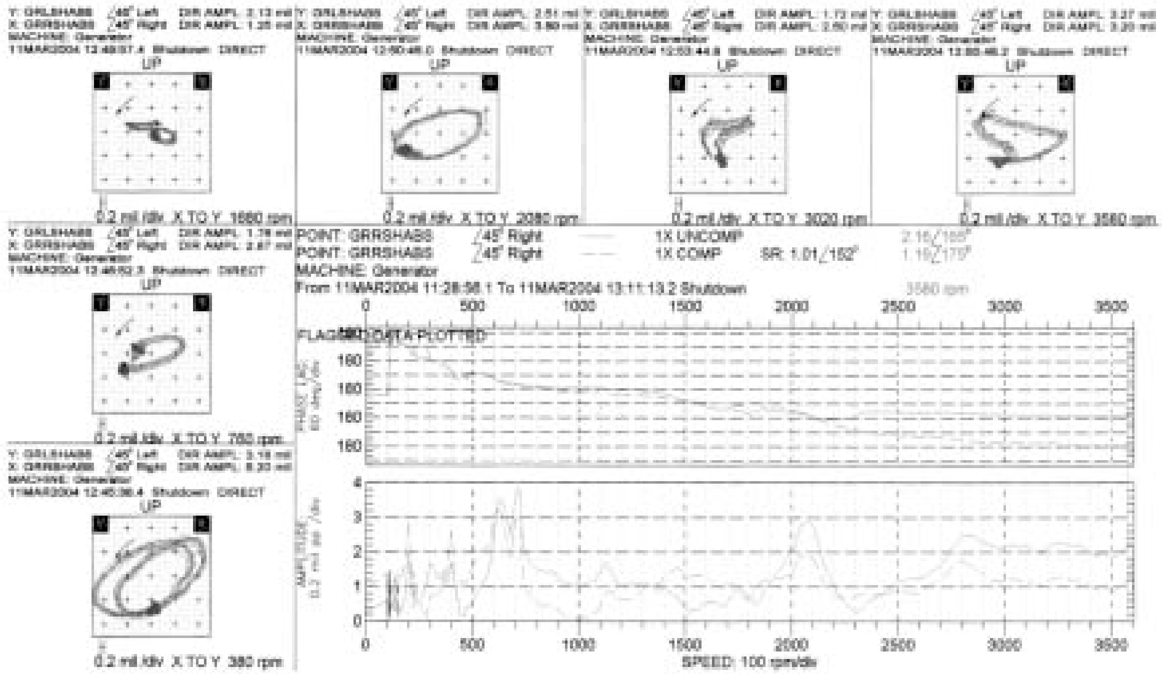


Figure 29: Field Balancing– Final Start Up

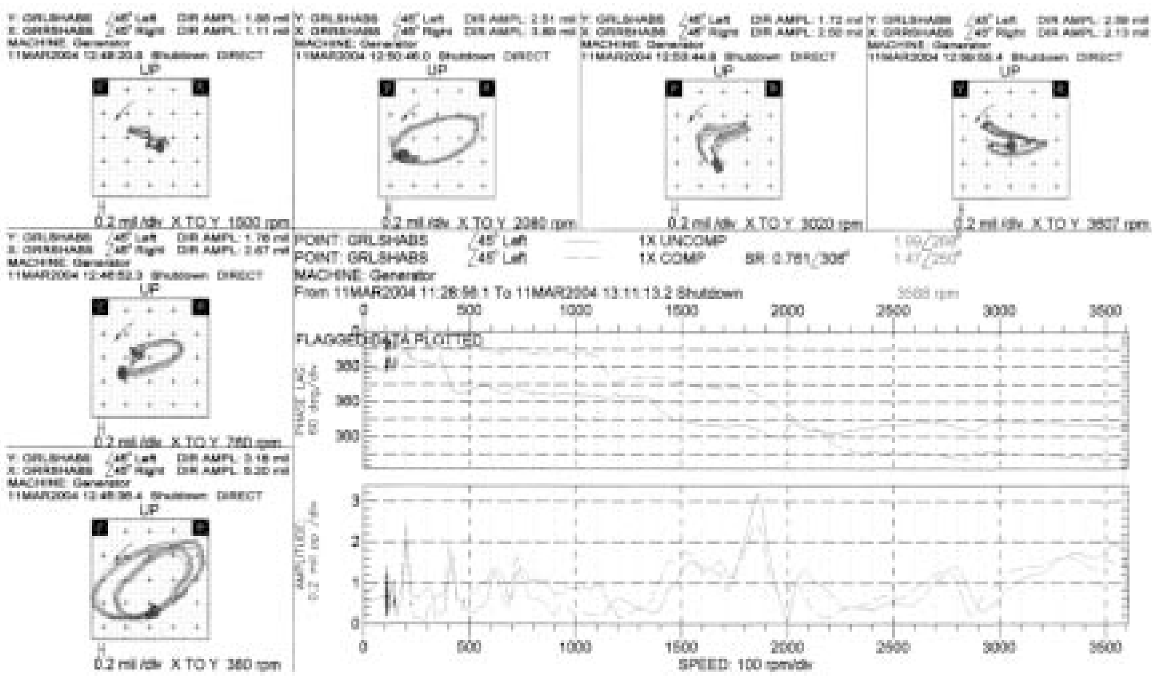


Figure 30: Field Balancing– Final Start Up