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DRAFT: DIAGNOSIS AND TREATMENT OF BOWED, MISALIGN, AND ECCENTRIC ROTOR TRAINS

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ABSTRACT

The intention of this paper is to demonstrate the advantages and the economic benefits of vibration root cause analysis and diagnosis, proper treatment of rotors during service outage, and follow up after commissioning of the units versus the general industry approach of field balance all vibration problems. During the course of the paper several diagnostic tools and repair procedures will be introduced and discussed. Special enfaces is made in the identification and treatment of rotor's bows, misalignments, and eccentricities since they represent the most common and less recognized vibration root cause. A case study will be used as a guide to facilitate the comprehension and to illustrate real industry results that can be achieved by systematically following the proposed methodology.

1) INTRODUCTION

Rotor bows, misalignments, and eccentricities within rotors and their couplings are the most common causes of vibration in large turbosets and perhaps also the least recognized during rotors repairs by plant managers and service shops. The general trend in the industry has been to treat most vibration issues as if they were the result of local unbalances in the rotor train, therefore, the solution most often employed is to simply try to balance the rotor or rotor train. Balancing theories have been developed and matured over the years and are now well known. Bishop [1], Kellenberger [2], Rieger and Zhou [3].
Bishop's theory and method of balancing is based on an uncompleted modal theory [2], utilizing N numbers of balancing planes, (N being equal to number of rotor's critical speed modes encounter in operation). Kellenberger's method differs in a way that requires N+2 balancing planes, it is also based on modal theory but it address the 2 rotor rigid modes by means of 2 extra balancing plains required to solve the rigid rotor modes due to effect of unbalances at low speed, prior to solving the modal response due to effect of unbalances at higher speeds. Although there was a heated debate in the past over which method is best suited for real life application, experience had proved both methods equally good within the manufacturing environment, but mostly ineffective when dealing with rotors which eccentricities exceed those recommended in the ISO 1940. The later category applies to the majority of rotors with a long service life.

From an engineering perspective there is no reason why a rotor will have to be balanced again and again in the field, after being balanced in a balance facility (regardless of the balancing method employed), in order to keep it operational. Aside from any engineering consideration one should also bear in mind that the cost of such rebalances in the field and the lost of revenue due to down time, make these rebalances costly and undesirable[1]. Therefore the question: what is it so often overlooked during rotor's repair that needs to be balanced on the field? automatically rise. The answer lays in the fact that almost all rotor repair processes in the service shops (OEM or Non-OEM) have evolved from the OEM's original manufacturing processes (that only apply to new rotors and that do not contemplate large deviations from the original design) leaving many critical rotor dimensions unchecked as there was no

[1] At an average sale price of 0.05 U$D/kWh a typical 300MW unit produce $360,000 a day

need to check them during manufacture. Some of these deviations become evident at first during balancing in a balance facility, but others might not become evident until after a rotor is "aligned" and assembled within the train. By this time it is too late to do any corrections on the rotors, and so, users and service shops resort to "Field Balancing". This type of situation should never occur if a rotor destined for service repair is properly inspected and repaired.

The best solution in the case of rotor train bows, misalignments and eccentricities is to correct the geometric deviations instead of attempt to balance them. Unless there has been a mistake during assembly (misalignment of two good rotors), correcting geometric deviation involves rotor machining and can only be accomplish during a major outage and at a service shop, this is the main reason why power plants are somewhat reluctant to do it and instead opt to field balance the rotor trains again and again. In addition to the aforementioned unwillingness from utilities, it has to be mentioned that in the past, the general approach to all vibration problems has been balancing; so there is a legacy of unwilling to invest in vibration diagnostic and root cause analysis in the industry.

2) CASE HISTORY

A large 750MW turboset had been suffering from vibration problems for several years. As a result of the vibration experienced by the train the HP front bearing had been failing at an approximate rate of once per year. The historical and pre-outage vibration condition of the rotor train can be seen on Table 1 along with other operational parameters.

The unit has been running in this condition for several years and several attempts to fix it had been made by different OEMs and by the utility resulting in large difference of opinions regarding the root cause of the problem, but surprisingly, they all agree that the problem can be resolved by balancing since most of the vibration was in the once per rev frequency. As a result of various recommendations the machine train has been field balanced at least 5 times and once its rotors were high speed balanced as individual components in a balancing bunker facility with no significant reduction on vibrations or improvement in its running condition.
A sketch drawing of the turboset is shown in Figure 1. Note the singular design of using just one journal bearing at the IP, LP1 and LP2 in order to save space, as a consequence relying on the previous rotor's journal and bearing for alignment.

3) ROOT CAUSE ANALYSIS: DIAGNOSIS

In order to diagnose the problem the first tool utilized was analyzing the start-up, shut-down and steady state vibration data and comparing it against know problem behavioral data. A first inspection of the data show that the problem was up to some degree due to rotor train bow. This conclusion was based on the hysteresis between hot and cold condition observed on the bode plots. The nature of the bow and the component that was bowed were not readily available from the vibration data but it was suspected on the IP rotor as possible culprit. Principles of bow like behavior are described by Gunter E.J. [4], Gunter et. al. [5], Ehrich E. [6], and Bently D. et al [7]. In essence a rotor bow will exhibit an increase in vibration amplitude that is roughly proportional to speed and for which there is no phase change, such characteristics can be easily observed in a bode plot when measuring displacement with proximity probes or velocity with seismic or accelerometer probes. A second clue to characterize this problem (when measuring displacement) is the appearance of a small dip before or after the first critical that can be seen depending whether the balance correction for the bow is overcompensating or under compensating the effect of the bow. This approach is very useful as it help to quickly identify the source of vibration.

Depending on rotor's stiffness when a rotor is bowed the best balance condition that can be achieved is a residual displacement measured at the journal in the amount of the bow. Gunter [4, 5]. Note we use the word *displacement* instead of vibration as this observed motion is not the result of any mechanical energy exchange but the simple wobbling of the journal due to being eccentric to the rotational center line. This fact imposes a theoretical limit of attainable residual displacement (mils pk-pk) that can be obtained by balancing. In other words if a rotor has 2 mils of bow at its center the best balance condition will produce 2 mils pk-pk of displacement at the journals (not vibration). This theoretical limit does not apply to the forces created by residual unbalance (CF forces at the bearings) that will be effectively driven to zero.

An interesting consequence of this behavior is that balancing by influence coefficient method does not work well when used in conjunction with proximity probes readings and in the presence of a bow since the solution cannot be driven to zero due to the bias created by the bow in the input measurement data (at least in principle). Balancing using influence coefficient (IC) can be understood as utilizing a least mean square optimization algorithm (LMS) in conjunction to a negative feedback loop scheme with the goal of driven all variables to zero; but it is well known that a negative feedback loop fail to converge to zero in the presence of an unaccounted measurement perturbation.

A second useful tool is to observe the bearing center line in relation to the journal centerline at stand still and at load condition.
When rotor misalignment is present it can be of two types: rotors' centerlines are parallel to each other but not collinear creating a crank where the two rotors meet (couplings) or they can intercept each other at the coupling but not be parallel to each other creating and angular misalignment between rotors. A combination of the two types is also possible and perhaps it is the most common case. When this happens the centrifugal forces developed as a result of one rotor driving the other in an eccentric position are enormous and most of the time the only solution is to realign the rotors. In some cases balancing can help with this situation but most often it does not. Because while balancing misalignments large local forces and stresses are develop due to placement of large amount of balancing weights caution should be used while attempting balancing misalignments.

When bearing misalignment is present several problems can occur. If a bearing is placed at the wrong elevation (vertical misalignment) then the most common problems rotor trains can exhibit are oil/steam instability when unloaded or increased operating temperature when overloaded. If the bearings are misaligned left-right (horizontal misalignment) the most common consequence is that they induce a rotor misalignment as they serve as reference for rotor alignment. In addition they can induce side forces that can excite rotor or pedestal resonances or simply induce rubs. In general, bearing misalignment produces a disruption of the oil film and therefore greatly affects oil film damping and stiffness characteristics.

In order to find out whether or not misalignment is present in a turboset observation of the journal centerline at stand still, low speed, and full speed, comes very handy. Using the concept of torque induce rotor self-alignment, when the rotor train is brought up to speed and free of constrains it tends to run in a straight line around its center of mass (or following a catenary curve in the case of a horizontal rotor) as this is the minimal potential energy configuration possible and therefore its natural state. This idea allows us to assume that when the turboset is at full speed and before load is applied to the generator the turboset is in a straight line configuration defining the optimal journal centerline. (Straight line means no deviation left or right and following the natural sag of a catenary curve). Once the optimal journal centerline has been established, then we can work back and find out where are the bearings in relation to the journals.

Looking at the journal centerline plots, also known as "shaft average centerline", and assuming that at full speed all rotors are in a perfect line. We can draw the journal centerline at stand still by looking at the position of the journals at turning gear using the full speed position as reference. If we further assume that all journals seat at the bottom of the bearing at stand still, (before any torque is applied and no movement has happened) then this line also defines the bearing centerline.
A 3D plot, 2D vertical and 2D horizontal views of the reconstructed journal and bearing centerline are shown in Figure 2.

The solid line (0-0 line) represents the journal centerline as measured by the proximity probe gap at full speed (normalized with respect to the catenary) . This line becomes the reference line for the remaining analysis. The dashed line represents the journal centerline as calculated based on probe gaps at turning gear (4~10 rpm) and the circles represent the bearing centerline (0 rpm).

It is clear that a straight rotor can not fit the dashed line that represent the journal centerline at turning gear or at stand still (circles) therefore the rotor has to be deformed at stand still (bowed misaligned or eccentric in some manner) in order to fit the observed configuration.

The present journal centerline analysis confirms the previous diagnosis, in addition, it is possible to estimate the amount of bow based on the centerline plot by fitting bowed rotors. Based on the best fit of a bowed rotor and geometric considerations the amount of bow at the HP and IP rotors were estimated in 0.002" and 0.004" respectively, in opposite angular directions. It was also found that the HI-IP and the IP-LP couplings had some degree of un-perpendicularity but no estimation was possible on the amount of angular misalignment. Angular misalignment or lack of perpendicularity at the couplings might reduce the bow.

In order to verify the existence of a bow and to analyze its possible remedy a computer model was created. A picture of the model is shown on Figure 1. Several scenarios were investigated, including HP and IP rotor bow and coupling misalignment. As a result of this, it was establish that the likely cause of the observed behavior was an IP rotor bow and a coupling angular misalignment between the IP and LP1 rotors. Figure 3 shows the result of the computer simulation. The magnitude of the stress at the #1 bearing babbitt produced by the simulated bow exceeds the yielding point of the material for the operating temperature which explains the #1 bearing failure rate.

At this point the presence of a bow had been confirmed with the use of 3 independent tests namely: observed dynamic behavior, reconstructive shaft centerline, and computer simulation.

4) PROPOSED SOLUTION AND JUSTIFICATION

Given that a rotor bow was diagnosed as the root cause of the exhibited vibrations problem then several methods to eliminate the bow were considered including: shot pining, stress reliving, corrective machining, and high speed balance.

Stress reliving is the only method that guaranties to bring the turbine rotor's forging back to its original stress free condition, minimizing the tendency to develop a new bow, although it poses two major practical problems: the rotor has to be debladed, it is time consuming and costly. Because of the above mentioned problems, stress reliving was discarded as an option.

Shot pining will partially achieve a stabilization of the bow preventing it from growing in the future (provided there are no external incidents or happenings) but will not correct the problem.

During the diagnose portion of the paper a computer model of the turboset was generated, whit the help of this model several options of corrective machining repair were then investigated; namely a) simply balancing of the rotors independently b) corrective machining of couplings and balancing c) rotors' centerline move, corrective machining of couplings and balancing.

Based on the experience of the authors and on the simulation of high speed balance each individual rotor separately; only balancing the rotors was discarded as a repair option very early on the investigation. Simply balancing each individual rotor separately in a balance facility will not produce an acceptable result leaving large forces to be resolved at bearings once all rotors are assembled together in the train. (See Figure 4)

Computer simulation of balancing, coupling corrective machining plus balancing, and journal centerline correction plus balancing are shown below in Figure 4, Figure 5, and Figure 6 respectively. In each case, the calculated displacement bode plots and bearing forces are shown.

From this analysis it is clear that the best course of action was to move the journal centerline of the rotors followed by high speed balance of each individual rotor. Although in theory there is no need to correct the journal centerline of the rotors and simply balancing them will produce a satisfactory result, in reality this is not possible. In order to perfectly balance the rotors an infinite number of balancing plane is needed; the subject HP and IP rotors only have 3 and 2 balancing planes available for balancing respectively. This is the reason why, not only for the subject rotors but for many others, balancing is not a real option when correcting a bow and, therefore, corrective machining is needed.

After corrective machining has been done then high speed balance of each individual rotor takes care of any residual unbalance due to machining tolerance errors.

It is recommended to perform a shot pining treatment to the forging to prevent the bow from growing in the future as this will rapidly undo any repair effort.

5) FINDINGS UPON ROTOR'S INSPECTION

The following is a summary of all the findings during the most recent outage of the subject rotors along with a brief explanation of the inspection procedures.

The first and most important inspection is the evaluation of runouts. Runouts are the single most telling pies of information about the rotors condition in the majority of cases and should not be overlooked or underestimated.

A full set of runouts are taken and evaluated. Runouts are taken at equally spaced angular degrees and at

several locations along the rotor length. In order to evaluate the runouts and be able to produce an eccentricity plot each individual sets of runouts is filter with a once and a twice per revolution filter that separate the eccentricity portion of the runout from the out of roundness and 2 pole effect in the case of 2 pole generators. Racic and Hidalgo [8,9]. This procedure can be used to evaluate any kind of rotor where concentricity and out of roundness are of concern. The runout evaluation and eccentricity plots are shown in Figure 7 and Figure 8.
The bow can easily be seen in both figures confirming the previous diagnosis. Opposite to what was diagnosed the magnitude of the bow in the IP case was only half of its prediction, the main reason of discrepancy is the fact that the coupling between the HP and IP rotors was badly off-square from cyclic bending stress and this cannot be seen in the diagnose.

A set of non-destructive tests were conducted to ensure the rotors are in good running condition from a material standpoint and that no other problem exist from years of running at high vibration levels.

6) CALCULATING THE JOURNAL CENTERLINE MOVE

With the new information on hand the computer model is updated and optimal centerline movements are calculated for the HP and IP rotors.
The optimization of the centerline move is a constrained multi-objective optimization; the goals are set to reduce the maximum effect of the 2 bows through the speed range (0-3600 rpm) as measured at the bearing forces and at the same time to minimize the amount of machining to reduce the risk of machining errors later during the correction.

$$Min\left(\lambda_1 \sum_{i=1..Nb} \left(\max(f_\omega)_i\right)^2 + \lambda_2 \sum_{j=1..Nc} \left(x_j - e_j\right)^2\right)$$

Subject to:

$$\left(x_j - e_j\right)^2 \leq K_j \quad j = 1..Nc$$

Where: λ are weighting coefficients, f_ω is the force at the bearing i and at speed ω, Nb is the total number of bearings, Nc is the total number of journals (or centerline references) moved, x is the original location of the journal center j, e is the move made to the journal center j, and K is the maximum machining allowable to the journal center j.

Constrains are set as follows: in order to change the journal centerline we have to reduce the size of the existing journals and other collinear critical surfaces, then there is a physical limit to how much machining can be done.

Termination criteria were set in accordance to what it is physically possible to machine ~0.0002" of journal's centerline movement and ~300Lb of centrifugal force at pick response.

The optimization algorithm largely use the FE model and linear simulation to calculate the forces produced by the journal centerline shifts at each speed point and then utilize a nonlinear programming algorithm (NLP) to find the optimal solution. Rao [10].

As a result of the optimization it was establish that the HP front journal had to be moved 0.0018" in the direction of the existing bow and that the IP dummy journal and coupling fit need to be moved 0.0040" also in the direction of its bow. The remaining HP rear and IP rear bearings were not moved. If we recall that these rotors weight ~20000Lb and ~50000Lb respectively it becomes clear the technical difficulty of handling and accomplishing such small moves of their centers of rotation.

After all machining was completed, including journal centerline moves and coupling squaring to the new rotor's centerline, the rotors were individually high speed balanced to account for any machining inaccuracy that may have occurred. The criterion for a satisfactory balance was based on turboset he minimization of transmitted forces to the bearings and not the displacement observed at the journals. In order to balance the IP rotor it was necessary to adapt a temporary stub shaft with a bearing since this rotor has only one bearing in its design. This activity alone requires a special attention during the execution.

7) MACHINE TRAIN START UP AND RESULTS

The turboset was reassembly paying close attention to the alignment of the rotors, although not required per OEM specs a swing check[2] was recommended to assure and verify a proper alignment of the HP to IP rotors. Unfortunately, due to time constrains, it was

[2] A swing check refers to the operation of rotating the coupled rotors while supporting the free end of the just coupled rotor with a sling allowing it to swing as a pendulum to check for alignment.

not done. Nevertheless first roll up to speed resulted in excellent vibration levels as can be seen in Figure 10 indicating excellent mechanical condition of the repaired rotors. As the machine was loaded and it warmed up, the vibration levels increased to the values shown on Table 2. This is not unusual on turbosets, when the rotor train and casing fully expand and every bearing finds its final position due to thermal expansion, vibration usually changes as the hot bearing and casing alignment will be different from the cold one; on the other hand the machine is now operated under full electrical and steam loads that produce additional external forces.

In order to evaluate the new alignment condition we followed the same shaft centerline reconstruction procedure described earlier on section 3. The results shown in Figure 9 use the same scale for an easier comparison. Since only the HP and IP turbines were realigned, leaving the LPs and rest of the train unchanged, the main change can be seen at the HP and IP turbines. It is now clear that this is not an optimal alignment and that further improvements in vibrations can be achieved if the turboset were realigned as a unit and the swing check performed. In addition to the full realignment removing some of the LP1 balance weights will improve the machine running condition as some of these correction weights were put to compensate for the IP-HP bow prior to this outage and are no longer needed.

The actual overall performance of the turboset is superior, this is reflected on smaller pedestal vibration under load and smaller vibration overall. A years old chronic vibration problem has been resolved without resorting to field balancing, but by addressing the problem at its root cause. The total cost of this repair as an schedule outage is estimated in half of the const of an emergency outage due to bearing failure and about the same as an emergency field balance shot. Therefore the savings over time are clear.

8) CONCLUSIONS

The advantages and the economic benefits of vibration root cause analysis and diagnosis, proper treatment of rotors during service outage, and the corresponding follow up after commissioning had been demonstrated with the help of a case study, for which, a years old chronic vibration problem had been solved. The economic benefits are clear since no fields balancing had been needed after the outage and the #1 bearing had not failed again.

A methodology for vibration diagnosis had been presented and discussed along with a verification process and a discussion on corrective actions highlighting the key aspects of each step. Each independent diagnostic method has been supported by the others greatly increasing the reliability of the overall diagnostic process.

A case study has been used to illustrate this process. The authors feel that the same methodology can be applied to other problem and that better engineering results are to be expected from its use in comparison to the current general industry approach of field balancing all vibration problems. A corresponding economic benefit is expected as well from the savings of avoiding unnecessary field balance and emergency repair outages.

	Journal Vibration [in pk-pk]		Bearing Cup [in/sec 0-pk]	Bearing Metal Temps [ºF]	
Bearing #	3600 rpm	Critical Speed	3600 rpm	Left	Right
HP Front	~0.010	~0.008	0.2	130(b) 178(t)	147(b) 161(t)
HP Rear	~0.010	~0.010	0.4	139(b) 154(t)	181(b) 153(t)
IP Rear	~0.014	~0.015	0.6	156(b)	182(b)
LP1 Rear	~0.005	~0.017	0.4	149(b)	189(b)
LP2 Rear	~0.012	~0.017	0.6	153(b)	181(b)
Gen Front	~0.007	~0.004	0.4		153(b)
Gen Rear	~0.012	~0.007	0.4		166(b)
Exciter Rear	~0.006	over 0.020	0.4		147(b)

Note: Bottom (b); Top (t)

Table 1: Rotor train vibration condition and bearing temperatures

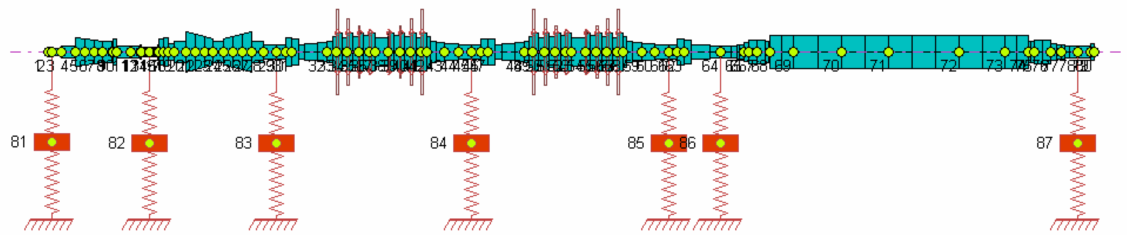


Figure 1: FE model of the turboset train

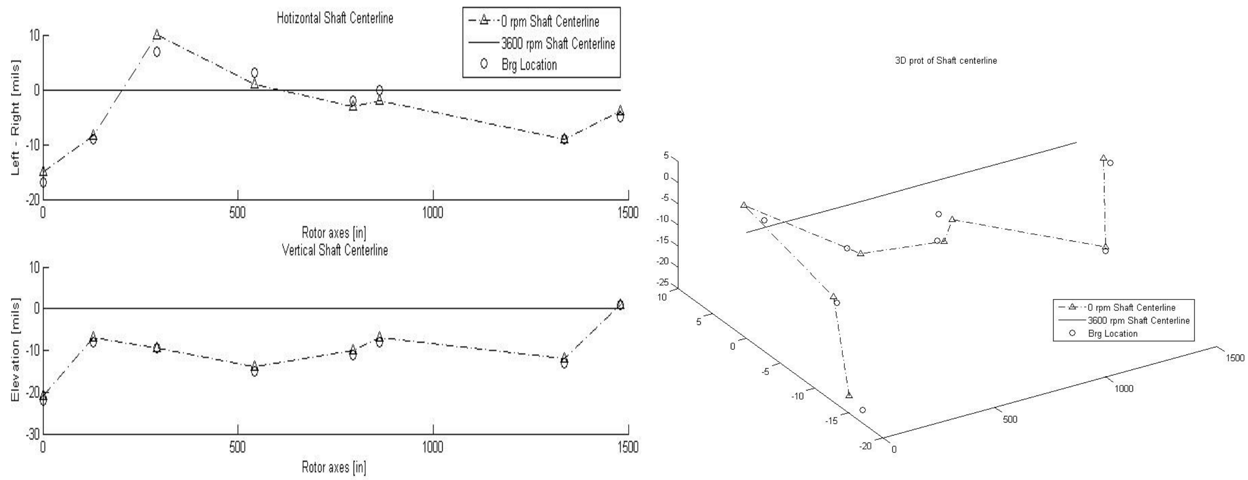


Figure 2: Journal and Bearing Centerline at Stand Still

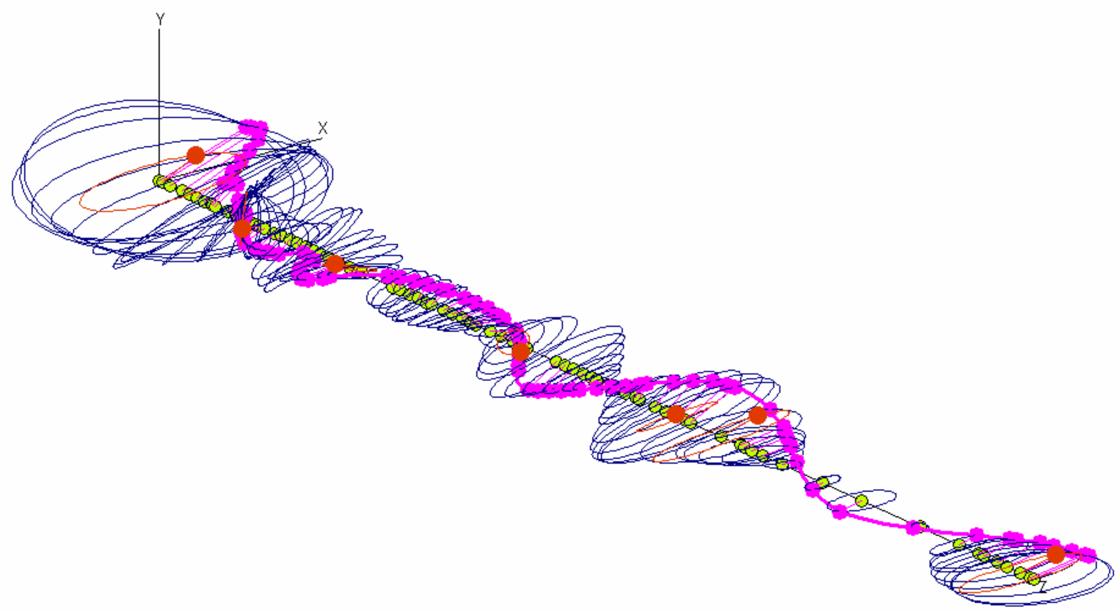


Figure 3: Synchronous rotor response to IP bow and coupling angular misalignment

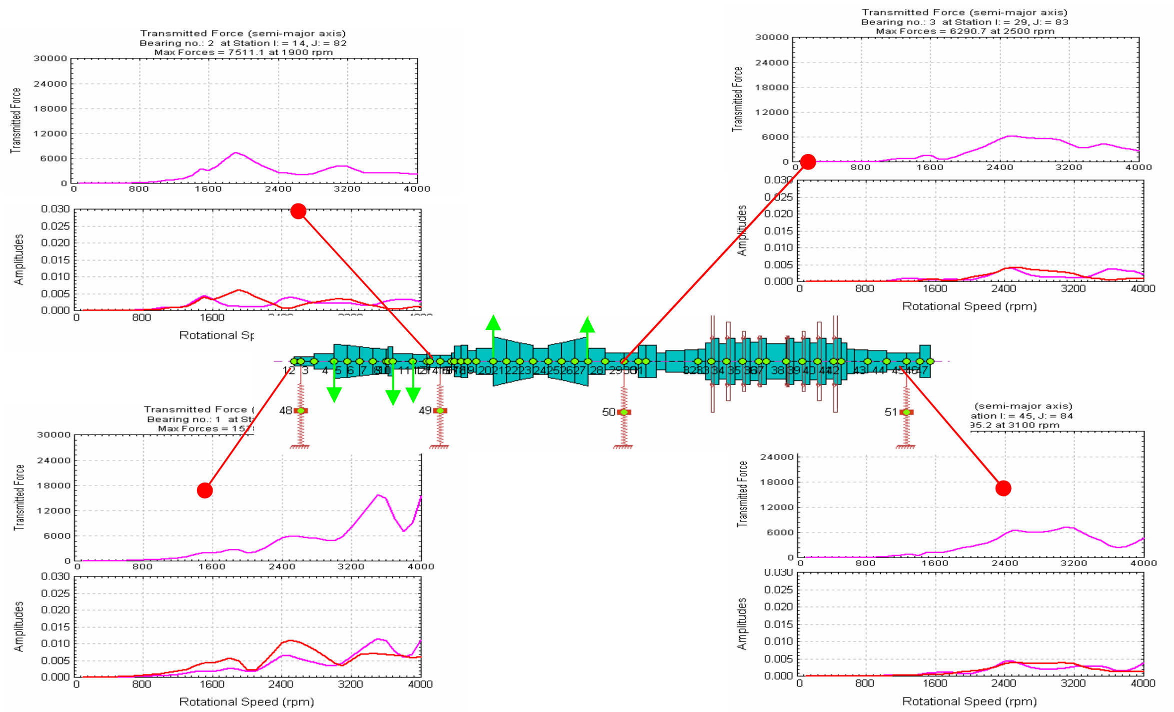


Figure 4: Individual rotor high speed balancing

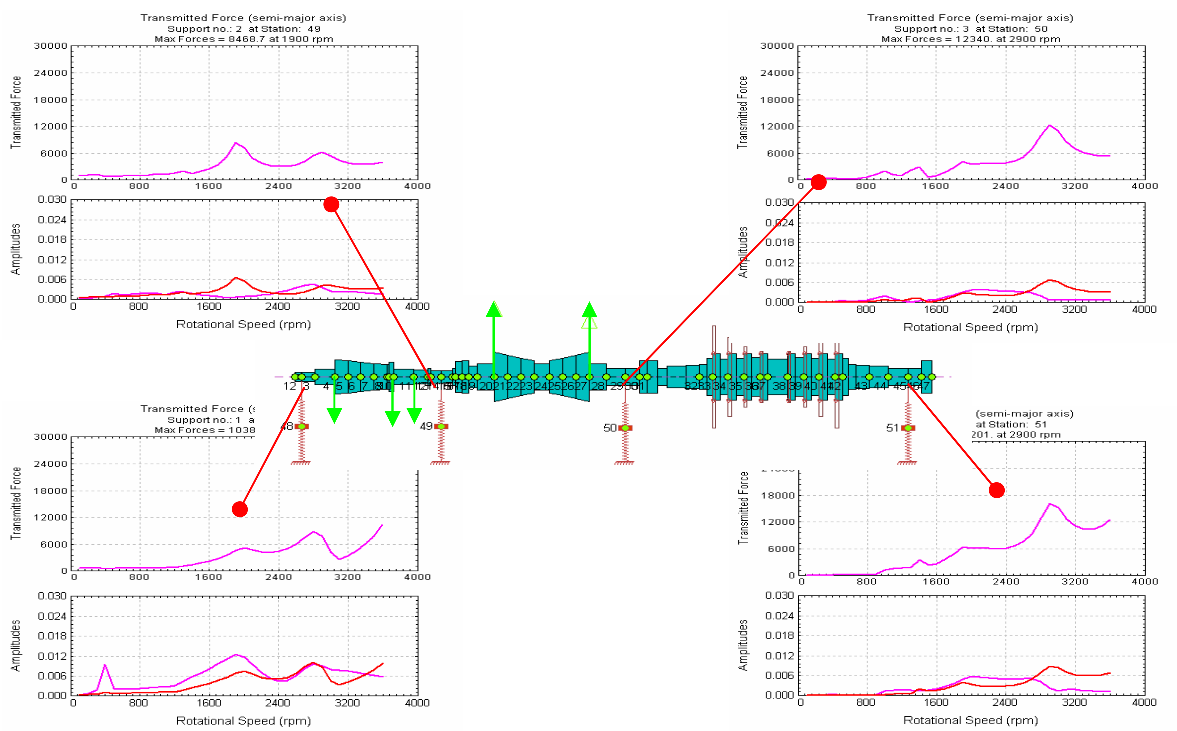


Figure 5: coupling machining and high speed balancing

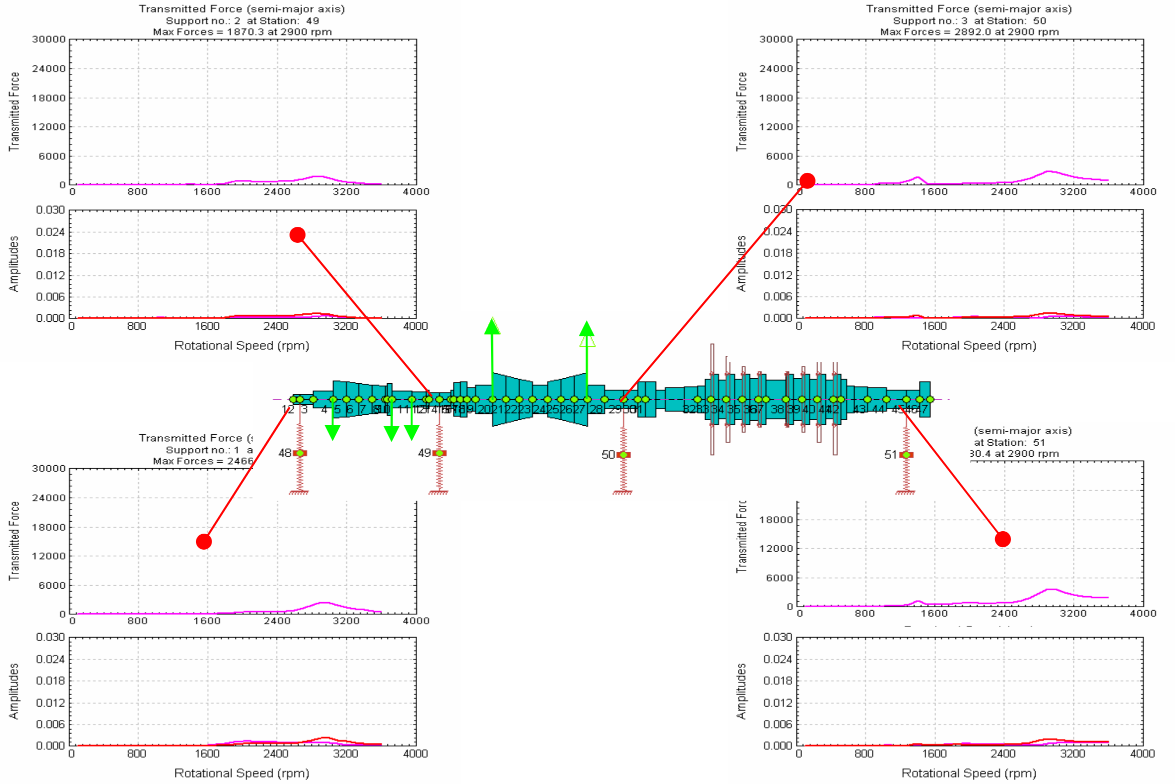
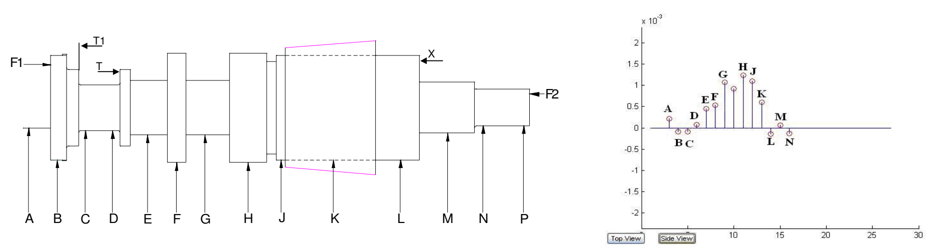
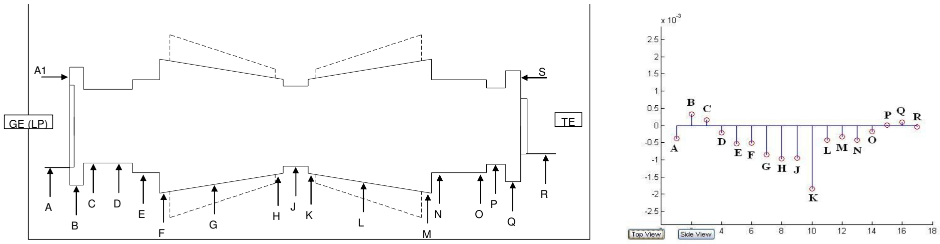


Figure 6: Journal centerline correction and high speed balancing



	Face		Fit	Cplg.	Journal						Inlet				Journal		Fit	Thrust					
	F1-L	F1-R	A	B	C	D	E	F	G	H	J	K	L	M	N	P	F2	T1-L	T1-R	T2-L	T2-R	X1-L	X1-R
0	0.0014		0.0006	0.0004	0.0000	0.0001	0.0009	0.0012	0.0022	0.0028	0.0030	0.0020	0.0010	0.0010	0.0003	0.0000	0.0004	0.0001		0.0007		0.0010	
45	0.0014		0.0008	0.0002	0.0000	0.0000	0.0008	0.0009	0.0020	0.0032	0.0045	0.0030	0.0015	0.0000	0.0003	0.0000	0.0006	0.0000		0.0007		0.0015	
90	0.0012		0.0006	0.0000	0.0000	0.0000	0.0005	0.0008	0.0010	0.0028	0.0015	0.0020	0.0010	0.0000	0.0000	0.0001	0.0008	0.0001		0.0005		0.0025	
135	0.0006		0.0001	0.0005	0.0002	0.0000	0.0003	0.0003	0.0006	0.0023	0.0020	0.0005	0.0000	0.0005	0.0001	0.0002	0.0007	0.0005		0.0002		0.0010	
180	0.0001		0.0002	0.0006	0.0002	0.0000	0.0001	0.0000	0.0000	0.0008	0.0000	0.0000	0.0000	0.0010	0.0003	0.0003	0.0004	0.0009		0.0000		0.0015	
225	0.0000		0.0001	0.0008	0.0002	0.0000	0.0000	0.0004	0.0002	0.0000	0.0010	0.0005	0.0000	0.0010	0.0001	0.0002	0.0001	0.0010		0.0000		0.0010	
270	0.0003		0.0001	0.0010	0.0002	0.0001	0.0006	0.0004	0.0011	0.0006	0.0005	0.0010	0.0005	0.0005	0.0001	0.0001	0.0000	0.0006		0.0001		0.0005	
315	0.0008		0.0000	0.0009	0.0002	0.0003	0.0009	0.0011	0.0017	0.0014	0.0015	0.0015	0.0005	0.0005	0.0003	0.0001	0.0001	0.0002		0.0003		0.0000	
0	0.0014		0.0006	0.0004	0.0000	0.0001	0.0009	0.0012	0.0022	0.0028	0.0030	0.0020	0.0010	0.0010	0.0003	0.0000	0.0004	0.0001		0.0007		0.0010	
Max	**0.0014**		**0.0008**	**0.0010**	**0.0002**	**0.0003**	**0.0009**	**0.0012**	**0.0022**	**0.0032**	**0.0045**	**0.0030**	**0.0015**	**0.0010**	**0.0003**	**0.0003**	**0.0008**	**0.0010**		**0.0007**		**0.0025**	
1X	**0.0008**		**0.0003**	**0.0004**	**0.0001**	**0.0001**	**0.0004**	**0.0005**	**0.0011**	**0.0016**	**0.0016**	**0.0012**	**0.0007**	**0.0003**	**0.0001**	**0.0001**	**0.0004**	**0.0005**		**0.0004**		**0.0008**	
Phase	35		51	257	223	313	1	5	5	54	38	25	27	245	337	188	92	210		32		103	
2X	**0.0000**		**0.0002**	**0.0001**	**0.0001**	**0.0001**	**0.0001**	**0.0000**	**0.0000**	**0.0001**	**0.0005**	**0.0005**	**0.0003**	**0.0004**	**0.0001**	**0.0000**	**0.0000**	**0.0001**		**0.0001**		**0.0004**	
Phase	0		40	312	312	313	307	321	339	329	34	63	59	0	0	335	320	22		29		52	
	-180		220	132	132	133	127	141	159	149	214	243	239	180	180	155	140	202		209		232	

Figure 7: HP Runout evaluation and eccentricity plot



	LP Coupling																HP Coupling		
	Face	Fit	Rim	Journal						Inlet				Seals		Dummy Journal	Rim	Fit	Face
	A1	A	B	C	D	E	F	G	H	J	K	L	M	N	0	P	Q	R	S2
0	0.0009	0.0011	0.0001	0.0000	0.0004	0.0018	0.0020	0.0010	0.0020	0.0020	0.0026	0.0030	0.0020	0.0010	0.0011	0.0003	0.0001	0.0008	0.0022
45	0.0010	0.0009	0.0002	0.0001	0.0003	0.0010	0.0015	0.0005	0.0010	0.0007	0.0005	0.0020	0.0018	0.0004	0.0006	0.0000	0.0000	0.0007	0.0046
90	0.0000	0.0000	0.0009	0.0004	0.0001	0.0003	0.0005	0.0000	0.0000	0.0000	0.0001	0.0010	0.0012	0.0003	0.0002	0.0001	0.0004	0.0012	0.0051
135	0.0002	0.0002	0.0010	0.0005	0.0000	0.0000	0.0000	0.0000	0.0000	0.0009	0.0000	0.0000	0.0006	0.0000	0.0000	0.0004	0.0004	0.0000	0.0040
180	0.0009	0.0004	0.0007	0.0004	0.0002	0.0003	0.0000	0.0005	0.0015	0.0015	0.0011	0.0010	0.0000	0.0004	0.0004	0.0003	0.0005	0.0015	0.0030
225	0.0009	0.0007	0.0005	0.0003	0.0005	0.0010	0.0010	0.0010	0.0017	0.0015	0.0030	0.0010	0.0010	0.0007	0.0005	0.0002	0.0003	0.0006	0.0008
270	0.0014	0.0010	0.0001	0.0000	0.0005	0.0013	0.0015	0.0020	0.0021	0.0023	0.0035	0.0020	0.0020	0.0010	0.0007	0.0000	0.0000	0.0008	0.0000
315	0.0012	0.0011	0.0000	0.0000	0.0004	0.0016	0.0020	0.0015	0.0018	0.0022	0.0031	0.0020	0.0021	0.0011	0.0004	0.0003	0.0001	0.0009	0.0005
0	0.0009	0.0011	0.0001	0.0000	0.0004	0.0018	0.0020	0.0010	0.0020	0.0020	0.0026	0.0030	0.0020	0.0010	0.0011	0.0003	0.0001	0.0008	0.0022
Max	**0.0014**	**0.0011**	**0.0010**	**0.0005**	**0.0005**	**0.0018**	**0.0020**	**0.0020**	**0.0021**	**0.0023**	**0.0035**	**0.0030**	**0.0021**	**0.0011**	**0.0011**	**0.0004**	**0.0005**	**0.0015**	**0.0051**
Evaluated Eccentricity																			
1X	**0.0005**	**0.0005**	**0.0005**	**0.0003**	**0.0002**	**0.0009**	**0.0011**	**0.0009**	**0.0010**	**0.0010**	**0.0019**	**0.0011**	**0.0010**	**0.0005**	**0.0003**	**0.0000**	**0.0002**	**0.0000**	**0.0026**
Phase	288	314	138	145	293	322	331	290	289	284	286	338	341	306	329	171	154	254	94
2X	**0.0001**	**0.0001**	**0.0001**	**0.0000**	**0.0001**	**0.0002**	**0.0001**	**0.0001**	**0.0004**	**0.0004**	**0.0001**	**0.0004**	**0.0003**	**0.0000**	**0.0003**	**0.0002**	**0.0001**	**0.0001**	**0.0002**
Phase	31	17	-58	-43	44	18	45	90	16	-18	26	21	88	0	22	-22	-21	29	46
Phase			122	137	224	198	225	270	196					180	202	158	159	209	

Figure 8: IP runout evaluation and eccentricity plot

	Journal Vibration [in pk-pk]		Bearing Cup [in/sec 0-pk]	Bearing Metal Temps [ºF]	
Bearing #	3600 rpm	Critical Speed	3600 rpm	Left	Right
HP Front	0.006	0.002	0.15	130(bottom) 145(top)	179(bottom) 150(top)
HP Rear	0.005	0.003	0.18	137(bottom) 154(top)	179(bottom) 153(top)
IP Rear	0.003	0.004	0.1	152(bottom)	175(bottom)
LP1 Rear	0.006	0.004	0.04	149(bottom)	189(bottom)
LP2 Rear			0.2	153(bottom)	181(bottom)
Gen Front			0.2		153(bottom)
Gen Rear			0.2		166(bottom)
Exciter Rear			0.2		147(bottom)

Table 2: Vibration of the entire turbo set (warm condition)

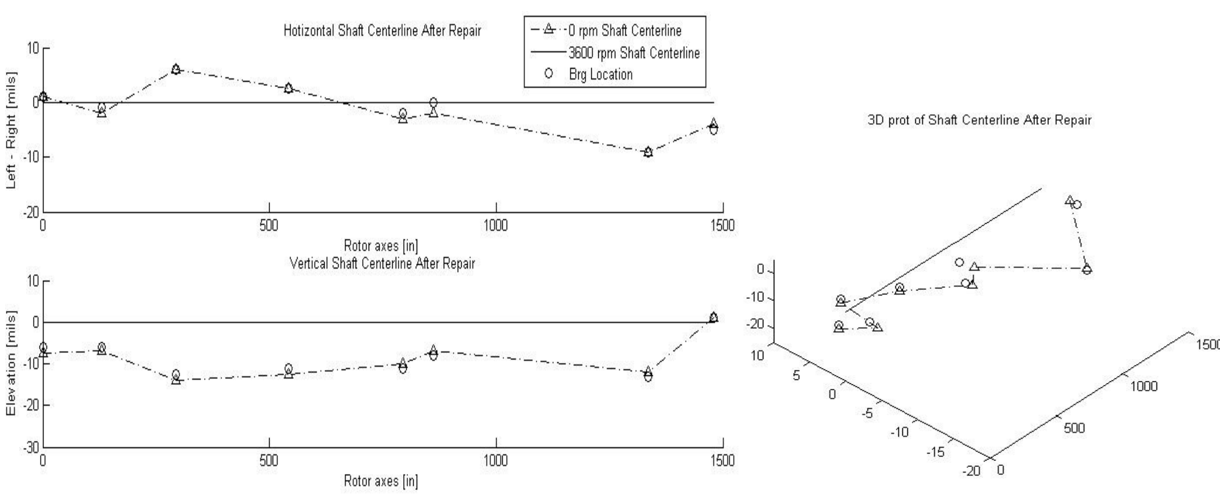


Figure 9: Journal and Bearing Centerline at Stand Still After Repairs

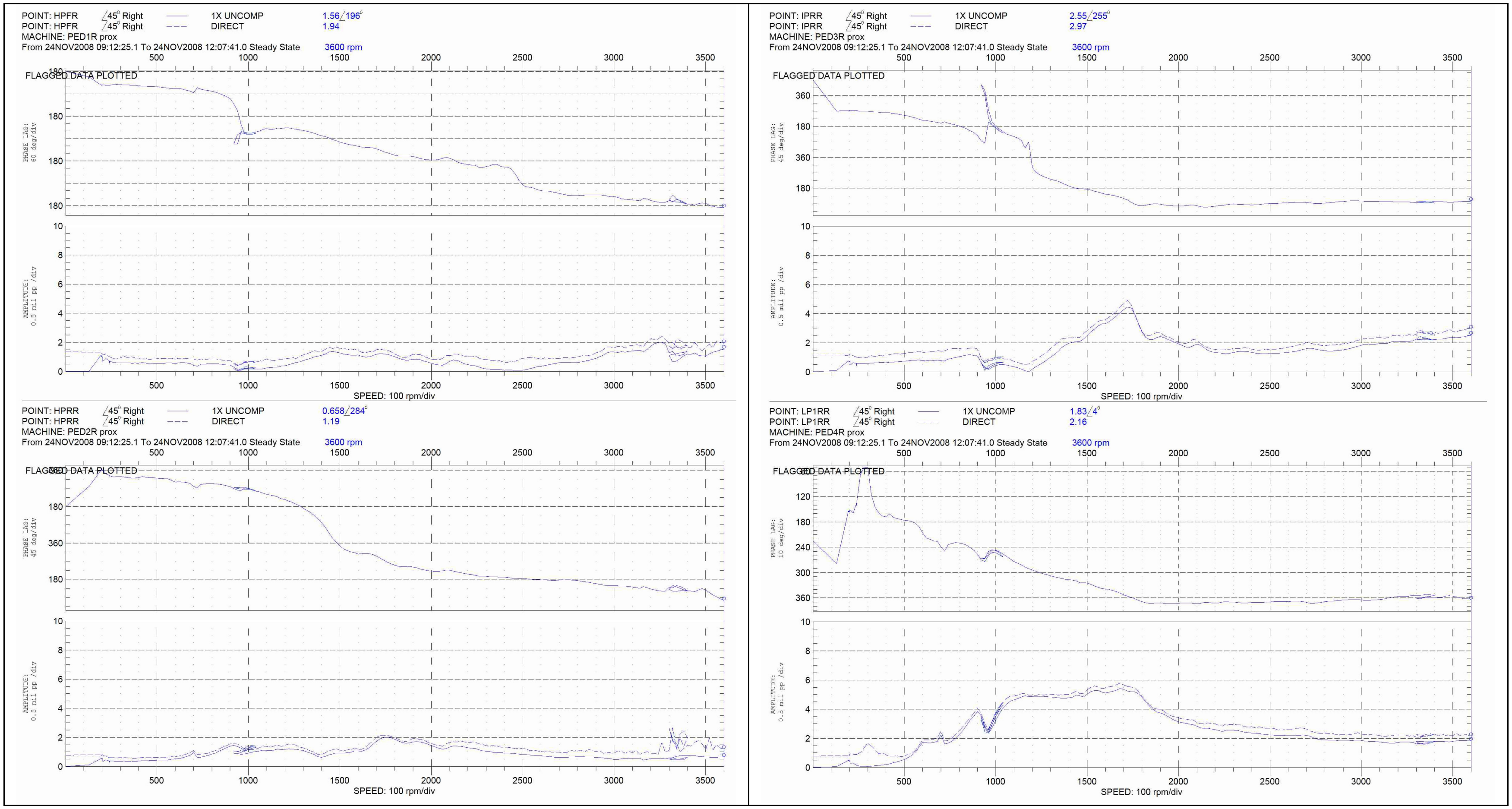


Figure 10: Start up bode plots of the HP, IP, and LP-1 turbine rotors

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