The Effect of Bearing Clearances On Its Dynamic Stiffness

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ABSTRACT

A modified, up rated steam turbine generator, had developed a rotor subsynchronous stability problem, with high vibration amplitudes. The up-rating consisted of a steam path replacement and replacement of all seal strips. Steam operating parameters were unchanged. The original bearings were re-used. Upon the turbine re-start the rotor became unstable from the onset. Since bearings were unchanged and installed with the pre-existing OD and ID clearances, the author concluded that the external disturbing excitation forces had increased. One way to combat subsynchronous instability in bearings is to restore or increase bearing's dynamic stiffness.

Keywords:

STEAM TURBINE, BEARING, DYNAMIC STIFFNESS,
SUBSYNCHRONOUS WHIRLING.

INTRODUCTION

The paper presents the cause of the problem and the approach used to solve a subsynchronous vibration problem on a 50 MW steam turbine generator with the process steam extractions for a paper mill. The steam turbine was up-rated from 38 to 50 MW and upon restart it had developed a rotor subsynchronous stability problem, with high vibration amplitudes. Prior to up-rating the turbine, the rotor was stable with low vibration amplitudes. The up-rating consisted of the steam path replacement and replacement of all seal strips. Steam operating parameters were unchanged. Both, front and rear bearings are of elliptical shape with a clearance of 1.4 mils per inch diameter. The original bearings were reused. Upon turbine restart, the rotor became unstable from the onset. Since bearings were unchanged and installed with the pre-existing OD and ID clearances, the Author had concluded that the external disturbing excitation forces must have increased. One way to combat subsynchronous instability in the bearing is to restore or increase bearing's dynamic stiffness if disturbances can not be eliminated.

When steam excitation forces acting on a rotor are small, then the gravity acting on the rotor is usually sufficient to create "bearing load", even if the bearing has large ID clearance and if it is loosely mounted. Steam disturbance excitation forces may become sufficiently large to affect rotor attitude angle and bearing loading, thus pulling the bearing over the threshold of stability. With the assumption that bearings are energy converters, when there are higher disturbance forces, in order to suppress journal motion, they convert more of the rotor's motion into oil heat. This is typically done by increasing bearing dynamic stiffness. In this case this was achieved by "pinching" the bearing shell i.e. reducing the assembly from positive clearance to -0.001–0.002 inches of preload, and by machining the bearing pads with higher preload and lower vertical clearance. A proper reduction of vertical clearance in an elliptical bearing allows formation of top oil wedge, which is adding to the vertical load from the gravity. The above approach is practical and successful up to a point when too much heat generated from oil "starvation" can endanger the integrity of the bearing itself.

Both bearings were reworked and installed with the appropriately reduced clearances. Upon the turbine restart there was still a trace evidence of the instability. The unstable bearing behavior was, for all practical purposes, solved by the "classic" approach. Since there is a practical limitation as to how much the bearing clearances can be reduced without endangering the integrity of the bearing babbitt, a more permanent solution may be to consider a retrofit to ServoFluid™ control bearing.

BACKGROUND

Many bearing assembly procedures in the industry exist requiring bearing assembly clearance. On some large bearings for steam turbines these clearances are up to 0.008" diametral in horizontal and up to 0.012" in vertical direction. In theory there is no problem with these clearances looking from the static journal and bearing stability. On a heavily loaded bearing there is no problem from the dynamics point of view either, as long as journal attitude angle is in the range of 15°–30°, and shaft unbalance is no larger than ~3.0 mils pk-pk. The problem with bearing looseness becomes evident most often on lightly loaded journals at the ends of the train with one or more of these conditions. (Fig.1):

a) Coupling alignment causes high rotor "swing".
b) High unbalance.
c) High external forces acting on rotor, like flow through seals, rub, etc., causing strong influence on gravity load.

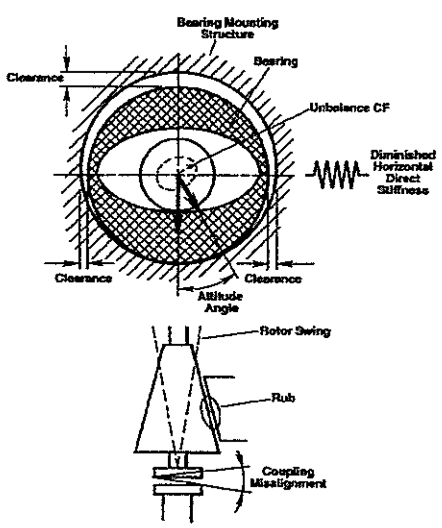


Figure 1. "Loose" Bearing Environment

The following is a representation of maximum vibration readings (major axis), in one real life case, on a machine with "loose" bearing shell before and after "pinching" the bearing:

	LOOSE mil/p-p	TIGHT mil/p-p
Shaft Relative	1.0	3.0
Bearing Shell	3.0	0.25
Shaft Absolute	4.0	2.5
Bearing Housing	0.05	0.25

TABLE 1. Variation in vibration amplitude readings between loose and tight bearing.

The sensors, proximity probes and seismic probes were mounted on the bearing shell. When the

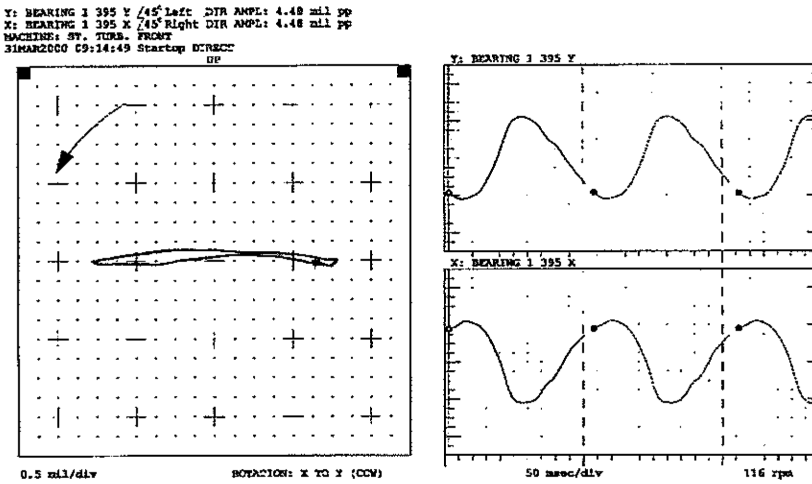


Figure 2. Typical Orbit of Shaft Running in "Loose" Bearing

bearing shell is loose and shaft and bearing are moving in phase, the relative vibration indication is diminished, while shaft absolute is increased to the sum of shaft and bearing motion.

When the bearing is "pinched", shaft absolute and relative displacement becomes almost identical. Dynamic force transmitted to bearing housing becomes real indication of the effect of the existing unbalance and system stiffness.

As seen in Fig. 2 there is a very high amplitude ratio in horizontal and vertical direction. The difference in amplitudes and their phase relation give us that highly elongated orbit shape, sometimes even looking as a straight line. This is due to very high anisotropy of the assembly.

The spectra in Fig. 3 also needs some explanation. As one of three or any combination of three external influences are acting on the rotor, they influence static stability of the journal (unloading). The journal, in a search of new stable position, is moving until (in elliptical bearing) forms a second, top wedge. The top oil wedge force pushes on the rotor downward, and the cycle continues. This cycling is usually at a broad band subsynchronous frequency, without defined peak.

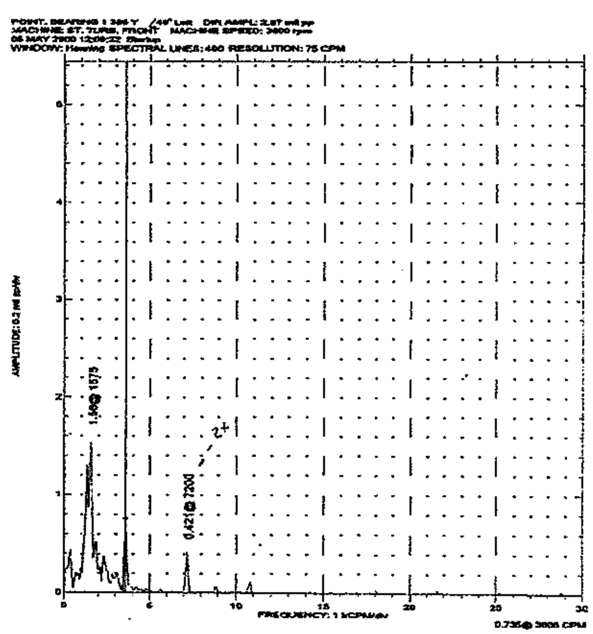


Figure 3. Typical Asynchronous Frequency Spectra of Shaft Running in Loose Bearing

Most of bearing computer programs automatically assume that the bearing shell is rigid when calculating bearing coefficients, and x or y direction stiffness is calculated based on the bearing profile and oil viscosity. The coefficients thus obtained are representing bearing characteristics with rigidly mounted bearing shell. When looseness is introduced, behavior of the same bearing becomes totally unpredictable.

An attempt to balance the rotor running in a loose bearing very often was unsuccessful. The reason is that in a "loose" bearing there are practically two sets of dynamic stiffness, sometimes requiring contradictory balance solutions. What that means is that the dynamic vector lag angle changes greatly, sometimes even over 90° in single revolution! The biggest culprit is collecting influence coefficients using horizontal (soft direction) readings. Since displacement is in phase with force acting against very soft spring, and with practically no damping, the response from trial weights always acts in the direction of force.

This new vector with the original vector gives a summation angle change, which never coincides with the angle of the trial weight move. Many balancers went full 360° of placing trial weights, and the solution was always 180° from it. Unless the correction weight was placed by knowledge or by chance within (5° opposite the actual unbalance, balancing is usually unsuccessful.

CASE HISTORY

With the advancements in vibration monitoring instrumentation, by monitoring shaft relative displacement, bearing shell and bearing structure as well as combined shaft absolute displacement, much more becomes "visible" in dynamic behavior of rotors.

Once prepared with the understanding of the effects of bearing profile, bearing assembly and rotor assembly, one can begin the analysis and interpretation of various plot presentations obtained and created from vibration sensors.

Unit Start-up

The case history is of a steam turbine-generator which underwent an upgrade from 35MW to 50MW. This entailed replacing of the turbine rotor and the balance of the steam path. Turbine bearings looked in reasonable good condition, and they were re-used. During the first turbine roll to speed a problem was encountered with the Unit tripping after reaching >4.0 mils p-p even at several hundred RPM. No attention was paid to data at this time, because of the rush to bring the Unit back on line. In one attempt the Unit was brought to speed, and oversped, but the turbine, already on a subsequent start, experienced a rub before reaching rated speed.

All these events could have been avoided if the vibration data had been scrutinized from the beginning.

This elongated orbit (Fig. 4) with horizontally oriented major axis of almost 0.006 in. p-p is a pure example of high shaft swings and a loose bearing.

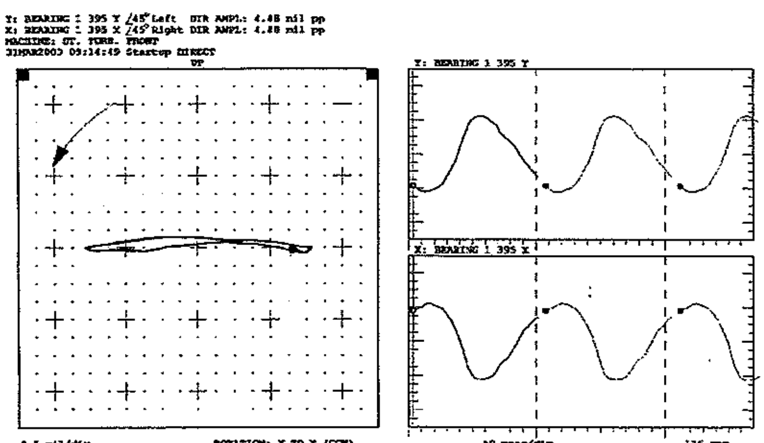


Figure 4. Journal #1 Orbit at 116 RPM

The journal rise (Fig.5) to 0.003 in. is the effect of jacking oil, then journal centerline moves almost 0.005 in. to the right due to the rotor's search for a stable position, while the loose bearing was "rolling".

Slow roll runout at start-up (Fig. 6) is about 0.001 in.,

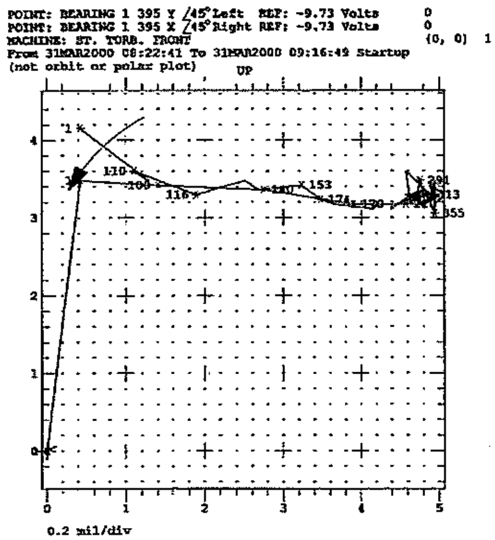


Figure 5. Journal #1– Centerline Plot to 355 RPM

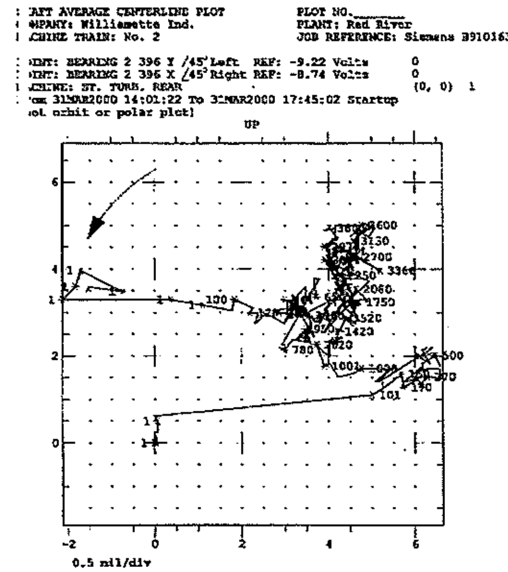


Figure 8. Journal #2 Centerline Change

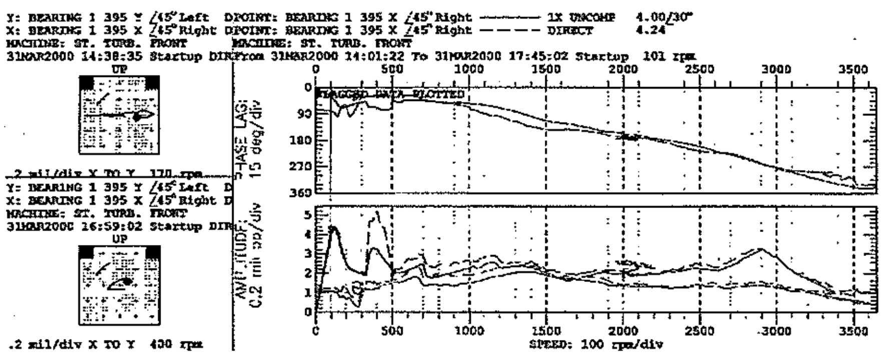


Figure 6. Bode Plot

but reaches over 0.004 in. at ~120 RPM during shut down.

From the start-up to shut down (Fig. 7) the journal centerline shifted over 0.008 in., indicating bearing looseness.

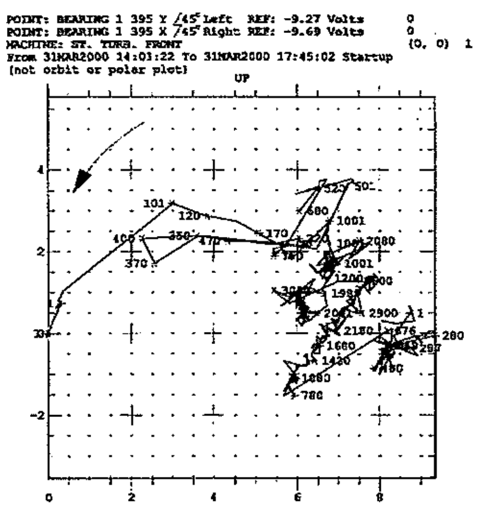


Figure 7. Journal #1 Centerline Change

By completing the picture with journal #2 centerline change (Fig. 8), it becomes clear that the entire rotor was shifted right, most likely due to misalignment at the turbine generator coupling.

This attempt to roll ended with the rotor rub at about 3200 RPM. Although overall vibration amplitudes were not high, it can be deduced that the rotor had shifted, because it could not be "retained" by the loose bearing. The orbit at the key location of the Bode Plot (Fig. 9) shows the moment of rub initiation through the partial shut down.

The plot pattern (Fig. 10)is almost identical to Fig. 9, and orbits indicate that rub occurred closer to bearing #2.

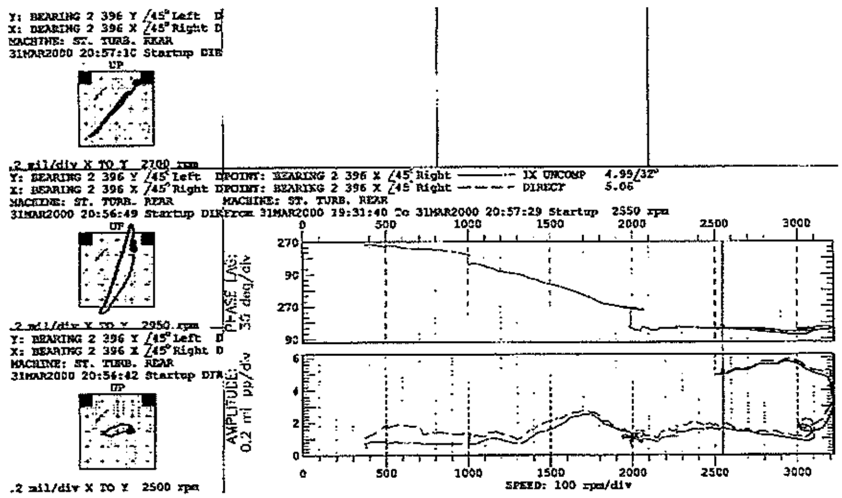


Figure 9. Bode Plot

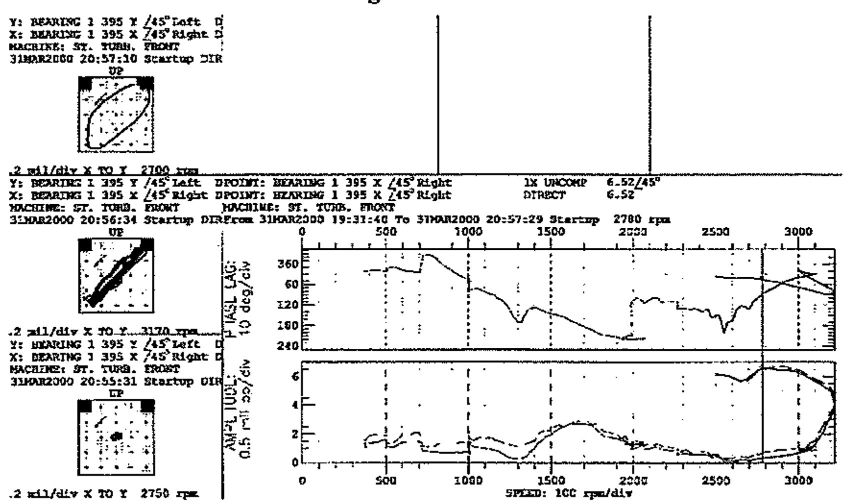


Figure 10. Bode Plot– Journal #2

Again, this plot (Fig. 11) also shows large shift of the journal centerline due to bearing looseness, and bearing large internal clearances.

Alignment Verification & Bearing #1 and #2 "Pinch"

From this plot (Fig. 12) it can be seen that the rotor "swing" at ~120 RPM is still there, but smaller and the rest of the roll up and coast down seems uneventful.

The Journal centerline still shifts, but only initially, then it stays primarily on the one side (Fig. 13). This is now the effect of shaft alignment and bearing internal clearances only.

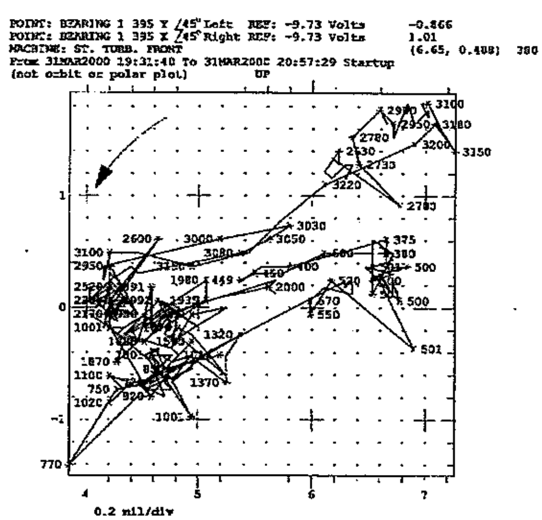


Figure 11. Journal #1– Centerline

Figure 14 shows the same data as above (Fig.13) only in the frequency domain. The selected orbit at ~120 RPM still shows the effect of rotor "swing" within large bearing ID clearances.

At rated speed the spectra show the low frequency instability (Fig. 15), but the overall amplitudes are too small to require any action.

The centerline again seems very unstable (Fig. 16), during run up and low load indicating that bearing has very little control over journal forced motion.

This plot (Fig. 17) shows clearly the effect of the main steam flow change on rotor position due to change in extraction steam flow, while speed and generator load remains constant.

The same steam action has different effect on outboard journal (Fig.18) than on inboard journal. This leads one to believe that the outboard journal gets easily unloaded by steam and when its bearing ID clearances are larger.

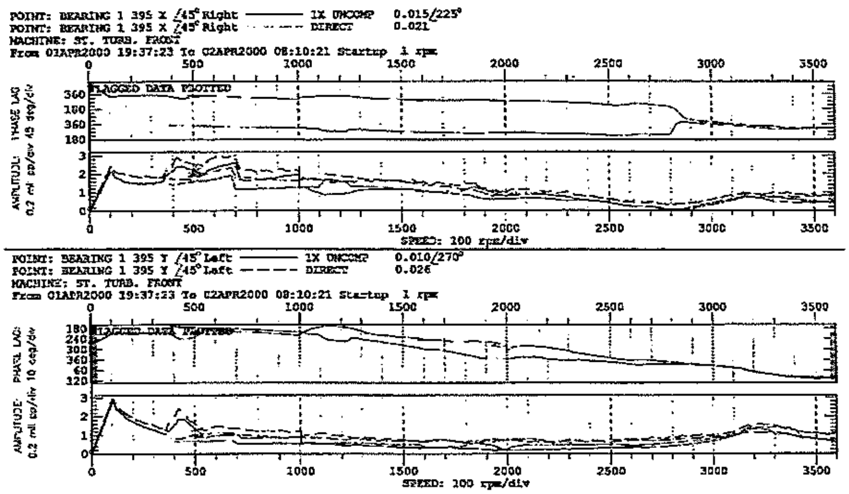


Figure 12. Bode Plot– Journal #1 & #2

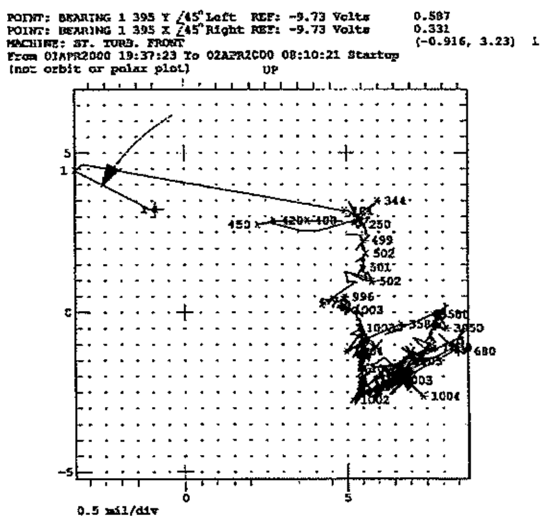


Figure 13. Journal #1– Centerline

When the Unit was loaded, the subsynchronous vibration became more prevalent. That can easily be seen in Figs. 19 and 20. although the overall amplitudes are still not large, the ratio of subsynchronous to synchronous grew considerably with higher load, and they grow even more with higher steam extraction.

The change in orbit location and the change in the major axis indicate unrestrained journal motion. Bearing clearances must be reduced in an attempt to put more load on the bearing (Figs. 21, 22 and 23).

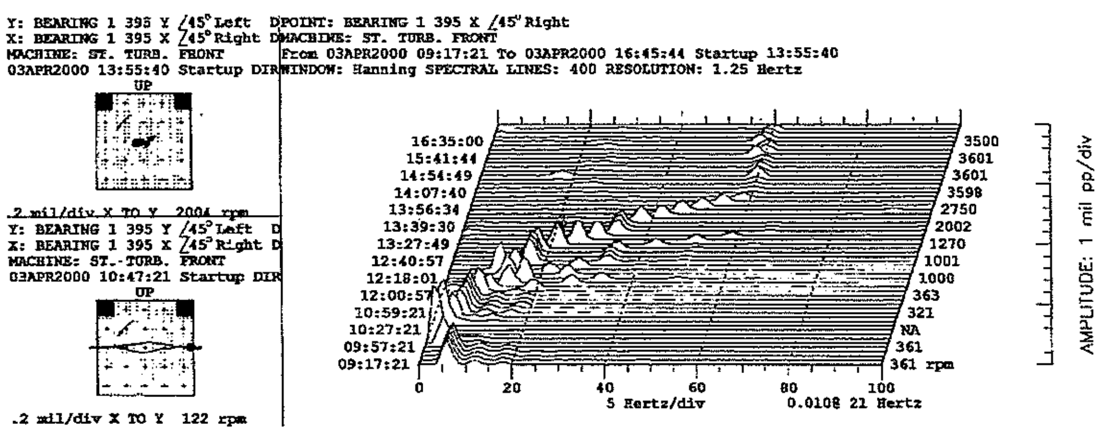


Figure 14. Waterfall– Journal #1

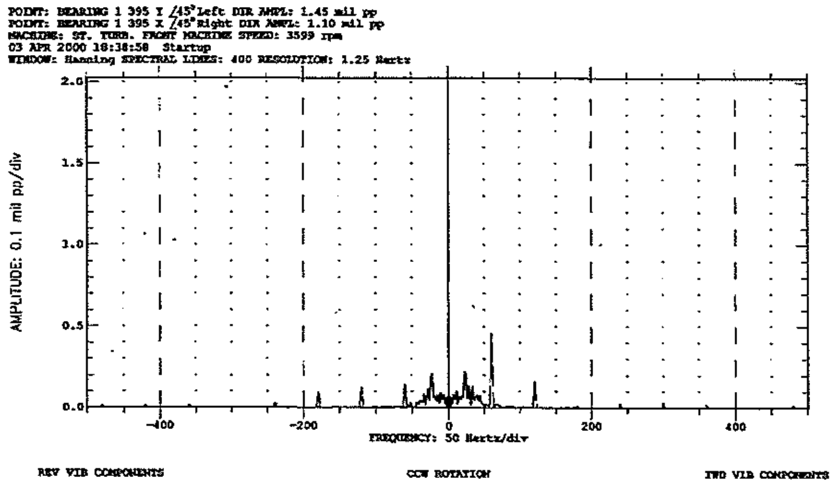


Figure 15. Frequency Spectra– Journal #1

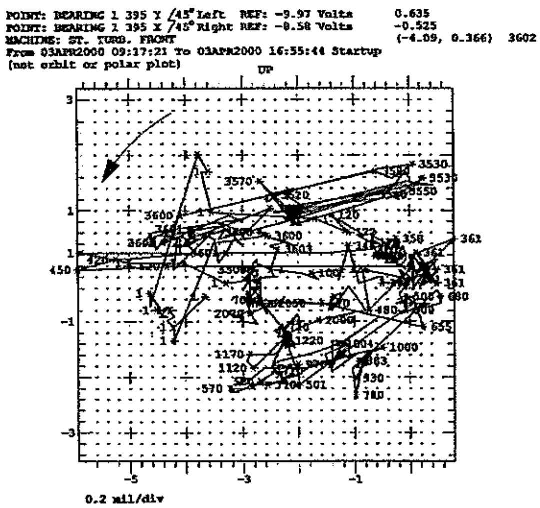


Figure 16. Journal #1 Centerline/Start-ups

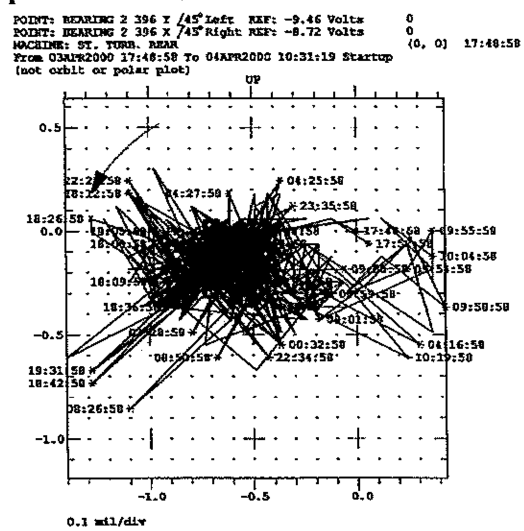


Figure 18. Journal #2 Centerline/Load and Extraction

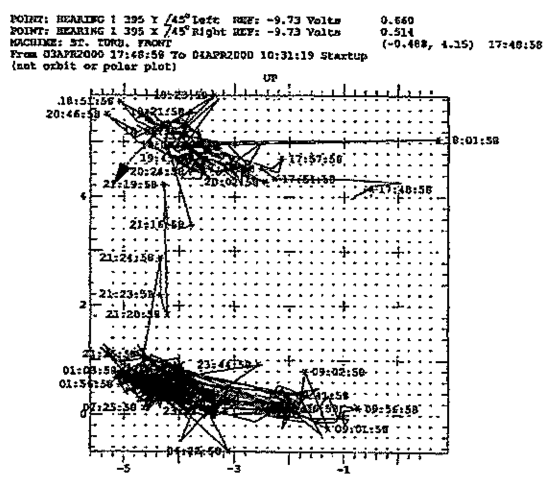
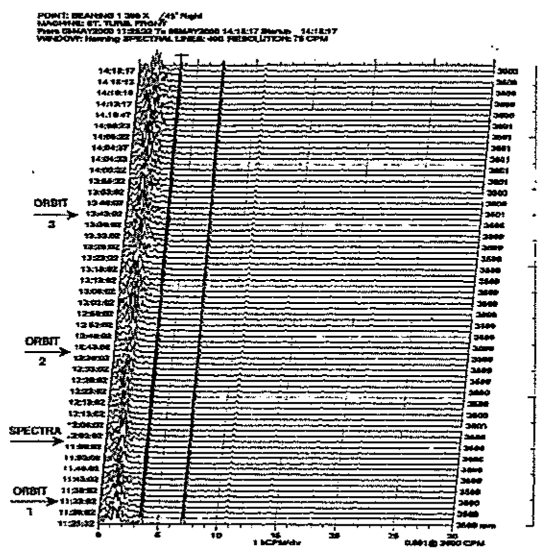


Figure 17. Journal #1 Centerline/Load and Extraction

Bearing ID Clearances and Dynamic Stiffness

In order to minimize or even eliminate residual subsynchronous vibration (whirling), several bearing coefficients calculations were performed to find the optimum bearing clearance. Most of the current bearing computer codes calculate stability threshold and predict whirling based on calculations of the critical journal mass. Theoretically, on elliptical bearings, for instance, the bearing is stable as long as there are no external disturbances present. (Fig. 24 & 25).



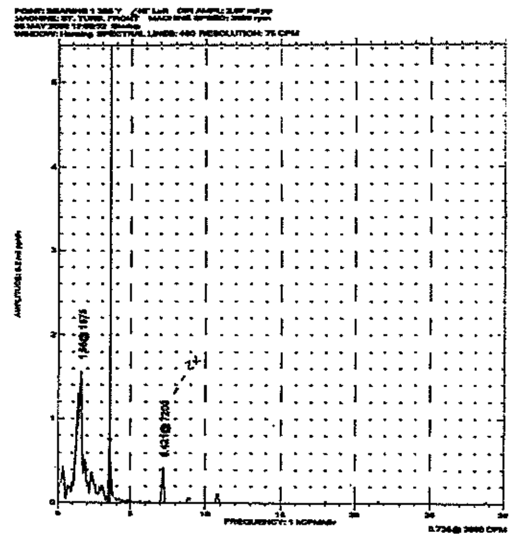


Figure 19, 20. Waterfall– Journal #1

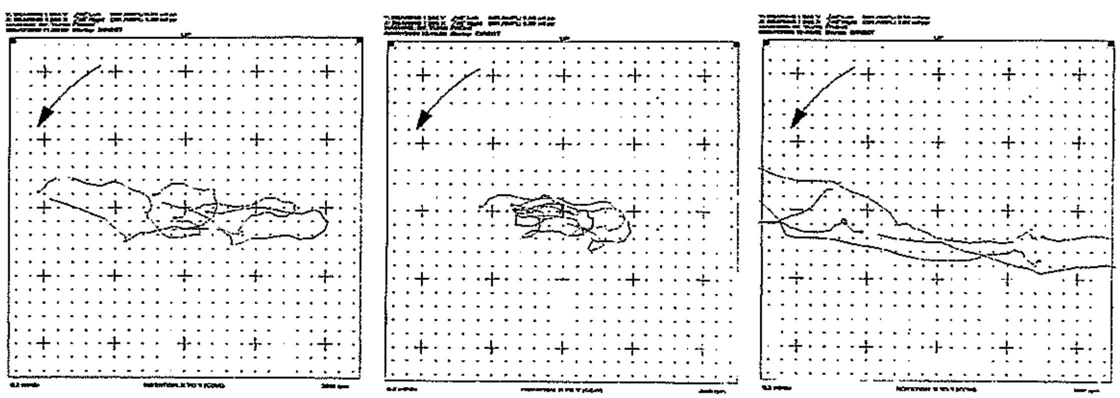


Figure 21, 22, 23. Orbits Journal #1

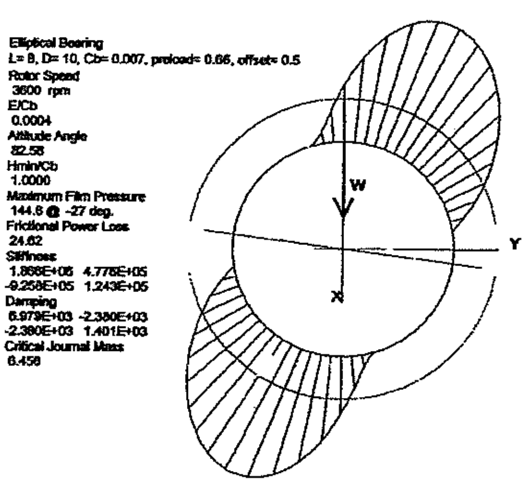


Figure 24.

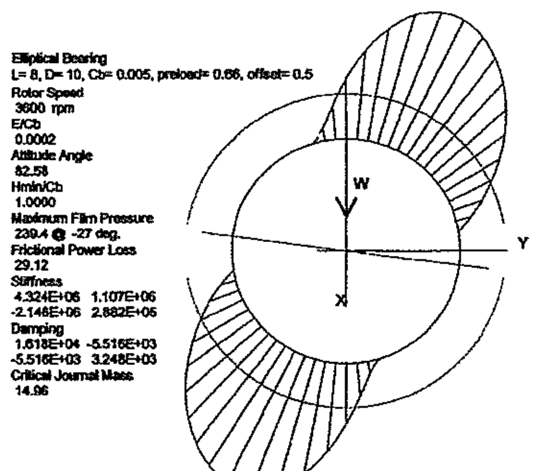


Figure 25.

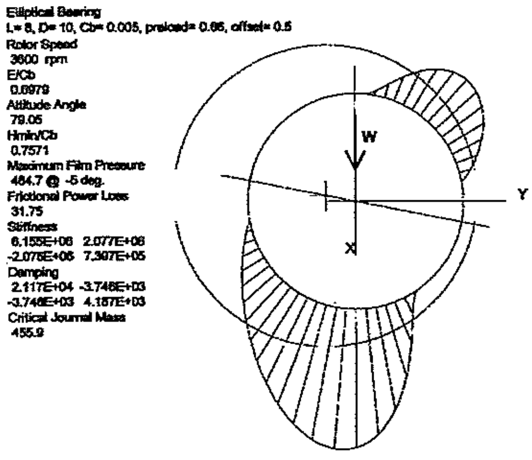


Figure 26.

In practice, when the external disturbances "unload" the bearing, it becomes exhibited by the journal centerline shift and by its unstable orbital motion.

While direct journal loading and direct bearing stiffness control the journal static position, the only bearing characteristic which can control journal dynamic displacement is bearing dynamic stiffness. Dynamic stiffness, by definition is the ability of bearing oil film to suppress journal displacement, which depending on rotor ratio of critical speed to its nominal speed (c_r/c_n), may lag centrifugal force from unbalance from 0 to 180°. Therefore, the solution to the problem of this case was to restore journal load and increase bearing dynamic stiffness, by first eliminating bearing assembly looseness and secondly closing bearing vertical clearance.

The stiffness and damping characteristics of installed bearings are mathematically described by the folowing equations; from which, one can intuitively see that bearing stiffness and damping can be increased by reducing clearances:

$$K_{BD}= \frac{n\Omega dl^3}{c^3} \frac{K_I e}{(1-e^2)^{5/2}}, (N/m,\ lb/in) \qquad (1)^*$$

$$D_{BD}= \frac{ndl^3}{c^3} \frac{D_I}{(1-e^2)^{3/2}}, \left(\frac{N{\cdot}sec}{m}, \frac{lb{\cdot}sec}{in}\right) \qquad (2)^*$$

Where:

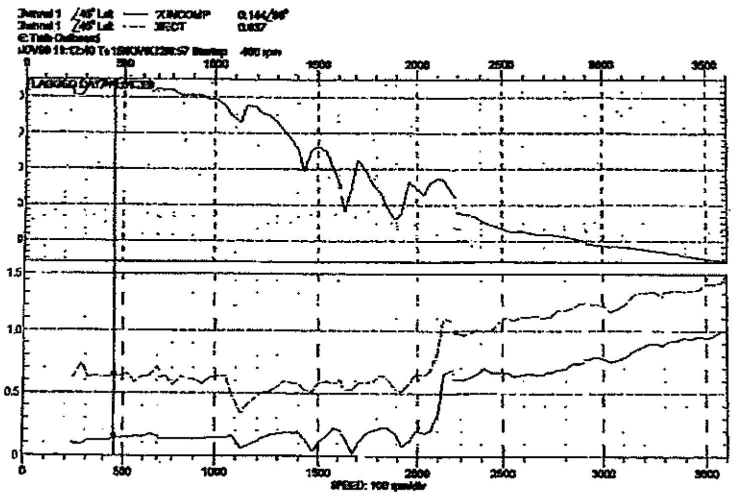
K_{BD}, D_{BD}	are radial stiffness and damping of the hydrodynamic bearing, respectively
K_I, D_I	are constants
d, c	are journal diameter and diametral clearance, respectively
l	is bearing length
n	is dynamic viscosity
e	is eccentricity ratio
Ω	is rotative speed

(Reference *: "ORBIT" First Quarter 2001, Volume 22, Number 1.)

Fig. 24 represents bearing clearances as found, while bearing shell was also loose. Pinching of the bearing showed some improvement, but did not eliminate whirling.

Fig. 25 shows the predicted result after closing vertical clearances and increasing bearing pad preload.

Fig. 26 shows the journal behavior in bearing with increased dynamic stiffness. No balancing or any change in external disturbances reduction was made, therefore, the whole bearing characteristics improvement can be attributed to increased bearing dynamic stiffness.



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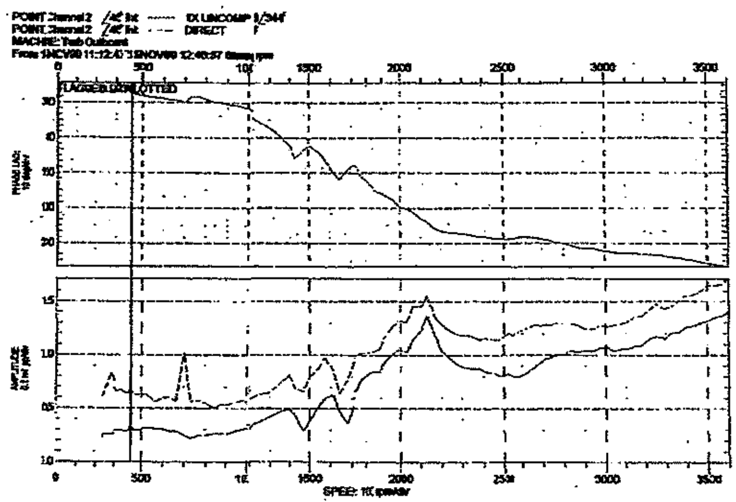


Figure 28.

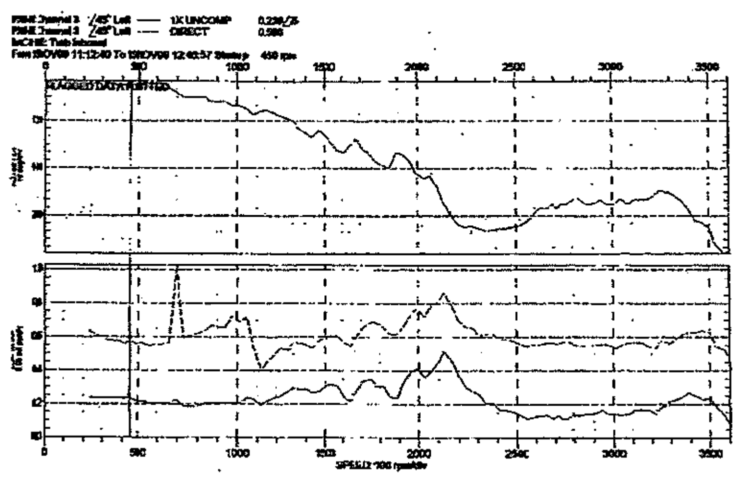


Figure 29.

CONCLUSION

Upon completion of the repair work, the turbine was restarted successfully.

Bode plots (Fig. 27-30) from the start-up show low speed runout very low and passing through critical speed uneventful.

Shaft centerline plots (Fig. 31, 32) show the turbine start-up and loading for journals #1 and #2 respectively. It is clear that the journals are now stabilized.

Plot 33 shows average spectra. The size of subsynchronous whirling is practically nil.

This case history shows how the real problem can be solved by application of modern vibration instrumentation. By understanding the rotor behavior through various graphical data displays and utilizing to the maximum currently limited ways of changing bearings behavior.

Orbit plots, Figs. 34, 35 and 36, show very stable orbits with small amplitudes through the full turbine load range.

ACKNOWLEDGEMENT

It is difficult to mention everyone who helped me over the years to grow and learn the intricacies of the work in the field of rotor dynamics, vibration instrumentation use and data analysis, as well as theory and practical application. Great thanks go to Dr. Donald Bently, Dr. Ed Gunter and Dr. Ron Eshelman, as well as other great authors of books in Rotordynamics too numerous to mention. Many thanks to my associates Ivo Dabelic and James Tinklepaugh for many hours spent in the field to collect data and assist in solving the rotordynamic problems.

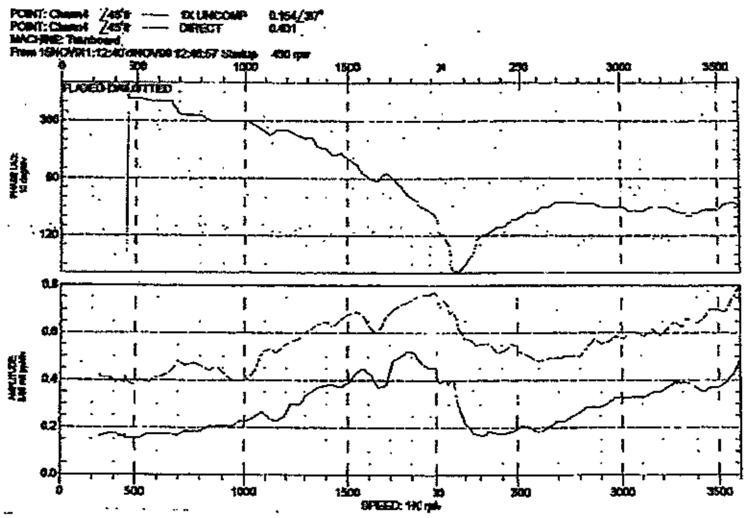


Figure 30.

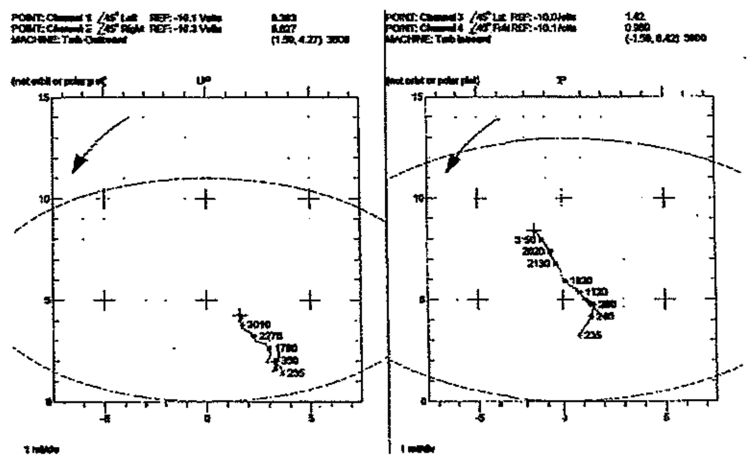
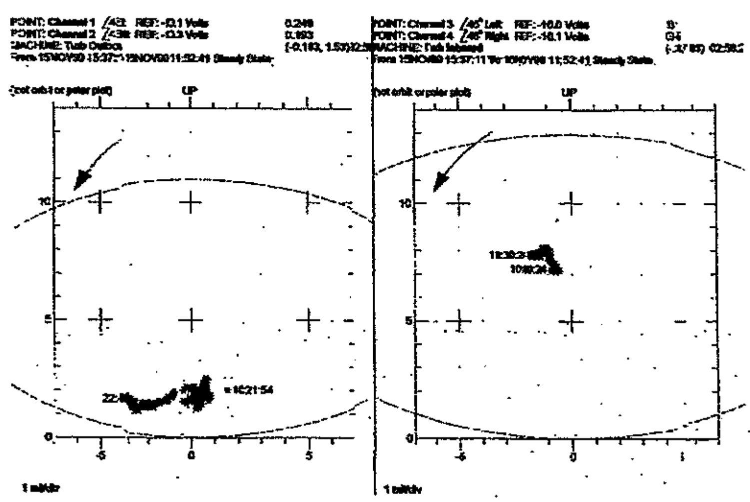


Figure 31.



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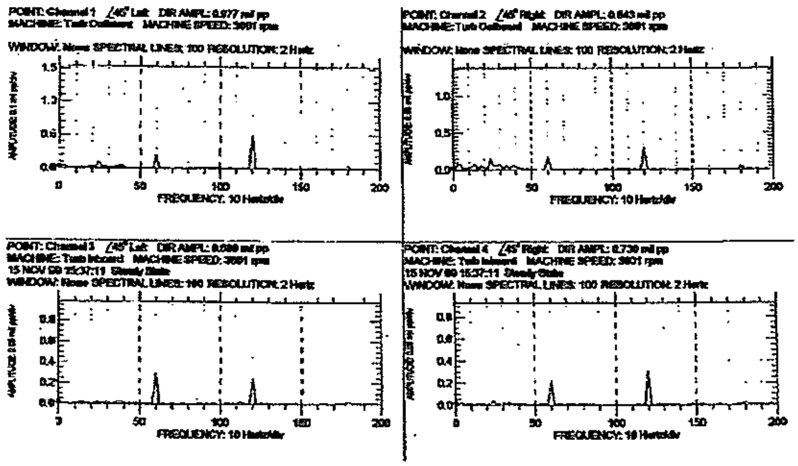

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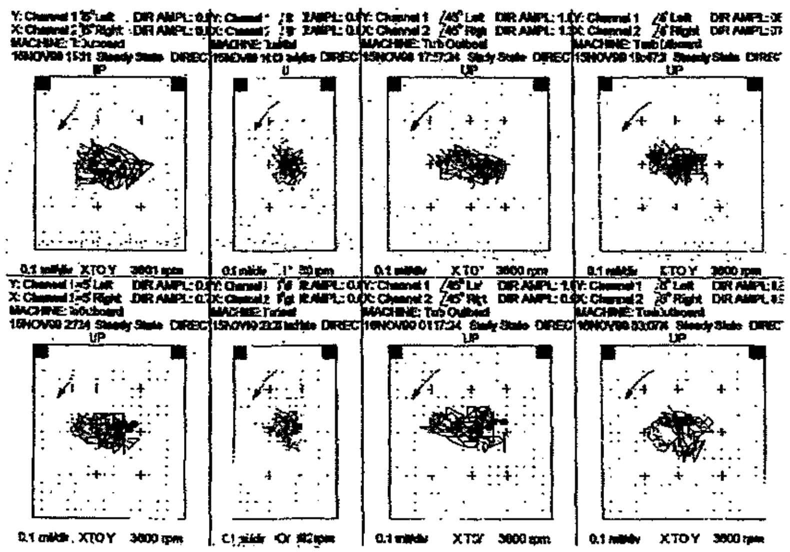

Figure 34.

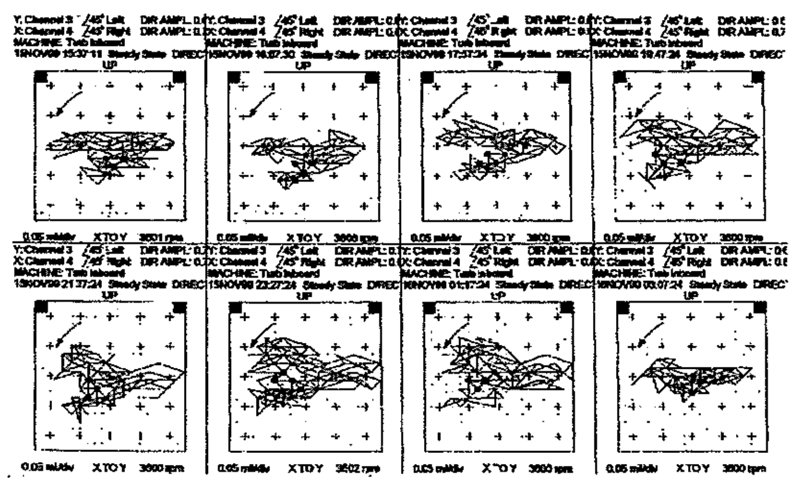

Figure 35.