# PASSING THROUGH A CRITICAL AT CONSTANT SPEED?

Zlatan Racic
Supervisor Operating Performance
Siemens Power Corporation
Bradenton, Florida

Born in Croatia and living in the United States since 1970. Mr. Racic graduated from MSOE in Milwaukee, Wisconsin with a BSMET and with a MBA degree from Nova University in Fort Lauderdale, Florida. His long career started as a merchant marine engineer, and currently he is employed at Siemens Power Corporation. In his line of work he has resolved numerous rotordynamic and vibration problems on practically any OEM rotating equipment.

## INTRODUCTION

This case history on a machine located in the Far East deals with a vibration problem on a combustion turbine driving an ~100 MVA generator. The problem observed was a "thermal vector"(at the CT exhaust end) which was changing with change in load. Several consultants had tried to correct the condition by balancing, but all they were getting was a shifting of the beginning and the end point of the "thermal vector".

## THE APPROACH

A first look at the Polar plot showing the run-up and loading to ~70 MW revealed a very low, 1st rigid mode critical at ~1100 RPM, a more pronounced 2nd rigid mode critical at ~2070 RPM and very good balance condition at operating speed of 3000 RPM (Fig.1). When the generator was synchronized, and loaded, the vibration amplitude and phase continued to change in a form very much resembling a "resonance". The only difference was that this was happening at a constant speed (Fig. 1,3).

When the Unit was partly unloaded, the vibration vector followed the load-up curve up to a point of trip. During deceleration, the balance condition changed, affecting also the 2nd rigid mode response at the turbine exhaust end (Fig. 2,3).

Subsequent runs confirmed the repeatability of the data.

With the assumption that if it looks like a critical and smells like a critical, it must be a critical, a trial weight was placed in the middle span of the turbine rotor at the angle based on critical speed location estimated from the polar plot (Fig. 1).

The result was amazing. (Fig.4,6). Not only the "loop" was gone, but the total length of the "thermal vector" had shrunk! A subsequent shut down showed also a reduced "critical at running speed", a further small improvement of 2nd rigid critical and slightly worsening and angular shift of the 1st rigid critical.

## ANALYSIS

When dealing with a "clean" thermal vector, the resultant seen on the polar plot is the vectorial sum of the residual unbalance vector and the changing thermal vector.

Balancing above cannot influence the total "vector spread" but only shift its location (Fig.7). But in this case balancing has reduced the total "vector spread", i.e., balancing had reduced the resultant amplitudes over a range of ~180°, at a constant speed, similar to the effect of reducing a typical resonance response.

Because the balance weights were placed at 210°. (90° behind the estimated peak of the displacement vector), the balance response seems to be like correcting a resonant response amplitude.

## CONCLUSION

The polar plot (Fig.1) shows the start-up curve before balancing and the rather odd vector change during the increase of load, until achieving steady state condition at 70 MW. It is almost a circular course from .27 in/s pk at 200° to .17 in/s pk at 340°. The initial attempts in balancing the end point of the steady state did not yield favorable results during load transient. The reason behind it is that any correction weight which would reduce vibrations at steady state will presumably deteriorate the transient vibration according to the laws of "superimposition".

When unloading the turbine, the vector did not follow back the path completely. The state of balance of the turbine, therefore, differs from the initial one. This fact explains sufficiently the observation that the vibration in the critical speed range of the 2nd rigid mode differs from that of the start-up.

Leaving out the above mentioned strange (circular) course of the vector change, the phenomenon, up to now, is understandable. There is a residual unbalance which causes vibrations in the critical speed range when starting up, there is a transient vector change when heating up the turbine shaft which goes temporarily in direction 180° and comes back after a while and there is a repeatable vector change due to load temperature which follows these parameters (see the difference between the 70 MW point and the TRIP point). The "superimposition" of these three components results in the total vector change depicted in Figure 8.

A correction weight at 210° ("at the high point of the velocity vector") was installed. Obviously, the vibrations in the critical speed range have been reduced by this shot for the start-up as well as for the shutdown. That is understandable since the vectors of both, at start-up and shutdown had been in the same quadrant before balancing.

Now, something happened, the physical explanation of which requires more sophisticated theories: The vector spread between the transient maximum point and the steady state point at 70 MW has shrunk by the installation of weights. As mentioned above, this could not have occurred as long as the simple laws of linear superimposition of the single components (independent from each other) apply.

When attempting balancing transient vibration ("Thermal vector") at constant speed, two cases are observed:

I. The transient maximum is reduced, the vector change remaining the same is only parallel shifted.

II. The transient maximum is reduced, the vector change has become smaller or almost disappears.

Case I. follows the simple laws of superimposition. Case II. can only be explained if more complex physical laws apply. One of them is the non-linear spring constant of the supporting system.

The critical speed in the non-linear system varies with the magnitude of vibration. The case is imaginable that a system is more sensitive to unbalance with higher vibrations than with smaller vibrations. The change of the state of balance, due to heating up the turbine shaft, may always be the same. But if the transient maximum can be reduced by field balancing, the sensitivity to unbalance can decrease and the turbine answers to the same change of the state of balance with a smaller vector spread. The phenomenon can be combined with the changes of the phase angle. Non-linear spring characteristic can occur in real life. Each joint between 2 machine components is a potential cause of non-linearity. Many parts of the machine are fitted with clearances, e.g., bearing shells in bearing casings.

The other possibility of Case II. is a simple change in system stiffness influenced by the system heating.

As the machine is heated up, the system support becomes "softer" and the critical speed migrates to the left (Fig.9).

As depicted in Figure 9, after the unit reaches operating speed, it operates with the amplitude of vibration on the left slope of the solid line "critical" at corresponding phase.

As often in our lives we do something because it works, even if we have no complete explanation why it is so. If more similar cases are studied in the future, hopefully we will be able to find the answer.

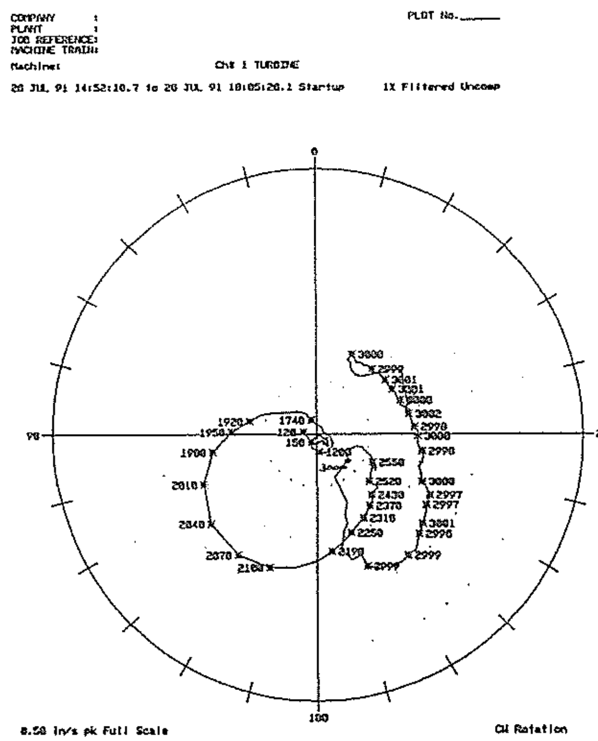


Fig. 1 Polar Plot of the Run before Balancing to Rated Speed, and through to Rated Load.

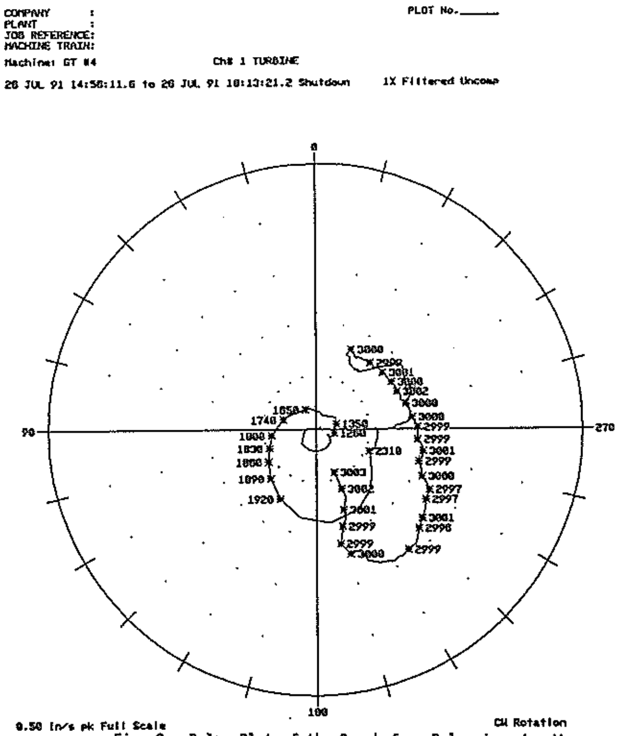


Fig. 2 Polar Plot of the Run before Balancing, Loading Unloading, Trip and Shut Down

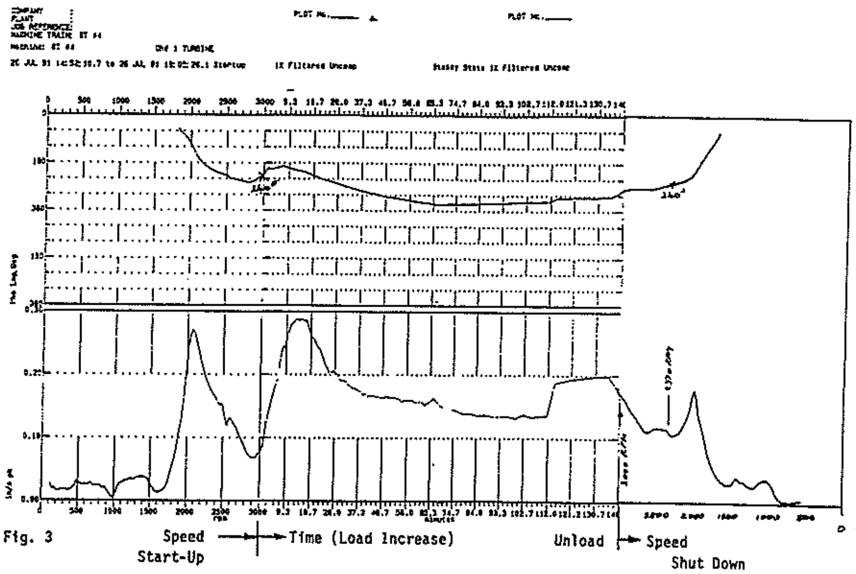


Fig. 3

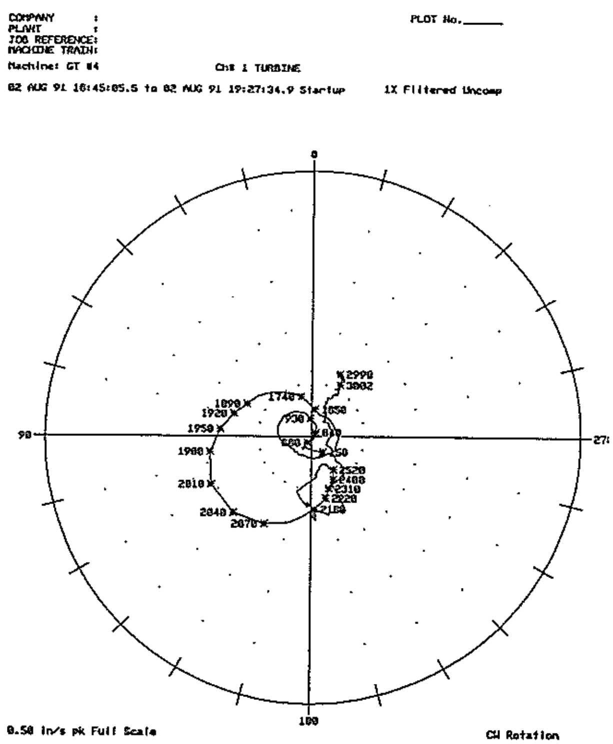


Fig. 4 Polar Plot after Balancing "The Critical", Start-up and Loading.

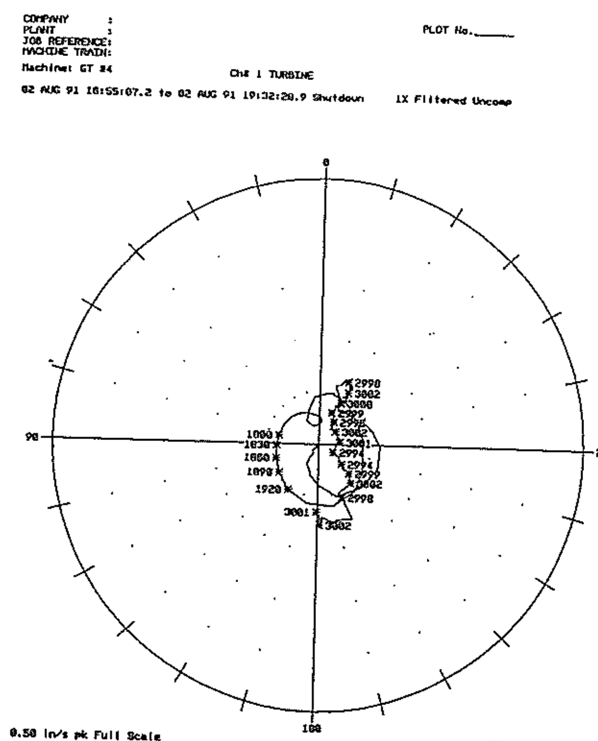


Fig. 5 Polar Plot after Balancing "Critical" Loading, Unloading and Shut Down.