The 8th IFToMM International Conference on Rotor Dynamics
September 12-15, 2010 / KIST, Seoul, Korea

ROTOR DYNAMIC MODELLING AS A POWERFUL SUPPORT TOOL FOR VIBRATION ANALYSIS ON LARGE TURBOMACHINERY

Kris Matthys
Laborelec
Linkebeek, Belgium
kris.matthys@laborelec.com

Marco Perucchi
DELTA JS AG
Zürich, Switzerland
mperucchi@delta-js.ch

Koenraad De Bauw
Laborelec
Linkebeek, Belgium
koenraad.debauw@laborelec.com

ABSTRACT

Laborelec is responsible for the follow-up of the vibration behaviour of a fleet of more than 100 shaft lines within the power generation division of GDF-Suez. A parallel monitoring of a large number of units permits a good comparison of the vibration behaviour of units of a similar type.

In recent years, Laborelec started to construct rotor dynamic models of shaft line designs that are used on a larger scale in the monitored fleet. The development was made in collaboration with the company DELTA JS, using their commercial rotordynamic software MADYN 2000 [1]. The goal was to have a better understanding of the dynamic behaviour of the units, as well as to have a powerful support tool available for the analysis of occurring vibration problems.

This paper describes some first experiences with the use of a rotor dynamic model for analysis purposes. A first case study describes the added value of the model for defining a fast and accurate balancing correction at the shaft end of a generator, which was then actually implemented in the power plant. A second case study describes the important role of the rotor dynamic model in the root cause analysis of vibration problems related to misalignment on a complex shaft line design. The model permitted to identify the critical speeds and bearings that have an important influence on the vibration behaviour.

INTRODUCTION

The role of efficient vibration monitoring in predictive maintenance on large turbogroups

As a subsidiary of the GDF-Suez group, Laborelec is a major partner of the GDF-Suez power generation division for different condition monitoring activities, of which vibration monitoring is a very important one. Laborelec vibration specialists have an on-line access to more than 80 local vibration monitoring systems, monitoring more than 120 shaft lines. The Laborelec Vibration Monitoring System (LVMS) was developed to assist the specialists for combining different essential tasks in this permanent monitoring activity: vibration measurements, visualisation of the measurements, alarming, data storage and analysis. One of the most important features is an easy remote data access, permitting an on-line expert's advice in case of important changes of vibration behaviour. The power plants can consult Laborelec in a 24/7 permanency for an immediate advice.

The wide range implementation of a similar monitoring system on a large diversity of machine types has a high added value in the troubleshooting activity. Vibration problems that are known for a certain machine design, or less common problems, can often reoccur after several years of trouble-free operation. The availability of historical data in a similar data plot format is an enormous advantage for the vibration expert. At present, Laborelec disposes for some of the units of historical data in a period of more than 15 years, in a compatible data format.

Even if vibration monitoring is certainly one of the most important tools for troubleshooting of the mechanical malfunctions of rotating machinery, it has its limits. Other techniques are often used in parallel, such as the analysis of selected operational parameters, the measurement of stator part displacements for assessing alignment variations, or impact testing for examining resonance conditions of stator parts or the foundation. In specific cases, also on-line torsional vibration monitoring is used.

The use of rotor dynamic analysis in the troubleshooting activity

For more complex root cause analyses, the deployment of a variety of measurements can give many answers, but also brings in additional questions. In recent years, Laborelec has collaborated on several occasions with the Swiss company DELTA JS for a parallel rotor dynamic analysis of known shaft lines in the context of a root cause analysis of vibration or bearing problems. The experience of DELTA JS with the analysis of a very wide range of rotating machinery enables a fast modelling, even with a minimal input of information. Even a coarse model of a shaft line is often already capable of revealing specific features of the vibration behaviour, with an important added value in the interpretation of findings of a measurement campaign.

Based on the positive results of these collaborations, Laborelec has started to make models of shaft lines that are used on a larger scale within the GDF-Suez fleet. The goal is to construct and validate a small library of rotor models that can be used quickly for an urgent need in vibration analysis. The models are made using the MADYN 2000 software of DELTA JS.

This paper describes some first experiences with the use of rotor dynamic modelling for analysis purposes.

CASE 1: MODAL ON-SITE BALANCING OF A GENERATOR SHAFT END

Introduction

This first case deals with a 50Hz air cooled generator rotor, directly driven by a gas turbine through an intermediate shaft. The generator had a major incident due to a cooling fan blade loss at nominal speed. This caused a significant damage to both rotor and stator and an unforeseen overhaul to the complete generator.

It was decided not to balance the rotor after the necessary repairs, due to a lack of time and the fact that the rotor repair was rather limited in comparison to the stator repair. Therefore the generator was mounted and coupled again to the gas turbine without doing an unbalance check, nor at low speed, nor at high speed. However, a weight chart was made for the mounting of the new fan blades to avoid unbalance problems.

During several attempts to start up the unit one didn't succeed to reach nominal speed due to automatic shutdowns at ±2980rpm caused by high relative shaft vibrations (>200 μm_{pp}) at the generator exciter end bearing (Figure 1).

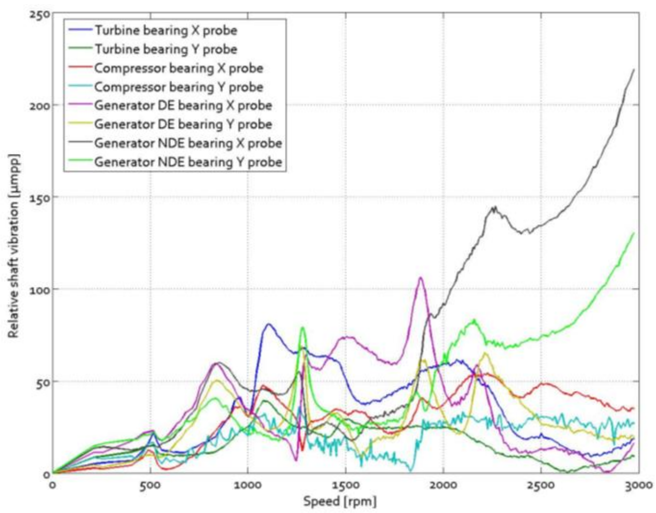

Figure 1: Relative shaft vibration amplitudes during startup

From the vibration data it seemed that the shaft line was exhibiting the influence of a lateral critical speed near 3000rpm. However, this lateral mode was never observed or measured into detail earlier, also not on similar units within the fleet. It was decided to define a balancing correction as a first urgent action. On-site balancing of a generator rotor is difficult because it is very time consuming to have access to the balancing planes at the rotor ends. Therefore, the existing rotor dynamic model of the shaft line was used to study other faster and easier possibilities to balance this rotor on-site.

Rotor dynamic model of the shaft line

The complete shaft line was modeled (Figure 2) by using the MADYN 2000 software. The model was based on scaled drawings. The four fluid film bearings were calculated with the ALP3T program [2]. The static load of the different bearings was determined by aligning the shaft only taking into account the flexibility of the different rotors, in a way that the couplings are free of bending moments. The present oil film thickness at nominal speed was not considered in this static calculation.

All bearing pedestals were modeled as pure stiffness and mass. This assumption was tested by performing impact tests on the bearing structure in both vertical and horizontal direction. No resonance frequencies were detected below 50Hz or at multiples of this frequency. Therefore it was decided to determine the static stiffness obtained at 50Hz and use this value for the complete frequency range of the different calculations.

There was only poor rotor dynamic information given by the manufacturer so it was not possible to completely verify the model; however, the calculated results fitted both the basic available rotor dynamic info and the measured vibration data quite well.

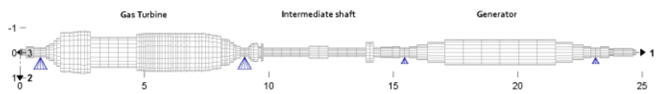

Figure 2: Rotor dynamic shaft line model

From the calculated eigenvalues at 50Hz two modes seemed to be present near the operating speed of the shaft line (Figure 3). The closest modes influencing the dynamic behaviour are at respectively 51.9Hz and 53.3Hz. The first eigenmode is the second vertical bending mode of the gas turbine and is unlikely to be causing high vibrations near the generator shaft end. The second eigenmode is a horizontal bending mode of the shaft end. From the eigenvalue analysis it is clear that the damping factor is poor and the mode deformation is almost completely planar (whirling factor of 0). This mode shape is shown in Figure 4 and shows clearly the planar deformation near the shaft end.

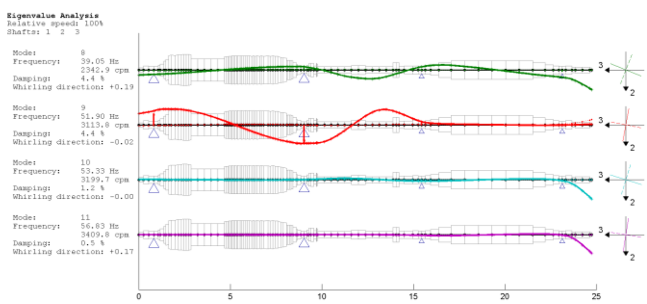

Figure 3: Eigenvalue analysis of the shaft line

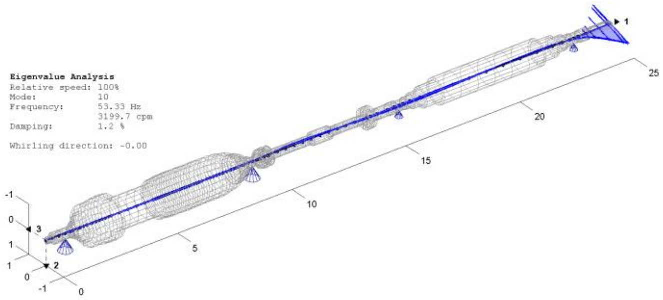

Figure 4: 3D mode shape of the bending critical of the shaft end

Based on the measured direct orbit shapes it was possible to conclude that it was this shaft end mode that was responsible for the high shaft vibrations causing the automatic shutdown of the unit during startup (Figure 5). One can clearly see that there is a local horizontal deformation at the generator NDE measuring plane near nominal speed, but for all the other measuring planes the relative shaft vibration amplitudes remain rather low.

From these results it was also clear that the "usual" balancing planes at both ends of the generator rotor were not recommended to balance this present unbalance. However, from local inspections of the shaft end it became clear that it would be possible to do a balancing test run with a mass connected at the end of the shaft line on a rather non-conformistic plane (Figure 6). This plane would be the best location because the mode deformation is the highest at the shaft end. This location would also make the balancing plane easily accessible for further adjustment of the mass.

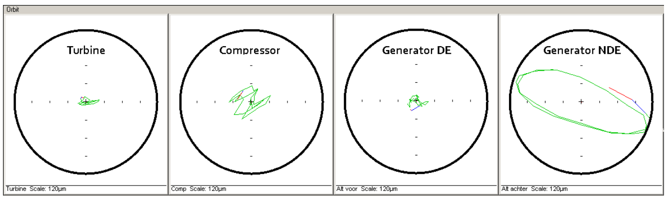

Figure 5: Direct orbit plots at 2980rpm

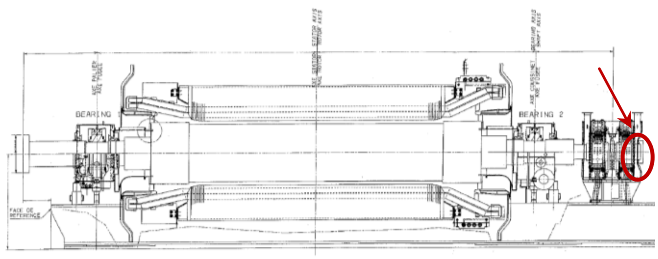

Figure 6: Chosen balancing plane of the generator shaft end

An unbalance calculation was done in order to have an idea of the expected response of the generator. The used weight for this calculation was determined by applying a G1 unbalance based on the weight of this free shaft end (Figure 7). This unbalance response shows that there is a certain influence at 50Hz due to this added unbalance, however to balance the shaft end, a correction mass of about 15x times higher would be needed to bring down the actual shaft vibration amplitudes to acceptable levels.

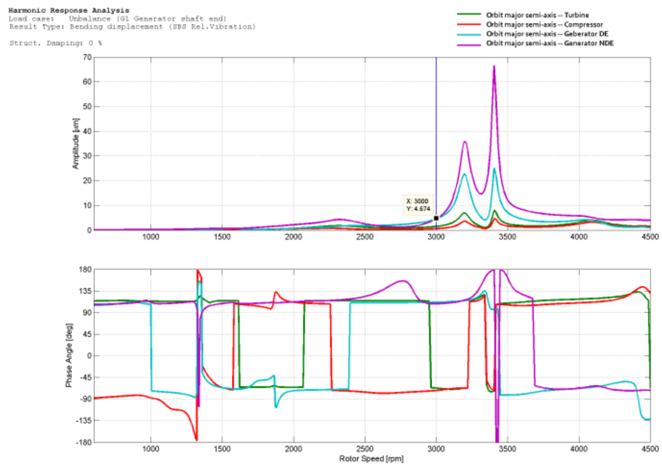

Figure 7: Unbalance response of a G1 unbalance of the shaft end

Balancing operation and result

The necessary mass was added at the chosen shaft end (Figure 8) in order to determine the desired dynamic behaviour. Due to the relatively small rotor radius it was necessary to use a significant mass, otherwise the obtained influence would have been too low.

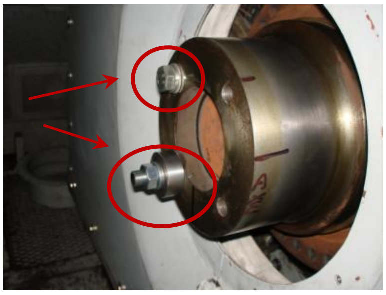

Figure 8: Picture of the used balancing plane

The resulting dynamic behaviour of the shaft line is shown in Figure 9 and shows that there is an important reduction of the relative shaft vibrations around 3000rpm. This confirms of course that this shaft end mode was the main reason for the increased shaft vibrations.

This balancing correction was done completely remotely. Only a local maintenance responsible went on-site to attach the balancing weight. This made it possible to react quickly on this vibration issue and reduced the unforeseen downtime of the unit to a minimum. The unit could be restarted the same day without any vibrations alarms and a more extensive balancing of the present residual unbalance of the rotor could be scheduled at a more appropriate moment in the maintenance planning.

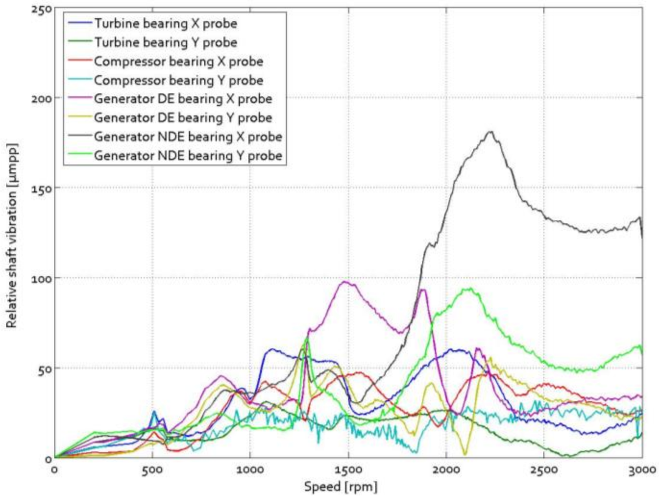

Figure 9: Relative shaft vibration amplitudes after balancing

A more detailed analysis of the vibration behaviour before and after the generator repair was made in the following days. This revealed that the influence of the unbalance distribution in the generator rotor before the repair partially compensated the impact of the shaft end unbalance on the vibration amplitude at operating speed. The modification of the generator rotor unbalance with the exchange of the fan blades created a situation where the unbalance distribution of both the rotor and the shaft end interacted in a negative way, thus causing high vibrations for the shaft end mode.

CASE 2: ALIGNMENT ISSUES ON A SINGLE SHAFT UNIT

Introduction

Misalignment is together with unbalance one of the most important malfunctions that appear within rotating machinery. Although a lot of study has been carried out on the subject of balancing or unbalance behaviour, much less information and research results exist for the interaction between (mis)alignment issues and its resulting dynamic behaviour [3,4,5,6]. This case shows the importance of rotor dynamic analysis in understanding misalignment and the effect it can have on the dynamic behaviour of a large turbo generator shaft line.

Rotor dynamic shaft line

The used rotor dynamic model is shown in Figure 10. The model was made with MADYN 2000 and includes a gas turbine coupled to a combined IP/HP turbine by means of an intermediate shaft. At the other side there is a LP turbine and the generator rotor. The location of the different fluid film bearings are indicated on the figure with blue triangles. All bearings are tilted pad bearings, except for the generator bearings, that are elliptical journal bearings. The static alignment was calculated by considering the gas turbine compressor end bearing and IP bearing at level 0. The other bearings are then lifted in such a way that all couplings have minimal bending moments. The thickness of the oil film was not considered in these static calculations.

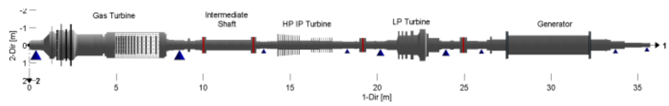

Figure 10: Rotor dynamic shaft line model

The different pedestal structures were modeled as pure mass and stiffness based on the impact tests on the bearings. For the modelling of the dynamic stiffness between the gas turbine compressor and turbine end bearing structure, a dynamic bearing support matrix was used [7,8]. Figure 11 shows the transfer function that has been used. The blue curve is the measured dynamic compliance in the vertical direction between the two gas turbine bearings, the red curve is the polynomial function that fits the measured curve. This polynomial curve is used to consider the influence of the support structure on the eigenvalues of the shaft line.

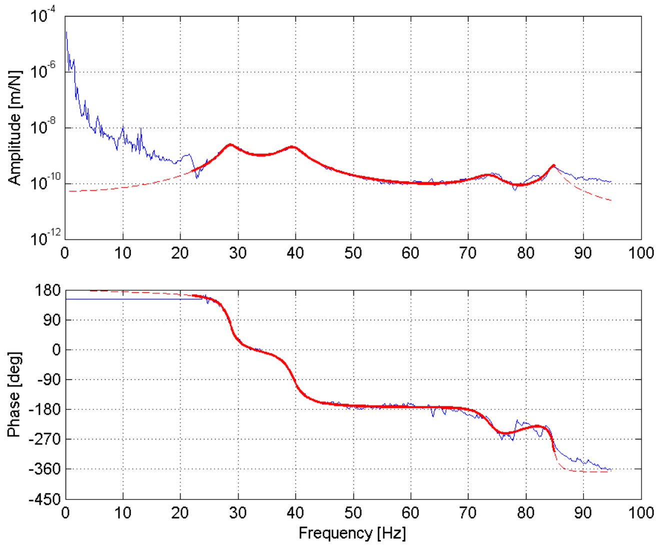

Figure 11: Dynamic bearing support transfer function

Findings by vibration monitoring

Based on the data of the LVMS vibration monitoring system installed on these type of units for several years, a lot of experience has been acquired already. One had noticed that sometimes during certain load conditions, the shaft vibration levels near the IP bearing increase slowly and attain high amplitudes (up to $250\mu m_{pp}$) near the bearings. Although the run up of the machine didn't reveal any important residual unbalance, it is only the first harmonic vibration component that changes during these situations. Once the machine is completely heated and the load of the unit is stable, the vibration amplitudes stabilize on a high level.

Figure 12 shows a typical plot for these kinds of phenomena, that have been observed on different units of the same type. It is clearly visible that all vibration amplitudes stabilize after a certain time but there is a visible influence of the load variation (active and reactive power) on the dynamic behaviour. The IP bearing vibrations attain the highest amplitudes, but the influence is also noticeable on the compressor and HP turbine bearing. The high vibration amplitudes can increase in many cases to values that are considered as unacceptable for long term operation, since they can be well above the B/C boundary of the ISO7919 standard.

Bearing temperatures are not mentioned in Figure 12, but the compressor radial bearing was at full load at 113°C and the IP bearing at 71°C. Based on these values it seems that the compressor bearing is significantly higher loaded than the IP bearing.

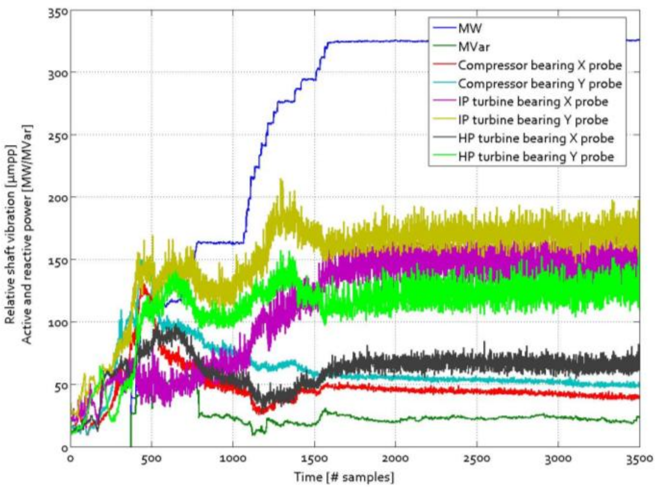

Figure 12: Relative shaft vibration amplitudes for different load conditions

Analysis of the available data did not reveal the exact cause of these increased vibration levels, yet alignment changes during the heating of the unit were suspected. However, no remarkable shaft position variation could be observed in the different bearings.

Even on-site balancing only had an influence on the dynamic behaviour during run up and run down of the shaft line. The observed vibration trend during the load increase remained present and the magnitude was the same. Balancing corrections on the intermediate shaft were however applied as a short term measure to control the vibration amplitudes in steady state operation.

Alignment measurements with a Permalign system (Pruftechnik) have been done during operation in order to measure the movement of the gas turbine and IP/HP stator at different load points. It was found that these parts show a significant relative movement. Figure 13 shows the situation for a certain load point were a relative horizontal displacement of almost 2 mm has been measured when comparing to the cold alignment conditions. This has of course an important influence on the dynamic behaviour of the shaft line near the intermediate shaft. These misalignment values were used in the model to verify their impact.

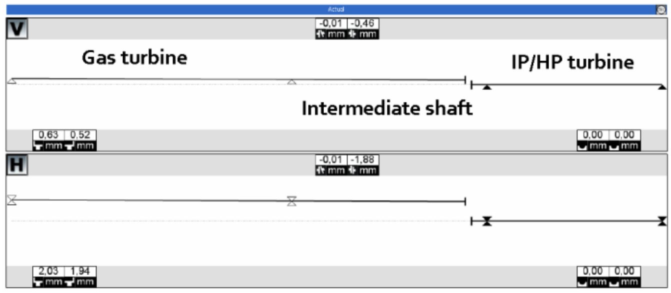

Figure 13: "Hot" alignment condition of the GT - IP/HP

Rotor dynamic calculations

It was suspected that a horizontal misalignment between the gas turbine and the IP/HP turbine causes the load sensitivity and therefore is responsible for the elevated vibration levels at the IP bearing. To find a possible explanation for this behaviour, different misalignment load cases were evaluated.

The calculated critical speeds near the running speed of the shaft line of a properly aligned shaft line are displayed in Figure 14-16. Based on this calculation it was possible to select the natural modes that have an influence on the behaviour of the unit at operating speed (50Hz):

- 3rd vertical bending mode of the gas turbine
- 3rd horizontal bending mode of the gas turbine
- 2nd horizontal bending mode of the IP/HP turbine

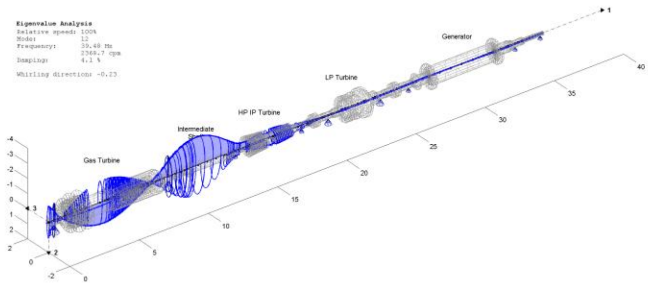

Figure 14: Mode shape of the 3rd vertical bending mode of the GT

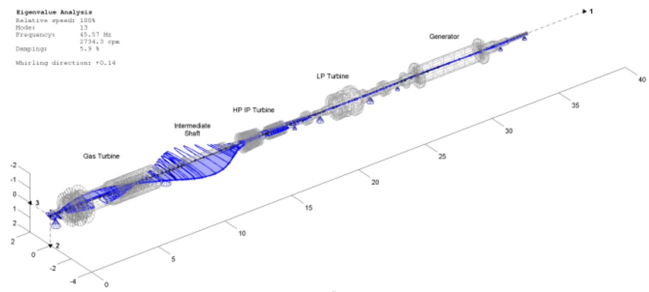

Figure 15: Mode shape of the 3rd horizontal bending mode of the GT

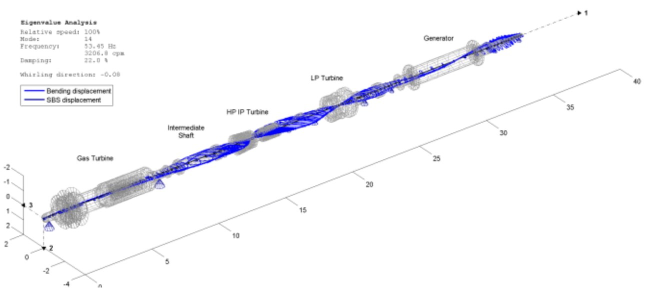

Figure 16: Mode shape of the 2nd horizontal bending mode of the IP/HP turbine

Only these modes will be considered in the different misalignment load cases (). Different misalignment conditions were calculated with the model based on the values obtained in the Permalign measurements. The used values are slightly increased in order to see the resulting effects more pronounced. It can already be concluded from these values that the 3rd bending mode of the gas turbine remains constant under these misaligned conditions. It is therefore less possible that this mode will be responsible for the load dependent vibration behaviour.

Table 1: Results of the misalignment load cases

	Aligned	CASE 1 3.75 / 1 mm	CASE 2 3.75 / 0.6 mm	CASE 3 3.75 / 0.5 mm	CASE 4 3.75 / 0.3 mm	CASE 5 7.5 / 1.2 mm
3rd bending GT V	39.5 Hz ; 4.1 %	39.5 Hz ; 4.1 %	39.5 Hz ; 4.1 %	39.5 Hz ; 4.1 %	39.6 Hz ; 4.1 %	39.6 Hz ; 4.1 %
3rd bending GT H	45.6 Hz ; 6.0 %	45.8 Hz ; 5.7 %	45.5 Hz ; 6.0 %	45.4 Hz ; 6.1 %	44.7 Hz ; 6.2 %	44.9 Hz ; 6.5 %
2nd bending HP IP H	53.2 Hz ; 22.7 %	54.7 Hz ; 23.7 %	53.2 Hz ; 22.5 %	52.9 Hz ; 21.8 %	53.5 Hz ; 17.3 %	52.1 Hz ; 19.4 %
Hor. force IP bearing	0 N	6.98 kN	-15.7 kN	-21.4 kN	-32.8 kN	-31.5 kN
Hor. stiffness IP bearing	0.51 MN/mm	0.6 MN/mm	0.45 MN/mm	0.41 MN/mm	0.25 MN/mm	0.32 MN/mm
Hor force HP bearing	0 N	-11.7 kN	-1.55 kN	0.1 kN	6.1 kN	-3.1 kN
Hor. Stiffness HP bearing	0.41 MN/mm	0.36 MN/mm	0.42 MN/mm	0.4 MN/mm	0.48 MN/mm	0.42 MN/mm

The values indicated with each load case mention the used horizontal displacement for respectively the turbine and compressor end bearing. Both values are in the positive horizontal direction of the bearing coordinate system because this is the only possible direction to lower the bearing stiffness and thus bringing the IP/HP second bending mode closer to the operating speed. This can be understood if one looks at the bearing pad orientation in . There is no supporting pad in the negative horizontal direction and therefore the dynamic behaviour is different from displacements in the positive bearing direction. However, this is only true if a certain load limit is not exceeded (approximately 50kN, obtained from the model), otherwise the bearing stiffness will increase again due to the upper and lower pad interaction (minimum oil film thickness).

This change in dynamic stiffness in these bearings is also visible in the orbit plots at different load conditions of the unit (Figure 18). At lower loads the compressor end bearing of the gas turbine shows a circular orbit which means that the bearing is behaving isotropic. This is reasonable because this bearing is a four tilted pad bearing with the load between the pads. The vertical and horizontal stiffness are equal and thus an isotropic behaviour can be expected. The IP bearing on the contrary shows a more elliptical orbit shape at lower loads which means that the bearing is behaving non-isotropic. This could also been seen from the horizontal and vertical stiffness and damping coefficients, which are different.

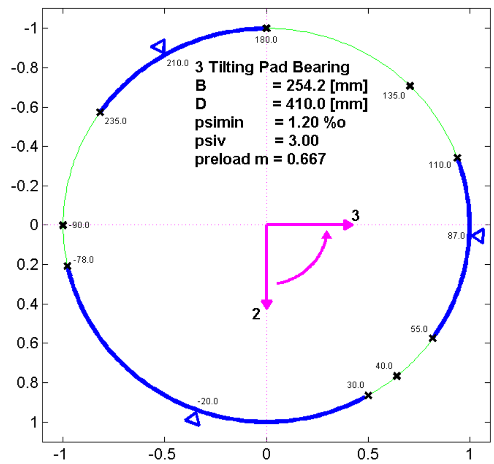

Figure 17: HP/IP bearing geometry

However when the load increases, the orbit shape for the compressor end bearing becomes smaller while the IP bearing becomes circular. This indicates that the bearing behaves isotropic, which can only be caused by a different shaft position within the bearing and thus a different vertical and horizontal bearing stiffness.

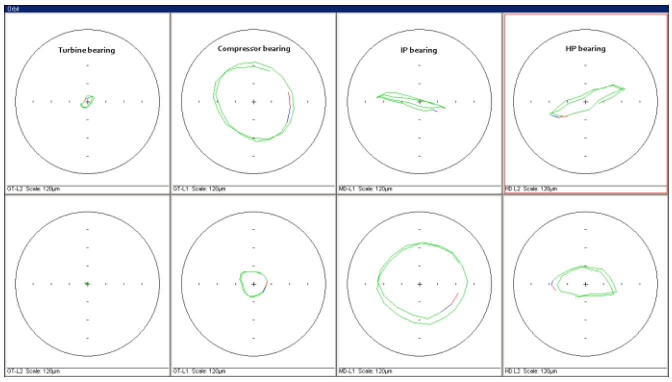

Figure 18: Orbit plots at different load conditions (up 110MW ; down 330MW)

shows that only cases 4 and 5 have an important influence on the 2nd horizontal bending mode of the IP/HP rotor. For case 4 this is done by influencing the damping factor, for case 5 this is by both a frequency decrease and a damping reduction. It is therefore most likely that the 2nd horizontal bending mode of the IP/HP rotor is responsible for the increased shaft vibrations near the IP bearing although the influence visible in the HP bearing doesn't correspond with the response of the model. The misalignment values used in case 5 are significantly higher than those found with the Permalign measurement but the study shows nevertheless that alignment deteriorations can alter the behaviour at nominal speed significantly.

At present, the results of both the study and the different alignment measurements are being compiled in order to define a modification of the alignment proposal for this type of shaft lines.

CONCLUSION

The use of rotor dynamic models in troubleshooting can certainly have a clarifying insight in complex vibration problems. Although most models are not "on the shelf" available for most maintenance departments in power plants, the experience shows that it is worth investing in having these models available for important, strategic shaft lines. The goal is not to predict the dynamic behaviour exactly, but more to search for trends or general parameter sensitivities. The previous cases are therefore examples of the practical use and input that can be expected from these kind of rotor dynamic models.

REFERENCES

1. Schmied, J., et. al., Application of MADYN 2000 to Rotordynamic problems of Industrial Machinery*, Proceedings of GT2007: ASME Turbo Expo*, Montreal, 2007.
2. Glienicke, J., Feder- und Dämpfungskonstanten von Gleitlagern für Turbomaschinen und deren Einfluss auf das Schwingungsverhalten eines einfachen Rotors, *Dissertation Techn. Hochschule Karlsruhe*, 1996.
3. Muszyńska, A., *Rotordynamics*, (2005) pp.184-207, Taylor and Francis Group.
4. Rybczynski J., Maps of Vibrational Symptoms of Bearing Misalignment Defects in Large Power Turboset, *Proceedings of ASME Turbo Expo*, Berlin, Germany, June (2008).
5. Endres N.M., Swan E.P., Impact of Wireless Laser Based Shaft Alignment on Vibration STG Coupling Failure, *Proceedings of PWR2007*, San Antonio, USA, July (2007).
6. Sekhar A.S. and Prabhu B.S., Effects of Coupling Misalignment on Vibrations of Rotating Machinery, *Journal of Sound and Vibration*, Vol. **185** (1995), pp. 665-671.
7. Nicholas J.C., Whalen J.K., Franklin S.D., Improving Critical Speed Calculations using Flexible Bearing Support FRF Compliance Data, *Proceedings of the 15^{th} Turbomachinery Symposium*, Texas A&M University, USA, pp. 69-78.
8. Vázquez J.A., Barrett L.E., Flack R.D., Including the Effects of Flexible Bearing Supports in Rotating Machinery, *International Journal of Rotating Machinery*, Vol. 7 (2001), No. 4, pp. 223-236.