BETA
Machinery Analysis

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Evolving Industry Calls for Advancements in Technology and Processes

Design of reliable compressor installations has become much more difficult and demanding over the years. Suppliers of specialized design services (pulsation studies) are faced with increasingly challenging demands from the various stakeholders.

Competing requirements

- the owner wants assured reliability, performance and efficiency to meet design objectives
- the equipment packager needs very rapid results to meet very tight scheduling
- the owner and the equipment packager want zero startup problems
- there is pressure to minimize cost of design modifications; source depends on the business arrangement
- the owner wants minimum capital cost
- the customer (owner, packager or EPC) wants low cost but high quality service

Meanwhile, the evolution of equipment has increased the need for more sophisticated and more accurate modeling methods. Accurately modeling compressors operating at ever higher speeds and with high horsepower ratings is technologically challenging. Modeling is easy enough - doing it well is difficult.

High operating speeds of current compressors require accurate prediction of mechanical natural frequencies well over 100 Hz. The higher the frequency the more difficult and error prone the exercise.

Concerning mechanical natural frequencies, the draft 5th edition of API 618 outlines a separation margin guideline that is one of the criteria that will dictate whether a forced response analysis will be required. There has been some debate in industry as to whether mechanical natural frequencies can be predicted accurately enough to rely on such a guideline. The problem is not really the technology. Modeling tools have long been available to do such analyses. Lack of knowledge concerning how to use the tools correctly; and the time and effort (cost) required has made these methods undesirable from a commercial point of view.

Advancements in modeling tools and computer power have enabled the evolution of modeling methods that are not only very accurate but efficient as well. However, these advances must be built on a large amount of experience and real world measurements before accurate results can be realized.

From the process perspective, delivery dates, imposed on the packagers by the end users and a very competitive market place, are getting ever shorter. As a result, if any level of acoustical and mechanical design analysis is going to be done, turnaround time for critical information has to be quick and the analysis process must minimize rework on the packager end.

The following article describes how Beta Machinery Analysis has evolved its state-of-the-art acoustical (pulsation) modeling methods. A subsequent article will describe Beta's mechanical (vibration) technology and processes.

Acoustical Analysis

In the early 1970's Beta Machinery Analysis developed the first digital acoustical program ("MAPAK") to be used as a commercial modeling tool in North America. The early versions of our MAPAK software were frequency based simulations that were limited to a small number of elements and had to be run on a mainframe computer at the University of Calgary. Although the early versions were crude by today's standards, the technology was leading edge and the accuracy was acceptable.

As higher horsepower, high speed compressors evolved it became evident that some of the simplifying assumptions that were made in the original software (assumptions necessary to allow the models to process in a reasonable amount of time) no longer provided enough accuracy for this new class of machines. Advancements in computing power allowed us to upgrade our frequency based program to again be able to predict pulsation and unbalanced forces to the degree of accuracy our standards demand.

In more recent years high speed pipeline and high pressure injection compressors, along with the evolution of new cylinder designs having dual inlet and outlet connections, have prompted yet another evolution in our acoustical modeling tools. To accurately predict the acoustical behavior of these types of systems a time domain version of MAPAK was developed. Field testing confirmed that our time domain model produced more accurate results, especially on these newer systems. However the average model took over 200 times longer to process than our frequency based simulation. A dedicated effort from our Research and Development Department resulted in development of a revolutionary processing tool which has allowed our more sophisticated time domain model to become a commercially viable tool. Today all acoustical models, no matter what type of system, are checked using our time domain program.

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INSIDE:

Case Study: Turbo Generator

Maintenance on the turbo generator had been reduced for several years for various reasons. The mechanical condition of this machine deteriorated to the point that field balancing was difficult. An innovative solution to control vibrations was required. This paper will chronicle the findings of field analyses and the results of troubleshooting efforts that Beta has been involved in from 1997 to present. It illustrates the importance of monitoring the mechanical condition of a rotating machine, and taking corrective and preventative action at the appropriate times.

Introduction

The subject machine is a 36 MW 3600 rpm steam turbine generator set that was originally installed in 1965. The turbo generator has two functions: generation of electrical power for the plant and production of low pressure utility steam (turbine exhaust steam).

The unit's performance and reliability had generally been good but overhauls since the early 1980's had yielded increasingly poor or troublesome results. This was due, in large part, to degradation and looseness of the soleplates and pedestals. The ongoing degradation resulted in shifting bearing support mechanical natural frequencies and variation in balance sensitivity.

An overall view of the turbo generator is shown in Figure 1. The near end is the opposite drive end bearing of the generator with the exciter cantilevered from it. At the far end is the steam turbine with the turbine's outboard standard containing timing and control devices at the far end (but not visible in the photograph).

A schematic of the unit showing the four bearings and the turbine's outboard standard is shown in Figure 2. The bearings will be referenced as DE for drive end, which may also be called inboard; ODE will be used for opposite drive end or the outboard bearings.

Fall of 1997

Until 1997, the unit generally ran in the 16 to 20 MW load range. It was overhauled and

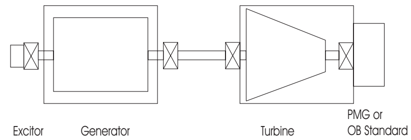

Figure 2. Schematic of the turbo generator.

rebalanced in 1997 by the manufacturer in preparation for loads to 36 MW. The balance results were poor with both radial and axial vibration at unacceptable levels. Our analysis in late 1997 found looseness in the generator's bearing pedestals and the turbine outboard standard. As a result the bearing pedestals were resonant near running speed. The generator was field rebalanced to acceptable levels in November 1997 at 36 MW. Thermal bowing of the generator rotor was found at higher loads. Particular attention was required to reduce axial vibration including use of a 3rd balancing plane for rotor straightening.

Figure 3 shows the dominant 1x component of the shaft relative measurements. As can be seen, the first critical of the generator rotor was in the 1700-1900 rpm range and was well damped. Also above 2000 rpm the shaft relative motion increased faster than (particularly the DE X probe readings), suggesting that rotor was approaching a second critical above 3800 rpm. The bump in the ODE X probe reading at 3100 rpm was due to an axial resonance of the opposite drive end bearing pedestal at 52 Hz. The turbine first critical was at 2100 rpm.

Prior to any field balancing, the generator bearing pedestal looseness was addressed. A bolting contractor was called in to correct the issues. Measurements of bolt stretch were performed during tightening. It was found that the bolts into the soleplate were approximately 1/8″ too long and were bottoming in the holes. During a previous overhaul, the washers had been removed! As well, the bolts into the transition piece were merely threaded into the transition piece, and the threads were stripped at one bolt. These conditions were corrected.

However, due to the looseness being there for many years the relative motion had fretted the surfaces. Therefore, after tightening some looseness was still present. Consideration was given to dismantling the unit and machining the mating surfaces. The decision was made to field balance the generator rotor without further correction of the pedestal looseness.

The balancing of the generator rotor began as a simple two plane process. Since a thermal bow in the rotor had been identified, measurements were taken after one hour of heat soak at full power (36 MW). The first stage of balancing was accomplished by adding correction weights to the DE and ODE generator balance planes. There was significant improvement in both shaft relative and bearing pedestal vibration magnitudes. With the existing looseness in the bearing pedestals a residual unbalance of less than 1.5 oz-in/1000 lb was desired but not yet accomplished (particularly at the ODE). Further adjustments of the weights were attempted, but any reduction in vibration at ODE was met with a corresponding increase at DE. The first stage correction weights are shown in Figure 4.

The second stage of the balance was also a two plane balance. The weights from the first stage were left. A trial weight was applied to the balance plane at the end of the slip ring which was outside the ODE bearing (cantilevered at the end of the exciter). The calculated correction weight was split so that part of the weight was on the slip ring plane and the rest was on the ODE balance plane. The weights were split such that an equal rotating force was generated by each, and they were in phase.

The radius at plane 2 was 9.75″, while at the slip ring it was 4″. This did not have the desired result. Then two weights were rotated relative to each other so that a couple was imposed, but the force was zero (i.e. 180 degrees apart, see Figure 5). This couple reduced the motion at the ODE, but did not increase the motion significantly at the DE. This resulted in an acceptable residual unbalance.

Figure 1. Overall view of the turbo generator.

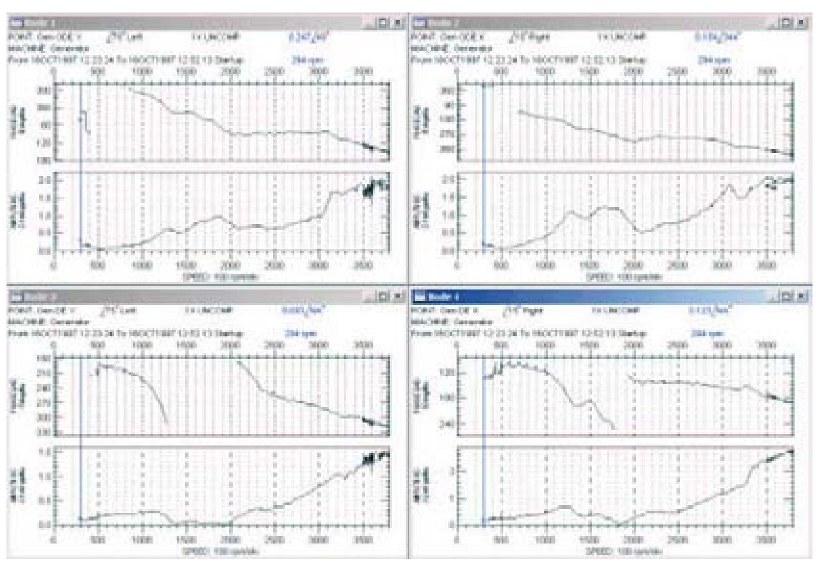

Figure 3. As found Bode plots of the shaft relative data for the generator in 1997.

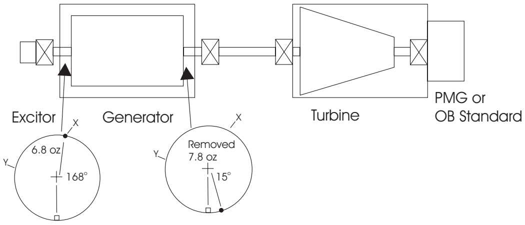

Figure 4. First stage balance weights on generator DE and ODE balance planes.

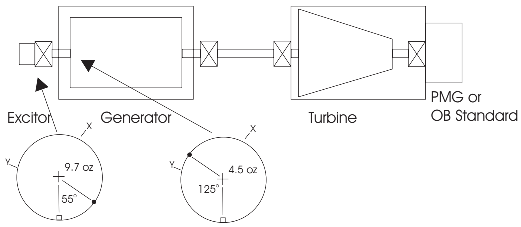

Figure 5. Second stage balance weights on generator ODE and slip ring (exciter) balance planes.

December 2001

The unit ran successfully at higher loads after the November 1997 rebalancing until a November 2000 power failure. The back-up DC lube pump failed and all generator bearings and journals were damaged on rundown, as was the turbine inboard journal. The journals were dressed in place and all bearings changed. After start-up, a step increase in vibration was recorded by plant personnel but levels remained acceptable.

The turbine rotor was removed for upgrade inspection in May/June 2001. A new turbine DE bearing was installed and a new timing wheel mounted closer to the turbine (in the outboard standard). During this shutdown, another consultant attempted to rebalance the generator rotor, including consolidation or removal of the 1997 balance weights.

The results were poor with both radial and axial vibration increasing to higher levels. Axial vibration, in particular, was at serious levels. They elected to install mass dampers on the generator bearing pedestals over further balance correction (see Figure 6). The mass dampers improved the high axial vibration.

Figure 6. Tuned mass damper installed on inboard generator pedestal in 2001.

The unit had been running in the 18 to 23 MW load range but plans were underway to return to higher loads. The plant requested our assistance in analyzing the turbo generator's condition including that of the mass dampers and developing recommendations for improved performance and reliability. That work was undertaken from December 10th to 12th, 2001.

It was originally expected that this would include rebalancing of the generator rotor. However, no information on shaft and bearing alignment, bearing clearances or balance weight changes and position had been received by the plant from November 2000 and June 2001 work. This prevented analysis of the problems encountered in the June 2001 balance work and the changes since our involvement in November 1997.

This analysis also indicated that the highest vibration was now found at the turbine inboard shaft and housing positions. It was decided to postpone rebalance of the generator rotor until more information was received and more time was available.

The plant requested that we improve the performance at the turbine drive end position with trim balancing. This required the use of a shaft balancing ring fabricated but not employed in 1997 to allow a 4th balancing plane if needed to improve axial vibration. The balancing ring allowed quick installation of weights without generator hood removal. We committed to attempt a one shot no trial weight trim balance to minimize power loss costs.

As shown in Figure 7 below, there was a large increase in turbine shaft relative amplitude as the unit passed through its critical speed during rundown. There was a simultaneous 180 degree change in phase, which is characteristic of resonance or a critical speed. The energy level seen at the turbine ODE during passage through the critical speeds was far more severe than expected. The energy of this occurrence could be felt in the building structure. The operators indicated they had noted this was occurring regularly since the June 2001 removal of the turbine and rebalancing of the generator.

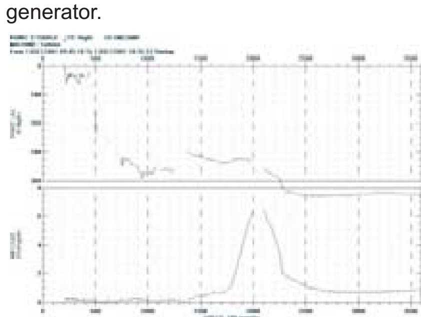

Figure 7. Bode plot of 1x component measured at the turbine opposite drive end bearing.

Full bearing clearance for the smaller turbine outboard journal is recorded as 9 mils. API General Purpose Steam Turbine Standard 611 states that the calculated unbalanced peak to peak rotor amplitudes at any speed from zero to trip shall not exceed 75 percent of the minimum design diametrical running clearances throughout one machine. For the turbine outboard bearing the allowable limit for shaft vibration is thus 6.75 mils p-p. The Bode plot indicates an as found shaft vibration of 8 mils p-p, exceeding the allowable limit. This unsafe condition required review of the forecasted trim balance response to avoid worsening this as found result (and hopefully improve it).

Analysis also indicated very large changes in shaft bowing/sets since 1997. Amplitudes had grown up to 200% and phases had changed by up to 145 degrees. Load trial results showed that the rotors should be balanced for specific load ranges. The change in thermal bowing was a result of deterioration of the generator rotor since the 1997 balancing. The rapid stop from full temperature and the resulting bearing seizure that occurred in the November 2000 power failure may have been a contributing factor. There was also indication of misalignment forces at both generator DE and ODE bearings.

A one shot balance of the turbine was undertaken. This involved installation of the 1997 balancing ring between the turbine and generator as shown in Figure 8.

Comparative operating deflected shape analysis of the shaft and housing vibration showed the largest improvement at the turbine drive end would result with the ring and trim balance weights positioned at the generator drive end side. A 5.8 oz. weight was installed on the shaft balancing ring, improving shaft relative vibration at the targeted turbine drive end bearing up to 76% and on the pedestal by up to 27%. It also improved the critical turbine ODE shaft relative and housing vibrations by up to 67% and 73% respectively. However, generator DE shaft relative and housing vibrations increased by up to 97% and 33% respectively. Phase reversal patterns indicated that further weight would continue to increase generator drive end vibration. The trim balancing avoided worsening the unacceptable conditions at the turbine outboard shaft during passage through the shaft critical speed.

Figure 8. Shaft balance ring installed between the turbine and generator.

Spring 2003

The generator windings failed on February 20, 2003. This was probably the final stage of the thermal bow that had been noted in 1997 and that had changed in the 2001 analysis. The generator rotor was shipped to the manufacturer for rewinding. After rewinding it was slow speed balanced at 375 rpm in the shop to help ensure that it could proceed safely through the critical speeds.

While the generator rotor was out, the looseness of the generator pedestals and soleplates was finally addressed. They were removed, and large voids and degradation of the grout were found (see Figures 9 and 10). The soleplates were re-grouted, and the pedestal spacer pieces were replaced with solid blocks.

The bearing clearances were checked and it was found that the turbine ODE bearing clearance was up to 0.0075″ above its nominal 0.009″ design clearance. This was due to the shaft at this position being 0.007" undersize. This likely dates back to the power failure damage, and contributed to the high sensitivity of the shaft at this position. The shaft relative vibration of the turbine at ODE was found above the API Standard 611 allowable limits in December 2001 analysis during passage through the critical speeds. This unsafe condition was shaking the building structure.

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Fig. 9 & 10. Two views of grout under the generator DE bearing pedestal as found in the spring 2003.

Prior to the April 2003 generator balancing, the high vibrations seen at the turbine ODE during passage through the critical in December 2001 had to be addressed. During the first start-up after the overhaul, the vibration of the turbine ODE was checked at 1400 rpm. Comparison to the value at 1400 rpm in December 2001 indicated that the vibration would again be excessive when passing through the critical.

To protect the turbine, the turbine outboard was balanced at 1400 rpm prior to passage through the critical speed. This was done with the installation of a balancing ring between the turbine ODE bearing pedestal and the PMG standard. This allowed installation of balance weights on the turbine rotor without opening the turbine. Field balancing of the turbine using the internal balance plane would have greatly increased the time as cool down and warm up time would have been required for each run. As well, the access covers to the turbine balance planes had not been opened since the early 1980's, indicating a significant probability of disassembly problems with the long term corrosion and heat effects.

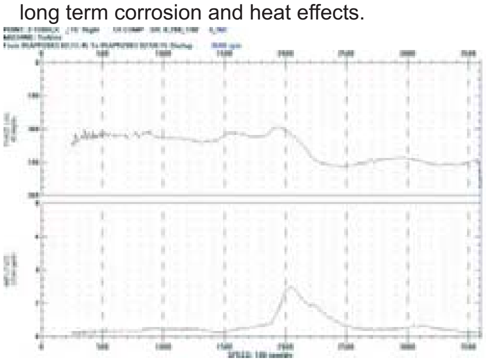

Figure 11. Bode plot of 1x component at turbine opposite drive end bearing after trim balancing with ring. Compare to Figure 7.

Figure 11 shows the 1x component of vibration after trim balancing of the turbine outboard using the balance ring. The maximum magnitude reached during the rundown through the critical was less than 3 mils p-p, compared to the more than 8 mils p-p observed in December 2001.

Following this, the generator was successfully field trim balanced at a power level of 18 MW. There was a large improvement in the generator bearing pedestal resonances with the refurbishment of the grout. In 1997, the generator drive end pedestal had an axial mechanical natural frequency of 52 Hz. After the repair in 2003, that had moved to 112 Hz.

Conclusions

Earlier or timelier work on several areas of this turbo generator would have greatly reduced the complexity and magnitude of the problems in later years. The generator bearing pedestal looseness had been noted by plant personnel in the 1980's, but was not finally repaired until 2003. The looseness caused ongoing shifts in the mechanical natural frequencies and vibration performance thereby making the generator rotor difficult to balance and requiring innovative solutions on the part of the field analysts. The generator rotor also developed a thermal bow that grew in magnitude, and finally ended with winding failure. The clearance in the turbine opposite drive end bearing was allowed to be excessive resulting in unacceptable vibrations through the turbine critical. Rather than overhaul the shaft and bearing, the problem was addressed with balance weights.

Advancements in Technology

(continued from page 1)

The shorter delivery times and an increase in the rental/lease market led to another advancement in our acoustical modeling capabilities. Ideally the best way to optimize the design of pulsation control devices is to do an acoustical model of the on-skid and off-skid components. These days it is common to have the pulsation bottles well into the construction phase before any information about the off-skid piping is available.

The traditional approach to this situation has been to design the pulsation control devices assuming an infinite pipe termination at skid edge. From an acoustical point of view this provides the most un-conservative approach - a poor design methodology. This can lead to serious startup problems and major rework, such as secondary volumes off-skid, modifications to pulsation bottles that are already built, or increased pressure drop due to additional orifice plates being utilized.

Beta Machinery Analysis has developed a proprietary technique to efficiently assess the sensitivity of the system to the off-skid components. If the skid edge sensitivity analysis indicates the system is not sensitive, the resulting pulsation bottles can be released for construction with a high degree of confidence that major changes will not be required once the off-skid components are modeled. If the skid edge sensitivity analysis indicates the system is sensitive all parties can be involved in a discussion to determine how best to proceed (eg. take a conservative approach to pulsation control to keep on schedule, expedite at least a preliminary off-skid layout, delay construction of pulsation bottles, etc.).

Test Data Essential

The development of accurate modeling tools and techniques relies on test data. Beta has always used pulsation data measured in the laboratory and the field to evaluate the accuracy of the simulation results and then, where needed, to adjust the process.

Suppliers without field service capability are missing this essential aspect. Also, those who rely on commercial software have a major problem in this area as they are unable to modify and correct their simulations.

Improving the Service Interface

Some of the increasing challenge has to do with service delivery rather than technology. However, technology can help here also. Beta has made meeting customer delivery schedules a high priority. Modeling improvements have addressed greater efficiency, as well as greater accuracy.

Use of the internet for communication of data, drawings, and results helps speed the process. In addition, Beta Machinery Analysis has a powerful internal computer network that supports a team approach involving any number of our engineering staff. This enables rapid turn around even on large, complicated jobs. Suppliers without these resources either cannot meet demanding schedules or are forced to take short cuts.

Summary

The reciprocating compression industry is an exciting and dynamic industry. The evolution that has taken place has presented many challenges from a design consultant point of view. With a dedicated Research and Development Department and a constant feedback loop between the design group, field services and R&D, Beta Machinery Analysis has been able to evolve with the industry.

Fast, good *and* cheap???

Fast and good, yes. Cheap? Well, you get what you pay for.

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