Brüel & Kjær Vibro

APPLICATION NOTE

Excessive vibrations on 250-MW turbo-generator bearings due to flexible steel foundations

Excessive bearing vibration was detected during commissioning of a 250 MW turbo-generator at the Huntly Power Station. Further investigation determined that it was resonance caused by the flexible steel foundations used for earthquake protection.The optimal solution was a dynamic vibration absorber that was mounted directly on the bearing pedestal. Although this solution prevented the bearings from being damaged prematurely, vibrations are still relatively high during run up.

The COMPASS monitoring system is used to carefully monitor the condition of the machine during these high vibrations as well as for performing tricky balancing runs.

Power station designed for earthquake protection

Huntly Power Station is owned and operated by Genesis Power Ltd., one of four State Owned Enterprises (SOE) that formerly made up the previous New Zealand Electricity Department. Huntly has four Parsons 250-MW units, the first of which was commissioned in 1981.

Each unit includes a 250-MW turbo-generator and a steam turbine driven main boiler feed pump. The turbo-generator is mounted on steel foundations to provide protection in the event of an earthquake. The COMPASS system is installed to monitor the turbo-generators and main boiler feed pumps.

Foundation design solves one problem but creates another

During early commissioning it was noticed that bearing 7 on the inboard side of the generator had a very high horizontal vibration (18 mm/s Peak) that peaked at the machine running speed (3000 RPM). Further investigation indicated that the high vibrations detected at bearing 7 were due to a horizontal resonance at 100 Hz (twice machine speed) in the flexible cross beam to which the bearing pedestal is mounted. This resonance was being excited by the generator gravity critical (a function of the differential stiffness inherent in the construction of two pole generators). Even though the installed transducers monitor vibrations only in the vertical axis at that time, the horizontal resonance rocked the bearing housing in such a way that a portion of it was seen in the vertical axis.

With the machine running at normal speed an

Fig. 1 Huntly Power Station

investigation was undertaken to determine how the bearing was moving. This involved measuring the 1´ and 2´ amplitude and phase on numerous points around the bearing pedestal and the steel

Fig. 2 Flexible steal-beam foundation for earthquake protection

foundation structure. These measurements indicated that the resonance was due to lack of horizontal stiffness of the bearing pedestal and the foundation hollow steel beam to which it was mounted (Fig. 2).

Simple solution

Major stiffening of the foundation columns, such as filling them with concrete, were only briefly considered as this would negate the earthquake protection afforded by the foundations. Adding an additional holding down bolts to each side of the pedestal to increase the stiffness had no effect. The only option available at a reasonable cost was to fit dynamic vibration absorbers to the bearing pedestal.

Dynamic vibration absorbers are relatively small auxiliary masses attached to the machine via a steel shaft that acts as a spring as shown in Fig. 3. The length of the shaft is tuned so that the auxiliary masses resonate at the frequency that is creating the original problem, which in Huntly's case was 100 Hz.

The effect of adding the absorber is to split the original resonance in two, one at a slightly higher frequency and one at a slightly lower frequency. If, as in the case of a synchronous machine, the running speed is constant then moving the resonance away from the running speed reduces the running speed vibration. What it does do, however is add a resonance into the machine run up (the upper frequency resonance does not present a problem since it is excited at speeds above the operating speed).

The design of a dynamic absorber is an optimisation between the physical constraints of the space available that governs the size of the masses, and the tolerance of the spring material to high cycle fatigue (placement shown in Fig. 4).

The degree of reduction in the vibration is governed by the damping between the absorber and the machine and the level of vibration that the spring can tolerate. Tuning in the case of Huntly is achieved by measuring the pedestal vibration while each mass is, in turn, moved on the threaded spring ¼ turn at a time. At the same time the vibration of the absorber is monitored to ensure the maximum value is not exceeded (200 mm/sec at 100 Hz). Because the 2× vibration problem was in the horizontal axis, the absorber spring has a narrower horizontal cross section to ensure that masses resonate only in the horizontal direction when tuned.

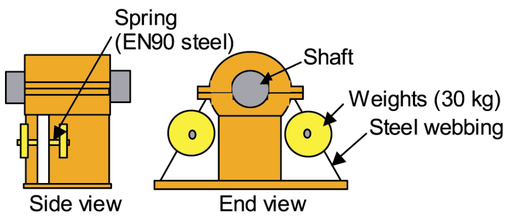

Fig. 3 Dynamic vibration absorber as seen from bearing 7 (these were mounted on all four units)

For safety reasons each weight has a wire strap that will prevent the weight falling two floors should the steel bar spring fail as shown in Fig. 4 (up to now there have been four failures).

Marginal results

As was previously mentioned the vibration absorber splits the 100 Hz horizontal resonance such that a lower frequency resonance of 91.5 Hz is introduced at a machine speed of 2750 RPM. When all four weights of the absorber have been optimally tuned and are within the vibration constraints of the absorber spring, the horizontal vibration of the pedestal at 100 Hz is reduced by approximately 5mm/s.

Even with a reduced 100 Hz horizontal resonance and a reasonably good state of balance, the overall vibration still is high at 24 mm/s at 2750 RPM during run up, as seen in Fig. 7. This is because the overall vibration is the vector sum of the horizontal resonance (Fig. 6) plus the 1× running speed vibration that results from being so close to the second critical speed of 2800 RPM, which cannot be reduced (see Table 1 and Fig. 5). The overall vibrations during run up at machines speeds from 2750 to 2850 RPM can be significantly higher if there is imbalance present.

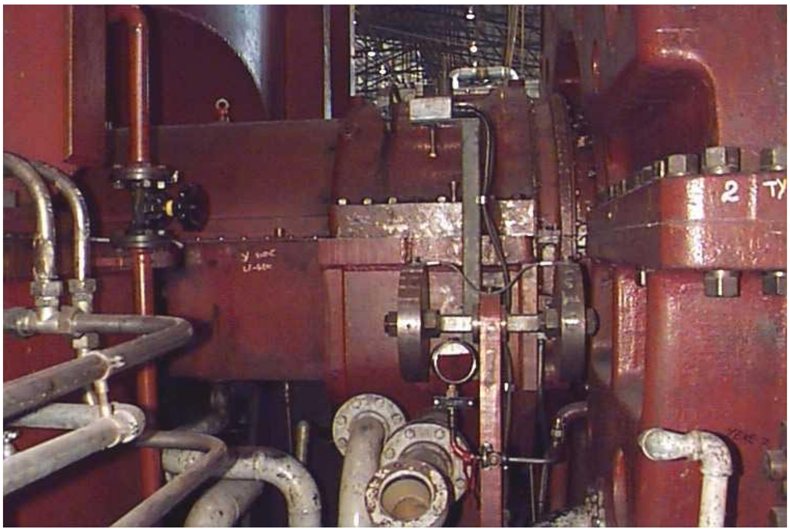

Fig. 4 Dynamic vibration absorbers as installed

In addition to the vertical and horizontal vibration, there is also a 1´ axial vibration of the bearings that occurs if there is imbalance present to any significant degree, resulting from the mode shape of the shaft. As in the case of the lateral stiffness, the axial stiffness of the two generator bearings is also fairly low. Even with a moderate state of imbalance, the axial motion of the bearing pedestals can be high and has been measured up to 30 mm/s peak. The vertical axis vibration sensors read the vector sum of the true radial vibration and the radial motion produced from the axial rocking of the pedestal.

When small balance changes have caused high vibration levels during peak loading and the machines cannot be removed from service, the axial vibration can be largely removed by attaching a large jacking screw to the top of the pedestal. This is tensioned against the end housing of the generator, thus stiffening the bearing pedestal in the axial direction. But the vibration energy does have the habit of popping up somewhere else, and this is typically in the adjacent bearing housing.

Machine Component	**Critical speeds (RPM)**			
	Mode	**OEM Original Calculations**	**Huntly Calculations**	**Tests before modification**
Generator shaft	1st	1476	1184	1200
	2nd		2822	2800
Generator bearing pedestal				2300
LP shaft	1st		1658	1400 - 1600
Exciter shaft	1st	2215	1748	1950
HP/IP Combined shaft	1st		1812	Not tested
	2nd		2004	Not tested

Table 1 Critical speeds of the turbo-generator components as originally specified by OEM for a rigid foundation compared to the lower critical speeds resulting from the flexible foundation (the machine's critical speeds are unaffected by the vibration absorber)

Conclusion

Although the flexible foundation structure achieved its objective of isolating the machine from the effects of an earthquake, they also created a raft of other ongoing problems for the machine and in particular to anybody trying to balance the generator rotor. Although mounting a dynamic vibration absorber on the generator bearing pedestals was the best solution to minimize the vibration problem, it was not completely eliminated.

The COMPASS system has an important role for monitoring imbalance and the condition of the bearings. More importantly the Compass system has allowed deferring generator balancing by several months to a more favourable load period. Prior to the Compass installation small balance changes would have forced the unit to be shut down because the vibrations went outside the range of the old installed monitoring equipment during the run up.

Acknowledgements

We wish to thank **Simon Hurricks**, Machine Dynamics Engineer at Genesis Power Ltd., for submitting this story.

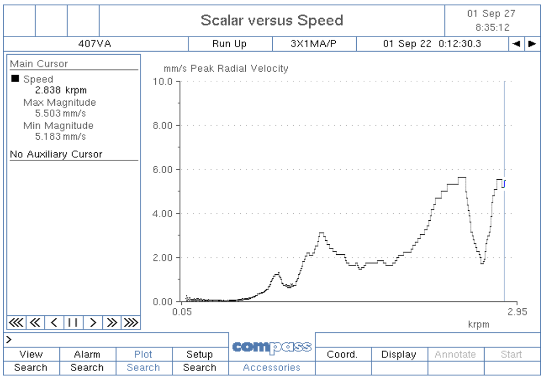

Fig. 5 Generator shaft 1× vertical-axis vibration during run up (second critical speed)

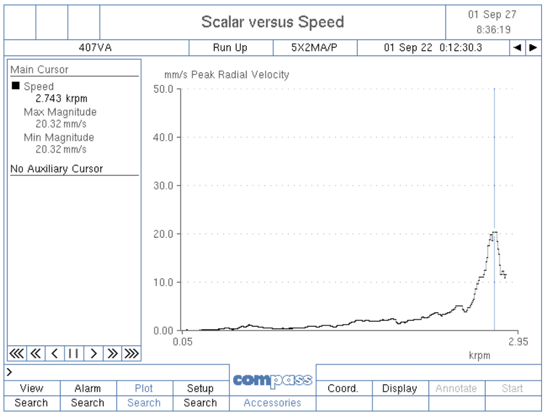

Fig. 6 Generator shaft 2× vertical-axis vibration during run up (dominant effect)

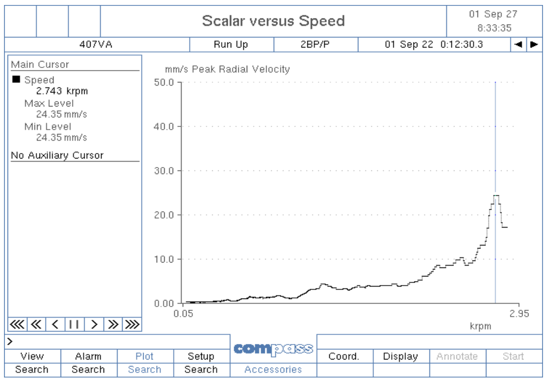

Fig. 7 Generator shaft band-pass vertical-axis vibration, 10 Hz-1kHz, during run up (vector sum of 1× and 2× running speed frequency components)

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