Low Cost Multiplane Balance of a Gas Turbine Auxiliary Driveshaft

A compressor train at an Amerada Hess Corporation plant suffered from serious vibration. The staff of Fern Engineering developed a low-speed balancing procedure to solve the problem, at a fraction of the cost of high-speed balancing.

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A mechanical drive gas turbine had been plagued with severe vibration for almost its entire history. The problem was finally isolated to a long slender shaft that operated at high speed. Even though the shaft had been balanced several times, the problem persisted. Multiplane, high-speed balancing was explored but it was deemed too expensive for an interim measure, and vendors where reluctant to conduct the procedure at the high shaft operating speed.

As an alternative, an analytical procedure was defined that duplicates the multiplane balance at a fraction of the cost. This procedure was implemented and the resulting balance reduced the vibration levels by 90%.

Background and Equipment Description

Amerada Hess Corp (AHC) purchased a gas processing plant in south central Louisiana, U.S.A. in 1999. The plant was first commissioned in the mid-1970s and features a Westinghouse W251BM gas turbine driving a Delaval gas compressor.

This installation is unique in that it has an expander/steam turbine that is solidly connected to the exhaust end of the gas turbine with no provision for decoupling. (As shown in Figure 1, the starter train consists of a long slender jackshaft, an auxiliary gearbox, a smaller shaft with a gear coupling, and the expander turbine.) The expander/steam turbine has a dual function. One function is that of starter for the gas turbine and the second is that of a "helper" turbine to supplement power during times of peak production. However, this "helper" feature has not been used in many years and starting is now the only function of the turbine. When the expander turbine is called on to start the gas turbine, natural gas expansion is used instead of steam to power the turbine because no warm-up is required with gas and the start up is rapid.

The long jackshaft is 89 inches long

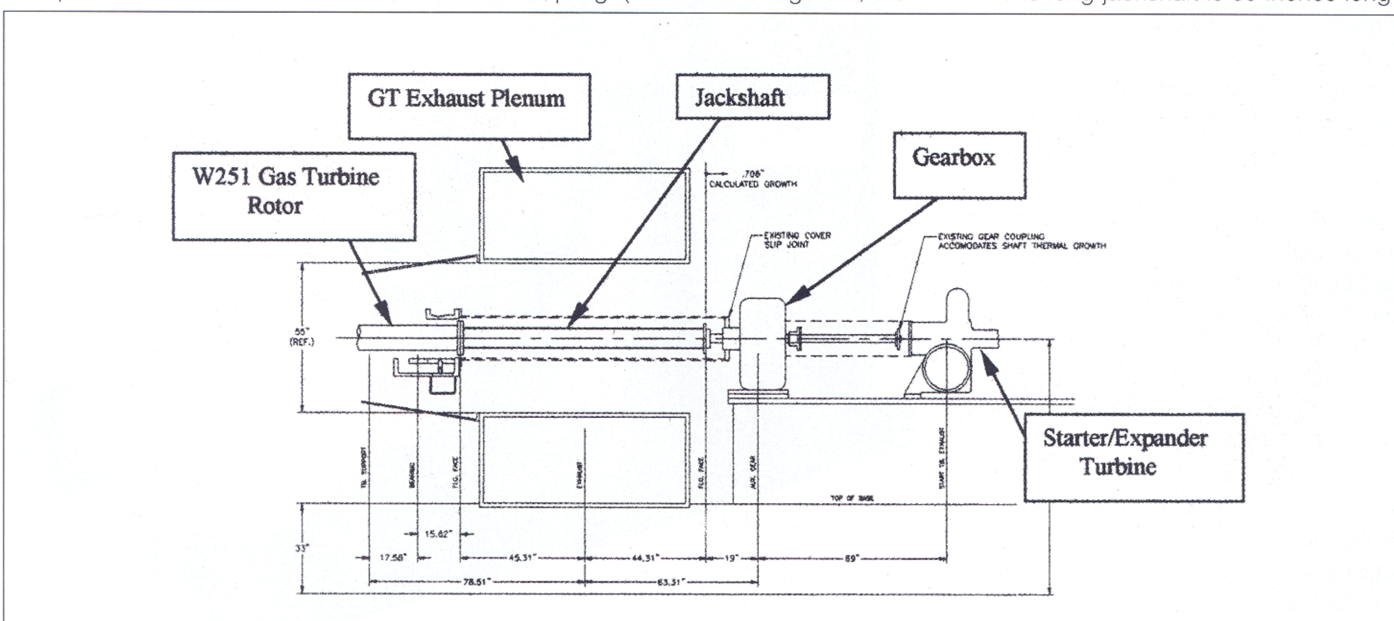

Fig.1. Starter drive train, viewed from the side.

x 71/2 inches in diameter, with one end solidly bolted to the aft end to the gas turbine shaft and the other end solidly bolted to the input side (pinion shaft) of the gearbox. The pinion shaft goes straight through the gearbox case with no gear ratio change to an output side. The pinion shaft has no thrust bearings so that it is free to move back and forth. The smaller gear-coupling shaft is solidly bolted to the output end of the pinion shaft, and the gear coupling end is connected to the expander turbine. The purpose of the gear coupling is to accommodate the large accumulated thermal growth of the entire gas turbine and starter train shafting.

Because the expander turbine is solidly coupled to the gas turbine with no decoupling provision, once the gas turbine has started the shafting, gearbox and expander turbine are continuously rotating with the gas turbine at 5,050 rpm.

Review of the past maintenance records revealed that severe vibration problems have plagued the exhaust end of the gas turbine and the starter train for many years, causing frequent and unplanned shutdowns. The OEM attempted to balance the gas turbine (at operating speed) on four separate occasions and was unsuccessful.

Identification of the Vibration Source

Amerada Hess Corp, the owner/client, contracted Fern Engineering (Fern) to help identify the problem and to suggest corrective measures. Engineers at Fern generated a finite element model of the entire gas turbine rotor and starter train, using ANSYS computer code, to evaluate the critical speeds and natural frequencies of the system. Results of the analysis revealed that the slender jackshaft was operating very near, but slightly below, its critical speed. It became apparent that the jackshaft required a precise and high-speed balance. Up to this point the jackshaft had been balanced and rebalanced several times, but only at conventional balance speeds of approximately 1,200 rpm.

AHC concluded that the addition of a clutch would alleviate the problem by decoupling the starter train once start-up was complete. However, this correction would require the turbine to be down for an extended time period, and could only be implemented at the next major outage, which was planned for the next year. AHC estimated that the procurement cycle for the hardware would be several months, due to clutch modifications which would be necessary for this application, which put delivery some time after an upcoming short routine outage.

Together, AHC and Fern developed an interim correction plan that featured high speed balancing (multiple plane balancing) of the jackshaft and restoration of the bearing fits and clearances. However, quotes from vendors to perform the high-speed balance were astronomical because of the necessary fixtures and precautionary measures dictated by the high speed. To get over this hurdle, Fern devised a procedure that duplicates the multiple plane high-speed balance, but is actually performed at conventional balance speeds.

Long slender shafts such as this jackshaft are difficult to machine without significant runout and wall thickness variations. And the culprit is usually the ID or central bore that is eccentric to the OD and the locating pilot diameters. Calculations show that the unbalance resulting from an eccentric bore is equivalent to displacing what would have been the weight of the bore by 1/2 of the difference of the wall thickness. The unbalance value is surprisingly large owing to the large weight of the would-be bore mass.

In a conventional balancing procedure, balance weights that compensate for the inherent unbalance are placed on the ends of the shaft. This placement of the balance weights is adequate, providing that the shaft operating speed is close to the balancing speed, or the shaft is considered a compact shape. However, with long, slender shafts that operate at speeds much higher than the balance speed, conventional balancing is inadequate. Because the balance weights are lumped at the shaft ends instead of being distributed along the shaft length directly opposing the unbalance, the shaft will deflect as in a simply supported beam with a distributed load, as shown in Figure 2.

Deflection of the shaft will occur as the speed increases beyond the balance speed, with the deflection being proportional to the square of the speed. At high speeds this deflection becomes significant. As a result, the

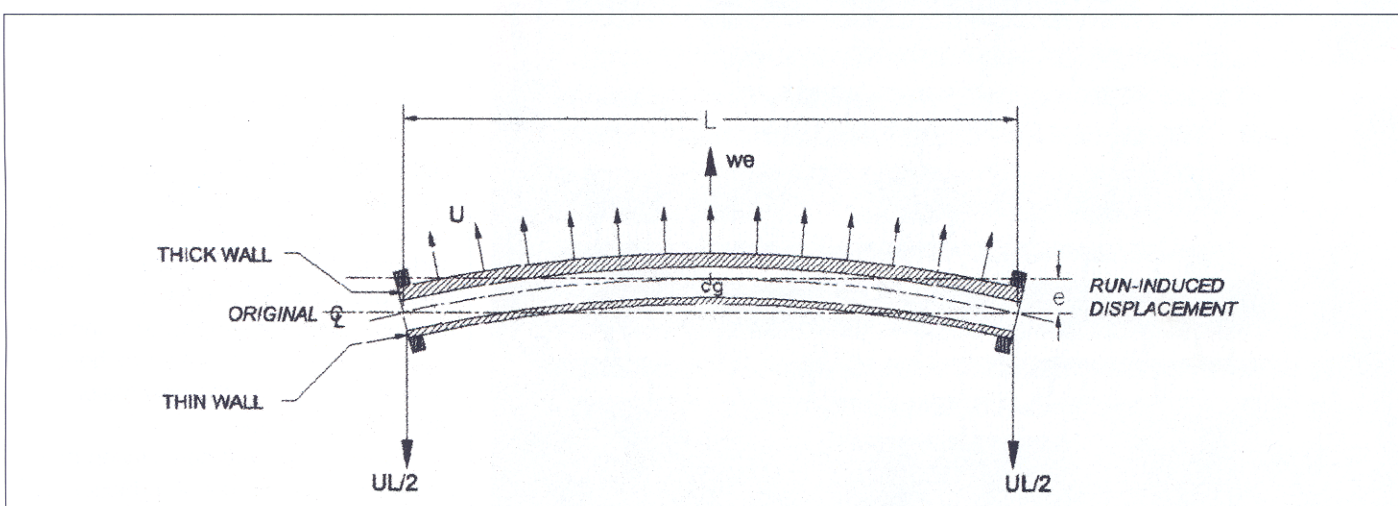

Fig.2. Shaft deflection due to an unbalance condition.

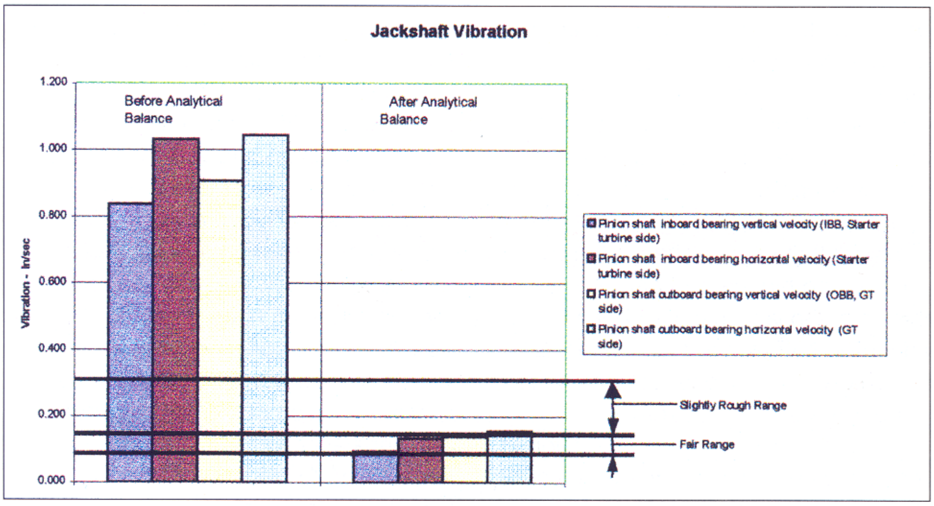

Fig.3. Jackshaft vibration levels, before and after balancing.

displaced center of gravity of the shaft creates a run-induced or deflection induced unbalance.

A Low-Cost Multiplane Balance Procedure

The corrective procedure involves defining three planes along the shaft tube (1/4 span, 1/2 span and 3/4 span) and measuring the wall thickness and run-out around the circumference at equal clock positions in these planes. Polar plots of the measurements are then made, and the associated center of gravity (cg) offset and its clock position are then calculated for each plane. This cg offset times the weight of shaft length (1/4 shaft tube) that it represents equals the unbalance. A compensating weight is then placed in that plane to negate the inherent unbalance. A final trim balance is then conducted at conventional balance speeds with those balance weights placed at the shaft ends in the normal manner.

As an exercise, the end balance weights were also predicted by calculation prior to the trim balance. The weights dictated by the conventional balance were remarkably close to the amount predicted by calculation, validating the procedure.

By following this procedure, five balance planes were created in contrast to two balance planes for a conventional balance. More importantly, the three additional planes are located at intermediate positions along the shaft length where a substantial part, and in this case a majority, of the unbalance exists.

Alignment and Gearbox Bearing Fits

A complete starter train alignment and bearing replacement had been done during an earlier outage for balancing and replacement of the jackshaft. During the time since that earlier outage, there were two bearing replacements, one scheduled and one unscheduled. The last was just two weeks before the balancing and replacement of the jackshaft. In both replacements, bearing wear consistent with high load and misalignment was apparent.

Alignment was rechecked and found to be within specification. Bearing to housing fits were checked and found to be 0.007 loose, whereas OEM requirements call for a tight fit. Apparently the years of high vibration resulted in wearing of the gearbox mating diameters. This problem caused the misalignment wear pattern observed at teardown, which contributed to the vibration during running.

Successful Implementation

During a short scheduled outage, the balanced shaft was installed by AHC and Fern, and a modified bearing installation procedure was instituted to account for the housing wear. (Line boring and resizing of the gearbox housing is scheduled during the next major maintenance outage.)

Vibration measurements taken before and after showed a major reduction of 90% in vibration levels. Figure 3 shows a side-by-side comparison of vibration levels before and after balancing. ■