A TORSIONAL STRAIN AND VIBRATION MONITOR FOR THE MOTOR-GENERATOR SHAFT OF THE ZERO GRADIENT SYNCHROTRON (ZGS)

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Introduction

A flywheel-motor-generator set supplies the power to pulse the ZGS ring magnet through a normal accelerating cycle. During the acceleration and flattop portions of the cycle, energy flows from the flywheel to the magnet. The direction of energy flow then reverses during the time that the magnetic field is decreasing.

This reversal of energy flow produces a reverse of the strain (twist) in the shaft connecting the generators to the flywheel. This in turn causes fatigue in the metal shaft.

The number of fully reversed stress cycles can be much larger than the number of accelerator cycles. This increase in number can be caused by shock excited ringing and by resonances. Shock excitation of torsional vibrations can be produced by sudden changes in the energy flow rate to the magnet. Resonances might be excited if the period of the ZGS pulse or if some part of the pulse is a multiple of a natural period of vibration of the rotating system.

Partially reversed stress cycles may be produced by the motor. This is especially important during motor startup when two times the slip frequency of the motor rotor matches any one of the natural vibration frequencies.

Partially reversed stress cycles may be produced at higher frequencies by the generators. The variations in generator torque are a function of the distorted waveshape generated by the rectifiers. This source of driven vibrations produces a very noticeable change in audible noise during the "invert" part of the ring magnet cycle.

The several sources of stress in the flywheel-motor-generator system can produce torsional vibrations that cover a wide range in frequencies. These extend from zero frequency to at least the twelfth harmonic of the generator voltage wave.

Motor-Generator Characteristics

The motor-generator set contains four main rotational inertias: the motor armature, the two generator armatures, and the flywheel. These are connected to each other by three lengths of steel shunting. Calculations performed by the manufacturer showed that several modes of torsional vibration are possible. The highest frequency was calculated to be 40.5 Hz. For comparison, the shaft rotates at 12.9-12.3 r/s.

The fatigue calculations for the 34-in diameter shafts connecting the flywheel to the generators indicated that the life of the shaft was unlimited if the amplitude of the torsional vibration was less than 20 mils (0.020 in) measured at the shaft surface. This corresponds to a twist of 1.2 mrad in the 8-ft long shaft.

General Principles

The general principles of the torsional strain and torsional vibration monitor are illustrated by the following example.

Suppose that two lines are ruled on the surface of the shaft, one near the flywheel and one near the generator. In addition, suppose that the two lines are parallel to the axis of the shaft and are at the same angular position at zero strain. If now the flywheel is prevented from rotating and a torque is applied to the generator end of the shaft, a torsional strain will result. This may be observed with stationary microscopes with graduated eyepieces. The strain will produce a displacement of one line with respect to the other. This displacement may be measured in thousandths of an inch.

Suppose now that the shaft is rotating at a fixed speed. If the shaft has no strain, the two ruled lines will pass by the axes of the microscopes at the same time. On the other hand, if the shaft has a strain, one line will cross the microscope axis before the other. The time interval between the two crossings will be proportional to the strain. The direction of the strain (twist) determines which ruled line crosses first.

Suppose again that the shaft is rotating at a fixed speed. Suppose also that we displace one microscope around the shaft by an amount larger than the displacement produced by the largest strain (twist). This will insulate that one ruled line always crosses its microscope axis before the other does. The result is a simplification of the electronic hardware.

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needed for measuring the time interval between the two crossings.

The above example illustrates the proportionality between the strain and the time interval between the lines crossing the microscope axes. It is apparent also that this time interval is inversely proportional to the rotational speed of the shaft. This last dependence can be removed by making the frequency of the clock pulses proportional to the rotational velocity of the flywheel.

The example described above would make only one measurement of strain per revolution of the shaft. This is inadequate since the above discussions of motor-generator characteristics and of driven vibrations indicate that frequencies as high as 700 Hz may be produced. Many sets of ruled lines are, therefore, required.

Description of the Monitor

A block diagram of one torsional strain monitor is given in Fig. 1. Two such systems are installed, one between the flywheel and generator No. 1 and the second between the flywheel and generator No. 2.

Lines were not ruled on the generator shafts for mechanical reasons. Instead, stainless steel bands containing engraved lines were attached to the shaft with epoxy cement. The bands, which were made up of 6-in long sections, were aligned using graduated scales in the eyepieces of the microscopes. Mechanical alignment was limited to an error of a few thousandths of an inch by variations in the width and in the spacing of the engraved lines.

The maximum safe strain was computed to be 20 mils; therefore, the range of the monitor was chosen as ± 40 mils. This range permitted the engraved markings on the bands to be spaced at 1/8 in so that the total number is over 800 per band. The frequency at which the lines pass a microscope axis is over 10 kHz so that a torsional vibration frequency of 1 kHz can be measured.

After the bands were aligned, the microscope eyepieces were replaced by the slit and photomultiplier systems shown in Fig. 1. One microscope position was then shifted an appropriate distance, and the amount of illumination was adjusted to give the best signal-to-noise ratio from the photomultipliers.

The signal from each photomultiplier is amplified and shaped so that one pulse is produced by the attached single-shot pulse generator each time the leading edge of an engraved mark passes the microscope axis.

A signal from BAND A in Fig. 1 sets the flip-flop, opens the gate, and starts the counter. A signal from BAND B then clears the flip-flop, stops the count, transfers the number in the counter into the register, and resets the counter. In this way, the result of one measurement remains in the register until the next measurement is completed.

The counter, register, and d-to-a converter operate in the one's complement number system. The reset of the counter is to negative full scale so that a normal count produces a zero number in the counter and register when there is no strain in the shaft.

The output of the d-to-a converter is displayed on an oscilloscope. The variations in strain, as a function of time, are available in graphical form for the powerhouse operator to study. The output of the d-to-a converter is also connected to comparators that operate the alarm and shutdown circuits.
Results

The ZGS ring magnet is operated so that the beginning of an accelerating cycle always occurs at a fixed speed of the generator shaft. Under these conditions, we may substitute a fixed-frequency clock for the variable one as shown in Fig. 1. The maximum error in strain thus introduced is slightly less than 1 mil. There is no significant effect on the measurements of torsional vibrations.

Fig. 2

Figure 2 shows the torsional strains and vibrations for a normal ZGS cycle with a 750-ms flattop. The left part shows the increase in strain during the accelerating portion of the ZGS pulse. The center section shows the smaller strain associated with flattop, while the right-hand portion shows the strain in the reverse direction during invert. The horizontal scale is 0.2 s/division, and the vertical is approximately 10 mils/division.

The acceleration and flattop parts of Fig. 2 show that an equilibrium level of torsional vibrations of 8 mils peak-to-peak is maintained during that time. Beats between the several shaft torsional vibration frequencies are apparent.

The invert portion of Fig. 2 shows larger amplitude torsional vibrations and includes higher frequency components. This is the result of pulsating torques from the generators due to the very distorted waveforms during the invert part of the cycle.

Fig. 3

Errors in the locations of the etched lines on the shaft can be expected to produce variations in the d-to-a output that look like torsional vibrations. Shaft wobble due to one or more out-of-balance rotating parts can be expected to produce similar variations of d-to-a output. Both of these sources will produce an apparent torsional vibration at the shaft rotation frequency.

The oscilloscope trace shown in Fig. 3 was taken in an attempt to evaluate these sources of error. A low-pass filter was inserted between the d-to-a output and the oscilloscope to remove most of the frequencies higher than the rotation frequency. The vertical sensitivity is ~ 7 mils/division, and the sweep speed is 0.2 s/division. The flattop is 150 ms. This trace indicates that an apparent torsional vibration of about 3 mils peak-to-peak amplitude may be due to these sources of error.

Fig. 4

Figure 4 shows in more detail the torsional strains during last part of the acceleration and first part of the flattop. The horizontal sweep speed is 50 ms/division, while the vertical sensitivity is ~ 10 mils/division. The top horizontal line is the reference voltage for a comparator. The bottom trace is the output of one of the alarms and is activated when the strain momentarily exceeds the reference.

Conclusions

The torsional strain and vibration monitors are used routinely by the Powerhouse operators when a new magnet program is being set up. The monitor display is studied during the time that program adjustments are being made in order to avoid exciting a shaft resonance. Routine rechecks ensure that conditions have not changed.

Observations with the monitors indicate that the damping capacity of the flywheel-motor-generator system during normal operation is larger than that expected from the internal friction in the steel shaft. The eddy current losses and the action of the amortisseur windings on the generator are effective sources of damping of the shaft torsional vibrations.

During motor startup, the monitor can provide data on the magnitude of the pulsations of torque from the motor. These pulses, which are produced by unbalances between the phases of the motor, sweep through all of the natural torsional vibration frequencies of the rotating system. The amplitudes of the vibrations produced indicate a satisfactory or unsatisfactory balance of the currents in the motor phases.