Title

BEVATRON MAGNET POWER SUPPLY TRANSIENT STRESSES IN THE BEVATRON 50,000-kva MOTOR GENERATOR

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INTRODUCTION

The UCRL Bevatron magnet is energized by two Westinghouse 50,000-kva motor generator sets. The drive shafts on these motor-generator sets are 27 feet long and vary from 20 to 40 inches in diameter between the flywheel and generator. The torsional vibration in the shafts has been investigated empirically in 1952 and theoretically in this report.

In October 1952, a Geiger torsiograph was used to measure the torsional vibration of the shaft of the west motor-generator with delayed inversion and with immediate inversion in the current pulse. The torsiograph was about 20 years old, and there were a lot of background vibrations requiring deductive interpretation of the records, so not too much confidence was placed on the test results. The purchase of a new torsiograph was considered but was decided against because of the high cost. The torsiograph results, reported on 10-4-52, stated that the peak torsional vibration stress was 6,000 psi with delayed inversion and 12,000 psi with immediate inversion, with a probable error of ± 6,000 psi. The test results were furnished to Westinghouse at their request.

The torsional vibration stresses were further investigated in June of 1954, since the east motor-generator set at the time was disassembled to repair damaged windings. The problem investigated on a theoretical basis, and the results are presented in this report. These calculations show that the transient torsional stress reaches 9,000 psi. This 9,000 psi happens to be the average of the torsiograph measurements for the delayed inversion and the immediate inversion current pulses. These calculations indicate that the Geiger torsiograph records were correctly interpreted in October 1952, and the agreement between the measurements and the calculations is good, particularly in view of the shortcomings in the measuring equipment used.

These calculations are limited to torsional considerations alone and do not take into account the bending stresses that might be caused if misalignments were to occur at the couplings, or other bending stresses, or stress concentrations at corners where diameter changes occur. Periodic inspection of the shaft with Xyglow equipment has been proposed and inspections have been made.

DESCRIPTION OF THE TRANSIENT STRESSES

Consider the motor-generator shaft rotating at 900 rpm with no load on the generator. Then an electrical load is put on the generator, increasing gradually from zero to 50 megawatts in 1.5 seconds. This load causes the shaft to slow down to, say, 880 rpm, and the shaft is twisted with the generator end lagging the flywheel end. As soon as the load reaches 50 megawatts, the voltage is reversed, causing the generator to act as a motor driven by power which decreases linearly from 50 megawatts to zero in 1.5 seconds. This reversal of power causes the shaft to twist in the opposite direction, with the generator end leading the flywheel end; the shaft speeds up to, say, 920 rpm; and torsional vibrations are set up. The problem is to determine the peak transient shear stress in the shaft immediately following the reversal in the direction of the power. The above relationships are shown diagramatically in Figs. 1 and 2.

CALCULATED STATIC STRESS FOR 50 MEGAWATTS

As the load is increased from zero to 50 megawatts, the torque on the shaft is increased to 390,000 ft-lb, which represents 3,000 psi shear stress.

$$S_2 = \frac{16}{\pi} \frac{T_2}{D^3} \quad = \text{stress when} \quad \theta = \theta_2 \text{ (see Fig. 1)}$$

D = 20 inches = min. shaft dia. between flywheel and generator

$$T_2 = \frac{P_2}{\omega}; \quad P_2 = 50 \times 10^6 \text{ watts} = 36.7 \times 10^6 \ \frac{\text{ft-lb}}{\text{sec}}$$

$$\omega = 900 \text{ rpm} = 94.2 \text{ radians/sec.}$$

$$T_2 = \frac{36.7 \times 10^6}{94.2} = 390,000 \text{ ft-lb} = 4.7 \times 10^6 \text{ in lb torque}$$

$$S_2 = \frac{16}{\pi} \times \frac{4.7 \times 10^6}{(20)^3} = \underline{\underline{3,000}} \text{ psi} = \text{static stress due to 50 megawatts at 900 rpm}$$

This stress would be 3,200 for 53 megawatts, corresponding to Pulse No. 8018 for which torsiograph measurements were taken in October 1952.

CALCULATED PEAK TRANSIENT STRESS

The shaft system is equivalent to a torsion-spring system, shown in Fig. 3, subjected to the following torques:

1. Inertia torque = $I \frac{d^2\theta}{dt^2}$, caused by the rotating masses.

2. External torque $K\theta_2$ = 390,000 ft-lb, which is the steady-state torque caused by 50 megawatts from the Bevatron magnet driving the generator as a motor.

3. Internal torque $K\theta$, which is the resisting torque of the shaft.

The summation of these torques equals zero, leading to the equation.

$$I \frac{d^2\theta}{dt^2} + K\theta + K\theta_2 = 0. \qquad (1)$$

This equation has the solution

$$\theta = C_1 \sin t \sqrt{\frac{K}{I}} + C_2 \cos t \sqrt{\frac{K}{I}} - \theta_2 . \qquad (2)$$

When

$$t = 0, \ \theta = \theta_2 \text{ and } \frac{d\theta}{dt} = 0, \text{ so that } C_2 = 2\theta_2 \text{ and } C_1 = 0$$

equation (2) becomes

$$\theta = 2\theta_2 \cos t\sqrt{\frac{K}{I}} - \theta_2.$$

When

$$\theta = \theta_3, \ \frac{d\theta}{dt} = 0; \text{ that is}$$

$$\frac{d\theta}{dt} = -2\theta_2\sqrt{\frac{K}{I}} \sin t\sqrt{\frac{K}{I}} = 0, \quad \text{and}$$

$$t\sqrt{\frac{K}{I}} = \pi; \quad t_3 = \frac{\pi}{\sqrt{K/I}} .$$

Consequently

$$\frac{\theta_3}{\theta_2} = -3.$$

Finally,

$$S_3 = \frac{\theta_3}{\theta_2} S_2 = 3 \times 3,000 = \underline{\underline{9,000 \text{ psi}}} \text{ for 50 megawatts,}$$

$$= 3 \times 3,200 = \underline{\underline{9,600 \text{ psi}}} \text{ for 53 megawatts (Pulse 8018).}$$

Thus the peak transient shear stress S_3 in the generator shaft is 9,000 psi for the rated 50 megawatts power. This shear stress is based on torsional considerations only, and it would be increased by any misalignment at the shaft couplings and by stress concentrations at corners where diameter changes occur.

This work was done under the auspices of the U. S. Atomic Energy Commission.

FIGURE CAPTIONS

Fig. 1. Relationship of current, volts, power, and angular twist.

Fig. 2. Angular twist of shaft = θ.

Fig. 3. Equivalent torsion spring system.

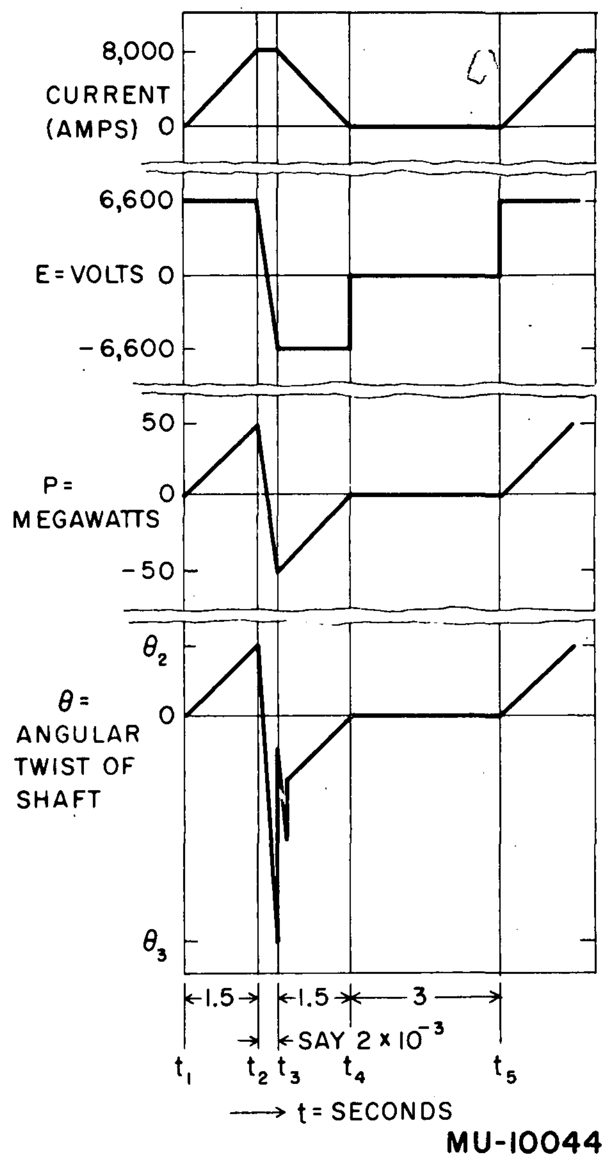

8,000
CURRENT
(AMPS) 0
6,600
E = VOLTS 0
- 6,600
50
P =
MEGAWATTS
0
-50
θ_2
θ =
ANGULAR
TWIST OF
SHAFT
0
θ_3
1.5
1.5
3
SAY 2×10^{-3}
t_1
t_2
t_3
t_4
t_5
t = SECONDS
MU-10044

Fig. 1

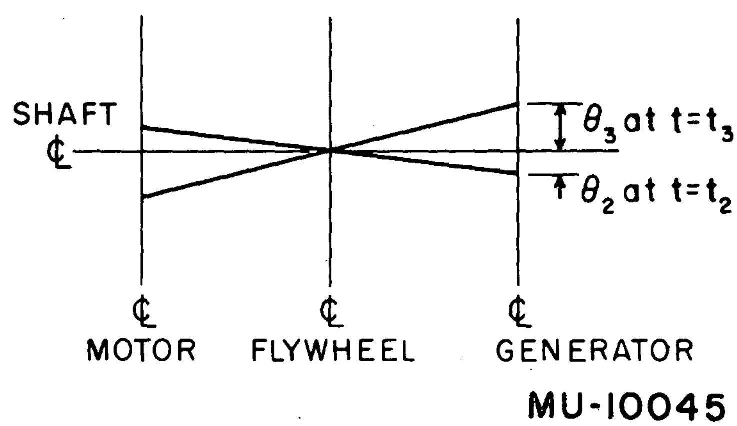
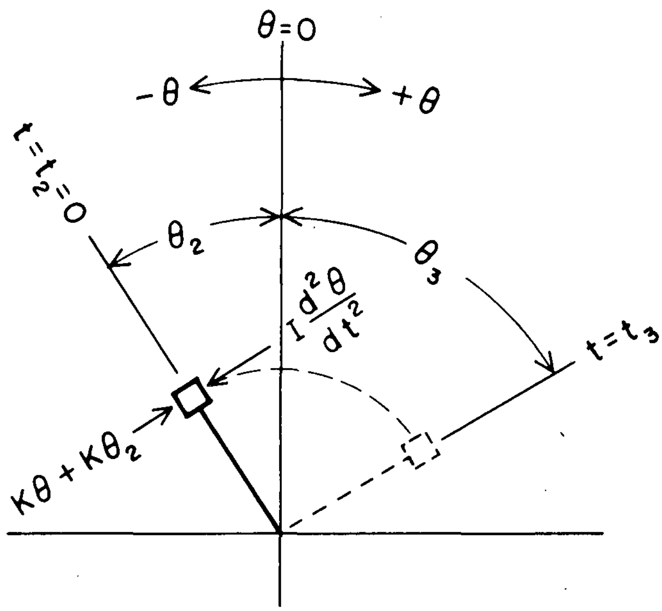


Fig. 2



K = SPRING CONSTANT OF SHAFT

θ = ANGULAR TWIST OF SHAFT

I = MOMENT OF INERTIA OF GENERATOR

t = TIME STARTING FROM $t_2 = 0$

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Fig. 3