THE MECHANICAL DESIGN OF A COMPENSATED PULSED ALTERNATOR PROTOTYPE

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Abstract

A prototype of a compensated pulsed alternator (compulsator) is presently under construction at the Center for Electromechanics (CEM) of The University of Texas at Austin. The unique machine configuration and peak output current (150 kA) generate large forces not typically seen by conventional rotating machines. The rotor is made of 2913 laminations shrink fitted on a vertical shaft. Since the rotor has an L/D of 3.2 and a maximum speed of 5400 rpm, these insulated laminations are clamped on the ends with large Belleville washers to increase the effective stiffness. The stator is mounted on a torque frame which allows it to rotate during discharge to reduce the forces transmitted to ground. The mechanical considerations and design of this machine are presented.

Introduction

The compulsator is a rotating energy storage device which provides high-voltage, high-current pulses by utilizing the principles of magnetic induction and flux compression. Although initially invented to power the flashlamps used in the Shiva Nova laser fusion facility at the Lawrence Livermore Laboratory, the compulsator is also presently under study as a power supply for other applications requiring compact, high-energy, high-power repetitive or single pulses. The engineering prototype compulsator under construction is a one-half scale model of one of the machines to be used in the Shiva Nova laser facility.

This paper presents the mechanical design. For details of the principal of operation, electrical design, and armature winding design, see references 1, 2, and 3 respectively.

Design

Figure 1 shows a cut-away isometric drawing of the compulsator without external connections such as oil lines and pumps for the bearings, motor-drive system, field coil connections and power supply, and output connectors to the flashlamp load. The three major mechanical components discussed are: 1) rotor, 2) back iron and 3) torque frame.

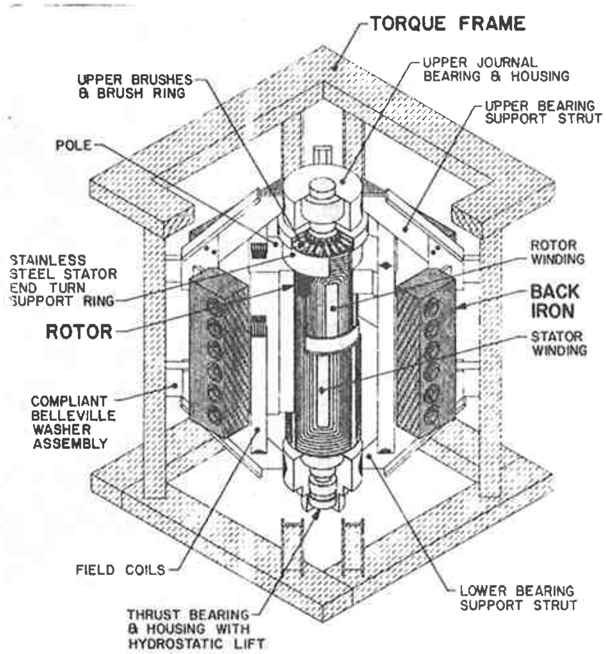


Figure 1. Cutaway Isometric of Compulsator

1. Rotor

In order to reduce eddy current losses, the rotor is made of 2913 steel (M-19) laminations, 0.036 cm (0.014 in) thick, 38.1 cm (15 in) in diameter and shrink fitted on to an AISI 4340 steel shaft heat treated to R_C 34. Because the nominal shaft diameter of 9.65 cm (3.8 in) is insufficient to keep the first rotor critical above the maximum operating speed of 5400 rpm, it is necessary to compress the laminations in order to increase the effective flexural modulus of the rotor, hence the rotor stiffness. The rotor cannot be allowed to pass through a critical because hysteritic losses from sliding of the lamination interfaces would result in a rotor instability[4]. The lamination preload is applied with two (one per end) large, titanium Belleville washers, 6.03 cm (2.375 in) thick and 38.1 cm (15 in) in diameter. Because of the Belleville washer configuration, the preload of $2.67x10^6$N (600,000 lb) preferentially loads the outside diameter of the laminations although the washer will be flattened to partially load the inner diameter also.

The required preload was not arbitrarily selected, but resulted from a series of tests performed on sample stacks of laminations. The two guiding design criteria were the interlaminar resistance and effective modulus versus load. With increasing load, the effective flexural modulus of the stack of laminations increases as the interlaminar resistance decreases.

The desired lamination core plating, C-5, could not be obtained on the schedule required for construction of the prototype. The plating received, C-0, was unacceptable and required that an interlaminar insulation be applied that could take the high loads. The first type of insulation tested was Sterling U-87/PS, an air dry varnish.

Figure 2 shows the measured overall resistance versus load of two separate stacks of laminations. The bottom curve is the test results of a stack of 1000 varnished laminations. Even as the load was held steady, the resistance continued to drop and at 1.33×10^6N (300,000 lb), measured 182 ohms. This was unacceptable since a value of 782 ohms was desired in order to keep the eddy current losses at an acceptable level.

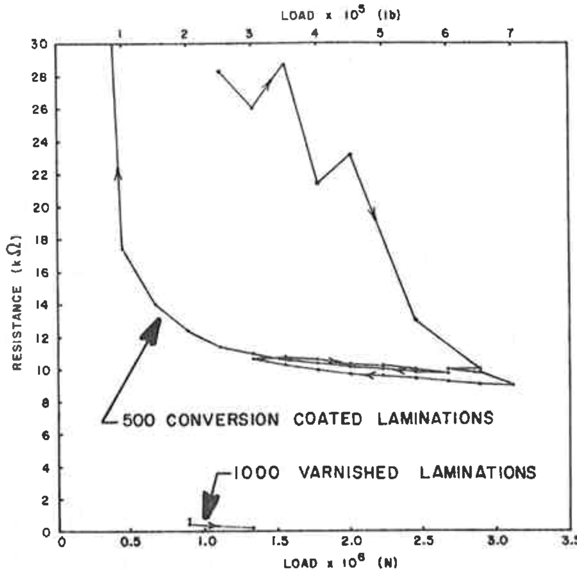


Figure 2. Resistance vs Load

The next insulation tested and finally used was a military refinish concentrate made by Atlanta Cutlery. It is a chemical conversion coating which phosphatizes the surface. The varnish previously applied was baked off the laminations before the chemical conversion coating was applied. The resistance versus load of a stack of 500 coated laminations is also shown in Figure 2 and at 3.11×10^6N (700,000 lb) measured 9,040 ohms, over an order of magnitude above the design goal of 391 ohms. Note that even as the load was fluctuated, the resistance remained relatively stable and repeatable. The resistance did register lower after the load was decreased than when the load was initially applied. This is probably due to an increasing number of small asperities breaking through the insulation as the load is increased and then remaining in contact with the adjacent lamination as the load is decreased.

During the same test with the chemical conversion coated laminations, the amount of axial compression versus load was measured and is shown in Figure 3a. When the stack is initially compressed, the total deflection is significant. This is a result of the air being squeezed out from between each lamination, small asperities being flattened, and any warpage in the lamination being flattened. When the same stack is then compressed a second time, even after sitting unloaded for one day, the total deflection is considerably less. The interesting fact is that the slope of the curve as the load is decreased is the same, indicating that after the stack is loaded, its mechanical characteristics are repeatable.

Figure 3b is an enlarged section of the first loading curve and shows some other interesting facts.

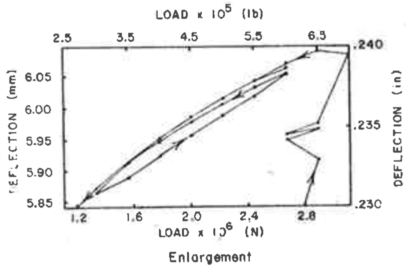


Figure 3b

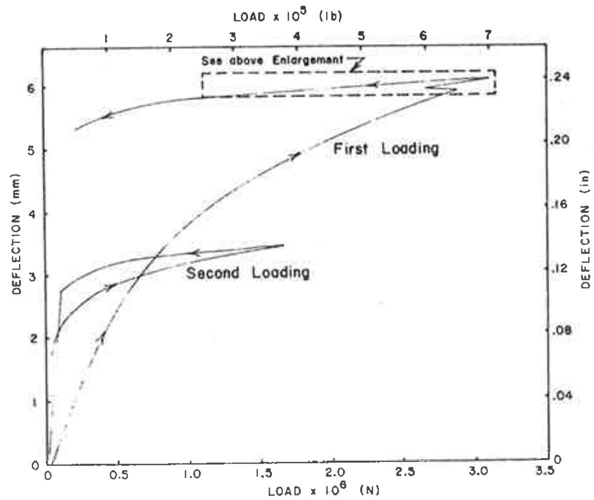


Figure 3a

Deflection vs Load

First note that as the stack is brought to full load, at two points the load is reduced and the stack still shows an increase in deflection. This was noticed in other tests not presented here and is due to the amount of time the stack is allowed to sit at load before the deflection measurement is made. If the load is held steady, the stack will continue to compress for many minutes. It is suspected this time element is a result of the air being squeezed out from between the laminations. Another important fact is that after the stack stops creeping and the load is fluctuated, the stack does not load and unload along the same curve, indicating some hysteresis in the stack.

Since the resistance remained high for all the loads tested, the level of load to be used is determined by mechanical limitations. The Belleville washers, which apply the preload, are held in place with two large nuts which also serve as the bearing journals. These nuts will be tightened using a stud tensioner loaned to CEM by the DuPont chemical processing plant in Victoria, Texas. The device works by stretching the stud (in our case, the shaft) and then "hand tightening" the nut down against the Belleville washer. The device has a 3.56×10^6N (800,000 lb) pulling capacity which produces the maximum stresses the modified 60° stub-tooth Acme threads and shaft can take. Due to relaxation in the threads as the load is transferred from the stud tensioner to the nut, the resulting preload will be less although the minimum desired is 2.67×10^6N (600,000 lb).

Clamped up sections of laminations 22.9 cm (9 in) long were bored with a taper of 6.86×10^{-3} cm (0.0027 in) on the diameter and the shaft then ground to match. After the application of the lamination insulation, the inner bore of the laminations were aligned for the shrink fit by pulling one lamination at a time up against two small ground shafts glued together and inserted down the bore. The entire stack was then clamped as tightly as possible without affecting the bore alignment. The alignment was checked by lowering the shaft in at room temperature until the tapers matched. The final shrink fit was done by chilling the shaft in liquid nitrogen and then dropping it into the laminations. Figure 4 is a picture of the rotor with the Belleville washers and nuts in place and the back iron in the background.

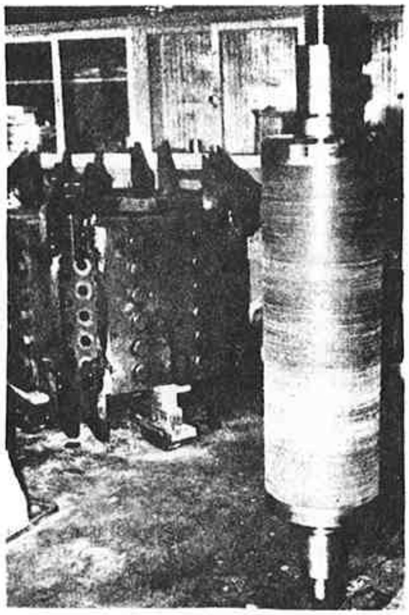


Figure 4. Rotor and Back Iron

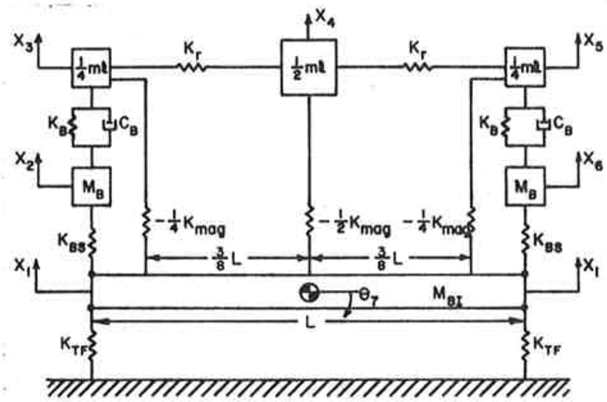
The inverse of the slope of the curve in Figure 3 can be considered an effective Young's Modulus, E_{eff}, of the stack of laminations in compression. Using the data obtained on the final unloading in Figure 3b, $E_{eff} = 1.88 \times 10^{10}$ MPa (2.72×10^{6} psi). Although what is actually desired is the flexural modulus, it could not be measured and the above E_{eff} should be sufficient for the dynamic calculations.

A discrete, lumped mass model of the rotor-bearing-support system (see Figure 5) was performed using a CDC 6600 computer to solve for the complex eigenvectors and complex eigenvalues. The torque frame, bearing supports, bearings (including damping), and rotor are included in the model. The first rotor critical is calculated to be 621 rad/sec, 10% above maximum operating speed.

The radial bearings for the machine are tilting pad, oil lubricated, hydrodynamic bearings made by Kingsbury, Inc. A special design feature incorporated into the bearing is spherical buttons on the back of the pads which allow for axial misalignment or cocking of the shaft. Each bearing is instrumented with a resistance temperature detector (RTD) embedded in the babbit to monitor pad temperatures.

The thrust bearing is a two sided, self aligning, tilting pad, hydrodynamic bearing made by Kingsbury, Inc. Each side is instrumented with a RTD and the loaded side has two load cells to measure steady state and dynamic loads.

Since this is an experimental machine and will be started and stopped many times, a high pressure oil inlet at the end of the shaft is used to lift the machine off the thrust bearing pads at zero speed to avoid excessive wear of the pads. This will only be used during start up.



K_{TF} = TORQUE FRAME STIFFNESS = 6.04×10^{7} N/M

K_{BS} = BEARING SUPPORT STIFFNESS = 1.29×10^{9} N/M

K_B = BEARING STIFFNESS = VARIED

C_B = BEARING DAMPING = VARIED

E_{EFF} = EFFECTIVE YOUNG'S MODULUS OF ROTOR = VARIED

K_R = ROTOR STIFFNESS; INCLUDES SHEAR DEFLECTION; DEPENDENT UPON E_{EFF} = VARIED

K_{MAG} = MAGNETIC SPRING STIFFNESS = 3.97×10^{7} N/M

ML = MASS OF ROTOR = 1.09×10^{3} KG

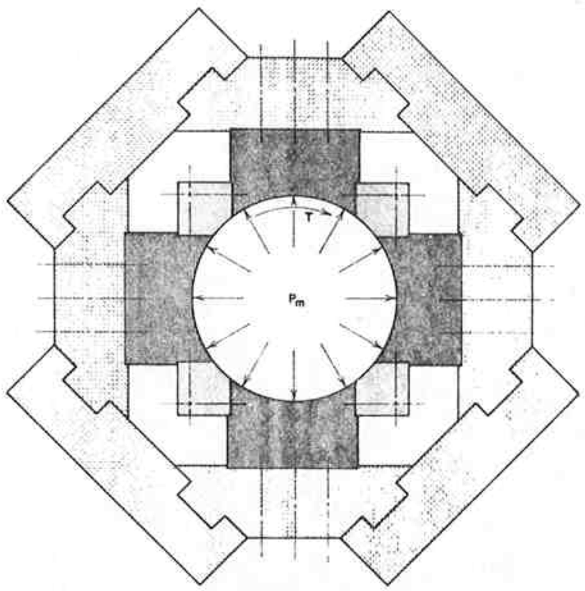
M_{BI} = MASS OF BACK IRON = 9.07×10^{3} KG

M_B = MASS OF BEARING HOUSING = 230 KG

Figure 5. Lumped Mass Dynamic Model

2. Back Iron

During a discharge, the back iron must withstand two forces generated from J X B forces and flux compression. These forces appear as a torque and an internal pressure applied at the inner diameter of the back iron where the stator conductors are located (see Figure 6). The back iron is designed to withstand these forces under steady state conditions for a peak fault current of 150 kA although this peak current only exists instantaneously during the pulse.



DISCHARGE FORCES: (FAULT MODE, 5400 RPM, 150 KA)
T = PEAK DISCHARGE TORQUE = 2.7×10^6 N-M
P_m = PEAK MAGNETIC PRESSURE = 20.7 MPa (3000 psi)

Figure 6. Back Iron

The reaction torque of 2.70×10^6 N-m (1.99×10^6 ft-lb) and internal pressure of 20.7 MPa (3000 psi) must be sustained with no relative movement of the pieces. The stator conductor is epoxied to the inner diameter[3] of the back iron and any slippage of a back iron member could initiate a crack in the epoxy. The ideal geometry for these forces would be a cylindrical vessel, but a casting could not be obtained in time. An irregular octagonal structure made of 16.5 cm (6.5 in) plate as seen in Figure 6 evolved with the sides interlocked with closely toleranced keys and slots. This allowed for most of the load to be taken by the keys in shear.

Another significant design feature of the back iron and poles is that they do not extend the length of the rotor. If this had been done, the axial forces on the end turns at each end of the rotor from the applied field and the current in the conductors would be very large and any asymmetry in the field would pull the rotor to one end and overload the thrust bearing. Therefore, the back iron only extends the active length of the conductors and the stator end turns are supported by stainless steel rings bolted to the end of the poles. Figure 7 shows the back iron supported in the torque frame.

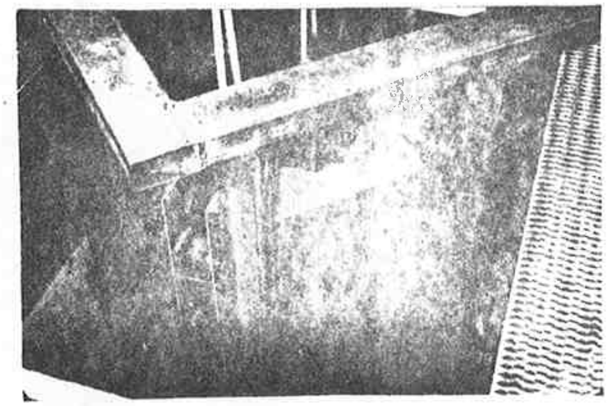


Figure 7. Back Iron In Torque Frame

3. Torque Frame

The torque frame is a structure designed to support the compulsator and allow a slight rotation of the back iron during a discharge (see Figures 1, 5 and 7). By allowing the back iron to rotate, the total peak load transferred to the torque frame as a result of the discharge torque is reduced from 6.82×10^6 N (1.53×10^6 lb) to 1.15×10^5 N (2.56×10^4 lb). As the back iron rotates, it compresses Belleville washers against I beams which form the structure of the torque frame. The springs are essentially serving as force attenuators. There are two sets of Belleville washers being compressed at each of the four corners of the torque frame located at a radius of 79.4 cm (31.3 in) from the center of the compulsator. The back iron is allowed to rotate 0.00733 radians, compressing the Belleville springs 0.582 cm (0.23 in). The torque frame is constructed of eight I beams (6I12.5), two per corner, which are connected at the top and bottom by a square formed from rectangular tubing. In addition to resisting the discharge torque, the frame must support the mass of the compulsator, approximately 9.07×10^3 kg.

Acknowledgements

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