Compulsator Rotor Assembly Method

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Abstract —High energy density requirements for fieldable electric gun applications have led to the development of air-core compulsators which utilize hybrid metal/composite rotor designs. In addition to supporting the required generator windings, these rotors also must store large amounts of kinetic energy. To satisfy energy density requirements, the tip speeds of the electric gun class compulsator rotor are appreciable with typical values exceeding 500 m/s for near term designs. High performance composites are utilized in the rotor structures and large amounts of radial pre-load is required to hold the rotor structure and electrical windings together at the tip speeds required.

This paper discusses the methodology, analyses, and tools required to successfully assemble and pre-load composite rotor structures using tapered press fits. Analyses and pre-assembly predictions are supported by data collected during composite rotor fabrication at The University of Texas at Austin Center for Electromechanics (UT-CEM).

INTRODUCTION

The Center for Electromechanics at The University of Texas at Austin (UT-CEM) has developed several generations of compulsators since the late 1970s [1,2,3,4,5,6]. The potential defense-based use of electric gun technology has resulted in an ongoing high power density air-core compulsator development effort that was initiated in the early 1980s [4,5,6]. The air-core designs feature high strength composite rotors with tip speeds operating in excess of 500 m/s in order to achieve desired specific energy and power densities. To function reliably, these rotors must respond as dependably as low speed steel rotors. Sudden balance shifts or significant non-linearity in the rotor composite materials would make the application fundamentally impractical. To realize the goal of a robust composite structure, the rotors utilize specially developed composite bands which are radially pre-loaded together via tapered interference press fits to form the final product. Properly engineered interferences enable the radial pre-load to be built into the rotors which enables them to remain stable at high speeds. For high speed composite rotors carrying windings, these pre-loads can exceed 5,000 psi. At the same time, it is most desirable to maintain the highest possible electromagnetic coupling between the rotor and armature windings. These factors together result in the requirement of very low taper angles (2° or less included angle), which results in long press distances required to assembly rotor rings.

ROTOR ASSEMBLY METHODOLOGY

Fig. 1 is a layout of a composite rotor assembly showing the four primary hardware components required for Class II tapered press fits. It should be pointed out that for Class II assemblies, each piece of hardware is engineered specifically for the ring being assembled. Table 1 describes the critical hardware components.

To prepare for a press fit, the first step is to determine the press load and geometry requirements of the hardware to be assembled. Initial estimates of the load requirements are typically based on 1D nested ring analyses while the final requirements are based on intensive finite element analyses of the entire rotor structure. UT-CEM uses two in-house developed assembly presses. The smaller press has a 48-in. stroke and 200-ton capacity and will accommodate up to a 36-in. diameter rotor up to 10 feet long. The larger press has a 48 in.-stroke, 560-ton capacity and will support rotors 144 in. in diameter up to 28 feet long.

After the composite ring sizes are determined, the required hardware is engineered and fabricated. Engineering analyses required includes 1D nested ring analysis and 2D axisymmetric FEA. After the hardware has been engineered, it is fabricated except for final machining of the lead-in ring, the inside of the push ring, and the inside of the compression ring (if used), respectively. These diameters are established after the banding and rotor dimensions have been physically acquired and measured. Prior to final set-up under the press, strain gages are applied to the banding and any tooling as desired. Strain gage data collected during the assembly is critical in confirming that designed pre-load levels are achieved. It is also used in determining the as-built characteristics of the rotor.

The completed banding, tooling, and rotor are first set-up under the press to perform a ‘dry fit’. Dry fit simply means that no epoxy is used while the components are pre-tested for correct fit. After all dimensions are validated, the components are disassembled and the surfaces of the rotor, lead-in ring and banding are coated completely with the epoxy. Application of epoxy is critical to lower the friction and help prevent damage between mating surfaces. The epoxy is a room 2-part temperature grade which allows about 1 hr of working time after the components are mixed. All components are then reassembled under the press and the platen is lowered into position above the press ring. At the onset of the press cycle, the data acquisition system is started and the banding is pressed onto the rotor. The press platen is controlled manually, and care must taken by the operator particularly during the last few inches of the fit; once the installed ring bottoms out, the press load can rise suddenly and damage the installed banding. Data acquired includes banding tangential and axial strains, press platen travel distance, and hydraulic ram pressure force.

UT-CEM has been developing and successfully using the tapered interference press fit method to assemble composite rotors for the past ten years. An example of a rotor assembled with this method is seen in Fig. 2. Fig. 3 is a chart which plots the various composite ring assemblies conducted at UT-CEM over the past decade.

In Fig. 3, radial pressure represents the peak installed interference pressure for an installed ring. Push distance ratio (PDR) is the ratio of the required press distance to the ring length. For the low taper angles and radial pre-load representative of CPA rotors, this value can easily exceed 1.0. Experience indicates that radial pressure versus percent push is a good way to represent overall difficulty for a given assembly operation. Also, these parameters may be used to separate the assemblies into classes. In general, assemblies below 2,500 psi peak radial interference pressure and 0.75 PDR are relatively easy to accomplish (Class I) and require only hand calculations and/or the use of the 1D nested ring analysis code to execute reliably. Fits exceeding 2,000 psi interference and approaching or exceeding 1.0 PDR are considered to be in the difficult range (Class II) and require more thorough engineering utilizing the finite element method. Of course, these are general guidelines, and each assembly operation must be evaluated individually.

ROTOR ASSEMBLY PHYSICS

It is important to consider that since a liquid epoxy film is used as a lubricant during the assembly process, the radial interference pressure results in a pressurized hydraulic boundary layer separating the installed banding from the rotor assembly. This epoxy boundary layer is usually a few mils thick and has been measured in several tapered press fit experiments. Installation of a tapered interference fit banding requires an applied force which exceeds the naturally occurring resisting forces. A diagram of these forces is shown in Fig. 4, where F_n represents the trapped epoxy hydraulic force which opposes ring axial motion against the taper and F_f is the resisting friction force. Assuming constant friction, resolving forces in the vertical direction produces the press force equation:

$$F_p = P_r * A * (\sin(\phi) + \mu * \cos(\phi)) \tag{1}$$

where,

	F_p	=	Applied press force
	P_r	=	Radial Interference Pressure
	A	=	Surface area of the banding bore
	μ	=	Friction coefficient
	Ø	=	Banding taper (half-angle)

From the geometry, the push distance is given by,

$$\text{Push Distance} = \frac{\delta}{\tan(\phi)} \quad \text{where; d = radial interference} \tag{2}$$

The value P_r represents the calculated value of radial pressure which varies linearly with the interference. Two components of (1) are worthy of note. First, the friction coefficient has been seen to vary somewhat during the installation procedure, especially during heavier interference fits. This is seen and discussed in the case histories below, and is presently a topic for research on its own. Second are the stick-slip components of (1). Whenever the μ cos(Ø) term is greater than the sin(Ø) term, the ring will not tend to slide back off the taper once the press load is removed. This tendency arises from the liquid epoxy hydrostatic fluid pressure in the banding interface. Extremely low values of friction require extra tooling to lock the platen down against the banding until the epoxy cures.

ASSEMBLY ANALYSES

As stated above, Class II assemblies generally require thorough engineering to ensure high success rates. Two dimensional axisymmetric Finite Element Analysis (FEA) is employed along with the 1D nested ring analyses on the rotor, installed band, and critical tooling components. The nested ring code is used to determine radial and hoop deflections and stresses that are expected in the installed band, rotor, and lead-in ring assemblies. The target lead-in ring design is one which matches the anticipated deflections of the rotor at the lead-in ring interface. As a ring is applied to the rotor, the rotor will deflect radially inward, and the presence of the taper makes this a gradual process. It is important that the lead-in ring remains slightly larger in diameter (a few mils, typically) than the start of the rotor. If the lead-in ring deflects too much, the installed band will be forced to rise over the rotor end-face and both rotor and applied band then become extremely susceptible to damage. Interface damage can be inferred from high apparent friction coefficients during the installation. Failed assembly attempts have generally resulted in noticeable damage to the ring being installed, and experience suggests that friction coefficients encountered in excess of 0.1 indicate a strong possibility of damage.

Past experience and recent detail design studies have shown that the last few inches of Class II press fits can be particularly stressful for the ring being installed. As the applied ring approaches its installed length on the rotor, less and less of the lead-in ring is under load from the ring. This results in the lead-in ring's tendency to spring open, resulting in excessive radial stresses in the applied banding. Also, longer lead-in rings tend to have bending deformations which are not accounted for in the 1D nested ring analysis. An example of this behavior is shown in Fig. 5 which plots an exaggerated deformed plot of a banding being applied to a rotor with 1 in. remaining to be installed. In this figure, lead-in ring deformations are seen along with noticeably high deformations in the end regions of the rotor banding as was discussed above.

CASE HISTORIES

Three recent cases of rotor banding assemblies are now presented. Cases 1 and 2 study a test and actual installation of a heavy interference fit banding onto a compulsator rotor assembled in 1993. Case 3 studies the procedure for a rotor assembled in 1997. Table 2 presents the physical data for these cases.

As seen in table 2 above and Fig. 1, cases 1 and 2 were considered Class II assemblies. Case 1 was designed as a practice attempt and utilized near full scale components and tooling, but not the real rotor used for case 2. Fig. 6 shows plots of predicted press loads based on assumed friction coefficients. Since many of the previous press fits recorded at UT-CEM tended to display a varying friction coefficient, two force curves were projected based on the relationship:

$$F_P = P_R A(\sin(\varnothing) + ae^{bx}\cos(\varnothing)) \tag{3}$$

where the friction coefficient is represented by the exponential function;

$$\mu = ae^{bx} \tag{4}$$

The constants for (4) were evaluated at an assumed low starting value of 0.005, and ending values of 0.1 and 0.2, respectively. In addition, a third linear curve was created assuming a constant friction coefficient of 0.1. In practice, this curve is important as it is used in real time during the press fit to judge whether or not the fit should be aborted. Essentially, if the real press load curve crosses the constant 0.1 friction curve during a fit, the procedure may be aborted depending on how much of the band is already installed. Of course, abort points must be clearly determined prior to proceeding with the procedure. The 0.1 critical friction value comes from UT-CEM's experience that damage is likely above this level as discussed previously.

For test case 1, a press load of 290 tons was expected for a friction value of 0.1 and about twice this for a friction of 0.2. During the actual event, the 0.1 friction curve was crossed 30 in. into the procedure. No abort occurred as it has been previously decided to abort up to 29 in. engagement. As reduced from the data, the installed ring indicated a friction approaching 0.2 at the end of the installation procedure, and in fact required all the press load capacity to install. Post test inspections revealed heavy damage to the installed band and the surface of the mock rotor.

The results of test case 1 were not very encouraging. However, a thorough 2D FEA revealed some fundamental problems with the stiffness and deflection patterns in the lead-in ring. In addition, subsequent experience showed that a particular woven glass material (produced by Randolph Austin, Inc.) would help to retain wetted epoxy at high pressures and acted to stabilize the friction coefficient. This combination produced the results shown in Fig. 7 for case 2. The case 2 results exhibit a much more constant friction which represents an average value between 0.05 and 0.06. These results indicate that the tapered interference fit process can be controlled and that even more severe pre-loading can likely be achieved using this method if needed.

Finally, case 3 is a more recent composite rotor application which leveraged all past tapered press assembly experience. Fig. 8 shows this hardware as it looked when it was set up beneath the large assembly press at UT-CEM. This assembly was considered to be in the Class II regime at 2,500 psi radial interference and 77% push. As the data shows in Fig. 9, this fit went exceptionally well. A very low and relatively constant friction coefficient of approximately 0.03 was back calculated for this assembly based on the data.

CONCLUSIONS

UT-CEM is developing the tapered press fit assembly method as a viable means of pre-stressing high quality composites to fabricate high performance composite rotors for air-core compulsators. The method may be readily leveraged into other applications as well such as high speed flywheel batteries and composite containment structures and should be amenable to compulsator rotor or composite flywheel production. Successful application of this method requires thorough engineering of the composites and tooling components.

There is still much work to be done. Present research is involved with combined analytical and empirical derivations of suitable friction models to aid in predicting press load requirements. In addition, research is focusing on limits of applicability based on current materials, with the hope of providing material requirements for expanding the method even further.

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Table 1 : Critical hardware components for class II fits

Component	Function
Press Ring	The press platen is applied to this directly. Transmits press load to the banding and provides clearance for the rotor shaft.
Push Ring	Composite interface between press ring and installed banding. Serves to protect banding and even out transmitted axial loads.
Lead-In Ring	Required for long press distances. Guides and expands banding up to engagement diameter of the rotor.
Compression Ring	Moderates deflections of lead-in ring to minimize end effects on banding during latter stages of press fit.

Table 2: Physical properties of the presented tapered press case studies

Case No.	Band ID (in.)	Band Length (in.)	Taper Angle (°)	Radial Interf. (in.)	Radial Press (psi)	% Push
1	27.863	19.5	0.1	0.116	4,000	170
2	25.00	19.0	0.1	0.11	4,000	166
3	22.00	39.0	0.1	0.045	2,500	77

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