CHAPTER 9

CONCLUSION AND FUTURE RESEARCH WORK

9.1 CONCLUSION

This research presented design, modelling, simulation and proto-typing procedures for FRG. Contributions in the major electrical and mechanical characterizations viz., electromagnetic static and dynamic characterizations, Computational Fluid dynamics (CFD), thermal analysis, and vibration analysis were proposed for FRG. In this chapter, a chapter wise conclusion is provided with possible future work.

Chapter 1 and 2, discuss the introduction of FRM and design of three phase 6/8 pole FR generator considered for analysis. The step-by-step procedure for creating model in finite element analysis is discussed. Then, a two- dimensional finite element static analysis of FRG is performed. The basic equations used to perform the magnetic field simulation, the assumptions based on which the formulation is done and setting up of the boundary conditions for the FRG analysis are given. The basic equations involved in the calculation of flux linkages, phase inductance, and torque are discussed. The results of simulations are documented.

Chapter 3 attempts to examine the effect of stator-rotor geometry on the minimization of cogging torque of the FRG. Modifications on the stator geometry and on the rotor geometry have been undertaken. With respect to the stator geometry modification, height (or thickness) of the permanent magnets of the stator poles have been varied and cogging torque minimization has been observed. With respect to the rotor geometry modifications, the following cases have been considered: a) rotor pole arc variation, and b) V-shaped punch on the rotor pole. The characterization of FRG under these cases for the observation of the effects on cogging torque has been done using three-dimensional finite element analysis. From the
results of simulation, the model to be selected for proto-typing in order to have less cogging torque has been decided.

First, the rotor pole arc variation is analyzed from $15.5^\circ$ to $25.5^\circ$. With the increase in the rotor pole arc, the peak to peak flux linkages as well as the peak to peak inductances are reduced, which reduces the reluctance torque of the generator. This reduces the cogging torque. Varying the rotor pole arc from $23.5^\circ$ to $25.5^\circ$ gives the minimum cogging torque, better electromagnetic torque performance and lesser core losses. Based on the analysis, the optimal rotor pole arc is found as $23.5^\circ$.

Proper choice of permanent magnet height results in substantial reduction of cogging torque as well as torque ripple. So, secondly, the generator has been analyzed for different permanent magnet heights varying from 2 mm to 5 mm. For the analyzed generator, the permanent magnet having the height of 22% (2 mm) of stator pole height gives 61% reduction in cogging torque and 14% reduction in torque ripple.

As a third case, V-shaped punch on the rotor pole is analyzed for different rotor hole depths. When reducing the depth of the hole, 84% of cogging torque is reduced and 31% of torque ripple is reduced. Thus the optimal depth of the hole is found that 9% (0.82 mm) of designed rotor pole width.

The optimized variables obtained are simulated in a single FRG machine in which it is found that cogging torque reduces up to 86%; but the torque ripple almost remains the same as in the conventional machine. When comparing all the models, the rotor pole arc of $23.5^\circ$ is more efficient to reduce the cogging torque for this generator. Hence, the experimental verification is carried out for the rotor pole arc $23.5^\circ$ model in the next chapter.

So it is concluded that (i) pole arc optimization can be done to finalize the rotor pole arc for a minimum cogging torque; (ii) Stator PM height can be made to be between 20% to 25% that of the main pole height for a performance of FRG with reduced cogging effects, and (iii) V-shaped...
punches on the either side of rotor pole, with a minimal depth, can contribute reduction in cogging torque.

FRG manufacturers can undertake these studies via simulation in order to finalize the proto-type dimensions.

Chapter 4 presents the dynamic characterization for the optimized rotor pole arc model. A same step-by-step approach given in the chapter 2 is used for performing the dynamic study. Stator coils are connected to an inductive load with an impedance of 172 ohm and the rotor and shaft together selected to make a motion component. The rotor is made to rotate at different speeds (1000-9000 rpm) for a specific time interval. In this simulation, time interval is set as 0-100 ms, with the step time of 0.02 ms. The solver runs the 2D transient with motion analysis for variable speeds. The performance of FRG, including induced voltage, speed vs efficiency characteristics and speed vs power characteristics with any load are studied. Voltage regulation is also addressed. The simulation results are validated with experimental results. The experimental verification is carried out for the rotor pole arc model. The experimental results are obtained and compared with the simulation results. FEA simulation results agree with prototype measured results with 6% tolerance.

Chapter 5 discusses the Computational Fluid Dynamic (CFD) analysis for predicting the air velocity inside the air region of FRG. The knowledge about the air velocity distribution inside electric machines is useful in undertaking various heat analyses. For instance, thermal analysis is fine tuned and a more realistic thermal simulation is made possible, when the air velocity distribution assisted thermal analysis is carried out. A 3-dimensional computational fluid dynamic analysis is used for this purpose. The profile of the turbulently circulating air inside the machine is captured.

From the simulation results it is observed that, at 9000 rpm the surface air velocity is varies from 3.39 m/sec to 6.78 m/sec and the velocity in the mid portion of air region are varies from 1.48 m/sec to 3.39 m/sec.

Even though the inside surface velocity varies between 3.39 m/sec to 6.78 m/sec, this variation is so rapid that it is almost about 6.78
m/sec only. So for the calculation of heat transfer coefficient the air velocity in the surface is fixed at 6.78 m/sec. The pressure inside the air region is 4.629 Pa. This CFD result can be used for through thermal analysis for the next section. For the simulation of thermal analysis, the accurate prediction of heat transfer coefficient is always good, and this is achieved only through the knowledge of air flow distribution.

Chapter 6 presents the prediction of steady state thermal characteristics of FRG using three dimensional finite element analyses. At first, the machine’s, inner surface heat convection coefficients are evaluated. Usually a lumped number is used as a constant velocity in the evaluation of convection coefficients. But practically the velocity is not constant. It varies turbulently and hence assuming it to be a constant will cause a big approximation. To avoid this limitation only, the air velocity at various inner walls is calculated using CFD, which fine tunes the thermal analysis.

The heat transfer co-efficient on the inner surfaces depends on the velocity of air with which they are in contact. This contact velocity varies and hence the Renold’s number (Re) accordingly vary. Reynold’s number is between 15010 - 21793. If Re is above 4000, the whirl of air is considered as turbulent, and hence the flow of air is turbulent in this machine. In the inner surface $h_{cv}$ is between 23.14 W/m$^2$k - 46.51 W/m$^2$k at the respective air velocities. These heat convection ($h_{cv}$) coefficients are set respective as boundary conditions for accurate thermal analysis.

It is to be noted that if the $h_{cv}$ is evaluated at the velocity of the operating speed, it will be a single lumped approximated value; now, as the $h_{cv}$ is found out in every inner wall portions of the machine at the respective varying air velocities, the simulation of heat distribution becomes more accurate and realistic. Static and dynamic heat distribution inside the machine can be effectively calculated now with the CFD results.

The heat flow rate $Q$ and the convection heat transfer co-efficient at the inner surfaces are specified. The ambient temperature was set as 35 degree centigrade and a finite element thermal analysis was carried out at 9000 rpm. The obtained steady state temperature is increased upto
$66^\circ$ C at full load 9000 rpm. Thus the maximum temperature of FRG is about $66^\circ$ C. Since F class insulation is used, the machine can withstand upto $155^\circ$ C, and hence the FRG is in thermal safety region.

To validate the simulation of thermal in the FRG, a practical thermal analysis was carried out on the prototype FRG. The generator temperature has been measured using LM35 temperature sensor. It has temperature range of 0-150$^\circ$ C. This sensor is mounted on the surface of the generator and connected to the LABVIEW tool through Digital to Analog Converter (DAC-USB 6006). From the DAC, the 5 V input is given to the first terminal of sensor. Second terminal is connected to analog terminal of DAC. All updates of analog output channels are software-timed. The third terminal is connected to GND, the ground-reference signal for the analog output channels. The average temperature is measured using LABVIEW tool. The measured no load temperature for 5000 rpm is 34$^\circ$C and for 9000 rpm $50^\circ$ C. These are in a safer ambient temperature limits.

Thermal analysis was also performed, both in simulation and a prototype, under loaded conditions. The generator is connected to the load and temperature is measured at a constant speed of 5000 rpm. The load temperature reaches 40$^\circ$ C after 3min and it reaches 55$^\circ$ C after 30 min. This means that the temperature rise does not affect the machine rating during continuous duty. The simulation results are good agreement with the experimental results.

Chapter 7 presents the three dimensional vibration analysis of FRG. The attraction and repulsion cycle caused by permanent magnets and salient rotor during the energy transformation leads to vibration and noise in the FRM, which can become destructive. To identify the frequencies of vibration which can cause the severe vibrations, a vibration analysis has to be done. The results of vibration analysis will list the frequencies (or speeds) to be skipped to prevent any structural damage which occurs when the frequency of the machine coincides with the natural frequency. This is called the modal study and this will ensure a quiet operation of the machine.

From the modal analysis, it is found that the vibration spreads
fully over the housing in three directions viz., horizontal, axial and radial at the frequency of 1488.62 Hz, 1568.78 Hz, 4119.94 Hz. At a modal frequency of 3372.43 Hz, the rotor and shaft undergo bending and arresting of the housing, which produce high acoustic noise. The severe shaft deformation at the modal frequency of 3848.85 Hz. It is observed here that shaft end vibration does not spread to rotor or housing, but as the pulley and connected loads will be put into vibration, the noise will be high. These frequencies are skipped during the operation.

Static stress analysis (load test) is also performed on FRG to monitor the limit violation of stress and deformation at different parts of the machine. Bearing is the additional model required for this analysis. Bearing is the element to hold the housing and rotor mass at the shaft location. So, bearing is simulated as four springs attached to the housing from the bearing locations. Spring stiffness of value of 2100 is assigned and the bearing is modeled. The load component on FRG machine, is of two components. They are,
(i) Torque due to the rotor
(ii) Centrifugal load due to rotating speed

The simulation is carried out for different rotating speeds of 900 rpm and 9000 rpm. The different parts of stress and deformations are obtained. From the results it is observed that, the stress and deformations of all the generator parts are in safe limit between 900 rpm to 9000 rpm.

An unbalanced rotor dynamic analysis is performed to determine the eccentricity of the rotor mass and it ensure that the vibration of the rotor is within the limit. The rotor unbalance force is computed assuming a balancing grade (Q) is 2.5 and damping ratio is 0.02, which are the usual standard values for high-speed machines. The weight of the rotor (w) is 3.23 kg. Under the operating speeds of 900 and 9000 rpm corresponding to balancing grade Q 2.5 (will vibrate at 2.5 mm/sec) the eccentricity e is computed as $e = \frac{v}{\omega}$, where $v$ - Velocity in m/s, $\omega$ - Angular speed (rad/sec).

From this equation the calculated value of eccentricity is 0.026 mm at 900 rpm and 0.0026 mm at 9000 rpm. As this eccentricity is of negligible value, it
is conclusive that the rotor dynamics of the considered FRG is in acceptable limit.

The unbalanced force is given by, $F = m\omega^2 e$, where $m$ = mass of the rotor (3.23/9.81=0.33 kg) and $e$ is eccentricity of the rotor. This gives the unbalanced force as 0.076 N at 900 rpm and 0.76 N at 9000 rpm.

This unbalanced force is applied over a frequency range of 1-10,000 Hz. This analysis is used to identify the possible vibrating and noise producing speed bandwidth which are skipped fast during acceleration. From the simulation results observed that large displacement occurs corresponding to the frequency of 1500 Hz. Since 1488.62 Hz and 1568.78 Hz are predominant, this corresponds to the horizontal mode and axial mode which obtained in the modal analysis. The critical speeds are identified as 11,164 rpm and 11,766 rpm. Hence it is inferred that, the FRG should not operate beyond these speeds. These natural frequencies are to be avoided for noiseless operations and it protects the machine from the fatigue failure of mechanical parts. So it will not affect the internal organs of human when operating the machines. The reported procedures for mechanical characterization of FRG in this thesis can be adopted for any pole combinations and dimensions of FRG to thoroughly perform mechanical study in three dimensions. This will help to fine tune the design to be declared for end product.

The above detailed electromagnetic and mechanical analyses were carried out on Losil 450 steel material. These were verified using an experimental set-up. Using this successful verification as a bench mark, a new material called cobalt-iron alloy (Hiperco 50A) was proposed for the FRG machine structure in chapter 8. This analysis is to improve the power density and reduce the weight of the generator for high speed aircraft application. This new material called cobalt-iron used as the stator-rotor material for FRG for the first time. The static and dynamic characteristics of this machine using FEA are documented and the same is verified with performance predictions. The measured output power of FRG with cobalt-
iron (Hiperco 50A) lamination confirms the increase in power by 15.5% and a reduction in weight by about 33% in comparison to the earlier used standard grade steel. In addition, it is shown that the stator and rotor core made with cobalt-iron (Hiperco 50A), results in increased power density, minimal loss, less material consumption and more resilient regarding magnetic and mechanical properties. The thermal and vibration analysis are also documented. The features of FRG developed with cobalt-iron alloy meet the application requirements of aircraft systems. Notwithstanding the fact that practical proof for the proposed cobalt-iron alloy material is absent, the analytical proofs and a respective comparison with the prototype proved results of Losil 450 is highly encouraging to propose cobalt-iron alloy for FRG stator-rotor construction. Hence, a prototyping of FRG with the proposed cobalt-iron alloy material as stator-rotor material is to be taken as an immediate future work. Thus the inference of this research can be used to fine tune the further application of soft magnetic materials in the FRM for automotive and aerospace industries.

9.2 FUTURE RESEARCH WORK

1. FRM prototype model for Hiperco 50A material will be done in the future.
2. FRM can be designed for the various aircraft and defence applications.
3. Various soft magnetic analysis can be carried out for improving the power density of the machine