Structural and Thermal Design of a Dual-Rotor, Constant-Frequency, Variable-Speed Generator

by

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Abstract

The mechanical design of a novel generator type is considered. This generator is intended to efficiently, reliably and cheaply convert a variable shaft-speed input into a constant electrical frequency output for application in the field of wind power. A optimization of the electrical characteristics has been performed and used as constraints on the mechanical design. Structural and thermal analyses of the design have been performed, and the design thereby specified to the point that a prototype can be constructed for testing.

Thesis Supervisor: James L. Kirtley, Jr.
Title: Professor of Electrical Engineering

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Chapter 1

Introduction

This thesis is a component of a project intended to develop a cheap, reliable, efficient means of converting a varying shaft speed into a constant frequency electrical power output. The mechanical and thermal aspects of the requirements of the proposed design are examined, and the design then specified with the intent that a prototype be built to test the validity of modeling and the usefulness of the device.

Chapter Two details the structural design of of the generator, based on analysis of the mechanical system.

Chapter Three describes the design and analysis of the cooling system for the generator.

Chapter Four concludes with an assessment of the design produced, and details the work that lies ahead on the project.

1.1 Motivation

In the field of electric power generation, there are number of cases in which the efficient conversion of mechanical to electrical power depends on the ability of the generator system to operate under conditions of a speed-variable mechanical input. Generally, it is also required that the output of the conversion system have a fixed electrical frequency to power, for example, the 60 Hz distribution grid. In such cases, the simple solution of a synchronous generator is undesirable, as its output frequency is directly coupled to the rotational speed of the input, so it must either operate inefficiently or at the wrong electrical frequency. Consequently, a number of alternate , more complicated designs have been proposed, each with benefits and drawbacks.

The particular application motivating this design is that of wind-generation of power. Wind power has substantial advantages over other generation techniques in that it is non-polluting, and fairly cost-efficient compared to other non-polluting power sources. However, the generation of wind power suffers from the problem of variable-speed mechanical input. With a standard fixed-pitch propeller, maximal energy is extracted from the airflow when the tip speed of the propeller has a constant ratio (which depends on the specific geometry of the propeller) to wind speed. Naturally, because the intended use of wind generation is to supply the distribution grid, the eventual electrical output of the conversion system must be 60 Hz.

The system proposed is one of a number of possible solutions to the problem (others are discussed in Section 1.3), and presents some substantial improvements in the areas of reliability, efficiency, cost, and complexity over existing solutions.

1.2 Design Overview

The design proposed in this thesis aims to convert the variable speed to a constant electrical frequency in the generator itself, by means of a novel machine design. The workings of the design, and the optimization of the mechanical system are described here, as they are the framework in which the design of the generator is undertaken[1]

1.2.1 Basic Configuration

The generator has, at its heart, a pair of rotors and a stator. One of the rotors is attached to the driveshaft and wound with the excitation coils, and the other, lying between the power rotor and the stator, is free-spinning, and supports a number of permanent magnets. Due to the two rotors, it would be very difficult from a mechanical standpoint to construct the generator as a standard radial flux machine, as the magnet rotor would then have to be a circular shell around the power rotor. Consequently, the generator employs axial flux.

The electromagnetic interactions in the machine take place parallel to the machine's driveshaft, between elements that are essentially discs mounted to the shaft or the housing. The advantage of such a geometry is that the number of rotating elements is not constrained

[1]The design of the electromechanical system, analysis of the electromagnetic theory of the machine, and optimization of the electromagnetic aspects of the geometry are due to Professor James L. Kirtley, Jr. (advisor for this thesis project), as they are his area of expertise, and outside the scope of my knowledge of electric machines.

by the difficulty of mounting a number of cylindrical shells. Thus, the rotor-rotor-stator arrangement is duplicated in reflection: rotor-rotor-stator-rotor-rotor, with two power rotors on the outside, and a magnet rotor between each power rotor and the stator. This allows more efficient utilization of the stator, where the majority of the electrical power is handled. For efficiency, the cores of the power rotor and stator are toroidal (this has the advantage of tightly confining the magnetic flux and lowering leakage), with interaction taking place between the annular faces. Consequently the magnets between them are also arranged in an annulus.

1.2.2 Operation

The full electromagnetic details of the machine operation are outside the scope of this paper. Moreoer, they are not particularly important to the mechanical design of the generator, so a greatly simplified version is here presented. This is done with the intent that the motivation for various design decisions be understood in the context of the operation of the machine, rather than an arbitrary set of specifications. A more complete and rigorous analysis of the machine's operation is presented in [1].

Obviously, as the machine is a generator, mechanical power is input through the driveshaft, to be converted as efficiently as possible to electrical power at the output. The power rotors are driven by the driveshaft, and carry an excitation current, which generates magnetic flux, moved through the armature coil of the stator by the spatial motion of the power rotor, thereby generating a voltage in the stator coil, and consequent output of electricity. The excitation current, carried to the rotors by means of slip rings on the driveshaft of the machine, also has a non-zero electrical frequency. This thereby generates an additional time-varying component of the magnetic field in the stator, as there is effectively a wave traveling electrically around the rotor as the rotor physically rotates.

Thus, the perceived frequency of magnetic flux in the reference frame of the stator is the sum of the mechanical frequency and the electrical frequency. Thus, by adjusting the electrical frequency of the rotor excitation current, we can adjust the electrical frequency of the output (which is the same as the apparent frequency of the magnetic flux). More importantly, by measuring the output frequency of the generator, and adjusting the input accordingly, we can keep the output of the generator at a constant 60 Hz, with, as it turns out, power input required at the rotor that is proportional to the electrical input frequency.

The above description requires unreasonably high mechanical mechanical speed without the addition of multiple pole pairs in the machine, so that one circling of the machine at any instant implies a traverse in electrical and magnetic phase of not 2π, but $2p\pi$, where p is the number of pole pairs. This has the effect of generating an output frequency which is p times as large as it would be with one pole pair, and requiring $2p$ separate magnets to generate the p pole pairs.

The permanent magnets in the rotor between the power rotor and the stator serve simply to drive magnetic flux efficiently across the gap between the two. Permanent magnets are, by comparison to electromagnets, very efficient, and capable of driving flux across much larger gaps of relatively low-permeability material (such as air). Their use in this machine allows the cores to be simply tori wound from strips of electrical steel, rather than complicated slotted affairs designed to reduce the air gap between the cores to a minimum.

1.2.3 Optimization and Parameter Determination

The constraints upon the design and construction of a prototype generator are largely due to the electrical optimization. Therefore, it was deemed best that the design and analysis of the electrical system be completed first, as it is more critical to the success of the prototype and more sensitive to parameter variation. Thus, Professor Kirtley constructed a numerical model of the electromechanical behavior based on the input of a number of geometric parameters. This model was then used in a Monte-Carlo optimization based on a number of parameters, including efficiency, total mass, outer diameter, current density in the windings, and peak flux density in the core, all of which were constrained to values which would result in reasonable designs.

The optimization produced a number of designs, representing a "frontier" of the design, in the sense that the algorithm evaluated each as roughly equivalent in terms of the output parameters. One of these designs was then selected as representing a good balance among the various competing factors. The specifics of this design (shown in Table 1.1) were taken to be the desired configuration of the prototype, and consequently, the mechanical design was based upon them.

Table 1.1: Parameters from Electrical Optimization

Parameter	Value
Rotor Inner Radius	0.27 m
Rotor Outer Radius	0.35 m
Stator Core Axial Length[2]	31 mm
Winding Thickness	5 mm
Rotor-Rotor and Rotor-Stator Air Gap	2 mm
Magnet Thickness	22 mm
Number of Pole Pairs	10
Efficiency	0.9008
Current Density	2.81 $\frac{MA}{m^2}$
Peak Gap Flux Density	0.88 T
Peak Core Flux Density	01.75 T
Magnet Mass	29.7 kg
Winding Mass	11.4 kg
Core Iron Mass	34.8 kg
Voltage per Turn	1.12 $\frac{V}{turn}$
Ampere-turns	2980 A·turn
Iron Dissipation	134.25 W
Stator Dissipation	243.25 W
Rotor Dissipation	614.72 W

1.3 Alternate Solutions

As mentioned, a number of other solutions to the problem of constant-frequency output from variable speed input have been developed. A brief aside is taken to explain the alternatives to the proposed design and consider the relative merits and difficulties of each scheme.

1.3.1 Continuously-Variable Transmission

Perhaps the most immediately obvious way to allow a variable speed propeller to produce a constant electrical frequency is to use a transmission. It is entirely feasible to design a so-called continuously-variable transmission (CVT), which allows any conversion ratio (not just a finite number of ratios) of rotational speed within a reasonable range. Obviously, by connecting such a transmission between the propeller shaft and the driveshaft of the generator, it becomes possible to use a standard generator, and keep the driveshaft speed constant while the propeller speed varies for maximum efficiency. Unfortunately, CVTs are mechanically complicated, and therefore prone to wear and failure, and fairly expensive as well, so they are less-than-optimal solution.

1.3.2 Power Electronics

Another fairly obvious solution is to leave the problem to the electrical side. It is quite simple to design a circuit which takes in power at one frequency and converts it to a different frequency, usually by way of converting to DC in the middle. Thus, the output frequency can be made to be completely independent of the input frequency over a very large range of potential input frequencies. The difficulty with such an approach is one of scale. It is a daunting prospect to build power electronics which can handle power on the order of a megawatt, which is well within the power-producing capabilities of modern windmills. Such electronics is also extremely expensive, though it presents perhaps the most versatile solution.

1.3.3 Variable-Pitch Propeller

A more subtle way of maintaining efficiency over varying wind speed derives from the observation that the constant of proportionality between the wind speed and optimal propeller speed depends on the geometry of the propeller. Thus, if the geometry of the propeller can be changed, the optimal tip speed at a given wind speed can be varied. Said another way, the propeller can be adjusted so that the optimal tip speed for the current wind speed is the speed which produces a 60 Hz output from the generator. This is an attractive option because it allows the use of a standard-design generator, while off-loading the complication into the propeller, which will likely be custom-produced anyway. Although in theory such a design is not terribly complex or expensive, attempts to produce such large variable-geometry propellers have not been entirely successful. Likely this will just be a matter of maturation of technology and designs. Attractive as this idea is, it still has considerably more mechanical complexity than the proposed design, and so there are definite benefits to reliability of using a special generator designed to do frequency conversion.

Chapter 2

Structural Design And Assembly Concerns

The primary objective of this thesis is to lay out the design necessary to construction of a prototype generator. Proceeding from the general geometry necessitated by the electromechanical requirements of the generator, I determine necessary dimensions of parts and methods of coupling between them. Of additional mechanical concern is the method of assembly, which is made non-trivial by the substantial weight of and attractive forces between parts of the generator.

2.1 Stator

The stator is the electrical output of the system. It consists of a tape wound silicon-steel core, and Gramme-Ring windings[1] (in an electrical configuration which has yet to be determined, but does not affect other design aspects). The core is to be constructed from 31 mm wide M-19 electrical sheet steel. The strip of steel will will be wound into a torus with inner radius 27 cm and outer radius 35 cm. As the steel is wound, the layers will be epoxied together, and the whole torus will be coated with a layer of fiberglass and epoxy to present a smooth surface for winding the coil.

Because the stator acts on both winding faces (one facing each rotor), the axial loads

[1] In a Gramme-Ring winding, the conductor travels in a spiral covering the whole surface of the core. That is to say, the conductor comes up on the inner radius of the core, traverses the top face from inner to outer radius, drops down the outer radius of the core, and returns to the inner radius across the bottom face.

on the operating stator should be balanced due to symmetry. However, the stator will experience the full torque of the input, and must, naturally, be prevented from rotation. At a reasonable upper limit of input torque of 300 N·m, force at the surface of the stator is only 860 N, which is not-unreasonable shear for a single $\frac{1}{4}$ in. bolt. Thus, the stator can be coupled to the housing by a mere handful of bolts, which will have their heads countersunk into the inner surface of the core and penetrate it radially outward, attaching to spacer blocks located around the outer surface of the core, where there can be gaps in the winding due to increased radius. To prevent possible deformation of the core, 20 bolts evenly spaced around the core will be used.

2.2 Power Rotors

The power rotor in the generator consists of a tape-wound silicon-steel core and windings with conventional end-turns.[2] The core is to be constructed from 15.5 mm wide 29 AWG M-19 electrical sheet steel, which will be identical to the stator core, but only half the axial length.

Expected axial force on the rotor core will be fairly high, due to the high flux density in the machine. The attractive force per unit area of core can be calculated using the Maxwell Stress Tensor:

$$F = \nabla \cdot \underline{\underline{T}} \tag{2.1}$$

where

$$T_{ij} = \mu_0 H_i H_j - \frac{1}{2}\mu_0 |H| \delta_{ij} \tag{2.2}$$

and δ_{ij} is the Croniker delta function. For our geometry, we can simplify this to $F_{axial} = \frac{1}{2}\frac{B_0^2}{\mu_0}$. The expected peak gap flux density is 0.88 T, so the axial force will be approximately 48 kN at worst.

Because the relative rotation gap is rather small, and thus the allowable deflection of the support structure is also small, the rotor core must be supported by a thick plate. Although it would be desirable from a mechanical design standpoint to doubly-support the rotor structure with a thrust bearing at a larger radius than the core, this approach is not

[2]Conventional end-turns return the conductor from outer radius to inner radius on the same side of the core that it travels from inner radius to outer radius, essentially forming a series of loops in the plane of one core surface.

practical. Firstly, such a bearing has to be of sufficiently large size and load capability to make it difficult to find and very expensive. Secondly, the bearing would have to be ferrous, and therefore interact with the magnetic fields of the generator, substantially reducing its efficiency. Finally, the mounting of the thrust bearing to the housing presents a substantial complication to the design. Thus, the best solution is to have the rotor assembly singly supported at the drive-shaft, and make the support structure a very stiff, thick plate.

The thickness of the plate is not particularly constrained, so weight can be saved by using an aluminum disc instead of a steel one, which would be thinner, but heavier for the same deflection. Simple manipulation requirements make the lighter option preferable.

Consideration must then be given to attachment of the core to the plate. The winding need not be completely uniform over the surface of the core, so U-bolts can be looped over the core, through holes in the plates, and secured. Collectively, these bolts must support the entire axial load of the core. Including a safety factor of 8, the total area of bolt needed so as not to exceed the elastic limit of the steel is 1920 mm^2. Using $\frac{1}{4}$ in. bolts, a total of 30 bolts will be needed to provide the desired strength.

Given a material and dimensions, we can use standard results for stress in and deflection of a disc with fixed inner edge and free outer edge. The loading conditions correspond to two uniform annular line loads, each carrying half of the total attractive force. The case of a single line load is solved in [2], and by superposition, we can simply add the resultant stresses and deflections. The analysis is given in more detail in Appendix A Under the conditions specified, a 6061-T6 aluminum disc 2" thick will have a maximum stress of 65 MPa and a deflection of 0.35 mm. That stress provides a safety factor of more than 4 (for possible concentrations, fatigue, and shocks) in the material strength, and does not allow the disc to impinge too substantially on the rotational gap.

Finally, to enable forced-convection cooling by flow through the rotational gaps, the magnet rotors must be perforated to allow air to flow through them axially. These can take the form of holes drilled just inwards of the inner edge of the core (where the bending moments in the disc are relatively low), so the loss of material does not appreciably impact the structural concerns.

2.3 Magnet Rotors

The magnet rotors freewheel on the driveshaft, connected by a separate shaft, and serve simply to drive flux between the stator and the power rotors. In operation, symmetry should make the net force on the rotor zero. Each of the twenty magnets, however, should be mounted so as to be able to take it's full portion of the axial load, about 2.5 kN. Mounting is also complicated by the fact that the magnet material is very difficult to machine.

The proposed solution is to machine an aluminum disc, of thickness 22 mm (as are the magnets), and radius 37 cm, into which are cut twenty windows (slightly oversize) to fit the magnets[3]. The magnets will be epoxied in place and each face of the disc will be covered by a thin (1 mm) sheet of aluminum, epoxied in place. The free-wheeling rotor disc is thus 24 mm thick. Because the magnets are more than 20 cm in perimeter, the cross-section of the sheet in shear is more than 20 mm^2, which gives a maximum stress of less than half the yield stress.

The bearings attaching the rotor assemblies to the driveshaft mostly support the axial loads related to maintaining the axial placement of the rotors, but also some inevitable radial loading, if merely from the mass of the magnets (about 15 kg per rotor), so tapered roller bearings are a good choice. The bearings can be press-fit into the rotor and rest against a shoulder left on the power-rotor face of the disc, of at least 1 mm thickness (actual thickness should be determined so as to make the thrust surface of the bearing flush with the stator-facing surface of the disc).

The magnet rotors must also be perforated to allow air to flow through them. This can be accomplished in the same manner as for the power rotors, with holes just inwards of the inner edge of the magnets.

2.4 Driveshaft

Taking a conservative upper-bound on the torque that the generator shaft will have to bear, the torsional loading will be about 300 N·m. Since we desire to have a fairly small shaft for reasons of weight, cost, and ease of assembly, the shaft will be of a high-strength material: steel. Magnetic flux near the shaft will be very small, so it is acceptable to use a ferrous shaft.

[3]The exact dimensions and shape of the magnets are not yet known, and will be limited by availability

Torque is transmitted from the shaft to the two rotor plates, and the load will split evenly due to symmetry, so each plate-shaft interface will need to support a torque of about 150 N·m. Additionally, the interface will have to support the attractive axial force exerted on the plate of about 50 kN. The combination of heavy axial and tangential loads suggests that the shaft should be stepped and keyed.

Design of the shaft proceeds as suggested in [3]. The elastic limit of ductile ferrous material in shear is well approximated by half it's elastic limit in tension. Using a lower-bound on $\sigma_Y = 200$ MPa for steel, we have a shear elastic limit of 100 MPa. The effective shear limit is further reduced by 25% due to the inclusion of a standard size keyway in the shaft. The recommended correlation is then

$$D = \sqrt[3]{\frac{5.1 K_t T}{S_s}} \approx 30 \text{ mm} \approx 1\tfrac{3}{16} \text{ in.} \tag{2.3}$$

where the factor $K_t = 1.25$ accounts for minor shocks in the torsional loading and fatigue, and S_s is the adjusted elastic shear limit. Transverse loading of the shaft is due only to the weight of the four rotors, and because the shaft is very short, the associated bending moment is insignificant.

For ease of procurement, using the American standard transmission shafting size of $1\frac{3}{16}$ in. is preferable. For such a shaft, the ANSI Standard preferred key is a $\frac{1}{4}$ in. × $\frac{1}{4}$ in. key, with a keyseat depth of $\frac{1}{8}$ in.. Expected shear force in the key at the interface is $\frac{T_{rotor}}{r_{shaft}} = 10$ kN, which will be spread over an area of $w_{key} t_{rotor} = 240\ \text{mm}^2$, for a stress of 41 MPa, substantially below the elastic limit for steel.

Design of the axial connection to the shaft requires that the local stress over the loading area not exceed the elastic limit of either the steel shaft or the aluminum rotor plate. The shoulder must have a large enough area to account for additional stress concentration due to the re-entrant corners, whose effect can be conservatively[4] assumed to create a stress concentration factor of 2. The area of the shoulder is given by $\frac{\pi}{4}(D_2^2 - D_1^2)$, so (including a factor of 8 for safety, fatigue, loading, and the like)

$$D_2 \geq \sqrt{D_1^2 + 20\frac{F}{\sigma_y}} \approx 44 \text{ mm} \approx 1\tfrac{3}{4} \text{ in.} \tag{2.4}$$

[4]Because the smaller diameter of shaft is not appreciably supporting the axial load, the effect of the re-entrant corner will be very small, even with little filleting. However, a fillet of radius 2 mm seems wise anyway, and for reasons of conservative design, the change in axial loading will be ignored.

Thus, the shaft should step up to $1\frac{3}{4}$ in..

The free-wheeling magnet-rotor assembly then bears on the $1\frac{3}{4}$ in shaft. It transmits no torque to the shaft, and very little axial load when the device is assembled (because magnetic forces on it from the rotor and stator are balanced). However, during assembly, it must bear axial load, and under operating conditions, it must be tightly constrained axially, as the clearance to the power rotor and the stator is only 2 mm. Thus, another step in the shaft is required. Because the load borne by that shoulder is small, the diameter need only step up to 2 in.

The spacing between the steps on the shaft determines the rotational gaps in the machine, and thus is fairly critical. The 2 in diameter section of the shaft lies between the stator-facing sides of the magnet rotors, so the distance between shoulders should include stator core width, two winding thicknesses, and twice the relative rotational gap, minus twice the distance from the face of a magnet rotor to the face of the magnet (assuming that the thrust surface of the magnet-rotor bearing is flush with the face of the rotor). All told, this is 43 mm $\approx$ 1.70 in.

The 1.75" diameter sections each extend from the stator-facing surface of the magnet rotor to the inner face of the power-rotor support plate. This distance is the thickness of the magnet-rotor, plus a rotational gap, plus a winding thickness, plus the width of the rotor core, a total of 46.5 mm $\approx$ 1.83 in.

The $1\frac{3}{16}$" diameter shaft should extend, on both sides, to the bearings in the housing end plates. On one side, it should extend another 3" to allow mounting of the slip rings that will transfer electric power to the rotors. On the other, after passing through the bearing, the shaft should extend another foot to allow for coupling of the prototype to the testbed, probably via bearings and a keyed gear, as it is expected that the dynamometer available for testing will not be able to handle the low speeds necessary for the machine. Thus, on one side, the shaft extends 6" beyond the mounting shoulder for the power rotor, and on the other, it extends 18"

2.5 Free-wheeling shaft

The free-wheeling shaft connects together the two magnet-rotors, thereby simplifying the problems of their axial alignment, as a single shoulder for each disc suffices to fully constrain both. It will be a thick-walled aluminum cylinder of the same length as the 2" section of the driveshaft (43 mm), of inner diameter slightly larger than the outer diameter of the bearings of the magnet rotor plates, and $\frac{1}{2}$" wall thickness. Bolts ($\frac{1}{4}$") passing through the rotor plates into tapped holes in the ends of the shaft will attach the plates. The two rotors are driven identically, so the free-wheeling shaft should not carry any substantial torque. Also, as the magnet rotors will be well balanced in axial forces, the axial loading of the free-wheeling shaft will also be small (although the shaft should, at its dimensions, easily support the full axial force between rotor and magnets).

2.6 Housing

The generator housing serves the function of connecting together the stator and the shaft assembly, as well as providing a means of mounting the generator to the test bed. For assembly reasons, the housing consists of several pieces: a cylindrical sleeve surrounding the rotating parts, spacer blocks connecting the stator to the sleeve, and end-plates with bearings to connect the shaft to the sleeve.

The sleeve must be of inner radius 38 cm so as to allow clearance to all the rotating parts. Additionally, the wall thickness must be sufficient to transmit all of the applied torque to the mounting. However, given the required diameter, very little thickness of aluminum is actually required to support the loads. The chief concern is supporting the weight of the device, and maintaining integrity near the stator-mounting and end-plate-mounting bolts. These concerns can be easily addressed with a 1 cm wall thickness, which will allow the luxury of determining the testbed-mounting implementation later. The axial length of the sleeve should leave adequate rotational clearance (at least 1 cm) between the back of the power rotor and the end plate. The overall length of the sleeve is then about 21.1 cm.

To allow the cooling air to escape (see Section 3.1), ventilation slots will be that which run around the circumference of the sleeve at the location of the 4 rotational gaps. These slots can take up about 50% of the circumference, and still leave plenty of material to support the loads. To allow removal of spacers from the assembly process (Section 2.7), the slots should be 2 mm wide. Additionally, the airflow must be forced to flow axially down the center of the machine, rather than around the end of the rotors. Consequently, a flexible plastic annular seal will be epoxied to the housing just outside the power rotors.

The spacer blocks, which serve to separate the stator from the sleeve, thus need to be 3 cm $\times$ 3 cm $\times$ 1.5 cm so as to span the minimum reasonable tangential distance given the diameter of the bolt-hole in the center. These blocks will easily support the mass of the stator and the associated operational torque.

The end-plates serve to constrain the driveshaft axially and radially with respect to the stator. Both axes should be fairly well balanced (except for the weight of the shaft assembly and slight mismatches in axial force on the stator), so a relatively low-load tapered-roller bearing can be used in each endplate. The endplates can be bolted to the sleeve by $\frac{1}{4}$" bolts around the periphery attaching to tapped holes in the sleeve. Additionally, forced convection will be accomplished by mounting a blower to each endplate, so a hole must be cut to which the blower can be mounted.

2.7 Assembly

Assembly of the generator poses a few interesting problems of its own. Since there are a large number of powerful magnets in the generator, the attractive forces between the magnet rotor and the stator and the two rotors will be very large, and require a screw jack to slide them into place in a controlled fashion.

The jack will consist of a steel tripod which spans the outer sleeve, with a threaded $1\frac{3}{4}$" hole at the joint. Through the hole will be inserted a $1\frac{3}{4}$" threaded rod, with a large hook on one end, and a handle on the other. This will attach to the rotors by steel cables which run around the hook and through the ventilation holes in the discs. Additionally, during assembly, HDPE spacers 1.9 mm thick will be placed in the rotational gaps to ensure correct spacing until the device is assembled and balanced properly. These spacers will be designed so as to fit through the ventilation slots in the outer sleeve, and will be attached to cords running through those slots, to aid in the removal of spacers after assembly.

Taken as a whole, the assembly process has a number of steps, whose order and exact details of implementation are important to the ease of the assembly process. The assembly should occur as follows:

1. Cores constructed from steel strip as described above
2. Magnet rotor assembly, best done as follows:
    (a) One faceplate epoxied in place

(b) Magnets epoxied into their pockets one at a time (to reduce the chance of two sticking together). If this stage of the assembly is done on a ferrous table, the magnets will supply their own clamping pressure. Of course, this *must* be done on a surface off of which the completed rotor can be slid; otherwise the combined force of the magnets will make lifting the rotor off impossible

(c) Second faceplate epoxied on. A large weight placed atop the rotor will supply clamping pressure

(d) Tapered roller bearing press-fit into place

3. End-plate bearings press-fit into place
4. Blowers bolted to end-plates
5. Rotor cores attached to their support plates with U-bolts.
6. Rotor coils wound
7. Stator attachment bolts inserted through core and into spacer blocks
8. Stator coil wound
9. Stator bolted to sleeve
10. HDPE spacers placed on one side of stator (generator axis held vertical for assembly of rotational elements)
11. Free-wheeling shaft bolted to one magnet rotor
12. Driveshaft inserted through magnet rotor to shoulder
13. Magnet rotor and shafts lowered into place with jack (This will require a hole in the assembly surface through which the driveshaft can pass on the bottom side)
14. HDPE spacers placed on magnet rotor.
15. Key inserted in driveshaft keyseat.
16. Power rotor located on shaft and lowered into place with jack
17. Plastic seal epoxied into housing
18. End-plate slid onto shaft and bolted to sleeve (no jack necessary because the flux is low near the endplate, and the endplates are non-ferrous)
19. Partially-assembled generator turned over
20. HDPE spacers placed on top side of stator
21. Second magnet rotor located on shaft and lowered into place with jack
22. Second magnet rotor bolted to free-wheeling shaft
23. HDPE spacers placed on magnet rotor
24. Key inserted in driveshaft keyseat
25. Second power rotor located on shaft and lowered into place with jack
26. Plastic seal epoxied into housing
27. End-plate slid onto shaft and bolted to sleeve
28. HDPE spacers removed

Chapter 3

Thermal Design and Analysis

3.1 Cooling Requirements and Thermal Design

The numerical modeling of the generator operation indicates that there will be a substantial amount of dissipation in the cores and windings of the machine. More than half of the dissipation occurs in elements that are effectively isolated from the environment, from a thermal conduction standpoint. The thermal resistance from the power rotors to the environs will be quite high without some kind of cooling scheme. The dissipation there, however, is expected to be more than 600 W, and the epoxies and wire insulation used should be kept below 85°C.

The need for some form of cooling is obvious. The only reasonable way to cool the machine is by forced air, as natural convection would not be sufficient to extract nearly 1 kW from the device without unacceptable temperatue rise. Liquid cooling, while much better in terms of potential heat-transfer rate, would either substantially impact the dynamics of the machine, or require complicated machining to create flow channels through the cores. Either way, substantial additional hardware would be required to pump the liquid, route the flows properly, and extract the waste heat before recycling it.

To maximize the effect of forced-convection-cooling, we would like to force the air as close as possible to the sources of dissipation, and over as much hot area as possible. Additionally, the flow should travel quickly to maximize the turbulence of the flow, and increase the capacity flow rate of air. These desires can be simultaneously satisfied by forcing air through the rotational gaps. The rotational gaps are as close as it is possible to get to the windings, where the majority of the dissipation (more than 85%) is expected to occur. Moreover, the gaps have quite small cross-sectional area due to their small height, and this will naturally speed up the flow as it is constricted.

In terms of the machine itself, the simplest way to force air radially through the gaps is to blow it in axially at the ends of the housing. If the rotor plates are perforated, the air will flow axially towards the stator until it reaches the 4 gaps, where the positive pressure in the center of the machine will force it radially outward, where slots cut in the side of the housing will allow it to escape to the environment. As described in Section 2.7, these slots will also allow for the extraction of the spacers used in the assembly process.

So that it can be forced through the relatively thin rotational gaps, the air must be prevented from flowing around the outside of the rotating elements. This can be accomplished by a loose seal against the back of the power rotor plates. Said seal could simply take the form of a flexible plastic annulus mounted to the motor housing at the back side of the power rotor. The pressure of air in the generator will then press the plastic gently against the plate, preventing the flow from going around the end of the rotor.

3.2 Thermal Analysis

With the most reasonable flow geometry selected, it remains to be seen how much air must be forced through the generator to keep its temperature acceptably low, and thus whether we can reasonably purchase blowers to accomplish the cooling.

3.2.1 Modeling and Approximation

Due to the very small size of the gap, the forced convection heat transfer will be dominated by the flow through the four gaps, where the velocity will necessarily be much higher. Thus, the critical flow geometry is that of radial flow between a pair of annular parallel plates. Unfortunately, this is a bizarre flow geometry, and consequently, there do not exist correlations for the heat transfer of such a flow. Thus, the following approximations were made to allow use of extant correlations:

- The change in radius from the inside to the outside of the annuli can be neglected. The cross-sectional area of each of the flows is thus assumed to be constant at $A_C = 2\pi \frac{r_o + r_i}{2} h$.
- The small size of the gap relative to the circumferential length allows approximation of the flow as between two infinite plates, for purposes of calculating heat transfer

per unit area. Combined with the previous condition, this allows consideration of the standard case of two infinite rectangular parallel plates with flow in the plane of the plates.

- The small size of the gap ensures that both sides of the gap will be at the same temperature. This is to say that the magnets, which experience essentially no dissipation, will be heated to the same temperature as the windings and cores.

- The thermal diffusivity of the windings and magnet plates is high enough that all surfaces involved in the heat transfer are at the same constant (spatially and temporally) temperature.

3.2.2 Analysis

With those assumptions made, the first step in the analysis is to determine the nature of the flow regime that occurs between the plates. For an initial estimate of flow velocity, a volumetric flow rate of 0.05 m^3/s is chosen. The Reynolds Number of the flow (based on the hydraulic diameter of the assumed parallel plate situation) is $Re_{D_h} \approx 3200$, which falls squarely into the transitional regime ($Re_{crit} = 2300$). The problem is further compounded by the short flow length (only 80 mm), which means that the flow will not have time to develop completely. For laminar flow, the expected non-dimensional thermal entry length is $L^+_{th} = 0.00797 = \frac{L_{th}}{Pr Re}$, so $L_{th} \approx 18$ m. Thus, Hausen's correlation for mean Nusselt number of transition flows in the thermal entrance region of parallel plates with uniform wall temperature, from [4], is used:

$$Nu_{m,T} = \frac{\bar{h} D_h}{k_f} = 0.116 \left(Re^{2/3} - 160\right) Pr^{1/3} \left[1 + \left(\frac{x}{D_h}\right)^{-2/3}\right] \tag{3.1}$$

Equation 3.1 gives the average heat transfer coefficient as $\bar{h} = 47.3 \ \frac{\mathrm{W}}{\mathrm{m^2 K}}$. Applying the P-NTU method for single-stream heat exchangers ($R_1 = 0$), the capacity flow rate of air is $C_1 = 58.2$ W/K, and the non-dimensional thermal size is $NTU = 0.254$. Thus, the efficiency is $P_1 = 0.20$. If the plates are to be at equal temperatures, the heat flux in each gap must be one quarter of the total heat dissipated, so $q_{gap} = 248$. Thus, the temperature rise of the plates is given by $T_{rise} = \frac{q}{P_1 C_1} = 21.0$ K

Chapter 4

Conclusion

The aim of this thesis project was to develop the mechanical and thermal design for a prototype of a novel type of generator. Proceeding from a description of the operation of the generator, functional requirements in the areas of structure and heat dissipation were determined. These functional requirements were analyzed, and appropriate design decisions made, specifying the design to about the limit without further determination of the testbed design and specific material and machining availability.

As designed, the generator meets all of the requirements set out, with substantial safety margins, making conservative approximations when necessary or expeditious. Thus, I am confident that the design presented will be workable in reality and allow construction of a prototype generator to test the predictions of our modeling.

4.1 Further Work

A number of tasks remain before the generator project, as a whole, is complete. The overall aim is to build and test a physical prototype of the generator to evaluate its potential usefulness, and the validity of our model. Three substantial tasks remain before this goal is achieved.

4.1.1 Testbed Design

Firstly, a means must be devised of testing the prototype, before the prototype is built, since the requirements of the testing setup may alter the design of the generator. A dynamometer will be available for testing of the generator under load, and it is capable of handling the expected power level of 10 kW. However, it is believed that the torque limit of the dynamometer is too low to allow it to interface directly with the generator. Hence, some design is necessary, for a transmission system and mounting technique. The testbed design is expected to be completed within the next month or so.

4.1.2 Construction

Naturally, it will be necessary to construct the prototype before it can be tested. This will require that all of the necessary materials be obtained (bearings, magnets, steel strip, and so forth), and that a machine shop be contracted to machine the larger parts of the generator (the sleeve and rotor plates, for example). Another concern is the need for a "small" power electronics package to drive the power rotor — "small" in comparison to the power levels of the generator; it will need to handle on the order of 1 kW. There is also the concern of the remaining parts needed for the testbed.

Once all of these parts have been obtained, the prototype will have to be assembled, which will be a long process, involving winding several hundred meters of steel strip, and sheer difficulty maneuvering the mass of the generator pieces, many of which will weigh over 100 lbs. Assuming that all of the needed parts can be obtained in time, it is hoped that the prototype will be assembled by the end of the summer.

4.1.3 Testing

Finally, the ultimate goal of the project will be reached in the testing phase. We hope to be able to characterize the efficiency, power handling capabilities, and transient behavior of the generator, in short to assess its suitability of the design concept for application to the wind power generation problem. Additionally, in this phase, we will be able to experiment with the control issues presented by the generator, as well as evaluate its possible utility for other functions, such as low-speed, high-torque motor applications without a gearbox. With luck, this final phase will be completed within a year, and we will have conclusively demonstrated the utility of the new design.

Appendix A

Power Rotor Plate Analysis

The power rotor plates must support very large loads during the operation and assembly of the generator. The structural conditions involve complicated derivations, so use is made of [2], which provides solutions to most reasonably conceivable constraint and loading conditions for a multitude of geometries. We assume that the rotor plates' attachment to the shaft is rigid, and that the loading of the disc is through the U-bolts. This will then effectively produce two annular line loads of $\frac{f}{4\pi r_0}$ and $\frac{f}{4\pi r_1}$, where f is the load force, and r_0 and r_1 are the radii of the annuli. The solutions for maximum stress and deflection of the perimeter for each line load are considered separately, then combined through superposition to yield the combined behavior due to both. The following MATLAB code implements the formulae given in Table 24 (pp. 332–373) of [2]. Unfortunately, the formulae in the fifth edition are in English units, so conversions are necessary and somewhat obscure the clarity of the code.

```
% Dimensions
a = 380/25.4; %r otor plate outer radius
b = 20/25.4; % radius of shaft support
r0 = 270/25.4; % radius of inner edge of core
r1 = 350/25.4; % radius of outer edge of core
t = 2;

% Material Properties (aluminum)
rho = .25;
E = 10e6;
nu = .33;

% Magnetics -> loading
mu0 = pi*4e-7;
Bo = .88;
qm = .5*Bo^2/mu0;        % q, the attractive pressure, in pascals
qtot = qm/6985;          % conversion to PSI
ptot = qtot*pi*(r1^2-r0^2)
```

```
w = ptot/2/r0/2/pi;

% disc constants used in expressions
D   = E*t^3/12/(1-nu^2);
C2  = 1/4*(1-(b/a)^2*(1+2*log(a/b)));
C3  = b/4/a*(((b/a)^2+1)*log(a/b)+(b/a)^2-1);
C8  = 1/2*(1+nu+(1-nu)*(b/a)^2);
C9  = b/a*((1+nu)/2*log(a/b)+(1-nu)/4*(1-(b/a)^2));
L3  = r0/4/a*(((r0/a)^2+1)*log(a/r0)+(r0/a)^2-1);
L9  = r0/a*((1+nu)/2*log(a/r0)+(1-nu)/4*(1-r0^2/a^2));

Mrb = -w*a/C8*(r0*C9/b-L9);
Qb = w*r0/b;
ya1 = -w*a^3/D*(C2/C8*(r0*C9/b-L9)-r0*C3/b+L3);
sigma1 = 6*Mrb/t^2;

% Solution repeated for second line load
r0 = r1;
D   = E*t^3/12/(1-nu^2);
C2  = 1/4*(1-(b/a)^2*(1+2*log(a/b)));
C3  = b/4/a*(((b/a)^2+1)*log(a/b)+(b/a)^2-1);
C8  = 1/2*(1+nu+(1-nu)*(b/a)^2);
C9  = b/a*((1+nu)/2*log(a/b)+(1-nu)/4*(1-(b/a)^2));
L3  = r0/4/a*(((r0/a)^2+1)*log(a/r0)+(r0/a)^2-1);
L9  = r0/a*((1+nu)/2*log(a/r0)+(1-nu)/4*(1-r0^2/a^2));
L11 = 1/64*(1+4*(r0/a)^2-5*(r0/a)^4-4*(r0/a)^2*(2+(r0/a)^2)*log(a/r0));
L17 = 1/4*(1-(1-nu)/4*(1-(r0/a)^4)-(r0/a)^2*(1+(1+nu)*log(a/r0)));

Mrb = -w*a/C8*(r0*C9/b-L9);
Qb = w*r0/b;
ya2 = -w*a^3/D*(C2/C8*(r0*C9/b-L9)-r0*C3/b+L3);
sigma2 = 6*Mrb/t^2;

% Solutions combined

sigma = sigma1+sigma2;
sigmam = sigma*6985 %value of total maximum stress, in pascals
ya = ya1+ya2;
yam = ya/39.4        %greatest deflection, in meters
```

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