

DESIGN OF HIGH-SPEED, DIRECT-CONNECTED, PERMANENT-MAGNET MOTORS AND GENERATORS FOR THE PETROCHEMICAL INDUSTRY

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Abstract - This paper re-examines some of the usual assumptions in deploying synchronous machines in multi-megawatt, high-speed applications. A viable design is presented with a focus on rotor capability and cooling system performance that can operate at common gas turbine speeds and power ratings. The models used in the analysis are validated with a similar topology machine, albeit a higher speed and lower power than the proposed design.

Index Terms — Permanent magnet, high speed, synchronous, motor, generator

I. INTRODUCTION

High-speed electric motors with variable speed drives have been shown to have an advantage over gas turbines as a prime mover for natural gas compressor applications when both environmental and economic factors are considered [1] [2]. Both synchronous and asynchronous systems have been considered for these applications, however there are significant differences among synchronous systems that warrant discussion.

The synchronous machines discussed by LaGrone [2] were discounted because of the complex rotor construction typical of wound-field synchronous machines. Permanent magnet (PM) machines do not suffer from many of the same difficulties, as they do not have field winding and a rotating rectifier. In addition, the modern synchronous PM machine designed for high speed and high power applications differs from its lower power brethren, such as the machine topologies studied by Melfi [3].

High-speed PM machines typically have surface-mounted magnets and are sleeved. Laminated rotor cores and/or embedded magnets do not offer either the radial stiffness or mass containment required for many applications. The sleeve provides the containment and a solid rotor core, or hub, provides stiffness. Common sleeve materials are non-magnetic high-strength alloys, pre-molded graphite composite, and wound-in-place carbon fiber. Of these, the carbon fiber winding offers the highest strength while providing minimal conducting path for eddy currents induced by air-gap flux disturbances.

While Arkkio [4] was primarily focusing on high-speed applications not more than one or two megawatts, the observation that machines will require a larger diameter than an equivalent PM machine should not be lost on the reader. As

the power rating becomes higher, the rotor speeds become lower, but the diameters grow. Arkkio suggests that the rotor tip speed of 250 m/s to be the upper limit for applying PM machines in favor of solid rotor induction machines. While Arkkio recommends a design value for sleeve stress, he does not directly connect the tip speed to the sleeve stress. Certain design choices, in particular magnet thickness and rotor construction, can influence the sleeve stress dramatically but have no effect on tip speed. For this reason, it is the sleeve stress that determines the limit of rotor containment (and ultimate speed) and the tip speed should only be used as a parametric indicator or rule-of-thumb.

The high-speed, sleeved PM machine has an intrinsically larger magnetic air-gap than the unsleeved PM machine due to the sleeve thickness and the increased magnet thickness required to force an equivalent amount of flux through the larger magnetic gap. This larger magnetic gap provides better demagnetization protection, especially under short-circuit conditions. The potential of PM machines to demagnetize is often used as a factor in dismissing them for applications where robustness is required. The machine design presented here benefits from a relatively large magnetic gap and the near immunity to demagnetization.

High-speed machines are operated at frequencies at least an order of magnitude above line frequency. This high-speed operation allows the machines to be smaller than conventional machines in the same power rating which yields a significantly higher power density machine than the conventional alternative. However, with the higher power density and frequency also come higher loss density. Special attention must then be paid to the choice of lamination material, coil construction, and cooling mechanism for what would otherwise be a typical stator and housing design. In the case of a high-speed PM machine, temperature sensitivity of the magnet material is an additional factor. For this reason, samarium cobalt is often the choice to realize higher temperature designs. In case of composite sleeve machines, the maximum sleeve operating temperature represents another design constraint.

II. DESIGN SPECIFICATION

The petrochemical industry would benefit from a high-speed, medium-voltage PM machine to drive centrifugal compressors or be directly driven by gas turbines. The PM machine has reduced system weight, higher operating efficiency, and smaller

envelope than a conventional solution. Directly coupling the motor to the compressor or the generator to the turbine eliminates the need for a gearbox and its inherent maintenance costs and performance penalties. By utilizing magnetic bearings the machine maximizes the benefits of a lube-free system. Magnetic bearings can operate at higher speeds with less loss than certain types of mechanical bearings, which are constrained by size and lubrication type.

Figure 1 shows a selection of gas turbine ratings which are well matched to an eight megawatt product family operating at a rated speed of no more than 15,000 RPM.

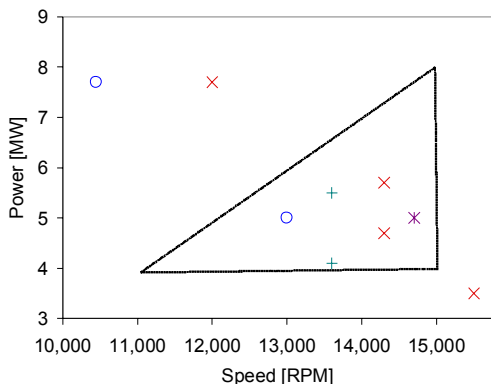


Fig. 1 Gas turbine ratings from various manufacturers [5] The triangle shows the target applications for a product family rated at 8MW and 15,000 RPM maximum.

TABLE I
Required Specification of High Speed Medium Voltage Motor

| Item | Specification |
|-----------------------|--|
| Power Rating | 8 MW |
| Rated Speed | 15 kRPM |
| Overspeed | 18 kRPM |
| Drive Rating | 6.6 kV, 1 kA |
| Sleeve Configuration | Wound carbon fiber |
| Cooling Configuration | Water/Glycol stator jacket Curtain air flow over end-turns Stator mid-stack air flow |
| Bearing Configuration | 2-Radial, 1-Thrust Magnetic Ball Bearing Touch Down |

III. DESIGN AND ANALYSIS

A. Electromagnetic Sizing

The electromagnetic sizing of the machine tends to come secondary to the mechanical limits of the rotor. The operational speed is driven by market analysis (Figure 1) and a rotor tip speed rule-of-thumb is used as a starting point for the rotor design. The rotor diameter is selected ultimately by iterating between the rotor containment and electromagnetic analyses. The rotor dynamic response determines the maximum rotor length, and hence the power limit for the product family. The stator design follows conventional lines, initially using a target current density and finally a detailed thermal analysis based on the electromagnetic loss estimate.

B. Electromagnetic Losses

Although the stator is very similar in construction to a conventional machine a few loss mechanisms must be specifically addressed. Relatively thin, low loss silicon steel is used to contain losses under the high-frequency operation. Special care must be taken in selecting the strand configuration in the multi-strand, multi-turn form-wound coils to contain strand losses under high-frequency operation. Commercially available lumped-parameter circuit simulators with core loss and copper eddy loss models were used to predict stator losses. These calculations were compared to results obtained with a commercial electromagnetic finite element analysis (FEA) software and published, closed-form analytical methods.

Rotor losses due to eddy-currents were predicted using a time-stepping, rotating-grid solver from the same commercial FEA software. The solution was obtained with a two-dimensional analysis that ignored the axial segmentation of the magnets and the electrical isolation between each other and the shaft. This approach overstates the losses and they were found to be insignificant when compared to the rotor windage.

C. Rotor Containment

A finite element analysis tool is used to model the winding process of the rotor sleeve with pre-impregnated carbon fiber tape including rotor temperature as a time dependent variable and carbon tape tension. Random generation of the rotor geometry node by node according to the manufacturing tolerances creates a more realistic system model.

The model static stresses found are inputs to a stress analysis tool to model the dynamic stresses in the rotor during operation. The rotor is modeled at nominal and over-speed conditions at varying temperatures.

This model is then verified by rotor burst testing. The rotor burst testing results coupled with tensile strength tests conducted on wound rings of the carbon fiber tape determine the material stress limits and therefore the rotor containment sleeve final dimensions.

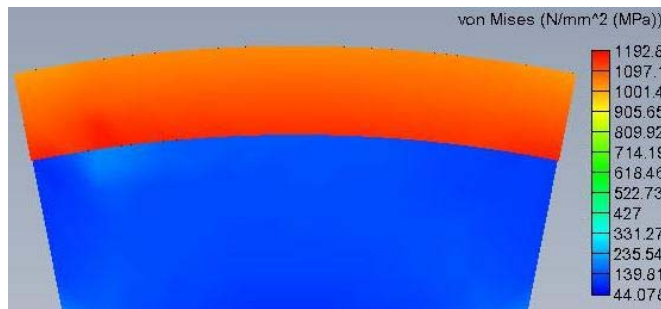


Fig. 2 Von Mises Stress Distribution at 18kRPM and 150°C

While the rotor tip speed for this design follows the rule-of-thumb of 250 m/s, the maximum allowable sleeve stress is constrained by the limits of the material and is above that chosen by Arkkio [4].

D. Rotor Dynamics

A commercially available FEA rotor dynamics software package is used to analyze the free-free natural frequencies and mode shapes of the machine. The solution approach of the tool is to lump the mass and inertia of a defined area to

create the nodes connected by massless beams. The total system model includes the rotor and bearings; however due to the nature of magnetic bearings the main housing support structure effect the rotor dynamic performance and therefore is not included in the total system model.

The magnetic bearings are modeled as dynamic supports with variable stiffness and damping. The magnetic bearings used in this machine (Figure 3) consisted of two radial support bearings, one to either end of the shaft and a separate active-thrust bearing at the coupled end to compensate for any axial loading of the coupled system. A coupling appropriate to the machine size was chosen and is modeled as a cantilevered weight.

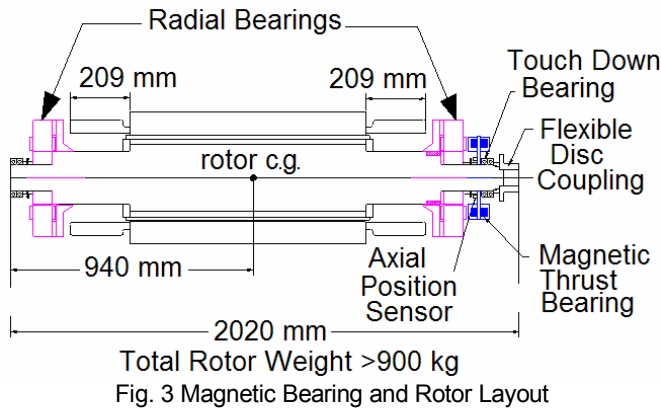


Fig. 3 Magnetic Bearing and Rotor Layout

The total rotor weight is over 900 kg and the bearing span is approximately 1.575 m. This resulted in a first forward bending mode close to the maximum operating speed of the machine. Therefore axial stiffening is added to the rotor resulting in a first forward bending mode of 21,184 cpm (Figure 4) which is 15% above the machine overspeed of 18,000 RPM.

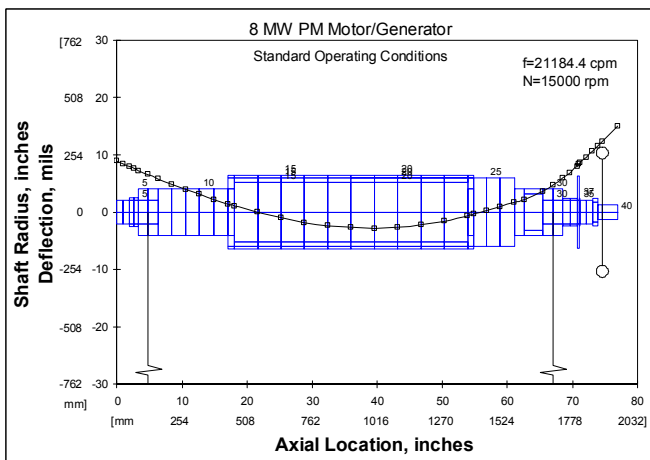


Fig. 4 First Forward Bending Mode with Geometry Plot Overlaid

The magnetic bearing control circuit design depends heavily on the expected system load response at the location of the support bearing actuators. The response plots shown, coupled end bearing (Figure 5) and non-coupled end bearing (Figure 6), are based on the maximum allowable imbalance per ISO 1940/1-1986 machine category G2,5 180° out of phase with

each other with a two plane balancing scheme.

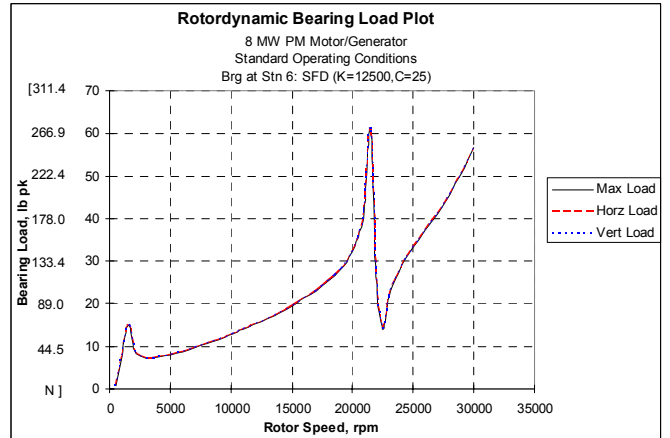


Fig. 5 Bearing Load Response Plot at Non-Coupled End

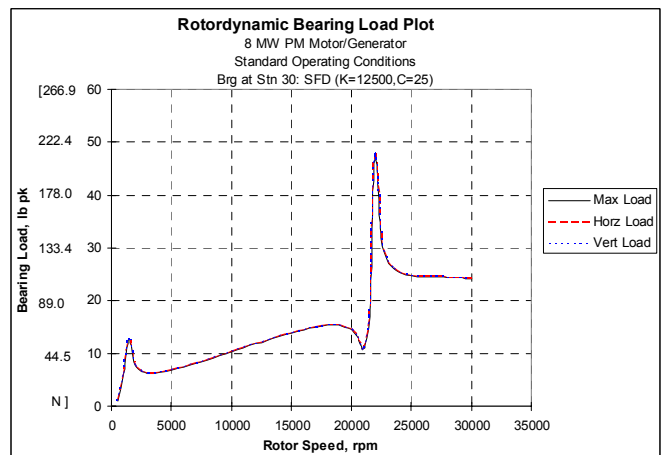


Fig. 6 Bearing Load Response Plot at Coupled End

E. Cooling

A lumped parameter model is used to model the machine geometry including rotor, stator, and cooling jacket to determine correct magnitude of cooling system approach. A maximum design temperature limit per coil insulation and carbon fiber analysis was set at 150°C.

A separate aluminum cooling jacket with a press fit to the stator back-iron extracts heat through a water/glycol cooling flow. The interference fit is determined for structural integrity as well as to reduce the insulating effects of a microscopic air gap, or contact resistance, between the cooling jacket and the stator back iron.

Curtain-air flow pulls heat out of the end turns without being choke-limited by the radial air-gap size. Although the wound-carbon-fiber sleeve acts as a thermal insulator, cooling flow through the radial air-gap minimally cools the rotor as well as cooling the stator tooth tips. Therefore cooling air flow is actively blown through a mid-stator-stack vent and into the radial air-gap and out over the stator end-turns (Figure 7). Loss inputs are determined and allocated per the electromagnetic analysis tools discussed in IV. B.

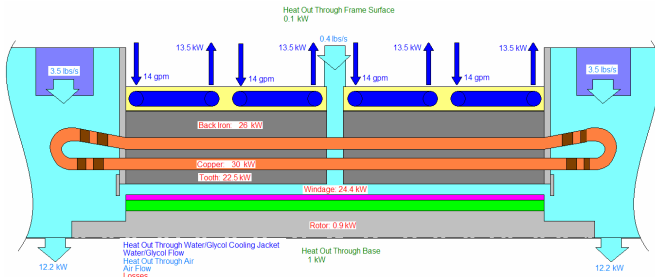


Fig. 7 Machine Cross-Section Showing Cooling System

The 3-D model of the entire machine is generated based on the cooling approach determined by the lumped parameter model. A finite-difference (FD) analysis tool is utilized by using the actual 3-D modeled assembly. Losses are represented as heat flux on the surface of the appropriate component, for example windage losses are modeled as heat flux on the surface of the rotor-containment sleeve. Cooling-flow inlets and outlets are defined; however the actual path and velocity of the flows were calculated by the FD model (Figure 8). The conclusions of the lumped-parameter model were confirmed and refined by the FD model analysis.

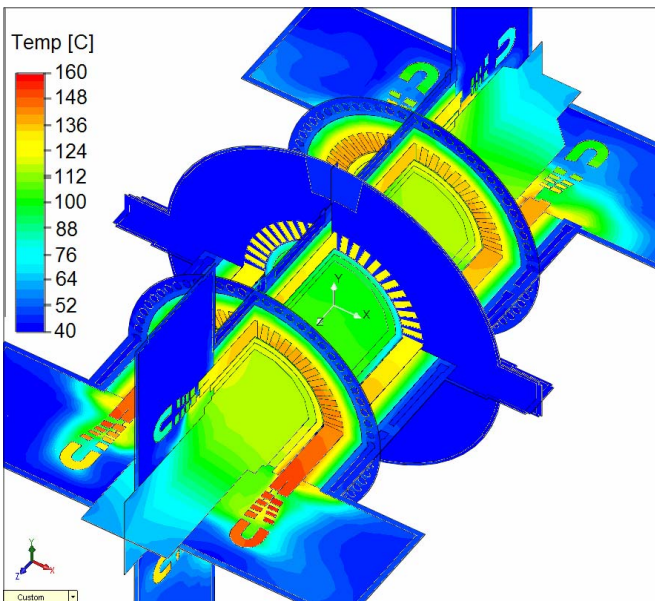


Fig. 8 Solid Model Results from Finite Difference Thermal Analysis of Machine

IV. EXPERIMENTAL RESULTS AND MODEL VALIDATION

A. Electromagnetic Model

Nelson [6] presented the details of the two-megawatt high-speed alternator (HSA) used to validate the electromagnetic model. It was shown that the open-circuit voltage was linear with speed both before and after a sudden short-circuit test and that no demagnetization was detected. The higher than predicted rotor field strength reported by Nelson was taken into account when designing the eight-megawatt machine by

adjusting the magnet material-properties, both in the drawing tolerance and in the electromagnetic model.

Electromagnetic aspects of the stator design are similar to or no different than a conventional machine and so model validation of stator quantities is of lesser concern and is not presented here.

B. Thermal Model

Two HSA machines were coupled together in a back to back configuration with one operating as a motor driving the other as a generator for testing purposes. In this configuration the cooling system was adjustable and thoroughly instrumented. The HSA machines were run with different cooling and load combinations per Table II. Both the lumped parameter and FD models were refined based on these tests until they reasonably matched the test data (Figure 9). In this way the lumped parameter and FD models were verified.

The lessons learned from the HSA testing and modeling such as the significance of the heat lost to the ambient air and through the machine feet were incorporated into the 8 MW machine thermal models.

TABLE II
Test Configuration for Cooling Model Validation

| Test Setup | Load [kW] | Liquid Cooling | Air Cooling |
|------------|-----------|----------------|-------------|
| 1 | 450 | Yes | Yes |
| 2 | No Load | Yes | Yes |
| 3 | No Load | Yes | No |
| 4 | No Load | No | Yes |

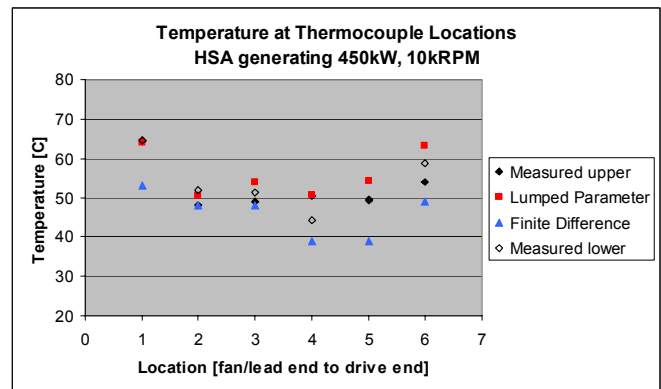


Fig. 9 Comparison Plot of Analytical Prediction and Test Data

V. CONCLUSIONS

This paper presents a viable medium-voltage, high-speed PM machine design well suited for replacement of, or coupling with, a variety of gas turbines.

The machines' stator consists of thin, low loss silicon steel laminations and multi-strand, multi-turn form-wound coils and was designed for high frequency operation using a commercially available FEA program.

The machine will be supported by one thrust and two radial magnetic bearings with ball bearings used as touch down

bearings during start up and shut down. The rotor-dynamic analysis completed shows the machine to be running sub-critical with a first forward bending mode 15% above the overspeed condition and over 20% above the nominal operating speed. The response plots show a 30 lbs peak reaction force over the operating speed range, which is well within the magnetic bearing control capabilities.

Cooling for the machine consists of cooling air entering through a mid stator vent as well as independent cooling flows over each of the end turn sections and a water/glycol cooling flow through a pressed on cooling jacket over the stator back iron. By independently blowing cooling air over the end turns the total flow rate is not restricted to the rotor air gap choke flow and can be significantly increased to meet the thermal requirements of the machine.

The analytical tools used in the design have been validated with existing hardware.

VI. REFERENCES

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VII. VITA

Cassandra Bailey graduated in 2003 from Northwestern University with a Bachelors of Science degree in Mechanical Engineering and is a member of ASME. She started as an Integrated Drive Generator Design Engineer with Hamilton Sundstrand designing an integrated variable speed transmission and AC generator for various military aircraft applications. Currently, she is with Direct Drive Systems as a Project Engineer designing high-speed medium-voltage permanent-magnet machines.

Dan Saban earned a BSEE degree from the University of Illinois - Urbana in 1992, a MSEE degree from Purdue University - West Lafayette in 1993 and a PhD degree in Electrical Engineering from the University of Wisconsin - Madison in 2006 where he specialized in electric motor analysis. Additionally, he holds an MSEE (2002) and a MSME (2003) from University of Wisconsin - Madison with focuses on

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Paulo Guedes-Pinto was graduated from the University of Sao Paulo in 1979 with a BSEE, majoring in electric machines. Paulo also pursued post-graduate studies in power systems and quality engineering. He has over 15 years of experience in the design and manufacture of motors and generators with Industrias Villares in Brazil and most recently with Direct Drive Systems. Paulo is registered professional engineer in Ontario, Canada and is a member of IEEE.