Prediction of Windage Losses of an Enclosed High Speed Composite Rotor in Low Air Pressure Environments

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Abstract

The frictional windage losses associated with non-ventilated airflows in the air gaps between the rotor and stator of a high speed rotating machine can greatly influence the rotor outer and stator inner surface temperatures. The characteristics of the radial and axial air-gap flows have been of general interest in many engineering applications. A rotating air gap flow is very complex, and in general, can be categorized as a continuum flow, slip flow, and free molecule flow, depending on the ratio of its mean free path to the air gap dimension. For a continuum flow between concentric rotating cylinders, secondary flow of rows of circumferential Taylor vortices in the air gap due to centrifugal flow instability of a curved flow at relatively high rotating speeds will typically be formed. As the air pressure in the air gap drops significantly, rarefied gas flow, departure from continuum flow, occurs when the mean free path becomes relatively large compared to the air gap dimension.

This paper has developed and summarized an analytical approach to predict high speed windage losses (rotor tip velocities up to 900 m/s) at low rotor cavity air pressures (0.1 torr to 10 torr). The predicted transient windage losses at various air pressures and high rotor speeds are compared with measured windage losses generated in continuum and slip flow regimes. The agreements between the predicted and measured windage losses are relatively well.

I. INTRODUCTION

Composite rotors have been commonly used in the modern designs of flywheel energy storage systems and pulsed power rotating electrical machines. High rotor tip speeds and lightweight machine structures are required to increase energy storage densities, which are especially critical to mobile systems. Given the large amount of energy stored in these high speed rotating systems, the rotors are generally built of high strength graphite epoxy composites. For a high speed electrical alternator, rotor field coils must be contained within multiple rotor bands under high centrifugal loading, and the high strength and stiffness associated with advanced composite materials make them suitable for such a rotor design [1].

Overheating of the rotor outer or stator inner surface composite material of a continuous duty flywheel or alternator is likely if the frictional windage loss generated in the air gaps can't be effectively removed. For a graphite epoxy rotor composite, the thermal limitation is the epoxy resin softening at elevated temperatures; as a result, machine's structural integrity at high rotating speeds degrades significantly as the resin loses its strength. For a non-ventilated high

speed composite rotor within a composite enclosure, due to the difficulties of dissipating the windage losses through composite materials which have low transverse thermal conductivities, the rotor must be operated in a partial vacuum environment to reduce the heat loss from windage drag such that the composites will not be overheated under a steady-state operating condition. However, practical limits on the rotating seals and vacuum pump capacities determine the achievable vacuum level in a mobile high speed rotating system. In order to realize a compact and mobile high speed flywheel or alternator system, a roughing vacuum pump is typically utilized to maintain a low operating pressure in the rotor cavities, and other auxiliary systems must be minimized. Under the practical capacity constraints of the shaft seals and vacuum pumps, it becomes important that windage losses at various rotor speeds and vacuum levels could be accurately predicted such that the thermal designs of the high speed composite rotors could be reliably analyzed.

In the past, experimental investigations have been conducted to measure the windage losses in rotating machines [2-5], primarily with rotor tip velocities lower than 350 m/s tested at air pressure levels near the atmospheric pressure. The windage test data for high speed rotors in partial vacuum environments is generally not available. The University of Texas Center for Electromechanics (UT-CEM) has designed and conducted a series of composite rotor spin tests to measure the windage losses and temperature distributions of a test setup at high rotor speeds and low air pressures [6,7]. The intent of the windage tests is to evaluate the windage loss prediction methods and investigate how the air-gap windage is distributed between the rotor and stator under the test conditions. The findings of the composite rotor spin tests will then be used to perform windage-related thermal design and analysis of a high speed electrical machine. This paper has developed and summarized an analytical approach to predict high speed windage losses (rotor tip velocities up to 900 m/s) at low rotor cavity air pressures (0.1 torr to 10 torr).

II. FLOW CHARACTERIZATION

The annular gap flow between concentric cylinders with a rotating inner cylinder and a stationary outer cylinder is commonly referred as a Couette flow, which results from a steady and relative motion of two parallel surfaces that are separated by a fluid. In general, the flow can be categorized as a continuum flow, slip flow, transition flow, and free molecule flow, depending on the degree of rarefaction present [8]. The division of a flow into various flow regimes is based on the value of an appropriate Knudsen number, which is a dimensionless parameter defined by the relation

$$Kn = \frac{\lambda}{L} \quad (1)$$

where λ is the molecular mean free path and L is a pertinent dimension of the flow field. The pertinent dimension L can be a body dimension or a characteristic length of the flow, such as the boundary layer thickness; however, the radial air gap dimension should be used for an annular gap flow. The Knudsen number serves to indicate the rarefaction effect.

The molecular mean free path is the average distance traveled by a gas molecule between collisions. For a gas in local thermodynamic equilibrium, it is defined by the relation [9]

$$\lambda = \frac{\mu}{\rho} \sqrt{\frac{\pi M}{2RT}} \quad (2)$$

where μ is the dynamic viscosity, ρ the density, M the molecular weight, $\bar{R}$ the universal gas constant, and T the temperature. By substituting the Sutherland law for the viscosity, a specific expression of this formula applicable to air in a range of 100 to 1900 K is obtained

$$\lambda = \frac{(3.1 \times 10^{-5})T^2}{(T + 110.4)P} \quad (3)$$

in which λ is in meters, T in Kelvins, and P in pascals.

The Knudsen number can also be related to the more conventional dimensionless parameters of fluid mechanics, the Mach number Ma and the Reynolds number Re , by the relation

$$Kn = 1.26 \sqrt{\gamma} \frac{Ma}{Re} \quad (4)$$

where γ is the specific heat ratio. The Mach number and Reynolds number indicate the importance of compressibility and viscosity effects, respectively. Based on experimental evidence [8], various flow regimes can be defined by the following limits

- (i) continuum flow if $Kn < 0.01$
- (ii) slip flow if $0.01 < Kn < 3$
- (iii) free molecule flow if $Kn > 3$

Relatively speaking, the density levels of the continuum, slip, and free molecule flows are ordinary, slightly or moderately rarefied, and highly rarefied.

The classical continuum flow of a linear Newtonian viscous fluid described by the Navier-Stokes momentum equations, derived by assuming the fluid as a continuous medium in which the viscous shear stresses are linearly proportional to the strain rates, represents the limiting case that the Knudsen number is negligibly small everywhere in the flow field. For an annular rotating continuum flow, various flow regimes exist, such as laminar, transition, vortex, and turbulent flows. Secondary flow of rows of circumferential Taylor vortices in the air gap due to centrifugal flow instability of a curved flow at relatively high rotating speeds will typically be formed.

As the air pressure in the air gap drops below the continuum flow limit, a rarefied gas flow occurs when the mean free path is no longer small or becomes relatively large compared to the air gap dimension. For a slightly or moderately rarefied gas flow, or the so-called slip flow, intermolecular and molecule-surface collisions are of equal importance, and the velocity and temperature of the air in the immediate neighborhood of the solid walls are different from those of the rotor and stator surfaces. The air appears to slip on the solid surfaces and introduces velocity and temperature discontinuities at the interfaces between the air and the boundary surfaces. The general effect of the velocity and temperature slip boundary conditions is to reduce the skin friction and heat transfer. As shown in equation (4), in the slip flow regime, the true rarefaction effect occurs in coincidence with either strong compressibility effect (large Mach number) or strong viscous effect (small Reynolds number).

For a highly rarefied flow, namely a free molecule flow, the mean free path is much larger than the characteristic length of the flow, at least several times greater than the radial air gap dimension. In the free molecule flow regime, no boundary layer is formed and the flow field is entirely undisturbed by the presence of the boundary surfaces. Under these conditions, molecular

collisions in the region between the solid surfaces are negligible and the flow phenomena are governed by the molecule-surface interaction. The molecular motion can be characterized by Maxwellian distribution, and the momentum and energy transfer occur solely from direct encounters between the gas molecules and the wall surfaces.

III. WINDAGE DRAG CALCULATIONS

A detailed description of calculating continuum flow windage drags on rotating disks and cylinders is contained in reference [10]. The windage frictional moment of a rotating disk is expressed in terms of a disk torque coefficient C_m , which is defined as

$$C_m = \frac{M_D}{\rho \omega^2 a^5} \quad (5)$$

where M_D is the disk moment on two sides of the disk, ρ the fluid density, ω the angular velocity, and a the radius of disk edge. Similarly, the windage frictional moment of a rotating cylinder is related to a cylinder torque coefficient C_f , which is

$$C_f = \frac{M_C}{\pi \rho \omega^2 a^4 L} \quad (6)$$

where M_C is the cylinder moment, a the radius of cylinder, and L the axial length of cylinder. The power loss from windage drag on a rotating machine is as follows:

$$P_w = (M_D + M_C) \omega \quad (7)$$

For non-ventilated enclosed disks, the presence of four different flow regimes, laminar flow merged boundary layers, laminar flow separated boundary layers, turbulent flow merged boundary layers, and turbulent flow separated boundary layers, have been experimentally verified. The actual disk flow regime is determined by the axial spacing ratio and disk Reynolds number. Experimentally obtained disk torque coefficients in various flow regimes have been summarized in reference [10]. For enclosed cylinders with zero axial flow in the gap, the laminar flow between rotating cylinders breaks down at certain rotational speeds and a stable flow of rows of circumferential Taylor vortices is formed [11]. The classical inviscid rotational instability criterion indicates that the Couette flow with the inner cylinder rotating and outer cylinder at rest is unstable. Viscosity somewhat stabilizes the flow. If the cylindrical air gap clearance is relatively small compared to the radius of cylinder, the flow stability is dependent on a single parameter, Taylor number, which is defined as

$$Ta = \left(\frac{\rho \omega r_i b}{\mu} \right) \left(\frac{b}{r_i} \right)^{0.5} \quad (8)$$

where r_i is the radius of rotating cylinder and b is the radial air gap clearance. The Taylor number is of the form of a clearance Reynolds number times the square root of the radial gap ratio. The laminar Couette flow persists up to a critical Taylor number of 41.3 before changing into a laminar Taylor vortex flow. Couette-type turbulent flow only occurs at very high rotational speeds. Empirical expressions of cylinder torque coefficients in laminar, transition, vortex, and turbulent flow regimes have been provided in reference [10].

As the Navier-Stokes solution is exact in the continuum flow regime where the Knudsen number of the flow is much less than 1.0, the free molecule solution is exact if the Knudsen number of the flow is much greater than 1.0. Within the free-molecule limit, where the molecular

collisions can be neglected and momentum and energy transfer result from molecule-surface interaction, the wall shear stress developed is given by [12]

$$\tau = \frac{1}{2} \rho u_o \sqrt{\frac{2kT}{\pi m}} = \frac{1}{2} \rho u_o \sqrt{\frac{2RT}{\pi}} \quad (9)$$

where ρ is the density, u_o the surface speed of rotating cylinder, k the Boltzmann constant, T the gas temperature, m the mass of molecules, and R the gas constant. The windage power loss for a free molecule flow in concentric rotating cylinders, $P_{w,C}$, can then be found by integration as follows:

$$P_{w,C} = \tau A u_o = \left[\frac{1}{2} \rho u_o \sqrt{\frac{2RT}{\pi}} \right] (2\pi a L) (u_o) \quad (10)$$

where a is the radius of rotating cylinder and L is the axial length of cylinder. Equation (10) can be rearranged by using ideal gas law and gives

$$P_{w,C} = P \left(\sqrt{\frac{2\pi}{RT}} \right) L \omega^2 a^3 \quad (11)$$

where P is the gas pressure and ω is the angular velocity. The power loss from windage drag on two end disk surfaces of a rotating inner cylinder, $P_{w,D}$, in an enclosure can be obtained by integrating the wall shear stress with respect to the disk surfaces

$$P_{w,D} = 2 \int \tau u dA = 2 \int \frac{1}{2} \rho u \sqrt{\frac{2RT}{\pi}} (u) (2\pi r dr) \quad (12)$$

where u is the velocity at a given radius of the disks. The integration of equation (12) gives

$$P_{w,D} = \left(\frac{P}{2} \right) \left(\sqrt{\frac{2\pi}{RT}} \right) \omega^2 a^4 \quad (13)$$

Couette flow in the slip flow regime with intermediate values of Knudsen number between the Navier-Stokes and free-molecule limits has been studied and approximate methods for treating the rarefied gas dynamics in this flow regime have been proposed. As the Knudsen number increases, the Couette flow solution goes smoothly from the Navier-Stokes solution to the free-molecule solution. A slip velocity correction can be included in the free-molecule wall shear stress equation to determine the windage shear in the slip flow regime [13]. The modified shear stress formula is written as follows:

$$\tau = \frac{1}{2} \rho u_o \sqrt{\frac{2RT}{\pi}} \left(\frac{2\lambda/L}{1 + 2\lambda/L} \right) \quad (14)$$

where λ is the molecular mean free path and L is the dimension of the radial air gap between concentric cylinders. The shear stress, expressed in equation (14), reduces to the Navier-Stokes stress value when $\lambda/L \ll 1$ and to the free-molecule stress value when $\lambda/L \gg 1$. Experimental results of Couette flow in concentric cylinders have confirmed the smooth transition between the Navier-Stokes and free-molecule limits that is predicted by the equation (14).

IV. WINDAGE TEST SETUP AND TEST PARAMETERS

A section view of the windage test hardware is shown in figure 1. The test rotor, 42.4 cm (16.694 in.) diameter and 12.1 cm (4.762 in.) long, is made of multiple preloaded composite

rings pressed onto a solid titanium hub. The rotor was attached to a spin pit turbine through a quill shaft. A steel stator enclosure with a G-10 composite liner surrounds the rotor and supports instrumentation. The stator was hung from a reaction torque transducer mounted on the spin pit lid. The torque transducer measured the transient windage drag on the rotor by monitoring the torque imparted on the stator assembly. Pressure transducers were utilized to provide transient air pressure measurement in the spin pit, axial air gaps between the rotor and stator end disks, and radial air gap between the concentric rotor and stator cylinders. The rotor rim and end disk surface temperatures were measured by five infrared sensors, which were mounted in the stator structure and were flush with the inside surface of the G-10 liner. Thermocouples were used to measure the stator temperature at various locations. The composite rotor was designed to be operated at a rotor tip velocity of 900 m/s at temperatures up to 93°C (200°F).

Windage tests have been performed in low pressure environments with air pressure levels at 10 torr, 1 torr, and 0.1 torr. Three peak rotor tip velocities, 333 m/s (15,000 rpm), 614 m/s (27,600 rpm), and 892 m/s (40,000 rpm) were selected for the windage tests. The conditions and assumptions used in the windage calculations are described as follows:

- Average rotor outer radius at rest = 8.347 in.
- Average stator inner radius = 8.487 in.
- Constant axial air gap = 0.484 in
- Initial air-gap air temperature before tests = 25°C
- At 27,600 rpm, average radial air-gap air temperature rises at 0.1 torr and 1 torr were calculated by Computational Fluid Dynamic (CFD) analysis [14] and were considered as baseline temperature rises
- Average radial air-gap air temperature rises at other speeds = (baseline temperature rises) $\times$ (rotor speed ratio)²
- Average axial air-gap air temperature rise = half of the average radial air-gap air temperature rise due to low disk surface velocities and large axial air gaps
- Average air-gap air temperature rise at 10 torr = 1.5 $\times$ (average air-gap air temperature rise at 1 torr)

A summary of windage test parameters are included in table I. Only the mechanical rotor spinning growths, based on the measured transient rotor speed curves, were considered in the calculation of the transient radial air gap size variation, and any potential air gap changes caused by thermal effect were neglected.

Depending on the magnitude of the rotor cavity pressure, the airflows in the radial and axial air gaps can be either continuum or slip flows. According to the previously described flow regime criterion, the flow regimes in the radial and axial air gaps are

- Continuum flows in both the radial and axial air gaps for windage tests with 10 torr air pressure in rotor cavity
- Slip flow in radial air gap and continuum flows in axial air gaps for windage tests with 1 torr air pressure in rotor cavity
- Slip flows in both the radial and axial air gaps for windage tests with 0.1 torr air pressure in rotor cavity

The air-gap flow regime changes from a continuum flow to mixed continuum and slip flows or a slip flow entirely as the air pressure dropped below a 2-torr level. The free molecule flow only occurs at very low pressures, typically below a pressure level of 0.001 torr. In the windage tests,

the partition between the continuum and slip flows was calculated to be approximately 2 torr, which was in line with the CFD prediction [14].

V. WINDAGE TEST RESULTS AND LOSS PREDICTIONS

Eight windage tests, with the planned rotor cavity pressures and rotor peak speeds shown in table I, were carried out and the actual transient rotor speeds, spin pit pressures, and rotor cavity pressures were recorded during each test. The intent of these windage tests was not only to validate the analytical windage predictions in different flow regimes, but also to investigate how the windage losses were distributed between the rotor and stator at the selected rotor peak velocities. In most of the tests, the rotor speeds were held at the peak values for prolonged period of time to allow for deep thermal diffusions in both the rotor and stator, and the time durations at the peak speeds were determined by the limiting temperatures of the rotor and stator composite material. The windage test results, which include the measured rotor speeds, measured spin pit pressures, measured rotor cavity pressures in the radial air gap, measured windage losses by stator torque transducer, and predicted windage losses based on the calculation methods described in section III, are plotted in figures 2 to 9. Although both the spin pit and rotor cavity pressures were measured, the transient rotor cavity pressures in the radial air gap were used in the calculations of the windage losses for both the rotating cylinder and end disks.

As shown in figures 2 to 9, the rotor cavity pressures in the radial air gap are higher than those in the spin pit for all the windage tests, and the pressure differences between the rotor cavity and spin pit become pronounced as the rotor speeds increase. It is obvious that the high rotor cavity pressures in the radial air gap were due to the additional static pressure heads caused by rotating centrifugal effect. Comparing the predicted and measured windage losses, the predicted losses are consistently higher than the measured losses for all the tests, with differences being generally within 15%. The differences between the predictions and measurements were mainly caused by two reasons. First, limited by the available rotor cavity pressure data in the axial air gaps, the measured radial air-gap pressures were used to calculate the windage losses in both the radial and axial air gaps; in doing so, the axial windage losses were over-predicted since the axial air-gap pressures are radius dependent and have a maximum value only at the outer edge of the end disks. However, the disk windage losses are highly radius dependent as indicated in equations (5) and (13), most of the disk windage losses are generated at large radii. Also, depending on the flow regimes, only 20% to 35% of the total windage loss in a windage test was contributed by the windage drags occurred in the axial air gaps. Therefore, the use of radial air-gap pressures in the axial air-gap windage loss calculations will certainly over-predict the losses; however, the over-predictions should not be significant. Secondly, the neglected dimensional changes of the radial air gap due to thermal effect in the windage loss calculations could also contribute to the discrepancies. It was observed that the measured stator inner-diameter surface temperatures were higher than the measured rotor outer-diameter surface temperatures. Depending on the air pressures, rotor speeds, and the time when the temperatures were taken, these temperatures differed between 1°C to 20°C. In general, the temperature differences were within 10°C. As a result of a warmer stator, the radial air gap dimension would be larger than those assumed in the loss calculations, in which only the rotor's spinning growths were considered. Larger actual radial air gaps in tests mean lower measured windage losses.

Sensitivity analyses on predicted windage losses of test#2 (27,600 rpm and 0.1 torr), test#5 (27,600 rpm and 1 torr), and test#8 (27,600 rpm and 10 torr) were conducted by varying the air-gap air pressure, air-gap air temperature, and radial air-gap dimension. The baseline air pressures, air temperatures, and radial air-gap dimension used in the windage loss calculations were those measured in the radial air gap, predicted by CFD analysis and estimated by assumed scaling relationship, and calculated by considering rotor's spinning growths at various speeds, respectively. The spin pit pressure and rotor cavity pressure in the radial air gap were assumed as the lower and upper bounds of the air pressures in the sensitivity analysis. A $\pm 20\%$ variation (or $\pm 20^\circ\text{C}$) of the baseline air temperature was assigned as the temperature uncertainty range. As for the radial air gap dimension, a ± 0.002 in uncertainty range, which included a machining tolerance and possible inaccuracy associated with the spinning growth calculation, was assumed in the sensitivity analysis. The sensitivity analysis results are summarized in table II. Within the assigned uncertainty ranges, the air pressure has a more dominant effect on the windage loss as compared to the air temperature and radial air gap dimension. Although the measured windage loss of the test#5 is outside the loss range of the sensitivity analysis, the difference is not significant. The windage tests performed at high rotor speeds are difficult to be controlled well and analyzed accurately. It is understood that some of the complex realities, such as the corner flows near the outer edges of the end disks and the eccentricities of the rotor at high speeds, were not taken into account in the windage loss calculations. However, the relative good agreement between the measured and predicted windage losses indicates that the windage drag calculation methods described in this paper can predict frictional windage losses well in the continuum and slip flow regimes.

VI. CONCLUSIONS

A series of composite rotor spin tests to measure the windage losses at high rotor speeds and low air pressures has been performed and analyzed. The intent of the windage tests is to validate the windage loss prediction methods described in this paper and to investigate how the air-gap windage is distributed between the rotor and stator under the test conditions.

The flow regimes associated with a wall-driven air-gap air flow are very complicated. Depending on the magnitudes of the air pressure, air temperature, air gap dimension, and rotor speed, the flow regimes are typically categorized as continuum flow, slip flow, and free molecule flow. The traditional Navier-Stokes shear stress calculation, which imparts zero-slip flow boundary conditions at the interfaces between the fluid and the solid, is only applicable to the continuum flow. As the air pressure drops below a 2-torr level in the windage test, the air-gap air begins to slip on the solid surfaces and introduces velocity and temperature discontinuities at the interfaces between the air and the boundary surfaces.

Analytical methods have been developed and summarized to predict the high speed windage losses in low air pressure environments. The predicted windage losses agree well with the test results, and the agreements are generally within 15% for all the windage tests. It is believed that the deviations between the predictions and measurements were caused by certain simplified assumptions used in the prediction calculations and some test complexities could not be reliably analyzed.

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TABLE I
SUMMARY OF WINDAGE TEST PARAMETERS

Test #	Rotor Cavity Pressure (torr)	Rotor Peak Speed (rpm)	Average Rotor Outer Radius (in)	Average Rotor Tip Velocity (m/s)	Average Radial Air Gap (in)	Axial Air Gap (in)	Average Radial Air-Gap Air Temperature Rise at Peak Speed (°C)	Average Axial Air-Gap Air Temperature Rise at Peak Speed (°C)
1	0.1	15,000	8.353	333	0.134	0.484	20	10
2	0.1	27,600	8.366	614	0.121	0.484	70	35
3	0.1	40,000	8.387	892	0.100	0.484	150	75
4	1	15,000	8.353	333	0.134	0.484	25	13
5	1	27,600	8.366	614	0.121	0.484	80	40
6	1	40,000	8.387	892	0.100	0.484	170	85
7	10	15,000	8.353	333	0.134	0.484	38	19
8	10	27,600	8.366	614	0.121	0.484	120	60

TABLE II
SENSITIVITY ANALYSIS RESULTS OF PREDICTED WINDAGE LOSSES

Windage Test	Measured Loss (W)	Predicted Baseline Loss (W)	Loss at Spin Pit Pressure (W)	Loss at Baseline Temperature +20°C (W)	Loss at Baseline Temperature -20°C (W)	Loss at Baseline Air Gap +0.002 in (W)	Loss at Baseline Air Gap -0.002 in (W)	Loss Range (W)
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#2 (at 1500s)	386	397	315	412	390	393	401	304~416
#5 (at 1500s)	557	646	604	659	632	639	652	583~665
#8 (at 900s)	1935	2097	1798	2079	2115	2092	2103	1775~2121

Figure 1. Section view of windage test hardware

Figure 2. Results of windage test #1 (15,000 rpm rotor peak speed at 0.1 torr)

Figure 3. Results of windage test #2 (27,600 rpm rotor peak speed at 0.1 torr)

Figure 4. Results of windage test #3 (40,000 rpm rotor peak speed at 0.1 torr)

Figure 5. Results of windage test #4 (15,000 rpm rotor peak speed at 1 torr)

Figure 6. Results of windage test #5 (27,600 rpm rotor peak speed at 1 torr)

Figure 7. Results of windage test #6 (40,000 rpm rotor peak speed at 1 torr)

Figure 8. Results of windage test #7 (15,000 rpm rotor peak speed at 10 torr)

Figure 9. Results of windage test #8 (27,600 rpm rotor peak speed at 10 torr)