ROTOR WINDAGE LOSSES FOR LUNDELL ALTERNATORS

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ABSTRACT

The Lundell alternator is being investigated for power system applications. At the high speeds required, power loss due to windage can become high and result in significiant heating of the alternator. This loss must be incorporated in the cooling design. The velocity profiles expected in these alternators are highly turbulent, and the Reynolds numbers are well above the levels previously reported; the available data could only be extrapolated. For this reason, a range of experimental data was generated to permit accurate calculation of the windage power loss for Lundell alternator design.

Windage tests were conducted on two concentric rotor-stator configurations in ambient air, producing Reynolds numbers as high as 100,000. Cylindrical rotors 10 and 12 inches in diameter were operated up to 24,000 rpm, and an 8 inch diameter Lundell rotor was operated up to 36,000 rpm. Gap-to-radius ratios of 0.01, 0.02 and 0.04 were tested. Tests were also conducted with axial slots machined on the stator surfaces to simulate alternator winding slots.

Using the data obtained, a method was developed to calculate the windage loss for any Lundell alternator given the geometry and cavity conditions.

THE LUNDELL ALTERNATOR is a solid-rotor brushless machine that is being investigated for use in future power system applications (1, 2).* Since there are no electrical or mechanical connections to the solid rotor, its peripheral speed is limited only by the maximum stress of the rotor material. Therefore, the Lundell alternator provides maximum reliability when directly coupled to a high-speed turbine. Windage losses become significant for large rotor diameters, high speeds, and high-density high-viscosity gases used in the rotor cavity for some applications. In addition to the reduction in system efficiency, high windage loss results in significant heat generation in the alternator. This heat load must be included in the alternator thermal design.

As shown in figure 1, the rotor of a Lundell alternator consists of two separate magnetic sections with north poles on one section and south poles on the other. The interpolar space is filled with a nonmagnetic metal to obtain a smooth rotor. For purposes of windage analysis the typical Lundell rotor is considered to consist of a series of cones and cylinders.

Taylor, Vohr and Wendt (3, 4, 5) have studied turbulent flow for concentric rotating cylinders with Reynolds numbers (Re) as high as 40,000 (here Re is defined using the gap as the characteristic dimension). These high Reynolds numbers, however, were obtained using large radial clearances. Reference (6) presents a preliminary study of windage losses for concentric rotating cylinders with Re as high as 100,000 and gap-to-radius ratios of 0.01 to 0.04. This is the range of gap sizes being considered for space power alternators. These small magnetic gaps are used to reduce field coil power, field coil leakage, and rotor pole leakage thereby reducing rotor diameter and weight. The choice of clearance must also allow for other design constraints such as magnetic spring rate (magnetic attraction force/unit eccentricity), synchronous reactance, windage loss and cooling gas flow rate.

No specific information was available on enclosed rotating conical sections. The most appropriate information came from the studies of Daily, et al., on enclosed rotating disks (7, 8).

To provide design engineers with information for the prediction of windage losses, drag coefficient versus Re data were generated for concentric cylin-

*Numbers in parentheses designate References at end of paper.

ders. A Lundell-shaped rotor-housing configuration was tested with gap-to-radius ratios of 0.01, 0.02 and 0.04. A correlation method was developed in reference (9) to calculate the total windage loss for this rotor. And this method has been used to predict, with good agreement, the measured windage loss from two differently sized Lundell rotors.

APPARATUS

The test apparatus, shown in figure 2, consisted of a solid rotor (the Lundell rotor is shown here) mounted on ball bearings lubricated with an air-oil mist and a concentric housing (stator) attached to a reaction-measuring support table. A more complete description of the test support equipment is given in reference (6).

ROTOR-HOUSING CONFIGURATIONS - Dimensions of the cylindrical and Lundell rotor and housing configurations tested are shown in figures 3 and 4, respectively. The rotors were made from a heat-treated forging of a low-alloy vanadium steel and the housings were aluminum. The housings were made so that they could be removed without changing the rotor alignment. The Lundell housing consisted of five individual sections, corresponding to the changes in cross-section, each split axially for ease in disassembly. All parts were doweled or keyed so that alignment of the parts could be reproduced upon reassembly.

TESTS PERFORMED - The 10-inch- and 12-inch-diameter cylindrical rotors were tested at speeds up to 24,000 rpm for each of three gaps corresponding to 0.01, 0.02 and 0.04 gap-to-radius ratio. After each test of the smooth-surface 10-inch-diameter housing, the housing was remachined with 60 equally-spaced axial slots. Tests were run for several slot-depth and -width combinations for each gap.

The Lundell rotor was tested at speeds up to 36,000 rpm for each gap. Tests were performed on the center cylindrical section of the housing and the complete housing assembly. After completing tests on the smooth-surface housing, the center cylindrical housing section was remachined with 60 equally spaced axial slots to simulate alternator winding slots. These slots were 0.119-inch wide by 0.010-inch deep.

Each test was rerun several times to verify reproducibility of the data. All tests were performed in ambient air.

INSTRUMENTATION - Instrumentation consisted of rotational speed, torque and temperature sensors.

Rotational speed was measured using a magnetic pickup and 60-tooth gear combination on the shaft of the drive system. The signal generated was sent to a counter and recorded. Rotational speed was controlled and measured to within 0.1 percent.

Torque measurements were made using the reaction torque device shown in figure 2. Strain gages are located on four flexure arms so that they sense torque about only one axis. This axis was made to coincide with the axis of the rotor and its housing. Torque developed about this sensing axis produced a proportional strain. All tare torques remained constant and were compensated for by calibration. The torque unit was calibrated by hanging accurately known weights from a calibration arm to produce known torques. Calibration was accurate within 0.03-inch-pound although measurements could be taken at less than 0.01-inch-pound increments. The strain gage output from the Wheatstone bridge circuit was measured on an IDVM - Integrating Digital Voltmeter. Low-frequency vibration resulted in a 1 to 2 percent oscillation in the signal output. Torque measurements on the cylindrical rotors at 5,000 rpm and the Lundell rotor at 8,000 rpm are approximately 5 percent accurate, the accuracy increasing with the speed. All temperatures were measured using iron-constantan type J thermocouples. Thermocouples were located axially along the housing at the centerline of the radial air gap.

RESULTS AND DISCUSSION

A typical Lundell alternator rotor consists of conical and cylindrical sections. Windage is the viscous loss in the annular gap between the rotor and the stationary outer housing. The frictional drag force, F, may be expressed as

$$F = \lambda A \frac{(\rho U^2)}{2} \qquad (1)$$

where

λ = drag coefficient
A = surface area of rotor
ρ = density
U = surface speed of rotor

For a cylinder where

$$T = FR \tag{2}$$

equation (1) can be rewritten in terms of measurable quantities as

$$\lambda = \frac{T}{\rho \pi \omega^2 R^4 L} \tag{3}$$

where

T = torque
ω = angular velocity of rotor
R = radius of rotor
L = length of rotor

Windage W is then

$$W = T\omega \tag{4}$$

The nondimensional flow parameter used to correlate the data is Reynolds number, defined as

$$R_e = \frac{Ud}{\nu} \tag{5}$$

where the characteristic dimension d is the radial gap and ν is the kinematic viscosity.

For the data presented, it was assumed that the pressure in the gap remained ambient since the housings were open-ended.

CYLINDRICAL ROTORS - Nondimensional curves of drag coefficient versus Reynolds number are plotted in figure 5 for both the smooth 10-inch- and 12-inch-diameter rotor-housing configurations. Three gap-to-radius-ratio 0.01, 0.02 and 0.04 are represented. Agreement between each gap-to-radius-ratio for both the 10-inch rotor and 12-inch rotor is within 5 percent. Since the length-to-diameter ratio of the 10-inch rotor is just over twice that of the 12-inch rotor, this ratio does not appear to be a significant factor in the drag coefficient. Above a Reynolds number of 15,000, deviation between the drag coefficients for the different gaps is less than ±5 percent. This result is supported by Taylor in reference (3). Based on previous works (e.g., Wendt, ref. (5)), the drag coefficients at Reynolds numbers below 10,000, increase with gap-to-radius ratio. This is seen in the 10-inch rotor data and the two smaller gaps of the 12-inch rotor. However, the torque values for the large gap (0.236 in.) of the 12-inch rotor were extremely low below 10,000 Reynolds number (less than 0.15 in.-lb) and subject to large errors.

The curves of drag coefficient versus Reynolds number change slope at an approximate Reynolds number of 5,000 to 7,000. This corresponds to the change from vortex to turbulent flow, as shown in reference (4). When the 10-inch housing was tested with various slot configurations, the change in slope occurred at progressively lower Reynolds numbers as the slot depth and width were increased. This resulted in higher drag coefficients at the higher Reynolds numbers. The maximum Reynolds number obtained was 110,000.

LUNDELL ROTOR - Figure 6 presents curves of drag coefficient versus Reynolds number for the 8-inch-diameter dylindrical section of the Lundell rotor. Three radial clearances are represented for both the smooth and slotted housings. The increased losses due to the axial slots were 30 to 50 percent at the higher Reynolds numbers.

The same trends for the smooth housing are seen here as in the curves of the 10-inch and 12-inch rotors. At Reynolds numbers above 15,000, the drag coefficients for the three radial clearances agree within ±5 percent for either the smooth or slotted housing. The transition to turbulent flow also occurs in the same Reynolds number range (5,000-7,000). However, the drag coefficients for the 8-inch-diameter cylindrical section are approximately 20 percent higher than for the 10-inch and 12-inch rotor. It is suspected that the presence of the 5-inch-diameter auxiliary cylinders caused an increased end effect on the cylindrical housing.

The complete Lundell rotor and housing were tested with both a smooth and slotted main cylindrical section at each of three gaps. The windage power loss for the 0.160 inch radial clearance unslotted housing configuration at 24,000 and 36,000 rpm was 1.6 and 4.7 kilowatt respectively. For the slotted housing at 36,000 rpm, the power loss rose to 6.0 kilowatt. The increase was equal to the additional loss contributed by the slotted cylindrical section. Different slot configurations will produce different overall power losses.

CALCULATION METHOD FOR WINDAGE LOSSES - In order to calculate the windage loss for a complete Lundell alternator, an empirical method was developed. In the development of a calculation procedure it was found that the drag coefficients for the 8-inch-diameter cylindrical section of the Lundell rotor provided the best correlation for the complete Lundell rotor using the available test data. It must be noted however, that for purely cylindrical windage calculations the 10- and 12-inch drag coefficient, should be used.

The empirical method developed to calculate the total Lundell-rotor windage losses treats the conical sections as cylinders having a diameter equal to the minor conical-section diameter, a length equal to the length of the conical surface and a gap equal to the clearance in the auxiliary-cylinder section. Windage from each of the cylindrical sections is then calculated using equations (3) and (4) with the drag coefficients found from the 8-inch-diameter cylindrical section. The drag coefficients are obtained by calculating the Reynolds number in that section and then using the curves in figure 6. A comparison between the measured and calculated values is given in table 1. The results calculated are within 10 percent of the experimental values.

To verify the applicability of this calculation method, the measured windage losses of two other Lundell alternators were compared. In reference (10) a three-phase 120/208-volt Lundell alternator with an output rating of 14.3 kilovolt-ampere and a 0.75 lagging power factor was tested at 36,000 rpm. The major diameter of this rotor was 3.3 inches. The test points were obtained using krypton in the rotor cavity at several pressure levels. Figure 7 shows the measured power loss for this alternator and the calculated loss as a function of cavity pressure. Good agreement is found with the higher-pressure test data and the deviation at the low-cavity-pressure point is within the experimental uncertainty for the low torque value recorded.

The other Lundell rotor used in the correlation was tested by AiResearch Manufacturing Company as part of a NASA contract. This rotor had a major diameter of 6.58 inches and was operated in air at 24,000 rpm over a range of pressure levels. Figure 8 compares the measured and the calculated windage losses. The results calculated are within 8 percent of the experimental values for the complete range of operating pressures.

SUMMARY

Three rotors, a 10-inch- and 12-inch-diameter cylinder and an 8-inch-diameter Lundell-type rotor were operated within stationary concentric housing. Three gaps were used for each rotor, corresponding to approximately 1, 2 and 4 percent of the rotor radius. The center cylindrical section of the Lundell housing was tested with a smooth surface and with axial slots to simulate alternator winding slots. It was found that power loss for the slotted stationary cylindrical (simulated armature) housing configurations operating in the turbulent flow regime was 30 to 50 percent higher than for the smooth housing. For concentric cylinders, the range of Reynolds number was extended to 100,000. Using the data obtained, a method was developed to calculate the windage loss for Lundell alternators.

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TABLE 1 - COMPARISON OF CALCULATED AND MEASURED WINDAGE LOSSES FOR LUNDELL-SHAPED ROTOR WITH SMOOTH CONFORMING HOUSING

	Radial air gap (in.)				
	0.039	0.080			0.160
Speed, rpm	24,000	12,000	24,000	36,000	24,000
Experimental loss, watts	1,819	259	1,675	5,110	1,593
Calculated loss, watts	1,894	278	1,743	5,140	1,485
$\frac{\text{Experimental-Calculated}}{\text{Experimental}}$, percent	-4.1	-9.3	-5.9	-0.58	6.8

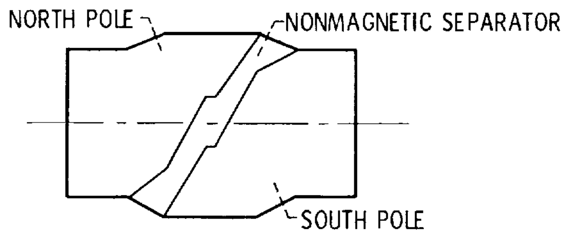

Figure 1. - Typical Lundell solid-rotor.

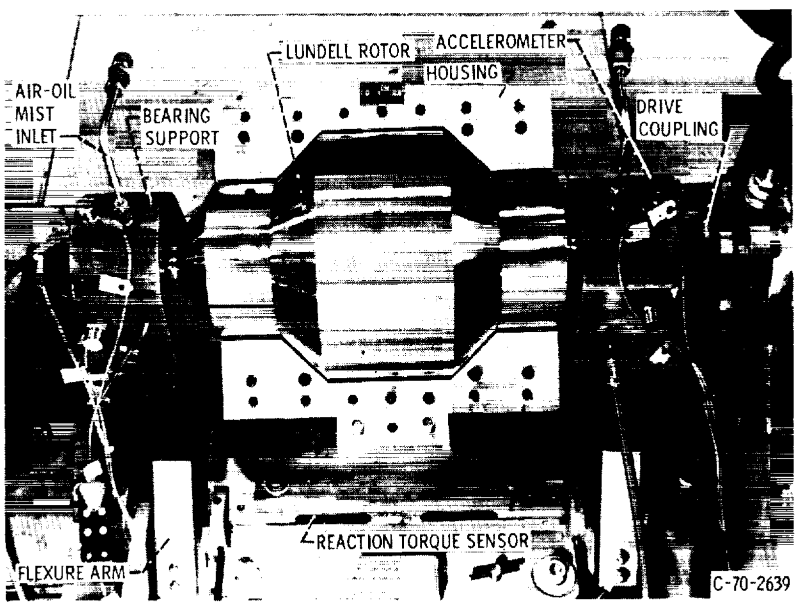

Figure 2. - Windage test apparatus for Lundell-type rotor.

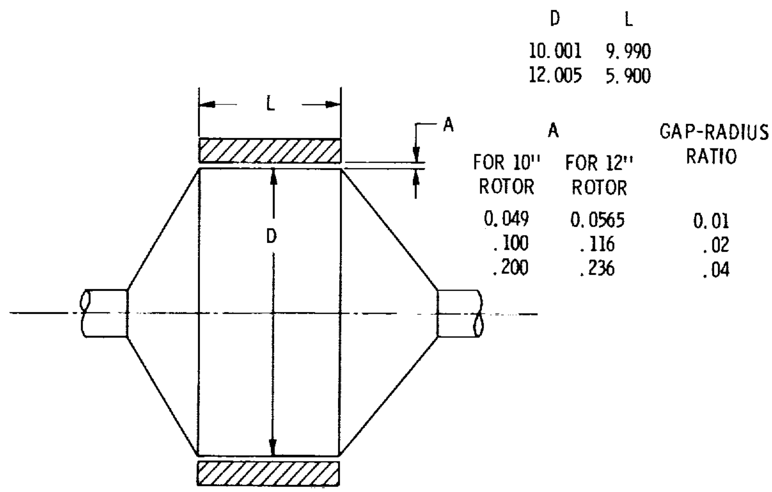

Figure 3. - Cylindrical rotor and housing configurations.

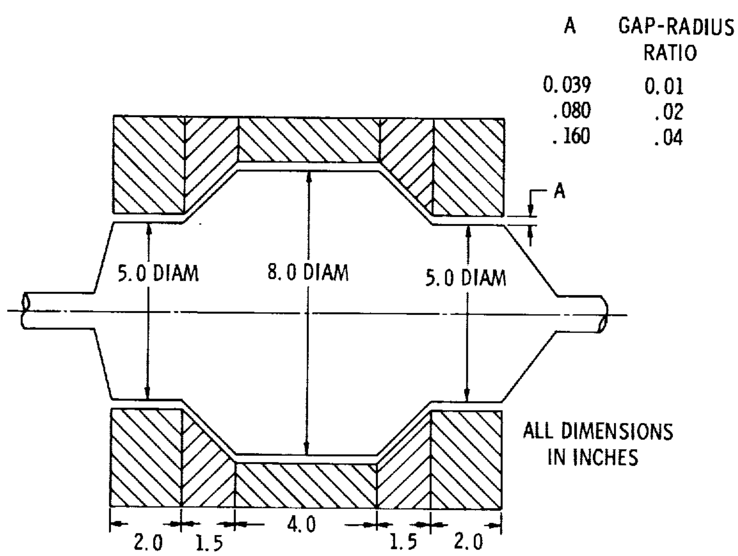

Figure 4. - Lundell rotor and housing configuration.

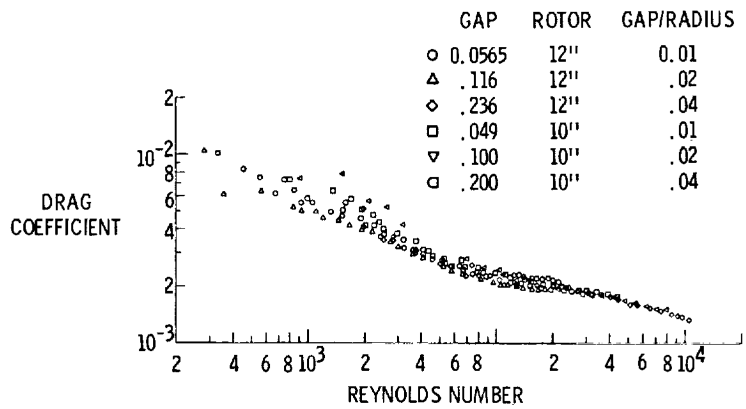

Figure 5. - Drag coefficient for a rotating cylinder in a smooth stationary cylindrical housing.

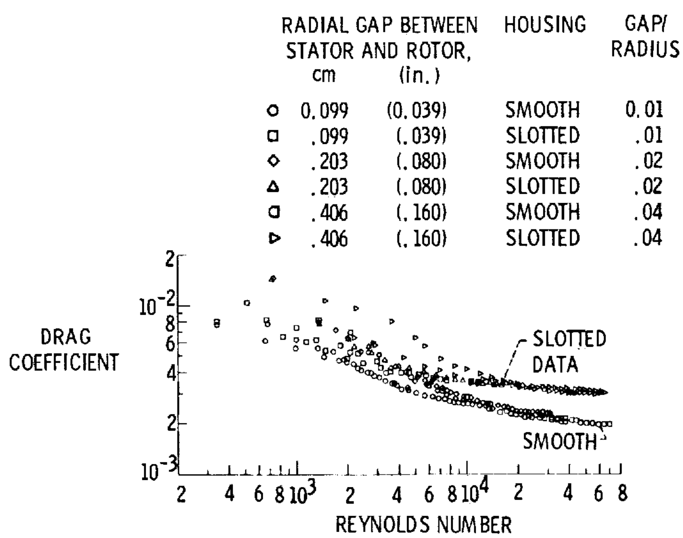

Figure 6. - Drag coefficient as function of Reynolds number for rotating cylinder in smooth and slotted stationary concentric housings. Rotor diameter, 20.32 centimeters (8 in.); rotor length, 10.16 centimeters (4 in.); slot depth, 0.025 centimeter (0.010 in.)

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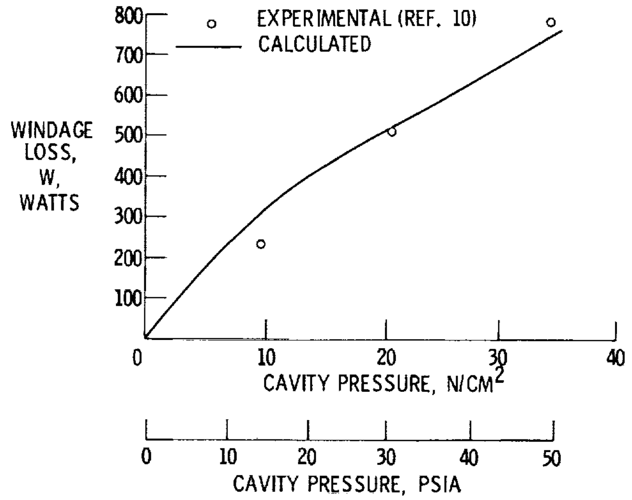

Figure 7. - Lundell alternator windage losses in krypton at 36000 rpm compared to analytical results.

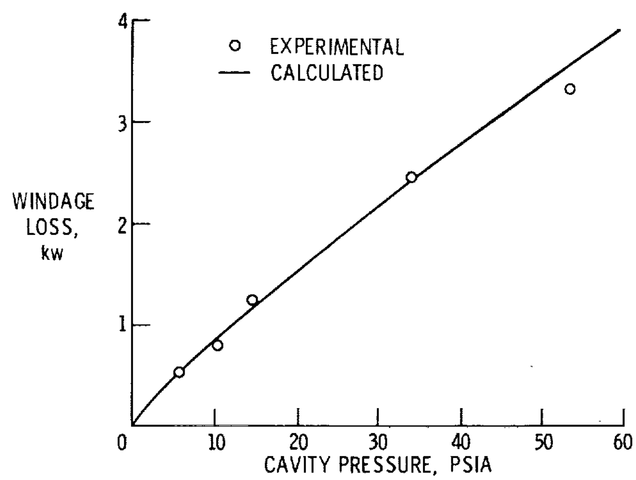

Figure 8. - Lundell alternator windage losses in air at 24000 rpm compared to analytical results.