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CORRELATION OF WINDAGE-LOSS DATA FOR A LUNDELL ALTERNATOR

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SUMMARY

Viscous torque (windage) of a Lundell-shaped rotor having a major diameter of 20.3 centimeters (8 in.) rotating within a concentric housing was measured at speeds up to 36 000 rpm. The center cylindrical section of the housing was tested with both a smooth surface and axial slots to simulate alternator winding slots. For concentric cylinders, the range of Reynolds numbers was extended to 70 000. Three radial clearances, 1, 2, and 4 millimeters (0.03, 0.08, and 0.16 in.), corresponding to 1, 2, and 4 percent of the rotor radius, were tested. Power loss for the slotted cylindrical housing was as high as 50 percent greater than that for the smooth cylindrical housing. For the rotor housing configurations tested the power loss approached a limit for slot depths greater than 0.25 millimeter (0.010 in.).

INTRODUCTION

The Lundell alternator is being investigated for use in Brayton-cycle space power systems (refs. 1 and 2). In high-power Brayton applications the potential for high windage losses is significant because of the large rotor diameters, the high rotational speeds, and the possibility of either high rotor cavity pressure levels or high-molecular-weight - high-viscosity gas in the rotor cavity. In addition to the system performance penalty, high windage loss results in significant heat generation in the electric generator, a loss that must be included in the generator thermal design.

The typical Lundell rotor comprises two geometric shapes, cylinders and cones. Taylor, Vohr, and Wendt (refs. 3 to 5) have studied turbulent flow (Reynolds numbers above 7000) for concentric rotating cylinders with Reynolds numbers as high as 40 000. These high Reynolds numbers, however, were obtained by using large radial clearances. Reference 6 presents a preliminary study of windage losses for concentric rotating cylinders with Reynolds numbers as high as 100 000 and gap-to-radius ratios of 0.01 to 0.04. Currently these are the gap sizes considered for space power alternators. These

radial clearances, or magnetic gaps, are used in order to reduce field-coil power and leakage, rotor pole leakage, and thus rotor diameter and weight. The choice of a radial clearance must also allow for other design constraints such as magnetic spring rate (magnetic attraction force per unit of eccentricity), synchronous reactance, windage loss, and cooling gas flow rate.

No information was available on enclosed rotating conical sections. The information most relative to conical sections was data from studies of enclosed rotating disks in the turbulent flow regime (refs. 7 and 8).

Therefore, to provide designers with additional information for the prediction of windage losses, a Lundell-shaped rotor housing was tested with gap-to-radius ratios of 0.01 to 0.04. The range of Reynolds numbers investigated was 1000 to 70 000.

SYMBOLS

A characteristic area (wetted area)

d characteristic dimension (rotor to stator, radial gap thickness)

F frictional force

K kinetic energy per unit volume

L length

R radius

Re Reynolds number

T torque

U velocity

W windage viscous power loss

λ drag coefficient (friction factor)

ν kinematic viscosity

ρ density

ω rotational speed

APPARATUS AND PROCEDURE

Rotor-Housing Configuration

The test rig (fig. 1) consisted of a solid rotor mounted on air-oil-mist-lubricated ball bearings and a housing (stator) attached to a "floating" support table.

Dimensions for the Lundell rotor-housing configuration are given in figure 2. The rotor was made from a heat-treated forging of a low-alloy vanadium steel. Surfaces were ground smoother than a 400-nanometer- (16-μin. -) rms finish. A variable-speed dc motor connected to the rotor by a splined coupling was used as the drive system.

The aluminum housing consisted of five individual sections, each split axially so that it could be mounted or removed without affecting the test rotor. All parts were doweled or keyed so that the alinement of the parts could be reproduced on reassembly. Also, a one-piece housing (fig. 3) representative of more conventional ground-based Lundell alternator configurations was tested. This housing differed from the previous housing in that it did not conform to the conical section of the rotor.

Testing Sequence

Successive tests were performed on the center cylindrical section, the center cylindrical section with the conical transition section(s), and the complete housing assembly. After completion of tests of the smooth-surface housing with the 0.99-millimeter (0.039-in.) air gap, the center cylindrical housing section was remachined with 60 equally spaced axial slots to simulate alternator winding slots. These slots were 3.02 millimeter (0.119 in.) wide by 0.25 millimeter (0.010 in.) deep. After testing with the slotted geometry the entire housing (five sections) was remachined to a 2.03-millimeter (0.080-in.) radial gap and the tests were repeated. This procedure was followed for a radial gap of 4.06 millimeter (0.160 in.) except that three slot depths, 0.13, 0.25, and 1.27 millimeters (0.005, 0.010, and 0.050 in.), were consecutively machined and tested. Finally, the one-piece nonconforming smooth housing with a 4.06-millimeter (0.160-in.) radial gap was tested. This test sequence is summarized in table I.

The rotor was run at speeds up to 27 000 rpm for the 0.99-millimeter (0.039-in.) gap and 36 000 rpm for the other two gaps. Each test was rerun several times to verify reproducibility of data. All tests were performed in ambient air.

INSTRUMENTATION

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

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

Torque measurements were made by using a reaction torque device (fig. 1). Strain gages were located on four flexure arms so that they sensed 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 or loads 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 3.4 millinewton-meters (0.03 (lb)-(in.)), although measurements could be taken at less than 1.1-millinewton-meter (0.01-(lb)-(in.)) increments. The strain-gage output from the Wheatstone bridge circuit was measured on an integrating digital voltmeter. Low-frequency vibration resulted in a 1- to 2-percent oscillation in the signal output. The resultant torque measurements taken at 8000 rpm were approximately 5 percent accurate, the accuracy increasing with speed.

All temperatures were measured by using iron-constantan type-J thermocouples. Thirteen thermocouples were mounted axially along the housing, each located at the centerline of the radial air gap. The thermocouple holes in the housing were 1 millimeter (0.04 in.) in diameter. Locations of the thermocouples are shown in figure 2.

A 1.72-newton-per-square-centimeter (2.5-psi) differential-pressure transducer was used to measure the head rise across the conical sections of the Lundell rotor. Locations of the pressure taps are also shown in figure 2.

THEORETICAL CONSIDERATIONS FOR ROTATING CONCENTRIC CYLINDERS

A typical Lundell alternator rotor consists of conical and cylindrical sections. The windage consists primarily of the viscous losses in the annular gap between the rotor and the stationary outer housing. The power loss due to drag on a rotating cylinder of radius R and rotational speed ω is given by

$$W = \omega RF \tag{1}$$

where F is the frictional force on the cylinder. By definition

$$F \equiv \lambda A K \tag{2}$$

where λ is the friction factor or drag coefficient, A is a characteristic area, and K is the dynamic head. Applying equation (2) to cylinders gives

$$F = \lambda 2\pi R L \frac{\rho U^2}{2} \tag{2a}$$

where $U = \omega R$. Combining terms results in

$$F = \lambda \rho \pi \omega^2 R^3 L \tag{3}$$

Substituting into equation (1) gives

$$W = \lambda \rho \pi \omega^3 R^4 L \tag{4}$$

Since the primary measured parameters in this test were torque and speed, equation (1) can be rewritten as

$$W = T\omega \tag{1a}$$

where $T = FR$. Solving equation (4) for the drag coefficient in terms of the measured parameters yields

$$\lambda = \frac{T}{\rho \pi \omega^2 R^4 L} \tag{5}$$

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

$$Re = \frac{Ud}{\nu} \tag{6}$$

where the characteristic dimension d is the radial gap.

DISCUSSION OF RESULTS

The total windage loss for a Lundell alternator comprises losses from the various axial sections of the rotor, which must be considered individually. These sections include the 20.3-centimeter- (8-in.-) diameter main cylinder, the two 12.70-centimeter- (5-in.-) diameter auxiliary cylinders, and the two identical conical transitions. The test results for each section are presented. Ultimately, a correlation of the data was used to estimate the windage loss for an actual alternator.

Main Cylindrical Section

The plots of torque as a function of speed for the 20.32-centimeter- (8.0-in.-) diameter cylindrical section are shown in figure 4. Data are presented for a smooth stationary concentric housing at each of the three radial clearances, 1, 2, and 4 millimeters (0.04, 0.08, and 0.16 in.). The plots show that the torque loss decreases as the radial clearance is increased.

Drag coefficients were obtained by using these torque data and equation (5). The drag coefficients are plotted as a function of Reynolds number for the 20.32-centimeter- (8.0-in.-) diameter cylindrical section in figure 5. Data are presented for both the smooth and slotted stationary concentric housing at each of the three radial clearances, 1, 2, and 4 millimeters (0.04, 0.08, and 0.16 in.). There were 60 equally spaced axial slots in the housing, 3 millimeters (0.119 in.) wide by 0.25 millimeter (0.010 in.) deep. Two assumptions were made in calculating the Reynolds number. The first assumption was that the pressure in the radial gap remained ambient since the housing was not enclosed. The second was that the average temperature in the gap was equal to the temperature at the center of the axial length of the gap. Typically there was a 6° C (10° F) variation in the gap temperature from the center to the end at 24 000 rpm and a 22° C (40° F) variation at 36 000 rpm. Maximum center temperatures were approximately 97° C (200° F) for the smooth housing and 110° C (230° F) for the slotted housing.

At Reynolds numbers above 1.5×10^{4}, the drag coefficients for the three radial clearances agree within 10 percent of each other for either the smooth or slotted housing. This independence of drag coefficient and radial clearance at high Reynolds number was also reported by Taylor (ref. 3).

Table II shows that at the same speed (24 000 rpm) the ratio of losses with slotted housings to those with smooth housings varied from 1.32 for the 1-millimeter (0.04-in.) radial clearance to 1.51 for the 4-millimeter (0.16-in.) radial clearance. Also, the data indicate that power loss for a smooth housing is more dependent on radial clearance than that for a slotted housing.

Figure 6 presents a comparison of the data with previously published information on drag coefficients for concentric rotating cylinders. Data were selected which most closely match the gap-to-radius ratio of 0.04. The curves indicate that drag coefficient may be a function of rotor diameter as well as gap-to-radius ratio and Reynolds numbers. Not enough data are presently available to determine an exact correlation.

The 4-millimeter- (0.16-in.-) radial-clearance housing was tested for four different slot depths (0, 0.13, 0.25, and 1.27 mm; 0, 0.005, 0.010, and 0.050 in.). Slot width (3mm; 0.119 in.) and number of slots (60, equally spaced) were maintained constant. Figure 7 shows the relation of slot depth to drag coefficient. The 0.13-millimeter (0.005-in.) slot produced approximately 20 percent increased losses, while the 0.25-millimeter (0.010-in.) slot produced a 50 percent increase. When the slot depth was increased to 1.27 millimeters (0.050 in.) the drag coefficients showed no further increase above that of the 0.25-millimeter (0.010-in.) slot.

Main Cylindrical Section and Transition Section

Tests were performed on a housing configuration consisting of the main cylindrical section and one conical transition section. A negligible difference was found between the torque values obtained for this configuration and that of the 20.3-centimeter (8.0-in.) cylindrical section alone, for each of the three gaps. Since the addition of the conical transition section must contribute a nonzero drag, it was concluded that the drag coefficient in the main cylindrical section had been lowered and the losses thereby reduced. As discussed in reference 9, the enclosed conical rotor section acts as a pump developing approximately 0.17 newton per square centimeter (0.25 psia) head rise at 24 000 rpm, which produces an axial flow. As mentioned by Gazley in reference 10, axial flow inhibits the development of Taylor vortices, reduces losses, and thereby decreases the drag coefficient.

When both conical transition sections were mounted to the main cylindrical section, through flow was eliminated because of symmetry. Figure 8 presents torque-speed curves for this configuration at each of the three radial clearances. The 2- and 4-millimeter (0.08- and 0.16-in.) curves fall very close. Based on the previous cylindrical data, this result is contrary to expectations. The torque values for a given rotor speed would be expected to decrease as the radial clearance is increased.

The losses due to the addition of the conical transition section were obtained by subtracting data for the main cylinder from those for the main cylinder plus the two transition sections. This calculation method produces inaccuracies of ±35 percent.

Lundell-Shaped Housing

The plots of windage loss as a function of speed for the complete rotor-unslotted-housing configuration are shown in figure 9. The power losses at 24 000 and 36 000 rpm for the 4-millimeter (0.16-in.) radial clearance are approximately 1.6 and 4.7 kilowatts, respectively.

Table III shows the loss distribution for the complete Lundell rotor at 24 000 rpm. The data for each section were obtained by the subtraction of sets of data (i.e., auxiliary cylindrical data equal complete Lundell data minus main cylinder and conical transition data). Table III shows that, for the rotor configuration tested, approximately 70 to 78 percent of the total loss occurs in the main cylindrical section. The conical transition section and the auxiliary cylindrical section contribute approximately 14 to 25 percent and 5 to 8 percent, respectively. The results also indicate that the windage is an inverse function of the radial clearance.

A one-piece housing (fig. 3) which was representative of more conventional ground-based Lundell alternator configurations and had a radial clearance of 4 millimeters (0.16 in.) and a smooth surface was tested. Losses were approximately 11 percent lower for the conforming housing. This was probably due to the additional vortices formed in the open area of the nonconforming housing.

Correlation of Windage Loss Data for Lundell Alternators

In order to calculate windage loss for a complete Lundell alternator it was necessary to determine a method for handling the conical transition sections. It was found that the loss contributed by these sections could be approximated by treating them as cylinders with a diameter equal to the minor conical section diameter and a length equal to the length of the conical surface. The gap dimension used was the radial clearance. Equation (4) could then be used to calculate the windage loss from each of the sections of the Lundell rotor. The drag coefficient to be used for each section is obtained by calculating the Reynolds number in that section and then using the curves for drag plotted against Reynolds number for smooth or slotted stators. Results calculated by this method proved to be within 7 percent of the experimental values. A comparison between calculated and measured values is given in table IV. Table III indicates that the experimental losses for the auxiliary section are less than the calculated losses obtained by using the drag coefficient presented in figure 5. A sample calculation is given in the appendix.

An additional comparison was made to verify the validity of the calculation method. The windage loss of a previously tested Lundell alternator was calculated, and the results were compared with the available experimental data, as reported by Repas and

Edkin in reference 11. The test alternator was a three-phase 1200-hertz 120/208-volt Lundell alternator with an output rating of 14.3 kilovolt-amperes and a 0.75 lagging power factor at 36 000 rpm. Dimensions for the machine are shown in figure 10. For the test, krypton was used as the gas in the rotor cavity.

Figure 11 shows the measured power loss for the alternator and the calculated loss as a function of cavity pressure and Reynolds number. The results are in good agreement with the test data at 20.7 and 34.5 newtons per square centimeter (30 and 50 psia). The deviation shown at 10.1 newtons per square centimeter (14.7 psia) is within the experimental uncertainties for such low torque values.

SUMMARY OF RESULTS

A Lundell-shaped rotor having a major diameter of 20.3 centimeters (8 in.) was rotated in ambient air at speeds up to 36 000 rpm. The center cylindrical section of the housing was tested with a smooth surface and with axial slots to simulate alternator winding slots. For concentric cylinders, the range of Reynolds number was extended to 70 000. Further, the radial clearances were reduced from those previously investigated to 1 to 4 percent of rotor radius, as appropriate to electric generators. It was found that

1. Power loss for the slotted stationary cylindrical (simulated armature) housing configurations operating in the turbulent flow regime was 20 to 50 percent higher than for the smooth housing.
2. For the rotor-stator configurations tested, the power loss approached a limit for slot depths greater than 0.025 centimeter (0.010 in.).
3. Power loss decreased as the radial clearance was increased.
4. Power losses for the conforming housing were approximately 11 percent lower than for the nonconforming housing, which was representative of more conventional ground-based Lundell alternator configurations.

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112-27.

APPENDIX - SAMPLE CALCULATION OF WINDAGE LOSS FOR LUNDELL-SHAPED ROTOR-HOUSING CONFIGURATION

The following conditions are given for the calculation of windage loss for the Lundell-shaped rotor-housing configuration:

Rotational speed, rpm	24 000
Radial air gap, d, m (in.)	0.00203 (0.080)
Radius of main cylindrical section, R, m (in.)	0.1016 (4.0)
Axial length of main cylindrical section, L, m (in.)	0.1016 (4.0)
Radius of auxiliary section, R, m (in.)	0.0635 (2.5)
Axial length of auxiliary section, L, m (in.)	0.0508 (2.0)
Axial length of conical section, L, m (in.)	0.038 (1.5)
Angle of conical section, deg	45
Gas in rotor cavity	air
Cavity pressure, N/cm^2 (psia)	9.86 (14.3)

The windage loss in the auxiliary section is calculated from the following:

(1) Gas temperature, estimated to be 328 K (130° F)

(2) Absolute viscosity μ, where

$$\mu = 1.93\times10^{-5}\ \frac{(N)(sec)}{m^2}\ (1.3\times10^{-5}\ \text{lbm/ft sec})$$

(3) Density ρ, where

$$\rho = \frac{P}{RT}$$

$$\rho = \frac{9.86\times28.97}{82.05\times328} = 1.055\ \text{kg/m}^3\ (0.066\ \text{lbm/ft}^3)$$

(4) Kinematic viscosity ν, where

$$\nu = \frac{\mu}{\rho}$$

$$\nu = \frac{1.93\times10^{-5}}{1.055}$$

$$\nu = 1.83\times10^{-5}\ \mathrm{m^2/sec}\ (1.97\times10^{-4}\ \mathrm{ft^2/sec})$$

(5) Reynolds number Re, where

$$\mathrm{Re} = \frac{Ud}{\nu}$$

$$\mathrm{Re} = \frac{(800\pi)(0.0635)(0.00203)}{1.83\times10^{-5}}$$

$$\mathrm{Re} = 1.76\times10^{4}$$

(6) Drag coefficient λ, where from figure 5

$$\lambda = 2.45\times10^{-3}$$

From equation (4) the windage loss is then

$$W = \lambda\pi\rho\omega^3R^4L$$

$$W = (2.45\times10^{-3})(3.14)(1.055)(800\pi)^3(0.0635)^4(0.0508)$$

$$W = 106\ \text{watts/auxiliary section}$$

For the windage loss in the conical section, the radius R, is the same as that for the auxiliary section, 0.0635 meter (2.5 in.); the drag coefficient λ is 2.45×10^{-3}; and the length of the conical section L is $0.038/\cos 45^{\circ}$, or 0.053 meter (2.1 in.). The windage loss is then

$$W = \lambda \pi \rho \omega^3 R^4 L$$

$$W = (2.45\times10^{-3})(3.14)(1.055)(800\pi)^3(0.0635)^4(0.053)$$

$$W = 113 \text{ watts/conical section}$$

For the windage loss in the main cylindrical section, the Reynolds number is

$$Re = \frac{(800\pi)(0.1016)(0.00203)}{1.83\times10^{-5}}$$

$$Re = 2.8\times10^4$$

and the drag coefficient is $\lambda = 2.3\times10^{-3}$. The windage loss is then

$$W = \lambda \pi \rho \omega^3 R^4 L$$

$$W = (2.3\times10^{-3})(3.14)(1.055)(800\pi)^3(0.1016)^4(0.1016)$$

$$W = 1305 \text{ watts}$$

The total windage loss is

$$W_T = 1305 + 2(106) + 2(113)$$

$$W_T = 1743 \text{ watts}$$

REFERENCES

1. Ingle, Bill D.; and Wimmer, Heinz L.: Experimental Performance of a 10 kW, 1200 Hertz Brayton Cycle Alternator and Controls for Space Power. Presented at the AIAA Fifth Intersociety Energy Conversion Engineering Conference, Las Vegas, Nev., Sept. 21-24, 1970.
2. Guentert, Donald C.; and Johnsen, Roy L.: Study of Brayton Cycle Power Generation System Using SNAP-8 Nuclear Reactor as an Energy Source. NASA TN D-5751, 1970.
3. Taylor, G. I.: Fluid Friction Between Rotating Cylinders. I - Torque Measurements. Proc. Roy. Soc. (London), Ser. A, vol. 157, no. 892, Dec. 2, 1936, pp. 546-564.
4. Vohr, John: Experimental Study of Superlaminar Flow Between Nonconcentric Rotating Cylinders. NASA CR-749, 1967.
5. Wendt, F.: Turbulente Strömungen Zwischen Zwei Rotierenden Konaxialen Zylindern. Ing.-Archiv., vol. 4, no. 6, Sept. 1933, pp. 577-595.
6. Gorland, Sol H.; Kempke, Erwin E., Jr.; and Lumannick, Stacey: Experimental Windage Losses for Close Clearance Rotating Cylinders in the Turbulent Flow Regime. NASA TM X-52851, 1970.
7. Daily, James W.; and Nece, Ronald E.: Roughness and Chamber Dimension Effects on Induced Flow and Frictional Resistance of Enclosed Rotating Disks. Tech. Rep. 27, Massachusetts Inst. Tech., Hydrodynamics Lab., May 1958.
8. Daily, J. W.; Ernst, W. D.; and Asbedian, V. V.: Enclosed Rotating Disks with Superposed Throughflow: Mean Steady and Periodic Unsteady Characteristics of the Induced Flow. Rep. 64-16, Massachusetts Inst. Tech., Hydrodynamics Lab. (AROD-2500-2, AD-443060), Apr. 1964.
9. Gorland, Sol H.; and Wood, James C.: Pumping Characteristics of a Rotating Truncated Cone. NASA TM X-52953, 1971.
10. Gazley, Carl, Jr.: Heat-Transfer Characteristics of the Rotational and Axial Flow Between Concentric Cylinders. Trans. ASME, vol. 80, no. 1, Jan. 1958, pp. 79-90.
11. Repas, David S.; and Edkin, Richard A.: Performance Characteristics of a 14.3-Kilovolt-Ampere Modified Lundell Alternator for 1200 Hertz Brayton-Cycle Space-Power Systems. NASA TN D-5405, 1969.

TABLE I. - SUMMARY OF TEST SEQUENCE

Test	Radial gap, mm		Slots	Slot size, mm		Center cylinder	Conical sections	Auxiliary cylinders
	Nominal	Actual		Depth	Width			
1	1	0.99	No	----	----	Yes	No	No
2	1	.99	No	----	----	Yes	Yes	No
3	1	.99	No	----	----	Yes	Yes	No
4	1	.99	Yes	0.25	3.02	Yes	No	No
5	1	.99	Yes	.25	3.02	Yes	Yes	No
6	1	.99	Yes	.25	3.02	Yes	Yes	Yes
7	2	2.03	No	----	----	Yes	No	No
8	2	2.03	No	----	----	Yes	Yes	No
9	2	2.03	No	----	----	Yes	Yes	Yes
10	2	2.03	Yes	.25	3.02	Yes	No	No
11	2	2.03	Yes	.25	3.02	Yes	Yes	No
12	2	2.03	Yes	.25	3.02	Yes	Yes	Yes
13	4	4.06	No	----	----	Yes	No	No
13	4	4.06	No	----	----	Yes	Yes	No
15	4	4.06	No	----	----	Yes	Yes	Yes
16	4	4.06	Yes	.13	3.02	Yes	No	No
17	4	4.06	Yes	.13	3.02	Yes	Yes	No
18	4	4.06	Yes	.13	3.02	Yes	Yes	Yes
19	4	4.06	Yes	.25	3.02	Yes	No	No
20	4	4.06	Yes	.25	3.02	Yes	Yes	No
21	4	4.06	Yes	.25	3.02	Yes	Yes	Yes
22	4	4.06	Yes	1.27	3.02	Yes	No	No
23	4	4.06	Yes	1.27	3.02	Yes	Yes	No
24	4	4.06	Yes	1.27	3.02	Yes	Yes	Yes
[a]25	4	4.06	No	----	----	---	---	---

[a]Test run with nonconforming housing.

TABLE II - LOSS COMPARISON OF SMOOTH AND SLOTTED

HOUSING FOR THREE AIR GAPS

[Sixty axial slots 0.302 cm (0.119 in.) wide by 0.025 cm (0.010 in.) deep spaced 6^{o} apart; rotational speed, 24 000 rpm.]

Housing	Radial air gap, cm (in.)					
	0.099 (0.039)		0.203 (0.080)		0.406 (0.160)	
	Torque					
	(N)(m)	(lb)(in.)	(N)(m)	(lb)(in.)	(N)(m)	(lb)(in.)
Smooth	0.56	5.0	0.52	4.6	0.44	3.9
Slotted	.74	6.6	.71	6.3	.66	5.9
Ratio of slotted- to smooth-housing torque	1.32		1.37		1.51	

TABLE III. - DISTRIBUTION OF LOSSES FOR FULL LUNDELL-SHAPED

ROTOR WITH SMOOTH CONFORMING HOUSING

[Rotational speed, 24 000 rpm.]

Section	Radial air gap, cm (in.)					
	0.099 (0.039)		0.203 (0.080)		0.406 (0.160)	
	Power loss, watts	Percent of total loss	Power loss, watts	Percent of total loss	Power loss, watts	Percent of total loss
20.32-cm- (8.0-in.-) diameter cylindrical	1420	78	1305	78	1109	70
Conical transition	257	14	228	14	399	25
12.70-cm- (4.00-in.-) diameter cylindrical	142	8	142	8	85	5
Total	1819	100	1675	100	1593	100
Transition section plus 12.70-cm- (5.00-in.-) diameter cylindrical sections	399	22	370	22	484	30

TABLE IV. - COMPARISON OF CALCULATED AND MEASURED LOSSES

FOR LUNDELL-SHAPED ROTOR WITH SMOOTH

CONFORMING HOUSING

[Rotational speed, 24 000 rpm.]

Section	Radial air gap, cm (in.)					
	0.099 (0.039)		0.203 (0.080)		0.406 (0.160)	
	Experimental loss, watts	Calculated loss, watts	Experimental loss, watts	Calculated loss, watts	Experimental loss, watts	Calculated loss, watts
12.70-cm- (5.00-in.-) diameter cylindrical	1420	1420	1305	1305	1109	1109
20.32-cm- (8.00-in.-) diameter cylindrical	142	230	142	212	85	182
Conical transition	257	244	228	226	399	194
Total	1819	1894	1675	1743	1593	1485
$\frac{\text{experimental - calculated}}{\text{experimental}}$	-4.1 percent		-5.9 percent		6.8 percent	

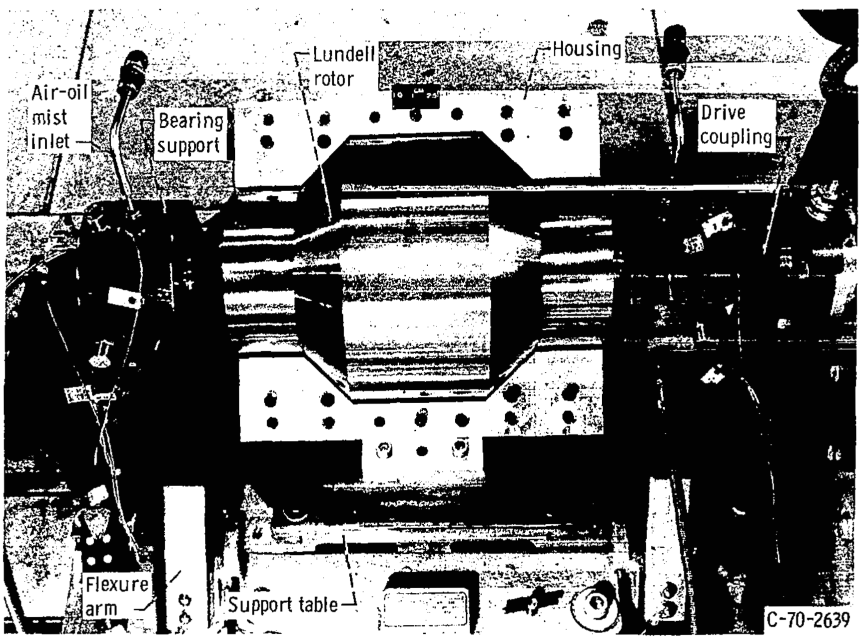

Figure 1. - Windage test apparatus for Lundell-type rotor.

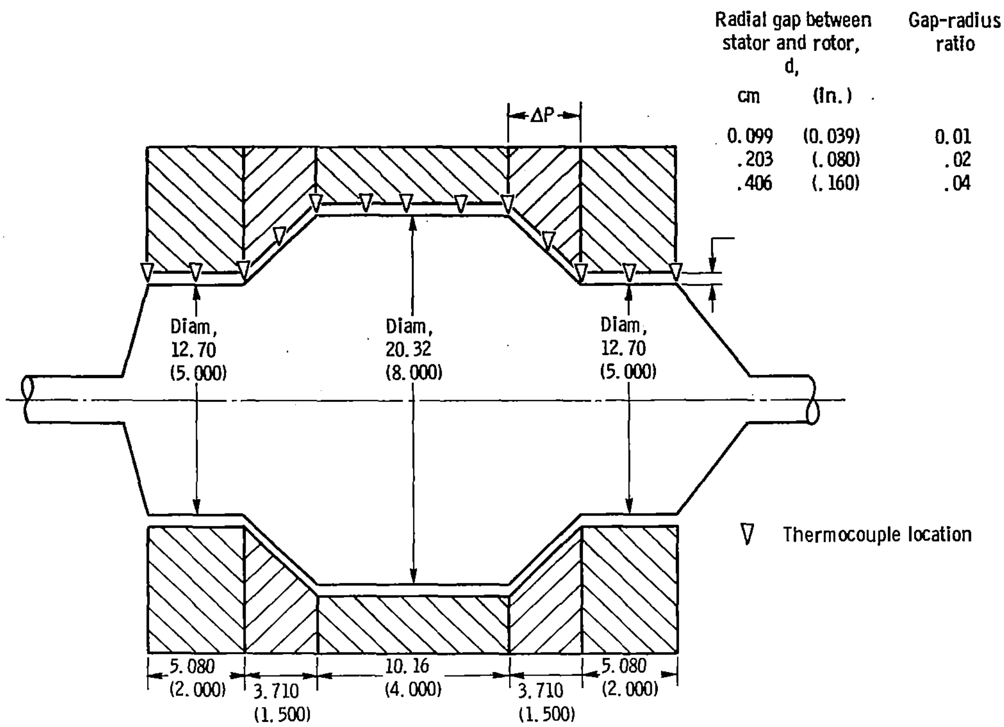

Figure 2. - Rotor-housing configuration. (Dimensions are in centimeters (in.).)

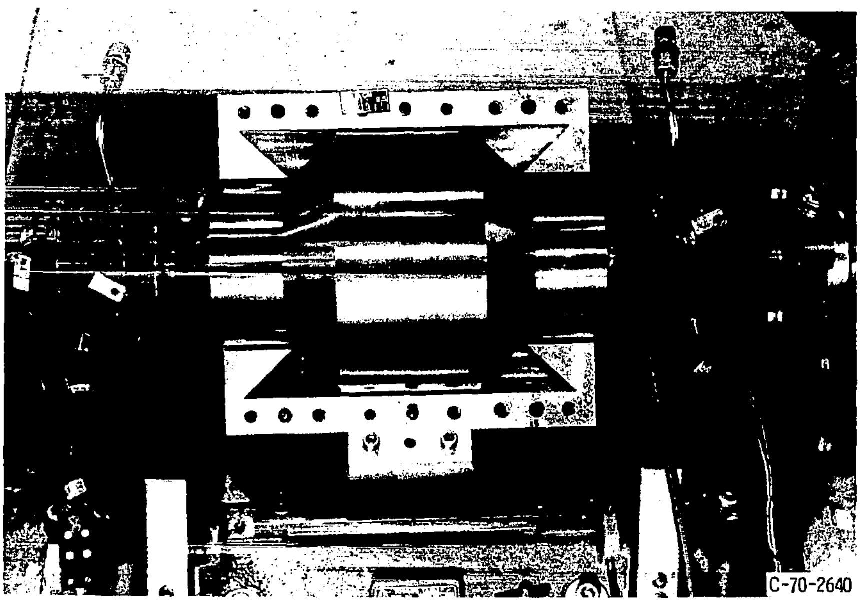

Figure 3. - Modified Lundell housing.

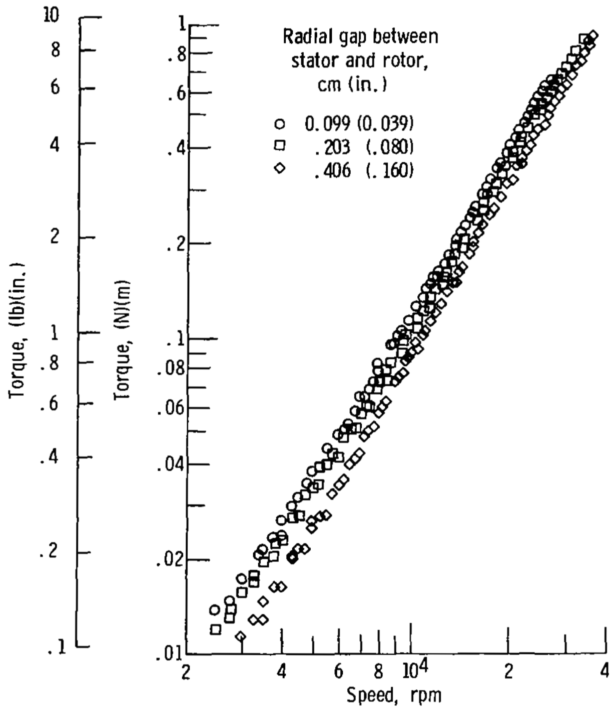

Figure 4. - Torque as function of speed for rotating cylinder in smooth stationary concentric housing. Rotor diameter, 20.32 centimeters (8.0 in.); rotor length, 10.16 centimeters (4.0 in.).

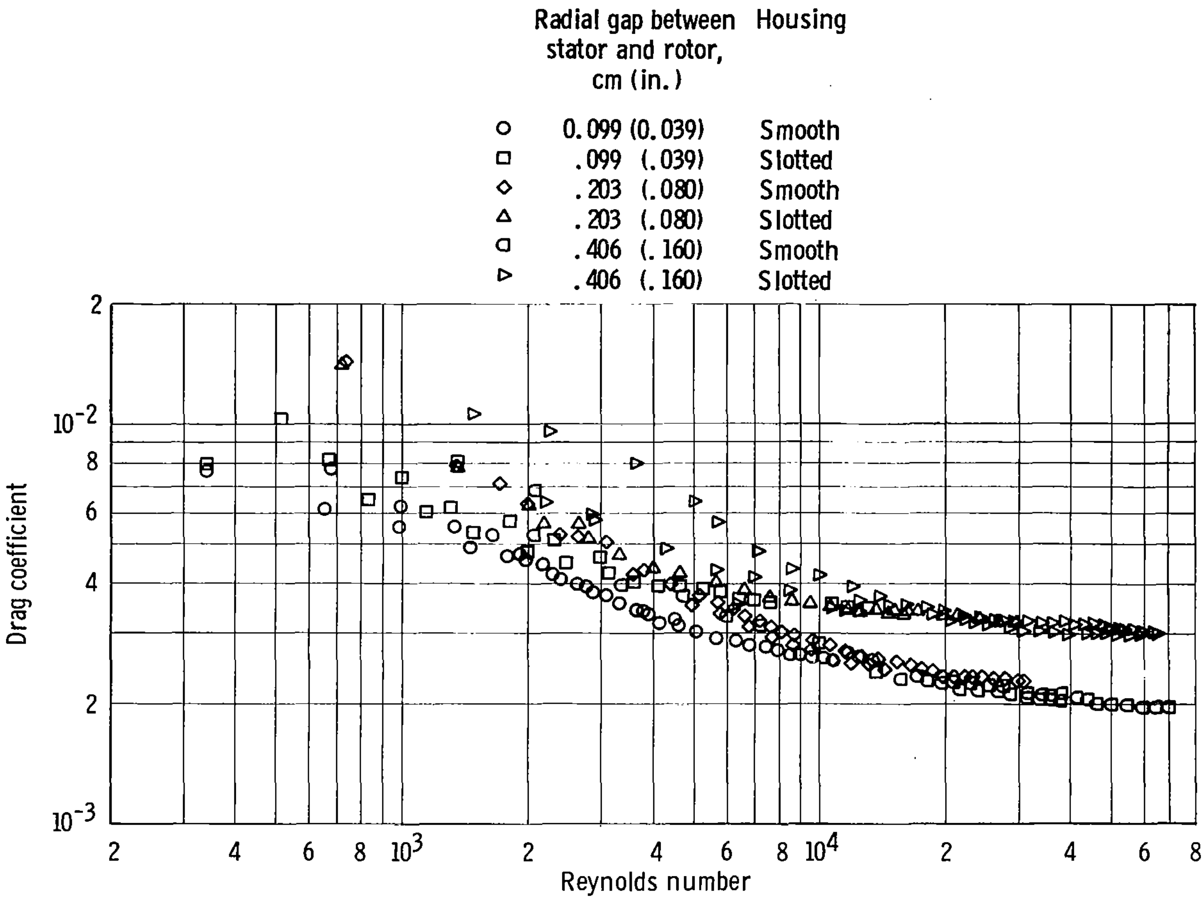

Figure 5. - Drag coefficient as function of Reynolds number for rotating cylinder in smooth and slotted stationary concentric housings. Rotor diameter, 20.32 centimeters (8 in.); rotor length, 10.16 centimeters (4 in.); slot depth, 0.025 centimeter (0.010 in.).

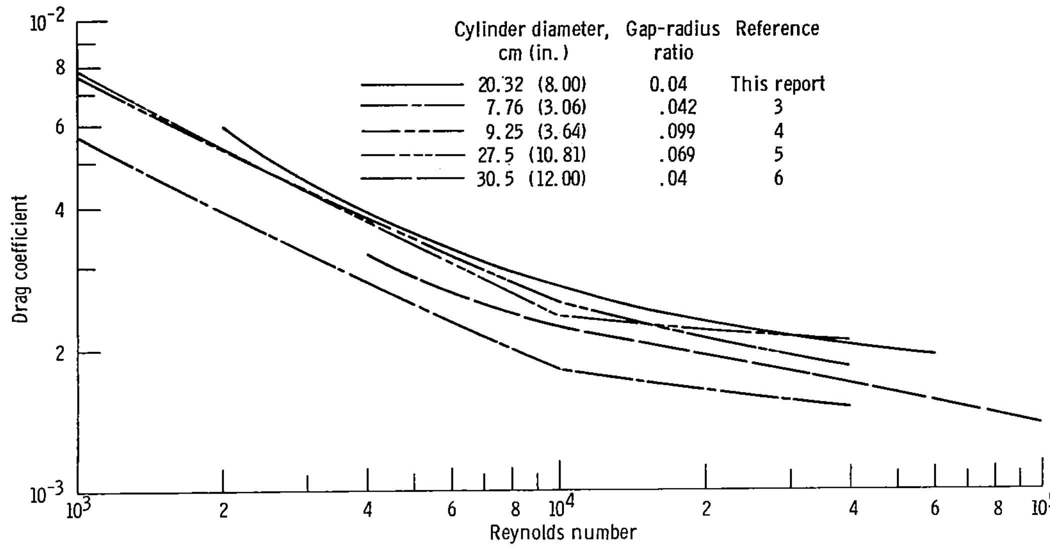

Figure 6. - Comparison of 20.32-centimeter- (8-in. -) diameter data with previously published information on drag coefficients for concentric rotating cylinders.

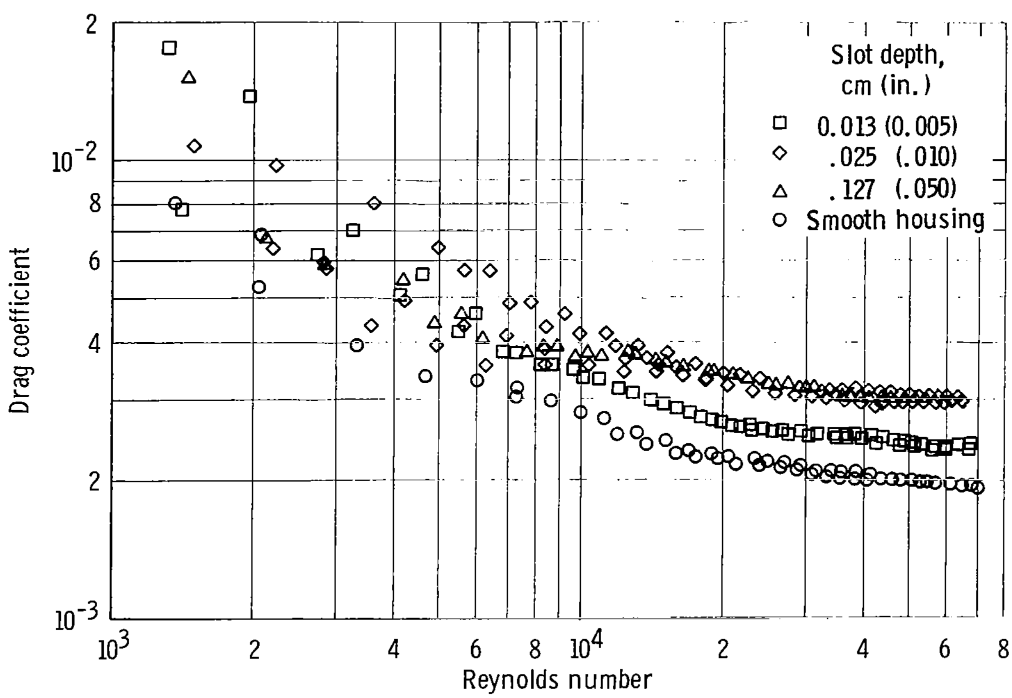

Figure 7. - Drag coefficient as function of Reynolds number for smooth cylinder rotating in stationary concentric housing for various slot depths in housing. Rotor diameter, 20.32 centimeters (8.00 in.); rotor length, 10.16 centimeters (4.00 in.).

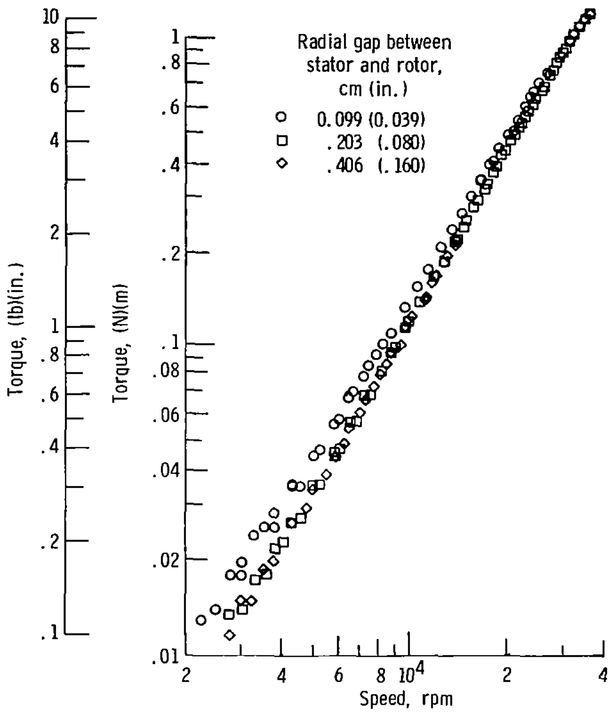

Figure 8. - Torque as function of speed for 20.32-centimeter- (8.00-in.-) diameter cylindrical section and both conical conforming transition sections.

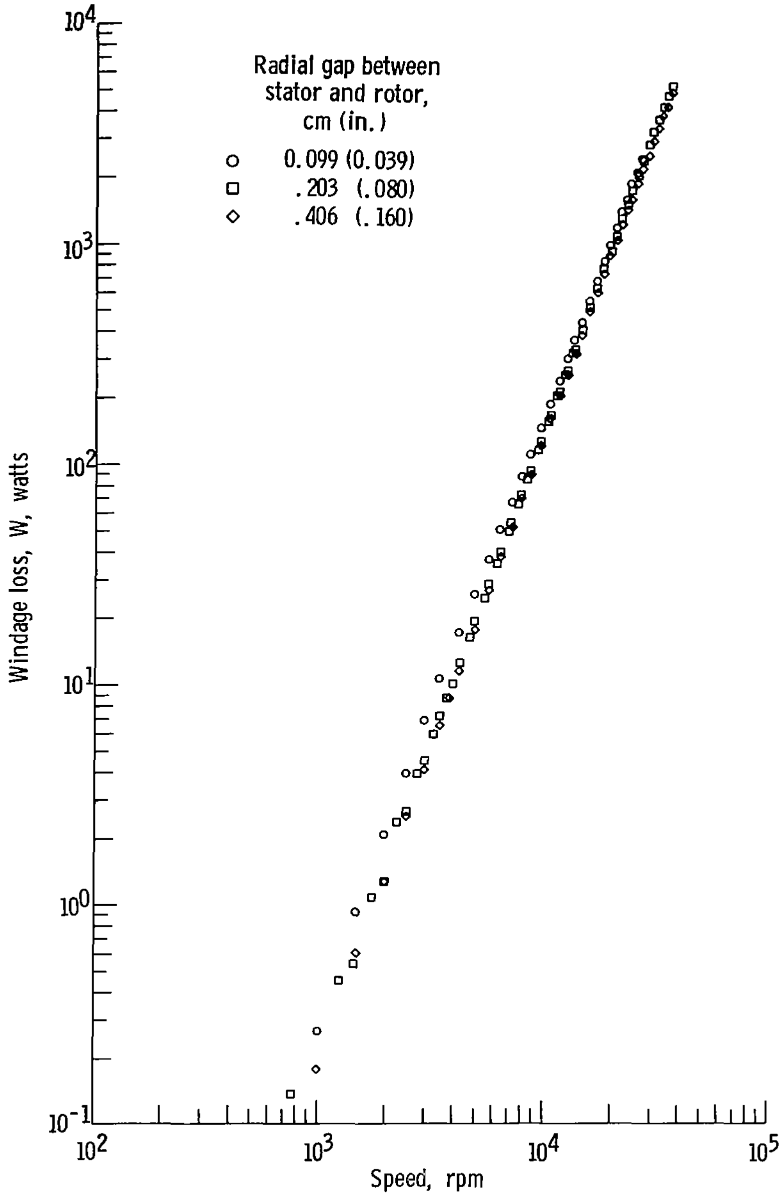

Figure 9. - Windage power loss for a Lundell-type rotor rotating in air within a stationary concentric unslotted housing.

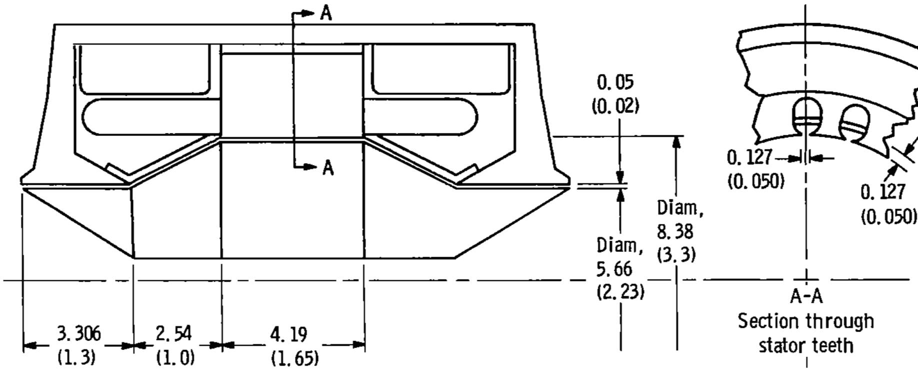

Figure 10. - Brayton test alternator of reference 11. (Dimensions are in centimeters (in.).)

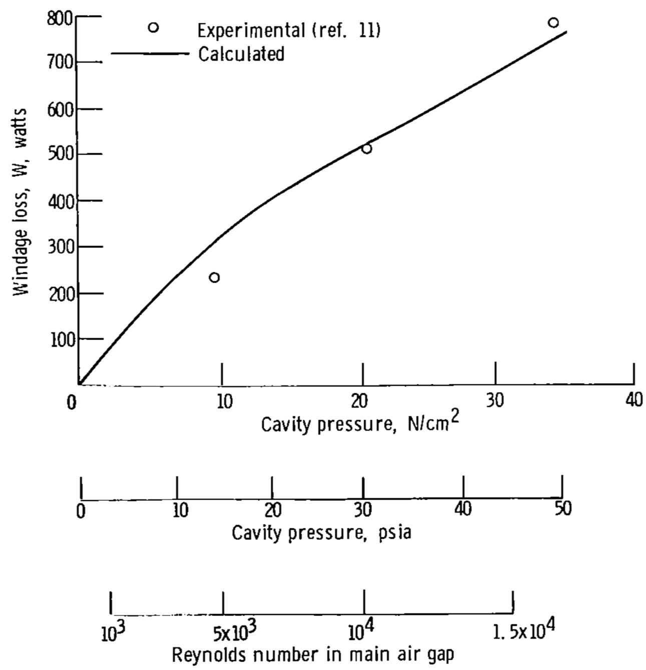

Figure 11. - Test alternator windage loss in kyrpton at 36 000 rpm compared to analytical results.

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