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EXPERIMENTAL WINDAGE STUDIES FOR HIGH-SPEED ALTERNATORS

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ABSTRACT

High-speed generators currently under development for space power systems have inert gas in the gap between the rotor and stator. At the speeds involved (24,000 rpm and 36,000 rpm), turbulent velocity profiles are developed, well above the levels previously investigated. Power losses due to windage become high and result in significant heating of the electric generator. This heating must be included in the generator thermal and electrical designs.

Turbulent flow (Reynolds numbers above 7000) for concentric rotating cylinders with Reynolds numbers as high as 40,000 have been previously investigated. For the most part, however, these high Reynolds numbers were obtained using large gap widths, (gap-to-radius ratio of 0.1 to 0.23). Current designs for space power system alternators require gap-to-radius ratios of 0.01 to 0.04 with Reynolds numbers as high as 200,000. Experimental data are needed to extend the existing information for the design of future electric generators.

Windage tests were conducted on two concentric rotor-stator configurations, in ambient air, producing Reynolds numbers as high as 100,000. A cylindrical rotor 12 inches in diameter and an 8-inch diameter Lundell-type rotor were operated at speeds up to 36,000 rpm. Gap-to-radius ratios of 0.01 to 0.04 were incorporated. Tests were also conducted with axial slots machined on the stator surface of the Lundell configuration to simulate electrical winding slots. The stators were mounted on a reaction-torque device so that viscous drag from the surface of the rotor would be measured. Tests were also conducted to determine the pumping characteristics of the conical sections of the Lundell rotor. The results are presented and discussed.

SYMBOL LIST

A	characteristic area (wetted area)
D	characteristic dimension (rotor to stator, radial gap thickness)
F	frictional force
K	kinetic energy/unit volume
L	length
R	radius
r_i	minor cone radius
r_o	major cone radius
T	torque
U	velocity
W	windage, viscous power loss
ΔP	pressure rise
λ	drag coefficient (friction factor)
ν	kinematic viscosity
ρ	density
ω	rotational speed

HIGH-SPEED GENERATORS currently under development for space power systems incorporate inert gas in the rotor-stator gap. The turbulent velocity profiles developed in this space are well above the levels previously investigated. At the speeds involved (24,000 rpm and 36,000 rpm), windage losses result in significant heating of the electric generator. This effect must be included in the generator thermal and electrical design.

Laminar flow between concentric rotating cylinders has been studied by Couette (1)*. Taylor (2,3), Vohr (4), and DiPrima (5) studied the vortex flow regime. The turbulent flow case (Reynolds numbers above 7000) was investigated by Taylor, Vohr, Wendt and Pai (2,4,6,7) with Reynolds numbers as high as 40,000. These high Reynolds numbers, however, were obtained using large radial gap clearances, gap-to-radius ratios of 0.1 to 0.23 (2,4). Reference 8 presents a preliminary study of windage losses for concentric rotating cylinders with Reynolds numbers as high as 100,000 using gap to radius ratios of 0.01 to 0.04. These are the gaps considered for designs of space-power alternators.

This paper presents results of tests conducted on two close-clearance concentric rotor-stator configurations. A 12-inch diameter cylindrical rotor was operated at speeds up to 24,000 rpm while an 8-inch major diameter rotor configuration representing a design for a smooth surface Lundell-type alternator rotor, Fig. 1, was operated up to 36,000 rpm. Gap-to-radius ratios of 0.01, 0.02, and 0.04 were tested. All tests were performed in ambient air and Reynolds numbers as high as 100,000 were obtained. The Lundell stator was tested with a smooth surface and with a surface machined with axial slots to simulate the electrical winding slots. The stators were mounted on a reaction-torque device so that viscous drag from the surface of the rotor would be measured.

The alternator cavity gas was considered as the possible means of removing the heat generated by windage of the Lundell rotor. The gas could be circulated through a heat exchanger by taking advantage of the pumping characteristics of the conical transition sections. Except for a preliminary study (9) no information was available on the pumping characteristics of enclosed rotating conical sections although studies of enclosed rotating disks operating in the turbulent flow regime can be found in Refs. 10 and 11. Results of the data presented in Ref. 9 are discussed here.

APPARATUS

The test apparatus (fig. 1) consisted of a smooth rotor supported at each end on ball bearings and a housing (stator) attached to a "floating" support table. Strain gages are mounted on four flexure arms which held the housing and support table so that it would pivot about the rotor axis. Reaction torque (viscous drag) between the rotor and housing was measured by means of a Wheatstone bridge circuit. A variable speed dynamometer consisting of a dc motor and two tandem gear units were used to drive the rotor. A splined coupling connected the rotor to the drive system. A more complete description of the apparatus is given in Ref. 8.

ROTOR-HOUSING CONFIGURATIONS

Dimensions of the cylindrical and Lundell rotor-housing configurations are shown in Figs. 2 and 3, 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. All parts were doweled or keyed so that the machined matching of the parts could be reproduced.

The Lundell housing was made in five separate and rigidly connected sections corresponding to the change in cross section of the rotor. A modified housing (fig. 4) was tested to determine the effects of "fairing in" the conical transition section of the rotor. To test the pumping effect of the conical sections of the Lundell rotor, five holes were made in the center cylindrical section of the housing. Flow enters the rotor-stator gap at each end of the rotor, passes through the conical pumping sections, and exits through the five radial outlets. Flow from the five outlets is manifolded together and piped through a flowmeter and flow control valve before being discharged back to the atmosphere.

TEST SEQUENCE

The 12-inch diameter cylindrical rotor was tested at speeds up to 24,000 rpm for each of the three gap widths. Several runs were made at each gap to verify the reproducibility of the data.

The Lundell rotor was run at speeds up to 36,000 rpm. Tests were performed using the cen-

*Numbers in parentheses designate References at end of paper.

ter cylindrical section of the housing and on the complete housing assembly. After a series of tests had been performed on the smooth surface housing, the center cylindrical housing section was machined with axial slots to simulate alternator winding slots. For the 0.039-inch and 0.080-inch gap widths (0.01 and 0.02 gap-radius ratio), sixty equally spaced slots 0.119-inch wide by 0.010-inch deep were machined. Three slot depths, 0.005, 0.010 and 0.050 inch, were incorporated into the 0.160 inch gap (0.04 gap-radius ratio) housing. Only a gap of 0.160 inch was used for the smooth surface, non-conforming housing, seen in Fig. 4.

The Lundell rotor and housing were also used to determine the pumping characteristics of rotating conical sections. Due to symmetry, the two conical sections act as identical pumps in parallel. With the flow control valve closed, these pumps will be dead-headed. At 18,000; 24,000 and 30,000 rpm flow was varied by means of the flow control valve and head-flow data were recorded. All testing was performed in ambient air.

INSTRUMENTATION

Instrumentation consisted of rotational speed, torque, temperature, flowmeter and differential pressure sensors. Speed was measured by means of a magnetic pickup and a 60-tooth gear arrangement on the shaft of the dynamometer. The signal generated was sent to a counter and recorded. Rotational speed could be controlled within 0.1 percent.

Torque measurements were made using a reaction torque device. 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. Any torque developed about this sensing axis will produce a strain proportional to the torque. All tare torques or loads remain constant and are compensated for by calibrations. The torque unit was calibrated by hanging accurately known weights from a calibration arm to produce known torques. The strain gage output from the Wheatstone bridge circuit was measured on an integrating digital voltmeter. Signal out put variation was 1 to 2 percent. Torque measurements above 1 in.-lb are approximately 5 per cent accurate. The accuracy increases at the higher torque levels.

All temperatures were measured using iron-constantan type J thermocouples. Bare-spike thermocouples were mounted on the housing extending halfway into the gap at several axial locations. Ambient temperature and the temperature at the flowmeter inlet were also measured.

Flow measurements were made using a turbine flowmeter. The output of the flowmeter was sent to a digital counter and recorded. Actual flow was then computed from a flowmeter calibration curve. Accuracy of flow measurement is estimated to be 5 percent.

A 2.5 psi differential pressure transducer was used to measure the pump head rise across the conical sections of the Lundell rotor. Location of the pressure taps are shown in Fig. 3. The pressure transducer was calibrated against a precision Bourdon tube pressure gage. An integrating digital voltmeter was used to measure the signal output. Transducer accuracy was 0.15 percent of full scale.

GENERAL COMMENTS FOR A CYLINDRICAL ROTOR

Torque due to viscous drag on a rotating cylinder of radius R and rotational speed ω is given by

$$T = FR \tag{1}$$

where F is the frictional force on the cylinder. The effect of pressure forces is assumed negligible (7). 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 kinetic energy/unit volume. Expanding Eq. (2),

$$F = \lambda(2\pi RL)(1/2\rho U^2) \tag{3}$$

where $U = \omega R$. Combining terms and solving for the drag coefficient in terms of the measured parameters, torque and speed

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

Reynolds number is defined as

$$Re = \frac{UD}{\nu} \tag{5}$$

The characteristic dimension D for concentric rotating cylinders is taken to be the radial gap thickness.

For the data presented, two assumptions were made. The first was that the pressure in the gap remained ambient since the housing was not enclosed. The second was that the average gap temperature was equal to the gap temperature in the center of the housing.

TWELVE-INCH DIAMETER CYLINDRICAL ROTOR

A non-dimensional curve is plotted in Fig. 5 of drag coefficient versus Reynolds number for each of the three gap sizes. Above a Reynolds number of 15,000 the deviation between drag coefficients for the different gaps is less than ±5 percent. This is supported by Taylor in Ref. 2. Below a Reynolds number of 10,000 the 0.116-inch gap (0.02 gap-radius ratio) had higher drag coefficients than the 0.0565-inch gap (0.01 gap-radius ratio). It would be expected based on previous work, (Wendt, (6)) that the 0.236-inch gap (0.04 gap-radius ratio) would be still higher; however, the torque values in this range for the 0.236 inch gap were extremely low (less than 0.15 inch-pound) and subject to large error.

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 Ref. 4. A maximum Reynolds number of approximately 110,000 was attained with the 0.236-inch gap housing. The effect of using the temperature at the center of the housing rather than an integrated average temperature was found to have a negligible effect on the curve presented in Fig. 5.

LUNDELL ROTOR

EIGHT-INCH DIAMETER CYLINDRICAL SECTION - Figure 6 presents curves of drag coefficient versus Reynolds number for the 8-inch diameter cylindrical section of the Lundell-type rotor. All three gaps are represented for both the smooth and slotted housings. For the 0.160-inch gap housing, only the slot depth of 0.010 inch is shown. The same assumptions are made here as for the drag coefficients plotted in Fig. 5.

The plotted points for drag coefficient of the slotted versus unslotted housings coincide for the lower Reynolds numbers in both the 0.039 and 0.080-inch gaps. Uncertainty for these data are approximately 5 percent. The 0.160-inch gap data in this Reynolds number range are subject to uncertainties of ±25 percent and may account for the different values between the slotted and unslotted case. The curves for the 0.039-inch gap separate at an approximate Reynolds number of 2,000 while the 0.080-inch gap separation point is approximately 4,000. Both points correspond to approximately 3,000 rpm. These effects are similar to those seen for varying roughness on drag coefficients in pipe flow. The increased losses due to the axial slots were 30 to 50 percent at the higher Reynolds numbers.

The form of the drag coefficient versus Reynolds number curves for the 8-inch cylindrical section with a smooth housing is similar to those for the 12-inch diameter rotor. At the higher Reynolds numbers the curves are within 5 percent and the transition to turbulent flow also occurs in the same range (5,000 to 7,000 Reynolds number). However, the drag coefficients for the 8-inch diameter cylinder are approximately 20 percent higher than for the 12-inch cylinder. Possible explanations for this anomaly are being investigated.

The 0.160-inch gap housing was tested for three different slot depths (0.005", 0.010", and 0.050"). Slot width and number of slots were maintained constant (0.119×60 equally spaced). Figure 7 shows a direct relation between slot depth and drag coefficient in the turbulent flow regime. The data for the lower Reynolds numbers would be expected to coincide for all the slot depths. The deviation shown is within the experimental uncertainties incurred for this Reynolds number range with the 0.160 inch gap housing. As the slot depth increases, the drag coefficient curves deviate from the curve for the smooth housing at lower Reynolds numbers. The 0.005-inch slot produced approximately 21 percent increased losses while the 0.010-inch slot produced a 50 percent increase. Apparently, the flow could not be turbulated further because increasing the slot depth to 0.050 inch produced no noticeable effect.

LUNDELL-SHAPED HOUSING

The plots of speed versus windage for the complete rotor-unslotted housing configuration are shown in Fig. 8. Windage power loss, W, was calculated using $W = T\omega$ (eq. (6)). The power loss at 36,000 rpm for the 0.160 inch gap housing is approximately 4.7 kW. Windage losses for this rotor can be scaled for other fluids at different pressures and temperatures. The slope of the curve is the power proportionality between windage and speed. Windage is seen to be an inverse function of gap width.

Power losses for the slotted Lundell housings were higher than for the unslotted housings, approximately 6.0 kW for the 0.160 inch gap housing with 0.010 inch slots at 36,000 rpm. The increase was equal to the additional loss contributed by the slotted cylindrical section. Different slot configurations will produce different overall power losses.

A modified Lundell housing (fig. 4) having a gap width of 0.160 inch and a smooth surface was tested and compared to the housing with the faired transition sections. Losses were approximately 12 percent higher for the modified housing.

PUMPING CHARACTERISTICS OF THE CONICAL SECTIONS

Head-flow performance of the conical sections with a 0.160-inch gap is shown by the nondimensional curve in Fig. 9 for three speeds-18,000; 24,000; and 30,000 rpm. Pressure coefficient is defined here as the ratio of the measured pressure rise to the kinetic energy of the fluid.

$$\text{Pressure coefficient} = \frac{\Delta P}{\frac{1}{2}\rho\omega^2\left(r_o^2 - r_i^2\right)} \tag{7}$$

where

ΔP = pressure rise
ρ = density
ω = angular velocity
r_o = major cone radius
r_i = minor cone radius

Flow coefficient is the ratio of the gas flow velocity to the rotor velocity. The actual term used is $Q_1/\omega r_o^3$ where Q_1 = inlet volume flow rate. It was assumed that the flows were identical for the two "pumps" in parallel. Temperature at the base of the cones could not be maintained constant during the test. The curve for 18,000 rpm was obtained at 86° F$\pm 6^\circ$ F; at 24,000 rpm the temperature was 115° F$\pm 15^\circ$ F; and at 30,000 rpm the temperature was 145° F$\pm 20^\circ$ F.

The dead-head (no-flow) pressure rise for the rotating truncated cones can be predicted within 10 percent by calculating the kinetic energy increase in the fluid. The assumption must be made that the average core velocity is approximately one-half the rotor speed. If either flow or head rise is given, the curves presented in Ref. 11 for enclosed rotating disks can be used to approximate the operating point. It was found that changing the gap width had no effect on the dead-head pressure rise.

SUMMARY OF RESULTS

Two rotors, a 12-inch diameter cylinder and an 8-inch diameter Lundell-type rotor, were rotated in ambient air at speeds up to 36,000 rpm within stationary concentric housings. Three gap widths were used for each rotor, corresponding to approximately 1 percent, 2 percent and 4 percent of the rotor radius. The center cylindrical section of the Lundell housing was tested with both a smooth surface and with axial slots simulating alternator winding slots. Results of the tests were:

1. Reynolds numbers above 100,000 were produced.
2. Drag coefficients for the 8-inch diameter cylinder were approximately 20 percent higher than for the 12-inch diameter cylinder.
3. Power loss for a slotted stationary cylinder in the turbulent flow regime can be 50 percent higher than for a smooth cylinder. The loss varies directly with slot depth but remains constant after a critical depth is reached.
4. Power loss is an inverse function of gap size.
5. Rotating truncated cones act as typical pumps, with head rise proportional to speed squared and flow proportional to speed. For a given flow, the operating point of a conical pump can be approximated using equations for enclosed rotating disks.
6. Gap width has a negligible effect on the dead-headed pressure rise.

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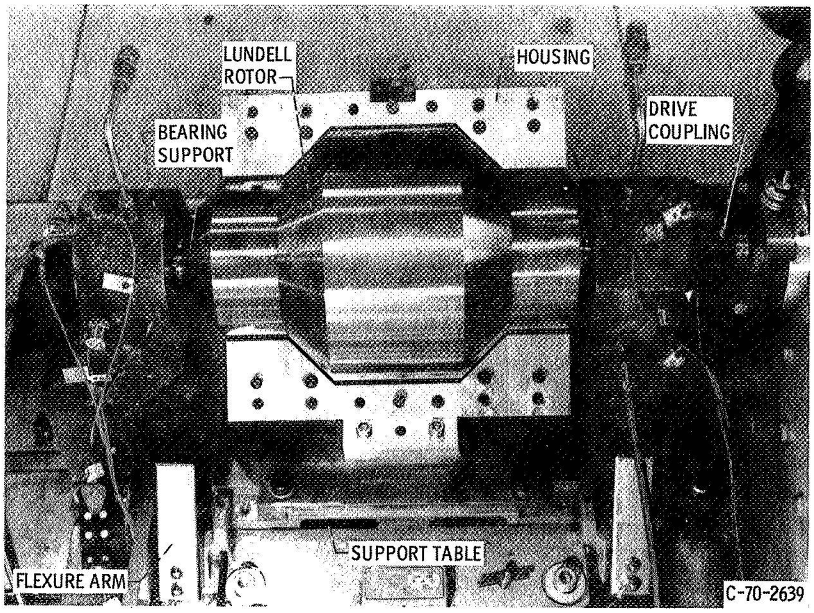
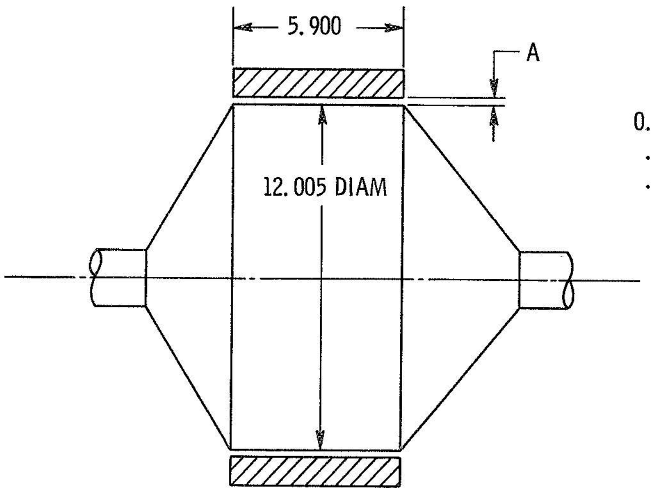


Figure 1. - Windage test apparatus for Lundell-type rotor.



A	GAP-RADIUS RATIO
0.0565	0.01
.116	.02
.236	.04

Figure 2. - Cylindrical rotor-housing configuration. (All dimensions are in inches.)

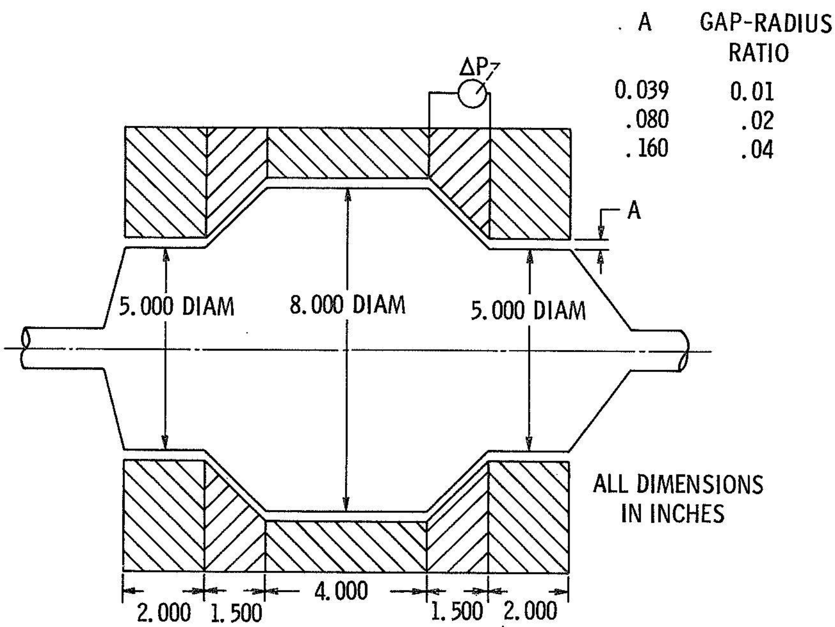


Figure 3. - Lundell rotor-housing configuration.

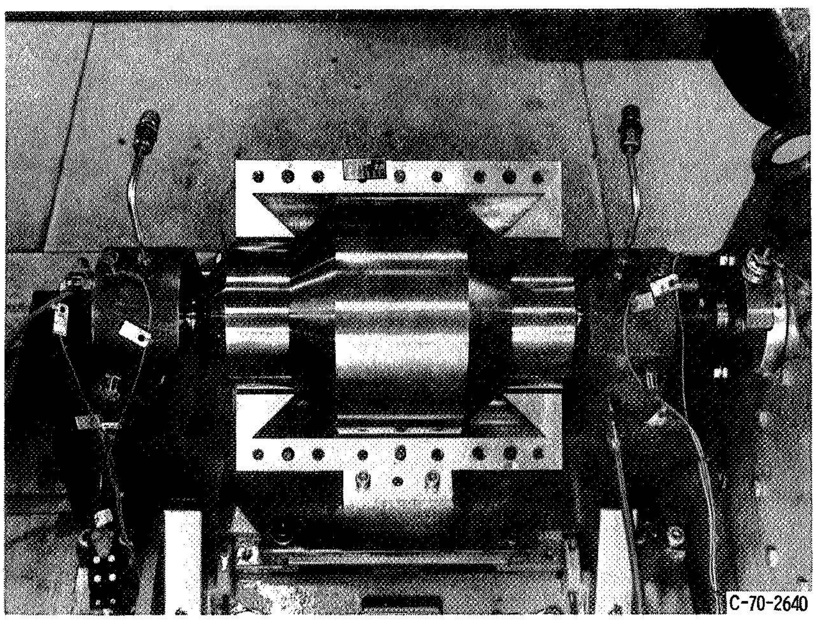


Figure 4. - Modified Lundell housing.

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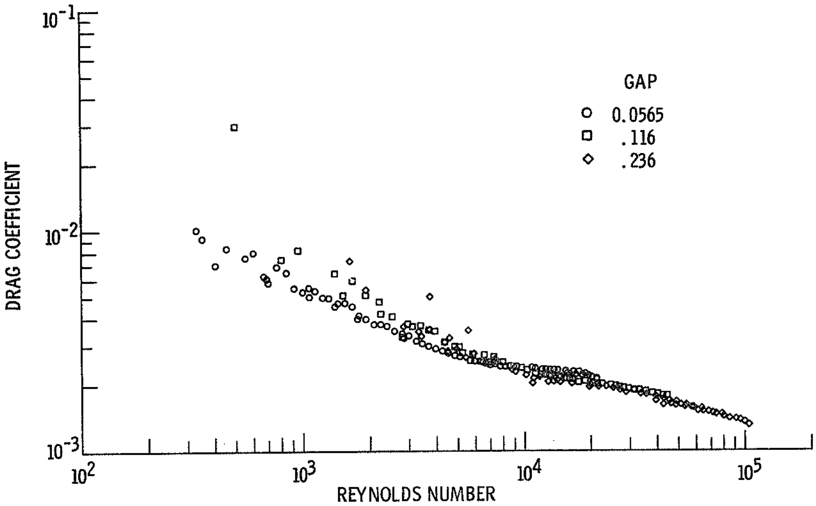

Figure 5. - Drag coefficient versus Reynolds number for a 12 inch diameter rotating cylinder in a stationary cylindrical housing.

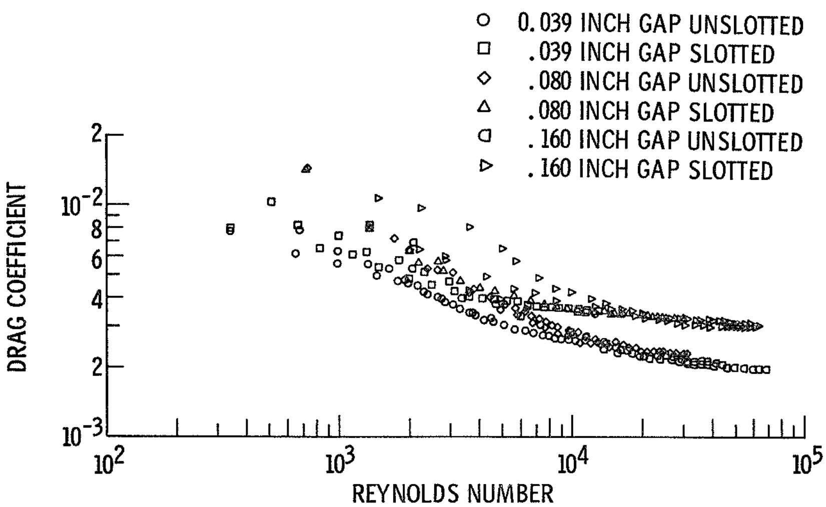

Figure 6. - Drag coefficient versus Reynolds number for a 8 inch diameter rotating cylinder in a smooth and a slotted stationary concentric housing. Slot depth 0.010 inch.

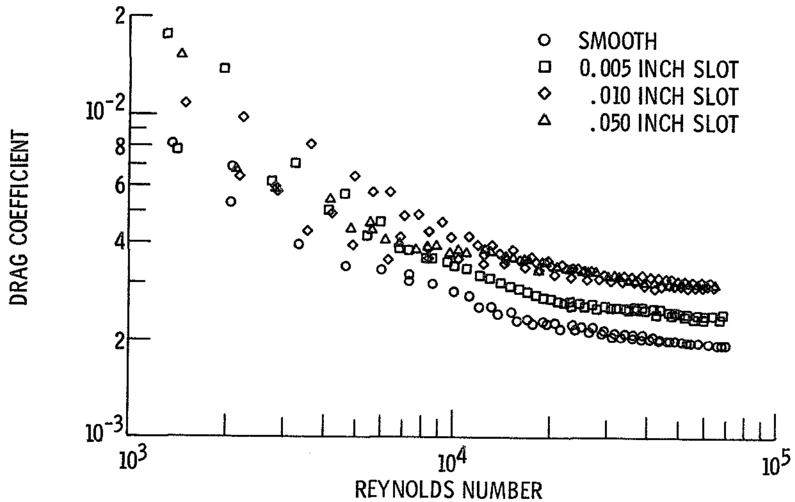

Figure 7. - Drag coefficient versus Reynolds number for a smooth 8 inch diameter cylinder rotating in a stationary concentric housing with a 0.160 inch radial gap for various slot depths in the housing.

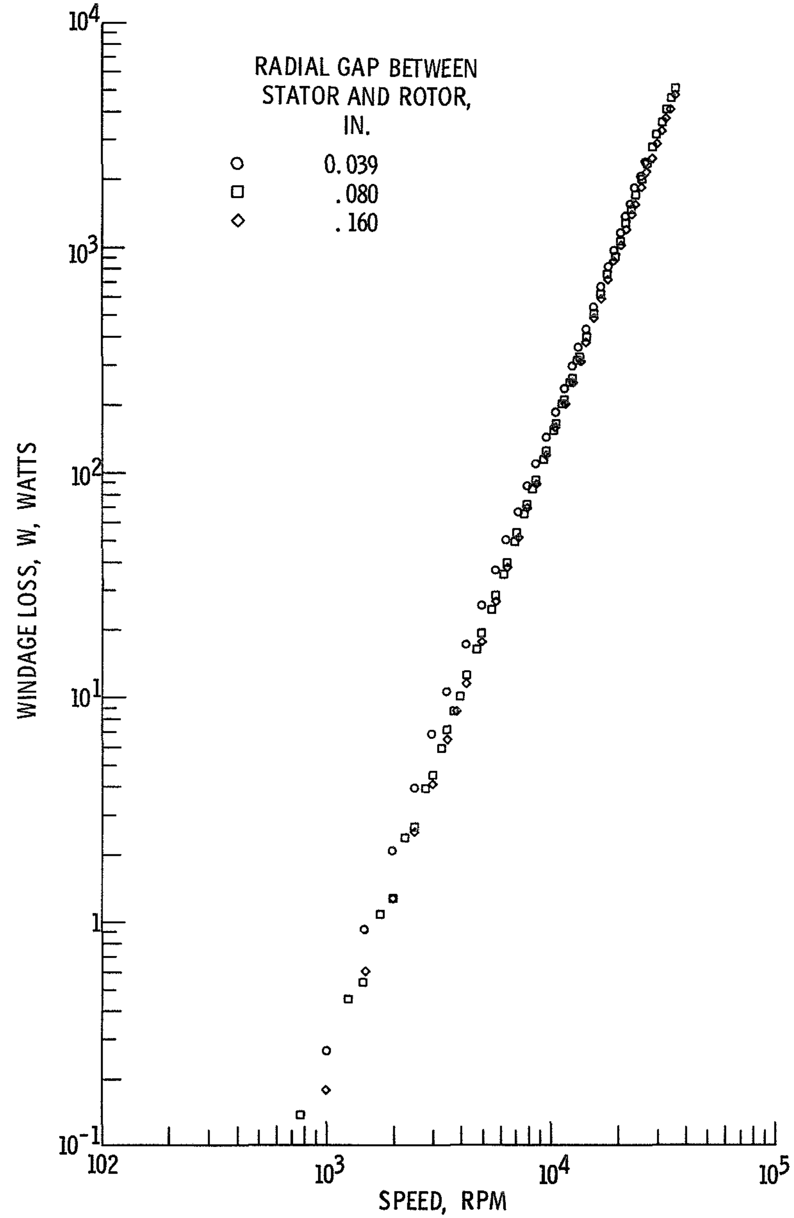

Figure 8. - Windage power loss for a Lundell-type rotor rotating in air within a stationary concentric unslotted housing.

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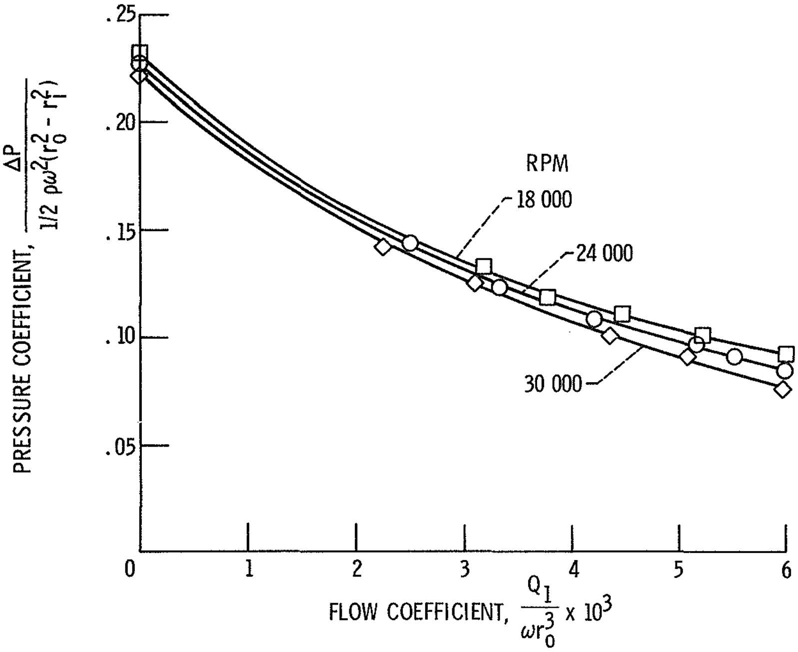

Figure 9. - Head-flow characteristics of rotating conical sections at three speeds.