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**DESIGN STUDY (OF) ELECTRICAL COMPONENT
TECHNOLOGY FOR 0.25 TO 10.0 MEGAWATT
SPACE POWER SYSTEMS**

Experimental Verification

By
C. C. Kouba
J. L. McCabria
N. V. Wheat

October 16, 1967

Aerospace Electrical Division
Westinghouse Electric Corporation
Lima, Ohio

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EXPERIMENTAL VERIFICATION

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October 16, 1967

Prepared Under
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San Francisco Operations Office
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Aerospace Electrical Division
Westinghouse Electric Corporation
Lima, Ohio

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Prepared by:

C. C. Kouba
C. C. Kouba, Engineer
Electromechanical Systems Dev.

J. L. McCabria
J. L. McCabria, Engineer
Electromechanical Systems Dev.

Norman V. Wheat
N. V. Wheat, Engineer
Electromechanical Systems Dev.

Approved by:

F. C. Stump
F. C. Stump, Program Manager
Electromechanical Systems Dev.

N. W. Bucci
N. W. Bucci, Engineering Manager
Systems R & D Department
Aerospace Electrical Division
Westinghouse Electric Corporation

FOREWORD

This Topical Report describes work under contract to the United States Atomic Energy Commission from September 1, 1966 to October 31, 1967 on a Design Study of Electrical Component Technology for 0.25 to 10 Megawatt Space Power Systems.

The work is directed toward defining representative Rankine nuclear space power electrical systems, the electrical components therein and their development problems. The particular work reported within deals with analytical verification experiments to provide data that would facilitate the accurate prediction of space power alternator performance and operating characteristics. The work was conducted by the Westinghouse Electric Corporation, Aerospace Electrical Division, under contract AT(04-3)-679 administered by the USAEC San Francisco Operations Office. Technical liaison is maintained with the USAEC Division of Space Nuclear Systems, Washington, D. C., Mr. G. S. Leighton, Liaison Officer, and with the Reactor Division of the San Francisco Operations Office, Mr. F. W. Hunton, Liaison Officer.

The report is written in three parts according to the three series of investigations:

1. Investigations of Alternator Magnetic Unbalance Forces and Flux Distributions by C. C. Kouba,
2. Investigations of Alternator Rotor Cooling Concepts by J. L. McCabria, and
3. Investigations of Alternator No-Load Losses by N. V. Wheat.

Other members of the Westinghouse Systems Research and Development Department assisting in the investigations include D. R. Basel, C. M. Gauthier, and A. E. King.

This report is assigned Contractors Designation Report Number WAED 67.51E.

ABSTRACT

A static magnetic mockup was prepared using the available generator components from the previous SNAP-50/SPUR 467-kVA experimental generator development program. The mockup was fitted with search coils to measure magnetic flux levels at a minimum of eighteen different locations within and external to the magnetic circuit. The magnetic unbalance forces and torques created by an eccentric and misaligned rotor and stator were also measured. The measured forces, torques and flux distribution, both useful and leakage, were compared with design predictions.

The static mockup of the above experiments was then modified to permit dynamic operation using water cooling and oil-lubricated bearings. The experiments were conducted to measure the no-load losses (pole face, magnetic and field coil) and corresponding temperature distributions at constant speed. The measured data was compared with the analytically calculated predictions.

A fluid dynamic test rig from the same generator development program was modified to include a rotor with radial flow pole passages. Experiments were run with "easy-fluids" that simulated potassium to determine flow characteristics of the rotor.

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1. INTRODUCTION

The Aerospace Electrical Division of the Westinghouse Electric Corporation has been under contract to the United States Atomic Energy Commission to define conceptual designs of complete electrical power systems and the electrical components therein which are suitable for Rankine nuclear space power systems in the 0.25 to 10.0 megawatt range. Three representative missions - interplanetary, planetary, and orbital - were specified in order that the conceptual definitions consider dependent design parameters typical of actual flight conditions. Work was carried on in design studies and in associated analytical verification experiments.

The analytical verification experiments were carried out to evaluate the adequacy of certain analytical techniques for a-c generator design. The experiments were specifically directed toward evaluating magnetic unbalance forces, magnetic flux distributions, generator losses at no-load condition, and rotor coolant flow concepts.

Early in 1965, the design of a 467-kVA potassium-cooled generator was completed^{(1-1)*} and the fabrication of two sets of components was initiated. The design details are shown in various figures throughout this report. The generator design was based largely on analytical considerations with only a minimum of experimental data. Mechanically speaking, the generator represented the technology developed since the initiation of the SNAP-50/SPUR program in late 1960. For example, the bearings and seals used were developed on data from the 50-kW (LMCD-II) generator,⁽¹⁻²⁾ rotor fluid dynamic tests,⁽³⁻⁴⁾ and in-house developments.^(1-3,1-4,1-5) In addition, the generator incorporated the features necessary to cool the rotor and to scavenge and maintain rotor cavity pressure below 0.02 psia. These features were developed on data from rotor fluid dynamic tests. Electrically speaking, the only experimental data available was that extrapolated from apparatus that differed considerably from this application in rating, speed, operating environment, and the type of material used in construction. Accordingly, in some design areas, an acute need existed for pertinent experimental data that would facilitate the accurate prediction of generator performance and operating characteristics.

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*Superscripts in parentheses refer to references at the end of the report.

This generator was designated the Experimental Generator and the two sets of components to be built were intended to:

1. obtain experience and test data on the fabrication and performance of the machine and,
2. to define problem areas peculiar to this type of machine.

Initial tests were to be performed using stoddard solvent at low speeds and at no load. Complete electrical and mechanical tests, using potassium as the coolant and lubricant, were to be used to verify the feasibility of the design concepts. Before the testing was started, however, the program was redirected and the hardware was diverted to a component technology test program on Air Force Contract AF33(615)-2326. At the conclusion of the 2326 contract, the status of the generator hardware was as follows:

1. The No. 1 stator was completed except for electrical winding and procurement of a bore seal.
2. The No. 1 rotor was completed except for slotting the pole faces and final polishing and balancing operations.
3. A special rotor fluid dynamic-test assembly was completed on which fluid-dynamic tests had been conducted and completed.
4. A portion of the No. 2 set of parts was started.

The present program diverted this existing hardware for use in the three verification test areas reported within.

2. INVESTIGATION OF ALTERNATOR MAGNETIC UNBALANCE FORCES AND FLUX DISTRIBUTIONS

by C. C. Kouba

2.1 MAGNETIC FORCE UNBALANCE AND FLUX DISTRIBUTION

2.1.1 Magnetic Force Unbalance

(a) Background and Test Objectives. In the normal selection or design of a bearing system, bearing load is an important parameter. The weight of the rotating member is an obvious load for the bearings when operating in a gravity application. In electromagnetic generators, a magnetic radial force or axial moment can result if the rotating member is made eccentric to or misaligned with the stator. (Misalignment refers to tilting of the rotor axis with respect to the axial centerline of the stator.) The magnitude of the radial force or axial moment is dependent upon the degree of eccentricity and/or misalignment between the rotor and stator and the level of field excitation. The magnetic force on the rotor will be present regardless of the conditions of gravity.

The test objective is to measure directly the net forces acting on the rotor of the 467 kVA experimental generator for various degrees of eccentricity, misalignment, and field current and to measure the unbalanced magnetic fields associated with these forces. The magnetic fields exist in the radial gaps of the generator. When the rotor is moved eccentrically to the stator bore, these fields become unbalanced and create a force on the rotor which acts in the same direction as the rotor displacement with respect to the stator.

(b) Test Description. Search coils were wound around the poles of the 467 kVA experimental generator rotor near the faces of the poles as shown in figure 2.1-1. The rotor is mounted in the stator through anti-friction bearings, load cells, and adjustable linkage, while the stator in turn is mounted in a fixture as shown in figure 2.1-2.

Anti-friction bearings were provided to facilitate the use of dial indicators which were mounted on the rotor to measure eccentricity and misalignment. The load cells are provided to measure the magnetic force imposed on the rotor. Adjustable linkage was provided to accurately adjust the eccentricity and/or misalignment of the rotor relative to the stator bore. All mounting fixtures and adapters were constructed of non-magnetic stainless steel to avoid any distortion of the generator flux distribution.

Figure 2.1-1. Rotor with Search Coils and Bearings

Figure 2.1-2. Stator Test Fixture

The magnetic field unbalance associated with a given rotor eccentricity was measured by connecting flux meters to the rotor search coils. The search coils wound around a diametrically opposite pair of poles were connected in series with the coil polarities such that the readings would be additive. After a set of readings were obtained for a given rotor eccentricity, the coils were reconnected with the coil polarities such that the readings would be subtractive and the test repeated. The net magnetic force acting on a pair of diametrically opposite poles is directly proportional to the difference of the squares of the flux quantity emanating from or entering the pole surfaces.

The net force acting on a diametrically opposite pair of poles is determined from the product of the flux measurements taken as described above since $(\phi_2 + \phi_1)(\phi_2 - \phi_1) = \phi_2^2 - \phi_1^2$. Dial indicators were mounted to the test fixture frame in order to measure the radial movement of the rotor as a result of load cell deflection under load. This deflection must be included to determine total rotor eccentricity under load.

Field current was increased in three ampere increments to 30 amperes. Fluxmeter, load cell, dial indicator, field voltage and field current readings were taken at each increment of field current. After each reading, the fluxmeters were reset to zero and held until just prior to raising the current to the next incremental level. The meter readings were added to obtain the total flux curve as a function of field current. This procedure was used to minimize errors caused by drifting of the meter pointers. The stator and rotor were demagnetized after each test run by reversing the polarity of the field current while decreasing the field current from 30 amperes to zero. Tests were run for the following values of rotor eccentricity and misalignment.

<u>Eccentricity</u> <u>(Inches)</u>	<u>Misalignment</u> <u>(Inches/Inch)</u>
0	0
0.005	0
0.010	0
0.015	0
0.020	0
0.030	0
0	0.00218

The load cells, load indicating meter, and fluxmeters were calibrated prior to performing the tests and checked at frequent intervals during the test.

(c) Calculation of Radial Force from Magnetic Field Test Data. When a magnetic field, B, exists in the space between two magnetized surfaces, there exists a force equal to

$$F = \frac{B^2 A}{72.13} \text{ lbs.} \quad (2.1-1)$$

B = flux density in kilolines per square inch
 A = cross sectional area of the gap in square inches.

Since the flux density, B, is equal to the uniform flux, ϕ , enclosed by the area, A, divided by that area, equation 2.1-1 can be written as follows:

$$F = \frac{\phi^2}{72.13(A)} \text{ lbs.} \quad (2.1-2)$$

The amount of energy per unit volume (i.e., energy density) in a given space is identical with pressure or force per unit surface area when the surface is taken at right angles to the direction in which the pressure can act to move a substance. Whenever there is an immediately available supply of energy, a mechanical force is in evidence. The energy stored in a magnetic field is

$$E = \frac{(B)^2}{72.13(Mr)} \frac{\text{inch-lb}}{(\text{inch})^3} \quad (2.1-3)$$

Mr = relative permeability

For a field density of 8.49 kL/inch² in space, where Mr = 1, there exists an energy density of 1.0 inch-lb/(inch)³ or 1.0 lbs/(inch)². Hence, the pressure tending to move the two bodies together will be equal to 1.0 pounds for each square inch of cross section of the gap. If the cross section is 10 square inches, the mechanical force of mutual attraction will be 10 pounds. Therefore, the force equals the energy density, E, times the cross sectional area, A. In air or free space where the relative permeability, Mr, is unity, the product of E times A results in equation 2.1-1.

Referring to figure 2.1-3 which is an end view of a 16 pole inductor alternator rotor (8 poles are located on each end of the rotor), a magnetic force, F, is shown perpendicular to each pole face, acting to pull the rotor towards the stator bore surface. There is no net magnetic force acting on the rotor if these forces are all equal. For this condition to exist, the magnetic field, B, at the surface of each pole must be equal. This occurs only when the rotor is perfectly concentric with the stator. When the rotor is moved eccentrically to the stator as shown in figure 2.1-4, the reluctance (equivalent to resistance in an electrical circuit) of the magnetic path for the bottom poles is less than that for the top poles due to the shorter length of radial pole gaps. Hence, the flux density and corresponding force at the surface of the bottom poles is greater than for the top poles, resulting in a net magnetic force acting downward in the direction of the eccentricity. As a matter of interest pertinent to this discussion, a 2-pole rotor will be examined. Figure 2.4-5 shows a 2-pole rotor positioned eccentrically to the stator. In this case, the flux path is common to each of the two poles and therefore, the flux density and corresponding force is the same for each pole. Therefore, no net radial magnetic force acts on a 2-pole rotor regardless

FIGURE 2.1-3. End View of 16-Pole Rotor

of the condition of eccentricity. A large axial moment does exist, however, which makes a 2-pole inductor generator impractical. We see then that the magnitude of the net radial force on an eccentric rotor depends on the number of rotor poles.

Figure 2.1-6 shows a stretched out view of one end of an inductor generator rotor with the corresponding flux density distribution in the radial gaps shown above. The density is at maximum over the pole faces and decreases to a minimum over the center of the interpolar space. The distribution at the other end of the rotor is identical except it is in the opposite direction and shifted 180 electrical degrees.

The magnitude of the force of attraction is proportional to the square of the flux density at all points in the radial gap. The flux density over the pole faces is normally greater than 10 times the density at the center of the inter-polar space and therefore the magnitude of the magnetic force over the pole faces is normally greater than 100 times the force over the center of the inter-polar space. For this reason the vicinity of the pole face becomes the prime area for investigation of magnetic force unbalance. The contribution of the inter-polar space away from the pole to the net radial force on the rotor is small.

FIGURE 2.1-4. Rotor Eccentric to Stator

FIGURE 2.1-5. Two Pole Rotor

FIGURE 2.1-6. Rotor Poles and Flux Distribution

In order to include the total flux, it would require that the flux density distribution be determined for at least half of the stator bore periphery. An integration of the flux density distribution according to equation 2.1-1 would then yield the force distribution. The flux distribution could be obtained with a Hall Effect Gaussmeter, but it would be a very difficult and tedious process.

It was concluded that the most accurate and efficient method for determining the magnetic field unbalance would be to measure the flux emanating from the rotor poles with search coils wound around the pole bodies near the tips of the poles.

A plot of per unit radial gap specific permeance for one pole pitch is shown in figure 2.1-7. The axis of abscissas is dimensioned in electrical degrees, there being 180 electrical degrees per pole pitch. The plot also represents the flux density distribution in the gap since flux density is directly proportional to permeance. The distribution corresponds to that of the test rig with the rotor set concentrically to the stator (the radial pole gap = 0.122 inch) and with the stator teeth and rotor poles unsaturated. The axis of ordinates coincides with the centerline of the rotor pole. One-half pole width corresponds to 60 electrical degrees for the test rig. The flux density does not drop immediately after the edge of the pole face is reached but instead it decays somewhat exponentially to the value at the center of the inter-polar space (180° point).

FIGURE 2.1-7. Radial Gap Per Unit Specific Permeance

This is due to the fact that the flux in the vicinity of the pole face edge bends to intercept the side of the pole and therefore the search coils wound around the pole bodies will measure an amount of flux greater than that emanating from the pole faces.

The flux intercepting the sides of the poles must be considered to determine the total flux linking the search coils. The flux distribution around the edge of a rotor pole for a radial pole gap of 0.122 inch is shown in figure 2.1-8. Ten conductors of a search coil are shown located against the pole body. The coils extend about 0.2 inch down the sides of the pole. In this plot, the fringing flux produces 33 flux linkages in the search coil. It would be equivalent to consider that the fringing flux entering the top one-third of the search coil height links all ten-turns of the coil. A good approximation of the total flux linking the search coil can be made by extending the half width of the search coil by a distance equivalent to the normal radial pole gap (0.122 inch) plus one-third of the coil height (0.66 inch). This distance corresponds to 15 electrical degrees along the pole pitch. Referring to figure 2.1-7, the total flux linked by a rotor pole search coil would correspond to the area under the curve up to the 75 degree point on the axis of abscissas.

It would be convenient to determine a correction factor for equation 2.1-2 which would account for the non-uniform distribution of the flux and for the effective increase in pole face area due to the fringing flux. This was performed in the following manner. A plot of radial gap specific permeance for a pole gap of 0.092 inch and 0.152 inch respectively is shown in figure 2.1-9. This plot approximates the conditions for the top and bottom poles of the test rig for a rotor eccentricity of 30 mils. As stated previously, the flux density distribution is identical to the permeance distribution in the radial gap. Since force is proportional to the square of the flux density at each point in the radial gap, the net force distribution can be found for the diametrically opposite poles by plotting the difference of the squares for each point on the curves. The net force distribution is shown in figure 2.1-10. The area under this curve is proportional to the total net force on the pair of poles. The search coils measure a quantity of flux represented by the area under the curves of figure 2.1-9 up to the 75 degree point on the axis of abscissas. The ordinates which give equivalent areas for a half pole width (60 degrees) were found. The difference of the squares of these two ordinates multiplied by 60 degrees is proportional to the force that would be calculated from equation 2.1-2 using the actual pole face area. The ratio of the area under the force distribution curve of figure 2.1-9 to the area calculated from the difference of the squares of the ordinates as previously described, gives the proper correction factor. For a rotor eccentricity of 30 mils, the correction factor is 0.834. This procedure was repeated for a rotor eccentricity of 5 mils (see figure 2.1-11 and 2.1-12) and resulted in a correction factor of 0.820.

FIGURE 2.1-8. Flux Distribution

FIGURE 2.1-9. Specific Permeance, 30 Mil Eccentricity

FIGURE 2.1-10. Force Distribution, 30 Mil Eccentricity

FIGURE 2.1-11. Specific Permeance, 5 Mil Eccentricity

FIGURE 2.1-12. Force Distribution, 5 Mil Eccentricity

The correction factors are nearly equal for both the largest and smallest rotor eccentricities tested and therefore a mean value of 0.83 was assumed to hold for all conditions of rotor eccentricity and misalignment.

Figure 2.1-13 shows the 16-pole test rotor positioned eccentrically to the stator bore by a distance e . A corresponding force diagram with concentrated forces acting at the center lines of the poles is also shown. Poles 1 and 9, 2 and 10, etc., are 180 degrees apart and therefore the forces acting on these poles may be subtracted directly to obtain the force diagram shown in figure 2.1-14.

FIGURE 2.1-13. Eccentric Rotor and Force Diagram

FIGURE 2.1-14. Force Diagram

The horizontal components of the composite forces cancel out due to symmetry leaving a net force which equals the sum of the vertical components. This force acts in the direction that the rotor was moved with respect to the stator (i.e., in the direction of the eccentricity).

The search coils wound around poles 1 and 9, 2 and 10, 3 and 11, etc., were connected in series with the coil polarities such that the readings would be additive.

After a set of flux readings were obtained for a given rotor eccentricity, the coils were reconnected with the coil polarities such that the readings would be subtractive and the tests repeated. For poles 1 and 9, for example, we find from equation 2.1-2 and the correction factor that

$$\begin{aligned} F_9 - F_1 &= \frac{0.83[(\phi_9)^2 - (\phi_1)^2]}{72.13(A)} & (2.1-4) \\ &= \frac{0.83(\phi_9 + \phi_1)(\phi_9 - \phi_1)}{72.13(A)} \text{ lbs} \end{aligned}$$

ϕ = flux in kilolines

where A is the area of the pole face. When this is done for each of the seven diametrically opposite pole pairs, the vector diagram of figure 2.1-14 is obtained. The vertical components of this diagram gives the net radial force acting on the rotor for the particular conditions of test.

(d) Test Results. Figure 2.1-15 shows a comparison of the total magnetic force acting on the rotor as measured by the load cells and as calculated from the flux measurements for various values of field current and rotor eccentricity. The force increases exponentially with field current up to about 15 amperes due to the practically linear relationship between field current and flux for low values of field excitation. Above 15 amperes field current, the effects of magnetic steel saturation begin to appear. That is, the rate of change of flux per ampere increase in field current starts to decrease. These effects can be deduced from the typical generator saturation curve shown in figure 2.1-16. The force on the rotor reaches a peak at about 21 amperes field current and then starts to decrease. This decrease in force is caused by high magnetic saturation of the rotor poles and the stator teeth immediately above these poles. That is, the flux from all the poles is approaching the saturation limit of the steel. The magnitude of the flux at each pole must be different to cause a net force on the rotor, and since the flux for each pole is approaching a common limit, the net force on the rotor decreases as the field current is increased.

Of most interest to the designer is the maximum magnetic force imposed on the rotor for a given rotor eccentricity. Figure 2.1-17 shows the maximum force on the rotor, as determined by test, plotted as a function of rotor eccentricity. The forces were taken from the load cell readings and the eccentricity is

FIGURE 2.1-15. Radial Magnetic Force on Rotor

FIGURE 2.1-16. Typical Generator Saturation Curve

FIGURE 2.1-17. Maximum Radial Magnetic Force on Rotor

comprised of the initial rotor displacement and the additional displacement caused by load cell deflection. The plot, as shown, is a straight line with a slope of 19.114 pounds per 0.001 inch eccentricity. Maximum force occurs at 21 amperes field current. Thus, for the 467 kVA experimental generator, which has a maximum rotor eccentricity tolerance of 0.006 inch, a magnetic force of up to 115 pounds on the rotor can be expected.

(e) Calculation of Magnetic Force Unbalance From Generator Design Data. Results of the magnetic force unbalance tests point the way to development of a calculation procedure for calculating the magnetic unbalance force (under static conditions) from design data. The first step is to determine the average radial pole gap for each pole of the rotor at a given eccentricity, e , (figure 2.1-13). This may be done graphically or programmed for computer derivation. Next the pole face flux density is determined for each pole. This information can be obtained by utilizing a program which calculates the no load saturation characteristics of the type of generator under consideration. In this manner, the saturation characteristics of the entire magnetic circuit are taken into consideration. A saturation curve is obtained for each of the radial pole gaps previously determined. Knowing the flux density at each pole face as a function of field excitation, the magnetic force on each pole is calculated using equation 2.1-1. The net magnetic force unbalance on the rotor is then determined by vector solution of the rotor force diagram (figures 2.1-13 and 2.1-14).

The procedure described above was used to calculate the magnetic unbalance forces on the rotor of the test fixture for a rotor eccentricity of 0.030 inch. Results are presented in figure 2.1-18. The calculated values are close to the test values (obtained from the load cells) up to 17 amperes field current at which point the calculated curve indicates premature saturation. This is due primarily to inexact steel saturation characteristics and assumptions which must be made in the no load saturation calculations. Nevertheless, the calculated peak magnetic unbalance force is within eight percent of the test value.

(f) Magnetic Moment on Rotor Due to Misalignment. Misalignment is defined as an angular displacement of the rotor axis of rotation from the centerline of the stator bore (see figure 2.1-19). This condition theoretically produces a force moment on the rotor along the axis of rotation. Tests were to be performed for rotor misalignments of 0.0002, 0.0005, and 0.0010-inch-per-inch. Starting with the low value of misalignment during test, the load cell readings were very small (1 to 3 pounds) so it was decided to misalign the rotor axis by 0.020 inch as measured from the stator locking rings (see figure 2.1-2) which gives approximately 0.00218 inch-per-inch misalignment. This value is more than twice the original maximum misalignment value to be used. This condition resulted in load cell readings of 10 and 17 pounds for the drive end and antdrive end respectively.

FIGURE 2.1-18. Radial Magnetic Force at 0.030 Inch Rotor Eccentricity

FIGURE 2.1-19. Rotor Misaligned With Stator Bore

Figure 2.1-20 gives dimensions pertinent to determination of the magnetic moment on the rotor and the corresponding force-moment diagram for the rotor. R_1 and R_2 represent the net magnetic forces on the rotor poles and P_1 and P_2 represent the load cell reaction forces.

FIGURE 2.1-20. Machine Dimensions and Force-Moment Diagram

R_1 and R_2 were calculated from the measured flux values using equation 2.1-4 and the force diagrams given in figure 2.1-21. R_1 and R_2 were also calculated from the load cell readings and the force-moment diagram shown in figure 2.1-20.

FIGURE 2.1-21. Force Diagrams

A comparison of the values of R_1 and R_2 as calculated by flux measurements and as calculated by the load cell readings is presented in the following table.

<u>Force</u>	<u>Calculated From Flux Readings</u>	<u>Calculated From Load Cell Readings</u>
R_1	33.73	35.16
R_2	55.97	42.16

The results shown in the above table verify that the reaction forces on the load cells (or bearings) result from the unbalanced magnetic forces acting on the rotor poles which result in turn from the change in radial gap lengths caused by rotor misalignment. Misalignments as great as 0.00218-inch-per-inch are not likely to occur in practice; therefore, it is concluded that magnetically induced axial force moments on the bearings due to misalignment will be negligably small compared to other forces acting on the bearings. This conclusion is based on the assumption that the generator radial gap is sufficiently large with respect to the misalignment condition, in this case the ratio of nominal radial gap (0.122 inch) to the misalignment (0.00218 inch-per-inch) is 56.

2.1.2 Magnetic Flux Distribution Tests

(a) Background and Test Objective. The generator output voltage at any given speed is a function of the magnitude of the change in magnetic flux cutting the stator conductors. This flux can therefore be considered the useful flux, while the difference

between this useful flux and the total flux is considered leakage flux. An important step in the initial design of any generator is to compute the values of both useful flux and leakage flux. It is necessary to experimentally verify these flux distribution calculations so that: (1) accurate predictions of generator performance and operating characteristics can be made well in advance of development testing, and (2) pertinent flux leakage data can be established for large inductor generators suitable for nuclear space electric power systems. Determination of this data now will improve the design accuracy of later generator configurations required by such programs.

(b) Operating Theory of a Radial Gap Inductor Alternator. Calculation of the flux in a particular component of an alternator cannot be made independently of other magnetic circuit components. The magnetic circuit is analogous to a non-linear electrical network where the current through a particular element is dependent on the arrangement and properties of other network elements. Analytical techniques to determine the flux distribution in alternators are available in standard textbooks and can be adapted, in many cases, to the solution for inductor type alternators. This report will present the general analytical considerations involved for magnetic circuit components of inductor alternators where the considerations involved depart significantly from conventional theory.

The construction of an inductor alternator consists of a stator frame which contains two armature stacks separated by an annular compartment, that carries the field coil winding as shown in figure 2.1-22. The stator frame, aside from being a structural member, completes the flux path from one stator stack to the other. It also supports the field winding.

The armature windings are located in slots contained in the armature stacks and normally extend from one stack to the other although each stack can be wound separately with winding interconnections made between stacks.

The rotor is normally machined from a solid forging of magnetic steel with a group of saliencies at each end of the rotor, which are in fact rotor poles similar to those of a conventional salient pole alternator. A 16-pole inductor alternator would have 8 poles at each end of the rotor displaced by 180 electrical degrees.

The main flux set up by the field coil MMF follows a path through the stator frame into the armature stack at one end of the alternator, through the radial gap into the rotor poles at the same end and then passes axially from one end of the rotor to the other, where the flux path is completed through the rotor poles, radial gap, and armature stack at that end of the alternator. Some of the flux does not follow the above described path, but instead "leaks" through the field coils, between the stator stacks, and around the ends of the armature stacks as shown typically by ϕ_2 , ϕ_3 , ϕ_6 , and ϕ_7 in figure 2.1-22. These flux quantities are called leakage flux because they are not useful in the generation of voltage, and yet they must be carried in parts of the magnetic circuit.

FIGURE 2.1-22. Magnetic Flux Paths and Search Coil Locations

Voltage is generated in the armature windings because the rotor presents a varying reluctance to the magnetic circuit of the armature as the rotor poles sweep past the windings during rotation. The rotor poles produce a flux density distribution in the radial gap as shown in figure 2.1-23.

Consider the flux density distribution over a double pole pitch (2λ) shown in figure 2.1-23 and the full pitch armature coil which is shown centered over a north pole. The armature coil extends the entire length of the alternator and therefore embraces not only the north flux represented by b but also the south flux represented by d and e . Therefore, the flux that is "useful" in generating voltage in the coil is represented by $b - (d + e)$.

It is convenient to consider the total flux over a double pole pitch as the base flux since the quantity repeats itself over the entire rotor periphery. Expressing the useful flux as a ratio of the total double pole pitch flux, we obtain the useful flux ratio "UFXRA".

FIGURE 2.1-23. Radial Gap Flux Density Distribution

$$\text{UFXRA} = \frac{b - (d+e)}{a+b+c} \quad (2.1-5)$$

Letting ϕ equal the flux represented by $a+b+c$, the equation for voltage generated in the armature winding is

$$E = 4(\text{FF})(f)(\phi)(N)(\text{UFXRA})10^{-8} \text{ volts} \quad (2.1-6)$$

where

FF = form factor
 f = frequency
 ϕ = flux in lines
 N = series turns per phase

The value of the useful flux ratio, as well as that of the total flux (ϕ), is required to determine the magnetic performance of an inductor alternator.

Figure 2.1-22 shows the major flux paths of an inductor alternator. The flux quantities are designated as ϕ_1 , ϕ_2 , ϕ_3 , etc. Flux quantity ϕ_1 is the sum of all other flux quantities, that is, it is the total flux in the alternator.

The leakage flux through the outer and inner field coil is designated by ϕ_2 and ϕ_3 respectively. Flux quantity ϕ_4 is the total leakage flux through the field coils and between the stator stacks. Flux quantity ϕ_5 is comprised of the total radial gap flux and the end leakage flux shown as ϕ_6 and ϕ_7 in figure 2.1-22.

The flux emanating from a rotor pole passes through the radial gap and teeth, and is then distributed in the depth behind slot (DBS) as shown in figure 2.1-24, before it enters the stator frame. Some of the flux is oriented in the circumferential direction and this distribution creates hysteresis and eddy current losses (core loss) in the DBS. That is, the flux alternates in direction through any given section of the DBS as the rotor is turned. Consideration of the flux distribution in the DBS is important in calculating alternator efficiency, since it can represent one of the higher machine losses, especially for high frequency machines.

FIGURE 2.1-24. Flux Distribution in Depth Behind Slot

The flux in the teeth does not change direction, but it does plusate with time as the rotor is turned, thereby, creating hysteresis and eddy current loss.

(c) Test Description. The essential requirement of these tests are to measure the distribution of the flux in the stator and rotor components of the 467 kVA experimental generator. To accomplish this, the stator was fitted with seven search coils and the rotor with 6 search coils. Field excitation was provided by means of a 100 volt, 100 ampere d-c power supply. Field current was increased in either three or five ampere increments to 30 amperes. Fluxmeters and field current readings were taken at each increment of field current. After each reading, the fluxmeters were reset to zero and held until just prior to raising the current to the next incremental level. The meter readings were added to obtain the total flux curve as a function of field current. This procedure was used to minimize errors caused by drifting of the pointers. The test rig was demagnetized after each test run by reversing the polarity of the field current from 30 amperes to zero.

(1) Search Coil Locations. Figure 2.1-22 shows the flux path of seven of the flux quantities to be measured and the location of the search coils to measure these quantities. The flux quantities are designated as ϕ_1 , ϕ_2 , ϕ_3 , etc. Flux quantity number one which is the summation of all other flux quantities was measured by coil number 1, which was wound around the outside diameter of the outer field coil. Search coils numbered 2A and 2B, 3A and 3B, 4A and 4B were connected in series opposition to measure the flux quantities ϕ_2 , ϕ_3 , and ϕ_4 respectively. Flux quantities 2 and 3 are the leakage flux through the outer and inner field coils respectively. Flux quantity 4 is the total leakage flux through the stator field coils and between stator stacks. Search coil number 5, wound around the rotor core outside diameter, measures the total flux ϕ_5 through the rotor core. Search coil numbers 6 and 7 were wound around the viscoseal outer diameters and measured the end leakage flux quantities 6 and 7 respectively. Coils numbered 8, 9, 10, and 11 were initially to be used in the magnetic unbalance force tests but new coil designations were later applied.

Figure 2.1-25 shows the flux path of the flux quantities ϕ_{12} and ϕ_{13} and the location of the corresponding search coils. Flux quantity number 12 is the flux through a single tooth and flux quantity 13 is the total of the flux through two teeth and one slot. The original intent in measuring ϕ_{13} was to determine the slot flux by subtracting twice the flux reading of coil 12. However a flux reading accuracy of ± 4 percent (see section on flux meter calibration) precluded accurate determination of the slot flux in this manner since the slot flux represents a small percentage of the tooth flux. Alternate schemes to determine the flux more accurately were devised but it was decided not to incorporate them into the fixture during test in order to facilitate testing and since slot flux is not a critical quantity in determining machine performance.

FIGURE 2.1-25. Tooth Flux

Figure 2.1-26 shows the locations of search coils 14 and 15 which were used to measure the flux distribution in the radial gap over a double pole pitch. In these tests, coils 14 and 15 were connected in series opposition and flux readings were taken for various field excitations. Then, coil 15 alone was connected to a flux meter and the test repeated. These test values were used to calculate the radial gap useful flux ratio. Coil 14 and 17 were connected in series opposition in order to determine the percentage of the flux embraced by coil 14 that enters the sides and faces of the poles instead of entering the rotor core diameter between poles. Coil 16 (not shown) was originally to be used to measure pole face flux but was later discarded.

The location of search coil 18 is shown in figure 2.1-27. Search coil 18 was wound through a stator slot channel, along the length of one stack, up the end of the stack through a lead channel in the field coil housing, across the frame over the stacks, and back into the slot channel through a split pin in the frame and stack clamping ring. When wound in this manner, coil 18 measures all of the circumferential flux (ϕ_{18} in figure 2.1-27) in the armature core (i.e., the DBS). To measure the flux distribution in the DBS, the rotor was turned by hand in increments of one-half slot pitch (one-half slot pitch = 12 electrical degrees) for a minimum of 15 slot pitches. This test was performed at a field excitation of 15 amperes. The maximum

FIGURE 2.1-26. Search Coils for Radial Gap Flux Distribution

FIGURE 2.1-27. Search Coil for Depth Behind Slot Flux

flux in the DBS was determined at other field excitations by rotating the rotor a minimum of 90 electrical degrees and measuring maximum flux meter deflection.

(2) Flux Meter Calibration. Five fluxmeters, manufactured by Sensitive Research, model FM, and one fluxmeter calibrator, model FC, were available for the test. Preliminary tests indicated that the inaccuracy of one meter precluded its use for further testing.

The multirange model FM is a direct reading fluxmeter for the measurement of total magnetic flux in maxwells (lines). The fluxmeter is actually a galvanometer movement without restraining springs. Deflection of the pointer is opposed only by a counter emf set up by the rotation of the moving coil in a magnetic field. An external coil (search coil) with a known number of turns is connected to the fluxmeter terminals. To achieve maximum accuracy, the search coil resistance must be built up to 15 ohms. Any change in flux linking the external search coil will be indicated by the meter pointer. The pointer remains at its maximum deflection unless manually actuated to return the pointer to zero. A lever switch is mounted on the meter box for this purpose. Power is supplied by an internal battery. There are a total of 100 divisions on the meter scale with 50 divisions on either side of the zero mark located on the center of the scale (see figure 2.1-28).

FIGURE 2.1-28. Facsimile Fluxmeter Scale

The theory of operation of the fluxmeters can be rather simply explained. The external search coil encloses a definite number of lines with a fixed number of turns. An internal coil encloses a definite number of lines created by an internal permanent magnet. It requires energy to change the flux linkages of this complete circuit, but because of the absence of springs and low friction, little latent energy is present. Therefore, when the external flux changes the number of lines linking the search coil, the internal coil, from the conservation of energy theory, takes up a position such that it cuts more or fewer lines, as the case may be. The final position of the pointer is achieved when the total lines being enclosed by the whole circuit, internal and external, is the same as originally existed. Since there are no restoring springs attached to

the pointer needle, the pointer may drift somewhat if exact balance of the movement is not achieved.

The model FC fluxmeter calibrator provides 0.2 percent accurate calibration points for checking the fluxmeters which have a guaranteed accuracy of +1.5 percent with the multiplier set on 100. No data is given for the other multipliers. The calibrator provides a check for basic scale linearity at 10 points (5 points on either side of zero) as well as for all ranges at full scale. Operation of the instrument is based on the complete flux reversal of a square loop core material driven well into saturation. Saturation density is an extremely stable characteristic thus assuring good long term repeatability. The instrument uses internal batteries as its excitation source. Excitation current is set by adjusting the meter rheostat to a prescribed setting.

The meters were calibrated periodically during the test period. Calibration points were the full scale points (50 divisions) for multipliers of 100, 30, 10 and 3 and scale points of 10, 20, 30, 40 and 50 with the meter multiplier set on one. Both the right and left meter scales were calibrated. Three calibration runs were performed on the four fluxmeters for a total of 216 calibration points. Of these, 208 were within +2 percent of the calibration values, and 213 were within +3 percent. The remaining three points were +4 percent, +6 percent, and +6 percent. From the above statistics, overall meter accuracy is considered to be within +2 percent of the calibrator flux value. This accuracy was obtained under controlled calibration conditions. In the actual tests, flux quantities were measured using the following procedure. Starting with a completely demagnetized test fixture, the current through the field coil was increased in 3 ampere increments to 30 amperes. After reading the fluxmeter at a given current increment, the fluxmeters were reset to zero and held until just prior to raising the current to the next incremental level. The meter readings were added to obtain the total magnetization curve as a function of field current. This procedure was used to minimize any errors caused by drifting of the meter pointers. The rate of drift of the meter pointers varied for each meter and was also dependent upon the position of the pointer on the meter scale.

It was considered desirable to determine the percent deviation of the meters from exact repeatability under actual test conditions. This test would allow evaluation of a combination of effects not present during calibration. These include meter drift over more prolonged periods of time than used under calibration condition, meter readability variations between four different persons, accuracy of setting and reading the current value from the dc power supply, and the effects of current drift due to heating of the test fixture field coil. Four wires were placed in a common sleeve and the sleeve was wrapped several times around the rotor core of the test fixture to form a common set of search coils. The four coils, one for each meter, were cut by exactly the same flux when the current was changed. One coil was connected

to each fluxmeter. The meters were read simultaneously during the test. Current through the field coil was increased in 3 ampere increments to 30 amperes with flux readings taken at each current. The test was repeated with 10 ampere increments to 30 amperes, and then repeated again in one 30 ampere increment. The search coil polarity was then reversed and the entire test repeated.

Test results are shown in figures 2.1-29 through 2.1-34. Figure 2.1-29 shows the variation between meters when read on the left scale. The percent deviation from exact repeatability for currents of 10, 20 and 30 amperes are shown at the bottom of the figures. The lowest fluxmeter reading was taken as the base value. Figure 2.1-30 shows the variation between meters when read on the right scales. Figures 2.1-31 through 2.1-34 show results for each individual meter on both the left and right scales. The average deviation (average of 18 points shown in figures 2.1-29 through 2.1-34) is five percent. The maximum deviation is 7.6 percent. If it is assumed, and it seems reasonable to do so, that the absolute flux value is located half way between the variation limits, then the average deviation would be +2.5 percent and the maximum deviation would be + 3.8 percent. The previously established fluxmeter tolerance of ± 2 percent would be included in these limits. Considering the stated tolerance of the model FC calibrator (± 0.2 percent), it is concluded that the recorded flux values are within ± 4.0 percent of their absolute values.

(3) Load Cells. The magnetic forces on the rotor were measured with two strain gage load cells in conjunction with a bridge amplifier meter. The load cells were model 3112-500 manufactured by Leban Instrument Company, Oak Park, Michigan and the meters were model BAM-1 manufactured by Ellis Associates. The rotor was suspended through the load cells by adjustable linkages used for setting rotor eccentricity. Although pinned attachments were used at the upper linkage support joints, some extraneous bending moments could be imposed on the load cells because of rigid attachment at the lower joints. This possibility was thoroughly investigated during the preliminary verification test runs. Perfect axial alignment of the rotor in the stator was not obtained at initial assembly. Excitation of the field coil produced magnetic forces which moved the rotor toward the aligned position. This motion was restrained by bending forces imposed on the load cells which in turn gave erroneous radial load indications. Shims were placed between the stator and the end bell to axially align the rotor. Alignment was assured when excitation of the field coil produced no axial rotor movement as measured by an 0.0001-inch per division dial indicator.

The load cells were calibrated by the manufacturer and a resistor was supplied with each cell for on-site calibration prior to each test run. Accuracy of calibration was verified in a Baldwin tensile test machine, recently calibrated to within 0.6 percent error, prior to and following the static tests. For verification, the cells were mounted in the jaws of the tensile

FIGURE 2.1-29. Fluxmeter Readings - Meters A, B, C, D, Left Scale

FIGURE 2.1-30. Fluxmeter Readings - Meters A, B, C, D, Right Scale

FIGURE 2.1-31. Fluxmeter Readings - Meter A

FIGURE 2.1-32. Fluxmeter Readings - Meter B

FIGURE 2.1-33. Fluxmeter Readings - Meter C

FIGURE 2.1-34. Fluxmeter Readings - Meter D

test machine through hinge pins which insured the application of tensile loads without imposing extraneous bending moments. The wiring harness, used in all testing, was used to connect the load cells to the amplifier meter during verification. The resistors were connected across the load cells to allow adjustment of the BAM-1 meter to read the appropriate calibration value. The calibration resistor was removed and load was applied to the cell in 50 pound increments. The BAM-1 meter was read at each value of load. Figure 2.1-35 indicates the results of verification with corrections included for tensile test machine error.

A simulated rotor weight of 80 pounds was applied to each cell. The calibration resistor was then placed across the load cell to verify utilizing the resistor calibration technique with the rotor tare loads on the cells. This calibration technique was used in all subsequent testing.

FIGURE 2.1-35. Load Cell Calibration Curve

(d) Test Results. Measured and calculated flux values for flux paths 1, 4, 5, 6, and 7 (figure 2.1-22) are plotted in figure 2.1-36. The curves are shown on a common graph to illustrate the relative magnitudes of the leakage flux quantities ϕ_4 , ϕ_6 , and ϕ_7 to the main path flux ϕ_5 and the total alternator flux, ϕ_1 . The leakage flux ϕ_4 through the field coils and between stacks is about 16 percent of the total alternator flux ϕ_1 . End leakage flux (ϕ_6 and ϕ_7) is about 2.5 percent of the total flux.

Values of stator leakage flux for flux paths 2, 3, and 4 as determined by test are plotted on a common graph, figure 2.1-37, to show the relative magnitude of the leakage flux through the outer field coil ϕ_2 , and the inner field coil ϕ_3 , to the total field and stack leakage flux ϕ_4 . Flux through field coils (ϕ_2 and ϕ_3) represents about 72 percent of the total stator leakage flux ϕ_4 .

The flux in path 5 is comprised of the total radial gap flux and a small amount of end leakage flux. The total radial gap flux was calculated from a flux plot of the radial gap area as portrayed in figure 2.1-7. End leakage flux, which is normally a small proportion of the total generator flux, is estimated by considering the air gap ampere turns as the driving MMF over a permeance path from the ends of the armature stacks to the rotor ends. The field leakage flux is obtained as follows. (See figure 2.1-38.)

The field coil conductors are uniformly distributed in the coil cross-section. The procedure is to calculate the flux through the incremented area dx , and integrate this expression over the total height of the field coil.

The flux leaking through the field coil is

$$\begin{aligned} d\phi_{LC} &= 3.19 \text{ (FLDNI)} \frac{(x)}{(\text{FLDH})(\text{LBS})} (2\pi)(R-x)dx \\ \phi_{LC} &= 3.19 \frac{(\text{FLDNI})}{(\text{FLDH})(\text{LBS})} \int_0^{\text{FLDH}} x(R-x)dx \quad (2.1-7) \end{aligned}$$

where

FLDNI = field ampere turns
 FLDH = field coil height
 LBS = length between stacks
 R = frame over field inner radius

The flux leaking from stack to stack is

$$\phi_{LS} = 3.19 \frac{\text{FLDNI}}{\text{LBS}} (\text{ALAM}) \quad (2.1-8)$$

where

ALAM = area of lamination below the field coil.

FIGURE 2.1-36. Alternator Flux Curves

FIGURE 2.1-37. Field Leakage Flux

FIGURE 2.1-38. Inductor Alternator Stator

The total flux leaking through the field coils and between stacks is

$$\phi_4 = \phi_{LC} + \phi_{LS} \quad (2.1-9)$$

The flux ϕ_{12} in a single tooth plotted as a function of field excitation is shown in figure 2.1-39. This flux is obtained with the tooth positioned directly over a rotor pole and is therefore the maximum flux seen by the tooth. The maximum tooth flux amplitude is required to determine the voltage performance of an alternator. Tooth flux was calculated by standard analytical procedures which consider the permeance of the radial gap and pole. As the rotor is turned such that the tooth in question is positioned directly between the poles on one end of the alternator, the tooth flux decreases to a minimum value which is greater than zero. The ac tooth flux amplitude is defined as the difference between the maximum amplitude (ϕ_{12}) and the minimum value described above. The ac tooth flux amplitude determines the hysteresis and eddy current loss in the armature teeth. The ac amplitude is about 92 percent of the maximum tooth flux amplitude as determined from the test curves of figure 2.1-40. The calculated ratio determined from a radial gap flux plot made for the test machine (see figure 2.1-7) is 93.6 percent.

The radial gap flux quantities measured by search coils 14, 15, and 17 are shown in figure 2.1-41. Refer to figures 2.1-23 and 2.1-26 for the location of the search coils and a

FIGURE 2.1-39. Single Tooth Flux

FIGURE 2.1-40. Single Tooth Flux

FIGURE 2.1-41. Radial Gap Flux

portrayal of the flux distribution measured. An interesting result of the test is that the differential flux between coils 14 and 17 is negligibly small which indicates that almost all of the radial gap flux enters the pole faces and sides with only a small quantity entering the rotor core diameter between poles. The flux measured by coil 15 corresponds to the flux quantity b shown in figure 2.1-23 and the flux measured by coil 14 corresponds to $a+b+c$. The flux values represented by a, c, d , and e are all equal due to symmetry and therefore the useful flux ratio can be determined from equation 2.1-5 using the search coil readings as follows:

$$\text{UFXRA} = \frac{b-(d+e)}{a+b+c} = \frac{\phi_{15}-(\phi_{14}-\phi_{15})}{\phi_{14}} \quad (2.1-10)$$

The useful flux ratio determined from test data and as calculated from design data is shown in figure 2.1-42. As mentioned previously, the useful flux ratio is a factor in the voltage equation for the alternator.

The most significant result of the flux distribution test is the determination of the ac flux amplitude in the armature depth behind slot (DBS). Test results were significantly different from the calculated values as shown by curve a in figure 2.1-43. The manner in which the radial gap flux enters the DBS and is distributed there is portrayed in figure 2.1-24. The flux oriented in the circumferential direction creates hysteresis and eddy current losses (core loss) in the DBS. The calculation procedure used to determine the circumferential flux was to assume, as a pessimistic first approximation, that the amplitude of the circumferential flux varied linearly with pole embrace with the function equal to zero for 200 percent pole embrace and unity for zero percent pole embrace. The basis for the assumption was that for a 200 percent pole embrace the poles on each end of the machine merge to form solid rings and the radial gap flux would be completely distributed by the teeth before entering the DBS and that for practically zero percent pole embrace almost all of the pole flux would be circumferentially distributed. The tests show that the accuracy of this approximation is not acceptable and that a more rigorous theoretical analysis is required. Consider the radial gap flux distribution shown in figure 2.1-23. Application of Fourier theory states that the radial gap distribution can be expressed as the summation of a dc flux component, a fundamental frequency ac flux component, and a series of odd numbered ac harmonic components. The following conclusions are drawn by considering each flux component individually. The dc flux component is uniformly distributed around the rotor periphery at each end of the alternator and therefore requires no circumferential distribution in the armature DBS, similar to the case of 200 percent pole embrace where the poles merge to form solid rings. The fundamental ac flux must travel a circumferential path in the DBS as shown in figure 2.1-44, which portrays

FIGURE 2.1-42. Useful Flux Ratio

FIGURE 2.1-43. Depth Behind Slot Maximum Flux Amplitude

the situation for one end of the alternator. The flux space distribution at the other end of the alternator is exactly the same. This can be visualized by shifting the poles in figure 2.1-44 by 180 electrical degrees (one pole pitch). Harmonics are also distributed circumferentially but do not span the entire pole pitch. They act to distort the sinusoidal distribution of the fundamental. For example, the third harmonic acts to make the apex of the DBS flux distribution wave more peaked.

Figure 2.1-45 shows the distribution wave measured in the DBS as a function of slot pitches, there being 24 electrical degrees per slot pitch. The data for this curve was obtained at a constant field excitation of 15 amperes by turning the rotor in increments of one-half slot pitch (12 electrical degrees) and measuring the circumferential flux in the DBS at each position with search coil 18.

FIGURE 2.1-44. Fundamental AC Flux Paths

FIGURE 2.1-45. Depth Behind Slot AC Flux Distribution

The calculation procedure for determining the flux amplitude in the DBS is as follows. A Fourier analysis is made of the radial gap flux distribution to determine the amplitude A_1 of the fundamental flux wave relative to the peak radial gap flux density B_{gp} . The average value of a sine wave is $2/\pi$ times the peak amplitude and the flux through the DBS is $1/2$ of the average fundamental flux. Therefore

$$\phi_{DBS} = \frac{(A_1)(B_{gp})(T_1)(SL)}{\pi}$$

where

ϕ_{DBS} = DBS flux amplitude

A_1 = amplitude of fundamental relative to the peak radial gap flux density

B_{gp} = peak radial gap flux density

T_1 = length of pole pitch

SL = stack length

(e) Summary. The net magnetic force acting on the rotor of the 467 kVA experimental generator for various degrees of eccentricity between the rotor and stator, and the associated unbalanced magnetic fields were measured. A force of 19.1 pounds per 0.001-inch eccentricity was found to be the maximum and occurred at a field current of 21 amperes. Misalignment (i.e., tilting of the rotor axis of rotation) of the rotor did not cause large reaction forces on the bearings. The theoretical relationship between the magnetic fields of an inductor alternator and the net force on the rotor was discussed. A generalized procedure for calculating the rotor forces from design data was formulated.

The flux distribution in the magnetic components of the 467 kVA experimental generator was measured and compared to calculated values. Good agreement was demonstrated except for the flux in the armature DBS where the measured value was about 60 percent of the calculated value. A new calculation procedure to determine the DBS flux was formulated and demonstrates good agreement with the measured flux.

2.2 FIELD COIL TEMPERATURES

2.2.1 Background and Test Objectives

Performance of the magnetic force unbalance and flux distribution tests provided an opportunity for measuring the thermal performance of the field coil cooling configuration. There were a number of considerations in the design of the field coil which required a careful analysis of the coil cooling design and that it be verified. First, the wire used was a copper base alloy, ZrCu, that had temperature limitations around 1000° to 1200°F.

Secondly, the calculated hot-spot temperature was of the order of 1000°F which left little room for error; since the analysis must be based on several assumptions regarding contact resistance, it is subject to unavoidable error, especially between insulations, both rigid and flexible. Thirdly, since the coil must be well insulated electrically, it is naturally in a poor thermal transport configuration; at high heat fluxes and for a heat source that increases in output as temperature increases, the problems of minimizing the hot-spot temperatures are compounded. Because of this, careful attention was paid to determining and measuring the suspected hot-spot temperatures.

The specific objectives of the tests were to:

- a) measure temperatures in the inner field coil to determine the radial heat transfer characteristics of the winding.
- b) measure temperatures in the outer field coil to determine the axial heat transfer characteristic of the winding to the cooling fin and to determine the radial heat transfer characteristics of the cooling fin.
- c) measure the temperatures in the coil and frame to determine the extent of the cooling capacity of the frame surrounding the coil.

The first two objectives, being of most importance, were achieved with the tests made. The latter objective was achieved qualitatively but not quantitatively.

The field coil has 316 turns in a series-wound split configuration, figure 2.2-1. Part of the coil is located outside the cooling tubes; the other part is located inside the cooling tubes. Alumina insulated copper cooling fins, which are brazed to the stator cooling tubes, extend into the outer field coil to act as heat sinks for the outer field copper losses. The thermal conduction paths on the inner field coil are short enough to obviate the need for cooling fins; a potting compound is included to fill the voids between the cooling tubes and inner coil outside diameter. The coil wire is rectangular 0.15 percent zirconium copper alloy that has been nickel coated to protect it during manufacturing insulation bakeout. The insulation is double-served, silicone bonded, S-glass. Rigid alumina and flexible silicone glass mica insulate the coil from the surrounding frame and stacks.

The cooling configuration of the coil is such that it was not specifically designed to operate in a vacuum. It was designed to operate in a gas ambient with what was hoped to be enough of a safety margin to allow at least partial vacuum operation. Tests, as described within, were to be made to check it out prior to finalizing the actual space-generator design.

2.2.2 Test Description

The primary tests were carried out during the static tests. Spot checks were made later during the no-load dynamic tests (Section 4) but were not as conclusive as the original static test results. A total of 35 thermocouples were mounted on or in the static test assembly, figure 2.1-2; seventeen of these were in the coil proper, the remaining eighteen were on the frame or cooling tubes. The location of the various thermocouples is illustrated schematically on figure 2.2-1.

The temperatures were monitored on a multipoint recorder, a dial indicator and a comparative potentiometer. The input power was monitored by measuring the coil volts and amperes. The output power was determined from the water flow rate (through a flowmeter) and the measured temperature rise across the coolant distribution manifolds. The test assembly was originally not insulated from the ambient but because of poor heat balances, was insulated by covering with a plastic foam box for the last static test. The power supply used was the same as that for the previously discussed magnetic flux distribution tests.

During the tests, it was determined that the coolant flow was uneven in the upper and lower cooling tubes. This was due to gravitational effects being predominant at the low flow rates necessary to obtain measurable water temperature rise; the pressure (head) drops through the lines were so small that they were of the order of less than one foot of water. As a result, more coolant was flowing through the lower tubes resulting in lower temperatures on the lower portion of the coil. Because of this, an unmeasured amount of heat was flowing circumferentially from the upper portions of the coil to the lower temperatures in the lower portion. Thus, average temperatures between the upper and lower coil had to be used to analyze the data relative to the radial inner coil heat flow.

The static tests were carried out at 10, 20, 30, and 35-amperes field current. The high 35-ampere current (rated 31 amperes at full-load) was necessary to get sufficiently high losses at the lower temperatures to simulate those expected during higher temperature potassium cooled operation. Because of this, the lower current ratings were not analyzed for comparative purposes; only the 30 and 35 ampere cases were analyzed in detail.

The temperatures measured are tabulated in table 2.2-1 along with those from the no-load dynamic tests. The temperatures of initial importance to the first two objectives of the static tests are indicated by double asterisks. Plots of these static test temperatures against field current indicated a steady rise with current without any unusual deviations; as a result, the values shown can be considered representative. Readings were not taken until at least one-half an hour after steady state conditions were initially indicated.

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2.2-1

FIGURE 2.2-1. Coil Schematic and Thermocouple Locations

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TABLE 2.2-1. Summary of Test Data Field Coil Temperatures (°F)

Thermocouple No.	Field Amps Recorder*	Static Tests				Dynamic Tests			General Location
		10 amps	20 amps	30 amps	35 amps	6000 rpm 25 amps	6000 rpm 20 amps	12,000 rpm 20 amps	
1	D	59.0	86.0	--	152.6	--	--	--	Inner field coil
2	D	--	101.3	--	235.5	--	--	--	Outer field coil
3	D	53.6	59.9	75.2	68.0	--	--	--	Frame O. D.
4	D	54.5	68.9	--	102.2	--	--	--	Frame O. D.
5	D	54.5	69.8	100.4	104.0	--	--	--	Frame O. D.
6	D	54.5	68.9	--	104.0	--	--	--	Frame O. D.
7	D	54.5	69.8	--	95.0	--	--	--	Frame
** 8	M	85.1	184.0	338.0	374.0	364.0	208.0	234.0	Inner field coil
** 9	M	74.3	148.0	266.0	329.0	344.0	198.0	254.0	Inner field coil
** 10	M	71.6	136.2	244.5	287.0	155.0	115.0	89.0	Inner field coil
11	M	59.9	97.7	178.0	217.0	300.0	165.0	89.0	Outer field coil
** 12	Open	--	--	--	--	--	--	--	Outer field coil
13	M	60.8	110.3	215.5	272.0	327.0	172.0	205.0	Outer field coil
** 14	M	59.9	93.3	170.5	190.4	112.0	92.0	220.0	Cooling fin
** 15	M	86.0	187.9	351.0	417.0	334.0	217.0	110.0	Inner field coil
** 16	M	77.0	162.5	308.0	426.0	87.0	80.0	252.0	Inner field coil
** 17	M	72.5	145.2	257.0	309.0	258.0	177.0	130.0	Inner field coil
18	M	60.8	104.0	199.5	252.0	277.0	160.0	220.0	Outer field coil
** 19	M	72.4	159.0	342.0	449.0	170.0	120.0	209.0	Outer field coil
20	D	66.2	121.1	230.0	300.0	125.0	97.0	266.0	Outer field coil
** 21	M	59.9	94.1	169.0	194.0	90.0	83.0	100.0	Cooling fin
** 22	D	78.7	157.0	--	322.0	--	--	--	Inner field coil
24	D	57.2	76.1	111.3	120.2	148.0	120.0	90.0	Stack
31	D	56.3	75.2	108.5	113.0	174.0	27.0	200.0	Stack
36	D	53.6	68.0	95.9	100.4	--	--	--	Frame cap
37	D	64.4	69.8	78.8	100.4	--	--	--	Frame cap
38	D	55.6	69.8	96.8	100.4	--	--	--	Frame cap
39	D	55.6	69.8	72.5	102.2	--	--	--	Frame cap
Cooling Water									
Inlet:									
Manifold	P	50.0	49.3	52.2	42.0	72.2	67.0	68.0	
Top Tubes	D	--	--	--	48.2	--	--	--	
Bottom Tubes	D	--	--	--	44.6	--	--	--	
Outlet:									
Manifold	P	54.7	60.1	71.4	58.0	97.0	--	120.0	
Top Tubes	D	--	--	--	63.5	100.0	75.0	130.0	
Bottom Tubes	D	--	--	--	63.5	73.0	65.0	95.0	
Side Tubes	D	--	--	--	62.2	--	--	--	
* M - Multipoint D - Dial Indicator P - Potentiometer									
** Temperature critical to analysis									
Note: All temperatures originally measured in °C.									

2.2.3 Analysis of Results

Rather than calculate various temperatures in the field coil for comparison with those measured, the approach was taken to find the equivalent conductivities which predict the measured temperatures and then use these to predict the hot spot temperatures for potassium coolant operation.

(a) Inner Field Coil. The inner field coil is cooled primarily by outward radial conduction to the cooling tubes running axially through the stator frame and coil. The wire is 0.045 by 0.144-inch strap that is wound on the flat side so that heat passing radially through the coil passes through the 0.045-inch length. The wire is coated with a protective nickel cover that is not more than 0.0002-inch-thick (included in the 0.045 by 0.144 dimensions). The insulation is 0.0055 inches thick on each side. The thermal conductivity of the three materials in series makes up the equivalent conductivity of the wire. That is:

$$k_{\text{wire (radial)}} = \frac{l_{\text{cu}} + l_{\text{ni}} + l_{\text{i}}}{\left(\frac{l}{k}\right)_{\text{cu}} + \left(\frac{l}{k}\right)_{\text{ni}} + \left(\frac{l}{k}\right)_{\text{i}}} =$$

$$\frac{\frac{0.0446}{212} + \frac{0.0004}{36} + \frac{0.011}{0.3}}{\frac{0.0446}{212} + \frac{0.0004}{36} + \frac{0.011}{0.3}} = 1.52 \frac{\text{BTU}}{\text{hr.ft.}^\circ\text{F}}$$

The thermal conductivities of the various materials are tabulated in table 2.2-2.

The radial temperature distribution within the inner field coil is determined with the equation:

$$t(r) = \frac{1}{\alpha} \left\{ \frac{1 + \alpha t_0}{B - \frac{AE}{D}} \left[-E/D J_0 \sqrt{B} r + Y_0 \sqrt{B} r \right] - 1 \right\}^*$$

---

* From Vol. I, NAS 5-1234 reports; re-derived for boundary conditions $dt/dr=0$ at $r=R_i$ and $t=t_0$ at $r=R_0$.

TABLE 2.2-2. Thermal Conductivity of Materials
at Static Test Conditions

Material	k (BTU/hr. ft. °F)	Source
ZrCu Alloy	212	Anaconda Copper Property Data File
Oxygen Free Copper	226	
Nickel	36	INCO Property Data File
Glass Serving Insulation	0.3	NASA-CR-54092
Air at 1 atm	0.023	Eckert & Drake, <u>Heat and Mass Transfer</u>
W839 Potting Compound	0.46	NASA-CR-54092
316 SST	9.4	SST Supplier's Data
Al ₂ O ₃ , 94%	6.0	NASA-CR-54092
Silicone Glass Mica	0.032	NASA-CR-54092
Glue (cement, 0.001" thk)	0.3	NASA-CR-54092

where

$$t(r) = T(r) - 25$$

$T(r)$ = temperature at any radius r , °C

$$t_o = T_o - 25$$

T_o = known temperature at outside radius r_o , °C

α = temperature coefficient of resistance, 1/°C

$$b = \alpha q_o / k_e$$

q_o = $I^2 R$ loss at 25°C, watts per cubic inch of coil
volume

k_e = equivalent conductivity of coil, watts/in/°C

R_i = inner radius, inches

R_o = outer radius, inches

$$A = J_o (\sqrt{b} R_o)$$

$$B = Y_o (\sqrt{b} R_o)$$

$$D = J_i (\sqrt{b} R_i)$$

$$E = Y_i (\sqrt{b} R_i)$$

J_o, Y_o, J_i, Y_i = Bessel Functions

The various constants used in the analysis of the test data are tabulated in table 2.2-3. The calculated equivalent conductivity, k_e , was calculated by using the following temperatures and corresponding radii from the tests:

$$\begin{aligned} r &= R_i = 6.52 \text{ inches} \\ T(r) &= T_i = \text{average of } T_9 \text{ and } T_{10} \\ R_o &= 6.95 \text{ inches} \\ T_o &= \text{average of } T_{10} \text{ and } T_{17} \end{aligned}$$

TABLE 2.2-3. Inner Field Coil Analysis Data

Test	Static	Static	Dynamic
Field Amps	30	35	20**
Average Coil Temperature, °C	149	193	81
*Calculated, α , 1/°C	0.00362	0.00363	0.00342
Calculated q_o , watts/in ³	12.8	17.4	5.7
Resulting Calculated k_e , watts/in °C (or) BTU/hr ft °F	0.01482 0.338	0.01347 0.306	0.00702 0.160
Volume of coil = 30.8 in³ Volume of wire in coil = 21.4 in³ Packing factor = 0.695 * $\alpha \equiv \frac{\left[\frac{\text{measured resistance}}{\text{room temp. resistance}} \right]^{-1}}{[\text{avg. temp., °C} - 25]}$ ** Average of both the 6000 rpm and 12,000 rpm cases.			

Since the calculated equivalent conductivities do not correspond to the insulated wire conductivity of 1.52 BTU/hr.ft. $^{\circ}$ F, one must conclude that there is an additional thermal conductance (resistance) to be considered (e.g., contact resistance). Likewise, since the calculated resistances differ between the static and dynamic tests, one must conclude that the difference is attributable to the fact that the two tests were run at different ambient pressure conditions. The static tests were in an atmospheric air ambient, the dynamic tests were in a moderate vacuum (approximately 0.5 psia) of helium/air mixture ambient. The lower calculated equivalent conductivity for the dynamic tests is certainly indicative of the greater resistance to heat flow in a vacuum. Comparison of the various conductivities lead to the conclusion that the simplified model on figure 2.2-2 might be used to simulate the equivalent coil conductance.

The contact conductance of 50 BTU/hr.ft ^{2} $^{\circ}$ F was first suggested by analyzing the no-load dynamic test data assuming there was no gas conductance path; the calculated conductance was 38.3 BTU/hr.ft ^{2} $^{\circ}$ F over an assumed 100 percent contact area. This correlated to an analysis using the relations of Clausing and Chao (reference 2-1) which predicted a contact conductance of 50 to 75 BTU/hr.ft. $^{\circ}$ F for a contact load of the order of one pound force. Using the lower value of fifty gives a contact area of 77 percent (38.3/50) which is intuitively reasonable. Continued analysis then leads to the length of the air gap between wire layers where there is no contact being 0.0013 inches long (which is also intuitively reasonable).

Using this simplified heat transfer model to predict the inner field coil temperatures for 625 $^{\circ}$ F potassium operation at full load (31 amperes in the field) resulted in the following conclusions. The inner field coil hot-spot temperature for a gas ambient would be 998 $^{\circ}$ F which agrees with the 1000 $^{\circ}$ F as originally predicted; in a vacuum ambient it would be in excess of this temperature, thereby exceeding a practical limit of 1000 $^{\circ}$ F. In fact, the analysis using the simplified model predicted a 2350 $^{\circ}$ F hot spot; this appears to be excessively high so one can not conclude the model on figure 2.2-2 is accurate for predicting the vacuum case temperatures. One can conclude, however, that in the vacuum ambient case, the temperature will be in excess of the 1000 $^{\circ}$ F limit which is sufficient to rule out vacuum operation for the coil configuration used.

Intermediate temperatures of interest to the design and test analysis are the temperatures between the field coil outside diameter and the coolant in the tubes. As this is a straightforward calculation of heat conduction, the analysis method will not be elaborated upon. The comparative results are shown in table 2.2-4. The detailed analysis indicated that, for the potassium case, about 88 percent of the temperature drop between the coil and cooling tubes occurs in the potting compound. For the water cases, it is only about 38 percent because of the larger film drop in the cooling tube.

FIGURE 2.2-2. Nodal Resistance Network Relating to Calculated Equivalent Conductances

TABLE 2.2-4. Calculated Inner Field Coil Temperatures on Outer Diameter

	Water Tests			Potassium
	Static	Static	Dynamic	Design
Field Amps	30	35	20	31
Avg. Coolant Temp., °F	62	50	95	625
Temp. on O.D. as:				
Concluded from tests, °F	202	224	115	-
Calculated, °F	172	208	175	731

(b) Outer Field Coil. In the outer field coil, the fin temperatures were analyzed to determine the effectiveness of the fins for removing heat from the center of the coil. A comparative thermal map of the fins was not made since it is difficult if not impossible to predict how the heat divides itself leaving the coil. It can divide itself radially in two directions and axially in two directions. In all four directions, electrical insulation layers, gaps and random contact areas make analytical predictions questionable.

Based on the measured temperatures of the fin and cooling water plus the calculated conductance down the fin and into the water, one finds the fins are only carrying out about 4 to 15 percent of the total heat generated in the outer coil. The rest must therefore be passing directly to the frame or cooling tubes. The reason behind this is that it is not only difficult for heat to get into the fin, but once in, it is almost as difficult to get it out; the tube and film wall surface areas that view the base of the fin are nearly as small as the fin but they have less than one-twentieth the thermal conductance. As a result, about 97 percent of the temperature drop between the water and fin outer tip occurred in the water film and stainless steel tube; the other 3 percent occurred in the fin itself. With potassium coolant to increase the film conductance, the temperature drop between the fin base and coolant is still 74 percent of the total between the fin tip and coolant.

Had the base of the fins been enlarged and flared so that heat coming down them could be spread over all the available tube surface area, the fins could have carried a maximum of 57 percent of the outer coil losses out; this is in contrast to the 15 percent (maximum) that was carried. As they were, however, the resistance was not great for potassium operation and the temperature rise between the 625°F coolant and the fin tip would

have been only 22°F assuming a proportionate increase in the outer coil losses from 1866 watts (30 ampere static test case) to 4100 watts (31 ampere design case).

The other area of high thermal resistance is the axial between the fin and edges of the wire. The conductance (or resistance) model set up to compare T_{21} and T_{19} is shown in figure 2.2-3 where the unknown conductance was expected to be dependent upon contact conductance, gas conductance, radiation, or any combination of the three. Analysis at the 30 and 35 ampere test cases indicated that, for the temperatures measured, the unknown conductivity was 0.113 BTU/hr.in²°F which corresponds to air over a 0.017 inch length. Since the mean length of the gap between the fin and centroids of the wire edge- radii calculates to be 0.015 inches, it would appear that the predominating heat transport mode was conduction through the gas. This was verified somewhat by the fact that whenever contact conductance or radiation was considered, the measured data did not correlate with the predictions.

The heat transport to the fins will be even less in a vacuum ambient since gas conduction appeared so predominate in the test data. Unfortunately, the tests at vacuum can not bear this out since the measured data during the dynamic tests does not correlate well either with itself or with the static test data. The data taken does indicate that during the dynamic tests, the fins were probably carrying out about half the heat compared to that for the previous static tests at the same power level. By using the same model, figure 2.2-3, to predict the hot spot in the outer field coil during full load potassium cooled operation (625°F coolant) we arrive at a minimum of 1020°F and a maximum of 1220°F as possible temperatures. For vacuum operation, these will be even higher. Thus one can state that, in all likelihood, the coil temperature would probably have exceeded its design temperature limit of 1000°F in its present configuration.

(c) Frame Temperatures. The frame temperatures were inconsistent and/or not complete enough to determine the direction and extent of the heat transport around, to and away from the frame. Heat balances in the earlier tests (10 and 20 amperes,

FIGURE 2.2-3. Nodal Resistance Model, Wire to Fin

static) indicated that the frame was picking up heat from the ambient. At 30 amperes, heat could be felt rising off the test fixture indicating a heat loss to the atmosphere. At 35 amperes, where the fixture was finally insulated, the ambient in the insulated box dropped to the mean water coolant temperature and a near perfect heat balance was achieved as seen in table 2.2-5. A heat balance on the field coil alone was not taken during the dynamic tests.

2.2.4 Conclusions

In its present configuration, the field coil would have exceeded its original design temperatures in a vacuum environment and would be very near (if not over) its design temperature for a gas ambient surrounding it.

Areas where predominating temperature drops occurred (and where improvements would do the most good) are in order of importance:

- 1) Outer coil - between fin and wire (too much resistance in gap between wire and fin insulation)
- 2) Outer coil - axially between wires (too much resistance in gaps between wire edges)
- 3) Inner coil - radially between wires (too low radial contact conductance)
- 4) Inner coil - in potting compound between coil and tubes (too long of path)
- 5) Outer coil - in fin at base (too small area)

In order to operate in a vacuum, an entirely different concept will be necessary to minimize the hot spot temperatures. The order above indicates that a fewer number of wide, thin straps operating at higher currents may be more appropriate by eliminating many of the contact joints typical of a small wire, many turn coil. Also, encapsulating the coil with potting compound under pressure to get a good fill might be necessary. The cooling fins could be eliminated with such a configuration since each electrical strap would be its own cooling fin.

Any configuration giving more positive contact between the wire and frame should be used if at all possible. Test data indicated the frame appears to be a better heat sink than one might first suspect.

TABLE 2.2-5. Summary of Test Data Heat/Power Balance

	Static Tests				Dynamic Tests		
					6000 rpm	6000 rpm	12,000 rpm
Input Power:							
Field Amps (I)	10.0	20.0	30.0	35.0	25.0	20.0	20.0
Field Volts (E)	18.9	42.9	81.2	105.5	72.4	40.7	48.2
Resistance (E/I), ohms	1.895	2.145	2.707	3.015	2.896	2.035	2.410
Avg. Temp. to give resulting resistance (°F)	80.0	155 ± 5	300 ± 10	380 ± 20	350 ± 15	125 ± 5	230 ± 10
Input Power (watts)	189.5	858.0	2436.0	3692.5	1810.0	814.0	964.0
Output Power:							
Coolant Flow (lb/min)	6.29	6.29	6.29	12.88	--	--	--
Temp. Rise (°F)	4.7	10.8	19.2	16.1	--	--	--
Output Power (watts)	520.0	1190.0	2125.0	3640.0	--	--	--
Percent of Input Power	274.0	139.0	87.4	98.6	--	--	--

3. INVESTIGATION OF ALTERNATOR ROTOR COOLING CONCEPTS

by J. L. McCabria

3.1 INTRODUCTION

A high-speed, liquid-cooled alternator with a solid rotor requires an effective cooling scheme for removal of the heat produced within the faces of the poles. A cooling scheme which offers distinct advantages for this task is illustrated in figure 3.1-1. With this scheme, the coolant enters the antidrive-end of the rotor along the axis of rotation. The flow is then diverted in a radial direction through the shaft extension and into a manifold which serves as a distribution duct for the coolant passages in the main body of the rotor. Each pole on the rotor is served by one of the parallel passages through the rotor. The passages are diverted outward and into the poles as shown in figure 3.1-1. The manifold on the outlet side of the rotor is similar to the inlet manifold except that it contains passages which diverts the coolant into the drive end bearing discharge chamber. Each of the manifolds contain radial vanes so that solid-body rotation of the fluid is maintained within the ducts. The advantages offered by this cooling concept are:

1. The energy losses in the face of the poles can be removed efficiently since the heat conduction paths between the pole faces and the coolant passages are short,
2. the core or the central portion of the rotor, which is inherently subjected to the greatest stress, is maintained at a low temperature which reduces the creep strain,
3. the coolant holes are conducive to the heat treatment process used in fabricating a rotor, and
4. this cooling scheme can be used in conjunction with bearing and shaft-seal systems that have been experimentally verified.

Complications associated with this cooling scheme are due to the high centripetal and Coriolis accelerations which are present in a high speed alternator. In effect, the coolant passage forms a centrifuge, a machine which uses centrifugal acceleration for separating materials of different density. In practical applications, the density distribution in the coolant will be nonhomogeneous since solid material within the system will be picked up and transported by the coolant. If these heavy materials are allowed to separate and accumulate in the passages, they may close the passages and suppress the flow of the coolant which would cause a failure of the system. In addition, the high centripetal acceleration which acts upon the fluid with the passages produces high pressures within the coolant passages. This means that a welded construction may be required.

Coriolis forces act upon the fluid since the coolant flows outward and inward from the axis of rotation. The effect of these forces is to cause secondary flows in the passages which have a radial component of flow. These secondary flows cause an increase in the flow resistance when compared with the resistance of a straight, non-rotating pipe.

An analysis of the rotor cooling concept illustrated in figure 3.1-1 is given in this report. The following considerations were included in this investigation: (a) secondary flow phenomenon in a rotating pipe based upon the inviscid-fluid theory, (b) the analogous case of secondary flows in a stationary curved pipe, (c) the frictional flow resistance in a rotating pipe based upon empirical formulae for curved pipe flow, (d) the pumping and throttling characteristic of the rotor, and (e) centrifugation of heavy materials within the rotor coolant passages. The experimental results of a fluid dynamic verification of the rotor cooling concept are also presented.

3.2 NOTES ON SECONDARY FLOW IN A ROTATING PIPE,* ANALYSIS AND METHOD

It can be shown that the Coriolis forces produce secondary flows in a pipe rotating about an axis perpendicular to the longitudinal axis of the pipe when a gradient of the velocity in the main pipe flow exists. These secondary flows can be thought of as small perturbations superimposed on the main pipe flow. In this interpretation, the secondary flow consists of small velocities acting in the cross-sectional plane of the

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* These notes were originally presented as part of a Westinghouse proprietary memo on the "Secondary Flow in a Rotating Duct by R. W. Detra and J. L. McCabria, Report No. AGT 2126, 9/30/55. Pertinent parts of the original report are presented here to give an analytical approach to the problem.

FIGURE 3.1-1 Cooling Scheme

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pipe perpendicular to its longitudinal axis. The effect of these small velocities is to affect a transport of the fluid particles within the cross section of the pipe, resulting in a deformation of the axial velocity distribution.

The fluid motion must satisfy the Eulerian equations of motion and the continuity equation. With the flow quantities expressed relative to the rotating system and the vector operator gradient expressed in terms of the derivatives in the rotating system, these equations, written in vector notation, become

$$\frac{\partial \mathbf{q}}{\partial t} + \mathbf{q} \nabla \mathbf{q} + 2\omega \times \mathbf{q} + \frac{d\omega}{dt} \times \mathbf{r} + \omega \times (\omega \times \mathbf{r}) = -\frac{1}{\rho} \nabla p + \mathbf{F} \quad (3.2-1)$$

and

$$\nabla \cdot \mathbf{q} = 0 \quad (3.2-2)$$

where

$\mathbf{F}$ = body force
 p = pressure
 $\mathbf{q}$ = vector velocity
 ρ = mass density
 ω = angular velocity of rotating system

To apply this system of four equations and four unknowns to the specific problem at hand, we expand the equations in terms of the cylindrical polar coordinates as indicated in figure 3.2-1. By considering only steady fluid motion in the absence of external forces and a constant angular velocity, we obtain the following governing equations:

$$\begin{aligned} \frac{u \partial u}{\partial r} + \frac{v}{r} \frac{\partial u}{\partial \theta} + \frac{w \partial u}{\partial z} - \frac{v^2}{r} + 2\omega w \sin \theta - \omega^2 r \sin^2 \theta &= -\frac{1}{\rho} \frac{\partial p}{\partial r} \\ \frac{u \partial v}{\partial r} + \frac{v}{r} \frac{\partial v}{\partial \theta} + \frac{w \partial v}{\partial z} + \frac{uv}{r} + 2\omega w \cos \theta - \omega^2 r \sin \theta \cos \theta &= -\frac{1}{\rho r} \frac{\partial p}{\partial \theta} \\ \frac{u \partial w}{\partial r} + \frac{v}{r} \frac{\partial w}{\partial \theta} + \frac{w \partial w}{\partial z} - \omega^2 z &= -\frac{1}{\rho} \frac{\partial p}{\partial z} \\ \frac{\partial (ur)}{\partial r} + \frac{\partial v}{\partial \theta} + r \frac{\partial w}{\partial z} &= 0 \end{aligned} \quad (3.2-3)$$

FIGURE 3.2-1. Coordinate System

Significant simplifications can be obtained in the previous equations if the following assumptions are made. We shall assume that the flow caused by the Coriolis force is secondary in nature or that the velocities of this flow are small compared to the principal axial velocity. It is further assumed that this secondary motion occurs only in planes perpendicular to the longitudinal axis of the pipe, and that there is no variation of the axial velocity with the axial distance z . Hence, the secondary flow velocities u and v can be interpreted as small perturbations superimposed on the main pipe flow. With $\partial w / \partial z = 0$ it follows from the continuity equation that the derivatives of the secondary flow velocities are of the same order of magnitude as the secondary flow velocities.

Neglecting all higher order terms in Equations 3.2-3, i.e. all squares, products and higher powers of the perturbation quantities, we obtain the following equations.

$$\left. \begin{aligned}
 \frac{w \partial u}{\partial z} + 2\omega w \sin \theta - \omega^2 r \sin^2 \theta &= - \frac{1}{\rho} \frac{\partial p}{\partial r} \\
 \frac{w \partial v}{\partial z} + 2\omega w \cos \theta - \omega^2 r \sin \theta \cos \theta &= - \frac{1}{\rho} \frac{1}{r} \frac{\partial p}{\partial \theta} \\
 \frac{\partial (ru)}{\partial r} + \frac{\partial v}{\partial \theta} &= 0
 \end{aligned} \right\} \quad (3.2-4)$$

Since $w \neq f(z)$, elimination of the perturbation pressure by cross differentiation yields

$$\frac{\partial}{\partial z} \left\{ \frac{\partial}{\partial r} (rwv) - \frac{\partial}{\partial \theta} (wu) \right\} = - \frac{\partial}{\partial r} (2r \cos \theta) + \frac{\partial}{\partial \theta} (2w \sin \theta)$$

$$\frac{\partial (ru)}{\partial r} + \frac{\partial v}{\partial \theta} = 0 \quad (3.2-5)$$

By introducing a stream function, which is a solution of the continuity equation, defined by

$$u = \frac{1}{r} \frac{\partial \Psi}{\partial \theta}; \quad v = - \frac{\partial \Psi}{\partial r}$$

there results from Equation 3.2-5 the following equation

$$\frac{\partial}{\partial z} \left\{ - \frac{\partial}{\partial r} \left(rw \frac{\partial \Psi}{\partial r} \right) - \frac{\partial}{\partial \theta} \left(\frac{w}{r} \frac{\partial \Psi}{\partial \theta} \right) \right\} = - \frac{\partial}{\partial r} (2r w \cos \theta) +$$

$$\frac{\partial}{\partial \theta} (2w \sin \theta) \quad (3.2-6)$$

Equation 3.2-6 can be integrated with respect to z , and after introducing the boundary condition that the secondary flow velocities are zero at the inlet, we obtain

$$\frac{\partial}{\partial \theta} \left(\frac{w}{r} \frac{\partial \Psi}{\partial \theta} \right) + \frac{\partial}{\partial r} \left(rw \frac{\partial \Psi}{\partial r} \right) = 2w \left\{ \frac{\partial}{\partial r} (r w \cos \theta) - \frac{\partial}{\partial \theta} (w \sin \theta) \right\} \quad (3.2-7)$$

In order to obtain a solution suitable for estimating the initial or starting effects of the secondary flow, we assume that the axial velocity w , which appears in Equation 3.2-7, is the inlet velocity only. Actually the axial velocity distribution changes from one cross section of the pipe to another, but this assumption will allow a good estimation of the initial effects of the secondary flow to be made. We consider a special case where the inlet axial velocity distribution is symmetrical about the longitudinal axis of the pipe so that $w = f(r)$ only. Equation 3.2-7 then becomes

$$\frac{\partial^2 \Psi}{\partial r^2} + \left(\frac{1}{r} + \frac{1}{w} \frac{\partial w}{\partial r} \right) \frac{\partial \Psi}{\partial r} + \frac{1}{r^2} \frac{\partial^2 \Psi}{\partial \theta^2} = \frac{2zw}{w} \frac{dw}{dr} \cos \theta \quad (3.2-8)$$

Substitution of the dimensionless quantities

$$\begin{aligned} \eta &= r/r_1 & r_1 &= \text{radius of pipe} \\ \beta &= w/w_0 & w_0 &= \text{maximum inlet velocity} \\ \delta &= z/z_1 & \Psi &= \delta r_1 w_0 \sigma(\eta) \cos \theta \end{aligned}$$

enables Equation 3.2-8 to be written in the following simple form

$$\frac{\partial^2 \sigma}{\partial r^2} + \left(\frac{1}{\eta} + \frac{1}{\beta} \frac{\partial \beta}{\partial \eta} \right) \frac{\partial \sigma}{\partial \eta} - \frac{\sigma}{\eta^2} = 2 \left(\frac{r_1 \omega}{w_0} \right) \frac{1}{\beta} \frac{\partial \beta}{\partial \eta} \quad (3.2-9)$$

where the associated boundary condition are

$$\begin{aligned} \sigma(0) &= 0 \\ \sigma(1) &= 0 \end{aligned}$$

Equation 3.2-9 can be solved approximately in the following manner. We assume a series solution of the form

$$\sigma(\eta) = \sum_{k=1}^n A_k \eta^k$$

This assumed solution satisfies immediately the first boundary condition, i.e. $\sigma(0) = 0$. Substitution of this assumed solution into 3.2-9 yields

$$\begin{aligned} &\sum_{k=1}^n (k-1)(k-2) A_k \eta^{k-2} + \\ &\frac{1}{\beta} \frac{\partial \beta}{\partial \eta} \sum_{k=1}^n k A_k \eta^{k-1} = 2 \left(\frac{r_1 \omega}{w_0} \right) \frac{1}{\beta} \frac{\partial \beta}{\partial \eta} \end{aligned} \quad (3.2-10)$$

The second boundary condition requires that

$$\sigma(1) = \sum_{k=1}^n A_k = 0 \quad (3.2-11)$$

Since β is known throughout the interval $0 \leq \eta \leq 1$, Equation 3.2-10 can be expanded in terms of the unknown constant coefficients A_k at any number of chosen values of constants within the interval $0 < \eta < 1$. Say we choose j values of η , then Equation 3.2-10 will yield j algebraic equations and these equations together with Equation 3.2-11 provide a system of $j + 1$ equations which is sufficient for solving for $k = j + 1$ constant coefficients.

The simultaneous solution of these algebraic equations for the constant coefficients can most systematically be handled using matrix notation.

In matrix notation the general problem can be expressed as

$$[d_{ij}] \{a_i\} = \{c_i\}$$

or

$$\begin{bmatrix} d_{11} & d_{12} & d_{13} & \dots & d_{1k} \\ d_{21} & & & & \\ & & & & \\ d_{k1} & \dots & \dots & \dots & d_{kk} \end{bmatrix} \begin{bmatrix} a_1 \\ a_2 \\ \\ a_k \end{bmatrix} = \begin{bmatrix} c_1 \\ c_2 \\ \\ c_k \end{bmatrix}$$

The solution is obtained after a Gauss-Jordan reduction of the matrix (d_{ij}) has been made so that it appears in the form

$$\begin{bmatrix} 1 & 0 & 0 & \dots & 0 \\ 0 & 1 & 0 & & \\ 0 & 0 & 1 & \dots & 1 \\ & & & & \\ 0 & & & & 1 \end{bmatrix} \begin{bmatrix} a_1 \\ a_2 \\ a_3 \\ \\ a_k \end{bmatrix} = \begin{bmatrix} c'_1 \\ c'_1 \\ \\ c'_k \end{bmatrix}$$

from which quite simply $a_1 = c_1$, $a_2 = c_2$, etc.

For the solution of the problem presented here, the matrix coefficients are

$$d_{ij} = (j-1)(j+1) \eta_i^{j-2} + \left(\frac{1}{\beta} \frac{d\beta}{d\eta} \right)_i \eta_i^{j-1} \quad \begin{matrix} j=1,2,3,\dots,k \\ i=1,2,3,\dots,k-1 \end{matrix}$$

$$d_{kj} = 1, \quad \begin{matrix} j=1,2,3,\dots,k \\ k=k \end{matrix}$$

$$c_i = \frac{2r_1\omega}{w_0} \frac{1}{\beta} \left(\frac{d\beta}{d\eta} \right)_i, \quad i=1,2,3,\dots,k$$

$$c_k = 0 \quad k=k$$

Since it was assumed in this analysis that the inlet axial velocity is prescribed for any specific problem, the above coefficients are very easily calculated after a choice has been made as to how many matching points, j , are to be used. The matrix reduction required to obtain the solution can be very easily programmed for solution on a digital computing machine and hence a definite time saving can be realized in obtaining the solution.

A numerical example has been solved to show the general trends of the solution of equation 3.2-9. For simplicity the inlet velocity distribution β was chosen to be parabolic being given by

$$\beta(\eta) = 1 - 0.3\eta^2$$

The large velocity gradient close to the wall of the pipe present in real flow, i.e. the region of large viscous losses, has been neglected.

The solutions of Equation 3.2-9, obtained by selecting various number of matching points, are shown by figure 3.2-2. The finite series solutions yielding these curves are as follows:

- (a) Matching points $\eta = 0, \eta = 1.0$

$$\frac{\sigma_2(\eta)}{2r_1\omega_0} = -0.1364(\eta - \eta^3)$$

- (b) Matching points $\eta = 0, \eta = 0.5, \eta = 1.0$

$$\frac{\sigma_3(\eta)}{2r_1\omega_0} = -0.0818\eta + 0.0258\eta^3 + 0.0558\eta^4$$

- (c) Matching points $\eta = 0, \eta = 0.333, \eta = 0.6667, \eta = 1.0$

$$\frac{\sigma_4(\eta)}{2r_1\omega_0} = -0.0846\eta + 0.0846\eta^3 - 0.0388\eta^4 + 0.0394\eta^5$$

- (d) Matching points $\eta = 0, \eta = 0.25, \eta = 0.50, \eta = 0.75, \eta = 1.0$

$$\frac{\sigma_5(\eta)}{2r_1\omega_0} = -0.0800\eta + 0.130\eta^3 + 0.1392\eta^4 - 0.1247\eta^5 + 0.524\eta^6$$

The convergence of the method of solution for the secondary flow is shown by figure 3.2-2. The solutions converge rapidly as the number of matching points is increased. The example shows, that with a smooth inlet-velocity distribution, the number of matching points required is relatively small. In this case the five points used to obtain the last solution are sufficient. However, if the entrance velocity is not so smooth as the one used here, the number of matching points necessary will increase.

Once the solution $\sigma(\eta)$ for the secondary flow has been obtained, an equation for the dimensionless secondary stream function can be written:

$$\psi = \delta r_1 \omega_0 \sigma(\eta) \cos \theta \quad (3.2-12)$$

Figure 3.2-2. Convergence of the Numerical Method of Solution for the Secondary Flow for $\beta = 1 - 0.3\eta^2$

Substitution for $\sigma(\eta)$ for this example enables Equation 3.2-12 to be written as follows:

$$\psi' = \frac{\psi}{2r_1^2 \omega \delta} = (-0.0800\eta + 0.0130\eta^3 + 0.1392\eta^4 - 0.1247\eta^5 + 0.5247\eta^6) \cos \theta \quad (3.2-13)$$

A plot of ψ' versus η for different values of θ can be constructed as shown by figure 3.2-3. For the example given here, values of η and ψ' are shown on cartesian coordinates for various values of θ . A graphical illustration of the secondary flow stream lines was constructed by plotting values of η and θ for a constant value of ψ' on polar coordinates as shown in figure 3.2-4. In general the stream lines are parallel with the wall in outer portion of the pipe except in the center of the pipe where the stream lines are for the most part directed parallel to the motion of the pipe.

The components of the secondary flow velocity can be determined with Equation 3.2-13 since

$$u = \frac{1}{r} \frac{\partial \psi}{\partial \theta} \text{ and } v = -\frac{\partial \psi}{\partial r}$$

For this example

$$\left. \begin{aligned} \frac{u}{V_0} &= (0.1600 - 0.0260\eta^2 - 0.2784\eta^3 + 0.2494\eta^4 - 0.1048\eta^5) \sin \theta \\ \frac{v}{V_0} &= (0.1600 - 0.0780\eta^2 - 1.1136\eta^3 + 1.2470\eta^4 - 0.6408\eta^5) \cos \theta \end{aligned} \right\} \quad (3.2.14)$$

where $V_0 = \omega z =$ tangential velocity of pipe.

A plot of u/V_0 versus η and v/V_0 versus η is given by figures 3.2-5 and 3.2-6 respectively. It can be noted from 3.2-5 that the secondary flow velocity reaches 42.5 percent of the tangential velocity of the pipe for this example. Hence, the secondary flow for this case cannot be interpreted as a small perturbation superimposed on the main pipe flow unless the tangential velocity of the pipe is low or the axial velocity of the main pipe flow is high.

The amount of distortion of a line of constant axial velocity at some point down stream of the inlet is determined by the following procedure.

Figure 3.2-7a represents two secondary flow stream lines

FIGURE 3.2-3. Secondary Stream Function Dimensionless Radius for $\beta = 1 - 0.3\eta^2$

FIGURE 3.2-4. Secondary Flow Stream Lines Due to a Velocity Distribution, $\beta = 1 - 0.3\eta^2$

FIGURE 3.2-5. Secondary Flow Velocity Due to a Velocity Distribution of $\beta = 1 - 0.3\eta^2$

FIGURE 3.2-6. Secondary Flow Velocity Due to a Velocity Distribution of $\beta = 1 - 0.3\eta^2$

$$q = \frac{\Delta\psi}{\Delta N}$$

Lines thru
points of
constant
axial velocity

FIGURE 3.2-7. Flow Stream Lines Schematic

where

ΔN = the normal distance between the lines
 ds = distance along the flow path
 q = velocity of the secondary flow = $\frac{\Delta \Psi}{\Delta N}$
 Δt = time required for the secondary stream to move
 from S_1 to S_2 = $\frac{\Delta N}{\Delta \Psi} ds$

The time required for the fluid to travel the distance dz in the pipe is given by

$$\Delta t = \frac{dz}{w}$$

Substitution for Δt in the secondary stream and in the main flow gives

$$\Delta t = \frac{\Delta N}{\Delta \Psi} ds = \frac{dz}{w}$$

$$\text{and} \quad \int_{S_1}^{S_2} \Delta N ds = \int_{z_1}^{z_2} \frac{\Delta \Psi}{w} dz \quad (3.2-14)$$

From Equation 3.2-13

$$\Psi = 2\Psi' r_1 \omega z$$

thus

$$\Delta \Psi = 2\Delta \Psi' r_1 \omega z$$

Equation 3.2-14 can now be written as

$$\int_{S_1}^{S_2} \Delta N ds = \int_{z_1}^{z_2} \frac{2\Delta \Psi' r_1 \omega z dz}{w} = \int_{z_1}^{z_2} \frac{2\Delta \Psi' r_1 \omega z dz}{w_0 \beta} = \omega \frac{\Delta \Psi' r_1}{\beta w_0} (z_2^2 - z_1^2) \quad (3.2-15)$$

The right side of the Equation 3.2-15 is equal to the area bounded by two stream lines and two normal lines drawn between the stream lines as shown by figure 3.2-7b.

For the example given here, values of w_0 , ω , r_1 , z_1 , and z_2 were assumed and areas for β equal to 0.988, 0.952, 0.892, and 0.808 were calculated. Distortion of the constant axial velocity lines was determined with the use of a planimeter as shown by figure 3.2-8.

FIGURE 3.2-8. Lines of Constant Axial Velocity in a Rotating Pipe

This theoretical treatment of the problem shows that the following factors should be considered when the rotor cooling scheme has passages which divert the coolant outward and inward from the axis of rotation:

- (a) The velocity gradients within the main flow produce a coriolis force distribution that causes secondary flows.
- (b) The secondary flows will cause an increase in the pressure drop per unit length of the pipe when compared with the flow resistance of a straight nonrotating pipe.
- (c) The phenomenon of secondary flows in a pipe rotating about an axis perpendicular to the axis of the pipe is similar to the phenomenon of secondary flow in a stationary curved pipe. Thus, it may be possible to use empirical formula which express the friction factor for flow through curved pipes to determine the friction factor for a rotating pipe.

3.3 FRICTION FACTOR FOR ROTATING PIPE

The analysis of the secondary flows in a rotating pipe given in Section 3.2 is based upon the inviscid-fluid theory. Hence, this analysis cannot contribute anything relative to the magnitude of the frictional flow resistance. However, it was demonstrated that the secondary flows in a rotating pipe are similar to the secondary flows in a stationary curved pipe. In curved pipe flow, the viscous forces produce a gradient of velocity within the main pipe flow which results in a distribution of centrifugal forces that produce secondary flows. In a rotating pipe, the viscous forces will produce a similar gradient of velocity which will result in a distribution of Coriolis forces that produce similar secondary flows. Empirical formulae which give the flow resistance for flow in a curved pipe can therefore be used to evaluate the friction factor for a rotating pipe if equivalent dimensionless parameter are used for the analogous case. Empirical formulae for the friction factor for turbulent and laminar flow in smooth curved pipes were presented by H. Ito, reference (3-1). His formulae are based upon the experimental results of several investigators. For turbulent flow he gives

$$f_c \frac{R}{r_o}^{1/2} = \frac{0.316}{[Re (r_o/R^2)]^{1/5}} \quad (3.3-1)$$

where

R = radius of curvature of the pipe axis
 r_o = radius of circular cross section of pipe
 $Re = 2 V_m \rho r_o / \mu$, Reynolds number
 ρ = density of fluid
 μ = viscosity of fluid
 V_m = mean axial velocity of fluid
 f_c = friction factor for a curved pipe

For laminar flow it was found that the experimental results agreed with Prandtl's empirical formula in reference (3-2) or

$$f_c / f_s = 0.289 [Re \sqrt{r_o / R}]^{0.36} \quad (3.3-2)$$

where f_s equals the friction factor for a straight pipe. It was also found that the curvature of the pipe effects the transition from laminar to turbulent flow by causing it to be delayed to higher Reynolds Numbers. H. Ito showed that the critical Reynolds Number is expressed by

$$Re \text{ (critical)} = 20,000 (r_o / R)^{0.32} \quad (3.3-3)$$

In order to use these formulae or similar formulae for a rotating pipe, one must determine an equivalent dimensionless parameter for r_o / R . To accomplish this task, we equate the Coriolis acceleration in a rotating pipe to the centrifugal acceleration in a curved pipe or

$$v^2 / R = 2v\omega$$

Thus,

$$R = \frac{v}{2\omega}$$

If we consider the fluid to be nonviscous for both cases and solid body rotation of the fluid in the curved pipe, we can say that

$$\frac{r_o}{R} = \frac{\omega r_o \omega}{V_m} \quad (3.3-4)$$

On the other hand, we could have equated the change of force with respect to velocity for each case since it is the distribution of these forces that causes the secondary flows. For the curved pipe

$$\frac{dF}{dv} = \frac{2\rho v}{R} - \frac{\rho v^2}{R^2} \frac{dR}{dv}$$

and for the rotating pipe we have

$$\frac{dF}{dv} = 2\rho\omega$$

Now letting dF/dv for a curve pipe equal dF/dv for a rotating pipe, we obtain

$$\frac{2\rho v}{R} - \frac{\rho v^2}{R} \frac{dR}{dv} = 2\rho\omega$$

For nonviscous flow and solid body rotation in the curved pipe

$$\frac{v}{R} = \text{constant and } \frac{dv}{dR} = \text{constant}$$

Thus, we can write the previous equation as

$$\frac{v}{R} = 2\omega$$

and solving for the radius of curvature of the curved pipe in terms of the variables of the rotating pipe also gives

$$R = \frac{V_m}{2\omega} \text{ and } \frac{r_o}{R} = \frac{2r_o\omega}{V_m}$$

We can now write H. Ito's equations as

$$f_r (V_m/2r_o\omega)^{1/2} = \frac{0.316}{[Re(2r_o\omega/V_m)^2]^{1/5}} \quad (3.3-5)$$

$$f_r/f_s = 0.289 [Re\sqrt{2r_o\omega/V_m}]^{0.36} \quad (3.3-6)$$

$$Re \text{ (critical)} = 20,000 (2r_o\omega/V_m)^{0.32} \quad (3.3-7)$$

A review of the publications on curved pipe flow shows that L. MacG. Trefethen, reference (3-1), Professor of Mechanical Engineering of the Tufts University compared data from curved pipes with data obtained with a rotating pipe. The data was compared on the basis that

$$\frac{r_o}{R} = \frac{2r_o\omega}{V_m}$$

The data for the analogous cases showed good agreement and it is reasonable, therefore, to assume that Equations 3.3-5, -6, and -7 can be used to estimate the friction factor for a radial passage in a rotor. Since these Equations have an empirical derivation, the values of the dimensionless parameters, Re and $V_m/2r_o\omega$, must be within the range of the values represented by the experimental data. For a high speed rotor, it is very

possible that the values of $2\omega r_o/V_m$ will be greater than the equivalent value of r_o/R for which experimental data exists. Experimental data obtained with radius ratios, r_o/R , of 0.061 and smaller were used by H. Ito for his correlations and the Reynolds Number for a radius ratio of 0.061 varied from 2400 to 81,000. This range of Reynolds Number is applicable for rotor cooling systems. However, the limitation on radius ratio and its equivalent value for a rotating pipe means that the speed of the rotor must be very low for the H. Ito's empirical equation to be useful. For example, the fluid dynamic test fixture described in Section 3.8 had a mean coolant velocity of 12.2 feet per second in a 0.107 inch diameter passage. Thus for

$$\frac{r_o}{R} = 0.061 = \frac{2r_o\omega}{V_m} = \frac{0.107 \times \omega}{12 \times 12.2}$$

the rotor speed, ω , must be less than 83.5 radius per second or 796 rpm. Since the test fixture was designed to operate at 24,000 rpm, the reliability of the data which is obtained with H. Ito's formula for the speed range of interest is open to question. The radius ratio required for an experiment using a curved pipe to simulate the flow conditions with the rotor at 24,000 rpm is

$$\frac{r_o}{R} = \frac{0.107 \times 800\pi}{12 \times 12.2} = 1.838$$

which is an impossible situation since the radius of curvature of the pipe axis would have to be less than the radius of the circular cross section of the pipe.

A search for information on the resistance to flow in a pipe rotating at high speeds was made. Unfortunately, data for conditions applicable to a rotor in a liquid cooled alternator was not found. Until high speed data is available, it is necessary to use the available curved pipe information even though the results will not be conclusive with respect to the feasibility of a rotor cooling system from the standpoint of flow resistance.

The friction factor for radial passage within the rotor for the fluid dynamic test fixture described in Section 3.8 are given in table 3.3-1. These values are based upon H. Ito's equations using stoddard solvent for the fluid and the maximum flow rate obtained during the experiments.

3.4 PRESSURE LOSSES WITHIN THE COOLANT PASSAGES IN THE FLUID DYNAMIC TEST FIXTURE WITH A STATIC ROTOR

A layout of the coolant passages for the fluid dynamic test fixture is given in figure 3.4-1. The pressure drop resulting from wall friction, right-angle bends, and accelerations

TABLE 3.3-1. FRICTION FACTOR FOR FLUID DYNAMIC TEST FIXTURE

$\frac{V_m}{2r_o \omega}$	ω rpm	f/f_s	f	Re	$Re_{critical}$
∞	0	1.00	.033	7770	2000
100	130	1.00	.0332		4575
16.4	797	4.38	.140		8160
12.0	1089	4.63	.147		9040
8.0	1630	4.98	.159		10290
4.0	3260	5.65	.181		12830
2.0	6520	6.40	.204		16020
1.0	13040	7.25	.232		20000
0.75	17400	7.63	.244		21900
0.544	24000	8.09	.258	7770	24300
Flow Rate = 2.73 gpm through eight parallel passages Main Flow Velocity = 12.18 ft/sec Fluid Density = 49.2 lbm/ft³ Viscosity of Fluid = 2.13 x 10⁻⁵ lbf sec/ft² Diameter of Passage = 0.00891 ft.					

FIGURE 3.4-1. Layout of Coolant Passages in Fluid Dynamic Test Fixture

of the fluid was calculated for a static rotor using the properties of stoddard solvent and Blasius formula for the friction factor. A flow rate equal to 2.73 gpm was used since this was the maximum flow rate experienced during the dynamic test described in Section 3.8. The pressure drop for each section of the flow path due to friction are given in table 3.4-1 and table 3.4-2 gives a summary of the pressure drop due to the acceleration around sharp corners. The loss occurring within a right-angle bend was assumed to be 1.25 times the dynamic head around the corner. The factor of 1.25 is typical for a square elbow or the intersection of two-cylinders as noted in reference (3-3). The total calculated pressure drop for a static rotor was 16.72 psi for these conditions.

In general most of the pressure loss is due to sharp turns and acceleration of the fluid, the pressure drop due to wall friction was only 19.7 percent of the total calculated loss.

A comparison on the calculated loss with the measured value is given by figure 3.4-2. It can be noted that the calculated pressure loss was within 10 percent of the actual loss.

3.5 PUMPING HEAD DEVELOPED WITHIN THE ROTOR DUE TO ROTATION

A distinct advantage of the rotor cooling concept under investigation is that it contains the features of a centrifugal pump. The flow path through the rotor, shown in figure 3.4-1, has the coolant entering the rotor along the axis of rotation and leaving the rotor at a point some distance from the axis of rotation. The centrifugal forces acting on the fluid between the axis of rotation and the exit port must be balanced by hydraulic force at the exit. Making a force balance on the fluid in a constant area passage gives

$$P_1 = P_2 + \frac{\rho R_2^2 \omega^2}{2gc} (1 - R_1^2/R_2^2) \quad (3.5-1)$$

where

- P_2 = pressure at point (2)
- P_1 = pressure at point (1)
- R_2 = radius at point (2)
- R_1 = radius at point (1)
- ρ = density of fluid
- ω = angular speed of rotation

For our case the flow enters the rotor on the axis of rotation or $R_1 = 0$ and for the flow path in the fluid dynamic test fixture R_2 was 1.035 inches. Figure 3.5-1 gives the head developed within the rotor for these conditions. It can be noted that a head of 248 lb_f/in^2 is developed at 24,000 rpm even when no heating of the coolant occurs. A significant increase

TABLE 3.4-1. Pressure Losses in Passages of Fluid Dynamic Test Fixture

Section	Dimensions (inches)	Area, in ²	Q, in ³ /sec	V in/sec	P _d psi	D _h in	R _e	f	ΔP _f , psi
Central Inlet Port	0.250 diam x 0.500	0.0491	10.91	214.0	1.680	0.250	26600	0.0248	0.0834
Hole- through Shaft Extension	0.156 diam x 5 '16	0.01912	2.626	137.2	0.696	0.156	10700	0.0310	0.434
Semicircular Hole in Manifold	0.141 diam x 1.825	0.00781	1.313	184.0	1.043	0.0866	7260	0.0342	0.7525
Rectangular Passage in Manifold	0.141 x 0.055 x 0.612	0.00776	1.313	169.0	1.054	0.0792	6670	0.0337	0.2750
Rotor Passage, Inlet	0.107 diam x 0.417	0.00899	1.313	146.0	0.786	0.107	7770	0.0336	0.1029
Rotor Passage, Outward	0.107 diam x 2.125	0.00899	1.313	146.0	0.786	0.107	7770	0.0336	0.525
Rotor Passage, Pole Tip	0.107 diam x 1.125	0.00899	1.313	146.0	0.786	0.107	7770	0.0336	0.278
Rotor Passage, Inward	0.107 diam x 1.125	0.00899	1.313	146.0	0.786	0.107	7770	0.0336	0.525
Rotor Passage, Outlet	0.107 diam x 0.417	0.00899	1.313	146.0	0.786	0.107	7770	0.0336	0.1029
Rectangular Passage in Manifold	0.141 x 0.055 x 0.612	0.00776	1.313	169.0	1.054	0.0792	6670	0.0336	0.2750
Semicircular Hole in Manifold	0.141 diam x 0.750	0.00781	1.313	184.0	1.043	0.0866	7260	0.0342	0.310
Outlet Port	0.141 diam x 0.203	0.01562	1.313	84.0	0.261	0.141	5900	0.0360	0.0135
TOTAL									3.2876
$P_d = 1/2 \rho V^2 = \text{dynamic head}$ $D_h = \frac{4 \times \text{Area}}{\text{Perimeter}} = \text{hydraulic diameter}$ $R_e = \frac{\rho V D_h}{\mu} = \text{Reynolds number}$ $Q = \text{volumetric flow}$ $V = \text{velocity}$ $\Delta P_x = \text{pressure loss due to friction}$									

TABLE 3.4-2. Acceleration and Turning Losses in Rotor Passage
for 2.73 gpm Flow

Location	Dynamic Head, psi	Acceleration and Turning Loss, psi
1. Inlet port	1.680	2.100
2. Holes through shaft	0.696	0.871
3. Entrance to semicircular holes	1.043	1.303
4. Entrance to rectangular passages	1.054	1.319
5. Entrance to rotor passages	0.786	0.984
6. Sharp corner in rotor passage	0.786	0.984
7. Sharp corner in rotor passage	0.786	0.984
8. Sharp corner in rotor passage	0.786	0.984
9. Sharp corner in rotor passage	0.786	0.984
10. Entrance to rectangular passage	1.054	1.319
11. Entrance to semicircular holes	1.043	1.303
12. Exit ports	0.261	<u>0.326</u>
TOTAL		13.43 psi

FIGURE 3.4-2. Pressure Drop Through Rotor for Fluid Dynamic Test Fixture

FIGURE 3.5-1. Comparison of Head Developed and the Pressure Losses Within The Rotor

of head results when a change in density of the coolant occurs within the passage. The coefficient of volume expansion of the fluid

$$\beta = -1/\rho (\partial\rho/\partial T)_p \quad (3.5-2)$$

and the temperature distribution within the passage determine the net head which will be produced by a density gradient. The example presented here is based upon the following specific assumptions with regard to the flow path given in figure 3.4-1.

- (a) the difference of temperature between points A and B equals 50°F,
- (b) the temperature increases linearly with the distance along the path from A and B, and
- (c) the coolant is stoddard solvent which has a coefficient of volume expansion of $31.7 \times 10^{-5} \text{ F}^{-1}$ and an average density of $49.2 \text{ lb}_m/\text{ft}^3$.

With these assumptions, the temperature rise between points on the coolant path becomes

$$\begin{aligned} \Delta T \text{ from A to A'} &= 50 \times 2.125/5.375 = 19.78^\circ\text{F} \\ \Delta T \text{ from A' to B'} &= 50 \times 1.125/5.375 = 10.45^\circ\text{F} \\ \Delta T \text{ from B' to B} &= 50 \times 2.125/5.375 = 19.77^\circ\text{F} \\ \Delta T \text{ from A to B} &= \underline{50.0^\circ\text{F}} \end{aligned}$$

and the average temperature between A and A' is equal to $T_1 + 9.89^\circ\text{F}$. Similarly, the average temperature between B' and B equals $T_1 + 40.11^\circ\text{F}$ which gives a difference between the average temperature of the coolant in the left and right side of 30.2°F . The pressure difference due to the change in the average density in corresponding sides of the flow path is given by

$$P_A - P_B = (\rho/2gc) R_A^2 \omega^2 (1 - R_A'^2/R_A^2) \beta \Delta T \quad (3.5-3)$$

where

- $P_A - P_B$ = pressure difference between point A and B,
- R_A' = radius at point A',
- R_A = radius at point A,
- ΔT = difference between the average temperature in the left and the right side.

The additional head resulting from a 50°F temperature rise is given by figure 3.5-1.

This example shows that a small change of density within and between the outward and inward passages can result in a significant pressure head which will help circulate the fluid

through the rotor. For the example shown here with stoddard solvent, a head of 42.6 psi is produced at 24,000 rpm with a 50°F temperature rise. The centrifugal head developed within the inlet passage was 248 psi at 24,000 rpm. When these heads are compared with a required head of 15.6 psi for 2.78 gpm flow in a static rotor, it becomes reasonable to conclude that the additional loss of head due to secondary flows within the passages will not be a critical problem. The calculations indicate that the required head at 24,000 rpm with 2.73 gpm flow will only be 28.5 psi. This brings up the question, "what is the nature of the flow when the pumping capacity of the rotor exceeds the throttling characteristics of the passages with a controlled supply of coolant?" For most cases the back pressure on the rotor will be considerably less than the head developed within the rotor when all of the coolant passages are filled with liquid. Figure 3.5-2 illustrates how the system will operate under these conditions. When the back pressure is too low, the radial passage on the inlet side of the rotor will operate with the flow of droplets in part of the passage. When drop flow occurs, the centrifugal force on the liquid drop does not have to be balanced by an increase of static pressure as the liquid moves outward. The centrifugal force in this case causes a radial acceleration of the drop which is constrained to rotate at a constant angular speed. The motion of the droplets can be described within a stationary coordinate system with coordinates of θ and r . Since there is no applied force in the "r" direction, the equation of motion in the radial direction is

$$m(\ddot{r} - r\theta^2) = 0 \quad (3.5-4)$$

where

$\theta = d\theta/dt = \omega = \text{angular speed} = \text{constant}$

$\dot{r} = dr/dt = v = \text{radial velocity of droplet}$

$\ddot{r} = d^2r/dt^2 = dv/dt = \text{radial acceleration of droplet}$

The general solution of Equation 3.5-4 is

$$r = c_1 e^{\omega t} + c_2 e^{-\omega t} \quad (3.5-5)$$

$$v = c_1 \omega e^{\omega t} - c_2 \omega e^{-\omega t} \quad (3.5-6)$$

If we let $t = 0$ at a position where $r = r_0$ and $v = v_0$, we have

$$r/r_0 = v_0/\omega r_0 \sinh \omega t + \cosh \omega t \quad (3.5-7)$$

$$v/v_0 = \cosh \omega t + r_0 \omega / v_0 \sinh \omega t \quad (3.5-8)$$

FIGURE 3.5-2. Droplet Flow in Radial Passages on Inlet Side of Rotor

These equations give the radius and radial velocity of a droplet starting with the point where the liquid no longer completely fills the passage. The example given here describes the motion of a droplet which forms at the entrance of the 0.156-inch diameter holes through the shaft extension in the fluid dynamic test fixture. For this case we shall consider

- (a) rotational speed = 24,000 rpm = 800 radians/sec,
- (b) 2.73 gpm flow through rotor,
- (c) distance from axis of rotation at the entrance of the 0.156 diameter holes equals 0.485 inches or $r_0 = 0.485$ inches,
- (d) distance from axis of rotation at the exit of the holes equals 0.787 inches, and
- (e) the holes are completely filled with liquid at their entrances or $v_0 = 11.43$ fps.

Under these conditions, equations 3.5-7 and 3.5-8 become

$$v/v_0 = \cosh \omega t + 8.89 \sinh \omega t$$

$$r/r_0 = 0.1125 \sinh \omega t + \cosh \omega t$$

The velocity of the droplets from the entrance to the exit of the holes is given by figure 3.5-3. It can be noted that the centrifugal force on the droplets increases their radial velocity from 11.4 to 133 fps with just 5/16 inches of travel within the passage. Very little of this velocity head would be recovered within the passages since viscous and turbulent action will consume most of the kinetic energy associated with the droplets. The loss of static pressure head which results from the flow of the droplets is given by

$$\Delta P = (\rho/2gc) R_2^2 \omega^2 (1 - R_1^2/R_2^2) \quad (3.5-9)$$

where R_1 = radius at point where drop flow starts

R_2 = radius at point where drop flow stops

For our case,

$$\begin{aligned} P &= \frac{49.2}{2 \times 32.2} \left(\frac{0.787 \times 800\pi}{12} \right)^2 \left(1 - \left(\frac{0.485}{0.787} \right)^2 \right) \\ &= 12950 \text{ lb}_f/\text{ft}^2 = 90 \text{ psi} \end{aligned}$$

FIGURE 3.5-3. Velocity of Droplet Flow

This example clearly demonstrates the manner by which a balance between the centrifugal head produced within the rotor and the back pressure on the rotor is maintained when the flow is controlled by external equipment. In most instances the back pressure on the rotor will be less than that required for a completely filled rotor passages. When this occurs, droplets or free surfaces will be produced in the passages on the inlet side of the rotor near the axis of rotation. With free surfaces and drop flow, the centrifugal force acting upon the liquid will produce a radial acceleration of the liquid and no increase of static pressure will be generated in this part of the passage.

3.6 CENTRIFUGATION OF HEAVY MATERIALS IN THE ROTOR COOLANT PASSAGES

A critical complication of a cooling scheme which has the coolant diverted outward and inward is due to the high centripetal acceleration present in this type of rotating system. In effect, the coolant passages form a centrifuge; a machine which uses centrifugal force for separating materials of different densities. The density distribution in the coolant will be non-homogeneous since solid materials within the system will be picked-up and transported by the coolant. Mechanical and other types of filtration will only limit the size of the particles which can be transported by the liquid. Unfortunately, the density of the solid materials may be much greater than the density of potassium. Therefore, the designer of the cooling system must cope with problem of preventing an accumulation of heavy material in the passages which could close and suppress the flow of the coolant.

Stoke's Law for the so-called "creep motion" of a particle can be used to determine if a particle will separate from the coolant. For this case the Reynold's number, based upon the dimensions of the particle and the relative motion of the fluid and particle, is very small which means the inertia forces associated with the fluid are very small compared with the viscous forces. The general differential equation of Navier-Stokes, in vector notation,

$$\rho \frac{DW}{dt} = F - \text{grad } p + \mu \nabla^2 w$$

can be integrated by neglecting the inertia forces.

Thus,

$$\mu \nabla^2 w = \text{grad } p$$

This equation in combination with the equation of continuity

$$\text{div } w = 0$$

was first solved by Stokes for the case of a sphere to give the drag on the particle as

$$D = 6\pi\mu rw$$

where

D = drag force
 r = radius of particle
 w = stream velocity relative to the particle
 μ = viscosity of the fluid

For our case the drag force must equal the net force acting on the particle. The centrifugal force on a sphere is

$$F_C = 4/3\pi r^3 \rho_p R \omega^2$$

where

F_C = centrifugal force
 ρ_p = density of particle
 R = radius from axis of rotation
 ω = angular velocity of rotor

In addition we have a net hydraulic force acting on the sphere. The net hydraulic force for this case is equal to the centrifugal force which would act upon the sphere if it had a density equal to the density of the fluid or

$$F_h = 4/3\pi r^3 \rho_f R \omega^2$$

where

F_h = hydraulic force
 ρ_f = density of fluid

We can now make a force balance on the sphere for the case where the acceleration of the sphere relative to rotor is zero.

$$6\pi\mu rw = 4/3\pi r^3 R \omega^2 (\rho_p - \rho_f)$$

The terminal velocity of the sphere becomes

$$w = \frac{2}{9} \frac{r^2 R \omega^2}{\mu} (\rho_p - \rho_f)$$

In order for a particle to exit from a position where the centripetal acceleration, $R\omega^2$, is a maximum, it must have a terminal velocity, w , which is less than the velocity of the coolant as illustrated in figure 3.6-1.

FIGURE 3.6-1. Maximum Allowable Terminal Velocity of Particle
in Coolant Passages

If we let the terminal velocity, w , equal the velocity of the fluid, v , we can solve for the maximum allowable particle size with a known density and centripetal acceleration for the coolant passage since

$$r = \left(4.5V\mu/R\omega^2(\rho_p - \rho_f) \right)^{1/2}$$

Table 3.6-1 gives typical diameter and speed for rotors in alternators of different ratings. This table also gives the maximum centripetal acceleration within a coolant passage which passes one-half of an inch below the surface of the pole face.

TABLE 3.6-1 Typical Diameter and Speed for a Rotor in Alternators of Different Ratings

Rating KVA	Speed RPM	Rotor Diameter Inches	Maximum Radius of Coolant Passage Inches	Maximum Acceleration ft/sec ²
400	24 000	11.572	5.286	2.78×10^6
1 200	19 200	16.818	7.909	2.66×10^6
6 000	9 000	34.548	16.774	1.24×10^6
12 000	7 200	47.241	23.170	1.10×10^6

The relative densities of the coolant, carbon, and some of the light compounds of potassium which may be present as particles are in table 3.6-2.

TABLE 3.6-2 Relative Densities of Potassium and Some of the Light Particles which May be Present in the Potassium

Material	Specific Gravity
Potassium	0.765
Potassium sulfide	1.8
Carbon	2.0
Potassium niokate	2.0
Potassium zirconate	2.0
Potassium monoxide	2.3
Potassium carbonate	2.4
Potassium silicate	2.5

A typical velocity for the potassium in the passages would be approximately 25 feet per second and with 600°F potassium the viscosity would be 0.545 lb_m/ft-hr. The maximum allowable particle size for several of the light compounds are given in table 3.6-3 for conditions stated previously.

TABLE 3.6-3. Maximum Allowable Diameter for Particles (inches)

Material	400 KVA	1200 KVA	6000 KVA
Potassium Sulfide	0.234×10^{-3}	0.239×10^{-3}	0.350×10^{-3}
Carbon	0.212×10^{-3}	0.216×10^{-3}	0.317×10^{-3}
Potassium monoxide	0.192×10^{-3}	0.196×10^{-3}	0.287×10^{-3}
Potassium carbonate	0.186×10^{-3}	0.190×10^{-3}	0.278×10^{-3}
Potassium silicate	0.180×10^{-3}	0.184×10^{-3}	0.270×10^{-3}

If one compares the allowable diameter for these particles with the size of a 10 micron (0.394×10^{-3} inches) mechanical filter, it becomes apparent that it is not feasible to prevent the separation of particles by limiting their size. This conclusion is supported by the fact that complex carbides of the metals and other compounds of the materials will be present which have a specific gravity greater than 2.5. Therefore, the only practical solution to the dilemma is to design the coolant passages so that the accumulation of the heavy material will not close or suppress the flow of the coolant. This can be accomplished by installation of an extra passage in the body of the pole where the centripetal acceleration is a maximum as shown in figure 3.6-2. In order to indicate how much volume is required for accumulation of particles, the following example is given.

Specific Assumptions

- (a) The system is charged with 50 lb_m of potassium which has 10 parts per million of impurities which have an average specific gravity of 2.0,
- (b) The surface area of the materials in contact with the potassium in the rotor coolant system is equivalent to 500 feet of one-half-inch-tubing,
- (c) 0.0001 inches of material is removed from the surfaces which are in contact with the potassium,

FIGURE 3.6-2. Flow Path for Coolant Within the Main Body of Poles

(d) Total surface area equal

$$500 \text{ ft} \times 12 \text{ in/ft} \times 1/2 \pi \text{ in} = 9,420 \text{ in}^2$$

(e) Total volume removed from surfaces

$$9,420 \text{ in}^2 \times 0.0001 \text{ in} = 0.942 \text{ in}^3$$

(f) Total impurities in initial charge

$$50 \text{ lb}_m \times \frac{10 \text{ parts}}{10^6 \text{ parts}} = 0.0005 \text{ lb}_m$$

$$0.0005 \text{ lb}_m \times \frac{1}{2 \times 0.0361 \text{ lb}_m/\text{in}^3} = 0.00695 \text{ in}^3$$

(g) Total volume required for accumulation of heavy materials

$$0.942 \text{ in}^3 + 0.00695 \text{ in}^3 = 0.949 \text{ in}^3$$

(h) For a 10 pole rotor with 1/4 diameter passages, the length of a passage in each pole must be equal to

$$\frac{0.949 \text{ in}^3}{10 \times \frac{\pi}{64} \text{ in}^2} = 2.03 \text{ in}$$

3.7 TYPICAL PRESSURES DEVELOPED BELOW THE SURFACE OF THE POLE FACES

The large centripetal acceleration which acts upon the coolant within rotor passages produces large pressures near the surface of the pole. The maximum acceleration for typical coolant passage are given in table 3.6-1. If one considers the coolant passage to be completely filled with potassium, the pressure at the point of maximum acceleration is given by

$$P = \frac{1}{2} \frac{\rho R_m}{g_c} A_m$$

where

R_m = maximum radius

A_m = acceleration at R_m

ρ = density of liquid coolant

The pressures developed near the pole face for the rotors given in table 3.6-1 are given in table 3.7-1.

TABLE 3.7-1. Maximum Pressure Within Rotor Coolant Passages
for Typical Alternators

Rating, KVA	Maximum Radius of Coolant Passages, Inches	Maximum Acceleration Ft/sec ²	Maximum Pressure Developed psi
400	5.286	2.78×10^6	6,280
1200	7.909	2.66×10^6	9,010
6000	16.774	1.24×10^6	8,910
12000	23.170	1.10×10^6	10,900

In general, the magnitude of the pressure requires a welded construction for the coolant passage. This means that the rotor must be a weldable alloy.

3.8 EXPERIMENTAL VERIFICATION OF COOLING CONCEPT

3.8.1 Description of Experimental Equipment

A test fixture which was used to verify the fluid dynamic performance of the rotor and related components for a 467-kVA generator, reference (3-4), was modified to accommodate the rotor cooling scheme under investigation. A cross-sectional view of the new fixture is shown in figure 3.8.1-1. The rotor for this fixture was supported on 2.0 inch diameter, pivoted-pad bearings and a duplex arrangement of anti-friction bearings was used to support the axial thrust on the rotor. The fixture contained four liquid seals and two vapor seals. A labyrinth seal was used in the drive shaft housing on a 0.864 diameter to prevent excessive leakage of the lubricant from the drive-end bearing. Viscoseals with helical threads on the housing and shaft were used between the bearings and rotor cavity to prevent the flow of liquid into the rotor cavity. In addition, vapor seals were used between the viscoseals and the rotor cavity to prevent vapor and gas from freely entering the cavity. The vapor seals were screw-type, molecular vacuum pumps which consisted of a smooth shaft running in a helical-threaded housing. These seals were cantilevered over the viscoseals to minimize the required length for the shaft extensions. This feature of the seal construction formed a slinger for the liquid that passed through the viscoseals. Volutes at the ends of the

slingers were utilized to collect the liquid when it left the lip of the slingers. A fourth viscoseal with helical threads on the housing and shaft was used on the anti-drive end of the rotor to maintain a liquid-vapor interface between the bearing and the scavenge system. This seal operated against the inlet pressure for the bearing while maintaining the vapor pressure of the rotor coolant on the discharge side of the seal.

The scavenge scheme for the rotor cavity and seal drains consisted of a dynamic slinger in the anti-drive end of the rotor. This slinger contained four equally-spaced, cylindrical wells. Each well intersected the axis of rotation at an angle of 30 degrees and was vented to the inlet side of the bearing. The centrifugal force acting on the liquid within the wells was sufficient to overcome the bearing pressure even with the vapor pressure of the rotor coolant within the central portion of the slinger. The low pressure reservoir within the center of the slinger provided a sink for the liquid in the rotor cavity at start-up and the liquid which passed through the viscoseals during start-up before an interface was established within the seals. It should be noted that the slingers on the vapor seals and dynamic action of the rotor within the rotor cavity provided sufficient velocity head to circulate the liquid from the various drain ports to the scavenge ports in the anti-drive end of the machine.

The dimensions of the flow path for the rotor coolant are given in figure 3.4-1. The rotor coolant entered the fixture on and parallel to the axis of rotation through a 0.125-inch diameter orifice in the supply port on the anti-drive end of the fixture. The coolant then passed through a central hole, 0.25 inch in diameter, in the end of the rotor whereupon it was diverted by four radial vanes through the shaft extension and into a manifold. This manifold served as a distribution duct for the passages in the main body of the rotor. It also formed the rotating portion of the liquid seal and provided a support for the rotating portion of the vapor seal. Radial grooves within the manifold provided the Coriolis reaction necessary to accelerate the coolant up to the tangential velocity of the circular holes in the body of the rotor. The rotor had eight equally spaced passage, 0.107 inch in diameter, which diverted the coolant radially outward from a diameter of 2.75 inches to a diameter of 7.0 inches and back again. A manifold, similar to the manifold on the inlet side of the rotor, was used to divert the coolant into the drive-end bearing discharge chamber. The discharge chamber formed a volute type of diffuser around the exit ports which converted part of the tangential velocity head associated with the coolant into static pressure.

The external flow circuits and instruments which were used for the bearings and rotor are shown in figure 3.8.1-2. Flow through each circuit was controlled by adjusting the throttle valves and the pump bypass valve. A constant displacement pump was used and the absolute pressures in the

FIGURE 3.8.1-1 FLUID DYNAMIC TEST FIXTURE

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FIGURE 3.8.1-2. Schematic Drawing of Test Loop and Instrumentation

fixture were controlled by adjusting the back pressures on the discharge side of the fixture.

A three-phase motor that was operated as a four-pole and eight-pole motor was used to drive the fixture. The speed of the motor for each speed range was controlled by varying the input frequency. All tests were performed using stoddard solvent for the coolant and lubricant.

3.8.2 Description of Tests

The tests were initiated on 2.13.67 and 4.3.67. On each of these occasions the test was terminated after the fixture stopped with a locked rotor. Inspection of the parts revealed a failure of a pivoted-pad bearing. High wear of both the journal and pads occurred in the circumferential direction. It was concluded that the failures were due to a self-loading action which was most likely caused by debris produced by scoring and tracking before the rotor reached sufficient speed to establish the proper film within the clearance. The bearings were altered by:

- (a) replacement of the stainless steel, type 416, journals with journals made of tungsten carbide,
- (b) replacement of the stainless steel pads with pads made of a tool steel which was heat treated to a hardness of RC 63, and
- (c) increasing the diametral clearance from 0.0030 ± 0.0002 to 0.0040 ± 0.0002 inches.

These changes were made to prevent scoring and tracking of the bearing surfaces. The modifications were complete on 4.14.67 and the tests were continued. Rotor flow calibrations were performed at 16,000, 20,000, and 24,000 rpm without incident or indication of any bearing or flow problems. Each calibration was performed by maintaining a constant input frequency for the drive motor and by varying the flow rate through the rotor. For each flow rate, the pressure on the discharge side of the rotor was varied to determine if this pressure effected the rotor flow rate.

3.8.3 Test Results

The test results for several rotor flow calibrations are summarized in table 3.8.3-1. The data is also graphically illustrated in figures 3.8.3-1, -2, -3, and -4. It was observed that the independent variable of the rotor flow calibrations was the pressure difference that acted upon the 0.125 diameter orifices in the rotor supply line. The flow through this restriction followed the standard formula for flow measurement of an

TABLE 3.8.3-1. Summary of Test Results

	Rotor Speed RPM	Rotor Flow GPM	Rotor Inlet Press psig	Orifice Discharge Press psig	Δp psi	$\sqrt{\Delta p}$ $1/2$ (psi)	No. 2 Bearing Discharge Press, psig	Rotor Discharge Press, psig
1	13483	1.04	1.0	-11.2	12.2	3.49	4.0	2.2
2	13462	2.24	29.0	-13.5	42.5	6.52	7.5	5.0
3	15,000 to 16,000	0.91	- 3.0	-12.9	9.9	3.15	10.0	8.2
4		0.894	- 3.9	-12.4	8.5	2.92	10.0	7.0
5		0.894	- 3.9	-12.9	9.0	3.00	14.0	10.0
6		0.877	- 2.9	-13.4	10.5	3.24	16.5	12.0
7		0.865	- 3.9	-12.9	9.0	3.00	10.0	7.5
8		1.95	21.0	-12.7	33.7	5.80	10.0	8.1
9		1.94	21.0	-11.9	32.9	5.74	10.0	7.0
10	15,000 to 16,000	1.93	21.0	-12.3	33.3	5.77	10.0	10.0
11		1.95	21.0	-12.7	33.7	5.81	15.0	12.0
12		1.91	21.0	-12.4	33.4	5.78	14.0	11.5
13		1.93	20.0	-12.7	32.7	5.72	10.0	8.1
14		2.62	47.5	-13.0	60.5	7.78	9.0	8.2
15		2.62	47.0	-12.6	59.6	7.72	7.5	7.0
16		2.63	48.0	-13.0	61.0	7.81	12.0	10.0
17	19,000 to 20,000	2.64	48.0	-13.0	61.0	7.81	13.5	12.0
18		2.61	48.0	-12.4	61.8	7.86	14.0	15.5
19		2.62	48.0	-13.0	61.0	7.81	9.0	8.2
20		0.95	- 2.5	-13.0	10.5	3.24	20.0	15.0
21		0.96	- 2.5	-13.0	10.5	3.24	27.0	21.0
22		0.93	- 2.0	-13.7	11.7	3.42	28.0	23.0
23		0.95	- 3.0	-13.4	10.4	3.22	25.0	19.5
24	19,000 to 20,000	1.95	23.0	-12.9	35.9	6.00	16.5	13.0
25		1.95	23.0	-13.0	36.0	6.00	21.0	17.0
26		1.95	23.0	-12.9	35.9	6.00	25.0	21.0
27		1.92	24.0	-13.5	37.5	6.12	29.0	25.0
28		1.95	23.0	-12.9	36.9	6.08	23.0	19.5
29		2.53	46.0	-12.8	58.8	7.67	16.0	13.0
30		2.54	46.0	-12.9	58.9	7.68	18.0	15.0
31	19,000 to 20,000	2.55	46.5	-12.9	59.4	7.70	21.0	17.1
32		2.55	46.0	-12.8	58.8	7.67	23.0	19.0
33		2.55	47.0	-12.5	59.5	7.71	26.0	21.5
34		2.56	47.0	-12.4	59.4	7.71	30.0	25.0
35		2.79	57.0	-13.2	70.2	8.38	17.5	15.0
36		2.83	58.0	-13.1	71.1	8.44	27.5	20.0
37	20011	1.82	16.5	-11.9	28.4	5.33	12.0	9.0
38	23955	0.71	0.5	- 4.7	5.2	2.28	16.5	11.0
39	23976	1.56	16.0	- 7.7	23.7	4.87	24.0	18.0
40	23385	2.21	35.0	-10.9	45.9	6.78	30.0	27.0
41	23254	2.64	55.0	-11.8	66.8	8.18	26.6	21.0

FIGURE 3.8.3-1. Rotor Flow Calibration at Speeds up to 16,000 rpm

FIGURE 3.8.3-2. Rotor Flow Calibration at Speeds Between 19,000 and 20,011 rpm

FIGURE 3.8.3-3. Rotor Flow Calibration at Speeds between 23,000 and 24,000 rpm

FIGURE 3.8.3-4. Rotor Flow Calibrations

incompressible fluid or

$$Q = A_t K_1 \sqrt{\frac{2gc}{\rho} \Delta P} \quad (3.8.3-1)$$

which for our case can be written as

$$Q = K_2 \sqrt{\Delta P} \quad (3.8.3-2)$$

where

- Q = volumetric flow rate through the rotor.
- ΔP = pressure difference which acts upon flow restriction in the stationary rotor supply line.
- K_2 = constant

The volumetric flow rate through the rotor as a function of the square root of the pressure difference for several calibrations is given by figures 3.8.3-1, -2, -3, and -4. These correlations show that the flow through the rotor was a linear function of the square root of the pressure difference as indicated by Equation 3.8.3-2. The important conclusion that is demonstrated by these data points is that the flow rate through the rotor was independent of the rotor speed as illustrated by figure 3.8.3-4. The only requirement for the rotor fluid supply system was to deliver the desired flow at a sufficient pressure to overcome the resistance of the stationary supply line in the anti-drive end of the fixture. Many of the data points were obtained with a controlled rotor flow rate and with a varying back pressure on the rotor (typical of data points 29 through 35). It was found that the flow rate was not affected by the back pressure on the rotor. This fact shows that the head developed with the rotor was greater than the combined pressure losses and the back pressure on the rotor. In fact, the rotor had to be operating with droplet on the inlet side of the rotor in order for the flow rate to be unaffected by the back pressure.

In general the test results showed that flow through the rotor was stable and that the additional flow resistance due secondary flow was not a problem for this rotor cooling concept.

3.9 SUMMARY AND CONCLUSIONS

A rotor cooling scheme which has the coolant diverted radially outward and inward from the axis of rotation was investigated. This scheme offers distinct advantages for cooling the pole of a solid rotor since the heat conduction paths between the pole faces and the coolant passages are short. The complications associated with this cooling scheme are due to the high centripetal and Coriolis accelerations which act upon the fluid in the passages.

In effect, the coolant passage forms a centrifuge which is capable of separation of the solid materials that are transported by the coolant. The only practical solution to this dilemma is to design the coolant passage so that the accumulation of the heavy material will not close or suppress the flow of the coolant. This can be accomplished by using extra passages in the body of the pole to serve as an accumulator for the heavy material.

The large centripetal acceleration which acts upon the coolant within the rotor also produces large pressures within the coolant passages. Examples given in this report indicate that a 12,000 kVA alternator would have 10,900 psi pressure within the passages at design speed. In general, the magnitudes of the pressures will require a welded construction of the coolant passages. Thus, the rotor material must be a weldable alloy.

Coriolis forces act upon the coolant since the coolant flows outward and inward from the axis of rotation. The effect of these forces is to cause secondary flows in the passages. These secondary flows cause an increase in the flow resistance when compared with the resistance of a non-rotating system. The secondary flow phenomenon in a rotating pipe was investigated using inviscid-fluid theory. It was found that the secondary flow in a rotating pipe is analogous to the case of secondary flows in a stationary curved pipe. Empirical formulae for the frictional resistance of curve pipes was used to determine the resistance of a rotating pipe by forming equivalent dimensionless parameters for each case. However, the range of the dimensionless parameters for a typical rotor is far above the range previously investigated in curved pipes. It was, therefore, necessary to extrapolate the available curved pipe data. Thus, the results cannot be considered to be conclusive with respect to the feasibility of the rotor cooling under investigation from the standpoint of flow resistance. The extrapolated data indicate that the friction factor for the radial passages are eight times greater than the friction factor for a non-rotating passage.

The calculated pressure losses in a rotor for the test fixture was compared with measured losses in the rotor under static conditions. It was found that the pressure drop due to wall friction was only 19.7 percent of the total pressure loss. The head developed within the rotor due to the location of the outlet and due to a change in density of the coolant was calculated and compared with the pressure loss data. The comparison showed that the pumping capacity was much greater than the throttling capacity and that pressure losses due to secondary flow would not present a problem. This conclusion was experimentally verified in a test fixture. The only requirement of the fluid supply system for these experiments was to deliver the desired flow at a sufficient pressure to overcome the resistance of the stationary supply line in the fixture. It was demonstrated by a series

of tests that the flow rate through the rotor was independent of rotor speed and was unaffected by the back pressure on the rotor at rotor speeds and pressures of practical interest.

4. INVESTIGATION OF ALTERNATOR NO-LOAD LOSSES

by N. V. Wheat

4.1 INTRODUCTION

The 467 kVA experimental generator departs from normal high power, high speed generator designs due to the high generated frequency used (3200 Hz) and the particular inductor type generator configuration. Theoretical calculation of the separate no-load losses in a generator of this type can be difficult to achieve with accuracy since there is very little experimental data in existence to substantiate the many assumptions which have to be made.

The separate losses expected to be encountered in this test mockup of the experimental generator are:

1. Stator iron losses
2. Pole-face loss
3. Windage loss
4. Friction loss
5. Copper losses (due to the field winding only, as the armature was not wound.)

4.1.1 Stator Iron Losses

Stator iron losses are separated into tooth loss and depth-behind-slots loss (referred to as D.B.S. loss) because different flux densities are encountered in these two locations.

Experience of testing 400 Hz generators has shown that the stator iron losses normally exceed results obtained by the Epstein square iron loss tests by a factor of approximately three times. This difference can be accounted for by: imperfect interlaminar insulation; flux harmonics (Epstein tests use pure sine wave excitation); welding of the stack of laminations; high interlaminar pressure. Thus, this factor of three was used to multiply the calculated values of the stator iron losses to obtain a seemingly more practical result.

4.1.2 Pole-Face Loss

Pole-face loss was expected to be high in this generator; hence, slotted pole-faces (figure 4.2-12) were used to increase the effective surface resistivity. The permeability of the pole-face material has a direct influence on the losses and the calculations involve the use of the average effective pole-face permeability. This value is difficult to obtain with accuracy (reference 4-1) and was obtained for these calculations by substituting values from previous test results. The effect of temperature on this permeability is uncertain and consequently had to be ignored.

4.1.3 Windage Loss

Windage loss is influenced by the rotor and cavity configurations together with the density and viscosity of the surrounding atmosphere and the rotor speed. Calculations were based on previous test results (reference 4-2) and variations from ideal conditions had to be ignored.

4.1.4 Friction Loss

Friction in the generator bearings varies with bearing preload, lubricant flow, temperature, speed and running time and therefore could not be calculated with any accuracy.

4.1.5 Copper Losses

Copper losses in the field winding are proportional to field current and field winding resistance and could only be calculated accurately for known field winding temperatures. During the tests these losses were calculated directly from field current and voltage thus taking into account winding resistance variation with temperature.

4.1.6 Test Requirements

The tests were performed in such a manner that the losses could be separated as far as possible by recording all input power and heat output, since all losses appear as heat (together with noise and vibration), and a loss balance could then be calculated.

4.2 THEORETICAL EVALUATION OF LOSSES

4.2.1 Pole-Face Loss - P (reference 4-1)

$$P = \frac{0.126 (B_y)^2 (\lambda)^2 (f)^{3/2} 10^{-12}}{\sqrt{\mu} \rho_{\text{eff}}} \text{ watts/cm}^2$$

where*

B_y = pole-face flux density in gauss.
 λ = wavelength of pole-face pulsations in cm.
 f = frequency of pole-face pulsations in Hz.
 μ = permeability of pole-face material in C.G.S. units.
 ρ_{eff} = effective resistivity of pole-face in ohms-cm.

Now,

λ is equal to the stator slot pitch which is 0.8 cm

and

f is given by:

$$f = \frac{N}{60} \times \text{No. of slots} = \frac{120N}{60} = 2N \text{ Hz}$$

where

N = rotor speed in rpm

also,

$$B_y = 0.0588 B_{gap}; B_{gap} = 0.508 B_{teeth} \text{ (obtained by equating areas)}$$

therefore,

$$B_y = 0.0299 B_{teeth}$$

The average effective value of $\mu = 14.0$ for H-11 rotor alloy was obtained from earlier tests as described, and the effective value of the resistivity due to the pole-face grooves (figure 4.1-1) is $\rho_{eff} = 2.63 \rho_{abs}$. As the resistivity of the pole-face material at 200°F is $\rho_{abs} = 49 \times 10^{-6}$ ohms-cm, we have a value of:

$$\rho_{eff} = 129 \times 10^{-6} \text{ ohms-cm}$$

Calculations were made at all values of flux density and speed for an assumed pole-face temperature at 200°F, and the variation of pole-face loss with temperature was calculated from the change in pole-face resistivity (reference 4-3). Results of the pole-face loss calculations are given in figures 4.2-1 and 4.2-2 together with test results.

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* See Section 4.9 for listing of symbols used in this section (Section 4).

FIGURE 4.2-1. Calculated Pole-Face Losses at 200°F and Test Results

FIGURE 4.2-2. Calculated Pole-Face Loss Temperature Correction

4.2.2 Windage Loss - W (reference 4-2)

Windage loss W, in watts, is given by:

$$W = 6350 K_1 K_2 K_3 (D/10)^{4.72} (N/10^4)^{2.86} (\rho_H/0.0765)^{0.86} (\eta/0.0435)^{0.14}$$

where

$K_1 = 0.98$, $K_2 = 1.03$, $K_3 = 0.31$ (geometry factors are determined from generator dimensions.)

Now, ρ_H for Helium gas at 760 mm mercury pressure is 0.0088 lb/ft³, and density is directly proportional to pressure, therefore at 30 mm mercury pressure

$$\rho_H = 0.000348 \text{ lb/ft}^3$$

and viscosity η for Helium at 200°F is 0.051 lb/ft hr (viscosity is proportional to $\sqrt{\text{temperature}}$).

Calculations were made at all values of speed and for pressures between 10 mm and 760 mercury at a temperature of 200°F. Results of the calculations and correction for temperature change are given in figures 4.2-3 and 4.2-4. Windage loss versus speed is plotted for calculated and test results in figure 4.2-11.

4.2.3 Stator Iron Losses

These losses are analyzed separately as D.B.S. loss and teeth loss.

(a) D.B.S. Loss. A curve of flux density in the D.B.S. (obtained from previous static flux distribution tests on the generator) is given in figure 4.2-5. At each excitation values, the corresponding D. B. S. flux density is found. This value is transferred to figure 4.2-6 (core-loss for Hiperco 27) together with the frequency and the corresponding loss in watts/lb is obtained. The loss is multiplied by the D.B.S. weight in pounds to give loss in watts, where:

$$\text{D.B.S. weight} = [\rho_S (\pi/4) (D_O^2 - D_I^2)] (\text{Stack Length}) (A.S.F.) (2) \text{ lbs}$$

where

ρ_S is the density of the core steel = 0.285 lbs/in³
 D_O is the O.D. of the D. B. S. = 14.11 inch
 D_I is the I.D. of the D. B. S. = 12.88 inch
stack length (one stack) = 3.23 inch
A.S.F. is the armature stacking factor = 0.92

FIGURE 4.2-3. Calculated Windage Loss Versus Pressure

FIGURE 4.2-4. Calculated Windage Loss Temperature Correction

FIGURE 4.2-5. A. C. Flux Density in the D.B.S. (from Test No. P6-004)

FIGURE 4.2-6. Core Loss for Hiperco-27, 0.004 Inch Thick
(from Epstein Square Tests on Standard
Samples at Room Temperature)

The multiplying factor of 2 is included as there are two stacks.

Total D.B.S. weight is found to be 44.34 pounds. D.B.S. loss is calculated for all excitation values and speeds and the results are shown in figure 4.2-7. Variation of D.B.S. loss with temperature has not been accurately determined over the range of 100° to 300°F and therefore cannot be included.

(b) Tooth Loss. A curve of flux density in the teeth (obtained as for the D.B.S.) for all excitation values is given in figure 4.2-8. At each excitation value, the corresponding tooth flux density is found. This value is divided by 2 (as the flux in the teeth does not change direction, but each tooth experiences an effective sinusoidal ac flux wave of $B_{\text{teeth}}/2$ amplitude) and transferred to figure 4.2-6 where the core-loss in watts/lb is read against the frequency. This loss is multiplied by the tooth weight in pounds to give loss in watts,

where

$$\text{tooth weight} = \{ [\rho_s (\pi/4) (D_i^2 - D_{i1}^2)] (\text{Stack length}) \\ (\text{A.S.F.}) (2) \} - \{ (\rho_s) (W_s) (h_s) \\ (\text{Stack length}) (\text{A.S.F.}) (2) \}$$

where

D_i is the I.D. of the D.B.S.	= 12.88 inch
D_{i1} is the I.D. of the teeth	= 11.51 inch
s is the number of slots	= 120
W_s is the width of slots	= 0.164 inch
h_s is the height of slots	= 0.685 inch

Total tooth weight is found to be 21.48 pounds. Tooth loss is calculated for all excitation values and speeds and the results are shown in figure 4.2-9. Again no temperature variation is given.

A curve of total electrical losses and test results, versus excitation, for 200°F, is given in figure 4.2-10.

4.3 TEST DESCRIPTION

The generator was mounted in the test fixture and coolant circuits and lubrication circuits were attached. The fixture was then mounted on an aluminum bed which was secured inside a thick walled vacuum chamber (figure 4.3-1). Piping for all coolant and lubricant circuits was vacuum sealed to the chamber together with all electrical leads and thermocouples. The thermocouples were attached to the generator at various strategic positions and to all coolant and lubricant pipes so that the total temperature change on all fluid paths could be recorded and also so that the temperatures of certain parts of the generator could be determined to establish heat flow paths and characteristics. Apparatus

FIGURE 4.2-7. Calculated Stator Iron Loss in D. B. S.

FIGURE 4.2-8. Flux Density in the Teeth (from Test No. P6-004)

FIGURE 4.2-9. Calculated Stator Iron Loss in Teeth

FIGURE 4.2-10. Calculated Total Electrical Losses and Test Results

FIGURE 4.2-11. Windage Loss Versus Speed Calculated and Test Results

FIGURE 4.2-12. View of Slotted Rotor Poles

FIGURE 4.3-1. View of Generator Mounted in Test Chamber

was provided for evacuating the chamber and providing an atmosphere of helium gas at 30 mm of mercury with a steady helium flow. The inlet and outlet helium temperatures were recorded. Apparatus was provided to pressurize the coolant and lubricant circuits. Flowrates of all fluids were recorded so that heat losses could be calculated individually for each flow path.

The generator was driven by a 24 hp 24,000 rpm electric motor situated outside the vacuum chamber with a vacuum seal between the drive shaft and chamber. A torquemeter was incorporated on the drive shaft to enable input mechanical power to the generator to be measured. The generator field was excited by a regulated d-c supply and all input electrical power to the generator was recorded. By these means, the input power (shaft power and field power) and the output power (fluid temperature change) could be recorded, enabling the loss balance to be made. The following tests were made to enable the losses to be separated as far as possible:

1. Complete test runs were made at 6000 rpm (two runs) and 12,000 rpm and an attempt was made to run a test at 18,000 rpm, only to result in a bearing failure. Therefore, the only results presented are for 6000 and 12,000 rpm.
2. Windage and friction losses were obtained by driving the generator at the two test speeds with zero field excitation, input power being measured on the torquemeter. This value can be compared with the output power measured on the fluid flow paths, taking exceptions for noise and vibration losses.
3. At each speed the field was excited to values of 0, 5, 10, 15, 20, and 25 amps and all power inputs and heat outputs were recorded at each point.

The heat flow out of the stator coolant circuit should equal the field input power and stator iron losses, together with radiated heat from the rotor (positive or negative depending on relative temperatures).

The heat flow out of the rotor coolant circuit should equal the pole-face losses together with some bearing heat due to friction, some heat from the motor shaft and any radiated heat from the stator (positive or negative depending on relative temperatures).

The heat flow out of the bearing lubrication circuit should equal the bearing friction loss together with stray losses from the rotor, rotor coolant and motor shaft (positive or negative depending on relative temperatures).

The heat flow through the helium circuit and vacuum pump should equal the windage losses together with stray heat from other sources in the generator.

Other losses and power inputs not measured or accounted for in the test procedure include: heat loss through the chamber outer surface; noise; vibration; heat loss in the field supply leads; heat gain through the motor shaft. The large thermal mass of the generator and chamber contributes to errors by absorbing heat under some conditions and rejecting heat under other conditions.

4.4 CALCULATION OF HEAT BALANCE FROM TEST RESULTS

Heat outputs and power inputs were calculated from the test results by means of the following equations.

1. Field power input $P_F = V \times I$ watts.
2. Shaft power input $P_M = \frac{2 \pi N T (746)}{33000 (12)}$
 $= 0.01185 N T$ Watts
3. Rotor heat output $P_R = (\text{Flowrate})$
 $(\Delta T_R 17.58) (62.5) / (7.481)$
 $= (146.8) (\text{Flowrate})$
 (ΔT_R) Watts
4. Stator heat output $P_S = (146.8) (\text{Flowrate})$
 (ΔT_S) Watts
5. Bearing tube lubricant $(\text{Flowrate}/60)$
heat output $P_L = (\Delta T_L) (62.5/7.481) (17.58)$
 $(0.519) (0.838)$
 $= (1.062) (\text{Flowrate})$
 (ΔT_L) Watts
6. Helium heat output $P_H = (\text{Flowrate}/60) (0.0088)$
 $(\Delta T_H) (17.58) (12.3)$
 $= (0.032) (\text{Flowrate})$
 (ΔT_H) Watts

where

I = Field current	Amps
V = Field voltage	Volts
N = Generator speed	RPM
T = Torque	In-lbs.
Rotor Flowrate	Gals/min
Stator Flowrate	Gals/min
Lubricant Flowrate	Gals/hr.
Helium Flowrate	Ft ³ /hr.

$\Delta T_R, \Delta T_S, \Delta T_L, \Delta T_H$ = Temperature rise °F

Lubrication oil density = 0.838 grams/ml

Lubrication oil specific heat = 0.519 BTU/LB-°F

Helium density at 760 mm mercury = 0.0088 Lbs/ft³

Helium specific heat at 30 mm mercury = 12.3 BTU/LB-°F

All power inputs and heat losses were calculated and are tabulated in tables 4.4-1 and 4.4-2.

Theoretically, for the heat balance,

$$\text{Total input power} = \text{Total heat output}$$

However, some heat cannot be accounted for as explained in Section 4.3. This is lost heat, which limits the accuracy of the heat balance and is tabulated in table 4.4-3.

TABLE 4.4-1. 6000 RPM Test Results

Power Input and Heat Output, Watts							
Field Current I	0	5	10	15	20	25	30
Field Power P_F	0	47	190	440	814	1810	2020
Motor Power P_M	411	582	1262	2215	3021	3180	3543
Total Input Power $P_F + P_M$	411	629	1452	2655	3835	4990	5563
Rotor Heat P_R	338	-	-	-	-	690	-
Stator Heat P_S	616	-	-	-	-	3820	-
Lube Heat P_L	46	-	-	-	-	46	-
Helium Heat P_H	2	-	-	-	-	6	-
Total Heat $P_R + P_S + P_L + P_H$	1002	-	-	-	-	4562	-
Separated Losses-Watts							
Pole-Face Loss [= $P_R - P_{R_0}$]	0	-	-	-	-	352	-
Total Elec. Loss [= $P_M - P_{M_0}$]	0	171	851	1804	2610	2769	3132
Windage Loss [= P_{M_0}]	411	-	-	-	-	-	-

TABLE 4.4-2. 12,000 RPM Test Results

Power Input and Heat Output, Watts							
Field Current I	0	5	10	15	20	25	30
Field Power P_F	0	48	196	482	964	-	-
Motor Power P_M	2278	2820	4320	6600	8650	-	-
Total Input Power P_F+P_M	2278	2868	4516	7082	9614	-	-
Rotor Heat P_R	366	477	514	770	0	-	-
Stator Heat P_S	366	477	1840	3480	6360	-	-
Lube Heat P_L	58	77	77	77	100	-	-
Helium Heat P_H	0	0	0	10	15	-	-
Total Heat $P_R+P_S+P_L+P_H$	790	1031	2431	4337	6475	-	-
Separated Losses-Watts							
Pole-Face Loss [= $P_R-P_{R_0}$]	0	111	148	404	-	-	-
Total Elec. Loss [= $P_M-P_{M_0}$]	0	542	2042	4322	6372	-	-
Windage Loss [= P_{M_0}]	2278	-	-	-	-	-	-

TABLE 4.4-3. 6000 and 12,000 RPM Test Results

Heat Balance							
Field I	Current Amps	Total Input Watts	Power	Total Heat Watts	Output Watts	Lost Heat Watts	% of Input
6000 RPM							
	0	411		1002		-591	144
	5	629		-		-	-
	10	1452		-		-	-
	15	2655		-		-	-
	20	3835		-		-	-
	25	4990		4562		428	8.6
12,000 RPM							
	0	2278		790		1488	65.4
	5	2868		1031		1837	64
	10	4516		2431		2085	46.1
	15	7082		4337		2745	38.8
	20	9614		6475		3139	32.6

4.5 SEPARATION OF LOSSES FROM TEST RESULTS

Pole-face losses will appear as heat in the rotor together with other heat from bearings, etc. This stray heat can be separated approximately by reading the value of the rotor heat output at zero excitation (P_{R0}), whereupon this value can be subtracted from all other values of P_R to give pole-face loss.

That is

$$\text{Pole-Face Loss} = P_R - P_{R0}.$$

Windage loss is represented by the generator input power at zero excitation (P_{M0}), neglecting the inclusion of friction which is difficult to separate.

That is

$$\text{Windage Loss} = P_{M0}.$$

Stator iron losses will appear as heat in the stator together with field coil power. The iron losses can be separated by subtracting the field power (P_F) from the stator heat (P_S) at each excitation point.

That is

$$\text{Stator Iron Loss} = P_S - P_F$$

Total electrical losses can be represented by the generator input power at each excitation point with the windage loss subtracted.

That is

$$\text{Total Electrical Loss} = P_M - P_{M0}.$$

This total electrical loss should be equal to the stator iron losses plus the pole-face loss.

That is

$$\text{Total Electrical Loss} = (P_S - P_F) + (P_R - P_{R0})$$

Any difference between these two results is due to lost heat and measurement errors.

4.6 ANALYSIS OF RESULTS

Two test runs were made at 6000 rpm to check the consistency of the test results over a period of three days and it was found that they were in very close agreement so only one set of the two results is presented for 6000 rpm - those being taken immediately prior to the 12,000 rpm test.

Test results are illustrated in figures 4.2-1, 4.2-10, and 4.2-11, for comparison with calculated losses. These test results are tabulated in tables 4.4-1 and 4.4-2.

4.6.1 Pole-Face Losses (figure 4.2-1)

At 6000 rpm, only one accurate test result was obtained and this shows a 34 percent decrease on the calculated figures. At this test point the lost heat was calculated at 8.6 percent of input power, giving a possible source of error on the test result. In this case, variations between calculated and test results will also be due to variations in the effective pole-face resistivity due to temperature rise of the pole-face and to variations in the average effective permeability with flux density.

At 12,000 rpm, three test results were obtained showing an increase of 25 percent to a decrease of 52 percent on the calculated values. As lost heat varied between 65 percent and 39 percent of input power over this range, very little accuracy can be attached to these results. However, they do agree with the 6000 rpm results in showing a major decrease on the calculated results, and the same comments apply in this case also.

4.6.2 Stator Iron Losses (figure 4.2-10)

These losses were not calculated separately from the test results but as they constitute about 86 percent of the total electrical losses shown in figure 4.2-10, the differences between calculated and test results is mainly due to differences in the stator iron losses.

At 6000 rpm, the six test results obtained show a maximum decrease of 40 percent on calculated results. This difference is mostly due to the use of an excessively high 'experience factor' used in calculating the losses in Section 4.2. If the experience factor was reduced to approximately two (2), the results would show a much better correlation.

At 12,000 rpm, four test results were obtained and again a decrease of up to 40 percent on calculated results was found. Again, use of an 'experience factor' of two would provide more useful calculated results.

4.6.3 Windage Losses (figure 4.2-11)

Test results showed a very marked increase over calculated results as shown in figure 4.2-11 - 27 times at 6000 rpm and 41 times at 12,000 rpm. Some variation will be due to small quantities of air, water vapor and oil vapor present in the chamber atmosphere thus affecting values of ρ_H and η , but evidently a very great percentage of the loss should be accounted for as friction. It is possible that the result at 6000 rpm is slightly higher than the true value due to measurement errors, as the results show an increase in proportion to $(N)^{2.46}$ to 12,000 rpm, whereas windage is proportional to $(N)^{2.86}$ and friction is proportional to $(N)^3$.

4.7 DISCUSSION ON GENERATOR BEARING FAILURE

During the run-up of the generator to 18,000 rpm from 12,000 rpm, severe bearing vibrations were recorded resulting in abortion of the tests. These vibrations were accompanied by a sudden rise in bearing temperature at the anti-drive end to over 350°C.

The generator was dismantled completely and various views of the damaged components are shown in figures 4.7-1, 4.7-2, 4.7-3, and 4.7-4. The anti-drive end bearing had disintegrated completely (figure 4.7-2) resulting in the loss of thrust constraint on the rotor, allowing it to damage the rotor coolant seal and the rotor stub shaft (figures 4.7-3 and 4.7-4). As the anti-drive end of the rotor was virtually unsupported, the drive end bearings rapidly overheated-evidence being found in discoloration of the bearing inner races. It was also found that the anti-drive end rotor windage baffle had been rubbing the rotor at three points near its periphery but a check revealed adequate clearance at these points. It can only be assumed that excessive bearing heat caused the baffle to distort, returning to its original shape on cooling.

The reason for the bearing failure is not accurately known and it may have been due to either of the following points:

1. Leakage of rotor coolant at the rotor seal may have caused reduced bearing lubrication resulting in overheating.
2. Failure of the phenolic bearing ball separators due to excessive temperature or wear.

It is not clear from the instrument recordings whether the vibration caused the excessive temperature or vice-versa and therefore no specific reason for the bearing failure can be deduced.

FIGURE 4.7-1. View of Generator After Bearing Failure

FIGURE 4.7-2. View of Generator Bearings After Failure

FIGURE 4.7-3. View of Generator After Bearing Failure

FIGURE 4.7-4. View of Generator After Bearing Failure

4.8 CONCLUSIONS

The no-load loss tests have proved to be very valuable in the verification of calculation procedures and in providing some basis for the various assumptions which had to be made in the calculations.

Comparison of test results with calculated results has yielded the following information related to calculation procedures and assumptions.

1. Stator iron losses were found to be about 40 percent lower than the calculated losses at both test speeds. The most likely reason for this difference is that the calculated results were excessively high. Examination of the calculation procedure (Section 4.2) will reveal that the only real variable involved is the flux density in the stator - the values of flux density used were obtained from static flux tests and results were accurate to ± 4 percent. It will also be found that the calculated results were multiplied by an experience factor of three (3). This factor was based on test results obtained from aircraft type generators and is used to account for losses involved in welded stator stacks, etc. The test results show that an experience factor of two (2) would provide calculated results much more consistent with those found in the test. This lower experience factor may be due to the fact that the stacks were not welded and were only under light pressure.
2. Pole-face losses were also found to be lower than calculated losses at both test speeds; a 34 percent decrease was found at 6000 rpm. This decrease can be due to either a decrease in the flux density in the pole-face or an increase in the pole-face permeability or resistivity. Pole-face flux density was calculated from tooth flux density and should be accurate to ± 4 percent. Calculations were made for 200°F pole-face temperature, and it is likely that the temperature may have exceeded 300°F - this would cause a 2 percent reduction in losses. The average effective value of the pole-face permeability used for the calculations was $\mu = 14.0$; as designated, this is an average value over the range of B_{yo} - $B_{y\max}$ and consequently will not hold true at flux densities higher or lower than the average B_y . If a value of $\mu = 28.6$ is substituted into the calculations for the test point, a corresponding decrease in pole-face loss of 30 percent is given, which is very consistent with test results.

The results of the test at 12,000 rpm show that the pole-face losses varied between 25 percent higher and 52 percent lower than calculated results - this is consistent with the varying value of μ , and substitution of values of μ between 9 and 60 into the calculations gives results more consistent with test results. It is evident that values of μ will have to be calculated for each test point to obtain more consistent results in future tests.

3. Windage losses as measured in the test (including friction) were very much higher than calculated values of windage alone. Factors affecting the calculated values are atmosphere density and viscosity and speed. None of these parameters varied by any significant amount from the calculation values during the tests which leads to the conclusion that the greater part of the measured windage losses was friction in the bearings and fluid friction in the rotor coolant system. In order to separate the windage losses from the friction losses, tests should have been made at various values of chamber pressure at each speed. Because the results of the test preclude separation of friction and windage, no definite conclusions can be made as to the accuracy of the calculations.

Accuracy of the test results was dependent on various factors and perhaps the main source of inaccuracies was the amount of heat lost from the various parts of the generator (heat lost is tabulated in table 4.4-3). Some of this loss would be due to the large radiating surface of the test chamber and some loss would be through external piping, shafts, etc., allowing conduction from the generator. Noise and vibration would also contribute to the amount of input power lost. Some of the temperature measuring instruments had to be re-calibrated during the tests. Often heat loss calculations involved very small temperature differences, measurements of which approached the limiting accuracy of the instruments. Measurements of input power on the motor were accurate to ± 1 percent and on the field to ± 1 percent, with the exception of heat loss in the field supply leads. Measurement accuracies on the whole were better than ± 5 percent.

Due to the large heat losses at some test points, the results obtained can serve only as an indication of the correct results and it is felt that these indications were adequate to correct the calculation methods as previously shown.

4.8 LIST OF SYMBOLS USED IN SECTION 4

<u>Symbol</u>		<u>Units</u>
B	Flux Density	kl/in ²
B _y	Pole-Face Flux Density	Gauss
D	Rotor Outside Diameter	Inch
D _i	D.B.S. Inside Diameter	Inch
D _{i1}	Teeth Inside Diameter	Inch
D _o	D.B.S. Outside Diameter	Inch
f	Frequency	Hz
h _s	Height of Slots	Inch
I	Field Current	amps
K ₁ ,K ₂ ,K ₃	Windage Constants	-----
N	Speed	rpm
P	Pole-Face Loss	watts
P _F	Field Winding Input Power	watts
P _H	Helium Atmosphere Heat Loss	watts
P _L	Lubricant Heat Loss	watts
P _M	Shaft Input Power	watts
P _{M0}	Shaft Input Power - Zero Excitation	watts
P _R	Rotor Heat Loss	watts
P _{R0}	Rotor Heat Loss - Zero Excitation	watts
P _S	Stator Heat Loss	watts
S	Number of Stator Slots	-----
T	Torque	in-lbs
V	Field Voltage	volts

<u>Symbol</u>		<u>Units</u>
W	Windage Loss	watts
W_s	Width of Slots	Inch
ΔT_H	Temperature Change - Helium Flow	$^{\circ}F$
ΔT_L	Temperature Change - Lubricant Flow	$^{\circ}F$
ΔT_R	Temperature Change - Rotor Coolant	$^{\circ}F$
ΔT_S	Temperature Change - Stator Coolant	$^{\circ}F$
λ	Wavelength	cm
μ	Pole-Face Permeability	C.G.S. units
η	Fluid Viscosity	lb/ft-hr
ρ_{abs}	Pole-Face Resistivity - Absolute	ohms-cm
ρ_{eff}	Pole-Face Resistivity - Effective	ohms-cm
ρ_H	Helium Density	lbs/ft ³
ρ_S	Stator Steel Density	lbs/in ³

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