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MASTER

AN INDUCTOR ALTERNATOR ROTOR FOR SPACE APPLICATION

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ABSTRACT

A discussion is presented of the technology and design criteria for the rotor in a radial gap, potassium cooled, inductor alternator as defined by recent work performed for the U.S. Atomic Energy Commission. Specific attention is given to the rotor material limitations, most favorable rotor speeds, and the reduction of pole face and windage power losses. Unique design considerations for the rotor cooling scheme are presented for alternator ratings between 0.4 and 12 MVA.

material for the rotor and maintaining stress intensities which are within the allowable limits of the material, the reduction of the pole face and windage power losses, and cooling the rotor effectively. These problems are not strange to a designer since the inductor alternator is not a recent innovation - patents were awarded in 1890 and 1896 to Messrs. W. M. Mordy and C. Steinmetz, respectively. However, they are unique at this time since they require excellent treatment and new technology for a high-temperature alternator for space applications.

ROTOR MATERIAL

INTRODUCTION

Development of the technology and design criteria for a potassium cooled version of a radial gap, inductor alternator was initiated in 1960 since this type of alternator was found to be most suited for an advanced high-temperature Rankine nuclear space power system. A significant part of the effort has been spent on the development of the rotating magnetic field structure for the alternator. The purpose of this paper is to review the technology for a rotor as defined by recent work performed for the U.S. Atomic Energy Commission under Contract AT(04-3)-679.

The essential duty of the rotating magnetic structure is to produce a periodic time varying induction within the stationary armature that contains coils of wire in which electricity is induced. This feat is accomplished by incorporating two, three or more magnetic north poles on one end of the rotor with an equal number of south poles on the opposite end. A typical arrangement for a rotor and stator for this type of alternator is shown in figure 1. This arrangement fulfills the requirements for potassium cooled alternator since all of the windings are in the stator compartment and can be hermetically separated from the rotor compartments which contains potassium vapor.

Analytical studies for power levels of 0.4, 1.2, 6.0 and 12.0 MVA and for coolant temperatures of 500°, 800°, and 1100°F were performed to determine the capabilities and limitations of an alternator for nuclear space power applications. Several rotor materials were investigated as part of the over-all study. Special attention was given to the system weight penalties associated with the magnetic and strength properties of the rotor materials and the coolant temperature. It was concluded from the results of these investigations that premium quality H-11 steel, AMS 6487, heat treated to a hardness of Rockwell C45 was the best rotor material for coolant temperatures between 500° and 800°F. The 800°F coolant provided a system weight advantage over lower coolant temperatures. A coolant temperature above 800°F was not considered practical for H-11 steel since the strength of this material decreases rapidly above this temperature. Materials which offered the best combination of strength and magnetic properties for a 1100°F coolant were investigated. However, it was found that these designs did not offer a system weight advantage over the designs with the lower coolant temperatures and the H-11 rotor material. It is therefore logical to forecast from the present technology that future rotors for high-temperature alternators for space applications will be made of H-11 steel and cooled with potassium at 800°F or below.

The problems which face the designer of a rotor are the selection of the best

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The dc magnetic properties for H-11 steel are given in figure 2. In general this material can support an induction of 85 kilolines per square inch through the mid section of the rotor core where the flux density is a maximum. The limitations that H-11 steel imposes upon a rotor from the standpoint of strength properties are graphically illustrated in figure 3. The values given in figure 3 represent the author's interpretation and extrapolation of the uniaxial tensile data given in references 1 and 2. Considerable extrapolation of the creep data was necessary to obtain the creep information displayed in figure 3, and these creep strengths should only be considered a "best estimation" of the creep strengths for 5 years of operation. The limitations are given in terms of allowable stress intensities at design speed and were based upon the following considerations:

- (a) The rotor should be capable of a 20 percent overspeed without catastrophic failure. To accomplish this, the stress intensity within the rotor at 120 percent of design speed must not exceed the minimum ultimate strength of the material. This limitation can also be satisfied by maintaining a stress intensity at design speed which is less than 69.5 percent of the minimum ultimate strength of the material as indicated in figure 3.
- (b) The stress intensity at critical locations, i.e., under the poles where plastic flow of the material will not affect the load support by the material, shall not exceed the minimum 0.2% offset yield strength of the material. Seven-two percent of the nominal value was used for the minimum value of 0.2% offset yield strength.
- (c) The creep strain in the rotor for 5-years operation shall be less than that required for a rubbing contact between the pole tips and the ceramic bore seal.

The initial dimensions of a typical alternator will permit a maximum creep strain of approximately 0.33% which means that the highly stressed portions of the rotor can have a creep strain of approximately 0.5%. It can be noted from figure 3 that with these creep strain limitations, the allowable stress intensity within a rotor will be limited by the creep strength of the materials for coolant temperatures above 775°F. Therefore, the designer of the alternator for this range of temperatures is faced with the problem of predicting the creep strain rate of the rotor if the maximum capability of the material is to be utilized. To accomplish this one must first establish the elastic state of

stresses and then establish the variance of the stresses when plastic deformation occurs. Since the plastic flow of the material is also a function of the stress intensity, this becomes a very complex problem with inter-related variables. It should also be pointed out that the stresses within a rotor are multiaxial (primarily biaxial) in nature which means that an equivalent stress intensity must be formed to obtain the plastic strain from simple uniaxial creep test data. The complexity of the problem is compounded by the fact that a miscalculation of the stress intensity by 10 percent can easily result in a 50 percent error in the estimated strain. Furthermore, it assumes that the temperature distribution within the rotor has been accurately defined. An error of 15°F can easily result in a 50 percent error in the estimated strain over a five year period. When we compound a possible error of 10 percent in stress and a 15°F error of temperature, the estimated creep strain can be in error by 125 percent. This assumes the material will exhibit at least average creep strain properties for H-11 steel and that accurate long time creep strain data for the average specimens was used for the calculations. The designer would not be expected to use just average material from the standpoint of creep strength, and it would be logical to assume that inferior material could be eliminated by testing the material during the procurement phase. A large number of long time creep tests will be required to establish the distribution and mean value of the creep strain at stresses and temperatures pertinent to the application. Even with complete and accurate uniaxial creep data on the material, it is apparent that strain calculations may be in error by a significant amount. In the past, the designers of high speed light weight machinery have based their predictions upon the creep strain rates observed during previous experiments on the actual or similar machines. There is little doubt that experiments on model and full scale rotors will be required before a rotor can be designed that utilizes the maximum capability of the H-11 rotor material. Until this information is available, a somewhat conservative rotor design will be required. The rotor dimensions and speeds for alternator ratings of 0.4 through 12.0 MVA are given in table 1. These designs are somewhat conservative from the standpoint of stress.

REDUCTION OF POLE FACE LOSSES

Losses which produce heat in the pole face are caused by two sources of time varying inductions. The first is due to the fact that as the pole face passes under a slotted armature, all points on the pole face experience a varying flux density. At

one instant a point on the pole face is under a tooth where the flux density is a maximum and at the next instant it is under a slot where the flux density is a minimum. A second source of time varying induction is produced by the flux that is generated by harmonic mmf's associated with the armature windings. Methods for calculating the pole face losses generated by these two sources of time varying inductions are given in reference 3. Some of the parameters which have a significant influence upon the magnitudes of the inductions are the ratio of slot opening to tooth pitch, the ratio of slot opening to radial gap, the ratio of radial gap to pole pitch, pole pitch and the direct axis synchronous reactance of the machine.

In general, these parameters can be varied to reduce the amplitude of flux variations on the surface of the poles. However, serious performance and weight penalties are incurred in the over-all design when these parameters are optimized only from the standpoint of pole face losses. The designer is therefore obliged to look for other methods to reduce the pole face losses. The most promising solution of the dilemma is to design the pole tip so minimum losses are incurred even when significant flux variation exists on the surface of the pole tips.

The losses within the pole face from the standpoint of material properties are primarily a function of $(\rho\mu)^{-1/2}$ where ρ equals the effective resistivity of the material along the eddy-current path and μ equals the effective permeability of the material when a small time varying induction is imposed upon the steady induction. H-11 steel, unfortunately, offers a very low ac magnetic permeability when subjected to the time varying inductions associated with the pole tips; therefore, this material must be considered inferior from the standpoint of pole face loss reduction. A comparison of the effective permeability of H-11 steel with a superior steel is given in table 2.

Two techniques can be used to increase the resistance of the eddy-current paths. The first technique consists of slotting the pole faces with grooves in a direction parallel to the motion of the rotor. The increase in resistance which can be achieved by slotting the faces is limited by how much material one can remove from the pole tips and by manufacturing limitations. Manufacturing techniques have been developed whereby a groove 0.005 inches wide by 0.10 inches deep can be made. With this width of groove, it is possible to use approximately 30 grooves per inch which increases the resistance of the eddy-current paths by a factor of 2.8 when compared with the resistance of a solid pole face. A

second technique which can be used to increase the resistance of eddy-current paths is to incorporate a pole tip of a laminated construction. The advantages offered by this technique are (1) a large increase in resistance can be obtained and (2) a material which offers superior a-c magnetic properties can be used. A comparison of the resistance of a grooved pole face with a laminated pole tip construction is given in table 3.

These techniques for reducing pole face losses were applied to alternators of different ratings and the results for two alternators are summarized in table 4.

It can be noted from table 4 that the pole face losses are excessive when provisions for the reduction of these losses are not incorporated on pole tip. Without provision for reducing the pole face losses, a significant weight penalty is imposed upon the system due to the decrease in performance of the alternator. An increase of alternator efficiency can be achieved by slotting the pole face; however, the losses are still large compared with losses associated with laminated pole tips. Laminating the pole tips reduces the losses by 20 to 30 folds and the system weight penalty associated with these losses becomes very small. In addition to the efficiency and weight considerations, the heat flux within the body of the pole is reduced and a complex cooling scheme is not required to maintain a reasonable pole face temperature. Without laminated pole tips, it would be very difficult to maintain a pole face temperature less than 950°F, the maximum allowable temperature for H-11 steel, with an 800°F coolant. With laminated pole faces it is possible to cool the rotor with coolant passages in the body of the rotor core.

A breakdown of the pole face losses for the 12.0 MVA alternator is given in table 5. It should be noted that the losses generated by harmonic mmf's from the armature windings are predominant which is typical for a high-rating, high-speed alternator of this type for space application. In general, this observation supports the conclusions presented by Barelo⁽⁴⁾ in 1955. It can also be noted from the table that the fifth harmonic predominates. If 120° phase belts had been used for the windings in place of 60° phase belts, even and odd harmonics would have been present and the combined loss for all of the harmonics would have been increased. Windings with a whole number of slots per pole per phase and 60° phase belts eliminates all even harmonics and sub-harmonics. The combined losses for cases without an integral slot winding were found to be excessive.

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WINDAGE POWER LOSS

The windage power loss within the rotor cavity must be predicted from model theory and by using experimental data from a similar rotor configuration. A limited amount of data, reference 5, exists on models very similar to the rotor configuration of interest. Unfortunately, the experiments were performed at higher Reynolds numbers and at lower Mach numbers than the numbers associated with the rotors for a potassium cooled alternator. It was therefore necessary to extrapolate the parametric form of the solution offered by the experimental data to gain some knowledge of the windage power losses for the cases of interest. The accuracy of the windage power loss calculations are limited accordingly; nevertheless, some heedful conclusions can be made at this time.

The rotor compartment pressure should be reduced below the saturation vapor pressure of the liquid potassium within the bearing compartments. With 800°F potassium in the bearings, the windage power loss in the rotor compartment is approximately one-half percent of the electrical output of the alternator. This amount of loss causes considerable aerodynamic heating of the poles and ceramic bore seal. In the large alternator (12 MVA) the pole faces would be approximately 1300° to 1400°F if parallel passages through the core were used to cool the rotor. Since H-11 steel must be limited to 950°F or below, it is obvious that coolant passages within the main body of the poles are required when provisions for reducing the pressure are not incorporated.

One method which can be used to reduce the pressure is to restrict the flow of vapor by inserting Holweck type molecular pumps between the liquid seals and the rotor compartment; the pumps act against the vapor pressure of the interface in the liquid seals. This type of vapor seal configuration is shown in figure 1. The molecular pumps are cantilevered over the pressure generating screw seals which confine the liquid potassium in the bearings. The molecular pump consists of a smooth shaft running in a helical threaded housing and a two-stage pump is obtained by using the outside and underside of a cantilevered impeller.

The pumping action is derived from the collision of molecules with the moving surface of the shaft where a momentum exchange provides an additional velocity component in the direction of rotation. Subsequent collisions induce a flow that is proportional to the peripheral speed of the shaft. The pump maintains its maximum pressure differential when the net flow, the induced flow minus the leakage flow across the

thread lands, is zero - which is the condition desired when the pump is employed as a shaft seal. At low pressures, e.g., 0.01 psia or less, pressure heads of 10:1 are easily achieved and heads of several thousand to one are not uncommon. Therefore, these pumps can be considered very effective as a vapor shaft seal at molecular flow conditions. However, molecular pumps are not as effective at greater pressures and pressure heads as high as 10:1 are very difficult to achieve. Typically, the head across two equal successive stages might be 2:1 and 6:1, respectively.

The vapor pressure of potassium at 800°F is 0.155 psia. Thus, two or three stages of pumps will be required to produce molecular flow conditions within the third or fourth stage. This series of pumps would require two cantilevered impellers. However, an over-all pressure ratio of 1000:1 could be achieved which would reduce the windage power loss by a factor of a thousand. Aerodynamic heating of the poles and bore seal would not be a problem under this condition.

COOLING CONSIDERATION

A high-speed rotor requires an effective cooling scheme to remove the energy produced within and transferred into the poles. A scheme which offers distinct advantages for this task is illustrated in figure 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 a separate passage which can be diverted outward and into the body of the pole to shorten the heat conduction path between the pole face and coolant. The manifold on the outlet end of the rotor is similar to the inlet manifold except that it contains passages which divert the coolant into the drive end bearing discharge chamber. The advantages offered by this cooling concept are:

- (a) 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;
- (b) 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;
- (c) the rotor has a centrifugal pumping action on the coolant since it enters the rotor on the axis of rotation and exits from the rotor at the radius of the shaft extension;

- (d) this cooling scheme can be used in conjunction with bearing and shaft-seal systems that have been experimentally verified.

The 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 densities. Since the liquid potassium coolant will pick up and transport solid material from within the system, the designer must cope with the problem of preventing these materials from accumulating in passages where they could suppress the flow of coolant and cause a failure of the system. 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. This problem was first solved by Stokes to give the terminal velocity of a sphere in a fluid. When his formula is applied to the case illustrated in figure 4, the terminal velocity of the particle at the point of maximum centrifugal acceleration must be less than the velocity of the coolant in order for the particle to travel to a position where the centrifugal acceleration is less. By letting the terminal velocity of the coolant, we can solve for the maximum particle size of known density that can exit from the coolant passage.

Table 6 gives typical diameters 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 and one-half of an inch below the surface of the rotor core. The relative densities of carbon, and some of the light-weight compound of potassium which may be present as particles are in table 7. 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-weight compounds are given in table 7 for conditions stated previously.

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 is to design the coolant passages so the accumulation of the heavy material will

not close or suppress the flow of the coolant. When the coolant is diverted outward into the main body of the poles, this can be accomplished by installation of an extra passage for storage of the dense materials as shown in figure 5a. When the coolant is not diverted outward, extra large manifolds on the ends of the rotor core can be used for storage of the dense materials as shown in figure 5b.

The large centripetal acceleration which acts upon the coolant within rotor passages produces large pressures near the surface of the pole. If one considers the coolant passage to be completely filled with potassium, the pressure at the points of maximum acceleration are as indicated in table 6. In general, the magnitude of the pressure requires a welded construction which means that the rotor material must be a weldable alloy.

Coriolis forces act upon the coolant since it flows outward and inward from the axis of rotation. The effect of these forces is to cause secondary flow in the passages which increases the pressure drop per unit of pipe length when compared with the flow resistance of a straight non-rotating pipe. An analysis of the secondary flow phenomenon was performed, reference 6, based upon inviscid-fluid theory. This analysis did not contribute anything relative to the magnitude of the frictional flow resistance; however, it was demonstrated that the secondary flow in a rotating pipe is analogous to the secondary flow in a stationary curved pipe. Thus, empirical formulae for the frictional resistance of curved pipes can be used to determine the resistance of rotating pipes at low speeds by forming equivalent dimensionless parameters for each case. The equivalent parameters are $2r\omega/V$ and r/R , where

r = radius of pipe
 R = radius of curvature of the pipe axis
 V = means axial velocity
 ω = angular velocity of pipe

For the 12 MVA rating, typical values for the rotor are $r/R = 2r\omega/V = 2 \times 0.125" \times 240 \text{ radian/sec} \div 300 \text{ in/sec} = 0.628$. To predict the pressure loss for this case one must have flow resistance data from a curved pipe which had a radius of curvature that was only 59 percent greater than the radius of the pipe. For lower ratings and higher design speeds, the analogous cases would represent impossible situations since the radius of curvature of the pipe axis would have to be less than the radius of the circular cross section. Thus, curved pipe data could not be used to predict the flow resistance for many of the cases of interest and the feasibility of these designs from the standpoint of pressure

loss was still open to doubt. To resolve this question, an experimental model was constructed and fluid flow tests were performed at speeds up to 24,000 rpm. The objective of the test was to prove that the pumping capacity of a rotor exceeded the throttling capacity of the coolant passages for a typical high speed rotor. It was demonstrated by a series of tests that the only pumping required by the external system was to deliver the desired flow to the central port on the antidrive-end of the fixture. The flow rate through the rotor was independent of rotor speed and was unaffected by the back pressure on the rotor at speeds and pressure of practical interest. These tests not only verified that the additional flow resistance due to secondary flow in the radial coolant passage will not be a critical problem but also demonstrated the distinct advantage of designing the rotor so that it has a pumping action on the coolant.

CONCLUSIONS

H-11 steel offers the best combination of magnetic and strength properties for potassium cooled inductor alternator for coolant temperatures of 800°F and below. Coolant temperatures above 800°F are not practical for this material since the strength decreases rapidly. Designs with rotor material more suited for higher coolant temperatures did not offer a system weight advantage over the designs with H-11 rotor material. The stress intensity within a rotor for a five-year life with coolant temperatures above 750°F will be limited by the creep strength of H-11 steel and the designer of the rotor must cope with the problem of limiting the creep strain of the rotor to less than that required for a rubbing contact between the pole tip and the ceramic bore seal.

Provisions for reducing the pole face losses are required since H-11 steel offers inferior ac magnetic properties for the reduction of pole face losses. Slotting of the pole faces reduce the losses; however, coolant passages in the body of the pole would be required. With a laminated pole face construction the rotor can be cooled with passages in the rotor core. The pole face losses generated by harmonic mmf's from the armature windings are predominant.

The rotor compartment pressure should be reduced below the saturation pressure of the liquid potassium within the bearing compartments. With 800°F potassium in the bearings, the windage power loss is approximately 0.5 percent of the electrical output of the alternator and excessive aerodynamic heating of the poles occurs. Three to four stages of molecular pumps can reduce the windage power loss by a factor of a thousand and heating of the poles will

not be a problem.

The primary complication with cooling the rotor is due to the centrifugation of dense particles within the coolant passages. The only practical solution to this dilemma is to design the coolant passages so the accumulation of the heavy materials will not close or suppress the flow of the coolant.

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Rating, MVA	0.4	1.2	6.0	12.0
Rotor Speed, RPM	24,000	19,200	9,000	7,200
Rotor Dia., In.	11.572	16.318	34.548	47.241
Pole Height, In.	2.210	3.250	6.793	8.966
Width of Pole, In.	2.418	3.514	4.532	4.963
Pole Length, In.	3.469	4.988	8.737	13.677
Axial Distance Between Poles	2.284	2.587	3.495	3.094
Number of Poles	10	10	16	20
Weight of Rotor, lbs.	156.3	456.8	3245	9192
Maximum Stress Intensity, psi	52,500	71,200	67,250	79,300
Percent of 0.2% offset Yield Stress	53%	72%	68%	80%
Percent of Allowable Stress Intensity for 0.5% Creep strain in 5-yr.	58%	79%	74%	88%
Specific Weight of rotor, lbs/kVA	0.392	0.380	0.541	0.765

Table 1. Rotor Dimensions and Stress

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Amplitude of Time Varying Induction on Surface of Pole (gausses)	H-11 Steel	Non-Grain-Oriented 4-1/4% Silicon Steel
0	1.8	350
130	7.1	520
182	8.1	580
242	9.0	642
394	11.0	739
578	12.8	803
802	14.7	885

Table 2. Effective Permeability
(c.g.s.) Steady Induction Equals
8.75 Kilogausses

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	H-11 Steel	4-1/4% Silicon Steel
Bulk Material	78.7	108
Effective Resistivity of Slotted Poles	304	---
Effective Resistivity of Laminated Pole Tip, 0.010 inch sheet by 0.5 inches depth	---	918

Table 3. Comparison of Effective
Resistivity, Microhms - cm

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Rating, MVA	0.400	12.000	Pole Tip Configuration
Rating, MW (Electrical)	0.340	10.200	
Efficiency, Excluding Pole Face Losses	92.51%	94.23%	
Pole Face Losses, Watts	8,145	129,727	Solid Pole Tips of H-11 Steel
Efficiency, including Pole Face Losses	90.51%	93.11%	
Losses per unit area, Watts/in ²	96.38	95.38	
Weight penalty associated with pole face losses, pounds	116.8	1780	
Temp. drop required to conduct heat from pole tips to rotor core, °F	648	2590	
Pole Face Losses, Watts	5,373	78,458	Grooved Pole Tips, 0.005 inch wide by 0.10 inches deep with 30 grooves per inch in H-11 steel
Efficiency, including pole face losses	91.18%	93.55%	
Losses per unit area, Watts/in ²	63.59	57.69	
Weight penalty associated with pole face losses, pounds	77.0	1073	
Temp. drop required to conduct heat from pole tips to rotor core, °F	427	1569	
Pole Face Losses, Watts	371	4373	Laminated, 4.25% Silicon Steel, Non- grain-oriented, Depth of lamination = 0.5 inches, Thickness = 0.010 inches.
Efficiency, including pole face losses	92.42	94.19	
Losses per unit area, Watts/in ²	4.392	3.215	
Weight penalty associated with pole face losses, pounds	5.3	59.9	
Temp. drop required to conduct heat from pole tips to rotor core, °F	29.6	87.4	

Table 4. Pole Face Loss Reduction

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Rating = 12.0 MVA
Laminated Pole Tips
Total Loss = 4,373 watts

Tooth Ripple Loss	5.46%
Load Harmonics	<u>94.54%</u>
Total	100.00%

<u>Harmonic Number</u>	<u>Percent</u>
5	85.70
7	10.70
11	1.63
13	0.56
17	0.21
19	0.14
23	0.75
25	<u>0.31</u>
Total	100.00

Table 5. Breakdown of Pole Face
Loss, 60° Phase Belt

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MVA	Speed (rpm)	Rotor Dia. (inches)	Max. Radius of Passages in Poles (inches)	Max. Acceleration (ft/sec ²)	Pressure Developed (psi)
0.4	24000	11.572	5.286	2.78×10^6	6280
1.2	19200	16.818	7.909	2.66×10^6	9010
6.0	9000	34.548	16.774	1.24×10^6	8910
12.0	7200	47.241	23.170	1.10×10^6	10900

MVA	Speed (rpm)	Rotor Core Diameter (inches)	Max. Radius of Passages in Core (inches)	Max. Acceleration (ft/sec ²)	Pressure Developed (psi)
0.4	24000	7.149	3.075	1.62×10^6	2130
1.2	19200	10.318	4.659	1.57×10^6	3140
6.0	9000	20.961	9.981	0.739×10^6	3160
12.0	7200	29.309	14.655	0.695×10^6	4360

Table 6. Typical Diameters, Speeds,
Maximum Accelerations, and Pressure
in Alternators of Different Ratings

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Material	Specific Gravity	0.4 MVA	Rating 1.2 MVA	6.0 MVA	Location of Passage
Potassium Sulfide	1.8	0.234×10^{-3}	0.239×10^{-3}	0.350×10^{-3}	Poles
Carbon	2.0	0.212×10^{-3}	0.216×10^{-3}	0.317×10^{-3}	Poles
Potassium Monoxide	2.3	0.192×10^{-3}	0.196×10^{-3}	0.287×10^{-3}	Poles
Potassium Caronate	2.4	0.186×10^{-3}	0.190×10^{-3}	0.278×10^{-3}	Poles
Potassium Silicate	2.5	0.180×10^{-3}	0.184×10^{-3}	0.270×10^{-3}	Poles
Potassium Sulfide	1.8	0.294×10^{-3}	0.311×10^{-3}	0.453×10^{-3}	Core
Potassium Silicate	2.5	0.226×10^{-3}	0.240×10^{-3}	0.350×10^{-3}	Core

Table 7. Maximum Allowable Diameter
for Particles, inches

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Figure 1. Typical Alternator

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Figure 2. DC Magnetization of H-11 Steel

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Figure 3. Allowable Stress at
Design Speed for H-11 Steel

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Figure 4.. Maximum Allowable Terminal Velocity of Particle in Coolant Passages

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Figure 5a. Flow Path for Coolant
Within the Main Body of Poles

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Figure 5b. Manifold for Coolant
Passages Within the Rotor Core

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