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DESIGN STUDY (OF) ELECTRICAL
COMPONENT TECHNOLOGY FOR 0.25
TO 10.0 MEGAWATT SPACE POWER
SYSTEMS. PARAMETRIC DESIGN
STUDY OF CANNED AC INDUCTION
MOTORS

H.C. 3.00, MN 065

By
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December 15, 1967

Aerospace Electrical Division
Westinghouse Electric Corporation
Lima, Ohio

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Aerospace Electrical Division
Westinghouse Electric Corporation

PARAMETRIC DESIGN STUDY OF CANNED
AC INDUCTION MOTORS

DESIGN STUDY
ELECTRICAL COMPONENT TECHNOLOGY
FOR 0.25 TO 10.0 MEGAWATT
SPACE POWER SYSTEMS

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Prepared Under
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for the
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FOREWORD

This report describes work under contract to the United States Atomic Energy Commission from September 1, 1966 to February 29, 1968, on a Design Study of Electrical Component Technology for 0.25 to 10 Megawatt Space Power Systems.

The work was directed toward defining representative Rankine nuclear space power electrical systems, the electrical components therein, and their development problems. This particular report describes the parametric design study of canned ac induction motors suitable for prime movers of centrifugal circulating pumps. 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 was 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.

Members of the Westinghouse Aerospace Electrical Division who participated in the motor parametric design study are T.C. Allen, R.C. Fear, W. W. Yates, And M. Tatarko.

This report is assigned contractors designation Report Number WAED 67.52E.

ABSTRACT

Parametric studies of canned ac induction motors for prime movers of centrifugal pumps for space nuclear electric power plants utilizing the Rankine conversion cycle is presented. The parametric data is suitable to determine minimum system weight as a function of shaft horsepower rating, frequency, synchronous speed, voltage, and coolant temperature for ratings between 5 and 250 horsepower.

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1. INTRODUCTION

The Aerospace Electrical Division of Westinghouse Electric Corporation has performed a design study to define conceptual designs of complete electrical power systems and associated electrical components suitable for Rankine nuclear electric space power systems from 0.25 to 10.0 megawatts. Three representative types of missions were considered so that design study parameters would reflect typical actual flight conditions. The types of missions considered were interplanetary, planetary, and orbital.

The design study was carried out in two successive stages:

- 1) Analytical parametric screening studies of various system electrical components to establish a spectrum of component design data for optimum system definition, optimum component design identification, and definition of design limitations for four system power levels - 0.25, 1.0, 5.0, and 10 Mwe.
- 2) Detailed design studies of selected component configurations to establish typical component conceptual designs for each system power level and to define development problems associated with each. The system electrical components studied were:
 - a) A-C generators
 - b) A-C motors
 - c) Frequency changers
 - d) Power conditioning equipment
 - e) Power transmission and control equipment
 - f) Electrical system and control protection equipment

The results of the studies will be applicable to (1) definition of the component development effort necessary to realize mission capabilities when defined and (2) identification and specification of the optimum electrical configuration for specific missions by the electrical system designer and nuclear system planner.

This report describes the a-c motor parametric study. Parametric design data was obtained for 5, 30, 60, 100, and 250 hp three phase induction motors. The data includes motor electrical weight, electrical weight plus system weight penalty, efficiency, and power factor as a function of frequency, synchronous speed*, voltage, and

* Hereafter, speed and RPM will denote motor synchronous speed unless otherwise stated.

coolant temperature over the following ranges:

<u>Variable</u>	<u>Range</u>
Frequency (Hz)	200 to 2000
Synchronous Speed (RPM)	4800 to 24,000
Voltage (L-N)	120 to 1000
Coolant Temperature (°F)	500, 800, 1100

A-C motors may have two applications where a nuclear electric space power plant is used. One application is the prime mover for centrifugal circulating pumps in the various basic powerplant loops. The other possible application is in power-using equipment. This parametric study was limited to consideration of canned motors for the centrifugal circulating pumps.

The a-c motor parametric data will serve three useful purposes:

- 1) Motor data can be combined with centrifugal pump data to define the characteristics of canned motor-pump packages. These packages can then be compared to electromagnetic pumps and turbine-driven pumps to assess the relative applicability and technological problems of each.
- 2) The motor parametric data will be helpful in defining conceptual powerplants using centrifugal pumps driven by motors.
- 3) Motor parametric data and design analyses will identify motor problems that will require advanced technology development to make the power plant a reality.

Westinghouse Aerospace Electrical Division has previously performed two limited parametric design studies (1), (2), (3)* on canned motors to drive centrifugal circulation pumps for nuclear electric power systems. One study (1) consisted of frequency and voltage parametric studies for two specific motor designs - 60 hp, 10,000 RPM and 30 hp, 15,000 RPM. The result of this parametric study gave motor weight as a function of frequencies from 400 to 3200 Hz and voltages from 100 to 1000 volts line-to-line (L-L) at an operating temperature of 1000°F. Effect of operating temperatures from 600°F to 1200°F on motor output for frequencies in the specified range was included. The other study (2), (3) consisted of a parametric frequency study for various 12,000 RPM, 200 volt L-L motors from 5 to 25 hp at 650°F average coolant temperature. Motor electrical weight, efficiency, power factor, and operating temperatures as a function of frequency from 400 to 1000 Hz was determined.

The scope of this study encompasses the range of data covered by previous studies and extends the range to the higher horsepower

* Superscripts refer to References, Section V.

ratings which conceivably might be required for powerplants in the neighborhood of five to ten megawatts. This study fills in gaps in the data not included in previous studies. All data, including data for points included in previous studies, is produced on a common basis to facilitate comparative evaluation.

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2. BASIS FOR THE PARAMETRIC STUDY

Initial efforts were directed toward establishing a basis for the parametric study. The basis provided a foundation and a plan for execution of the parametric study. Parametric study procedures and limitations, motor design guidelines, significant study parameters, and possible motor design problem areas were defined. The basis for the parametric study was formed from evaluation of the following information.

2.1 PRELIMINARY STUDIES

2.1.1 Review of Background Information

A review of previous parametric or design studies (1), (2), (3), (4), (5) on high temperature, liquid-alkali-metal-cooled, a-c induction motors was conducted to ascertain applicability or usefulness of information contained in those reports to the parametric screening study. The information was found to be useful for:

- a) Providing motor design guidelines for high temperature motors operating in unusual or hostile environments of the type pertinent to this design study and eliminating much preliminary "trial and error" design work which would have otherwise been necessary. (Some of these guidelines are discussed in Appendix C.)
- b) Determining procedures for generating parametric data in conjunction with item a) above. (The parametric study plan of approach is outlined in paragraph 3.1.)
- c) Determining significant parameters for the parametric study. (These parameters are defined in paragraph 2.2.2.)
- d) Anticipating probable problem areas (e.g., interface temperature rise, outgassing of certain motor components, unique material and fabrication requirements) so that appropriate allowances could be made for these problem areas in generating the parametric data and so that their effects would be reflected in the parametric study results.
- e) Helping to choose materials for the various motor components for purposes of the parametric study. (These materials are discussed in paragraph 2.3.)

It was found that motor performance data produced by these previous studies could only be used as a comparative check on data produced by the present study rather than as a direct and integral part of the data. The reasons for this were:

- a) Advances in materials development(6),(7),(8) for application to high temperature machinery since the previous studies plus preliminary results of current work* indicated improved material choices for some motor components. High temperature properties of materials significantly affect motor performance; therefore, motor performance points determined using different materials properties could not be used in direct conjunction with each other.
- b) Certain design requirements which affected motor performance (e.g., shaft diameter, rotor laminated stack inside diameter, soak condition, etc.) applicable to the former studies were no longer applicable to the present parametric study.
- c) The design procedures in the former studies emphasized producing highly optimized designs for specific SNAP-50/SPUR applications. It was not the intent in the present parametric study to produce truly optimized motor designs, but to generate motor performance data on representative motor designs over a larger range of power.

2.1.2 Preliminary Parametric Designs

After the review of previous studies was completed, approximately 140 preliminary parametric motor designs were made using a digital computer motor design program independently developed for use in previous studies. These designs represented a cross-section of the anticipated design requirements for the subsequent parametric study, and were used to (1) identify desirable alterations to the computer design program, (2) help to establish certain motor design parameter limitations, such as flux densities, current densities, and temperatures for the various design requirements, and (3) help choose motor component materials.

2.2 MOTOR DESIGN CONSIDERATIONS

Application of a-c induction motors to the nuclear electric powerplant has required extension of conventional motor design technology to elevated temperature regions. Closer attention must be given to the interdependences between electrical design, mechanical design, and materials considerations than in conventional motor design. Electrical design is more sensitive to mechanical design considerations, such as fabrication techniques, and materials considerations affect both electrical and mechanical design by imposing more design limitations than usual. This is because of elevated temperature, alkali metal compatibility, and low weight, high-performance requirements imposed on both electromagnetic and structural components. Design allowances must be made for these unique design considerations.

* Materials Development under NASA Contract NAS3-6465 and a concurrent WAED independently funded in-house program.

The effect of these allowances is illustrated by the discussion of the selected motor design configuration, materials, and motor design parameters contained in the following paragraphs.

2.2.1 Basic Concept

The a-c induction motor design concept was used as a basis for performing the parametric study. The basic design utilizes a three-phase, wye-connected stator winding and a squirrel cage rotor. The stator and rotor electromagnetic components are protected from the effects of contact with alkali metal in the liquid or vapor phases by hermetically sealed assemblies which incorporate a ceramic bore seal for the stator and a refractory metal cladding system around the rotor. Figure 1 is a rough sketch of the longitudinal cross-section of a typical motor of this type.

The stator assembly consists of a primary winding and a laminated stack supported in a hermetically sealed cavity formed by the stator frame, end bells, bore seal, and bore seal end extensions. The heat exchanger frame (or outer half of the motor frame) and motor stator frame jointly form the stator heat exchanger.

The primary windings (stator conductors) are electrically insulated with high temperature resistant inorganic insulation. The stator laminations are electrically insulated by interlaminar insulation to reduce eddy-current losses.

The bore seal, with its end extensions, forms the final sealing element for the stator assembly. The bore seal itself is made from a refractory ceramic composition for high electrical resistance and for alkali metal compatibility.

The rotor assembly consists of the electromagnetic components (laminations, conductor bars, and end resistance rings) surrounded by a cladding system to form a hermetically sealed cavity. The cladding system, which includes the rotor shaft, end plates and rotor can, is required to protect the rotor electromagnetic components from the alkali metal vapor in the rotor cavity. The electrical losses of a metal "can" in the magnetic air gap will not be detrimental provided contact is maintained with the rotor laminations. The rotor conductor bars and resistance end rings require high mechanical strength to withstand the rotational stresses at elevated temperature.

The bearings are of the sleeve type suitable for operation with liquid alkali metal as lubricant and coolant.

The seals are expected to be a hydrodynamic, non-contacting type suitable for operation with alkali metal where the fluid being sealed forms its own barrier. These seals are required to exclude liquid metal from the air gap during operation.

The shapes of the stator wire and rotor conductor bar and the stator slot configuration are different than those found in conventional motor design.

Rectangular wire, called strap, was chosen for the stator conductor instead of the more conventional round wire. Previous studies(2),(3) conducted by Westinghouse Aerospace Electrical Division determined that a wire-wound motor for this application would be impractical. The reasons for this were:

Figure 1. AC Motor Basic Design Concept

- a) Greater wire-wound stator resistance from less efficient slot use; i.e., inability to completely fill the available slot winding space with conductor because of the circular cross-section of the wire. The greater stator resistance results in larger stator losses.
- b) Higher resistance to heat transfer within the slot because of less equivalent thermal conductivity and less cross-sectional area available for heat transfer from the conductors to iron. This results in higher slot temperatures for a given coolant temperature.

The combination of factors above causes losses and temperatures to be of impractical proportions for the wire-wound designs.

Normally, the wire-wound stator would have an offsetting advantage. Because it is possible to use partially closed slots with a wire-wound motor, it is possible to make the teeth thinner without increasing the slot opening. Thus, the teeth can be thinned to the point where they are worked at the maximum magnetic density determined by the steel properties. More winding area can be obtained in this manner. Thinning the teeth of the strap-wound motor results in wide slot openings because a strap to fit the wider slot must be inserted through the slot opening. The wide slot opening causes the flux to bunch under the teeth and increase the effective length of the air gap. However, there are two factors which reduce the significance of this advantage for wire-wound designs:

- a) The motor designs being contemplated for this application have long air gaps compared to normal standards because space must be allowed for the bore seal and rotor can as well as for running clearance. The long air gap tends to reduce the detrimental effect of wider slot openings required by strap designs.
- b) A slot configuration different than the conventional slot and similar to that employed in the generator will be employed to partially close the slot while still using strap. This slot configuration with the stator rectangular wire is illustrated in figure 2.

The better utilization of winding space and the better heat transfer for the strap-wound motor more than offsets the advantages of the wire-wound motor. Therefore, it was concluded that the motor should be strap wound.

Parallel sided teeth in the rotor laminations result in a trapezoidal slot shape. The rotor conductors have a trapezoidal shape to fully utilize the available conductor area. Trapezoidal conductor bars decrease the electrical resistance at high operating temperatures by providing maximum conductor cross-sectional area. This maintains high operating speed (decreases slip). The rotor slot configuration with trapezoidal bar is illustrated in figure 3.

Figure 2. Stator Slot Configuration

Figure 3. Rotor Slot Configuration

The air gap of the motor is extremely wide by ordinary standards because of the presence of the ceramic stator bore seal and the rotor cladding in the electrical "air" gap. The wide air gap results in high exciting current and low power factor. The high temperatures cause the stator winding resistance to be higher than normal; this high resistance in combination with the high exciting current results in increased no-load copper losses.

The inorganic insulation used in the stator slots and on the stator conductors requires much more space than conventional insulation; therefore, the available stator conductor area is less than in a conventional motor. For this reason, the ratio of steel area to copper area is larger than it would be in a conventional design.

The rotor cladding acts like another rotor cage and contributes some useful torque. However, ripples in the flux field caused by the slot openings generate high frequency losses in the cladding of such a magnitude that they cannot be neglected.

The motor cooling arrangement assumed for the parametric study is the arrangement conceived during past studies (1), (2), (3), (4), (5) performed by the Westinghouse Aerospace Electrical Division. Essentially, it consists of separate heat sinks for the stator and rotor assemblies formed by the stator heat exchanger and the interior of the motor shaft, respectively. All heat generated in the stator is assumed removed at the stator heat exchanger by the stator coolant. This includes heat generated in the winding end extensions, which is assumed to flow entirely into the stator stack and then into the stator coolant. Likewise, all heat generated in the rotor is assumed removed at the shaft inside diameter by the rotor coolant. In other words, for purposes of the parametric study, it is assumed that there is no net heat exchange between the stator and rotor assemblies by conduction, convection, or radiation. The nature of heat transfer within the motor based on this cooling scheme is discussed in the following paragraphs.

Stator losses occur in the conductors, the stator iron, and the stator bore seal. The major portion of the stator losses occur in the conductors. This heat is generated uniformly throughout the conductor lengths including the end turns. Iron losses occur in both the teeth and the depth-behind-slot (DBS) of the punching. It is assumed that the iron losses do not vary from punching to punching so that heat generation in the iron is uniformly distributed axially along the stack.

The rotating electrical field cutting the stator bore seal can produce losses in the bore seal, depending on the material selected. The high electrical resistance of the bore seal material - a ceramic composition - keeps these losses to a negligible level.

The heat transfer paths within the stator are axial, circumferential, and radial as shown on figure 4. All the heat generated in the winding end turns is assumed conducted axially inward to the stack along each conductor since there is no provision for removing the heat directly to the stator frame other than by radiation. For purposes of the parametric study the net heat exchange between the winding end extensions and the stator frame and end bells was neglected.

Figure 4. Stator Heat Transfer Paths

The additional heat from the end turns causes a temperature rise in the punchings at the outer portions. Therefore, there is axial heat flow from punching to punching through the interlaminar insulation toward the middle of the stack.

Heat from the top coil in the slot flows primarily in a circumferential direction through the wire insulation and the slot liner to the teeth on either side of the coil, thence radially outward along the teeth to the DBS. The heat from the bottom coil flows both circumferentially through wire insulation and slot liner to the teeth on either side and radially outward through wire insulation and slot liner to the DBS. Heat from the teeth loss flows radially outward through the punching-frame interface and frame to the coolant.

The rotor losses occur in the conductor bars, end rings, at the tips of the punching teeth, and in the rotor can. The major portion of the rotor losses occurs in the conductor bars where heat is uniformly generated. The rotor can losses may be assumed to be distributed uniformly over the entire can. Heat transfer paths within the rotor are circumferential, radial, and axial as shown in figure 5.

Heat from the conductors flows circumferentially through an oxide coating on the conductor bar surface and through the conductor bar-rotor tooth interface to the rotor teeth. Heat from the rotor can combines with this heat to flow radially inward to the rotor coolant through the punching-shaft interface and the shaft itself. Heat from the end rings flows inward to the conductor bars or into the shaft depending on whether the end ring inside diameter contacts the shaft. This is determined by permissible geometrical proportions of the ring that yield the correct volume required by electrical design considerations.

An aspect of heat transfer of primary concern is the temperature drop or resistance to heat flow across a joint or interface.

In passing from the stator conductors to the stator iron, heat must flow through three interfaces: (1) conductor-wire insulation interface, (2) wire insulation-slot liner interface, and (3) slot liner-tooth interface. Westinghouse Aerospace Electrical Division experience has been that temperature drop across the conductor-wire insulation interface is insignificant; however, the others are usually significant.

Interface thermal resistance is again of concern in cooling the rotor. Here there are four interfaces to be considered. These interfaces are (1) the conductor bar-tooth interface, (2) the rotor can-punching interface, (3) the punching-shaft interface, and (4) the resistance end ring-shaft interface, if present. The punching-shaft interface is the most critical; calculations (4), (5) show that the conductor bar-tooth interface is not a problem. Sufficient contact pressure can be maintained between the conductor bar and tooth to provide adequate heat transfer. Provided contact pressure is maintained between the rotor can and punchings (even as low as 40 psi), sufficient heat transfer can be maintained across the rotor can-punching interface.

Figure 5. Rotor Heat Transfer Paths

Environment must be considered when evaluating heat transfer across an interface. For instance, under hard vacuum, it is difficult to determine the exact amount of heat transfer between surfaces forming a mechanical joint. It is probably valid to conclude conditions will be worse than when the joint is surrounded by some gas such as air, argon, nitrogen, etc. In this regard, it is important to know the conditions that exist within the sealed stator and rotor assemblies. There are several possibilities:

(a) Stator Cavity Open to Space Vacuum. Outgassing at elevated temperatures of slot components in the confined spaces of the stator slot may cause the pressure to build up in the voids and crevices to a value which produces minimum electrical strength and a phenomenon called electrical discharge corona occurs. Even though the outgassing products may promote heat transfer, conditions must be maintained such that corona effects are minimized and breakdown is prevented.

(b) Stator and Rotor Cavity Evacuated and Hermetically Sealed. In this instance the danger of rupture of component cladding with pressure buildup from outgassing exists unless the unit is designed to maintain a pressure differential.

(c) Preliminary Outgassing of Stator and Rotor Components, Evacuation, Backfilling with Inert Gas, and Hermetically Sealing. This introduces unknowns relative to fabrication techniques and the possibility of the problem described in (2) occurring cannot be totally eliminated. It is also undesirable from a systems standpoint since the non-condensable gas could eventually permeate into the turbine condenser system.

For purposes of the parametric study, it was assumed that the stator and rotor cavity are hermetically sealed and that solutions to avoid corona pressures or excessive outgassing pressure build-up have been found.

The squirrel cage, a-c induction motor is attractive for service as the prime mover of centrifugal pumps for nuclear electric powerplants for the following reasons.

- a) Performance characteristics (speed-torque relationships) closely match the speed-torque requirements of centrifugal pumps where constant pump delivery is a primary objective.
- b) The basically uncomplicated electromechanical configuration makes it easier to provide protective cladding for the electromagnetic components, and the absence of electrical commutating devices such as slip rings, commutators, brushes, or electronic switching devices vulnerable to hostile environments makes this motor design inherently more reliable for long unattended life.

2.2.2 Motor Design Parameters

Motor design parameters may be divided into three general categories:

- a) The design parameters for which parametric relationships have the most significance to the overall nuclear power-plant system definition. Also, these parameters must facilitate comparison of motor-driven centrifugal pump packages to electromagnetic pumps or turbine-driven centrifugal pump packages. These design parameters will be referred to as system design parameters in subsequent portions of this report.
- b) The design parameters that must be adjusted in the design process to obtain desired motor performance for specific combinations of system design parameters. These design parameters will be referred to as variable internal design parameters.
- c) The design parameters for which constant values must be assigned throughout the parametric study, or known relationships to other parameters established. These parameters may take on constant values according to or be known functions of geometry, materials, temperature, voltage, or speed. These properties will be referred to as invariant internal design parameters.

(a) System Design Variables. The requirements and limitations imposed on the motors by the plant prime mover system and associated equipment were included in the selection of design conditions. Particular emphasis was placed on determining the motor requirements relative to the following system considerations.

- 1) System/component weight penalty per kilowatt of loss.
- 2) Load transients and prime mover response rate.
- 3) System pumping requirements.
- 4) Radiation levels.
- 5) Radiator penalties and low temperature cooling limitations.
- 6) Weight and volume limitations.

The three motor characteristics that usually have the most influence on system design are weight, efficiency, and power factor. These characteristics are functions of motor design parameters; therefore, parametric relationships between these motor characteristics were derived by varying certain selected motor design parameters.

The basic motor design parameters for which parametric relationships have the most significance to system design are listed in table 1. These are the system design parameters.

These particular parameters were chosen because (1) they comply with the design specifications, (2) they coincide with design parameters selected for parametric design studies of other system electrical components (e.g., generator, frequency changer), and (3) they are natural, basic input items to the motor design computer program.

Table 1. System Design Parameters

Parameters	Range
Rating (horsepower)	5,30,60,100,250
Frequency (Hertz)	200,400,600,800,1000,2000
Speed (RPM)	4800 to 24,000
Voltage (L-N)	120,240,480,1000
Coolant Temperature (°F)	500,800,1100

Intermediate speed points between 4800 and 24,000 RPM are not shown because they depend on specific combinations of integral number of poles and frequency; therefore, the intermediate speed points examined varied according to the frequency used. Table 2 shows speeds for the various possible combinations of frequency and number of poles.

An additional motor characteristic that is significant to system design is motor power losses. These losses appear as waste heat which must be accommodated by the powerplant's waste heat rejection loops. Power loss is a function of motor design parameters; therefore, a parametric relationship with weight, efficiency, and power factor as a function of the system design parameters could have been determined. However, since the output power (rating) is known, efficiency can serve as a measure of the power losses. For this reason, power loss data was not explicitly presented.

(b) Variable Internal Design Parameters. The variable internal design parameters that affect the motor design the most are

- 1) Stator stack outside diameter
- 2) Stack length (stator and rotor)
- 3) Number of slots (stator and rotor)

Table 2. Possible AC Motor Synchronous Speeds for Various Combinations of Frequency and Number of Poles

Poles Freq.	2	4	6	8	10	12	14	16	18	20	22
200	12000	6000	4000								
400	24000	12000	8000	6000	4800						
600	36000	18000	12000	9000	7200	6000					
800	48000	24000	16000	12000	9600	8000	6857				
1000	60000	30000	20000	15000	12000	10000	8571	7500	6667	6000	
2000			40000	30000	24000	12000	17143	15000	13333	12000	10000

- 4) Pitch or coil throw
- 5) Turns per coil
- 6) Stator stack inside diameter
- 7) Ratio of iron to copper (stator and rotor)

A discussion of their effect on motor design and on other parameters is contained in Appendix B.

(c) Invariant Internal Design Parameters. Members of this group of design parameters which have the most effect on motor design and performance are:

- 1) Maximum torque ratio (ratio of maximum torque to rated torque)
- 2) Stator winding current density
- 3) Flux densities (stator and rotor)
- 4) Rotor Resistance
- 5) Air gap length (including bore seal and rotor can thicknesses)
- 6) Stator slot component insulation dimensions
- 7) Maximum rotor punching inside diameter/shaft diameter
- 8) Minimum bending radius of stator winding hair pin end extensions
- 9) Windage
- 10) Bearing friction

These parameters are defined and the basis of selection of their values or relationships to other design parameters for the parametric study are discussed in Appendix C.

2.3 MATERIALS

2.3.1 Materials Selection

Careful attention must be given to selection of motor materials not only because of the effect of their contribution to determining

practical motor design performance but also because materials limitations most often determine the upper limit of practical design feasibility. Table 3 is a summary of the materials selection for the parametric study listing the basic motor component, component material choice (different materials selections for different temperature ranges where applicable), and a short statement of reason for choice. An effort was made to use materials consistent with those used in the generator for similar applications wherever possible.

There are numerous alloys which could have been selected for the motor structural components - depending upon individual combinations of temperature alkali metal coolant involved, and relative thermal expansion characteristics. One design consideration that is different from the generator is that the motors will be required to operate in all the power plant system loops; therefore, consideration of more than one alkali metal coolant must be given in the choice of motor materials. This could be done either by selecting a satisfactory material for all contemplated coolants or by selecting different materials for each coolant. Usually, a tradeoff is involved between obtaining the best overall combination of properties and obtaining compatibility with the largest number of different alkali metals. For purposes of the parametric study, one material was selected for all coolant designs to (1) reduce the number of design variations between parametric designs and (2) to simplify handling material property data in the analysis. It was assumed the probable coolants involved would be lithium, potassium, and eutectic sodium-potassium (NaK) based on heat transfer media used in the SNAP-50/SPUR power plant and others^{(9),(10)}. Consideration of a single material with long term resistance to all three alkali metals restricted the material choice to a refractory metal because of lithium compatibility requirements. Of the three refractory metal alloys considered - columbium, tantalum, and molybdenum - tantalum T-111 was chosen because it: (1) exhibits mechanical properties at elevated temperature as good as the other candidate materials including good creep strength favorable for long life service, (2) shows promise of requiring no post-weld heat treatment which permits less complicated canned motor fabrication techniques, and (3) is being used more extensively in current liquid metal technology programs which may indicate extensive use throughout future powerplant systems. In this respect, fabricating the motor using tantalum T-111 would tend to prevent undesirable bi-metallic systems.

Table 3. AC Motor Component Material Selection for Parametric Design Study

Component	Material		
	500°F Coolant	800°F Coolant	1100°F Coolant
1. Stator Frame	<p>Tantalum T-111 Basis:</p> <p>Long term compatibility with system alkali metal coolants (Li, K, NaK) contemplated in the past (coolants for present systems have not been defined) together with good mechanical properties at elevated temperature.</p>	<p>Tantalum T-111 Basis:</p> <p>Same as 500°F</p>	<p>Tantalum T-111 Basis:</p> <p>Same as 500°F</p>
2. Stator Laminations	<p>Hiperco 27 Basis:</p> <p>Based on results of work presented in Reference 6.</p>	<p>Hiperco 27 Basis:</p> <p>Same as 500°F</p>	<p>Hiperco 27 Basis:</p> <p>Same as 500°F</p>
3. Stator Interlaminar Insulation	<p>Aluminum Orthophosphate Basis:</p> <p>Based on results of work presented in Reference 7.</p>	<p>Plasma-arc Sprayed Alumina Basis:</p> <p>Based on results of work presented in Reference 7. Better stability and minimum outgassing at the higher temperature levels.</p>	<p>Plasma-arc Sprayed Alumina Basis:</p> <p>Same as 800°F</p>
4. Stator Conductor	<p>Inconel 600 Clad Silver Basis:</p> <p>Based on results of work presented in Reference 7.</p>	<p>Inconel 600 Clad Silver < 1400°F DS Copper (Cube) > 1400°F Basis:</p> <p>Based on results of work presented in Reference 7. DS Copper required for metallurgical stability at temperatures > 1400°F</p>	<p>Inconel 600 Clad Silver < 1400°F DS Copper (Cube) > 1400°F Basis:</p> <p>Same as 800°F</p>
5. Stator Conductor Insulation	<p>Anadur-refractory oxide and glass frit over E-glass fiber serving with silicone resin bonding. Basis:</p> <p>Based on results of work presented in Reference 7.</p>	<p>Anadur < 1150°F Plasma-arc Sprayed Alumina on preformed coil > 1150°F Basis:</p> <p>Based on results of work presented in Reference 7. Plasma-arc sprayed alumina required at temperature > 1150°F for long term stability.</p>	<p>Anadur < 1150°F Plasma-arc Sprayed Alumina on preformed coil > 1150°F Basis:</p> <p>Same as 800°F</p>

Table 3. AC Motor Component Material Selection for Parametric Design Study - Continued

Component	Material		
	500°F Coolant	800°F Coolant	1100°F Coolant
6. Stator Slot and Phase Insulation	Alumina, 98.5% Purity Basis: Based on results of work presented in Reference 7.	Alumina, 98.5% Purity Basis: Same as 500°F	Alumina, 98.5% Purity Basis: Same as 500°F
7. Stator Slot Wedges	Alumina, 98.5% Purity Basis: Based on results of work presented in Reference 7.	Alumina, 98.5% Purity Basis: Same as 500°F	Alumina, 98.5% Purity Basis: Same as 500°F
8. Stator Bore Seal	Alumina, 99% Pure Basis: Based on results of work presented in Reference 8. Offers alkali metal compatibility with high electrical resistance.	Alumina, 99% Pure Basis: Same as 500°F	Beryllia, 99.8% purity, high density Basis: Based on results of work presented in Reference 8. Required for long term, alkali metal compatibility at higher temperature levels.
9. Rotor Can	Tantalum T-111 Basis: Good mechanical properties at elevated temperatures combined with compatibility with all probable alkali metal coolants.	Tantalum T-111 Basis: Same as 500°F	Tantalum T-111 Basis: Same as 500°F
10. Rotor Conductor Bar	DS Copper (Cube) Basis: Based on results of work presented in Reference 7. Good mechanical properties and stability at elevated temperatures to resist centrifugal loads in rotor for long life.	DS Copper (Cube) Basis: Same as 500°F	DS Copper (Cube) Basis: Same as 500°F
11. Rotor Resistance End Ring	Cube Basis: Based on results of work presented in Reference 7. Good mechanical properties and stability at elevated temperatures to resist centrifugal loads in rotor for long life.	Cube Basis: Same as 500°F	Cube Basis: Same as 500°F

Table 3. AC Motor Component Material Selection for Parametric Design Study - Concluded

Component	Material		
	500°F Coolant	800°F Coolant	1100°F Coolant
12. Rotor Laminations	Hiperco 27 Basis: Based on results of work presented in Reference 6.	Nivco Basis: Based on results of work presented in Reference 6. Required for better strength and creep strength at higher temperature levels.	Nivco Basis: Same as 800°F
13. Rotor Shaft	Tantalum T-111 Basis: Good mechanical properties (including fatigue properties) at elevated temperatures combined with compatibility with all probable alkali metal coolants.	Tantalum T-111 Basis: Same as 500°F	Tantalum T-111 Basis: Same as 500°F

Results of the preliminary motor designs indicated that stator conductor hot spot temperatures, which usually occur in the winding end extension hair-pin turns, might exceed the maximum temperature (1400°F) permissible for long term stability of the Inconel 600-clad silver wire. For those designs, two alternatives were possible:

- (1) Increase the size of the motor or otherwise improve cooling so that the hot spot temperature was reduced to less than 1400°F or,
- (2) substitute dispersion-strengthened (DS) copper for the clad silver conductor. Alternate number (1) might have caused many high temperature designs to appear unfavorable or be eliminated from consideration because of excessive weight. Therefore, it appeared that despite the slightly inferior electrical properties of DS copper, alternative number (2) would be the better of the two. There is one serious drawback to use of DS copper; forming stator winding coils from pre-insulated DS copper does not appear feasible at the present time, and there are currently no techniques developed capable of insulating formed coils of DS copper. However, for purposes of the parametric study, it was considered more reasonable to expect that such techniques could be developed, if required, than to eliminate potential designs due to unfavorable weight considerations.

Finally, it should be pointed out that problems with application techniques of plasma-arc sprayed alumina to conductors will have to be solved before this form of insulation can be considered available for practical use for temperatures above 1150°F.

2.3.2 Materials Properties

Pertinent properties data on the materials contained in table 3 was extracted from the three topical reports (6), (7), (8) published under contract NAS3-4162 and from results of work currently being concluded under contract NAS3-6465.

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3. EXECUTION OF THE PARAMETRIC STUDY

3.1 PROCEDURE

The objective of the motor parametric study was to obtain motor specific weight, efficiency, and power factor as a function of the system design parameters - frequency, speed, and voltage - at each of the coolant temperatures for each of the motor ratings to be studied.

The parametric studies were carried forward in a cyclic process to obtain this parametric data. This process is described in the following paragraphs.

Two of the system design parameters were held invariant while motor designs were obtained for an excursion of the third system design parameter over a range of values previously defined for that parameter. This parameter excursion is called a variable study (e.g., frequency study). The variable studies were performed sequentially, using the most attractive value of the variable parameter from the previous variable study as one of the invariant parameters in the succeeding variable study. Completion of one set of all three variable studies constituted one cycle; thus, a cycle consisted of frequency, speed, and voltage studies. Successive cycle studies were performed until the most attractive value of the variable parameter from each of the variable studies repeated itself in a subsequent cycle. This cyclic process was performed for each motor rating (5, 30, 60, 100, and 250 hp) at each of the study coolant temperatures (500°, 800°, and 1100°F).

Designs which gave the best combination of electrical weight and efficiency were used as a basis for choosing the best frequency, speed, or voltage from each of the variable studies. Where these parameters were opposed (i.e., where a choice between two designs was involved, one with lowest weight and one with highest efficiency), system weight penalties were used to determine the choice. Where two or more designs were so similar as to make a choice difficult, detailed thermal analyses were performed in an attempt to sufficiently amplify inherent differences in efficiencies. Application of the system weight penalty was also used in these situations as an alternative. Curves of system weight penalty as a function of motor efficiency that were used in the parametric study are shown in figures 6, 7, and 8 for 500°, 800°, and 1100°F coolant, respectively.

A comprehensive step-by-step procedure was formulated for obtaining the parametric data according to the cyclic process described in the preceding paragraphs. The procedure steps are summarized below.

- 1) Choose speeds for 5, 30, 60, 100, and 250 hp ratings at coolant temperatures of 500°, 800°, and 1100°F based on preliminary designs.

Figure 6. AC Motor System Weight Penalty for 500°F Coolant

Figure 7. AC Motor System Weight Penalty for 800°F Coolant

Figure 8. AC Motor System Weight Penalty for 1100°F Coolant

- 2) Choose materials, material dimensions, and fixed parameters.
- 3) Perform first-cycle frequency study using the following system design parameters:

Ratings	5, 30, 60, 100, and 250 hp
Volts	120 L-N
RPM	As chosen for each rating in step 1
Frequencies	200, 400, 600, 800, 1000 and 2000 Hz
Coolant Temp.	500°F

Use component temperatures obtained from preliminary designs.

- 4) Find actual ratings of designs from step 3 by assuming rated torque = $1/2.25$ maximum torque.
- 5) Rerun designs of step 3 as required to obtain the desired ratings within ± 20 percent.
- 6) Compare designs from step 3 and 5 in terms of lbs/hp and efficiency to obtain a best frequency for each rating. Perform detailed temperature calculations as necessary.
- 7) Perform first-cycle speed study using the following system design parameters:

Ratings	5, 30, 60, 100, and 250 hp
Volts	120 L-N
Frequencies	As chosen for each rating from step 6
Speeds	Four or five speeds for each rating and temperature corresponding to frequency chosen from step 6 and 2, 4, 6, 8, and 10 poles (24000 rpm maximum)
Coolant Temp.	500°F

- 8) Find actual ratings of designs from step 7 by assuming rated torque = $1/2.25$ maximum torque.
- 9) Rerun designs of step 7 as required to obtain desired ratings within ± 20 percent.
- 10) Compare designs from steps 7 and 9 to choose a best speed for each rating. Perform detailed temperature calculations as necessary.

- 11) Perform first-cycle voltage study using the following system design parameters:

Ratings	5, 30, 60, 100, and 250 hp
Frequencies	As chosen for each rating from step 6
Speed	As chosen for each rating from step 10
Voltage	120, 240, 480, and 1000 volts (L-N)
Coolant Temp.	500°F

- 12) Find actual ratings of designs from step 11 by assuming rated torque = $1/2.25$ maximum torque.
- 13) Rerun designs of step 11 as required to obtain the desired ratings within ± 20 percent.
- 14) Compare designs from steps 11 and 13 in terms of lbs/hp and efficiency to obtain best voltage for each rating. Perform detailed temperature calculations as necessary.
- 15) Repeat steps 3 through 14 from the second and subsequent cycles as necessary, except begin each cycle with values of speed and voltage from steps 10 and 14, respectively, of the preceding cycle. Maintain coolant temperature at 500°F.
- 16) Perform steps 1 through 15 for 800°F and 1100°F.

Figure 9 displays diagrammatically the course of the actual parametric study. The values shown between the blocks for each study represent input parameters which were selected from the previous study and held constant in the subsequent study (except in the second cycle speed studies). The top value is the parameter associated with the best design chosen in the previous study. The lower value represents the remaining parameter held constant in the previous study and carried over as a constant in the subsequent study.

Second cycle voltage studies were not required for any of the 500°, 800°, and 1100°F coolant studies and in some instances, particularly the 500°F coolant designs, it was possible to terminate the second cycle studies with the second cycle frequencies studies.

Examination of figure 9 will reveal why it was possible to delete portions of the second cycle studies from the total parametric study. Whenever a particular second cycle study resulted in a combination of parameters which matched the best parameters from previous studies, there was obviously no advantage in continuing further.

Approximately at the time of initiation of the 800° and 1100°F first-cycle voltage studies, it became apparent that the second-cycle frequency studies must be initiated at once to permit completion of the 800° and 1100°F studies according to schedule. Based on results of the previous studies, the number of frequencies to be studied in the second cycle frequency studies was reduced from six to three (400, 600, and 800 Hz), and it was assumed that the results of the first-cycle voltage studies would indicate 120 volts as the best voltage selection. The 800°F - 5, 30, 60, and 100 hp design and the 1100°F - 5, 30, and 60 hp design second-

Figure 9. AC Motor Parametric Study, 500°F Study

1. 5 HP

2. 30 HP

Figure 9. AC Motor Parametric Study, 500°F Study - Continued

3. 60 HP

4. 100 HP

Figure 9. AC Motor Parametric Study, 500°F Study - Continued

5. 250 HP

Figure 9. AC Motor Parametric Study, 800°F Study - Continued

1. 5 HP

2. 30 HP

Figure 9. AC Motor Parametric Study, 800°F Study - Continued

Figure 9. AC Motor Parametric Study, 800°F Study - Continued

5. 250 HP

Figure 9. AC Motor Parametric Study, 1100°F Study - Continued

1. 5 HP

2. 30 HP

Figure 9. AC Motor Parametric Study, 1100°F Study - Continued

3. 60 HP

4. 100 HP

Figure 9. AC Motor Parametric Study, 1100°F Study - Concluded

5. 250 HP

cycle frequency studies were initiated on this basis in parallel with the first cycle voltage studies. As shown in figure 9, for all but one of the 800°F designs (100 hp) and one of the 1100°F designs (30 hp), the voltage assumptions proved correct. For those designs where the voltage changed, the new voltage was used as an input to the second-cycle speed studies when run. The frequency studies with the incorrect voltages were not rerun, however. This did not affect the resulting trends shown by the second-cycle frequency studies, and the actual magnitudes of the weights and efficiencies could be corrected to new values corresponding to the new voltages using first cycle data. The broken lines in figure 9 indicate where the voltage studies were by-passed so that the frequency studies could be performed in parallel.

Theoretically, the synchronous speed should be held constant at the speed determined by the first-cycle speed studies throughout the second-cycle frequency studies. However, permissible numbers of pole-frequency combinations did not permit matching the speed to input speeds for every frequency. Alternatively, for those frequencies not permitting the exact desired speed, the number of poles were chosen so that the nearest possible speed to the desired speed was obtained. When the difference between a higher speed and the desired speed and the difference between a lower speed and the desired speed were about equal, the higher speed was usually chosen because previous data had, in general, indicated that the higher speeds produced the better designs.

Detailed temperature calculations were run on selected designs throughout the parametric study to act as a guide in determining the most favorable designs. In most instances, thermal instability* or excessively high temperatures were used as justification for eliminating designs from consideration. However, there were exceptions where the results of the computer design thermal program were disregarded. For example, an 800 Hz, 12,000 rpm design was chosen as the best design in the 1100°F, 100 hp first-cycle frequency study. In the first-cycle speed study all 1100°F, 100 hp designs at speeds greater than and including 12,000 rpm were thermally unstable. Because of slight design differences which were inadvertently incorporated in the process of design iteration between the first cycle frequency and the first cycle speed studies, the designs were just different enough to cause one to cool and the other to become thermally unstable. The thermal instability of the speed study 12,000 rpm design indicated by the computer was ignored and 12,000 rpm was chosen as the best speed.

* Thermal instability means that the computer design program failed to reach a steady-state temperature distribution within a reasonable number of design iterations. Additional iterations were not made to limit the amount of detail design calculations made on the parametric screening designs. The designs are still valid with the approximate temperatures calculated.

Another example involved the 1100°F, 250 hp designs. Based on initial design temperature estimates and other internal design parameters held common between all the parametric designs to provide a common basis for comparison, none of the 1100°F, 250 hp first-cycle frequency study designs were thermally stable when detailed temperature calculations were made. Rather than revising the design temperatures and doing the study over, the results of the detailed thermal analysis were ignored and the studies continued on the basis of the best of the designs using the original design temperature estimates. Throughout the 1100°F, 250 hp studies, selected designs were adjusted until they did become thermally stable to determine cooling weight penalties. These designs were generally 20 to 50 pounds heavier, but less than 1.0 pound heavier on a specific weight basis (pounds per horsepower output).

In one case - the 500°F, 250 hp designs - all of the methods described above to choose the designs with best combination of weight and efficiency failed to produce any significant differences between the 400 and 600 Hz designs; therefore, both frequencies are indicated.

3.2 COMPUTATION METHOD

The motor designs were obtained using a Westinghouse Aerospace Electrical Division independently developed digital computer program. The program combines basic, fixed, strap-wound motor design with temperature calculations.

The electrical design portion of the program designs a motor using empirical and other basic formulas which have been developed at Westinghouse Aerospace Electrical Division and found to give good results. The original program has undergone some modification for use in designing motors for unusual applications like the space nuclear electric power plant because of changes in materials and the extreme environments. For instance, the program was changed to give a trapezoidal bar in the rotor, to use an input diameter as the magnetic inside diameter of the rotor, and to use all available area for slot space to obtain low-slip designs. Allowances for insulation were also changed and some of the empirical formulas were modified to give closer agreement with actual conditions because of the large air gap and extreme temperatures. Provisions were added to the program for calculating the losses in a stator and rotor can and the additional torque from the rotor can was included in the output. The effect of partially closed stator slots was also included.

The temperature calculations are in the form of individual subroutines for the stator and rotor which perform steady-state, three-dimensional heat transfer and temperature distribution analyses. These analyses are based on input electrical losses in the form of heat generation rates which are determined in the main electrical design routine. The average temperatures of certain predominant heat generating motor components are determined and fed back to the main electrical design routine where they are used to redetermine

electrical losses and other motor performance aspects.

A more detailed description of the motor design computer program is contained in Appendix D.

As indicated in the step-by-step procedure outlined in section 3.1 on parametric study procedure, the detailed temperature calculation routine was used only at certain discrete times during the parametric study to (a) perform spot checks on and adjust average temperatures assumed for the stator windings, rotor conductor bars, and rotor resistance end rings and to (b) amplify differences in efficiencies between very similar motor designs obtained in some variable studies to permit selection of the best value of that variable for use in the subsequent variable study. The reasons why the detailed temperature calculation subroutines were not used for all of the parametric designs are as follows:

- 1) The temperature calculation subroutine requires considerable specific design information, much of which cannot be defined until subsequent conceptual design analyses are made. Examples of the required information are stator frame thickness, interface or interference fit pressures (stresses), and heat transfer coefficients. If assumptions are used for these figures, the significance of the detailed nature of the temperature calculation is reduced. Estimated average temperatures of the stator windings, rotor conductor bars, and resistance end rings based on a limited amount of sample detailed temperature calculations using the temperature calculation subroutines was considered satisfactory for purposes of the parametric study. The detailed input design information mentioned above was estimated based on information obtained from a previous motor design study(4),(5).
- 2) Reduction of effort and associated elapsed time required to prepare computer input data.

4. RESULTS AND CONCLUSIONS

4.1 RESULTS

The results of the parametric study are graphically displayed in Appendix A, figures A-1 through A-50. Motor specific electrical weight (pounds per output horsepower), specific electrical weight plus weight penalty, efficiency, and power factor are plotted as functions of frequency, speed, and voltage for the three coolant temperatures for each of the five motor ratings studied. A machine plotter was used to plot data points, label axes, and print titles. The dashed curves and shaded areas were done by hand. Table A-1, Appendix A, contains ranges of stator stack outside diameters, stack lengths, and average stator winding temperatures (when calculated) that were determined during the parametric study.

The data in figures A-1 through A-50, Appendix A, should be regarded as broad, general trends instead of specific, final design data for two related reasons:

- a) wide variations of most of the data
- b) lack of opportunity for design refinement

Motor design is essentially a trial and error process even with the aid of a design computer program. It is impractical to program all of the motor design decision-making process into the design computer program because of the nearly infinite number of possible motor design parameter combinations at the beginning of the design process; therefore, a substantial amount of design decision must be left to designer judgement as the design process progresses. In a normal motor design process, the designer may make many design adjustments and refinements to the first rough design which meets basic design requirements before the best possible design is obtained. During this design refinement process unintentional design "accidents" or variations invariably occur because it is difficult to accurately predict all of the effects of the design adjustments. This is because of the complex design parameter interrelationships, some of which cannot be predicted analytically or empirically. These design "accidents" or variations must be eliminated by further design adjustments.

The number of design adjustments of each design in this parametric study was restricted to two or three to keep the magnitude of the task within practical proportions. This amount of design adjustment was not enough to completely eliminate all the effects of design accidents, and these effects cause discontinuous variations in the data. Some of the discontinuous variation in the

design data was unavoidable because of the coarseness of the possible design steps in changing a particular rating at a given coolant temperature from one design point to the next. Variations in the data caused by this type of design adjustment tends to average out in the specific weight plus system weight penalty curves, although all of this variation could not be eliminated.

Additional design refinement would be required to eliminate all of the discontinuous design variations caused by design accidents. Some designs would require more refinement than others, particularly those exhibiting more weight and less efficiency compared to other designs of the same rating and coolant temperature within the same variable study. Some improvement in weight and performance could be expected for most of the designs, especially in the high-weight, low-efficiency designs, but in some cases further refinement would cause increased weight and increased efficiency. The latter effect probably would occur in designs with thermally-stable, steady-state temperatures greater than permitted by insulation systems or component strengths. In this case, current density of the particular design probably would have to be reduced and/or size increased to reduce the temperature to acceptable levels. Lower temperatures and larger size tends to improve efficiency. A measure of the specific weight penalty incurred in the latter type of design adjustment was obtained by performing further design refinement on certain discrete designs selected from designs which were all thermally unstable at a particular coolant temperature within a particular variable study. (An example is the 250 hp, 1100°F coolant, first cycle study designs, figure A-41.)

The wide variations in most of the data made it very difficult to draw definite, smooth curves through the data points. Instead, shaded envelopes were established around the more erratic data points to indicate general trends. Identification of individual coolant temperature trends were rarely attempted for these cases. Where the data appeared reasonable based on theoretical design expectation and judgement, curves were drawn for individual coolant temperatures. These curves are shown as dashed lines (e.g., figure A-1, Efficiency Versus Frequency) to indicate that they should be considered as estimated curves. Again, the primary purpose of these curves is to show trends instead of definite design data.

The trends indicated by the shaded envelopes were established with the aid of some rules of thumb. The general shape of the curves were established from sets of points which appeared reasonable based on theoretical design expectations. Then, with some exceptions, the vertical locations of the shaded envelopes and estimated curves were determined by points of minimum weight, maximum efficiency, and maximum power factor. Therefore, the envelopes and estimated curves presumably include design points of possible attainment, but not necessarily obtained in this parametric study. Lower temperature motors theoretically should

weigh less and be more efficient than higher temperature motors. The lower bounds of the envelopes were established by minimum weight and maximum efficiency 500°F coolant points, and the upper bounds were established by minimum weight and maximum efficiency 1100°F coolant points. If minimum weight 800°F coolant points happened to be less than 500°F coolant minimum weight points because of design variation, the lower bounds of the specific weight envelopes were adjusted slightly below the minimum weight 800°F coolant points because 500°F coolant designs theoretically could be designed lighter. According to the same reasoning, if 800°F coolant maximum efficiency points were greater than maximum efficiency 500°F coolant points, the upper bounds of the efficiency envelopes were adjusted upward until slightly higher than the 800°F coolant maximum efficiency points. Adjustment of envelope bounds between 1100°F coolant points and 800°F coolant points were in the reverse direction. In general, curves for individual coolant temperatures should not cross. However, there were some exceptions where valid design reasons and the data indicated that the curves might cross. For example, in some speed studies the 500°F coolant points were obtained at a frequency less than for the higher coolant temperatures. At the higher speeds the tendency of a low frequency, small-number-of-pole combination design to produce a heavier motor than a higher frequency, greater-number-of-pole combination tended to predominate because end extension weight was a greater percentage of total weight; therefore, the higher temperature coolant designs weighed less than the 500°F coolant designs. In these cases, a dashed curve for the 500°F coolant temperature data is shown through the envelope to indicate the relationship.

Even though there is much discontinuous variation in the data, it might be regarded as more realistic than if continuous data were forced by additional design refinement. This is because any difficulties encountered in making a specific design conform to a continuous curve in a parametric study of this type would also be encountered if the attempt was made to obtain the same specific design for actual manufacture. Design refinement would probably be terminated before the design conformed to a smooth parametric relationship. Furthermore, the individualistic nature of designers might cause wide variations in the same design determined by different designers because of the strong dependence of design adjustment on designer judgement.

The trends of the specific weight, specific weight plus weight penalty, and efficiency curves for the frequency and speed studies, Appendix A, reflect the following opposing effects on weight and efficiency caused by varying the number of poles.

If speed is maintained constant, increasing the number of poles (and frequency) requires more excitation current. More excitation current causes larger no load copper losses. Larger no load copper losses cause lower efficiency and the weight increases because size must be increased to keep current density the same.

This effect is accentuated in the motor designs for this parametric study because a larger air gap is required to accommodate the stator bore seal and rotor can and because of the higher temperatures. (See Appendix C, Invariant Internal Motor Design Parameters.) The effect of no load copper losses on efficiency tends to diminish in importance as rating increases because no load copper losses become a smaller percentage of full load losses. (See Appendix B, Variable Internal Motor Design Parameters.)

On the other hand, for constant speed, the winding end extension length becomes smaller as the number of poles are increased. A smaller winding end extension promotes better efficiency in two ways: (a) the reduced amount of copper reduces copper losses and (b) cooling is facilitated which produces lower temperatures and less copper losses. The smaller end extensions also weigh less. The beneficial effects of smaller end extensions is independent of rating.

The frequency studies (performed at constant speeds) indicate that the best combination of weight and efficiency occur at frequencies between 400 and 800 Hertz for all the ratings studied. The lower end of this frequency range, 400 Hertz, appears the most attractive for the smaller ratings (5, 30 hp); the higher frequencies (600, 800 Hertz) appear the most attractive for the larger ratings (60, 100, 250 hp). Evidently, the excitation current effect is more predominate for the 5 and 30 hp ratings, but approximately at 60 hp, the end extension effect begins to predominate.

The best combination of weight and efficiency for the lower ratings (5, 30 hp) would be expected to occur at progressively higher frequencies for higher coolant temperatures. As stated previously, motor size tends to increase with temperature because of reduced steel densities and other temperature effects. Usually this causes reduced air gap flux densities which reduces the influence of the excitation current effect and causes the end extension effect to become more predominant. The data does not reflect this expected trend probably because of the wide variation in the data.

The 200 Hertz, two-pole designs were difficult to obtain, especially for the larger ratings. Even when designs were obtained, they often proved thermally unacceptable (except for 5 hp designs) because of the relatively large winding end extensions. Therefore, the 200 Hertz, two-pole designs were omitted from further consideration after the first cycle frequency studies.

The speed studies (performed at constant frequencies) show that motor weight generally tends to decrease with increasing speed provided that no limiting effect such as cooling requirements occurs. The reason is the fundamental relationship between motor output horsepower, motor output torque, motor volume, and speed, which is

$$\text{Torque} \sim \frac{\text{Output Horsepower}}{\text{Speed}} \sim \text{Motor Volume}$$

For a given rating, greater speed requires less motor volume (size) to maintain the same output torque. Reduction of motor size by increasing speed is limited by cooling requirements. (See Appendix B.) Heat transfer within a motor is partially governed by the size of heat transfer areas determined by motor size. By reducing motor size these areas can become too small to permit steady-state rejection of heat, and the motor will overheat. The winding end extension effect discussed previously also influences cooling since the end extensions are a major source of motor heat.

Cooling requirement limitations begin to manifest themselves at the 30 hp, 800°F coolant, 400 Hertz, two-pole, 24,000 rpm design. In general, the data indicates that as motor rating and/or coolant temperature is increased maximum speed must be reduced and/or frequency-number of poles must be increased to provide thermally acceptable designs. It can be seen from table A-1, Appendix A, that many of the higher speed designs accepted as thermally stable for purposes of the parametric study actually have steady-state temperatures greater than permitted by winding insulation system life requirements or by other component strength requirements. No attempt was made to adjust these designs to reduce their temperatures to levels compatible with wire insulation or other component strength requirements for reasons stated previously. However, if the designs were adjusted by increasing size and reducing current density the effect might be to reduce the speeds where the lightest and most efficient designs occur.

The points of maximum efficiency would be expected to occur at the highest speeds studied (24,000 rpm) for the lower ratings (5, 30 hp), and gradually move toward lower speeds as ratings increase because of the opposing excitation current-end extension effect. The same trend would be expected in the specific weight plus weight penalty curves because of end extension effects. This trend is not apparent in the speed studies data except for a slight indication that it begins to occur at ratings between 100 and 250 hp. Again, the data cannot be considered conclusive because of the wide variation.

The voltage studies show that increasing the voltage causes weight to increase and efficiency to decrease in the smaller ratings (5 to 30 hp). As rating increases, the rate of change of weight and efficiency with respect to increasing voltage becomes less; there is only a slight weight penalty and efficiency remains the same for larger voltages in the 100 and 250 hp ratings.

Larger voltages impose greater penalties on the lower ratings than on the higher ratings. Higher voltages require more turns in the winding which require more conductors per slot. More than one parallel path is normally not used in the smaller ratings; therefore, the additional number of conductors per slot required by the additional turns is greater for the smaller ratings than for the larger ratings, where more than one parallel path may be employed. Since

stator conductor insulation thickness and slot clearance allowed per conductor remains constant, more room must be provided for the additional insulation and clearance imposed by the additional conductors per slot. Since the additional number of conductors per slot required by higher voltages is more for the smaller ratings than for the larger ratings, the size of the smaller ratings must be increased more to accommodate the additional space required by the additional insulation and clearance.

Impractical 5 hp designs were obtained for all coolant temperatures at voltages greater than 240 volts L-N because additional space required by the additional insulation caused the stator teeth to become too narrow for practical fabrication. Furthermore, since impractical 500°F coolant, 60 hp designs were obtained for voltages greater than 480 volts L-N, it is probable that all 30 hp designs should be limited to 480 volts L-N from practical considerations.

All the frequency and speed studies show that power factor decreases with increasing number of poles. This is because more poles require more excitation mmf.

4.2 CONCLUSIONS

4.2.1 General Trends

The following conclusions may be obtained from the results of this parametric study discussed in the previous paragraphs.

- a) The most attractive frequencies for the ratings and coolant temperatures studied appear to be between 400 and 800 Hertz. The specific frequency selected from this range for a specific design is influenced by the number of poles required, which is determined by the required design speed (see next item). The best designs appear to have between 2 and 8 poles. Low frequency-small number of pole combinations are best for the smaller ratings and lower coolant temperatures (<800°F) where possible; higher frequency-greater number of pole combinations are best for the larger ratings and higher temperatures (>800°F).
- b) The most attractive speeds are the highest speeds consistent with cooling, strength, and pump operating limitations. Where cooling difficulties are encountered, the best alternative appears to be to maintain the same speed and increase the frequency and number of poles to facilitate cooling. When the frequency and number of poles cannot be increased more without exceeding the maximum values specified in item a) above, then the next lowest speed, highest frequency-pole combination for the larger ratings and the next lowest speed, lowest frequency-pole combination for the small ratings are the next best alternatives.

- c) For minimum weight-maximum efficiency designs a voltage of 120 volts (line-to-neutral) appears the best of all voltages studied for all ratings studied. However, there is very little penalty in weight or efficiency incurred by using voltage as high as 1000 volts (line-to-neutral) for the higher ratings (greater than 60 hp).
- d) Designs with the minimum number of poles have the maximum values of power factor.

4.2.2 Recommended System Design Parameters

Table 4 shows recommended system design parameters for each of the ratings studied at each of the three coolant temperatures considered and typical specific electrical weights and efficiencies for each design. The values of frequency, speed, and voltage selected gave the best combination of minimum weight and maximum efficiency based primarily on the specific electrical weight plus weight penalty curves in figures A-1 through A-50, Appendix A, and the conclusions outlined in section 4.2.1. Selection of the parameters was limited to electric motor considerations because of the lack of centrifugal pump parametric data. Retention of a high power factor was not a consideration in selection of the parameters, although selection of the highest possible speeds tends to select designs with higher power factors because high speed designs have the smallest number of poles for a given frequency. The values listed in table 4 should be regarded as recommended values only, not as final, firm, inflexible choices. Among other factors, these selections are contingent upon further design adjustment, thermal analysis, and stress analysis, and upon the specific design requirements.

Motor Rating, HP (kWe)	5 (3.7)			30 (22.4)			60 (44.7)			100 (74.6)			250 (186.4)		
Coolant Temperature (°F)	500	800	1100	500	800	1100	500	800	1100	500	800	1100	500	800	1100
Frequency (Hertz)	400	400	400	400	800	600	800	800	800	800	800	800	600	600	800
Synchronous Speed (rpm)	24000	24000	24000	24000	24000	18000	24000	24000	16000	24000	24000	12000	12000	12000	8000
Voltage (Volts L-N)	120	120	120	120	120	120	120	120	120	120	120	120	120	120	120
Specific Electrical Weight (lbs/hp)	2.06	2.39	2.91	0.92	0.87*	1.33	0.5*	0.55	0.87	0.48*	0.52	0.79	0.54*	0.57	0.88*
Efficiency (Percent)	76.0	72.5	63.1	83.8	74.4*	74.5	81.5*	83.1	77.0	85.0*	84.8	82.0	89.1	89.1	80.8*

* Estimated values

5. REFERENCES

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APPENDIX A
MOTOR PARAMETRIC CURVES AND TABULATED DATA

BLANK

FIRST CYCLE DATA 12000 RPM SYNCHRONOUS SPEED 120 VOLTS L-N

COOLANT TEMPERATURES +500°F Δ800°F □1100°F

FIGURE A-1. 5 HP AC MOTOR FREQUENCY STUDY

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FIRST CYCLE DATA 12000 RPM SYNCHRONOUS SPEED 120 VOLTS L-N

COOLANT TEMPERATURES +500F Δ 800F $\square$ 1100F

FIGURE A-2. 5 HP AC MOTOR FREQUENCY STUDY

FIRST CYCLE DATA FREQUENCY 400/600 HERTZ 120 VOLTS L-N

COOLANT TEMPERATURES + 500°F Δ 800°F □ 1100°F

FIGURE A-3. 5 HP AC MOTOR SPEED STUDY

FIRST CYCLE DATA FREQUENCY 400/600 HERTZ 120 VOLTS L-N

COOLANT TEMPERATURES + 500F ▲ 800F ◻ 1100F

FIGURE A-4. 5 HP AC MOTOR SPEED STUDY

FIRST CYCLE DATA FREQUENCY 400/600 HERTZ 24000/18000 RPM SYNCHRONOUS SPEED

FIGURE A-5. 5 HP AC MOTOR VOLTAGE STUDY

5-9

FIRST CYCLE DATA FREQUENCY 400/600 HERTZ 24000/18000 RPM SYNCHRONOUS SPEED

COOLANT TEMPERATURES + 500°F Δ 800°F □ 1100°F

FIGURE A-6. 5 HP AC MOTOR VOLTAGE STUDY

SECOND CYCLE DATA VARIOUS SYNCHRONOUS SPEEDS 120 VOLTS L-N

COOLANT TEMPERATURES +500F Δ 800F $\square$ 1100F

FIGURE A-7. 5 HP AC MOTOR FREQUENCY STUDY

SECOND CYCLE DATA VARIOUS SYNCHRONOUS SPEEDS 120 VOLTS L-N

COOLANT TEMPERATURES +500F Δ 800F $\square$ 1100F

FIGURE A-8. 5 HP AC MOTOR FREQUENCY STUDY

SECOND CYCLE DATA FREQUENCY 400 HERTZ 120 VOLTS L-N

COOLANT TEMPERATURES +50°F Δ800°F □1100°F

FIGURE A-9. 5 HP AC MOTOR SPEED STUDY

SECOND CYCLE DATA FREQUENCY 400 HERTZ 120 VOLTS L-N

COOLANT TEMPERATURES +500F Δ 800F $\square$ 1100F

FIGURE A-10. 5 HP AC MOTOR SPEED STUDY

FIRST CYCLE DATA 12000 RPM SYNCHRONOUS SPEED 120 VOLTS L-N

FIGURE A-11. 30 HP AC MOTOR FREQUENCY STUDY

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FIRST CYCLE DATA 12000 RPM SYNCHRONOUS SPEED 120 VOLTS L-N

FIGURE A-12. 30 HP AC MOTOR FREQUENCY STUDY

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FIRST CYCLE DATA FREQUENCY 400/600 HERTZ 120 VOLTS L-N

COOLANT TEMPERATURES +500F Δ800F □1100F

FIGURE A-13. 30 HP MOTOR SPEED STUDY

FIRST CYCLE DATA FREQUENCY 400/600 HERTZ 120 VOLTS L-N

COOLANT TEMPERATURES + 500°F Δ 800°F □ 1100°F

FIGURE A-14. 30 HP AC MOTOR SPEED STUDY

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FIRST CYCLE DATA FREQUENCY 400/600 HERTZ 12000/18000 RPM SYNCHRONOUS SPEED

FIGURE A-15. 30 HP AC MOTOR VOLTAGE STUDY

FIRST CYCLE DATA FREQUENCY 400/600 HERTZ 12000/18000 RPM SYNCHRONOUS SPEED

FIGURE A-16. 30 HP AC MOTOR VOLTAGE STUDY

SECOND CYCLE DATA VARIOUS SYNCHRONOUS SPEEDS 120 VOLTS L-N

FIGURE A-17. 30 HP AC MOTOR FREQUENCY STUDY

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SECOND CYCLE DATA VARIOUS SYNCHRONOUS SPEEDS 120 VOLTS L-N

COOLANT TEMPERATURES +500F Δ 800F $\square$ 1100F

FIGURE A-18. 30 HP AC MOTOR FREQUENCY STUDY

SECOND CYCLE DATA FREQUENCY 600 HERTZ 240 VOLTS L-N

COOLANT TEMPERATURES +500F Δ 800F □ 1100F

FIGURE A-19. 30 HP AC MOTOR SPEED STUDY

SECOND CYCLE DATA FREQUENCY 600 HERTZ 240 VOLTS L-N

COOLANT TEMPERATURES + 500F Δ 800F □ 1100F

FIGURE A-20. 30 HP AC MOTOR SPEED STUDY

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FIRST CYCLE DATA 12000 RPM SYNCHRONOUS SPEED 120 VOLTS L-N

FIGURE A-21. 60 HP AC MOTOR FREQUENCY STUDY

75

FIRST CYCLE DATA 12000 RPM SYNCHRONOUS SPEED 120 VOLTS L-N

FIGURE A-22. 60 HP AC MOTOR FREQUENCY STUDY

FIRST CYCLE DATA FREQUENCY 400/600 HERTZ 120 VOLTS L-N

COOLANT TEMPERATURES + 500°F Δ 800°F □ 1100°F

FIGURE A-23. 60 HP ALUMINUM MOTOR SPEED STUDY

77

FIRST CYCLE DATA FREQUENCY 400/600 HERTZ 120 VOLTS L-N

COOLANT TEMPERATURES + 500°F Δ 800°F □ 1100°F

FIGURE A-24. 60 HP AC MOTOR SPEED STUDY

78

FIRST CYCLE DATA FREQUENCY 400/800 HERTZ 12000/24000 RPM SYNCHRONOUS SPEED

COOLANT TEMPERATURES + 500F Δ 800F □ 1100F

FIGURE A-25. 60 HP AC MOTOR VOLTAGE STUDY

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FIRST CYCLE DATA FREQUENCY 400/800 HERTZ 12000/24000 RPM SYNCHRONOUS SPEED

COOLANT TEMPERATURES + 500°F ▲ 800°F □ 1100°F

FIGURE A-26. 60 HP AC MOTOR VOLTAGE STUDY

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SECOND CYCLE DATA VARIOUS SYNCHRONOUS SPEEDS 120 VOLTS L-N

FIGURE A-27. 60 HP AC MOTOR FREQUENCY STUDY

SECOND CYCLE DATA VARIOUS SYNCHRONOUS SPEEDS 120 VOLTS L-N

COOLANT TEMPERATURES + 500F Δ 800F □ 1100F

FIGURE A-28. 60 HP AC MOTOR FREQUENCY STUDY

g2

SECOND CYCLE DATA FREQUENCY 600 HERTZ 240 VOLTS L-N

COOLANT TEMPERATURES + 500F Δ 800F $\square$ 1100F

FIGURE A-29. 60 HP AC MOTOR SPEED STUDY

g^h

SECOND CYCLE DATA FREQUENCY 600 HERTZ 240 VOLTS L-N

COOLANT TEMPERATURES + 500F Δ 800F $\square$ 1100F

FIGURE A-30. 60 HP AC MOTOR SPEED STUDY

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FIRST CYCLE DATA 12000 RPM SYNCHRONOUS SPEED 120 VOLTS L-N

TEMPERATURES + 500F Δ 800F □ 1100F

FIGURE A-31. 100 HP AL MOTOR FREQUENCY STUDY

FIRST CYCLE DATA 12000 RPM SYNCHRONOUS SPEED 120 VOLTS L-N

FIGURE A-32. 100 HP AC MOTOR FREQUENCY STUDY

86

FIRST CYCLE DATA FREQUENCY 400/800 HERTZ 120 VOLTS L-N

COOLANT TEMPERATURES + 500°F Δ 800°F □ 1100°F

FIGURE A-33. 100 HP AL MOTOR SPEED STUDY

87

FIRST CYCLE DATA FREQUENCY 400/800 HERTZ 120 VOLTS L-N

COOLANT TEMPERATURES + 500°F Δ 800°F □ 1100°F

FIGURE A-34. 100 HP AC MOTOR SPEED STUDY

FIRST CYCLE DATA FREQUENCY 400/800 HERTZ 12000/24000 RPM SYNCHRONOUS SPEED

COOLANT TEMPERATURES + 500F Δ 800F □ 1100F

FIGURE A-35. 100 HP AC MOTOR VOLTAGE STUDY

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FIGURE A-36. 100 HP AL MOTOR VOLTAGE STUDY

SECOND CYCLE DATA VARIOUS SYNCHRONOUS SPEEDS 120/240 VOLTS L-N

COOLANT TEMPERATURES + 500°F Δ 800°F □ 1100°F

FIGURE A-37. 100 HP AC MOTOR FREQUENCY STUDY

SECOND CYCLE DATA VARIOUS SYNCHRONOUS SPEEDS 120/240 VOLTS L-N

FIGURE A-38. 100 HP AC MOTOR FREQUENCY STUDY

SECOND CYCLE DATA FREQUENCY 600 HERTZ 240 VOLTS L-N

COOLANT TEMPERATURES + 500F Δ 800F $\square$ 1100F

FIGURE A-39. 100 HP AC MOTOR SPEED STUDY

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SECOND CYCLE DATA FREQUENCY 600 HERTZ 240 VOLTS L-N

(COOLANT TEMPERATURES + 500F Δ 800F □ 1100F)

FIGURE A-40. 100 HP AC MOTOR SPEED STUDY

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FIRST CYCLE DATA 12500 RPM SYNCHRONOUS SPEED 120 VOLTS L-N

FIGURE A-41. 400 HP AC MOTOR FREQUENCY STUDY

FIRST CYCLE DATA 12000 RPM SYNCHRONOUS SPEED 120 VOLTS L-N

COOLANT TEMPERATURES + 500F Δ 800F □ 1100F

FIGURE A-42. 250 HP AC MOTOR FREQUENCY STUDY

FIRST CYCLE DATA FREQUENCY 400/600 HERTZ 120 VOLTS L-N

COOLANT TEMPERATURES + 500°F Δ 800°F □ 1100°F

FIGURE A-43. 200 HP AC MOTOR SPEED STUDY

FIRST CYCLE DATA FREQUENCY 400/600 HERTZ 120 VOLTS L-N

FIGURE A-44. 250 HP AC MOTOR SPEED STUDY

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FIRST CYCLE DATA FREQUENCY 400/600 HERTZ 9000/18000 RPM SYNCHRONOUS SPEED

COOLANT TEMPERATURES + 500°F Δ 800°F □ 1100°F

FIGURE A-45. 250 HP AC MOTOR VOLTAGE STUDY

FIRST CYCLE DATA FREQUENCY 400/600 HERTZ 9000/18000 RPM SYNCHRONOUS SPEED

COOLANT TEMPERATURES + 500°F Δ 800°F □ 1100°F

FIGURE A-46. 250 HP AC MOTOR VOLTAGE STUDY

SECOND CYCLE DATA VARIOUS SYNCHRONOUS SPEEDS 480 VOLTS L-N

COOLANT TEMPERATURES + 500°F Δ 800°F □ 1100°F

FIGURE A-47. 250 HP AC MOTOR FREQUENCY STUDY

SECOND CYCLE DATA VARIOUS SYNCHRONOUS SPEEDS 480 VOLTS L-N

COOLANT TEMPERATURES: + 500°F Δ 800°F □ 1100°F

FIGURE A-48. 250 HP AC MOTOR FREQUENCY STUDY

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SECOND CYCLE DATA FREQUENCY 600 HERTZ 480 VOLTS L-N

FIGURE A-49. 250 HP AC MOTOR SPEED STUDY

SECOND CYCLE DATA FREQUENCY 600 HERTZ 480 VOLTS L-N

COOLANT TEMPERATURES † 500F △ 800F □ 1100F

FIGURE A-50. 250 HP AC MOTOR SPEED STUDY

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Table A-1. AC Motor Dimension and Temperature Data

Rating (hp)	Study	Cycle	Coolant Temp- erature (°F)	Rotor O.D. Range (in.)	Stator O.D. Range (in.)	Stack Length Range (in.)	Average Winding Temp- erature (°F)	Average Rotor Bar Temp- erature (°F)
5	Freq.	1	500	2.15-3.47	4.37-5.25	3.25-5.50	775-992	549-617
5	↓	↓	800	2.11-3.81	4.50-5.25	3.62-5.80	947-1136	879-1106
5	↓	↓	1100	2.71-3.82	4.87-6.20	2.87-5.12	1417-1591	1139-1156
5	Speed	1	500	2.16-3.90	4.50-6.00	2.75-4.37	757-826	529-610
5	↓	↓	800	2.38-4.22	4.50-5.87	2.50-4.37	1108(a)	854(a)
5	↓	↓	1100	2.21-4.37	4.54-6.30	2.50-4.75	1629(a)	1166(a)
5	Volt	1	500	2.43-5.38	4.50-7.50	2.62-3.62	884(b)	535(b)
5	↓	↓	800	2.32-5.18	4.50-7.70	2.37-2.87	1112(c)	860(c)
5	↓	↓	1100	2.34-3.70	4.60-6.25	2.50-3.75	1634(c)	1166(c)
5	Freq.	2	500	1.78-2.33	3.80-4.40	2.37-2.62	887(c)	536(c)
5	↓	↓	800	2.33-3.56	4.30-4.80	2.50-3.62	1112(c)	860(c)
5	↓	↓	1100	2.27-3.26	4.55-4.80	2.50-3.62	1634(c)	1166(c)
5	Speed	2	800	2.49-3.09	4.50-5.60	2.62-3.00	788(c)	536(d)
5	↓	↓	1100	2.73&3.00	4.90&5.10	2.87&3.12	1634(c)	1166(c)
30	Freq.	1	500	3.53-4.95	5.62-6.75	4.75-10.12	889-924	526-599
30	↓	↓	800	4.06-5.41	6.50-7.87	4.25-7.12	1089-1199	924-1015
30	↓	↓	1100	4.31-5.81	6.75-8.75	4.12-6.25	1451-1526	1177-1221
30	Speed	1	500	3.50-6.33	6.00-8.50	3.62-5.87	800-1258	554-600
30	↓	↓	800	4.32-6.43	6.60-8.70	4.25-6.62	1249(d)	892(d)
30	↓	↓	1100	3.27-6.21	7.00-8.75	4.00-6.50	1626-1688	1174-1337
30	Volt	1	500	3.61-4.59	6.12-7.70	4.37-5.12	875(b)	733(b)
30	↓	↓	800	3.98-4.69	6.75-8.12	4.37-5.12	1256(c)	896(c)
30	↓	↓	1100	3.31-5.15	6.60-7.70	3.62-5.12	1688(c)	1337(c)
30	Freq.	2	1100	3.33-4.71	6.50-7.00	3.50-4.75	1688(c)	1337(c)
30	Speed	2	1100	3.74&4.23	6.60&7.50	3.62&4.50	1688(c)	1337(c)
60	Freq.	1	500	4.11-6.75	7.12-10.50	4.50-9.50	726-937	568-583
60	↓	↓	800	5.05-6.67	7.50-9.50	5.87-8.62	1060-1138	900-1018
60	↓	↓	1100	5.03-6.79	7.50-10.00	5.37-8.50	1536(e)	1276(e)

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Table A-1. AC Motor Dimension and Temperature Data (Continued)

Rating (hp)	Study	Cycle	Coolant Temp- erature (°F)	Rotor O.D. Range (in.)	Stator O.D. Range (in.)	Stack Length Range (in.)	Average Winding Temp- erature (°F)	Average Rotor Bar Temp- erature (°F)
60	Speed	1	500	4.38-7.26	7.00-9.25	4.25-7.87	852-1015	588-606
60	↓	↓	800	4.27-6.72	6.50-9.18	4.25-6.87	1348(b)	966(b)
60	↓	↓	1100	3.91-6.72	6.60-9.30	4.12-7.12	1604-1608	1142-1173
60	Volt	1	500	4.65-6.05	7.00-8.50	5.37-6.62	983(b)	704(b)
60	↓	↓	800	4.04-4.72	6.25-7.25	4.37-5.50	1238-1273	910-911
60	↓	↓	1100	4.96-5.19	7.35-8.35	5.12-5.25	1607(c)	1175(c)
60	Freq.	2	800	4.10-4.46	6.40-7.20	4.25-4.50	1346(c)	968(c)
60	↓	↓	1100	4.20-5.03	7.00-8.00	4.37-5.37	1698(e)	1166(e)
60	Speed	2	1100	4.24&5.15	7.15&7.65	4.25&5.75	1742(c)	1202(c)
100	Freq.	1	500	5.50-8.09	8.25-11.50	5.12-10.25	1112(c)	635(c)
100	↓	↓	800	5.99-8.77	8.87-11.50	5.62-8.75	1122-1231	942-980
100	↓	↓	1100	6.23-8.51	8.75-11.40	6.12-8.87	1617(e)	1247(e)
100	Speed	1	500	4.71-8.38	7.75-10.50	5.00-8.75	881-1129	594-688
100	↓	↓	800	4.62-7.58	7.40-10.35	4.75-8.00	1309-1382	881-1037
100	↓	↓	1100	4.35-7.70	7.65-10.50	4.50-8.12	1616(c)	1247(c)
100	Volt	1	500	5.08-5.48	8.25-8.50	5.25-7.12	1135-1195	832-872
100	↓	↓	800	4.51-4.85	7.40-7.65	4.62-6.00	1266-1380	886-994
100	↓	↓	1100	6.12-6.22	9.00-9.40	6.37-6.87	1616(c)	1247(c)
100	Freq.	2	800	4.53-5.70	7.30-8.30	4.75-5.87	1382(c)	1040(c)
100	↓	↓	1100	5.63-7.19	8.75-9.15	5.62-7.37	1615(c)	1247(c)
100	Speed	2	1100	6.11&6.25	8.10&8.05	7.12&6.50	1634(c)	1247(c)
250	Freq.	1	500	6.74-11.50	10.50-15.00	6.87-10.50	723-1132	539-810
250	↓	↓	800	7.05-10.41	10.50-13.75	7.62-9.87	1427(c)	1031(c)
250	↓	↓	1100	7.56-10.89	10.50-15.00	8.12-11.12	1814(c)	1544(c)
250	Speed	1	500	6.42-10.26	10.00-12.75	6.27-10.87	1065-1189	613-1427
250	↓	↓	800	7.08-10.54	10.00-13.20	7.25-10.87	1323-1429	849-894
250	↓	↓	1100	7.17-10.78	10.15-13.60	7.12-11.00	1832(c)	1292(c)

Table A-1. AC Motor Dimension and Temperature Data (Concluded)

Rating (hp)	Study	Cycle	Coolant Temp- erature (°F)	Rotor O.D. Range (in.)	Stator O.D. Range (in.)	Stack Length Range (in.)	Average Winding Temp- erature (°F)	Average Rotor Bar Temp- erature (°F)
250	Volt	1	500	7.77-7.98	10.50-10.75	9.37-9.75	1100-1170	744-752
250	↓	↓	800	8.22-8.55	11.50-11.60	8.62-9.75	1251-1255	870-877
250	↓	↓	1100	6.70-6.97	9.80-10.30	6.87-10.50	1832 (c)	1292 (c)
250	Freq.	2	500	7.73-10.06	10.50-12.25	9.00-10.37	1086-1119	543-554
250	↓	↓	800	8.45-9.53	11.45-12.15	8.62-10.25	1256 (c)	878 (c)
250	↓	↓	1100	6.48-8.15	9.70-11.30	6.87-8.12	1832 (c)	1292 (c)
250	Speed	2	1100	6.72&8.12	9.85&11.00	7.125&8.62	1832 (c)	1292 (c)
(a) Calculated at 18,000 RPM only (b) Calculated at 120 Volts only (c) Indicates estimated temperature values (d) Calculated at 12,000 RPM only (e) Calculated at 800 Hz only								

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APPENDIX B

VARIABLE INTERNAL MOTOR DESIGN PARAMETERS

To illustrate and to provide an insight into the complexity of formulating a meaningful procedure for generating motor parametric data (discussed in Section 3.1) a discussion of the effects of some of the significant variable internal motor design parameters on motor design is presented in the following paragraphs.

The main internal design variables which affect the design of the motor include: stator stack outside diameter, stack length, number of slots, pitch or coil throw, turns per coil, stator stack inside diameter, and the ratio of iron to copper in both the stator and the rotor. The effect of each is discussed below.

(1) Stator Stack Outside Diameter - The following basic equation applies to any induction motor:

$$\text{Flux} = \frac{(\text{Constant}) (\text{Voltage})}{(\text{Frequency}) (\text{Turns})}$$

To determine the effect of stator stack outside diameter (motor size) variations, the frequency and flux density are held constant. Visually, when the diameter is changed, all dimensions in the motor change proportionately. The variation will not be exactly as described below since the relations are greatly oversimplified. However, the relations do indicate trends.

- a. Diameter is reduced -- direct first power variation.
- b. Flux decreases because of constant density and less area due to both diameter and length reduction. Flux varies directly as the diameter squared.
- c. Turns increase -- frequency and voltage are constant; therefore, because flux reduces, turns must increase inversely as the diameter squared.
- d. Resistance increases. More turns means that finer wire must be used, giving a direct squared effect based on the turns change or an inverse fourth power change based on diameter. However, area is also reduced (second power inverse effect based on diameter) and length is reduced (first power direct effect based on diameter). Therefore, the resistance increases inversely as the fifth power of the diameter.
- e. Mutual reactance (which governs exciting or no-load current) and leakage reactance increase directly as the turns squared which makes it an inverse fourth power effect based on diameter. However, area (of the flux path) varies directly as the diameter squared and the length directly

as the first power. This makes permeance vary inversely as first power of the diameter. Since reactance varies directly with permeance, the net effect is that reactance increases inversely as the diameter cubed.

- f. The cooling area, which has the major effect on temperatures since most of the temperature rise is across insulation and interfaces whose length remains constant, reduces directly as the diameter squared.
- g. From d. and e., the secondary resistance over leakage reactance ratio increases inversely as the diameter squared.
- h. For a negligible resistance (in the total impedance), the exciting current varies inversely with reactance or varies directly as the diameter cubed. Negligible resistance is usually the case.
- i. From d. and h., the no-load copper losses decrease directly as diameter to the first power.
- j. The output decreases directly as the diameter cubed.
- k. Therefore, for constant efficiency, full load losses would decrease as the diameter cubed.

It is possible to conclude from the effects above that as the diameter is reduced, the no-load losses become more and more predominant in the total losses because output is reduced as a third power effect and no-load losses as a unity power effect. If no-load losses are predominant, cooling becomes more difficult as diameter is reduced because the cooling area is reduced as a second power effect while no-load losses are reduced as a first power effect. Therefore, at some point, it becomes nearly impossible to cool the motor.

Conversely, as the diameter increases, the load losses will increase as a third power effect, eventually making the no-load losses a small factor. Cooling area will increase only as a second power effect. Therefore, eventually motors become too large to cool easily.

Motor weight and the point at which cooling difficulties are encountered can be affected by design manipulation. For instance, if a larger copper-to-steel ratio is used in arriving at the shape of the slots, larger and fewer slots are obtained. This results in more efficient slot utilization because of less insulation area and higher efficiency. However, this improvement is limited by an increase in exciting current because of the proportional increase in the slot openings to the total steel surface at the air gap. In other words, Carter's Coefficient - a factor multiplying the air gap length to account for the flux bunching in the teeth - increases.

(2) Stack Length - It is advisable to maintain the stack length approximately equal to the stator inside diameter because of practical manufacturing reasons and because experience has shown that better performance is obtained. This is a compromise between two offsetting tendencies. The torque capability of a motor is

approximately proportional to the total volume of material used. Therefore, for the same output, if the outside diameter increases, the stack length must decrease and vice versa. From a purely electrical standpoint, the actual optimum design would tend toward very short stack lengths. The larger diameter, shorter stack length motor tends to have more copper space but also tends to have longer end extensions which contribute more strength, fabrication, and cooling difficulties. The motor running temperatures tend to be lower if, for a given required volume, the diameter is proportionally smaller and the stack length longer. This results in smaller temperature rises above a given coolant temperature because the radial cooling paths of the stack and the winding end extensions are shorter. To obtain a true comparison between individual designs, the stack length should be maintained nearly constant. This is extremely difficult to accomplish because it makes fine design changes very difficult when considering the motors' typically small number of turns-per-coil which must be a whole integral. Therefore, the stack lengths are limited to an equivalent dimension between the stator outside diameter and the stator inside diameter.

(3) Number of Slots/Pitch/Turns per Coil - It is difficult to obtain fine adjustments in the total series conductors per phase and the resultant total flux when the turns per coil is small. Contributing to the difficulty in making fine design changes is the necessity for limiting the number of parallel paths. The use of more parallel paths allows more turns-per-coil permitting a finer adjustment. More parallel paths result in more conductors per slot with increased insulation penalties and increased cooling difficulties. The flux level can be adjusted by changing the pitch and by slightly varying the number of slots. With the same total flux, the torque output can be increased by shortening the stack length and increasing flux densities.

(4) Stator Inside Diameter/Iron-to-Copper Ratios - Increasing the stator inside diameter has the effect of lowering stator copper area, increasing rotor copper area, and increasing the slot opening. For any given design, if the speed is low, the inside diameter can be increased to increase rotor copper and reduce slip.

For any given design having sufficient speed, the iron-to-copper ratio in the rotor can be increased. Since the maximum rotor magnetic density is fixed by the steel, the stack length must be decreased to continue working the steel at that flux density. The consequent higher gap density results in higher output but the speed decreases because the rotor copper is decreased. Another related effect of smaller stack length is the decrease in cooling area which produces higher temperatures.

APPENDIX C

INVARIANT INTERNAL MOTOR DESIGN PARAMETERS

This appendix contains definition of certain motor design parameters which were assigned constant values or definite relationships to other design parameters for consistent use throughout the parametric study. Brief discussions of the basis for selection of the constant values or relationships are given wherever applicable. Some of these methods merely consisted of assumptions or rules of estimation based on past experience.

1. MAXIMUM TORQUE RATIO

For centrifugal device applications it is usually satisfactory to have a maximum torque of 200 percent of rated torque (for small motor ratings). This usually gives close to optimum efficiency and power factor. It also allows for some variation in voltage and frequency. Variations in voltage and frequency cause higher losses, which become worse as the ratio of maximum torque to rated torque decreases. For larger ratings and for general purpose motors that require some starting torque, this ratio of maximum torque to rated torque is sometimes increased to 250 percent or higher to obtain better efficiency. This is done at the sacrifice of a higher starting current. However, since larger motors generally are difficult to cool, higher efficiency is more desirable. A ratio of maximum torque to rated torque of 2.25 was selected for this study. Since the output torque varies with the volume of steel, it was necessary to hold this ratio constant to obtain a true comparison between designs.

2. STATOR WINDING CURRENT DENSITY

It was necessary to provide some common basis for comparison between designs in the parametric study, and current density in the stator windings is one of the better choices for such a common design parameter.

The current density should be as large as possible to obtain maximum performance for a motor of a given physical size. However, more heat per unit volume is produced with larger current densities so that the maximum current density is limited by ability to reject the heat. In other words, for the same performance, larger current densities tend to produce lighter, less efficient motors and smaller current densities tend to produce heavier, more efficient motors.

The results of the preliminary parametric designs indicated that a current density of approximately 10,000 amperes per square inch was a good compromise between the opposite tendencies described above for the prescribed coolant conditions of the parametric study.

3. STATOR AND ROTOR FLUX DENSITIES

The rotor flux densities were determined from the rotor steel magnetic properties at the temperatures of interest so that the rotor steel was worked near its maximum capability. Stator flux densities were not worked to their maximum magnetic-capability. Stator flux densities are somewhat influenced by rotor design and designs which utilize the maximum capability of the stator steel tend to have unacceptable adverse effects on other design factors. Stator flux densities for this parametric study were selected at values from the preliminary parametric designs which appeared to give the best designs from weight and efficiency considerations.

4. ROTOR RESISTANCE

Rotor resistance was adjusted to give reasonable starting torque and slip for each design. Reasonable starting torque and slip was determined by design judgement and experience.

5. AIR-GAP LENGTH

The following formula is used to determine the electrical air-gap length:

$$\text{Air-Gap Length} = \text{Rotor-Can Thickness} + \text{Stator Bore Seal Thickness} + \text{Running Clearance} + \text{Bore-Seal Clearance.}$$

a. Rotor Can Thickness. A simple, brief analysis was made to roughly determine the rotor can thickness requirements as a function of rotor diameter, speed, and temperature. The criterion used was the limitation of rotor can deformation from creep to less than 0.1 percent in 40,000 hours since comparison between stress rupture strength and creep strength from available data on tantalum T-111 alloy (the rotor can material) disclosed creep strength to be the governing factor.

The most significant stress in the rotor can is the tangential stress component at the inside diameter of the can; therefore, the analysis was limited to the tangential stress component. The tangential stress component was considered the sum of two parts, each resulting from different causes, (1) interference fit between the rotor can and rotor stack outside diameter and (2) centrifugal loading (body forces), i.e.,

$$\sigma_t = \sigma_{tp} + \sigma_{tn} \quad (C-1)$$

where

$$\begin{aligned} \sigma_t &= \text{total tangential stress component (psi)} \\ \sigma_{tp} &= \text{tangential stress from interference fit (psi)} \\ \sigma_{tn} &= \text{tangential stress from rotation (psi)} \end{aligned}$$

Since the rotor can thickness is small compared to the rotor can diameter, the stress may be assumed uniformly distributed across the wall and the approximate "hoop-stress" equations apply.

$$\sigma_{tp} = \frac{p_c D}{2t} \quad (C-2)$$

where p_c = interface contact pressure from interference fit (psi)
 D = mean diameter of rotor can (in)
 t = rotor can wall thickness (in)
 σ_{tp} = tangential stress from interference fit (psi)

and,

$$\sigma_{tn} = \frac{\gamma \omega^2 D^2}{4g} \quad (C-3)$$

where γ = specific weight of rotor can material (lbs/in³)
 ω = angular velocity, rad/sec
 D = mean diameter of rotor can (in)
 g = gravitational constant
 σ_{tn} = tangential stress from rotation (psi)

If an appropriate design stress, σ_d , is substituted for σ_t in equation C-1 and if expressions for σ_{tp} and σ_{tn} given by equations C-2 and C-3, respectively, are substituted into equation C-1, rotor can thickness, t , can be determined.

The design stress is given by,

$$\sigma_d = \frac{\sigma_c}{1.5} \quad (C-4)$$

where σ_c = creep strength of Tantalum T-111 alloy at temperature and time of interest necessary to limit deformation to 0.1 percent.
 1.5 = design factor to lend some conservatism to the analysis because of its approximate nature and to allow for unknown design conditions such as shock, vibration, stress concentration, and inaccuracy in material property data.

The allowable creep strength, σ_c , was taken from the Larson-Miller plot of vacuum creep and rupture data (figure C-1) for the appropriate rotor can temperatures (assumed as coolant temperature (°F) + 100°F) and 40,000 hours.

The resulting computations, based on an assumed interface pressure of 50 psi showed that for most of the diameter, speed and temperature combinations to be investigated in the parametric study, required rotor can thicknesses, t , was much less than 0.010 inch. At about the 1100°F, 15,000 rpm, 7.5 inch diameter point, the required can thickness begins to increase rapidly. However, as creep

deformation takes place, the interference fit will relax causing the interface pressure to decrease which in turn decreases the tangential stress. The can may then carry larger amounts of centrifugal loading without exceeding the permissible creep strength level.

Fabrication considerations take on increasing importance when the required thicknesses for mechanical strength are so small (< 0.010 inches). Since it appears feasible to deep draw rotor cans with 0.010 to 0.015 inch walls, it was decided to specify a rotor can thickness of 0.015 inches for the entire range of rotor diameters, speeds, and temperatures contemplated for the parametric study.

b. Stator Bore Seal Thickness. The ceramic stator bore-seal thickness is dictated by the minimum manufacturable thickness which is impermeable to the liquid metal vapor. The bore seal thicknesses used for the motor parametric study were as follows:

<u>Bore Seal O. D.</u> <u>(inches)</u>	<u>Thickness (inches)</u>	
	<u>Alumina</u>	<u>Beryllia</u>
4.0	0.024	0.070
6.0	0.036	0.080
8.0	0.048	0.090
10.0	0.060	0.100
12.0	0.072	0.110

c. Running Clearance. An empirical relation incorporated into the motor design computer program was used to determine the "running" radial clearance between the rotor can O.D. and the bore seal I.D. for all ratings in all studies. This equation has proved satisfactory in past motor design and fabrication at Westinghouse Aerospace Electrical Division and is considered compatible with the objectives of the present parametric study. The relation is

$$RUN = 0.005 + \frac{0.0042 (I.D.)}{\sqrt{P}}$$

where

RUN = running radial clearance (inches)

ID = stator ID (inches)

P = number of poles

The running clearance varied between 0.010 and 0.020 inches throughout the parametric study.

d. Bore Seal Clearance. A previous motor design study (4), (5) showed that the bore seal will not contact the stator stack I.D. at operating temperature because of the differential thermal expansion between the stator and the bore seal. Based on the results of the previous study a radial clearance between the bore seal outside diameter and the stator inside diameter of 0.002 inches was assumed throughout the parametric study.

6. STATOR SLOT INSULATION COMPONENT DIMENSIONS

In general, these dimension parameters were used with the bare wire dimensions to determine stator slot dimensions. The insulation dimensions were based on mechanical strength, electrical strength, and fabrication considerations. These parameters and their values used for the parametric study are listed below along with brief, amplifying statements on their functional significance where it is considered important.

a. Stator Slot Liner Thickness and Tolerance. The following thicknesses were chosen to provide the necessary electrical strength at temperature, to resist corona glow and electrical breakdown between the conductors and stator iron under the unique stator cavity environment, and to reflect current fabrication limitations:

<u>Voltage</u>	<u>Slot Liner Thickness (inches)</u>
120 V (line-to-neutral)	0.018
240 V (line-to-neutral)	0.018
480 V (line-to-neutral)	0.018
1000 V (line-to-neutral)	0.030

b. Stator Conductor Insulation Thickness. The following thicknesses were chosen to provide sufficient electrical strength at temperature, to resist corona glow and electrical breakdown between the conductors and stator iron under the unique stator cavity environment, and to reflect current fabrication limitations:

<u>Voltage</u>	<u>Insulation Thickness (inches)</u>	
	<u>Anadur</u>	<u>Plasma-arc Sprayed Alumina</u>
120 V (line-to-neutral)	0.011	0.008
240 V (line-to-neutral)	0.011	0.008
480 V (line-to-neutral)	0.011	0.008
1000 V (line-to-neutral)	0.011	0.008

The dimensions above are nominal dimensions. In addition to the bare wire dimensions and the slot insulation component dimensions listed under a and b above, the stator slot dimensions included allowances for proper tolerances on the slot component dimensions and for proper clearances between the various slot components for fabrication purposes.

7. ROTOR STACK INSIDE DIAMETER/SHAFT DIAMETER

The rotor stack inside diameter and the shaft outside diameter coincide in the final design layout. However, in the theoretical design process the rotor stack inside diameter is determined from electrical design considerations independently of the shaft outside diameter, which is determined from mechanical design considerations. The rotor stack inside diameter is the necessary diameter to permit the rotor core to carry the flux. This diameter, called the rotor magnetic inside diameter, is determined by the motor design computer program, and represents the maximum permissible shaft diameter. On the other hand, the rotor shaft must be large enough to support the rotor stack mass within acceptable limits of deflection, to adequately transmit the maximum prescribed torque, and to accommodate an adequate coolant passage for the rotor. The theoretical values of these two parameters rarely coincide. The required shaft outside diameter may exceed the maximum magnetic inside diameter in some cases. When this happens redesign of the motor is required.

Required shaft diameters as a function of motor rating for all the different combinations of design parameters could not be determined prior to starting the parametric study because of the lack of data on probable rotor mass and losses. Therefore, for purposes of the parametric study the rotor magnetic inside diameter was assumed to be the required shaft diameter. As the parametric study progressed, the rotor magnetic inside diameters determined by the motor design computer program were monitored to ensure that they were adequate by actually determining shaft diameters from mechanical design considerations for selected representative designs.

8. MINIMUM BENDING RADIUS OF STATOR END EXTENSION HAIR PIN TURNS

This factor is determined by coil forming techniques. There is a limit to the sharpness of bends which may be made in the insulated conductor when forming the coils. This factor affects the mean length of turns in the stator coil; therefore, it affects the weight of the windings and winding resistance. The following guidelines were used to determine the hair pin bending radii throughout the parametric study:

Estimated Bend Radii Based on Conductor Dimension

BEND RADII

(where X = conductor dimension measured in radial direction)

Conductor Width To Thickness Ratio	Winding Direction	
	Edge-wise	Flat-wise
1 : 1	See Note 3	See Note 3
1.5 : 1	1.5X	1.5X
2 : 1	1.5X	1.5X
2.5 : 1	2 X	1.5X
3 : 1	2 X	2 X

Notes:

- 1) Recommended maximum conductor dimension is approximately 0.3 inch for Anadur insulation on typical clad conductors.
- 2) Bend radius is measured to the conductor excluding the insulation.
- 3) Conductor width-to-thickness ratio of 1:1 is not recommended for pre-formed coil applications because of conductor twisting problems.

9. WINDAGE

This is the term used to designate motor losses from viscous friction forces acting between the fluid (liquid or vapor) and the spinning rotor or between portions of the fluid with imparted velocity and the stationary stator and end bells. Windage affects motor efficiency by absorbing some of the useful motor output. The total motor windage losses were considered to consist of two parts,

- a. rotor end losses
- b. losses in the annulus between the rotor can and the stator bore seal.

Since the "canned" rotor surfaces will be smooth, the total losses for both ends can be considered equivalent to the power absorbed on both sides of a submerged rotating disk. These losses are given by,

$$P = K\rho\omega^3r^5 \quad (C-5)$$

where

$$K = 9.2 \operatorname{Re}_r^{-0.615}, \operatorname{Re}_r < 10^5 \quad (C-6)$$

$$K = 0.0525 \operatorname{Re}_r^{-0.167}, \operatorname{Re}_r \geq 10^5 \quad (C-7)$$

$$\operatorname{Re}_r = \text{Reynolds Number} = \frac{\omega r^2}{\nu} \quad (C-8)$$

ρ = mass density of fluid (slugs/in³)
 ω = angular velocity (radius/sec)
 r = radius of rotor (in)
 ν = Kinematic viscosity of fluid (in²/sec)
 P = losses (lbs-in/sec)

The losses in the annulus between the rotor can outside diameter and the stator bore seal are given by,

$$P = \rho\omega^3r^5 \left(\frac{\pi L}{4r} \right) f \quad (C-9)$$

where

L = length of annulus = rotor length (in)
 f = frictional coefficient

$$f = \frac{8}{Re_b} = \frac{8}{Ta} \sqrt{\frac{b}{r}}, \quad (Ta \leq 41.1) \quad (C-10)$$

$$f = (0.31) \sqrt{\frac{b}{r}} Ta^{-0.123}, \quad (41.1 < Ta \leq 90) \quad (C-11)$$

$$f = (1.52) \sqrt{\frac{b}{r}} Ta^{-0.478}, \quad (Ta > 90) \quad (C-12)$$

$$Ta = \text{Taylor's number} = Re_b \sqrt{\frac{b}{r}} \quad (C-13)$$

$$Re_b = \text{Reynolds number} = \frac{\omega r b}{\nu} \quad (C-14)$$

b = running radial clearance between rotor can outside diameter and stator bore seal inside diameter (in)

Equations C-9 through C-14 were used to prepare the curves of windage loss as a function of rotor diameter and stack length in figures C-2, C-3, and C-4 for 500°, 800°, and 1100°F coolant conditions, respectively. These curves were used to estimate the windage losses throughout the parametric study.

10. FRICTION

This term is used to designate motor losses which occur in the motor bearings and seals. Non-contacting screw-type seals are anticipated for application in these motors, and for the range of seal diameters expected in these parametric designs the losses could be assumed negligible.

The bearing friction losses were computed by using a Westinghouse Aerospace Electrical Division developed computer program.

The program utilizes relationships derived for plain cylindrical bearings to calculate a number of bearing performance parameters associated with the bearing power loss. The results have been compared with experimental results for a four pad, pivoted-pad bearing and found to be in good agreement.

Some pertinent assumptions had to be made to provide input for the bearing loss computer program. These assumptions were,

- a. The bearing length to diameter (L/D) ratio was assumed to be 0.75 in all cases.
- b. The bearing radial clearance was assumed to be 0.002 inch in all cases.
- c. The bearing diameter was determined by assuming a design bearing load = 15 psi and for the initial parametric studies where no rotor weight data was available, the bearing diameter was assumed equal to the shaft outside diameter through the stack plus the nominal clearance specified under assumption "b" above.
- d. The fluid (lubricant) properties used were a composite, weighted average of the properties for liquid lithium, potassium, and NaK, since the motor may be required to operate with any of these liquid metals as lubricants.

The curves shown in figures C-5, C-6, and C-7 for the 500°, 800°, and 1100°F coolant condition were prepared using this computer program. These curves were used to estimate bearing friction loss throughout the parametric study.

Figure C-1. Larson-Miller Plot of Vacuum Creep and Rupture Data for Several Columbium and Tantalum Base Alloys 0.012 Inch Thick Samples. (Reference: NASA-CR-54093, Bore Seal Technology Report, December 1964)

FIGURE C-2. Windage Losses for 500°F Coolant Condition

FIGURE C-3. Windage Losses for 800°F Coolant Condition

FIGURE C-4. Windage Losses for 1100°F Coolant Condition

FIGURE C-5. Bearing Friction Losses for 500°F Coolant Condition

FIGURE C-6. Bearing Friction Losses for 800°F Coolant Condition

FIGURE C-7. Bearing Friction Losses for 1100°F Coolant Condition

APPENDIX D

MOTOR DESIGN COMPUTER PROGRAM

This appendix contains a discussion of some of the fundamentals involved in the motor design computer program employed in the motor parametric study.

The program essentially consists of a main electrical design routine supported by optional heat transfer and temperature distribution subroutines for the stator and rotor components.

The basic design provisions of the computer program start with minimum input data such as the outside diameter of the stator punchings, voltage, frequency, number of phases and poles, can thicknesses, friction and windage, and rated horsepower. The program then designs a motor using empirical formulas and standard materials. However, these empirical formulas are based on motors used in ordinary environments and do not give a suitable motor design for extreme environments. Additional input data for the computer program must then be used for material constants and variations in clearances made necessary by the use of non-standard materials.

Other input data for the computer program allows the designer to fix any dimension of the motor at any compatible value. Other dimensions are then calculated with the input values fixed. Considerable versatility is thus built into the computer program allowing the designer to optimize a motor using any material for any environment.

The motor performance calculation is based on the per-phase equivalent circuit shown in figure D-1.

The core loss resistance, RM_1 , may be calculated with the following formula.

$$RM_1 = \frac{3(VAG_1)^2}{SCL} \quad (\text{ohms})$$

where

SCL = total stator core loss

VAG_1 = air gap voltage

The loss in the stator can is regarded as core loss and is represented in the circuit in the same manner as the stator core loss.

The rotor can loss consists of two parts. The part which is due to the fundamental field is represented by the resistance $RCANR$. The rotor can, in relation to the fundamental field, acts as another rotor cage and provides useful torque. This component of the can loss is treated as another cage occurring prior to the rotor leakage reactance. The can resistance is divided by the slip in the same manner as the rotor cage and the torques from both branches are added to the output.

RSICH = Hot stator per phase resistance
 RST1 = Stray load loss resistor
 XS1 = Stator leakage reactance
 XM1 = Mutual reactance
 RM1 = Core loss resistance
 RCAN = Stator bore seal resistance
 RCANR = Rotor can resistance
 XR1 = Rotor leakage reactance
 RERIH = Hot rotor ring resistance
 RBARH = Hot rotor bar resistance
 S = Slip
 EA1 = Input phase voltage - RMS
 VAG1 = Air-gap voltage - RMS

Figure D-1. Per Phase Equivalent Circuit

The other component of rotor can loss is due to the slot ripple. It acts like a load on the motor and hence is subtracted from the output similar to a friction or windage load. This component is not represented in the circuit.

The basic method for calculating can losses is derived from a paper by Robinson, Rowe, and Donelan(11). The stator can loss formula for the motor is

$$W = \frac{15.5(B)^2(n)^2(D)^3(L)(t)(10^{-16})(ks)}{p}$$

Where

B = Maximum gap density in lines per square inch
 N = Speed of rotating field in revolutions per second
 D = Diameter of can in inches
 L = Length of stack in inches
 t = Thickness of can in inches
 p = Electrical resistivity of can in ohm-inches
 ks = Reduction factor for end effects (empirical)
 W = Can loss in watts

The formula for rotor can loss is

$$\text{Losses (watts)} = \frac{2.6(bt)(Kcc-1)^{2.33}(Kr)(W)}{5.04(be)}$$

Where

bt = Stator tooth width
 be = Stator slot opening
 Kcc = Carter's Coefficient
 W = Losses as calculated by previous formula
 Kr = Reduction factor for mmf opposition in the rotor

This formula is based on the assumption that any rotor can losses are caused by slot ripple. The formula is mainly empirical but was basically derived from theoretically correct considerations. The only basic change from the fundamental formulas in Reference 11 is that the expression containing Carter's Coefficient for rotor losses is raised to the 2.33 power instead of the second power. This change was based on an investigation conducted by Westinghouse Aerospace Electrical Division(2).

The stray load loss resistor is calculated using an empirical method. The circuit is solved using the resistance as shown in the equivalent circuit except that the series equivalent resistance of the branch consisting of RST1 and XS1 is multiplied by the square of the power factor.

The computer calculates and prints out total electrical weight. This includes everything in the motor from the stator outside diameter to the rotor inside diameter. It does not include cans, frame, end bells, bearings, seals, shaft, terminals, or leads.

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The computer continually corrects for saturation of the magnetic steel. The constants and flux densities printed out are exact for the full load point only because of the change in flux densities with load. The actual design method using the computer program requires trial and error; an optimum design can only be obtained by varying parameters, noting the effect, and making further changes based on the effect. The optimizing procedure is influenced by inputs from material information and mechanical analysis. Design engineer judgement determines when the optimum design is achieved.

The temperature calculation-subroutine performs a steady-state, three-dimensional heat transfer and temperature distribution analysis. Steady-state temperatures are obtained for more than 80 discrete nodal points distributed throughout a symmetrical section of the motor stator and rotor. The nodes are connected by a three-dimensional grid system which simulates the heat transfer paths within the symmetrical sections. Solutions are obtained by solving a system of simultaneous equations written for the grid system based on the electrical analogy to heat transfer. Variation of material properties with temperature, such as thermal conductivity and hardness, is taken into account. During the calculation process, properties of materials in the vicinity of nodal points are continually corrected to match the temperature calculated at the nodal points.

In addition to definition of heat path geometry and coolant information in the computer input sheets, equations of required material properties as a function of temperature must be supplied.

The detailed temperature calculation subroutines are intended primarily to obtain fine temperature distributions for use in final design analysis, such as thermal stress calculations and analysis of differential thermal expansion. This type of analysis usually occurs after conceptual designs are established. As stated in section 3.2 on computation methods, the unwieldiness of the required input information discouraged using these routines for all parametric designs. An option provided for the overall design program permits the temperature calculations for the stator, rotor or both, to be omitted from the design calculations, provided the average temperatures of the stator windings, rotor conductor bars, and rotor resistance end rings are provided as input data to the electrical design routine. Estimated temperatures used as input may not correspond to actual temperatures determined by considering the final electrical loss distribution. The purpose of the temperature calculation subroutines is to determine the actual temperature distribution corresponding to the magnitude and distribution of the electrical losses of the final electrical design; this illustrates the advantage of using the temperature calculation routines - the final temperature distribution corresponds to the final electrical design losses when the temperature calculation subroutines are used. Close approximations to the actual temperatures were obtained for temperature input by employing the temperature calculation subroutines at discrete times

during the parametric study to adjust the input temperature values.

A simplified flow diagram of the entire motor design program including the temperature calculation subroutines is shown in figure D-2. Figure D-2 illustrates the trial and error process used by the computer program to ensure that temperatures used to determine motor performance in the main electrical design routine correspond to actual calculated temperatures based on electrical motor losses.

The meaning of the term "thermal instability" can now be more clearly understood. Most iterative processes like the trial and error process above may converge or diverge. Definition of convergence for this process is established as agreement between calculated temperatures and temperatures used in the main routine within an acceptable tolerance (10°F). The number of iterations permitted to achieve convergence must be limited to a reasonable number to conserve computation time. Convergence or divergence and their respective rates may depend on actual physical aspects of the design or on purely mathematical characteristics. Design judgement was used to determine which of the cases above applied. Thus, if the temperatures readily converged within the allowable number of iterations, the design was considered thermally acceptable. If the temperatures failed to converge within the allowable number of iterations, but the results of the last iteration were close, the design was considered acceptable, although it was regarded as thermally unstable. Of course, if the temperatures diverged the cooling provisions were considered inadequate, and the design was regarded as thermally unacceptable.

Figure D-2. Simplified Flow Chart of Strap Wound Induction Motor Design Computer Program