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**INVESTIGATION OF ELECTROMAGNETIC LAUNCHER
BEHAVIOR FOR IMPACT FUSION**

**Annual Report
for period July 1, 1983 to May 1, 1984**

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INVESTIGATION OF ELECTROMAGNETIC LAUNCHER BEHAVIOR FOR IMPACT FUSION

Abstract

A program to develop an ultrahigh velocity accelerator (SUVAC), based on the electromagnetic railgun accelerator concept and sponsored by the U.S. Department of Energy, has been initiated at Westinghouse R & D Center. The program involves the construction over a 4-year period (July 1983 - June 1987) of a multi-stage railgun accelerator which has the potential of accelerating a 1-g projectile to about 30 km/s (Mach 100). The scientific objective of the program is to use the accelerator so built as the experimental apparatus to investigate the potential technical problems of accelerating macroparticles to velocity presently thought to be required to produce impact fusion. The program is part of a joint program with the University of Washington to develop the scientific and technological basis to achieve controlled thermonuclear fusion by hypervelocity impact. This report summarises the progress made in the first year of the program. It covers work done in the period 7-1-83 to 5-1-84.

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1 Executive Summary

A program to develop an ultra-high velocity accelerator based on the electromagnetic railgun concept, sponsored jointly by the U.S. Department of Energy and the Westinghouse Electric Corporation, has been initiated at the Westinghouse R & D Center, Pittsburgh, Pennsylvania. The program involves the construction over a 4-year period (July 1983 - June 1987) of a multi-stage railgun accelerator (Figure 1) which has the potential of accelerating a 1-g projectile to about 30 km/s (Mach 100). The scientific objective of the program is to use the accelerator so built as the experimental apparatus to investigate the potential technical problems of accelerating macroparticles to velocity presently thought to be required to produce impact fusion (> 100 km/s). The program is part of a joint program with the University of Washington to develop the scientific and technological basis to achieve controlled thermonuclear fusion by hypervelocity impact. This report summarises the progress made in the first year of the program. It covers work done in the period 7-1-83 to 5-1-84.

1.1 Principal Accomplishments in Year 1

The following important milestones have been achieved:

1. Engineering design for the entire 25 m long accelerator has been completed. This involves detailed system study and design^{1, 2, 3, 4, 5}, and the detailed design of the accelerator barrels^{6, 7}. A projection of the system is shown in Figure 2. Figures 3 and 4 show the detailed design of the barrels for the booster and the main sections of the accelerator.
2. Engineering design for the entire 10-stage distributed power supply system has been completed (Figure 2). This involves the detailed design^{8, 9} of the booster power supply subsystem (0.4 MJ, Figure 5) and a system of twenty one capacitor energy modules (2.1 MJ, Figure 6).
3. The booster accelerator, the booster power supply and its vacuum system has been fabricated, assembled and fully installed.
4. Testing and commissioning of the booster accelerator and its power supply system has begun.

These program milestones have been achieved ahead of schedule and funding profile (Table 4). A detailed list of the tasks which have been completed in order to achieve the above major program milestones are given in Table 2.

1.2 SUVAC Design Highlights

Detailed system study led to the selection of a multi-stage synchronous-wave-drive accelerator scheme which employs a distributed power supply system. The circuit schematic of the system is shown in Figure 1. The carrier of the Lorentz force which provides the propulsion is a plasma armature. Parametric study of plasma-arc dynamics conducted led to a concept for initiating the plasma armature which is novel to railgun technology. The concept involves generating the plasma armature by rapid combustion of a seeded propellant mixture in the presence of a strong electric current. The seeded propellant mixture is chosen so that the resultant plasma armature will contain low atomic weight ionic species including Li^+ , H^+ , N^+ and O^+ . The amount of the propellant is chosen so that the resultant plasma armature is relatively wide and cold. By making the plasma armature temperature sufficiently low, gross material erosion of the barrel can be avoided. Though low plasma temperature leads to high plasma resistivity, the large width of the plasma armature results in a low resistance for the plasma armature, thereby reduces the resistive loss in the armature and gives the accelerator an acceptable level of efficiency. The hard-drive conditions necessary to accelerate the projectile to the required velocity of 30 km/s in a distance of 25.6 m place stringent demands on the mechanical and thermal design of the accelerator. An innovative concept employing self-synchronised magnetic containment effect to enhance the stiffness of the barrel has been developed and applied in the critical design of the accelerator. To increase the accelerator efficiency in the first stage when the projectile velocity is low, the magnetic flux which produces the propulsive Lorentz force is augmented by using a secondary winding in the barrel. Innovative designs were also employed in the vacuum system which is required to house the accelerator. The result is that the entire system, accelerator-vacuum-tanks-power-supply, is completely modular in nature and can be fabricated and assembled modularly. It is planned to carry out the assembly and experimentation of the accelerator in 4

phases. The first phase involves the construction of the booster stage of the accelerator, SUVAC-I. The second phase will see the implementation of SUVAC-II which will consist of the first 6 m of the accelerator. SUVAC-III, consisting of the first 12 m constitutes the third phase. In the 4th phase, we will see the realization of the entire accelerator, SUVAC-IV. The principal characteristics of these four metamorphic states of the accelerator is shown in Table 1.

In terms of the science and technology of EM accelerator, it is noteworthy that the DoE sponsored program has already made the following significant contributions in advancing the art of railgun accelerator design:

1. Advanced system design which includes optimization of the plasma armature with respect to system performance¹.
2. The use of radiative transport data, which are generated by computer models of the plasma armature, to effect a rational thermal design of the accelerator.
3. Mechanical design of the accelerator includes detailed analysis of the effects of the plasma pressure^{6, 7}
4. The systematic parametric exploration of flux augmentation for improving accelerator efficiency⁴.
5. The use of pulsed magnetic containment effects for the first time in railgun accelerator design⁷.
6. Creation of a sophisticated multi-stage DES railgun simulation code²

1.3 Major Milestones Set for Year 2: 7-1-84 to 6-30-85

Major program milestones have been set for Year 2. They are:

1. Test & Commission SUVAC-I Power Supply	5/84
2. Test & Commission SUVAC-I Accelerator	6/84
3. Test & Commission SUVAC-II Power Supply	8/84
4. Test & Commission SUVAC-II Vacuum System	9/84

5. Test & Commission SUVAC-II Accelerator System	10/84
6. Experiment with Velocity Attempt at 14 km/s	11/84
7. Test & Commission SUVAC-III Power Supply:	1/85
8. Test & Commission SUVAC-III Vacuum System	3/85
9. Test & Commission SUVAC-III Accelerator	4/85
10. Experiment with Velocity Attempt at 20 km/s	6/85

The list of tasks which need to be completed and the dates for their completion in order to achieve the above program milestones is shown in Table 3.

The emphasis in Year 2 is again the construction of the experimental facility. The physical phenomena which come into play in the use of railgun to achieve ultra-high velocity are complex. Though the main physical problems can be identified, their detail theoretical analyses would appear to be premature at present without some experimental inputs. Our chosen approach is to attempt through systematic experimentation elucidation of these complex phenomena; to generate experimental data which will simplify the formulation of theoretical models. Once the experimental facility is completed, systematic experimentation to explore the scientific problems of accelerating macroparticle to ultra-high velocity (20 km/s - 100 km/s) can be conducted. We expect to initiate appropriate theoretical programs in ultra-high velocity physics to support the experimental program in the second or third year of the overall program.

The status of the entire 4-year program (7-1-83 to 6-30-87) is shown in Table 4.

Figure 1. The schematic of a multi-stage electrically continuous railgun. The simple railgun comprises a pair of metallic rails connected to a power supply and a pulse forming network. A conducting armature provide the electrical short between the rails. When a current flows in the rails, magnetic field is induced in the region between the rails and this acts on the current in the armature to produce the Lorentz force which accelerates the armature. In the case of the multi-stage railgun accelerator, current is fed sequentially into a series of locations along the rails. The aim is to sustain a constant current through the armature (so that the acceleration is constant) in such a way that only a short length of the rails behind the armature carries any significant amount of current. This reduces resistive losses and losses due to residual magnetic energy. Most significantly it leads to high piezo-kinetic efficiency resulting in minimal length for the accelerator.

WESTINGHOUSE-DOE ULTRA-HIGH-VELOCITY EM LAUNCHER
(25.6 m, 30 km/s)

Figure 2. SUVAC: Westinghouse-DoE Scientific Ultrahigh Velocity Accelerator.

Dwg. 4269B20

WESTINGHOUSE-DOE 30 km/s ULTRA-HIGH VELOCITY LAUNCHER—FIRST STAGE BOOSTER

Figure 3. Ultrahigh Velocity Barrel for SUVAC-I, the Booster Stage.

WESTINGHOUSE-DOE 30 km/s ULTRA-HIGH VELOCITY ELECTROMANETIC LAUNCHER

Figure 4. Ultrahigh Velocity Barrel for Main Accelerator Sections of SUVAC.

Figure 5. SUVAC-I Booster Power Supply Subsystem.

Figure 6. Capacitor Modules for Distributed Power System for SUVAC.

Table 1. Metamorphic States of the Westinghouse-DoE SUVAC Accelerator.

	Length ¹	Peak Current	Maximum Velocity	Commission ² (Planned)
SUVAC-I	1.83 m	280 kA	7.5 km/s	6/84
SUVAC-II	5.49 m	420 kA	13.5 km/s	10/84
SUVAC-III	10.98 m	450 kA	20 km/s	4/85
SUVAC-IV	25.6 m	450 kA	30 km/s	6/86

Remark:(1) Except for SUVAC-I and SUVAC-IV, experimentation with SUVAC-II and SUVAC-III will be carried out with the accelerator equipped with a "muzzle" section to help dissipating residual magnetic energy.

(2) During commissioning, tests are conducted only to ascertain that operations of the various components meet design specifications. Velocity attempt with minor system adjustments such as inductance, projectile geometry, plasma generation package, trigger delay etc. will be carried out only during planned series of experimentation.

Table 2. TASKS ACCOMPLISHED: 7-1-84 to 5-1-84.

System Design and Analysis

Accelerator Design and Analysis

Design of Inter-Stage Current Injection Module

Design of Distributed Power Supply

Design of Triggering and Firing System

Design of the New Laboratory

SUVAC-I Booster Power Supply: Fabrication, Assembly, and Installation

Booster Accelerator (SUVAC-I): Fabrication, Assembly, and Installation

Distributed Power Supply: Component Fabrication Completed

Table 3. TASKS TO BE ACCOMPLISHED IN YEAR 2.

	Planned date of completion
Fabricate SUVAC-II Vacuum System	7-15-84
Fabricate SUVAC-II Accelerator	8-1-84
Assemble SUVAC-II Power Supply	8-1-84
Test & Commission SUVAC-II Power Supply	8-15-84
Assemble SUVAC-II Vacuum and Bus System	8-15-84
Assemble SUVAC-II Accelerator	9-1-84
Develop Inter-Stage Trigger	8-15-84
Develop Instrumentation	9-1-84
Develop Data Reduction and Analysis Package	9-1-84
Test & Commission SUVAC-II Vacuum System	9-8-84
Test & Commission SUVAC-II Accelerator System	10-1-84
Experiment with Velocity Attempt at 15 km/s	11-15-84
Assemble SUVAC-III Power Supply:	12-1-84
Test & Commission SUVAC-III Power Supply:	1-1-85
Fabricate SUVAC-III Vacuum System	1-1-85
Fabricate SUVAC-III Accelerator	2-1-85
Assemble SUVAC-III Vacuum and Bus System	2-1-85
Assemble SUVAC-III Accelerator	3-1-85
Test & Commission SUVAC-III Vacuum System	3-15-85
Test & Commission SUVAC-III Accelerator	4-15-85
Experiment with Velocity Attempt at 20 km/s	6-30-85

Table 4. Program Status Summary.

	Contract Requirements (A.R.O)	Program Performance (A.R.O.)
Conceptual Design	5 months	3 months
Detailed Design	9 months	6 months
Test Firing SUVAC-III	21 months	On Schedule
Velocity Attempt: 20 km/s	27 months	On Schedule
Test Firing Entire Accelerator SUVAC-IV	36 months	On Schedule
Velocity Attempt: 30 km/s	45 months	On Schedule
Final Report	48 months	On Schedule

2 Ultrahigh Velocity Accelerator: Fundamental Considerations

The basic problems of obtaining ultra-high velocity for macroparticles using the railgun technique arise essentially from the unique requirements placed upon the accelerator which may be summarised as follows:

1. Because of the very high velocity required, the armature is necessarily a plasma instead of a solid conductor.
2. The use of a plasma armature gives rise to a host of plasma dynamic effects which are relatively unknown at present: plasma-rail boundary effects at ultra-high velocity ($> \text{Mach } 100$), MHD phenomena and instabilities, plasma kinetic phenomena and instabilities, etc.
3. At very high velocity the junction ("breech") voltage which includes the back EMF induced by the motion of the armature can become rather large. This may induce break-down at the junction (breech) shunting the armature current.
4. Friction between the projectile and the bore at ultrahigh velocity and the simultaneous requirement of sealing the plasma behind the projectile.
5. To obtain very high velocity, multi-stage accelerators are mandatory. Besides shunting the armature current thereby reducing its acceleration, arcing ahead of the projectile presents an unique problem for the multi-stage accelerator in that it complicates the sensing of the projectile position by either electromagnetic means or optical means. Real-time projectile-armature position information is needed to control firing of a multi-stage accelerator. In its broadest sense, this requirement extends also to various 'self-activating' switching schemes. Arcing ahead of the projectile may be caused by one or combination of several mechanisms: plasma blow-by past the projectile, shock wave ahead of the projectile, or ultra-violet radiation from the plasma armature.
6. To obtain the very high velocity required in a reasonably short

distance, very high acceleration is necessary. This requires very aggressive driving conditions. Very high current concentration is required. This usually leads to very high plasma temperature and pressure. The combination of these two plasma attributes, high temperature and high pressure, place stringent demands on the accelerator design and material technology. In particular, barrels with a high degree of stiffness are required to contain the mechanical loading and to provide adequate sealing of the plasma behind the projectile.

7. Because of the need for insulation between the rails thus leading to the presence of gaps, gas sealing to prevent plasma blow-by becomes a major problem in the presence of high plasma pressure, high plasma temperature and magnetic loading on the rails.
8. Aerodynamic drag and ablation of the projectile at ultrahigh velocity: this may be reduced or eliminated by using a sufficiently high vacuum. The need of a vacuum complicates the design of the accelerator and the current transmission manifold. High-current and high-voltage vacuum feed-throughs are required. The use of vacuum will also eliminate arcing ahead of the projectile due to gasdynamic shocks and UV ionization. However, plasma blow-by still remains to be problematic.
9. Because of the need for a high vacuum, vacuum compatible materials are required in the construction of the accelerator. In particular, dielectric materials which fulfil simultaneously the requirements of high compressive strength, high flexural strength, high shock resistance, high arcing resistance, low outgassing rate, and high tolerance machinability, are required. These requirements tend to be conflicting demands. Application of advanced material concepts and technology are required.
10. The length-to-bore ratio of the accelerator is necessarily huge.

In a sense, the challenge of ultrahigh velocity (20 km/s - 200 km/s) for macroparticles is simply that it has never been done before.

3 System Study and Design

Details of the system analysis conducted for the SUVAC launcher system are given in the references^{1, 2, 3, 10, 4}. Highlights of the study are presented below.

3.1 Selection of System Concept

Two candidate accelerator schemes were examined:

1. Single stage accelerator (Figure 7)
2. Multi-stage energy injection but electrically continuous accelerator (Figure 8)

In the single stage accelerator scheme, current from a single power source is fed into the breech of the gun and flows through the entire length of accelerator at the instant of projectile exit from the muzzle. This scheme was rejected in the present program in terms of its limited potential for extrapolating such a scheme to achieve the very high velocity which is required for impact fusion and other applications:

1. It has low piezo-kinetic efficiency (the ratio of average pressure to peak pressure is low). This leads to a system which either is very long or has a very high bursting force at the breech, or requires an enormous amount of energy to drive an elaborate pulse forming network.
2. It has low energetic efficiency due to large losses in the form of residual magnetic energy in the circuit if this energy is not recovered and large resistive losses. In the ultimate application where repetitive launching at a rate of 0.1 Hz to 0.01 Hz may be required, the magnitude of these losses also translates to problems in thermal management of equivalent magnitude.

In a multi-stage system, current is fed into the accelerator at a number of locations by a distributed system of energy stores (Figure 8). As the projectile-armature flies past the current injection point, the current from the power source feeding that stage is turned on and adds to the decaying current from the previous stage. The armature current is maintained nearly constant, though

Figure 7. Single Stage Railgun.

Candidate concept considered but rejected: Poor piezokinetic and energetic efficiency, leading to large bursting force at breech and/or large energy losses in circuit. Limited potential for extrapolation to velocity required for impact fusion.

Figure 8. Multi-stage Electrically Continuous Railgun.

System concept selected: Favorable piezokinetic and energetic efficiency, leading to good launcher dynamics and reduced thermal dissipation in launcher. Potential for extrapolation to velocity required for impact fusion worthy of serious investigation which is the object of the present program.

significant amount of current only flows in a limited length of the rails (typically less than the length of about 2 stages) behind the projectile. This occurs when the condition for synchronous wave drive is satisfied. This is achieved by tuning the impedance of the power source to match the rail impedance and the motional impedance of the armature. Overall this results in the accelerator having high piezo-kinetic, high energetic efficiencies, and favorable launcher dynamics.

3.2 System Specifications

As the projectile mass and dimensions are somewhat arbitrary at this stage, we select a projectile dimension of about 1 cm^3 and a projectile mass of 0.8 g - 1.0 g as the baseline to define a system potentially capable of achieving 30 km/s. The mass of the plasma armature to be used was optimized during the system study and was determined to be in the range 0.2 g to 0.4 g depending upon the ionic species used. For the purpose of system study, we used a maximum total mass of 1.4 g for the projectile and armature. If we assume the length of the accelerator to be 25.6 m (84 ft), the projectile will experience a mean acceleration of $1.76 \times 10^7 \text{ m/s}^2$. The average propulsive force required would be 24.6 kN, and the mean propulsive pressure would be 246 MPa. This is a manageable pressure load on the projectile, the rails and the barrel dielectric using state-of-the-art railgun materials. The length 25.6 m was therefore selected as a baseline for the system design.

As reliable data for material friction and plasma-rail boundary layer effects at velocities in excess of 10 km/s were and are not available at present (the development of such data is indeed one of the goals of the present program), it was decided that these effects would be ignored as issues in the detailed engineering design of the accelerator, as design analyses attempting to include these effects at present would be either too cumbersome or too tentative to make them not very useful. This is in accord with the program philosophy that the objective of the program in part is to develop an experimental facility to investigate ultrahigh velocity phenomena, thereby enabling the scientific development of ultrahigh velocity accelerator technology. We do, however, allow for these "ultrahigh velocity losses" in the overall program strategy by designing the entire system based upon only 70 % of the total energy available from the

power supply. Together with the option of reducing the mass of the projectile-armature combination to 1 g instead of 1.4 g, we should have enough excess energy to cover losses due to these effects and other ultra-high velocity effects not known at present.

3.3 Accelerator Efficiency and Choice of Ionic Species for the Plasma Armature

Details of this study are presented in the reference⁵. A 1-D MPD model was developed for the plasma armature and was used to study the dependence of the intrinsic accelerator efficiency upon the ionic species used for the plasma armature. The intrinsic accelerator efficiency is defined as the percentage of the kinetic energy gained by the projectile (excluding the armature) over the total energy dissipated in the accelerator (excluding residual magnetic energy). This efficiency is indicative of the efficiency of one stage in a multi-stage accelerator, where the residual magnetic energy of one stage is transported down to the next stage by inductive coupling between the consecutive stages. The principal results are shown in Figures 9 and 10. Figure 9 shows that accelerator efficiency is a moderately strong function of the ionic mass, decreasing with increasing ionic mass, and a function of the ionization potential, increasing with decreasing ionization potential. Both results are primarily manifestation of the same physical effect, that is, there are more particles per unit mass of the armature for ions with smaller atomic mass or ionization potential. A greater number of particles imply a greater volume, and therefore a wider plasma for a fixed bore. A wider plasma armature provides a larger area of current collection. Though the plasma temperature is lower leading to a higher plasma resistivity, the net armature resistance is lower due to the greatly increased area of current collection. Figure 10 shows the optimum mass of the plasma armature for a specific ionic species. The optimum mass is a function of some of the gun parameters including the bore cross-section, the rail inductance gradient and the current used. For a Li^+ plasma, the optimum armature mass is about 50 mg for the SUVAC-I barrel (booster), of which the bore cross-section is 9.5 cm x 9.5 cm, the rail inductance gradient is 0.75 $\mu\text{H/m}$ and the current used in the computation is 300 kA (it was found later that for SUVAC-I, a peak current of

Figure 9. Effects of ionic mass and ionization potential on intrinsic accelerator efficiency.

Curve 743744-A

Figure 10. Effects of mass of plasma armature on intrinsic accelerator efficiency.

280 kA is sufficient to achieve the design goal). It is seen from Figure 10 that for masses of the armature less than the optimum, accelerator efficiency degrades rapidly with decreasing mass of the armature. There appears to be a critical armature mass above which the accelerator efficiency is a weak function of the armature mass until the assumptions of the model break down. The practical implication of this is obvious, i.e. there is a minimum armature mass which should be generated for the plasma armature, and slight excess of this mass due to factors outside experimental control is not important. This simplifies the practical problem of developing the plasma generation package. In practice, the plasma armature is usually a mixture of several species of ions due to practical constraints on the selection of materials for initiating the arc and contamination of the arc by ablation products of rail, barrel and projectile materials. The above results for a single-species plasma, however, provided insight and guidance to the selection of the appropriate materials and quantity for generating the plasma armature.

3.4 Flux Augmentation

The simple railgun is essentially a 1-turn linear motor. The use of additional turns to increase the driving magnetic field gives rise to what is called augmented railguns. The increase in the magnetic field per unit current manifests itself as an increase in the effective inductance gradient L' which appears in the propulsive force expression, $1/2 L' I^2$. By increasing the force per unit current, flux augmentation has the potential of improving the efficiency of the accelerator by minimising resistive losses in the circuit, since the augmented railgun uses a smaller current to provide the same accelerating force.

There are, however, complications to the simple theory. The use of a smaller current tends to lower the plasma temperature and hence increases the plasma resistivity and losses in the plasma armature. The percentage improvement in performance is also a function of the armature velocity. Since addition of the augmentation windings constitutes considerable complications in the construction of the barrel, the breech and the muzzle, a detailed study was carried out to find how the accelerator performance varies with the degree of flux augmentation, so that the appropriate degree of augmentation can be selected.

The details of the study are presented in reference¹. The principal results are shown in Figure 11 which shows the accelerator efficiency as a function of the effective inductance gradient L' , indicative of the degree of augmentation. A typical value of L' for a simple single-turn railgun designed for aggressive driving conditions is $0.3 \mu\text{H/m}$. The results shown in Figure 11 were obtained with the help of the same plasma dynamics code used in Subsection 3.3, thus the effects of variation in the plasma armature properties have been taken into account. The results show that, for the bore dimensions and payload of the SUVAC accelerator, the fractional improvement in accelerator efficiency flattens out at a value of L' approximately between $0.7 \mu\text{H/m}$ and $1 \mu\text{H/m}$, suggesting that additional effort to increase augmentation above this range is not warranted. This range of L' can be obtained by a 1-turn augmentation winding in the barrel making the barrel a 2-turn linear motor. As the improvement is most pronounced when the velocity is below 10 km/s , we decided that augmentation would be used only in the first stage (SUVAC-I) of the accelerator.

3.5 Minimum Acceptable Value of L' from System Standpoint

In barrel design, attempts to use rail geometry with high values of L' tend to accentuate both the mechanical and thermal problems in the design. They also tend to increase the resistance of the rails, thus reducing some of the improvement in efficiency. Railgun barrel design thus tends to be an exercise in trade-offs between the electromagnetic, plasma dynamical, structural, thermal, material and efficiency requirements. To define the bounds on the barrel design study, we need to ascertain at the outset the minimum values of the driving inductance gradient L' which could be tolerated from the system standpoint, that is, to achieve the system objective of 30 km/s with a 1.4 g projectile-armature using only 70% of the total energy available (2.5 MJ). This amounts to an average system efficiency of 36% . From Figure 11, this requirement can be met if the value of L' is at least $0.65 \mu\text{H/m}$ for the first stage, and $0.34 \mu\text{H/m}$ from the second stage onward, provided the first stage could deliver a velocity of 8 km/s or more. These requirements now define the baseline parameters for detailed design of the system.

Figure 11. Effects of Flux Augmentation on Accelerator Efficiency.

3.6 Cryogenic Rails

The potential improvement to the accelerator efficiency which might be caused by the use of cryogenic rails was systematically examined in the system study. Superficially, cryogenic rails hold the promise of lower rail resistance and higher accelerator efficiency. This simple inference is however complicated by several effects. The magneto-diffusion skin effect for pulsed current is induced and accentuated by the velocity of the fast-moving armature. This effectively reduced the area of current conduction in the rails thereby increasing its resistance. This velocity skin effect worsens with increasing conductivity of the conductor. It is therefore not clear a priori that the use of cryogenic rails would necessarily improve the efficiency of the accelerator. The trade-offs become even more complicated when one takes into account the relative contributions to losses by the resistance in the external circuitry and the use of shorter length of rails per stage made possible by the use of a multi-stage accelerator.

The principal result of this study is shown in Figure 12, which shows how the intrinsic accelerator efficiency varies with the resistivity of the rails as manifestation of the degree of cryogenic refrigeration of the rails. A 2-m accelerator section is assumed. A projectile of 1.2 g is accelerated by a 0.2-g Li^+ plasma armature carrying a current of 400 kA and subjected to a rail inductance gradient of $0.33 \mu\text{H/m}$. The resistance of the external circuitry is assumed to be $1 \text{ m}\Omega$, the losses in which is taken into account in calculating the intrinsic accelerator efficiency. The various curves pertain to different values of mean velocity for the projectile as a result of the constant acceleration given to it over the length of the 2-m section. Velocity-skin effect has been taken into account, the details of which is given in the reference⁵. The resistance of the plasma armature is determined from the plasma armature model mentioned earlier.

The principal feature of the results is shown in Figure 12 is that the improvement to accelerator efficiency is minuscule over a wide range of cryogenic temperatures, from liquid hydrogen to room temperature. Consequently, we do not plan to pursue this concept any further in the present program.

Figure 12. Effects of Cryogenic Rails: Efficiency vs Rail Resistivity.

3.7 System Simulation and Tuning

3.7.1 Development of an Advanced DES Multi-Stage Simulation Code:PARADES

Before the detailed design of the system could be undertaken, a computer code which can simulate in detail the performance of a multi-stage, electrically continuous, distributed energy store railgun system had to be developed. A circuit schematic of the system is shown in Figure 13.

Details of the simulation code developed are given in the reference³. Capabilities of the code include:

1. Capability to simulate distributed capacitor store system.
2. Capability to simulate distributed inductive store system.
3. Capability to simulate true crow-bar action giving accurately the coupling of the oscillatory shunted capacitor circuit to the non-oscillatory load, thus taking into account effects of the internal inductance and resistance of the capacitor power supply including those of the switches on the commutation of the crow-bar current into the crow-bar switch and their effects on the rail current.
4. Capability to simulate accurately the effects of augmentation through a mutual inductance formulation of the coupling between the augmentation winding and the inner firing rails rather than the usual, and somewhat erroneous, use of an "equivalent" L' in the circuit equations which govern a simple railgun. The use of the latter procedure gives rise to erroneous muzzle current and hence erroneous armature current and consequently erroneous acceleration profile for the projectile. It would also give rise to erroneous estimates of the muzzle voltage. PARADES takes care of all these effects and gives accurately the muzzle current, the armature current and the muzzle voltage.
5. Capability to simulate the effects of the muzzle shunt.

Figure 13. Schematic of a Multistage Electrically Continuous DES Railgun.

6. Sophisticated switching and controls available to the user in running the simulation, such as delayed current injection at any given stage, control of crow-bar trigger either for fixed time delay or with self-activation.
7. Extensive graphics post-processor available to display graphically and interactively the results of the simulation.

3.7.2 Results of Simulations and Parametric Tuning

Detailed simulation study and parametric tuning of the system to obtain the required velocity and a matched-impedance synchronous-wave drive is an iterative and a tedious process and was carried out in our system design task, the details of which are described in the references^{3, 1}. The process, which also interacted closely with the barrel design task, eventually defined the system and its electrical parameters which are shown in Table 5. The simulated rail current, armature current, current in the transmission line (bus current), the capacitor voltage, the interstage junction voltage, the acceleration, the velocity and the position-time history of the projectile are shown in Figures 14,15, 16, 17, 18, 19,20 and 21 respectively. This concludes the system design task for this phase of the program. We expect to review and continue to conduct system study and simulation and to re-tune the system parameters throughout the entire experimental program when experimental data of the system performance become available to close the design loop.

Table 5. Definition of the System Parameters: Phase I.

Stage	Length	Inductance	Capacitance
	(m)	(μ H)	(μ F)
1	1.83	6.5	8.00
2	1.83	3.5	3.96
3	1.83	4.0	3.96
4	1.83	4.0	3.96
5	1.83	3.5	3.96
6	1.83	3.5	3.96
7	1.83	3.5	3.96
8	2.74	2.7	5.61
9	2.74	2.2	5.94
10	7.32	1.8	5.94

Figure 14. Simulated rail current vs armature position by PARADES.

Figure 15. Simulated armature current vs armature position by PARADES.

Figure 16. Simulated bus current vs time by PARADES.

Figure 17. Simulated capacitor voltages of the distributed power supply.

Figure 18. Simulated inter-stage junction voltage by PARADES.

Figure 19. Simulated projectile acceleration profile by PARADES.

Figure 20. Simulated projectile velocity profile by PARADES.

Figure 21. Simulated projectile position-time history by PARADES.

4 Barrel Design: Concept Selection

Details of the barrel concept selection and design process are reported in the references^{6,7}. Some of the basic considerations have already been described in Section 2. Besides attempting to meet the requirements of those fundamental aspects, the barrel concept selection process was further influenced by the following considerations:

1. That it is an experimental device.
2. As such it should provide easy access for instrumentation.
3. It should allow rapid assembly and disassembly.
4. The design should allow for the possibility of pre-stressing the entire barrel, so that any excessive bore deformation can be controlled by an appropriate assembly procedure if it is found to occur later experimentally.
5. Ease of machining should be maximized to minimize the cost of construction.

Based upon all these considerations, the final concepts selected which best implement the specifications of the system study are as shown in Figure 3 for the booster accelerator, and in Figure 4 for the barrel for the main accelerator.

5 Barrel Design: Electromagnetic Analysis and Formulation of the Structural Loading on the Barrel

To obtain the mechanical loading on the barrel, knowledge of the electromagnetic field produced by the current flowing in the rail and in the plasma armature is required. In the absence of a 3-D finite element EM modelling code, an approximation (upper bound) to the pressure loading on the bore due to the plasma armature was obtained as follows:

1. Using a 2-D finite-element EM modelling code (WEMAP), the electromagnetic field in the plane of the barrel cross-section due to a harmonic current flowing in the rails of infinite length was computed (Figure 22). The actual EM field in the time domain due to a current

pulse is a superposition of the EM fields in the frequency domain weighted according to the spectrum of the pulse. The actual time-domain synthesis of the field was not carried out. We performed all our analysis in the frequency domain and adopted a worst-case approach, which frequently implies the use of the results corresponding to the high frequency limit. Because of the very high velocity of the armature, we expect the high frequency results to be close to a true time-domain analysis.

2. By calculating the total magnetic energy stored in the system per unit length, the inductance gradient L' of the system is obtained.
3. The value of L' is obtained thus for a number of frequencies. The asymptotic value of L' at the high frequency limit is assumed to be the lower bound of the effective L' in the force formula $1/2 L' I^2$ (Table 6)
4. The net force acting on the back of the projectile is calculated as $1/2 L' I^2$.
5. The 2-D magnetic field calculated in (1) is not a true representation of the the field immediately behind the plasma armature as it does not include the field contributed by the current flowing in the plasma armature and thus cannot be used directly to obtain the magnetic pressure distribution acting on the plasma armature. However, we expect that the geometry of the magnetic field obtained in (1) is similar to the real case except perhaps at the corners where curvature of the current filaments is very large. On the basis of this conjecture, we scale the magnetic pressure acting on the back of the plasma armature in proportion to B^2 where B is as calculated in (1), so that the total force acting on the plasma armature is given by $1/2 L' I^2$.
6. By MHD equilibrium, this magnetic pressure distribution at the rear boundary of the plasma armature is transformed into an equivalent gas pressure distribution acting on the back of the projectile.
7. The mechanical load on the rails within the plasma is the sum of

magnetic pressure and gas pressure which is constant assuming the plasma to be in MHD equilibrium (Figure 23).

8. The pressure loading on the insulators in between the rails consists of the plasma gas pressure only. If the current density within the plasma armature is constant in the direction parallel to the rails (though allowed to vary in the plane of the bore cross-section), the magnetic field would decrease linearly from the rear boundary of the plasma armature to zero at the projectile. In this case, by MHD equilibrium, the gas pressure would decrease quadratically from a peak value immediately behind the projectile to zero at the rear boundary of the plasma. Integrating over a distance of two inches (the assumed spacing between supporting plate bolts), the average pressure on the dielectric inserts due to the plasma can be determined at a cross section behind the projectile. This pressure was used in the two dimensional finite element structural analysis. Figure 24 shows the pressure on the dielectric inserts due to the plasma.
9. Due to induced current in the steel plates, the magnetic field exerts a magnetic pressure on the inner surface of the steel plate. This magnetic pressure on the inner surface of the steel plates can be calculated from the magnetic field intensity along these surfaces. The magnetic fields are obtainable from the WEMAP program mentioned previously. These pressures are depicted in Figure 25. They are found to be low but their analysis have been included for completeness.

6 Barrel Design: Self-synchronised Pulsed Magnetic Containment

The basic concept here is to exploit the eddy current induced in the conducting steel outer structure by the 'moving' front of the magnetic field following the motion of the armature to confine and compress the flux generated by the rail current to the cavity within the steel structure. The result is that the rails are repelled from the inner walls of the steel cavity with forces which tend to neutralize the force of repulsion between the rails caused by the magnetic pressure

Table 6. Inductance gradient L' vs frequency for SUVAC main barrel.

Case 1: Steel outer containment structure (design selected)

Frequency (kHz)	Inductance Gradient $\mu\text{H/m}$
1	0.378
3	0.351
5	0.343
10	0.336

Case 2: Insulating outer containment structure

Frequency (kHz)	Inductance Gradient $\mu\text{H/m}$
1	0.411
3	0.389
5	0.384
10	0.378

Figure 22. Magnetic flux plot of SUVAC main barrel at 5000 Hz.

Figure 23. Resultant plasma and magnetic pressure on rails.

Figure 24. Average gas pressure on dielectric inserts due to plasma.

Figure 25. Magnetic pressure on inner surface of steel plates.

and the plasma gas pressure on the rails. The effects are modelled by using the high-frequency capability of WEMAP, and are clearly seen by comparing the pressure loads on the rails with the conducting steel structure (shown in Figure 23) and without the conducting steel structure (shown in Figure 26). Without the steel structure, the net force on the rail in the vicinity of the plasma armature is approximately 11,300 lb/in. In the presence of the pulsed magnetic containment effect induced by the steel, this load is reduced to 4,400 lb/in, a reduction of nearly 3 times. We have been conservative in these estimates. We expect that a more exact 3-D transient analysis of the effect will show a greater reduction of the repulsive force on the rails. The structure behaves as if it has been stiffened by the compression of flux by the conducting containment structure.

The presence of the steel will also shape the magnetic field directly above and below the plasma armature, perhaps reducing its pressure on the dielectric. This would constitute magnetic containment in relation to the dielectric. We have not analysed this effect in detail.

This magnetic containment is not achieved without paying a price, which consists of a reduction in the driving inductance gradient L' . The essence of the technique is to shape the magnetic field near the armature so that the resultant gas pressure in the plasma armature is highly anisotropic towards the direction of propulsion. The full exploitation of the technique will have to await more exact analysis based on 3-D modelling of the field due to the rails, the plasma armature, and the conducting containment structure, and shaping the containment structure, the rails, and the containment cavity to give the optimal results. At present, we content ourselves by placing the steel structure sufficiently far from the rails so that the reduction in L' is somewhat minimal, about 10 % (Figure 6), and yet closed enough to induce a significant degree of magnetic containment of the rails and the plasma armature.

7 Barrel Design: Analytical Sizing of the Barrel

To initially size the dimensions of the barrel, hand stress calculations were performed using simple beam theory. A more refined structural analysis was done later using finite element techniques. Analytical sizing is important in minimizing the costly iterations of finite-element design and analysis.

Figure 26. The resultant pressure rail loadings without the steel structure.

7.1 Beam on Elastic Foundation Analysis

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The stresses and deflection along the length of the steel support plates behind the copper rails were determined initially using a beam on an elastic foundation analysis and then verified through a finite element model.

The model used for the beam on an elastic foundation was a beam fixed at one end (the breech end of the rails) and with a uniformly distributed load from a to b as shown in Figure 27. The beam is semi-infinite and is assumed to extend from the left end to a length beyond that portion affected by the loading.

The stiffness of the foundation on which the beam rests was calculated based on the individual stiffnesses of the bolts used to support the barrel structure. The stiffness was also determined from the forces and deflections obtained from the finite element analysis. The determination of the foundation stiffness is described below.

The spring constant of the bolts was obtained from the following relationship:

$$1/K_B = 1/E(0.4D/A_1 + L_1/A_1 + L_T/A_M + 0.4D_M/A_M) \quad (1)$$

where:

D = nominal bolt diameter

A_1 = cross-sectional unthreaded area

A_M = cross-sectional minor thread area

D_M = minor thread diameter

L_1 = length of unthreaded portion

L_T = length of threaded portion being clamped

Figure 27. Beam on elastic foundation model for the barrel.

The stiffness of the thru bolt at the base of the barrel is determined as follows: $D = 0.625$ in., $A_1 = 0.307$ in.², $A_M = 0.240$ in.², $D_M = 0.553$ in., $L_1 = 6.0$ in., $L_T = 1.0$ in. Therefore from equation (1):

$$K_{B \text{ bottom}} = 1.14 \times 10^6 \text{ lb./inch} \quad (2)$$

Since the bolts are spaced at two inches,

$$K_{B \text{ bottom}} = 0.57 \times 10^6 \text{ lb./inch/inch} \quad (3)$$

For the shorter bolts at the top of the barrel: $D = 0.50$ in., $A_1 = 0.196$ in.², $A_M = 0.1486$ in.², $D_M = 0.435$ in., $L_1 = 0.50$ in., $L_T = 1.25$ in. Using equation (1) again,

$$K_{B \text{ top}} = 2.21 \times 10^6 \text{ lb./inch}$$

These bolts are also spaced at two inches. Therefore,

$$K_{B \text{ top}} = 1.10 \times 10^6 \text{ lb./inch/inch} \quad (4)$$

The stiffness of the top plate can be estimated from:

$$\Delta = PL/EA \quad \text{or} \quad K_{\text{plate}} = P/\Delta = EA/L$$

$$K_{\text{plate}} = (27.6 \times 10^6 \text{ lb./in.}^2)(1.125 \text{ in.})/(5.0 \text{ in.})$$

$$K_{\text{plate}} = 6.21 \times 10^6 \text{ lb./inch/inch} \quad (5)$$

Using the above calculated stiffnesses the following model (28) can be set up to determine the resultant foundation stiffness. The total foundation stiffness is:

$$K_T = ((1/1.10 \times 10^6) + (1/12.41 \times 10^6))^{-1} + 1.14 \times 10^6$$

$$K_T = 2.15 \times 10^6 \text{ lb./in./in.} \quad (6)$$

The total foundation stiffness can also be determined from the displacements calculated using the finite element model in conjunction with the loading.

$$F_{\text{TOT}} = k\delta$$

$$\delta = \text{deflection of side plate} = .00279 \text{ inches}$$

Figure 28. Model for the Foundation Stiffness.

$$F_{TOT} = 6706 \text{ lbs./in.}$$

$$\text{Hence, } k = 2.40 \times 10^6 \text{ lb./in./in.} \quad (7)$$

The equations and rotations used in the beam on an elastic foundation model are shown in the table below. The bending stress, shear stress and deflection of the steel support plate on the elastic foundation are tabulated on the following page. The stiffness used was $2.40 \times 10^6 \text{ lb./in./in.}$ derived from the finite element model.

7.2 Stresses on Upper Plate Bolts

With reference to the coordinate system and geometry shown in Figure 24, where x and y are the coordinates in the plane of the bore cross-section and z is measured along the axis of the barrel, the plasma pressure $P(x,y,z)$ at a position (x,y,z) was assumed to vary as:

$$P(x,y,z) = P_{\max}(x,y) (z - z_a)(2l - z + z_a)/l^2 \quad (8)$$

where z_a is the position of the rear boundary of the plasma armature and l is the

Results of Beam on an Elastic Foundation Analysis

X	BENDING	SHEAR	DEFLECTION
(in)	STRESS	STRESS	
	(psi)	(psi)	(in)
1.0	-9,410	3,170	-.0007
2.0	-6,020	2,520	-.0012
3.0	-3,840	2,080	-.0015
4.0	-2,440	1,770	-.0016
5.0	-1,540	1,530	-.0017
6.0	-970	1,310	-.0016
7.0	-620	1,070	-.0014
8.0	-430	750	-.0011
9.0	-310	490	-.0008
10.0	-220	320	-.0004
15.0	-40	40	-.0001

Shear, moment, slope, and deflection formulas for semi-infinite beams on elastic foundations

NOTATION: All notation is the same as that for Table 7. No length is defined since these beams are assumed to extend from the left end, for which restraints are defined, to a length beyond that portion affected by the loading. Note that M_A and R_A are reactions, not applied loads.

The following constants and functions, involving both beam constants and foundation constants, are hereby defined in order to permit condensing the tabulated formulas which follow

k_s = foundation modulus (pounds per square inch per inch); b_s = beam width (inches), and $\beta = \left(\frac{k_s b_s}{4EI}\right)^{1/4}$. (Note: See page 94 for a definition of $\langle x - a \rangle^n$)

$$\begin{aligned} F_1 &= \cosh \beta x \cos \beta x & A_1 &= 0.5e^{-\beta a} \cos \beta a & B_1 &= 0.5e^{-\beta b} \cos \beta b \\ F_2 &= \cosh \beta x \sin \beta x + \sinh \beta x \cos \beta x & A_2 &= 0.5e^{-\beta a} (\sin \beta a - \cos \beta a) & B_2 &= 0.5e^{-\beta b} (\sin \beta b - \cos \beta b) \\ F_3 &= \sinh \beta x \sin \beta x & A_3 &= -0.5e^{-\beta a} \sin \beta a & B_3 &= -0.5e^{-\beta b} \sin \beta b \\ F_4 &= \cosh \beta x \sin \beta x - \sinh \beta x \cos \beta x & A_4 &= 0.5e^{-\beta a} (\sin \beta a + \cos \beta a) & B_4 &= 0.5e^{-\beta b} (\sin \beta b + \cos \beta b) \end{aligned}$$

$$\begin{aligned} F_{a1} &= \langle x - a \rangle^0 \cosh \beta \langle x - a \rangle \cos \beta \langle x - a \rangle & F_{b1} &= \langle x - b \rangle^0 \cosh \beta \langle x - b \rangle \cos \beta \langle x - b \rangle \\ F_{a2} &= \cosh \beta \langle x - a \rangle \sin \beta \langle x - a \rangle + \sinh \beta \langle x - a \rangle \cos \beta \langle x - a \rangle & F_{b2} &= \cosh \beta \langle x - b \rangle \sin \beta \langle x - b \rangle + \sinh \beta \langle x - b \rangle \cos \beta \langle x - b \rangle \\ F_{a3} &= \sinh \beta \langle x - a \rangle \sin \beta \langle x - a \rangle & F_{b3} &= \sinh \beta \langle x - b \rangle \sin \beta \langle x - b \rangle \\ F_{a4} &= \cosh \beta \langle x - a \rangle \sin \beta \langle x - a \rangle - \sinh \beta \langle x - a \rangle \cos \beta \langle x - a \rangle & F_{b4} &= \cosh \beta \langle x - b \rangle \sin \beta \langle x - b \rangle - \sinh \beta \langle x - b \rangle \cos \beta \langle x - b \rangle \\ F_{a5} &= \langle x - a \rangle^0 - F_{a1} & F_{b5} &= \langle x - b \rangle^0 - F_{b1} \\ F_{a6} &= 2\beta \langle x - a \rangle \langle x - a \rangle^0 - F_{a2} & F_{b6} &= 2\beta \langle x - b \rangle \langle x - b \rangle^0 - F_{b2} \end{aligned}$$

$$\text{Transverse shear} = V = R_A F_1 - \gamma_A 2EI\beta^3 F_2 - \theta_A 2EI\beta^2 F_3 - M_A \beta F_4 + LT_V$$

$$\text{Bending moment} = M = M_A F_1 + \frac{R_A}{2\beta} F_2 - \gamma_A 2EI\beta^2 F_3 - \theta_A EI\beta F_4 + LT_M$$

$$\text{Slope} = \theta = \theta_A F_1 + \frac{M_A}{2EI\beta} F_2 + \frac{R_A}{2EI\beta^2} F_3 - \gamma_A \beta F_4 + LT_\theta$$

$$\text{Deflection} = y = \gamma_A F_1 + \frac{\theta_A}{2\beta} F_2 + \frac{M_A}{2EI\beta^2} F_3 + \frac{R_A}{4EI\beta^3} F_4 + LT_y$$

Expressions for R_A , M_A , θ_A , and γ_A are found below for several combinations of loading and left end restraints. The loading terms LT_V , LT_M , LT_θ , and LT_y are given for each loading condition.

Left end restraint Loading, reference no.	Free	Guided	Simply supported	Fixed	Loading terms
2. Uniformly distributed load from a to b	$R_A = 0$ $M_A = 0$ $\theta_A = \frac{-w}{EI\beta^3} (B_3 - A_3)$ $\gamma_A = \frac{-w}{2EI\beta^4} (B_2 - A_2)$	$R_A = 0$ $\theta_A = 0$ $M_A = \frac{-w}{\beta^2} (B_3 - A_3)$ $\gamma_A = \frac{w}{2EI\beta^4} (B_1 - A_1)$	$M_A = 0$ $\gamma_A = 0$ $R_A = \frac{w}{\beta} (B_2 - A_2)$ $\theta_A = \frac{w}{2EI\beta^3} (B_4 - A_4)$	$\theta_A = 0$ $\gamma_A = 0$ $R_A = \frac{-2w}{\beta} (B_1 - A_1)$ $M_A = \frac{w}{\beta^2} (B_4 - A_4)$	$LT_V = \frac{-w}{2\beta} (F_{a2} - F_{b2})$ $LT_M = \frac{-w}{2\beta^2} (F_{a3} - F_{b3})$ $LT_\theta = \frac{-w}{4EI\beta^3} (F_{a4} - F_{b4})$ $LT_y = \frac{-w}{4EI\beta^4} (F_{a5} - F_{b5})$

From: R. J. ROARK and W. C. YOUNG, *Formulas for Stress and Strains*,
Fifth Edition, McGraw-Hill, 1975, page 143.

length of the plasma armature. The pressure distribution $P_{\max}(x,y)$ is obtained from the electromagnetic analysis described earlier and can be evaluated at the surface of the dielectric inserts where $x = h/2$. We assumed that 10 inches is a typical length of the plasma armature. (See Section 9)

The spacing of the bolts on the upper plate was selected at two inches. To determine the load acting on the bolts one can integrate the pressure distribution over the spacing of the bolts b just behind the projectile in the z -direction as follows:

$$F = \int_{z_b-b}^{z_b} \int_{-w/2}^{w/2} P(h/2, y, z) dy dz$$

where w is the bore width (0.375 in). We note that the average pressure just behind the projectile is,

$$P_m = \int_{-w/2}^{w/2} P(h/2, y, z_b) dy = 42,700 \text{ psi.} \quad (9)$$

After integrating, the load acting on a two inch depth of the upper steel plate is 31,600 lbs. Each two inch section of plate is supported by two bolts, one on each side of the plate. Therefore the stress in the bolts is given by:

$$\sigma_1 = F/A_{\text{bolt}} = 31,600/(\pi(d_{\text{bolt}})^2/4)$$

Assuming both bolts share the load equally,

$$\sigma = \sigma_1/2 = 80,500 \text{ psi.} \quad (10)$$

The plasma load on the dielectric insert will also act to shear the top plate bolts mounted parallel to the plate. The shear area is given by

$$A_s = 2 \times \pi d^2/4$$

For a 1/2 inch diameter bolt, $A_s = .393 \text{ in}^2$. The shear load as previously determined is 31,600 lbs.

$$2\sigma_{\text{shear}} = 31,600 \text{ lb.}/.393 \text{ in}^2 = 80,407 \text{ psi.} \quad (11)$$

Again this is conservative and does not allow for the interaction of the entire length of plate and the additional bolts.

7.3 Stresses on the Upper Plate

The upper plate was analyzed as a beam with a uniform load partially distributed over its length. The depth of the beam was taken to be two inches, the spacing between adjacent bolts. It was realized that this model did not account for the interaction of the remainder of the plate with the two inch section but it was felt that the model was conservative (thus inherently safe) and provided a starting point for sizing the plate thickness.

The beam model is shown in Figure 29.

where:

$$a = 1.36 \text{ in.}$$

$$b = 1.54 \text{ in.}$$

$$c = 1.36 \text{ in.}$$

$$l = 4.20 \text{ in.}$$

Figure 29. A Beam Model of the Upper Steel Plate for SUVAC Main Barrel.

The distributed load w acting on the plate is calculated from the maximum pressure on the dielectric inserts described in Section 7.2 above. The maximum average pressure at the bore is given by expression (9) to be 42,700 psi. Thus the load on the dielectric per linear inch axially is the product of this pressure and the bore width, i.e. $42,700 \text{ psi} \times 3/8 \text{ in} = 16,000 \text{ lb/in}$. This load is transmitted by

the dielectric to the steel plate via the contact area between the dielectric and the steel plate. We assumed that the load is spread evenly over the contact area, of which the width is 1.54 in. Thus the applied pressure on the steel plate over this area is $P_1 = 16,000/1.54 \text{ psi} = 10,400 \text{ psi}$. The load per linear inch laterally on the steel plate over a 2-in depth is thus 20,800 lb/in. This is the load w in the beam model for the upper plate (Figure 29).

The reaction R_1 on the beam is given by:

$$R_1 = (wb/2l) \times (2c + b)$$

$$R_1 = 16,244 \text{ lb.}$$

The maximum bending moment is given by:

$$M_{\max} = r_1(a + (R_1/2w))$$

After substituting, $M_{\max} = 28,400 \text{ in-lb.}$

For a 1.25 inch thick plate the maximum bending stress is calculated as:

$$\sigma_{\max} = Mc/I = (28,400)(1.25/2) \times 1/((1/12)(2)(1.25)^3)$$

$$\sigma_{\max} = 54,550 \text{ psi.}$$

7.4 Stresses on Side Mounting Plates

The loading used to determine the stresses on the steel side plates of the barrel was calculated from the resultant force on the metallic rails shown in Figure 2. Taking an average value of this resultant pressure over the face of the rail gives a pressure of 14,485 psi. For a 2 inch bolt spacing and 0.375 inch rails this pressure yields a total load of:

$$F = (14,485 \text{ lbs/in}^2)(2 \text{ in.})(0.375 \text{ in.})$$

$$F = 10,863 \text{ lbs.}$$

This load is transmitted over a length of 2.210 inches on the side steel plate. The loading on the side plate then is a distributed load of 4938 lb/inch.

The side plate is modelled conservatively as a beam simply supported at both ends with a uniform distributed load $w = 4938 \text{ lb/inch}$ as shown in Figure 30.

Figure 30. Beam Model for Side Mounting Plate.

The maximum bending moment is given by:

$$M_{\max} = wl^2/8$$

$$M_{\max} = (4938 \text{ lb/inch})(2.210 \text{ in.})^2/8$$

$$M_{\max} = 3014 \text{ in-lb}$$

The maximum bending stress then is:

$$\sigma_{\max} = ((3014 \text{ in-lb})(c/2))/(1/12)(2)(c^3)) \quad (12)$$

where c is the plate thickness. For a plate thickness of 1.340 inches the maximum bending stress is:

$$\sigma_{\max} = 5040 \text{ psi} \quad (13)$$

This is a relatively low stress and should give rise to relatively small deformation of the steel, giving the structure the high degree of stiffness which it requires for precision bore dynamics.

7.5 Stresses on Main Thru Bolt at Base of Barrel

The total load of 10,863 lbs. acting over a two inch depth calculated in Section 7.4 was used to determine the stresses in the bolt at the base of the barrel. For a 5/8 inch diameter bolt the stress on this bolt is:

$$\sigma = F/A = (10,863 \text{ lbs.})/((\pi/4).625^2)$$

$$\sigma = 35,409 \text{ psi.} \quad (14)$$

This is again a relatively low stress for a high-strength, high-stiffness super-grade aircraft steel bolt and should incur very small deformation in the bolt.

8 Barrel Design: Detailed Finite Element Structural Analysis

Structural analysis of both the augmented (SUVAC-I, the booster) and non-augmented (SUVAC main barrel) was performed to estimate the maximum stresses and deflections occurring at firing. In both cases a two dimensional model was used representing a 1 inch thick cross-section of the barrel. A linear static finite element analysis of such a section loaded with the maximum magnetic and gas pressures developed in the barrel is expected to give a conservative estimate of the maximum stresses and deflections of the barrel. In order to have the top plate of the barrel restrained, the section had to be taken at the bolt location for both barrel types.

Both barrels are symmetric about a vertical plane at the center of the bore. Half the barrel was modeled and restraints were placed at line of symmetry to represent the actual barrel. This greatly reduced the cost and time of analysis with essentially no loss of accuracy.

In order to gain insight into the stress and deflection distributions along the length of the barrel, a three dimensional non-augmented model was attempted in addition to the 2D models.

All three models contained three materials; steel, copper and G11 epoxy insulation. The material properties used were the same for each model and are as follows:

	Steel	Copper	G11
Young's modulus	3×10^7	1.5×10^7	2.2×10^6
Poisson's ratio	0.3	0.355	0.16
Density	0.286	0.322	0.07

Both barrels were subjected to very complex loading patterns. Loads were determined using WEMAP and included as input in the structural analysis for all three models. Gas pressure distributions were imposed on the bore walls and on the faces of gaps (slip surfaces) which exist at the interface of the copper and the insulating material. Magnetic pressure distributions were imposed on all four surfaces of the conducting rails and the inner faces of the main barrel enclosure. Using PATRAN-G for our finite element modeling and ANSYS for finite element analysis as we were, it was most convenient to approximate the load distributions with cubic equations and distribute equivalent loading forces on the appropriate nodes. A one inch thick plate model was created, and each force represented the product of the pressure at that location and the area of an element face at that node.

Each configuration contains interfaces where like or unlike materials contact each other as the individual parts are assembled. These gaps or interfaces, may be in contact with each other or separated by some small distance depending on how the assembly deforms under load. In addition, these surfaces may slide relative to each other even if they are contacting. Both these conditions cannot be modeled by a simple linear elastic analysis. Closed gaps, contacting surfaces, without sliding respond and are modeled as continuous material and we were fortunate not to encounter gaps where sliding was of major concern. Open gaps are modeled as independent unconnected pieces of material and we noted significant instances of these in the models. For these analyses open gaps were included in each model based on experience and initial analytic runs were made. After inspecting the results of the original analyses, the gaps were modified, opened or closed, as required to give a proper representation of the assembly.

8.1 SUVAC-I: Augmented Barrel

Only a two dimensional model of the augmented barrel was analysed. Figure 31 is the final model used for this analysis. Initially, the top and bottom steel plates of the barrel were chosen to be only 1/2 inch thick. Analysis of the initial model projected maximum deflections in the bore on the order of 50 mils and a maximum stress of 300,000 psi in the top plate on the plane of symmetry at the top surface. This was unacceptable so the thickness of the top and bottom steel plates were doubled and the loads were reviewed to verify their validity. The final model includes some adjustments to the loads as well as the thicker sections.

One inch thick plates were used for this entire model except at the reduced side section. The two remaining elements in that section are 0.6 inches thick to give an equivalent cross-sectional area of the 1/2 inch bolt which holds these pieces together. Open gaps are included as indicated in Figure 32. The difference in the gap pattern at the top and bottom corner of the steel is due to the bolt that holds the top plate on. The bolts were not modeled, instead a closed gap was used to restrain the top plate. The long vertical gaps in this model are a marginal representation of the true assembly. Inspection of the results reveals that this is actually a closed gap with a large amount of sliding. However, since the pressure of the two surfaces against each other is small and the relative sliding is great, we feel that this model is the best possible without going to an expensive non-linear analysis and is within the accuracy required.

Gas pressure loads were included representing the gas that enters the open gaps. Because of the short period of time the gaps actually open, we feel that the gas pressure does not reach the entire length of the open gap. The area where gas pressure in the gap contributes the most significant loading is in the open gap between the primary rail and the epoxy. It is doubtful that the gas would have enough time to penetrate much beyond this. Thus, gas pressures were included on both sides of the open slip surfaces from the bore around the primary rail. Figure 33 shows the loads due to gas pressure and Figure 34 shows the loads due to magnetic pressures. Figure 35 shows the restraints applied to the problem. Each circle highlights a restrained node and each number indicates that no deflection is allowed in the related degree of freedom. The degree of freedom are:

IDE: 1.GEOMETRY MODEL 2.ANALYSIS MODEL 3.DISPLAY 4.NEUTRAL SYS. 5.END
AUGMENTED RAIL

Figure 31. The finite element structural model for SUVAC-I.

IDE: 1.GEOMETRY MODEL 2.ANALYSIS MODEL 3.DISPLAY 4.NEUTRAL SYS. 5.END
AUGMENTED RAIL

Figure 32. The FE structural model for SUVAC-I showing the slip surfaces.

1	X Translation	4	Rotation about X
2	Y Translation	5	Rotation about Y
3	Z Translation	6	Rotation about Z

Figure 36 shows the deflection due to the gas pressure and Figure 37 shows the deflection due to the magnetic forces. Figure 38 shows the total deflection expected when firing the launcher and is simply the vector sum of the previous figures by superposition. The deflections are shown with a magnification of 30 in all the 3 figures. The maximum displacement is experienced by the dielectric insert near the bore with a magnitude of 21 mils. Figure 39 shows the Von Misses stress expected due to both magnetic and gas pressure loads.

8.2 Non-augmented Barrel: SUVAC Main Barrel

8.2.1 2D Model

This barrel is much like the augmented barrel with some open and some closed slip surfaces, complex load distributions and bolted connections modeled with closed gaps. Open gaps are as indicated in Figure 40 and where decided upon after initial analysis. As previously mentioned, the barrel cross-section was taken at a bolting location so that the top plate is restrained. Since the bolts entering from the side of barrel give the top plate a longer span, this case was modeled on the assumption that it is worst case. Again the line of symmetry was modeled by restraining the nodal motion as indicated in Figure 41 but the lower support structure's large bolt was not include. Nodal restraints in the nut's bearing area were used to restrain this section and replace the bolt.

Figure 42 shows the loads due to the gas pressure and figure 43 shows loads due to magnetic pressures. The deflection due to gas pressure and the deflection due to magnetic pressures are shown in figures 44 and 45 respectively. The total displacement by superposition is figure 46 and the Von Mises stress from the total load is seen in figure 47.

IDE? 1.GEOMETRY/ MODEL 2.ANALYSIS MODEL 3.DISPLAY 4.NEUTRAL SYS. 5.END
 FORCE, , P.L. 2
 IDE? 1.GEOMETRY/ MODEL 2.ANALYSIS MODEL 3.DISPLAY 4.NEUTRAL SYS. 5.END
 GAS PRESSURES

Figure 33. Gas pressure loads on SUVAC-I finite element model.

XME? 1.GEOMETRY MODEL 2.ANALYSIS MODEL 3.DISPLAY 4.NEUTRAL SYS. 5.END
 FOR C... P.L. 1
 XME? 1.GEOMETRY MODEL 2.ANALYSIS MODEL 3.DISPLAY 4.NEUTRAL SYS. 5.END
 MAGNETIC LOADS

Figure 34. Magnetic pressure loads on SUVAC-I finite element model.

Figure 35. Restraints in the FE element structural model for SUVAC-I barrel.

Figure 36. Deformations of SUVAC-I (x 30) due to gas pressure.

Figure 37. Deformations of SUVAC-I (x 30) due to magnetic pressure.

Figure 38. Total deformations of SUVAC-I (x 30) due to combined loads.

Figure 39. SUVAC-I: Von Mises Stress due to combined gas and magnetic load

OBJ? 1.GEOMETRY MODEL 2.ANALYSIS MODEL 3.DISPLAY 4.NEUTRAL SYS. 5.END
NON-AUGMENTED RAIL

Figure 40. FE model for SUVAC main barrel showing the slip surfaces.

Figure 41. FE structural model for SUVAC main barrel showing the restraints.

Figure 42. Gas pressure loads on the FE model for SUVAC main barrel.

Figure 43. Magnetic pressure loads on the FE model for SUVAC main barrel.

Figure 44. Deformations (x60) of SUVAC main barrel due to gas pressure loads

Figure 45. Deformations (x60) of SUVAC main barrel due to magnetic loads.

Figure 46. Deformations (x60) of SUVAC main barrel due to combined loads.

NON-AUGMENTED RAIL VON MISES STRESS

RAIL GUN 6TH 2D ANALYSIS: MODELED WITH FEWER CRACKS
SUBCASE 1 X 1.000 PLUS SUBCASE 2 X 1.000 SUBCOM
)DESTRUCTIVE ANALYSIS MODEL 3.DISPLAY 4.NEUTRAL SYS. 5.END
NO 0000UGRM NON-AUGMENTED RAIL VON MISE
RU

100004.	- H
50177.	- P
80000.	- C
70402.	- D
70515.	- L
05457.	- F
55740.	- G
51053.	- M
44065.	- I
20078.	- J
31101.	- K
24203.	- L
17416.	- N
10500.	- N
2041.	- O

Figure 47. Von Misses stress in SUVAC main barrel due to combined loads.

9 Barrel Design: Plasma-Rail Interaction and Plasma Armature Generation Technique

9.1 Plasma Radiation, Velocity-skin Magnetic heating, and Rail Survivability

In its entirety, rail damage by the moving arc is a result of several complex processes occurring at the rail-armature interface. There is, for example, the possibility of formation of highly localised arc-roots which could give rise to discrete spots on the rail which are intensely hot. As these hot spots move, tracks are created. Such tracks have been observed. However, in the region of most severe damage where the arc velocity is low, observed damage is typically indicative of gross melting of the rail, with features of the re-solidified rail surface suggestive of the effects of high pressure, interfacial instability of the metallic surface during the molten state, and turbulence. As this class of damage is the most severe, it seems logical to address it (gross melting) first before addressing effects caused by the fine structure of the plasma armature.

This issue was recently examined by Thio¹⁰ in a program sponsored by the Army Research Office. The fundamental results of that study are being extended and applied in the present program to develop a practical plasma generation package and in the selection of suitable rail materials for the accelerator. For the sake of completeness and ease of reference, we quote below the principal results of that study.

The study examined the six ions, Cu^+ , Al^+ , C^+ , H^+ , O^+ , and Li^+ , as candidate ionic species for use in railgun plasma armature. Cu^+ and Al^+ have been popularly used in past experiments in plasma driven railguns^{11, 12, 13, 14, 15, 16, 17, 18, 19}. C, H, and O are typical ablation products of the bore and projectile materials caused by the plasma armature.

In developing the trend of the general results, the study used the bore dimensions of SUVAC-I as a reference. A current of 300 kA is assumed. With these parameters, the temperature of the plasma armature and its length are calculated for the candidate set of ionic species as a function of the mass of the plasma armature. Figures 48 and 49 shows a sample of the results for the case of Li^+ , Al^+ , and Cu^+ plasma armatures.

For a given plasma mass, say 200 mg, the temperature of the plasma armature varies from a low value of 13,000 K for the Li plasma to the high values of 19,900 K and 26,100 K for the Al and Cu plasma respectively. The higher temperatures of the Al and Cu plasmas will give rise to higher radiation intensity and therefore higher rail heating rates. However, the cumulative radiative temperature rise also depends on the dwell time of the plasma armature. The longer the dwell time, the greater is the radiative temperature increment. For the same velocity of the plasma armature, the dwell time is inversely proportional to its length. Looking at Figure 49, it can be seen that the Cu vapor plasma has the shortest length. For the same total armature mass of 200 mg, the Cu armature is 4 cm long, the Al plasma has a length of 6.5 cm, while the Li plasma has a length of 13.5 cm, very much in the reverse order of their temperatures.

The temperature rise in the rail due to black-body radiation from these 3 plasma armatures is shown in Figures 50 and 51. Figure 50 shows the case for copper rails, whereas Figure 51 shows the case for tungsten or molybdenum rails. In all these cases, a velocity of 5 km/s is assumed for the plasma armature. The results are valid only before the onset of rail melting. Non-linear effects associated with temperature-dependent material properties are ignored. These non-linearities will cause perturbations to the absolute values of the linearised results but we do not expect them to change the relative nature of the results.

For all cases of rail materials, the rail temperature rises extremely rapidly for the Cu plasma armature, and only slightly less so for the Al plasma armature. For Cu vapor plasma, the radiative rail temperature rise exceeds the melting point of all of the three rail materials, namely copper, molybdenum and tungsten, well before the fly-by of the armature is complete. The results indicate that in this case, the use of Cu vapor plasma armatures would certainly cause gross melting of the rail, and substantial damage to the rails can be expected to occur at this point. On the other hand, in the case of the Li plasma, the radiative temperature rise is relatively minor, a mere 260 deg rise for copper rails and 460 deg for tungsten or molybdenum rails, well below their melting points. The higher temperature rise in the case of the tungsten or molybdenum rails is due to the lower thermal conductivity of these materials. For the Al vapor plasma, the

Figure 48. Plasma armature temperature vs plasma mass.
For different ionic species under the maximum current operating
condition of SUVAC booster stage

Figure 49. Plasma armature length vs plasma mass.
For different ionic species under the maximum current operating
condition of SUVAC booster stage

Figure 50. Rail temperature rise due to radiation from plasma armature.

Figure 51. Rail temperature rise due to radiation from plasma armature.

rail temperature rise due to its radiation is much greater and marginal on the melting of copper rails.

In addition to plasma radiative heating, there is a second contribution to rail heating, that due to transient heating by the magnetic field induced by the motion of the plasma armature - velocity-skin effect heating as analogous to skin-effect heating by induced eddy current in stationary conductors.

From a knowledge of temperature rise T_k due to velocity-skin-effect heating and the melting point T_m of the rail material, the permissible temperature rise due to plasma radiation before onset of melting can be found. Given the radiative flux of the plasma armature, time to onset of surface melting is then determined. Further, given the length of the plasma armature, the minimum velocity of the armature to prevent rail surface melting can be calculated.

For the bore dimensions and gun parameters (L') of SUVAC-I with 300 kA of current, the minimum velocity required of the plasma armature to avoid causing rail surface melting for the five ionic species, Cu, Al, O, H and Li are presented in the case of copper and molybdenum rails in Figures 52 and 53 respectively. The initial temperature of the rail is assumed to be 0 degree C.

We note that, if we use an armature mass of 400 mg and copper rails, the velocity v_m for an Al plasma is about 1.7 km/s, and for a Cu plasma is about 10 km/s, whereas for a Li plasma the required velocity can be as low as 120 m/s. For molybdenum rails, the requirements are somewhat reduced: about 500 m/s for Al plasmas, 37 m/s for Li plasmas, but still rather high for Cu plasmas, about 3 km/s is required.

In the hard-drive conditions planned for SUVAC-I, the rail could also fail due to thermal softening by the temperature rise in the rails well before melting occurs. It is therefore not sufficient to know only the value of v_m , but also the expected temperature rise for any velocity of the armature above the minimum velocity v_m .

Using only 200 mg for the plasma armature under the peak design

Figure 52. Minimum armature velocity for rail survivability.

Figure 53. Minimum armature velocity for rail survivability.

operating conditions of SUVAC's booster, the maximum rail temperature rise for copper and tungsten rails are shown in Figures 54 and 55 respectively for a range of armature velocity from 0 to 10 km/s. The dashed curves indicate that the temperature rise has exceeded the melting point of the material starting from an initial temperature of 0 deg C. For tungsten rail, we see that the rail temperature can be kept below 2000 deg C for a Li plasma armature travelling at velocities above approximately 1 km/s. Whereas for Al, C, or O plasmas, substantially higher temperatures (over 2500 deg C) occur. For Cu^+ vapor plasma of the same mass (200 mg), the rail will always melt for armature velocity below 10 km/s. The moderately lower temperature rise associated with the use of Li plasma may be tolerated. A Li plasma with a higher mass would lower the rail temperature rise even more and this can be used. We note that the results for molybdenum rails coincide with those for tungsten within about 2 % in most cases.

9.2 The Proposed Plasma Armature Generation Package

Based on these results and the results obtained in the system study on the dependence of accelerator efficiency on the plasma species (Section 3.3), we propose to use a Li^+ -based plasma armature in the advanced ultra-high velocity accelerator. Though it is possible to generate the plasma armature entirely from blowing a fuze made out of raw lithium, this has two disadvantages: bare lithium oxidizes spontaneously in air, making practical handling difficult if not impossible. Secondly, blowing the lithium fuze involves the abrupt solid-to-vapor transition which is explosive in nature and is highly destructive to rails at the point of transition. We propose therefore to generate a Li^+ -based plasma armature by using a mixture of lithium nitrate, lithium carbonate, or lithium hydride and nitrocellulose or nitroglycerin in appropriate proportions.

Figure 54. Maximum temperature rise vs plasma armature velocity.
For different ionic species with the same armature mass under
the same peak operating conditions of SUVAC's booster

Figure 55. Maximum temperature rise vs plasma armature velocity. For different ionic species with the same armature mass under the same peak operating conditions of SUVAC's booster

10 Distributed Power System Design

10.1 SUVAC-I Booster Power Supply Subsystem

Details of the design of the capacitive energy module for the booster stage, SUVAC-I, are given in the reference⁸. The design emphasises modularity, serviceability, low-cost, interfaceability, and safety. One of the main design constraint of the module is that the module should be operable as a stand-alone facility which can be used for powering other on-going or planned pulse power experiments by Westinghouse.

The energy module for SUVAC-I comprises four sub-modules, each of which is a 100-kJ pack, giving a total energy of 400 kJ for the booster stage (Figure 56). Each sub-module consists of 10 Maxwell Laboratories high energy density pulsed capacitors (Cat. 33796) of 200 μ F each, giving a sub-total of 2 mF per sub-module and a total of 8 mF for the entire SUVAC-I module. The module is designed to operate at 10 KV, though only 8.5 KV is currently planned for powering the SUVAC-I booster accelerator. The schematic and circuit parameters of the entire module is shown in Figure 57.

Each sub-module is equipped with a ignitron for turning on the current from that sub-module. This ignitron is the main output switch. A second ignitron in each sub-module provides the crow-bar action and is the crow-bar switch. Ideally, the crow-bar ignitron is closed when the capacitor voltage is zero during the discharge. At this point, all the energy stored in the capacitors has been transferred into the external circuit, and by closing the crow-bar ignitron at this instant, one attempts to shunt the capacitor so that all the transferred energy would remain in the external circuit. However, this condition is rarely met in practice due to a finite forward bias required to activate the ignitron and the finite amount of inductance in the capacitor-crowbar circuit leading to trapping of a finite amount of energy in this circuit after the crowbar switch is closed. Detail computer simulations were made to assist in the selection of the circuit parameters and are given in the reference⁸.

In the present design, the maximum current capacity of the module is limited by the number and type of ignitrons used for switching and crow-barring.

Based on cost and present requirements, GL-37207A (GE catalog) ignitrons designed for fast capacitor discharge were selected. These ignitrons are rated for current up to 300 kA and a charge transfer of 200 C per shot before their service life (nominally rated at 200 shots) will be seriously affected. The use of four of these ignitrons for crowbar essentially limits the operating current output from the module to 1 MA. Only 280 kA is needed for the booster stage, SUVAC-I.

The transmission plates and bus work are designed to carry 1 MA from each sub-module (see later Section 10.3) giving a total current output from the module of 4 MA. By virtue of this design, the capacitor module is failsafe even in the case of fault current. For the commonly occurring faults including flash-overs across the capacitor terminals, the built-in inductance of the transmission manifold, the switches and the capacitor plates have been designed to limit the current to 4 MA in this module. The capacitor transmission plates and the crowbar ignitrons contribute a total parallel equivalent internal inductance of less than 50 nH, which is a sufficiently low inductance to limit the crowbar commutation time and ripple to an acceptable level for our applications. The transmission manifold of the module is designed in such a way that there are sufficient inductance for current isolation between each sub-module. Not only does it limit the current in case of fault, it also enhances the firing of the crowbar ignitrons.

The module is equipped with an external inductor whose inductance can be varied between 1 μ H to 8 μ H. The variability of these inductances provide a wide range of tunability of the system parameters for the purpose of experimentation and is highly desirable to the experimental program. It allows pulses of variable length and amplitude to be generated. The capacitance of the module can also be varied by varying the number of sub-modules used in the experiment. The specifications of this power subsystem is shown in Table 7.

10.2 Capacitor Energy Modules for SUVAC Main Accelerator

For the main stages of SUVAC accelerator, a highly flexible system of distributed energy stores have been designed. The emphases in the design are again modularity, serviceability (including accessibility), low-cost, interfaceability, and fail-safe operation. Details of the design are given in the reference⁹.

Table 7. SUVAC-I BOOSTER CAPACITOR MODULE.

Specifications		
Voltage	10 kV	
Number of submodules	4	
Capacitance per submodule	2 mF	
Energy per submodule	100 kJ	
Total capacitance,	8 mF	
Total energy	400 kJ	
Maximum output current (switching limited)	1.0 MA	
Maximum output current (transmission limited)		4.0 MA
Total internal (crowbar) inductance	50 nH	
Variable external inductance	0.5 to 8 μ H	

Table 8. The 2.1-MJ Distributed Energy System for SUVAC Main Accelerator.

Specifications		
Voltage	10 kV	
Number of submodules	21	
Capacitance per submodule	2 mF	
Energy per submodule	0.1 MJ	
Total system energy	2.1 MJ	
Maximum output current per submodule	300 kA	
Internal (crowbar) inductance	120 nH	
Variable external inductance	1 to 4 μ H

Figure 56. SUVAC-I Capacitor Bank.

Figure 57. Circuit Schematic and Parameters for SUVAC-I Capacitor Module.

By considerations based upon system requirements, accessibility, ease of assembly and disassembly, switching requirements, complexity of the bus work and capacitor plates, practicability, and cost, it was decided that 100 kJ is a convenient unit of energy for the system. Maxwell Laboratories state-of-the-art high-energy-density low-inductance pulsed capacitors (Cat. 33843) were identified to be the capacitors for the application. Each of these capacitors are rated at 330 μ F, 10 kV, 16.5 kJ, 100 kA, < 30 nH, and a life of at least 10,000 cycles at 20 % voltage reversal.

Each of the SUVAC module consists of 6 of these capacitors, arranged with a rectangular base area of 24 in x 28 in and placed on insulating materials on a cart equipped with casters (Figure 58).

The current transmission plates are mounted atop the capacitors and connected to the anode and cathode (ground ring) of the capacitors, respectively. The capacitors are equipped with a central anode stud and a special ground ring for low-inductance electrical connection to be made.

The capacitor modules have provisions for the switch assembly mounted off to one side of the cart. This framework provides the support for the ignitrons and their coaxial mounts. One small transmission plate carry the current from the capacitor module to the ignitron anode and another is used as the positive output terminal of the module of the capacitor bank. The design of the modules had to be done in such a fashion as to permit the assembly to be configured with a single ignitron or more (upto a maximum of 3) as required if the capacitor modules are to have crowbar capability. A feature of the design is that it allows for easy changes to be made to the module to accommodate changes to the system parameters if required by the experimental program in the future. We expect that minor adjustments of the system parameters would be required throughout the experimental program depending upon the results of actual experimentation.

Figure 58. Mobile Capacitor Energy Modules for SUVAC Main Accelerator.

10.3 Electrical Connection to the Capacitor Terminals

Electrical connection of the capacitor to the transmission plate is made through a brass hex nut. The length of the brass pieces was selected to provide the modules with as much plate separation as is possible without exceeding the maximum inductance (30 nH) which is permitted for the capacitor plates. The first hex nut and washer near the base of the capacitor anode stud serve as a seal for the O-ring around the stud to prevent the dielectric oil from leaking out of the capacitor. The replacement to the capacitors original seal nut was done to gain the necessary thread length for proper engagement of the major current carrying hex nut. The beville washers serve as a constant preload on the jam nut to prevent any vibration during the pulse from loosening the assembly. The jam nut is used to lock the top hex brass extender threads to the capacitor anode stud terminal. The insulation used to prevent flashover between the capacitor anode and the ground ring consists of a plexiglass tube surrounding the stud assembly with RTV silicone sealant at the base of the capacitors plastic header assembly piece. The plexiglass tubing is also sealed against the top transmission plate.

10.4 Containment of magnetic pressures

Due to the high currents expected from the capacitor modules, The capacitor modules had to be designed with some form of constraint to contend with the plate separating forces. The transmission plate constraints were designed to contain the magnetic pressure developed from a current of 1 MA at the output of the module. The actual current expected in a capacitor module will not exceed 250 kA under normal operation. Since the forces are a function of the square of the current, This provided us with a safety margin of nearly 16 times the normal operating forces. An additional factor is the pulsed nature of the current waveform. Even without a detailed analysis of the inertial effects, a much greater safety margin is evident. The maximum expected fault current expected through a capacitor module will not exceed 2 MA by virtue of the choice of internal inductance for the capacitor modules. In a fault, the time that the plates see the load is on the order of 1 ms at most. Due to the inertial of the transmission plates, the plates will move no more than a fraction of an inch under fault conditions and this was considered sufficient safety for the modules. It is,

however, of prime importance that, during normal operating conditions, the plates will sustain as little motion as possible to ensure the integrity of the electrical contacts. This requirement is met by over-designing the constraints to contain times the force 16 times over and DC.

11 Construction of Accelerator: Plan of Action and Status Report

The funding cycle, the cost of the accelerator, the schedule of program milestones, manpower capacity and management, and the requirement for preliminary testings to close the design loop, all contribute in varying degree of complications in developing a strategy in the fabrication of the accelerator. The final strategy selected is as follows:

1. Fabricate booster accelerator (Year 1)
2. Fabricate booster vacuum tank (Year 1)
3. Assemble booster accelerator and vacuum tank (Year 1)
4. Test firing booster accelerator (Year 1)
5. Fabricate SUVAC-II (2nd and 3rd stages) and muzzle section (Commencement: Year 1, 4th Quarter; Completion: Year 2, 1st Quarter)
6. Fabricate vacuum tanks for SUVAC-II (Commencement Year 1, 4th Quarter; Completion: Year 2, 1st Quarter)
7. Assemble SUVAC-II, muzzle section and vacuum tanks (Year 2, 1st Quarter)
8. Test firing SUVAC-II (Year 2, 2nd Quarter)
9. Formulate design modifications if necessary (Year 2, 2nd Quarter)
10. Fabricate SUVAC-III (4th, 5th and 6th stages) (Year 2, 2nd Quarter)
11. Fabricate vacuum tanks for SUVAC-III (Year 2, 2nd Quarter)

12. Assemble SUVAC-III, muzzle section and vacuum tanks (Year 2, 3rd Quarter)
13. Test firing SUVAC-III (Year 2, 3rd Quarter)
14. Formulate further design modifications if neccessary (Year 2, 4th Quarter)
15. Fabricate SUVAC-IV (7th, 8th, 9th and 10th stages) (Year 3, 1st quarter)
16. Fabricate SUVAC-IV vacuum tanks (Year 3, 1st Quarter)
17. Assemble SUVAC-IV and vacuum system (Year 3, 2nd Quarter)
18. Test fire SUVAC-IV (Year 3, 4th Quarter)

Fabrication of the booster accelerator, SUVAC-I, is fully completed together with its own muzzle assembly for containing the muzzle blast. Fabrication of the vacuum tank for this stage is also completed. The entire system, the accelerator, the muzzle assembly, the vacuum tank and the ballistic range has been assembled and installed. The system is ready for testing and commissioning in the month of May 1984. In June, the system will be ready for systematic experimentation.

Fabrication of the additional stages (2nd and 3rd) and the "muzzle section" required to implement SUVAC-II has been initiated since mid-April 1984. The "muzzle section" is essentially just another stage of the accelerator barrel equipped with muzzle assembly but without a power supply. Its function is to prolong the length of the accelerator so that the residual magnetic energy stored in the system can be dissipated in a more controlled way instead of an abrupt "explosion" at the muzzle. This is neccessary because the last stage (3rd stage) of SUVAC-II is designed to provide current to flow in the 3rd and 4th stages of the complete accelerator SUVAC-IV. A muzzle assembly will also be provided to absorb any residual energy in the form of a muzzle blast. Delivery of these barrel sections is expected in July 1984, when assembly will begin.

Fabrication of the additional vacuum tanks required for SUVAC-II has also begun since mid-May 1984. They will also be delivered in July 1984.

12 Construction of the Distributed Power Supply: Plan of Action and Status Report

Factors similar to those which dictate the choice of strategy in accelerator construction also provided the constraints in developing the plan of construction of the distributed power supply, with one additional requirement: the development of the power supply is required to pre-date the accelerator development in all phases of the experimental program. Because of the complexity of its control, charging and safety requirements, its development, testing and commissioning sequence are necessarily longer and requires more effort. Its development is tied in closely with the development of the new laboratory because of its requirements for space, power, warning, interlocks and safety system. For these reasons, construction of the distributed power supply and the development of the new laboratory has begun as early as January 1984.

All the components for SUVAC-I power supply subsystem consisting of the four capacitor sub-modules has been completely fabricated, assembled and installed. Testing and commissioning is currently underway. Results of these tests will soon be reported.

As for the capacitor energy modules for the main accelerator (SUVAC-II, SUVAC-III, SUVAC-IV), a complete set of the capacitor transmission plates, the carts, the frames, the insulating base, and the ignitron plates for all the 22 modules has been fabricated. Orders for the electrical and electronics components required to build the control and firing circuits sufficient for the next two stages of the accelerator (i.e. to implement SUVAC-II) have been placed and 90 % of these components have been received. Delivery of the remainder is expected in June. Mechanical assembly of six of the capacitor modules have been completed. Assembly of the electrical and electronics equipments and the development of the programmable controller for firing SUVAC-II is on the way, and is expected to be ready by mid-July when testing and commissioning of the SUVAC-II power supply subsystem will commence.

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