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Final Report

April 1966

THE HIGH-G LINEAR ACCELERATOR--A FEASIBILITY STUDY

By: E. G. CHILTON

Prepared for:

OFFICE OF CHIEF OF ENGINEERS
DEPARTMENT OF THE ARMY
WASHINGTON, D.C. 20315

CONTRACT DA 49-129-ENG-556

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141

STANFORD RESEARCH INSTITUTE

MENLO PARK, CALIFORNIA

SRI

ERRATA

for Final Report, April 1966, "The High-G Linear Accelerator--A Feasibility Study" for Office of Chief of Engineers, Department of the Army, under Contract DA-49-129-ENG-556 (SRI Project PHU 5656).

Page 1 Line 4 - read attained instead of obtained.

43 Under "Multiple Tank Pneumatic Drive," first line, last word should read "analysis"

56 Table V, heading in column 2 should read "Can it Fulfill the Requirements"

57 Paragraph 2, line 2 should read ..... "a total of $175,000."

58 Penultimate paragraph, fifth line, first word should read "accompanying."

58 Last paragraph, first line, fourth word should be "case."

58 Last line - "alignment."

60 First line should read ..... "from $80,000 to $14,000,000."

60 Last paragraph, first line, penultimate word should read "probable."

65 Equation (A-1), delete "/" between τ and ω

65 Equation (A-2), replace "ρ" by "p"

65 Equation (A-5), replace "$(1/\tau\omega/^2$" with "$(1/\tau\omega)^2$"

68 Equation (A-7), after ρ add $_v$ to read "ρ_v"

112 Add square root signs to second equation:

$$M_2 = \frac{V_2}{\sqrt{kRT_2}} = \frac{V_2}{49.02\sqrt{T_2}}$$

112 Center of page, following "and the mass velocity, G, by" the equation should read "$G = w/A = \rho V$"

113 Add square root sign in fourth equation:

$$G' - W/A = p_0\sqrt{g_c \frac{k}{RT^0}}\ M,/\ldots..$$

113 In fifth equation read

$m(K.E.)_f$ in place of $m(K.D.)_f$

118 Add brackets to right hand side of first and second equations:

$$\ldots.. = \frac{1}{\eta_s} Mc_p T_a \left[\left(\frac{p_{oi}}{p_a}\right)^{\frac{k-1}{k}} -1\right]$$

$$(W_k)_2 = \frac{1}{\eta_s} m c_p T_a \left[\left(\frac{p_{oi}}{p_a}\right)^{\frac{k-1}{k}} -1\right]$$

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A FEASIBILITY STUDY

Stanford Research Institute Contract DA-49-129-ENG-556

ERRATA SHEET

Page	Paragraph	Line	Comment
10	Table II	10	Peak Power, delete "(10^{-3})", "115,000" "52,000", insert "116,000,000","52,000,000"
10	Table II	11	Delete "94,000" insert "95,000"
43	Multiple Tank Pneumatic Drive	1	Delete "snalysis" insert "analysis"
56	Table V	1, Col 2	Delete "if" insert "it"
57	Research Cost	2	Delete "195,000" insert "$175,000"
58	Vehicles	5	Delete "accompnaying" insert "accompanying"
58	Air Bearing Guide Rails	1	Delete "base" insert "case"
58	"	6	Delete "laignment" insert "alignment"
60	First	1	Delete "$14,000" insert $14,000,000"
60	Incremental cost of system variation	1	Delete "probably" insert "probable"
65	Equation A-1		Delete the slash between tau and omega
65		10	Delete "definite" insert "definition"
65	Equation A-3		Delete p^2 in the numerator and insert rho squared
65	Equation A-5		Delete the slash and insert a bracket for the one over tau omega term squared in the denominator
68	Equation A-7		Add the subscript "v" to rho
112		15	Delete "C" insert "G" in Equation G=W/A

STANFORD RESEARCH INSTITUTE

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A FEASIBILITY STUDY

Prepared for:

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CONTRACT DA 49-129-ENG-556

By: E. G. CHILTON

SRI Project PHU-5656

Approved: R. B. VAILE, JR., ASSOCIATE DIRECTOR
CHEMICAL, THEORETICAL, AND APPLIED PHYSICS

Copy No. ....

ABSTRACT

Inertial guidance systems for tactical missiles must perform during the boost period when, for less than one second, the entire missile is accelerated at very high rates. It is desirable to propel the guidance system on a test vehicle which can be accelerated in a horizontal straight line for a few tenths of a second at up to 200 g, whose motion can be carefully controlled, and which can be brought to a stop after the test. Therefore, the feasibility of various propulsion methods was studied and a comparison made of vehicles driven by linear electric motors, compressed gas, and rockets. This study indicates that the highest probability of success lies with a combination pneumatic and constant frequency linear induction motor drive.

The installation would consist of a pneumatic piston which accelerates the test vehicle to about 200 ft/sec, after which the induction motor drive takes over. The electrical system comprises a three-phase alternator rated at about 280 megawatts for short-time operation, a linear stator along which the test vehicle moves, and a power control circuit which selectively feeds power from the alternator to the stator. Beryllium fins on the vehicle make up the translator of the linear motor. A Doppler-radar control system provides feedback control. Deceleration is by dc coils in the stator. It is estimated that the entire unit can be designed and built in about four to five years at a cost of about $15 million.

Alternate systems which show promise are based on variable frequency ac and on hybrid fuel rockets. Either or both of these alternate systems may be less costly than the constant frequency systems, but both require more research and development effort and are less assured of success.

ACKNOWLEDGEMENT

This research was performed for the Office of the Chief of Engineers under Contract DA 49-129-ENG-556. The work was monitored by Mr. Harold H. McCauley of OCE. Among the contributors to this study at Stanford Research Institute (SRI) were: H. T. Albachten (multiple tank pneumatic drive); J. S. Arnold (power and control systems); E. J. Klein (single tank pneumatic drive); V. M. Lieskovsky (air bearing supports); A. E. Moon and C. R. Self (economic and cost studies); H. L. Murphy and J. E. Van Zandt (buildings, foundations and structure); H. R. Ross (linear induction motor); V. Sanford and E. E. Spitzer (detail designs); T. D. Witherly (rocket sled drive, pneumatic booster). In addition, contributions were made by the following consultants: Dr. S. Beckwith, Mr. H. C. Hitchcock and Dr. W. G. Hoover (electrical power systems); Dr. R. H. Eustis (MHD power generation); and Dr. D. B. DeBra (requirements of inertial guidance systems).

CONTENTS

TABLES

FIGURE CAPTIONS

I INTRODUCTION

Tactical missiles used by the U. S. Army are accelerated by powerful rockets during a very short, initial portion of their flight. Typically, accelerations as high as 200 g are maintained for 0.1 to 0.2 seconds, after which the missile has obtained supersonic velocity and proceeds on its path without further power. The inertial guidance system must perform its function during the acceleration period while power for guidance is available. This exacting task must be performed in the high-g environment.

The Army Missile Command at Redstone Arsenal, Huntsville, Alabama, whose functions include the testing of tactical missiles and their guidance systems, is in need of a device which can simulate the acceleration portion of a missile's flight. Today, guidance systems are tested statically (one g maximum) and/or test-fired in the complete missile. Firing tests are unsatisfactory not only because of their high cost but also because they do not cover a wide and controllable "g" range, nor is it usually possible to recover the guidance system for later study. What is needed is a device which carries the guidance system or some of its components, together with suitable measuring and recording equipment. This device or vehicle must be accelerated for periods of about 0.1 to 1 sec along a straight line at a constant preselected g-value. It would then be slowed down at a rate lower than the one at which it was speeded up.

The need for such a device has been recognized since before 1960. In that year the Missile Command investigated some proposals for the design and construction of such a unit. A feasibility study[1]* of one proposed design showed the difficulties likely to be encountered with linear magnetic drives. Early in 1965 the Missile Command decided to obtain a thorough survey of methods that might have a chance to meet their

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* References are listed at end of report.

requirements and requested the Office of the Chief of Engineers (OCE) to commission and supervise this study. The result was Contract DA 49-129-ENG-556 between Stanford Research Institute (SRI) and OCE. This is the final report under that contract.

The specifications on which this study were based are shown in detail in Appendix G and are condensed in Table I. Two kinds of payload were specified: the complete guidance system weighing up to 200 lb including a jig which can rotate the system about an axis in the direction of motion; and certain of its components weighing up to 50 lb. The complete system must reach acceleration levels of 60 g, the components 200 g. Acceleration periods should last for 0.6 and 0.15 sec, respectively, and longer for lower accelerations. The exacting limits on control and cross-axis acceleration should be noted.

TABLE I

BASIC REQUIREMENTS

	Payload 50-lb	Payload 200-lb
Maximum Acceleration	200 g	60 g
Duration of Acceleration	0.15 sec	0.6 sec
Control	±5 g	±0.5 g
Minimum Vehicle Diameter	12 in	25 in
Allowable Cross-Axis Acceleration	3 g	
Allowable Peak Temperature	200°F	
Allowable Magnetic Field	2 gauss	
Number of Test Runs	8 per day	
Number of Test Runs	1000 per year	

II SUMMARY, CONCLUSIONS AND RECOMMENDATIONS

Basic requirements can be met with today's technology and a moderate amount of additional research and development, with the possible exception of the cross-axis acceleration limit whose achievement is not assured. The system will be costly and will require several years to design and build.

Analysis of the basic requirements indicates the need for an energy storage system capable of delivering power at a level of more than 100 megawatts for fractions of a second. This can be accomplished mechanically (flywheel; compressed gas) or chemically (fuel and oxidizer). Electrical energy storage (batteries or capacitors) is not practical.

Three basic propulsion methods have been investigated: the linear induction motor powered by a flywheel driven generator or a magneto-hydrodynamic power generator (MHD); the compressed gas driven piston in an evacuated tube; and the rocket sled. Two variants of each of these methods and some combinations have been analyzed for feasibility. Specifically, we have looked at:

- Constant frequency linear induction motors
- Variable frequency linear induction motors
- Single tank pneumatic systems
- Multiple tank pneumatic systems
- Conventional solid fuel rockets
- Hybrid fuel rockets

Potentially all these methods are capable of storing enough energy and delivering enough force to reach the required acceleration levels. However, none of them meet all the requirements at today's state of the art. The amount of research and development required to make each method acceptable, the likelihood of success of the research, estimates of flexibility and the cost of the resulting installation were used to select among the methods. The constant frequency linear induction motor with a pneumatic booster was selected for highest feasibility, with the variable frequency motor and the hybrid fuel rocket close seconds.

The constant frequency system would consist of:

- A 240 cycle 14 kV 3-phase alternator capable of delivering 280 MW of power in the short circuit mode for about 0.12 sec and 130 MW for 0.5 sec. Its rotor is about ten feet in diameter, 3.8 feet long and weighs about 70 tons. It is big enough to act as its own flywheel with a nominal speed of 960 r/min which is reduced less than ten percent by the most severe power demand contemplated. The alternator is brought up to speed by a direct coupled 1000 hp motor taking its power from the line.
- A power controller, conceived as sets of inverse-parallel connected ignitron pairs in each phase. The controller transmits only a fraction of each half cycle, the size of the fraction selected by the power demanded at the vehicle. It is likely that the ignitrons can be replaced by silicon controlled rectifiers.
- A pneumatic booster which accelerates the test vehicle to about 200 ft/sec after which the magnetic drive takes over.
- The linear stator consisting of two sets of opposed magnetic cores with horizontal flux gaps. The stator is 1,050 feet long and consists of three sections: a 75-foot-long, high power section for accelerations in excess of 60 g; a 275-foot-long, medium power section; and a 700-foot-long, deceleration section. The pole pitch in the accelerator section is about 40 inches. Shunt capacitors totalling 6.2×10^6 kVAR are provided along the acceleration sections for power factor correction.

- The vehicle, which holds the payload, a data tape recorder, and a magnetic shield to protect payload and recorder. A pair of translators, attached like wings to the vehicle, transmit the propulsive power. They are made of beryllium alloy, because of its high strength, high heat capacity, and moderate electrical resistivity. The vehicle runs on four, cylindrical, hydrostatic air bearings fed from a pressure tank inside the vehicle.
- A Doppler radar acting to measure the velocity of the moving vehicle.
- A control center, which programs the desired acceleration pattern and sets the power controller accordingly, compares the actual velocity as measured by the radar with the desired pattern and adjusts the power controller to suit, records the measured velocity and acceleration, and performs the appropriate sequence switching.
- Auxiliaries, such as buildings and support structures, safety devices and dollies to move the vehicle outside the stator.

The performance of air bearings at high speeds and their effect on cross-axis acceleration and the detail design of the translators require further research. Development work is needed on the power controller, and the analysis of the high transients produced in the system.

The cost of the research tasks is estimated at $175,000. Development, design and the construction are estimated at about $14,800,000; of this sum, more than half is required for the stator and the associated capacitors. Whenever possible, cost quotations have been obtained from manufacturers. The greatest uncertainty in the cost estimate is the power controller ($1,500,000), quotations for which have ranged from $80,000 to $14,000,000.

It is recommended that

- Research programs be initiated to study the two unsolved problems for the constant frequency linear induction motor,

  (1) The design of air bearings and their effect on cross-axis acceleration,*

  (2) The design of the translator, selection of material, method of construction and electrical and mechanical properties;*

- Methods for converting high power levels of dc or constant frequency ac to controlled variable frequency ac be thoroughly investigated to permit the use of a variable frequency linear induction motor drive;*

- The possibility of using hybrid fuel rockets be studied, with particular attention to the rapid control of thrust.

- The Army's requirements be carefully reviewed. Appreciable savings can be obtained if payload weight or duration of acceleration can be even slightly reduced, or if the requirement for control of acceleration can be relaxed. A large saving could be obtained if acceleration control and measurement could be confined to the deceleration portion of the run.

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* For details see Appendix F.

III THE PROBLEM

STATE OF THE ART

Acceleration testing is not a new art. A tabular comparison of some existing acceleration devices is shown in Fig. 1. Among common devices which will apply controlled accelerations to a test object are impact testers and vibration exciters, neither of which will meet durations specified for the project. Centrifuges are also common, but their non-linear motion is unacceptable because of its effect on directional gyros and gyro-accelerometers in the guidance packages.

There are other devices which can produce linear accelerations of longer duration. Among these are the catapult, the light-gas gun, the vacuum tube, and the rocket sled. Catapults are generally designed for low accelerations (1 g or less) and when control is not important. To gain better control and a smoother takeoff, two linear induction motor driven catapults were built by Westinghouse Electric Corp. in the early 40's.[2] The light-gas gun is specifically designed to accelerate small particles (simulating meteorite dust) to hypervelocities. The evacuated tube as an accelerating device has been demonstrated by Duke University[3] for the Army and it has also been proposed as a missile launching device, although not at the values of accelerations needed here.

Rocket sleds, finally, such as those now in operation at Holloman Air Force Base, New Mexico, and the Naval Ordnance Test Station, China Lake, California, can provide the necessary acceleration and for a long enough time. Control, particularly feedback control, however, is difficult to achieve. Nevertheless, rocket sleds come near enough to satisfy the requirements that they were included in the feasibility study.

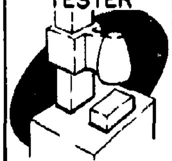
	CATAPULT 	ROCKET SLED 	CENTRIFUGE 	IMPACT TESTER 
MAXIMUM ACCELERATION	NO	YES	YES	YES
MAXIMUM FORCE	YES	YES	YES	YES
DURATION	YES	YES	YES	NO
CONTROL	NO	NO	YES	YES
LINEARITY	YES	YES	NO	YES
X-AXIS ACCELERATION	NO	NO	YES	YES

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FIG. 1 COMPARISON OF EXISTING ACCELERATORS

PROJECT ORGANIZATION

Three basic propulsion methods were initially selected for feasibility study--the linear induction motor, the pneumatic tube, and the rocket sled. During the actual study, all three methods were investigated simultaneously during the first three months. Then the comparison was made, pointing to the constant frequency linear motor as the system of highest success possibility. Thereafter we confined our studies to that system alone. It was carried to enough technical detail that a cost analysis could be made and the remaining research problems could be identified.

The hybrid fuel rocket did not come to our attention until about a month before the termination of the project. Hence, its discussion is necessarily short and incomplete.

BASIC CONSIDERATIONS

Some basic consequences can be drawn from the requirements on Table I (p. 2). For instance, an acceleration of 60 g applied for 0.6 sec moves the vehicle a distance of $60(32.2)(0.6)^2/2 = 348$ ft, at which time its velocity is $60(32.2)(0.6) = 1160$ ft/sec or just supersonic. If the vehicle so accelerated were to weight 750 lb, the force required to accelerate it (neglecting friction and air resistance) would be $60(750) = 45{,}000$ lb; the power required would rise to a peak of $45{,}000(1160) = 52 \times 10^6$ ft-lb/sec (95,000 hp or 71 MW). The total energy applied and stored as kinetic energy is $71(0.6)/2 = 21.3$ MW-sec or about 6 kW-hr. These and other conclusions are summarized in Table II.

The estimated total length of the track consists of 350 ft for acceleration and 700 ft for deceleration. This implies a deceleration rate of 30 g from the peak velocity of 1160 ft/sec reached at the end of the 60 g run. Total vehicle weights are based on an estimated empty vehicle weight of 550 lb. If this weight can be reduced, both force and power required will go down in direct proportion.

TABLE II

FORCE AND POWER REQUIREMENTS

Payload (lb)	50		200
Acceleration (g)	200		60
Acceleration (ft/sec^2)	6,440		1,930
Time at Acceleration: t (sec)	0.15		0.6
Maximum Velocity: v_m = at (ft/sec)	965		1,160
Accel. Track Length: L = 1/2 v_mt (ft)	73		348
Total Track Length (ft)(est.)		1,050	
Gross Weight of Test Vehicle (lb)(est.)	600		750
Force Required: F (lb)	120,000		45,000
Peak Power Required: P = Fv_m (lb-ft/sec)(10^{-3})	115,000		52,000
(hp)	210,000		94,000
(kW)	157,000		71,000

The power required to drive the vehicle goes from zero at the start to the enormous values shown in Table II at the end of the acceleration run. Since such burst of power cannot be drawn from normal electrical power supply lines, it becomes necessary to store the energy for the run in a system from which it can be withdrawn at the rates required. Mechanical, chemical and electrical storage methods were considered.

We concluded early in the study that electrical or electro-chemical storage, batteries or capacitors, was impractical. Conventional storage batteries, such as lead-acid cells, will not deliver high power rates. A notable exception is the sintered nickel-cadmium cell which is reported to deliver very high, short duration currents. However, even this cell would not be able to deliver the power required unless the total energy stored was 10 to 100 times that needed.

Capacitors have the problem of voltage decrease as energy is withdrawn and hence here, too, considerable excess energy must be stored if power is to be withdrawn at an increasing rate. Based on ten times the required energy, or a total stored energy of 60 kW-hr or 216x10^6 joules and a voltage of 20 kV, the capacity needed would be about 1.1 farads. At an estimated cost of about 15$/μF, this capacitor bank would cost $16 million, not to mention the equipment required to charge the capacitors and the cost of the system that can use the dc power.

The mechanical energy storage systems include flywheels and compressed gases. These systems also prefer to deliver energy at decreasing rates not at the increasing rates demanded by the linear accelerator. Therefore, it will be necessary to store a considerable excess of energy. (The system selected stores about five times the energy required in each run.)

Among chemical energy storage systems only rocket engines--both solid and hybrid--were investigated. The rocket engine is not power limited but thrust limited.

Very considerable system simplification and large cost reductions could be obtained if the requirements permitted uncontrolled acceleration at relatively low levels followed by controlled deceleration. The objections to this approach appear to lie in the difficulty of establishing instrument zeros while the vehicle is travelling, of uncaging gyros without exerting disturbing torques, or of measuring the effects of the uncontrolled acceleration. It was not a function of this study to evaluate these objections.

IV THE LINEAR INDUCTION MOTOR

Acceleration control--to better than 1% in a few milliseconds--is one of the most difficult requirements of the desired accelerator. Such close control can be achieved most readily in electrical systems.

Almost any type of electric motor can be "unfolded" or developed and made into a linear motor. The squirrel-cage induction motor has the specific advantage of not requiring slip rings on its rotor. Thus, if the rotor is changed into the moving element, no power pick-offs will be required.

The principle of the linear induction motor has been known and applied for over 20 years.[2] Even prior to that time, electromagnetic guns were proposed and tested in France, Germany, and Japan. In the U.S., research in magnetic propulsion of projectiles was pursued intensively in the late 50's and 60's.* Some of the most thorough studies to date were performed by Laithwaite[4] and his staff at the University of Manchester. We have extrapolated Laithwaite's work into power levels several orders of magnitude greater than his. This can be done with confidence because the basic analyses and equations have been shown to be valid.

The linear induction motor is a flattened out (or unfolded) version of the familiar polyphase induction motor. One element--either the stator or rotor--is stationary and extends along the entire acceleration section. The other element is attached to the vehicle and is, therefore, appreciably shorter than the track. This leads to end effects which are absent on the rotary machine. Also, the linear motor is less able to absorb the repulsive or attractive forces that exist between stator and rotor unless a two-sided stator is used (see Fig. 2). The two-sided stator has the additional advantage of providing a greater and more uniform magnetic flux than would be possible with a single-sided stator.

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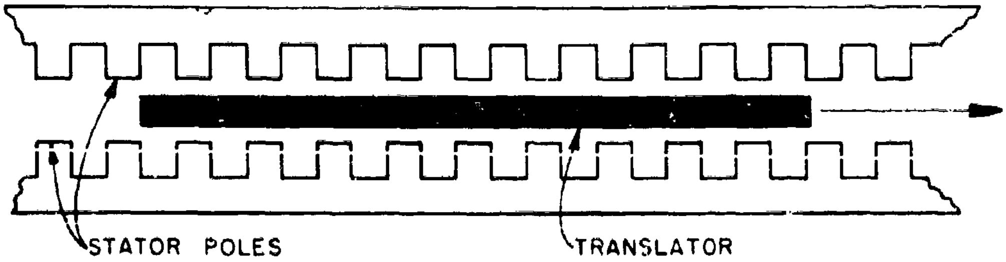
* A good bibliography of U.S. and foreign studies may be found in Ref. 5.



1. TRANSLATOR OF CONDUCTING MATERIAL – NO SLOTS
2. USE OF DOUBLE-SIDED MOTOR
3. USE OF NON-FERROUS CONDUCTING MATERIAL FOR TRANSLATOR

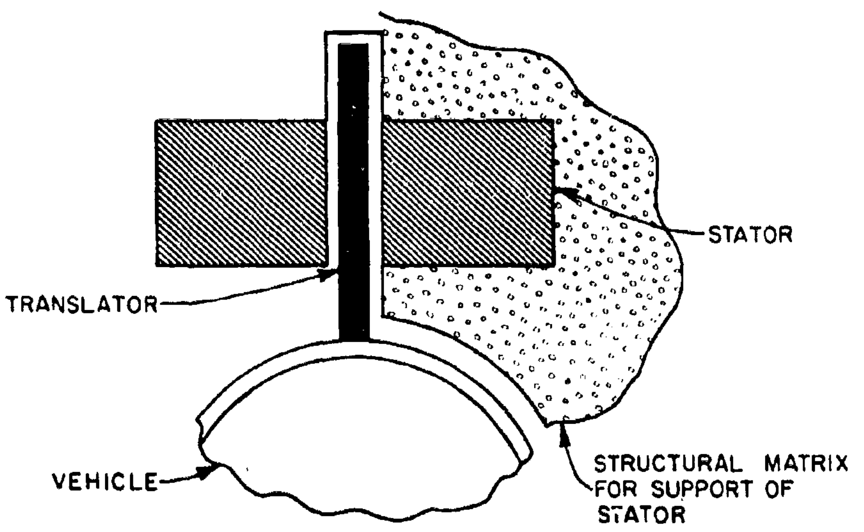


FIG. 2 BASIC DEVELOPMENT OF LINEAR MOTOR

In the present application there is little doubt as to which should be the stationary and which the moving element. The squirrel-cage rotor, which requires no electric power pick-offs and can be made much lighter per unit length than the stator, is ideally suited for the moving element. Since it moves along a straight line and does not rotate, we shall henceforth call it a translator instead of a rotor.

The force or thrust developed in the linear motor (see Appendix A) is a complicated function of stator and translator geometry, of the flux developed in the gap, the resistivity of the translator, and the slip or relative velocity between the travelling magnetic field and the translator. Fig. 3 shows the effect of slip and translator resistivity on thrust.

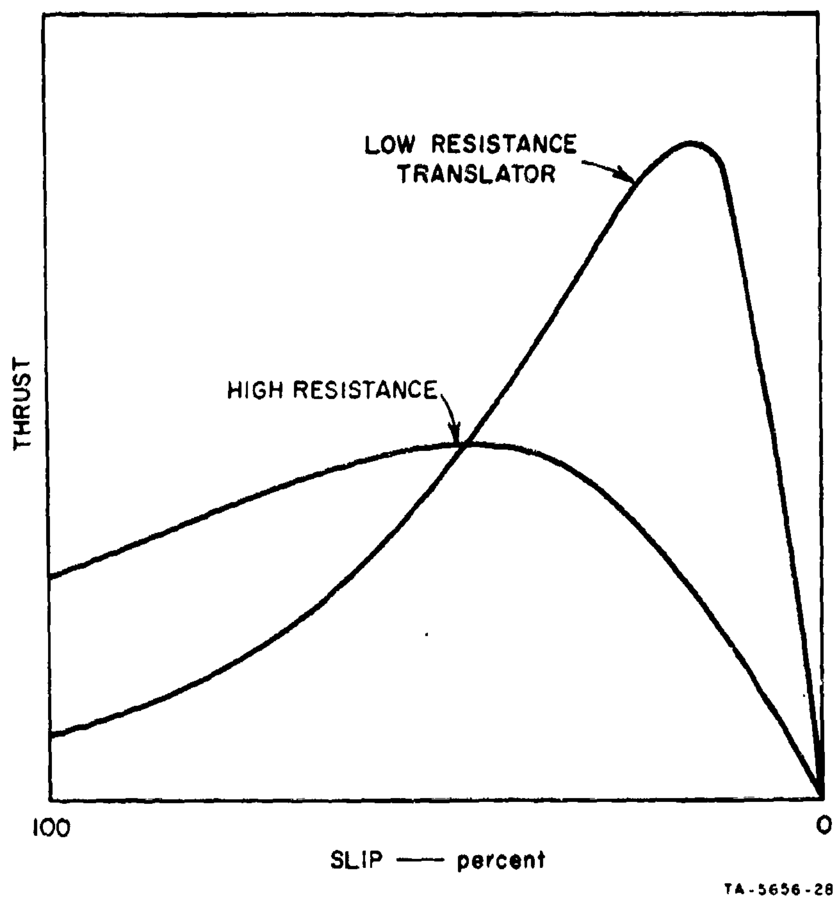


FIG. 3 THRUST-SLIP CURVES FOR LINEAR INDUCTION MOTOR

The synchronous speed of the magnetic field is given by

$$v_s = 2pf \qquad \text{(ft/sec)}$$

where p is the pole pitch in the stator (ft) and f is the frequency of the field (c/sec). Synchronous speed is constant if pole spacing and frequency are constant. The speed of the translator, however, varies linearly from zero to a maximum. Hence the slip, expressed in percent of the synchronous speed, v_s, decreases linearly from 100% to some value greater than zero. In a constant acceleration motor, thrust should be held constant regardless of slip. This can best be accomplished with a high resistivity translator, provided it is used at slips in the vicinity of the peak thrust. It implies that the translator is brought to a reasonable slip by means other than the magnetic drive and that the peak velocity be substantially below synchronous speed. It also implies substantial heat energy delivery to the translator and a consequent temperature rise during the run.

The alternatives are a variable resistivity translator (similar to a wound rotor induction motor), or a variable synchronous speed. The first requires additional weight of the control gear on the vehicle. Synchronous speed can be changed most effectively, and continuously, by providing variable frequency power.

Deceleration can best be accomplished in a stator of similar configuration, but excited with dc, so that the translator alternately cuts magnetic fields in opposite directions. The resultant decelerating force is nearly constant for a large range of speeds and reduces substantially only after the speed has slowed to a small fraction of its peak velocity. From that point on mechanical braking methods must be used.

Possible power arrangements are shown in Fig. 4. The simplest one, shown by the heavy lines, will be discussed in detail. The possibilities for variable frequency power will be discussed later.

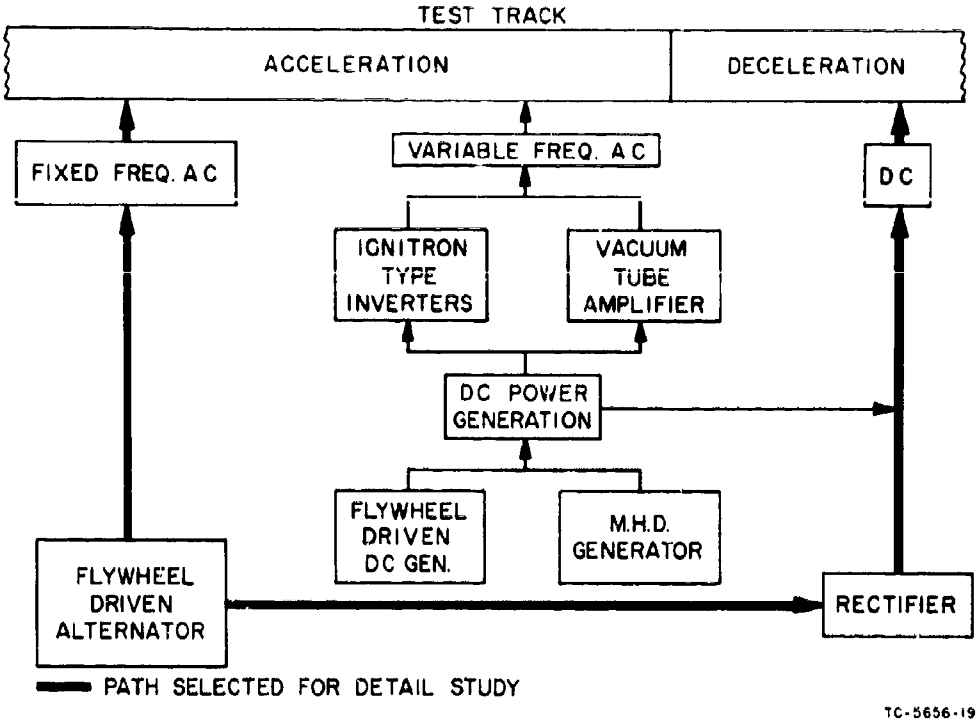


FIG. 4 POSSIBLE POWER ARRANGEMENTS FOR LINEAR MOTOR

CONSTANT FREQUENCY LINEAR INDUCTION MOTOR

Components

The heart of the proposed facility is the power and control system. This is shown schematically in Fig. 5 and consists of:

(1) Primary power supply

(2) Power controller for both acceleration and deceleration

(3) Stator, including the pneumatic starter

(4) Vehicle and its translators and bearings

(5) Sensing system and signal processor

(6) System controller and data recorder.

In addition, the system needs

(7) Buildings and foundations

(8) Stator constraining structure

(9) Auxiliaries such as vehicle handling equipment, safety devices, final mechanical braking, translator cooling facilities.

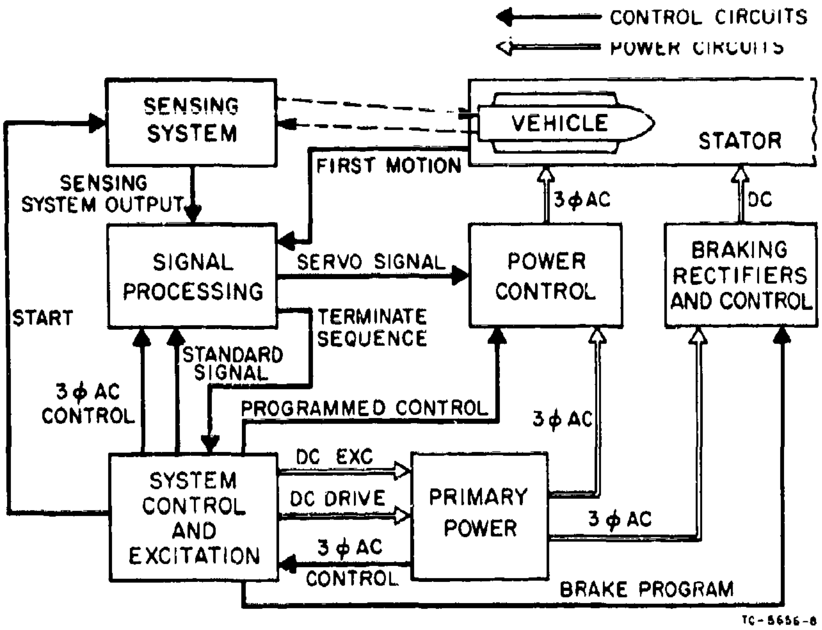


FIG. 5 POWER AND CONTROL SYSTEM — CONSTANT FREQUENCY MOTOR

In the following paragraphs each of these parts is discussed separately, despite the fact that they are closely interrelated and that, during the study, the calculations for each part had to be revised repeatedly until a coherent system resulted. Details of the calculations are reported in Appendix A.

(1) The primary power supply includes energy storage, conversion and distribution equipment. It consists mainly of the alternator, the alternator drive motor, the rectifiers to supply dc power for excitation of the alternator field and for the alternator drive motor (if a dc motor is selected), and the necessary circuit breakers and distribution facilities.

The alternator stores mechanical energy in the angular momentum of its rotor and delivers this energy as electrical energy to the power controller and thence to the track. Since the introduction of the vehicle into the track will appear very similar to a symmetrical short circuit to the alternator, the electrical, magnetic, and mechanical characteristics of the alternator construction should be similar to those of a machine designed for short-circuit testing of circuit breakers, transformers, and other power system components. Several such machines have been built to date.

The power output of the alternator must be compatible with the power requirements of the track. Table III lists the alternator specifications.

TABLE III

ALTERNATOR SPECIFICATIONS

Output Requirements	
Peak voltage-current product	50-lb payload: 280 MVA, 0.12 sec 200-lb payload: 130 MVA, 0.5 sec
Voltage	Not to exceed 14 kV line-to-line
Power factor	Will vary from leading to lagging
Frequency	240 Hz at rated speed
Phases	Three
Suggested Alternator Characteristics	
Type	Short circuit testing
Speed	960 r/min
Rotor moment of inertia	56,000 ft-lb-sec^2
Rotor dimensions	10 ft diam x 3.8 ft long
Rotor weight	70 tons

The alternator is to be operating near a laboratory where inertial components are being tested. Considerable torque is transmitted to the alternator foundation when the rotor is decelerated rapidly. To minimize interference with the testing of inertial components, adequate isolation between the laboratory and alternator foundations must be provided.

The alternator drive motor is used to bring the alternator up to speed in preparation for a test run. Normal operation of the drive motor requires that it drive the alternator from its run-down speed of approximately 860 r/m to the pretest speed of 960 r/m in less than one hour. The motor and its controls must also be capable of accelerating the alternator from a full stop to the pretest speed. This results in a motor rating of about 1000 hp, primarily because of alternator windage losses.

At this time it appears that a dc motor directly coupled to the alternator shaft will most economically meet the drive motor requirement. A wound-rotor induction motor could be used instead of the dc motor, however. The dc motor voltage rating must be the same as the alternator field voltage in order to accomplish the economies sought.

The rectifier unit converts ac power from the local utility service into dc power to drive the alternator drive motor and to excite the alternator field. The single rectifier can be used for both purposes since both are not used at the same time. Total output requirement of the recitifier is about 1000 kW.

(2) The power controller receives power from the alternator in the power supply and delivers controlled amounts of this power to the stator, according to commands from the system controller.

Power control may be accomplished by means of a series regulator made up of two inverse-parallel ignitrons in each phase. Because of power requirements, more than one set may be required per phase. The ignitron is a mercury-pool rectifier with an ignition electrode which allows the arc in the tube, and hence current flow, to be initiated at any desired instant. Power control is accomplished by varying the angle, or the time after the positive-going zero crossing, at which the arc is initiated. Fig. 6 shows a schematic of one phase of the inverse-parallel regulator. Fig. 7 shows idealized voltage and current waveshapes occurring in the circuit. Ignitron firing circuits are reqyired to translate the commands from the controller into proper firing times for the ignitrons.

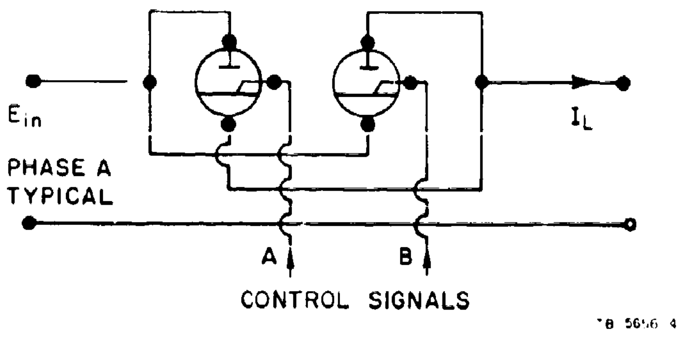


FIG. 6 INVERSE-PARALLEL IGNITRON REGULATOR

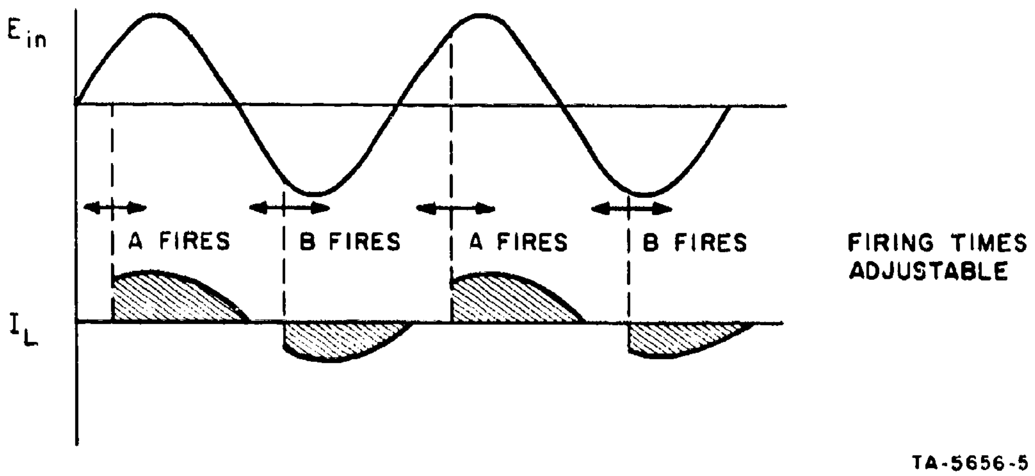


FIG. 7 VOLTAGE AND CURRENT WAVEFORMS

The power controller can supply dc power to the decelerating section of the track by firing only one of the ignition pairs per phase. The output can be mechanically switched from the acceleration to the deceleration sections.

Design of the power controller is complicated by the transient nature of the service. Accurate timing of the ignitrons and protection from extreme surges of current and voltage are a severe design problem.

(3) The stator consists of an accelerating section and a decelerating section. The accelerating section produces the moving magnetic field which interacts with the vehicle translators (fins) to produce the accelerating thrust. The decelerating section produces a stationary magnetic field which interacts with the fins to slow the vehicle.

The acceleration section of the stator is built in two portions, a 75-ft portion for accelerations up to 200 g and a 275-ft portion for accelerations up to 60 g. The core is made from laminated silicon steel and weighs approximately 1300 tons. A typical core and coil design is shown in Figs. 8 and 9. An experienced builder of electrical machinery may well select other configurations as more efficient or less costly.

Stator pole pitch is 40 inches and there are 12 slots per pole, 2 coils per slot. The phase belts are 60^0 and each coil spans 10 slots to give 5/6 chording (for optimum waveform). There will be 8 coils per phase-pole-stator--connected with two sets of four series coils in parallel. Typical coil design is shown in Fig. 10.

The magnetizing current would result in a reactive power requirement of 9.04×10^3 MVAR at the required flux density (1.7 webers/meter^2) for testing the 50-lb payload at 200 g's, if the entire track were energized. Because only 75 feet of the track length is required for this test, however, the remaining coils can remain unenergized, resulting in a reduction of the reactive power to 3.3×10^3 MVAR. The entire track length is required for testing the 200-lb payload at 60 g's. However, with a lower thrust (45,000 instead of 120,000 lb) we can either reduce the flux density and keep the stator width constant or reduce stator width but keep flux density high. It is economical to reduce the stator size even

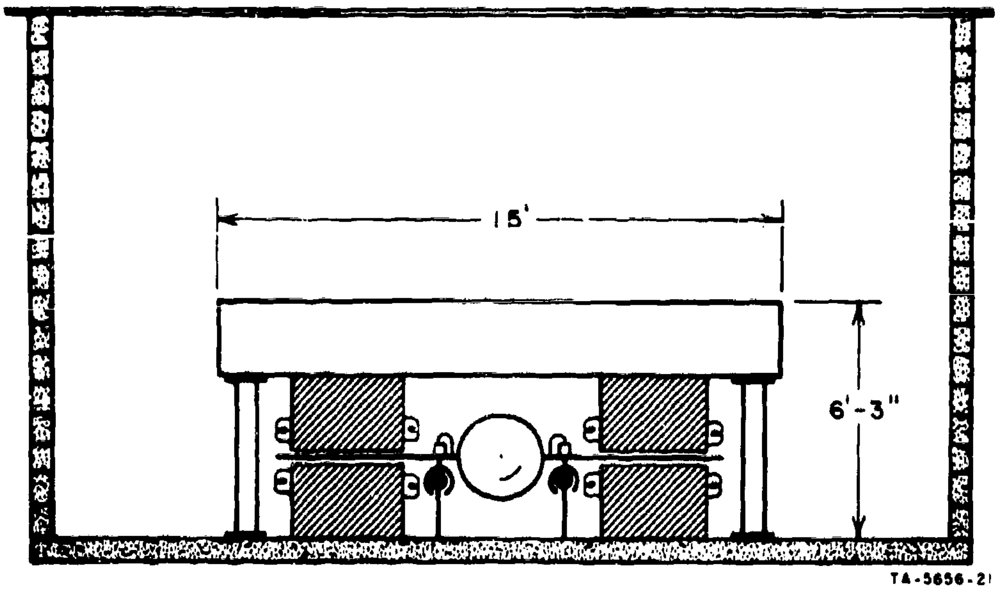


FIG. 8 ACCELERATION SECTION OF STATOR — CROSS-SECTIONAL VIEW

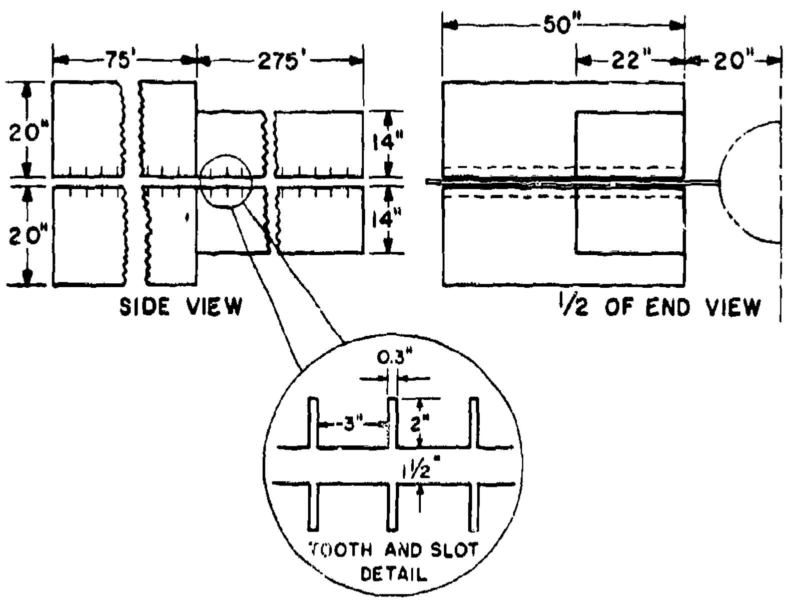
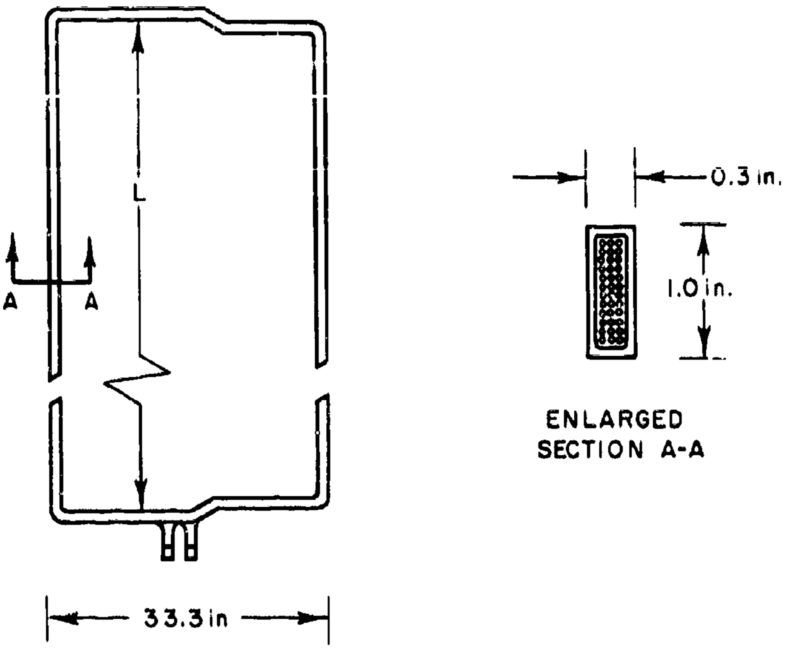


FIG. 9 ACCELERATION SECTION OF STATOR — MAGNETIC STRUCTURE



NOTES:

1. 75% of coil cross section is copper, remaining 25% is insulation & air.
2. Insulation is for 14 kV nominal machine rating.
3. All conductors in coil fully transposed.
4. Each coil is one turn.
5. Coil dimension L: 58 in. 30 in.
   Copper weight per coil: 13.2 in. 9.16 lb
   Number of coils required: 1104 3984
6. Total weight of copper required: 51,100 lb

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FIG. 10 STATOR COIL DESIGN DETAILS

though the savings in steel are partially offset by higher capacitor costs. The reactive power required for the entire stator is 6.2 x 10^3 MVAR. It must be supplied by capacitors, with suitable switching to open unneeded sections of track and to arrange capacitors in the proper series or parallel connections to attain the proper voltage and KVAR ratings in the active section of the track. Details of the switching have not been worked out.

In addition to the maximum horizontal reactive force of 60,000 pounds along each side of the track, each pair of pole pieces attracts its mate with a force of about 70 psi when the stator is excited but does not deliver power. At the place where the translators are absorbing power, the pole pieces repel each other with a pressure of 1400 psi. To support these forces a suitable retaining structure must be constructed. (See page 32.)

The decelerating section of the stator produces a direct-current magnetic field in the path of the fins which causes the kinetic energy of the vehicle to be transformed into heat energy and the vehicle to slow down. The pole pieces are 22 inches wide, 14 inches high, and 40 inches long. Since the coils will be energized by dc, rather than ac, there is no need to use laminated construction and high-quality steel. Estimated weight of the steel in the decelerating section is 1,450 tons. Very preliminary analysis indicates that coils on the decelerating section will be spaced to give the same 40-inch pole pitch as on the accelerating section. This arrangement will result in a requirement for 848 coils with the approximate shape and copper content of the 30-inch coils used in the accelerating section. Decelerating coils need not be stranded and transposed, however. Copper weight for the decelerating section is 7,770 pounds.

The pneumatic starter (Fig. 11) supplies the accelerating force during the early period of the run. Supplying the force pneumatically, rather than electrically, during this time period is desirable because the electrical drive is very inefficient at the low speeds and the total amount of heat which must be stored in the fins is reduced substantially. Also, the high repulsive forces between the stator cores would be difficult to resist at low speeds.

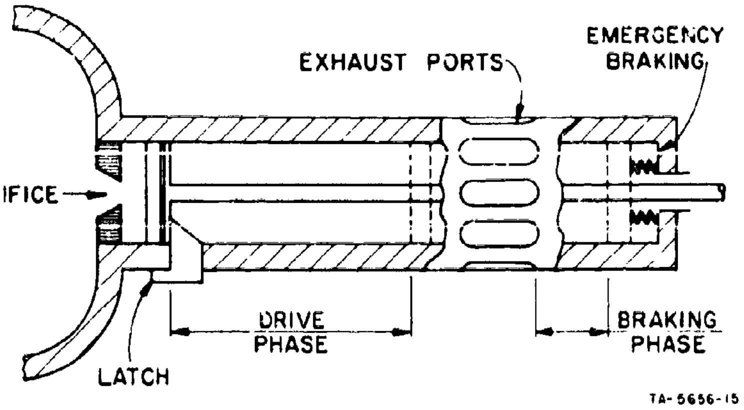


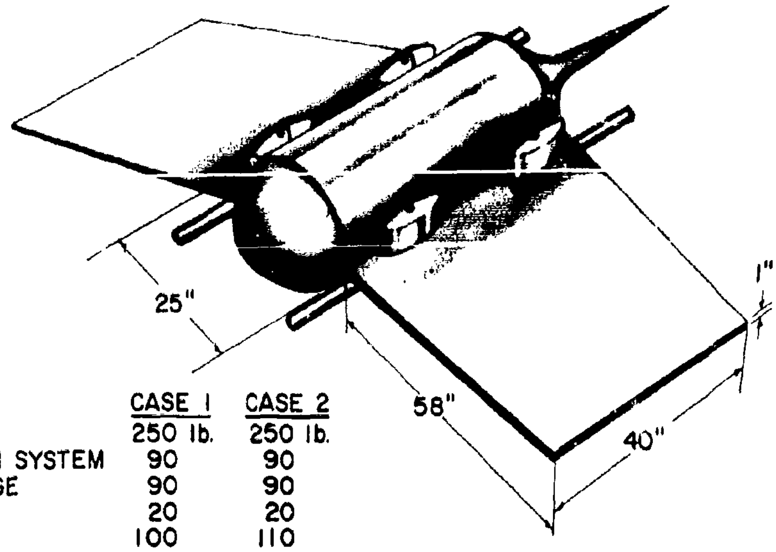
FIG. 11 PNEUMATIC STARTER

The pneumatic system would consist of a storage reservoir (about 250 ft^3 at 800 psi), a variable area orifice, a mechanism to lock the piston at the start, and the piston-cylinder system. The cylinder should have large ports in its wall between the drive phase and the braking phase so that the back pressure would be minimal during the drive and the drive pressure would be vented during braking.

The guide rails for the air bearings, although a part of the stator, are discussed in the "vehicle" section below in connection with the air bearings.

(4) The vehicle consists of the fins, or translators; the air bearing outer races, or slippers, together with the air supply for the bearings; a magnetic shield; a recorder; and a structure which provides rigid interconnection between the fins, bearings and test article, and provides protection and aerodynamic fairing to the payload. Fig. 12 is a sketch of the vehicle showing its principal dimensions. Fins are swept back to minimize end effects. (The apparent dihedral is an optical illusion. Fins are flat and horizontal.)

It is on the fins, or translators, that the accelerating force is developed; they provide the conducting material for the driven element in the linear motor. The fins are constructed of beryllium because of the



WEIGHTS:	CASE 1	CASE 2
FINS	250 lb.	250 lb.
SUSPENSION SYSTEM	90	90
GAS STORAGE	90	90
RECORDER	20	20
BODY	100	110
PAYLOAD	200	50
TOTAL	750 lb.	610 lb.

FIG. 12 TEST VEHICLE CONFIGURATION

high specific heat of this metal. The density of the cross section is reduced and the resistivity is increased by using laminated, honeycomb or sponge metal in the construction. However, the precise configuration and construction of the fins are still subject to further research. Because beryllium and its oxides are poisonous to humans, the fins must be clad with a thin layer of stainless steel. The stainless steel used must be nonmagnetic to avoid undesirable forces on the fins. The weight of the fins is estimated at 125 pounds each or 250 pounds total.

Since the weight of the fins is so high, a trade-off study was made to indicate what combination of size and number of fins would produce a maximum force-to-weight ratio. The details of this study are included in Appendix A.

Air bearing supports are proposed to guide the vehicle down the track without severe lateral vibrations. Such bearings work extremely well at low velocities, but have not been used to our knowledge at near sonic speeds. Therefore, research will have to be performed to test such bearings. Assuming suitable air bearings can be developed, they would provide the potential for a low-friction support and guide for the vehicle. The outer races, or slippers, with their air supplies, are carried on the vehicle. The inner races, or guide rails, support the vehicle on an air film which is contained by the slippers. Details of the slippers are shown in Fig. 13.

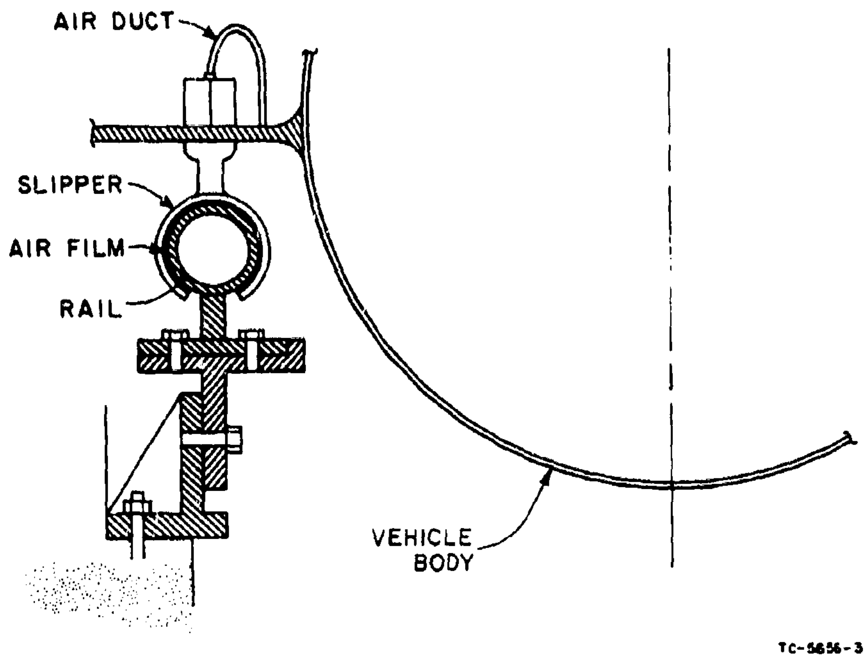


FIG. 13 SLIPPER DESIGN DETAIL

Four slippers are used on the vehicle. All four allow a small motion of the vehicle with respect to the slipper in the vertical direction. This motion is constrained by spring-dashpot units that minimize dynamic loads on the bearings. Maximum travel on the spring-dashpots is limited so that the slipper will bottom on the inner race before the fin touches the stator. In addition to vertical motion, the slipper on one side of the vehicle can move with respect to the vehicle in the horizontal plane. This horizontal freedom will allow for thermal expansion of the vehicle. Pressurized bottles are carried on board the vehicle to provide air for the bearings. Such an internal supply is necessary to allow the bearing to operate properly at low speeds. At higher speeds, air will be ducted into the shaped bearing gap at the front end of the slipper, thus providing additional bearing stiffness. The slipper construction consists of an outer shell and a replaceable insert, as shown in Fig. 13. Detection and recording of bottoming by a slipper is desirable and may be accomplished with bridge-type oscillators, using the capacitance between slipper and rail as sensor.

The tubular rails are centerless ground to attain the degree of surface finish required. The rails are attached to the supporting brackets by bolts and threaded holes. Adjusting screws are provided on the brackets for alignment. Since the clearance between slipper and rail is only a few thousandths of an inch, care must be taken in the design, construction and installation to minimize misalignment at the joints.

The degree to which air bearings can minimize cross-axis acceleration and in what frequency range requires additional study and will depend on the detail design of the bearings. The amount of lateral motion in the air bearings will also determine the accuracy to which drifts of gyros in the test package can be measured. These studies are part of the air bearing investigation proposed.

The magnetic shield is a thin cylinder of high permeability alloy which encircles the test article in order to divert magnetic flux from the stator around the test article. Maximum flux density through the test article is specified to be 2 gauss. Flux density in the area between the stator elements is estimated at 300 gauss. Magnetic shields should be tailored to fit as closely as possible around the test articles and tape recorder, both to reduce weight and to increase effectiveness.

A magnetic tape recorder is installed in the vehicle to record the performance of the articles under test. Typical of the recorders we would select is the Northam Model M53 recorder. This is a self-contained, seven-channel recording system with a frequency response of about 20,000 Hz.

(5) The sensing system and signal processor determine the velocity of the vehicle and compare a voltage which represents this velocity with a voltage which represents the desired velocity of the vehicle. This comparison results in an error signal, which is processed by the System Controller into a command signal for the Power Controller. A schematic diagram is shown in Fig. 14.

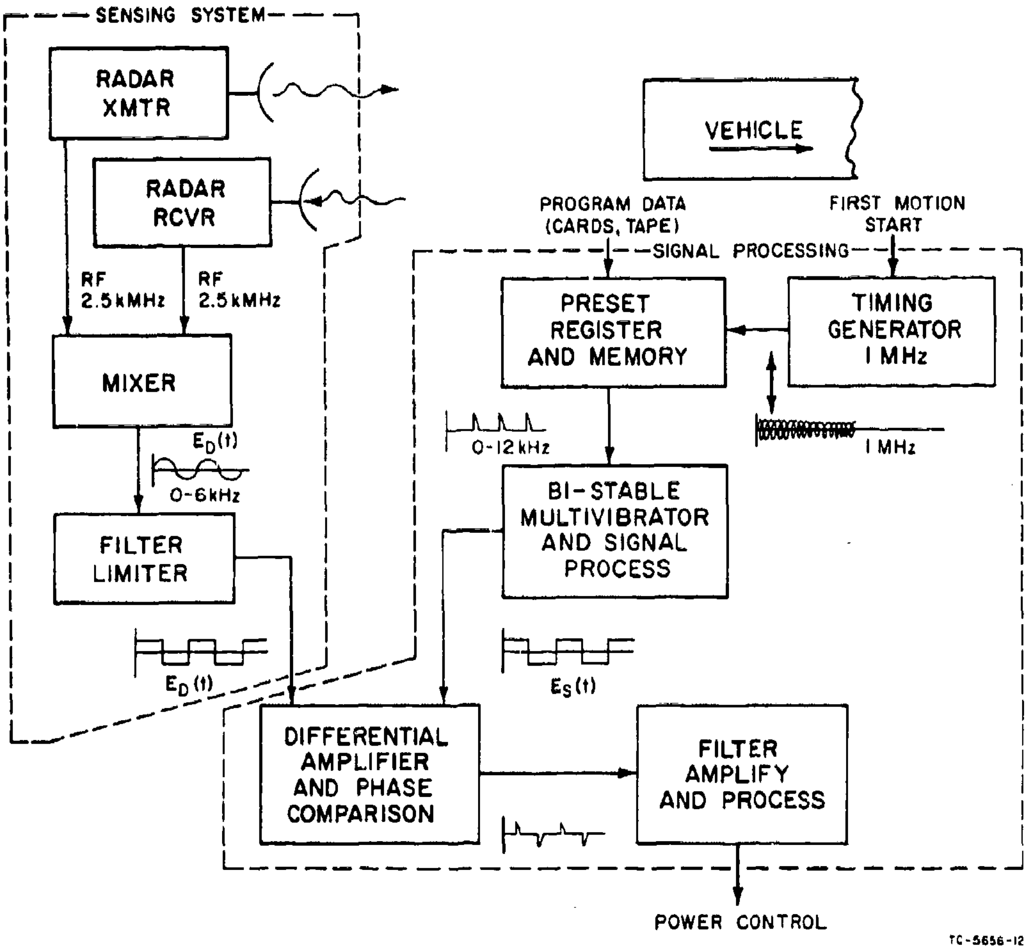


FIG. 14 CONTROL CIRCUIT DETAIL

Velocity measurement of the vehicle is accomplished by means of a Doppler radar system. The continuous-wave Doppler radar technique appears to be well-suited.[6] As applied to the present problem, a high frequency transmitter produces electromagnetic radiation from a fixed transmitting antenna that is located near the rear of the vehicle in its starting position. Energy is reflected from the vehicle toward the source, and is picked up by a receiving antenna. As the vehicle starts and moves along its path, the path length traversed by the radiation from the transmitting antenna to the vehicle and back to the receiving antenna increases. The electrical phase angle between the transmitted and received signal varies with the path length, and is most readily interpreted as a new alternating signal frequency that goes from zero at the start of the run to some value that depends upon the peak velocity of the vehicle. Calculations indicate that a radio frequency of 2500 megacycles will give satisfactory time resolution early in the run, and that the resolution improves as the velocity increases. The Doppler frequency at a velocity of 300 m/sec would be about 7000 Hz.

The techniques for applying cw Doppler methods are well established. The present requirement may not be satisfied by existing apparatus because of our interest in the low frequency end of the output signal, which is of little interest in more usual applications. Basically, the RF portion of the apparatus must contain a signal generator of good stability. An output power of a few watts is easily obtainable and will minimize noise problems. Receivers, mixers, amplifiers, and output circuits must be adapted to the problem requirements; the techniques are well established. Antenna design for this system may be complicated by the space requirements and the desirability of narrow radiation beams. Specular reflection from the vehicle or the use of a corner reflector built into the vehicle can be used to increase the received signal. Radiation absorbing materials may be necessary on parts of the structure to minimize unwanted RF signals.

The signal processing system has the function of generating a correction signal, based on the Doppler information, for application to the vehicle drive system. The technique proposed is that of preparing a standard signal $E_s(t)$, comparing this with the Doppler signal $E_d(t)$, and using the differences to produce the control signal. One way to reduce this to practice is detailed in Appendix A.

(6) The system controller acts in conjunction with the sensor and signal processor and the power controller to control the acceleration of the vehicle. In addition, the system controller commands switching and sequence operations in the power controller and power supply for the start, accelerate, run, decelerate, and recovery modes of operation. To perform these functions, a digital computer with a memory of about 32 thousand words and an access time of 5 μsec will be required, together with input-output equipment. The GE C4050 process control computer (or equivalent) with a magnetic tape input and additional memory and digital signal conditioners and input controllers is acceptable. Suitable print-out of vehicle velocity and acceleration can be provided by compatible equipment.

(7) Buildings and foundations require special attention because of the large forces and moments developed in the stator and by the power generator.

Three major buildings will be required as indicated in Fig. 15:

- The stator building, approximately 25 x 1100 ft and 14 ft high.
- The power building, approximately 20 x 50 ft and 18 ft high.
- The laboratory and control building, approximately 20 x 25 ft and 9 ft high.

Wall, roof, foundation, and utility requirements are included in Appendix A.

(8) The stator constraining structure must be designed with considerable care because it must not only support the heavy static load of the stator core, but the severe dynamic loads during stator operation. It can be shown (see Appendix A) that the excited stator poles attract one another with a force equivalent to 70 psi (or 42,000 lb/ft in the first

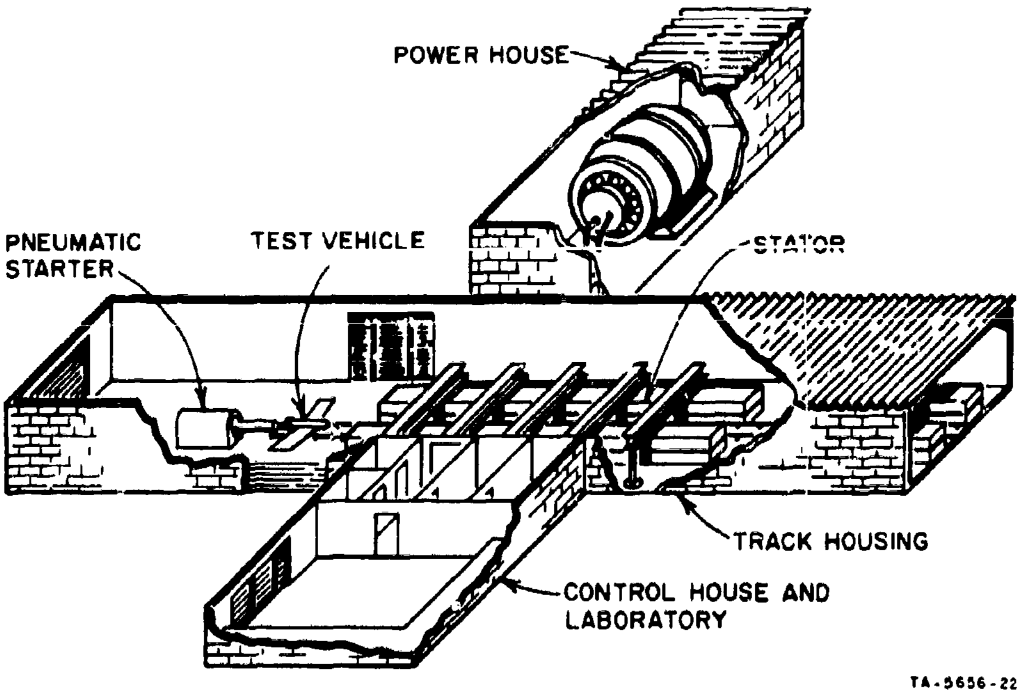


FIG. 15 BUILDING LAYOUT

75 feet of track). When the translators pass through, the attractive force changes into a momentary repulsive force equivalent to about 1400 psi (or nearly three million pounds total). This repulsive force is reduced as it pushes the poles apart. However, any widening of the gap causes a reduction in propulsive force, and must, therefore, be resisted. The inertia of the pole material contributes to resisting this high dynamic force, but is more effective at the higher speeds when the duration of the force at one place is shortest.

A complete dynamic analysis of the stator was beyond the scope of this study. However, for cost estimating purposes, we have computed that built-up plate girders with a section modulus of as much as 700 in^3 may be required across every foot of the stator to maintain dimensional integrity. These girders are held to the foundation by tension bars of about 50 in^2 cross section. The entire supporting structure is estimated to weigh 6 tons per foot of stator length (for the acceleration section).

It is believed that these estimates are conservative, for they do not take into account the bending strength of the stator core itself, which is likely to be significant.

(9) Auxiliary facilities important to the system include:

- Equipment to handle the vehicle before and after each run. One concept of a vehicle dolly is shown in Fig. 16. It consists of short extensions to the air bearing rail on a portable dolly so that the vehicle can slide onto the dolly from the rail or vice versa. Compressed gas and electrical power are supplied through a quick disconnect umbilical.

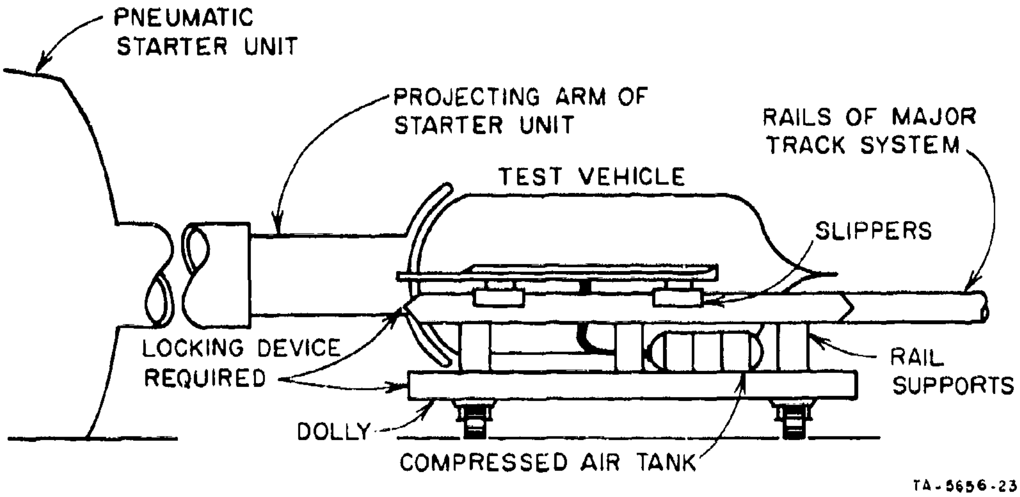


FIG. 16 TEST VEHICLE DOLLY

- Safety devices will be concerned primarily with the handling of high power electrical circuits, the presence of strong magnetic fields and the high speed of the test vehicle. Most of these are routine. If a power failure occurs or the control mechanism breaks down after the vehicle has been accelerated but before the braking system becomes operative, the vehicle could continue down the track unimpeded except by air friction. Since such an event is unlikely, we believe that it will be adequate to provide an earth barricade at the end of the stator.

• The dc decelerating system becomes less effective at low speed and must be aided by mechanical or pneumatic devices that bring the vehicle to a complete stop. We propose sets of pneumatic tires, set into the end of the stator which engage the translator and, by inertia and braking, bring the vehicle to a complete stop. Water cooling sprays will be required to keep the wheels cool and to reduce the translator temperature to a point where it can be handled.

Unresolved Problems

The major unresolved problems of the constant frequency linear induction motor are the high-speed air bearings and the translator material configuration. Statements of work and cost and time estimates to solve these problems are given in Appendix F.

There are many other technical problems that will have to be solved. We believe that most of these can be solved within the present state of-the-art and do not require research.

The Operating Cycle

Operation of the high-g linear accelerator facility can be envisaged as follows:

Tenant and non-tenant customers will contract through the director of the facility for scheduling and costing of projected component testing.

At scheduled times, customers will arrive at the facility with the components which are to be tested. These components will be taken to the laboratory facility where they will undergo varying degrees of inspection and pre-run testing using devices such as servo-tables and tilt tables and test instruments such as ohmmeters, oscilloscopes, oscillographs, and vacuum tube voltmeters. These devices and test instruments, together with associated tools, laboratory power supply, wiring, etc., will be provided in the laboratory by the facility. Given an expected workload of 1,000 runs per year, or approximately 4 runs per regular work day, we can assume the need for providing up to approximately 5 such inspection and test stations.

While at the inspection and test station, the components which are to be run will be installed in adapter sections which will be designed to fit inside the test vehicle where they will hold the component in the preferred position; typically these adapter sections will include power supplies, inverters, and instruments for recording the characteristics of the component to be measured during the run.

When the test vehicle is available, it can be conveniently rolled inside the laboratory adjacent to an inspection and test station. The component to be tested (already fitted into its adapter section) will be hoisted by overhead crane from the station and placed aboard the test vehicle. When installation of the adapter section and compressed air bottles has been completed, the test vehicle will be withdrawn from the laboratory to the stator building.

Within the stator building, (see Figure 15) the vehicle transport dolly will be emplaced as an extension of the test track, the rails of the vehicle transport dolly interfacing with the air bearing guide rails of the major track system (see Figure 16). The vehicle transport dolly will then be locked in place between the major track system and the pneumatic starter system. Using the air bottle stored on the test vehicle dolly to pressurize the air bearing, the test vehicle will be moved to the rear so that the aft fairing of the vehicle interfaces with the drive facing of the pneumatic starter piston.

At this point an electrical umbilical cord will be attached to the test vehicle. This cord will transmit signals for uncaging gyros, starting recorders, releasing compressed air into the vehicle air bearings, etc. Then the external compressed air bottle will be disconnected from the vehicle.

Next, all personnel will be evacuated from the accelerator building. Electrical sensors will show the console operator when all accelerator building doors have been closed and a visual check of the start end of the accelerator will then be made by the console operator through the observation window common to the laboratory control panel room and the accelerator building.

User representative monitors will be able to make static readiness (performance) checks measured through signals passed through the umbilical.

Status of all systems will be displayed on the console. When all systems are ready, the console operator will initiate the computer controlled test run procedure. The procedure will include the pneumatic start, application of field voltage to the alternator, control of acceleration, energizing the decelerating section, indication of run being complete, and switchover for return of alternator speed, tank pressure, etc., to the desired values for the next run.

The test vehicle will come to rest against the receiving unit at the end of the test track. The last portion of the track will be another test vehicle dolly, locked in place at the end of the track.

Attaching the test vehicle dolly's compressed air bottle to the test vehicle will allow facility personnel to extract the front of the vehicle from the receiving unit. The test vehicle dolly may then be unlocked, the dolly drive system attached to the side of the dolly, and the vehicle and dolly then withdrawn from the accelerator shed through a side door adjacent to the track at that point.

At the laboratory, after all on-vehicle tests have been performed, the test component adapter section will be detached from inside the test vehicle and the component and adapter section will be hoisted to the inspection and test station where further post-run testing can be carried out. The test vehicle is ready to receive the next component to be tested, and the cycle of operations can begin anew.

Data reduction with either analog or digital printout will be performed by a tape deck, a paper-recorder, and the circuits and printer of the systems controller.

VARIABLE FREQUENCY LINEAR INDUCTION MOTOR

If it is possible to vary the frequency of the stator current, the linear induction motor can be operated at a small slip (see Fig. 3) with a low resistivity translator. This yields the following major advantages over the constant frequency system:

- The motor operates at high efficiency so that the power generated need be only slightly greater than the power used.
- Very little power is converted to heat in the translator so that no heat storage problem exists. This means that the translators can be made lighter and smaller, thus further decreasing the thrust and power requirements.
- Speed control is positive and can be achieved directly by controlling the stator current frequency, regardless of small variations in thrust.

Calculations show that, with a 40-inch pole pitch, a frequency range from 20 to 180 Hz gives a speed range from about 130 to nearly 1200 ft/sec. Translators could be constructed from aluminum which would reduce the weight slightly. The reduced weight combined with the increased efficiency would reduce peak power requirements from 280 MW to about 200 MW, and the energy delivered from 65 MW-sec to about 25 MW-sec.

These evident advantages are obtainable only if a method can be found to generate variable frequency power at the high levels and with the rapid frequency control demanded. Of several possible ways we investigated, one method appeared particularly well suited to this problem: dc generation by a magneto-hydrodynamic (MHD) generator, followed by a variable frequency oscillator.

The principle of the MHD generator is illustrated in Fig. 17. It has no moving mechanical parts. A moving ionized gas stream flows between the poles of a strong magnet. Power is extracted by electrodes at right angles to both the magnetic flux and the gas flow direction. It is created at a potential of several kilovolts. Details of two proposed designs in the 40 to 100 MW range are shown in Appendix C. Although an MHD generator of this size has not yet been built, the manufacturers who have built smaller units are confident that extrapolation to the desired power is well within the state of the art.

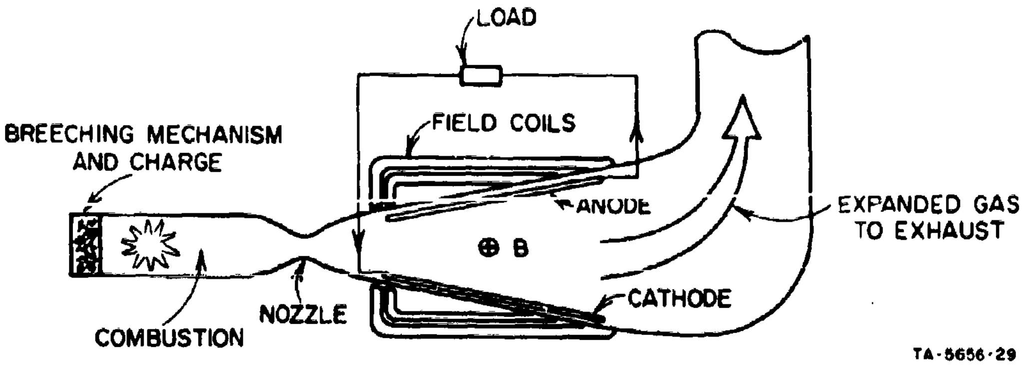


FIG. 17 PRINCIPLE OF MHD GENERATOR (fuel would be solid or liquid)

More difficult is the conversion of dc to variable frequency ac. Vacuum tube oscillators are a distinct possibility, for power tubes (triodes) with output in excess of one megawatt are already available. Inverters using high power solid state devices such as silicon-controlled rectifiers may also be possible. We recommend a study specifically aimed at the feasibility of dc conversion to variable frequency ac and capable of rapid frequency control. A specification and cost estimate for such a study are included in Appendix F.

The possibility of generating ac directly with MHD generators is remote. One could rotate the magnet about the ionized gas jet. This would require the rotation of a heavy magnet at close to 11,000 r/min to achieve a frequency of 180 Hz, a difficult task at best. Another method would produce an oscillating jet by varying periodically the flow of oxidizer. Even if this is possible, it would not produce ac but varying amplitude dc, not a great improvement over constant voltage dc.

Other problems with the variable frequency system are switching and power factor correction. Because of power factor, it is not practical to excite the entire stator simultaneously as is proposed for the constant frequency system. Each section must be excited just before the translator enters, soon enough to establish the magnetic field. Also, the total capacity connected with each stator section must be adjustable so that, as different velocity profiles are chosen and correspondingly different frequencies are assigned to the sections, the proper power factor corrections are made.

It is evident that the variable frequency drive would be more efficient and easier to control than the constant frequency induction motor. But the technical problems, particularly the conversion of dc to variable frequency ac, cannot be solved rapidly, nor is a successful solution assured.

V PNEUMATIC SYSTEMS

At first glance, propulsion by compressed gas appears to be very attractive. Compressed gas provides efficient, inexpensive energy storage; if the vehicle is visualized as a piston in a tube, its construction is simple and light, so that the pressure requirements are nominal; braking could be accomplished by gas compression in front of the vehicle; techniques for compressed gas pumping are well-established and economical. Also, there is some evidence[3] that a gas-propelled vehicle in a tube travels smoothly along the tube without oscillating between the tube walls.

Past experience, however, points either to short period, high acceleration devices, such as the compressed gas gun, or to longer period, low-g systems, such as the pneumatic tube. Neither system is normally concerned with acceleration control.

We have studied feasibility of two different pneumatic systems. Both use a cylindrical vehicle in a tube. One type of system would use a single, large reservoir at one end of the tube for gas storage; a second type would use a number of storage tanks placed at intervals along the tube. Both systems are based on air, but if shock phenomena become limiting, helium, with sonic velocity 2 to 3 times that of air, could be used.

Acceleration control in the single reservoir system would be accomplished by varying the mass flow rate through a single large valve at the entrance to the tube. Valve action would be preprogrammed since it would not be feasible to close-loop-servo the system. In the multiple tank system, control would be accomplished by adjusting the pressure and temperature in each tank prior to the test and by having the test vehicle itself trigger the opening of successive valves as it travels along the tube. Here also feedback control would be difficult or impossible.

Analyses (see Appendix D) have indicated that it would be necessary to evacuate the tube before each run to reduce the pressure build-up ahead of the vehicle. We have also checked the lateral stability of such a vehicle in the tube and have found that proper design and weight balance can result in a stable configuration.

Gas dynamics analyses of the pneumatic drives were performed to determine if the required pressures, temperatures, and storage volumes are reasonable. Three modes of operation were considered:

Case		I	II	III
Payload	lb	50	200	50
Tube diameter	in	12	25	25
Acceleration	g	200	60	200
Est. vehicle wt.	lb	50	200	200
Total weight	lb	100	400	250

The first and second cases correspond to the requirements of the project (Table I, page 2). Case III proposes to use the same vehicle and tube as Case II for the payload and conditions of Case I. The analyses in all cases are based on the most extreme flow and velocity conditions, i.e., at the end of the positive acceleration phase of the most demanding tests with each payload. Details of the calculations are shown in Appendix D.

SINGLE RESERVOIR SYSTEM

The system would consist of the storage reservoir, from which compressed gas flows through a control valve into the tube. The vehicle is propelled by the gas pressure in the evacuated tube. Braking methods have not been investigated for this system.

The analysis proceeds from the system requirements and computes the minimum volume, pressure and temperature of air in the storage tank that can provide adequate propulsion force for the three cases. The calculation assumes adiabatic flow in the tube and in the valve.

This assumption is reasonable for the tube. However, a quick-acting adiabatic flow valve does not exist. Its development would be a major achievement, useful not only for the linear accelerator but also for blow-down wind tunnels and for the control of many dynamic processes. The development of such a valve is the major obstacle in this propulsion scheme.

The calculations (Appendix D) show that Case II makes the most severe demands on storage vessel volume and the vacuum system. Depending on the temperature to which the vessel and the air in it can be heated, the reservoir would require a minimum volume of somewhere between 2,000 and 6,000 cubic feet. Typical reservoir requirements taken from the calculations are shown below:

Case		I	II	III
Initial pressure in tube	psia	1.5	0.15	1.5
Reservoir Volume	ft^3	172	2960	643
Initial Reservoir Pressure	psig	422	159	285
Initial Reservoir Temperature	^oF	157	256	152

Estimates of power of compressors and vacuum pumps (also detailed in the appendix) shows that power requirements are quite nominal (less than 100 hp total).

MULTIPLE TANK PNEUMATIC DRIVE

The same principles used in the single reservoir system snalysis are applicable to the multi-tank system. However, the flow geometry is so complex that a quantitative analysis is impossible.

The basic idea of the multi-tank system is to distribute the gas storage volume among a number of tanks located along the tube. Each tank would be connected to the tube by one or more quick-acting valves and nozzles. If air is used as the working fluid, it will be necessary to maintain a fairly high temperature in the reservoirs near the end of the run to keep the local sonic velocity greater than vehicle velocity. For the same reason, the flow area of the valves should be comparable to the tube cross section.

The interval between tanks must be sufficiently short that the acceleration decay due to pressure drop in the interval will not exceed the tolerance limits, i.e., the acceleration-displacement curve for the vehicle would be similar to that shown in Fig. 18.

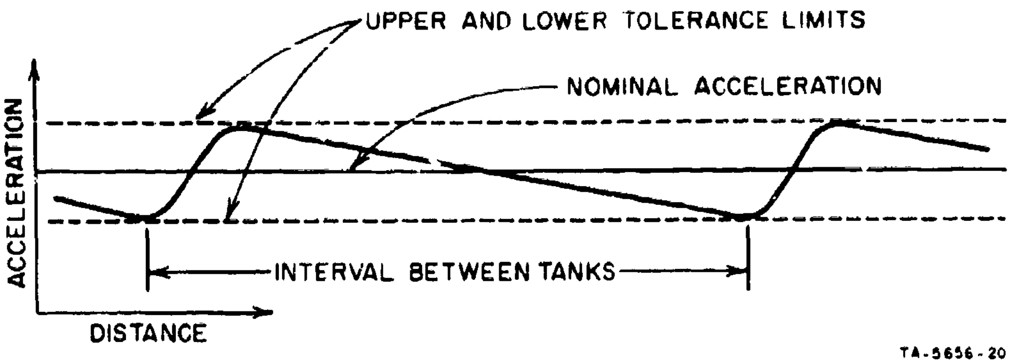


FIG. 18 ACCELERATION VERSUS DISTANCE — MULTIPLE TANK SYSTEM

Without a very difficult and lengthy analysis, any estimates of tank size, pressure, temperatures and tank-spacing must be based on a number of fairly arbitrary assumptions. Some of these assumptions are:

- As the vehicle passes any particular valve, this valve opens instantaneously.
- The valve (orifice) coefficient is 0.92.
- Gas from the tank expands into the volume behind the vehicle adiabatically and instantaneously. It then starts to apply acceleration to the vehicle.
- Friction drop in the tube under these unsteady conditions is twice the steady flow friction drop.

On the basis of these assumptions we have calculated the number and size of tanks necessary to accelerate the vehicle in each of the three limiting cases, so that the variation in acceleration does not exceed allowable limits. The spacing range is also listed below.

Case		I	II	III
Number of tanks		10	50	8
Range of tank spacing	ft	16-4	13-1.5	20-6
Tank volume	ft^3	2400	5000	5000
Range of tank pressure	psi	180-390	45-90	105-240

In view of the assumptions, these values are to be considered very tentative. To refine them, an extensive research effort would be required, but even then, it does not seem likely we could predict reliably the pressures and temperatures required in the various tanks to obtain a given acceleration level. Therefore, a considerable trial and error period would be required on the completed facility before a range of constant acceleration patterns can be achieved. It seems unlikely that a feedback control mechanism can be used.

SUMMARY OF PNEUMATIC DRIVES

Both single and multiple tank systems appear technically feasible. The principal problem would be valve design since the valves must provide a smooth transition from zero flow to maximum flow in a very short time. Gas flow control is further complicated by the fact that flow through an orifice or nozzle is generally a function of both upstream and downstream conditions. Because of these factors we feel that the degree of acceleration control achievable by either of the pneumatic drives can be determined only by extensive model studies.

The limitation of sonic velocity is not really restrictive, since helium, with a sonic velocity almost three times that of air, could be substituted for air if necessary.

VI ROCKET SLEDS

In the sense that a rocket sled is nothing but a rail bound version of a flying missile, it would appear at first sight to be well-suited to this project. In contrast to other systems, rocket motors are not power-limited and will produce substantially the same thrust, regardless of speed. It is easy to show that the accelerations required can be achieved readily on existing rocket sleds and with existing rocket motors. In fact, test durations can be increased since existing rocket sled tracks are considerably longer than required. However, three major drawbacks are apparent:

- It is difficult, if not impossible, to provide feedback acceleration control on solid rocket motors.
- The vibration (lateral acceleration) of existing rocket sleds far exceeds the requirements of the test vehicle.
- The cost of solid rocket engines is too high to contemplate their use for the large number of routine tests desired.

EXISTING SOLID ROCKET TEST SLEDS

The evaluation of rocket sled tracks is based on our own experience with test tracks and on information obtained from operators of existing rocket sled tracks. The problem was discussed informally with personnel at the two major tracks now operating, the Air Force Missile Development Center (AFMDC) track at Holloman, New Mexico, and the supersonic Naval Ordnance Research Track (SNORT) at China Lake, California. Other suitable heavy duty tracks, not now operational, are the AFFTC track at Edwards, California and the HSRS track at Hurricane Mesa, Utah, either of which could be reactivated in a relatively short time.

The acceleration produced by a particular rocket motor can be decreased by carrying ballast on the sled, or increased by hold-down at start until the proper amount of propellant mass has been expended. Acceleration in flight can be controlled by several techniques, used singly or in multiple. These include the programmed release of ballast, staging of rocket firings, and various types of braking that can be used during the positive as well as the negative acceleration phases. These control techniques are usually preprogrammed and provide acceleration control to about ±10% peak acceleration. It is doubtful if they could be close-loop-servoed.

Braking can be achieved by flaps which increase the air drag, water brakes, or retro-rockets.

To perform the proposed testing on rocket sleds it is probable that a dual-rail, multi-motor sled would be used for tests up to about 75 g, and a monorail, single-motor sled for tests above 75 g. At the present time, and in the foreseeable future, suitable motors for the dual-rail sled are available without cost to military users. This situation results from the obsolescence and/or limited shelf life of the motors. The monorail sled would require one high-performance motor per run at a cost of about $3000. Using SNORT as an example, the cost of operations (support personnel labor, overhead, etc.) would be about $3000 to $4000 per day, in addition to the cost of the rocket motors. Our experience indicates that no more than two runs per day can be made on the average.

Neither sled would be particularly volume or weight limited and the dual-rail sled could be designed to carry as many as four of the 50-lb payloads at one time by adding rocket motors as required. It would also be possible, by firing sustainer rockets in stages, to extend test durations far beyond the specified values.

HYBRID ROCKET SLEDS

There is a possibility that hybrid rocket sleds may overcome several if not all of the drawbacks of the existing solid fuel rocket sleds.

Hybrid rockets consist of a solid fuel and a liquid oxidizer. If properly designed, the major parts of the rocket engine ought to be reusable many times, thus reducing the cost per run to a value between \$50 and \$500, depending on the amount of fuel and oxidizer needed.

Acceleration control is possible by controlling the liquid oxidizer flow. We visualize a big and a small flow line in parallel. The valve on the big line is preset to give a flow just smaller than required to maintain the desired thrust. The valve on the small line is used as a trim valve and is controlled by a control circuit on the vehicle actuated by an instrument accelerometer. The time constant of the feedback control system depends primarily on the speed with which a small change in valve position is converted into the desired change in thrust. We are told that this may be 1 to 10 msec, but it must be confirmed by tests.

Two preliminary design suggestions by United Technology Division of United Aircraft Corporation are shown in Appendix E. A preliminary research and development program for such a hybrid rocket is included. The estimated cost of the feasibility research is \$285,000. This is an unofficial rough estimate.

The problem of crosswise acceleration is identical on all rocket sleds and is also similar to that of the linear induction motor drive. Hence, the air bearing rail proposed in Chapter IV may be as applicable for the rocket sled as for the induction motor.*

* Tests with air bearing slippers on a rocket sled at Holloman AF base have been planned and may be underway.

VII COMPARISON OF PROPOSED METHODS

A summary of the technical and cost data developed on this project is shown in Table IV. Its arrangement is based on a list of requirements of the research contract.

The methods were also compared on the basis of the following criteria:

- Availability -- How likely is it that the system can be built and will perform as planned? How much research is required?
- Controllability -- Is it going to maintain constant acceleration? Can it be controlled during the run?
- Flexibility -- How versatile is it? How easy is it to change conditions between runs? How often can a run be made?
- Cost -- How expensive is it to build and to operate?
- Growth Potential -- How far and how early can the system be extended to higher accelerations and/or longer duration runs?

The comparison of the methods according to these criteria is based on limited information. Hence, the results are open to question and re-evaluation.

Availability. Only the constant frequency linear induction motor promises to meet the technical requirements in the near future with a minimum of additional research (selection of the translator material). All other systems require a moderate or extensive research effort before they can be built. The support (lateral vibration) problem is common to all systems and should be investigated in any case. Air bearings promise to present a useful solution.

TABLE IV

HIGH-G LINEAR ACCELERATOR
SUMMARY OF COMPARATIVE EVALUATION

	Applicable to all methods	Linear Induction Motor -- Constant Frequency	Linear Induction Motor -- Variable Frequency (with pneumatic starter)	Pneumatic Drives	Hybrid Fuel Rocket Sled
Accelerator length	1050 ft				
Package carrier dimension	25" O.D. x ~ 5'	2 Translators 40" long x 50" span	Translators, size not determined	No translators required	Hybrid rocket adds ~ 4 ft in length
Package carrier weight		550 lb	350 lb + translators	~ 200 lb	275 lb, including fuel
Propulsion system requirements	See details in report				
Sound	All noisy	At max. rates--shock wave interaction with stator. Also mechanical noise from forces in electric machinery. Electrical noise alone estimated at 110/dB.[1]		Noisy at valves (125/dB)[1]	High noise level 180/dB[1]
Safety		Routine protection from high voltages (20 kv) and magnetic fields required. Fence is adequate		No apparent safety hazard	Problems similar to missiles using equivalent fuel
Outline drawings	See details in report				
Method of returning package		Dolly as shown in Fig. 16		Not investigated[2]	Dolly in Fig. 16
Problem areas	See text for details	Air bearings, fin material	Air bearings, DC gen., Freq. converter	Valves, Aerodynamics, Control	Air bearings, Onboard control
Evaluation of solving problem areas	See text for details	Solution seems near (175 KB[3] -- 10 months)	Solutions are possible (300 KB[4] -- 10 months)	Considerable research req. (no cost estimate--18 months)	Solutions seem near (365 KB--10 months)
Comparison table	See also Table V				
Space for propulsion and control	Effects on buildings is compared to linear induction motor drive. Site should be flat and horizontal for all methods.	Power and control house, 20x50 ft.	Similar	Similar	None
Track, construction, enclosure		Very heavy concrete base ~ 25x1050 ft--roofed brick shed	Similar	Similar--but lighter concrete base	Similar to pneumatic drive
Foundation stability		Stable to 1 ft/1000	Same	Same	Same
Utility requirement		~ 1000 kW + routine	~ 2000 kW + routine	~ 100 kW + routine utilities	Routine and ventilation of enclosure

Propulsion system performance	This has been major basis for feasibility. Hence, all systems meet acceleration requirements.				
Power input requirements	Because of very high short duration power requirements, energy is stored in each system. Steady power requirements are low in every case, and present a very small fraction of the cost.				
Control accuracy and complexity		Desired accuracy can probably be achieved by Doppler radar feedback control		Control accuracy not easily achieved[2]	Not investigated[2]
Cost of carrier	Cost of carrier vehicle is expected to be a very small portion of the total cost of the system, regardless of the system selected. Cost estimates were made only for the linear induction motor with constant frequency. In this system the carrier cost is about 0.2% of total system cost.				
Bearing requirements	Air bearings				
Aerodynamic complexity		Carrier shaped for transsonic flow		Evacuation required	Similar to linear motor
Test repetition suitability	All systems at least once an hour	Depends only on generator. Half-hour intervals possible	Depends on solution of primary power. One hour certain	Evacuation time of tube and pump-up time limiting	Limited only by safety operations
Maintenance requirements		Control circuit tube replacement is major concern	Not estimated.[2] Appears high	Not estimated[2]	Not estimated.[2] Likely to be low
Operation complexity		Complex	Very complex	Relatively simple	Simplest
Potential for higher g		Limited by length of track and power available	Same as constant frequency—but also has frequency limit	Can be extended if helium is used instead of air	Limited only by specific impulse of fuel
Estimated design cost		(1,400 KB[3])[5]	Not estimated[2]		
Construction cost		13,600 KB	Not estimated[2]		
Operating cost		90 KB/year[6]	Not estimated[2]		

NOTES: [1] Based on data from the Handbook of Noise Control.
[2] Items not investigated because detail design and analysis is prerequisite for evaluating the item. Such a detail analysis was not part of the scope of this study.
[3] KB means thousand dollars.
[4] Includes contingency of 155 KB for switching, power factor correction and other problems.
[5] Design cost is difficult to break out of construction. Number is based on an estimated 8% of total construction cost plus research costs.
[6] Excludes overhead, amortization and land costs.

Controllability. The response time required of this control system is estimated to be 1-10 msec.* This type of control is relatively easy to achieve in the induction motor systems. There is a good chance that the hybrid rocket can also meet these specifications but research is required to substantiate this. The other systems do not lend themselves readily to feedback control.

Flexibility. Scheduling of runs of the linear induction motor system is limited by the time required to bring the alternator up to speed (or to recharge the MHD generator). One run every 30 min may be possible. It is as easy to change conditions between runs as to keep them the same. Flexibility of the pneumatic systems is limited by the problem of valve adjustment and pressure and temperature control in the air reservoirs. Experience with rocket sleds indicates that safety requirements in the handling of explosives are the major limits on run frequency; these problems are minimal in hybrid rockets since both fuel and oxidizer are safe as long as they are separated.

Cost. A detailed cost analysis of only one system--the constant frequency linear induction motor--has been possible under this contract. It shows that initial costs are high but operating costs are low. The other systems can only be ranked relative to the induction motor system at this time.

Growth Potential. Higher accelerations than those presently contemplated can be produced either by reducing the vehicle weight or by increasing thrust, or both. The linear induction motor is not well suited to provide higher than design thrusts without change in the generator itself (unless the generator was overdesigned to begin with). Nor can it extend run duration without a change in track length. The limits of run duration are shown in Fig. 19. For the presently contemplated track the solid lines apply. If the track is extended, the run duration will be limited by power availability (dotted lines).

* Actual requirements depend on rate of change of applied forces, such as drag, friction, motor thrust, etc., and on the allowable variations in acceleration (see Table I).

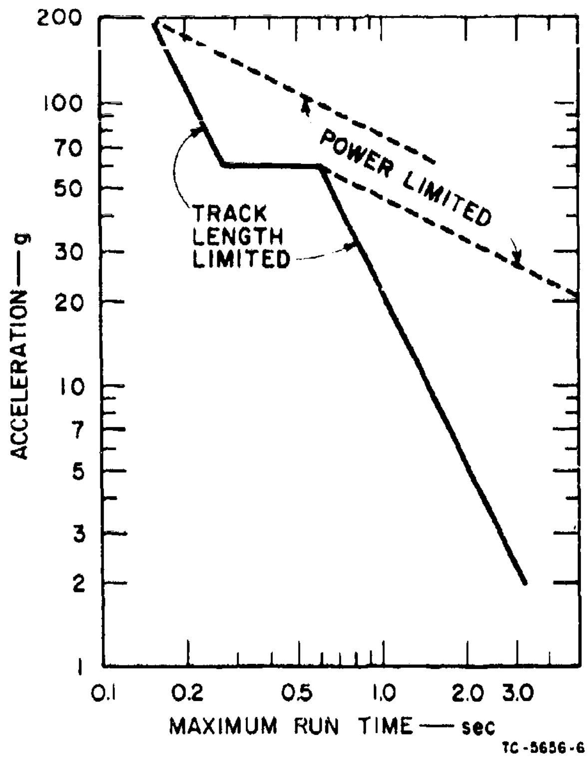


FIG. 19 MAXIMUM RUNNING TIME AT CONSTANT ACCELERATION

To increase acceleration in the pneumatic system it will be necessary to use gases with a higher velocity of sound than air, such as helium. The creation of shocks will limit the air system to the accelerations for which it is designed (see Appendix D). Test duration is track and power limited, similar to the induction motor drive.

Acceleration levels of a rocket sled are limited only by the specific impulse available and duration by the length of the track.

Overall Comparison

Comparative evaluations are shown in Table V. Numerical figures of merit were assigned and added to give a tentative overall evaluation. It seems clear the the six methods considered fall into two groups: the pneumatic schemes and the existing rocket sled least likely to provide an adequate answer, and the two linear induction motors and the hybrid rocket sled most likely. We, therefore, recommended research that can lead to a definite selection among these last three schemes.

TABLE V

FEASIBILITY COMPARISON

Method	Can if Fulfill the Requirements		Amount of Research Required to Make Method Workable	Likelihood of Early Success (within 12 months)	Relative Cost		Flexibility	Final Rating
	Acceleration	Control			Initial	Operating		
Pneumatic (single tank)	Yes [1]	No [0]	Large [0]	No [0]	Mod. [1]	Low [2]	Small [0]	4
Pneumatic (many tanks)	Yes [1]	No [0]	Large [0]	No [0]	Mod. [1]	Low [2]	Small [0]	4
Rocket Sled (standard)	Yes [1]	No [0]	Large [0]	No [0]	Low [2]	High [0]	Small [0]	3
Rocket Sled (hybrid)	Yes [1]	Yes [2]	Moderate [1]	No [0]	Low [2]	Mod. [1]	Mod. [1]	8
Electro-Mag. (constant freq.)	Yes [1]	Yes [2]	Slight [2]	Yes [1]	High [0]	Low [2]	High [2]	10
Electro-Mag. (variable freq.)	Yes [1]	Yes [2]	Moderate [1]	Yes [1]	High [0]	Low [2]	High [2]	9

VIII COST ANALYSIS
(Constant Frequency Linear Induction Motor)

The costs of the High-G Linear Accelerator are separated into four categories for convenience of discussion. These are:

1. Research costs
2. Design, development and investment costs
3. Incremental costs of system variation
4. Annual operating costs

The total cost is estimated at about $15,000,000. At a 6% discount rate and with an estimated 1000 runs per year, this amounts to about $2000 per run if amortized over ten years and $1200 per run if amortized over twenty years.

1. Research costs for the air bearings and the translator are $80,000 and $95,000 respectively, a total of $195,000. Estimated time for the research is 10 months for each, and for both.

2. Design, development, and investment costs are predicated upon (1) an investigation of current and near term projected labor and material costs and delivered prices of current and roughly similar system components, and (2) discussions with General Electric, Westinghouse, Allis-Chalmers, and individual authorities in the field of heavy electrical equipment construction and costing.

Cost estimates presented in the following paragraphs and in Appendix B must be considered inexact because of the inexact design information upon which they are based. Developing more precise information is beyond the scope of the present effort, since detailed design is not required to reach a decision on technical feasibility.

Design and development include all operation required to develop the given component from conceptual stage to a finished and acceptable product; specifically, this entails engineering design, drafting, fabricating, assembly, and testing of each component. Investment includes transportation, installation, and on-site testing of each subsystem and of the total operating system.

The eight primary subsystems and costs are summarized in Table VI.

TABLE VI

DESIGN, DEVELOPMENT, AND INVESTMENT COSTS

Subsystem	Cost (\$1000)	
Vehicles (2)	125	
Air Bearing Guide Rails	300	
Stator	8,232*	
Power Supply	1,800*	
Sensing, Processing, and System Controller	334	
Power Controller	1,500*	
Pneumatic Starter	45	
Miscellaneous (Bldgs., Vehicle Handling Apparatus, Lab Equipment)	316	
Total D & D Cost		12,652
Installation charges (for items noted with *)		2,160
Total D, D & I Cost		14,812

Vehicles - \$125,000

The current concept calls for the construction of two test vehicles, of different sizes, each conforming to the same general design, and each constructed of aluminum body and beryllium translators. The cost shown reflects inclusion of necessary test component adaptor sections and accompnaying tape recorders, power supplies, and inverters. These costs assume a successful initial research effort on the operating characteristics of the translators.

Air Bearing Guide Rails - \$300,000

As in the base of the vehicles, the costs of the air bearing guide rails assumes a successful research effort on the dynamics of the slipper-on-rail air bearing design. It includes the fabrication and precision alignment of stainless steel tubular rail sections and carbon steel supporting brackets (adjustable). The preponderance of these costs will be incurred in the installation and laignment procedure.

Stator Assembly - $8,232,000

This figure includes the accelerating portion of the accelerator, the decelerating portion, the capacitors, and the supporting and enclosing steel structures.

The iron and copper used in the deceleration section accounts for approximately $2.9 million of this cost. The manufacturers who provided SRI with cost estimates apparently estimated the costs per pound of this section to be the same as those of the acceleration section. SRI believes that the costs of the deceleration section of the stator subsystem can be reduced, perhaps by as much as 1/2, through design simplification, less costly materials, and more precise price estimating.

Power Supply - $1,800,000

Included here are the alternator, drive motor and rectifier. The major portion of this cost is for the alternator. Discussions with manufacturers reveal that cost estimates are usually made on the basis of dollars per kW of rated output, with premiums paid for special design features. The rated output used is usually the steady-state rating, a figure which is not readily derived from the transient requirements for the linear acceleration test facility. Based upon another method of rating, SRI derived a cost which is about two-thirds the estimated cost listed here. In contrast, one manufacturer estimated a cost which is about one and one-third times the listed estimated cost.

Sensing, Processing, and System Controller - $334,000

This figure covers the doppler radar sensing system and the information processing operation control computer. The control computer is provided with input/output accessories, printer, and control console. Each portion of this subsystem is within the current state of the art, and can be made available with a minimum of developmental effort. Programming and program testing are included in this figure.

Power Controller - $1,500,000

This includes system design, manufacture of the necessary regulator modules, and delivery. This cost estimate is subject to wide variance.

Estimates on this unit have ranged from $80,000 to $14,000. Such a wide range indicates the need for more detailed study of the system requirements and the preparation of a subsystem design.

Pneumatic Starter - $45,000

This cost covers the design, fabrication, installation and testing of the pneumatic starter unit, including the necessary compressor, compressor motor, and supporting structure.

Miscellaneous - $316,000

This total covers the accelerator building, laboratory building, and power building; laboratory test, inspection and handling equipment, test data reduction apparatus, and vehicle handling equipment. Test data reduction apparatus include a tape deck and signal conditioners to transfer information from data tapes into the system controller computer as well as an oscillograph for analog data readout. Five laboratory test stations are provided and equipped with test equipment to support the anticipated 1000 runs per year.

Installation - $2,160,000

This cost covers necessary on-site assembly, emplacement, alignment, and testing of major equipment items.

3. Incremental costs of system variation. Two most probably system variations have been considered and costed out. These are (1) the incremental cost per foot of test track, and (2) the incremental cost of all additions to subsystems necessary to accommodate one additional pound of test vehicle weight. The incremental cost figures are listed below.

$ 8,500	Per foot of acceleration section of track (These costs appear to be constant for all reasonable additions or reductions, i.e., ±25%.)
$ 1,000	Per foot of deceleration section of track
$10,000	Per pound of test vehicle weight (These costs are roughly constant within a small range about the design weight.)

4. Annual operating costs. These costs include staff salaries, repair, maintenance, and utilities. The staff is assumed to consist of a supervisory engineer, four technicians including one sensor technician, two maintenance men, and a secretary. Operating cost details are shown in Table VII. Job descriptions are included in Appendix B.

TABLE VII

ANNUAL OPERATING COSTS*

Salaries (approximations of expected government salary rates)		
1 Supervisor/Engineer	$12,000	
1 Senior Electronics Technician (supervisory)	9,000	
3 Electronics Technicians ($7,500 each)	22,500	
2 General Maintenance/Custodians ($5,500 each)	11,000	
1 Secretary/Bookkeeper	5,000	
Total Salaries		$59,500
Administrative materials, maintenance and repair expendables	2,000	
Utilities	2,000	
Replacement of ignitrons, thyratrons and other power system components (1 ignitron and 1 thyratron per month)	13,000	
Replacement of test equipment (assume 5-year life for all instruments)	10,800	
Replacement of dollies and dolly drive systems (assume 3-year life for all)	1,700	
Total Direct Costs		29,500
Total Annual Operating Costs		$89,000

* Does not cover allocated burden or investment amortization charges.

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APPENDICES

APPENDIX A

CONSTANT FREQUENCY LINEAR INDUCTION MOTOR TECHNICAL DETAILS

The following calculations are based to a large extent on the work of Laithwaite and his associates.[4,8] They have derived these equations from fundamentals and have shown experimentally that they are correct.

TRANSLATOR DESIGN

The fundamental equation for the thrust is Eq. 5, Ref. 4:

$$F = \frac{\rho_r J_s^{\,2}\ (w\alpha p)}{2v_s}\ N_v\ \frac{\sigma}{\sigma^2+(1/\tau/\omega)^2}\ G\ (\alpha, \tau\omega, p, \sigma) \qquad \text{(A-1)}$$

(For definite of terms, see the nomenclature at the end of the Appendix.) The function G is a very long and involved albegraic expression (see Ref. 4) which is substantially constant for a given design.

J_s, the peak surface current density in the stator, can be expressed in terms of the maximum field strength, B_m, in the stator:

$$J_s = \pi B_m g/\mu_0\ p \qquad \text{(A-2)}$$

where g is the gap spacing.

Also, the effective translator surface resistivity, ρ_r, can be expressed in terms of $\tau\omega$ and the various dimensions:

$$\rho_r = \frac{4\ p^2\ \mu_0 \omega d w}{\pi(\tau\omega)g^2(w+p)} \qquad \text{(A-3)}$$

Remembering also, that

$$v_s = 2\ f\ p = \omega p/\pi \qquad \text{(A-4)}$$

and substituting A-2, A-3, and A-4 into A-1, we obtain

$$F = \frac{2\pi^2\ Bm^2}{\mu_0(\tau\omega)}\quad \frac{\sigma}{\sigma^2 + (1/\tau\omega/^2}\quad \alpha\ N_v\ G\ d\ \ \frac{w^2}{w + p} \qquad \text{(A-5)}$$

The quantity in square brackets is plotted in Fig. A.1.

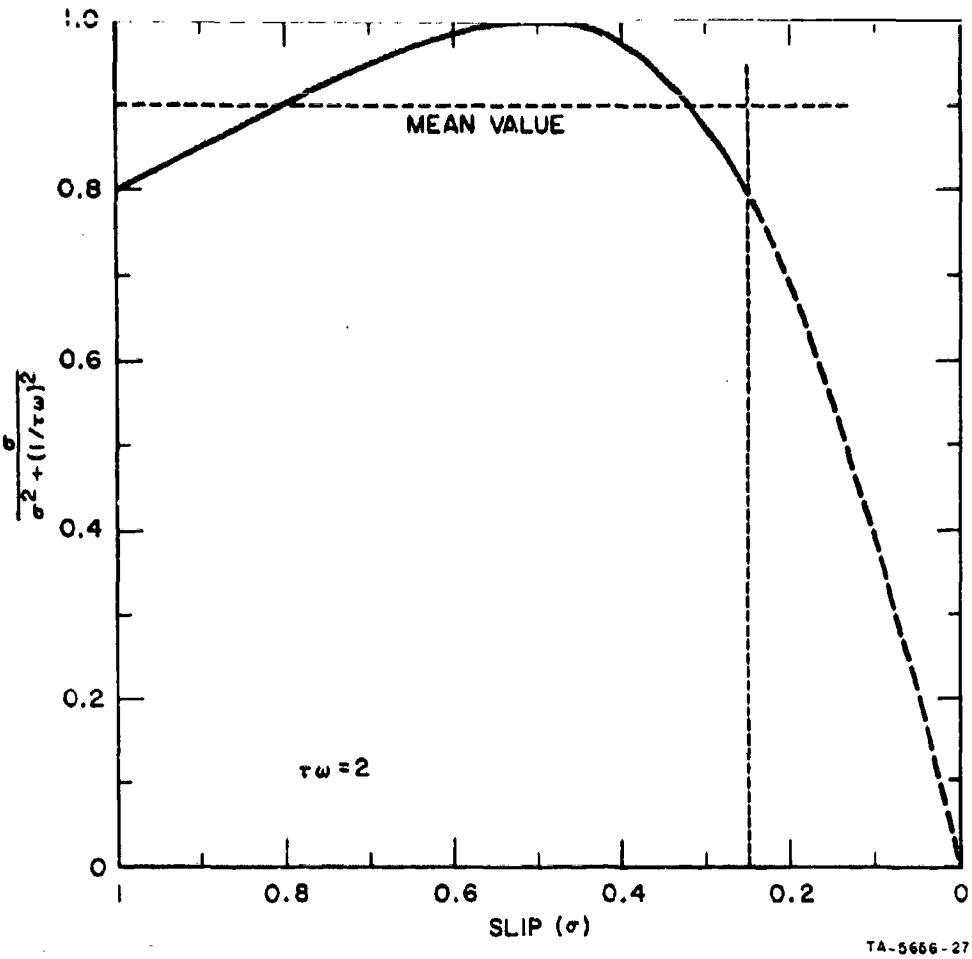

FIG. A-1 VARIATION OF SLIP FUNCTION

Translator Resistivity. From page 465 of Ref. 4 we calculate that a value of $\tau\omega = 2$ will make the square bracketed expression in A-5 constant to within ±10% down to a slip $\sigma = 0.25$. Hence, it is assumed the translator material is so chosen as to make $\tau\omega = 2$. (For the dimensions of the proposed system, this corresponds to a surface resistivity $\rho_r \sim 10^{-2}\ \Omega/m^2$.)

Field Strength. Field strength is limited by saturation effects. We believe a value of $B_m = 1.7$ webers/m² (17,000 gauss) can be reached.

Translator Length. A first order analysis on the acceleration of translators of various lengths shows that constant thrust can only be achieved when the length is an integral multiple of the pitch ($\alpha = 1,2$, etc.). This analysis neglects end effects which, however, can be minimized by skewing the translator.

Pitch. The pitch must be such that, at the peak vehicle velocity, the slip σ is 0.25. This means that the synchronous speed, $v_s = v/(1-\sigma) = 4v/3$. With a peak velocity of 360 m/sec (1180 ft/sec), v_s = 480 m/sec. The pitch is related to v_s by Eq. A-4. Since the translator must be at least one pitch long, we would like to make p as small as practical. On the other hand, the higher the frequency the greater the reactive power and the more difficult the generator design. We selected a pitch of one meter and a frequency of 240 c/s.

Translator Thickness and Width. For electrical efficiency the translator should be as wide as possible. For structural reasons, a narrow one is preferred. Hence, we have investigated translators whose width was of the same order of magnitude as the pitch. The trade-off analysis following this chapter indicates that, for reasonable structural design, two translators are preferable to three or four. (A single translator is undesirable because the driving force will not be concentric with the vehicle and will cause a large pitching or yawing moment.) We have selected two translators, each 1.47 m wide (58 inches), 1.0 m long (40 inches) and 2.5 cm thick (1 inch).

Temperature Rise. It is shown below that the heat absorbed by the translators during a run can amount to as much as 54,600 BTU (for the 60 g case). With a specific heat for beryllium of 0.52 BTU/°F/lb, this will cause a temperature rise of approximately 400°F. These estimates are conservative for they make no allowance for the heat saved during pneumatic start-up, which is considerable.

<u>Translator Number, Width and Thickness--A Trade-Off Analysis</u>. The total weight of the vehicle is made up of the translators, W_T, and the fixed weight, W_F, of the shell and components

$$W = W_T + W_F \tag{A-6}$$

The weight of the translators can be expressed in terms of their dimensions:

$$W_T = N_v \rho \ (w + 2c)\ p\ d \tag{A-7}$$

where w is the effective width in the stator and c is the overhang (assumed equal on both sides of the stator).

If the vehicle acceleration is a, we have, by Newton's law,

$$Wa/g_c = F \tag{A-8}$$

where F is as given in Equation A-5, and can be expressed as

$$F = C\ N_v d\ \frac{w^2}{w+p} \tag{A-9}$$

where
$$C = \frac{2\pi^2\ B_m^2}{\mu_o\ (\tau\omega)} \left[\frac{\sigma}{\sigma^2 + (1/\tau\omega)^2}\right] \alpha\ G \tag{A-10}$$

Combining A-6 through A-9 yields:

$$W_F + \rho_v p N_v d\ (w + 2c) = \frac{g_c}{a}\ C\ N_v\ d\ \frac{w^2}{w+p} \tag{A-11}$$

To utilize this equation we will now make the following assumptions:

- B_m is constant at 1.7 webers/m provided the gap does not exceed 4 cm. This means that the maximum translator thickness, d, must be less than 4 cm.
- The term in square brackets is constant (0.9) and G is constant (0.85). (See Ref. 4.)
- The translator width, w, should be in the vicinity of $p = 1$ m.
- The length of the translator is p. This is already inherent in Equation A-7 and implies $\alpha = 1$.
- The number of translators should be greater than one to permit centering the resultant driving force in the vehicle.

- The average density of the translator is that of pure beryllium.
- The overhang, c, is 0.15 m.
- The weight of vehicle shell and contents is 125 kg.

With these assumptions, we can compute d for a variety of widths, w, and translator numbers, N_v. This is done in Table A-1. The table also shows the weight of the translators and indicates that the wider the translators the lower their weight. However, their mechanical stiffness becomes less and less, and the cost of the stator goes up. Therefore, we have selected two 2.5-cm-thick translators, 1.17 + 0.3 = 1.47 m long.

TABLE A-1

TRADE-OFF COMPUTATIONS FOR TRANSLATOR DIMENSIONS

Effective Translator Width: w (m)	Total Translator Width: w + 2c (m)	Total Translator Weight: W_T (kg)	Translator Thickness: d (cm)		
			$N_v=2$	$N_v=3$	$N_v=4$
0.7	1.0	361		d > 4.0	
0.8	1.1	244		4.00	3.00
0.9	1.2	193		2.90	2.17
1.0	1.3	156	3.26	2.17	1.65
1.1	1.4	135	2.62	1.74	1.31
1.2	1.5	120	2.16	1.44	1.08
1.3	1.6	111	1.87	1.25	0.94

For a more sophisticated trade-off analysis, one should include a mechanical stiffness criterion and a cost criterion including the stator width. This could not be done under the present contract.

STATOR DESIGN

Accelerating Section

Having obtained or assumed ω, p, w, B_m and g ($g \approx d + 0.75 = 3.25$ cm) and the overall length of the accelerating sections, we can compute the electrical requirements of the stator. Engineering practice suggests three-phase excitation with 12 slots per pole side, two coils per slot. The phase belts are 60° and each coil spans ten slots to give 5/6 chording for optimum wave form.

The flux per pole is obtained from

$$\phi = B_m \, wp \, 5/6 \approx 1.7 \text{ weber}$$

The number of turns per phase coil is given by[7]

$$N_p = \frac{2 \quad E}{3 \; \omega \; (K_p K_d) \; \phi} \approx 4$$

if the applied voltage, E, is 1.4×10^4 V(RMS) and the winding and pitch distribution factors, $K_p \, K_d$, are assumed to be 0.935.

The series current is computed from the assumption that the total magnetomotive force appears in the gap. The equation is derived in Ref. 7 and, after suitable modification for the linear machine, is

$$I_s = \frac{\pi^2 \quad 2 \quad g_{eff} \; \phi \; k_i}{24 \; (K_p \; K_d) \; wp \; N_v \mu_o} = 2.25 \times 10^3 \text{ A(RMS)}$$

with $g_{eff} = 1.06 \; g = 3.45 \times 10^{-2}$ m to allow for the fringing factors of the slots, and $k_i = 1.02$ to correct the magnetizing current for core and pole tooth saturation.[7]

Reactive power is found from the relation:

$$P_{VAR} = 2\omega \frac{B_m^2}{2\mu_o} \; g \; w \; L \qquad \text{(A-12)}$$

where L is the stator length.

For the 200 g case, the stator length is 74 ft = 22.5 m, and

$$P_{VAR} = 3.3 \times 10^9 \ \text{VAR} = 3.3 \times 10^3 \ \text{MVAR}$$

For the 50 g case, the field strength, B_m, can be reduced to 1.07 webers/m² but L is increased to 350 ft = 107 m and

$$P_{VAR} = 6.2 \times 10^9 \ \text{VAR} = 6.2 \times 10^3 \ \text{MVAR}$$

The partition of power, or the way the power is utilized is illustrated in Fig. A-2. The power supplied to the stator is estimated to be some 2% lower than the power generated due to losses in the control system and the capacitors. Some stator power goes into copper and iron losses. Of the power applied to the vehicle, a portion goes into heat and only the remainder is available for thrust.

Iron and copper losses are computed from Ref. 10 as about 25 MW and 10 MW, respectively. The total power applied to the vehicle is mav_s, or about 240 MW for the 200 g case. The power available for thrust is, of course, $mav = mav_s(1-\sigma)$. It starts at zero and reaches its peak of 180 MW at the maximum vehicle velocity.

The heat that must be absorbed in the translators during acceleration is, for the 200 g case,

$$H = \int_0^{0.15} mav_s \, \sigma \, dt$$

Since σ varies linearly from 1 to 0.25 as t changes from 0 to 0.15, $dt = -0.2 \, d\sigma$.

Therefore,

$$H = mav_s \int_1^{0.25} -0.2 \, \sigma d\sigma = 22.5 \ \text{MW sec} \quad \text{or} \quad 21{,}300 \ \text{Btu}.$$

For the 60 g case the corresponding heat is 36.0 MW sec or 34,200 Btu.

Decelerating Section

The vehicle is decelerated by dynamic braking; that is, by exciting the winding with dc so that the vehicle coasts through a steady field in which the polarity flips at each pole. The thrust-slip curve is approximately as shown in Fig. A-3.

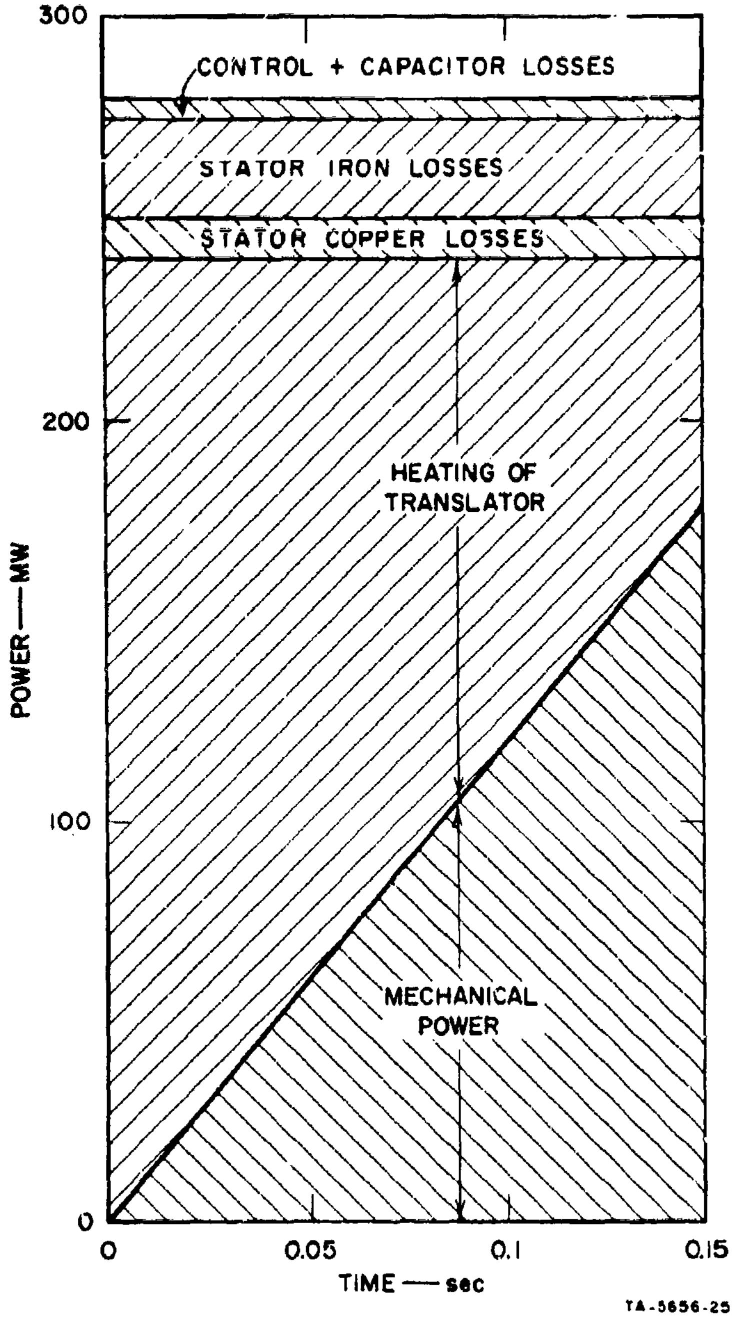

FIG. A-2 POWER PARTITION — 200 g PAYLOAD

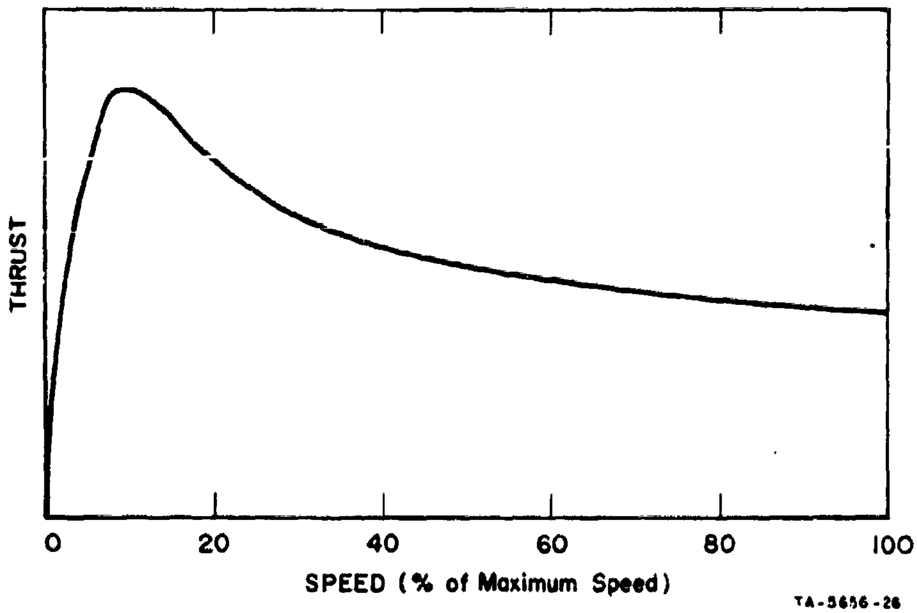

FIG. A-3 THRUST-SPEED CURVE FOR DECELERATION SECTION

The speed at which the peak braking force occurs can be computed[7] to be about 10% of maximum speed. Thus, it is estimated that the place where mechanical braking has to take over occurs at about 5% of maximum speed, i.e. at a velocity of (0.05)(1160) = 58, i.e. at about 60 ft/sec.

The additional heat load imposed on the translators is about equal to the kinetic energy stored in the vehicle. This amounts to 13.5 MW-sec for the 200 g case and 21.6 MW-sec for the 60 g case. Therefore, the maximum heat load on the translators occurs in the 60 g case, and amounts to about 57.6 MW-sec or 54,600 Btu.

ALTERNATOR

The alternator must be able to deliver 280 MW for about 0.12 sec or 130 MW for 0.5 sec (see Table III) or 33.6 and 65 MW-sec, respectively. In mechanical units, 65 MW-sec = 6.63×10^6 kg-m.

Rotor Inertia. The rotor inertia is selected so that the speed reduction of the rotor does not exceed 10% of its initial speed after the maximum amount of energy has been withdrawn. The kinetic energy stored at the initial speed, N, is

$$E_i = \frac{1}{2} I \ (2\pi N/60)^2 \qquad (\text{kg-m})$$

where I is the moment of inertia in m-kg-sec^2 and N is in r/min. At the end of the run, the kinetic energy remaining is

$$E_e = \frac{1}{2} I \ (2\pi \ 0.9N/60)^2$$

Therefore, the energy used up is

$$E_i - E_e = (1 - 0.9^2) \ E_i = 0.19 \ E_i$$

Assuming an overall alternator efficiency of 95%, the energy delivered will be:

$$E_d = 0.19 \times 0.95 \ E_i = 0.18 \ E_i$$

This energy must equal the maximum energy demanded by the system, or 6.63 x10^6 kg-m. Therefore,

$$E_i = \frac{1}{2} I \ (2\pi N/60)^2 = 36.8 \times 10^6 \text{ kg-m}$$

or

$$I = 6.72 \times 10^9/N^2 \ \text{kg-m-sec}^2 \qquad (A\text{-}13)$$

Rotor Speed. Rotor speed is limited by the limit on surface velocity of the rotor. Thus, it is a practical rule that the surface velocity $2\pi RN$ should not exceed 9,000 m/min. Combining this condition with equation A-13 and assuming the rotor to be a solid circular cylinder of constant density, ρ, we can obtain a relation between its radius, R, and its length, h; using $I = \pi\rho R^4 h/2$,

$$\rho R^2 h/8\pi = 83$$

If the average density is that of steel ($\rho = 7.8 \times 10^2$ kg-sec^2/m^4)

$$R^2 h = 2.7 \ \text{m}^3$$

A reasonable selection meeting these requirements is R = 1.5m (~ 5 ft), h = 1.2 m (~ 4 ft), N = 960 r/min.

Drive Motor. The size of the drive motor is determined by friction and windage losses in the alternator and by the energy required to bring it up to speed in a reasonable period of time.

If the energy delivered, E_d, is to be made up in ten minutes, the power required is $65 \times 10^3/10 \times 60 \approx 100$ kW. Windage and friction losses are estimated at 0.5% of the steady rated power of the alternator. This is likely to be about 0.5% of 80 MW, or 400 kW. Thus, our estimate of the power required is 500 kW or about 700 hp. Since this is a very coarse estimate, we have specified a 1000 hp drive motor.

SENSING SYSTEM AND SIGNAL PROCESSOR

A functional diagram of the sensing and processor and the controller is shown in Figure 14. Tracing the operation sequence, we assume that the vehicle is in its start position, all circuits are ready, and that appropriate program data has been put into the memory unit. The radar is operating and the mixer output is some value (say zero), established by the transmitter frequency and the antenna-vehicle geometry. When the vehicle motion begins, a signal from a first motion switch gates the timing generator "on," which produces a precise timing signal at, say, 1 MHz. This signal is counted in the preset register, and causes the delivery of an output pulse whenever the count coincides with the numbers that have been programmed into the memory. These output pulses trigger the bi-stable multi-vibrater to produce the standard signal, $E_s(t)$. Motion of the vehicle and reflecting surface produce maxima and minima in the signal output of the Doppler mixer, which is Doppler signal, $E_d(t)$. The two signal voltage square waves are combined in a differential amplifier and phase comparator, to produce the feedback signal which will be applied to the power control system.

The system accuracy is affected by the following factors:

(a) The sinusoidal Doppler output, $E_d(t)$ depends upon a knowledge of the radiation wave length and the system geometry. These must be measured and maintained within established limits.

(b) The square wave Doppler output, $E_d(t)$, must have axis crossings that coincide in time with those of the sinusoidal $E_d(t)$. This is controllable within any reasonable limits by the choice of appropriate amplifier gain ahead of the limiter.

(c) The accuracy of the timing signal generator, which should be crystal controlled.

(d) The rise time and delay in pulse forming, counting, and triggering circuits.

(e) The accuracy with which program data are prepared.

Analysis of the specific circuitry chosen for the system will be necessary at a later stage of the development. There appears to be no reason why a maximum time error of one timing generator cycle (approx. 10^{-6} sec) cannot be attained, as the digital techniques that are described are inherently very fast.

Because of the one-time nature of this device, it may be advantageous to procure off-the-shelf test equipment, i.e. signal generators, couplers, detectors, etc., and assemble the system from these components.

BUILDING AND SITING REQUIREMENTS

Stator Building

(1) 25 ft wide x 14 ft high x 1150 ft long (approx.). Accelerator approx. 17 ft wide with 4-ft walk and maintenance aisles either side; no unusual height requirements other than short overhead craneway or lifting hoist at head end of accelerator.

(2) Reinforced concrete moisture-proofed floor capable of supporting accelerator loads of 50,000 lb/linear ft (accelerator and supporting frame approximate 17 ft in width).

(3) Floor slope shall not exceed one milliradian (0.1%) in longitudinal direction and shall have a transverse slope for drainage of washdown water of 1/8 inches per foot.

(4) Allowable differential settlement between any two points shall not exceed one milliradian (1 foot per 1000) and grade change at any point shall not exceed one/tenth of this value, except that final average slope after settlement and in longitudinal direction shall not exceed requirement (3) above. (Accelerator frame may have leveling jacks to compensate for settlements.) "Final" slope may be taken as 20 years after application of load.

(5) No unusual requirements for walls and roof; construction and materials compatible with Base standards, except that there shall be no windows other than viewing ports from laboratory (sky lights allowable).
(6) Enclosure shall be weathertight, have forced air ventilation and space heaters (or equal). No unusual light requirements. (Under particular temperature variations and design of accelerator's expansion/contraction characteristics, air conditioning may be required.)

Power Building

(1) 20 ft wide x 50 ft long x 18 ft high (approx.).
(2) Reinforced concrete moisture-proofed floor capable of supporting a 170,000-lb generator/alternator (over an approx. base area of 12 ft x 24 ft) with a 500,000 ft-lb overturning torque between 12 ft frame points. (Approx. size of reinforced 2-way slab: 8 inches.) Floor shall be separated from stator slab to minimize vibration transfer.
(3) No unusual requirements for walls and roof: construction compatible with Base standards.
(4) Enclosure shall be weathertight; forced air ventilation; space heaters (or equal).

Laboratory Space

(1) Approximately 20 ft x 25 ft x 9 ft high to accommodate 6 people and instrumentation; no unusually large instruments contemplated. Test, gage and computer bench along one wall, approx. 4 ft x 10 ft.
(2) Floor may be concrete slab on grade or supported wood frame; live loads approx. 75 psf.
(3) Enclosure to be air-conditioned, provided with male and female lavatories. No unusual requirements for water, power, gas, lighting and sewage disposal.

Outside Storage, Parking and Loading Areas

(1) Asphaltic or concrete surfacing shall be provided; no unusual requirements. Parking space for 10 vehicles.

Site

(1) Natural ground shall be flat to rolling.

(2) Finished grade shall have level pad areas to accommodate stator, power and lab buildings, and outside storage, parking and loading spaces. Pad elevations may vary with respect to each other, except for stator, lab and loading dock (at head end of accelerator) which shall be at the same grade. Power and stator building pads shall have edge slopes sufficient for positive and rapid disposal of drainage water away from buildings.

(3) Due to the critical stator tolerances differential settlements must be minimized. The soils underlying the stator building should be generally of sandy loam to rocky material, uniform in nature, and should be free of expansive clays. Bearing capacity alone will generally govern on all other facilities.

(4) The tail end of stator building should be either

...adjacent a natural or excavated earth slope

(or)...provided with an earthen enbankment approx. 10 ft high and 30 ft (average) thick

(and)...generally restricted from personnel.

(5) Site shall have a security fence and shall be a minimum of 10 ft from all sides of stator building.

(6) The site shall be provided with:--

(a) Electric service -- sufficient to power a 1000-hp 3-phase motor; no other abnormal loads. Also, for air compressor, air conditioning, interior and yard lighting, and general lab equipment.

(b) Domestic water service -- drinking, lavatories, and lab accommodating approx. 6 people and wash-down and yard maintenance. No unusual pressure, flow, treatment or quality requirements.

(c) Gas service -- air conditioning, space heaters, laboratory; no unusual requirements.

(d) Sanitary sewer lateral -- domestic wastes only, to accommodate approx. 6 people.

(e) Telephone, fire and safety warning systems -- no unusual requirements.

Nomenclature

Appendix A

a	acceleration	m/sec^2
c	translator overhang	m
d	translator thickness	m
f	frequency	cps
g	gap width	m
g_c	gravitational constant	= 9.807
g_{eff}	effective gap width	m
h	rotor length	m
k_i	correction factor	
m	mass of vehicle	$kg\text{-}sec^2/m$
p	pole pitch	m
v	vehicle velocity	m/sec
v_s	synchronous velocity	m/sec
w	effective translator width	m
B_m	field strength	$webers/m^2$
C	constant defined in Eq. A.10	
E	voltage, energy	
E_i	initial energy	kg-m
E_e	energy at end of run	kg-m
E_d	energy delivered	kg-m
F	stator thrust	kg
G	function in Eq. A.1	
H	heat absorbed in translator	W-sec or BTU
I	rotor moment of inertia	$kg\text{-}m\text{-}sec^2$
J_s	peak surface current density	A/m^2
K_d	winding distribution factor	
K_p	pitch distribution factor	
L	stator length	m
N	rotor speed	r/min

Nomenclature Appendix A (Continued)

N_v	number of translators on vehicle	
P_{var}	reactive power	VA
R	rotor radius	m
W	total weight of vehicle	kg
W_F	weight of vehicle shell and components	kg
W_T	weight of translators	kg
α	ratio of translator length to pole pitch	
μ_o	= $4\pi \times 10$	henries/m
ρ	rotor mass density	$kg\text{-}sec^2/m^4$
ρ_r	translator surface resistivity	$ohms/m^2$
ρ_v	translator weight density	kg/m^3
σ	slip	
$\tau\omega$	translator time constant	
ϕ	flux	weber
ω	radial frequency	rad/sec

APPENDIX B

COSTS AND PERSONNEL

This appendix contains

- Cost breakdowns and basis for cost estimates
- Copies of letter estimates from major manufacturers
- Job descriptions for the operating personnel

COST ESTIMATES

Subsystem cost estimates are based on catalog prices and manufacturers' estimates of unit costs where available. Otherwise, educated guesses were used. The cost estimate should not be used as a basis for establishing bid prices.

TABLE B.1

COST BREAKDOWN

Item	Unit Cost	Parts Cost	Sub-System Cost	System Cost
		$	$	($1000)
Vehicles				125
A. Fin Body Assembly (2 req.)	27,050$		54,100	
Fin material—278 lb Beryllium	80$/lb	22,200		
Fin machining and cladding 250 lb	10$/lb	2,500		
Body and nose fairing—				
Materials and machining 50 lb	5$/lb	250		
Air bearings 40 lb	10$/lb	400		
Compressed air bottle, titanium		1,700		
B. Adapter Section (5 req.)	14,200$		71,000	
Rear fairing and adapter 60 lb	10$/lb	600		
Tape recorder		7,000		
Signal conditioning amplifiers (8 each)	700$	5,600		
Power supply, batteries and inverters		1,000		
Air Bearing Guide Rails (From Westinghouse estimate, excludes foundations)				300
Stator				8,232
A. Stator Iron and Copper (includes material and construction)			5,500,000	
Accelerating section 1300 tons	2,000$/ton	2,600,000		
Decelerating section 1450 tons	2,000$/ton	2,900,000		
B. Capacitors—6.2 million kVA at 240 Hz	0.323$/kVA		2,000,000	
C. Supporting Structure			732,000	
Structural steel 2400 tons	275$/ton	660,000		
Installation 2400 tons	30$/ton	72,000		
Power Supply				1,800
A. Alternator, 280,000 kW peak (steady state rating 80,000 kW)	21$/kW		1,680,000	
B. Drive Motor, Control, Exciter			120,000	
Sensing, Processing and System Controller				334
A. Sensing Circuits			9,400	
Microwave signal generator, HP 8616B		1,450		
Directional coupler, HP S750		150		
Crystal detector, HP S24A		175		
Amplifier, HP 462A		325		
Waveguide, 25 ft	10$/ft	250		
System design and test, 1.5 man-months	2100$/m-m	3,150		
Installation and checkout, 2 man-months	1950$/m-m	3,900		
B. Processor and System Controller*			324,000	
Central processor C 4050		71,200		
Magnetic tape input D 4530 A1		18,920		
Memory add-on, 4096 words capacity C 4050WX (7 each)	20,000$/unit	140,000		
Printer M4260		38,000		
Console M4741		2,500		
Analog input channel		18,430		
Digital input channel		5,100		
Output channels		4,880		
Computer program development		25,000		

* Estimates are based on General Electric components.

(Continued)

TABLE B.1 COST BREAKDOWN (Continued)

Item		Unit Cost	Parts Cost	Sub-System Cost	System Cost
			$	$	($1000)
Power Controller					
(Estimated equivalent steady state rating is 60,000 kW)		25$/kW			1,500
Pneumatic Starter					45
A. Vessel				34,100	
Material, carbon steel	74,800 lb	0.14$/lb	10,500		
Labor	74,800 lb	0.30$/lb	22,400		
Flange			1,200		
B. Cylinder and Auxiliaries				10,900	
Cylinder—material and machining	805 lb	3$/lb	2,650		
Compressor			1,650		
Flexible bellows			5,[illegible]00		
Water pump, tank and valve			1,000		
Miscellaneous					316
A. Buildings and Grounds				251,200	
Accelerator bldg.—1125' x 25'			177,100		
Power bldg., incl. utilities			16,300		
Laboratory bldg., incl. utilities			15,600		
Parking areas, walkways, fencing and landscaping			19,200		
A&E fees (10% of costs)			23,000		
B. Data Reduction Facility				15,650	
Sanborn 8 channel recorder			6,250		
High-gain dc amplifiers (4 each)		700$/unit	2,800		
AC amplifiers and demodulators (4 each)		400$/unit	1,600		
Magnetic tape transport			5,000		
C. Laboratory Equipment				42,125	
100 kVA power supply 400 Hz		40$/unit	4,000		
Oscilloscopes	(5 each)	1000$/unit	5,000		
Volt-ohm-milli-ammeter	(5 each)	50$/unit	250		
Vac tube voltmeter	(5 each)	75$/unit	375		
Digital voltmeter	(5 each)	2500$/unit	12,500		
Servo tables	(2 each)	7500$/unit	15,000		
Tilt tables	(2 each)	2500$/unit	5,000		
D. Vehicle Handling				6,000	
Propulsion units	(2 each)	1500$/unit	3,000		
Dollies	(3 each)	1000$/unit	3,000		
E. Vehicle Stopping Components				1,000	
Installation — Based on Total					
Cost of stator, power supply and power controller (excluding supporting structure, 20% of cost)					2,160
Annual Salary Costs					
(Salaries are approximations of expected governments salary rates)					
1 - Supervisor/Engineer			12,000		
1 - Senior Electronics Technician (supervisorial)			9,000		
3 - Electronics Technicians ($7,500 each)			22,500		
2 - General Maintenance/Custodians ($5,500 each)			11,000		
1 - Secretary/Bookkeeper			5,000		
					60

GENERAL ELECTRIC
COMPANY

235 MONTGOMERY STREET, SAN FRANCISCO, CALIFORNIA 94106 . . . TELEPHONE 415 -434-2211

INDUSTRIAL
SALES
DIVISION

January 27, 1966

Stanford Research Institute
Menlo Park, California 94025

Attention - Mr. Albert E. Moon,
Systems Economy Research Program

Gentlemen:

This will confirm our telephone conversation of January 25 relating to your letter of January 17.

Several of our developmental and test systems application engineers in Schenectady have reviewed the specifications accompanying your letter of January 17 and have prepared a very approximate engineering estimate based strictly on the hardware concepts described in your systems specifications. This estimate covers these major items of equipment engineered as a system.

Linear stator, with guide rail, vehicle, and pneumatic starter

Power supply consisting of 280,000 Kva alternator with suitable drive motor.

Capacitors, 2,000,000 KVAR, 240 cycles.

Power controller with sensing and signal processing functions

ESTIMATED PRICE--$12,500,000

We have not attempted to estimate installation costs or have we included any facilities such as foundations, buildings, etc.

We agree with the thought expressed in your letter that the linear induction system appears to be a sound approach. We see some alternative possibilities for power and control system refinements but their merits can only be established by an extensive and penetrating engineering study.

Yours very truly,

G. L. CROW,
INDUSTRIAL SALES

km

Westinghouse Electric Corporation

Hendy Avenue
Sunnyvale, Calif.
735-2226

February 7, 1966

Stanford Research Institute
333 Ravenswood Ave.,
Menlo Park, Calif.

Attention: Mr. Albert E. Moon,
Systems Economy Research Program,
Management Sciences Division

Subject: High G Linear Acceleration Test Facility

Gentlemen:

It is a pleasure to respond to your letter of 17 January 66 which requests cost estimates for the subject test facility. This data is presented to you for planning purposes only and is considered to be budgetary in nature. We have evaluated the various subsystems and submit the following information:

SECTION 2000 - AIR BEARING GUIDE RAIL:

Air bearing guides and air bearing rails including air pressure vessel, but excluding rail support foundation.

Estimated cost - $ 300,000

SECTION 3000 - STATOR:

Linear stator, but excluding installation and support.

Estimated cost - $5,500,000

SECTION 4000 - POWER SUPPLY:

14 KV alternator, drive-motor and related controls.

Estimated cost - $2,500,000

SECTION 6000 - SYSTEM CONTROLLER:

Computer, including program and operating console.

Estimated cost - $ 300,000

Stanford Research Institute
Mr. Albert E. Moon

February 7, 1966

SECTION 7000 - POWER CONTROLLER:

(A) Sequence Switching - Under today's technology, the system control could most effectively be accomplished with the use of silicon controlled rectifiers for the required power at 14 KV. The SCR system installed is estimated to cost in the order of $14,000,000.

It may not be within today's state-of-the-art to control this power (14 KV) by the use of ignitrons in inverse-parallel configuration. The problem of control with a load of varying power-factor would have to be solved with a fairly extensive development program.

(B) Power Factor Correction - Installed power factor correction for the specified 14 KV and a capacity of 3500 MVAR for a 250 CPS system is estimated at $7,000,000.

With reference to Section 5000 - "Sensing and Signal Processing", we will require additional time to review the CW Doppler radar technique described. This evaluation and estimate would be prepared by our Aerospace Division located at Westinghouse Defense Center in Baltimore, Md. Regarding Section 1000 (Vehicle) and 8000 (Pneumatic Starter), Westinghouse would elect to sub-contract non-critical parts.

Although the major portion of the over-all system is within present state-of-the-art, the very demanding interfaces may make it quite difficult to obtain the desired and required results from several individual suppliers. For this reason, you or your client may find that the responsibility for such a system should be assigned to one supplier. Westinghouse is both capable and interested in undertaking the production of this system; many of the individual components are presently within our existing Product Lines. A "Turn-Key" contract is proposed, however the alotted time and information has precluded our submitting an estimated cost for such a contract with this letter. Here again, the important interfaces point out the wisdom of utilizing a single supplier, experienced in managing the "total package" concept.

Stanford Research Institute
Mr. Albert E. Moon

February 7, 1966

Our Sunnyvale Division has a considerable amount of experience as a major contractor for Test Facilities; you and others associated with this program are cordially invited to visit us and see our facilities. The undersigned will be glad to make the necessary arrangements.

Thank you for your consideration. We hope that the foregoing will fill your present needs. In the event you have questions or require further information, please call us at any time.

Very truly yours,

WESTINGHOUSE ELECTRIC CORPORATION

R.W. Sumner.

Robert W. Sumner,
Atomic, Defense & Space Group

RWS:bam

BRIEF JOB DESCRIPTIONS OF OPERATING PERSONNEL

A. (1) Supervisor/Engineer—This job calls for a person with both technical knowledge encompassing the entire high-g linear accelerator system and administrative proficiency. Among the tasks to be performed by this individual are:

1. Scheduling of facility usage
2. Preparation of facility operating budget
3. Accounting for facility expenses and revenues
4. Purchase of necessary facility equipment and supplies (as budgeted)
5. Selection of other facility personnel
6. Overseeing operating of all tests

B. (4) Electronics Technicians (1-senior-supervisor, 3-non-supervisory) —These jobs encompass operating, maintaining and repairing electromechanical and mechanical portions of the facility. Among the tasks to be performed by these individuals are:

1. Operation of the accelerator control console
2. Assistance with pre-run testing and inspection of components to be tested.
3. Assistance with installation of components and adapter sections in test vehicles
4. Operation of test vehicle dolly and drive system for purposes of attaching and detaching dolly's to front and rear of test track
5. First echelon maintenance and inspection of power facilities
6. Assistance with post-run testing and inspection of components
7. Operation of overhead electrical crane hoist within laboratory
8. Inspection and replacement of ignitrons, thyratrons and other devices as req .red.

C. (2) General Maintenance/Custodian—These jobs encompass light maintenance of operating and non-operating portions of the entire facility. Among the tasks to be performed by these individuals are:

1. Attachment and removal of test vehicle dollies from ends of test track (using dolly drive system)
2. Use of motor-compressor unit for filling compressed air bottles
3. Assistance as required, in moving test components, maintaining test and recording equipment
4. Cleaning and maintaining physical orderliness in all areas of test facility, inside and outside of buildings
5. Painting, light carpentry, plumbing and other light repair and improvement tasks as required by the facility director

D. (1) Secretary/Bookkeeper—This job encompasses typing, taking dictation, light expense, billing and revenue bookkeeping and other administrative assistance duties as required by the facility director.

APPENDIX C

MHD POWER GENERATOR — ESTIMATES

This appendix presents copies of a series of letters from two leading manufacturers of MHD generators, the Avco-Everett Research Laboratory and the Hercules Powder Company. These letters contain ideas for short period dc and ac power generation by MHD generators and cost estimates.

AVCO-EVERETT RESEARCH LABORATORY

2385 REVERE BEACH PARKWAY · EVERETT, MASSACHUSETTS 02149

March 8, 1966

Professor R. H. Eustis
Stanford University
Mechanical Engineering Department
Stanford, California

Dear Bob:

Thank you for your letter of March 4 regarding Dr. Chilton's report.

I have examined my letter to you of October 7, and it is quite agreeable with me that Dr. Chilton include it in his report. I do believe that we can achieve some better fuel economy than is quoted on page 3 of the letter. I think we should also point out that the estimated costs are on a single prime contract basis for the completed installation.

Subsequent to writing to you on October 7, Dr. Chilton informed us that he was interested in alternating current, and I wrote to him on January 4 describing a method by which we could supply A.C. I believe you received a copy of that letter. I think the price estimated in my letter to Dr. Chilton is actually more appropriate to the objective he has in mind and would suggest that letter might also be included.

We would of course be happy to cooperate with you and Stanford Research Institute in any further work on this subject.

Sincerely,

T. R. Brogan

/mcd

A DIVISION OF AVCO CORPORATION

AVCO-EVERETT RESEARCH LABORATORY

2385 REVERE BEACH PARKWAY · EVERETT, MASSACHUSETTS 02149

January 4, 1966

Dr. E. G. Chilton, Manager
Mechanics Department
Physics Division
Stanford Research Institute
Menlo Park, California 94025

Dear Dr. Chilton:

With reference to your letter of December 1, 1965, concerning the possibilities of obtaining AC from the MHD generator power supply that we had discussed with Professor R. H. Eustis and which is described in my letter to him of October 7, we have considered the problem and find that a complete system capable of supplying 100 Megawatts, 3 phase at a nominal voltage 2 to 6.9 KV, with variable frequency 20 cps to 120 cps with a one-second duty cycle could be supplied at an estimated cost of $2,200,000.00 to $2,500,000.00. Conversion of the DC to AC can be accomplished using a conventional inverter. The frequency is controlled by controlling the triggering rate, and further, the triggering sequence as well as the generator controls (i.e. seed flow rate) can be combined to yield the 5 to 10 milliseconds response times you desire. Samples of the variable to be controlled can be used in a feedback-servo system connected to the trigger control circuit and generator controls to obtain the desired variation of the controlled variable.

A DIVISION OF AVCO CORPORATION

In considering the problem of supplying AC the following operating conditions for this inverter were set on the basis of your letter of December 1, 1966, our telephone conversation, and the correspondence with Professor Eustis:

OUTPUT REQUIREMENTS:

1.	Output circuit configuration	3 phase - 3 line
2.	Frequency	20 to 120 cps - controlled
3.	Power	100 Megawatts, at 120 cps (in operation, power proportional to frequency)
4.	Duty cycle	One second - repeatable after all components have returned to ambient temperature. (Longer pulses at lower power level could also be accommodated.)
5.	Voltage	4 KV - L-L (not important to the cost estimate)
6.	Power factor	0.9 lagging

INPUT CONDITIONS:

1.	Input circuit configuration	DC - 2 line
2.	Power	110 megawatts (10% loss assumed in the system between the generator and load)
3.	Voltage	2 KV (The cost estimate is relatively insensitive to the magnitude of this voltage.)
4.	Duty cycle	One second

In arriving at an estimated cost of the inversion equipment, full advantage was taken of short-term heat sinking capabilities of the switching elements, of the large increase in transformer rating for two-second periods, and of the reduction in the required voltage rating of reactive supplies for short term operation. The estimate was based on readily available components and is conservative since the calculation was done on the basis of full load conditions (100 MW) for one second. Since the nature of the filtering requirements were unknown, the cost estimate does not include special allowances for filtering or for other than ordinary controls. Unless these requirements are elaborate, however, the cost should be nominal.

Please do not hesitate to contact us if you have further questions concerning this estimate.

Sincerely yours,

Thomas R. Brogan

Thomas R. Brogan
Project Director
MHD Power Generation

cc: Prof. Robert Eustis, Stanford Univ.

where V is the velocity of the piston*

c is the local speed of sound in front of the piston

c_i is the speed of sound in the undisturbed medium in front of the piston (1126 ft/sec assumed).

Only right-running waves (in the direction of motion) occur in this case, since as shown later, the tube is so long that no reflections from its end will occur during the acceleration portion of the run.

With $k = 1.4$

$$\frac{c}{c_i} = \frac{k-1}{2}\frac{V}{c_i} + 1 = 0.2\frac{V}{c_i} + 1$$

and with a constant acceleration, a,

$$\frac{V}{c_i} = \frac{at}{c_i}$$

The isentropic relationships give

$$\frac{p}{p_i} = \left[\frac{c}{c_i}\right]^{\frac{2k}{k-1}}$$

$$\frac{T}{T_i} = \left[\frac{c}{c_i}\right]^2$$

where p and T represent the state of the air directly in front of the package. The local Mach number in front of the package is given by

$$M = \frac{V}{c} = \frac{V}{c_i} \Big/ \frac{c}{c_i}$$

The pressure, force and Mach number for the three cases in question** are given in the following table. Case I is calculated for three different initial pressures in the tube: 0.1, 0.5, 1 times atmospheric pressure; and Case II is calculated for 0.01, 0.1, 0.5 and 1 times atmospheric pressure. Case III would be the same as Case I.

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* For nomenclature see the end of this Appendix.

** Case I: 50-lb payload, 50-lb vehicle
Case II: 200-lb payload, 200-lb vehicle
Case III: 50-lb payload, 200-lb vehicle

TABLE D.1

BACK PRESSURE CALCULATIONS

CASES I AND III

a = 200 g   t = 0.15 sec

Time t	Track Distance s	Mach Number	Drag Pressure * $(p_b)_{0.01}$	$(p_b)_{0.1}$	$(p_b)_{0.5}$	$(p_b)_{1.0}$
sec	ft		lbf/ft^2	lbf/ft^2	lbf/ft^2	lbf/ft^2
0.01	0.3	0.056		229	1145	2290
0.02	1.3	0.093		249	1245	2490
0.05	8.0	0.271		312	1560	3120
0.10	32.2	0.513		452	2260	4520
0.15	72.4	0.730		640	3200	6400

CASE II

a = 60 g   t = 0.6 sec

0.1	9.6	0.165	26.8	268	1340	2680
0.2	38.6	0.321	33.8	338	1690	3380
0.3	86.8	0.465	42.0	420	2100	4200
0.4	154	0.604	52.2	522	2610	5220
0.5	241	0.731	64.0	640	3200	6400
0.6	347	0.855	78.6	786	3930	7860

* $(p_b)_{0.01}$ denotes the pressure on the front of the package when the initial pressure ahead of the package was 0.01 atmosphere.

Table D.1 shows that for all cases and under all initial pressure conditions, the motion of the piston relative to the air directly in front of it is always subsonic.

For Case I, the first pressure signal will have traveled $x = c_1 t = 1126 \times 0.15 = 169$ ft at the end of the acceleration run. Since the length of the deceleration section should be at least twice the minimum required acceleration length, i.e., $2 \times 72.4 \approx 145$ ft, and must be traversed twice by the shock, it can be concluded that no reflection of pressure waves will reach the piston during its acceleration run. It is also not significant whether the end of the tube is kept closed or open, apart from the necessity of maintaining a vacuum.

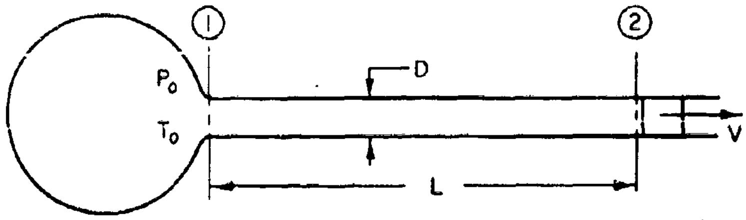
The same considerations are valid for Case II, since the distance traveled by the signal is $x = c_1 t = 1126 \times 0.6 = 675$ ft, the minimum acceleration distance is 347, and $2 \times 347 \approx 700$ ft.

For Case I, the theoretical accelerating force is $F = ma = 100 \times 200 = 20{,}000$ lb. Therefore, the maximum theoretical force due to pressure buildup in front will be 2.5%, 12.5% and 25.1% of the theoretical accelerating force with the initial pressure in front of the package at 0.1, 0.5, and 1 times the atmospheric pressure, respectively. For Case II, the theoretical accelerating force is $F = 400 \times 60 = 24{,}000$ lb. The theoretical force due to pressure buildup in front will be 1.1%, 11.1%, 55.7% and 111.5% for initial pressures of 0.01, 0.1, 0.5 and 1 times atmospheric pressure.

For all cases it appears that evacuation of the tube is necessary to avoid large drag forces.

GAS DYNAMICS -- SINGLE RESERVOIR SYSTEM

In the single reservoir system the tube is considered to be a constant-area, adiabatic duct with friction, fed by an isentropic convergent nozzle[9] (Fig. D-1). Numerical data are based on air as the working fluid, although the analysis, in general, would be applicable to helium or some other gas as well.



TA-5656-17

FIG. D-1 SCHEMATIC DIAGRAM OF SINGLE RESERVOIR PNEUMATIC SYSTEM

We assume perfect gas behavior for the air with the constants,

$$R = 53.3 \frac{\text{ft-lbf}}{\text{lb}^{o}}$$

$$k = \frac{c_p}{c_v} = 1.4$$

$$c_v = 0.17 \frac{\text{Btu}}{\text{lbm}^{o}\text{R}}$$

From the specifications, we obtain the acceleration, a, the test duration, t, the payload mass, W_p, and the package diameter. We assume the tube diameter, D, is equal to the specified package diameter; and the total mass, W_T, is twice the payload mass for Cases I and II. For Case III the tube diameter is the same as that for Case I and the total mass is equal to the 50-lb payload, plus the estimated vehicle mass for the 200-lb payload. From these parameters, we compute the cross sectional area, A,

$$A = \frac{\pi}{4} D^2$$

the controlled test distance, L,

$$L = \frac{1}{2} at^2$$

the velocity at the end of the controlled run, V_2,

$$V_2 = at$$

and the pressure required for acceleration, p_a,

$$p_a = \frac{W_T a}{A}$$

Then the total pressure required, p_2, is given by

$$p_2 = p_a + p_b$$

where p_b is the buildup ahead of the vehicle as obtained above.

We next assume a temperature, T_2, such that the Mach No., M_2 is less or equal to 1.0, the maximum velocity without choking due to friction. M_2 is obtained from the relationship,

$$M_2 = \frac{V_2}{kRT_2} = \frac{V_2}{49.02 \quad T_2}$$

Also T_2 should be larger or equal to $32^{o}F$ to avoid freezing.

The air density, ρ^2, is given by

$$\rho_2 = p_2/RT_2$$

the air flow rate, w, by

$$w = \rho_2 A V_2$$

and the mass velocity, G, by

$$C = w/A = \rho V$$

Assuming constant viscosity, μ, we obtain the Reynolds number, N_R, independent of position along the duct

$$N_R = \frac{\rho D V}{\mu} = G \frac{D}{\mu} \text{ with } \mu_{air} \cong 12 \times 10^{-6} \frac{\text{lbm-sec}}{\text{ft}}$$

Estimating the roughness, ϵ,

$$\epsilon \cong 10^{-5} \text{ ft}$$

we can obtain the relative roughness, ϵ/D.

Then from Shapiro's[9] Figure 6.15, we obtain the friction factor, $4f$, and compute the friction parameter, $4f\ L/D$. From Shapiro's Table B.4, with M_2 as the argument we obtain the dimensionless flow parameters at point 2. Then for point 1, the tube entrance,

$$\left(4f \frac{L_{max}}{D}\right)_1 = \left(4f \frac{L_{max}}{D}\right)_2 + (4f\ L/D)_{1 \text{ to } 2}$$

Entering Table B.4 with $(4f \frac{L_{max}}{D})$, we obtain M_1 (consistent with the previously determined M_2 and the dimensionless flow parameters at point 1).

Then

$$T_1 = \frac{(T/T^*)_1}{(T/T^*)_2} \times T_2, \quad p_1 = \frac{(p/p^*)_1}{(p/p^*)_2} \times p_2, \quad \text{etc.}$$

The final conditions in the storage tank are obtained from Table B.2, for the isentropic flow through the convergent nozzle.

$$p_{of} = (p_o/p)_{M=M_1} \times p_1$$

and

$$T_{of} = (T_o/T)_{M=M_1} \times T_1$$

The check the analysis to this point, we compute the mass flow rate, G', from stagnation conditions and compare with G computed previously. From equation 4.16,[9]

$$G' - W/A = p_o \quad g_c \frac{k}{RT_o} \quad M_1/(1 + \frac{k-1}{2} M_1{}^2)^{\frac{k+1}{2(k-1)}}$$

The initial conditions in the storage tank are obtained by applying the First Law of Thermodynamics. Assuming an adiabatic process and equating the energy change of the air to the work done on the vehicle.

$$(M - m)\, c_v\, (T_{oi} - T_{of}) + mc_v\, (T_{of} - \bar{T}_{1,2}) - m(K.D.)_f$$
$$= \bar{p}_2 AL$$

where M is the initial air mass in the storage tank and m is the air mass in the tube at the end of the run.

Then

$$m = \bar{\rho}_{1,2} AL$$
$$mc_v \Delta T = m \times 0.17\ (T_{of} - \bar{T}_{1,2}) = m\Delta u$$
$$m(K.E.)_f = \frac{AL(\rho_1 V_1)\ \bar{V}_{1,2}}{2g_c \times 778}$$
$$\bar{p}_2 AL = \frac{p_a + p_2}{2} AL \times \frac{1}{778} \quad W_k$$

So that $$(M - m)\, c_v \Delta T_o = W_k + m(K.E.)_i - m\Delta u$$

If we assume various values for ΔT, the temperature drop in the storage tank, we can then determine the initial temperature in the storage reservoir, T_{oi}, the initial pressure required, p_{oi}, assuming an isentropic process,

$$p_{oi} = \left(\frac{T_{oi}}{T_{of}}\right)^{\frac{k}{k-1}} \times p_{of}$$

the total mass of air, M,

$$M = \frac{(M - m)\Delta u}{c_v \Delta T_o} + m$$

the storage volume, V_s,

$$V_s = \frac{M}{\rho_{oi}} = \frac{MRT_{oi}}{p_{oi}}$$

and, if the storage tank is spherical, the diameter of the sphere, D_s,

$$D_s = \sqrt[3]{\frac{6V_s}{\pi}}$$

Numerical data are presented in Table D-2 for $\Delta T_o = 25°F, 50°F, 75°F$.

TABLE D-2

PNEUMATIC DRIVE DATA

	PARAMETER	CASE NUMBER I	CASE NUMBER II	CASE NUMBER III
a	acceleration (g)	200	60	200
t	time (sec)	0.15	0.6	0.15
W_p	payload mass (lbm)	50	200	50
W_T	total package mass (lbm)	100	400	250
D	diameter (in.)	12	25	25
A	area (ft^2)	0.785	3.42	3.42
L	acceleration distance (ft)	72.3	347	72.3
V_2	peak velocity (ft/sec)	964	1,158	964
P_a	acceleration pressure (lbf/ft^2)	25,500	7,020	14,600
P_2	total pressure (lbf/ft^2)	26,000	7,100	15,240
P_2	total pressure ($lbf/in.^2$)	181	49	106
M_2	Mach number	0.885	1.0	0.87
T_2	temperature (°R)	490	555	490
ρ_2	density (lbm/ft^3)	0.994	0.240	0.583
w	mass flow rate (lbm/sec)	752	950	1,925
G	mass velocity ($lbm/sec\text{-}ft^2$)	958	278	562
N_R	Reynolds number	80×10^6	46×10^6	94×10^6
ϵ/D	relative roughness	10^{-5}	0.5×10^{-5}	0.5×10^{-5}
$4f$	friction factor	0.005	0.005	0.005
$4fL/D$	friction parameter	0.36	0.87	0.87
$(T/T^*)_2$	temperature ratio	1.037	1.0	1.037
$(p/p^*)_2$	pressure ratio	1.15	1.0	1.15
$(V/V^*)_2$, $(\rho^*/\rho)_2$	velocity, density ratios	0.901	1.0	0.901
$(4fL_{max}/D)_2$	friction parameter	0.028	0	0.028
$(4fL_{max}/D)_1$	friction parameter	0.388	0.87	0.898
M_1	Mach number	0.63	0.51	0.51
$(T/T^*)_1$	temperature ratio	1.11	1.14	1.14
$(P/P^*)_1$	pressure ratio	1.67	2.10	2.10
$(V/V^*)_1$, $(\rho^*/\rho)_1$	velocity, density ratios	0.66	0.54	0.54
T_1	temperature (°R)	525	633	540
P_1	pressure (lbf/ft^2)	37,700	14,900	27,900
V_1	velocity (ft/sec)	705	625	600
ρ_1	density (lbm/ft^3)	1.36	0.445	0.97
P_{of}	final stagnation pressure (lbf/ft^2)	49,250	17,800	33,300

TABLE D-2 (Concluded)

	PARAMETERS	CASE NUMBER I	II	III
T_{of}	final stagnation temperature (°R)	567	666	562
G'	mass velocity (lbm sec-ft^2)	958	278	562
m	air mass (lbm)	66.8	402	192
$mc_v\Delta T$	internal energy change (Btu)	681	4,920	1,730
$m(K.E.)_f$	kinetic energy (Btu)	927	6,370	2,340
W_k	work (Btu)	1.88×10^3	10.75×10^3	4.75×10^3
$\Delta T = 25°F$				
$(M-m)\Delta u$	internal energy change (Btu)	2.126×10^3	12.20×10^3	5.36×10^3
T_{oi}	initial temperature (°R)	592	691	593
T_{oi}	initial temperature (°F)	132	231	133
P_{oi}	initial pressure (lbf/ft^2)	57,000	20,300	40,200
P_{oi}	initial pressure (lbf/in.^{2})	395	140	280
M	initial mass (lbm)	567	3,270	1,450
V_s	storage volume (ft^3)	314	5,930	1,140
D_s	sphere diameter (ft)	8.4	22.5	13.0
$\Delta T = 50°F$				
T_{oi}	initial temperature (°R)	617	716	612
T_{oi}	initial temperature (°F)	157	256	152
P_{oi}	initial pressure (lbf/ft^2)	60,800	22,900	41,100
P_{oi}	initial pressure (lbf/in.^{2})	422	159	285
M	initial mass (lbm)	417	1,837	823
V_s	storage volume (ft^3)	226	3,060	655
D_s	sphere diameter (ft)	7.6	18.0	10.8
$\Delta T = 75°F$				
T_{oi}	initial temperature (°R)	642	741	637
T_{oi}	initial temperature (°F)	182	281	177
P_{oi}	initial pressure (lbf/ft^2)	75,500	25,800	51,400
P_{oi}	initial pressure (lbf/in.^{2})	524	179	357
M	initial mass (lbm)	233	1,355	612
V_s	storage volume (ft^3)	106	2,080	404
D_s	sphere diameter (ft)	5.9	15.9	9.2

VACUUM AND COMPRESSOR REQUIREMENTS FOR PNEUMATIC DRIVE

Evacuation of tube

The analysis is based on empirical data provided by a vacuum pump manufacturer (F. J. Stokes Corporation).

We consider Case II as the most demanding and assume a final pressure of 0.01 atmosphere and a maximum evacuation time, t, of 50 minutes. From the Stokes data,

the pump-down factor, $p_f \cong 5$

and the needed pumping speed, $S = \dfrac{V \times p_f}{t}$

where V = tube volume = $AL - 3.42 \times 347 \times 2.5 = 2970\ ft^3$ (assuming a total tube length 2.5 times the test section).

Substituting in the equation for S,

$$S_{min} = \frac{2970 \times 5}{50} = 297\ cfm$$

A suitable pump would then be the Stokes Model 612(535 cfm) or equivalent with a 20 hp motor. The actual pumping time is then

$$t = \frac{2970 \times 5}{535} = 28\ minutes$$

Compressor Power

For the purpose of this analysis we assume air as the working fluid, behaving as a perfect gas with the constants

$$R = 53.3\ \frac{ft\text{-}\#}{lb\,^oR}$$

$$k = \frac{c_p}{c_v} = 1.4$$

$$c_v = 0.17\ \frac{Btu}{lb\,^oR}$$

$$c_p = 0.24\ \frac{Btu}{lb\,^oR}$$

Since the air will be stored at elevated temperatures, we assume an adiabatic compression process with a reasonable isentropic efficiency, rather than a more efficient isothermal process. Using the same nomenclature as before, the compressor work required at initial start-up is

$$(W_k)_1 = \frac{1}{\eta_s} M c_p (T_{oi} - T_a) = \frac{1}{\eta_s} M c_p T_a \left[\left(\frac{p_{oi}}{p_a}\right)^{\frac{k-1}{k}} - 1\right]$$

where the isentropic efficiency, $\eta_s = \frac{\Delta h \text{ isentropic}}{\Delta h \text{ actual}}$,

and the compressor work to repeat a run is

$$(W_k)_2 = \frac{1}{\eta_s} m c_p T_a \left[\left(\frac{p_{oi}}{p_a}\right)^{\frac{k-1}{k}} - 1\right]$$

Assuming the work is to be done in time, t, the average power requirement is

$$(hp)_{1,2} = \frac{(W_k)_{1,2} \text{ (ft-lbf)}}{t(\text{sec})} \times \frac{1}{550} \frac{\text{sec-hp}}{\text{ft-lbf}}$$

The tabulated numerical data are based on the assumptions,

$\eta_s = 0.80$

$T_a = 530\ ^oR$ (ambient temperature)

$p_a = 14.7\ \text{lbf/in}^2$ (ambient pressure)

$t = 50$ min.

We obtain m, M, and p_{oi} from the data in Table D-2 and compute horse power, as shown in Table D-3.

TABLE D-3

COMPRESSOR POWER

	Parameter		Case Number I	II	III
Trial No. 1 (ΔT_0 = 25°F)	m	(lbm)	66.8	402	192
	M	(lbm)	567	3270	1450
	p_{oi}	(lbf/in^2)	395	140	280
	$(hp)_1$		66.3	222	143.5
	$(hp)_2$		7.8	27.3	19
Trial No. 2 (ΔT_0 = 50°F)	M	(lbm)	417	1837	823
	p_{oi}	(lbf/in^2)	422	159	285
	$(hp)_1$		50.3	134.5	82.4
	$(hp)_2$		8.1	29.4	19.2
Trial No. 3 (ΔT_0 = 75°F)	M	(lbm)	233	1355	612
	p_{oi}	(lbf/in^2)	524	179	357
	$(hp)_1$		31.1	106	68.3
	$(hp)_2$		8.9	31.5	21.5

We conclude that the power requirements for evacuation and compression would not be prohibitive.

NOMENCLATURE

A = area

a = acceleration

c = sonic velocity

c_p = specific heat at constant pressure

c_v = specific heat at constant volume

D = diameter

ϵ = roughness

f = friction factor

g_c = proportionality constant in Newton's second law

G = mass velocity, W/A

h = enthalpy

hp = horsepower

K.E. = specific kinetic energy

k = ratio of specific heats, c_p/c_v

m = mass

M = Mach No., mass

N_R = Reynolds nu...ber

p = pressure

R = gas constant

S = pump speed

s = entropy

T = temperature

t = time

u = internal energy

V = velocity, volume

w = mass flow rate

W_k = work

W_p = payload mass

W_T = total vehicle mass

ρ = density

μ = viscosity

η = efficiency

Subscripts:

a = acceleration

b = build-up

f = final

i = initial

o = stagnation state

s = isentropic

1,2 = state points

Special symbols:

* superscript = state at which Mach No. is unity

Barred quantities = average value

lbf = pounds force

lbm = pounds mass

APPENDIX E

PROPOSED HYBRID ROCKET PROPULSION

This appendix presents two proposed hybrid rocket designs by United Technology Center (UTC), a division of United Aircraft Company.

The first uses an aluminum-ammonium perchlorate fuel in a polybutadiene binder and hydrogen peroxide as oxidizer. The second uses nitrogen tetroxide as oxidizer. It is somewhat more efficient but produces corrosive gases.

The design details are direct copies of the proposals presented by UTC.

SRI PROPULSION UNIT

DESIGN SUMMARY

REQUIREMENTS

Vehicle & Payload Wt. Less Propulsion	G's	Burn Time, sec
350 lbs	60	0.6
200 lbs	200	0.15

PERFORMANCE

Isp Delivered (1000/14.7)	255 sec
Isp Delivered (465/14.7)	242 sec
C* Delivered (1000 psia)	5680 ft/sec
F @ 60 G's	28,000 lb
F @ 200 G's	64,000 lb
Pc @ 60 G's	465 psia
Pc @ 200 G's	1000 psia (max)
Fuel	Al/AP/PBD (25/25/50)
Oxidizer	H_2O_2
Mixture Ratio	0.85

NOZZLE DATA

Throat Diameter	7.51 in
Exit Diameter	16.0 in
Area Ratio	4.6:1
Type	15° Conical

CASE AND TANK DATA

Material	HP9-4-45 Steel
Usable Strength	250,000 psi
Density	0.286 lb/in^3

DESIGN CRITERIA

Isp Efficiency	94% Max
C* Efficiency	97% Max
Min. Safety Factor	2.0

SRI PROPULSION UNIT

WEIGHT SUMMARY

Oxidizer Usable	31.7 lbs
Unusable	0.6
Fuel Usable	37.3
Unusable	0.6
Combustion Chamber	23.9
Helium Gas and Sphere	3.2
Oxidizer Tank	2.3
Helium Flow Control Unit	1.0
Oxidizer Flow Control & Injector Assy	5.0
Grain Liner, Supports & Insulation	8.5
Nozzle Assembly	8.0
Total Weight	122.1
Usable Propellant	69.0
Mass Fraction	0.565

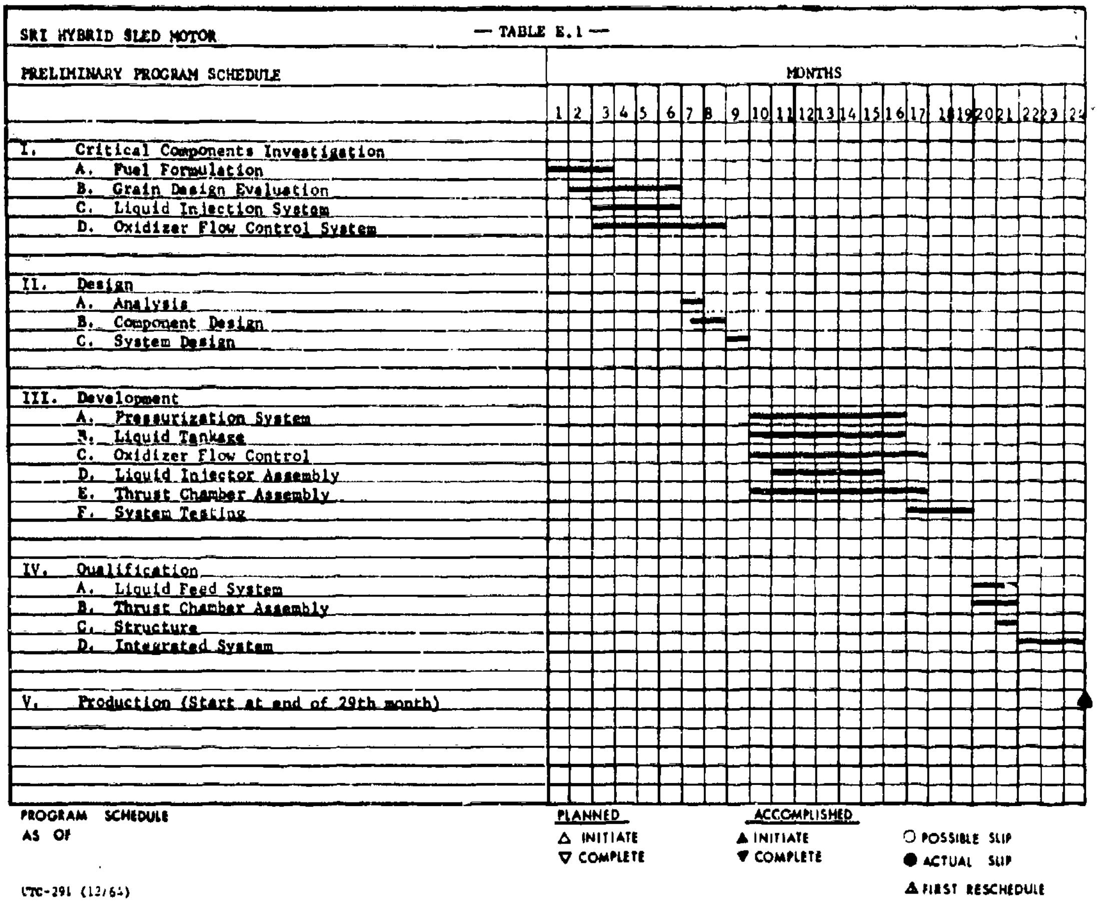
DESIGN SUMMARY

CASE	I	II
Payload Weight, lbs	200	50
Structure Weight, lbs	150	150
Propulsion System Weight, lbs	119.9	87.9
Acceleration	60g	200g
Nominal Thrust (70°F), lbs	28,250	57,600
Allowable Thrust Variation, lbs	±235	±1440
Allowable Fuel Grain Initial Temperature, °F	+55 to +85	+24 to +116
Burn Time, sec	0.6	0.15
Chamber Pressure, psia	252	1,000
Expansion Ratio	6.3	6.3
Nozzle Throat Diameter, in	6.79	6.79
Nozzle Exit Diameter, in	17.10	17.10
Nozzle Length, in	16.3	16.3
Propellant	N_2O_4 - 60% AP/10% Al/10% TFTA/20% PBD	
Mixture Ratio	0.7	0.7
Theoretical Isp (1000-14.7), sec	270	270
Delivered Isp, sec	242	251
Pressurant	Helium	Helium
Total Propellant, lbs	71.5	35.09
Mass Fraction	0.584	0.391

SUMMARY OF WEIGHTS

	I	II
Usable Propellant	70.10	34.40
Unusable Propellant	1.40	.69
Nozzle	9.25	9.25
Chamber	10.00	12.20
Chamber Insulation	4.03	4.03
Phenolic Sleeves	0.27	0.31
Flanges, Nuts, Bolts	7.30	7.30
Injector and Lines	3.30	5.40
Oxidizer Tank	8.10	8.10
Pressurant Tank	1.89	1.89
Valves	4.30	4.30
Total	119.94	87.87
Usable Propellant	70.10	34.40
Mass Fraction	0.584	0.391

— TABLE E.1 —


SRI HYBRID SLED MOTOR
PRELIMINARY PROGRAM SCHEDULE
MONTHS
1 2 3 4 5 6 7 8 9 10 11 12 13 14 15 16 17 18 19 20 21 22 23 24
I. Critical Components Investigation
A. Fuel Formulation
B. Grain Design Evaluation
C. Liquid Injection System
D. Oxidizer Flow Control System
II. Design
A. Analysis
B. Component Design
C. System Design
III. Development
A. Pressurization System
B. Liquid Tankage
C. Oxidizer Flow Control
D. Liquid Injector Assembly
E. Thrust Chamber Assembly
F. System Testing
IV. Qualification
A. Liquid Feed System
B. Thrust Chamber Assembly
C. Structure
D. Integrated System
V. Production (Start at end of 29th month)
PROGRAM SCHEDULE
AS OF
PLANNED
△ INITIATE
▽ COMPLETE
ACCOMPLISHED
▲ INITIATE
▼ COMPLETE
○ POSSIBLE SLIP
● ACTUAL SLIP
△ FIRST RESCHEDULE
UTC-291 (12/64)

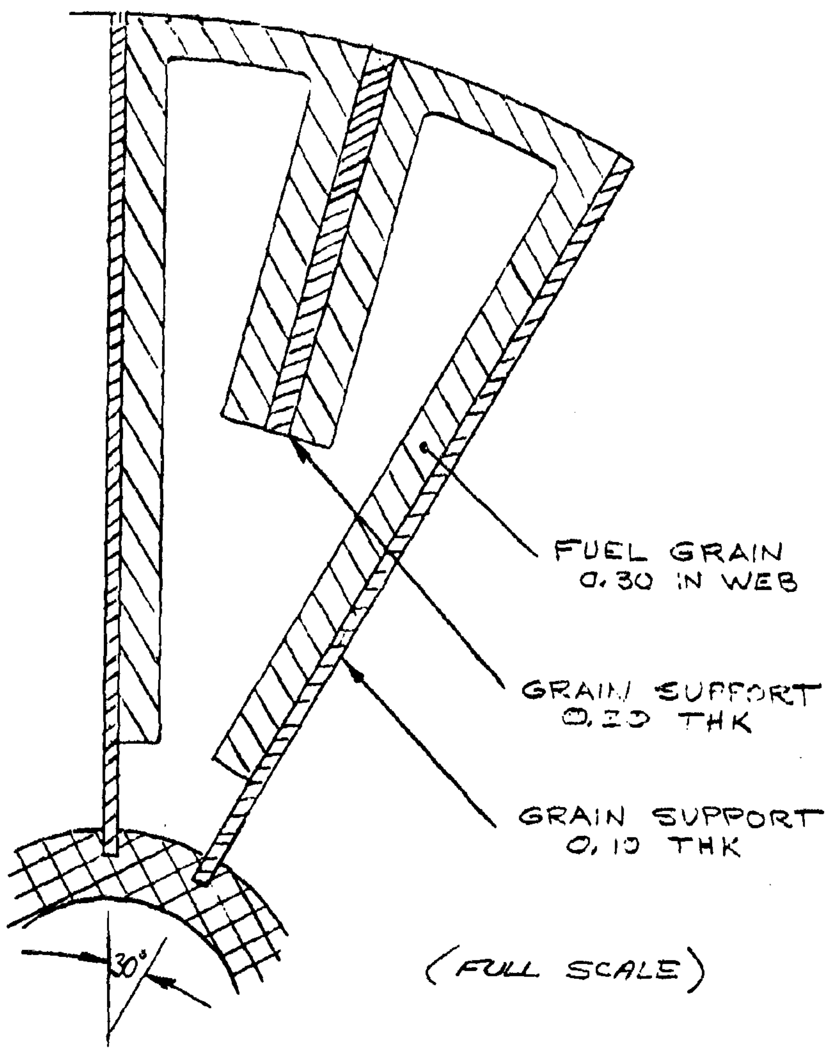


FIG. E-1 SRI PROPULSION UNIT GRAIN CONFIGURATION (24-spoke wheel)

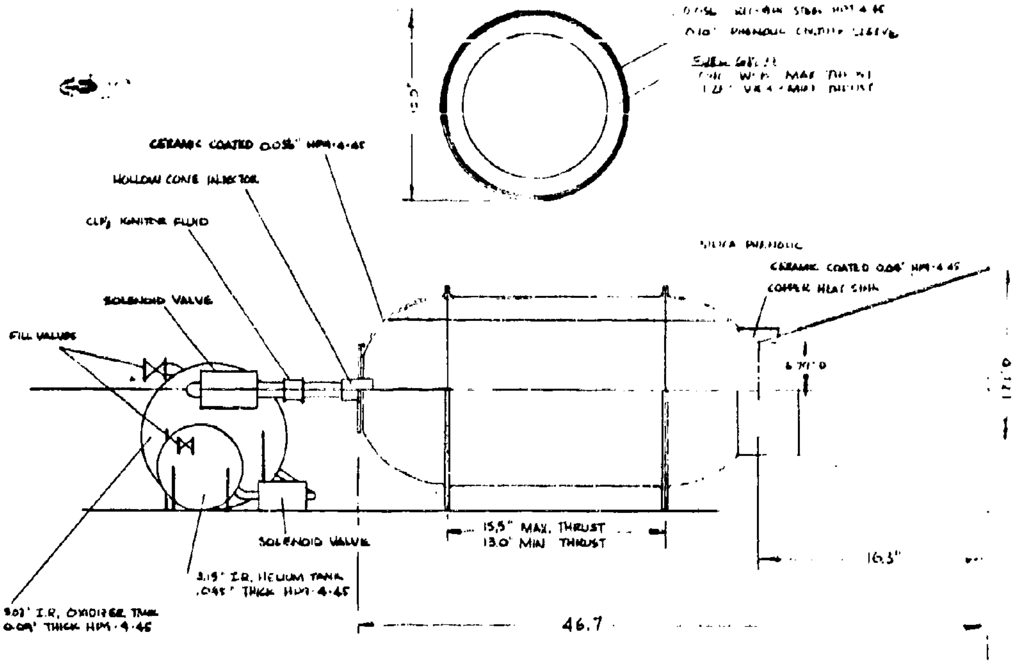


FIG. E-2 SRI PROPULSION UNIT — DESIGN I

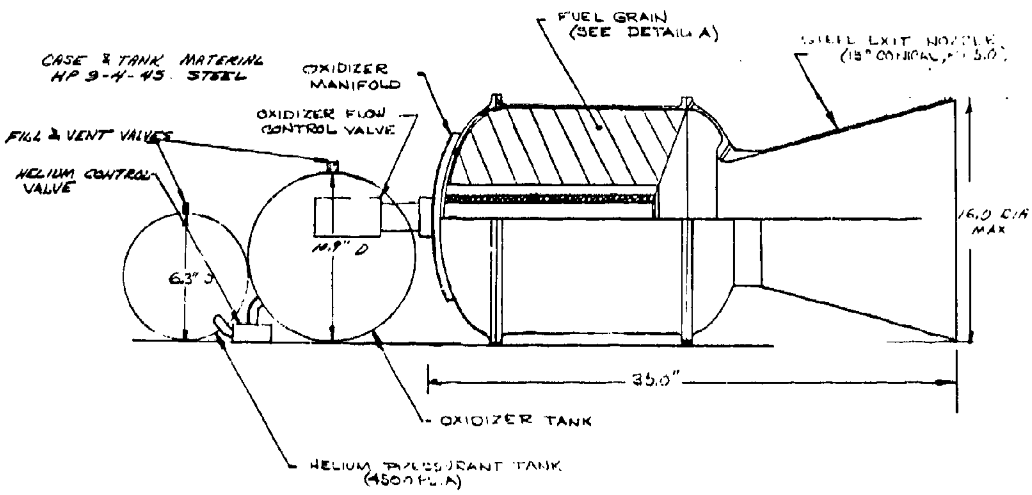


FIG. E-3 SRI PROPULSION UNIT — DESIGN II

APPENDIX F

UNSOLVED RESEARCH PROJECTS -- WORK STATEMENTS AND COST ESTIMATES

To simplify the soliciting of follow-up research work in the areas where additional research is proposed, we have prepared work statements and cost estimates for three areas of investigation:

- Construction methods and material selection for the constant frequency translator
- Design and analysis of high-speed air cushion bearings
- Feasibility of a high energy variable frequency power supply.

TRANSLATOR DESIGN AND PERFORMANCE

A thorough analytical and experimental study is required to determine the best material, geometry and configuration for the translator on the linear induction motor. This problem cannot be divorced from the power system design, for several of the required properties are directly dependent on the choice of the system.

For a first choice, beryllium appears to offer some outstanding advantages, for it combines light weight, high strength, and high heat capacity. The last is particularly important in a constant frequency system for the heat developed in the translator becomes of major concern. Since the heat is generated in a very short period of time, very little of it can be transferred out of the translator, either by convection or conduction, and a high heat capacity is essential. As the translator heats up, both its electrical and its mechanical properties change. It is important to know the effect of these changes. Methods of cooling the translators after completion of the run also need study.

If beryllium is selected, it will have to be encapsulated or clad because of its high toxicity. This may also have the advantage of increasing the surface resistivity of the translator. The selection and effects of cladding materials must be studied. Closely related are the structural geometry and methods of manufacture: laminated construction, sintering, or honeycomb methods may be preferable to the use of solid sheet. The possibility of alloying should be investigated.

Statement of Work

1) Investigate materials and methods of construction that can produce a light, strong translator with high heat capacity and with a surface resistivity in the order of 10^{-2} ohms/m .

2) Evaluate material properties, such as flexural and fatigue strength, density and modulus of elasticity.

3) Evaluate thermal properties, such as coefficient of thermal expansion, heat capacity and thermal diffusivity in the temperature range $-100^{\circ}F$ to $+600^{\circ}F$. If made of several different materials, evaluate thermal stresses.

4) Expose material samples to an ac magnetic field and evaluate the electrical characteristics of the material.

5) Propose methods for rapid cooling of the material without causing corrosion or undue thermal stresses.

It is estimated that this study can be performed in ten months at a cost of $90,000.

DESIGN OF AIR CUSHION BEARINGS

One of the most critical items in the design of a high-g linear accelerator is the supports and bearings on which the vehicle runs. Experience with rocket sleds indicates that solid friction bearings have intolerably high crosswise accelerations. Air film lubrication appears to provide satisfactory answers if the experience with low-speed air bearings can be extrapolated to velocities very near sonic. Research and development are needed to justify the use of air bearings for this service and determine optimum design parameters.

The following performance requirements must be met:

- The velocity range for these bearings is from zero to 1200 ft/sec.
- Under low-speed operation each bearing may support as much as 300 lb. At higher speed, upward or downward, dynamic loads of as high as 1000 lb may be experienced. The bearing must not bottom under any of these conditions.

- Bearings may be required to withstand a side thrust of about 500 lb.
- Overall weight of the air support system, including lubricating air for a total period of 5 sec, should not exceed 120 lb.
- Bearing inserts should be readily replaceable and should wear preferentially over the support rail.

Statement of Work

1) Develop a complete air bearing support system for the high-g linear accelerator to meet the performance requirements mentioned above.

2) Optimize the design on a weight basis.

3) Determine the load-deflection and stability characteristics for these bearings in transverse linear motion and rotation about an axis at right angles to the rail. Both must be evaluated as functions of air flow rate.

4) Study the dynamics of the complete system to determine the probable cross-axis accelerations that may be transmitted to the vehicle. Specify what additional shock isolation may be required.

5) Perform laboratory tests on prototypes of these bearings at speeds of up to 100 ft/sec.

6) Consider the use of pressurized gases other than air.

7) Design in detail a complete gas bearing system for the high-g accelerator.

It is estimated that this work can be performed in 10 months at a cost of about $80,000.

VARIABLE FREQUENCY POWER SUPPLY

Considerable simplification in vehicle design and power savings are possible for the high-g linear accelerator if the proposed constant frequency linear induction motor can be replaced by a variable frequency system. Since variable frequency ac is not easily generated, particularly at the power levels required, a feasibility study is needed that evaluates various possible methods of providing variable frequency ac at the 100 MW level for periods of up to 0.5 sec and at frequencies varying from about 20 to 200 Hz. The primary power may be either dc or constant frequency ac.

Among the possible systems to be studied are:

- Vacuum tube power amplifiers. With modern high power tubes available this approach appears to present no fundamental innovations.
- Gaseous and solid state inversion systems. The gaseous and solid state conductive inverters typified by ignitrons, thyratrons and silicon controlled rectifiers have not been developed into the power magnitudes required.

Statement of Work

1) Investigate the technical and economic feasibility of various methods to convert dc or constant frequency ac to variable frequency ac in the 100 MW power range for periods of up to 0.5 sec and at output frequencies from 20 to 200 Hz.

2) Compare methods on the basis of: probability of successful construction, controllability (how easy is it to change frequency and how rapidly can the changes be made?), and cost, both capital and operating.

3) Prepare detail flow charts and/or make breadboard experiments to indicate feasibility and show what additional work must be done to make the method successful.

It is estimated that this research can be done in 7 months at a cost of about $65,000.

APPENDIX G

DESIGN REQUIREMENTS

The following lists are copies of design requirements and desiderata furnished by the contracting agency to SRI:

TECHNICAL REQUIREMENTS OF THE FACILITY

The desired operational requirements of the linear accelerator shall meet the following specifications:

(1) Both linear acceleration for 0.15 sec and deceleration for 0.15 sec of a 50-lb payload at 200 ± 5 g's in sequence or individually.

(2) Both linear acceleration for 0.6 sec and deceleration for 0.6 sec of a 200-lb payload at 60 ± 0.5 g's.

(3) Minimum controlled linear acceleration and deceleration of either a 50- or 200-lb payload will be 2 g's.

(4) The payload shall be cylindrical and shall not exceed 20-in. in diameter x 4-ft in length for the 200-lb payload and 12-in. in diameter x 2-ft in length for the 50-lb payload.

(5) The acceleration and deceleration profile shall be programable in 1 g steps.

(6) The acceleration and deceleration between 2 and 60 g's may be maintained by presetting of power requirements.

(7) The system shall provide for test package data links and the propulsion system shall not produce detrimental test package environment such as spurious vibrations, heating and interference with data recovery system.

ADDITIONAL DESIDERATA*

LATERAL TRANSLATIONS BOTH VERTICAL AND HORIZONTAL

(1) Desire less than 2 g peak to peak over a frequency range of 5 to 1200 Hz of sine motion and less than 4 g peak to peak from 1200 Hz to 3000 Hz.

(2) Desire less than 4 g peak to peak over frequency range of 5 to 3000 Hz of white noise vibration. These values are at the 60 to 200 g level.

(3) Desire less than 1 g PP of sine vibration perpendicular to acceleration direction of 5 to 1200 Hz and less than 2 g PP from 1200 Hz to 3000 Hz.

(4) Desire less than 2 g PP white noise from 5 to 3000 Hz for acceleration levels 2 to 60 g.

ANGULAR TRANSLATIONS ABOUT THE ACCELERATION DIRECTION APPLIES TO BOTH VERTICAL AND HORIZONTAL

(1) Alignment position at time of application of g force must be held to ±0.1 mil,** 1 σ of the average g vector during the run.

(2) During a run, the maximum peak to peak angle excursion cannot exceed 1 mil for any g setting from 2 to 200 g. During acceleration phase, the average excursion cannot be more than ±0.1 mil from the average of the entire acceleration phase.

(3) The transition from acceleration to deceleration phase will allow a reorientation of the average deceleration vector not to exceed 2 mil (desire less).

(4) The allowable excursion from the deceleration vector will be identical to the acceleration requirements.

(5) Roll deviation cannot exceed a band of 2 mil during a complete run.

(6) There are no terminal alignment requirements.

---

* These are not design requirements.
** A mil is 1/1000 radian or about 3.5 minutes of a degree.

ACCELERATION REQUIREMENTS

(1) 2 to 60 g ± 0.5 g linear. Vibration acceleration in this direction cannot exceed 1 g PP sine over range of 5 to 1200 Hz and 2 g, PP sine from 1200 to 3000 Hz. 2 g PP white noise is maximum value from 5 to 3000 Hz.

(2) From 60 to 200 g ± 5 g linear. Vibration acceleration cannot exceed 5 g PP sine over range of 5 to 3000 Hz.

(3) 5 g PP white noise over range of 5 to 3000 Hz.

Combinations of the above motions cannot result in a coning effect below 100 Hz which exceeds 0.1 mil maximum.

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4. DESCRIPTIVE NOTES (Type of report and inclusive dates)

Final Technical Report

5. AUTHOR(S) (Last name, first name, initial)

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[redacted]

11. SUPPLEMENTARY NOTES	12. SPONSORING MILITARY ACTIVITY Office of Chief of Engineers Department of the Army

13. ABSTRACT Inertial guidance systems for tactical missiles must perform during the boost period when, for less than one second, the entire missile is accelerated at very high rates. It is desirable to propel the guidance system on a test vehicle which can be accelerated in a horizontal straight line for a few tenths of a second at up to 200 g, whose motion can be carefully controlled, and which can be brought to a stop after the test. Therefore, the feasibility of various propulsion methods was studied and a comparison made of vehicles driven by linear electric motors, compressed gas, and rockets. This study indicates that the highest probability of success lies with a combination pneumatic and constant frequency linear induction motor drive.

The installation would consist of a pneumatic piston which accelerates the test vehicle to about 200 ft/sec, after which the induction motor drive takes over. The electrical system comprises a three-phase alternator rated at about 280 megawatts for short-time operation, a linear stator along which the test vehicle moves, and a power control circuit which selectively feeds power from the alternator to the stator. Beryllium fins on the vehicle make up the translator of the linear motor. A Doppler-radar control system provides feedback control. Deceleration is by dc coils in the stator. It is estimated that the entire unit can be designed and built in about four to five years at a cost of about $15 million.

Alternate systems which show promise are based on variable frequency ac and on hybrid fuel rockets. Either or both of these alternate systems may be less costly than the constant frequency systems, but both require more research and development effort and are less assured of success.

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