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A.R. Susoeff, R.S. Hawke, J.J. Morrison,
G. Dimonte, and B.A. Remington

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DESIGN OF AN ELECTROMAGNETIC ACCELERATOR FOR TURBULENT HYDRODYNAMIC MIX STUDIES*.

A.R.Susoeff, R.S.Hawke, J.J.Morrison, G.Dimonte and B.A.Remington
Lawrence Livermore National Laboratory
Livermore CA

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ABSTRACT

An electromagnetic accelerator in the form of a linear electric motor (LEM) has been designed to achieve controlled acceleration profiles of a carriage containing hydrodynamically unstable fluids for the investigation of the development of turbulent mix. The Rayleigh-Taylor instability is investigated by accelerating two dissimilar density fluids using the LEM to achieve a wide variety of acceleration and deceleration profiles. The acceleration profiles are achieved by independent control of rail and augmentation currents. A variety of acceleration-time profiles are possible including: 1) constant, 2) impulsive and 3) shaped. The LEM and support structure are a robust design in order to withstand high loads with low deflections and to mitigate operational vibration. Vibration of the carriage during acceleration could create artifacts in the data which would interfere with the intended study of the Rayleigh-Taylor instability. The design allows clear access for diagnostic techniques such as laser induced fluorescence (creating 2D images with high spatial resolution), radiography, shadowgraphs and particle imaging velocimetry (a direct measure of fluid flow field). Electromagnetic modeling codes were used to optimize the rail and augmentation coil positions within the support structure framework. Results of contemporary studies for non-arcing sliding contact of solid armatures are used for the design of the driving armature and the dynamic electromagnetic braking system. A 0.6MJ electrolytic capacitor bank is used for energy storage to drive the LEM. This report will discuss a LEM design which will accelerate masses of up to 3kg to a maximum of about $3000g_0$, where g_0 is acceleration due to gravity.

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INTRODUCTION

Lawrence Livermore National Laboratory is conducting a project to investigate the development of turbulent mix influenced by the Rayleigh-Taylor instability in hydrodynamically unstable systems [1]. An electromagnetic solid armature accelerator in the form of a linear electric motor (LEM) has been designed to achieve controlled acceleration profiles of a carriage containing various dissimilar density fluids. The information gathered from tests conducted on the LEM, for example, will be used in support of studies of fluid instabilities for inertial confinement fusion (ICF) [2,3]. Quantitative understanding of hydrodynamically driven turbulence is one of the key unsolved problems in ICF. Previous investigators have studied mix effects using various types of accelerators, however, reliable control of the acceleration rate was difficult to attain [4,5,6,7,8]. We initially considered using hydraulics to accelerate the carriage module but were concerned over controlling the acceleration rate and profile and the high mass of the total accelerated package. Hydraulic acceleration was abandoned in favor of electric drive and control.

With electric drive the acceleration profiles are achieved by independent control of rail and augmentation currents and a variety of acceleration and deceleration profiles are possible. The acceleration and payload mass are limited by the available electrical energy and floor loading capacity directly underneath the LEM. The overall design continues to evolve with time. Continuing refinements are expected.

Our initial working point consists of two phases. The first phase includes developing a concept test model and designing a mechanical structure which will contain the force resulting from accelerating a mass of up to 3kg to a maximum of about $3000g_0$, where g_0 is acceleration due to gravity. Phase 1 will serve to familiarize personnel with the operating characteristics of the apparatus, evaluate diagnostic results and make necessary modifications. During Phase 2 the LEM will be taken to the design limit imposed by the power supply and mechanical structure. A test platform incorporating the appropriate reaction mass is under construction for this purpose.

Evaluation of overturning forces consistent with local seismic risk criteria was done for the capacitor bank and LEM. This analysis defines the method of anchoring the major components to a concrete floor. Appropriate mitigation measures are included in the apparatus and facility design.

DESIGN PARAMETERS FOR THE APPARATUS

Basic Requirements

The fluid mass to be accelerated is about 0.7kg. The carriage module containing the fluid is driven vertically downward. Acceleration distances up to 1 meter long are required. The main structural element of the carriage module is the cell, with a working fluid cavity of 7cm by 7cm by 14cm long. The size of the cavity was determined from modeling the expected instabilities for various candidate test fluids [9].

Mass estimates, based on acceptable cell wall deflection, have been made for low accelerations, i.e. $100g_0$ to $750g_0$, as well as $3000g_0$ for the carriage module. We estimated the maximum total payload mass to be 3.0kg. The mechanical force acting on the armature (F_a) and reacting on the LEM and support structure is found from the product of mass and acceleration:

$$F_a = ma \quad (1)$$

When the aggregate mass for the test fluids, carriage module, containment materials and armatures is 3kg, a force of 88kN ($\approx 20,000$ lbs) is required to achieve an acceleration rate of $3000g_0$. This force is used to calculate stress and deflection on the support structure and reaction forces transmitted to the floor. An impulse (≈ 5.0 ms) launch acceleration up to $3000g_0$ over a distance of 10 cm will induce the highest stress in the support structure. The LEM design evolved from the establishment of upper acceleration and mass limits and application of these limits to stress and deflection on the apparatus.

Capacitor Bank

The amount of energy transfer possible from the existing capacitor bank effects mass and acceleration. A four module bank made up of electrolytic capacitors is used for the energy storage to drive the armature and the augmentation coils. Each module consists of 960, 1500 μ fd, 450 volt electrolytic capacitors. Maximum energy storage and current per module is 150kJ and 100kA. A five conductor, four circuit, flat plate transmission line transfers energy from the electrolytic capacitor bank to the LEM. The modules are configured to drive the rails and augmentation coils separately.

Evaluating a case, for example, to accelerate 3kg to $1000g_0$ requires the simple rail and driving armature circuit to conduct currents in the neighborhood of 240kA. By adding augmentation coils to the system the required rail current drops to less than 50kA. Hence a much lower mass armature can now be used.

Linear electric motor (LEM)

The basic apparatus is shown schematically as Fig.1. The electrolytic tough pitch copper alloy rails are spaced 10cm (4.0inch) apart and they are parallel to better than $125\mu\text{m}$ (0.005inch). Separate pairs of rails are used for acceleration and braking. A tapered ramp is incorporated on both ends of the rails to allow smooth mechanical transition of the armature or brake contact face. The armature is to make non-arcing sliding contact with the rails. Augmentation coils are installed outside the rail positions. All components are fastened to a stiff, robust aluminum frame designed to restrain mechanical deflections to less than $125\mu\text{m}$ (0.005inch) at maximum design load condition with a ninety degree field of view throughout the travel length of the carriage module for diagnostics access. A modular design was developed where stiffness could be easily increased as needed. Vibration of the carriage during acceleration could create artifacts which can interfere with the intended study of the Rayleigh-Taylor instability. Our robust design mitigates operational vibration and allows for additional damping measures to be taken if necessary.

Augmentation Coils

Electromagnetic modeling codes were used to optimize the locations of the rail and augmentation coils with respect to each other and the support structure framework [10]. Augmentation coils have been designed for the impulse (short time duration $\approx 5\text{ms}$ and short drive length $\approx 10\text{cm}$) and continuous drive (long time duration $\approx 40\text{ms}$ and ≈ 1.0 meter long drive length) modes of operation. The augmentation coils are also designed to withstand high loads with low deflections.

Fig.1. A schematic layout of the LEM configured for an impulse test. Appropriate tie bars, gussets and bracing are not shown for clarity. The view (a) looking down from the starting point and (b) from the front.

Carriage module

Two module types are shown schematically in Fig.2. A module is made up of: 1) a cell

(a)

Note: Compliant material and seal between window, reaction plate and cell not shown.

(b)

Fig.2. Carriage cell types. A side view (a) as seen by the diagnostics of the "soft guide shoe" design for accelerations up to 750g₀. Oblique view (b) of the 3000g₀ "hard guide shoe" design looking from the laser access end; but the dynamic brake, reaction flange and laser window are not shown for clarity. The aluminum "hard" shoes have a lower coefficient of friction with the guide track but are more complex to fabricate. We will begin by using the plastic compliant surface type shoe design for the 750g₀ case.

which contains the test fluid, 2) windows to allow diagnostic access to the action of the fluids and 3) a place to mount the acceleration armature, dynamic brake and the guide module through the LEM is shown as Fig.3.

1) *Cell*: The cell is the main structural element for attaching the other components making up the module. The mechanical forces created during acceleration and braking are reacted against the cell. A slot on one end of the cell allows transmission of a laser light sheet into the cell cavity, by way of an optically clear window, to illuminate the contents.

Fig.3. A schematic cross-section of the LEM. The view to the left of the line-of-sight shows the acceleration components. The deceleration rail is shown on the right, omitting similar parts for clarity. Geometry and part location were guided by modeling the components in the magnetic field resulting from current flow in the rails and augmentation coils. The rails of the LEM are supported by a stiff insulation material. By nearly balancing the forces acting on the rail a low modulus, low tensile strength, non-magnetic material such as acetal can be used as a rail support. The augmentation coil mounting technique allows flexibility of location throughout the length of the I-beam support structure.

2) *Windows*: Optically clear windows form two sides of the cell and access through the laser slot. Candidate materials for the windows include: polycarbonate, acrylic, fused quartz and sapphire. Individual selection is based on strength, stiffness, the optical quality, laser light attenuation and compatibility with the test fluids. Acceleration loads effect bending and shear properties and hence total mass.

3) *Seals*: Two fluid sealing methods are used (see Fig.2). Leakage for low accelerations ($100g_0$ to $750g_0$) is controlled with an o-ring gasket . In order to reduce mass we intend to use a two-part cure silicone rubber to effect a seal at $3000g_0$.

4) *Guide shoes*: The guide shoes (see Fig.2) are a form of cantilever leaf spring. They are designed to preload the cell to the expected maximum side loading while maintaining minimum drag. We estimated the potential side loads to be less than 10 percent of the acceleration loading. Guide shoes are fastened to the cell. They restrain lateral motion and balloting together with the guide tracks (see Fig. 2 and Fig.3).

Experimental diagnostics

Clear optical access for various diagnostic techniques is a key feature of our design. Radiography and shadowgraphs provide a record of turbulent mix distribution. Two state-of-the-art diagnostic techniques are used to characterize the fluid instabilities. Laser induced fluorescence (LIF) [11] records 2D images with high spatial resolution and particle imaging velocimetry (PIV) [12] is used to make direct measurement of the fluid flow field from which the turbulent energy can be deduced.

DESIGN METHOD

LEM Support structure

The main support frame of the LEM is designed to carry the full design load limit at maximum current. The reaction surface for the LEM is a pair of aluminum I-beams ≈ 3 meters (10 feet) long oriented vertically (refer to Fig.1). The end of each beam is fastened and reacted through a thick plate. These plates form the datum to fix the separation distance of the I-beams. The lower I-beam end plate acts as the base platform. A welded aluminum frame is attached to the back of each I-beam. Each frame is triangular in shape to allow a 90 degree field of view of the carriage module as it

traverses the length of the LEM. An aluminum skin is attached to the frame to increase stiffness.

Shear motions causing misalignment of adjoining components are negated with steel dowel pins installed at assembly. Bolts and screws are assembled with disc springs to insure tensile loading of each fastener and reduce the probability of loosening during repeated electric pulsing of the LEM

Each I-beam has a series of symmetric hole patterns to fix and locate components. This method of assembly was chosen so components could be easily removed and replaced when expended and so parts could be shimmed in relation to each other to obtain the required basic dimensions.

Carriage module guides

The guide track structure is fabricated from non conducting plastic material. The individual guide tracks are mounted on the rail support and shimmed into position. The guide shoe and guide track material is matched for low coefficient of friction and wear properties [13]. Initially we are using acetal and nylon.

Armature/dynamic brake

Testing conditions for the experiments require a nonarcing sliding contact armature. Results of contemporary studies for non-arcing sliding contact of solid armatures are used for the design of the driving armature and the dynamic electric braking system. We compared two methods of establishing the proper initial mechanical contact force: 1) using current density [14] and 2) current only [15]. Each armature (see Fig.4) is designed to fit individual test parameters such as: mass, acceleration, current, rail separation distance and action. We have designed the armature to remain compliant against the rail while in operation within a predetermined stress limit. We used half the tensile yield strength for the armature alloy as maximum allowable stress. Each armature contact leg is a cantilever leaf spring type design in which initial mechanical preload is obtained by deflecting the spring (armature leg) a specified distance.

The carriage module is brought to a stop when electric energy is applied to activate the dynamic brake. This occurs when the braking armature is on the rail surfaces and

Fig.4. General design of an armature (a) and brake (b). Each component is optimized for expected current, action and minimum mass.

commutation is verified. Once a current path is established magnetic loading of the armature occurs maintaining the contact force required for non-arcing sliding contact.

The armature and dynamic brake (see Fig.4) are electrically isolated from the cell. The acceleration armature can also serve as the brake up to the tensile and shear limits of the fasteners. The tensile stress during high g_0 deceleration with our current design results in fastener failure so the brake is mounted on opposite end of the cell. While a mass penalty is incurred from this design forces generated during either acceleration or braking are applied in compression against the cell requiring fasteners to only locate the parts.

The leg of the armature is designed for a deflection of $\approx 250\mu\text{m}$ (0.010inch). This figure was arrived at from the ability to make the rails parallel to within $\approx 125\mu\text{m}$ (0.005inch). This allowance gives us confidence that the minimum force required to maintain non-arcng contact between the sliding surfaces can be maintained while staying within the stress limits of the armature material.

The minimum required leg cross-section is determined from calculation of the action properties. Armature width is arbitrarily chosen to be about 10 percent less than the width of the rails to insure full face contact side to side. The leg has a rectangular cross-section.

The length and thickness of each armature contact leg is determined by algebraic manipulation of formulas for moment of inertia, section modulus and maximum allowable deflection and stress for a static beam [16,17]. The armature leg is assumed to be a beam with one end fixed and one end simply supported for the magnetic case and one end fixed and one end free to find the cross-section from the given deflection and stress values for the mechanical loading case. Physical and mechanical properties for the armature and rail materials needed to effect the design include: density, modulus of elasticity and tensile yield strength for the armature and compressive yield strength for the rail. Since the rails are made of ductile material it is assumed the design compressive yield strength is equal to the tensile yield strength.

The armature design discussed in this report will drive a 2kg mass at $750g_0$. Rail current is 46kA. The magnetic force per unit length acting on the armature leg (F_m^*) is found by dividing the mechanical force found using (1) by the amount of rail separation (W) which is 10cm (4.0 inch):

$$F_m^* = [F_a/W] \quad (2)$$

For this case $F_a = 14.7\text{kN}$ (≈ 3310 lbs) and $F_m^* = 14.7\text{kN/m}$ (≈ 840 lb/in). These values are used to determine the stress and deflection in the armature from mechanical and magnetic loading.

The solid phase current action of the armature material is used to calculate the cross-section area (A_a) using:

$$A_a \geq [\int I^2 dt / 0.33 A I_s]^{0.5} \quad (3)$$

An action integral ($\int I^2 dt$) for 46kA current was calculated to be 9.72 A²s. The action to melt ($A I_s$) used for this calculation is 1E17 A²sec m⁻⁴ [18,19,20] and is multiplied by 0.33 to give a safety factor for loss of material strength due to resistive heating during current transfer. At 46kA an armature cross-sectional area (A_a) of at least 17.7mm² is needed. Current density is ≈ 2600 A/mm².

Applying our current density at 46kA calculation to Barber's J-P curve [14] gives a contact pressure of ≈ 30.0 N/mm². Dividing this value by the required area results in a mechanical contact force of 531N (≈ 120 lbs).

The second method to apply force for non-arcing sliding contact is the "g/A rule " used to 30kA, by Marshall in homopolar generators [15]. The mechanical armature contact forces to conduct the required current for the 46kA case is ≈ 453 N (≈ 101 lbs)

Since we must have sliding contact with no arcing it is conservative to use the higher contact force value for mechanical installation. This value is used to determine the length of the armature leg. Further refinement of contact force requirements is expected as we operate this apparatus and gain experience.

The length and thickness of the armature leg is determined by algebraic manipulation of static case beam formulas knowing the maximum allowable stress (σ) on the armature leg, the required bending (δ) on the end of the leg and a given width (b). A safety factor is included by taking half the tensile yield strength for a particular armature alloy as the maximum allowable stress.

The armature cross-section is a rectangle. The moment of inertia (I) and section modulus (Z) are found using [16]:

$$I = [bh^3/12] \quad (4)$$

$$Z = [bh^2/6] \quad (5)$$

We fixed the armature width (b) to be 9mm (0.36 inch). Substituting into (4) and (5) we have equivalents for armature thickness (h):

$$I = 0.03h^3$$

$$Z = 0.06h^2$$

The length of the armature leg can be found having established width and expressing thickness (h) in terms of (4) and (5) for a deflection (δ) of 0.010 inch at the end of the leg at a stress of 251 MPa (36500 lb/in²), one-half the tensile yield strength of 502 MPa (73000lb/in²) for aluminum alloy 7075-T6. The modulus of elasticity (E) of aluminum is 6.9 GPa (1.0e7 lb/in²).

Deflection (δ) of a beam with one end fixed and one end free is found using [16]:

$$\delta = [W_a L^3 / 3EI] \quad (6)$$

Rearranging (6) to solve for length (L)

$$L = [\delta 3EI / W_a]^{0.33} \quad (7)$$

The maximum stress (σ) on this beam is found by using [10]:

$$\sigma = [W_a L / Z] \quad (8)$$

Rearranging (8) to solve for length (L):

$$L = [\sigma Z / W_a] \quad (9)$$

Solving the equations simultaneously gives us an armature length of 24.7 mm (0.974 inch), a thickness (h) of 5.9 mm (0.231 inch) and an area of 146 mm². This area is more than enough to satisfy the action to melt. Knowing the cross-section area the deflection (δ) and stress (σ) on the armature leg as a result of magnetic loading is calculated.

Under this condition the armature leg is considered a beam fixed at one end and simply supported at the other end. The magnetic load calculated using (2) is uniform across the beam. Deflection and stress are found using [16]:

$$\delta_{\max} = [F_m^* L^3 / 185EI] \quad (10)$$

$$\sigma_{\max} = [(F_m^* L / 8Z)] \quad (11)$$

Resulting in values of $\delta_{\max} = 130\mu\text{m}$ (0.0011 inch) and $\sigma_{\max} = 215 \text{ MPa}$ (31122 lb/in²).

The armature legs are connected together by a cross beam, which also serves as a mounting bracket to attach the armature to the carriage cell module. The beam is designed to deflect less than 25 μm at the center under full load.

The sum of the mechanical and magnetic deflections establish the angular requirements (see Fig.5) to keep the armature face in close contact with the rail during installation and operation. The minimum angle in this case is 0.95 degree.

The armature drag force is calculated from the sum of the mechanical and magnetic loading forces. We use a static coefficient of friction of 0.2. The drag force in this case is about 3 percent of the armature driving force. The significance of this amount of drag caused by the armature contact load against the rail and its effect on the expected carriage module acceleration is yet to be determined.

FUTURE WORK

Design of the phase 2 test platform is underway. We have developed a microwave time-position diagnostic to monitor carriage travel for future adaptation to the LEM [21]. Installation for both is pending future funding.

Fig.5. Armature contact face angle. Current flow is intentionally directed through the very end of the leg.

SUMMARY

An electromagnetic accelerator in the form of a linear electric motor (LEM) has been designed to achieve controlled acceleration profiles of a carriage containing hydrodynamically unstable fluids for the investigation of the development of turbulent mix. The information gathered from tests conducted on the LEM will be used in support of studies of fluid instabilities for inertial confinement fusion (ICF). Quantitative understanding of hydrodynamically driven turbulence is one of the key unsolved problems in ICF. The wide variety of mass and acceleration rates will allow us to add to the fluid instabilities data base and check models for designing non-arcing sliding contacts.

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