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DESIGN ASPECTS OF THE NEW SERD CATAPULT

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INTRODUCTION

IMAGINE, IF YOU WILL, EVEL KNIEVEL in a new variation of his vehicle jump stunt. He first accelerates for about 300 feet, then ballistically launches from a steep ramp on the ground with sufficient velocity to easily clear the top of the Washington Monument. To make this event even more spectacular, he uses a METRO bus, instead of his customary motorcycle. Then 45 seconds later, after an assumed soft landing, he repeats the act as an encore.

All he needs for this unusual feat is a highly controllable energy source that can deliver 60 million foot pounds in three seconds — every 45 seconds. *Fantastic?* Not at all — because in fact similar launch requirements are satisfied hundreds of times a day at sea as U.S. NAVY carriers routinely catapult F14-A aircraft into flight.

Dependency on high energy catapults for launching became necessary with the advent of carrier borne jet aircraft following World War II. A progressive evolution of catapult technology has followed the continuing changes of aircraft and ship requirements, resulting in challenging system design problems with unique solutions.

CATAPULT BACKGROUND

The present mechanisms for accelerating the 74,500 pound F14-A aircraft to a take-off velocity of 225 feet per second is a C-13 steam catapult. The C-13's motive force is produced by steam pressure acting on *two* shuttle-connected pistons which move in long underdeck cylinders.

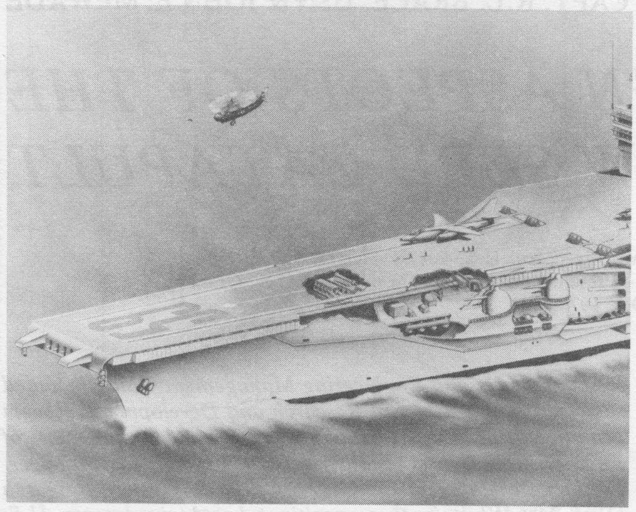


Figure 1. Steam Catapult Installation.

On launch command, the shuttle force breaks a tensioning restraint and accelerates the aircraft over a fixed distance to a predetermined take-off velocity. The shuttle force may be altered from launch to launch in order to accommodate changes in aircraft weight, stress limitations, or required end speed, thus necessitating appropriate setting of pressure for each. Acceleration, which is restricted by aircraft and pilot limitations, must be sufficient to attain the minimum take-off velocity plus a 10% to 15% safety margin.

The installation arrangement of the C-13 catapult on a carrier is shown in Figure 1. Typically, a carrier of this size will have *two* catapults in the bow and *two* in the waist area of the angled deck.

A slot in the flight deck allows a "shuttle" power run of 308 feet and braking distance of 5 feet. The entire propulsion and braking mechanism is contained in a 48-inch deep x 52-inch wide (1.2m x 1.3m) trough.

Launch acceleration begins with the opening of a fast acting control valve to allow rapid admission of steam into the cylinders. The cylinders are slotted to allow passage of a structure interconnecting the "shuttle" and pistons. The slots are pressure sealed by a spring loaded flexible strip, except where opened for a short distance forward of the pistons by a cam on the structure, thereby maintaining pressure integrity behind the pistons.

Fixed to the "shuttle" is a tapered piston, called a "spear", which, at the end of the power stroke, enters into a single cylinder filled with water thereby producing a high rate of deceleration and bringing the "spear" to a complete halt.

The propulsion cylinders are assembled from 12-foot Stainless Steel Sections, bolted together and rigidly supported on adjustable structural mounts. Steam, at 550psi, originates in the ship's steam power plant and is supplied to the catapult receiver tanks through heavy 8-inch diameter piping. Below, mounted on the Gallery Deck, is the retraction engine, and on the Main Deck the steam receivers.

Ancillary system requirements for the C-13 (Not shown in Figure 1) include equipments for Gallery Deck air conditioning, condensate drainage, system preheating, and added fresh water conversion capacity.

Major interface considerations of a C-13 catapult installation are:

WEIGHT (including provision for deck reinforcements)	= 2,800 tons (24.90 x 10^6 N)
SPACE OCCUPANCY (excluding remote support items)	= 80,000 Ft^3 (2265 m^3)
FRESH WATER REQUIREMENTS (1 operational day)	= 80 tons (711.7 x 10^3 N)
INSTALLED COST per catapult (1976 $'s)	= $10,300,000

The steam catapult, which was introduced to the Fleet in the early 50's, has a proven record of dependability. It has also adapted well to performance upgrading, as aircraft loads and velocities increased. Notwithstanding, certain inherent characteristics, most of them attributable to the use of steam, have engendered criticism. These include: a high topside weight; a burdensome (particularly in non-CVN ships) demand for fresh water; installation and maintenance difficulties in keeping the propulsion cylinders precisely aligned; increased corrosion susceptibility; exposed high temperature areas with related fire hazards; and other factors; all diminishing or hampering operational effectiveness to some degree.

The SERD Catapult

Development of a new generation catapult, called SERD for STORED ENERGY ROTARY DRIVE, has been authorized by the Naval Air Systems Command as a future replacement for the C-13 steam catapult. The SERD catapult will use the principle of stored flywheel launch energy, a concept previously developed for lower energy shorebase catapults. Responsibility for design and development has been assigned to the Ship Installations Department of the Naval Air Engineering Center (NAEC), Lakehurst, N.J., where, to date, analytical studies in support of a firm system concept and preliminary component designs have been completed.

The purpose of this paper is to describe the SERD catapult design concept, the expected future impact on the ship design and air operations, and the analytical methods and results used to predict end performance.

The System Concept

FUNCTIONAL DESCRIPTION —

Figure 2 depicts a SERD catapult installation on the Gallery and Flight Decks of a carrier. The principal subsystems are: 1) Power train, 2) Launch ("Tape") Drive System, 3) Waterbrake, and 4) Shuttle Retract System. Energy for the launch is initially stored in a flywheel, then, on command, transmitted as an accelerating torque through a gear reducer, clutch, and reel, and as a tow force a fabric "tape" connected to a track constrained "shuttle." The waterbrake is used to stop

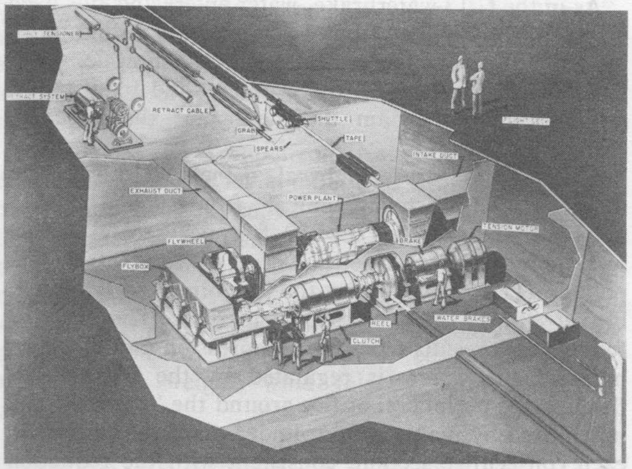


Figure 2. SERD Catapult Installation.

the "shuttle" at the end of the stroke, and the retract system to return it to battery position.

Figure 3 schematically illustrates the functional details of the system. The power train consists of a 10,000HP gas turbine powerplant coupled to an integral flywheel and 2-stage gear reducer assembly, called a "flybox". The output shaft of the "flybox" is connected to the input of a friction clutch rated at 500,000 foot pounds (677,500 Nm) torque and capable of dissipating 35,000,000 foot pounds (25,812,500 J) of frictional heat loss energy with each launch. Rotating at an initial turbine speed of 3600rpm, which decelerates at launch to 2,844rpm, the power train will supply a maximum of 106,000,000 foot pounds of energy (78,175,000 J) to the drive system. Of this amount, approximately 60% is available for aircraft acceleration. In addition, to maintain the stored energy, the turbine must supply 1,210,000 ft. lbs. of energy per second (892,375 J) in overcoming flywheel aerodynamic and bearing friction losses. The maximum launch rate is *one* per 45 seconds.

The launch dynamic characteristics are determined by the amount of clutch slip which is regulated by a

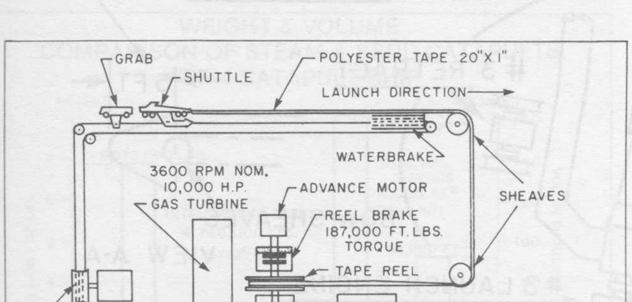


Figure 3. SERD Catapult Functional Schematic.

"servo control" to impart a predetermined torque program to a "tape" winding reel. The "tape" from the reel is directed under and around *two* reversing sheaves forward of the launch position, then onto the "shuttle," pulling it forward in an underdeck track. The reel assembly includes a disc brake, rated at 187,000 foot-pounds (253,385 Nm) torque to prevent "tape" backlash which would otherwise result from force oscillations in the "tape" during "shuttle" braking. An advance motor, also part of the reel assembly, permits "shuttle" positioning and initial hook-up tensioning of the aircraft in readiness for launch.

The "tape" towing medium is a woven polyester webbing with a breaking strength of 1,250,000 pounds (5,560,000 N). The "tape" is approximately 500 feet (152.4 meters) in length, 20-inches (.500 meters) wide, and 1-inch (0.0254 meters) thick. The fiber surface is coated with a thermoplastic polyester elastomer to reduce possible damage from abrasion, moisture absorption, and contact with aircraft fuel. Weight of the "tape" is approximately 8.3 pounds per foot (12.35 kg/m).

On launch command, a "Servoed Actuater" causes the clutch to begin frictional engagement, increasing initial (pre-breakaway) "tape" tension to as high as 60,000 pounds (266,880 N). At a predetermined force level the deck secured tensioning link separates, resulting in rapid acceleration buildup. Thereafter, clutch torque and "tape" tension continue to rise, then level off, following the analog signal launch force program. In the F14-A launch, for example, the "tape" force at the reel rises to 154,000 pounds (685,000 N) in 3.2 seconds, holds constant for the remaining 0.8 seconds of the launch stroke, then goes to zero in 0.1 second.

Prior to breakaway, the "tape-reel" radius is approximately 1 foot (0.305 m). After motion begins the radius increases rapidly with rotation, and then less rapidly as the circumference of each lap becomes greater, until at the completion of the fixed 305 feet (93 m) launch stroke, the reel radius is 3.35 feet (1.02 m). Since reel radius is a factor in determining torque, the changing radius is anticipated in the clutch control program so as to obtain proper tow force throughout launch.

When the "shuttle" reaches the end of stroke the reel brake is activated and the clutch disengaged, thus causing the "shuttle" to decelerate and the aircraft to separate from the tow connection. Immediately after, "shuttle" waterbraking begins.

The waterbrake consists of *two* water-filled cylinders, approximately 45 feet (13.72 m) in length, into which *two* side-by-side piston "spears", mounted on and beneath the "shuttle," enter and decelerate on contact with the water (Figure 4). The cylinders are machined from HP9-4-20 high strength (195 KsiY) alloy steel billets in 82-inch (2.08 m) sections. The inside diameters range from 7.41 inches (18.82 cm) at the entrance aperture, down to 6.85 inches (17.4 cm) in the final section. Unlike the C-13 waterbrake, in which a single end supported "spear" engages a short water-filled cylinder, the SERD cylinders require slotting (as in the C-13 steam propulsion cylinders) so as to allow passage of a cross structure connecting the "spears." Within the central slotted section, a flexible slot sealing strip, which prevents water ahead of the "spears" from escaping, is displaced by a cam located behind the "spears" on the cross structure.

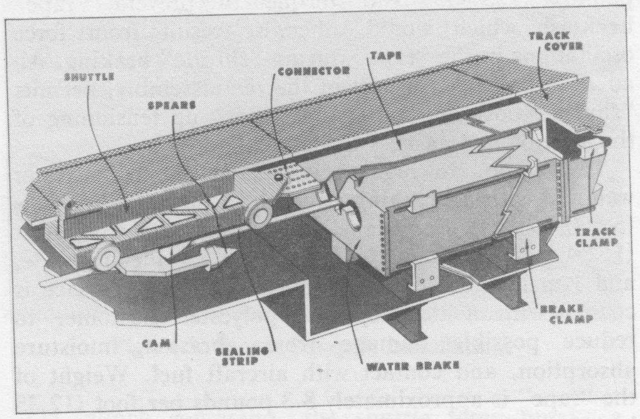


Figure 4. SERD Catapult Waterbrake.

As in the C-13 waterbrake, water enters and fills each cylinder through a frontal circular manifold with inwardly directed nozzles. During a launch cycle, water flow is maintained to fill the cylinders and to sustain a retaining pressure dam until entry of the "spears" occurs.

The "spears" are cylindrical, 6 inches (16.5 cm) in diameter, with steep conical fronts. The design maximum entry speed is 259 feet per second (78.94 m/s), producing a kinetic energy of 15,000,000 foot pounds (11,062,500 J) for the combined reel, clutch, brake, "tape," and "shuttle" inertias. The braking force, which results from the compression and displacement of the water, is regulated by the escape flow through the peripheral orifice around the "spears," and maximized by making the cylinder diameter smaller in each consecutive section, consistent with the reduction in velocity during braking. Depending on the entry velocity of the "spears," braking will require up to 40 feet (12.19 m) of travel and occur in 0.4 seconds, or less. Maximum water pressure will be 6,500 psi (18.65 N/m²), with a "tape" tension at the shuttle of 500,000 pounds (2,224,000 N) in the highest energy case.

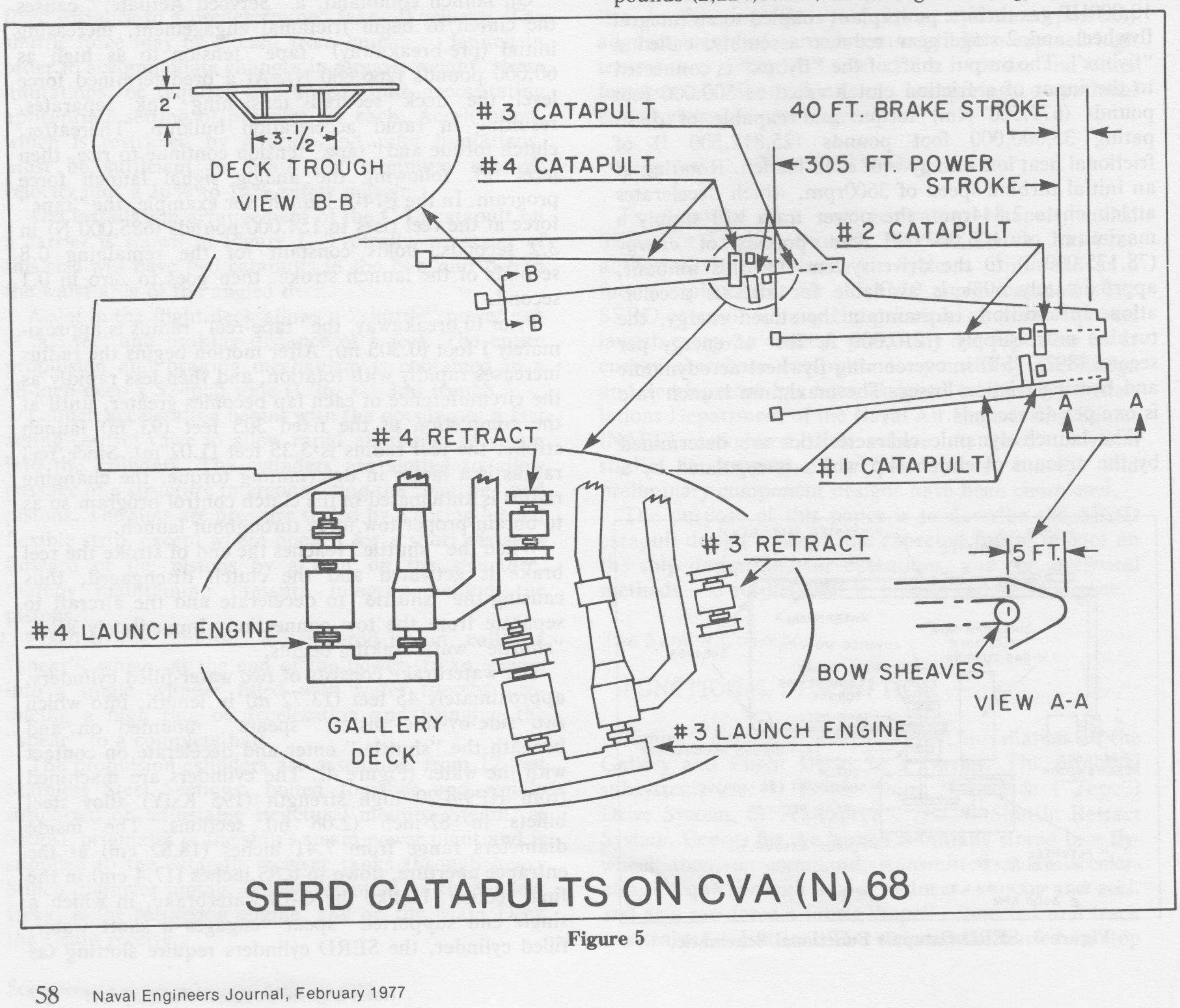


Figure 5

When the "shuttle" stops, a cable drive "grab" assembly, guided by the same track as the "shuttle" and powered by a hydraulic drive motor, moves forward, latches on the "shuttle," and retracts it to the initial pre-launch position.

Auxiliary SERD components (Not shown in Figures 2 & 3) are necessary for cooling and lubrication of the drive components. Also, an extensive sensor and interlock system will guard against abnormal machinery operation. The clutch torque control servo will be responsive to the signal difference between a torque sensor and an electronic function generator, the latter programmable by signal inputs corresponding to aircraft type, weight, wind velocity, ship speed, et cetera.

SHIPBOARD INTERFACE FACTORS —

Figure 5 illustrates how the SERD catapult, as presently conceived, would fit on the deck plan of a modern carrier such as *USS Nimitz* (CVN-68). The plan shows *two* "stub booms" for the bow catapults and *one* for the angled deck, which support sheaves for reversing "tape" travel. Additional sheaves may be needed for "tape" guidance depending upon the final selection of machinery locations within a particular class of ship.

Optimal locations of machinery spaces warrant detail analysis of the ship space and structural constraints VERSUS catapult performance factors. For example, the "tape" reel to bow sheave distance should be a minimum in order to minimize launch inertia. Studies of ship design characteristics indicate that the distance may vary from 75 to 150 feet within different class ships.

Gallery Deck machinery will have a height clearance of 9½ feet (2.85 m). Area per catapult will be 2,880 ft² (267.55 m²). Efficient utilization of space dictates that catapult launch machinery pairs be consolidated so as to obtain common work and access areas, supervisory stations, et cetera.

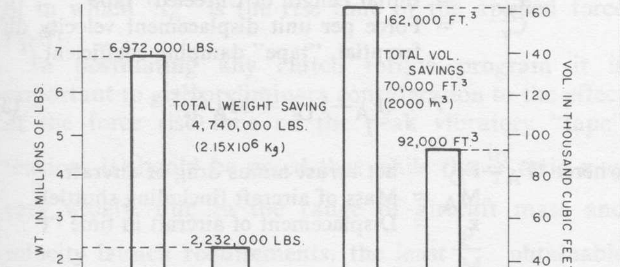


Figure 6.

The underdeck trough, in which the "tape" and "shuttle" are guided, will have a depth of 2 feet (0.61 m), half that of the C-13 trough. The change will materially reduce and simplify the underbridging structure of the deck. Also, since the trough catapult components will require far less alignment precision, a lighter and simpler trough structure will suffice.

As illustrated by the graphical comparisons in Figure 6, changing the trough structure, and replacing the C-13 steam drive cylinders, steam tanks, piping, et cetera, with SERD catapult components, will result in weight and volume reductions of 4,740,000 pounds (21,083,520 N) and 70,000 ft³ (1982 m²). These reductions may be traded for an improvement in the ship's righting moment, increased armor, increased habitation and storage space, and many other options.

Eliminating the high pressure steam generating requirement will increase the ship's propulsion energy capacity, or, alternatively, permit design of lower capacity systems for future ships. Reduction of air conditioning and elimination of the water conversion space and power similiarly create new options for ship design improvements.

A number of operational benefits will result from a changeover to the SERD catapult:

- Elimination of aircraft steam ingestion.
- Elimination of flight deck obscurations.
- Elimination of temperature stabilization delays.
- Elimination of fuel spill trough fires.
- Elimination of overboard discharge of lubrication oil.
- Elimination of (steam) increased corrosion activity.

The gas turbine engine, selected over other prime mover alternatives because of having independence from the ship's propulsion power plant, presents the main ship-interface design problems. Principal among these will be the turbine air intake ducts requiring low pressure drop, flow distortion, and water ingestion, complicated by the relatively large sectional area of 30 to 50 ft² (2.78 to 4.65 m²) (depending upon final engine specifications) which must penetrate the ship shell between the 03 and 02 levels.

A similar problem will exist for each turbine exhaust, which must be isolated from the air intake, to avoid exhaust reingestation, and to be water and air cooled to restrict outlet temperatures to a safe limit.

Also, within the ducts and surrounding the turbine, extensive noise reduction measures will be needed. If not attenuated, the airborne noise power in the vicinity would be 20 to 30dB. above the deafness avoidance level.

Fuel for the gas turbine is regular JP-5 which can be supplied by piping from nearby Flight Deck fueling lines. About 1,000 gallons (3,750 L) will operate the the turbine for a launch sequence of 16 aircraft — approximately 5% of the same aircrafts' fuel consumption during launch and subsequent flight. This presents only a *minor* addition to the ship's fuel storage requirement.

System Kinematics

SERD catapult kinematics encompass launching and braking functions. The launching kinematics will be discussed first.

LAUNCHING —

As shown in Figure 3, the power train couples to the launch drive components at the clutch where independent control of torque is exercised. This allows the launch drive variables on either side of the clutch to be considered separately, since both are independent, yet mutually responsive to the same torque.

The clutch output response kinematics are influenced by the attendant parameters of component inertia, friction, and compliance. Some are fixed, such as the aircraft and "shuttle" inertias. Others, as in the "tape" and reel inertia, aerodynamic friction, et cetera, are variable.

On the clutch input side the essential parameters are flywheel speed and inertia, representing stored kinetic energy.

In developing the optimum system parameters, it is appropriate first to postulate a trial torque program for the launch, then to test the program ability to obtain the desired acceleration, velocity, and stroke requirements. With that accomplished, the effect on the input side may be considered in terms of flywheel design speed and inertia requirements.

1) Clutch Output Analysis

Analysis of dynamic systems with inertia and compliance generally involves the use of differential equations to express force, motion, and time relationships, frequently solvable by formal methods of Calculus. Solutions of differential equations for the SERD catapult, however, because of parameter variability, are practical only by iterative integration methods.

The iterative integration method uses repetitive computation and integration of incremental changes in variables during time intervals made sufficiently small to allow approximation of varying rates and coefficients as constants. Throughout the process, coefficients are updated from Tables according to the state of the related variable. During each time interval, ordered element responses are sequentially computed and transferred from element to element in the dynamical system. Because of the multiplicity of steps to be taken, the iterative solution method is adaptable only to computerized analytical procedures.

Two types of clutch output models may be used in the analysis. The first, illustrated in Figure 7(a), is simple to use and well suited to conceptual sizing and parametric "trade-off" studies. The model simplistically assumes a mass-spring-mass representation of the launch system. The unreeled "tape" is a spring with parallel damper (representing viscous friction internal to the "tape"). At opposite ends are masses representing the aircraft (including "shuttle") and the reel with

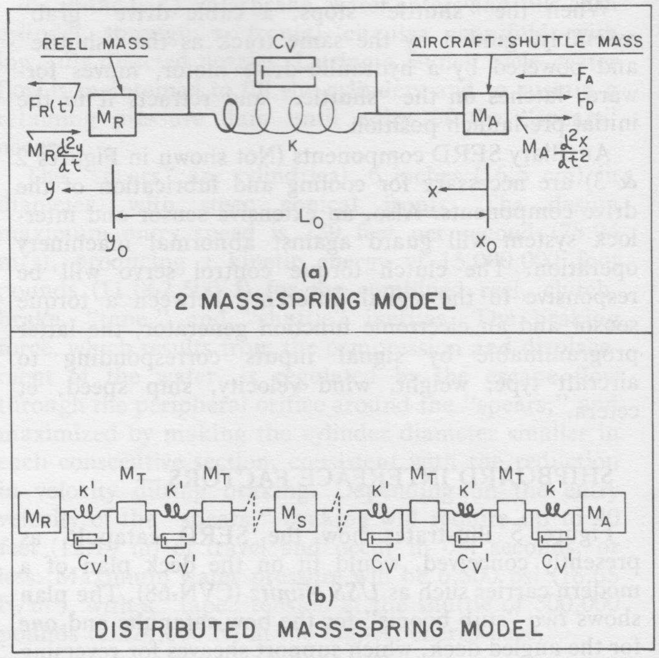


Figure 7.

contained "tape" (including clutch and brake component masses). The end masses also include the unreeled "tape" mass, proportionally divided.

The basic differential equations describing the 2 Mass-Spring Model parameters of Figure 7(a) are:

$$F_R(t) = M_R \frac{d^2y}{dt^2} + T \qquad (1)$$

where: $F_R(t)$ = Time variant (signified by (t)) force component of clutch torque.
M_R = Lineal Mass of reel and contained "tape."
T = Tape Tension.
y = Lineal displacement of M_R in time t.

$$T = K(y—x) + C_v \left[\frac{dy}{dt} — \frac{dx}{dt}\right] \qquad (2)$$

where: K = Spring Constant [Tape C.S. Area x Modulus ÷ Length (L)].
L = Length of Unreeled "tape" = L_o—y.
L_o = Initial Length of Unreeled "tape".
C_v = Force per unit displacement velocity differential, "tape" damping coefficient.

$$T + (F_A—F_D) = M_A \frac{d^2x}{dt^2} \qquad (3)$$

where: $(F_A—F_D)$ = net thrust minus drag of aircraft.
M_A = Mass of aircraft (including shuttle).
x = Displacement of aircraft in time "t".

Referring again to the model of Figure 7(a), when force $F_R(t)$ is applied in Δt, mass M_R moves distance "y," in accord with Equation (1). The spring tension T, caused by the displacement "y," is expressed in Equation (2). The effect of the spring tension, applied to the aircraft, is to cause the displacement "x," as shown by Equation (3). By approximating rates as constants:

$$\frac{d^2(x,y)}{dt^2} = \frac{2\Delta(x,y)}{\Delta t^2} \quad \text{(Acceleration)}$$

$$\frac{d(x,y)}{dt} = \frac{\Delta(x,y)}{\Delta t} \quad \text{(Velocity)}$$

and $F_R(t)$ as constant, during each of "j" iterations of Δt, so that:

$$F_R(+) = \sum_{n_{\Delta t}=1}^{n_{\Delta t}=j} \Delta F_R(t) + F_{R_o}$$

Solutions for incremental displacements Δx and Δy may be obtained from which "x" and "y" may be determined as:

$$x,y = \sum_{n_{\Delta t}=1}^{n_{\Delta t}=j} \Delta(x,y) + x_o, y_o$$

By repetitively solving, summing, and reapplying the results to each of the equations, in the indicated order, complete histories of all variables can be developed.

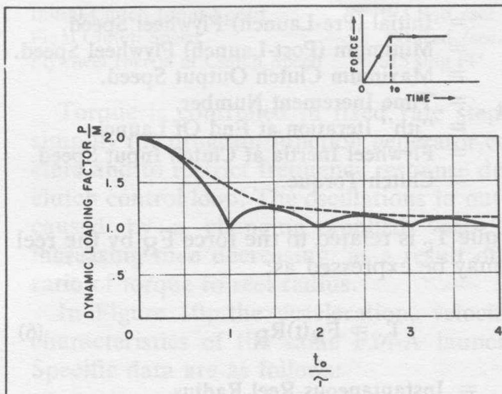
Another relationship useful to the 2 Mass-Spring model analysis is that of the natural vibration period, expressed as:

$$\tau = \frac{2\pi}{\sqrt{K_X\left(\frac{1}{M_A} + \frac{1}{M_R}\right)}} \tag{4}$$

As described in Reference [1] it can be shown that a vibration of period τ will occur whenever a tension reversal occurs, causing an oscillatory exchange of stored "tape" (spring) energy and mass kinetic energy. The force amplitude of the vibrations, relative to the applied force, can be described as a peak loading factor $\frac{P}{M}$, uniquely related to the ratio $\frac{t_o}{\tau}$ (as shown in Figure 8) in which "t_o" is the rise time of the applied force $F_R(t)$.

In postulating any clutch torque program it is important to give preliminary consideration to the effect of the force rise time on the peak vibratory "tape" tension. It should be noted that while the $\frac{t_o}{\tau}$ ratio may vary widely, due to the range of aircraft mass and velocity launch requirements, the least $\frac{P}{M}$ obtainable will provide the most efficient utilization of launch energy from the standpoint of aircraft stress capacity, which is the limiting factor in the system.

The second solution technique for analysis of clutch output behavior uses the model shown in Figure 7(b)



RELATIONSHIP OF $\frac{P}{M}$ TO $\frac{t_o}{\tau}$ IN A 2 MASS-SPRING SYSTEM

Figure 8.

which provides more accurate representation of the true conditions and accordingly yields more precise results. The "tape" is modeled as a series of distributed masses and springs at fixed distance intervals (normally 10 feet (3.048 m)), with fixed position sheave masses also present. The terminal masses represent the aircraft and "shuttle," and the reel with contained "tape."

Equations (1), (2), and (3) are applied as previously described except that the M_R and M_A coefficients are now descriptive of the input and output masses of each distributed 2 Mass-Spring model. Thus, during the time increment Δt in which $F_R(t)$ is postulated, a series of computations in sub-intervals of Δt are conducted, thereby tracing the response effect from mass to mass, over the entire "tape" length.

The results of this method of analysis effectively reveal such subtleties as superimposed transmitted and reflected tension waves causing tension to vary over the length of the "tape." Reflected waves, for example, appear when a forward moving wave encounters an increased mass, which causes a reversal of tension differential across the driving mass, thus causing a force wave to move in the opposite direction.

2) Clutch Input Analysis

As observed from the input side, the clutch torque obtained from frictional slippage requires a proportional flywheel deceleration rate. Energy produced by the slippage is dissipated as heat in the clutch. For the least energy lost to slippage the clutch input and output speeds must converge, with a common "lock-up" speed occurring immediately at the end of the launch stroke.

From the foregoing rationale, the initial and final flywheel speeds may be determined from the relationships:

$$\omega_{ii} - \frac{1}{J}\sum_{n_{\Delta t}=1}^{n_{\Delta t}=j} T_c \Delta t = \omega_{if} = \omega_{of} \tag{5}$$

where: ω_{ii} = Initial (Pre-Launch) Flywheel Speed.
ω_{if} = Minimum (Post-Launch) Flywheel Speed.
ω_{of} = Maximum Clutch Output Speed.
$n_{\Delta t}$ = Time Increment Number.
j = "jth" Iteration at End Of Launch.
J = Flywheel Inertia at Clutch Input Speed.
T_c = Clutch Torque.

Clutch torque T_c is related to the force F_R by the reel radius, and may be expressed as:

$$T_c = F_R(t)R_R \tag{6}$$

where: R_R = Instantaneous Reel Radius

$$= \sqrt{R_{Ri}^2 + \frac{(L_o - y)c}{\pi}}$$

R_{Ri} = Reel Radius, fully unreeled at start of launch.
$(L_o - y)$ = Length of unreeled "tape."
c = "Tape" Thickness.

Since the energy requirements will vary with different combinations of mass and velocity launch factors, the time of the launch and clutch output speed will also vary, thereby necessitating appropriate changes to the clutch torque program. Hence, as shown by Equation (5), the initial flywheel speed ω_{ii} must be anticipated prior to launch, and the prime mover revolutions per minute (rpm) adjusted accordingly.

It should be noted that Equation (5) does *not* provide for the effect of flywheel frictional torque, although it is quite substantial. In effect, the analysis assumes that an offsetting torque supplied by the prime mover exists at all times when the flywheel is rotating, except when necessary to reduce the initial flywheel speed for a lower energy launch requirement.

Input energy to the clutch, E_i, which includes the slippage loss and the energy transmitted to the output, may be expressed by the relationship:

$$E_i = \sum_{n_{\Delta t}=1}^{n_{\Delta t}=j} T_c \omega_i \Delta t \tag{7}$$

where: ω_i = Instantaneous clutch input speed during launch [Equation (5)].

Figure 9. Clutch Torque & Speeds versus Time.

The portion of input energy lost to slippage, E_{ic}, may be expressed by the relationship:

$$E_{ic} = \sum_{n\Delta t=1}^{n\Delta t=j} T_c(\omega_i - \omega_o)\Delta t \qquad (8)$$

where: $(\omega_i - \omega_o)$ = Instantaneous clutch slippage speed during launch.

ω_o = Instantaneous clutch output speed = $\frac{\Delta y}{\Delta t}\left(\frac{1}{R_R}\right)$

3) COMPUTED RESULTS

Figure 9 illustrates the clutch torque program, as developed, for launching a 74,500 pound (331,376 N) F14-A aircraft at 135 knots (68.67 m/s). Also shown are the clutch input and output speeds. Specific data are noted as follows:

Pre-Breakaway Clutch Torque	60,000 Ft.lbs.	(81,600 Nm)
Maximum Clutch Torque	490,000 Ft.lbs.	(666,400 Nm)
Initial Clutch Input Speed	103 Rad./sec.	
Final Clutch Speed	80 Rad./sec.	
Flywheel Inertia at Clutch Speed	46,515 slug Ft²	(63,240 kg m²)

Torque is controlled in fixed rate steps in order to simplify the program function generator control parameters and to restrict frequency response demands of the clutch control loop. The oscillations in output speed are caused by a changing applied "tape" force, first increasing then decreasing, as a result of the changing ratio of torque to reel radius.

In Figure 10, the acceleration, velocity and stroke characteristics of the same F14-A launch are shown. Specific data are as follows:

Stroke	= 305 ft	(92.96 m)
Peak Acceleration	= 2.8G's	
$\frac{P}{M}$ Loading Factor	= 1.27	

A comparison of the Clutch Output Speed Curve in Figure 9, with that of the Aircraft Acceleration in Figure 10, indicates that the vibration periods are the same, and that the frequency increases as the length of unreeled "tape" decreases, as normally expected.

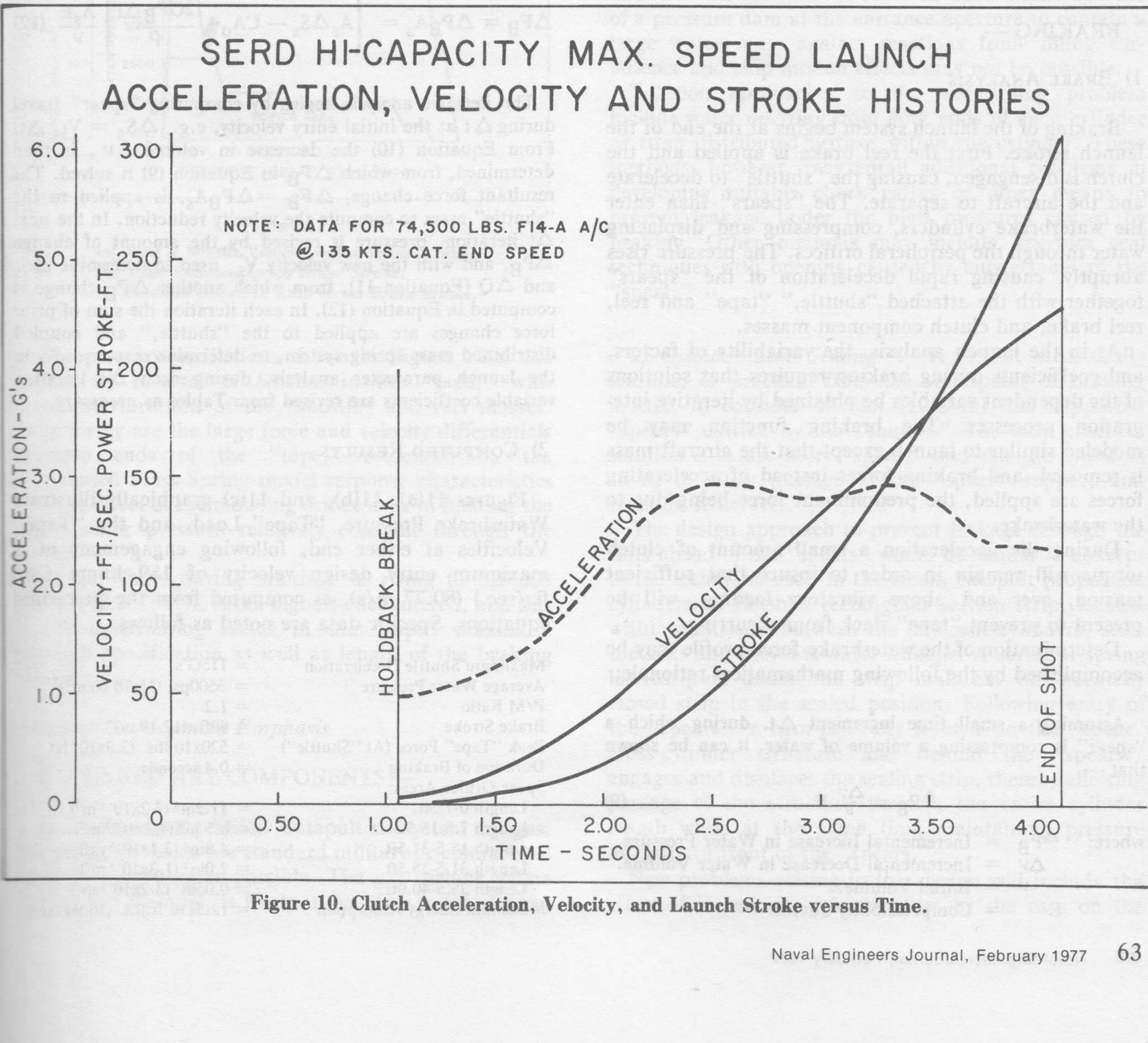


Figure 10. Clutch Acceleration, Velocity, and Launch Stroke versus Time.

An analysis of the total launch energy requirements for the F14-A launch provides the following distribution:

Pre-Launch Stored Energy	$300x10^6$Ft.lbs.	($221.3x10^6$ J)
Discharged Energy at Launch	$98x10^6$Ft.lbs.	($72.3x10^6$ J)
Energy Lost in Clutch	$31x10^6$Ft.lbs.	($22.9x10^6$ J)
Energy Gained by Aircraft	$59x10^6$Ft.lbs.	($43.5x10^6$ J)
Miscellaneous Parasitic Losses	$8x10^6$Ft.lbs.	($5.9x10^6$ J)

In addition to restoring the energy discharged at launch during the 37 second period for recharge, the prime mover will be required to support continuous bearing friction losses in the fear reducer (between flywheel and clutch) and aerodynamic drag losses of the flywheel. These losses cause a constant torque load on the prime mover requiring 2,200HP ($1.64x10^6$ Nm/s).

The peak input power will occur during recharge, consisting of the combined demands of energy recharge plus that of the friction and drag, i.e., Horsepower = $\frac{98x10^6}{550x37} + 2{,}200 = 7{,}015$ ($5.23x10^6$ Nm/s) (37 sec. Recharge) [A prime mover rating of 10,000HP has been assumed, in anticipation of added requirements for auxiliary power takeoffs].

BRAKING —

1) Brake Analysis

Braking of the launch system begins at the end of the launch stroke. First the reel brake is applied and the clutch is disengaged, causing the "shuttle" to decelerate and the aircraft to separate. The "spears" then enter the waterbrake cylinders, compressing and displacing water through the peripheral orifices. The pressure rises abruptly, causing rapid deceleration of the "spears" together with the attached "shuttle," "tape" and reel, reel brake, and clutch component masses.

As in the launch analysis, the variability of factors, and coefficients during braking requires that solutions of the dependent variables be obtained by iterative integration processes. The braking function may be modeled similar to launch except that the aircraft mass is removed, and braking forces instead of accelerating forces are applied, the predominant force being due to the waterbrake.

During the deceleration a small amount of clutch torque will remain in order to insure that sufficient tension, over and above vibratory loading, will be present to prevent "tape" slack from occurring.

Determination of the waterbrake force profile may be accomplished by the following mathematical rationale:

Assuming a small time increment Δt, during which a "spear" is compressing a volume of water, it can be shown that —

$$\Delta P_B = \frac{\Delta \nu}{\nu} E \tag{9}$$

where:
ΔP_B = Incremental Increase in Water Pressure.
$\Delta \nu$ = Incremental Decrease in Water Volume.
ν = Initial Volume.
E = Compressibility Factor.

During the same Δt interval the decrease in volume will be equal to the volume penetration of the "spear," less the volume of water passing through the peripheral orifice separating "spear" and cylinder, i.e. —

$$\Delta \nu = \left(A_s \, \Delta S_s\right) - \Delta Q \tag{10}$$

where:
A_s = Effective Area of "Spear."
ΔS_s = Travel of "Spear."
ΔQ = Orifice Outflow Volume.

and, from the classical orifice flow relationship:

$$\Delta Q = CA_o \sqrt{\frac{2GP_B}{\rho}} \Delta t \tag{11}$$

where:
C = Orifice FLow Coefficient.
A_o = Orifice Area.
G = Gravitational Constant.
ρ = Density of Water.

Substituting Equations (10) and (11) into Equation (9) and solving for the change in brake force ΔF_B:

$$\Delta F_B = \Delta P_B A_s = \left[A_s \Delta S_s - CA_o \sqrt{\frac{2GP_B \Delta t}{\rho}}\right] \frac{A_s E}{\nu} \tag{12}$$

The iterative analysis begins by computing "spear" travel during Δt at the initial entry velocity, e.g., $\Delta S_s = V_{is} \, \Delta t$. From Equation (10) the decrease in volume, $\Delta \nu$, is then determined, from which ΔP_B in Equation (9) is solved. The resultant force change, $\Delta F_B = \Delta P_B A_s$, is applied to the "shuttle" mass to compute the velocity reduction. In the next Δt iteration, pressure is revised by the amount of change ΔP_B, and with the new velocity V_s', used to determine ΔS_s and ΔQ (Equation 11), from which another ΔP_B change is computed in Equation (12). In each iteration the sum of prior force changes are applied to the "shuttle," and coupled distributed mass-spring system, to determine response. As in the launch parameter analysis, during each Δt iteration variable coefficients are revised from Tables as necessary.

2) Computed Results

Figures 11(a), 11(b), and 11(c) graphically illustrate Waterbrake Pressure, "Tape" Load, and the "Tape" Velocities at either end, following engagement at a maximum entry design velocity of 159 knots (265 ft./sec.) (80.77 m/s), as computed from the preceding equations. Specific data are noted as follows:

Maximum Shuttle Deceleration	= 115G's
Average Water Pressure	= 5500psi (15.78 N/m²)
P/M Ratio	= 1.2
Brake Stroke	= 40ft. (12.19 m)
Peak "Tape" Force (At "Shuttle")	= $520x10^3$lbs. ($2.3x10^6$ N)
Duration of Braking	= 0.4 seconds
Spear Orifice Areas:	
Length 0-7.5ft.	= 11.2in² ($7.2x10^{-3}$ m²)
Length 7.5-15.5ft.	= 6.5 in² ($4.2x10^{-3}$ m²)
Length 15.5-31.5ft.	= 4.8in² ($3.1x10^{-3}$ m²)
Length 31.5-35.5ft	= 2.0in² ($1.3x10^{-3}$ m²)
Length 35.5-40.0ft	= 0.05in² ($3.2x10^{-5}$ m²)
Maximum Energy Absorption	= $12.5x10^6$ft.lbs. (16.94 Nm)

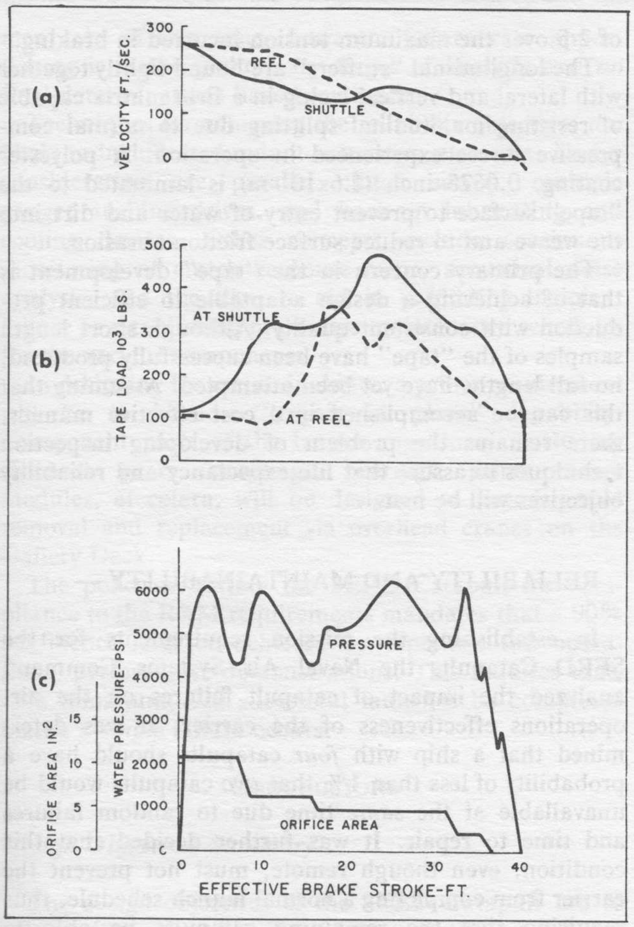


Figure 11.
(a) Reel & Shuttle Velocity versus Brake Stroke.
(b) Tape Tensions versus Brake Stroke.
(c) Pressure & Orifice Area versus Brake Stroke.

The pressure oscillations shown in Figure 11(a) are caused by reversal of tension in the "tape," with resultant vibrations of the "shuttle" and reel masses. Noteworthy are the large force and velocity differentials between ends of the "tape," evidenced by the distributed Mass-Spring model response characteristics and the effect of the reducing orifice area in holding the mean water pressure relatively constant through the stroke.

"Tape" force during braking, in rising to 520,000 pounds ($2.3x10^6$ N), is the highest encountered, and this is the determining factor in the "tape" maximum strength specification as well as length of the braking stroke.

Areas of Development Emphasis

EXTENDED ART COMPONENTS —

In developing the SERD Catapult subsystem designs, the policy will be to use standard military or commercial components wherever possible. The gas turbine prime mover, for example, is likely to be of an existing type, possibly qualified for shipboard applications. In other areas, it is apparent that some components will require capacities and ratings necessitating an extension of the design art. Components in this category are principally the waterbrake, clutch, and tow "tape." The design problems and considered initial approaches for each are presented in the following discussions:

WATERBRAKE—The basic waterbrake technology was developed for the steam catapult. As in the C-13 catapult, a 5-foot (1.52 m) end supported "spear" engages a waterfilled cylinder of equal length and is decelerated to a stop. In the SERD Catapult, the increase in braking inertia due to the "tape" and other drive components, together with the strength limitation applied to the "tape," results in a waterbrake having dual cylinders and "spears," and braking distance of 40 ft. (12.2 m). These changes produce design problems as follows:

1) *Waterfilling* — As a result of the increased length, filling and maintaining the cylinders full with water between braking cycles will require a greater pressure and flow rate than that of the C-13 waterbrake. The use of a pressure dam at the entrance aperture to contain a large water mass against reactions from filling turbulence and ship motion effects may not be possible.

Solution approaches to the waterfilling problem include water *inletting* from both ends of each cylinder or from distributed orifices within the cylinder. These methods, however, may entail additional problems in developing suitable check valves or other means to prevent leakage under the high pressures caused by braking. Other methods may include aperture seal techniques with opening caused by engagement of the "spear."

2) *Cylinder Slot Sealing* — It is not structurally feasible, or practical from the standpoint of braking inertia, to consider 40 foot (12.2 m) end supported "spears" carried by the "shuttle." The only practical solution is to use short "spears" and slot the cylinders so as to allow passage of the "spear" support and reaction structure.

The design approach to prevent leakage through the slots ahead of the "spears" would be similar in concept to the technique used in the steam catapult propulsion cylinders. A flexible, rectangular section strip, located within a channel between the adjacent cylinders, seals the slots and prevents water leakage. A series of spring loaded pins under the strip maintain the normally closed strip in the sealed position. Following entry of the "spears," a cam centrally located on the "spear" cross-cylinder structure and behind the "spears," engages and displaces the sealing strip, thereby allowing passage of the structure through the entire cylinder length while at the same time maintaining pressure integrity in front of the "spears."

New problems arising in this design will include the effect of impact at high velocities by the cam on the sealing strip. Another problem will be that of preventing loss of seal due to vibration propagated through the strip following the initial impact.

In dealing with these problems consideration will be given to the use of special hardened materials for the cam, and the possible use of pneumatic or hydraulic methods of seal pressurization to inhibit wave propagation in the strip.

CLUTCH—The planned initial design approach to the clutch will be that of a dry multi-disc friction type. The requirement to transmit a torque of 500,000ft.lbs. (677,500 Nm) in the "slip mode" of operation, which will generate approximately 45,000Btu's (47.5×10^6 J) of heat per launch cycle, is unique. While clutches of this torque rating are commonly available, they *cannot* be operated in a "slip mode" with any reasonable life expectancy due to the high rate of wear in combination with the heat output and temperature rise.

Other design constraints include an inertia of 700 slug ft^2 (949 kgm^2) on the clutch output side, an overall space limit envelope of 168 inches x 80 inches x 64 inches (4.27 m x 2.03 m x 1.63 m), and a life of 10,000 maximum energy engagements under maximum duty cycle conditions.

The principal factor influencing clutch disc performance is heat, caused by the slippage between input and output discs. The heat rate will produce temperature gradients causing mechanical stresses throughout the disc. In addition, the temperatures, if allowed to rise sufficiently, will degrade material strength enough to allow stresses in excess of yield, thus resulting in distortion and more heat and consequent rapid failure.

The approach concept to the problem of removing clutch heat in the dry friction clutch is to provide air cooling at 250°F (121°C) at a rate of 12,000 CFM (5.66 m^3/s) directed over the clutch discs. By ducting between components, a major part of the air requirement could be obtained by bleeding the gas turbine at a suitable stage (assuming ample power reserve), and cooling it by water injection. The remainder could be obtained with a motor driven compressor.

As a back-up for the dry clutch development, a method of converting the design to that of an oil filled wet clutch is contemplated. The use of oil in contact with the friction discs may yield improved temperature control and more uniform and repeatable torque as a function of disc pressure.

"TOW TAPE" — The present "tow tape" design concept is that of a woven polyester fibre tape, 20-inches (0.51 m) wide and 1-inch (0.025 m) thick. This material was selected over others, including maraging steel, because of a superior strength to weight ratio, greater modulus of elasticity, smaller bending radius, and stability of properties over a wide range of environmental conditions.

The "tape" weave design consists of 2,400 longitudinal tensile bundles, called "stuffers," each having a breaking strength of 550lbs. (2446 N), or 1,320,000lbs. (5,870,400 N) total, thereby providing a factor of safety of 2.5 over the maximum tension incurred in braking.

The longitudinal "stuffers" are bound tightly together with lateral and vertical lacing in a firm matrix capable of resisting longitudinal splitting due to normal compressive forces experienced in operation. A polyester coating, 0.0625 inch (1.6×10^3 m) is laminated to the "tape" surface to prevent entry of water and dirt into the weave and to reduce surface friction abrasion.

The primary concern in the "tape" development is that of achieving a design adaptable to efficient production with consistent quality. Although short length samples of the "tape" have been successfully produced, no full lengths have yet been attempted. Assuming that this can be accomplished in a cost effective manner, there remains the problem of developing inspection techniques to assure that life expectancy and reliability objectives will be met.

RELIABILITY AND MAINTAINABILITY —

In establishing the mission requirements for the SERD Catapult, the Naval Air Systems Command analyzed the impact of catapult failures on the air-operations effectiveness of the carrier. It was determined that a ship with *four* catapults should have a probability of less than 1% that *two* catapults would be unavailable at the *same* time due to random failures and time to repair. It was further decided that this condition, even though remote, must not prevent the carrier from completing a normal launch schedule, thus requiring that the remaining catapults be able to assume the full launch burden in the interim.

From these and related analyses, determinations of the maximum launch rate and basic Reliability and Maintainability (R&M) requirements were made. As specified by the Naval Air Systems Command, the Mean Cycles (Launches) Between Failure (MCBF) is to be *not less* than 400 and the Mean Down Time (MDT) for maintenance *not in excess of* 4.5 hours for Fleet service approval.

In developing a catapult design to meet these basic R&M requirements, extensive use will be made of mathematical models to assist in defining and controlling R&M design specifications at the sub-system and lower assembly levels. Apportionment of the system failure rate ($\frac{1}{MCBF}$) within such models will begin with standard components on which failure rate data is furnished by the manufacturer, or obtained by testing. The remainder of the allotted failure rate will then be divided between the non-standard components in relation to the judged design complexity and severity of problems to be overcome. A subsequent analysis will then be made on component failure modes and effects to determine whether consequent damage to the catapult or aircraft or injury to personnel could result. Components in this category will be given preference by reducing the failure rate apportionment, or, if warranted by the criticality of the effect, will be given a separate and independent percentage reliability requirement. As of this time, however, failure rate apportioning is in the preliminary stages of analysis and *no* final determinations have been made.

Techniques for achieving the reliability objectives are likely to include redundant subsystem elements. Under consideration are parallel clutch servos, multiple ganged turbine drives, and back-up lubricating and cooling devices. Other concepts include continuous monitoring of clutch response error, acoustical noise analysis, and the like, to aid in predicting incipient failure or establishing preventive maintenance schedules for the various components.

In designing for maintainability, a prime requirement will be to have a "repair in place" capability. Component parts of the launch machinery, including flywheel, gears, couplings, bearings, prime mover modules, et cetera, will be designed to permit rapid removal and replacement via overhead cranes on the Gallery Deck.

The policy in testing the SERD Catapult for compliance to the R&M requirements mandates that a 90% confidence level be achieved by repetitive full operational performance demonstrations. This requires that, as a minimum, 920 successful launches be completed before a *single* failure occurs.

CONCLUSIONS

Design Objectives Summary

The design objectives of the SERD Catapult are to achieve improved ship interface campatibility and operational effectiveness relative to the existing steam catapult. Important gains will be a large reduction in topside weight, greatly increased resistance to shock, and elimination of dependency on ship steam and water. Air operations will benefit by the elimination of aircraft steam ingestion, deck obscurations, trough fires, and temperature stabilization delays. The Program also provides the opportunity to make major improvements in Reliability and Maintainability through application of current technologies in failure prediction modeling and human factors design.

Risk Assessment Summary

Prospects for a successful development program appear good based upon the following summary factors:

1) Prior experimental efforts in connection with land-based systems have demonstrated the feasibility, although at lower energy capacities, of the flywheel powered, clutch controlled, "tow tape" driven catapult. There is no evidence to indicate that there is a limit to the energy delivery capacity in this concept.

2) The SERD Catapult design has been soundly formulated through use of computer simulation programs, permitting analysis of optimum system and component parameters. This capability to obtain accurate insight into performance factors and stress levels in advance of a final design greatly reduces the attendant technical risks, as in any system of such complexity.

3) Areas of component design requiring an extension in the state-of-the-art have been identified, and a systematic program of design and test has been launched. None of the items impose theoretical restrictions against increasing of capacities, or changes to suit environmental constraints. Manufacturability aspects are simultaneously being given consideration in the selection of candidate design solutions. Ship installation factors are also receiving attention through coordination meetings with representatives of the Naval Sea Systems Command. The results of these meetings have been to resolve most of the design approaches to be taken toward satisfying mutual interface requirements.

REFERENCE

[1] Frankland, J.M., "Effects of Impact on Simple Elastic Structures." David W. Taylor Model Basin Report No. 481, April 1942.