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NAVY DEPARTMENT
OFFICE OF NAVAL RESEARCH

AN INVESTIGATION OF PROBLEMS CONNECTED WITH
LAUNCHING AND RECOVERING CARRIER AIRCRAFT

Contract No. Nonr 850(00)

Arthur D. Little, Inc.
Cambridge 42, Massachusetts

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NAVY DEPARTMENT
OFFICE OF NAVAL RESEARCH

**AN INVESTIGATION OF PROBLEMS CONNECTED WITH
LAUNCHING AND RECOVERING CARRIER AIRCRAFT**

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C-58620

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Arthur D. Little, Inc.
Cambridge 42, Massachusetts
August 20, 1953

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Arthur D. Little, Inc.
RESEARCH - ENGINEERING - INVESTIGATION



CAMBRIDGE 42, MASSACHUSETTS

August 20, 1953

Office of Naval Research
Department of the Navy
Washington 25, D. C.

Attention: Code 461

Subject: Contract NONR 850(00)

Gentlemen: C-58620

In accordance with the terms of the subject contract,
we submit herewith a technical report on the work performed by
Arthur D. Little, Inc. on the investigation of problems associated
with launching and recovering carrier borne aircraft.

Respectfully submitted,

Arthur D. Little, Inc.

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CONFIDENTIALSUMMARY

In keeping with the terms of the contract and in view of the great bulk of information on specific problems of catapulting and arresting gear, this report is directed toward a broad theoretical evaluation of both conventional and novel mechanisms with a view to finding the major advantages and drawbacks inherent in various schemes for launching and recovering aircraft.

The report opens with a few comments on carrier aviation in general, and the influence of plane speed and size on the carrier dimensions. From here the discussion moves to a consideration of the basic arrestation problem - the provision of an energy sink and engine by means of which energy can be transferred from the aircraft to the sink - and the conclusion is reached that the energy should be dissipated thermally to either an atmospheric or liquid sink.

Before dealing with proposed engine types, attention is paid to the possible means of linking the aircraft to the engine. The advantages of applying arresting loads to various parts of the fuselage and undercarriage are weighed and it is concluded that the orthodox tail hook is preferable to landing gear or fuselage attachment. An alternative scheme to the cross deck pendant is proposed and takes the form of a coarse net actuating a series of small engines. This scheme has the advantage that all engines on board the carrier would operate during each arrest and each engine would absorb only a part of the total energy.

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An analytical study of a number of mechanical and electromagnetic engines follows.

The study evaluates the ability of each system to dissipate thermal energy at high rates while providing a constant retardation of 3g and considers the structural feasibility of each system. Some attention was paid to the inertia of the devices since this is important at high engagement velocities.

The study revealed the following facts.

1. Friction devices, though structurally simple, would overheat rapidly since there does not seem to be a really satisfactory cooling method. Of the reeved cable types, the hydraulic ram is most satisfactory. All reeved cable engines will apply high inertia loads to the aircraft on engagement and a method is suggested whereby these loads may be reduced.
2. A rotary viscous shear device would be extremely bulky and complicated.
3. A braked hydraulic coupling would require an elaborate control device to insure constant retardation, would produce high inertia loads, and would need a friction brake to bring it to a complete stop.
4. A multiple damped spring system is simple, has low inertia, but presents structural difficulties.
5. Electro-magnetic devices are, in general, extremely heavy and require auxiliary power.

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Some consideration is given to the popular proposal that a high speed air stream be provided on the flight deck by a suitable wind machine. The scheme is shown to be impracticable due to the extremely long "wind corridor" needed.

The section dealing with launching problems opens with a discussion of possible energy sources and the energy storage problem.

An evaluation of the hydraulic cable-drive catapult is presented before passing to the consideration of less orthodox gears.

The following devices were given attention.

- (1) Linear steam and water turbines
- (2) The slotted cylinder mechanism — actuated by steam, an internal combustion gas generator, and a hydraulic fluid.
- (3) Flywheel driven "lead screw"
- (4) Electrically driven engines including
 - (a) Linear motor directly coupled to shuttle
 - (b) Rotary motor with cable linkage

The slotted cylinder mechanism appears to be the most attractive catapult type, and would be most suitably powered by a gas generator. The combustion of a hydrocarbon in oxygen is the most attractive gas generating system.

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CONFIDENTIALINTRODUCTION

The development of both civil and military aircraft has been characterized by ever increasing speeds, sizes and payloads, which bring in their wake increased handling difficulties. Long take-off and landing runs present a problem which, though serious in the case of land based planes, is critical for carrier based aircraft. Since the difficulties associated with lengthening a flight deck are more complex than those arising from an extension of an airport runway, the problem must be solved by the development of more effective launching and retrieving methods.

Lack of maneuverability in an aircraft which is required to make a sure and rapid landing in a limited area must inevitably place a burden on the pilot: a burden which can be removed to some extent by better ground control of his approach.

A more basic attack has evolved in recent years and entails a critical re-evaluation of all the components which make up a military plane. Consolidation of much of the auxiliary equipment will result in larger payloads for a given flight weight.

Under the broad program of study and research which the Navy Department has built around the problems of carrier aviation, Arthur D. Little, Inc. undertook a study of some of the problems connected with launching and retrieving aircraft. The present report deals with the results of the study.

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CONFIDENTIALSCOPE OF THE STUDY

The scope of the study is defined in the contract as follows:

"The contractor shall furnish the necessary personnel and facilities for and, in accordance with any instructions issued by the Scientific Officer or his authorized representative, shall conduct a study of the various problems connected with launching and arresting aircraft in carrier operations. This research should include, but not necessarily be limited to methods of supplying or absorbing the required energy, (1) to launch heavy, high-speed, high-lift aircraft at the required speeds, and (2) to retrieve them with the smallest amount of arresting energy to do the job efficiently. The study should take into consideration the heavier, high-speed aircraft now in the stages of prototype for the 1956-60 era of carrier operations."

It was felt that, since there has not been a similar study made in the past, it would be advisable to incorporate in the report some general considerations of the problem. From this point of view, the report should provide a general background for anyone who might make more detailed studies of the problems. It should, by its basic considerations, point up means of launching and retrieving which might be worthy of further development.

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CONFIDENTIALWATER BASED PLANES

Until recently, experience with water based aircraft had been limited to fairly large and relatively slow aircraft operating from protected bases close to the shore. Extending this knowledge to the design of high-speed aircraft, intended to operate from unprotected water, would seem to imply greatly strengthened hulls with an attendant weight penalty. However, the introduction of water skis on a recent design appears to have reopened the whole question of high-speed water based planes. This development, while contributing to the effectiveness of a joint sea and land operation, cannot be construed as rendering obsolescent the orthodox system of carrier aviation.

The utility of such aircraft is limited by the condition of the sea (the present limitation being five-foot, high-frequency waves) and it is doubtful if they could operate in the heavier seas which still permit flight deck operations.

Planes of this type will probably make their greatest contribution during an offshore operation when land bases are unavailable.

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CONFIDENTIALCONVENTIONAL AIRCRAFT

In considering the launching and retrieving process, it serves little purpose to divide the aircraft into groups which specify their purpose. The multiplicity of types, shapes and sizes, serves only to complicate the basic problem of supplying and receiving large amounts of kinetic energy. In general, we may say that two broad categories exist - a high-weight low-speed group and a low-weight high-speed group - and even these are not clearly defined. The average fighter belongs to the second group while the attack aircraft is characteristic of the first group. However, as planes grow and develop, much overlapping occurs and many fighters now have both a high top speed and a high flight weight.

In this respect, it is interesting to note the improvement in carrier striking power which can be attained if the size of the planes can be reduced without loss of effectiveness. If it be assumed that the number of planes which a carrier can carry is proportional to its deck area, i.e. proportional to the two thirds power of its relative weight, and if it be further assumed that the weight of the plane is cut in half, then a given carrier can provide a striking force about 60% greater with light small planes than with heavier aircraft. Or expressing this another way, one carrier equipped with light planes need have only half the displacement of one carrying heavier planes for the same striking power.

The problems associated with the arrestation of heavy high-speed aircraft are greater than those involved in the launching process. The large amount of energy to be absorbed implies massive

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equipment with large inertia and this large inertia produces severe impact stresses at high entry speeds. Strengthening the aircraft to withstand these stresses results in a weight penalty to the aircraft and calls for a further increase in the energy capacity of the arresting gear.

In contrast with this, the penalty incurred in the launching process is paid by the ship-borne gear rather than the aircraft. The high residual kinetic energy in large capacity catapults results in inefficiencies which show up as a high consumption of the primary fuel and large auxiliary equipment.

Efforts to lighten the flight weight of an aircraft with a given payload are greatly to be advocated and it is notable that this trend is growing steadily.

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CONFIDENTIALINFLUENCE OF CARRIER ON AIRCRAFT DESIGN

The kinetics of flight deck operations involves the following variables:

Aircraft gross weight	W_g
Aircraft stall speed	V_s
Acceleration in g's during launch and retrieve	α
Length of catapult and arresting run	L
Length of carrier	L_c

If the aircraft is assumed to be launched and arrested at the stalling speed V_s we may write;

$$W_g = \frac{1}{2} C_L \rho S V_s^2 \quad (S \text{ being the wing area})$$

$$\text{or } \frac{W_g}{S} = \frac{1}{2} C_L \rho V_s^2$$

the left-hand side is λ the wing loading.

$$\begin{aligned} \text{Since the length of launching run is } L &= \frac{1}{2} \alpha g t^2 \\ &= \frac{1}{2} \alpha g \left(\frac{V_s}{\alpha g} \right)^2 \\ L &= \frac{1}{2} g \frac{V_s^2}{\alpha^2} \end{aligned}$$

$$\text{thus } \lambda = (C_L \rho g) \alpha L$$

For a given catapult length and launching acceleration, the maximum wing loading of the aircraft is fixed; in this way the catapult and arresting gear capacity control a fundamental aircraft design factor.

Taking the representative figures

$$\lambda = 39 \text{ LB/FT.}^2$$

$$\begin{aligned} C_L &= 1.7 \\ \alpha &= 3 \\ L &= 100' \end{aligned}$$

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Aircraft operating with wing loadings of 60-100 lb/ft.² require that the product αL lie in the range 460-765, or if $\alpha = 4$ the catapult length must be between 115 and 190 feet.

If the permissible acceleration incorporated in the structural design of the aircraft can be increased, while keeping the launching and arresting length fixed, the wing loading may be increased in direct proportion. This increased wing loading will show up in two different ways, (a) as an increase in the military load which can be carried, and (b) in the greater weight of the structure which has to be strengthened to withstand the higher accelerations. Another approach is to redesign the aircraft to the same gross weight, or for the same military load, and to accept the profit of higher wing loading in the form of reduced wing size. This may be the preferable attitude since the "growth factor" will be smaller for the lighter aircraft. (The term "growth factor" has been defined in a paper by I. H. Driggs as the ratio of gross weight to military load, and might be considered as an index of the effectiveness of a particular design.) It has been shown that smaller growth factors can be achieved with a low gross weight than with a large gross weight.

A recent design proposal for a day fighter in which the launching acceleration was in the order of 9 g's had a gross weight several thousand pounds less than it would have been had the design been based on lower acceleration limits. In this case the necessary wing area was so small that it was no longer necessary to make provision for folding wings and the elimination of the folding mechanism resulted in a substantial weight saving.

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CONFIDENTIALINFLUENCE OF AIRCRAFT ON FLIGHT DECK SIZE

While it is true that existing carrier facilities largely control the design of new aircraft intended to operate from then, it is equally true that carrier design is influenced by considerations of aircraft scheduled for future design. If the flight deck area is divided into a forward, or launching, area and an after, or landing, area, the lengths of these sections will depend on the take-off and landing speeds and the accelerations permitted during these operations. The length of the forward part will be larger than the catapult length by some factor which allows for handling and parking needs, while the length of the after part will likewise be longer than the arresting gear runout to allow for multiple hooking points (pendants) barricade area, handling etc.

There are simple relations between speed acceleration and launching (or retrieving) run but the factors previously mentioned vary indiscriminately from ship to ship and are of a rather indefinite origin.

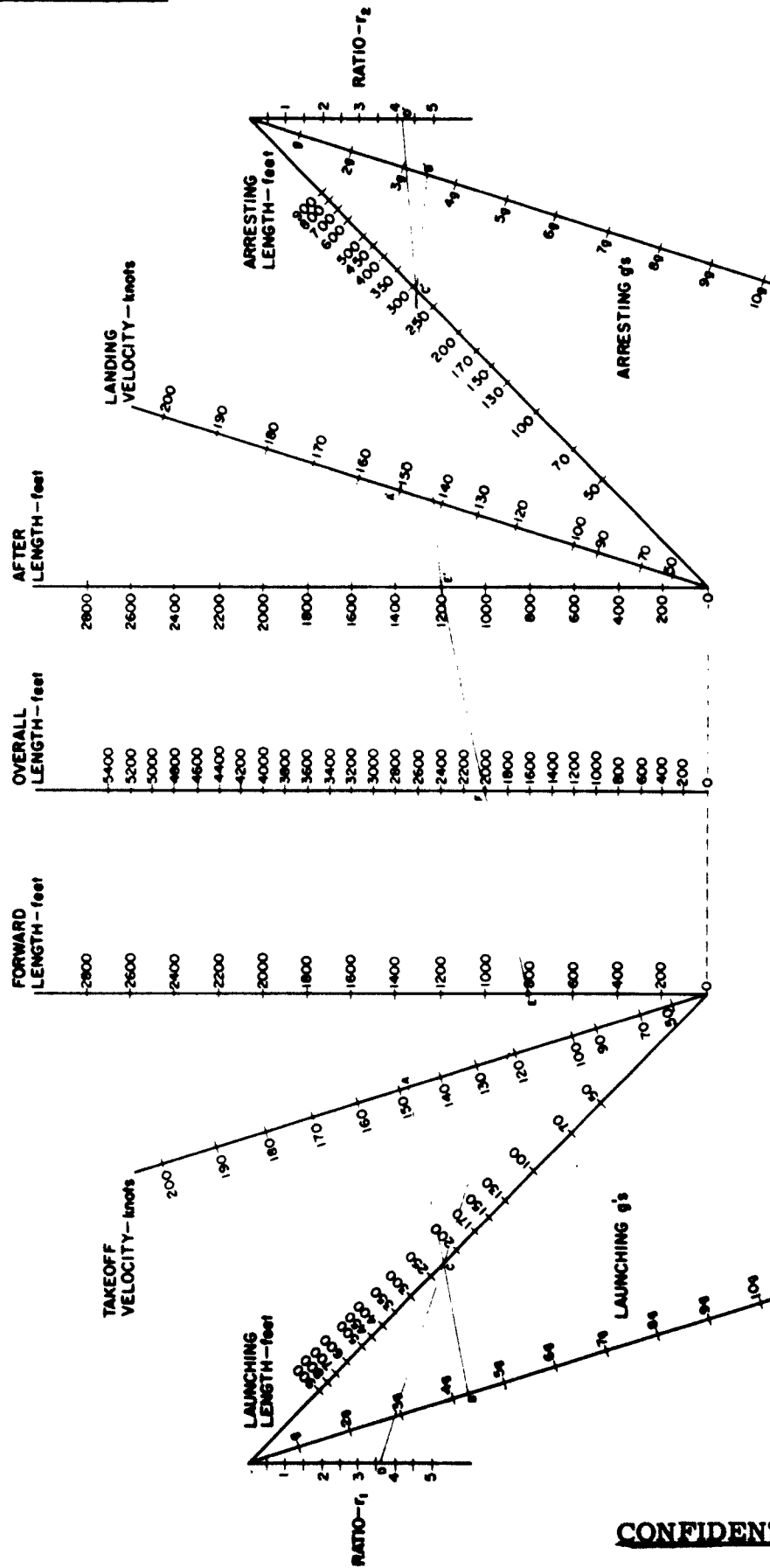
A simple nomograph has been prepared and may be used to find the approximate flight deck size necessary to handle an aircraft with known landing and take-off speeds, which may be launched and arrested at given accelerations.

To find the flight deck required for a given aircraft, connect the take-off speed A with the mean launching acceleration B, giving the point C. Choosing a suitable factor D join this to C by a line giving the point E which represents the length of the forward section of the flight deck.

Taking now the landing speed A' and connecting this

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FLIGHT DECK NOMOGRAPH

FIGURE 1

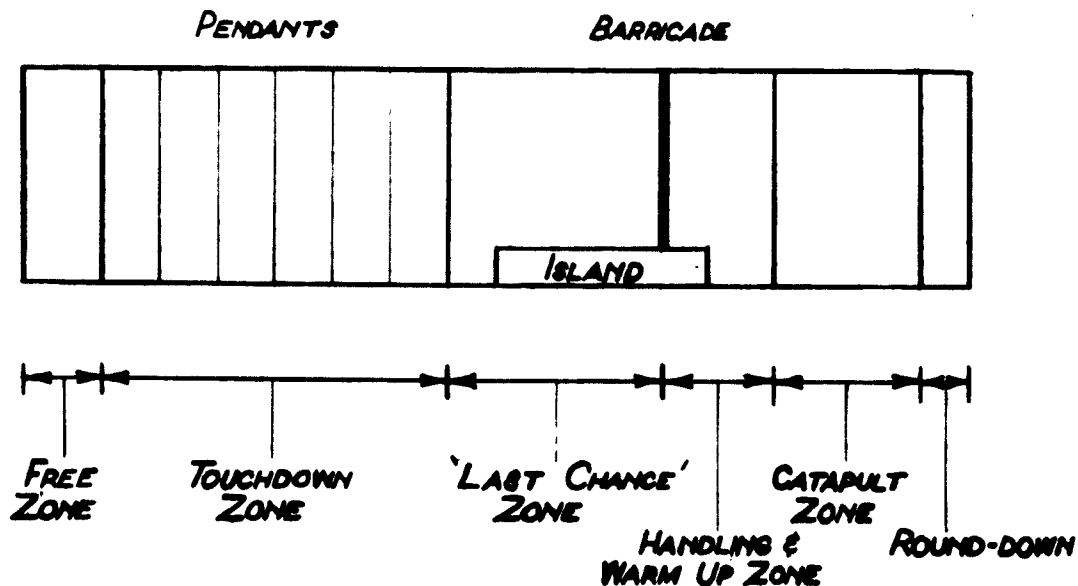
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with the mean landing acceleration B' gives the point C' which when joined to the factor D' gives E' , representing the length of the after section.

If the points E and E' are connected, the resulting point F represents the flight deck length.

It can now be seen that attempts to reduce the flight deck length must involve a reconsideration of the obscure factors which cause this length to increase beyond the minimum catapult length and arresting runout.

Consider the approximate layout of an average deck

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At the after limit of the flight deck there is a region, which we may call a 'free' zone, in which it is inadvisable for a pilot to attempt to land since a slight error in judgment (or a pitching deck) could cause the aircraft to hit the stern of the carrier. It should be pointed out that the extent of this free zone will increase as the landing speed goes up, assuming the human reflex time and the response time of the aircraft to remain the same. Another way of saying this is that, as approach speeds increase, most normal landings will occur farther forward on the deck.

With regard to the touchdown zone, it is obvious that one pendant would be sufficient if it could be assured that every aircraft picked up this wire. Since no such assurance can be made, many wires must be provided to increase the probability of hooking on after touchdown.

There does not appear to be any clear basis on which to choose the number of wires or the distance between them and it is concluded that the arrangement is a matter of opinion based on experience. If this is true, it would perhaps be worthwhile to re-evaluate the choice of the number of pendants and their separation, since it may be possible to reduce both. While it is true that such a step would increase the possibility of barrier engagement and consequent plane damage, the shorter flight deck would result in a smaller initial cost of the carrier. Needless to say, the injury to flight personnel brought about by barrier engagement must be taken into consideration, and if past experience shows this to be a hazardous event, the proposal would be ruled out.

With regard to what has here been called the "last chance"

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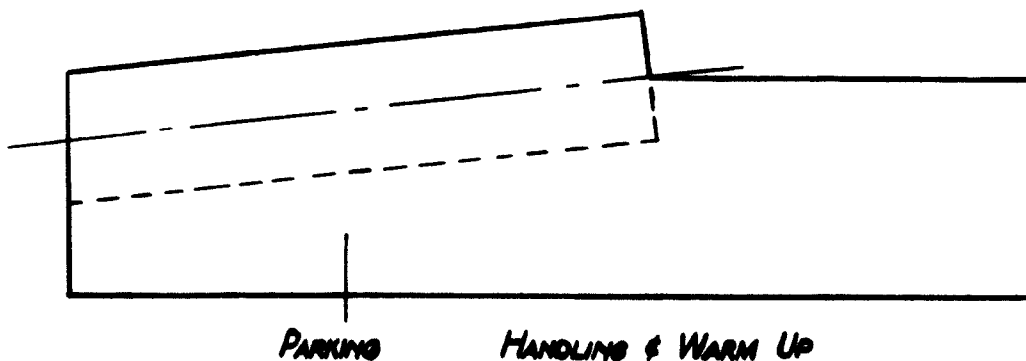
zone, this is equal in length to the runout on the last wire and any aircraft landing in this region will contact the barriers and barricade.

The handling and warm up zone immediately forward of the barricades will have a length which may vary with the size of the aircraft being launched and the launching frequency. A small handling zone should be added to the other advantages accruing to the use of small aircraft.

We have already seen that the length of the catapulting zone is set by the physical laws connecting take off speed and mean acceleration and any reduction of this length must come through an increase in catapult capabilities.

Of the "round-down" of the flight deck, little can be said other than that it exists.

A novel flight deck configuration which bears the name "canted deck" has many advantages over what has now become known as the "axial deck". As shown below, the landing area is skewed relative to the axis of the carrier, thereby eliminating the possibility of a landing aircraft colliding with planes parked on the main deck.

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Elimination of this hazard permits the removal of barricades during a normal landing and, since it is no longer mandatory to arrest every aircraft which touches down, the number of deck pendants may be reduced. Not only is the touchdown zone shortened, but the run-out zone may overlap (or more accurately, lie alongside) the handling and warm up zone associated with the catapult area.

Additional parking space is a further advantage of the arrangement.

It can thus be seen that the canted arrangement yields a shorter overall flight deck than the axial scheme.

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CONFIDENTIALPHYSIOLOGICAL AND PSYCHOLOGICAL CONSIDERATIONS

Accelerations of the order of three or four g's which are at present encountered in flight deck operations cause little or no discomfort to flight personnel, but, if the trend towards higher accelerations becomes firmly established, the whole question of crew reaction to high "g" loading will become pressing.

Previous investigations of human tolerance of acceleration have been confined largely to the study of structural damage and the question of psychological reaction has received little attention. In the case of catapulting and arresting aircraft, psychological effects are of considerable importance since it is essential that the crew be able to function and react normally as soon as possible after the acceleration ceases. This is particularly important in the catapulting operation since the pilot must take over control of the aircraft soon after leaving the deck. During the arresting operation this is not so critical since the plane is now safely at rest. Brief discussions with medical personnel of the U.S. Navy have indicated that the limitations on g set by psychological reactions are much lower than those set by physiological considerations. Tentative limits suggested were of the order of 8 to 10 g in the first case and about 40g in the second case. The maximum acceleration is not the only critical factor involved; it has been found that the rate of build up and the duration play an important though rather obscure part.

Documentary information on this subject is limited, and, as might be expected when dealing with the nebulous field of psychology, results are not of a quantitative nature. The writers of this report

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would urge that an extensive study of this subject be made, since it is likely to form the fundamental limitation of the launching and arresting operations.

CONFIDENTIALARRESTING GEARGeneral Considerations

When an aircraft of weight W approaches a carrier with a relative velocity V , a quantity of kinetic energy $\frac{W}{2g} V^2$ must be removed from the aircraft before it comes to rest with respect to the carrier. The problem of arrestation may be stated in general terms by the question "how may this energy be removed in a safe, rapid and efficient manner"?

This general problem may be subdivided into two further problems, viz.;

(1) What may be done with the energy removed from the aircraft? This leads to a discussion of possible sinks for the energy and the form in which it (the energy) should be delivered to the sink.

(2) What type of mechanism can be used to accept the kinetic energy of the aircraft, convert it to a form suitable for transfer to the sink and carry out this transfer?

A complete quantitative evaluation of all the proposed solutions to these basic problems is obviously beyond the scope of this contract, however, in cases where quantitative study is not presented, qualitative argument will be used to compare various proposals.

Energy Sinks

The question arises as to whether the energy liberated by arrest should be stored for subsequent use or simply dissipated.

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Although the possibility of storing the energy appears attractive, the value of any scheme must be judged on the basis of utilization, size and simplicity. The relatively small amounts of energy released when an aircraft is arrested, together with the relative infrequency of landings (compared with the ship's total operating time) would make it undesirable to install special equipment to store and re-use the energy. Some idea of the magnitudes involved can be gained from the fact that a strike of 50 aircraft each having a landing weight of 20,000 lbs. and a landing speed of 100 kts, liberates a total of 450×10^6 ft. lbs. of energy which is equalled by the heating value of some 3 or 4 gallons of fuel oil. However, the energy cannot be stored as efficiently as in fuel oil but must be stored mechanically, thermally or electrically.

If stored mechanically - say by elastically stressing a mass of metal (e.g. a spring) the volume of metal required can be shown to be extremely large. For example, if the yield stress of the metal in tension is 60,000 psi and $E = 30 \times 10^6$ psi, a volume of 52,000 cu. ft. is required to store 450×10^6 ft. lbs. of tensile strain energy.

Again, if the energy is stored as the potential energy of a compressed gas with pressure limits of 1 atmosphere and 6000 psi the volume of compressed gas and therefore of the storage vessel must be 253 cu. ft.

If electrical storage cells of output rating 12 watt hrs/lb weight are used a total cell weight of some 7 tons would be necessary. The primary objection to energy storage is that the supply and demand

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are not geared together and it is obviously unwise to provide a sizable and costly apparatus for this purpose if its utilization would be quite low.

In general, it may be said that the bulk of the energy should be dissipated, a fraction of the total being used to restore the arresting mechanism to the battery position.

In dissipating the energy, a "sink" is necessary and three possible candidates present themselves - the ship, the sea and the atmosphere. The method of dissipation may be to impart mechanical, thermal or electrical energy to the sink.

I. Dissipation by mechanical means

(a) To the sea

The most direct way to accomplish this is to land the aircraft directly on the water and take it on board the carrier when zero relative speed has been reached. Such a scheme is only applicable to amphibious or water based aircraft and would require minor modifications to the carrier (cranes, handling equipment, etc.). Furthermore, a much smoother sea is required for the operation of water based aircraft than for flight deck operation with orthodox aircraft.

Assuming the use of more or less conventional aircraft, landing on a carrier flight deck - the energy of arrestation may be used to pump sea water against some head, and this head released as kinetic energy when the water is discharged back to the sea. If a 20,000 lb. aircraft is arrested from 100 kts. in 2 seconds, the average rate of energy release is 4.5×10^6 ft.lb./sec. If sea

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water is pumped against a head of 500 psi, the rate of pumping must be 63 cu. ft/ sec. (28,500 gal. /min.) to absorb this energy. The physical size of such an installation would preclude its use on board ship.

(b) To the atmosphere

The arresting forces may be used to operate an air pump which imparts potential or kinetic energy to the surrounding atmosphere. If the pressure rise across the air pump is 500 psi, the rate of pumping will be 355 standard cu. ft/sec.

(c) To the ship

Some of the total kinetic energy of the aircraft will be transferred to the carrier and is quantitatively equal to the product of the arresting force and the distance moved by the carrier. For a given aircraft, landing with a definite speed on a specified carrier which is also moving at a definite speed, there is no way of varying the arresting technique to increase or decrease the carrier's gain in K.E. due to the impact.

This can be seen quite easily from the following. The forces applied to the aircraft are the same (in the line of flight) as the forces applied to the carrier and act during the same period of time. Since the impulse is thus the same for carrier and aircraft, the change in momentum must be the same.

Aircraft

$$M(V_1 - V_2)$$

Carrier

$$M(V_2 - V_1)$$

$$\text{and } V_2 = V_2$$

$$\text{thus } V_2 = \frac{MV_1 + MV_1}{M + M}$$

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However, the remainder of the K.E. might be used to augment the thrust in some way - by reaction against the atmosphere or against the sea. The tremendous capacity required when thrusting against air or water has just been pointed out and it seems unlikely that augmentation of the ship's speed is at all feasible.

II. Dissipation by electrical means

This requires the conversion of the aircraft's K.E. to electrical energy which is then dissipated by an electrical discharge (a) through the atmosphere (b) to the hull or (c) to the sea. The personnel hazards associated with the first two are immediately apparent and would eliminate them as possibilities. The last named would require the provision of an electrode, insulated from the hull and immersed in the sea.

III. Dissipation by thermal means

This requires conversion of the aircraft energy to heat which is then dissipated through coolers to any of the three sinks.

(a) To the sea

If the temperature of a quantity of sea water can be raised from 70°F to - say - 150°F, the amount of such coolant required would be about 9 gallons per second or about 18 gallons per arrest. It is quite possible to store enough coolant for a complete strike and then discharge this at the conclusion of operations.

(b) To the atmosphere

The weight of air necessary to absorb this energy over the same temperature range is 303 lbs/sec. or 8080 s.c.f./arrest.

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CONFIDENTIAL(c) To the hull

If the displacement of the carrier is - say - 20,000 tons, it has a thermal energy capacity of some 4×10^9 ft. lb. per °F rise in temperature. This is enough to absorb the energy of several hundred arrests without an appreciable temperature increase. However, the actual mechanism of transferring the heat to the hull is more difficult than in the case of the other sinks.

In general, it may be concluded that the most effective form of sink is a quantity of fluid (having specific heat and boiling point as high as possible) to which the energy is transferred in the form of heat. This conclusion now focuses attention on the second question - the mechanism by which the kinetic energy of the plane is converted to heat and transferred to the sink. In dealing with this problem, it is necessary to consider first how the mechanism will be actuated - how will it accept the kinetic energy of the aircraft? Assuming that there is some positive coupling between the aircraft and the "converter" what form can this "linkage" have.

Linkages

The ideal linkage would be such as to apply the arresting forces uniformly over the whole structure of the aircraft much in the way that aerodynamic drag forces are applied. In this respect, it has been suggested that a high velocity airstream be created along the carrier deck so that when an aircraft with a stall speed V_g enters this airstream, it may come to rest with respect to the carrier while as yet it has flying speed with respect to the relative wind. This proposal is considered later in the report, the present discussion being

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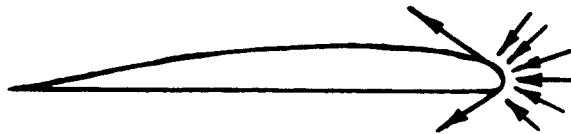
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confined to linkages positively coupled to the aircraft.

Considering the orthodox configuration of an aircraft to consist of fuselage, lifting surfaces, stabilizing surfaces and landing gear, the linkage may be coupled to any of these elements. The point of attachment must obviously not be such as to result in high structural loads in the aircraft.

Wing Attachment

There appear to be more disadvantages than advantages to the proposal that the arresting force should be applied by contacting the leading edge of the wings. Considering a 20,000 lbs. airplane with a 30 ft. wingspan being arrested at 3g's; if the arresting force of 60,000 lbs. is uniformly distributed spanwise, the horizontal bending moment at the wing roots will be 225,000 lbs.ft. If the load is distributed over a spanwise strip of width 6" the surface loading on the leading edge will be 4000 lbs/sq. ft.



Any attempt to reduce the moments by concentrating the loads toward the wing root will increase the surface loading on the leading edge. With high-speed wings of thin section and relatively sharp leading edge, the surface loadings will be even higher than the figure quoted.

If the arresting gear is to be of use, it must be able to accept a great variety of aircraft types, and the difficulties of wing

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contact with delta or swept wing aircraft and propeller driven planes are quite apparent.

There is little to be gained by installing arresting hooks in the wings since the gear necessary to operate such hooks would require a strong, thick wing even though a thin wing was desirable from an aerodynamic standpoint. Turning to the stabilizing surface as a coupling point for the arresting linkage, it is obvious that the aircraft would have to fly under the linkage. Aircraft which fly at a very steep angle of attack (such as the F7U) would find it impossible to fly under the gear and still have the stabilizer fin engage the linkage. Furthermore, the number, shape and size of the stabilizers vary so much from plane to plane that it would be virtually impossible to insure that a linkage placed at a fixed height above the deck would engage on every aircraft.

The leading edge of the stabilizer is quite short in comparison with the wings so that the arresting forces would produce an extremely high surface loading, while the moment produced by these forces would cause a plane with a tricycle undercarriage to settle back on its tail.

The selection of a coupling point can be narrowed to a choice between a hook attached to the fuselage and the main landing gear.

In a discussion with representatives of the Grumman Co., it was learned that present undercarriages are not strong enough to withstand frequent arrests and that during barrier engagements, the undercarriage structure is allowed to yield plastically. During

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barrier engagement, however, the arresting cable catches just above the tires, so that, if a scoop mechanism were added to the undercarriage to lift the cable higher, the bending moments could be considerably reduced. The weight thus saved by the elimination of the hook and dashpot assembly could be used to increase the strength of the main landing gear.

The Grumman Co. supplied an interesting tabulation of weight penalties paid by a typical aircraft to enable it to operate from a carrier. The figures quoted give the range of penalties for five aircraft built by different companies.

<u>ITEM</u>	<u>% FLIGHT GROSS WEIGHT</u>		
Folding wing	1.5	to	2.1
Catapult	.4		.5
Arresting gear	.6		1.5
Barrier	.1		1.0
Landing gear	1.6		2.3

The actual landing gear weight for a land based plane may lie between 5% and 7% of the design gross weight of the aircraft; the figures quoted here refer to the additional weight arising from the higher sinking speeds encountered in carrier aviation.

Elimination of the tail hook assembly would permit a 10% to 20% increase in undercarriage weight and, if the necessary strengthening of the undercarriage can be carried out with a smaller increase, a net weight saving can be accomplished.

A direct answer to this question can only be given by someone familiar with the detailed design of an aircraft but it seems unlikely that a substantial decrease in weight can be achieved in this manner.

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Assuming that the arrester linkage will be coupled to some hook arrangement attached to the aircraft, attention will now focus on the form of the linkage itself. At this point, it is necessary to take account of the fact that the flight path of the aircraft during the approach and touchdown is subject to considerable variation, the extent of this variation depending on the techniques of the L.S.O. and pilot, the condition and handling characteristics of the aircraft, and the condition of the weather.

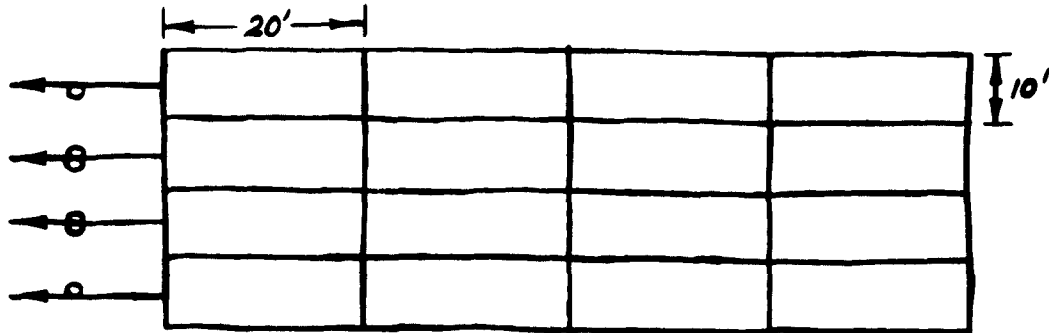
No accurate statistics are available on the point of touchdown but experienced observers have indicated that the bulk of all normal landings occur within an area bounded by lines 20 ft. on either side of the deck center line and by the first 3 wires on a conventionally rigged deck. Abnormal landings may be further off center and/or contact may be made with a late wire.

If an aircraft landing at any point in this "normal" area is to pick up the linkage, the linkage would almost necessarily have to be of the cross-deck pendant type. If the cross-deck pendant be eliminated by the use of multiple hooking points (such as rings) many of these rings have still to be coupled together if the number of converters or arresting engines is to be kept to a minimum. A possible method of providing multiple hooking points is to cover the landing area with a large net of very coarse mesh, the boundary of this net being attached to one or more converters. An apparent disadvantage of this scheme would be the mass of any net strong enough to take the arresting forces for heavy aircraft.

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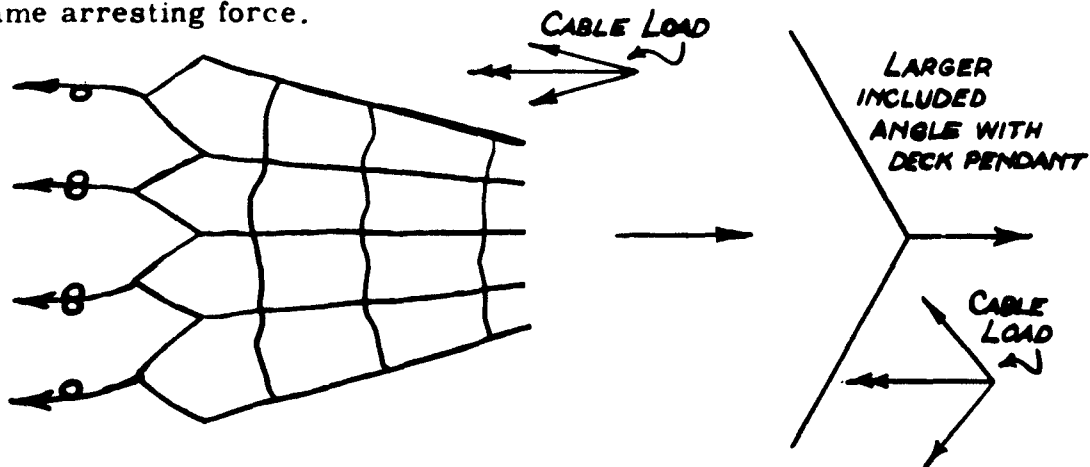
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For example, consider the net of the following dimensions



The transverse members and the side members should have approximately the same strength as present cross deck pendants while the fore and aft members could be considerably weaker. To prevent abrasion of the transverse members, due to frequent wheel and hook contact, they would require to be covered by a hard but flexible sheath, whereas the axial members could be of nylon or some strong fibre. If the transverse and side members are $1\frac{3}{8}$ " wire rope their weight will be 1000 lbs. but doubtless a lighter material could be found.

One advantage that this system has over the cross deck pendant system is that smaller cable loads are required for the same arresting force.



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A further advantage is that the arresting engines coupled to the net can be made of small capacity. When handling a light aircraft two engines could be used and others cut in (in pairs) to accomodate heavier aircraft. This point brings up one serious objection to conventional carrier installations. At present a carrier may contain as many as nine engines, each capable of arresting the heaviest aircraft the ship can handle, yet during each arrest only one of these engines is used. A more desirable situation is to have a small number of engines all of which are active during arrest. Not only is the total arresting capacity more effectively used by this method, but each engine can be quite small.

It will be shown later that the inertia of an arresting gear is proportional to its maximum energy capacity, so that if a gear of large capacity is used to arrest a light aircraft, unduly high inertia loads will be applied to the plane. The obvious advantage of multiple units of small capacity is that they may be grouped to provide the arresting capacity demanded during any landing and the inertia loads will be matched to the size of the aircraft being taken aboard.

Limitation of the Cross -Deck Pendant

The development of arresting gears designed to accept heavy aircraft at high speeds has pointed up a need for an increase in the performance of cross-deck pendants and considerable attention has recently been given to the dynamics of this element. Among these should be mentioned the report by F. O. Ringleb "Dynamics

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of a Moving Cable" and a more recent British report by J. Thomlinson "Behavior of Ropes under Longitudinal and Transverse Impact". Due to the complexity of the subject, this study does not include independent work on cable dynamics but rather briefly summarizes the general conclusions and applies them to the problem of landing planes on carriers at increasingly higher speeds.

Cross-deck pendants are required for any presently envisioned system of arresting aircraft. Under present landing speeds, the attrition rate of these cables is fairly high, due mainly to a burning of the cables by the sliding of the arresting gear hook. For this reason, they are made detachable from the remainder of the cable reeving system through the use of connecting shackles at the deck edge. Furthermore, it is noted that the cross deck pendants, although originally straight, become twisted after use. This would seem to indicate that the bending stress in the cable where it passes over the airplane arresting gear hook exceeds the yield point of the material. Neither of these effects appears to limit the potentiality of the arresting gear pendant, the excess energy available between the yield point and the ultimate strength of the cable gives a margin of safety between obvious indications of a defective cable (such as broken wires and elongation of the cable) and the possibility of cable breakage.

The major limitation upon the cross-deck pendant is the stress induced in the cable by the transverse impact of the plane. Since the cable is accelerated from rest to plane speed

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almost instantaneously, severe impact stresses arise which are distributed through the whole cable system but are particularly severe in the pendant. Studies have shown that, at the moment of impact, an extensional wave is generated and travels along the pendant with a characteristic velocity, to be followed at a lower velocity by a "kink" or triangular shaped displacement of the cable. The point of impact (the apex of the triangle) moves forward with the aircraft and the base of the triangle lengthens as the kink moves toward the deck edge. The extensional stress which is set up is a function of the plane speed, the speed of sound in the cable, and the mass per unit length of the cable. There is a limiting velocity of the plane corresponding to which any given ideally supported cable will be subjected to a stress in excess of its strength and therefore will fail. In practical arresting gears, it is found that the stress waves set up by the plane are reflected back and forth through the cable from the shackles, sheaves and tie points, so that the maximum stress in the deck pendant is frequently three or four times the initial impact tension. The complexity of the system has made analytical evaluation of the stresses very difficult and the high velocity of the tension waves has made the experimental approach equally difficult.

For the purpose of setting theoretical limits upon the maximum speed at which arrestation may take place, it will be of value to consider some information following a table prepared by Thomlinson, indicating the critical velocities for transverse impact on a cable having no initial tension. The values in the first column are those

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required to give a stress in the cable of half the breaking strength; i. e. approximately the yield strength of the cable. Those in the second column represent the speeds which might be accepted by good present arresting gear having an augmentation of stress to three times that of the ideal free cable.

TYPE OF ROPE	CRITICAL TRANSVERSE IMPACT VELOCITY (knots)	CRITICAL TRANSVERSE IMPACT VELOCITY (knots)
	<u>($\sigma = 1/2$ T.S.)</u>	<u>($\sigma = 1/2$ T.S.)</u>
		Augmentation due to Reflection = 3
Steel Cable 1 1/16" dia.	250	110
Nylon	600	264

It is apparent that the critical transverse velocity for the augmented stress is very close to the present landing velocities. Since it will be necessary to increase the allowable landing velocity of planes if the optimum possible gains in plane size and performance are to be obtained, it becomes of interest to see where this improvement might be expected. One line of attack would be to lessen the magnitude of the reflected waves. This might involve a different type of arresting gear and special attention to the design or elimination of shackles and sheaves. Another solution would be to use ropes having a high strength-to-weight ratio and low modulus of elasticity. Nylon has these characteristics to a remarkable degree as is indicated in the table. Whether it has the other properties desired of a cross deck pendant such as abrasion resistance and small size is open to question.

The ill-effects of impact can be countered to some extent

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by setting the cable in motion prior to its contact with the aircraft, thereby reducing the relative impact velocity. Such a scheme would involve some device to eject the cable and a triggering system actuated by the approaching aircraft.

If the aircraft design is to make use of the advantages accruing from a high stalling speed, some solution to the impact problem must be found.

CONFIDENTIALCONVERTERS

It has been pointed out that the most attractive method of dissipating the energy is by thermal means either to a liquid sink or to a gaseous (atmospheric) sink. The question of the form of mechanism necessary to convert the aircraft's K.E. to thermal energy now comes up for consideration. Several schemes have been proposed and each of these will be considered in some detail. It would be inadvisable to embark on a detailed design of each system so that only the approximate size and characteristics of each device can be investigated here.

A desirable property of the converter is that it offer a constant or nearly constant resistance during the period in which it acts. Any system which is sensitive to either the velocity of the aircraft or its runout will not inherently possess this property since the velocity decreases from a maximum to zero and the runout varies from zero to a maximum. Examples of these two types of resistance are (a) viscous shearing forces in a liquid and (b) the resistance of a compressed spring. The possibility of compounding these two types of resistance appears attractive and will be discussed in this section.

When considering the properties of various devices, "standard" arrest conditions will be chosen as follows:

Weight of aircraft = 70,000 lbs.

Relative Entry velocity = 170 feet/second

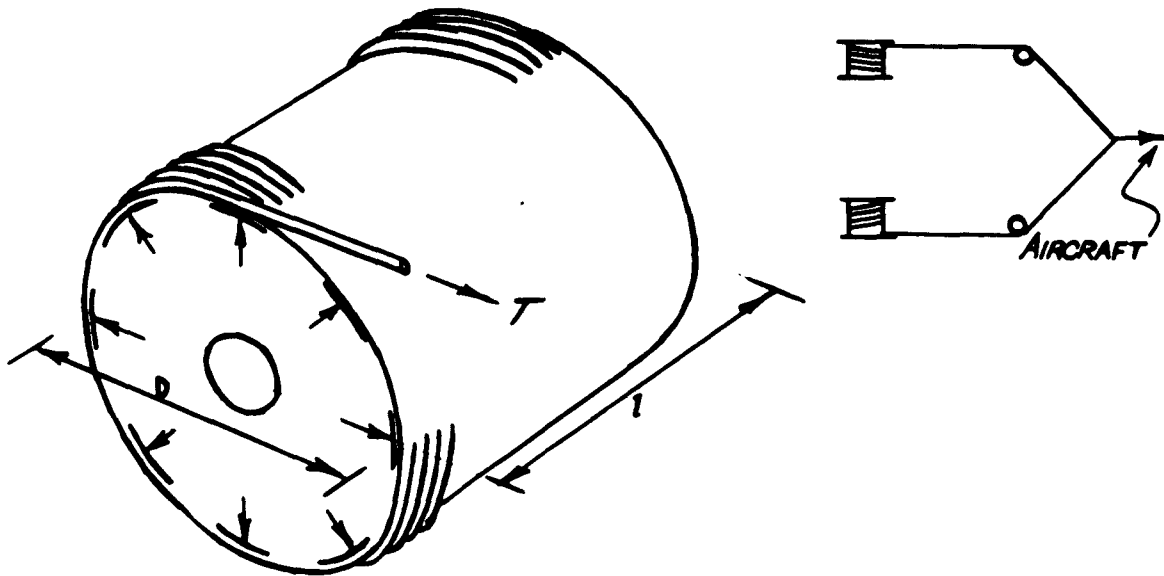
Arresting forces for 3g arrest ~~is~~ 200,000 lbs.

Energy released on arrest ~~is~~ 32×10^6 ft. lb. = 45,400 B.T.U.

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CONFIDENTIALROTARY FRICTION DEVICE

The mechanism consists of a hollow cylinder (around which the cable is wrapped) having friction pads bearing against its inner surface. As envisaged, these devices would work in pairs, one attached to each end of the cable as follows:



If L = total length of cable to be paid out ins.

d = diameter of cable used ins.

T = cable tension lb.

then $T = \frac{\pi}{4} d^2 \sigma_c$ where σ_c = limiting stress

assumed constant for all cables.

The energy capacity of the device is $TL = \frac{\pi}{4} d^2 \sigma_c L = H \text{ lb.ins.}$ $\frac{\pi}{4} d^2 L$

is the volume of the cable = $\frac{W}{\rho}$ if W is the total weight of cable and

ρ is the cable density (assumed constant for all cables) lb./in.³.

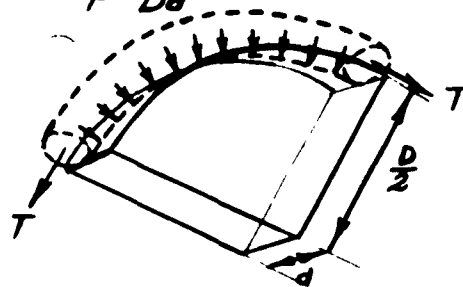
The energy absorbed is thus $\frac{\sigma_c}{\rho} W$

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So that the weight of cable is proportional to the energy capacity of the device.

Assuming that the drum will generally fail in buckling, the maximum external pressure it can withstand is $\frac{2EA^3}{(1-\nu^2)D^3}$ (Timoshenko Vol. II pg 219)

The pressure corresponding to the cable tension is p where $p \cdot \frac{D}{2} d = T$ or $p = \frac{2T}{Dd}$



if this is the critical pressure

$$\frac{2T}{Dd} = \frac{2EA^3}{(1-\nu^2)D^3}$$

$$\text{or } h^3 = \frac{2T(1-\nu^2)D^3}{Dd2E} = \frac{T}{E} \cdot \frac{D^2}{d} (1-\nu^2)$$

For steel cable, the manufacturers advise that the cable should not be bent to a diameter less than 18-25 times the cable diameter.

Taking an average figure, we can say $D = 20d$

$$\text{so that } h^3 = \frac{T}{E} 400 (1-\nu^2) d$$

$$\text{introducing the value of } T ; h^3 = \frac{\pi}{4} \cdot \frac{\sigma_c}{E} \cdot 400 (1-\nu^2) d^3$$

$$\text{or } h = d \sqrt[3]{\frac{\pi}{4} \cdot \frac{\sigma_c}{E} \cdot 400 (1-\nu^2)}$$

Assuming the values $E = 30 \times 10^6 \text{ lb/in}^2$, $\nu = \frac{1}{3}$, $\sigma_c = 90,000 \text{ lb/in}^2$

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we get $h = 0.94d$

Since $D = 20d$ the drum circumference = $\pi \cdot 20d$ ins.

the number of coils on the drum is then $\frac{L}{20\pi d}$

the length of the drum is then $d \cdot \frac{L}{20\pi d} = \frac{L}{20\pi}$

The weight of the drum is now $\pi D \times h \times \frac{L}{20\pi} \times \eta$ (where η = density lb/in.³)

$$\begin{aligned} &= 20\pi d(0.94d) \frac{L}{20\pi} \cdot \eta \\ &= 0.94\eta L d^2 \text{ lbs.} \end{aligned}$$

$$\begin{aligned} \text{Total weight of moving parts} &= 0.94\eta L d^2 + \frac{\pi}{4} d^2 L \rho \\ &= L d^2 (0.94\eta + 0.78\rho) \text{ lbs.} \end{aligned}$$

It was shown earlier that the energy capacity of the device was

$$H = \frac{\pi}{4} L d^2 \sigma_c \quad \text{so that } L d^2 = \frac{4H}{\pi \sigma_c}$$

In terms of energy capacity then the weight of moving parts is

$$\frac{4H}{\pi \sigma_c} (0.94\eta + 0.78\rho)$$

or the inertia weight per unit of energy capacity is

$$\frac{4}{\pi \sigma_c} (0.94\eta + 0.78\rho)$$

or introducing the values

$$\begin{aligned} \sigma_c &= 90,000 \text{ psi} \\ \eta &= .283 \text{ lb/in.}^3 \text{ if steel or } .096 \text{ lb/in.}^3 \text{ if aluminum} \\ \rho &= .154 \text{ lb/in.}^3 \left\{ \begin{array}{l} 8 \times 19 \text{ extra flexible hoisting} \\ \text{rope} \end{array} \right\} \end{aligned}$$

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"specific" inertia weight = 5.47×10^{-6} lb./in.lb. or approximately
 66×10^{-6} lb./ft.lb.

Considering the Stationary Friction Surfaces

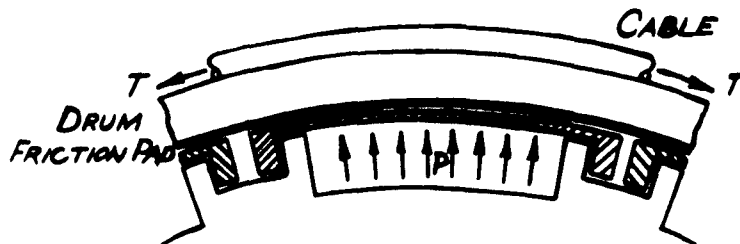
For purposes of heat transfer from the rubbing surfaces, it is essential that the friction pads be as thin as possible while having sufficient mechanical strength to withstand the friction loads.

If the total area of friction pads equals the area of the inner surface of the drum = $\pi D l$

and if the loading pressure is P then $\pi D l P \mu$ = friction force = T

$$\begin{aligned} \text{the cable tension} &= \frac{\pi d^2 \sigma_c}{4} \\ \therefore P &= \frac{\frac{\pi d^2 \sigma_c}{4}}{\pi D l \mu} = \frac{\pi}{4} \cdot \frac{d}{L} \cdot \frac{\sigma_c}{\mu} \end{aligned}$$

Assume the pads to be loaded critically in tension



If S = number of segments of thickness t and σ is the maximum

$$\begin{aligned} \text{tension allowable, } t \sigma S &= T \\ \text{or } t &= \frac{T}{\sigma S} = \frac{\frac{\pi d^2 \sigma_c}{4}}{\sigma S} \end{aligned}$$

if $\sigma = \sigma_c$ i.e. same critical tension for cable as for pad material

$$t = \frac{\pi}{4} \cdot \frac{d^2}{S}$$

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The value of the system will depend on its ability to dissipate heat to the cooling water. If the heat cannot be dissipated rapidly enough, the temperature of the friction surfaces will rise to unduly high values causing distortion and loss of mechanical properties of the various members.

The rate of conversion of kinetic energy to heat is TV where T is the cable load and V the cable velocity (fps). The rate of dissipation through the loading pad will be $\frac{ka}{t} \Delta\theta$ B.T.U./sec.

where $\Delta\theta$ = temperature difference across pad thickness

a = total area of pads sq. ft.

t = pad thickness

k = conductivity of metal in pads B.T.U./sq.ft./ft./°F/sec.

If all aircraft are arrested at the same g loading $[\propto g]$ the arresting force will be $\propto W$ where W = aircraft weight

$$\Delta\theta = \frac{TVt}{Jka} = \propto WV \cdot \frac{\pi d^2}{4} \cdot \frac{1}{15} \cdot \frac{1}{k} \cdot \frac{1}{20\pi l d} \quad [J = 778 \text{ ft.lb./B.T.U.}]$$

$$= \frac{\pi}{80} WV \frac{d}{l k}$$

We have seen that $TL = H$ = total energy capacity

$$L = \frac{H}{T} \text{ or } 20\pi l = \frac{H}{\propto W} \text{ or } l = \frac{H}{20\pi \propto W}$$

also, since $\frac{\pi d^2}{4} \sigma_0 = T = \propto W$; $d = 2 \left(\frac{\propto W}{\pi \sigma_0} \right)^{\frac{1}{2}}$

If a single device absorbs all the kinetic energy of the aircraft $H = \frac{WV^2}{2g}$

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Substituting $\Delta\theta = 40\pi^{\frac{1}{2}} \cdot \frac{g^{\frac{1}{2}}}{\sigma_c^{\frac{1}{2}}} \cdot \frac{1}{sk} \cdot \frac{W^{\frac{3}{2}}}{V^{\frac{3}{2}}} \propto \frac{1}{V^{\frac{3}{2}}}$

taking $\sigma_c = 90,000 \text{ psi} = 1295 \times 10^6 / 16 \text{ lb/in}^2$

$$\Delta\theta = 64.4 \frac{1}{sk} \cdot \frac{W^{\frac{3}{2}}}{V^{\frac{3}{2}}} \cdot \alpha^{\frac{1}{2}} \quad F = 3000 \frac{1}{sk} \cdot \frac{W^{\frac{3}{2}}}{V^{\frac{3}{2}}}$$

when $\alpha = 3$

At first glance it might appear strange that the temperature difference decreases with increasing velocity, if the aircraft weight is kept constant. However, this can be explained as follows.

If the aircraft weight is constant, this implies a constant arrest force for all entry speeds, which, in turn, implies a constant cable diameter. Since the energy to be absorbed increases as the square of the velocity, the cable runout (length) must increase as the square of the velocity which results in the friction area increasing with V^2 . The thickness of the pad decreases as the pad length l increases, i.e. decreases with V^2 .

thus in the expression $\Delta\theta = \frac{T}{k} V \left(\frac{t}{s} \right) \quad \left(\frac{t}{s} \right) \propto \frac{1}{V^2}$

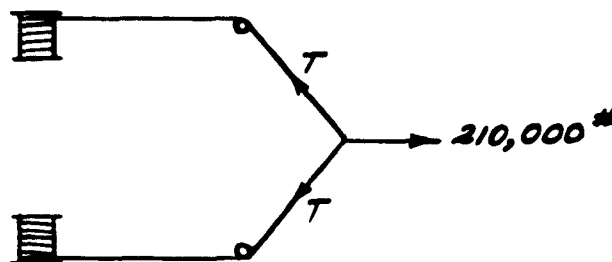
so that $\Delta\theta \propto \frac{T}{k} \cdot \frac{1}{V^2}$

The temperature rise in a friction device designed for low weight, high-speed aircraft would be smaller than that designed for high-weight, low-speed planes - the energy capacity being constant. However, since the inertia weight would be constant and accelerations would be larger for the high speed aircraft, inertia

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effects would be greatest with the high speed planes.

Consider the major dimensions of a rotary friction device designed to accept a 70,000 lb. aircraft at an entry speed of 170 fps ($\approx$ 100 kts) and carry out arrest at 3g. If the cross deck pendant system is used, a pair of absorbers must accompany each pendant as shown.



If T is chosen to be 150,000 # this will give a mean arrest force of 210,000 #

Total energy capacity per machine	$= 15 \times 10^6$ ft. lbs.
Cable size	$= 1 \frac{1}{2}$ " nominal dia.
Cable length (wrapped)	$= 100$ feet
Drum diameter	$= 28$ inches
Drum length	$= 19$ inches
Drum thickness	$= 1.41$ inches
Inertia weight per machine	$= 990$ lbs.
Total inertia neglecting deck pendant	$= 1980$ lbs.
No. of friction pads arbitrarily chosen as	$= 12$
Tangential length of each pad	$= 7$ "
Axial length of each pad	$= 19$ inches
Radial thickness of pad	$= .008$
Loading pressure	$= 450$ psi

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(It would probably not be feasible to have the pad thickness less than .05" so this figure will be chosen)

Maximum temperature difference across pad $\Delta\theta = \frac{TVt}{Jk\delta} = \frac{150,000 \times .004 \times 170}{\frac{1}{144} \times 11.08 \times 778} \text{ } ^\circ\text{F}$

taking k for steel = $\frac{1}{144}$ BTU/sec. /sq. ft. /ft. thickness / $^\circ\text{F}$

$$\Delta\theta = 1700^\circ\text{F}$$

Fluid pressure to give desired friction force = 450 psi (μ assumed = .2)

COOLING SYSTEM

The device may be cooled by circulating water behind the friction pads, the water pressure being used to load the pads.

If a temperature rise of 10°F is allowed in the cooling water, the total quantity of water to absorb 15×10^6 ft. lbs. = 1930 lb. or 31 cu. ft.

If the flow area is 150 sq. in.; i.e. allowing the cooling ducts behind the friction pads to have a radial width of approximately 2" then

$$(\text{velocity of cooling water}) \times (\text{arrest time}) \times \frac{150}{144} = 31 \text{ cu. ft.}$$

The arrest time will be approximately 2 seconds so that the velocity of the cooling water will be $\frac{144 \times 31}{150 \times 2} = 15 \text{ fps}$

Taking the mean temperature of the cooling water as 80°F , the heat transfer coefficient between pad and coolant at 15 fps will be 2000 BTU/hr / ft.² / $^\circ\text{F}$. If the effective heat transfer area is 75% of friction

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surface, i.e. = 8.7 sq. ft., mean temperature difference between

$$\text{coolant and pads} = \frac{15 \times 10^6}{2 \times 778} \times \frac{3600}{2000 \times 8.7} = 2000^\circ\text{F}$$

This points up the basic difficulty with any friction device - namely the inability to provide large heat transfer coefficients while maintaining low temperatures and reasonable surface areas.

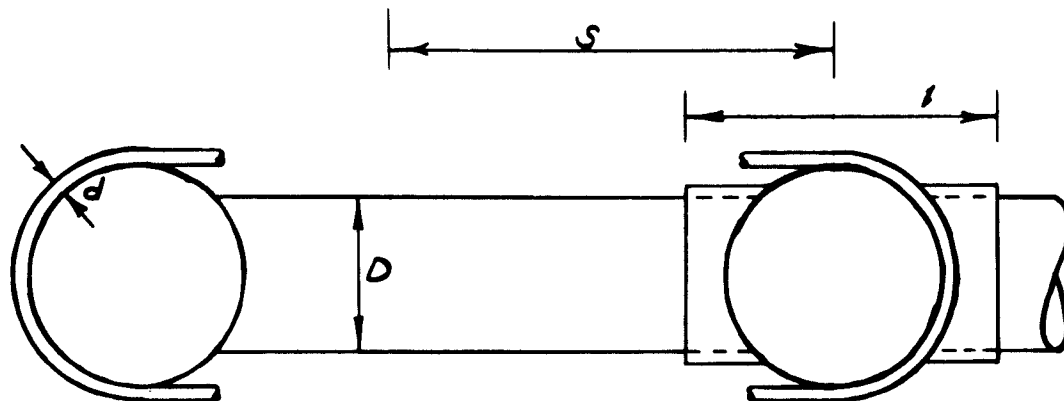
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CONFIDENTIALLINEAR FRICTION DEVICE

This is envisaged as a long stationary tube of circular or rectangular cross-section having a series of friction pads pressed against its inner or outer surface. The advantage of such a device is that it provides a larger heat transfer surface. In the rotary device, each element of heat transfer surface is acting all during the arrest whereas with the linear system each element works during a part of the arrest only.

The moving member may be coupled directly to the aircraft, in which case the stroke of the engine is equal to the runout of the plane, or, the engine may be reeved to provide a short stroke.

Considering a cable tension T produced by a cable of diameter d and stress σ_c operating an engine whose configuration is as shown



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Reeving ratio = R

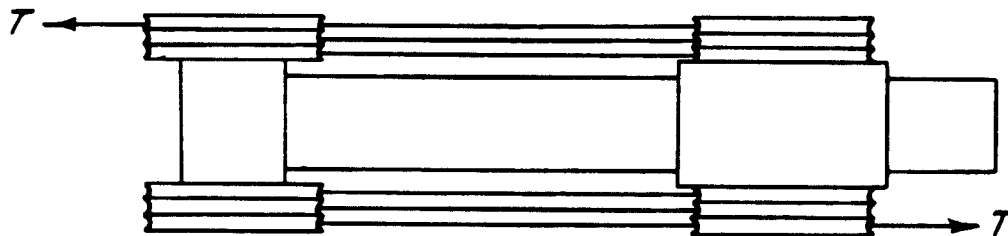
Length of friction pad = l

Stroke = S

Diameter of friction tube = D

Coefficient of friction = μ

If the device is reeved so that one engine is used per deck pendant,



Number of pulleys per 'side' = $\frac{R}{2}$ stationary and $\frac{R}{2}$ moving.

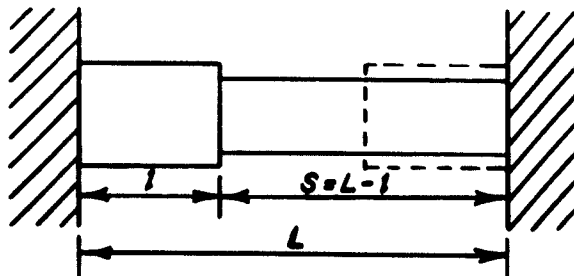
Friction force per 'side' = RT giving total friction force in engine

$$= 2RT$$

For a stroke length S a length of cable RS is paid out per 'side' or a total length per engine of $2RS$ so that the energy capacity of the engine is = $2RST = H$

If the pressure loading between the friction surfaces is P the energy converted to heat is $\pi D l P \mu S = 2RST = 2RS \frac{\pi}{4} \sigma_c d^2$

Consider an engine having a fixed overall length L



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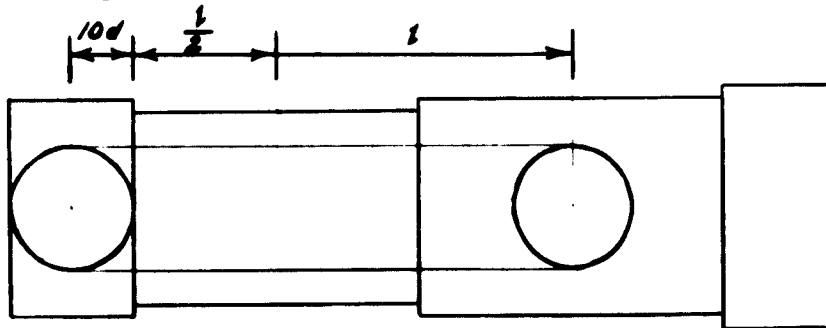
If the length of the pad is l , the length of the stroke S is $L-l$;
if l be increased, the stroke must decrease to maintain the same
overall length L .

$$\begin{aligned}\text{The energy dissipated per stroke is } \pi D l \rho \mu S \\ = \pi D \rho \mu (lL - l^2)\end{aligned}$$

differentiate this w.r. to l and equate to zero for a maximum
 $L = 2l$ thus $S = 2l - l = l$.

Maximum energy absorption is obtained when $S = l$.

The energy absorbed thus becomes $\pi D \rho \mu l^2$
and the configuration is

INERTIA WEIGHTS

$$\text{Length of cable per side} = \frac{R}{2} \left[2\pi(10d) + 2\left(10d + \frac{l}{2}\right) + 2l \right]$$

If the rotary inertia effects of a pulley are approximated by a length
of cable equal to the perimeter of the pulley, the effective length of
cable becomes,

$$\begin{aligned}\frac{R}{2} \left[2\pi(10d) + 2\left(10d + \frac{l}{2}\right) + 2l \right] + R(20\pi d) \\ = R \left[d(30\pi + 10) + \frac{3}{2}l \right] \text{ per "side."}\end{aligned}$$

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The average acceleration over the whole length of the cable is approximately half the acceleration at the free end (actually $\frac{n+1}{n} \cdot \frac{1}{2}$). Thus the equivalent inertia length referred to the free end is

$$\frac{R}{2} [d(30\pi + 10) + \frac{3}{2} l]$$

$$\text{or weight} = \frac{\pi}{4} d^2 \frac{R}{2} [d(30\pi + 10) + \frac{3}{2} l] \rho$$

$$\text{effective weight for two sides} = \frac{\pi}{4} d^2 R \rho [d(30\pi + 10) + \frac{3}{2} l]$$

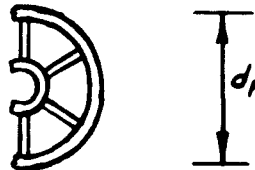
$$\text{Thickness of friction pads} = \frac{PD}{2\sigma}$$

$$\text{weight} = \pi D l \frac{PD}{2\sigma} \eta = \pi D^2 \frac{lP}{2\sigma} \eta$$

WEIGHT OF PULLEY

Consider a series of geometrically similar pulleys of the

general shape

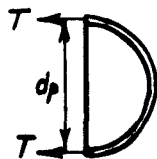


The strength of the rim can be considered as a circular arc in buckling and the spokes as buckling columns.

For the rim, the critical pressure is $\frac{8EI}{d_p^3} \left(\frac{4\pi^2}{\alpha^2} - 1 \right)$ (Timoshenko VII)

Since I varies as d_p^4 the strength varies as d_p . For the spoke, the buckling load varies as $\frac{1}{(\frac{d_p}{2})^2}$ i.e. as d_p^2

The size of the pulley controls the size of cable which can be

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accommodated, viz;

thus the radial pressure acting on the

rim is $\frac{2T}{d_p}$; in terms of the cable diameter T varies as d^2 and d_p as d so that the applied pressure varies as d . The load on the spoke varies as T ; i.e., as d^2 . Thus the strength and the load vary in the same way for both rim and spoke, so that, if a pulley is scaled up geometrically, its strength is always equal to the applied load. Since the weight is proportional to the cube of the linear dimension, we can say that the weight of the pulley is $= \gamma d^3$ where γ is a constant and d is the cable diameter.

The weight of the moving crosshead is then $\pi D^2 \frac{1P}{2\sigma} \eta$ + $R\gamma d^3$. The inertia weight referred to the free end of the cable is

$$\frac{1}{R^2} (\pi D^2 \frac{1P}{2\sigma} \eta + R\gamma d^3)$$

so that the total (two sets of cables and pulleys plus one crosshead)

$$\text{referred inertia weight is } \frac{\pi}{4} d^2 P R [d(30\pi + 10) + \frac{3}{2} l] + \frac{1}{R^2} (\pi D^2 \frac{1P}{2\sigma} \eta + R\gamma d^3)$$

$$\text{Energy capacity is } \pi D l^2 P \mu = 2 R l \frac{\pi}{4} \sigma_c d^2$$

Inertia weight/unit of energy capacity =

$$\frac{P}{2l\sigma_c} [d(30\pi + 10) + \frac{3}{2} l] + \frac{1}{R^2} [\frac{P}{l} \cdot \frac{\eta}{2\sigma\mu} + \frac{d}{l} \cdot \frac{2\gamma}{\pi\sigma_c}]$$

For a given arrest condition, i.e. a specified arrest force and cable runout, the diameter d and the product $Rl = L$ are specified. The diameter D is controlled by the allowable pressure

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between the friction surfaces, the higher being this pressure, the smaller will be the value of D . As the reeving ratio R is increased, the first term becomes larger while the second and third terms decrease.

The effects of reeving ratio on the "specific inertia weight" will be dependent on the respective values of d and D so that no general conclusions can be drawn from an inspection of the relation.

Heat Dissipation

The rate of heat dissipation in the linear friction device will be the same as that in the rotary friction device. However, there is a difference in that most of the heat transfer surface in the linear device only works during part of the arrest time. Surfaces near the ends of the friction tube are heated during a small part of the arrest interval while surfaces in the middle of the tube work all during the interval.

A reasonably realistic evaluation of the time-temperature relation at the friction surfaces for either rotary or linear systems is far beyond the scope of this work, but it may be pointed out that the thermal loads will (on the average) be smaller for the latter device. Consider the major dimensions of a linear device, of energy capacity comparable with that of the rotary mechanism considered earlier and having the same pressure loading.

Total energy capacity of machine	$= 30 \times 10^6$ ft. lbs.
Cable force	$= 150,000$ lbs.

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Cable size	= 1 1/2" nominal dia.
Cable paid out during arrest (per side)	= 100 ft.
Length of friction tube chosen as	= 20 ft.
Length of stroke	= 10 ft.
Reeving ratio	= 10:1
Length of friction pad	= 10 ft.
Dia. of friction tube (if $P = 450$ psi)	= 89 inches
Total inertia weight (effective)	= 940 lbs.

The advantages of reeving show up clearly in the small effective inertia weight even though the equipment itself is quite massive.

Referring to the expression for specific inertia weights, the terms are $1.18 \times 10^{-6} + 1.31 \times 10^{-6} + .058 \times 10^{-6} + .018 \times 10^{-6}$ lbs./in.lb.

The first two terms arise from the rotary inertia of the pulley and from the inertia of the cable; as can be seen, these terms contribute the major part of the inertia loads.

Large cable runouts and pulleys can be eliminated by using a different configuration of the absorbing device - namely an unreeved system. Such a system will now be considered.

UNREEVED LINEAR FRICTION DEVICE

If the friction pad is directly coupled to the aircraft, the length of cable (other than the deck pendant) is virtually zero. Consider two such devices, one on each side of the deck, having an effective stroke of 100 ft. and a resistance of 150,000 lbs. If the same pressure of 450 psi is used

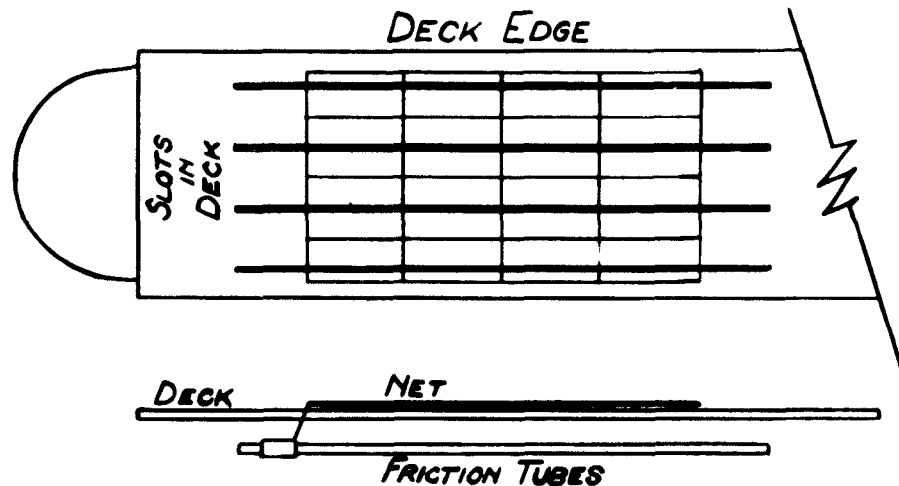
$$\mu \pi D l \times 450 = 15 \times 10^4 \quad \text{where } \mu = 0.2$$

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let $l = 2D$ then $D = 16.3''$

A 16" diameter tube with a 32" long friction pad would have an extremely low inertia.

Consider this system in conjunction with the arresting net proposed earlier.



If 4 friction tubes were used, each could work at a loading pressure of 250 psi having the dimensions 16" diameter and 32" long and would be required to dissipate 8×10^6 ft. lbs. of energy. This corresponds to a heat transfer through the tube of 25 B.T.U./sq. ft.

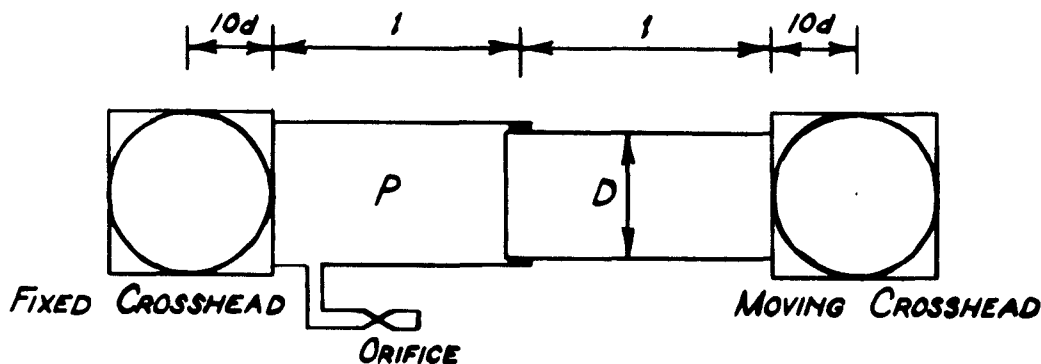
The advantage of multiple units acting simultaneously has been pointed out earlier. The rate of dissipation in each unit can be small and the maximum instantaneous temperature rise at the friction surface is smaller than with a single machine. The scheme has, of course, the advantage that the arrest capacity can be geared to the aircraft being accepted, but has the disadvantage that the inertia weight of the net may be rather large when handling light, high-speed aircraft.

CONFIDENTIALHYDRAULIC DEVICES

It appears that the problems of heat transfer to the fluid sink are a major drawback to the operation of any mechanical friction device. These problems do not arise if the energy can be dissipated by viscous shear forces within the fluid itself. This implies the provision of large velocity gradients within the fluid and the success of any scheme depends on how effectively these gradients can be introduced and on the viscosity of the fluid medium.

In the orthodox hydraulic gear in present use, the velocity gradients are introduced by forcing a fluid through an orifice, (the resistance of the fluid supplies the arresting force.)

Consider an arrangement similar to the reeved linear friction device.



- If
- R = reeving ratio
 - P = fluid pressure
 - d = diameter of cable
 - D = diameter of ram
 - L = length of cable paid out; i.e. $L/2$ per side

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then

$$\frac{\pi}{4} D^3 P = 2RT = 2R \frac{\pi}{4} d^3 \sigma_c$$

and energy capacity

$$\frac{\pi}{4} D^3 P L = 2R L T = \left(\frac{\pi}{4} d^3 \sigma_c \right) 2R L$$

For a given arrest force, if P is chosen D follows automatically providing R is fixed; P and D now being fixed, the length of stroke is decided by the energy capacity. There is no optimum combination of the various parameters since each is fixed more or less independently. For example, the arrest force does not depend on the aircraft's velocity but only on its weight.

For purposes of comparison with other systems, the effective inertia force will be evaluated.

Weight of Ram

Required wall thickness $h = D \sqrt[3]{\frac{P(1-\nu^2)}{2E}}$

weight of tube = $\pi D^2 l \eta \sqrt[3]{\frac{P(1-\nu^2)}{2E}}$ neglecting end closure

weight of moving pulleys = $R \gamma d^3$

effective inertia at aircraft = $\frac{1}{R^2} \left[\pi D^2 l \eta \sqrt[3]{\frac{P(1-\nu^2)}{2E}} + R \gamma d^3 \right]$

wrapped length of cable per side = $\frac{R}{2} [2\pi(10d) + 40d + 4l]$

If rotary inertia of pulley is approximated by a length of cable equal to the perimeter of the pulley, the effective length of cable per side

becomes $\frac{R}{2} [2\pi(10d) + 40d + 4l] + R(20\pi d)$

total length per machine = $R[60\pi d + 40d + 4l]$

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$$\text{Equivalent inertia length at aircraft} = \frac{R}{2} [60\pi d + 40d + 4l]$$

$$\begin{aligned} \text{total referred inertia} = & \frac{1}{R^2} \left[\eta (\pi D^2 l) \sqrt{\frac{\rho(1-\nu^2)}{2E}} + R \gamma d^3 \right] \\ & + \frac{\pi}{4} d^3 \frac{1}{2} [60\pi d + 40d + 4l] \end{aligned}$$

$$\text{Since energy capacity} = \frac{\pi}{4} D^2 P l = 2 R I T = 2 R I \frac{\pi}{4} d^2 \sigma_c$$

the referred inertia per unit of energy capacity is

$$\frac{\eta}{R^2} (0.195 \times 10^{-2}) \left(\frac{4}{\rho^{1/2}} \right) + \frac{\gamma}{\sigma_c} \cdot \frac{4}{\pi} \cdot \frac{d}{R L} + \frac{\rho}{\sigma_c} \left[15\pi \frac{d}{l} + 10 \frac{d}{l} + 1 \right]$$

Referring to the standard arrest conditions which require $d = 1 \frac{1}{2}$ " and a runout $L = 200$ ft. (i.e.; 100 ft. of cable per side), and choosing $R = 10 : 1$ and a limiting pressure of 10,000 psi, the specific inertia becomes

$$.0475 \times 10^{-6} + .018 \times 10^{-6} + 2.92 \times 10^{-6} = 2.98 \times 10^{-6} \text{ lbs.}$$

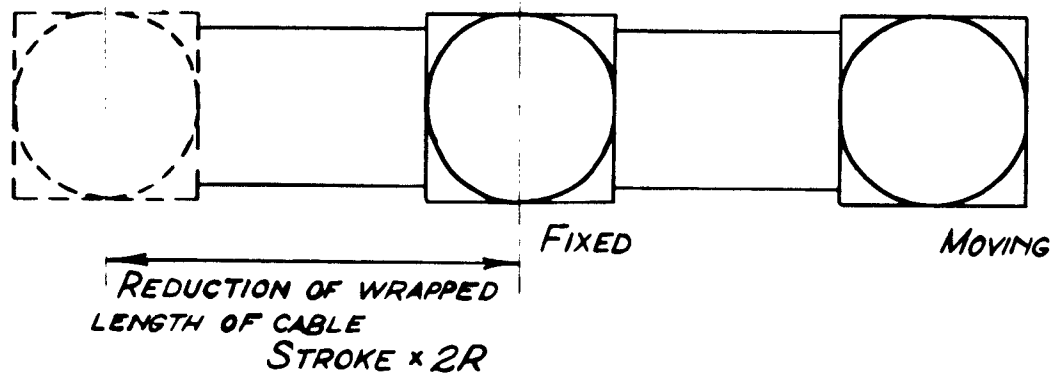
per in. lb. of energy.

Thus the effective inertia weight of an engine capable of absorbing 30×10^6 ft. lb. is 1070 lb.

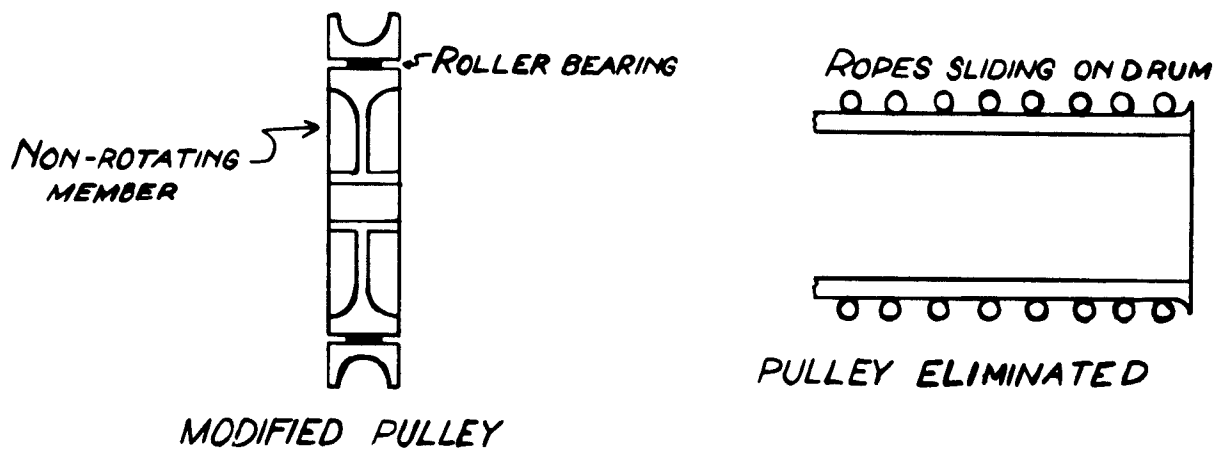
An interesting point arises from the figures given for a specific inertia. In both the friction and the hydraulic devices, the inertia of the cable and pulley made up the bulk of the inertia weight of the system. The contribution of the moving crosshead and the ram (or friction pad) was comparatively small. A major step in the reduction of the effective inertia weight of a reeved system would obviously be to reduce the effective length of wrapped cable and the rotary inertia of the pulleys. In the orthodox hydraulic gear, this can be achieved by moving the fixed crosshead to the open end of

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the cylinder and modifying the pulleys.



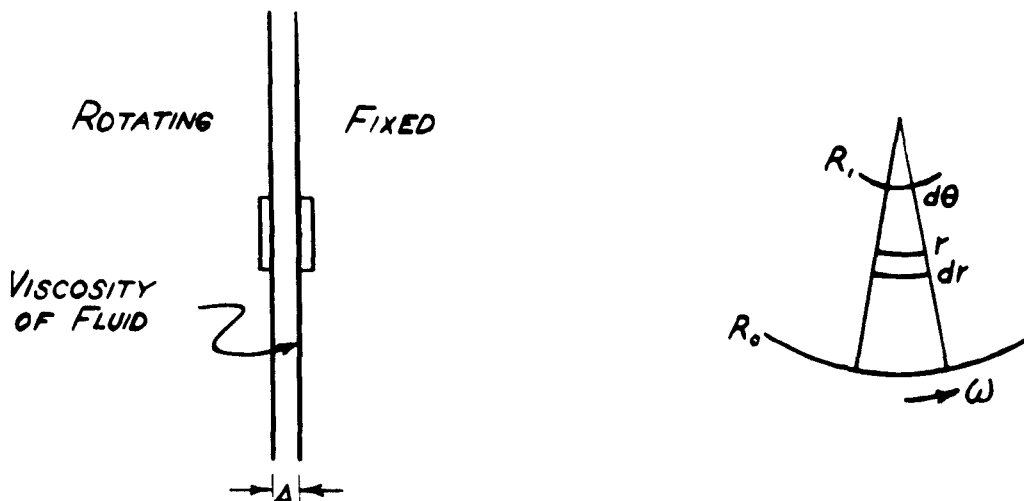
for device considered reduction = 200 feet



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CONFIDENTIALROTARY SHEAR IN FLUID

In the orthodox hydraulic gear, shear stresses are introduced into the fluid by forcing it at a high velocity through an orifice. Velocity gradients may be produced by rotating a disc in a viscous fluid, the resulting drag forces being used to arrest the aircraft. Consider a pair of discs one of which is rotating and the other fixed, the intervening space being filled with a viscous fluid.



Element $r \cdot dr \cdot d\theta$ has velocity gradient $\frac{\omega r}{\Delta}$ so that the fluid friction associated with the element is

$$r \cdot dr \cdot d\theta \frac{\omega r}{\Delta} \mu = \mu \frac{\omega}{\Delta} r^2 \cdot dr \cdot d\theta$$

$$\text{torque} = \mu \cdot \frac{\omega}{\Delta} \int_0^{2\pi} \int_{R_i}^{R_o} r^2 \cdot dr \cdot d\theta = 2\pi\mu \frac{\omega}{\Delta} \cdot \frac{1}{4} (R_o^4 - R_i^4)$$

$$\text{Rate of dissipation of energy} = 2\pi\mu \cdot \frac{\omega^2}{\Delta} \cdot \frac{1}{4} \left(\frac{R_o^4 - R_i^4}{J} \right) \text{ heat units/sec.}$$

This dissipation occurs in a volume $\pi(R_o^2 - R_i^2)\Delta$.

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$$\text{Rate of dissipation per unit volume} = \frac{2\pi\mu \cdot \frac{\omega^2}{\Delta} \cdot \frac{1}{4}(R_o^2 - R_i^2)}{\pi(R_o^2 - R_i^2)\Delta}$$

and if c = specific heat

β = density

$$\text{the rate of temperature rise is } \theta = \frac{\mu}{2} \cdot \left(\frac{\omega}{\Delta}\right)^2 \frac{R_o^2 + R_i^2}{J} \cdot \frac{1}{c\beta}$$

As a specific example, consider the outer radius of the disc to be 1 1/2 ft. and the inner radius 1/2 ft., i.e. such as would fit inside the drum dealt with earlier.

Then, if $c = .5 \text{ BTU/lb. / } ^\circ\text{F}$; $\beta = 56.2 \text{ lb./cu.ft.}$; $\mu = 3$ poises and if the maximum rate of temperature rise is to be limited to 20°F per second:

$$20 = 3/40 \times 10^{-6} \left(\frac{\omega}{\Delta}\right)^2 \frac{2.5}{.5 \times 56.2 \times 778}$$

$$\frac{\omega}{\Delta} = 7460$$

At a maximum rim speed of 170 fps (landing speed of a/c)

$$\omega \approx 120 \text{ rads/sec}$$

$$\text{and } \Delta = 0.0161 \text{ ft.} = 0.193''$$

Number of pairs of discs required to give a resisting force of 150,000 lb. = n

$$\text{torque} = TR_o = n \cdot 2\pi\mu \cdot \frac{\omega}{\Delta} \cdot \frac{1}{4}(R_o^2 - R_i^2)$$

where n = number of pairs of discs

$$\text{i.e. } n = \frac{150,000 \times 1.5 \times 4}{2\pi \times 0.00627 \times 7460 \times 4.987} = 612$$

Some 600 pairs of discs would be required. The values assigned to the variables were based on the following considerations.

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The fluid viscosity figure of 300 centipoises corresponds to a medium oil at room temperature. Since this property of the oil is quite sensitive to temperature, the allowable temperature rise was limited to 20°F in 2 seconds. This corresponds to a maximum rate of temperature rise of 20°F/sec. at the beginning of the arrest falling linearly to zero at the end of the arrest, i.e., after 2 seconds.

The values of density and specific heat are normal for a medium oil.

An earlier part of this work showed that a rotary device should be contained in a drum 3 ft. in diameter and this controlled the choice of R_o and R_i .

The principal objection to such a device is the large number of discs needed to dissipate the energy. The physical size of the drum (3' dia. and approximately 10' long) can be reduced by using a fluid of higher specific heat and/or a smaller variation of viscosity with temperature. Possible fluid media are the silicone oils, with their low temperature coefficient and a specific heat of .6. The use of such a medium effectively increases the amount of energy which can be dissipated in a unit volume of the fluid and further improvement in the absorber can be obtained by increasing the velocity gradients within the fluid. (See section of Turbo Machinery)

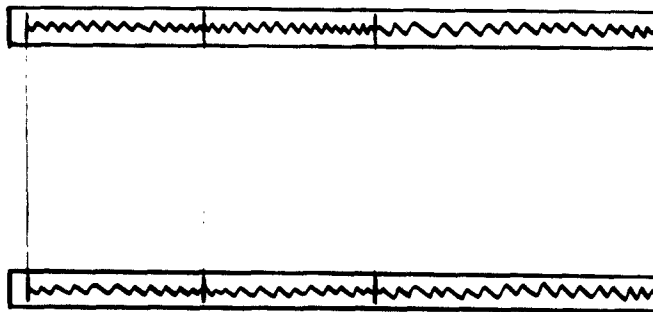
The large inertia associated with a cable and drum arrangement has already been pointed out and for this reason alone the rotary shear device would be unfeasible.

CONFIDENTIALDAMPED SPRING SYSTEM

A system which absorbs energy by viscous action in a fluid will offer a retarding force which is some function of the velocity of actuation. As the velocity of the aircraft falls off, the retarding forces will also fall off.

A simple shock absorber, consisting of a perforated plate moving through a viscous fluid, will offer a high resistance when first engaged by the aircraft and, if it can be so arranged that a second disc will be picked up after a certain distance, the diminishing drag can be restored to an effective value. A system of coupled shock absorbers might be adjusted to give a reasonably constant retarding force and such an arrangement will be studied in this section.

The system, as envisaged, would consist of two long slotted cylinders having several discs or free pistons sliding within. Corresponding discs in each cylinder would be coupled by deck pendants while all the discs in a cylinder would be coupled by axial springs.



When the aircraft enters the gear, one of the discs is given a large impulsive velocity producing a large viscous drag.

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As the aircraft velocity diminishes under the action of the arresting force the viscous drag falls off accordingly, but the compression of the coupling spring alleviates the condition by accelerating another disc and introducing additional drag. When all discs are in motion, the final (storage) spring would be compressed and the decreasing drag forces would be compounded by the mounting spring force.

By proper compounding, i. e. by a correct choice of the damping and spring factors, the variation in arrest force may be kept low but it (the arrest force) can never be maintained constant. The retardation characteristic of such a system will depend on four factors.

- (1) The mass of the aircraft being arrested.
- (2) The damping factor associated with each piston or disc.
- (3) The spring constants of the coupling and storage springs
- (4) The velocity at which the aircraft enters the gear.

The dynamic behavior of the gear may be described by a linear differential equation of order $(n + 1)$ where n is the number of discs, the constants in the equation being functions of the factors listed above. If more than two discs are used, it becomes impossible to obtain the solution of this equation explicitly and the displacement - time relationship must be arrived at by numerical means. To demonstrate the properties of various examples of this gear, numerical solutions have been obtained for a three disc system with different damping factors and spring constants.

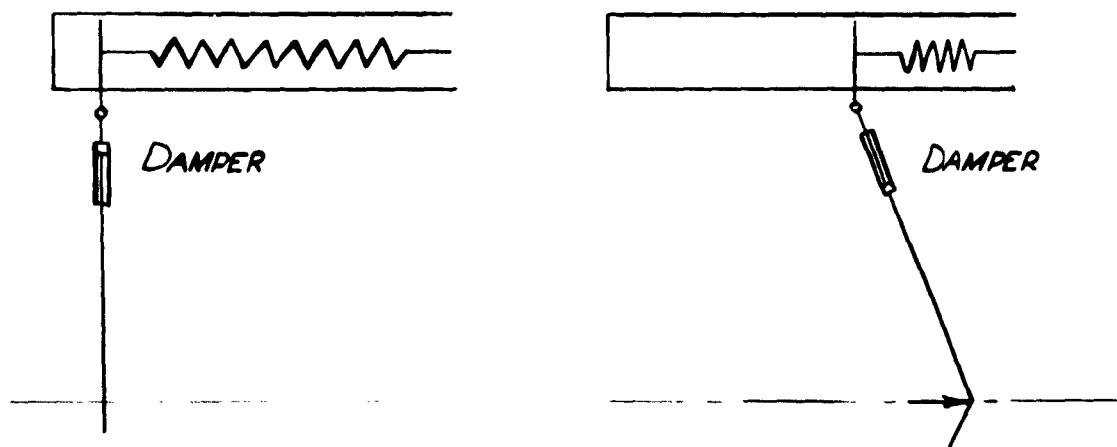
The curves shown do not illustrate the optimum compounding but merely show that an acceptable arrest characteristic can be achieved with this system. These curves refer to an idealized system

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of three weightless discs with pure viscous damping (drag proportional to velocity) the discs being connected by weightless springs. A mass corresponding to half of a 70,000 lb. aircraft impacts the end disc.

The practical realization of this idealized system presents some difficulty. Experience obtained with the slotted cylinder steam catapult should minimize the problems involved by the continuous sealing necessary with this type of arresting gear.

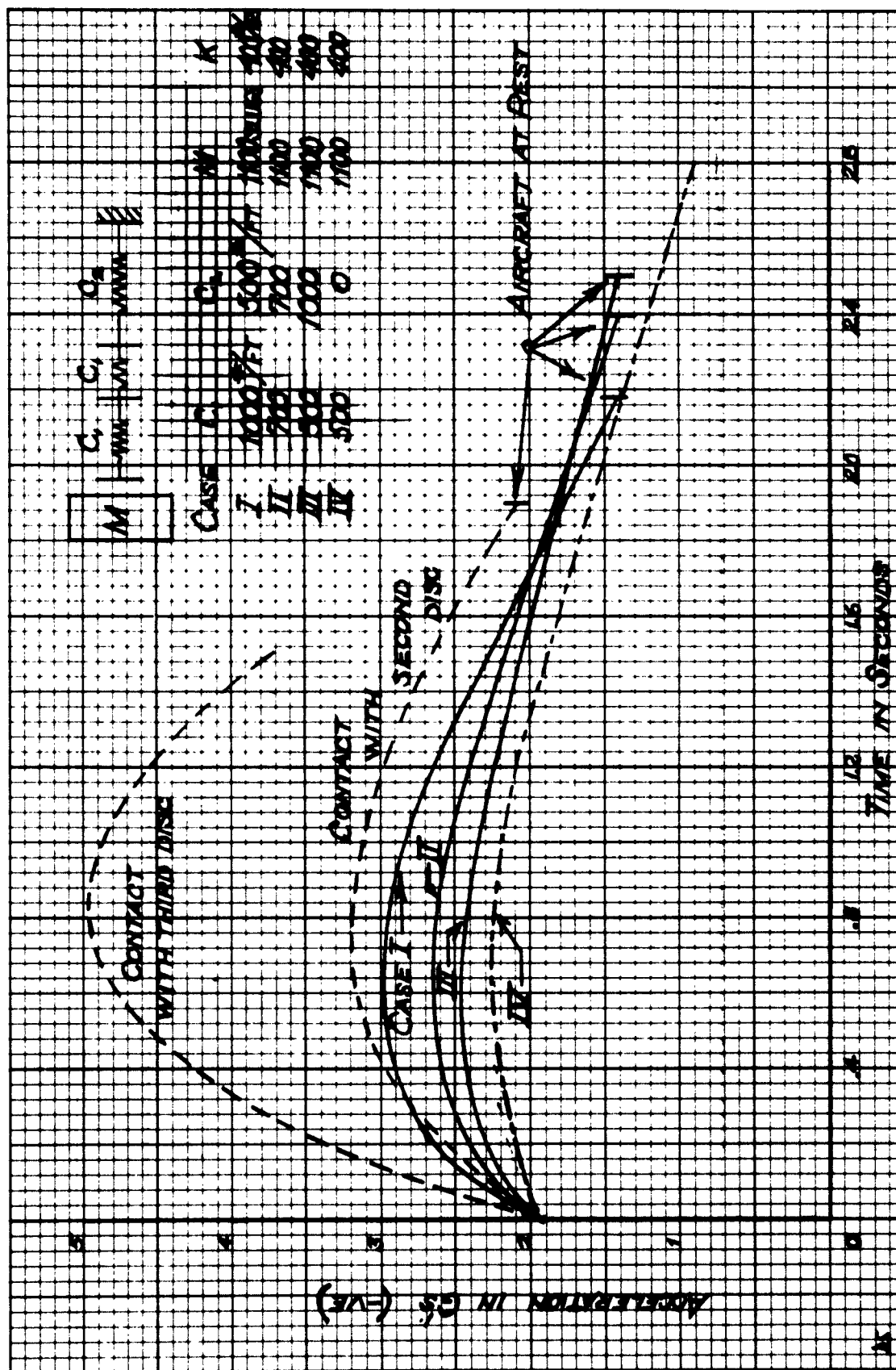
The deck pendant connecting two corresponding discs must not be kept taut but must be capable of extending when picked up by the aircraft. Hydraulic dampers, of the type used experimentally by the British, besides allowing the pendant to extend, would diminish the shock loads arising from high speed entry of heavy aircraft.



The major difficulty would appear to be the provision of coupling and storage springs having the necessary length and stiffness but which will not be too heavy.

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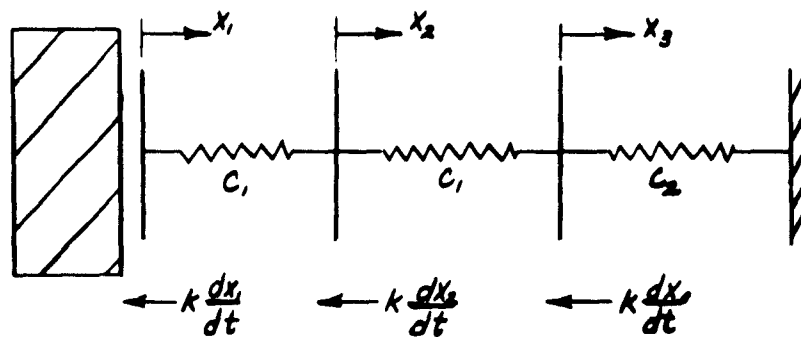
ARREST CHARACTERISTIC OF DAMPED SPRING SYSTEM

Figure 2

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CONFIDENTIALDamped Elastic System

(3 discs and fixed spring)



First disc

$$-c_1(x_1 - x_2) - k \frac{dx_1}{dt} = M \frac{d^2 x_1}{dt^2}$$

Second disc

$$c_1(x_1 - x_2) = c_1(x_2 - x_3) + k \frac{dx_2}{dt}$$

Third disc

$$c_1(x_2 - x_3) = k \frac{dx_3}{dt} + c_2 x_3$$

$$(MD^2 + KD + c_1) x_1 - c_1 x_2 = 0$$

$$c_1 x_1 - (KD + 2c_1) x_2 + c_1 x_3 = 0$$

$$c_1 x_2 - (KD + c_1 + c_2) x_3 = 0$$

Eliminating x_2 and x_3

$$\left[D^4(MK^2) + D^3(3mc_1k + c_2mk + k^3) + D^2(mc_1^2 + 4c_1k^2 + 2Mc_1c_2 + c_2k^2) + D(3c_1^2k + 3c_1c_2k) + c_1^2c_2 \right] x_1 = 0$$

The general solution will be of the form:

$$x_1 = Ae^{-\alpha_1 t} + Be^{-\alpha_2 t} + e^{-\alpha_3 t} (S \cos \omega t + F \sin \omega t)$$

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where $-\alpha_1, -\alpha_2, -\alpha_3 + i\omega, -\alpha_3 - i\omega$ are the roots of a 4th order polynomial with coefficients as in the differential equation above; and A, B, S, F, satisfy the boundary conditions

$$\text{at } t=0 \quad x_1 = x_2 = x_3 = 0, \quad \frac{dx}{dt} = V \quad (\text{the aircraft entry vel.})$$

NOTE: In this analysis M will be half the mass of the aircraft since two duplicate systems will be used side by side.

Numerical solutions have been obtained for the following values of stiffness and damping.

		<u>CASE 1</u>	<u>CASE 2</u>	<u>CASE 3</u>	<u>CASE 4</u>
k	lb/fps	400	400	400	400
c ₁	lb/ft.	1000	700	500	500
c ₂	lb/ft.	500	700	1000	0

In each case, an aircraft of approximately 70,000 lbs. weight (M = 1100 slugs) at an entry velocity of 170 f.p.s. ($\approx$ 100 knots) was considered.

The optimum combination of damping and stiffness cannot be deduced from the general solution but the final example, Case 4, gives an acceptable (g) characteristic.

DIMENSION OF DISCS

If A is the cross sectional area of the absorber cylinder, then a disc moving at velocity V along the cylinder will displace fluid at a rate VA (vol./sec.). If the disc is provided with orifices which permit the transfer of fluid from one side to the other, the

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movement of the disc will be resisted by a force which depends on the disc size, the number and size of the orifices and the viscosity of the fluid. If these factors combine to give a low critical velocity for the orifices, the orifice flow during arrest may be turbulent and the disc resistance will be proportional to the square of the disc velocity. To maintain the condition of resistance proportional to velocity, the orifice flow must be kept laminar.

For laminar flow rate Q through an orifice of length l and diameter d , a pressure difference

$$\Delta P = 128 \frac{\mu l}{\pi d^4} Q \quad \text{is required, } \mu \text{ being the}$$

fluid viscosity.

Since the disc displaces fluid at a rate VA , the flow rate through each of n orifices is $\frac{VA}{n} = Q$

$$\Delta P = 128 \frac{\mu l}{\pi d^4} \cdot \frac{VA}{n}$$

$$\text{the resistance to the disc} = \Delta P \cdot A = 128 \frac{\mu l}{\pi d^4} \cdot \frac{VA^2}{n}$$

$$\text{or resistance per unit velocity} = \Delta P \cdot \frac{A}{V} = 128 \frac{\mu l}{\pi d^4} \cdot \frac{A^2}{n}$$

Let the area of the disc A be 50% greater than the area of orifices

$$\text{then } 1.5 \left(n \frac{\pi}{4} d^2 \right) = A$$

$$\text{and } k = 128 \frac{\mu l}{\pi d^4} \cdot 2.25 n^2 \pi^2 \frac{d^4}{16}$$

$$\text{Since } k \text{ must be } 300 \text{ lb/ft/sec.} \quad \therefore 300 = 18 \pi \mu l n$$

$$\text{The velocity through the orifice is } \frac{VA}{n \frac{\pi}{4} d^2} = 1.5 V$$

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So that the Reynolds number is $1.5Vd \cdot \frac{\rho}{\mu}$

Since the fluid will be required to be of fairly high viscosity yet relatively insensitive to temperature, a silicone oil will be chosen (G. E. type 9996)

$$\mu = .04 \text{ lb. (force) sec. / ft.}^2$$

$$\rho = 1.85 \text{ slugs/cu. ft. (Sp. Gr. = .96)}$$

Allowing for a maximum disc velocity of 200 ft. /sec.

$$R = 13,900 d$$

$$l.n. = 132.5 \text{ ft.}$$

Choosing a Reynolds number of 300, the necessary orifice diameter is .0226 ft. or approximately 1/4".

A length of 1/2 ft. gives 265 holes of 1/4" diameter.

A disc 5" in diameter will be required.

Since laminar flow will not be established until the fluid has traveled some distance along the orifice passages, the discs will require a length somewhat greater than 5".

$$\text{The transition lengths is } .058 R \cdot d \\ \approx 4"$$

so that the first 4" of the orifice passages will not have developed flow. Since the resistance in the undeveloped region will not vary as the velocity, this will tend to upset the direct velocity proportionality that is required. Any attempt to reduce critical length must involve a reduction in the Reynolds number either by decreasing the diameter of the orifice or increasing the fluid viscosity. The former step results in a smaller disc diameter and it must be pointed out that a 5" diameter is already quite small. With regard to increasing

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the fluid viscosity, the value of 1900 centipoises used here is close to the practical limit.

Springs

Spring of stiffness 500 lbs./ft. and 100 lbs./ft. having the compression lengths of approximately 10 ft. and 100 are outside the range of normal mechanical spring practice.

For example, if the coupling springs were made in nests of three having a mean diameter of, say, 6" and a stiffness of 165 lb/ft each, and working to a maximum fiber stress of 100,000 psi, a wire diameter of 5/8" and approximately 100 coils would be required. Each spring could then withstand 2000 lbs. force before failure.

The storage spring is required to withstand a much higher maximum force as well as provide a large compression length.

To provide the maximum thrust, a nest of large springs would be required. For example, if 12 springs, each of stiffness 84 lb./ft. were used, they would require to have a mean diameter of 10" with a wire diameter of 1" to withstand the required load of 4000 lbs. each. Further 200 coils would be required.

An air spring provides a much lighter energy storage device but suffers from the drawback that it is non-linear.

If 'a' is the cross sectional area of the air column which has a length l and a pressure p .

$$p(a l)^n = C \quad \text{a constant}$$

which on differentiation give $\frac{dp}{dl} = \frac{nc}{a^n l^{n+1}} + 1$

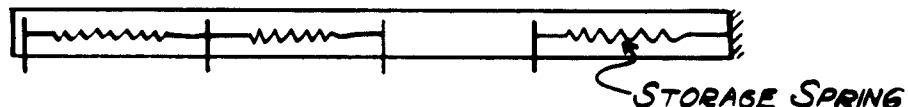
or the "stiffness" which is $-a \frac{dp}{dl} = \frac{nc}{a^{n+1} l^{n+1}}$

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which varies inversely as l^{n+1} As the compressed length l diminishes, the stiffness increases. For isothermal compression $n=1$ and the stiffness is proportional to $\frac{1}{l^2}$, while for adiabatic compression $n=1.4$ and the stiffness $\frac{1}{l^{2.4}}$. Actual conditions will lie somewhere between these two limits.

Since l is required to diminish by $1/2$ or more the stiffness will vary by a factor of at least 4 during compression.

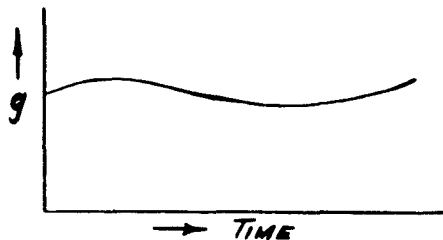
The difficulty of providing a storage spring having the required stiffness and compression length can be eliminated to some extent by using a modified Case 4. The system envisaged consists of three discs with coupling springs, and a storage spring which is not positively coupled to the third disc but is placed at a considerable distance from it.



When the aircraft picks up a disc, the arresting force would vary as shown in Case 4. After an appropriate interval - say $1\frac{1}{2}$ seconds - the whole assembly would have travelled about 170 feet and have slowed to about 75 fps. At this time, the storage spring would be contacted and the residual kinetic energy of the system would be given up to this spring. If the simplifying assumption be made that all discs are travelling at the same speed

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(75 fps) when the storage spring is contacted, a stiffness of 1750 lbs/ft. would bring the aircraft to rest in a further 45-50 feet. The maximum retardation at the end of the stroke would be 2.5 g. The "compounded" arrest curve would now have the general shape:



In general, the damped spring system has a definite advantage in that a single engine is used for all arrests thus ensuring 100% utilization and eliminating the multiple capacity of present systems. Further, the amount of cable (and its associated elastic effect) is greatly reduced.

The main problems to be solved before such a device would be entirely satisfactory are.

1. Development of coupling and storage springs having the necessary characteristics but which are light and compact.
2. Adequate sealing of a slotted cylinder containing several pistons.
3. Provision of a positive locking device to prevent "walk-back" of the aircraft when brought to rest.
4. A simple method of returning the gear to the battery condition after arrest. With the connected storage spring, this problem does not arise, but with the unconnected storage

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spring some device must be provided. A continual circulation of oil through the cylinders would take care of the return stroke.

A completely detailed analysis of this system is outside the scope of this present study but it is felt that the information presented warrants further study.

CONFIDENTIALTURBO-MACHINERY

The term "turbo-machinery" usually implies those rotary machines which impart kinetic or static head to a fluid, this head being used later to do useful work. The application of such machinery to the process of arrestation would involve an exchange of K.E. between the aircraft and the pump, which would then discharge a fluid under a high static and kinetic head. The energy would be dissipated by passing the fluid through a let-down valve and converting the potential and kinetic energies to thermal energy through the action of viscous forces.

However, when considering power levels of the order of 10,000 H. P. and pressure levels not exceeding five or six hundred psi., the necessary pumping rates become very high. This was pointed out earlier when it was shown that a 20,000 lb. aircraft being arrested from 100 kts. in 2 seconds would require a pumping rate of 67,500 gal./min. against a head of 500 psi.

If the quantity of fluid is to be kept to a minimum it is advisable to produce, directly, high rates of momentum transfer within the body of the fluid rather than go through the process of producing a static head and converting this to a turbulent kinetic head. A desirable condition is achieved by operating a fluid coupling with the "driven" member fixed. In such a case, the fluid in its passage through the moving blades attains a high tangential momentum which is destroyed upon entering the fixed blade passages. The momentum transfer between moving and fixed blades produces shearing forces which provide the resisting torque.

A necessary condition for the proper functioning of this

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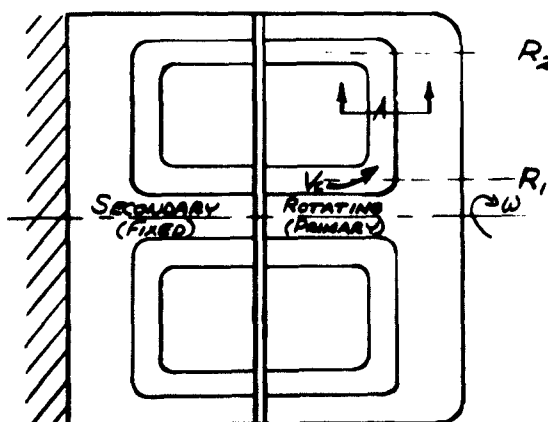
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device is a high circulation velocity of the fluid within the coupling. This would appear to be a drawback when using the device as an arresting engine, since the circulation velocity will be zero initially and will build up gradually as the driving member is actuated by the aircraft hooking a deck pendant. As the aircraft (and consequently the coupling) slows down, the circulation velocity will fall off so that the arresting forces will decrease with decreasing aircraft speed. However, if the coupling or "fluid brake" can reduce the speed to say 10% of the entry speed, the plane can finally be brought to rest by some simple friction device.

The following analysis has been carried out to see if the circulation can be built up rapidly enough and to find the general retardation characteristics of the "fluid brake".

To insure that the fluid in the primary element will have the velocity of the element, it is necessary to have radial blades dividing the circulatory path into segments.

Consider one of these paths:



A = cross sectional area of one circulatory path, constant for any point.

V_c = circulation velocity, a function of time.

I = mass moment of inertia of rotating disc, fluid in disc and aircraft.

ω = angular velocity

n = number of paths

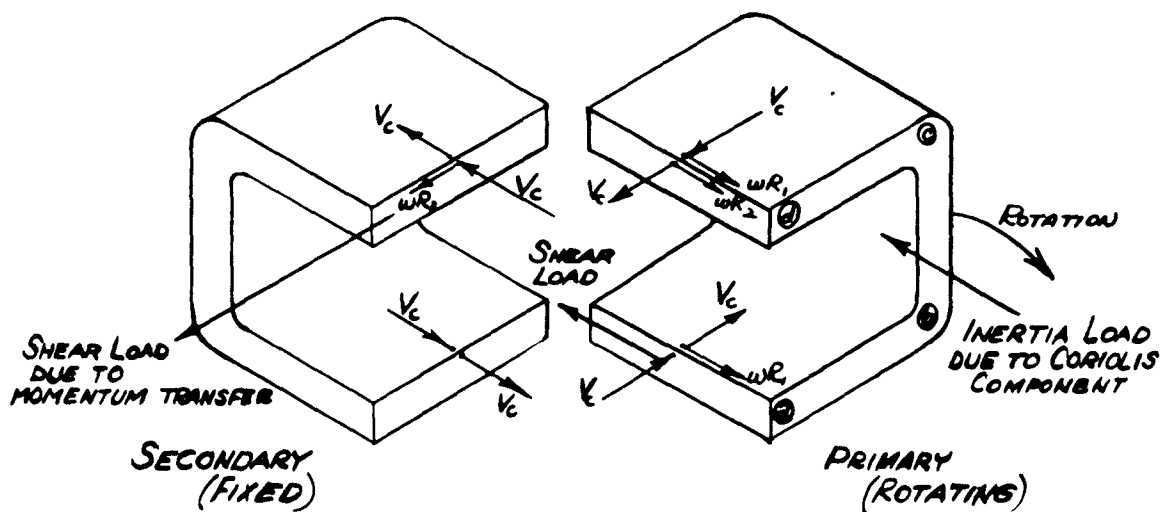
ρ = mass density of fluid

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Assume that the primary element is given an impulsive velocity, that is $\omega = \omega_0$ at time $t = 0$ and the circulation velocity $V_c = 0$ at $t = 0$. Then at any time t , the fluid at entry to the moving element has an axial velocity of V_c and no tangential velocity. Fluid at exit from the moving element has the same axial velocity, V_c , as at entry, but has acquired a tangential velocity, ωR_2 , equal to the rotating disc velocity.

Consider the momentum transfer at inner and outer radii,



For the rotating primary element, the momentum transfer over the flow cross sectional area is equal to the shear load at the inner radius,

$$\text{the shear load} = \rho A V_c (\omega R_1)$$

and the resulting torque for n paths

$$= n \rho A V_c \omega R_1^2$$

For steady state condition ($\omega = \text{constant}$), there is no change in momentum as the fluid moves from point a to b . Moving from b to c , the fluid is accelerated in the tangential direction due to a Coriolis component of $2V_c\omega$. Then, the inertia force due to the fluid between

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b and c .

$$= \int_{R_1}^{R_2} P A dr \cdot 2 V_c \omega$$

and the resulting inertia torque for n paths is

$$n A P V_c \omega (R_2^2 - R_1^2)$$

The fluid is not accelerated in the tangential direction while flowing from c to d , therefore, the total torque developed due to fluid flowing from a to d is

$$n P A V_c \omega R_1^2 + n P A V_c \omega (R_2^2 - R_1^2)$$

that is, the net torque on the rotating primary element is

$$n P A V_c \omega R_2^2$$

For the fixed secondary element, fluid entering the element at the outer radius, R_2 , has a tangential momentum of $\rho \omega R_2$ per unit volume. After the fluid has entered the fixed element, it has lost its tangential momentum, therefore, the momentum transfer per unit volume is $= \rho \omega R_2$

The rate of momentum transfer is equal to the shear load

$$= \rho \omega R_2 \cdot V_c A$$

and the resulting torque on the fixed secondary for n paths is

$$n P A V_c \omega R_2^2$$

As expected, the net torques on the primary and secondary elements are equal.

In a time dt , the primary element rotates through an angle $d\theta$. The work done by the torque, then, is

$$n P A V_c \omega R_2^2 \cdot d\theta$$

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but $\frac{d\theta}{dt} = \omega$

$$\text{Work} = nPAV_c \omega^2 R_2^2 dt$$

The work done by the torque must equal the loss in kinetic energy of the rotating mass in time dt . The rotating mass includes not only the coupling structure but the fluid trapped in the rotating member.

$$\Delta K.E. = -\frac{d}{dt} \left(\frac{\omega^2 I}{2} \right) dt = -I\omega \cdot \frac{d\omega}{dt} \cdot dt$$

equating

$$nPAV_c \omega^2 R_2^2 \cdot dt = -I\omega \frac{d\omega}{dt} \cdot dt$$

$$nPAV_c \omega R_2^2 = -I \frac{d\omega}{dt} \text{ ----- } \textcircled{0}$$

It has been assumed that the primary element is given an impulsive velocity. That is, the circulation velocity $V_c = 0$ at time $t = 0$. The rotational velocity ω and consequently the centrifugal force exerted by the fluid, varies with time. The centrifugal force is the only force available to produce a change in the circulation velocity as well as to overcome the frictional resistance of the fluid flow path.

The force to accelerate the mass, PAI of the fluid is
 $(PAI) \frac{dV_c}{dt}$

The friction resistance is assumed to be a function of the square of the circulation velocity. The friction force, then is
 $= kV_c^2$

where k takes account of the length and area of the flow path and its surface condition.

The centrifugal force developed by the rotating element is

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$$\int_{R_1}^{R_2} \omega^2 R dM = \int_{R_1}^{R_2} \omega^2 R (PA dR) = \frac{PA\omega^2}{2} (R_2^2 - R_1^2)$$

$$\frac{PA\omega^2}{2} (R_2^2 - R_1^2) = PAI \frac{dV_c}{dt} + kV_c^2 \text{-----} (2)$$

Substitution in equation (2) of the value of V_c and $\frac{dV_c}{dt}$ from (1) gives

$$\frac{PA\omega^2}{2} (R_2^2 - R_1^2) = \frac{-II_1}{nR_2^2} \left[\frac{1}{\omega} \cdot \frac{d^2\omega}{dt^2} - \frac{1}{\omega^2} \left(\frac{d\omega}{dt} \right)^2 \right] + k \left(\frac{I_1}{nPA R_2^2} \right)^2 \left[\frac{1}{\omega^2} \left(\frac{d\omega}{dt} \right)^2 \right]$$

--- (3)

Introducing constants a, b, c

$$a\omega^2 = -\frac{b}{\omega} \cdot \frac{d^2\omega}{dt^2} + \frac{b+c}{\omega^2} \left(\frac{d\omega}{dt} \right)^2 \text{-----} (4)$$

$$\text{or } \frac{a}{b} \omega^2 = + \frac{1+c}{\omega^2} \left(\frac{d\omega}{dt} \right)^2 - \frac{1}{\omega} \cdot \frac{d^2\omega}{dt^2} \text{-----} (4)$$

Consider the successive differentiation of ω^{-K}

$$\frac{d}{dt}(\omega^{-K}) = -K\omega^{-K-1} \frac{d\omega}{dt}$$

$$\frac{d^2}{dt^2}(\omega^{-K}) = K(K+1)\omega^{-K-2} \left(\frac{d\omega}{dt} \right)^2 - K\omega^{-K-1} \frac{d^2\omega}{dt^2}$$

$$\therefore \frac{\omega^K}{K} \cdot \frac{d^2}{dt^2}(\omega^{-K}) = \frac{K+1}{\omega^2} \left(\frac{d\omega}{dt} \right)^2 - \frac{1}{\omega} \cdot \frac{d^2\omega}{dt^2}$$

This is identical with the *RHS* of equation (4) if $(K+1) = \frac{c}{b} + 1$

$$\text{i.e. } K = \frac{c}{b}$$

$$\therefore \frac{\omega^K}{K} \cdot \frac{d^2}{dt^2}(\omega^{-K}) = \frac{a}{b} \omega^2$$

$$\text{or } \frac{d^2}{dt^2}(\omega^{-K}) = \frac{a}{b^{\frac{c}{K}}} \omega^{2-K}$$

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Substituting Ω for ω^{-K} and multiplying both sides by $2 \frac{d\Omega}{dt}$

$$2 \left(\frac{d\Omega}{dt} \right) \left(\frac{d^2\Omega}{dt^2} \right) = \frac{2ac}{b^2} \Omega^{1-\frac{1}{K}} \frac{d\Omega}{dt}$$

$$\frac{d}{dt} \left[\left(\frac{d\Omega}{dt} \right)^2 \right] = \frac{2ac}{b^2} \Omega^{1-\frac{1}{K}} \frac{d\Omega}{dt}$$

$$\frac{d\Omega}{dt} = \sqrt{\frac{2ac}{b^2} \cdot \frac{\Omega^{2-\frac{1}{K}}}{2-\frac{1}{K}}} + C,$$

$$\text{or } \frac{d\omega}{dt} = \frac{-\omega^{K+1}}{K} \sqrt{\frac{ac}{b^2} \cdot \frac{1}{(1-\frac{1}{K})} \omega^{2(1-K)} + C} \quad \text{--- (5)}$$

$$\text{at } t=0, \quad \omega = \omega_0 \quad \text{and} \quad \frac{d\omega}{dt} = 0 \quad \therefore C = \frac{-ac}{b^2(1-\frac{1}{K})} \omega_0^{2(1-K)}$$

$$\frac{d\omega}{dt} = -\sqrt{\frac{ac}{b^2(1-\frac{1}{K})}} \cdot \frac{\omega^{K+1}}{K} \cdot \omega_0 \sqrt{\left(\frac{\omega}{\omega_0} \right)^{2(1-K)} - 1}$$

Introducing the dimensionless parameters

$$\frac{\omega}{\omega_0} = \omega_R$$

$$\omega_0 t = \phi$$

$$\frac{d\omega_R}{d\phi} = \frac{1}{\omega_0} \cdot \frac{d\omega}{dt}$$

$$\text{i.e. } \frac{d\omega_R}{d\phi} = -\sqrt{\frac{ac}{b^2(1-\frac{1}{K})}} \cdot \frac{1}{K} \cdot \omega_R^{K+1} \sqrt{\omega_R^{2(1-K)} - 1}$$

This equation is not integrable for all values of K but a fair approximation can be obtained by numerical methods. To do this, values must be chosen for the constants $a, b, \text{ and } c$.

Choice of Constants

$$a = \rho \frac{A}{2} (R_2^2 - R_1^2), \quad b = \frac{I_1}{\pi R_1^2}, \quad c = \frac{K I^2}{\pi^2 A^2 \rho^2 R_2^4}$$

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The friction factor k associated with the flow path is $\rho \frac{Af l}{2D}$ where

ρ, A, l have already been defined

and D = hydraulic radius

f = a term denoting the degree of roughness etc. of the wetted surfaces.

$$\text{Now } K = \frac{C}{b} = \frac{k l}{\pi A^2 \rho^2 R_2^2 l}$$

and if the mass of the aircraft m_a is considered to act as an inertia at the radius R_2 neglecting the inertia of the brake

$$\begin{aligned} K &= \frac{C}{b} = \rho \frac{Af l}{2D} \cdot m_a R_2^2 \cdot \frac{1}{\pi A^2 \rho^2 R_2^2 l} \\ &= \frac{f}{2} \cdot \frac{1}{D} \cdot \frac{m_a}{\pi \rho A l} \end{aligned}$$

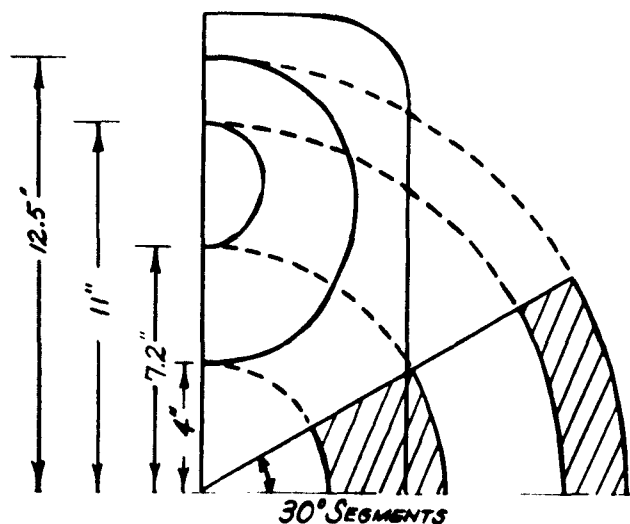
the term $\pi \rho A l$ is the mass of fluid in the brake m_f .

$$\therefore K = \frac{f}{2} \cdot \frac{1}{D} \cdot \frac{m_a}{m_f}$$

Now

$$\begin{aligned} \frac{\partial C}{b^2} &= \frac{k(R_2^2 - R_1^2)}{2 \rho A^2 l^2} = \frac{\rho A f l}{2 D} \cdot \frac{(R_2^2 - R_1^2)}{2 \rho A l^2} \\ \therefore \frac{\partial C}{b^2} &= \frac{f}{4} \cdot \frac{1}{D l} (R_2^2 - R_1^2) \end{aligned}$$

When considering a rotary friction device for the standard arrest conditions (see page 42) it was shown that a drum of diameter 28" and length 19" was large enough. Using the same drum dimensions gives the following flow path.

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Using these tentative demensions

$$A = .0638 \text{ sq. ft.}$$

$$l = 1.56 \text{ ft.}$$

$$D = .285 \text{ ft.}$$

$$R_1 = .48 \text{ ft.}$$

$$R_2 = .98 \text{ ft.}$$

$$\rho = 1.94 \text{ slugs/cu. ft.}$$

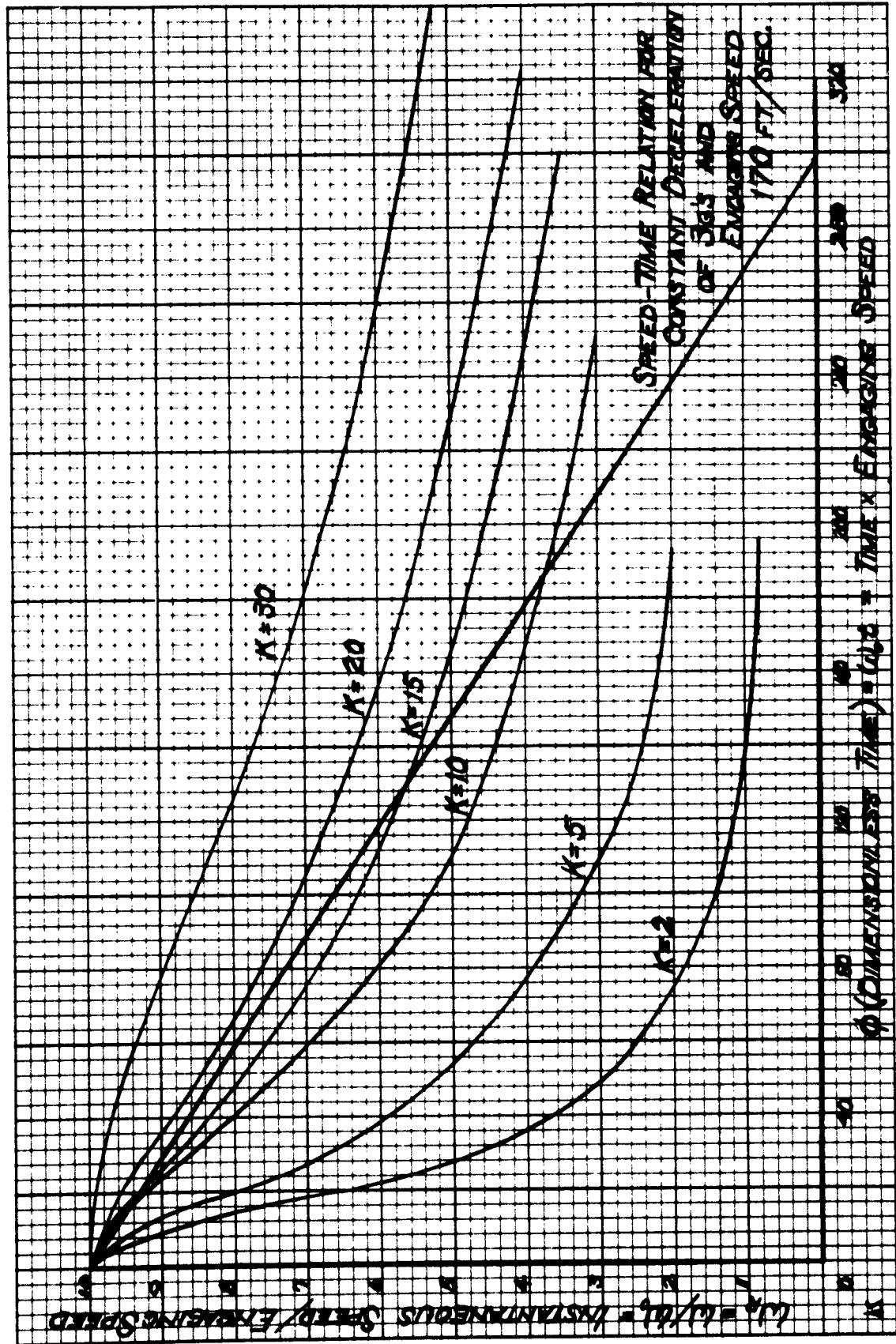
and choosing

$f = .02$, approximate solutions of the

speed time relation have been plotted for values of the ratio $\frac{m_a}{m_f}$, viz.

$\frac{m_a}{m_f}$	giving	K
36.6		2
91.5		5
183		10
274		15
366		20
549		30

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SPEED DECAY WITH BRAKED FLUID COUPLING

Figure 3

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None of the plotted curves gives a desirable arrest characteristic. The curves for $K = 15$ and $K = 20$ are fairly good over the early part of the arrest period but provide very low arrest forces at low speeds. Since the values of K indicate the amount of fluid circulating in the brake - a low K corresponding to a large circulating mass and vice versa - the possibility of varying K during the arrest becomes of interest. If the brake started off with $K=20$ this would correspond to 22.9 gallons of fluid (considering the "standard" arrest of a 70,000 lb. aircraft). If the final value of K was reduced to 2, this would correspond to 229 gallons of fluid. So that approximately 206 gallons of fluid would have to be pumped into the brake during the arrest period of 1.76 seconds. This implies a mean pumping rate of 7040 gallons/minute.

Referring back to the expression $nPA_1 = m_f$ (the mass of fluid in the brake), with the assumed flow path dimensions and a volume of 229 gallons of fluid, the number of flow paths required is 307 or, taking 12 buckets per wheel, some 25 wheels. Taking an axial length of 6" per wheel, the brake would be 12' - 13' long. However, this would probably be divided into two units, each connected to one end of a deck pendant.

The drawbacks inherent in the scheme can now be seen. Attention has already been drawn to the large inertia effects associated with a drum and a wrapped cable, and in this case the drum would be larger than the minimum necessary to house the arrest cable.

The difficult job of metering fluid to the brake during the short period of arrest would require extensive design and development work before a satisfactory solution could be obtained.

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CONFIDENTIALELECTRO MAGNETIC DEVICES

If the aircraft is allowed to actuate an electrical generator of some kind, the resulting electrical energy may be dissipated through a water cooled loading resistance. As the aircraft slowed down, the generated potential would drop off and this would necessitate some variation of the loading resistance to maintain a constant retardation. If the generating equipment be thought of in terms of standard electrical apparatus, operating at perhaps 100% overload for short intervals, capacities in the range of 15,000 to 30,000 HP would result in prohibitively large installations. Not only would the total installed weight be large but the mass of moving parts would be so great that substantial inertia loads would be applied to the arrested aircraft.

Since the field densities needed are greater than those offered by the residual magnetism of normal materials, additional power will be needed to supply field windings and it is basically unsound to supply additional energy to an apparatus whose sole purpose is to dissipate large amounts of energy received from another source. If the generated current is allowed to supply the field windings, this will increase the difficulties associated with control of the retardation.

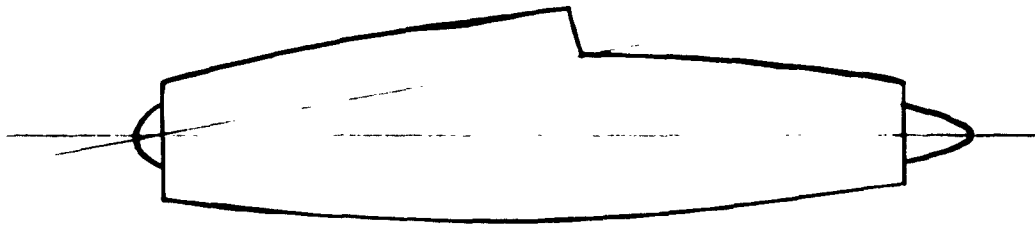
In general, it may be said that excessive installed weight and effective inertia will preclude the use of electro-magnetic arresting gear.

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CONFIDENTIALCHANGES IN CARRIER CONFIGURATION

Two variations of the orthodox flight deck have been proposed which, besides contributing materially to the effectiveness of carrier operations, simplify to some extent, the actual process of arrestation.

The "angled" or "canted" deck, as the name implies, consists of a landing area inclined at an angle to the centerline of the carrier. The after end of the landing area is at the intersection of the two axes while the forward end is cantilevered out over the port side.



The principal advantage of this scheme is that once an aircraft has touched down, it is not imperative that it be arrested but may take off and go around again. Since the forward part of the landing area is no longer needed for parking aircraft, there is no longer any danger of a crash when an airplane fails to hook an arresting wire. This reduced necessity for catching an aircraft which has touched down can simplify the arresting problem by allowing a reduction in the number of arresting wires and engines. One drawback would be the possibility of taking a longer time to land a complete strike of aircraft since some aircraft might make several passes. A reduction in the number of

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arrest points (deck pendants) lends added interest to the damped spring system considered earlier, also to the coarse net mentioned previously.

Although the angled deck requires that a slightly different landing technique be learned by the pilots (e. g. power will be cut after a pendant has been hooked instead of before touchdown) the difficulties are not insurmountable.

The "flexible deck", as the name suggests, is a relatively soft area of the deck, formed by stretching a sheet of heavy rubber over a number of inflated rubber tubes. A single pendant, placed some 6 or 7 feet above the deck catches the hook of an aircraft flying with close to stall power. As the aircraft loses flying speed, it sinks down and finally drops on to the flexible area. The vertical and horizontal speeds at the time of contact with the deck can be controlled by the height at which the aircraft hooks on and the rate at which it is arrested. If arrested rapidly at a considerable height above the deck, the sinking speed will be high when touchdown occurs. If arrested slowly, close to the deck, contact may take place at a low sinking speed but with a high forward speed with some danger of burning the rubber mat. To the obvious advantage of reduced arresting capacity (single pendant and engine) must be added the benefits accruing from the elimination of the aircraft undercarriage since this is no longer necessary with a flexible deck. The reduced weight is offset somewhat by the slight stiffening needed to enable the fuselage to withstand the deck impact. However, this is a minor point and an appreciable net saving in weight can be effected.

The canted deck and flexible deck have been given

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careful tactical consideration by organizations experienced in these matters. Since the technical aspects of the arresting problem are not materially altered by these configurations, this report will not dwell at length on either system.

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CONFIDENTIALMETHODS OF REDUCING APPROACH VELOCITY

As the top speed of new aircraft goes higher and higher, the stalling speed also increases, thereby increasing the amount of energy to be absorbed by the arresting gear. The possibility of using some device to reduce the approach velocity while maintaining sufficient lift is certainly attractive. Any device used in addition to flaps and slots - such as boundary layer control - necessarily imposes a weight penalty on the aircraft, and this line of thinking has resulted in the suggestion that the wind over the deck be increased to perhaps 100 kts. by the installation of a suitable wind machine on the carrier. Assuming that such a condition could be achieved, an aircraft with a stalling speed of 130 kts would have an approach velocity of 30 kts. relative to the carrier and could be arrested quite easily by a fairly light gear. However, the scheme must be considered in the following way.

In "still" air, the aircraft is flying at 130 kts and has the K.E. associated with its mass at this velocity. As it enters the high speed air stream, the drag forces are suddenly increased (proportional to V^2 where V is the relative wind) and the aircraft slows down under these increased drag forces until an air speed of 130 kts is again reached. During the time that the airplane is slowing down, it moves through an appreciable distance relative to the carrier so that the wind machine must be capable of producing a region of high velocity air equal in length to this distance.

Assume the carrier to be steaming at V_c and producing a wind over deck of V_d , the velocity of the air stream relative to the sea is then $V = V_d - V_c$. Assuming that there is no natural wind and

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that the aircraft is flying at its stalling speed V_s in still air, then:

$$\text{drag forces on aircraft} = k V_s^2 = \text{engine thrust.}$$

On entering the high speed stream, the aircraft velocity becomes

$$(V_s + V) \quad \therefore \text{new drag forces} = k(V_s + V)^2$$

If the engines are cut immediately on entering the high speed stream,

$$\text{the net retarding force is } k(V_s + V)^2$$

After a short time, the aircraft has slowed to a velocity v and the

$$\text{retarding force is } k(v + V)^2$$

Therefore:

in an interval dt the aircraft moves a distance dx and the work

$$\text{done by the retarding forces} = k(v + V)^2 \cdot dx$$

$$= \text{loss in K. E.} = -d\left(\frac{1}{2} m v^2\right) = -m v dv$$

$$\therefore k(v + V)^2 dx = -m v dv$$

$$\therefore dx = -\frac{m}{k} \int_{v_0}^{v_1} \frac{v dv}{(v + V)^2}$$

$$\text{or } x = -\frac{m}{k} \left[\ln(v + V) + \frac{V}{v + V} \right]_{v_0}^{v_1}$$

If $v_0 = V_s$ and $v_1 = V_s - V$

$$\begin{aligned} x &= -\frac{m}{k} \left[\ln\left(\frac{V_s}{V_s + V}\right) + \frac{V}{V_s} - \frac{V}{V_s + V} \right] \\ &= \frac{m}{k} \left[\ln\left(1 + \frac{V}{V_s}\right) - \frac{V^2}{V_s(V_s + V)} \right] \end{aligned}$$

As an example, take the case where a high speed stream equal to the stalling speed of the aircraft is produced

$$x = \frac{m}{k} \left[\ln 2 - \frac{1}{2} \right] = .193 \frac{m}{k}$$

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This must be the length of the high velocity air stream plus the distance moved by the carrier.

Considering the resistance factor k ; if the aircraft is represented by a flat plate of area A its resistance is $\rho A v^2$ so that $k = \rho A$

$$\text{or } \frac{m}{k} = \frac{m}{\rho A} = \frac{1}{\rho \left(\frac{A}{m}\right)} = \frac{1}{\rho g \left(\frac{A}{W}\right)}$$

now $\frac{A}{W}$ is given as 0.5 sq. ft. / 1000 lbs. for large aircraft so that

$$\text{we may write } x = 0.193 \cdot \frac{1}{0.0005 \rho g}$$

$$\text{which for standard atmosphere} = \frac{.193}{.0005 \times 32.2 \times .00238} = 5040 \text{ ft.}$$

The provision of a mile long airstream having a velocity of about 100 kts. is obviously out of the question. In view of the relatively small effect of the wind machine, it seems unlikely that a combination of wind machine and arresting gear would be justified. Furthermore, the mechanical difficulties associated with the provision of a high-speed air stream are not justified by the arresting effects gained.

CONFIDENTIALCATAPULTSGENERAL CONSIDERATIONS

Since the launching problem is obviously the converse of the arresting problem, we may look at it from the standpoint that, rather than being supplied with an energy source and trying to find a sink, we are now supplied with a sink and must find a suitable energy source. The fundamental energy source on board ship is, of course, fuel oil and this must be the starting point for any process requiring other than man-power unless additional chemical sources are carried on board. The ultimate form which this chemical energy must take when delivered to the aircraft is mechanical in nature and the launching gear should be considered as all the equipment necessary to carry out the conversion and transfer. Equipment already exists on board ship to carry out the first step of the conversion and energy is available in thermo-mechanical form (steam) and in electrical form. However, since a short time power demand of 29,000 hp is required for the launch of a 70,000 lb. plane at 100 kts in 2 secs., and may overload the ship's facilities, an energy accumulator or reservoir may be necessary. This reservoir will be charged relatively slowly, prior to the actual launch, while during the launch the energy will be released rapidly through the mechanism which delivers it to the aircraft.

The discussion which follows will deal with types of reservoir and delivery mechanisms both conventional and novel.

"Standard" launch conditions will be taken as follows:

Aircraft weight = 70,000 lbs.

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End speed (relative to carrier) = 100 knots = 170 ft./sec.

Energy delivered = 32×10^6 ft. lb.

Launching force for 3g launch = 200,000 lbs.

MECHANICAL STORAGE OF ENERGYStrain energy of solid medium (metallic spring)

The storage of energy by mechanical stress in a metallic spring is readily shown to be impracticable. Consider a weight of metal W having density ρ and elastic modulus E , stressed in tension to σ .

$$\text{Strain energy} = \frac{\sigma^2}{2E} \cdot \frac{W}{\rho}$$

With the figures

$$\sigma = 60,000 \text{ psi (yield)}$$

$$E = 30 \times 10^6 \text{ psi}$$

$$\rho = .283 \text{ lb./in.}^3$$

The weight of metal necessary to store 32×10^6 ft. lb. is 900 tons. The case of pure tension is a highly idealized condition since most mechanical springs operate in bending and torsion and this latter condition produces a much lower utilization of elastic capacity due to the non-uniformity of the stress distribution.

An increase in elastic energy capacity (for the same stress) can only be obtained by a reduction of elastic modulus and density. However, the range of densities and moduli which give acceptable capacities can only be obtained using gaseous media.

CONFIDENTIALStrain energy of gaseous medium (air spring)

If a weight of air W is compressed and expanded polytropically ($PV^n = c$) between the pressure limits P_0 and P_1 the energy involved in the process is

$$= \frac{n c \pi}{n-1} \left[(P_1)^{\frac{n-1}{n}} - (P_0)^{\frac{n-1}{n}} \right]$$

If the pressure limits are chosen as 5000 psi and 4000 psi the energy involved in compressing and expanding one pound of air between these limits with the polytropic index 1.2 is 18,000 ft. lb. The weight of air necessary to store 32×10^6 ft. lb. becomes 1785 lbs. The volume limits, starting with atmospheric air (14.7 psi 15°C) will be 218 cu. ft. and 182 cu. ft. requiring an air flask some 4' dia. and 20' long. Such a cylinder would have a wall thickness of approximately 2" and would weigh some 10 tons so that the weight of gas can be neglected in comparison with the weight of the air flask.

Kinetic energy of solid medium (flywheel)

If a large mass can rotate about one axis at a speed ω rads/sec., it will have a kinetic energy $1/2 I \omega^2$ associated with it, where I is the mass moment of inertia about the axis of rotation. The maximum speed at which the mass can be allowed to rotate is that at which the centrifugal stresses reach the strength limit of the material.

For a solid disc, the radial and tangential stresses caused by rotation are

$$\sigma_r = \sigma_t = \frac{\rho v^2}{8} \cdot \frac{3+\nu}{8}$$

Where ρ = density of material

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V = peripheral velocity

ν = Poisson's ratio

If the maximum stress which the material can withstand is σ the maximum allowable peripheral velocity is

$$\sqrt{\frac{8\sigma}{\rho(3+\nu)}}$$

If the diameter of the disc is D the maximum rotational speed ω is

$$\frac{2}{D} \sqrt{\frac{8\sigma}{\rho(3+\nu)}}$$

If the flywheel has length L its mass moment of inertia is

$$\frac{\pi}{32} D^4 L \frac{\rho}{g} = \frac{D^2 W}{8g} = I$$

The maximum kinetic energy which can be stored is thus

$$\frac{1}{2} I \omega^2 = \frac{1}{2} \left(\frac{D^2 W}{8g} \right) \frac{4}{D^2} \cdot \frac{8\sigma}{\rho(3+\nu)}$$

or taking $\nu = \frac{1}{3}$ $K.E. = \frac{3}{5} \cdot \frac{\sigma W}{\rho}$

Using the same yield stress (60,000 psi) and $\rho = .283$ lbs./cu. in. the weight of flywheel necessary to store 30×10^6 ft. lb. becomes 2830 lbs.

An obvious drawback to the use of a flywheel is the falling off in speed as energy is given up. Since the aircraft is gaining in speed during this time, the linkage between the flywheel and shuttle must provide a variable gear ratio.

CHEMICAL STORAGE

This is undoubtedly the most efficient means of storing energy. For example, the combustion of 2 lbs. of fuel oil having a calorific value of 19,000 BTU/lb. releases 30×10^6 ft. lb. - the energy required for the "standard" launch conditions. The heat of

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combustion is used to raise the reaction products to a high pressure which can then be used to actuate the launching mechanism. Unfortunately, the overall efficiency of such a system is low, since the reaction products, which are exhausted after the launch, retain a large part of the total thermal energy. This inefficiency results in a proportionate enlargement of the store of fuel and makes it important that the fuel be readily available and fairly inexpensive.

If an efficiency of 20% were achieved, 10 lbs. of fuel oil would have to be stored and used per standard shot. Considering a fuel/oxygen ratio of 1:3 a combined weight of 40 lbs. of fuel and oxidant would be required per shot.

Comparing this with a powder propellant having a heat of detonation of 700 BTU/lbs. and an overall efficiency of 20%, the weight of charge per shot would be 275 lbs., which points up the logistics problem associated with the use of powder propellant.

Ideally the fuel would be a monopropellant (i.e. one which supplies its own oxygen) and should be in a liquid form for ease of handling. If the reaction in a fuel of this type is easily initiated, problems of safety will arise, while if the reaction is initiated with difficulty, operation of the catapult will be erratic and uncertain.

At the present stage of development, the combustion of fuel oil in stored oxygen appears the most satisfactory fuel system, in that no outstanding hazards, handling problems or combustion technique are associated with its use.

ELECTRICAL STORAGE

The energy requirement of 32×10^6 ft. lb. is equivalent

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to 43×10^6 watt-sec. During the launching period of 1.76 seconds the power required will increase uniformly from zero to 48,600 kw. Obviously such a demand could not be supplied directly from the ship's electrical system without back up from an energy storage device.

If energy-storage devices were available whose energy could be used during launching and which could be recharged between launchings so as to smooth out the power demands on the ship's system, the assumptions of 100 percent efficiency and a launching every 30 seconds give an average power required from the ship's system of 1430 kw. The energy-storage device would have to be able to store 40×10^6 watt-seconds (29.4×10^6 lb. ft.) and discharge this energy at a maximum rate of 47,200 kw.

Unfortunately it is not easy to store bulk energy electrically. Energy may be stored in the electric field of a capacitor or in the magnetic field of an inductor. With the former, the stored energy density is given by

$$w_{elec.} = \frac{1}{2} \epsilon E^2 \text{ ----- } \textcircled{1}$$

where $w_{elec.}$ is the energy density in watt-seconds per cubic meter of dielectric medium, ϵ is the permittivity (or capacitivity) of the dielectric in rationalized MKS units (coulombs²/newton-meter²), and E is the potential gradient in volts per meter. The maximum permissible potential gradient is determined by the dielectric strength of the medium as an insulator. For example, air at atmospheric pressure will withstand a maximum potential gradient of about

In a magnetic field, the energy density is given by

$$w_{\text{mag}} = \frac{1}{2} \frac{B^2}{\mu} \quad \text{--- (2)}$$

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material and an air gap in series nearly all of the energy is stored in the air gap, the magnetic material serving merely as a means for guiding and concentrating the field. In such a circuit, the maximum flux density which can be produced is determined by saturation of the magnetic material. For example, it is uneconomical to work silicon steels at flux densities much above 100 kilolines per square inch (1.55 webers per square meter). At this flux density in a magnetic circuit comprising steel and air paths in series and assuming the same flux density in both parts, by Equation (2) the energy density in the air is 9.5×10^5 watt-seconds per cubic meter. The volume of magnetized air required to store 40×10^6 watt-seconds would then be about 42 cubic meters. In order to keep the losses in the magnetizing coil from being excessive, the gap length would have to be relatively short and its cross-sectional area would therefore have to be large.

Thus, neither of the suggested electrical means for energy storage appears to be practicable. The storage of energy in electrochemical form in a storage battery should also be considered. The limiting feature here is the short-time power-output capacity. A rough calculation can be based on a good quality heavy-duty 6-volt automobile battery. Such a battery rated 125 amperes for 20 minutes is capable of an output of 300 amperes for 3 minutes. Assuming 5 volts available at the 300 ampere discharge rate gives a peak power output of 1500 watts. The peak power required of the storage element during launching is about 47,000 kw. Hence, about 30,000 car batteries would be required weighing about 35 pounds each and occupying about 500 cubic inches each. The total weight would be about

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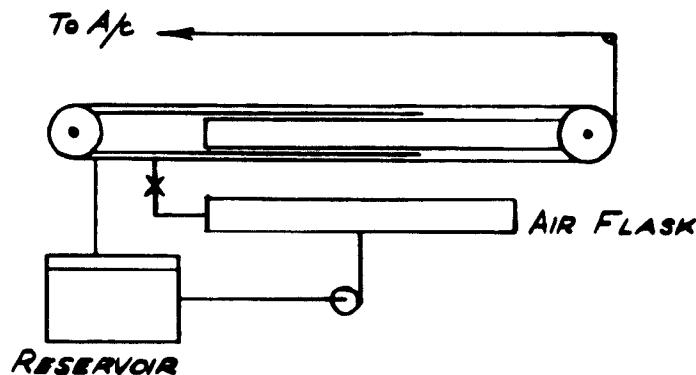
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1,000,000 pounds and the space occupied would be about 9,000 cubic feet packed solidly.

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CONFIDENTIALLAUNCHING MECHANISMSORTHODOX HYDRAULIC SYSTEMS

In this system the energy accumulator is an air spring, the thrust from this spring being transmitted through a fluid medium to a hydraulic ram. The movement of the ram is geared to the shuttle by means of a reeved cable system.



The limitations of hydraulic, cable driven catapults has long been recognized, two of the major drawbacks being

1. The large amount of energy lost as K.E. in the cable, crosshead, etc.
2. The variations in launching force caused by cable elasticity.

A natural consequence of the first point is that pumping and storage equipment must be larger than would appear necessary from considerations of the aircraft kinetic energy. The construction of launching gears to handle heavy, high-speed, aircraft is severely penalized because of this, since the reduced efficiency will cause the equipment weight to scale up more rapidly than the useful energy output.

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N. A. F. Report M-4635 deals with the limitations of this type of catapult and the graphs in Figure 4 have been prepared from results quoted in that report.

These curves show that (at high speeds) considerably more energy appears as K. E. of the launching mechanism than as K. E. of the load. At the lower speeds which are of interest for present day aircraft, the energy of the gear is still a substantial fraction of the total although less than 50%. One interesting point illustrated by the curves is the large amount of energy associated with the cable sheaves compared with that associated with the ram and crosshead.

If a catapult has an equivalent mass of cable M_1 (which includes towing and retrieving strands and the rotary inertia effects of the sheaves), the energy associated with the cable, sheaves etc. at the end of the launching run is approximately $1/2 M_1 (\frac{V}{2})^2$ where V is the shuttle end speed. This assumes that the average velocity of the cable in the machine is half the end velocity - a more accurate estimate would be a summation of the energy of the reeves within the machine.

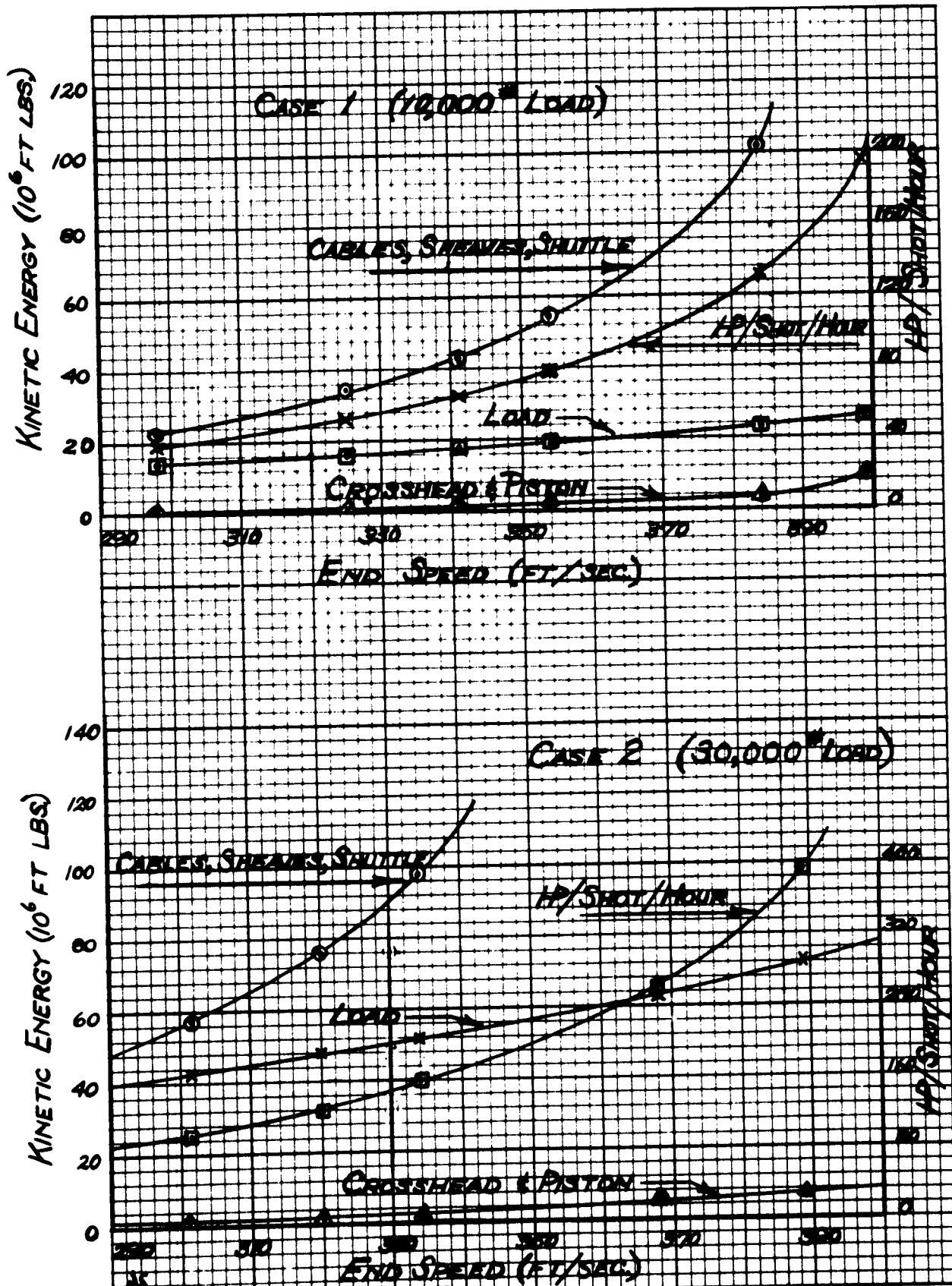
If the mass of the ram and crosshead assembly is M_2 the energy associated with it is $1/2 M_2 (\frac{V}{10})^2$ for a reeving ratio of 10.

$$\text{Thus "cable" K.E.} = \frac{M_1}{4} V^2$$

$$\text{"ram" K.E.} = \frac{M_2}{200} V^2$$

Taking the representative figures of 5000 lbs. and 20,000 lbs. for the weight of cable and ram respectively, it can be seen that the energy associated with the cable is 12 1/2 times the energy associated with the ram.

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ENERGY DISTRIBUTION IN HYDRAULIC REEVED CATAPULT
FROM N.A.F. REPORT 11-4635

Figure 4

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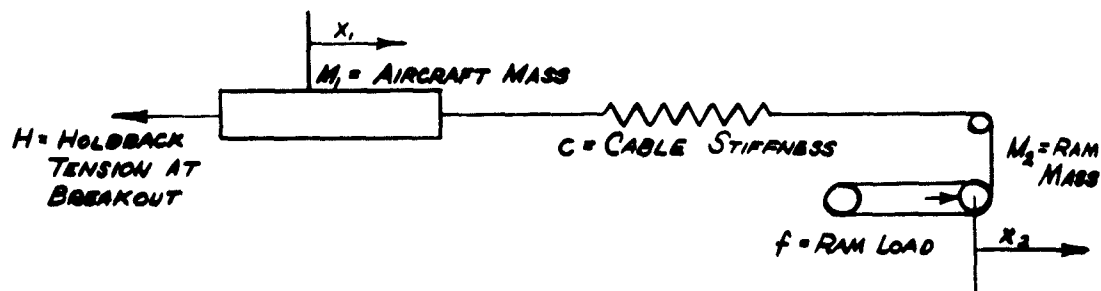
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Reducing the reeving ratio while keeping the output (aircraft) K.E. constant results in a more efficient catapult but this is offset somewhat by space needed to accommodate the increased stroke of the longer ram. The use of a piston sliding in a slotted cylinder and directly connected to the launching shuttle results in a greatly improved distribution of energy within the launching mechanism, since the elimination of cable, sheaves, ram etc. effects a large reduction of this "parasitic" moving mass.

The slotted cylinder mechanism is considered further at a later stage of this report.

Fluctuations in launching force - the second point mentioned - are obviously undesirable. In addition to the unwelcome structural and physiological effects of "jerk" (or rate of change of acceleration), the maximum "g" loading allowed by the aircraft design is not utilized during the full launching run. This will result in an unnecessarily long launching time and run for a given end speed.

Neglecting the damping effects of aircraft drag, friction in the launching gear, cable hysteresis etc. a simplified dynamical system is as follows.



REEVING RATIO = 72

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When pressure is admitted to the launching cylinder, the ram will accelerate - stressing the cable and loading the aircraft until the holdback ring breaks. The acceleration experienced by the aircraft has been analyzed for a set of conditions closely approximating those encountered in practice. The ram pressure is assumed to build up to its maximum value F according to the form: $\text{force} = F(1 - e^{-k(t+t_0)})$. This implies that at zero time, the nominal launching force is $\frac{F}{k}(1 - e^{-kt_0})$ and is equal to the holdback tension H . Time is measured from the instant the holdback breaks and at this instant the aircraft is assumed to be at rest. The velocity and acceleration of the ram are assumed to be negligible at the instant the holdback breaks.

The resulting acceleration of the aircraft has roughly two components - one an exponential build up and the other a high frequency sinusoidal oscillation. The period of this oscillation is

$$2\pi\sqrt{\frac{m_1 m_2}{c(\kappa^2 m_1 + m_2)}}$$

and has a total fluctuation

$$2\sqrt{\left[\frac{H}{m_1} - \frac{Fc}{m_1 \frac{m_2}{\kappa} \omega^2} + \frac{(F - \kappa H)c}{m_1 \frac{m_2}{\kappa} (\kappa^2 + \omega^2)}\right]^2 + \left[\frac{c\kappa(F - \kappa H)}{m_1 \frac{m_2}{\kappa} \omega (\kappa^2 + \omega^2)}\right]^2}$$

where

$$\omega^2 = \sqrt{\frac{c(\kappa^2 m_1 + m_2)}{m_1 m_2}}$$

The equation was derived in the following manner:

Consideration of the equilibrium of the masses m_1 and m_2 gives

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$$c(n x_2 - x_1) = m_1 \frac{d^2 x_1}{dt^2}$$

$$F[1 - e^{-k(t+t_0)}] - nc(n x_2 - x_1) = m_2 \frac{d^2 x_2}{dt^2}$$

giving a single equation

$$D^2[D^2 + \omega^2]x_1 = \frac{ncF}{m_1 m_2} [1 - e^{-k(t+t_0)}]$$

where D is the operator $\frac{d}{dt}$

$$\text{and } \omega^2 = \frac{c(n^2 m_1 + m_2)}{m_1 m_2}$$

this has the general solution

$$x_1 = C_1 + C_2 t + C_3 \sin \omega t + C_4 \cos \omega t + \frac{ncF}{m_1 m_2 \omega^2} \cdot \frac{t^2}{2} - \frac{ncF[e^{-k(t+t_0)}]}{m_1 m_2 k^2 (k^2 + \omega^2)}$$

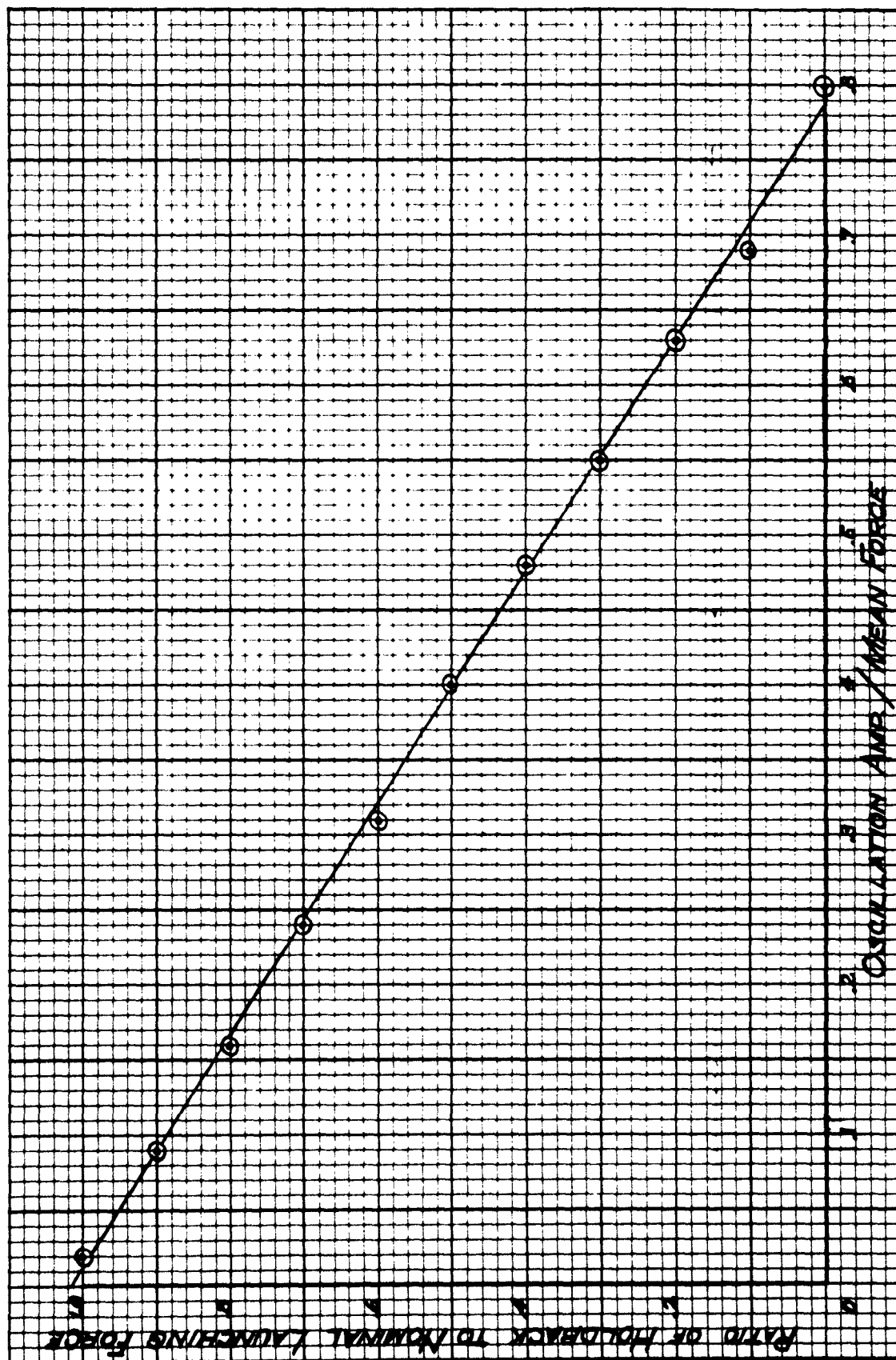
$$\text{and } \frac{d^2 x_1}{dt^2} = -C_3 \omega^2 \sin \omega t - C_4 \omega^2 \cos \omega t + \frac{ncF}{m_1 m_2 \omega^2} - \frac{ncF e^{-k(t+t_0)}}{m_1 m_2 (k^2 + \omega^2)}$$

After the constants of integration have been evaluated, the value of the total amplitude of the sinusoidal portion can be evaluated.

If this fluctuation is expressed as a fraction of the nominal acceleration, the resulting dimensionless ratio can be plotted as a function of $\frac{H}{F/n}$ - the ratio of the holdback tension to the nominal accelerating force. Figure 5 shows such a plot for aircraft launched from a catapult of the H4B type.

The graph implies that very low fluctuations could be obtained by using a holdback tension almost as great as the launching force.

Figure 6 gives a comparison of the theoretical launching force with some experimental values obtained during a launch

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EFFECT OF HOLDBACK TENSION ON OSCILLATORY LOAD

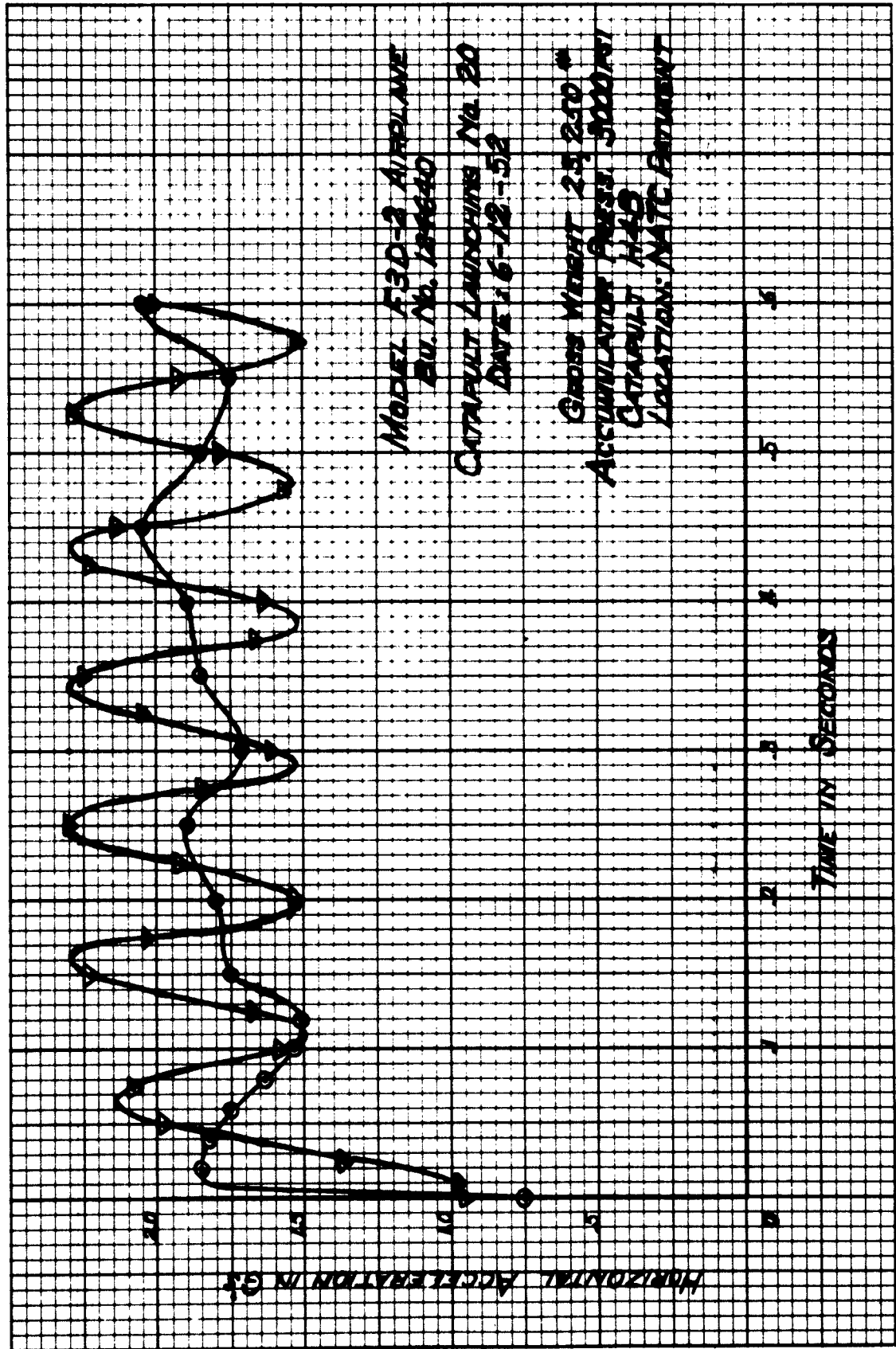
Figure 5

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at Patuxent. The agreement is fairly good.

Energy losses in the catapult and fluctuations in the launching force can be considerably reduced by eliminating the cable system altogether. The configuration of the system would now be in the form of a slotted cylinder containing a moving piston to which the aircraft is attached.



COMPARISON OF ACTUAL AND THEORETICAL LAUNCHING ACCELERATIONS
(v)
(v)
Figure 6

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The reader is assumed to be familiar with the structural features of this type of catapult, and this discussion will be limited to the mechanics of the system.

Two general systems of operation can be envisaged - the direct cycle and the storage cycle. The direct cycle is here taken to mean the system in which thermal energy is released from the primary fuel during the launching process only and at the rate required by the conditions of the launch. The storage cycle is that in which thermal energy is released from the primary fuel at a continuous - and fairly low - rate, all during flight deck operations and makes use of an accumulator between the combustion chamber and the catapult.

DIRECT CYCLE

This almost necessarily implies that combustion of the primary fuel takes place within the catapult cylinder or an extension of the cylinder. The high rate of energy release required for the 'standard' launching conditions chosen involves burning rates close to detonation. At the same time, the desirability of controlled rate of release (necessary for constant acceleration) places a restriction on the use of explosive fuels - solid propellants such as cordite etc. A more readily controlled burning rate can be obtained by using liquid fuels which can be metered to the combustion chamber at the required rate.

Much has been said on the problems of safety and

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logistics with regard to 'explosive' fuels. There is no theoretical basis on which to choose an optimum combination of three such dissimilar variables as efficiency, safety, and ease of supply, and any choice must be dictated largely by personal opinion based on experience. The potential saving in installed weight offered by the absence of an energy accumulator makes the direct cycle an attractive catapult system but the success of such a device hinges on the choice of fuel.

A British Admiralty Report - "Sources of Power for Aircraft Catapults" - presents the results of an extensive study and comes to the conclusion that the combustion of diesel oil in stored oxygen offers many advantages. This is considered the best alternative to a satisfactory liquid mono-propellant if such cannot be found.

If the composition of the fuel oil is assumed to be 85% C and 15% H_2 , combustion will require 3.5 lbs. O_2 /lb. fuel or 15 lbs. air/lb. fuel. Assuming combustion to be carried out at constant pressure, the fuel-oxygen mixture will have a final temperature of 9200°R and the fuel-air mixture a temperature of 4100°R. The useful work output will be 2660 BTU/lb. and 4530 BTU/lb. respectively (if the oil has a heat of combustion of 18,500 BTU/lb.), so that the oxygen system will require 15.5 lbs. fuel and the air system 9 lbs. fuel for the standard launch of 30×10^6 lbs.

Since the temperature attained is greater than that which could be allowed in a slotted cylinder, it becomes necessary to add some form of diluent. Dilution can be accomplished by using a large amount of excess air (or oxygen) or by injecting water into the

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combustion zone. The cost of generation of liquid oxygen would preclude its use as a diluent so that three systems only are of interest.

1. Fuel-air mixture with air diluent
2. Fuel-air mixture with water diluent
3. Fuel-oxygen mixture with water diluent

The table below gives the amount of fuel necessary for a standard launch with the various combinations and if a maximum demand of 200 shots per day is assumed, this allows some estimate to be made of the capacity of the auxiliary equipment - oxygen generator, air compressor etc.

SYSTEM	TEMP.	COMBUSTION		O ₂	H ₂ O
		FUEL PER SHOT	AIR PER SHOT	PER SHOT	PER SHOT
1 fuel/air + air	1500°R	6 lbs.	392 lbs. 5220 scf	--	--
2 fuel/air + water	1500°R	13 lbs.	194 lbs. 2600 scf	--	118 lbs.
3 fuel/oxygen + water	1500°R	18 lbs.	--	63 lbs.	200 lbs.
1 fuel/air + air	2000°R	6.5 lbs.	295 lbs. 3930 scf	--	--
2 fuel/air + water	2000°R	11.2 lbs.	168 lbs. 2240 scf	--	76 lbs.
3 fuel/oxygen + water	2000°R	15.7 lbs.	---	55 lbs.	148 lbs.
1 fuel/air + air	2500°R	7.1 lbs.	237 lbs. 3160 scf	--	--
2 fuel/air + water	2500°R	8.4 lbs.	126 lbs. 1680 scf	--	38 lbs.
3 fuel/oxygen + water	2500°R	14.62 lbs.	---	51 lbs.	110 lbs.

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These figures are based on combustion at atmospheric pressure and will be different for combustion at pressures of interest for slotted cylinder operation. However, thermodynamic data for hydrocarbon combustion products at high pressure are limited and time does not permit an extensive study of high pressure conditions.

Taking the maximum requirement (per shot) for the various cycles as

1. 6000 s.c.f. of air,
2. 5000 s.c.f. of air -- 150 lbs. of water,
3. 70 lbs. liquid oxygen - 250 lbs. of water,

and a demand of 200 shots per day at a maximum rate of two per minute, the auxiliary capacities may be listed as follows:

- (a) Using a fuel/air plus air system operating at a pressure of 2000 psi, an adiabatic horsepower of 4,250 HP would be required to supply the air. If multistage compression is employed using intercooling, the process will be more nearly isothermal and a smaller power will be necessary. The isothermal horsepower required is 3750 H.P. Discussions with representatives of Ingersoll-Rand, Inc. have revealed that two "six frame" machines would serve this purpose and would require some 3000 HP each. Two machines would occupy a volume of approximately 20,000 cubic feet and would weigh about 180 tons. A considerable improvement in volume and weight can be effected by carrying out the initial compression (say to 200 psi) in a turbo-compressor and this might reduce the total weight of the installation by as much as half.

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- (b) A fuel/air/water system operating at the same pressures would require air compressors of half the previous capacity but would in addition require a water pump of some 45 HP. This would be a relatively small installation and the whole system would be about half the size of the fuel/air/air system.
- (c) A fuel/oxygen/water system would require a liquid oxygen plant of seven tons/day capacity. An existing ten tons/day unit occupies a total volume of 4500 cu. ft. and weighs approximately twenty tons. The water pump would increase these figures by several hundred cu. ft. and a few tons.

On the basis of weight and volume, the fuel/oxygen/water system clearly has the advantage but suffers slightly from the fact that new techniques are involved in the production, handling and storage of liquid oxygen. However, it is felt that this is outweighed by the space advantage.

It should be pointed out that the discussion just presented does not consider the inefficiencies introduced by heat losses from the launching cylinder. However, since the same conditions will apply for each system, the comparisons drawn and the conclusions reached are not invalidated.

STORAGE CYCLE

Since the fuel is now burned at a relatively slow rate, and externally to the catapult, the ship's boilers may be used instead of a special combustion chamber. However, the use of a special chamber need not be ruled out since this would enable catapult

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operation to be independent of the ship's boilers and would permit launching to be undertaken at the minimum notice.

The important feature of the storage cycle is, of course, the accumulator. Taking the contents of accumulator and cylinder together, we may conveniently divide them into (a) the working medium and (b) the storage medium. We will call the system single phase when the working and storage media are both gaseous or both liquid, and two phase when one is liquid and the other gaseous.

The following is an attempt to arrive at the best system for a storage cycle and is based on considerations of size and efficiency for a given useful output. Size and weight are doubly important where the accumulator and cylinder are at flight deck level and the question of efficiency assumes importance when we realize that this controls the size of the auxiliary gear necessary to recharge the system.

The following assumptions will be made:

- (1) The maximum pressure will be the same in each system.

If all systems operate at the same strength level, the lightest unit will be that which involves the smallest total volume. The required volume per unit of energy output will be assessed.

- (2) The minimum pressure is the same in each system.

This is equivalent to saying that each case will provide the same variation in "g" during the launching stroke.

- (3) The useful work output (aircraft K. E.) is the same in each case.

The ratio of recharging work to useful output will be evaluated.

If the energy added to the accumulator during the launching period

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can be neglected we may say

work done = decrease in internal energy.

$$\int_{\text{Initial}}^{\text{final}} p \, dv = - \int_{\text{Initial}}^{\text{final}} c_v \, dt = E$$

SINGLE PHASE SYSTEM

(a) Gas

$$\text{Expansion work} = \frac{p_1 v_1 - p_2 v_2}{n-1} = E$$

$$\text{or } \frac{v_2}{E} = \frac{n-1}{p_2 \left[\left(\frac{p_1}{p_2} \right)^{\frac{n-1}{n}} - 1 \right]}$$

The volume v_2 is the total volume of accumulator and cylinder. If the upper pressure is P atmospheres and a pressure variation of 10% is allowed, then, taking $n = 1.4$

$$\frac{v_2}{E} = \frac{0.00699}{P} \quad (\text{ft. lb. sec. units})$$

It is assumed that the cylinder is completely scavenged and the accumulator is recharged with gas initially at atmospheric pressure.

Work necessary to raise atmospheric volume v_c from pressure p_0 to pressure p_2 and volume v_c where

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V_c = (volume of accumulator)

$$= \frac{P_2 V_c - P_0 V_c}{n-1} = \frac{P_2 V_c}{n-1} \left[1 - \left(\frac{P_2}{P_0} \right)^{\frac{1-n}{n}} \right]$$

Work necessary to raise volume $V_0 + V_c$ from P_2 to P_1 and volume $V_1 = E$

$$\text{Total recharging work} = E + \frac{P_2 V_c}{n-1} \left[1 - \left(\frac{P_2}{P_0} \right)^{\frac{1-n}{n}} \right]$$

$$\text{Ratio } \frac{\text{recharging work}}{\text{work output}} = 1 + \frac{\left[1 - \left(\frac{P_2}{P_1} \right)^{\frac{1}{n}} \right]}{\left[\left(\frac{P_1}{P_2} \right)^{\frac{n-1}{n}} - 1 \right]} \left[1 - \left(\frac{P_2}{P_0} \right)^{\frac{1-n}{n}} \right]$$

(b) Liquid

Since the compressibility of most liquids is low, the value of $\int P dv$ will be low and extremely large volumes would be required. It is not necessary to show this analytically and the system can be discarded.

TWO PHASE SYSTEMS

(a) Gaseous storage medium - liquid working medium.

The volume of liquid must be $\geq V_c$ so that the gas volume before expansion $\approx V_2 - V_c$, and V_0 after expansion.

$$\text{Expansion work done} = \frac{P_1 V_1 - P_2 V_2}{n-1} = E \quad \text{as before}$$

$$\text{and } V_2 = \frac{E(n-1)}{P_2 \left[\left(\frac{P_1}{P_2} \right)^{\frac{n-1}{n}} - 1 \right]}$$

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in this case v_2 will be v_1 and equals $.00688 \frac{E}{P}$ cu. ft.

the volume before expansion $(v_1 - v_2) = v_1 = .00654 \frac{E}{P}$

so that total volume $= v_1 + v_2 = 2v_1 - (v_1 - v_2) = .00722 \frac{E}{P}$

The energy required to be replaced will be E plus the pressure energy of the fluid; since the latter is small the efficiency will approach 100% and the ratio of recharging work-to-work output will approach 1.

(b) Liquid storage - gaseous working medium.

If a body of liquid is raised to its saturation temperature at a pressure P_1 and the pressure allowed to fall to P_2 the saturation temperature will decrease and some of the liquid will be converted to vapor. The drop in pressure is accomplished by allowing the vapor to expand into the launching cylinder and after the launch, the cylinder is assumed to be complete scavenged.

The work done may be calculated from the change in internal energy and can be found from thermodynamic tables if the process is assumed to be isentropic.

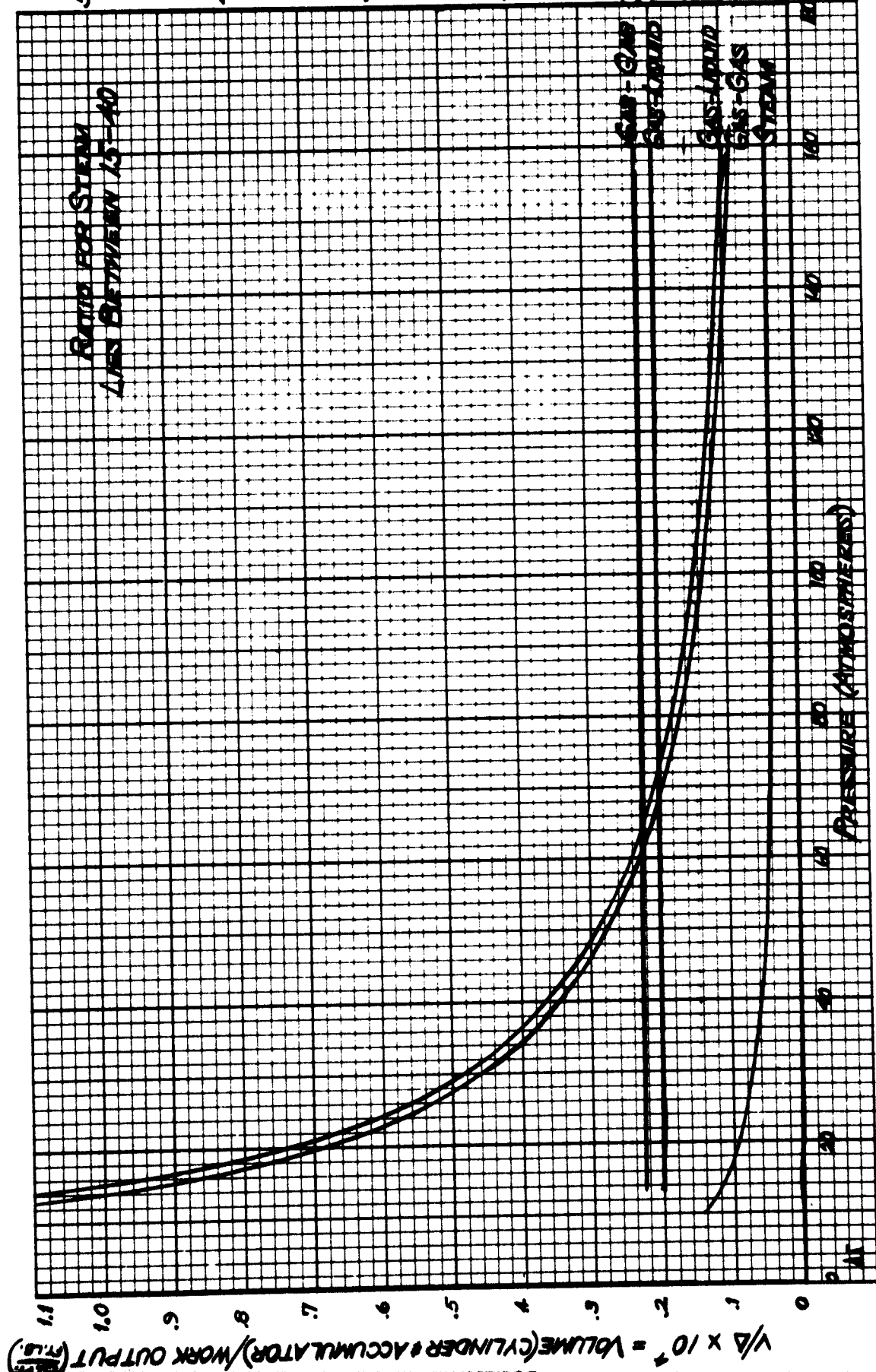
The work necessary to recharge the system is the drop in the total enthalpy of the accumulator.

The necessary information for a water-steam system has been obtained from steam tables and is shown graphically in Figure 7

Comparison of the various systems on the basis illustrated reveals that a catapult operating at pressures in the region of 10-50 atmospheres would most effectively use a steam cycle. Use of steam would result in the smallest unit but a very inefficient unit. At higher pressures, the gas-gas and gas-liquid systems

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RECHARGING WORK / WORK OUTPUT



VOLUME OF CYL. & ACCUMULATOR FOR VARIOUS SYSTEMS

Figure 7

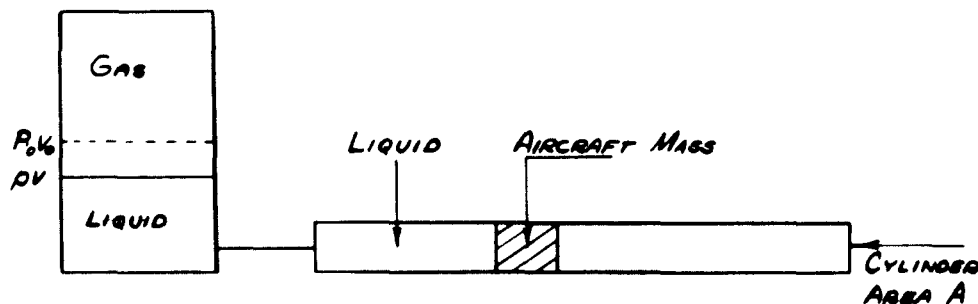
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give competition on a weight basis while maintaining a strong lead on an efficiency basis.

Although the gas-liquid system requires a slightly greater volume than the all gas unit, it has definite advantages in its greater efficiency and the ease with which liquids can be pumped to high pressures. Operations at high pressures with an all gas system would introduce difficult pumping problems.

Aside from pumping and valve losses, inefficiencies will arise in the gas-liquid system from two main sources. There will be a loss of head due to the friction of the liquid in the launching cylinder, and some of the energy delivered by the expanding gas will appear as kinetic energy in the moving liquid.

An approximate analysis of the system can be made as follows.



Assuming the contents of the accumulator to have negligible velocity and ignoring the static head of the liquid, the motion of the cylinder contents can be described by the equation.

$$(m_a + PAx) \frac{d^2x}{dt^2} + f\rho \frac{A}{D} x \left(\frac{dx}{dt} \right)^2 = PA = AP_0 \left(\frac{h_0}{h_0 + Ax} \right)^n$$

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where m_s = mass of aircraft + shuttle etc.

ρ = fluid density

A = piston area

D = piston diameter

f = friction factor at appropriate velocity

p = fluid pressure.

This equation cannot be integrated explicitly but an approximate numerical solution has been obtained using the following values.

P_0 = 2000 psi = 288,000 lbs./sq. ft.

V_0 = 1800 cu. ft.

ρ = 1.94 slugs/cu. ft. (water)

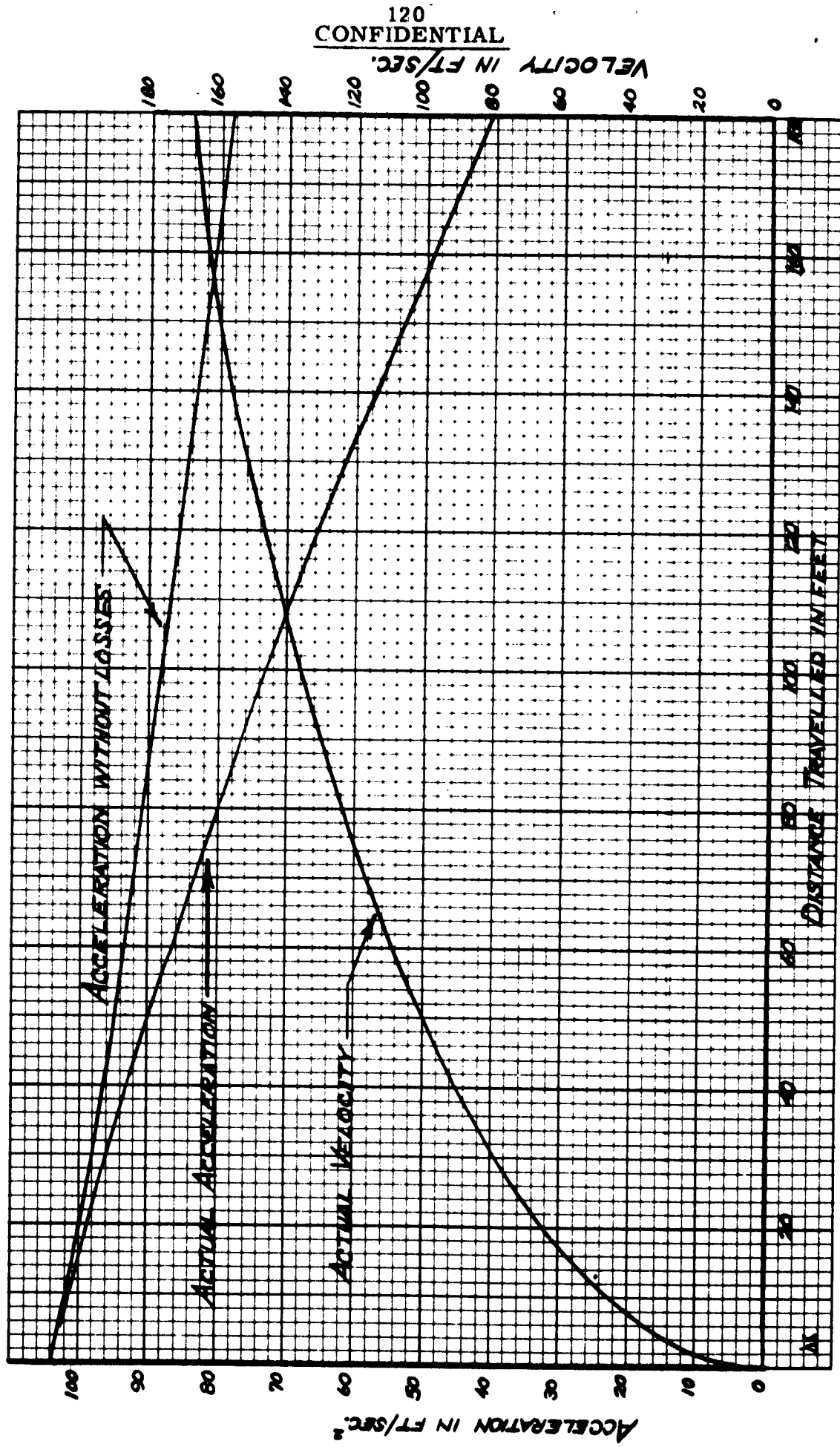
D = 1 ft.

A = $\frac{\pi}{4}$ sq. ft.

m_s = 2180 slugs (70,000 lbs. aircraft)

Figure 8 illustrates this solution by plotting the acceleration of the aircraft versus travel and comparing it with the acceleration for a weightless, frictionless, fluid. The areas under these two curves represent respectively the useful and ideal work done and give a measure of the efficiency of the system.

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CHARACTERISTIC OF WATER SLOTTED-CYLINDER CATAPULT

CONFIDENTIALWATER POWERED LINEAR TURBINE

A two-phase gas-liquid system has been suggested in connection with a linear turbine and is discussed in Report No. M5231 (Naval Aircraft Factory). In this system the potential energy is converted to kinetic energy by accelerating the liquid. The liquid impinges on blades on the moving shuttle and produces the necessary acceleration.

A single stage turbine of this type has a theoretical efficiency of 37% so that, for the same conditions of pressure and energy output it would require a considerably larger accumulator than a slotted cylinder mechanism. Staging of the turbine increases the efficiency to 57% but involves a row of stationary blades running the full length of the launching track.

Aside from questions of efficiency and size, a comparison of this unit with the slotted cylinder must take into account the mechanical difficulties involved in each system.

The use of high operating pressures will give rise to strength and sealing problems in the case of slotted cylinder, and to break up of the water jet in the case of the turbine. One point might be mentioned in the matter of jet velocity and this is the possibility of a serious water hammer arising from rapid closing of the control valves in a stream of 700-1400 ft. /sec.

LINEAR STEAM TURBINE

Attention has already been drawn to the extremely low efficiency of the slotted cylinder steam system in which steam is

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used non-expansively. The possibility of using steam expansively in a linear impulse reaction turbine might appear attractive but this can be shown to be even less efficient than the non-expansive system.

For maximum efficiency in an impulse turbine, the blade speed should be half the nozzle velocity and in an impulse-reaction turbine, the velocities should be equal (assuming equal pressure drops across nozzle and bucket). For the blade velocities involved in catapulting, large quantities of low pressure steam would be necessary to give a reasonable efficiency while providing the required thrust. The quantity of steam can be reduced by increasing the pressure but this reduces the overall thermal efficiency.

A single impulse stage, operating with steam expanded from 500 psi to atmospheric pressure, would require close to 1000 lbs. of steam per second to give a thrust of 240,000 lbs. when the shuttle is stationary. This is on the basis of zero inlet and exit blade angles. As the shuttle gathers speed, the weight of steam necessary to provide this thrust would increase somewhat.

Since the steam velocity resulting from a 500 psi drop in a single stage is of the order 4000 ft./sec. and the maximum shuttle speed is only 170 ft./sec., the resulting efficiency will be extremely low and can only be raised by staging the pressure drop. This will have the added advantage of decreasing the weight of steam necessary to provide the thrust. However, the problem of staging such a turbine, requiring, as it does, thousands of blades distributed along the track, would rule out the system as a feasible catapult.

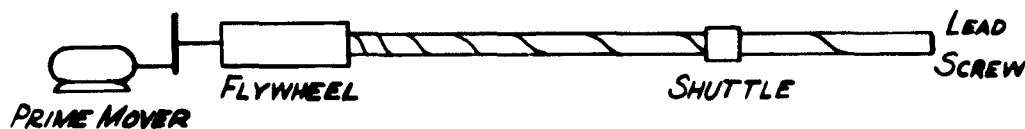
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CONFIDENTIALFLYWHEEL DRIVE

The use of a flywheel as an energy storage medium has been considered in connection with reeved cable driven catapults and is discussed in Report No. M-5030 (NAMC).

Earlier in this report the undesirable properties of a cable drive have been pointed out, and if these are to be avoided, the flywheel must deliver its energy through some other form of linkage - possibly an electrical or a rigid mechanical system. Since the flywheel will be rotating at high speed initially, and the aircraft will be at rest, any clutching arrangement will have a high slip and will result in a large energy loss. To avoid slip, the linkage must provide a variable gear ratio and this ratio must vary in such a way as to take account of the simultaneous increase in aircraft speed and decrease in flywheel speed.

Consider a linkage in the form of a long lead screw and flywheel coaxially mounted.



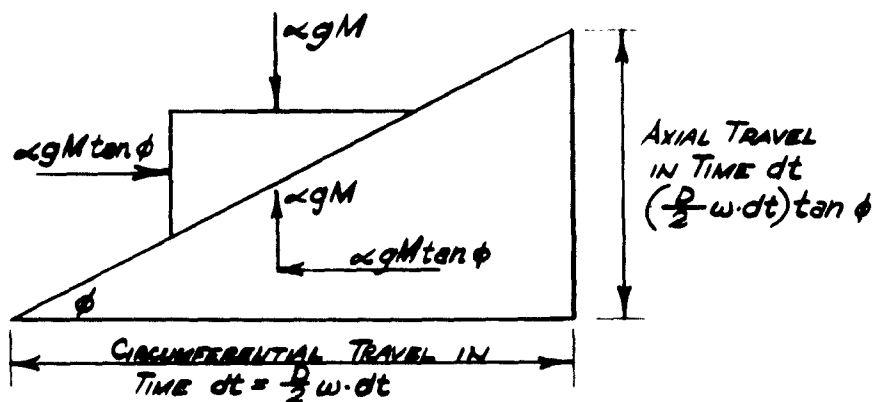
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The pitch of the screw must vary continuously along its length, i.e., the developed "thread" must have a form somewhat as shown below.



If a mass M (the aircraft) is to be accelerated at a constant acceleration αg , the inertia (i.e. launching) force is $\alpha g M$, so that the forces on the mass and screw are

NOTE. Friction neglected



The reaction torque on the lead screw is $\frac{D}{2}(\alpha g M) \tan \phi$ where D is the diameter of the screw.

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In an interval of time dt , the screw will rotate through an angle $\theta = \omega \cdot dt$ where ω = angular velocity in rads/sec. at the instant considered. The circumferential travel of the screw $\theta \cdot \frac{D}{2} = \frac{D}{2} \omega \cdot dt$ and the advance of the mass is $(\frac{D}{2} \omega \cdot dt) \tan \phi$ i.e. its axial velocity is $\frac{D}{2} \omega \tan \phi$. Differentiating this with respect to time gives the axial acceleration

$$\frac{D}{2} \left[\omega \sec^2 \phi \cdot \frac{d\phi}{dt} + \tan \phi \cdot \frac{d\omega}{dt} \right] = \alpha g \text{-----} (1)$$

The angular acceleration of the flywheel and screw caused by the reaction torque is given by

$$-I \cdot \frac{d\omega}{dt} = \frac{D}{2} (\alpha g M) \tan \phi \text{-----} (2)$$

where I is the mass moment of inertia of the rotating parts and the minus sign indicates a retardation.

$$\text{From } (1) \text{ \& } (2) \quad \omega \frac{d\phi}{dt} = \frac{\frac{2\alpha g}{D} - \tan^2 \phi \left(-\frac{\alpha g DM}{2I} \right)}{\sec^2 \phi}$$

the left-hand side may be written as $\omega \cdot \frac{d\phi}{d\theta} \cdot \frac{d\theta}{dt} = \omega^2 \frac{d\phi}{d\theta}$
 so that $\omega^2 = \frac{\frac{2\alpha g}{D} \cos^2 \phi + \frac{\alpha g DM}{2I} \sin^2 \phi}{\frac{d\phi}{d\theta}}$

differentiation with respect to θ gives

$$2\omega \frac{d\omega}{d\theta} = \frac{\left(\frac{d\phi}{d\theta} \right)^2 \left(-\frac{2\alpha g}{D} \cdot 2 \cos \phi \sin \phi + \frac{\alpha g DM}{2I} 2 \sin \phi \cos \phi \right)}{\left(\frac{d\phi}{d\theta} \right)^2} - \frac{\frac{d^2\phi}{d\theta^2} \left(\frac{2\alpha g}{D} \cos^2 \phi + \frac{\alpha g DM}{2I} \sin^2 \phi \right)}{\left(\frac{d\phi}{d\theta} \right)^2}$$

Equation (2) may be written as

$$\frac{d\omega}{d\theta} \cdot \frac{d\theta}{dt} = -\frac{\alpha g MD}{2I} \tan \phi$$

$$\text{or } 2\omega \frac{d\omega}{d\theta} = -\frac{\alpha g MD}{I} \tan \phi$$

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equating the two expressions

$$-\frac{gDM}{2I} \tan \phi = \frac{\left(\frac{d\phi}{d\theta}\right)^2 (2 \cos \phi \sin \phi) \left(\frac{DM}{2I} - \frac{2g}{D}\right) - \frac{d^2\phi}{d\theta^2} \left(\frac{DM}{D} \cos^2 \phi + \frac{gDM}{2I} \sin^2 \phi\right)}{\left(\frac{d\phi}{d\theta}\right)^2}$$

which may be rearranged

$$\frac{\frac{d^2\phi}{d\theta^2}}{\left(\frac{d\phi}{d\theta}\right)^2} = \frac{\frac{DM}{I} + \left(\frac{DM}{I} - \frac{4}{D}\right) \cos^2 \phi}{\frac{1}{2} \left[\frac{DM}{I} - \left(\frac{DM}{I} - \frac{4}{D}\right) \cos^2 \phi \right]} \cdot \frac{\sin \phi}{\cos \phi}$$

Taking the left-hand side and multiplying by $\frac{d\phi}{d\theta}$

$$\frac{\frac{d^2\phi}{d\theta^2} \cdot \frac{d\phi}{d\theta}}{\left(\frac{d\phi}{d\theta}\right)^2} = \frac{\frac{1}{2} \frac{d}{d\theta} \left[\left(\frac{d\phi}{d\theta}\right)^2 \right]}{\left(\frac{d\phi}{d\theta}\right)^2}$$

integrating gives $\frac{1}{2} \ln \left(\frac{d\phi}{d\theta} \right)^2$ or $\ln \left(\frac{d\phi}{d\theta} \right)$

Taking the right-hand side, multiplying by $\frac{d\phi}{d\theta}$ and integrating

w.r. to θ gives

$$2 \int \frac{\left[\frac{DM}{I} + \left(\frac{DM}{I} - \frac{4}{D} \right) \cos^2 \phi \right] \sin \phi}{\left[\frac{DM}{I} - \left(\frac{DM}{I} - \frac{4}{D} \right) \cos^2 \phi \right] \cos \phi} \cdot d\phi$$

making the substitution $\cos \phi = x$ this transforms to

$$-2 \int \frac{a + bx^2}{(a - bx^2)x} \cdot dx \quad \text{where} \quad \begin{aligned} a &= \frac{DM}{I} \\ b &= \frac{DM}{I} - \frac{4}{D} \end{aligned}$$

which integrates to $\ln \left[-c \frac{(a - bx^2)^2}{2bx^2} \right]$

equating $\frac{d\phi}{d\theta} = -c \frac{(a - bx^2)^2}{2bx^2}$

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by the substitution of $t = \tan \phi$ and integrating.

the expression transforms to

$$-c\theta = \frac{b}{a} \int \frac{dt}{[t^2 + \frac{a-b}{a}]^2}$$

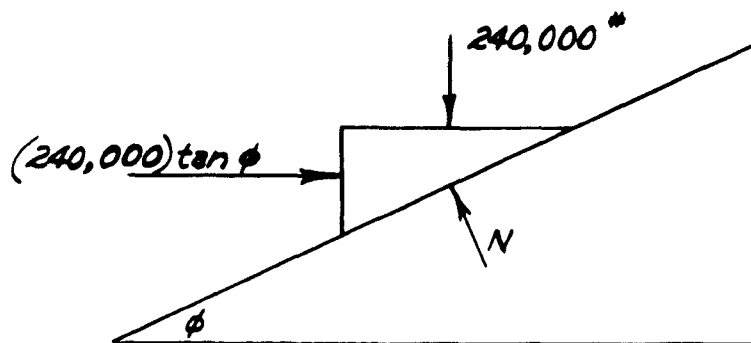
which integrates to

$$-c\theta = \frac{\tan \phi}{2K[K + \tan^2 \phi]} + \frac{1}{2} \left(\frac{1}{K} \right)^{\frac{1}{2}} \tan^{-1} \left[\left(\frac{1}{K} \right)^{\frac{1}{2}} \tan \phi \right] + c_2$$

where $K = \frac{a-b}{a} = \frac{4I}{D^4M}$ and c, c_2 are constants of integration.

The constants can be determined from the boundary conditions that the acceleration is $3g$ and the velocity at the end of the stroke is 170 ft./sec. Assume that the weight of aircraft with shuttle and linkage is $80,000$ lbs.

$$\text{Force} = ma = \frac{80,000 (3g)}{g} = \underline{240,000 \text{ lbs.}}$$



$$N = \frac{240,000}{\cos \phi}; \quad \text{where } \tan \phi < 0.14 \frac{N}{\cos \phi} \approx N \quad (\text{within } 1\%)$$

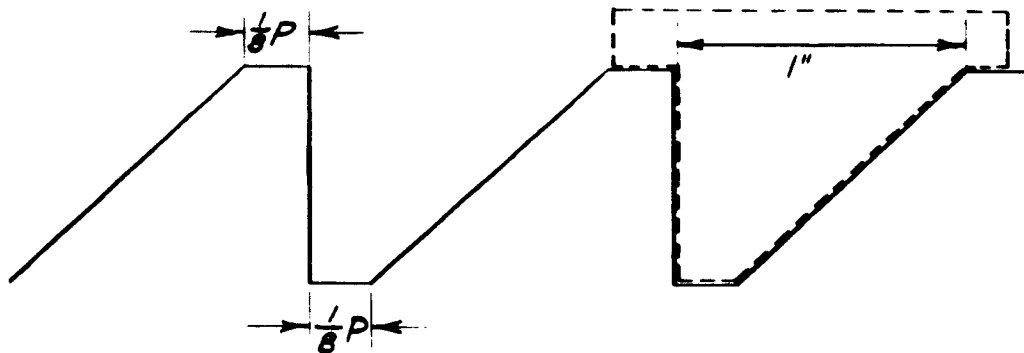
Assume that the allowable shear in the thread is $20,000$ psi.

$$\frac{240,000 \text{ lbs.}}{20,000 \text{ psi.}} = 12 \text{ sq. in.}$$

Thus approximately 12 sq. in. of root area is required.

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Assume that buttress-type threads will be used.



$$\frac{7}{8}P = \frac{5}{4}'' \quad \therefore P = 1\frac{1}{2}''$$

say that $1\frac{1}{2}''$ is minimum pitch.

At the beginning of the stroke, the acceleration is $3g$ and the minimum pitch is $1\frac{1}{2}''$.

One revolution will cause the shuttle to move $1\frac{1}{2}''$ at $3g$ so that

$$\text{time required will be } \sqrt{\frac{1.5}{12} \left(\frac{1}{1.5 \times 32.2} \right)} \quad \text{from } s = \frac{1}{2} at^2$$

$$\therefore t = \frac{1}{19.7} \text{ sec.}$$

The cylinder must be rotating at 19.7 rps

$$\text{or } \omega = 2\pi(19.7) = 124 \text{ rads/sec.}$$

At the end of the stroke, the shuttle must be moving at 170 ft./sec.

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Assume that the maximum allowable pitch angle is 45° , then the required rps will be $\frac{170}{2\pi R}$; where R is the radius of the cylinder.

$$\omega = 2\pi \left(\frac{170}{2\pi R} \right) = \frac{170}{R}$$

In order to launch the aircraft under the specified conditions, the flywheel must transmit 30×10^6 ft. lbs. to it (the aircraft).

Thus
$$\frac{1}{2} \omega_1^2 I - \frac{1}{2} \omega_2^2 I = 30 \times 10^6 \text{ FT.LB.}$$

$$\frac{1}{2} (124)^2 I - \frac{1}{2} \left(\frac{170}{R} \right)^2 I = 30 \times 10^6 \text{ FT.LB.}$$

In a thin cylinder $I = m R_c^2$; where R_c is the mean radius of the cylinder; however, $R \approx R_c$,

$$m = 2\pi R t (l) \frac{\rho}{g}$$

t = thickness, l = length, ρ = density

$$m = 2\pi (150) \frac{490}{32.2} R t = 14,300 R t$$

$$I = 14,300 R^3 t$$

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let the minimum thickness be 3"

$$I = 14300(.25)R^3 = 3580R^3$$

$$\frac{1}{2}(124)^2(3580)R^3 - \frac{1}{2}\left(\frac{85}{R}\right)^2 3580R^3 = 30 \times 10^6 \text{ FT. LB.}$$

$$2750R^3 - 5170R = 3000$$

$$R = 1.6'$$

$$\omega \text{ at end of stroke} = \frac{170}{1.6} = 106 \text{ rad/sec.}$$

$$I = 14,300(.25)(1.6)^3 = 14,700 \text{ lbs. ft. sec.}^2$$

$$\text{K.E. beginning of stroke} = \frac{14700}{2}(124)^2 = 113 \times 10^6 \text{ ft. lb.}$$

$$\text{K.E. end of stroke} = \frac{14700}{2}(106)^2 = 83 \times 10^6 \text{ ft. lbs.}$$

$$\text{weight} = 2\pi (1.6) 150 (1/4) (490) = 185,000 \text{ lbs.}$$

The flywheel device would thus weigh considerably less than the conventional launching device.

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There are several things to be considered in the evaluation of the flywheel device.

- 1.) The method of introducing the shuttle on to the screw.
- 2.) The problem of maintaining bearing between the threads and the follower on the shuttle.
- 3.) A method of supporting the 150' long cylinder.
- 4.) Source of power to put energy into the flywheel.
- 5.) Influence rotating cylinders may have on the behavior of the ship.

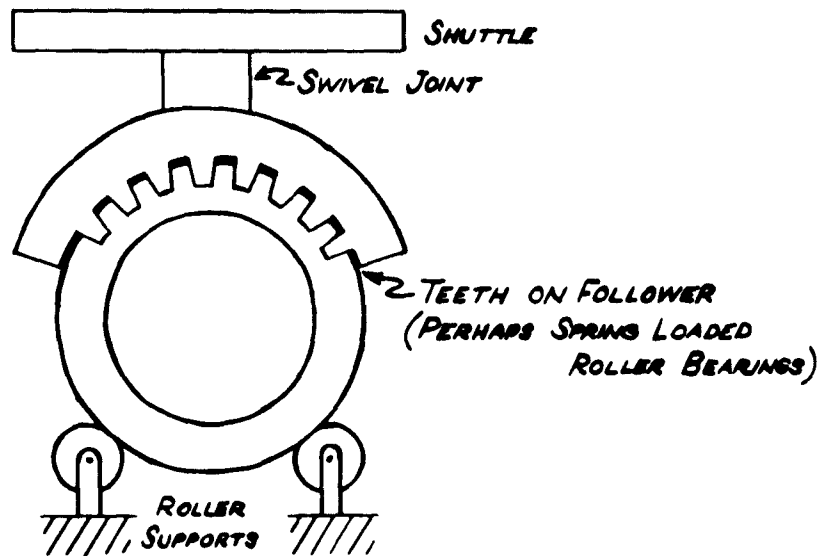
In introducing the shuttle on to the screw, the shuttle could be allowed to move five or six inches before taking up the full load of the aircraft (this could be done by a hydraulic springing device); thus the initial impact loading on the follower and threads would be much reduced and the threads would not have to be very thick at the beginning of the stroke where the pitch is small. A switching device could be used to engage the shuttle and start it moving from rest.

Since the pitch of the thread will vary continuously along the length of the cylinder, the follower will have to be mounted on a swivel joint which will allow it to follow the thread; it will have to be spring loaded in order to maintain approximately constant bearing. Also since the surface of the thread will have some curvature, roller bearings may have to be used to maintain contact between the shuttle and the thread.

Support for the 150' long cylinder could be given by rollers placed at appropriate intervals along the length - perhaps

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as shown in the cross-section below



The amount of power required to build up the speed of the flywheel will depend on the time allowed between launchings. If 30 seconds are allowed, approximately 2000 H. P. will be required and this amount can be supplied easily by electrical or steam apparatus.

A force perpendicular to the axis of rotation will be exerted on the supports whenever the ship pitches. This force is due to the gyroscopic action of the cylinder and can be computed in the following manner.

The moment of a gyroscopic couple (C) is equal to $I\omega\Omega$ where I = polar moment of inertia of mass, ω = angular velocity of spinning and Ω = angular velocity of precession.

$$\text{Thus } \text{FORCE} = \frac{I\omega\Omega}{150} = \frac{14700(124)\Omega}{150} = 12,200\Omega$$

If we assume that the ship pitches 10° in 5 seconds

$$\Omega = \frac{10}{360} (2\pi) \frac{1}{5} = 0.034 \text{ rads./sec.}$$

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Force = $12,200 (.034) = 415 \text{ lbs.}$ $\therefore$ Torque = $67,500 \text{ lbs.ft.}$

It does not seem likely that a force of 415 lbs. would affect the behavior of the ship in any significant fashion.

From a consideration of the foregoing remarks, the fly-wheel device appears to be a feasible launching mechanism and it is suggested that the matter be investigated further.

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CONFIDENTIALELECTRO-MAGNETIC SYSTEMS

Mechanical forces are associated with the storage of electrical energy in either an electrostatic or a magnetic field. The former phenomenon can be ruled out immediately as a means for producing large forces, because energy cannot be stored compactly in an electrostatic field, as previously shown. In what follows, attention will be devoted entirely to devices using the magnetic field as the force-producing medium. The basic phenomena are the forces exerted on ferromagnetic material tending to align it with or bring it into the position of the densest part of the magnetic field, and the forces exerted on current-carrying conductors in a magnetic field or between the magnetic fields produced by current-carrying conductors.

There are several conceivable schemes by which these phenomena might be used for aircraft launching. These schemes can be classified into two types, with several variations in each type: (1) devices for producing linear motion directly, and (2) devices for producing rotary motion (as in conventional electric motors), together with an arrangement of drums and cables.

One type of linear launching device which may be considered briefly is a variation of the electric gun. It might consist of a long solenoid with an iron plug traveling down its length and with some kind of an arrangement of sliding contacts to energize the section of the solenoid winding in front of the iron plug and de-energize the section behind it. The general nature of the device is shown in Figure 9 in which the small circles indicate in cross-

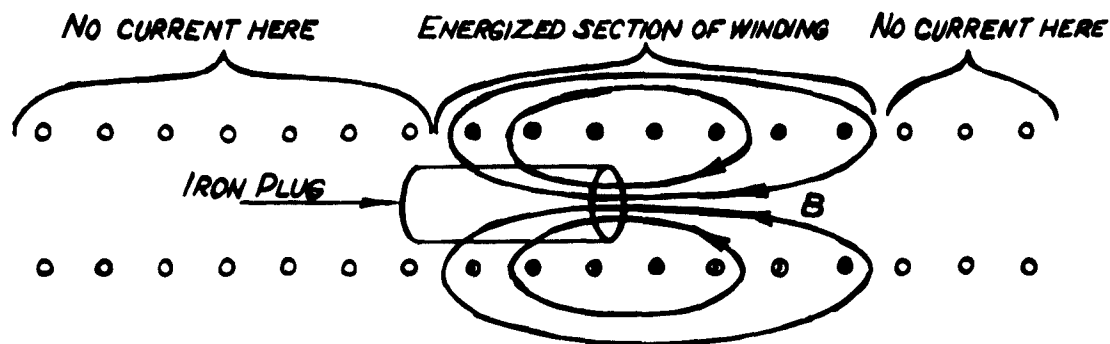
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Figure 9 - Schematic arrangement of electric gun.

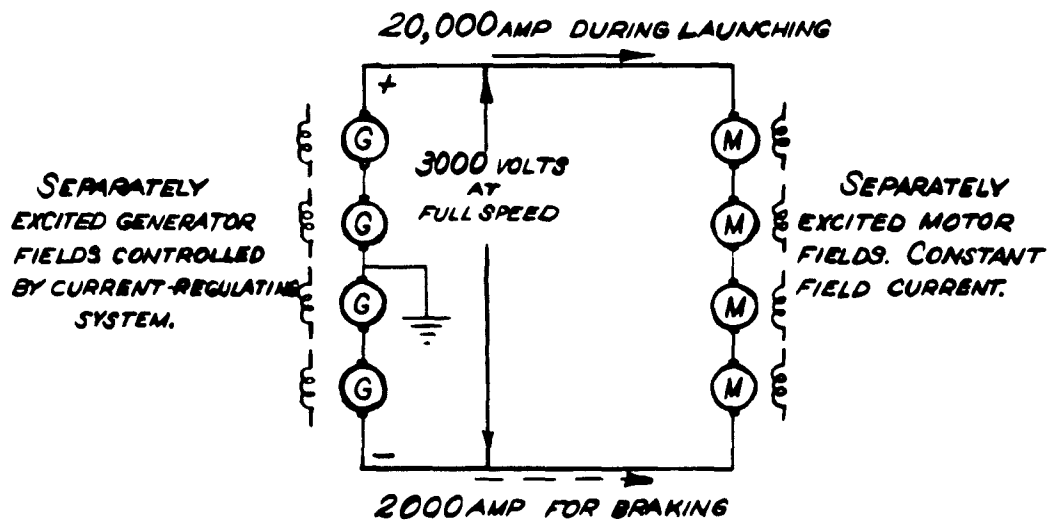


Figure 10 - Series d-c system.

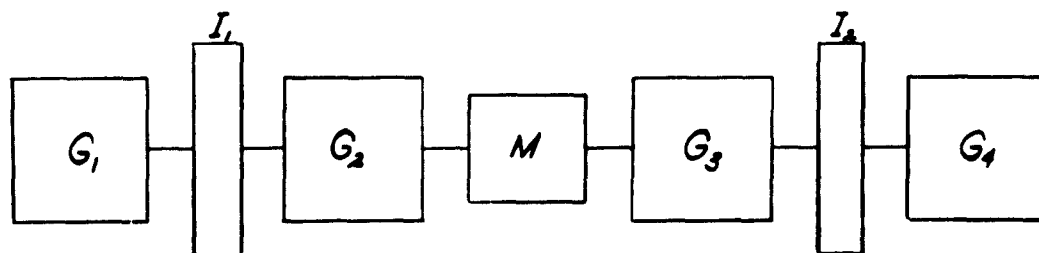


Figure 11 - Schematic arrangement of motor-generator set with four generators and two fly wheels.

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section a portion of the solenoid winding. Current directions in the energized portion of the winding are shown by dots and crosses indicating the points and tails of arrows pointing in the direction of the current. The magnetic field B entering the right-hand end of the iron plug produces a force f pulling it to the right. In English units

$$f = 0.0139B^2A$$

where f is the force in pounds, B is the flux density in kilolines per square inch, and A is the area of the right-hand surface of the plug in square inches. With most magnetic materials, the flux density cannot be much above 100 kilolines per square inch, because of magnetic saturation. If the retarding forces exerted by flux leaving the plug are neglected, the surface area on the right-hand face of the plug necessary to create a force of 210,000 pounds (70,000-pound aircraft accelerated at 3g) is, from Equation 4,

$$A = \frac{f}{0.0139B^2} = \frac{210,000}{(0.0139)(100)^2} = 1510 \text{ in.}^2$$

or 10.5 square feet. This, of course, neglects the very considerable weight of the iron plug and the force necessary to accelerate it. For example, if the plug were four feet long it would weigh about 20,000 pounds. Hence this scheme or any variation of it based on the attraction of iron or any other ferromagnetic material by a magnetic field is impracticable. The magnetic force intensities which can be exerted by this phenomenon are only about 140 pounds per square inch.

Another possibility for a linear launching device is a flattened-out electric motor utilizing the forces between the magnetic fields produced by two sets of current-carrying conductors

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on the stationary and moving parts respectively. Considerably greater forces can be produced this way than by the attraction of iron in a magnetic field. One launching device of this type has already been built by Westinghouse and is called the Electropult. It works on the induction-motor principle and consists of a shuttle car riding on a track about 500 ft. long. The track is a developed squirrel-cage winding and the shuttle car which is 12 ft. long, 3 1/2 ft. wide and 5 1/2 inches high, has a developed 3-phase 8-pole winding. It is designed to launch a 20,000 pound aircraft at 90 miles per hour in a 400 foot run, the remaining 100 feet of track being required to stop the shuttle car by electrical braking. The duty cycle is one plane per minute for one hour. Because of heating, three cars are used, each in turn launching 20 planes. The power supply is three, 4-pole, 80-cps, 3-phase generators each driven by a 1750-hp, 2400-rpm aircraft engine with a flywheel. Each generator has a short-time rating of 1200 kw. The three generators are operated in parallel.

This system is small compared with the one being investigated in this report. The acceleration is only about 22 feet/sec.², the force developed about 15,000 pounds and the power at top speed about 3600 hp.

For the sake of completeness, the possibilities of flattened-out motors of other types should be mentioned. Conceivably, a flattened-out synchronous motor could be built, but this would require a power supply whose frequency could be increased at a uniform rate. A full-size generator driven by a prime mover

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whose speed could be increased at a uniform rate would be needed. If such a variable-speed prime mover were available, it would be better to use it to drive a wind-up drum of some kind and use cables to transmit the force directly to the airplane being launched, thereby eliminating the generator and motor. There would also be electrical difficulties in synchronizing the motor and generator at the very low speeds at the beginning of the launching run.

It is possible to conceive of a flattened-out d-c motor with a stationary armature and commutator laid out flat and a car carrying field poles and brushes sliding along the commutator. Or the inverse arrangement can be imagined. However, in view of the difficulties in building even conventional types of d-c motors in very large sizes this scheme would seem to be impracticable.

The only linear electrical launching device with any possibilities, then, seems to be the Westinghouse Electropult, or some variation of it. The weight, size, excessive losses inherently associated with a squirrel-cage induction motor operating with a large value of slip, and complications in increasing the power by a factor of about 20 seem to make this scheme unattractive.

This leaves rotating electric motors with windup drums and cables to be considered. The synchronous motor with variable-frequency supply seems to be out of the question, for reasons already given.

Induction motors can easily be built in large sizes and have been applied successfully to very large wind-tunnel drives. The difficulty, however, is that an induction motor is not readily

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adaptable to variable-speed drives. The speed of an induction motor in revolutions per second is given by

$$n = n_s(1-s)$$

where n_s is the synchronous speed, and s is the slip expressed as a fraction of synchronous speed. The synchronous speed in revolutions per second is

$$n_s = \frac{2f}{N}$$

where f is the frequency of the supply in cycles per second, and N is the number of poles of the motor winding. The synchronous speed can be varied by varying the frequency but this brings up the difficulty of obtaining a variable-frequency supply. Pole changing is commonly used to obtain two, and sometimes as many as four, synchronous speeds by sharing stator-winding connection, but this scheme appears to be practicable only for steady-state operation at any one of several selected speeds. Variation of the slip s appears to be the only practicable means of speed control for any aircraft-launching mechanism.

But a serious disadvantage of an induction motor used in the normal way with its rotor short circuited is its inherent inefficiency when running at a large slip. In this respect, it is exactly analogous to a slipping clutch. When running at a slip s (expressed as a fraction of synchronous speed) the fraction s of the electric power supplied to the primary (stator) winding is generated as electric power in the secondary (rotor) winding. If the secondary is short-circuited as in the normal induction motor, this power is then converted to I^2R heat and lost.

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This means, then, that an induction motor with short-circuited rotor and constant-frequency power supply accelerating a mass (including its own rotor) from standstill to near synchronous speed dissipates an amount of energy in rotor heating equal to the kinetic energy it has imparted to the moving mass. The energy efficiency cannot possibly be greater than 50 per cent. In a system as large as the one under consideration, this inherent property of the induction motor with short-circuited rotor is indeed a serious disadvantage. Not only does it cause a very serious heating problem in the motor but also it doubles the energy required from the power source and doubles the size of the energy-storage equipment required to smooth out the power flow.

Numerous schemes have been invented for recovering the slip-frequency electric power generated in the rotor. Basically they all comprise a means for introducing adjustable voltage of slip frequency into the rotor circuits of a wound-rotor induction motor whose speed is to be regulated. They all involve at least one auxiliary variable-speed commutator-type machine of some kind. The slip-frequency electric power generated in the rotor windings of the main induction motor is delivered to the auxiliary machines where it is either converted to mechanical power and applied to the drive or changed back to line-frequency power and returned to the line. For applications where the speed range is not too wide, or when the power drops off rapidly with decreasing speed (as in wind tunnels, for example), the auxiliary machines can be considerably smaller than the main induction motor. The aircraft-launching mechanism, however,

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requires full torque over a speed range from zero to full speed. Under these conditions the auxiliary machines would have to be as big as the main induction motor. Therefore, any slip-energy recovery scheme seems to be impracticable.

Finally, d-c motors should be considered. Undoubtedly a d-c system could be built. In many ways, it would be similar to the d-c reversing blooming mill drives used in steel mills. Higher speed motors would be used, however. Such a system would require several motors operating mechanically in tandem, because of the difficulty of building very large d-c machines. The motors could be supplied by several d-c generators which could be equipped with flywheels. Since large d-c machines cannot be built for voltages much above 1000 volts, the system could be arranged as a series circuit. For example, Figure 10 shows four motors operating mechanically in tandem and electrically with their armatures in a series loop supplied by four generators. The four generators could be part of a flywheel motor-generator set, as shown schematically in Figure 11. The motor *M* of Figure 11 could be a wound-rotor induction motor supplied from the ship's electrical system and with sufficient resistance inserted in its rotor circuits so that the speed of the motor-generator set would drop to, say, 85 per cent of synchronous speed during launching and thereby allow the inertia of the set to give up some of its stored energy so as to reduce the peak load on the ship's electrical system. The motor *M* can then be made considerably smaller than the combined generators. This arrangement is much like that used in reversing blooming mills.

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In general it may be said that any catapult system based on conventional rotating electrical machinery would be bulky, involving large topside weights and undesirable cable drives. Unconventional "flattened out" electrical machines, while eliminating cable drives, are quite massive and introduce complicated control systems. In short, electrical catapults for shipboard use may be ruled out entirely.

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CONFIDENTIALGENERAL CONCLUSION AND RECOMMENDATIONS

There does not appear to be a more satisfactory method of applying arresting loads than the conventional tail hook assembly. Although elimination of the tail hook arrangement would allow strengthening of the undercarriage to withstand arresting loads, a more satisfactory arrangement from a weight standpoint is the use of undercarriageless aircraft in conjunction with a flexible deck and having a conventional tail hook.

The final disposition of the energy given up by the arrested aircraft should be as thermal energy to a liquid sink.

Engines which carry out the conversion of kinetic to thermal energy by the friction of rubbing surfaces will, in general, involve extremely high surface temperatures and, as a result, will no doubt behave erratically.

The standard hydraulic engines would give improved performance if the effective inertia of the gear were reduced. Some modifications are suggested in the report.

A rotary shear device would be bulky and structurally complex while offering no advantages over conventional gears. A braked hydraulic coupling would require an elaborate control device to give a desirable arrest characteristic and, in common with all rotary devices would introduce large inertia loads.

A system of coupled shock absorbers as described in the report seems the only novel gear offering any promise. It has the advantage of fairly low inertia and would take up comparatively little space. However, its advantages are offset by some structural

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difficulties. It is nevertheless felt to be worthy of further consideration.

Electro-magnetic devices are, in general, extremely heavy and require auxiliary power. It is probably safe to say that electrical equipment may be discarded in favor of hydraulic and mechanical apparatus.

In a catapult system, the slotted cylinder configuration offers the most promise both for direct cycle and storage cycle operation. For operation on the former system consideration should be given to the air (or oxygen) and fuel oil combination, while a storage cycle based on compressed air storage and fluid actuation appears to have advantages. In general, a direct cycle would have an advantage over the storage cycle since it saves accumulator weight.

A flywheel-lead screw would be an efficient though fairly complex arrangement. It may well find some application where high accelerations with short launching runs are required, for example, in the launching of guided missiles.

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CONFIDENTIALPERSONAL OPINIONS

Throughout this report the aim has been to present facts from which factual conclusions are derived. However, there seems to be a wish for a statement of personal conclusions which may have been developed by the personnel working on the project as a result of their experience. Following the suggestions of the staff of the Ships Installation Branch of BuAer, the following observations are presented:

1.) With Regard to Methods of Launching Aircraft from Carriers

The present program of catapult development would appear to meet all requirements of the foreseeable future. We believe that thinking should be directed toward higher launching velocities and can see no serious basic restriction upon developing the slotted cylinder catapult to give any desired end speed. The launching acceleration could be appreciably higher than that now used and eight times the gravitational acceleration is considered realistic. It would appear reasonable to limit the launching weight to less than 30,000 pounds for the higher take-off velocities. While the steam catapult will require less development to attain new design limits, it is essentially a heavy unit, requires large amounts of steam and demands a warm up period. There appears to be strong logic in favor of supplying the basic energy (combustion of fuel) at the point of use rather than in boilers remote from the catapult. Accordingly there appear to be impelling reasons for accelerating the program for an internal combustion catapult. The problems of securing large volumes of high pressure air and of generating liquid oxygen in shipboard should be actively tackled.

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CONFIDENTIAL2.) With Regard to Methods for Recovering Aircraft on Carriers

It appears to the writers that the greatest problems facing carrier operation lie in the field of aircraft recovery. Not only is the greatest pilot skill required to bring his plane down in an exact location on a small, pitching deck, but the velocity limits in landing are more apparent than those of launching. The penalties in plane weight paid for carrier operation in both the landing gear and the arresting apparatus are much greater than those for catapulting.

There appear to be possibilities for improvement in arresting gears by the development of equipment which has lower reflected stress waves in the pendant. This will require continued analytical and experimental work and the development of new arresting gears. The slotted cylinder arresting gear proposed in this report would appear to deserve careful consideration because it has properties which might significantly reduce reflected stress waves.

Although an increase in deceleration during the arrest would result in smaller landing areas, this step is not to be recommended since, (1) an exact touch-down location cannot be chosen, (2) some deck area is necessary to regain flying speed and control of a plane which fails to hook a pendant, and (3) strengthening the arresting equipment adds a higher percentage to the weight than the catapulting requirements. The necessarily longer run outs associated with increased landing speeds should be accommodated by the carrier. The use of flexible mats as a method of reducing the landing gear requirements would appear to be highly desirable.

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CONFIDENTIAL3.) With Regard to Carrier Development

The problems of landing aircraft at increasingly higher speeds on carriers become so difficult that the relative freedom of the canted deck becomes almost mandatory. With this, the available engagement length should be reduced and it would appear reasonable to restrict the number of cross deck pendants to three or four. By the use of flexible mats to accept high sinking velocities gently, it becomes reasonable to raise the cross deck pendant well above the deck thereby giving the pilot greater freedom of level to compensate for his increased speed.

4.) With Regard to Plane Development

Since the penalty which a plane pays for its catapulting equipment is low, it is recommended that the limitations upon take-off speed be imposed by the ability of the pilot to take high accelerations rather than a limitation of the catapult. To gain advanced experience and to be up to the front in plane development, it would appear desirable to carry through, in as simple terms as possible, the design of a fighter plane suitable for 8 g's acceleration and high take-off velocity. Particular care should be given to trying to secure a low landing speed.

5.) With Regard to Water Landing of Planes

The limits on launching velocity of planes appear to be far above what the present and immediate future can effectively utilize but there appear to be definite limitations on the speed at which a plane can be recovered - both with regard to the arresting system and the time which a pilot has to reach a definite location.

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The question naturally arises as to whether some less restrictive landing system could not be developed. Were the water plane, it would form an admirable surface since it presents nearly ideal deceleration characteristics. However, this perfect condition does not exist but is disturbed by waves. Consideration should be given to whether a high speed plane might be developed which, without too great a penalty, would be capable of landing in the waves. The present F2Y program should be watched closely and further development carried toward this ideal.

An effective carrier striking force might consist of light, high-performance planes launched at a high speed from small carriers and landing on the water. The carrier would serve as a base. Special methods for rapidly recovering the planes aboard the carrier from the sea would have to be developed.

Attention should also be given to developing smooth seas suitable for the landing of planes.

6.) With Regard to Other Areas Requiring Investigation

In addition to the studies relating directly to the design of the plane and the carrier there are some studies which should be made of related matters. In the psychological field further studies should be made of the influence of a constant acceleration for an interval of time upon the alertness and response of humans. The information should be utilized in the program for new planes.

7.) With Regard to Future Studies

The work done on this contract to date, and the discussions with interested personnel, have indicated that studies of the broad type initiated on this subject should be continued by setting up

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a group to study not only the launching and recovery problem, but the whole field of operation from carriers. Such a group could be composed of personnel from a civilian organization advised by representative of the Navy Department, and would evaluate the need for research and development on the various aspects of carrier aviation. The work of the group should be of a general nature and should point out specific problems worthy of intensive study. To be fully effective, the group should inspire vigorous criticism from all Naval Bureaus, since the resulting stimulation would result in a valuable fresh approach.

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