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Report 2121

SOME DESIGN PRINCIPLES OF GROUND EFFECT MACHINES
SECTIONS A, B, C, D, F, AND G

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# DAVID W. TAYLOR NAVAL SHIP RESEARCH AND DEVELOPMENT CENTER

Bethesda, Md. 20084

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## SOME DESIGN PRINCIPLES OF GROUND EFFECT MACHINES
## SECTIONS A, B, C, D, F, AND G

by

Harvey R. Chaplin and Allan G. Ford

See 1473 in back

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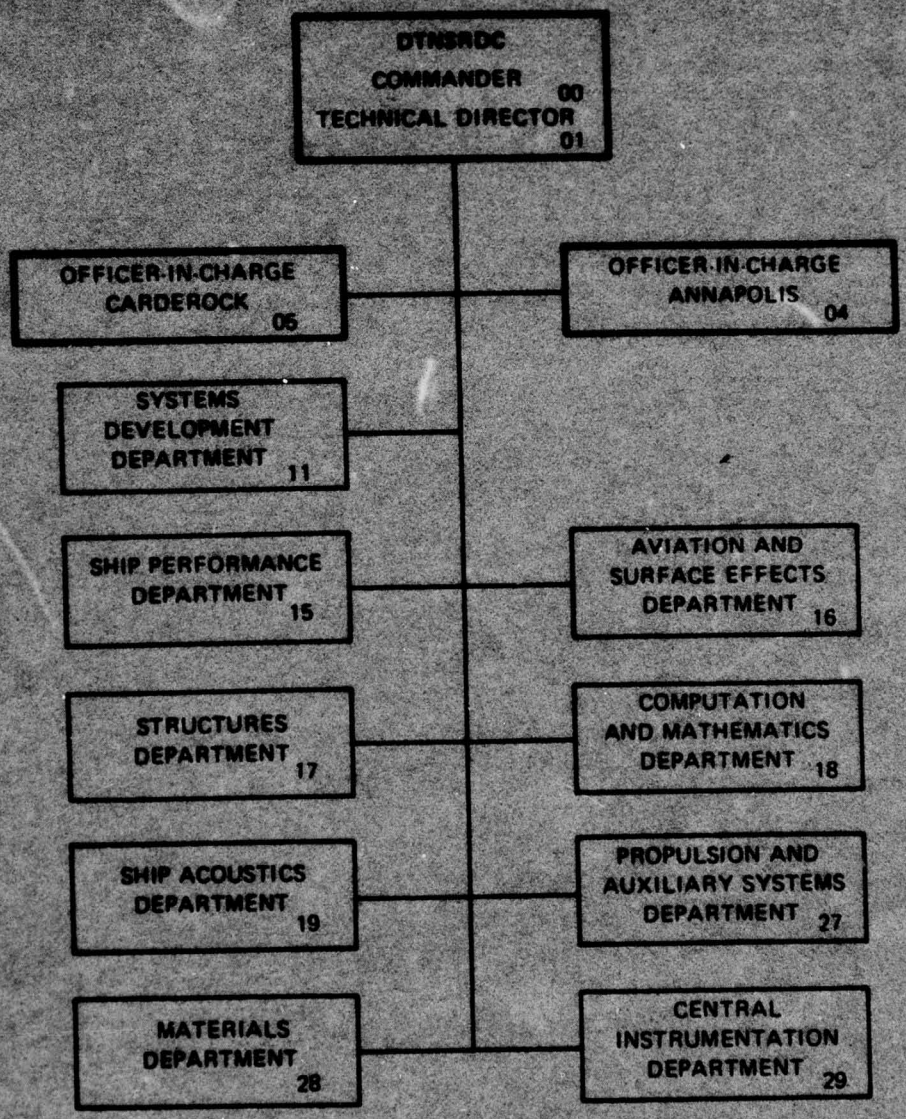
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AVIATION AND SURFACE EFFECTS DEPARTMENT
RESEARCH AND DEVELOPMENT REPORT

March 1974

Report 2121

## MAJOR DTNSRDC ORGANIZATIONAL COMPONENTS



(1)

# SOME DESIGN PRINCIPLES OF GROUND EFFECT MACHINES
## SECTION A — INTRODUCTORY SURVEY

by

Harvey R. Chaplin and Allen G. Ford

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## Foreword

This report is based on a lecture series presented by the authors at the von Kármán Institute for Fluid Dynamics, Rhode-Saint-Genèse, Belgium, in May 1965; and at the University of Maryland, College Park, Maryland, in July 1965. The lectures were prepared under the joint auspices of the David Taylor Model Basin and the Naval Air Development Center. They were presented in Belgium under the joint sponsorship of the von Kármán Institute and the Advisory Group for Aerospace Research and Development (AGARD); and in Maryland under the sponsorship of the Assistant Secretary of the Navy for Research and Development.

The revised lectures will be presented as follows:

A. Introductory Survey
B. Air Cushion Mechanics
C. Internal Aerodynamics
D. Drag
~~E. Drag Optimization for Sidewall GEM (CAB)~~ Never published
F. Cushion Contributions to Stability
G. Seakeeping
H. Performance Summary

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## NOTATION

| | |
|---|---|
| $\ell$ | cushion length, ft |
| b | cushion beam, ft |
| S | cushion area, $ft^2$ |
| C | cushion perimeter, ft |
| h | "daylight" gap, ft |
| $\Delta p$ | cushion pressure, $lb/ft^2$ (gage) |
| $p_t$ | total pressure at nozzle exit, $lb/ft^2$ (gage) |
| W | gross weight, lb |
| $S_g$ | gap area (hC), $ft^2$ |
| $V_o$ | forward velocity, ft/sec |
| $V_k$ | forward velocity, knots |
| $V_c$ | cushion-reference velocity $\left(\sqrt{\frac{2}{\rho}\Delta p}\right)$, ft/sec |
| $\rho$, $\rho_a$ | air density, $slugs/ft^3$ |
| Q | flow quantity rate, $ft^3/sec$ |
| P | total power, lb-ft/sec |
| $P_p$ | propulsion power, lb-ft/sec |
| $P_c$ | cushion power, lb-ft/sec |
| $P_i$ | ideal total power, lb-ft/sec (corresponding to efficiencies of unity) |
| $P_{p,i}$ | ideal propulsion power, lb-ft/sec |
| $P_{c,i}$ | ideal cushion power, lb-ft/sec |
| $q_a$ | dynamic pressure $(\rho V_o^2/2)$, $lb/ft^2$ |
| $C_D$ | total drag coefficient $\left(\frac{D}{q_a S}\right)$ |
| D | total drag, lb |
| $C_{D_o}$ | coefficient of "other" drag excluding wave drag $\left(\frac{D - D_w}{q_a S}\right)$ |
| $D_w$ | wavemaking drag, lb |
| $C_c$ | discharge coefficient $\left(Q \div S_g V_c\right)$ |

| | |
|---|---|
| $K_{c,i}$ | ideal cushion power parameter $\left(d\vartheta_c \div \Delta p/p_t\right)$ |
| $\propto$ | symbol for "is proportional to" |
| $\gamma_w$ | weight density of water, $lb/ft^3$ |

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# INTRODUCTION

It is conventional to begin any discussion of Ground Effect Machines with a reminder that these vehicles have, as yet, enjoyed only a very short history of development. While this might become tiresome to those who interest themselves in very many such discussions, it is nevertheless one of the facts which it is absolutely essential to keep in mind, if one is to acquire or maintain a rational perspective of this important new field.

The history of serious study and development, having pertinence to the current state of affairs, goes back, in fact, only about eight or nine years; to Cockerell's initial success in attracting official support for his ideas, in the United Kingdom, and to the beginnings of serious research programs in the United States. From a purely historical point of view, of course, one must go back much further, perhaps a century or more, to find the first of a remarkable number of independent conceptions of GEM's of various types; but these seem to have had little or no influence on the modern development of the field.

It is not very meaningful, of course, to compare the current state of GEM development with that of the airplane in the year 1911, as is sometimes done. The environment for technological development, and the pace at which it is possible to carry such developments forward, have changed a very great deal since the Wright brothers flew their first airplane. Even so, nine years (or more appropriately, six years since SR-N1 made her first flight) is still a very short time, indeed, in the life of an important new form of transportation. Some consequences of this youthfulness of the field are:

1. The state of the art must be regarded as primitive (the remarkable practical success of some of the British Hovercraft notwithstanding).

2. Conclusions and opinions of almost every kind should be reached and held tentatively. Physical principles will not change, of course; but the apparent practical significance of some of the most fundamental principles has been changing rapidly, and this must be expected to continue as the field continues to develop and expand.

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3. In common with most fields of technology, whether primitive or advanced, the most important facts known about Ground Effect Machines are rather simple, elementary facts; but also, the most important of the questions which remain unanswered are rather simple, elementary questions. It follows that the role of subtle and sophisticated mathematical analysis is, as yet, comparatively unimportant in this field. The simple, commonsense approach is usually far more rewarding; and even the simplest approaches lead quickly into areas where the uncertainties of the underlying experimental facts exceed the inherent mathematical errors of first-order analysis.

We will try to follow the simplest and clearest paths available, seeking to expose the main ideas in their elementary forms, and relying on intuitive plausibility in lieu of proof.

In fact, if we wish to pursue our subject to the point of conclusions, we have little choice but to rely on intuitive approaches at many stages along the way. But let the reader beware! Conclusions reached in such a manner are not to be accepted as permanent truths. They are to be regarded as a form of tentative interpretation of available facts, which must be constantly reinterpreted as new facts come along.

SCOPE

Not the least of the vagaries of the Ground Effect Machine field is the meaning of the term Ground Effect Machine (GEM). This term is, in fact, only one of a number of terms which have been used rather loosely and interchangeably in reference to the same general class of vehicles. A partial listing of some of the more common terms is:

Ground Effect Machine
Hovercraft
Air Cushion Vehicle
Surface Effect Ship
Hydroskimmer

Various authors have elected to consider one or another of these terms to be descriptive of a very broad class of vehicles, using others to describe subclasses within the broad class; but there seems to have been little or no agreement on this among different authors.

We elect to consider that all of the above terms are synonymous, describing the same broad class of vehicles, which we proceed to define as follows:

Ground Effect Machine (GEM) — Any vehicle which causes air pressure higher than atmospheric to be exerted on the surface of the earth directly below it, so that the integral of this additional air pressure, over the area of the vehicle planform projected on the earth's surface, produces a force comparable to (say, greater than half) the total weight of the vehicle. The air-filled space between the earth's surface and the vehicle is called the "air cushion."

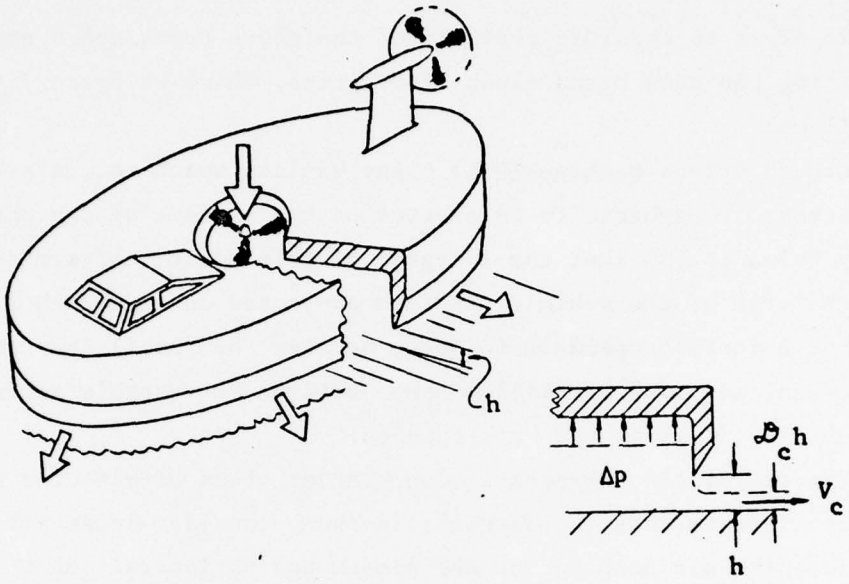
The definition embraces a much broader class of vehicles than we will consider in detail. We will, in fact, consider in detail only vehicles with air cushions of the plenum and peripheral jet types, and certain design variations of these involving sidewalls and other forms of mechanical "cushion containers." (This narrower class of vehicles includes, for example, the Westland Hovercraft, SR.N1 through SR.N6; the Denny sidewall craft D-1 and D-2; the U. S. Navy's XR-1 "Captured Air Bubble"; the Bertin B-6; and a large number of similar craft.)

Before embarking on detailed consideration of the fluid dynamic design principles, however, it will be profitable to consider briefly some of the more obvious characteristics, similarities, and differences of various types of craft, including some which lie beyond the scope of what we will later consider in detail.

The plenum type craft is comparatively very easy to understand, yet possesses most of those characteristics of Ground Effect Machines which must be understood, if one is to understand Ground Effect Machines at all. We will therefore consider the Plenum first, and in somewhat more detail than other types of craft, for purposes of this introductory survey.

## SIMPLE PLENUM CRAFT

Consider a craft as sketched below:




The cushion is contained in a large cavity, between the ground, the base of the craft, and the "plenum walls" extending down from the base. Air is pumped directly into the cushion through a compressor, and escapes through a "daylight" gap at the lower edge of the cushion, at velocity $V_c$ . If we choose the primary variables to be:

| | | |
|---|---|---|
| Cushion Area | S | $ft^2$ |
| Cushion Perimeter | C | ft |
| Daylight Gap | h | ft |
| Cushion Pressure | Δp | psfg |
| Discharge Coefficient | $\mathcal{B}_c$ | – – |

Other properties of the system follow directly

| | | |
|---|---|---|
| Lift = Weight | $W \doteq \Delta p\ S$ | lb |
| Gap Area | $S_g = hC$ | $ft^2$ |
| Reference Velocity | $V_c = \sqrt{\frac{2}{\rho_a} \Delta p}$ | ft/sec |
| Flow Quantity | $Q = V_c\ S_g \mathcal{B}_c$ | $ft^3$/sec |
| Ideal Cushion Power | $P_{c,i} = \Delta p\ Q$ | lb-ft/sec |

These relationships become slightly modified with forward speed, $V_o$ , when the dynamic pressure

$$q_a = \rho_a V_o^2/2$$

becomes comparable in magnitude to the cushion pressure $\Delta p$. At moderate speeds, however, we can write:

Cushion Specific Power (Ideal)

$$\frac{P_{c,i}}{W V_o} = \frac{S_g}{S} \mathcal{D}_c \div \sqrt{q_a/\Delta p} \quad \left(\text{Plenum, } \frac{q_a}{\Delta p} << 1\right) \qquad [A\text{-}1]$$

The ideal propulsion power is the product of total drag and velocity. If we define a total drag coefficient

$$C_D \equiv \frac{D}{q_a S}$$

then

$$P_{p,i} = C_D q_a S V_o$$

and we can write

Propulsion Specific Power (Ideal)

$$\frac{P_{p,i}}{W V_o} = \frac{D}{W} = C_D \frac{q_a}{\Delta p} \qquad [A\text{-}2]$$

also,

Total Specific Power (Ideal)

$$\frac{P_i}{W V_o} = \frac{P_{c,i}}{W V_o} + \frac{P_{p,i}}{W V_o} \qquad [A\text{-}3]$$

The specific power (often called "equivalent drag/lift ratio") is one of the most important of the various nondimensional vehicle performance parameters. For a vehicle of given gross weight W, the weight of fuel $W_f$ consumed during a trip of a given distance is directly proportional to the total specific power $\frac{P}{W V_o}$ (see Figure A-1).

The simple form of Equations [A-1] through [A-3] conceals many complications with which we will deal in the succeeding sections. The cushion pressure and cushion power can vary somewhat with forward speed; and the total drag coefficient $C_D$ is composed of many components (external aerodynamic drag, momentum drag, wavemaking drag, hydrodynamic drag, etc.) which vary in different ways with speed. (Note, $C_D$ is constant if D is proportional to $V_o^2$.) Also, we have only the ideal specific power, with no allowance yet for the efficiency of the propeller, compressor, ducts, etc.

Nevertheless, an excellent beginning toward insight into the nature of Ground Effect Machines can be had by constructing a graph of these relationships, and interpreting it as though the cushion pressure, cushion power and total drag coefficient were independent of speed.

This is done in Figure A-2, a logarithmic graph of ideal specific power versus the speed parameter $V_k/\sqrt{\Delta p}$. The speed in knots $V_k$ equals approximately 17.2 $\sqrt{q_a}$ for standard air density. We consider $V_k/\sqrt{\Delta p}$ to be nondimensional, being merely a short way to write

$$\frac{V_o}{1.69} \div \left[\Delta p \left(\frac{0.00238}{\rho_a}\right)\right]^{\frac{1}{2}}$$

On this graph, Equation [A-1] gives the solid lines of slope -1; Equation [A-2] gives the solid lines of slope +2; and Equation [A-3] gives the dashed curves, for total ideal specific power, which become asymptotic to the cushion specific power lines at low speeds and asymptotic to the propulsion specific power curves at high speeds.

For reasons which will become clear in the succeeding paragraphs, we have substituted an "ideal cushion power parameter" $K_{c,i}$ (which is identically equal to $\mathcal{B}_c$ for the plenum) in place of the discharge coefficient $\mathcal{B}_c$ in Figure A-2.

The discharge coefficient for a simple plenum is typically about 0.6 in magnitude, the extreme range of possibilities being about 0.5 to 1.0. Thus, for purposes of a very broad look at vehicle performance, we can consider $K_{c,i}$ constant.

Figure A-2 shows, then, the dominant influence on performance of the gap area ratio $S_g/S$, and of the drag coefficient $C_D$. If we regard each of the dashed curves as representing a specific design, then the minima of these curves are "optimum" points, in the sense of providing minimum fuel consumption during a given trip. Comparing different "designs" on this basis, the following (approximate) trends can be traced on Figure A-2:

1. With air gap ratio $S_g/S$ fixed, the optimum speed decreases with increasing drag coefficient in proportion to $C_D^{-1/3}$; and the minimum specific power increases as $C_D^{1/3}$.

2. With drag coefficient $C_D$ fixed, the optimum speed increases with increasing air gap ratio in proportion to $\left(S_g/S\right)^{1/3}$; and the minimum specific power increases in proportion to $\left(S_g/S\right)^{2/3}$.

(These interpretations depend upon the cushion power and drag coefficient remaining nearly constant in the vicinity of the optimum point. It will be shown later that this is usually approximately valid, provided $V_k/\sqrt{\Delta p} < 10$, except, in the case of overwater operation, near the "hump speed," and provided the cushion pressure is "moderate." The meaning of "moderate cushion pressure" will be clarified shortly.)

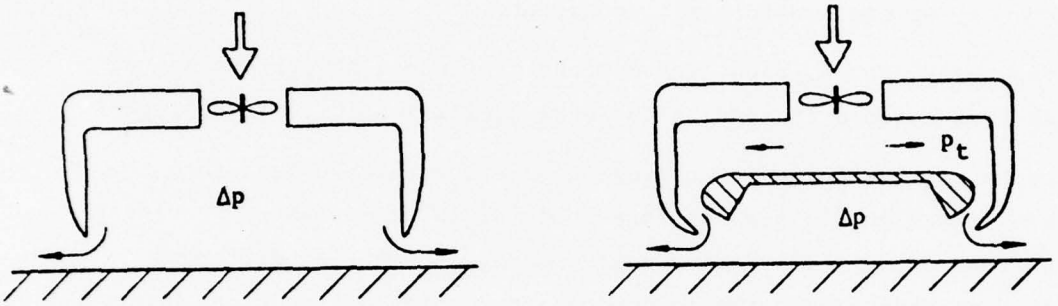
Further discussion of Figure A-2 will be postponed until we have introduced the peripheral jet.

PERIPHERAL JET CRAFT

For the plenum craft, we found the cushion power to be proportional to the product of the 3/2-power of cushion pressure and the discharge coefficient,

$$P_{c,i} = \Delta p \sqrt{\frac{2}{\rho} \Delta p}\, \mathcal{A}_c\, S_g$$

It is possible to reduce the discharge coefficient very substantially, by discharging the air from the compressor at the periphery of the cushion, instead of directly into the cushion.




Plenum

Peripheral Jet

This reduction of the discharge coefficient is accomplished, however, at the expense of increasing the total pressure of the compressor output (in psfg) from $\Delta p$ to $p_t$, $p_t > \Delta p$. The cushion power thus is proportional to

$$P_{c,i} \propto \sqrt{\Delta p}\, p_t\, \mathcal{Q}_c$$

Rewriting this in the form

$$P_{c,i} \propto \sqrt{\Delta p}\, \Delta p\, \frac{\mathcal{Q}_c}{\Delta p / p_t}$$

comparing with the expression for the plenum given previously, and noting further that $\Delta p = p_t$ for the plenum, it is found to be convenient to write

$$\left.\begin{aligned} P_{c,i} &\propto \sqrt{\Delta p}\, \Delta p\, K_{c,i} \\ K_{c,i} &\equiv \frac{\mathcal{Q}_c}{\Delta p / p_t} \end{aligned}\right\} \text{Plenum or Peripheral Jet}$$

In Section B, we shall consider in detail the relationships which determine the quantities $\beta_c$ , $\Delta p/p_t$, and their quotient $K_{c,i}$ for the peripheral jet. For the present, let us presuppose those results by noting that, in practical design, $K_{c,i}$ is likely to be approximately the same (perhaps 20 percent less) for the peripheral jet as for the plenum craft.

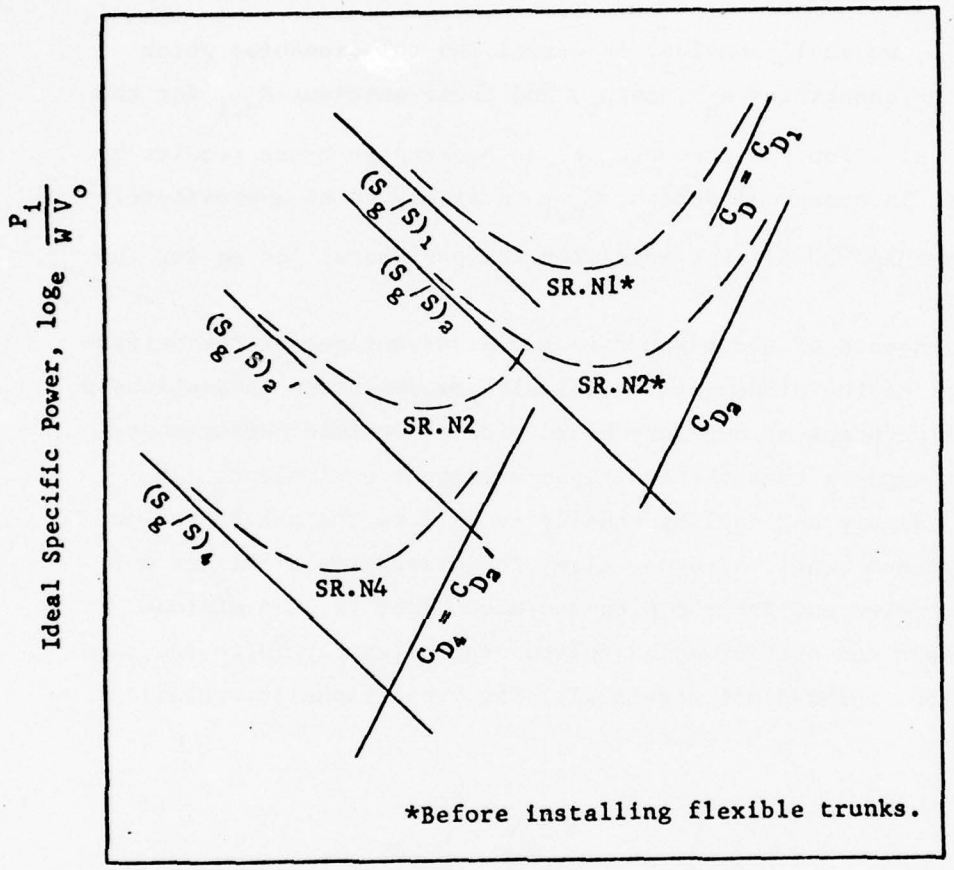
Other aspects of the advantages and disadvantages of the peripheral jet versus the plenum craft will also be discussed in Sections B and C. For purposes of our very broad view of vehicle performance, however, it appears that the two types are about equivalent. In particular, Figure A-2 applies equally as well to the peripheral jet as to the plenum craft, as does, also, the discussion of Figure A-2, which was carried out for the plenum craft. That is, the minimum specific power and optimum speed follow approximately (with the same qualifications pointed out previously) the proportionality relationships:

$$\left(\frac{P_i}{W\,V_o}\right)_{opt} \propto \left(\frac{S_g}{S}\right)^{2/3} C_D{}^{1/3}$$

$$\left(\frac{V_k}{\sqrt{\Delta p}}\right)_{opt} \propto \left(\frac{S_g}{S}\right)^{1/3} C_D{}^{-1/3}$$

The subscript "opt" denotes "optimum" in the sense of minimizing the specific power, thus minimizing the fuel consumed per mile.

It is very enlightening to trace, qualitatively, one of the main trends of Hovercraft development as in the following graph which should be compared with Figure A-2.




The SR.N1 was allowed to have a relatively high drag coefficient $C_{D_1}$ for reasons of economical construction, and had an air gap area $S_g$ judged just sufficient to avoid water contact under "normal" conditions. This resulted in a rather large air gap ratio $(S_g/S)_1$, because the SR.N1 was so small.

The SR.N2 was more streamlined, giving originally a smaller drag coefficient $C_{D_2}$. In the original design, the air gap height ("daylight" clearance) h was about the same as for SR.N1, but the air gap area ratio $(S_g/S)_2$ was smaller, because the SR.N2 was larger. (For a given planform shape and given daylight clearance h, the ratio $S_g/S$ is inversely proportional to the linear dimension of the planform.)

However, before SR.N2 was put in service, Westland changed the design, introducing flexible trunks and reducing the air gap ratio substantially to $(S_g/S)_2'$. The daylight clearance was no longer large enough to avoid water contact under "normal" conditions, so the total drag coefficient under average expected wave conditions was increased substantially $\left(\text{to } C_{D_2'}\right)$, on account of the additional hydrodynamic drag.

The SR.N4 (projected 150-ton channel ferry) might have, according to information given in the press, about the same daylight clearance, h; but again, a substantial decrease in air gap area ratio $\left(\text{to } (S_g/S)_4\right)$, because of the increased planform dimensions. The average total drag coefficient might go either up or down, relative to SR.N2, depending on how the respective average operating wave conditions were defined. In the qualitative graph presented above, it has been assumed that $C_{D_4} = C_{D_2'}$.

It should be stressed again that the above graph is only qualitative, and undoubtedly contains discrepancies as to the detailed relationships between specific craft. The general trend, however, toward ever smaller gap area ratios $S_g/S$, ever smaller specific power, and ever smaller optimum nondimensional speeds is correctly represented.

Also correctly represented is the sharp break in the trend of development which occurred with the introduction of flexible trunks. One should not consider that flexible trunks caused this break in the trend; but rather, that they permitted it. There was a substantial gain in operating economy to be had by reducing the air gap area $S_g$, even at the expense of greatly increased contact with the surface (and hence increased drag) and decreased (nondimensional) speed. Flexible trunks made this decreased gap area feasible from the standpoint of structural integrity, structural weight, and motions (passenger comfort). (Decreasing the air gap also has other advantages, besides reduced specific power, as we shall see in the succeeding sections; in particular, reduced size and weight of machinery and ducting.)

Of course, specific power is not the only performance parameter which is important. There are three parameters which are of paramount importance: specific power, ratio of empty weight to gross weight, and speed (dimensional). Every successful vehicle is the result of a favorable balance between these three parameters.

The unfavorable trend toward lower nondimensional speeds, as the specific power is favorably reduced, has tended to be offset by increased cushion pressure so that the dimensional speed has not decreased. Actually, a gradual trend toward higher actual speeds with larger vehicles is justified, as we shall see in later sections. In fact, if Figure A-2 were universally valid, we would not have to worry very much about speed. We could accept almost any design change, within reason, which improved the specific power; and then win back the desired speed by increasing the cushion pressure. (The structural problems generally tend to become more tractable also, with an increase in cushion pressure.) However, among the various qualifications placed on Figure A-2 was the requirement that the cushion pressure be "moderate." It is now necessary to clarify this requirement, which has to do with the wavemaking drag.

The practical validity of Figure A-2, and of the various arguments we have carried out in connection with this figure, depends upon the following conditions: In the vicinity of the optimum point,

a. The total drag is varying nearly in direct proportion to $V_k^2$.

b. The cushion power is nearly constant.

Since some of the components of the total drag are definitely following decidedly different laws from the $V_k^2$ law, Figure A-2 is meaningful only if these components are a relatively small part of the total drag, in the vicinity of the optimum speed.

The wavemaking drag, in particular, varies in proportion to a negative power of $V_k$ at speeds above "hump speed"; therefore, Figure A-2 does not apply at all if the wavemaking drag is very significant, in the vicinity of the optimum speed. Moreover, for a given planform shape and size, the wavemaking drag $D_w$ divided by the gross weight W varies directly with the cushion pressure $\Delta p$:

$$\frac{D_w}{W} \propto \Delta p$$

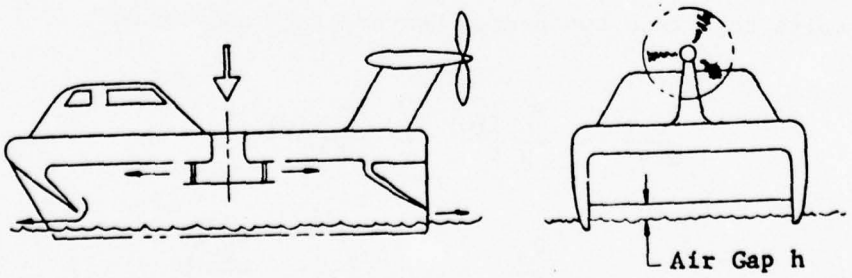
Therefore, when we say Figure A-2 is restricted to "moderate" cushion pressures, we mean specifically that it is restricted to cushion pressure sufficiently moderate that the wavemaking drag plays only a minor role in the determination of the optimum performance point.

The Westland family of Hovercraft SR.N1 through SR.N4, which were discussed above, and also SR.N5 and SR.N6, which are reversions to a smaller size, satisfy the condition of "moderate" cushion pressure reasonably well. However, it would seem that, projecting that line of development into the future, to larger and larger vehicles, the wavemaking drag would become more and more important, and optimum performance would be determined less and less adequately by the simple cushion power versus propulsion power tradeoff represented in Figure A-2.

Rather than attempt to follow the subtle changes likely to take place in that line of development, however, it will be simpler for our present purpose to turn to a different line of development in which we already find the cushion power playing a minor role, while the wavemaking drag plays a very major role.

CAPTURED AIR BUBBLE (CAB), OR SIDEWALL GEM

Many of the early ideas and inventions of Ground Effect Machines included the idea of minimizing the air gap by use of slender sidewalls, which extended into the water to block the escape of air from the cushion.




Plenum-Type Sidewall Ground Effect Machine

In the intensive early exploration of GEM's (from 1957 to 1961, say), the sidewall GEM received its due share of attention; but most

investigators quickly concluded (perhaps prematurely) that, because of the hydrodynamic drag of the sidewalls, these craft would be suitable only for relatively low speeds, perhaps of the order of 50 knots.

Two separate trends have developed to change this outlook. First, as we have just noted, the foremost developers of "ordinary" GEM's (we will call them "full-peripheral" GEM's to denote more or less uniform air gap h all around) are finding it profitable to accept much more severe hydrodynamic drag, in the interest of reducing the air gap area $S_g$, than would have been guessed by most experts a few years ago. Secondly, more recent research on sidewall GEM's (especially under the Captured Air Bubble research program at the U. S. Naval Air Development Center) has produced greatly improved understanding of the sidewall GEM. The sidewall GEM, or CAB now appears to be a very strong contender for future maritime GEM applications.

For the full-peripheral GEM (at its current stage of development, with "moderate" cushion pressures), the most essential thing to be understood is the tradeoff between cushion power and propulsion power. For the CAB, the most essential thing to be understood is the tradeoff between the wavemaking propulsion power and the remainder of the propulsion power.

Previously we dealt with the ideal total specific propulsion power

$$\frac{P_{p,i}}{W\,V_o} = \frac{D}{W} \doteq C_D \left( \frac{1}{17.2} \frac{V_k}{\sqrt{\Delta p}} \right)^2$$

We must now split this into two parts, "wavemaking" and "other"

$$\frac{P_{p,i}}{W\,V_o} = \frac{P_{p,i(w)}}{W\,V_o} + \frac{P_{p,i(o)}}{W\,V_o}$$

$$\doteq \frac{D_w}{W} + C_{D_o} \left( \frac{1}{17.2} \frac{V_k}{\sqrt{\Delta p}} \right)^2$$

For a given planform shape, the wavemaking drag can be represented functionally; that is,

$$\frac{D_w}{W} \approx \frac{\Delta p}{\ell} \; f\left( \frac{V_k}{\sqrt{\ell}} \right)$$

where $V_k/\sqrt{\ell}$ is the "speed-length" ratio (a nondimensional speed parameter, in the sense that it is derived from the nondimensional Froude number $V_o/\sqrt{g\ell}$ by inserting a constant value for the acceleration of gravity g). (Similarly, $\Delta p/\ell$ is nondimensional in the sense of being derived from $\Delta p/(\gamma_w \ell)$ by inserting a constant value for the weight density of water $\gamma_w$).

For simplicity we shall consider only a single planform shape (rectangular, $\ell/b = 2$) and the corresponding single graph of $\frac{D_w}{W} \div \frac{\Delta p}{\ell}$ versus $V_k/\sqrt{\ell}$ (Figure A-3) for purposes of this introductory discussion.

When we seek to represent the CAB on a graph like Figure A-2, we find the problem slightly complicated by the addition of the two new variables $\Delta p/\ell$ and $V_k/\sqrt{\ell}$ . However, one of these is eliminated by relating the speed-length ratio $V_k/\sqrt{\Delta p}$ as follows:

$$V_k/\sqrt{\ell} = \left(V_k/\sqrt{\Delta p}\right) \sqrt{\Delta p/\ell}$$

Hence, only the single new variable $\Delta p/\ell$ need appear on the graph. Such a graph is presented in Figure A-4, which shows, for simplicity, only single representative values of the variables $C_{D_o}$ and $K_{c,1}\left(\frac{S_g}{S}\right)$.

Figure A-4 displays straight lines of slopes -1 and +2, just like Figure A-2; but now, the intersection between these lines is faired by a more complicated curve of total ideal specific power, strongly influenced by the wavemaking drag "hump." In fact, we now find <u>two</u> minima in the "total" curve when the cushion is heavily loaded. The minimum at the higher speed is the one which we choose to call the optimum performance point. (In practice, the air gap ratio $S_g/S$ might be reduced as the speed is reduced to such an extent that the lower minimum disappears,

and the specific power improves continuously with speed reduction, at speeds below "hump speed." In fact, the cushion power has usually been neglected entirely from the optimization studies for CAB's, except that a 5 or 10 percent power margin is added at the end.)

One sees in the example of Figure A-4 that an increase in cushion loading parameter $\Delta p/\ell$ has a very slightly unfavorable effect on the value of the minimum specific power, at the higher speed minimum. On the other hand, an increase in $\Delta p/\ell$ has a noticeably favorable effect on the (dimensional) optimum speed. However, at very high loadings, the severe wavemaking drag hump complicates the design of the propulsion system.

Keep in mind that Figure A-4 gives only one example of $C_{D_o}$ and $K_{c,i}\left(\frac{S_g}{S}\right)$. It is easy to see that the picture can change considerably with changes in these variables.

The alert reader will now ask, Wherein has our preliminary treatment of CAB's, as reflected in Figure A-4, and our preliminary treatment of full-peripheral GEM's, as reflected in Figure A-2, revealed any fundamental differences between the two types of craft? We have no answer, except to admit that we have found no fundamental differences at all! In treating the CAB, we have been compelled to bring in the concept of wavemaking drag, because

a. We expect such small values of cushion specific power that the component of propulsion specific power associated with wavemaking drag becomes significant by comparison, and

b. We expect somewhat higher values of the parameter $\Delta p/\ell$ for the CAB, which makes the wavemaking drag larger in absolute magnitude.

However, these are differences of <u>degree</u>, not differences of <u>kind</u>. The fact is that we might just as well have introduced wavemaking drag into Figure A-2 (had we not been striving for simplicity); and the fact is that Figure A-4 is every bit as valid for a full-peripheral GEM (with $K_{c,i}\left(\frac{S_g}{S}\right) = 0.002$, $C_{D_o} = 0.5$ and rectangular cushion, $(\ell/b = 2)$ as it is

for a CAB. Of course, such a small value for the air gap area ratio is not at all representative of current full-peripheral GEM's, but the trend of development seems to point somewhat in that direction.

Note that, since the cushion power is such a small fraction of the CAB's total power, there is little or nothing to be gained from sophistication in the method of delivering air to the cushion. Designers will probably choose to let the air be delivered directly from the compressors to the cushion, as in the full-peripheral plenum craft and as indicated schematically in the sketch at the beginning of our discussion of CAB's.

Also, referring to that sketch, it is apparent that the plenum walls at the bow and stern will preferably be compliant with the uneven surface (waves) over which the craft travels. Two of the proposed solutions are:

a. To spring-suspend the bow and stern plenum walls mechanically, in such a way that they will plane on the wave surfaces, as skiis, at a favorable angle of attack for minimum drag; or

b. To use flexible-fabric bow and stern plenum walls arranged to yield freely to wave impacts; in other words, to fit the CAB with flexible trunks.

## TRENDS

The influence of flexible trunks, and/or of sidewalls and flexible bow and stern seals, can be viewed at this point as aspects of a consistent general trend: Reduction of the air gap area ratio, reduction of the specific power, and reduction of the nondimensional speed. Progress to larger and larger vehicles can be seen to result in a further continuation of this same trend.

This trend is indicated in Figure A-5, which identifies the domain of current GEM's to be specific powers of the order of 0.2 to 0.3 (roughly the same as helicopters) with speed-pressure ratios of the order of 5 to 10; and implies a future extension of this domain into much lower specific powers and somewhat lower nondimensional speeds. It is emphasized again that reduction of nondimensional speed does not necessarily imply lower

actual speed in knots. In fact, to the extent that the reduction in nondimensional speed results from progress to larger vehicles, the speed in knots will be found to increase.

A vertical dashed line on Figure A-5 marks the value (17.2) of the speed parameter at which the air dynamic pressure associated with forward speed equals the quantity W/S (cushion loading, wing loading, or disc loading for the various vehicles represented) at sea level standard conditions. To the right of this line lie ground effect vehicles, such as the "Ram Wing" and "Wing in Ground Effect" (WIG), which are sustained by pressures induced by the forward motion rather than by a compressor-fed cushion. It was once thought that the future evolution of GEM's might proceed in this direction; that is, toward more aircraft-like vehicles. However, the current trend appears to be definitely toward more ship-like GEM's, as indicated in Figure A-5.

Aerodynamics Laboratory
David Taylor Model Basin
Washington, D. C.
March 1966

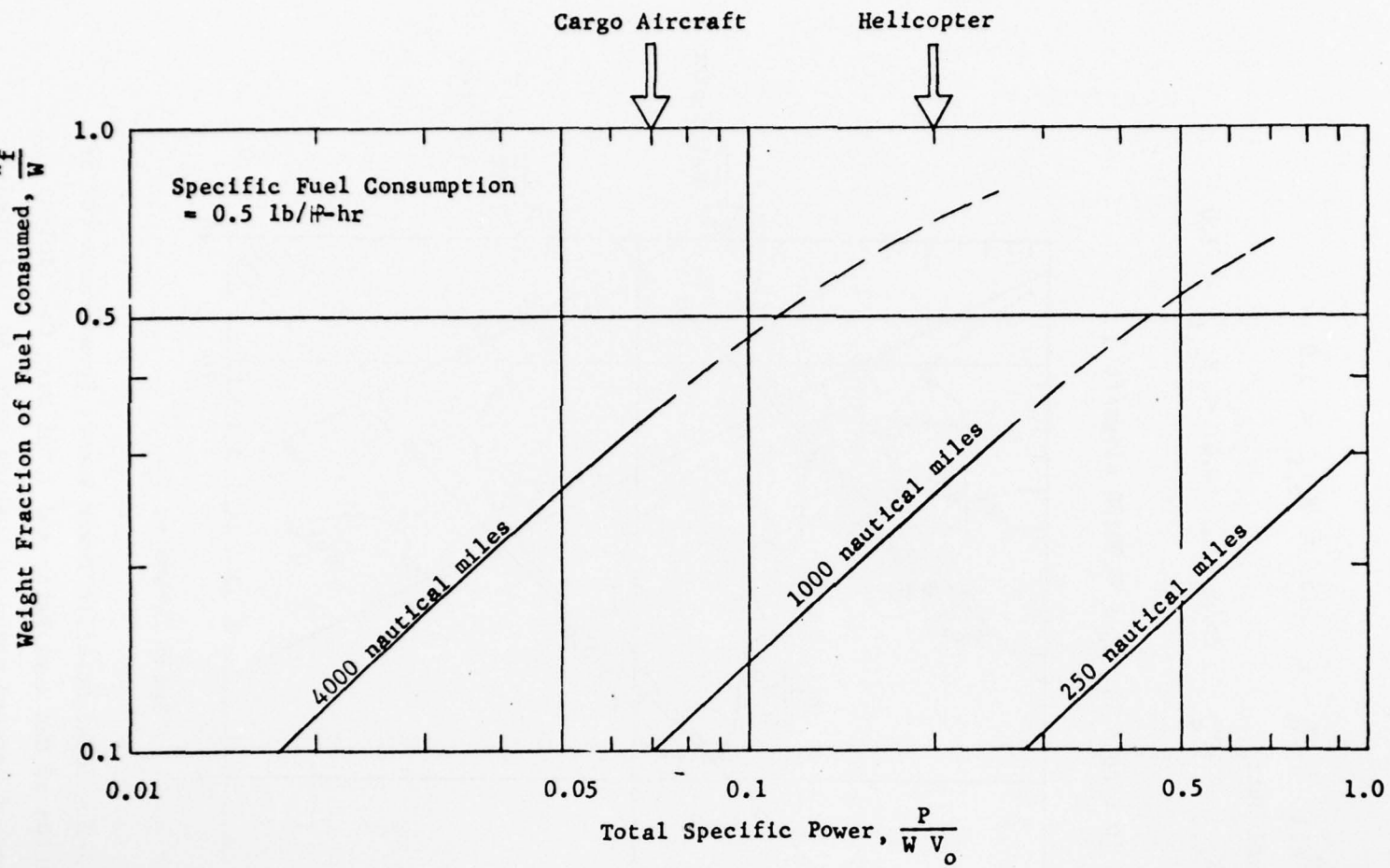



Figure A-1 - Fuel Consumed in Various Distances as Function of Total Specific Power

Definitions and Expected Ranges

Plenum:

$$K_{c,i} \equiv \mathcal{O}_c \quad , \quad 0.5 < K_{c,i} < 1.0$$

Peripheral Jet:

$$K_{c,i} \equiv \mathcal{O}_c \div \Delta p/p_t \quad , \quad 0.4 < K_{c,i} < 1.0$$

$$[W \doteq \Delta p\ S \quad ; \quad \rho_a = 0.00238 \text{ slugs/ft}^3]$$

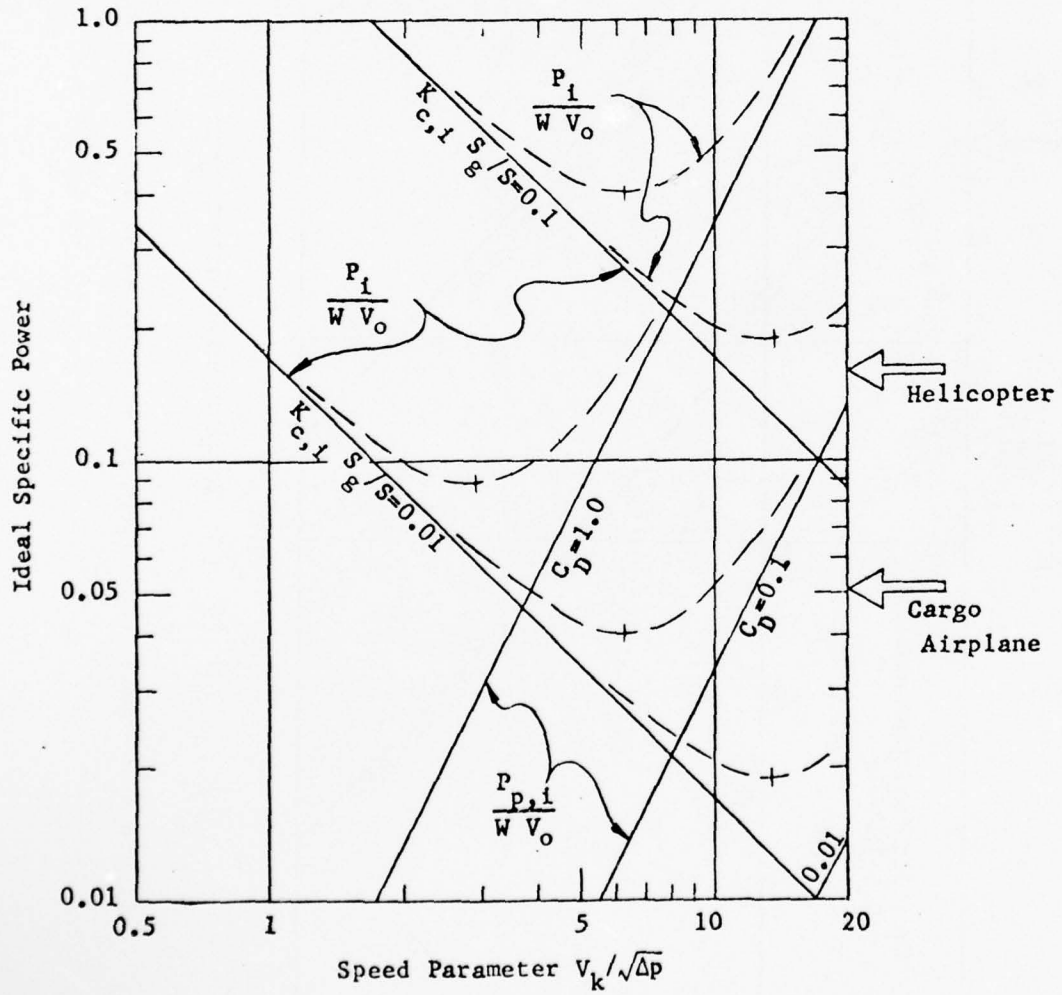



Figure A-2 - Ideal Specific Power Versus Speed Parameter as Function of Gap Area Ratio and Total Drag Coefficient (Effect of forward speed on cushion power is neglected.)

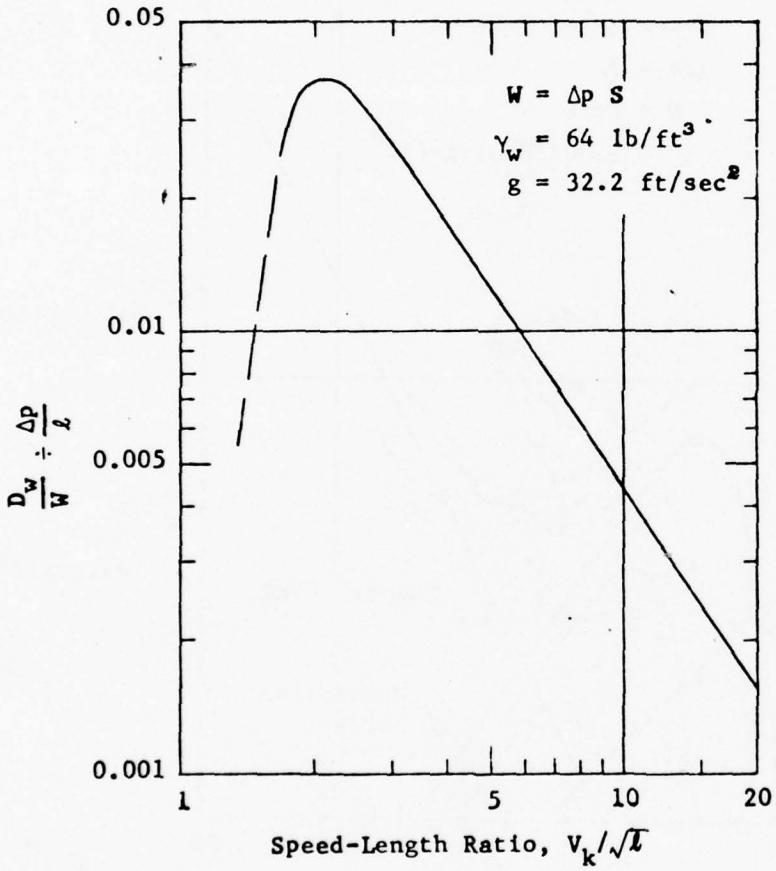



Figure A-3 - Wavemaking Drag for a Rectangular Cushion of Length/Beam Ratio $\ell/b = 2$
(According to Newman and Poole, DTMB Report 1619, March 1962)

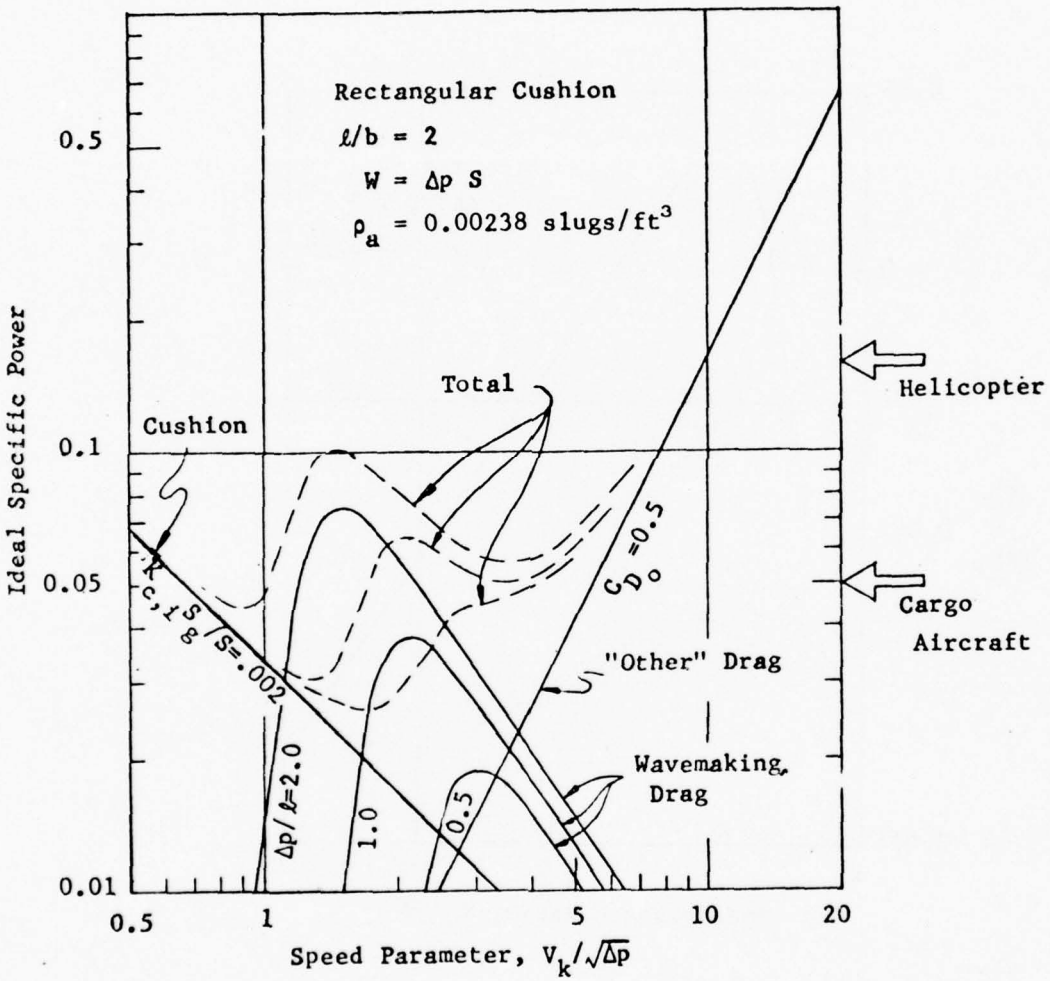



Figure A-4 - Ideal Specific Power Versus Speed Parameter for a CAB Craft With Various Cushion Loadings, $\Delta p/\ell$

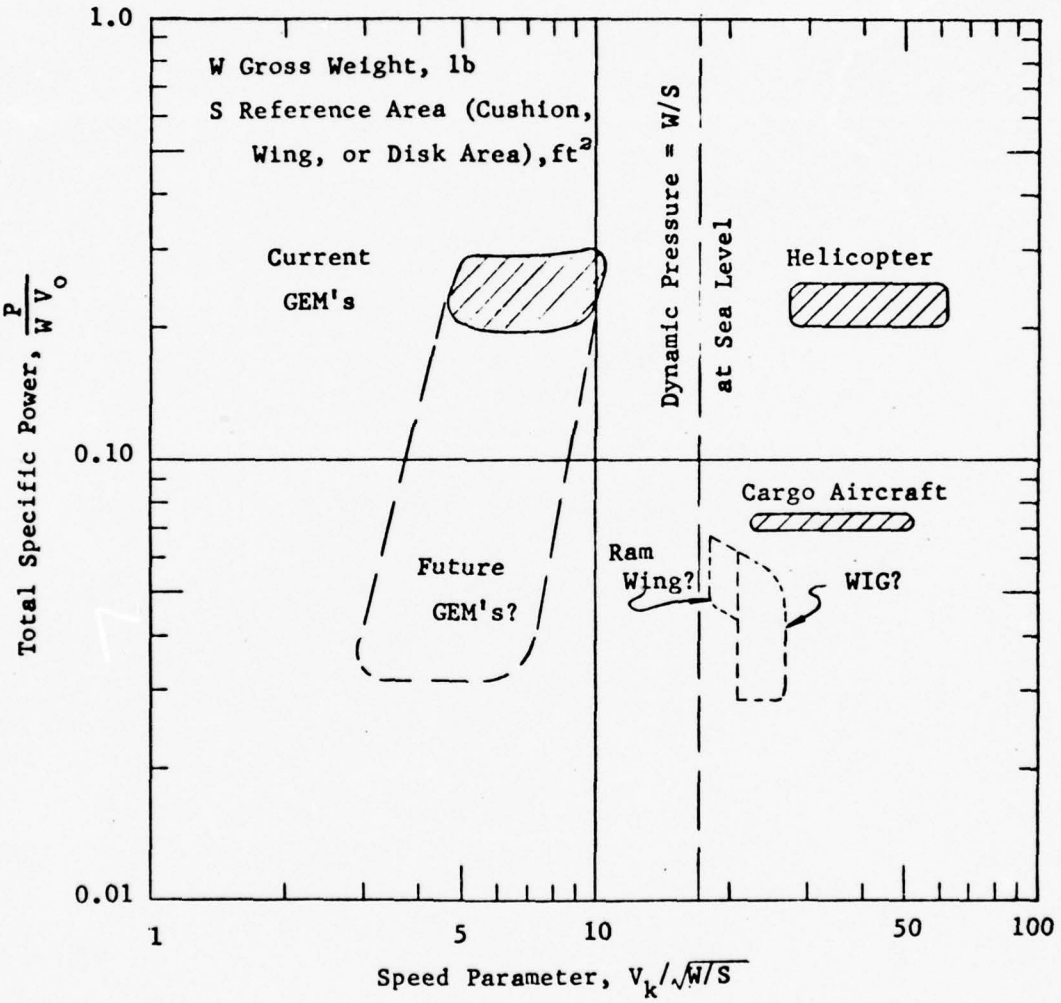



Figure A-5 - Approximate Loci of the Design Operating Points of Various Existing and Proposed Vehicle Types

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# SOME DESIGN PRINCIPLES OF GROUND EFFECT MACHINES
## SECTION B — AIR CUSHION MECHANICS

by

Harvey R. Chaplin and Allen G. Ford

APPROVED FOR PUBLIC RELEASE: DISTRIBUTION UNLIMITED

April 1966

Report 2121B
Aero Report 1100B

## Foreword

This report is based on a lecture series presented by the authors at the von Kármán Institute for Fluid Dynamics, Rhode-Saint-Genèse, Belgium, in May 1965; and at the University of Maryland, College Park, Maryland, in July 1965. The lectures were prepared under the joint auspices of the David Taylor Model Basin and the Naval Air Development Center. They were presented in Belgium under the joint sponsorship of the von Kármán Institute and the Advisory Group for Aerospace Research and Development (AGARD); and in Maryland under the sponsorship of the Assistant Secretary of the Navy for Research and Development.

The revised lectures will be presented as follows:

A. Introductory Survey
B. Air Cushion Mechanics
C. Internal Aerodynamics
D. Drag
~~E. Drag Optimization for Sidewall GEM (CAB)~~
F. Cushion Contributions to Stability
G. Seakeeping
H. Performance Summary

## NOTATION

| | |
|---|---|
| $\ell$ | cushion length, ft |
| b | cushion beam, ft |
| S | cushion area, $ft^2$, measured, in plan view, to outer edge of nozzle exit |
| $\Delta p$ | cushion pressure, psfg |
| $p_t$ | total pressure of air supply to cushion at nozzle exit or cushion entry, psfg |
| $V_o$ | flight velocity in earth axes, ft/sec |
| $V_k$ | flight velocity in earth axes, knots |
| $\rho$, $\rho_a$ | air density, $slugs/ft^3$ |
| $\rho_w$ | water density, $slugs/ft^3$ |
| $q_a$ | reference air dynamic pressure $\left(\rho_a V_o^2/2\right)$, $lb/ft^2$ |
| $q_w$ | reference water dynamic pressure $\left(\rho_w V_o^2/2\right)$, $lb/ft^2$ |
| u | local air velocity, ft/sec |
| q | local dynamic pressure $(\rho u^2/2)$, $lb/ft^2$ |
| $S_g$ | daylight gap area (cushion perimeter times mean daylight gap (hC), $ft^2$) |
| h | "daylight" gap, feet (local value or mean value as appropriate) |
| C | cushion perimeter, feet |
| $\theta$ | peripheral jet efflux angle, measured positive inward from vertical, degrees |
| t | nozzle thickness, feet |
| x | nozzle thickness parameter $\left(\frac{t}{h}(1 + \sin\theta)\right)$ |
| Q | total air volume flow rate through cushion system, $ft^3/sec$ |
| $\mathscr{C}_c$ | cushion discharge coefficient $\left(\frac{Q}{S_g\sqrt{\frac{2}{\rho}\Delta p}}\right)$ |

| | |
|---|---|
| $K_c$ | cushion power parameter $\left(\frac{\rho_c}{\Delta p/p_t}\right) \cdot \eta_{int}$ |
| $R_c$ | ram recovery parameter $\left(\rho_c \eta_r/\eta_c\right)$ |
| $C_j$ | jet reaction coefficient |
| W | gross weight, pounds |
| w | cushion loading $\left(\frac{W}{S}\right)$, lb/ft$^2$ |
| $L_e$ | external aerodynamic lift, pounds |
| $D_e$ | external aerodynamic drag, pounds |
| $C_L$ | external aerodynamic lift coefficient $\left(L_e/q_a S\right)$ |
| $C_{D_e}$ | external aerodynamic drag coefficient $\left(D_e/q_a S\right)$ |
| $P_c$ | total shaft power supplied to cushion system, lb-ft/sec |
| $P_p$ | total shaft power supplied to propulsion system, lb-ft/sec |
| P | total shaft power $\left(P_c + P_p\right)$, lb-ft/sec |
| $\eta_r$ | ram recovery efficiency |
| $\eta_d$ | duct efficiency |
| $\eta_c$ | compressor efficiency |
| $\eta_{int}$ | internal efficiency |

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## SUMMARY

The energy and momentum relationships governing air cushion performance are reviewed. The exponential-theory equations are recommended for calculation of cushion pressure and jet reaction, and a modified equation is proposed for calculation of the volume flow rate.

The question of off-design cushion performance is discussed. It is pointed out that off-design performance is of paramount practical importance and that the widely held notion that peripheral jet cushions are more efficient than simple plenum cushions is not necessarily valid in this context.

The danger of drawing erroneous conclusions from the cushion performance equations, which (except for the plenum) do not apply to off-design operation, is emphasized.


## INTRODUCTION

There have been almost as many formulations of the fluid dynamic relationships governing the behavior of air cushions as there are fluid dynamicists working in this field. The air cushion poses intriguing and important problems which — like most problems of fluid mechanics — can become extremely complex if one seeks to formulate them exactly.

The fact is, however, that experimental measurements of the important quantities characterizing air cushion performance fall into a rather simple pattern, compatible with comparatively simple theoretical treatments.

We will consider three separate formulations: The "Thin-Jet Theory," which is of limited practical value, but provides an easily understood introduction to some main ideas of the peripheral jet; the "Exponential Theory," which comes closest of all the simple theories to providing solutions fully satisfactory for design application (but fails to correctly predict one of the essential quantities in the case of very thick jets); and Plenum Theory. Finally, we will use the results of the Plenum Theory to arrive at a modification of the Exponential Theory results, providing simple formulas which are suitable for practical design application.

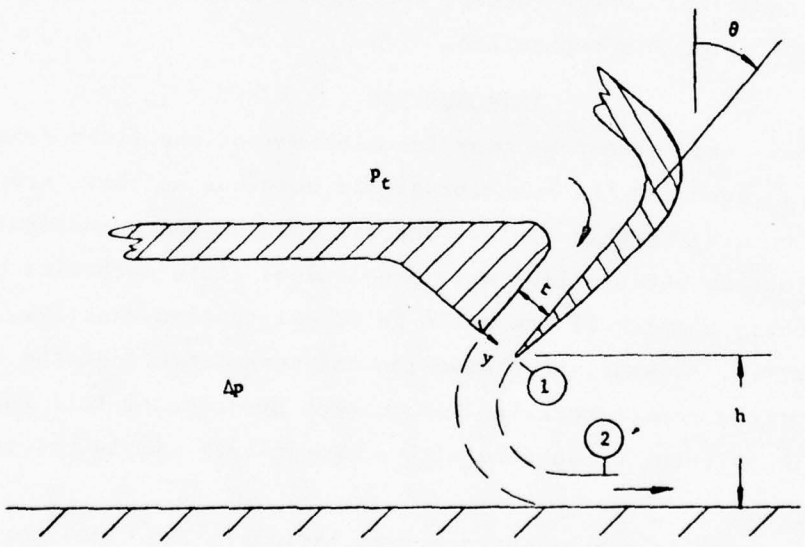
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All theoretical considerations will be pursued under the assumption that air will behave as an incompressible, inviscid fluid. Moreover, we will employ throughout the approximation that the airflow at the cushion periphery is essentially two-dimensional.

## THEORETICAL FORMULATIONS

### THIN-JET THEORY

Consider an air cushion bounded top and bottom by impervious surfaces, and separated from the outside atmosphere by a peripheral jet ("air curtain"), as shown.




Neglecting mixing between the jet and its surroundings, and assuming the cushion to be in static equilibrium at pressure Δp (no air entering or leaving cushion), it will be seen that the jet must curve and become tangent to the ground, as shown in the sketch.

Bernoulli's equation applied to the jet is

$$p + q = p_t$$

where $p_t$ is the total pressure (assumed to be uniform throughout the jet), p is local static pressure, and q is local dynamic pressure ($\rho u^2/2$, where u is the magnitude of the local velocity). (All pressures are "gage," measured relative to the outside atmosphere.)

Thus, the dynamic pressure along the outer free boundary of the jet ($p = 0$) is $q = p_t$; and along the inner free boundary ($p = \Delta p$), it is $q = p_t - \Delta p$.

Now if we introduce the assumption

$$\Delta p \ll p_t$$

it will be seen that

$$q \doteq p_t$$

throughout the jet; and the magnitude of the jet momentum per unit cushion periphery at nozzle exit

$$j_1 = \int_o^t \rho u^2 \, dy \doteq 2 \, q_1 \, t \doteq 2 \, p_t \, t$$

will also be nearly uniform along the free path of the jet.

The horizontal pressure force $\Delta p \cdot h$ per unit cushion periphery must be balanced by the change in horizontal component of jet momentum:

$$\Delta p \cdot h = j_1 \sin \theta + j_2 \doteq 2 \, p_t \, t \, (1 + \sin \theta)$$

or

$$\frac{\Delta p}{p_t} \doteq 2x \quad , \quad x \ll 1$$

where [B-1]

$$x \equiv \frac{t}{h} \, (1 + \sin \theta)$$

The nondimensional quantity x is called the "nozzle thickness parameter." We are clearly justified in replacing the original assumption ($\Delta p \ll p_t$) by the condition $x \ll 1$.

Equal in importance to the cushion pressure ratio $\Delta p/p_t$ is the discharge coefficient $\mathcal{O}_c$, which relates the volume rate of flow Q, through the cushion system, to the cushion pressure. We define

$$\mathcal{O}_c \equiv \frac{Q}{S_g\sqrt{\frac{2}{\rho}\ \Delta p}}$$

where $S_g$ (which equals hC in the case under consideration) is the area of the "daylight" gap at the edge of the cushion.

We have already seen that, if $x \ll 1$, the jet velocity at the nozzle exit is nearly uniform and of magnitude $\sqrt{\frac{2}{\rho}\ p_t}$. Hence,

$$\left.\begin{aligned} Q &\doteq \sqrt{\frac{2}{\rho}\ p_t}\ tC \\ \mathcal{O}_c &\doteq \sqrt{\frac{p_t}{\Delta p}}\ \frac{t}{h} \end{aligned}\right\}, \quad x \ll 1$$

Substituting from Equation [B-1], we obtain

$$\mathcal{O}_c \doteq \frac{1}{1 + \sin\theta}\ \sqrt{\frac{x}{2}}, \quad x \ll 1 \qquad [B\text{-}2]$$

We have already found an expression for the jet momentum, which immediately provides an approximate solution for the jet reaction coefficient

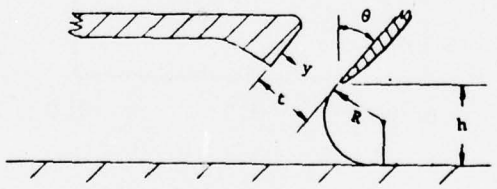
$$C_j \equiv \frac{\int_o^t (\rho u^2 + p)dy}{\Delta p\ t} \doteq \frac{1}{x}, \quad x \ll 1 \qquad [B\text{-}3]$$

EXPONENTIAL THEORY

For values of the nozzle thickness parameter x of interest for practical design (say, $x > 0.2$), the thin-jet theory does not give satisfactory results. In order to obtain practical design formulas, it is necessary to take account of the variation in static pressure across the jet. Different investigators have done this in many different ways, including direct postulation of the static pressure distribution, or postulation of the radius of curvature R(y) of the jet streamlines at the nozzle exit, whence the pressure variation can be calculated from the relation

$$\frac{\partial p}{\partial y} = - \frac{\rho u^2}{R} = - 2 \frac{q}{R}$$

The most successful of the simple theories of this kind was due to Stanton-Jones and Elsley, who assumed that an adequate approximation could be obtained by setting




One can then write, from the Bernoulli equation and the above relations:

$$\frac{\partial p}{\partial y} = - \frac{\partial q}{\partial y} \doteq - \frac{q}{h} (1 + \sin \theta)$$

$$= - 2 q \frac{x}{t}$$

where $x \equiv \frac{t}{h} (1 + \sin \theta)$, as before. This equation is readily integrated to obtain

$$\log_e q = 2x \frac{y}{t} + \text{Constant}$$

where the constant can be determined from the boundary conditions

$$q\ (0) = p_t - \Delta p$$

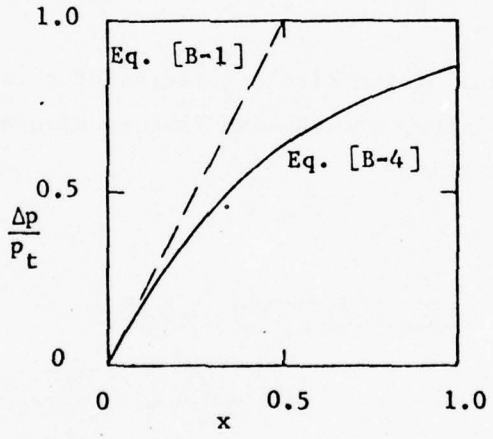
$$q\ (t) = p_t$$

yielding

$$q = (p_t - \Delta p)\ e^{2x\ \frac{y}{t}}$$

and

$$\frac{\Delta p}{p_t} = 1 - e^{-2x} \qquad [B\text{-}4]$$




The quantity of volume flow per unit periphery is:

$$\frac{Q}{C} = \int_o^t u\ dy = \sqrt{\frac{2}{\rho}} \int_o^t \sqrt{q}\ dy$$

$$= \sqrt{\frac{2}{\rho}\ (p_t - \Delta p)} \int_o^t e\ x\ \frac{y}{t}\ dy$$

$$= \frac{t}{x} \sqrt{\frac{2}{\rho}\ (p_t - \Delta p)}\ (e^x - 1)$$

Continuing, the discharge coefficient can be calculated

$$\mathcal{D}_c = \frac{Q/C}{\sqrt{\frac{2}{\rho}\,\Delta p}\; h}$$

or

$$\mathcal{D}_c = \frac{t/h}{x}\sqrt{\frac{1 - \Delta p/p_t}{\Delta p/p_t}}\,(e^x - 1)$$

Substituting for $\frac{\Delta p}{p_t}$ from Equation [B-4] gives, after some reduction

$$\mathcal{D}_c \doteq \frac{1}{1 + \sin\theta}\sqrt{\tanh\frac{x}{2}} \qquad [B\text{-}5]$$

(Note: Do not use this equation if $x > 0.4$. See Equations [B-7] and [B-5'])

The jet reaction coefficient is given by

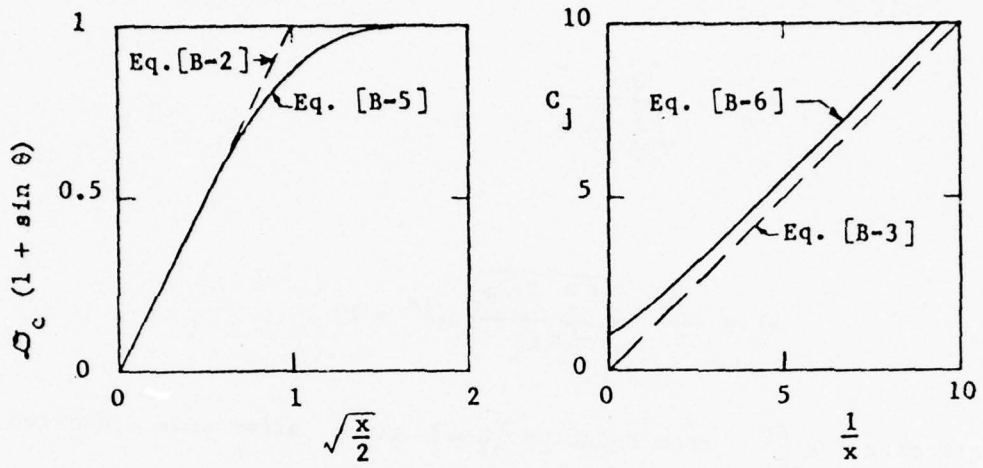
$$C_j = \frac{1}{\Delta p\; t}\int_o^t (\rho u^2 + p)\,dy = \frac{1}{\Delta p\; t}\int_o^t (p_t + q)\,dy$$

$$= \frac{p_t}{\Delta p}\,\frac{1}{t}\int_o^t \left[1 + \left(1 - \frac{\Delta p}{p_t}\right) e^{2x\,\frac{y}{t}}\right] dy$$

$$= \frac{p_t}{\Delta p}\,\frac{1}{t}\left[t + \frac{t}{2x}\,e^{-2x}\left(e^{2x} - 1\right)\right]$$

or, finally

$$C_j = \frac{1}{1 - e^{-2x}} + \frac{1}{2x} \qquad [B\text{-}6]$$




Note that Equations [B-4] through [B-6] agree exactly with Equations [B-1] through [B-3] in the limit $x \to 0$.

It is found that Equation [B-4] agrees remarkably well with experiment in the whole range $x > 0.2$, provided adequate account is taken of non-uniformities in total pressure over the nozzle exit. (In the range $x < 0.2$, the equation is valid for the ideal fluid considered; but, for real fluids, mixing effects become important in this range.)

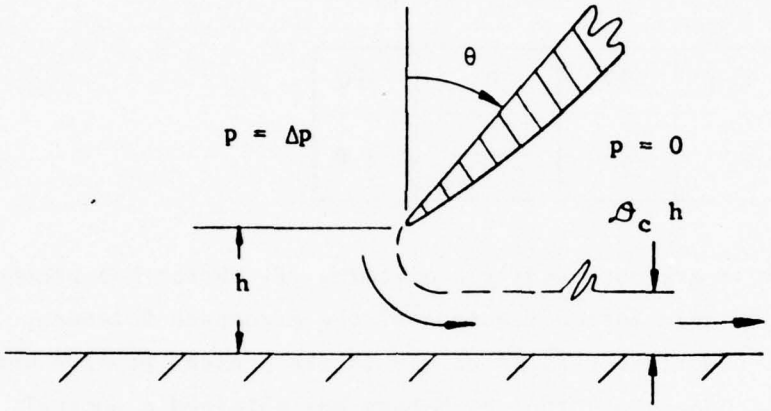
Equation [B-5], on the other hand, is found to agree with experiment in a more limited range, $0.2 < x < 0.4$, say. The Plenum Theory will provide the basis for a less restricted formula for $\mathcal{D}_c$.

Experimental data available from which to draw conclusions with regard to the validity of Equation [B-6] are limited. The equation is almost certainly reliable in the range $x < 0.4$, and it seems quite likely to be valid for most practical purposes in the whole range $0 < x < \infty$. In any event, it will be shown later that the jet reaction coefficient plays a rather small role in most practicaal design problems.

PLENUM THEORY

It is clear, on physical grounds, that the peripheral jet craft and the plenum craft become indistinguishable, from the standpoint of fluid mechanics, when the former's nozzle thickness parameter x becomes very

large. The plenum craft could be defined as the limiting form of the peripheral jet craft, in the limit $x \to \infty$.




Equation [B-4] gives

$$\lim_{x \to \infty} \left( \frac{\Delta p}{p_t} \right) = 1.0$$

and Equation [B-6] gives

$$\lim_{x \to \infty} C_j = 1.0$$

The results in both cases agree with our intuition and experience. However, Equation [B-5] gives

$$\lim_{x \to \infty} \mathcal{D}_c = \frac{1}{1 + \sin \theta}$$

which does not agree with experience, except in the isolated case $\theta = 90^\circ$. It is therefore to the discharge coefficient $\mathcal{D}_c$ that we will direct our attention.

Unfortunately, the seemingly simple fluid dynamic problem of discharge through a sharp-edged orifice presents rather formidable mathematical difficulties when one attempts direct solution.

Classical hydrodynamics provides solutions in the following discrete cases:

Discharge Coefficient for Plenum Craft

| $\theta$, deg | 90 | 0 | -90 |
|---|---|---|---|
| c | $\frac{1}{2}$ | $\frac{\pi}{\pi + 2}$ | 1.0 |

Further solutions are not readily available. F. Ehrich has presented a numerical method of calculation (Journal of the Aerospace Sciences, November 1961), but numerical results of use in the present problem were not included. It is understood that A. Gabbay has obtained a general solution (University of London Thesis, 1960), but this is not available to the present author at this time.

We will therefore proceed to select, rather arbitrarily, a formula which is consistent with the facts at hand. In addition to the discharge coefficients presented above, some additional inferences can be drawn from consideration of the pressure forces acting on the orifice lip. These considerations are discussed in Section F. We will presuppose them here by stating that, if the formula for $\mathcal{D}_c$ is written in the form

$$\mathcal{D}_c = \frac{1}{2}\left[1 + \frac{\cos\theta}{f(\theta)}\right]$$

where $f(\theta)$ is a function yet to be determined, it can be argued on physical grounds that

$$\frac{df(\theta)}{d\theta} > 0 \quad , \quad \theta > -\frac{\pi}{2}$$

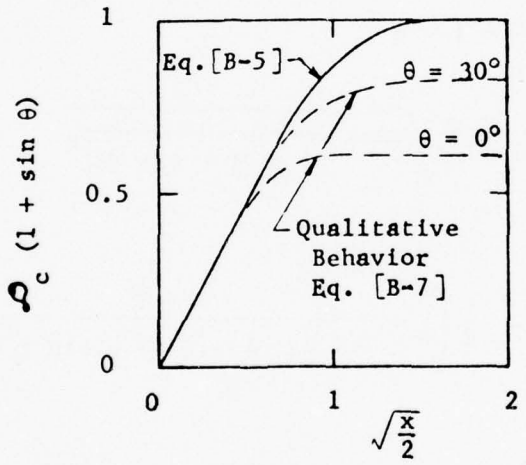
and that $f\left(\frac{\pi}{2}\right)$ is of order unity. A function, within the range of interest,

which satisfies these conditions, as well as the facts presented in the table, is

$$f(\theta) = \frac{\pi + 2}{\pi - 2}(1 + \sin\theta) - \sin\theta\cos\theta$$

We will therefore adopt the relationship

$$\lim_{x \to \infty} \mathcal{D}_c = \frac{1}{2}\left[1 + \frac{\cos\theta}{\frac{\pi + 2}{\pi - 2}(1 + \sin\theta) - \sin\theta\cos\theta}\right] \qquad [B\text{-}7]$$




There is no question of the adequacy of Equation [B-7] for practical design purposes. The empirical nature of this equation should be borne in mind, however, if one is tempted to apply it to analysis of more subtle academic questions. For example, it will be shown in Section F that the quantity $2\mathcal{D}_c - 1$ has a certain significance. While we can be confident that the percentage error incurred in predicting $\mathcal{D}_c$ itself by Equation [B-7] is small, a substantial percentage error might be incurred in predicting $2\mathcal{D}_c - 1$. (Care has been taken, however, to avoid any order-of-magnitude error in this particular quantity.)

MODIFIED EXPONENTIAL THEORY

The Exponential Theory gives results (Equations [B-4] through [B-6]) which are suitable for design application, except that Equation [B-5]

$\mathcal{C}_c$ cushion discharge coefficient $\left(\frac{Q}{S_g\sqrt{\frac{2}{\rho}\Delta p}}\right)$



gives incorrect discharge coefficients in the limit $x \to \infty$. We have determined from the Plenum Theory that the correct limits should be approximately as given by Equation [B-7]. It will be convenient to have a new expression for discharge coefficient which agrees with Equation [B-5] for small and moderate values of x, but approaches agreement with Equation [B-7] as $x \to \infty$. Such an expression (Equation [B-5']) is given below as part of a complete set of *air cushion relationships* suitable for design application:

If $x > 0.2$:

$$\frac{\Delta p}{p_t} = 1 - e^{-2x} \qquad [B\text{-}4]$$

$$\mathcal{C}_c = \bar{\mathcal{C}}_c \sqrt{\tanh \frac{x}{2\left[\bar{\mathcal{C}}_c(1 + \sin\theta)\right]^2}} \qquad [B\text{-}5']$$

where

$$\bar{\mathcal{C}}_c \equiv \lim_{x\to\infty} \mathcal{C}_c \doteq \frac{1}{2}\left[1 + \frac{\cos\theta}{\frac{\pi + 2}{\pi - 2}(1 + \sin\theta) - \sin\theta\cos\theta}\right] \qquad [B\text{-}7]$$

$$C_j = \frac{1}{1 - e^{-2x}} + \frac{1}{2x} \qquad [B\text{-}6]$$

These relationships, which will provide the basis for most of our subsequent considerations of the air cushion, are presented graphically in Figures B-1 through B-3, along with the quantity $\frac{\mathcal{C}_c}{\Delta p/p_t}$, which has a special significance in connection with the cushion power — our next topic of discussion.

Note particularly the fact that we no longer need to draw any sharp distinction between cushions of the plenum type and of the peripheral jet type. The above relationships yield cushion properties which gradually approach those of the classical plenum type, as x is made larger and larger.

## PERFORMANCE CHARACTERISTICS

### TOTAL LIFT AND CUSHION POWER

The total lift must equal the gross weight W, in an equilibrium condition. It is composed of the sum of the base-pressure reaction, the jet reaction, and the external aerodynamic lift.

$$W = \Delta p\ S + (C_j - 1)\ \Delta p\ S_N \cos\theta + C_L\ q_a\ S$$

or

$$W = \Delta p\ S \left[ 1 + (C_j - 1) \frac{S_N}{S} \cos\theta + C_L \frac{q_a}{\Delta p} \right] \qquad [B\text{-}8]$$

where $S_N$ is total nozzle area and $q_a$ is free-stream dynamic pressure of the ambient air.

The cushion power $P_c$ is the shaft power supplied to the cushion system compressor(s). In the absence of any losses, this would simply be the rate of work required to raise the total pressure of air at volume flow rate Q from $q_a$ (free-stream total pressure) to $p_t$ (nozzle-exit total pressure); that is,

$$P_c = Q\ (p_t - q_a)$$

However, there will inevitably be losses in the internal system. These will be discussed at some length in Section C. For the present, we will simply introduce and apply the following efficiency parameters, to be defined in Section C:

Recovery efficiency, $\eta_r$

Duct efficiency, $\eta_d$

Compressor efficiency, $\eta_c$

Internal efficiency, $\eta_{int} = \eta_c\ \eta_d$

The cushion power becomes:

$$P_c = Q\left(\frac{p_t}{\eta_d} - \eta_r\, q_a\right) \div \eta_c$$

Recalling the previous definition

$$\mathcal{O}_c \equiv \frac{Q}{\sqrt{\frac{2}{\rho}\Delta p}\, S_g}$$

we have, after some reduction

$$P_c = \Delta p \sqrt{\frac{2}{\rho}\Delta p}\, S_g \left[\frac{c}{\Delta p/p_t} \div \eta_{int} - \mathcal{O}_c \frac{\eta_r}{\eta_c}\frac{q_a}{\Delta p}\right] \qquad [B\text{-}9]$$

The significance of the quantity $\frac{\mathcal{O}_c}{\Delta p/p_t}$, shown graphically in Figure B-3, is now apparent. For design conditions common to most of the important current vehicles; namely,

$$\frac{S_g}{S} \ll 1$$

$$\Delta p \doteq \frac{W}{S}$$

$$\frac{q_a}{\Delta p} \ll 1$$

the cushion power is almost directly proportional to this quantity. If, subject to these conditions, we were to define an "optimum jet geometry" providing minimum cushion power for given weight W and cushion area S, Figure B-3 would show this optimum to correspond roughly to the geometry: $\theta \doteq 90°$, $x \doteq 0.7$, $\frac{t}{h} \doteq 0.35$. However, a rather large number of practical considerations intervene to render any such simple concept of an "optimum geometry" practically meaningless.

Equation [B-9] has the weakness that it contains the quantity $\Delta p$, the cushion pressure, which is not really a primary design variable, although, as we have already noted, $\Delta p$ is usually approximately equal to the primary variable "cushion loading" w, defined as follows:

$$w \equiv \frac{W}{S}$$

We can rectify the situation by substituting for $\Delta p$ from Equation [B-8], but, in so doing, we will introduce the important simplifying approximation

$$W \doteq \Delta p \; S \left(1 + C_L \frac{q_a}{\Delta p}\right) \quad , \quad \frac{S_g}{S} \ll 1$$

or

$$\Delta p \doteq w \left(1 - C_L \frac{q_a}{w}\right) \quad , \quad \frac{S_g}{S} \ll 1 \qquad [B\text{-}8']$$

(The justification for the condition $\frac{S_g}{S} \ll 1$ is as follows: Referring to Equation [B-8], it is evident that the approximation [B-8'] is valid provided

$$(C_j - 1) \frac{S_N}{S} \cos \theta \ll 1$$

Noting that

$$S_N \leq tC$$

$$S_g = hC$$

$$\frac{S_N}{S_g} \leq \frac{x}{1 + \sin \theta}$$

we see that Equation [B-8'] is valid provided

$$x(C_j - 1) \frac{S_g}{S} \frac{\cos \theta}{1 + \sin \theta} \ll 1$$

Now referring to Equation [B-6], we find

$$\lim_{x \to 0} \left[ x\left(C_j - 1\right) \right] = 1$$

$$\lim_{x \to \infty} \left[ x\left(C_j - 1\right) \right] = \frac{1}{2}$$

The ambitious reader can convince himself that the quantity

$$x\left(C_j - 1\right) \frac{\cos \theta}{1 + \sin \theta}$$

is of order unity or lower, under all conditions of practical interest; hence, the condition $\frac{S_g}{S} \ll 1$ is sufficient.)

Combining Equations [B-9] and [B-8'], we obtain

$$P_c = w \sqrt{\frac{2}{\rho} w}\, S_g \left[ \left(1 - C_L \frac{q_a}{w}\right)^{\frac{3}{2}} \frac{\rho_c}{\Delta p / p_t} \div \eta_{int} - \left(1 - C_L \frac{q_a}{w}\right)^{\frac{1}{2}} \rho_c \frac{\eta_r}{\eta_c} \frac{q_a}{w} \right] \quad , \quad \frac{S_g}{S} \ll 1 \qquad [B\text{-}9']$$

It is convenient to rewrite this relationship in terms of the nondimensional "cushion specific power":

$$\frac{P_c}{W V_o} = \frac{S_g}{S} \left[ \left(1 - C_L \frac{q_a}{w}\right)^{\frac{3}{2}} \frac{K_c}{\sqrt{q_a / w}} - \left(1 - C_L \frac{q_a}{w}\right)^{\frac{1}{2}} R_c \sqrt{q_a / w} \right] \quad , \quad \frac{S_g}{S} \ll 1 \qquad [B\text{-}10]$$

where

$$K_c \equiv \frac{\rho_c}{\Delta p / p_t} \div \eta_{int}$$

$$R_c \equiv \rho_c \frac{\eta_r}{\eta_c}$$

An easier insight into the relationship given by Equation [B-10] is obtained by substituting $q_a \doteq (V_k/17.2)^2$ for standard air density, and introducing a further approximation, valid when $(q_a/w) \ll 1$; that is,

$$\frac{P_c}{WV_o} \doteq \frac{S_g}{S} K_c \frac{17.2\sqrt{w/\ell}}{V_k/\sqrt{\ell}} \quad , \quad \left\{ \begin{matrix} (q_a/w) \ll 1 \\ \\ (S_g/S \ll 1 \end{matrix} \right\} \qquad [B\text{-}10']$$

where w is expressed in pounds per square foot and $V_k$ in knots. The condition $q_a/w \ll 1$ is not quite satisfied throughout the whole operating range of practical vehicles, so Equation [B-10'] is not suitable for detailed design; but it correctly expresses the dominant behavior of the cushion specific power. The essence of the truth is expressed in the not-quite-true statement, "The cushion specific power is directly proportional to the ratio of air-gap area to cushion area, and inversely proportional to the speed-length ratio $V_k/\sqrt{\ell}$."

The total specific power is the sum of cushion specific power and propulsion specific power

$$\frac{P}{WV_o} = \frac{P_c}{WV_o} + \frac{P_p}{WV_o}$$

and is directly proportional to the fractional weight of fuel consumed per mile. (The factor of proportionality for ordinary engines is approximately 1/650. Roughly speaking, then, if $\frac{P}{WV_o} = 0.1$, about 10 percent of the gross weight will be consumed as burned fuel in traveling 650 nautical miles.) Thus the specific power and the load-carrying capacity of the vehicle completely determine the range-payload characteristics and direct operating costs.

We will consider the problem of estimating the propulsion specific power in later Sections.

NON-UNIFORM NOZZLE PARAMETERS

In the derivation of the cushion performance relationships

(Equations [B-4] through [B-10]), it was implied that the quantities t, $\theta$, h, and $p_t$ were uniform around the periphery.

NOZZLE THICKNESS PARAMETER x UNIFORM - Actually the results are valid under a somewhat less limiting restriction; namely, the restriction that the parameter

$$x \equiv \frac{t}{h} (1 + \sin \theta)$$

be uniform around all parts of the free periphery, and that air be supplied to all parts of the nozzle at uniform total pressure $p_t$.

However, if the jet angle $\theta$ varies around the periphery, then it is necessary to use an average value of the discharge coefficient, calculated as follows: We define a curvilinear coordinate c, measured along the cushion periphery (in plan view), such that

$$\oint dc \equiv C$$

$$\oint h \, dc \equiv S_g$$

Then we define

$$(\mathcal{A}_c)_{av} \equiv \frac{1}{S_g} \oint \mathcal{A}_c \, h \, dc$$

and insert this value in place of $\mathcal{A}_c$ in Equations [B-5] through [B-10]. (In other words, we take a weighted average of $\mathcal{A}_c$ around the periphery, with air gap h as weighting factor.)

PRODUCT $\left[p_t \left(1 - e^{-2x}\right)\right]$ UNIFORM - A still further generalization is possible. If the air is supplied to different parts of the nozzle at different total pressures $p_t$ (that is, from different compressor systems all operating at the same compressor efficiency $\eta_c$ and duct efficiency $\eta_d$), Equations [B-4] through [B-10] are still applicable provided the quantity

$$p_t \left(1 - e^{-2x}\right)$$

is uniform around all parts of the free periphery. Furthermore, it is assumed that the above weighted average will be used in place of $\mathcal{O}_c$, and a weighted average

$$\left(\frac{\mathcal{O}_c}{\Delta p/p_t}\right)_{av} \equiv \frac{1}{S_g} \oint \frac{\mathcal{O}_c}{\Delta p/p_t} \, h \, dc$$

will be used in place of $\dfrac{\mathcal{O}_c}{\Delta p/p_t}$ .

This generalization is of practical importance, for example, in case peripheral jets are to be used around most of the free periphery, while the cushion is allowed to function as a plenum type at the stern. (This is one way of avoiding the tendency of stern nozzles to snag on obstacles or scoop up water.) Equations [B-4] through [B-10] are applicable to this case, using the weighted-average parameters $\left(\mathcal{O}_c\right)_{av}$ and $\left(\dfrac{\mathcal{O}_c}{\Delta p/p_t}\right)_{av}$, provided the air which escapes under the stern is continually replaced by a separate compressor delivering air directly to the cushion at a total pressure $p_t \doteq \Delta p$.

OFF-DESIGN CUSHION PERFORMANCE

When the conditions specified in the preceding paragraphs are met; that is, either x and $p_t$ are uniform or $p_t\left(1 - e^{-2x}\right)$ is uniform around the free periphery, the cushion is said to be operating "on-design." Roughly speaking, the physical significance of on-design operation is that the nozzle conditions at each section of the periphery are just sufficient to maintain the cushion pressure $\Delta p$ without having air entering or leaving the cushion itself.

When these conditions are not met (off-design cushion operation), the jet is relatively stronger in some sections and weaker in others; and equilibrium is established through the mechanism of air entering the cushion from the strong sections of the jet and escaping under the weak sections. (Note that we define off-design cushion operation independently of whether or not the compressors are operating on-design. Compressors will be discussed in the next section.)

Certain conclusions can be drawn immediately.

1. Considering peripheral-jet machines in operation over a level surface:

a. If h, t, and $\theta$ are uniform around the free periphery, then the cushion will operate on-design so long as it is in level attitude, regardless of operating height.

b. If h, t, or $\theta$ varies around the free periphery, then only one operating height and attitude can provide on-design cushion operation. (There are mathematical exceptions to this rule, but not practical exceptions.)

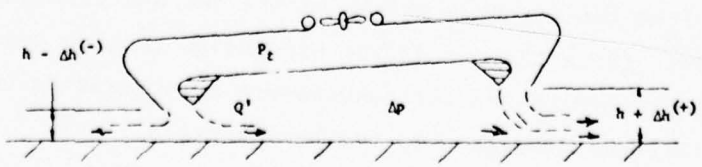
c. Practically speaking, peripheral-jet machines cannot provide on-design operation over an irregular surface.

2. In principle (and in practice, within reason), plenum type machines provide on-design cushion operation regardless of operating height, attitude, or surface irregularities.

In other words, peripheral-jet cushions practically never operate on-design, whereas plenum cushions practically always operate on-design. We must therefore re-examine our previous conclusion, that peripheral-jet types are more efficient, in light of the penalty to be paid in practice due to off-design conditions.

It is fairly easy to see that there is a penalty for off-design operation. It is rather difficult to calculate the precise magnitude of the penalty. We will consider only a few of the simpler cases, in order to get some idea of the magnitude of the penalty.

TWO-DIMENSIONAL GEM IN PITCH - When a sidewall craft, on-design in level trim, is placed at a finite trim angle, a part of the air from the low-end nozzle will flow through the cushion and pass out under the high-end jet.



If the cushion pressure $\Delta p$, total pressure $p_t$, and volume flow rate Q remain unchanged, then the relationship between the air-gap decrease $\Delta h^{(-)}$ at the low end and increase $\Delta h^{(+)}$ at the high end can be estimated, in the case $x \ll 1$, as follows:

The momentum balance across the low-end jet gives

$$\left.\begin{aligned} \Delta p \left( h - \Delta h^{(-)} \right) &\doteq \Delta p\, h - 2\, \rho\, Q' \sqrt{\frac{2}{\rho}\, p_t} \\ \Delta h^{(-)} &\doteq \frac{2 \rho\, Q'}{\Delta p} \sqrt{\frac{2}{\rho}\, p_t} \end{aligned}\right\} \quad x \ll 1 \quad .$$

If we assume the total pressure of the portion Q' of the air is reduced to $\Delta p$ in traversing the cushion, we have:

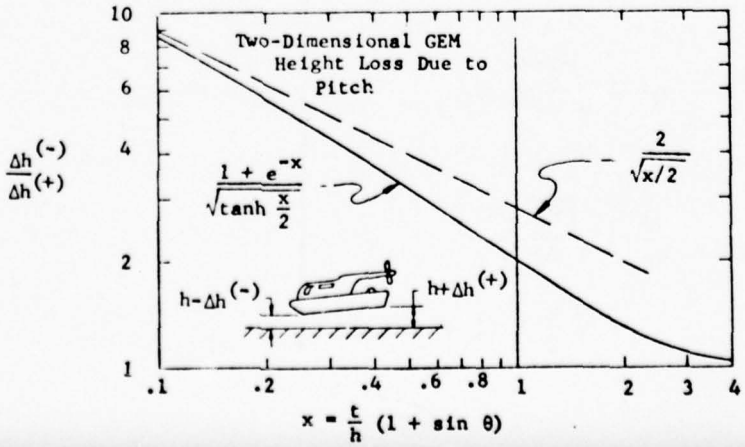
$$\Delta h^{(+)} \doteq \frac{Q'}{\sqrt{\frac{2}{\rho}\, \Delta p}}$$

$$\frac{\Delta h^{(-)}}{\Delta h^{(t)}} \doteq 4 \sqrt{\frac{p_t}{\Delta p}}$$

or, since $\Delta p \doteq 2x\, p_t$ when $x \ll 1$,

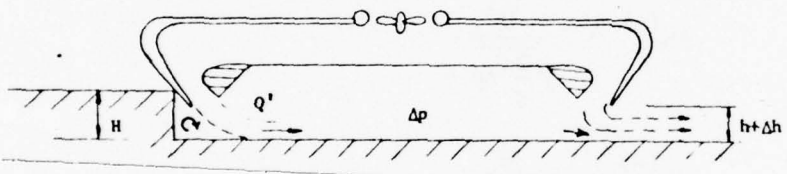
$$\frac{\Delta h^{(-)}}{\Delta h^{(+)}} \doteq \frac{2}{\sqrt{x/2}}$$

This relationship is plotted, along with a modified relationship giving the correct behavior $\frac{h^{(-)}}{h^{(+)}} \to 1.0$, as $x \to \infty$.



We cannot rely very heavily on either of the relationships presented in the graph, except in cases of rather thin nozzles (small x); but evidently the loss of average air gap with finite pitch angles is substantial in such cases. (It is interesting to note that this behavior would give rise to a strong coupling between pitch and heave, in case of dynamic motions; that is, a pitching oscillation would excite heave oscillations.)

OBSTACLE CLIMBING - Consider a sidewall craft attempting to traverse an obstacle just the proper width to pass between the sidewalls.

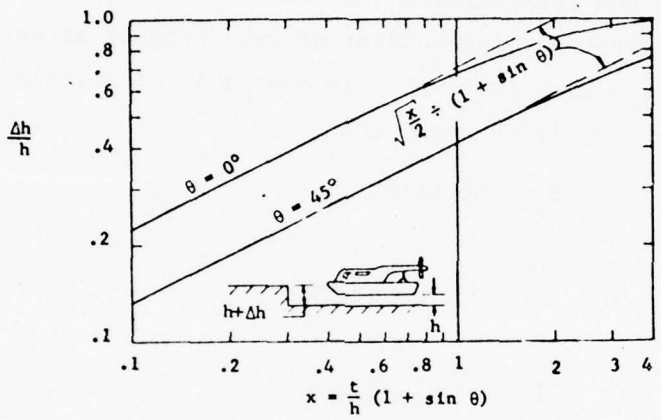



For simplicity we assume that level trim is maintained and total pressure $p_t$ remains constant as the obstacle is approached. In the situation sketched, we have

where

$$\left.\begin{aligned} \Delta h &\doteq \frac{Q'}{\sqrt{\frac{2}{\rho}\,\Delta p}} \\ Q' &\doteq \sqrt{\frac{2}{\rho}\,p_t}\;t \\ \frac{\Delta h}{h} &\doteq \sqrt{\frac{p_t}{\Delta p}}\,\frac{t}{h} \\ &= \sqrt{\frac{x}{2}} \div (1 + \sin\theta) \end{aligned}\right\} \quad x \ll 1$$

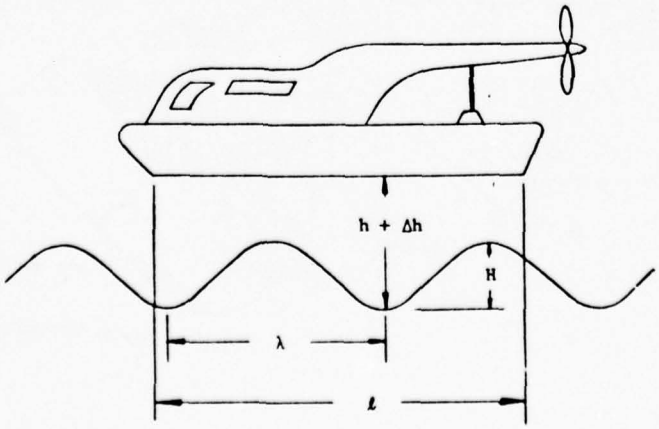
(Here, h is air gap corresponding to level-trim operation at a distance from the obstacle.) This relationship is shown graphically below, together with an arbitrarily modified relationship consistent with the obvious limiting case, $\frac{\Delta h}{h} \to 1$, as $x \to \infty$.




Two-Dimensional GEM Step Obstacle Negotiable in Level Trim

FLIGHT OVER WAVES - A problem of considerably greater practical interest is posed by the case of a full-peripheral-jet craft (no sidewalls, etc.) moving without water contact normal to the wave crests of an idealized sinusoidal sea. We consider the case $\frac{\ell}{b} >>$ ; amd $\frac{\ell}{\lambda} >> 1$, so that the area of the air gap is

$$S_g \doteq 2\,\ell\left(h + \Delta h - \frac{H}{2}\right)$$



Unfortunately, as is often the case when we come to a problem of practical importance, this problem is considerably more complicated than the artificial examples of the preceding two paragraphs. We will content ourselves with some deductions concerning the qualitative behavior. We will, as before, adopt the reasonable assumption that the quantities Q and $p_t$ remain fixed (that is, independent of wave height) at values producing an air gap of h feet in level trim over a level surface.

For the plenum, $x \to \infty$, it is clear that

$$S_g = \text{Constant}$$

Therefore

$$h + \Delta h = \frac{H}{2} + \text{Constant}$$

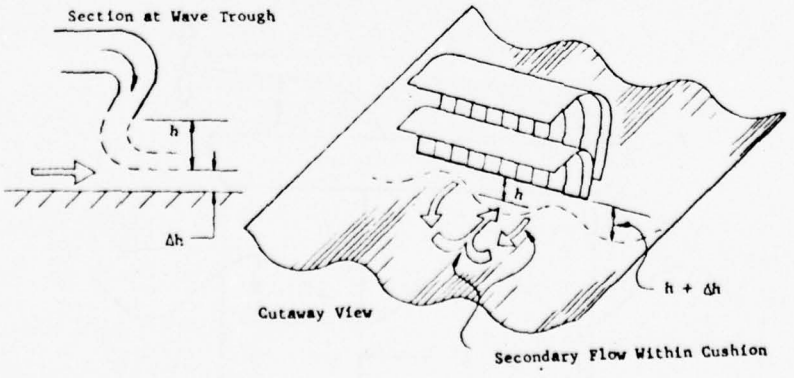
but

$$\Delta h = 0 \text{ when } H = 0$$

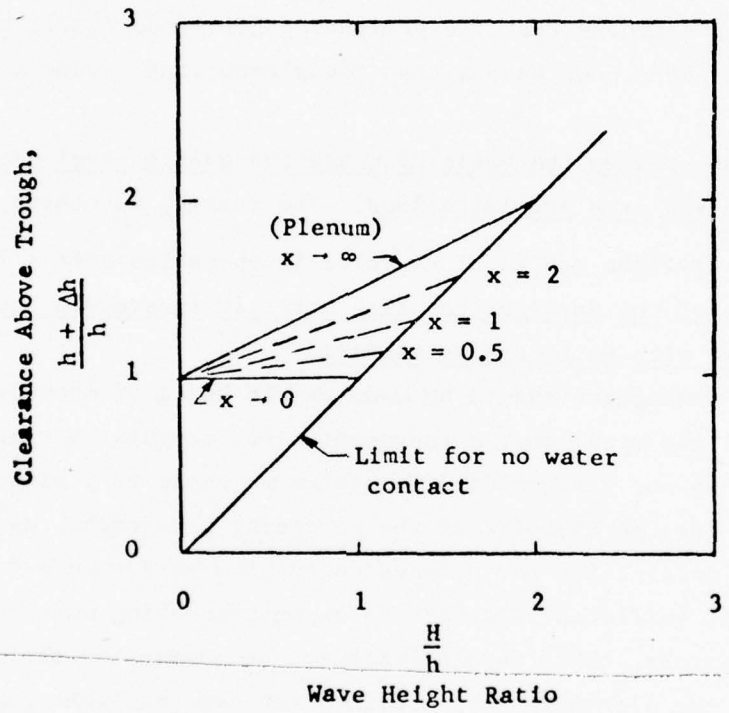
so

$$\Delta h = \frac{H}{2}$$

Also, for a very thin jet, $x \to 0$, it is clear that $\Delta h \doteq 0$, since the portion of the air gap corresponding to $\Delta p$ in the wave troughs must be filled by a volume flow of air diverted from the jet near the wave crests; and, as $x \to 0$ , the volume flow of air available also approaches zero.



This behavior is graphed below, together with intuitive estimates of the behavior when x is finite. It must be borne in mind, however, that these intuitive estimates have no claim to validity, quantitatively.




Note: Dashed lines are merely intuitive estimates.

PLENUM VERSUS PERIPHERAL JET - Our considerations of off-design cushion performance have raised some extremely important questions regarding the widely accepted notion that peripheral-jet craft are more efficient than plenum craft.

There is no doubt that if they are compared on the basis of equal weight, cushion area, and daylight gap area, and also on the basis of

operation over a level surface, then the peripheral jet is substantially more efficient. Equations [B-10] and [B-10'] show the cushion power to be roughly proportional to the quantity $\mathcal{D}_c \div (\Delta p/p_t)$; and Figure B-3 shows that this quantity is likely to be only 70 to 80 percent as large, for well designed peripheral jets, as for plenums. (Moreover, we will see in the following sections that the peripheral jet lends itself to slightly smaller, lighter compressors than the plenum, and produces less momentum drag.)

However, comparisons on the basis of operation over a level surface with equal daylight gap area are fallacious! The reason, of course, is that a significant daylight gap is of no value in operation over a level surface. The value of the daylight gap lies entirely in avoiding or minizing the contact with an irregular surface.

A more proper comparison should be made on the basis of operation, with equal daylight gap area, over a representative irregular surface. Unfortunately this is not easy to do. Even when we chose very simple irregular surfaces for the examples of the preceding paragraphs, we did not succeed in determining the off-design cushion performance for peripheral jets with sufficient accuracy to support anything more than very general conclusions. What we did learn can be summarized as follows: At constant power, the plenum ($x \to \infty$) provided the same daylight gap area over an irregular surface as over a level surface; whereas the thin peripheral jet ($x \to 0$) lost as much as half its daylight gap area in moving from a level to an irregular surface. (In fact, we could easily construct other examples in which the loss would be more than half.) However, the crucial question, of how much is lost with intermediate jet thicknesses, of the order of $x \doteq 1$, say, remains inadequately answered. Almost certainly, a noticeable fraction — and perhaps most — of the peripheral jet's apparent 20 to 30 percent level-surface advantage is bound to disappear when one succeeds in making proper comparisons; and the "optimum" nozzle thickness parameter will undoubtedly be found to be still thicker than the value of $x \doteq 0.7$, which would be deduced from Figure B-3.

Thus, the question of which is better, plenum or peripheral jet, may be largely an academic question. If the answer is peripheral jet, it will likely be a very plenum-like peripheral jet, with performance very like a simple plenum.

It is extremely important to bear in mind, however, that peripheral jets with thin nozzles will suffer a severe penalty in off-design cushion operation. The importance of keeping this in mind is emphasized by the facts that we have adopted cushion performance formulas, Equations [B-4] through [B-10], which are based upon on-design operation; and that considerations of the internal aerodynamics and the momentum drag provide arguments in favor of thinner nozzles, as we shall see in the following sections.

Finally, it is necessary to remember that the whole concept of on-design cushion operation (and on-design cushion performance as given by Equations [B-4] through [B-10]) is only a convenient idealization. In actual practice, air cushions (except for simple plenum types) practically never operate on-design.

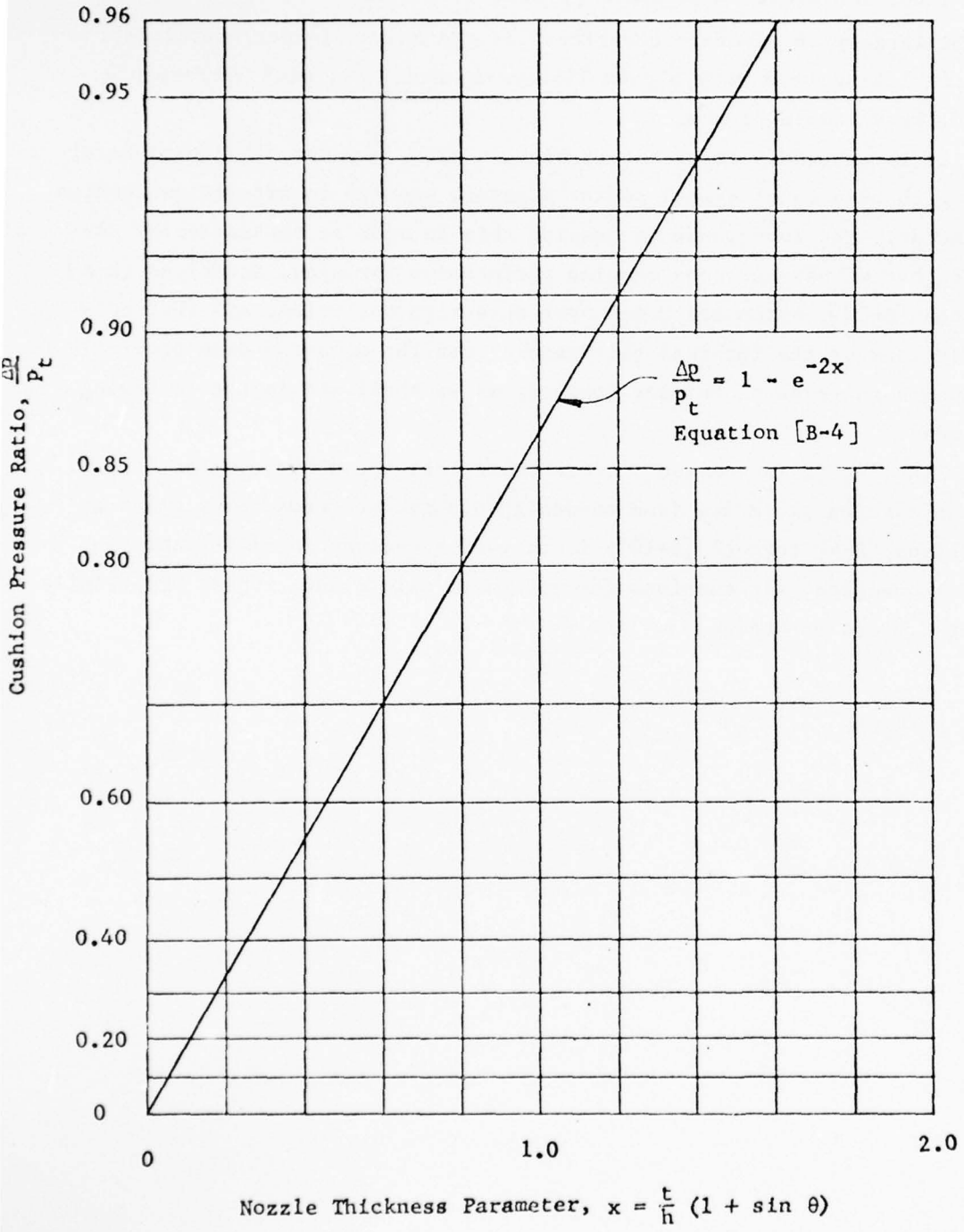



Figure B-1 - Air Cushion Pressure Ratio as a Function of Nozzle Thickness Parameter

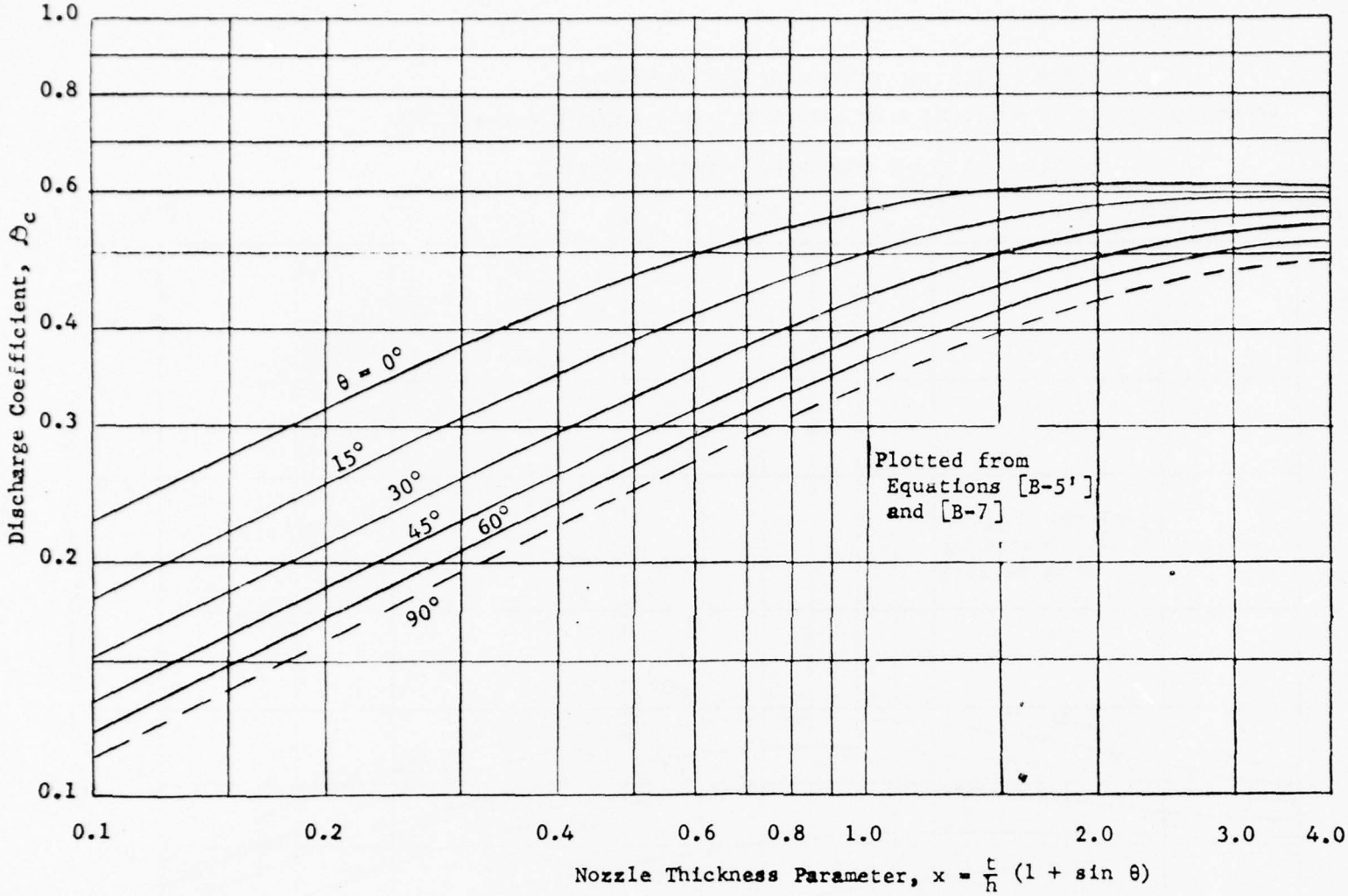



Figure B-2 - Discharge Coefficient as a Function of Nozzle Thickness Parameter and Peripheral Jet Efflux Angle

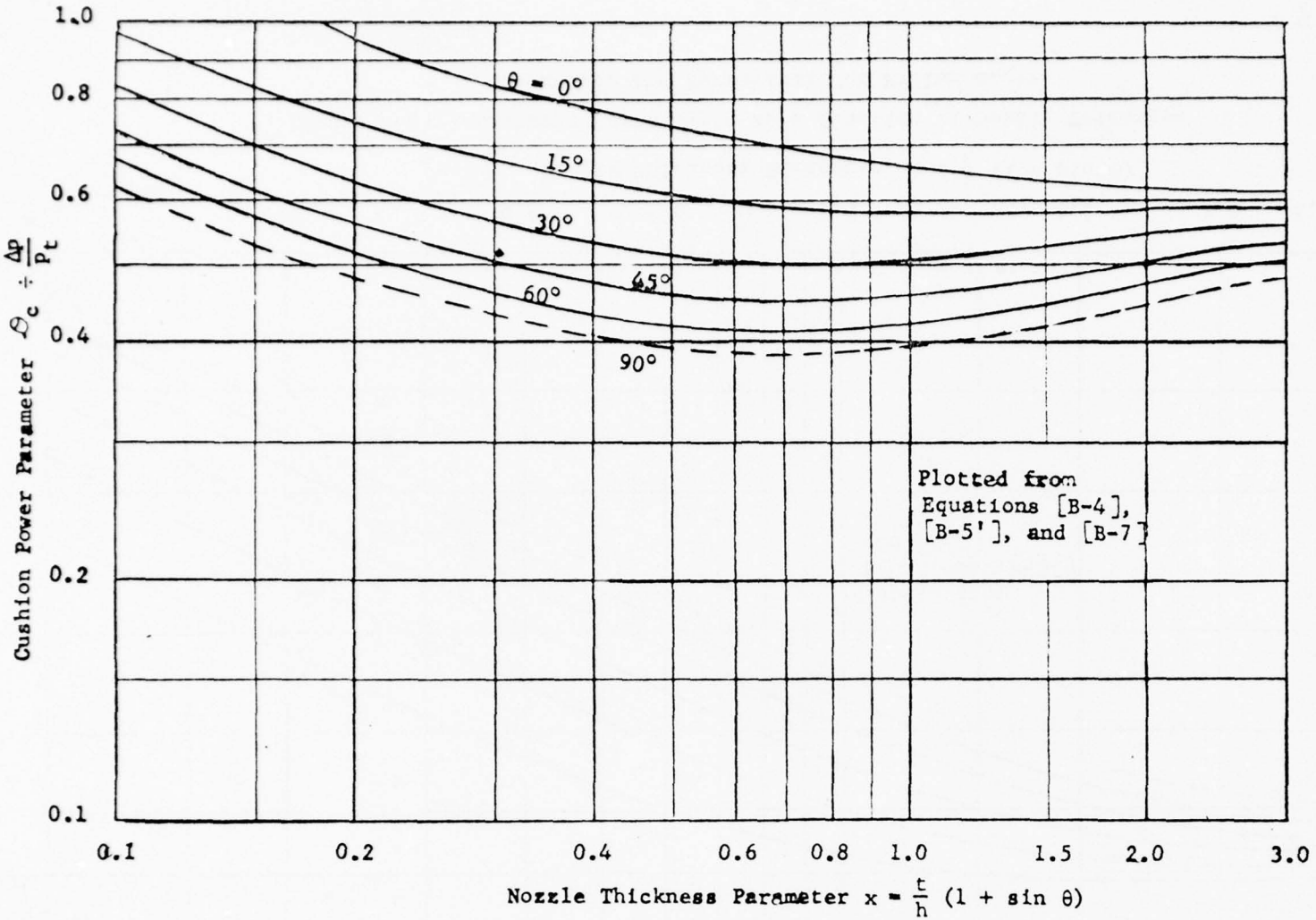



Figure B-3 - Cushion Power Parameter as a Function of Nozzle Thickness Parameter and Peripheral Jet Efflux Angle

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13 ABSTRACT

The energy and momentum relationships governing air cushion performance are reviewed. The exponential-theory equations are recommended for calculation of cushion pressure and jet reaction, and a modified equation is proposed for calculation of the volume flow rate.

The question of off-design cushion performance is discussed. It is pointed out that off-design performance is of paramount practical importance and that the widely held notion that peripheral jet cushions are more efficient than simple plenum cushions is not necessarily valid in this context.

The danger of drawing erroneous conclusions from the cushion performance equations, which (except for the plenum) do not apply to off-design operation, is emphasized.

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|---|---|---|---|---|---|---|
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| Air Cushion Vehicle | | | | | | |
| Peripheral Jet Craft | | | | | | |
| Plenum-Type Craft | | | | | | |
| Performance Characteristics | | | | | | |
| Nozzle Thickness | | | | | | |
| "Daylight" Gap | | | | | | |
| Peripheral Jet Efflux Angle | | | | | | |
| Air Volume Rate of Flow | | | | | | |
| Cushion Power | | | | | | |
| Cushion Pressure | | | | | | |
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# SOME DESIGN PRINCIPLES OF GROUND EFFECT MACHINES
## SECTION C — INTERNAL AERODYNAMICS

by

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## Foreword

This report is based on a lecture series presented by the authors at the von Kármán Institute for Fluid Dynamics, Rhode-Saint-Genèse, Belgium, in May 1965; and at the University of Maryland, College Park, Maryland, in July 1965. The lectures were prepared under the joint auspices of the David Taylor Model Basin and the Naval Air Development Center. They were presented in Belgium under the joint sponsorship of the von Kármán Institute and the Advisory Group for Aerospace Research and Development (AGARD); and in Maryland under the sponsorship of the Assistant Secretary of the Navy for Research and Development.

The revised lectures will be presented as follows:

A. Introductory Survey
B. Air Cushion Mechanics
C. Internal Aerodynamics
D. Drag
~~E. Drag Optimization for Sidewall~~ GEM (CAB)
F. Cushion Contributions to Stability
G. Seakeeping
H. Performance Summary

## NOTATION

| | |
|---|---|
| $R$ | compressor reference dimension: effective radius from the compressor axis to the compressor blades, ft |
| $\Omega$ | compressor rotational speed, radians/sec |
| $V_b$ | reference blade velocity ($\Omega R$), ft/sec |
| $q_b$ | reference blade dynamic pressure ($\rho_a V_b^2/2$), lb/ft$^2$ |
| $V_n$ | mean radial velocity through blade row, ft/sec |
| $\lambda$ | internal advance ratio ($V_n/V_b$) |
| $S_1$ | compressor flow area at inlet face, ft$^2$ |
| $S_2$ | compressor reference flow area at blade row (including area occupied by blades), ft$^2$ |
| $S_3$ | compressor exit area, ft$^2$ |
| $S$ | cushion area, ft$^2$, measured, in plan view, to outer edge of nozzle exit |
| $S_g$ | daylight gap area (cushion perimeter times mean daylight gap ($hC$), ft$^2$) |
| $\rho_a$ | air density, slugs/ft$^3$ |
| $H_{\ell,i}$ | mean inlet total pressure loss, lb/ft$^2$ |
| $H_{\ell,i_o}$ | reference inlet total pressure loss, lb/ft$^2$. Value of $H_{\ell,i}$ when $V_o = 0$ |
| $H_{\ell,d}$ | duct loss, lb/ft$^2$. Mean loss of total pressure between compressor exit and nozzle exit (or cushion entry) |
| $q_1$ | compressor inlet reference dynamic pressure, lb/ft$^2$ $\left[\lambda^2 q_b \left(\frac{S_2}{S_1}\right)^2\right]$ |
| $q_3$ | compressor exit reference dynamic pressure, lb/ft$^2$ $\left[\lambda^2 q_b \left(\frac{S_2}{S_3}\right)^2\right]$ |
| $q_r$ | cushion reference dynamic pressure, lb/ft$^2$ $\left[\bar{\alpha}_c^2 \Delta p\right]$ |

$K_{\ell,i}$ inlet loss coefficient $\left(H_{\ell,i_o}/q_1\right)$

$K_{\ell,d}$ duct loss coefficient $\left(H_{\ell,d}/q_3\right)$

$\eta_r$ ram recovery efficiency $\left(\frac{q_o - H_{\ell,i}}{q_o - H_{\ell,i_o}}\right)$

$\eta_d$ duct efficiency $\left(\frac{p_t}{\Delta p_t + \eta_r q_o}\right)$

N number of identical compressors in operation

Z number of compressor blades

$c_b$ blade chord, ft

$\sigma$ compressor solidity ratio $\left(\frac{Z\ c_b}{2\pi\ R}\right)$

$\beta$ blade angle, deg

$\phi$ reference relative flow angle $\left(\equiv \tan^{-1}\ \lambda\right)$

$\alpha$ airfoil angle of attack, deg

$c_\ell$ airfoil sectional lift coefficient

$c_d$ airfoil sectional drag coefficient

$P_c$ shaft power supplied to compressor, lb-ft/sec

$\Delta p$ cushion pressure, $lb/ft^2$

$\Delta p_t$ total pressure rise across compressor, $lb/ft^2$

Q volume flow rate, $ft^3/sec$

$\eta_c$ compressor efficiency $\left(\frac{Q\ \Delta p}{P_c}\right)$

$\eta_{int}$ internal efficiency $\left(\eta_d\ \eta_c\right)$

$C_c$ cushion discharge coefficient $\left(\frac{Q}{S_g\ \sqrt{\frac{2}{\rho}\ \Delta p}}\right)$

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## SUMMARY

The power dissipated through losses in the inlet-compressor-ducting system is discussed. A simple illustrative derivation of the characteristics of a hypothetical compressor is carried out. These characteristics are then used to illustrate the construction of combined internal characteristics charts.

The dominant influence of the air gap area in determining the necessary size of the compressor(s) is emphasized.

## INTRODUCTION

The "ideal" cushion power is the product of the volume flow rate times the total pressure at nozzle exit (or cushion entry), $Q\ p_t$. The actual shaft power can easily be several times larger than this if the compressor and ducting are not properly matched to the cushion requirements.

A detailed consideration of compressor design and performance would be beyond the scope of the present study. On the other hand, some facets of the internal aerodynamics of GEM's can be explored most conveniently in terms of explicit compressor characteristics, and it is rather important to understand, qualitatively at least, why the compressor behaves as it does. To this end, a simple approximate derivation of the characteristics of a typical compressor will be carried out.

## ILLUSTRATIVE COMPRESSOR CHARACTERISTICS

For this purpose we will consider only one type of compressor, the airfoil-type radial compressor sketched in Figure C-1. Moreover we will restrict ourselves to the case of moderate solidity and moderate internal advance ratio (Say, $\sigma \leq 0.6\ \lambda \leq 0.4$) so that we can neglect swirl, mutual interference between airfoils, and higher orders of $\lambda$ without errors so large as to obscure the primary effects.

Within these limitations, the aerodynamic radial force coefficient for each blade is roughly equal to the sectional lift coefficient of the equivalent two-dimensional airfoil at the same angle of attack (see Figure C-1):

$$C_r \equiv \frac{\text{Radial Force per Blade}}{q_b\ (\text{Blade Area})} \doteq c_\ell$$

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The total pressure rise across the blade row is

$$\Delta p_t = \frac{\text{(Radial Force per Blade)} \times \text{(No. of Blades)}}{S_2}$$

or

$$\Delta p_t \doteq \sigma c_\ell \, q_b \qquad [C\text{-}1]$$

Similarly, the tangential force coefficient (see vector diagram, Figure C-1) is given approximately by:

$$C_t \equiv \frac{\text{Tangential Force per Blade}}{q_b \text{ (Blade Area)}} \doteq \lambda c_\ell + c_d$$

The shaft power required to overcome the tangential blade forces is:

$$\text{(Tangential Force per Blade)} \times \text{(No. of Blades)} \times V_b$$

$$\doteq \sigma(\lambda c_\ell + c_d) \, q_b \, V_b \, S_2$$

In addition, we must add the power dissipated in friction on the housing of the rotating fan. Taking an average coefficient of friction $C_f$ we can express the power dissipated on one side of the lower disc as:

$$\int_0^{1.26\,R} \frac{1}{2} \rho_a \left( V_b \frac{r}{R} \right)^3 C_f \, 2\pi r \, dr \doteq 4 C_f \, q_b \, V_b \, R^2$$

$$\doteq 3 C_f \, q_b \, V_b \, S_2$$

Using arbitrary estimates for $C_f$ and for the number of equivalent half-discs to represent the rotating housing, we obtain for the total shaft power input

$$P_2 \doteq \sigma c_\ell \, q_b \, V_b \, S_2 \left[ \lambda + \frac{c_d}{c_\ell} + \frac{0.02}{\sigma c_\ell} \right] \qquad [C\text{-}2]$$

The useful power output is $\Delta p_t \, Q$ , where Q , the volume rate of flow, is:

$$Q = V_n \, S_2 = V_b \, S_2 \, \lambda$$

Therefore

$$\Delta p_t \, Q = \sigma \, c_\ell \, q_b \, V_b \, S_2 \, \lambda$$

The compressor efficiency is

$$\eta_c \equiv \frac{\Delta p_t \, Q}{P_c}$$

$$= \frac{\lambda}{\lambda + \frac{c_d}{c_\ell} + \frac{0.02}{\sigma \, c_\ell}}$$

Collecting the various results, we have

$$\eta_c \doteq \frac{\lambda}{\lambda + \frac{c_d}{c_\ell} + \frac{0.02}{\sigma \, c_\ell}} \qquad [C\text{-}3]$$

$$\frac{P_c}{V_b \, q_b \, S_2} \doteq \sigma \, c_\ell \left[ \lambda + \frac{c_d}{c_\ell} + \frac{0.02}{\sigma \, c_\ell} \right] \qquad [C\text{-}2']$$

$$\frac{\Delta p_t}{q_b} \doteq \sigma \, c_\ell \qquad [C\text{-}1']$$

where Equations [C-2'] and [C-1'] have been simply rewritten from Equations [C-2] and [C-1] in a form convenient for construction of a nondimensional compressor characteristics chart. The construction

of the chart proceeds as follows: For selected values of $\beta$ and $\sigma$, a value of $\alpha$ is chosen; $c_\ell$ and $c_d/c_\ell$ are read from Figure C-2; $\lambda$ is calculated from

$$\lambda \doteq \tan^{-1}(\beta - \alpha)$$

and Equations [C-1′], [C-2′], and [C-3] are applied to obtain one point on each of the graphs of Figure C-3. This process is repeated for successively different choices of $\alpha$ until complete lines are traced out.

The following points must be emphasized in regard to the approximate theory outlined above, and the compressor characteristics chart (Figure C-3) derived therefrom:

1. The compressor which we have considered (Figure C-1) is not a real compressor, in the sense of having ever been constructed and tested. It is a purely hypothetical compressor chosen for an illustrative example (a) because it lends itself to a very simple approximate theoretical analysis and (b) because the theoretical characteristics so derived are quite typical of the characteristics of the real compressors likely to be used in GEM applications. (The main qualitative differences which one would probably find if he constructed and tested our hypothetical compressor are these: The pressure-rise chart of Figure C-3 would show a slightly higher peak in the curve of constant $\beta$ for each successively higher value of $\beta$, due to effects of higher orders of $\lambda$, which we have neglected; and the power-input chart would show somewhat lower ordinates in the region $\lambda \to 0$, due to swirl effects, which we have neglected.)

2. The theoretical method of analysis we have used is not suitable for serious design application. Far more accurate (but, necessarily, far more complex) methods are readily available in innumerable textbooks, reports, etc. The value of the derivation presented here is only that it leads to predictions of the compressor characteristics which are qualitatively correct and which are comparatively very easy to understand.

## DUCT LOSSES

Now that we have settled on a usable description of the compressor characteristics, the next step is to examine how these characteristics must be matched to the internal flow characteristics of the vehicle as a whole.

Consider a schematic diagram of the whole flow system:

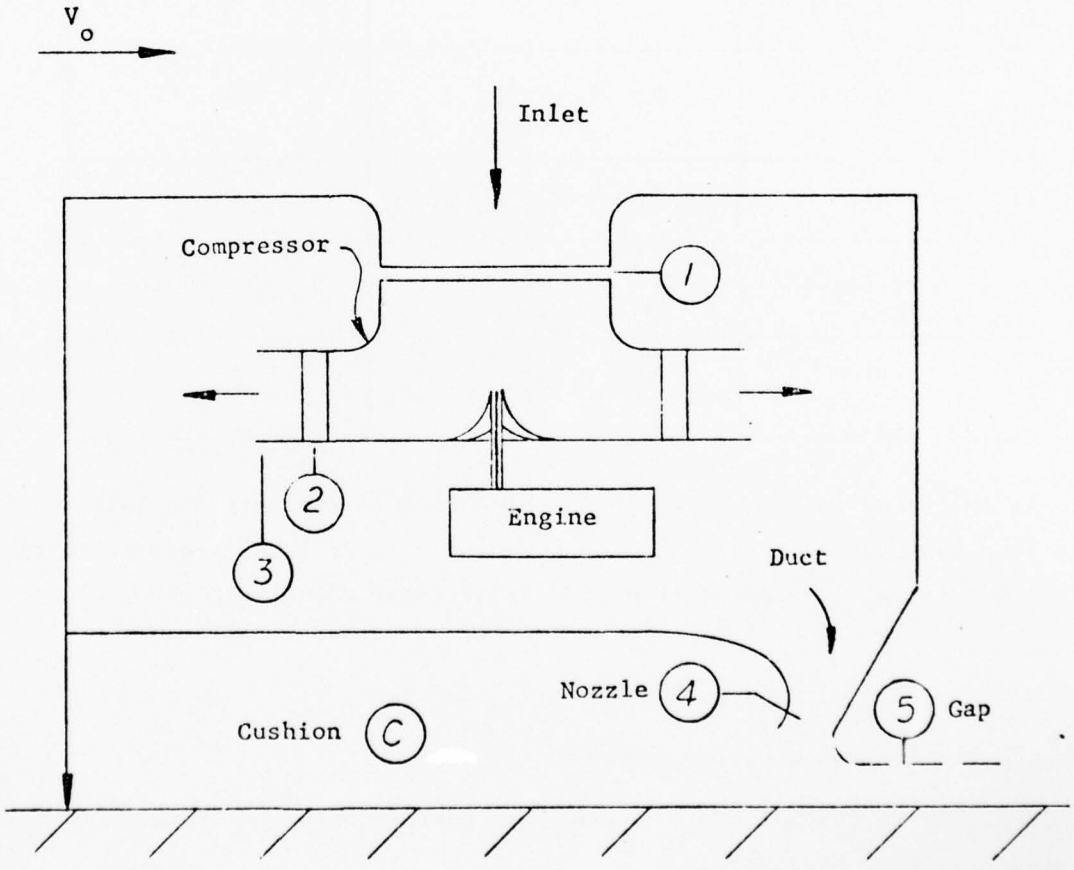


Table C-1

Total Gage Pressure and Volume Flow Rate at Several Stations

| Station | Description | Total Gage Pressure | Volume Flow Rate |
|---|---|---|---|
| (0) | Free Stream | $q_o$ | – |
| (1) | Compressor Inlet Face | $q_o - H_{\ell,i} \equiv \eta_r \, q_o - H_{\ell,i_o}$ | Q |
| (2) | Blade Row | – | Q |
| (3) | Compressor Exit | $q_o - H_{\ell,i} + \Delta p_t$ | Q |
| (4) | Nozzle | $q_o - H_{\ell,i} + \Delta p_t - H_{\ell,d} \equiv p_t$ | Q |
| ( ) | "Daylight" Gap | – | $Q = \mathcal{O}_c \sqrt{\frac{2}{\rho} \Delta p} \, S_g$ |
| (C) | Cushion | $\Delta p = p_t \left( \frac{\Delta p}{p_t} \right)$ | – |

As indicated in the table, it is convenient to separate the inlet loss into two parts: the loss which occurs with zero free-stream velocity, and an additional loss associated with free-stream velocity:

$$H_{\ell,i} = H_{\ell,i_o} + \left(1 - \eta_r\right) q_o$$

where $\eta_r$ is the "ram recovery efficiency."

Next, it is convenient to define loss coefficients, in terms of reference dynamic pressures:

$$q_r \equiv \tfrac{1}{2} \rho \left( \frac{Q}{S_g} \right)^2 = \mathcal{O}_c^2 \, \Delta p$$

$$q_1 \equiv \tfrac{1}{2} \rho \left( \frac{Q}{S_1} \right)^2$$

$$q_3 \equiv \tfrac{1}{2}\rho\left(\frac{Q}{S_3}\right)^2$$

$$K_{\ell,i} \equiv \frac{H_{\ell,i_o}}{q_1} = \frac{H_{\ell,i_o}}{q_r}\left(\frac{S_1}{S_g}\right)^2$$

$$K_{\ell,d} \equiv \frac{H_{\ell,d}}{q_3} = \frac{H_{\ell,d}}{q_r}\left(\frac{S_3}{S_g}\right)^2$$

With these definitions, we can write

$$\begin{aligned}\Delta p_t &= p_t + H_{\ell,i_o} + H_{\ell,d} - \eta_r q_o \\ &= p_t + q_r\left[K_{\ell,i}\left(\frac{S_g}{S_1}\right)^2 + K_{\ell,d}\left(\frac{S_g}{S_3}\right)^2\right] - \eta_r q_o \\ &= p_t\left\{1 + \frac{\Delta p}{p_t}\mathscr{L}_c^2\left[K_{\ell,i}\left(\frac{S_g}{S_1}\right)^2 + K_{\ell,d}\left(\frac{S_g}{S_3}\right)^2\right]\right\} - \eta_r q_o\end{aligned}$$

But we have previously defined:

$$\Delta p_t = p_t \div \eta_d - \eta_r q_o$$

Therefore,

$$\eta_d = \frac{1}{1 + \frac{\Delta p}{p_t}\mathscr{L}_c^2\left[K_{\ell,i}\left(\frac{S_g}{S_1}\right)^2 + K_{\ell,d}\left(\frac{S_g}{S_3}\right)^2\right]} \qquad [C\text{-}4]$$

Example:

Consider a vehicle equipped with N identical compressors of the type shown in Figure C-1; that is,

$$S_1 = 0.785\ R^2\ N$$

$$S_s = 1.7\ R^2\ N$$

$$\eta_d = \left[1 + \frac{\Delta p}{p_t}\ \mathscr{C}_c^{\,2}\left(\frac{S_g}{S_s}\right)^2\left(K_{\ell,d} + 4.7\ K_{\ell,i}\right)\right]^{-1}$$

Further, assume $K_{\ell,i} \equiv 0.02$ and $K_{\ell,d} = 1.0$. (These assumptions are reasonable for a typical peripheral-jet design with ducting which consists of nothing more elaborate than a large unobstructed "plenum" through which the air flows to reach the nozzles; and also, for a plenum-type GEM in which air is delivered more or less directly from the compressor to the cushion. The assumption that $K_{\ell,d} = 1$ is approximately equivalent to assuming that all of the kinetic energy in the compressor efflux is lost through viscous action.) Then,

$$\eta_d = \frac{1}{1 + \frac{\Delta p}{p_t}\ \mathscr{C}_c^{\,2}\left[0.38\left(\frac{S_g}{R^2\ N}\right)^2\right]}$$

It is possible to understand, from this formula, one of the fundamental facts about GEM's (which, by the way, has led to the downfall of many an amateur builder): <u>In moderate-sized GEM's with significant daylight clearance, the compressors require a very great deal of space</u>.

Let us add some more specifics to our example: Suppose $\frac{\Delta p}{p_t} = 0.8$, $\mathscr{C}_c = 0.5$, the average daylight clearance (in feet) is h, the cushion is rectangular with length/beam ratio $\ell/b = 2$, and there is only a single compressor ($N = 1$).

Then we can write

$$\eta_d = \left[1 + 2.74\ \frac{(bh)^2}{R^4}\right]^{-1}$$

$$R = 1.285\ \sqrt{bh}\left(\frac{\eta_d}{1 - \eta_d}\right)^{1/4}$$

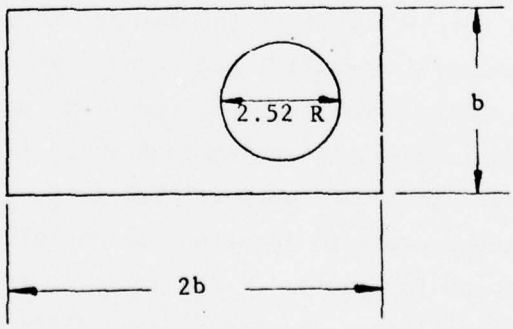
Tabulating from this formula, approximately, for h = 0.5 foot, the maximum diameter of the compressor (2.52 R) is given for various combinations of beam and duct efficiency $\eta_d$.

Table C-2

Compressor Maximum Diameter Required To Achieve Given Values of Duct Efficiency for Various Sizes of Single-Compressor Designs

$\left[h = 0.5 \text{ foot}; \frac{\Delta p}{p_t} = 0.8; \mathcal{M}_c = 0.5; \frac{\ell}{b} = 2; K_{\ell,i} = 0.02; K_{\ell,d} = 1.0\right]$

| b, in feet | Maximum Diameter (2.52 R) in feet, for: | | | |
|---|---|---|---|---|
| | $\eta_d = 0.9$ | $\eta_d = 0.8$ | $\eta_d = 0.7$ | $\eta_d = 0.5$ |
| 10 | 12.6 | 10.2 | 9.0 | 7.3 |
| 20 | 17.8 | 14.5 | 12.7 | 10.2 |
| 40 | 25.2 | 20.5 | 18.0 | 14.5 |
| 80 | 35.7 | 29.0 | 25.5 | 20.6 |
| 160 | 50.4 | 41.2 | 36.2 | 29.1 |




Note: Compressor of type illustrated in Figure C-1

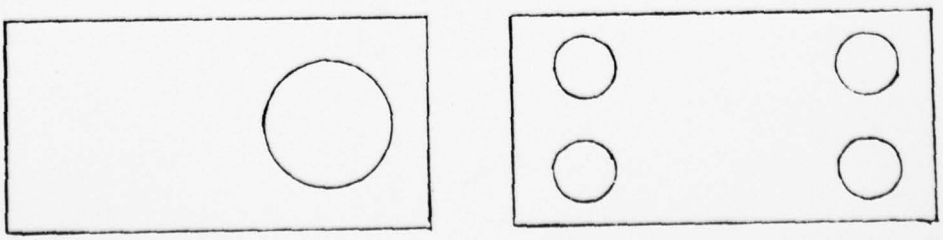
It will be very much worth while to spend a few minutes contemplating this table and the development which led up to it. Of the various assumptions which were used in constructing Table C-2, only two can be manipulated over a sufficiently wide range, in a practical design, to really strongly influence the result. These are the assumptions that $h = 0.5$ foot and $K_{\ell,d} = 1.0$.

The choice of daylight clearance h is governed primarily by considerations of tradeoff between cushion power and propulsion power (to be discussed in the succeeding sections); but obviously one cannot afford to ignore the consideration of space occupied by the compressor. Observe that the compressor dimensions vary in proportion to $\sqrt{h}$. That is, if h is increased from 0.5 foot to 2.0 feet, the compressor diameters in Table C-2 must be doubled; and if h is decreased from 0.5 foot to 0.125 foot, the diameters can be halved.

Similarly, the compressor dimensions vary approximately in proportion to $\left(K_{\ell,d} + 0.09\right)^{1/4}$. For example, if the duct loss coefficient $K_{\ell,d}$ could be reduced from 1.0 to 0.23, the compressor diameters could be reduced to three-fourths of the values listed in Table C-2. This is not, by any means, impossible; but it nearly always turns out to be impractical in actual designs. It requires smooth ducts, with no sharp bends or sudden changes in stream-tube area, leading from the compressors to the nozzles. The difficulty is that to do this usually requires placing the compressors in locations which interfere with efficient utilization of space for payload. For example, for a single-compressor design, highly efficient ducting is very difficult unless the compressor is placed near the center of the planform. It is sometimes said that the need for efficient ducting can be avoided by the use of an efficient diffuser to decelerate the compressor efflux, but this is deceptive. When one considers the total space occupied by the compressor and diffuser combined, it usually turns out to be more economical of space to use a larger compressor with no diffuser. (The same conclusion does not necessarily apply to considerations of weight and economy.)

Of course, each design presents a slightly different problem, and one should not be too hasty in assuming that sophisticated ducting and/or diffusers will not be worth while. The point is that it is far more dangerous to naïvely assume that a value of $K_{\ell,d}$ substantially less than unity will be achieved, unless and until one has very carefully determined precisely how this will be achieved and what it will cost. In fact, the careless designer may well end up with a value of $K_{\ell,d}$ much greater than unity, if he fails to assure that there is adequate space in his ducting for the stream tubes to reach their destination without "squeezing through" constricted passages before reaching the nozzle (in the case of peripheral-jet types) or the air gap (in the case of plenum types).

Finally, let us reflect for a moment on the implications of Table C-2, if, instead of a single large compressor, one uses a number of smaller compressors. In fact, the designer usually will choose a multiple-compressor design, except in cases of very small vehicles, for a variety of reasons: weight and balance, reliability, economy, convenience of fabrication and maintenance, power transmission problems, engine-compressor matching, etc. However, use of multiple compressors does not generally alleviate the problem of space occupied by the compressors; just the contrary, in fact. For example, if four compressors were used, the diameter of each would be half the value listed in Table C-2. Taking into account that these four half-size compressors must be spaced far enough apart to avoid excessive interference between their respective effluxes, it is easy to see that they will generally entail more compromise of the useful space aboard the vehicle than one full-size compressor would have done.



The following conclusions are indicated:

1. The duct efficiency depends on a number of design details, but is by far most strongly dependent on the ratio of "daylight" gap area to compressor exit area, $S_g/S_3$ .

2. The proportionate planform space occupied by compressors (if good duct efficiency is maintained) is thus primarily dependent on the ratio $S_g/S$ . Roughly speaking, this space problem tends to become oppressive when $S_g/S > 0.05$, say, unless unusual measures are taken to avoid duct losses.

## OVER-ALL INTERNAL EFFICIENCY

Equation [C-4] expresses the duct efficiency in terms of relationships between the dimensions of the compressor and the dimensions of the vehicle as a whole. It will now be instructive to examine the over-all internal efficiency.

$$\eta_{int} \equiv \eta_d \, \eta_c$$

We have an approximate expression for $\eta_c$ (Equation [C-3]) for a compressor of the type shown in Figure C-1. It will be convenient to re-express Equation [C-4] in terms of the compressor performance parameters. This can be approached directly, but it is considerably easier to start over from the beginning.

Let us write

$$p_t = \Delta p_t - H_{\ell,i_o} - H_{\ell,d} + \eta_r \, q_o$$

$$= \Delta p_t - K_{\ell,i} \, q_1 - K_{\ell,d} \, q_3 + \eta_r \, q_o$$

but

$$q_3 = \lambda^2 \, q_b \left( \frac{S_2}{S_3} \right)^2$$

$$q_1 = \lambda^2 \, q_b \left( \frac{S_2}{S_1} \right)^2$$

We can now write

$$\eta_d \equiv \frac{p_t}{\Delta p_t + \eta_r q_o}$$

$$\doteq 1 - \frac{\lambda^2}{\Delta p_t / q_b} \left[ K_{\ell,i} \left( \frac{S_2}{S_1} \right)^2 + K_{\ell,d} \left( \frac{S_2}{S_3} \right)^2 \right] \left( \frac{1}{1 + \frac{\eta_r q_o}{\Delta p_t}} \right) \quad [C\text{-}4']$$

or, for the compressor of Figure C-1, assuming $\frac{\eta_r q_o}{\Delta p_t} \ll 1$

$$\eta_d \doteq 1 - \frac{\lambda^2}{\Delta p_t / q_b} \left[ 2.96 \, K_{\ell,i} + 0.63 \, K_{\ell,d} \right]$$

Let us assume, as before, that

$$K_{\ell,i} = 0.02$$

$$K_{\ell,d} = 1.0$$

for a simple duct system, giving

$$\eta_d \doteq 1 - 0.689 \frac{\lambda^2}{\Delta p_t / q_b}$$

which allows us to assign a value of $\eta_d$ immediately to each point of the pressure-rise chart (Figure C-3) and to plot the product $\eta_d \eta_c = \eta_{int}$, instead of $\eta_c$, on the efficiency chart.

Next, we can convert the pressure-rise chart of Figure C-3 to a cushion-pressure chart, and thus obtain a system characteristics chart for the whole vehicle, analogous to the compressor characteristics chart for the compressor alone. This conversion proceeds as follows. Note that

$$\lambda = \frac{V_n}{V_b} = \frac{\mathcal{C}_c \sqrt{\frac{2}{\rho} \Delta p} \; S_g}{V_b \; S_a}$$

$$= \frac{\mathcal{C}_c \; S_g}{S_a} \sqrt{\frac{\Delta p}{q_b}}$$

or

$$\left( \frac{\Delta p}{q_b} \right)_{\text{demand}} = \lambda^2 \div \left( \frac{\mathcal{C}_c \; S_g}{S_a} \right)^2$$

where the subscript "demand" denotes that this relationship is derived from the fact that a certain cushion pressure $\Delta p$ acting across gap area $S_g$ with discharge coefficient $\mathcal{C}_c$ demands a corresponding value of $\lambda \; V_b \; S_a$, from continuity considerations. We need also an "availability" relationship, giving the cushion pressure which the compressor can sustain. This is easily obtained as follows

$$\frac{\Delta p}{q_b} = \frac{\Delta p}{p_t} \frac{p_t}{q_b} = \frac{\Delta p}{p_t} \frac{\Delta p_t}{q_b} \frac{p_t}{\Delta p_t}$$

or

$$\left( \frac{\Delta p}{q_b} \right)_{\text{available}} = \frac{\Delta p}{p_t} \frac{\Delta p_t}{q_b} \eta_d \left( 1 + \frac{\eta_r \; q_o / q_b}{\Delta p_t / q_b} \right)$$

"Internal System Characteristics Charts" constructed from these relationships are presented in Figure C-4. These charts provide fairly complete information on the internal system performance, and interrelationships between the main design parameters, subject to the simplifying assumptions which have been introduced.

The following points are worth noting, in regard to Figure C-4:

1. *Within the range considered, most aspects of internal performance improve with increasing solidity, σ.* We have, therefore, not established an "optimum" solidity. We may surmise, however, that the

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AD A038709



performance shown for $\sigma = 0.6$ is not significantly different from optimum. Higher solidities were not investigated because a more complex compressor theory would have been required.

2. For $\sigma = 0.6$, there is clearly an "optimum" combination of blade angle $\beta$ and advance ratio $\lambda$, from the standpoint of providing maximum internal efficiency. Figure C-4 does not include enough values of $\beta$ to define this optimum combination precisely, but it is near $\beta = 20°$, $\lambda = 0.22$.

3. One usually should not choose this optimum combination as the design point, because:

(a) The optimum point lies uncomfortably close to the stall point, and intermittent stall might be experienced in unsteady operating conditions. (Roughly speaking, for a given $\beta$ the compressor is stalled at all values of $\lambda$ lower than the peak of the $\frac{\Delta p}{q_b}$ curve, and is said to be "full stalled" when $\lambda = 0$.)

(b) If the design point corresponds too closely to the maximum pressure output of the compressor, the full-stalled pressure output might be inadequate for take-off.

(c) This choice of the design point might preclude operating the lift system at less than full power, or preclude flight at gross weight greater than the design value (most desirable capabilities, because they provide extended range and/or payload in unusually favorable operating conditions). (Note, in this regard, the tremendous advantage which a variable-$\beta$ compressor would enjoy! Review, also, Figure C-3 from this viewpoint.)

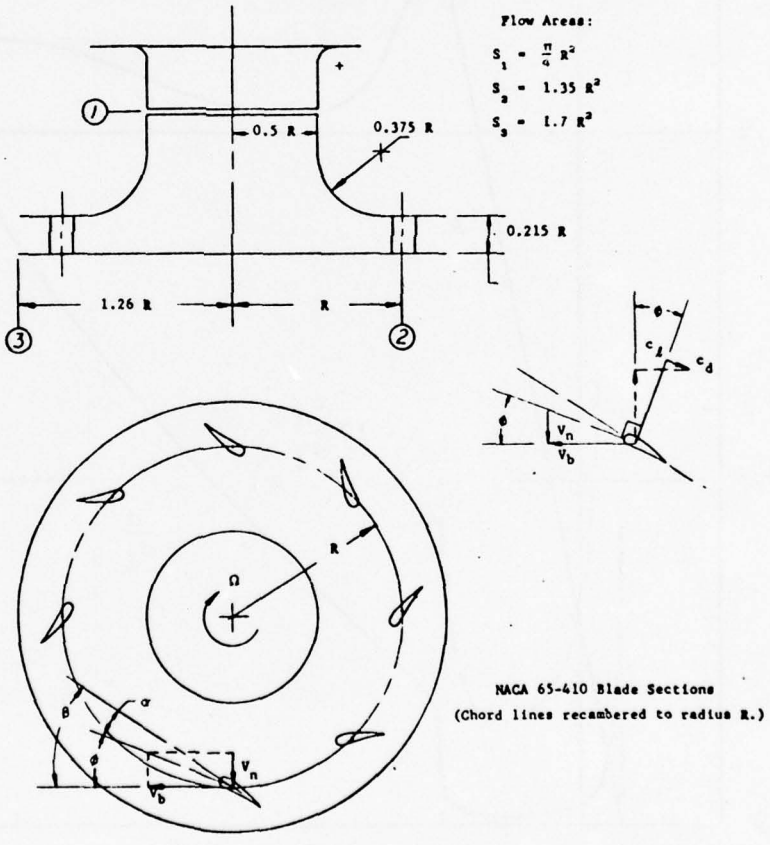
(d) Careful study of the relationships illustrated in the Cushion Pressure Chart of Figure C-4 will reveal that choice of higher-than-optimum values of both $\beta$ and $\lambda$ permits use of smaller compressors, with savings in weight and space. The tradeoff of internal efficiency versus compressor weight and size is an extremely important one. We will not attempt to formulate this tradeoff quantitatively herein, because the quantitative benefit of a given weight or size reduction is very difficult to determine, and it varies greatly from one application to another. This tradeoff must be considered with utmost care, however, in actual design efforts.

At the risk of becoming monotonous, a final reminder must be given of the assumptions and simplifications which led up to Figure C-4:

1. Only one type of compressor (Figure C-1) was considered, and only limited ranges of its allowed variables ($\beta$, $\lambda$, and $\sigma$) were considered. A somewhat more complete picture could be obtained by considering a broader range of variables, but this would require introduction of more complex theory. In defense of the presentation given, it may be pointed out that the system characteristics derived herein are qualitatively representative of most practical systems. In fact, the simple theory for axial compressors is exactly analogous to that given for the airfoil-type radial flow compressor, the main first-order difference being a different numerical relationship between the areas $S_1$, $S_2$, and $S_3$; and the centrifugal compressor is simply an extremely high-solidity version of the compressor considered here.

2. Only one of the infinite number of possible combinations of duct loss coefficients is reflected in Figure C-4, namely $K_{\ell,i} = 0.02$, $K_{\ell,d} = 1.0$. This combination is quite representative of current vehicles, but it must be remembered that current vehicles represent a relatively primitive state-of-the-art. (Hovercraft were first seriously considered around 1957!) In a previous discussion, the merits of more sophisticated ducting were somewhat discounted, from the standpoint of alleviating the space problem imposed by large compressors. However, the same arguments do not necessarily apply to questions of compressor weight and economy. The relationships given in this section permit the reader, with fairly nominal effort, to construct other "system characteristics charts," similar to Figure C-4, but with different assumptions regarding duct loss coefficients; and, if he wishes, he could use manufacturers' compressor characteristics in place of the illustrative characteristics, derived herein. An exercise of this kind is definitely recommended for readers with serious design intentions.

Aerodynamics Laboratory
David Taylor Model Basin
Washington, D. C.
April 1966




NACA 65-410 Blade Sections
(Chord lines recambered to radius R.)

Figure C-1 - Illustrative Radial-Flow Compressor

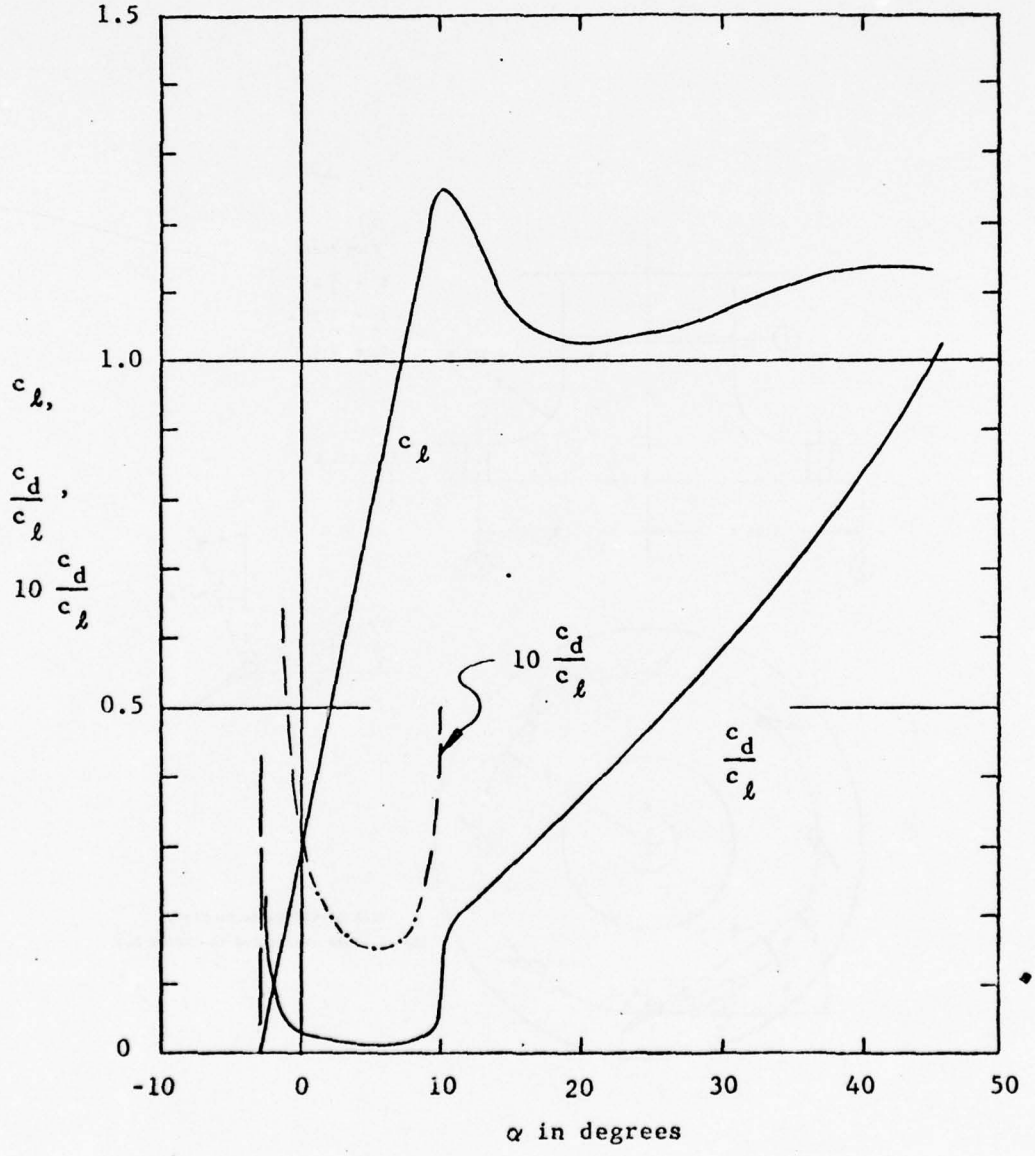
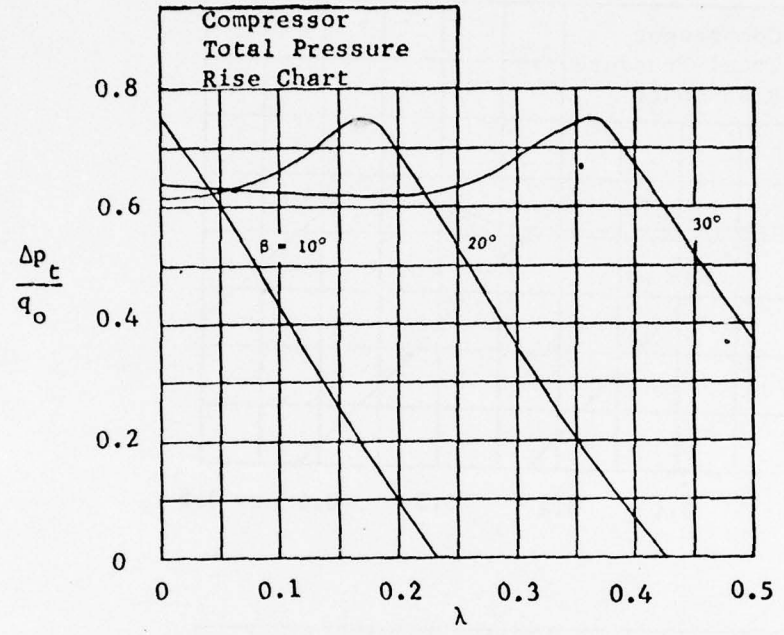



Figure C-2 - Assumed Characteristics of NACA 65-410 Profile



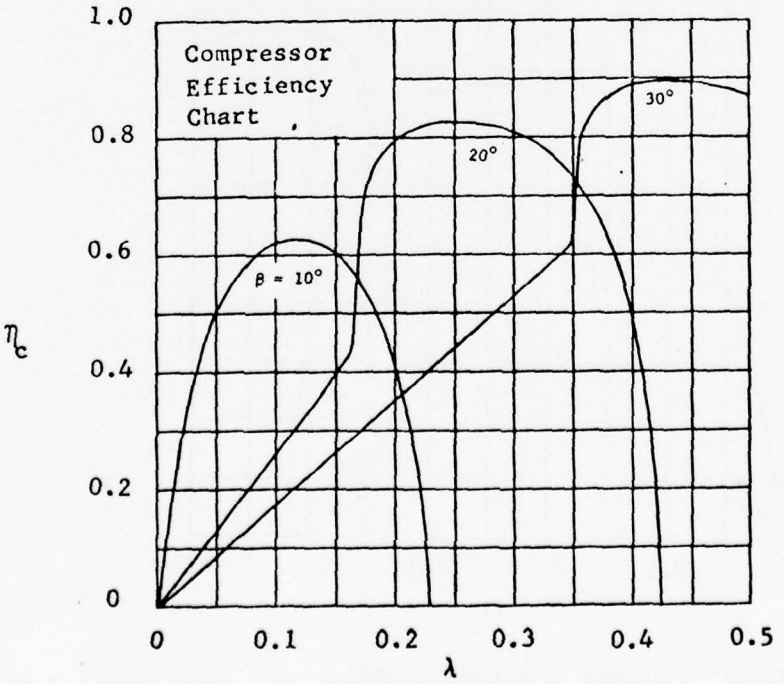
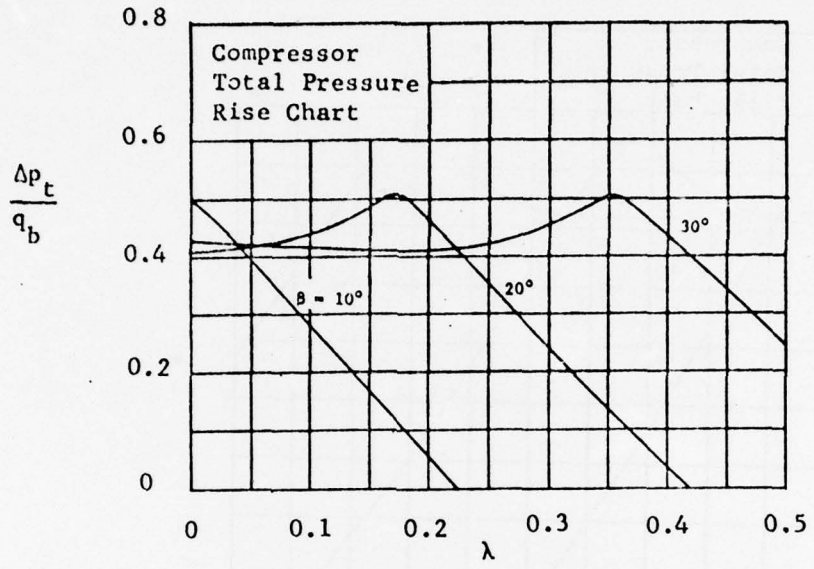




Figure C-3 - Compressor Characteristics Charts

[Charts based on compressor illustrated in Figure C-1.]

(a) $\frac{\Delta p_t}{q_b}$ and $\eta_c$ versus $\lambda$ for $\sigma = 0.6$



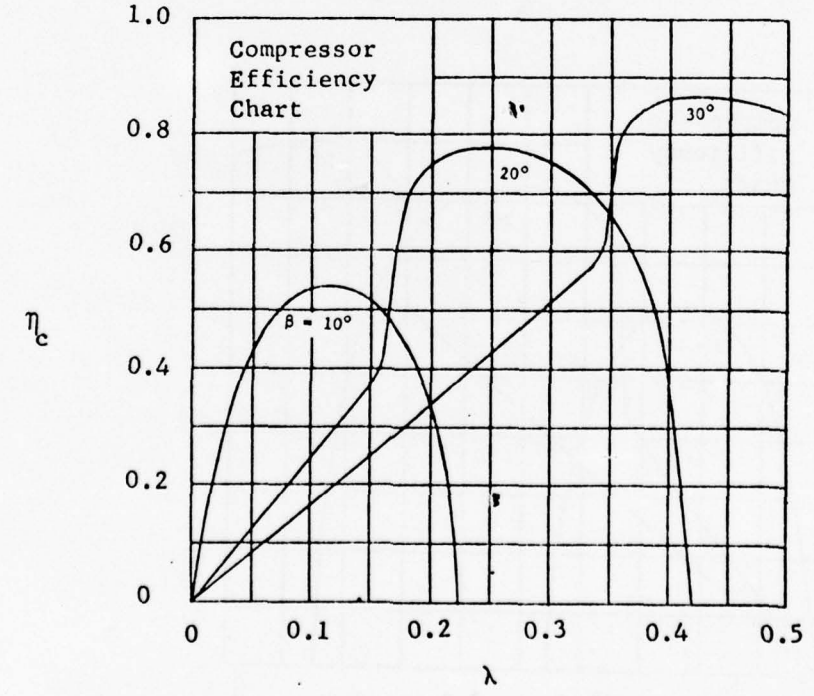
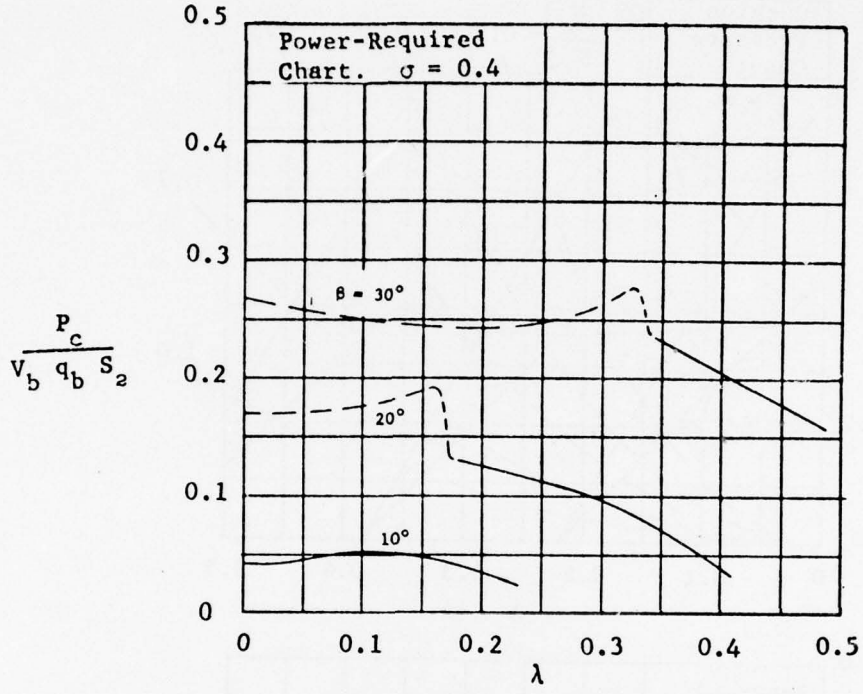
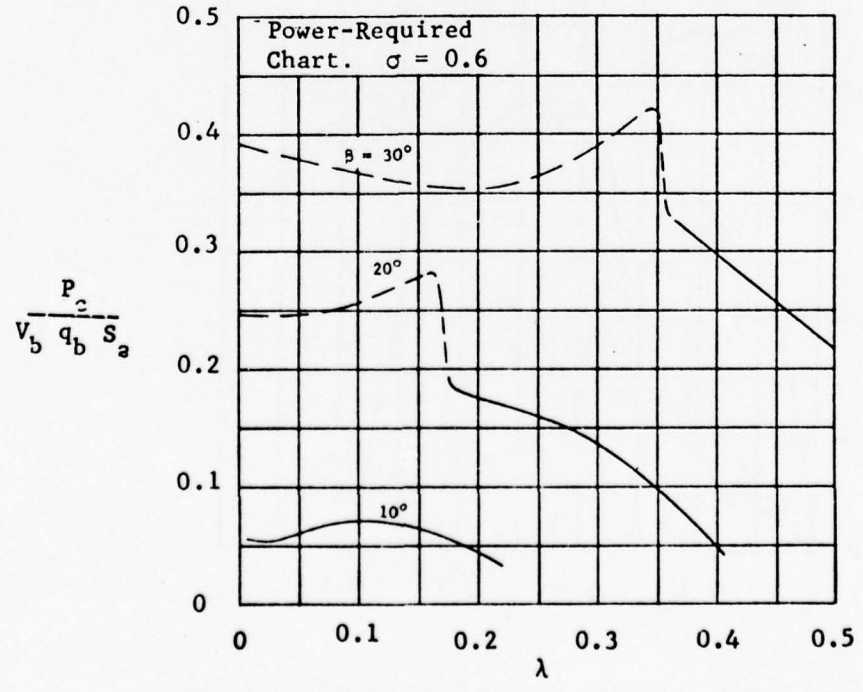




Figure C-3 (Continued)

(b) $\frac{\Delta p_t}{q_b}$ and $\eta_c$ Versus $\lambda$ for $\sigma = 0.4$



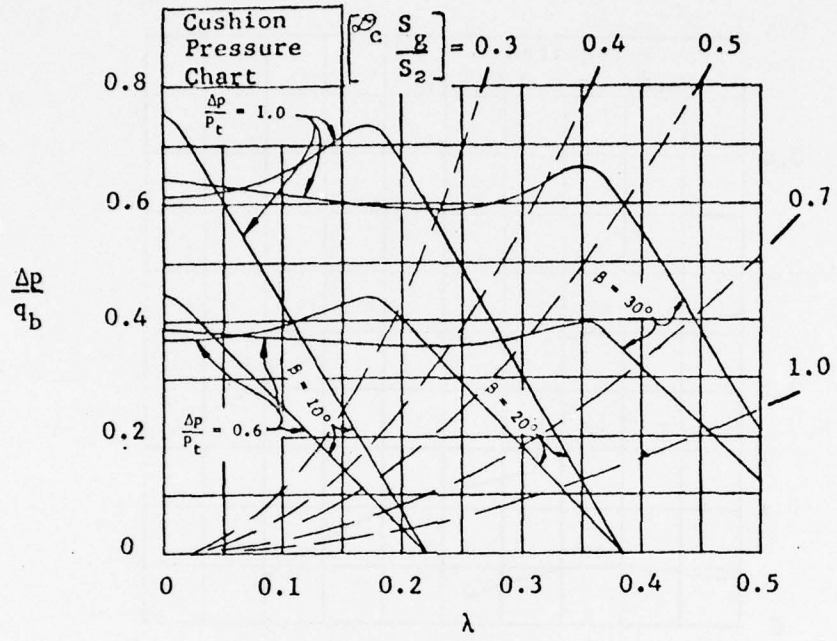

$$\frac{P_c}{V_b\ q_b\ S_a}$$




$$\frac{P_c}{V_b\ q_b\ S_a}$$

Figure C-3 (Concluded)

(c) Power-Required Parameter for σ = 0.4 and 0.6



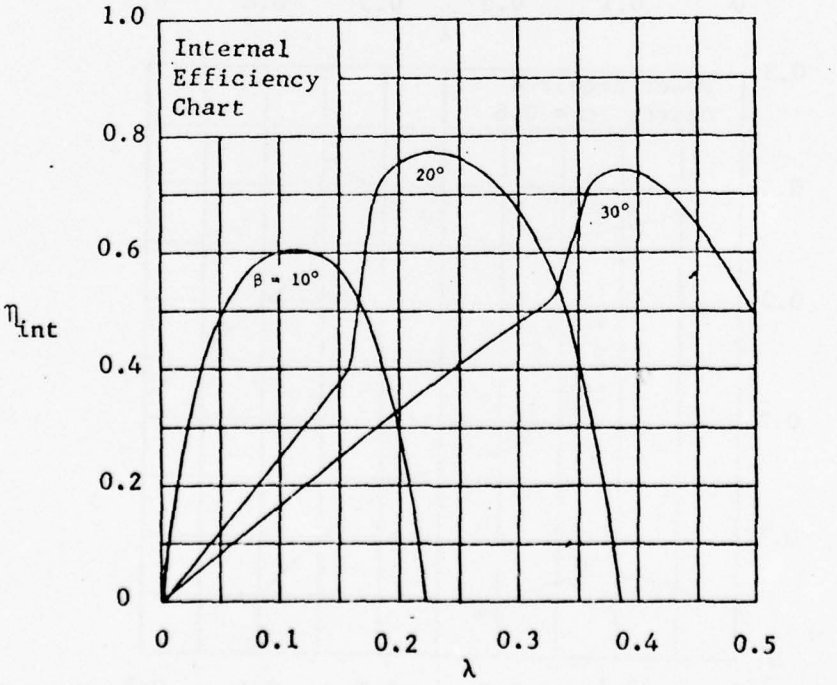
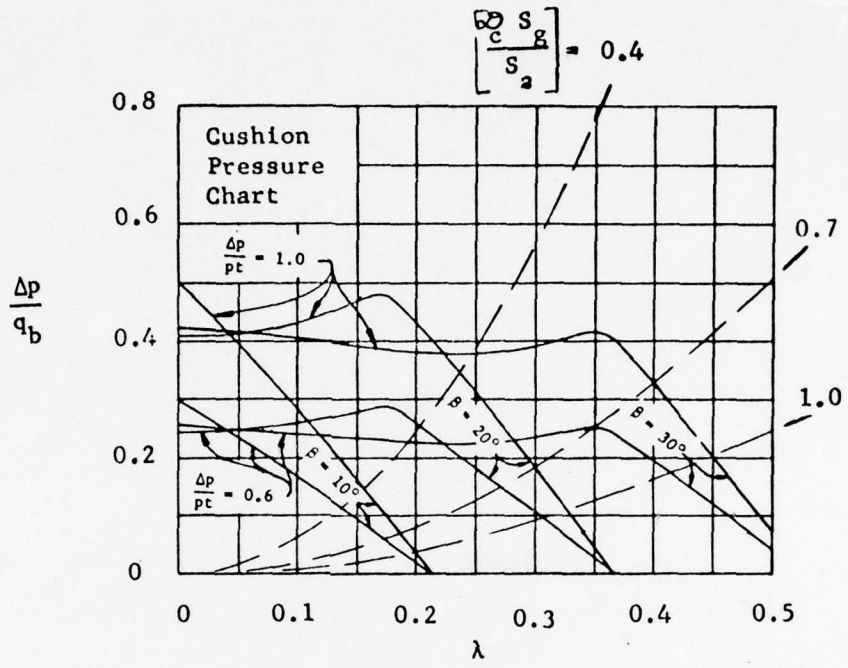




Figure C-4 - Internal System Characteristics Charts

[Charts based on compressor illustrated in Figure C-1; $K_{\ell,i} = 0.02$; $K_{\ell,d} = 1.0$; $\eta_r\, q_o \ll \Delta p_t$ .]

(a) $\sigma = 0.6$



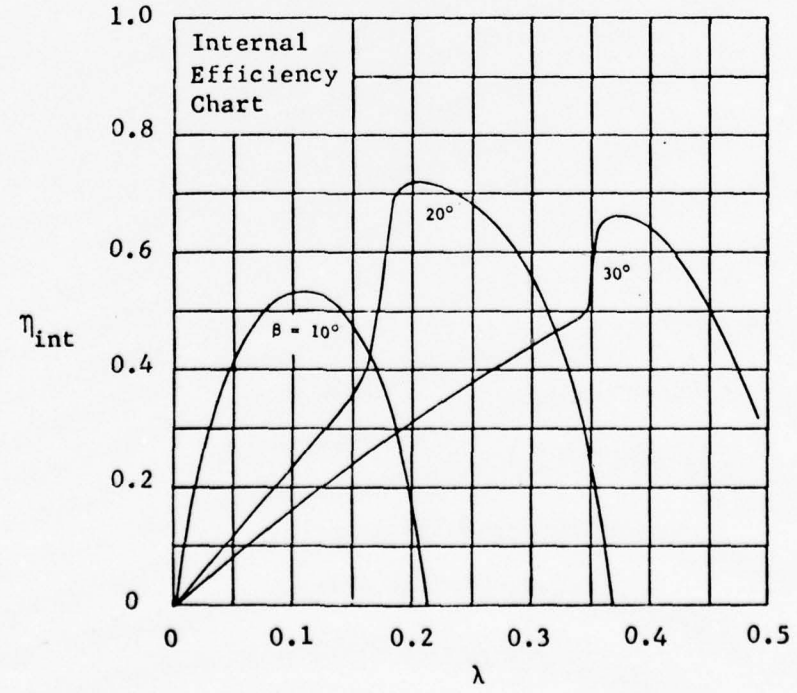




Figure C-4 (Concluded)

(b) $\sigma = 0.4$

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SECTION D — DRAG

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13 ABSTRACT

Six vehicle drag terms are derived. These apply to the sidewall captured air bubble air cushion vehicle and the full-peripheral air cushion vehicle.

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| 14 KEY WORDS | LINK A | | LINK B | | LINK C | |
|---|---|---|---|---|---|---|
| | ROLE | WT | ROLE | WT | ROLE | WT |
| Air Cushion Vehicles | | | | | | |
| Full-Peripheral ACV | | | | | | |
| Drag | | | | | | |
| Wavemaking Drag | | | | | | |
| Hydrodynamic Drag | | | | | | |
| External Aerodynamic Drag | | | | | | |
| Ram Drag | | | | | | |
| Captured Air Bubble | | | | | | |
| Sidewalls | | | | | | |
| Trunks | | | | | | |

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# SOME DESIGN PRINCIPLES OF GROUND EFFECT MACHINES
## SECTION D — DRAG

by

Harvey R. Chaplin and Allen G. Ford

APPROVED FOR PUBLIC RELEASE: DISTRIBUTION UNLIMITED

June 1966

Report 2121 D
Aero Report 1100 D

## Foreword

This report is based on a lecture series presented by the authors at the von Kármán Institute for Fluid Dynamics, Rhode-Saint-Genèse, Belgium, in May 1965; and at the University of Maryland, College Park, Maryland, in July 1965. The lectures were prepared under the joint auspices of the David Taylor Model Basin and the Naval Air Development Center. They were presented in Belgium under the joint sponsorship of the von Kármán Institute and the Advisory Group for Aerospace Research and Development (AGARD); and in Maryland under the sponsorship of the Assistant Secretary of the Navy for Research and Development.

The revised lectures will be presented as follows:

A. Introductory Survey
B. Air Cushion Mechanics
C. Internal Aerodynamics
D. Drag
E. ~~Drag Optimization for Sidewall CEM (CAB)~~
F. Cushion Contributions to Stability
G. Seakeeping
H. Performance Summary

## NOTATION

| | |
|---|---|
| $a$ | wave amplitude, ft. (Half of the trough to crest height.) |
| $b$ | beam of pressure region or bubble, ft |
| $C_{D_e}$ | external aerodynamic drag coefficient (based on cushion area, S) |
| $C_{D_t}$ | trunk drag coefficient (trunk drag/$q_a S$) |
| $C_f$ | skin friction coefficient |
| $C_L$ | lift coefficient (lift/$q_a S$) |
| $D_e$ | external aerodynamic drag, lb |
| $D_r$ | ram or momentum drag, lb |
| $D_{s,a}$ | additional sidewall drag, lb |
| $D_{s,w}$ | secondary wavemaking drag (drag due to retaining air bubble), lb |
| $D_t$ | trunk (or skirt) drag, lb |
| $D_T$ | total drag, lb. (The appropriate sum of the component drags.) |
| $D_w$ | wavemaking drag, lb |
| $\rho_c$ | nozzle discharge coefficient of ACV base |
| $F_\ell$ | length Froude number ($V/\sqrt{g\ell}$) (nondimensional) |
| $F_{\sqrt{S}}$ | area Froude number ($V/\sqrt{g\sqrt{S}}$) (nondimensional) |
| $f_\ell$ | wave drag parameter $\left(f_\ell \equiv \frac{D_w/L}{\frac{4}{\rho_w g}\frac{p}{\ell}}\right)$ (see Figure D-2) |
| $f_{\sqrt{S}}$ | wave drag parameter (see Figure D-3) |
| $G$ | geometry factor ($G = 1$ for full-peripheral ACV; $G < 1$ for CAB ACV) |

| | |
|---|---|
| $H$ | average wave height, trough to crest, ft |
| $h$ | height or daylight clearance, ft |
| $h_a$ | additional sidewall depth, ft (see Figure D-5) |
| $h_b$ | maximum wave depression under craft, ft (see Figures D-5 and D-6) |
| $L$ | lift, lb ($L = W$) |
| $\ell$, $\ell_B$ | length of pressure region (or bubble), ft |
| $\Delta\ell$ | short length, ft |
| $\ell_s$ | sidewall wetted length, ft |
| $\ell'$ | shortened wetted length of ACV with daylight clearance, ft (see Figure D-6) |
| $n$ | number of sidewall wetted sides |
| $P_p$ | propulsive power, lb-ft/sec |
| $p$ | pressure, $lb/ft^2$ |
| $q_a$ | dynamic air pressure ($\frac{1}{2} \rho_a V^2$), $lb/ft^2$ |
| $q_w$ | dynamic water pressure ($\frac{1}{2} \rho_w V^2$), $lb/ft^2$ |
| $R$ | resultant lift force, lb |
| $S$ | cushion base area, $ft^2$ |
| $S_g$ | area of discharge (daylight clearance area), $ft^2$ |
| $V$ | velocity, ft/sec |
| $V_j$ | jet velocity of air discharge from ACV base, when fully contracted, ft/sec |
| $W$ | vehicle weight, lb |
| $w$ | specific weight of ACV $\left(\frac{W}{S}\right)$, $lb/ft^2$ |

| | |
|---|---|
| $\alpha$ | vehicle pitch angle or angle of attack, deg |
| $\eta_p$ | propulsive efficiency |
| $\lambda$ | wave length, ft |
| $\rho_a$ | mass density of air, slugs/ft$^3$ |
| $\rho_w$ | mass density of water, slugs/ft$^3$ |
| $\gamma$ | weight density of water ($\rho_w$ g), lb/ft$^3$ |

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## SUMMARY

Six vehicle drag terms are derived. These apply to the sidewall captured air bubble air cushion vehicle and the full-peripheral air cushion vehicle.


## INTRODUCTION

The subject of the drag of air cushion vehicles is broken up into six parts. The various components of drag, which apply to a skirted full-pheripheral air cushion vehicle (ACV) and to a captured air bubble (CAB) type of sidewall vehicle with only slight modification, are as follows: wave drag, $D_w$ ; a secondary wavemaking drag associated with the drag of that amount of sidewall required to retain the bubble (or cushion), $D_{s,w}$ ; additional sidewall drag, $D_{s,a}$ ; the trunk (or skirt) drag of the full-peripheral ACV, $D_t$ (a geometry factor of less than one is used for the fore and aft ski drag in the CAB type craft); the external aerodynamic drag (air profile drag made up of form drag and skin friction drag), $D_e$ ; and the ACV ram or momentum drag, $D_r$ . The total drag, $D_T$ , is an appropriate sum of the drags. All the drag terms will ultimately be referred to vehicle weight, W (which equals the vehicle lift, L), to give the drag/weight (or drag/lift) ratio.

The propulsive power $P_p$ is the product of the total drag $D_T$ , the vehicle velocity V, and the reciprocal of the propulsive efficiency, $\eta_p$ . The specific propulsive power (nondimensional) is given by:

$$\frac{P_p}{WV} = \frac{D_T}{W} \div \eta_p$$

## WAVEMAKING DRAG

A region of pressure, proceeding at some speed V over a body of otherwise calm water, generates waves. If the pressurized region is considered to be at rest, and the otherwise calm water has a velocity of -V, then the wave pattern generated is fixed in space.

Our interest in this wave is that an ACV pressure region generates such a wave, and further that the pressure region ultimately acts back on the vehicle to give a resultant air pressure force tilting back somewhat from the vertical. This tilted force has, in addition to a lift component, a drag component, which will be called a wavemaking drag, $D_w$ .

The simplest wavemaking case is that of a two-dimensional (infinite beam and infinite aspect ratio) short pressurized region of length $\Delta\ell$, pressure p, and velocity V. This was treated by Lamb (Reference 1, pages 402-403) and by Wagner (Reference 2) and others. Lamb notes that at a distance of about half a wavelength from the origin, on the downstream side, a simple-harmonic wave profile is fully established. The amplitude "$\Delta a$" is given by

$$\Delta a = \frac{2p\Delta\ell}{\rho_w V^2} \qquad [D\text{-}1]$$

Note that the amplitude is proportional to $p\Delta\ell$, and to the reciprocal of the square of velocity. The wave length $\lambda$ is proportional to the square of velocity as follows:

$$\lambda = \frac{2\pi V^2}{g} \qquad [D\text{-}2]$$

The wavemaking drag/lift ratio is proportional to $\Delta a/\lambda$, and is given by

$$\frac{D_w}{L} = \frac{p\, g\, \Delta\ell}{\rho_w V^4} \qquad [D\text{-}3]$$

Hence, the wavemaking drag is

$$D_w = \frac{g\, b\, (p\, \Delta\ell)^2}{\rho_w V^4} \qquad [D\text{-}3a]$$

Note that the wavemaking drag $D_w$ is proportional to the square of the pressure p, and is inversely proportional to the fourth power of velocity. On the logarithmic plots (Figure D-2, for example), this fourth-power relationship can be easily verified for the two-dimensional case.

The next simplest wavemaking case is that of a two-dimensional (infinite beam and infinite aspect ratio) uniform pressurized region of finite length $\ell$. Lamb (Reference 1, p 404) indicates a method for integrating the previous results, taking into account not only amplitudes, but also phases of the component waves generated by different and separated $\Delta\ell$ regions. Lamb says that "we can easily deduce the requisite formulae"; but in fact the process is rather complex and tedious, and even the final formulas are long.

It does, however, yield formulations of surface shape in three regions, ahead of the pressure region of length $\ell$, within it, and behind it. The one simply stated result is the wavemaking drag/lift ratio.

$$\frac{D_w}{L} = \frac{h}{\ell} = \frac{2p}{\ell\rho g}\left(1 - \cos\frac{g\ell}{V^2}\right) \qquad [D\text{-}4]$$

where h and $\ell$ are shown in Figure D-1. This formula reduces to equation [D-3] when $\ell$ is very small (or when $g\ell \ll V^2$, $\frac{V^2}{g\ell} \gg 1$, or $\frac{V}{\sqrt{g\ell}} \gg 1$). Note that, since lift L is proportional to pressure, the wave drag $D_w$ is proportional to the square of the pressure p. Figure D-1, taken from Lamb, shows the surface shape taken by the water for $\ell$ equal to $\frac{V^2}{g}$ or for a Froude number $V/\sqrt{g\ell}$ of unity. In this case the pressure length $\ell$ is 0.159 times wave length.

The most difficult of the wavemaking problems, that due to a constant pressure region over a finite region, is the most useful. Fortunately, this work has been accomplished in recent years at the David Taylor Model Basin (Reference 3). Figure D-2 shows these results for a rectangular distribution of constant pressure. The wave drag parameter $f_\ell$ plotted on the ordinate (drag/lift ratio divided by pressure/length ratio and constants) is plotted against the length Froude number $V/\sqrt{g\ell}$. It can be noted that the wave drag $D_w$ is proportional to lift L, to pressure p, the wave drag parameter $f_\ell$, and is reciprocally proportional to pressure (or bubble) length $\ell_B$. The value of the wave drag parameter $f_\ell$ depends only on the length/beam ($\ell$/b) ratio and Froude number $F_\ell$. The speeds of maximum $f_\ell$ are referred to as hump speeds or Froude numbers. In the sub-hump region, the slopes of the lines in Figure D-2 were

adjusted to the slopes of early British Hovercraft experimental data. (See Reference 4.) The actual below-hump speed shape of $f_\ell$ is probably different from that shown in Figure D-2. The above simplification is only justified by the lack of importance of the below-hump speed values to the present task at hand. Secondary drag humps and valleys can exist, however, and they could be of considerable significance for some purposes.

Figure D-3 is also a plot of Newman's results (Reference 3) for a rectangular pressure region, but here the pressure or bubble length $\ell$ is replaced by $\sqrt{S}$, where S is the total pressure area. This replacement of $\ell$ by $\sqrt{S}$ is done both in the ordinate, abbreviated $f_{\sqrt{S}}$, and the abscissa $F_{\sqrt{S}}$, and results in a numerically different plot from that of Figure D-2. The advantage of Figure D-3 is that equal area pressure regions S, at the same area Froude number $F_{\sqrt{S}}$, have equal speeds; whereas, this is not true of Figure D-2. This could prove to be of some advantage in the comparison of vehicles of equal planform areas at the same speeds.

Figure D-4 is a plot similar to that of Figure D-3 but for elliptical planform shapes.

The resultant ratio of wave drag to lift (or drag/weight) is written, based on Figure D-2. For Figures D-3 and D-4, similar expressions apply.

$$\frac{D_w}{W} = \frac{4p}{\rho_w g \ell} \cdot f_\ell \qquad [D\text{-}5]$$

If external aerodynamic lift $C_L\, q_a\, S$ provides partial vehicle support, then the cushion pressure is less than the vehicle specific weight w, which is (W/S). In terms of the vehicle specific weight w, the wave drag/weight ratio is

$$\frac{D_w}{W} = \frac{4}{\rho_w g} \frac{w}{\ell_B} \left(1 - \frac{C_L\, q_a}{w}\right)^2 f_\ell \qquad [D\text{-}6]$$

## HYDRODYNAMIC DRAG

### SIDEWALL DRAG IN A CAPTURED AIR BUBBLE TYPE OF AIR CUSHION VEHICLE

Figure D-5 shows a CAB ACV in an above-hump speed condition. The angle $\alpha$ is exaggerated to point out two regions of sidewall wetting and drag.

A primary function of sidewalls (or sideboards or side skegs) is to prevent loss of air along the side of the vehicle. This is done by a triangularly wetted sideboard section of maximum wetted height $h_b$; that is, wetted on the outside only. This wetted area is referred to as sidewall due to the bubble. The assumption here is that the outside water line continues horizontally; whereas, the inside water line is turned down by the angle $\alpha$.

Below this triangular region of sideboard due to the bubble, is a region fully wetted on both sides that is referred to as additional sidewall, which is fully wetted, both inside and outside.

ADDITIONAL SIDEWALL DRAG — The amount of additional sidewall depth $h_a$ (Figure D-5) in smooth water would be held to a minimum at high speeds, but increased for transiting the hump speed. In waves at high speed, fore and aft seals or skis (skirts or trunks at zero "daylight" clearance) are set very close to the bottom of the sidewalls, but waves of height H will nevertheless wet an equivalent height $h_a$ of approximately one-half the average wave height.

$$h_a = \frac{1}{2} H \qquad [D\text{-}7]$$

The reason for this is that (for the above ski setting) the bottom of the sidewall runs at the wave trough, and the fore and aft skis or seals move up and down rapidly, staying very close to the water surface.

The additional sidewall drag is given by the product of the skin friction coefficient $C_f$, the dynamic pressure in water $q_w$, and the wetted area. Wetted area is the product of sidewall length $\ell_s$, wetted

height $h_a$, and number of wetted sides n. The additional sidewall drag/weight ratio is then, for n equal 4:

$$\frac{D_{s,a}}{W} = \frac{C_f \, q_w \, n \, \ell_s \, h_a}{\ell_B \, w \, b} \qquad [D\text{-}8]$$

$$\frac{D_{s,a}}{W} = 4 \, \frac{\ell_B}{b} \, \frac{\ell_s}{\ell_B} \, C_f \, \frac{q_w}{w} \, \frac{h_a}{\ell_B}$$

SECONDARY WAVEMAKING DRAG OR SIDEWALL DRAG DUE TO RETAINING THE AIR BUBBLE – Figure D-5 shows triangularly wetted sidewall sections of maximum wetted height $h_b$ that are wetted on the outside only. These triangular sections prevent air loss along the vehicle sides. Their total water wetted area for the whole vehicle is $h_b \, \ell_B$ . The drag is the product of the drag coefficient $C_f$, water dynamic pressure $q_w$, and wetted area $\ell_B \, h_b$ .

$$\frac{D_{s,w}}{W} = \frac{C_f \, q_w \, \ell_B \, h_b}{\ell_B \, w \, b} = \frac{\ell_B}{b} \, C_f \, \frac{q_w}{w} \, \frac{h_b}{\ell_B} \qquad [D\text{-}9]$$

From Figure D-5, it can be seen that

$$\frac{h_b}{\ell_B} = \frac{D_w}{W} \qquad [D\text{-}10]$$

Thus,

$$\frac{D_{s,w}}{W} = \frac{\ell_B}{b} \, C_f \, \frac{q_w}{w} \, \frac{D_w}{W} \qquad [D\text{-}11]$$

The expression (D-11) is for a CAB type of ACV with sidewalls, and no appreciable daylight clearance h.

Figure D-6 shows a sidewall ACV with some daylight clearance h, but with the sidewall still wetted to some extent on the outside. The wetted height is $(h_b - h)$; the wetted length is $\ell'$, and

$$\ell' = \frac{h_b - h}{h_b/\ell_B} = \frac{\left(\frac{D_w}{W}\ell_B - h\right)}{\frac{D_w}{W}} \qquad [D\text{-}12]$$

The wetted area is the product of $\ell'$ and $(h_b - h)$.

$$\text{Wetted Area} = \frac{\left(\frac{D_w}{W}\ell_B - h\right)^2}{\frac{D_w}{W}} = \frac{\ell_B{}^2\left(\frac{D_w}{W} - \frac{h}{\ell_B}\right)^2}{\frac{D_w}{W}} \qquad [D\text{-}13]$$

In this case

$$\frac{D_{s,w}}{W} = \frac{C_f\, q_w}{\ell_B\, w\, b} \cdot \frac{\ell_B{}^2\left(\frac{D_w}{W} - \frac{h}{\ell_B}\right)^2}{\frac{D_w}{W}}$$

Or

$$\left.\begin{aligned} \frac{D_{s,w}}{W} &= \frac{\ell_B}{b} C_f \frac{q_w}{w} \frac{\left(\frac{D_w}{W} - \frac{h}{\ell_B}\right)^2}{\frac{D_w}{W}} \quad , \quad \left(\frac{D_w}{W} - \frac{h}{\ell_B}\right) \geq 0 \\ &= 0 \quad , \quad \frac{D_w}{W} - \frac{h}{\ell_B} < 0 \end{aligned}\right\} \qquad [D\text{-}14]$$

This equation is valid for a more general case than (D-11). It is valid for a sidewall ACV with some daylight clearance h; but, nevertheless, with some wetted area, as shown in Figure D-6. When $h/\ell_B$ equals and then exceeds $D_w/W$, this drag becomes and remains zero.

SIDEWALL WAVEMAKING DRAG — Because of the high fineness ratios of CAB type ACV sidewalls, and low immersed depths when "on the bubble,"

sidewall wavemaking drag is usually negligible. Nevertheless, a form for computing it is available, based on wave-resistance theory for thin bodies (Reference 5). The resistance R is given in terms of beam B, draft H, length L, and the resistance coefficient r as follows:

$$R = r\left[\frac{8\rho g}{\pi}\ \frac{B^2\ H^2}{L}\right] \qquad [D\text{-}15]$$

The resistance coefficient r is plotted in Figure D-7 as a function of the reciprocal of Froude number squared for a parabolic thin body form and a "full" form.

FORM DRAG — Form drag of water-immersed portions of sidewalls is also usually negligible because of the high fineness ratios and low immersions. The profile drag, made up of skin friction and pressure form drag, then approximates the skin friction drag. One form of drag estimate by Hughes (Reference 6) is:

$$\frac{\text{Pressure Form Drag}}{\text{Profile Drag}} = 1.16\,\frac{t}{\ell} \qquad [D\text{-}16]$$

where t and $\ell$ are body thickness and length, respectively.

TRUNK OR SKIRT DRAG IN FULL-PERIPHERAL AIR CUSHION VEHICLES

The most important new element which the British have introduced in air cushion vehicle technology is flexible trunks, or skirts, as illustrated in Figure D-8. These trunks have in no way changed the basic principles of air cushion vehicles but they have had a very profound effect on the design choices that the designer is likely to make. The most obvious effect on the designer's choice is in the question of the "daylight" gap (the clearance between the bottom of the skirt and the water) and in the hard structure clearance (clearance between the hard structure of the vehicle and the water). The introduction of trunks has caused the designer to choose much lower values of the daylight gap and much higher values of the hard structure clearance. These choices involve design tradeoffs; for example, the choice of daylight gap

clearance involves the tradeoff between lift power and propulsion power. Using small values of daylight clearance, the lift power is greatly reduced. On the other hand, as soon as the vehicle operates not on a smooth surface, but in a seaway, the flexible trunk drags through the water and there is an increase in propulsive power required.

The only definitive information readily available on the subject of the drag associated with pulling flexible trunks through waves is represented in Figure D-9. It comes from the tests of the VA-3 Hovercraft conducted by Republic Aviation under contract to the Office of Naval Research.

In the left graph of Figure D-9, a trunk drag coefficient, referred to cushion area and the dynamic pressure of air, is plotted versus the ratio of wave height to cushion length for two different values of daylight clearance. It was then surmised that air cushion vehicles tend to average out the waves over which they fly, and therefore significant trunk drag should not occur until the wave height exceeds twice the daylight clearance. On this basis, in the right graph in Figure D-9, the same data were replotted versus the ratio of the wave height, less twice the daylight clearance, to cushion length. It was found that the data then fell nearly on a single line, which could be represented by a very simple nondimensional formula. This formula is an empirical expression of the VA-3 trunk drag. It is not known whether the drag characteristics of the VA-3 trunks are favorable or unfavorable compared with other designs. Since, however, these are the only definitive data available, this formula will be applied directly to estimate performance in a sea state.

The trunk drag coefficient $C_{D_t}$, is a function of the average wave height H, the ACV length $\ell$, and the daylight clearance h. The drag coefficient $C_{D_t}$ is the drag per unit cushion area S, and per unit air dynamic pressure $q_a$.

$$C_{D_t} = 6.6 \left( \frac{H - 2h}{\ell} \right)^{1.2} \qquad [D\text{-}17]$$

The trunk drag/weight ratio is then,

$$\left.\begin{aligned} \frac{D_t}{W} &= \frac{C_{D_t}\, S\, q_a}{\ell_B\, w\, b} = \frac{C_{D_t}\, q_a}{w} \\ &= 0.0012\ \frac{C_{D_t}\, q_w}{w} \end{aligned}\right\} \qquad [D\text{-}18]$$

Thus,

$$\left.\begin{aligned} \frac{D_t}{W} &= 6.6\ \frac{q_a}{w}\left(\frac{H - 2h}{\ell}\right)^{1.2} \\ &= 0.00792\ \frac{q_w}{w}\left(\frac{H - 2h}{\ell}\right)^{1.2} \end{aligned}\right\},\quad (H - 2h) \geq 0$$

$$= 0\ , \qquad (H - 2h) < 0 \qquad\qquad [D\text{-}19]$$

CAPTURED AIR BUBBLE AIR CUSHION VEHICLE OR SKI DRAG

The fore and aft seals or skis on CAB ACV's and models have been of a mechanical type principally, but the advantages of low-weight fabric materials, relative to high-frequency response, are evident. For this reason an analogy can be drawn to full-peripheral ACV trunks, and equation [D-19], times a geometric factor G, can be used tentatively for CAB type ACV seal or ski drag. This geometry factor G is less than 1, and depends on the length-to-beam ratio of a CAB ACV. If we put G in equation [D-19], it takes a value of 1 for the full-peripheral ACV. The factor G is tentatively given as $1/(1 + \ell/b)$.

EXTERNAL AERODYNAMIC DRAG

The external aerodynamic drag is given by the product of the drag coefficient $C_{D_e}$, the reference base cushion area S, and the air dynamic pressure $q_a$. The external aerodynamic drag/weight ratio is then,

$$\frac{D_e}{W} = \frac{C_{D_e}\, S\, q_a}{\ell_B\, w\, b} = \frac{C_{D_e}\, q_a}{w} \qquad [D\text{-}20]$$

## RAM OR MOMENTUM DRAG

The ram or momentum drag arises from bringing a constant mass flow rate of air $\dot{m}$ from a velocity V relative to the ACV to a zero velocity relative to the ACV. Thus,

$$D_r = \dot{m}\, V = \rho\, V_j \left( \mathcal{C}_c\, S_g \right)\, V_o \qquad [D\text{-}21]$$

where $\rho\, V_j$ is the mass flow rate per unit area near the ACV base (where the flow is fully contracted), $V_o$ is the vehicle velocity, and $\mathcal{C}_c\, S_g$ is the fully contracted flow area. $S_g$ is the daylight clearance area, and $\mathcal{C}_c$, the discharge coefficient.

If the vehicle were fully cushion supported, the weight W would equal wS, and the ram drag weight ratio would be

$$\frac{D_r}{W} = \frac{\left(2\ \rho_a\right)^{\frac{1}{2}} \left(\tfrac{1}{2}\rho_a\, V_j^{\,2}\right)^{\frac{1}{2}} \mathcal{C}_c\, S_g \left(\frac{2}{\rho_a}\right)^{\frac{1}{2}} \left(\tfrac{1}{2}\rho_a\, V_o^{\,2}\right)^{\frac{1}{2}}}{wS}$$

$$= \frac{2\, p^{\frac{1}{2}} \left( \mathcal{C}_c\, S_g \right) q_a^{\frac{1}{2}}}{wS} \qquad [D\text{-}22]$$

$$= 2\, \mathcal{C}_c \left( \frac{q_a}{w} \right)^{\frac{1}{2}} \frac{S_g}{S}$$

If external aerodynamic lift $C_L\, q_a\, S$ provides partial ACV support, the cushion pressure p is less than the specific weight w. Then,

$$\frac{D_r}{W} = 2\, \mathcal{C}_c \left(1 - C_L \frac{q_a}{w}\right)^{\frac{1}{2}} \left(\frac{q_a}{w}\right)^{\frac{1}{2}} \frac{S_g}{S} \qquad [D\text{-}23]$$

## DRAG SUMMARY

The total drag/lift ratio $D_T/W$ is given as an appropriate summation of the previously derived component lift/drag ratios. These component ratios are repeated here for convenience.

$$\frac{D_w}{W} = \frac{4}{\rho_w g} \frac{w}{\ell_B} \left(1 - \frac{C_L q_a}{w}\right)^2 f_\ell \qquad [D\text{-}6]$$

$$\frac{D_{s,a}}{W} = 4\left(\frac{\ell_B}{b}\right)\frac{\ell_s}{\ell_B} C_f \frac{q_w}{w} \frac{h_a}{\ell_B} \qquad [D\text{-}8]$$

$$\left.\begin{aligned} \frac{D_{s,w}}{W} &= \frac{\ell_B}{b} C_f \frac{q_w}{w} \frac{\left(\frac{D_w}{W} - \frac{h}{\ell_B}\right)^2}{\frac{D_w}{W}} \quad , \quad \left(\frac{D_w}{W} - \frac{h}{\ell_B}\right) \geq 0 \\ &= 0 \quad , \quad \left(\frac{D_w}{W} - \frac{h}{\ell_B}\right) < 0 \end{aligned}\right\} \qquad [D\text{-}14]$$

$$\left.\begin{aligned} \frac{D_t}{W} &= 6.6 \frac{q_a}{w} \left(\frac{H - 2h}{\ell}\right)^{1.2} G \quad , \quad \left(\frac{H - 2h}{\ell}\right) \geq 0 \\ &= 0 \quad , \quad \left(\frac{H - 2h}{\ell}\right) < 0 \end{aligned}\right\} \qquad [D\text{-}19]$$

where

$G = 1$ for full-peripheral ACV

$$G = \frac{1}{1 + \ell/b} \quad \text{for CAB ACV}$$

$$\frac{D_e}{W} = \frac{C_{D_e} q_a}{w} \qquad [D\text{-}20]$$

$$\frac{D_r}{W} = 2\rho_c \left(1 - C_L \frac{q_a}{w}\right)^{\frac{1}{2}} \left(\frac{q_a}{w}\right)^{\frac{1}{2}} \frac{S_g}{S} \qquad [D\text{-}23]$$

Aerodynamics Laboratory
David Taylor Model Basin
Washington, D. C.
December 1965

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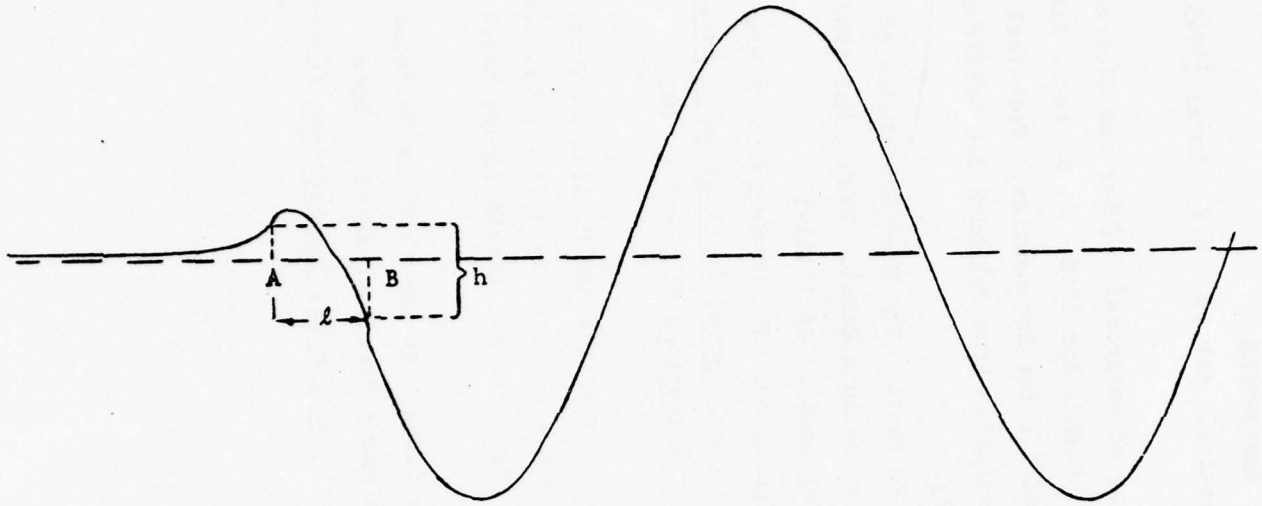



Figure D-1 - Two-Dimensional Gravity Wave for a Pressure Region A to B, for a Froude Number $V/\sqrt{g\ell}$ of Unity

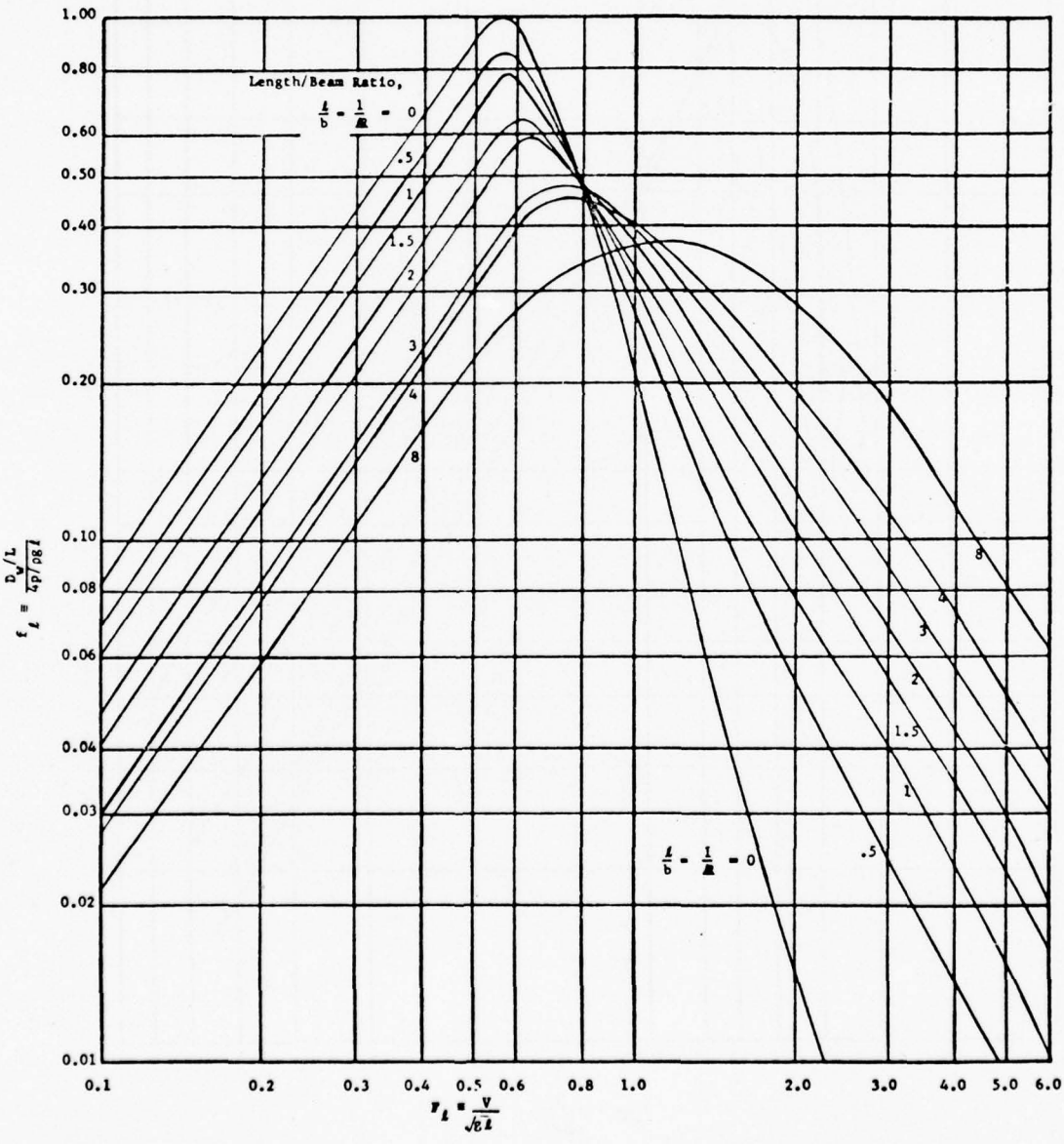



Figure D-2 - Wave Drag Results, Based on Pressure Region Length $\ell$, for Rectangular Pressure Planforms

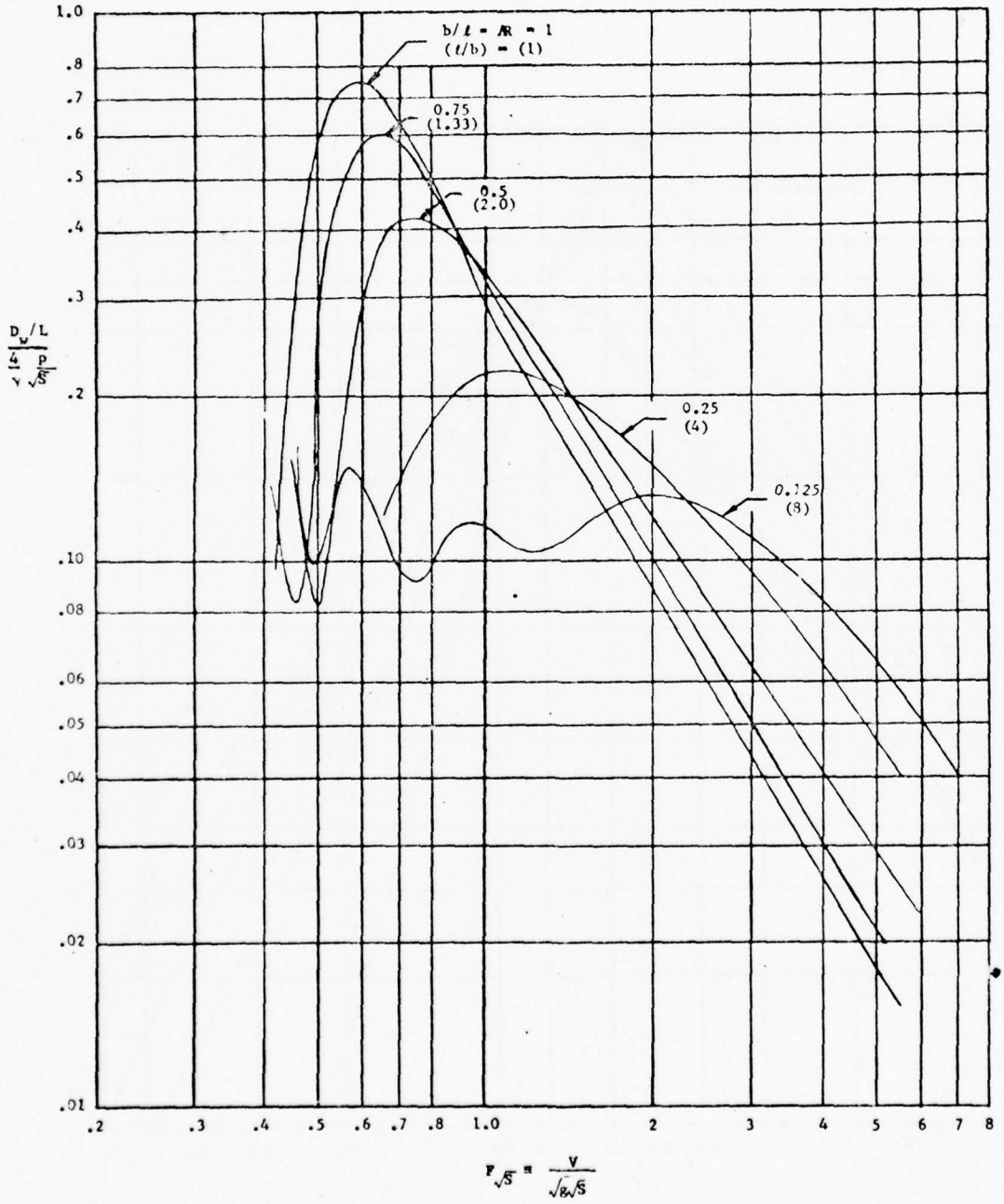



Figure D-3 - Wave Drag Results, Based on $\sqrt{S}$, for Rectangular Pressure Planforms

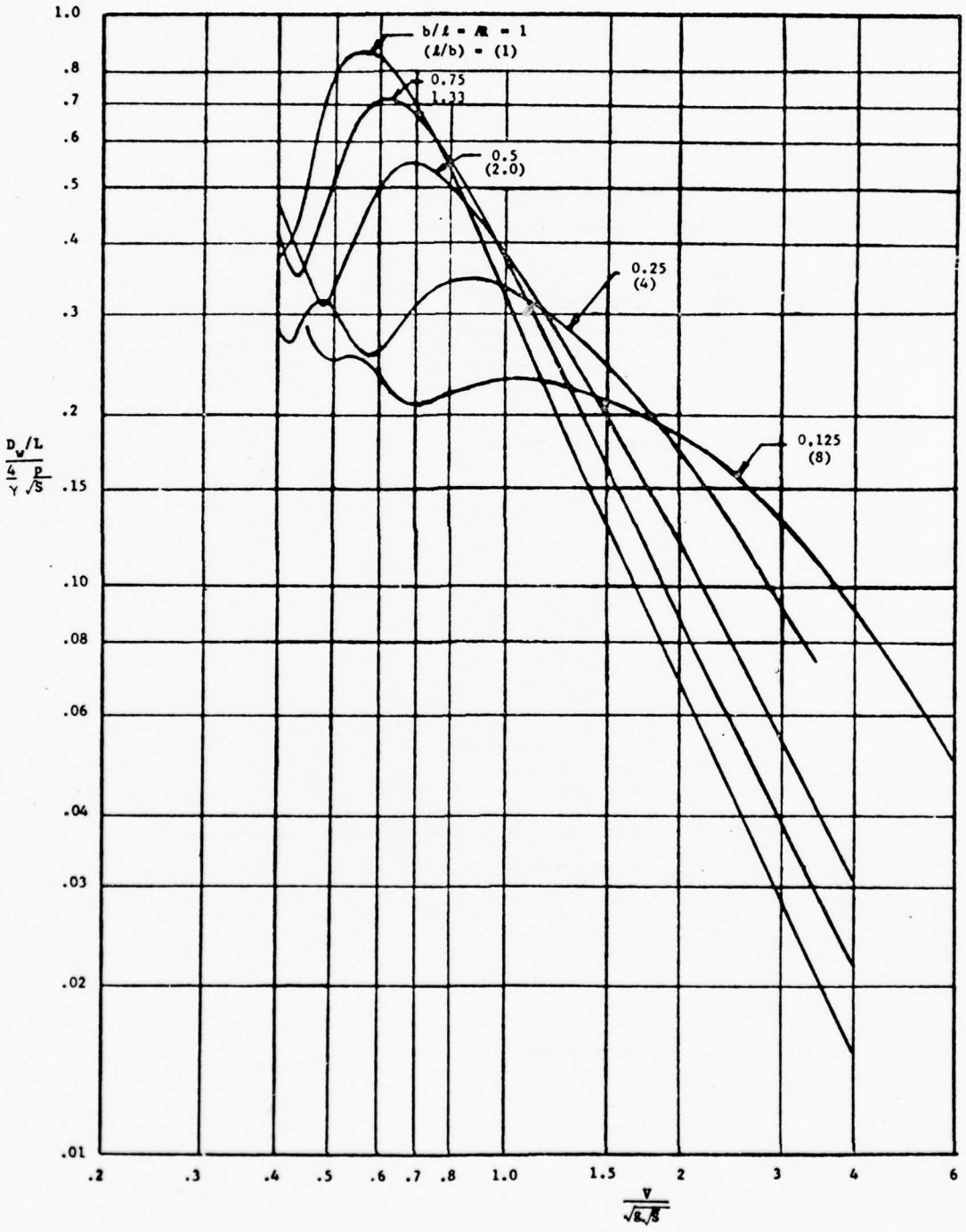
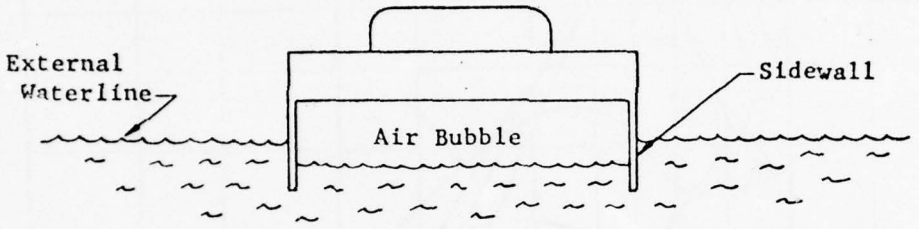
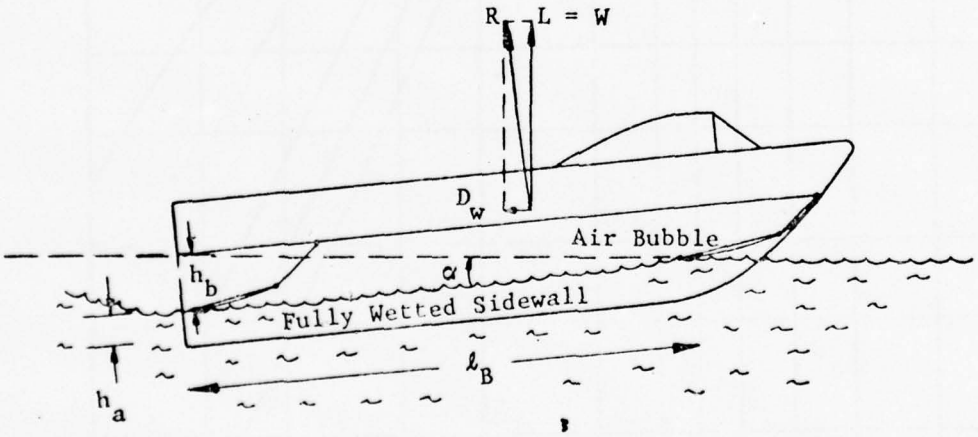



Figure D-4 - Wave Drag Results, Based on $\sqrt{S}$, for Elliptical Pressure Planforms




Sectional View




Side View

Figure D-5 - Sidewall CAB ACV Showing Sidewall Wetted Areas

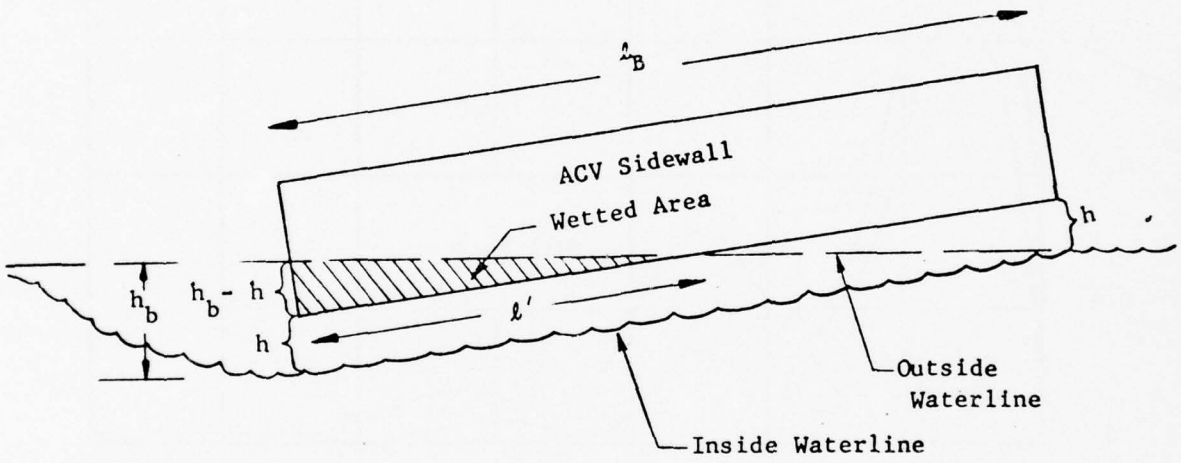



Figure D-6 - Sidewall ACV With Daylight Clearance, but With Some Wetted Area

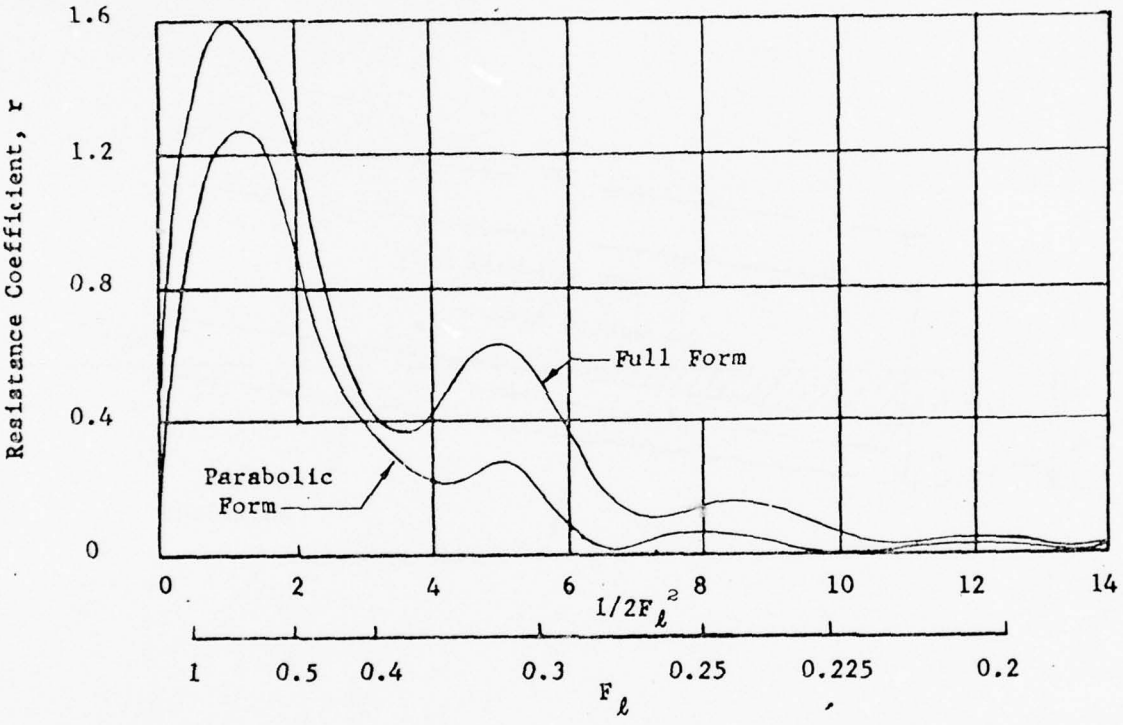



Figure D-7 - Wave Resistance Coefficient for Thin Bodies

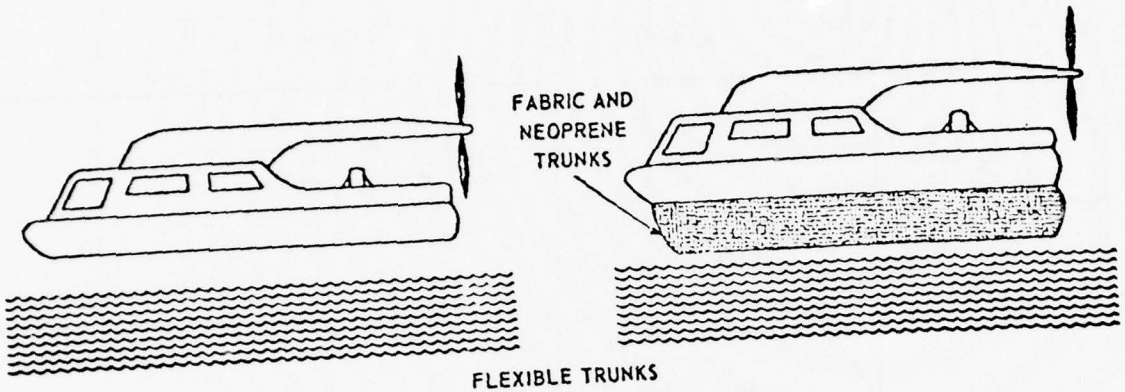



Figure D-8 - Full-Peripheral Vehicles With and Without Trunks

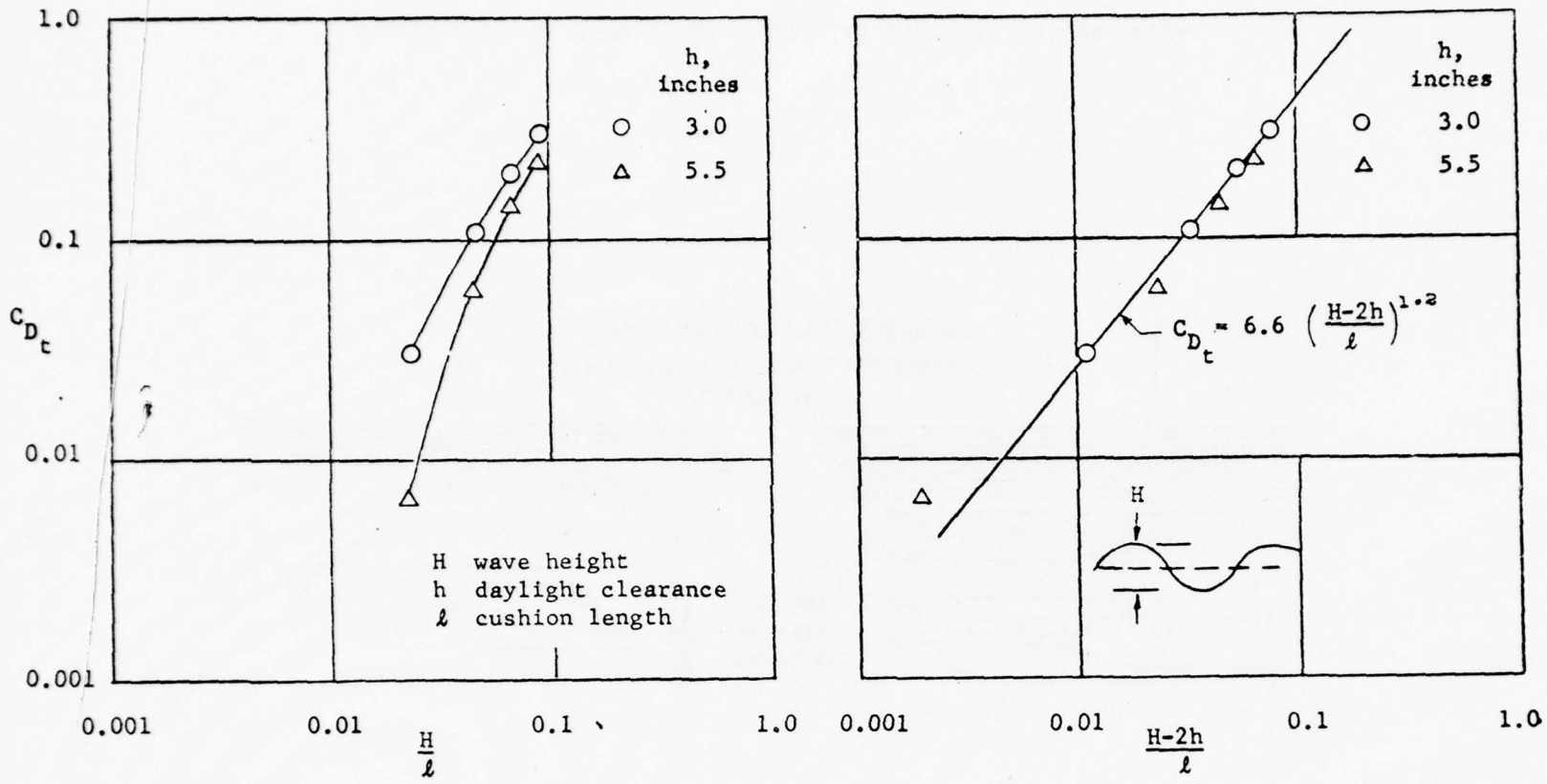



Figure D-9 — Trunk Drag Coefficient (VA-3 Data)

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| Internal Efficiency | | | | | | |
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The power dissipated through losses in the inlet-compressor-ducting system is discussed. A simple illustrative derivation of the characteristics of a hypothetical compressor is carried out. These characteristics are then used to illustrate the construction of combined internal characteristics charts.

The dominant influence of the air gap area in determining the necessary size of the compressor(s) is emphasized.

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# SOME DESIGN PRINCIPLES OF GROUND EFFECT MACHINES
## SECTION F - CUSHION CONTRIBUTIONS TO STABILITY

by

Harvey R. Chaplin and Allen G. Ford

APPROVED FOR PUBLIC RELEASE: DISTRIBUTION UNLIMITED

February 1967

Report 2121F
Aero Report 1100F

## Foreword

This report is based on a lecture series presented by the authors at the von Kármán Institute for Fluid Dynamics, Rhode-Saint-Genèse, Belgium, in May 1965; and at the University of Maryland, College Park, Maryland, in July 1965. The lectures were prepared under the joint auspices of the David Taylor Model Basin and the Naval Air Development Center. They were presented in Belgium under the joint sponsorship of the von Kármán Institute and the Advisory Group for Aerospace Research and Development (AGARD); and in Maryland under the sponsorship of the Assistant Secretary of the Navy for Research and Development.

The revised lectures will be presented as follows:

A. Introductory Survey
B. Air Cushion Mechanics
C. Internal Aerodynamics
D. Drag
~~E. Drag Optimization for Sidewall GEM (GAB)~~
F. Cushion Contributions to Stability
G. Seakeeping
H. Performance Summary

## NOTATION

| | |
|---|---|
| $\ell$ | cushion length, ft |
| b | cushion beam, ft |
| S | cushion area, $ft^2$, measured, in plan view, to outer edge of nozzle exit |
| C | cushion perimeter, ft |
| $\Delta p$ | cushion pressure, psfg |
| $p_t$ | total pressure of air supply to cushion at nozzle exit or cushion entry, psfg |
| $V_o$ | flight velocity in earth axes, ft/sec |
| $V_k$ | flight velocity in earth axes, knots |
| $\rho, \rho_a$ | air density, slugs/$ft^3$ |
| $S_g$ | daylight gap area (cushion perimeter times mean daylight gap, h), $ft^2$ |
| t | nozzle thickness, ft (also used for time, seconds) |
| h | daylight gap, ft (local value or mean value as appropriate) |
| $\theta$ | peripheral jet efflux angle, measured positive inward from vertical, degrees |
| x | nozzle thickness parameter $\left(\frac{t}{h}\ (1 + \sin\theta)\right)$ |
| Q | total air volume flow rate through cushion system, $ft^3$/sec |
| $\mathscr{C}_c$ | cushion discharge coefficient $\left(\frac{Q}{S_g\sqrt{\frac{2}{\rho}\Delta p}}\right)$ |
| W | gross weight, pounds |
| y | heave displacement, ft, measured from mean vertical position |
| $\alpha$ | pitch angle, radians |
| $\phi$ | roll angle, radians |
| $k_h$ | heave stiffness coefficient, $-\frac{\partial}{\partial h}$ (Lift), lb/ft |
| $C_h$ | heave damping coefficient, $-\frac{\partial}{\partial \dot{h}}$ (Lift), lb-sec/ft |
| M | pitching moment, lb-ft |
| $\mathscr{L}$ | rolling moment, lb-ft |

| | |
|---|---|
| $(\dot{})$ | $\frac{d(\ )}{dt}$ |
| $\epsilon$, $\bar{h}$ | amplitude of simple wave, ft |
| $\omega_o$ | natural frequency of heave oscillation, radians/sec |
| $\omega_{o,\alpha}$ | natural frequency of pitch oscillation, radians/sec |
| $\omega_{o,\phi}$ | natural frequency of roll oscillation, radians/sec |
| $\omega$ | frequency of encounter, radians/sec |
| $\lambda$ | wave length, ft |
| g | acceleration due to gravity, ft/sec$^2$ |

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## SUMMARY

A review is undertaken of some of the more important elementary relationships which determine the influence of the air cushion on the pitch, heave, and roll stability of ground effect machines. The analyses presented yield only rough approximations of the quantities needed for design. Further, the analyses are restricted to cases of vehicles which are not in contact with the surfaces over which they travel, whereas the influences of surface contact are of primary importance in most practical problems.

## INTRODUCTION

The problem of predicting the contributions of the cushion to the stability characteristics has never been formulated in a fully satisfactory manner. It is regretted that it is not practical to fully rectify that unfortunate situation at present.

The difficulty is twofold. First, there are some aspects of the problem which are not sufficiently well understood to support a fully satisfactory formulation. In the second place, even for those aspects which are reasonably well understood, it is found that the different elements of the cushion system interact with one another in a complex manner which makes it very difficult, if not impossible, to arrive at simple and reliable design formulas. (There is a third difficulty concerning nonlinearity, but it would be idle to dwell on this when we cannot even deal properly with the linearized approximations.)

The most important contributions of the cushion to stability involve the following motions:

Heave - vertical translational motion ($y$).

Pitch - rotation about the lateral axis ($\alpha$).

Roll - rotation about the longitudinal axis ($\phi$).

In each of these motions, the cushion may contribute "stiffness" (a force or moment opposing the displacement from equilibrium in proportion to this displacement) and "damping" (a force or moment opposing the motion in proportion to the velocity of the motion).

We will ignore effects of deformation of the water surface under loads imposed by the cushion. These effects are important at low forward speeds.

## HEAVE MOTION WITHOUT WATER CONTACT

The easiest of these motions to understand, by far, is heave. Moreover, some of the principles which govern heave motion are also of primary importance in the pitch and roll motions. We will therefore consider the heave motion first, and somewhat more carefully than the other motions. Nevertheless, it will be found that, even in the case of heave, we will have to content ourselves with a rather superficial treatment of the subject in order to avoid an unduly laborious development.

### HEAVE STIFFNESS

The heave stiffness coefficient is defined as:

$$k_h \equiv -\frac{\partial}{\partial h}(\text{Lift}) \doteq -S\frac{\partial}{\partial h}(\Delta p)$$

This definition is not mathematically meaningful, however, until we have specified which independent variables the base pressure is considered to depend upon. Reflecting upon the relationships derived in Sections B and C, it will be seen that our problem would be simplest if we could write:

$$\Delta p = \Delta p(h, \dot{h}, Q)$$

or

$$\Delta p = \Delta p(h, \dot{h}, p_t)$$

It is easy to see, however, that in heave motions of practical interest, neither $p_t$ nor Q are independent of $\dot{h}$. For example, assuming compressor rotational speed $\Omega$ constant, examination of Figure C-4 will show that, as h is decreased (and both $\mathcal{A}_c$ $S_g$ and $\lambda$ are therefore decreased), the total pressure $p_t$ will ordinarily increase, whereas Q will decrease. However, it may not always be safe even to assume $\Omega$ constant, because the compressor torque will ordinarily oscillate during heave oscillation. For a thorough analysis one must write:

$$\Delta p = \Delta p(h, \dot{h}, \text{throttle setting})$$

and then derive a system of coupled differential equations describing the behavior of engine, compressor, and cushion, including an accounting for the angular momentum of engine and compressor, and the characteristics of the engine governor system, when appropriate.

This is not terribly difficult to do for any one well-defined vehicle, but it is rather laborious, and the result is so complex and so dependent on the peculiarities of the particular system considered that it is difficult to draw any simple conclusions which are of general interest.

In the present study, we will restrict ourselves to a more superficial treatment of the problem as follows: It was noted above that, in a real heave oscillation (with no compressor stalling), $p_t$ increases with decreasing h while Q decreases with decreasing h; whereas, if h is held artificially fixed, Q will obviously increase with increasing $p_t$. We may surmise from this that the correct value of the stiffness coefficient will lie somewhere between the value calculated by holding Q fixed and the value calculated by holding $p_t$ fixed. We can therefore draw at least some qualitative conclusions by comparing results of these two calculations.

1. Q Held Fixed - In this case it is convenient to use the following relationship from Section B:

$$\Delta p = \frac{1}{2} \rho \left( \frac{Q}{hC} \right)^2 \div \mathcal{A}_c^2$$

This relationship is easy to work with in the two limiting cases $x \ll 1$ and $x \gg 1$. In the first case,

$$\frac{1}{\mathcal{A}_c^2} \doteq 2 \frac{h}{t} (1 + \sin \theta)$$

and

$$\left. \frac{\partial(\Delta p)}{\partial h} \right]_{Q \text{ fixed}} \doteq - \frac{\Delta p}{h} \quad , \quad x \ll 1$$

In the second case,

$$\mathcal{A}_c^2 \doteq \text{Constant}$$

and

$$\frac{\partial(\Delta p)}{\partial h} \doteq - 2 \frac{\Delta p}{h} \quad , \quad x \gg 1$$

2. $p_t$ Held Fixed - In this case the convenient relationship from Section B is

$$\Delta p = p_t \left(1 - e^{-2x}\right)$$

and

$$\left.\frac{\partial(\Delta p)}{\partial h}\right]_{p_t \text{ fixed}} = -\frac{\Delta p}{h}\,\frac{x\, e^{-2x}}{1 - e^{-2x}}$$

These various relationships are plotted in Figure F-1, in the form of the nondimensional heave-stiffness parameter $k_h \frac{h}{\Delta p\, S}$ versus nozzle thickness parameter x. The qualitative behavior of the "Q-fixed" curve is indicated by a dashed line. (This curve could be determined accurately from the relationships given in Section B, but at the expense of some labor; and, since we are only entitled to draw qualitative conclusions anyway, it hardly seems worth while.)

It is seen in Figure F-1 that the heave stiffness parameter calculated with Q fixed covers a range from 1.0 to 2.0 as x increases from zero; whereas, with $p_t$ fixed, it covers a range from 1.0 to zero as x increases from zero. The most natural assumption to make is that, with throttle setting fixed, the heave stiffness parameter will be relatively independent of x and will have a value of roughly

$$k_h \frac{h}{\Delta p\, S} \doteq 1.0 \qquad [F\text{-}1]$$

HEAVE DAMPING

The heave damping coefficient is defined as

$$C_h \equiv -\frac{\partial}{\partial \dot{h}}(\text{Lift}) \doteq -S\,\frac{\partial}{\partial \dot{h}}(\Delta p)$$

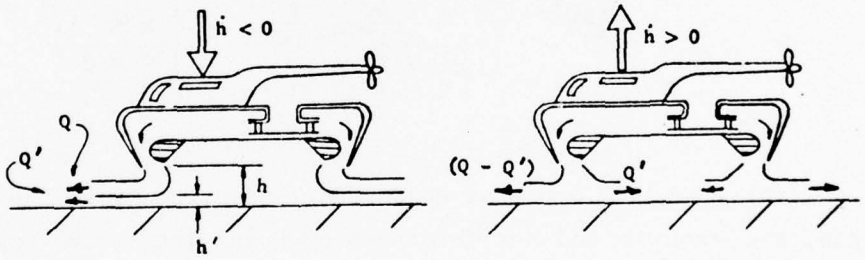
The same difficulties as to mathematical meaning of this difinition (concerning the independence of variables), which were discussed above for the case of heave stiffness, obviously apply here. As before, we will content ourselves with obtaining rough estimates of the damping

(1) with Q held fixed, and (2) with $p_t$ held fixed; and will rely on the fact that the "real" case will lie somewhere between these extremes.

The damping force arises primarily from the changing volume of the cushion. The rate of change of cushion volume is approximately

$$Q' \doteq - \dot{h} S$$

It is necessary to consider separately the case $\dot{h} < 0$ (vehicle moving downward), when the flow Q′ "squeezes out" under the normal efflux Q; and the case $\dot{h} > 0$, when a portion Q′ of the normal efflux Q is diverted back into the cushion.




1. Q Held Fixed - We will consider the four cases, $x \ll 1$, $\dot{h} \lesseqgtr 0$ and $x \gg 1$, $\dot{h} \lesseqgtr 0$.

(a) $\underline{x \ll 1, \dot{h} < 0}$ - As the flow Q′ approaches ambient pressure, it approaches the velocity $\sqrt{\frac{2}{\rho} \Delta p}$ and displaces a thickness

$$h' \doteq \frac{Q'}{\sqrt{\frac{2}{\rho} \Delta p}\, C}$$

The cushion pressure will be roughly the same as if $\dot{h}$ were zero and the daylight clearance were $h - h'$; that is,

$$\Delta p + \delta(\Delta p) \doteq \Delta p \frac{h}{h - h'} \doteq \Delta p \left(1 + \frac{h'}{h}\right)$$

$$\doteq \Delta p \left(1 + \frac{Q'}{\sqrt{\frac{2}{\rho} \Delta p}\ hC}\right)$$

$$\doteq \Delta p \left(1 - \frac{\dot{h}}{\sqrt{\frac{2}{\rho} \Delta p}} \frac{S}{hC}\right)$$

$$\frac{\partial(\Delta p)}{\partial \dot{h}} \doteq - \frac{\Delta p}{\sqrt{\frac{2}{\rho} \Delta p}} \frac{S}{hC} \quad , \quad \begin{cases} Q \text{ fixed,} \\ x \ll 1, \\ \dot{h} < 0 \end{cases}$$

(b) $\underline{x \ll 1,\ \dot{h} > 0}$ - As discussed in Section B, in the thin-jet case, the momentum balance for the peripheral jet can be expressed

$$\Delta p\ h = j_1 \sin \theta + j_2$$

where

$$j_1 = 2\ p_t\ t$$

is the momentum per unit periphery of the jet leaving the nozzle, and $j_2$ (taken equal to $j_1$, in Section B) is the net momentum per unit periphery of the jet moving outward after ground contact. In the present case,

$$j_2 \doteq j_1 \left(\frac{Q - 2Q'}{Q}\right)$$

$$Q \doteq \sqrt{\frac{2}{\rho} p_t}\ t\ C$$

$$\Delta p\ h \doteq 2\ p_t\ t\left[\sin\theta + 1 - \frac{2\ \dot{h}\ S}{\sqrt{\frac{2}{\rho}\ p_t}\ t\ C}\right]$$

$$\frac{\partial(\Delta p)}{\partial \dot{h}} \doteq -4\ \frac{p_t}{\sqrt{\frac{2}{\rho}\ p_t}}\ \frac{S}{hC} = -\frac{4}{\sqrt{\frac{\Delta p}{p_t}}}\ \frac{\Delta p}{\sqrt{\frac{2}{\rho}\ \Delta p}}\ \frac{S}{hC}$$

$$\frac{\partial(\Delta p)}{\partial \dot{h}} \doteq -\frac{4}{\sqrt{1 - e^{-2x}}}\ \frac{\Delta p}{\sqrt{\frac{2}{\rho}\ \Delta p}}\ \frac{S}{hC}\ , \begin{cases} Q \text{ fixed}, \\ x \ll 0, \\ \dot{h} > 0 \end{cases}$$

(c),(d) $\underline{x \gg 0,\ \dot{h} \gtrless 0}$ - These two cases can be treated as one. When the jet is very thick, $\mathcal{O}_c \doteq$ Constant, $\Delta p \doteq p_t$, and the base pressure can be determined from:

$$Q + Q' \doteq \mathcal{O}_c\ \sqrt{\frac{2}{\rho}\ \Delta p}\ hC$$

$$= Q - \dot{h}\ S$$

$$\Delta p \doteq \frac{1}{2}\ \rho\left(\frac{Q - \dot{h}\ S}{\mathcal{O}_c\ hC}\right)^2$$

$$\frac{\partial(\Delta p)}{\partial \dot{h}} \doteq -2\ \frac{\Delta p}{\sqrt{\frac{2}{\rho}\ \Delta p}}\ \frac{S}{hC} \div \mathcal{O}_c \qquad \begin{cases} Q \text{ fixed}, \\ x \gg 0, \\ \dot{h} \lesssim 0 \end{cases}$$

2. $p_t$ Held Fixed

(a) $\underline{x \ll 1,\ \dot{h} \lesseqgtr 0}$ - In Section C, we saw that

$$\left.\begin{aligned} \Delta p &\doteq p_t(2x) \\ Q &\doteq \sqrt{\frac{2}{\rho} p_t}\ tC \end{aligned}\right\} x \ll 1$$

It therefore makes no difference, when $x \ll 1$, whether one holds $p_t$ fixed or holds Q fixed; the relationship between the cushion pressure and h is the same. Therefore, the relationships derived above apply equally here; that is

$$\frac{\partial(\Delta p)}{\partial \dot{h}} \doteq -\frac{\Delta p}{\sqrt{\frac{2}{\rho}\Delta p}} \frac{S}{hC}, \quad \begin{cases} p_t \text{ fixed}, \\ x \ll 1, \\ \dot{h} < 0 \end{cases}$$

$$\frac{\partial(\Delta p)}{\partial \dot{h}} \doteq \frac{2}{\sqrt{1 - e^{-2x}}} \frac{\Delta p}{\sqrt{\frac{2}{\rho}\Delta p}} \frac{S}{hC}, \quad \begin{cases} p_t \text{ fixed}, \\ x \ll 1, \\ \dot{h} > 0 \end{cases}$$

(b) $\underline{x \ll 1,\ \dot{h} \gtrless 0}$ - In this case we have seen that $\Delta p = p_t$. Therefore

$$\frac{\partial(\Delta p)}{\partial \dot{h}} \doteq 0, \quad \begin{cases} p_t \text{ fixed}, \\ x \gg 0, \\ \dot{h} \lesseqgtr 0 \end{cases}$$

An intuitive interpretation of the foregoing limiting-case relationships is given in Figure F-2. Note that, in the case $\dot{h} < 0$, the situation is quite similar to that found for heave stiffness; that is, the range of possible values of the heave damping parameter is more or

less symmetrically disposed around the value 1.0, and we can be reasonably confident that we would not go too far astray by choosing this constant value as an estimate for practical purposes. However, the situation is considerably more complicated in the case $\dot{h} > 0$. In the range $x < 1$, the value of the damping parameter is strongly dependent on x, and can assume large values when x is very small. Thus we are faced, not only with a situation in which the damping parameter can vary over a wide range, but also with a situation in which (since, in general, the damping coefficient will vary during a cycle of oscillation as $\dot{h}$ passes through zero and changes sign) we are not entitled to assume simple harmonic motion, even when the amplitude of the motion is small.

Nevertheless, we are faced with the choices of (a) proceeding on the assumption that the damping parameter is constant, (b) undertaking a very complex investigation which takes into account the variation in damping parameter, or (c) abandoning the investigation of heave motions.

We will, of course, follow course (a), assuming, in fact, that the damping parameter is equal to 1.0; that is,

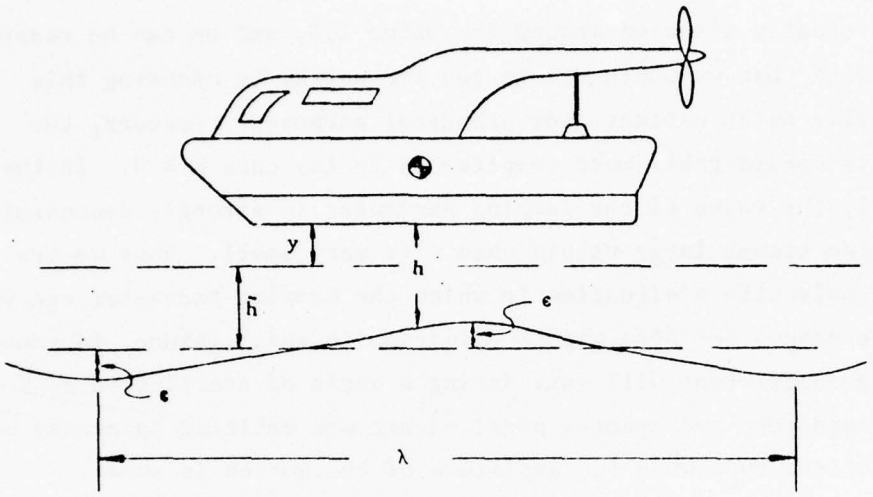
$$C_h \doteq \frac{\Delta p\, S}{\sqrt{\frac{2}{\rho}\Delta p}} \frac{S}{hC} \qquad [F\text{-}2]$$

Note that, despite the considerable range of uncertainty involved in this assumption, it is least likely to give the correct order of magnitude, under conditions of practical interest.

WAVE-EXCITED HEAVE MOTION

Let us assume that the vehicle is executing a steady-state heave oscillation, excited by passage over a regular sinusoidal surface wave system of small amplitude.

1. Wave Length >> Cushion Length ($\lambda \gg \ell$) - In this case we can measure the mean daylight clearance h from the bottom of the craft directly below the c.g. to the local water surface.




(Dimensions distorted for clarity)

The linearized equation of motion is:

$$\frac{W}{g}\ddot{y} + C_h \dot{h} + k_h (h - \bar{h}) = 0$$

where

$y$ is displacement of craft above mean position

$h$ is distance of craft above local water surface

$\bar{h}$ is distance of mean position of craft above mean level of water surface.

If the wave system passes under the craft with frequency $\omega$ radians/second, we can (with proper choice of time origin) write

$$h = \bar{h} + y - \varepsilon \sin \omega t$$

$$\dot{h} = \dot{y} - \varepsilon\omega \cos \omega t$$

The equation of motion then becomes

$$\frac{W}{g}\ddot{y} + C_h \dot{y} + k_h y = k_h \varepsilon \sin \omega t + C_h \varepsilon\omega \cos \omega t$$

Now, assuming $W \doteq \Delta p\, S$, we can write Equations F-1 and F-2

$$k_h \doteq \frac{W}{h}$$

$$C_h \doteq \frac{W}{\sqrt{\frac{2}{\rho}\Delta p}} \frac{S}{hC}$$

The equation of motion can then be written

$$\ddot{y} + 2\zeta\omega_o \dot{y} + \omega_o^2 y = \omega_o^2 \epsilon\left(\sin \omega t + 2\zeta \frac{\omega}{\omega_o} \cos \omega t\right)$$

where

$$\omega_o \doteq \sqrt{\frac{g}{h}} \qquad [F\text{-}3a]$$

$$\zeta = \frac{1}{2}\sqrt{\frac{\rho g h}{2\,\Delta p}} \frac{S}{hC}$$

This equation is easily solved, yielding

$$\frac{y}{\epsilon} = \left[\frac{1 + \left(2\zeta \frac{\omega}{\omega_o}\right)^2}{\left(1 + \frac{\omega^2}{\omega_o^2}\right)^2 + \left(2\zeta \frac{\omega}{\omega_o}\right)^2}\right]^{\frac{1}{2}} \sin\left[\omega t - (\phi_2 - \phi_1)\right] \quad , \quad \lambda \gg \ell$$

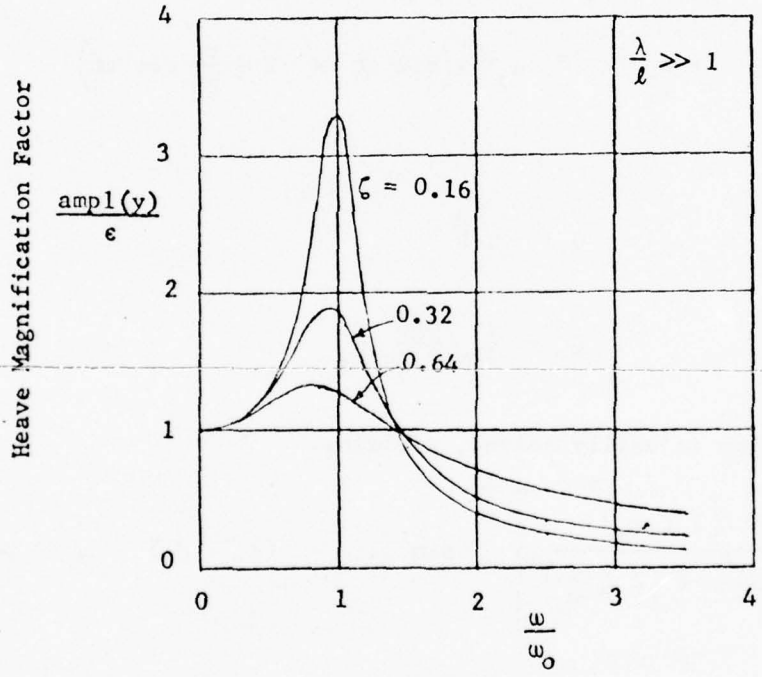
$$\phi_1 = \tan^{-1}\left(2\zeta \frac{\omega}{\omega_o}\right) \qquad [F\text{-}3b]$$

$$\phi_2 = \tan^{-1}\left(\frac{2\zeta \frac{\omega}{\omega_o}}{1 - \frac{\omega^2}{\omega_o^2}}\right)$$

Example - Suppose $h = 0.5$ foot; $\Delta p = 0.5\ \ell$; $\frac{\ell}{b} = 2$, $S = \frac{1}{2}\ell^2$; and $C = 3\ell$. (These assumptions are roughly consistent with currently

operating vehicles.) Then the motion is approximately as follows, for craft of various sizes:

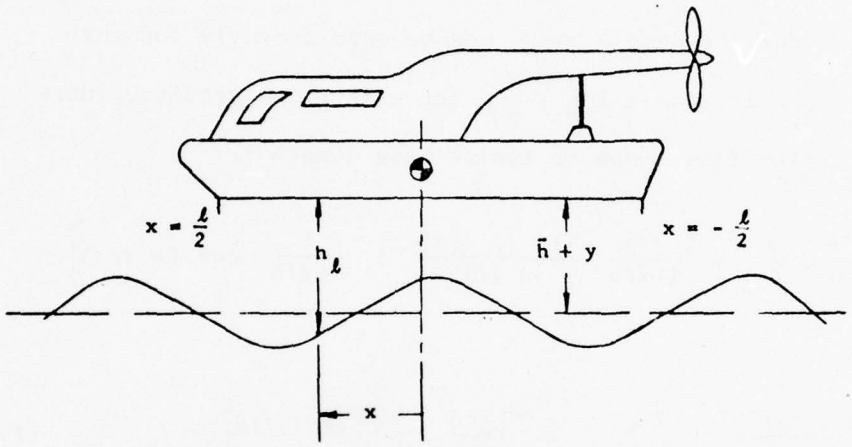
| | | | | |
|---|---|---|---|---|
| $\ell$ (ft) | = | 25 | 100 | 400 |
| $\zeta$ | = | 0.16 | 0.32 | 0.64 |
| $\omega_o$ (radians/seconds) | = | 8.0 | 8.0 | 8.0 |




Note the distinct resonance effect, which tends to become less pronounced for larger craft, and note the tendency for the heave motion to become very small at high frequencies of encounter.

2. Efffect of Wave Length - If the wave length is not very large compared with the cushion length, it is necessary to consider the variations in daylight clearance along the length of the craft; that is, instead of measuring h at the center of the craft, we must calculate the average value of the local gap h along the entire periphery.

For a rectangular cushion this is fairly easy.




For a particular point along the periphery we can write:

$$h_\ell(x, t) = \bar{h} + y - \epsilon \sin\left(\omega t + 2\pi \frac{x}{\lambda}\right)$$

and thus calculate the average gap:

$$h(t) = \frac{\ell}{b+\ell}\frac{1}{\ell}\int_{-\frac{\ell}{2}}^{\frac{\ell}{2}} h_\ell(x,t)dx + \frac{1}{2}\frac{b}{b+\ell}\left[h_\ell\left(\frac{\ell}{2},t\right) + h_\ell\left(-\frac{\ell}{2},t\right)\right]$$

or

$$h = \bar{h} + y - \left[\frac{\ell/b}{1+\ell/b}\frac{\sin(\pi\ell/\lambda)}{\pi\ell/\lambda} + \frac{1}{1+\ell/b}\cos(\pi\ell/\lambda)\right]\epsilon \sin\omega t$$

(Note: We are averaging along the periphery, which is appropriate for that part of the excitation which arises from heave stiffness. A smaller excitation term arises from heave damping, and for this term the average should be taken over the whole cushion area. We will neglect this refinement.)

Note that, in the limit $\frac{\lambda}{\ell} \to \infty$, this degenerates to the same formulation we used previously; that is,

$$h \doteq \bar{h} + y - \epsilon \sin \omega t \quad , \quad \lambda \gg \ell$$

Thus, our results for the case $\lambda \gg \ell$ can be used directly for the present case by substituting $\epsilon F\left(\frac{\ell}{b} , \frac{\ell}{\lambda}\right)$ for $\epsilon$ in those results, where the heave attenuation factor due to finite wave length is:

$$F\left(\frac{\ell}{b} , \frac{\ell}{y}\right) = \left[\frac{\ell/b}{1+\ell/b} \frac{\sin (\pi \ell/\lambda)}{\pi \ell/\lambda} + \frac{1}{1+\ell/b} \cos (\pi \ell/\lambda)\right]$$

In other words:

$$\left[\frac{\text{ampl } (y)}{\epsilon}\right]_{\text{arbitrary } \frac{\lambda}{\ell}} = \left|F\left(\frac{\ell}{b} , \frac{\ell}{\lambda}\right)\right| \left[\frac{\text{ampl } (y)}{\epsilon}\right]_{\frac{\lambda}{\ell} \gg 1} \qquad [F\text{-}4]$$

The heave attenuation factor is plotted in Figure F-3. For a craft of practical length/beam ratio, it is found that there is one range of wave lengths slightly longer than the cushion length, and a second range of wave lengths about two-thirds the cushion length, for which the heave motions will be practically nil regardless of frequency of encounter $\omega$. (Indeed, there are an infinite number of such ranges of shorter and shorter wave lengths. However, these further ranges are of limited interest, because the frequency of encounter will generally be so high as to preclude significant heave motions anyway.)

Before leaving the subject of heave damping, it may be well to review what we have done.

1. By making broad (and weakly justified) assumptions regarding the characteristics of the engine-compressor-ducting system, we were able to obtain a simple expression for the heave stiffness, Equation [F-1].

2. With still more sweeping assumptions, we obtained a simple expression for heave damping, Equation [F-2].

3. These results enabled us (with the further assumptions; craft moving over a small-amplitudal sinusoidal-wave sea, traveling normal to wave crests) to represent the heave motion by an ordinary differential equation typical of a simple displacement-excited, damped spring-mass system; and to solve this equation, first in the case $\lambda >> \ell$ (Equation [F-3]); and then in the case of arbitrary $\lambda$ (Equation [F-4]).

4. Certain conclusions were indicated:

(a) The natural frequency in heave is approximately

$$\omega_o \doteq \sqrt{g/h}$$

(This is borne out by experience.)

(b) The nondimensional heave damping coefficient is

$$\zeta \doteq \frac{1}{2} \sqrt{2 \frac{\rho g h}{\Delta p}} \frac{S}{hC}$$

(It is difficult to determine whether or not this is in quantitative agreement with experience; but, at least the effects of the design parameters S/C, h, and $\Delta p$ appear to be qualitatively borne out by experience.)

(c) A resonance effect is predicted, involving relatively high heave response when the encounter frequency $\omega$ is nearly equal to natural frequency $\omega_o$, and a decay in heave response toward zero as $\omega$ increases to still higher values. (This is borne out by experience. However, this resonance effect is of little practical significance, because motions under conditions which, according to our analysis, would produce severe resonance effects are usually, in practical cases, dominated by water-contact effects. Resonance effects are seldom noticeable in currently operating vehicles, for this reason. There are also some indications from experience that "coupling effects" between pitch motion and heave motion can be significant.)

(d) An attenuating effect depending on planform shape and ratio of cushion length to wave length is predicted (Figure F-3), due to the fact that the craft tends to respond to changes in <u>average</u> daylight clearance h, rather than to changes at any one point. (This is, at least qualitatively, borne out by experience.)

REMARKS

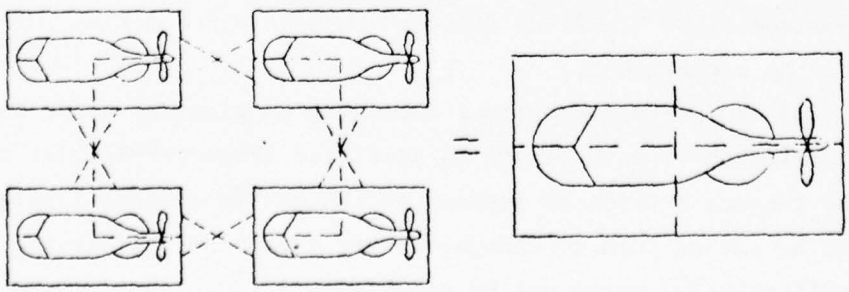
The fact that we have obtained results which are generally consistent with experience provides some encouragement that the main ideas employed in the analysis were more or less valid. It is to be hoped that the analysis itself has been useful to the extent of allowing some of these ideas to be formalized, and some of their consequences to be explored. However, the reader is warned not to expect anything more from this type of analysis. At almost every step of the analysis, we have resorted to simplifying assumptions which far overstep the bounds of what is normally considered to be allowable and justifiable. It would be a dangerous mistake to suppose that the results can be safely applied for design purposes, or as a basis for further, more detailed analyses.

## PITCH AND ROLL MOTIONS WITHOUT WATER CONTACT

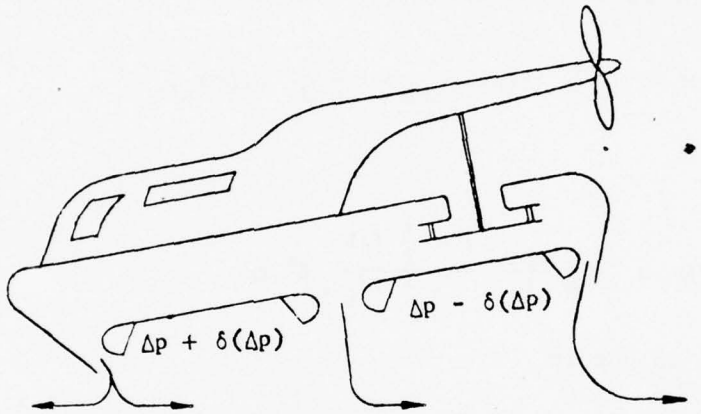
The following discussion will be pursued in as brief and simple a manner as possible, attempting only to convey some of the main ideas involved in pitch and roll motions in the absence of water contact, without attempting to develop these ideas to the point of design application.

### PITCH MOTIONS

The most common form of attitude stabilization is "compartmentation." The underlying idea is this: Suppose one took four individual craft, each of which was stable in heave, and connected them together with a rigid framework. The resulting combination would be stable in pitch and roll, since each individual unit would tend to seek and hold a certain daylight clearance h. A similar result is achieved by introducing



appropriate "barriers" within the cushion of a single craft. These barriers can be vertical air jet curtains, or impervious walls ("baffles") which impede flow of air from one compartment to another.




---

PITCH STIFFNESS -- Suppose the cushion is rectangular, with perfect barriers which divide the cushion into four equal compartments and permit no flow whatever from one compartment to another. Then each compartment is effectively an individual cushion with the same ratio $\frac{S}{hC}$ (measured in level attitude) as the total cushion.

If the vehicle pitches bow-up to a small angle $\alpha$ (radians), the mean daylight clearance of the fore and aft compartments, respectively, is

$$h_{fore} \doteq h + \frac{\frac{\ell}{2}\,\alpha\,\frac{b}{2} + \frac{\ell}{4}\,\alpha\,\frac{\ell}{2}}{\frac{1}{2}\,(b+\ell)}$$

$$= h + \frac{1}{2}\,\frac{1 + \frac{1}{2}\,\ell/b}{1 + \ell/b}\,\ell\,\alpha = h + \delta h$$

$$h_{aft} \doteq h - \frac{1}{2}\,\frac{1 + \frac{1}{2}\,\ell/b}{1 + \ell/b}\,\ell\,\alpha = h - \delta h$$

Applying our previous estimate for heave stiffness we obtain

$$\delta(\Delta p)_{fore} = -\delta(\Delta p)_{aft} = -\frac{\Delta p}{h}\,\delta h$$

The pitching moment is

$$M = \frac{\ell}{4}\frac{S}{2}\,\delta(\Delta p)_{fore} - \frac{\ell}{4}\frac{S}{2}\,\delta(\Delta p)_{aft}$$

or

$$M = -\frac{1}{8}\frac{\Delta p S}{h}\,\frac{1+\frac{1}{2}\,\ell/b}{1+\ell/b}\,\ell^2\,\alpha$$

Assuming $W \doteq \Delta p S$, we obtain

$$\frac{M}{W\ell} \doteq -\frac{1}{8}\,\frac{1+\frac{1}{2}\,\ell/b}{1+\ell/b}\,\frac{\ell}{h}\,\alpha$$

Now, to allow for the fact that the compartmenting barriers are <u>not</u> perfect, let us introduce an empirical "barrier effectiveness factor" $E_b$ and write

$$\frac{M}{W\ell} \doteq -\frac{1}{8}\,E_b\left(\frac{1+\frac{1}{2}\,\ell/b}{1+\ell/b}\right)\frac{\ell}{h}\,\alpha \qquad [F\text{-}5]$$

(It should be pointed out that the heave stiffness estimate introduced in the derivation of Equation [F-5] was based on a qualitative consideration of the compressor-ducting characteristics, which does not apply to the pitch and roll cases, unless each cushion compartment is supplied from a common system. If all compartments are fed from one system, then the appropriate estimate would be based on the "$p_t$ Fixed" curve of Figure F-1 rather than the "Throttle Setting Fixed" curve, and the pitch stiffness could be much less than that given by Equation [F-5].)

PITCH DAMPING -- In the same way that the estimate of heave stiffness was extended to provide an estimate of pitch stiffness, we can extend the estimate of heave damping to provide an estimate of pitch damping. However, that exercise will be omitted, since the principles involved should be obvious, by this time, to the serious reader; and the result is of rather limited practical value for the various reasons discussed earlier.

PITCH NATURAL FREQUENCY -- We will consider pitch motion only to the extent of estimating the natural frequency. This we can do by considering the equation of motion for free oscillation in pitch, neglecting damping:

$$I_y \ddot{\alpha} - \frac{\partial M}{\partial \alpha} \alpha = 0$$

From Equation [F-5] we have

$$\frac{\partial M}{\partial \alpha} \doteq - \frac{1}{8} E_b \left( \frac{1 + \frac{1}{2} \ell/b}{1 + \ell/b} \right) W \frac{\ell^2}{h}$$

Also, we can write the moment of inertia about the lateral axis in the form

$$I_y = \frac{W}{g} \bar{r}_y^2$$

where $\bar{r}_y$ is the "radius of gyration" about the lateral axis. The equation of motion becomes

$$\ddot{\alpha} + \frac{1}{8} E_b \left( \frac{1 + \frac{1}{2} \ell/b}{1 + \ell/b} \right) \left( \frac{\ell}{\bar{r}_y} \right)^2 \frac{g}{h} \alpha = 0$$

and the solution is:

$$\alpha = \text{Constant} \cdot e^{i \omega_{o,\alpha} t}$$

where

$$\omega_{o,\alpha} = \sqrt{\frac{E_b}{8} \frac{1 + \frac{1}{2}\,\ell/b}{1 + \ell/b}} \quad \frac{\ell}{\bar{r}_y} \sqrt{\frac{g}{h}} \qquad [F\text{-}6]$$

(Example - Suppose $E_b = \frac{1}{2}$, $\ell/b = 2$, $\bar{r}_y = \frac{\ell}{3}$ ; then

$$\omega_{o,\alpha} = 0.61 \sqrt{\frac{g}{h}} = 0.61 \quad \omega_o$$

This result is roughly consistent with experience. Most experiments have indicated the natural frequency in pitch to be between half to three-quarters of the natural frequency in heave, $\omega_o$ .)

ROLL MOTIONS

ROLL STIFFNESS -- The development of an expression for roll stiffness follows exactly the above development for pitch stiffness, with the roles of length and beam interchanged. The result can be written down by inspection from Equation [F-5], substituting rolling moment $\mathfrak{L}$ for pitching moment M; roll angle $\phi$ for pitch angle $\alpha$; beam b for length $\ell$, and vice versa.

$$\frac{\mathfrak{L}}{Wb} \doteq - \frac{E_b}{8} \left( \frac{\ell/b + \frac{1}{2}}{\ell/b + 1} \right) \frac{b}{h} \quad \phi \qquad [F\text{-}7]$$

ROLL NATURAL FREQUENCY -- Similarly, the result for natural frequency in roll, $\omega_{o,\phi}$ , can be written down by inspection from Equation [F-6].

$$\omega_{o,\phi} \doteq \sqrt{\frac{E_b}{8} \left( \frac{\ell/b + \frac{1}{2}}{\ell/b + 1} \right)} \quad \frac{b}{\bar{r}_x} \sqrt{\frac{g}{h}} \qquad [F\text{-}8]$$

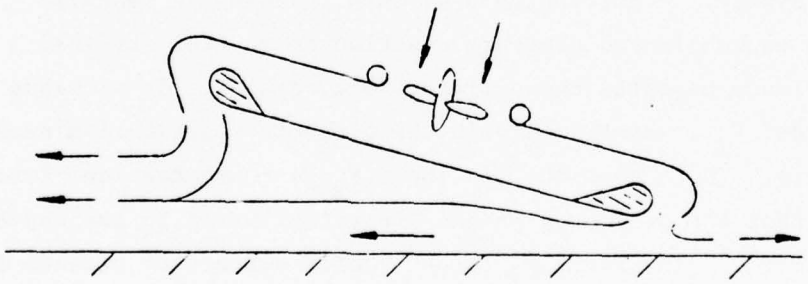
ATTITUDE STABILITY WITHOUT COMPARTMENTATION

The modern design trend for air cushion vehicles is toward extremely small daylight clearance, accepting some degree of water contact at practically all times. Under these conditions, it becomes comparatively unimportant precisely what degrees of pitch and roll stiffness the cushion provides in the absence of water contact. A

serious question arises as to whether the design complications and other penalties associated with compartmenting jets or baffles are really justified.

On the other hand, it is certainly most undesirable to have the craft become actively unstable in pitch and roll, in those rare instances when the craft might operate free of contact with the surface. The craft might become practically unmanageable under such circumstances unless positive, fast-acting pitch and roll controls were provided - which in themselves are usually unwanted design complications.

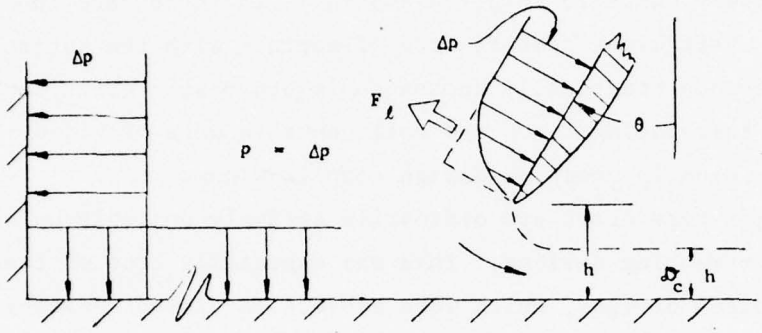
Peripheral jet type craft are ordinarily actively unstable unless fitted with compartmenting devices. This was especially true of the old-style flat-bottom designs, which were subject to violent destabilizing effects of the "cross flow" of air through the cushion from the low-side jet toward the high side. Under certain conditions, a "venturi throat" type of flow could establish itself along the low side of the base, forming a low-pressure region which sucked the low side down to the surface.



The modern peripheral-jet design trend toward deep flexible trunks, with deeply recessed base in between, should presumably greatly alleviate these cross-flow effects, but apparently compartmentation is still required to avoid some degree of attitude instability.

Plenum type craft, however, usually exhibit a reasonably acceptable degree of pitch and roll stiffness, even if compartmenting devices are omitted from the design. There are several phenomena which can contribute to attitude stability of plenum type craft, but we will consider only the most important one, which involves pressure forces acting near the lower edge of the plenum wall.

Consider the flow discharge from an infinite plenum chamber through a sharp-edged orifice:




The efflux through the orifice of height h "necks down" to a vena contracta of height $\mathcal{C}_c$ h . It is easy to show that there must be a reaction force per unit length of the plenum of 2 $\Delta p$ $\mathcal{C}_c$ h on the whole plenum system. A portion $\Delta p$ h of this reaction is readily identified as an unbalanced pressure reaction on the far vertical wall of the plenum opposite the orifice. The remainder is ascribed to a "lip force" $F_\ell$ , identified with the pressure distribution near the orifice lip. (Note that the lip force $F_\ell$ is a hypothetical force in the sense that the pressures on the lip actually act in the opposite direction from $F_\ell$ . The Force $F_\ell$ is the vector difference between the actual lip force and the lip force which would act if the pressure were uniformly equal to $\Delta p$.) The magnitude of $F_\ell$ is determined from

$$\frac{F_\ell}{\ell} \cos\theta + \Delta p\, h = 2\, \Delta p\, \mathcal{C}_c\, h$$

or

$$\frac{F_\ell}{\ell} = \Delta p\, h\, \frac{2\,\mathcal{C}_c - 1}{\cos\theta}$$

where $\ell$ , in the present context, is the length (normal to the plane of the sketch) of the plenum.

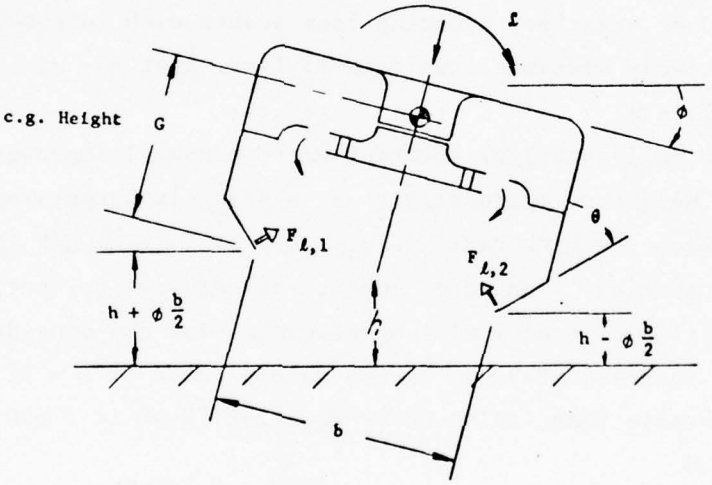
Using the relation from Section B:

$$\mathcal{D}_c \doteq \frac{1}{2}\left[1 + \frac{\cos\theta}{\frac{\pi+2}{\pi-2}(1+\sin\theta) - \sin\theta\cos\theta}\right]$$

we get

$$\frac{F_\ell}{\ell} \doteq \frac{\Delta p\, h}{\frac{\pi+2}{\pi-2}(1+\sin\theta) - \sin\theta\cos\theta}$$

We are now in position to calculate the roll stiffness of a plenum type vehicle with $\ell/b \gg 1$.




Referring to the sketch, we have

$$£ = -\left(F_{\ell,1} - F_{\ell,2}\right) G\cos\theta + \left(F_{\ell,1} - F_{\ell,2}\right)\frac{b}{2}\sin\theta$$

$$F_{\ell,1} \doteq \Delta p\left(h + \phi\frac{b}{2}\right)\frac{\ell}{\frac{\pi+2}{\pi-2}(1+\sin\theta) - \sin\theta\cos\theta}$$

$$F_{\ell,2} \doteq \Delta p\left(h - \phi\frac{b}{2}\right)\frac{\ell}{\frac{\pi+2}{\pi-2}(1+\sin\theta) - \sin\theta\cos\theta}$$

$$F_{\ell,2} - F_{\ell,1} = \frac{\Delta p \, \phi \, b \, \ell}{\frac{\pi + 2}{\pi - 2}(1 + \sin\theta) - \sin\theta \cos\theta}$$

or, if $W \doteq \Delta p \, b \, \ell$

$$\frac{\mathcal{L}}{Wb} \doteq - \frac{\frac{G}{b}\cos\theta - \frac{1}{2}\sin\theta}{\frac{\pi + 2}{\pi - 2}(1 + \sin\theta) - \sin\theta\cos\theta} \phi \quad , \quad \ell/b \gg 1 \qquad [F\text{-}9]$$

This relationship is illustrated in Figure F-4. According to this formula, the plenum type craft will always have stable roll stiffness when $\theta$ is zero or negative, becoming less stable with increasing $\theta$ until it becomes actively unstable when $\theta$ is so large that the vectors $F_\ell$ point above the c.g.

Note that roll stability (without water contact) improves with increasing c.g. height or decreasing beam, a slightly surprising result, perhaps. (However, a high ratio of c.g. height to beam G/b is very dangerous in practice, when water contact effects are present.)

When vehicles of practical length/beam ratios are considered, it is found that Equation [F-9] is approximately valid if $\theta = 0$. The craft will be less stable than indicated by Equation [F-9] if $\theta > 0$, more stable if $\theta < 0$.

When pitch is considered, a result for the case $b/\ell \ll 1$ can be written down by inspection from Equation [F-9] (replacing $\mathcal{L}$ with M and b with $\ell$); but this case is of little practical interest. However, we can write down a useful formula for the isolated case $\theta = 0$, in which case

$$\frac{M}{W\ell} \doteq - \frac{\pi - 2}{\pi + 2}\frac{G}{\ell} \phi \quad , \quad \theta = 0° \qquad [F\text{-}10]$$

The above developments have been based on the assumption that the walls of the plenum are sufficiently deep compared to the gap h that the pressure drop occurring near the lip is felt almost entirely on

the walls, and not on the base of the vehicle. If the walls are shallow compared to the gap h, so that the pressure drop is felt on the base, then the vehicle can be much more stable; but of course there is a corresponding loss of efficiency of the plenum in providing lift.

We saw in Section B that there is no clear line of demarcation between the aerodynamics of peripheral jet craft and plenum craft. The latter can be thought of as simply a form of the former, but with very thick jets.

By the same token, it must be realized that the same considerations which led to Equations [F-9] and [F-10] for plenum craft, apply also (in modified form) to ordinary peripheral-jet craft, becoming more and more important as the nozzle thickness parameter, $x = \frac{t}{h}(1+\sin\theta)$, is increased.

REMARKS

The remarks made at the end of the study of heave motions apply equally here; that is, the derivations of formulas concerning pitch and roll motions were intended primarily as a means of formalizing some main ideas, and exploring some of their consequences. The formulas are not likely to be reliable for design application.

It is even more important not to lose sight of the following fact: The whole of this section has concerned itself with motions in the absence of water contact. Modern design trends seem to point toward practically all types of GEM's operating in intimate contact with the water (or whatever surface they are traveling over) at practically all times. The contributions of the cushion to stability (except, perhaps, to heave stability) thus become much less significant than the forces and moments imposed by contact between the water (or other surface) and the trunks, sidewalls, seals, hull, or whatever portions of the vehicle experience the contact.

These extremely important effects are discussed (not adequately, by any means, but to the best of the authors' ability) in a later section.

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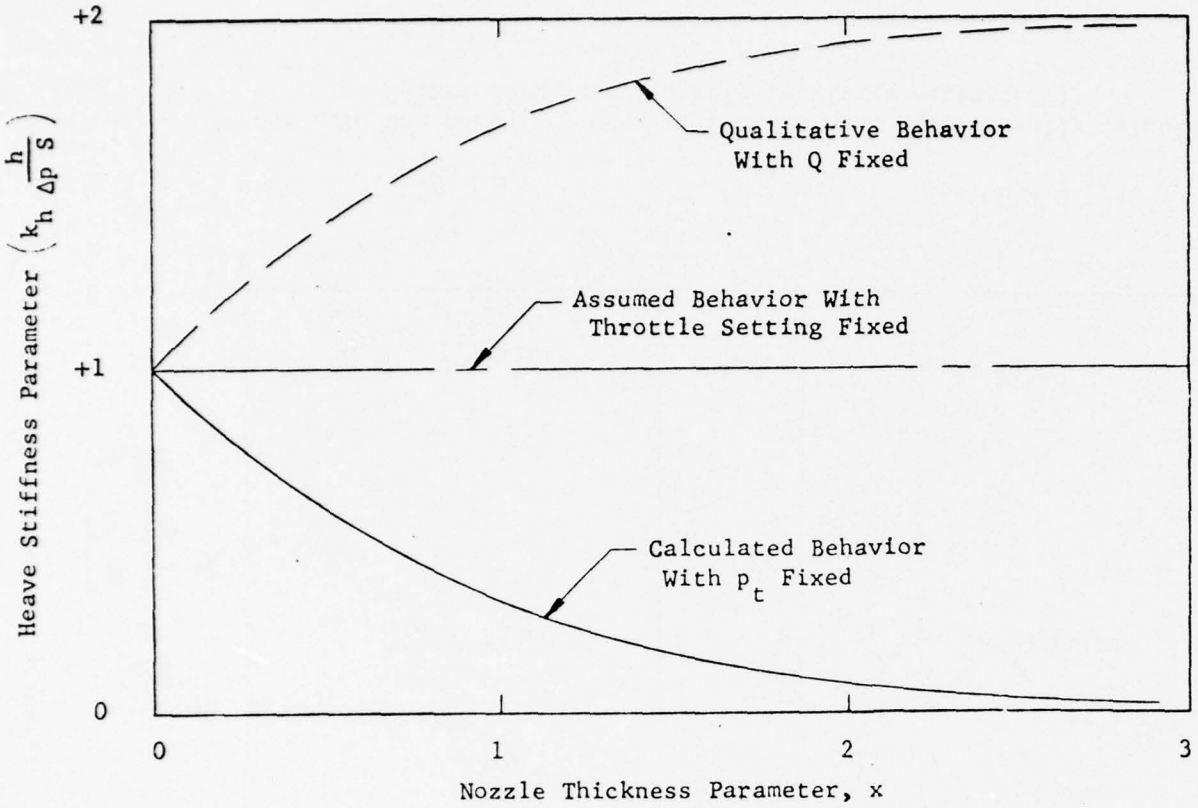
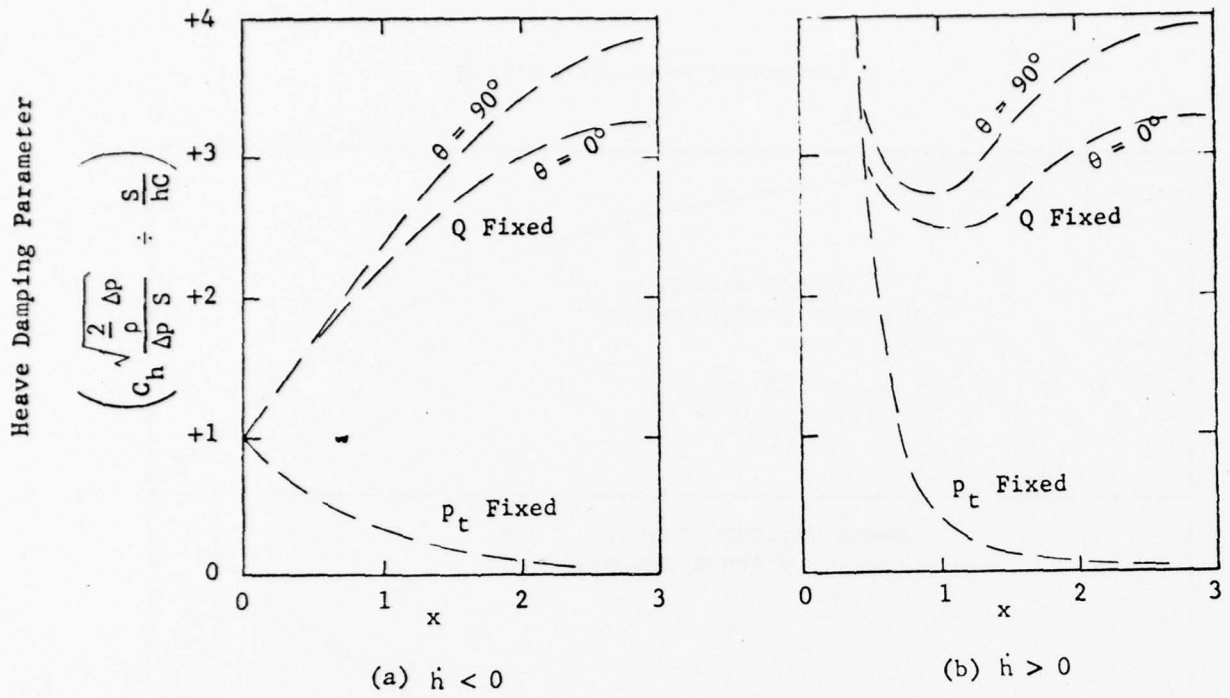
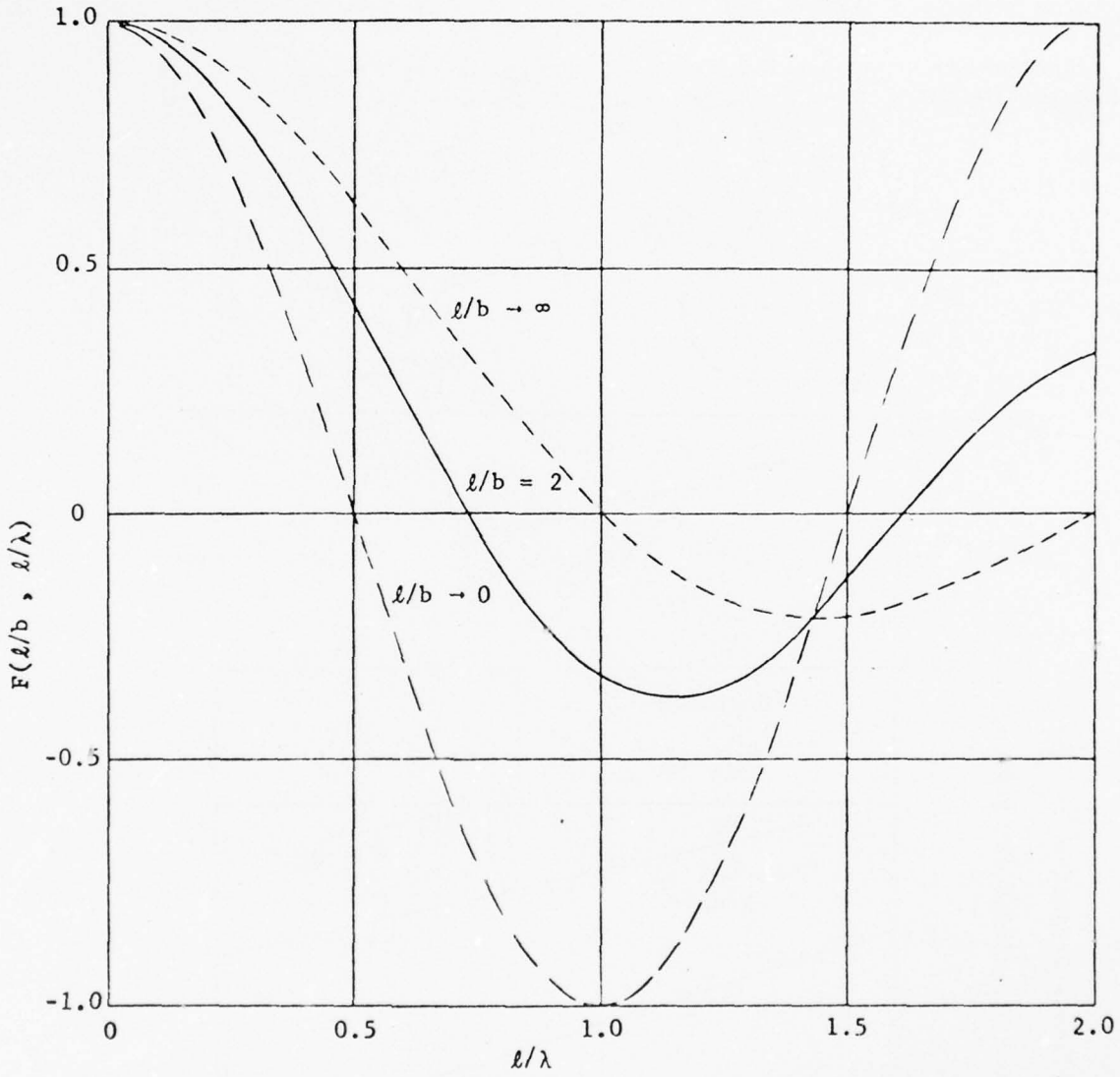



Figure F-1 - Estimation of the Heave Stiffness



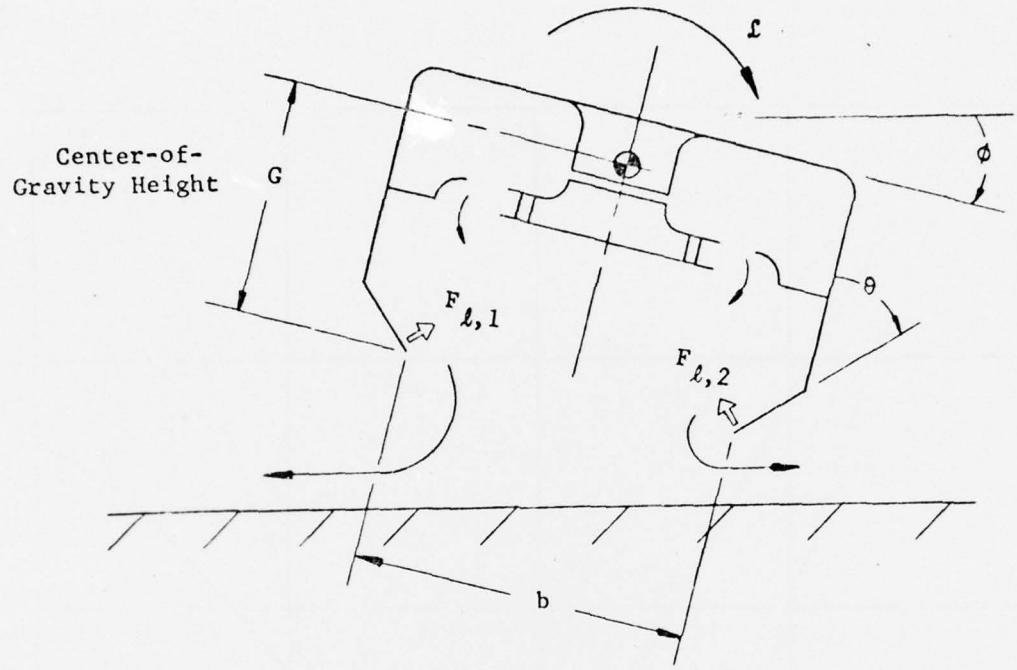

(Note: Only the behaviors at $x \ll 1$ and $x \gg 1$ were actually calculated. The dashed curves are largely intuitive estimates.)

Figure F-2 - Estimation of Heave Damping




(Change of sign of F implies 180° change of of phase between wave and motion.)

Figure F-3 - Heave Attenuation Factor Due to Finite Wave Length for a Rectangular Cushion



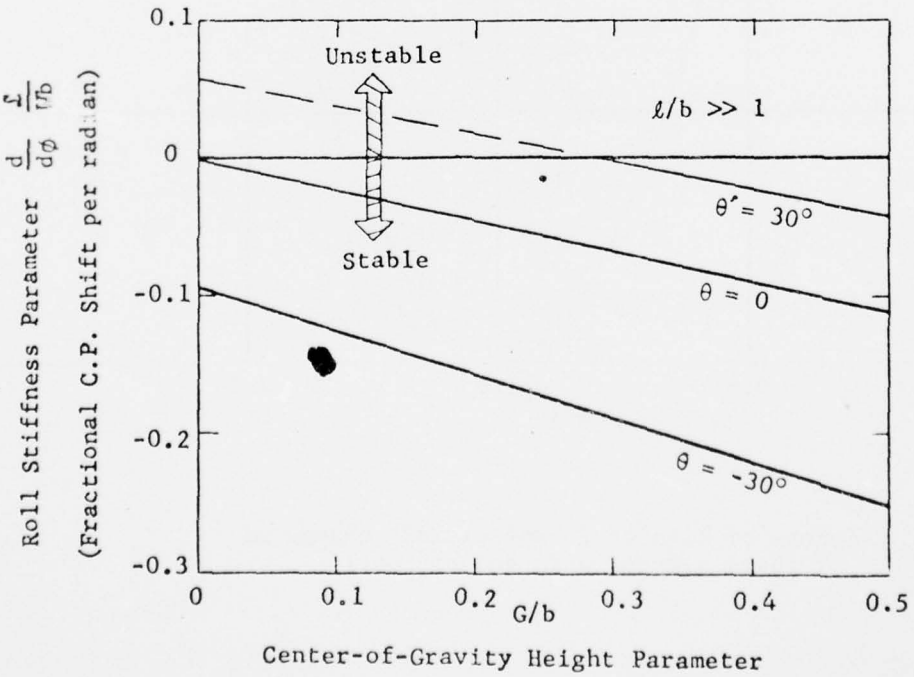




Figure F-4 - Roll Stiffness for a Plenum Type Craft Without Compartmentation

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13 ABSTRACT

A review is undertaken of some of the more important elementary relationships which determine the influence of the air cushion on the pitch, heave, and roll stability of ground effect machines. The analyses presented yield only rough approximations of the quantities needed for design. Further, the analyses are restricted to cases of vehicles which are not in contact with the surfaces over which they travel, whereas the influences of surface contact are of primary importance in most practical problems.

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# SOME DESIGN PRINCIPLES OF GROUND EFFECT MACHINES
## SECTION G — SEAKEEPING

by

Harvey R. Chaplin and Allen G. Ford

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July 1966

Report 2121G
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## Foreword

This report is based on a lecture series presented by the authors at the von Kármán Institute for Fluid Dynamics, Rhode-Saint-Genèse, Belgium, in May 1965; and at the University of Maryland, College Park, Maryland, in July 1965. The lectures were prepared under the joint auspices of the David Taylor Model Basin and the Naval Air Development Center. They were presented in Belgium under the joint sponsorship of the von Kármán Institute and the Advisory Group for Aerospace Research and Development (AGARD); and in Maryland under the sponsorship of the Assistant Secretary of the Navy for Research and Development.

The revised lectures will be presented as follows:

A. Introductory Survey
B. Air Cushion Mechanics
C. Internal Aerodynamics
D. Drag
~~E. Drag Optimization for Sidewall CEM (CAB)~~
F. Cushion Contributions to Stability
G. Seakeeping
H. Performance Summary

## NOTATION

| | |
|---|---|
| $\bar{h}$ | local wave elevation, feet |
| $\bar{\bar{h}}$ | amplitude of sinusoidal wave component, feet |
| $\mu$ | stationary frequency of sinusoidal wave component, radians/sec (frequency with which wave propagates past a fixed point) |
| $\omega$ | frequency of encounter of sinusoidal wave component, radians/sec (frequency with which wave is encountered by a moving point) |
| $\lambda$ | wave length, feet |
| $x$ | distance along horizontal axis in direction opposite to wave propagation, feet |
| $g$ | acceleration of gravity, ft/sec$^2$ |
| $c$ | velocity of wave propagation, ft/sec |
| $\rho_w$ | mass density of water, slugs/ft$^3$ |
| $i$ | arbitrary integer |
| $m_o$ | mean-square of $\bar{h}$, ft$^2$ |
| $V_w$ | wind velocity, knots |
| $[A(\mu)]^2$ | spectral density of wave system, as function of stationary frequency of components, ft$^2$-sec |
| $[A(\omega)]^2$ | spectral density of wave system as function of frequency of encounter of components, ft$^2$-sec |
| $H$ | wave height of composite wave, feet |
| $H_{av}$ | mean of all values of H |
| $H_{1/3}$ | mean of the highest one-third values of H |
| $H_{1/10}$ | mean of the highest one-tenth values of H |
| $V_o$ | velocity of moving point (craft), ft/sec; taken positive in direction opposite to wave propagation |
| $\alpha$ | pitch angle, radians |
| $y$ | heave displacement, feet |

| | |
|---|---|
| $M_\alpha$ | pitch magnification factor for linear response to a sinusoidal wave |
| $M_y$ | heave magnification factor for linear response to a sinusoidal wave |
| $m_{o\alpha}$ | mean-square value of $\alpha$, radians$^2$ |
| $m_{oy}$ | mean-square value of y, ft$^2$ |
| S | cushion area, ft$^2$ |
| $\ell$ | cushion length, feet |
| b | cushion beam, feet |
| $\tilde{\ell}$ | mean cushion length (S/b), feet |

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SUMMARY

The analytical and experimental tools available for application to GEM seakeeping problems are reviewed briefly. The one available set of seakeeping data for a full-scale GEM is reviewed, and is found to constitute encouraging evidence that GEM seakeeping problems will be tractable.

## INTRODUCTION

The term "seakeeping" refers to the riding qualities of a vehicle which travels on a "seaway"; that is, in the ocean-surface environment. A vehicle with good seakeeping qualities moves through or over the ocean waves with a minimum of the violent motions which would cause discomfort to the passengers and interfere with the functions of crew and equipment, a minimum of the structural loads which require additional investment in structural weight at the cost of payload, and a minimum of the additional resistance which causes degradation of speed and range.

The question of the seakeeping characteristics of Ground Effect Machines is probably the most crucial of the questions which will affect the future economic and/or military importance of these vehicles. It is not a simple question, of course, and one cannot expect a simple answer. Even for displacement ships, the progress toward scientific methods of predicting the seakeeping qualities of a given hull form has been comparatively recent; and much still remains to be done in the problem of taking proper account of seakeeping in the initial design. The science of ship design is one of the oldest sciences, yet still depends, in many aspects, on gradual evolution and the "designer's eye."

The very short history of Ground Effect Machines has permitted only the beginnings of an evolutionary process, and only a very few designers (mostly in the British Hovercraft projects) can lay claim to even the most rudimentary bases for development of a "designer's eye." Nevertheless, this short history has already produced encouraging evidence that the seakeeping problems of GEM's will be found tractable, to a degree that few would have dared to hope, before. This early evidence

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of tractability pertains largely to the remarkable cushioning effects of the air cushion itself, plus the remarkable early successes in the development of flexible understructure.

Unfortunately, the important information concerning seakeeping of GEM's has usually found its way into the literature in very fragmentary forms. Despite the predominant importance of this subject, the following discussion will have to be limited to a very brief review of the most significant of the available mathematical methods, plus a rather superficial examination of the most readily accessible experimental evidence.

## DESCRIPTION OF NATURAL WAVES

The following material is intended as an introduction to the simpler aspects of this complex subject, for benefit of those readers who are unfamiliar with the problems of mathematical description of the sea.

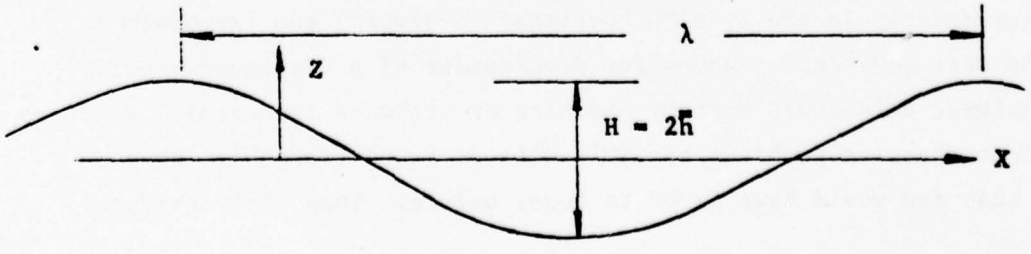
Unless otherwise stated, this material is based on Vossers' "Fundamentals of the Behavior of Ships in Waves" (Reference 1).

### SIMPLE WAVES

The simplest non-trivial solution for motion of the surface of a body of water is the one-dimensional sinusoidal wave propagating in the (negative, by choice) direction of the x-axis. The wave elevation measured from the mean is

$$\bar{h} = \bar{\bar{h}} \sin\left(\frac{2\pi}{\lambda} x + \mu t\right)$$

where $\bar{\bar{h}}$ is the amplitude of the wave (half the wave height H), $\lambda$ is the wavelength, and $\mu$ is the frequency (in radians/sec) with which the wave propagates past a fixed point.



If the wave "steepness" is sufficiently small $\left(\frac{H}{\lambda} \leq \frac{1}{20}\right)$, and the water sufficiently deep $\left(\text{depth} \geq \frac{\lambda}{2}\right)$, the frequency is given by

$$\mu = \sqrt{2\pi g/\lambda} \quad , \quad \text{radians/sec}$$

The velocity of propagation is obtained from

$$\frac{c}{\lambda} = \frac{\mu}{2\pi}$$

or

$$c = \sqrt{\frac{g\lambda}{2\pi}} = \frac{g}{\mu}$$

and the energy stored in the wave motion, per unit area of the sea surface, is

$$E = \frac{1}{2} \rho_w \bar{h}^2$$

(This is, in other words, the energy which would be required to establish the wave motion, by a conservative process.)

Natural waves are not precisely sinusoidal. The trochoidal wave form is used for some purposes, as a somewhat better approximation of natural wave form. However, the deviations from sinusoidal are very slight, except in cases of extremely steep waves, and the sinusoidal representation is far more convenient for purposes of mathematical analysis.

IRREGULAR WAVES

The sea displays many characters. Some of the common terminologies are:

Wind Sea — A sea which, formerly calm, is building up under the action of winds; characterized by a high degree of irregularity.

Swell — A sea in process of subsiding, in relatively calm wind; comparatively regular, often comparatively long-crested.

Fully Arisen Sea — A sea which has been exposed to a more or less constant wind condition for some time, and has reached a condition of equilibrium between addition of energy from the wind and dissipation of energy from the viscous action of the water.

Long-Crested Sea — Waves all propagating in generally the same direction, so that relatively long, parallel, unbroken crests are visible. (Sometimes used to denote mathematically one-dimensional waves, which, of course, do not occur from natural causes.)

Short-Crested Sea (Confused Sea) — Waves propagating in a variety of directions, so that crests are short, broken, and distinctly three-dimensional in nature.

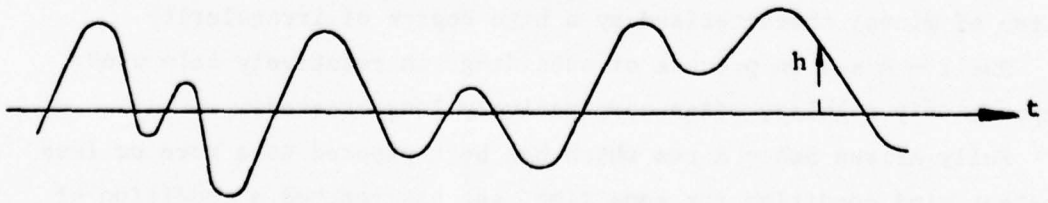
The most fundamental property of the sea's surface is randomness. In principle, it would be possible to mathematically reproduce the point-by-point motion of any finite area of ocean surface over any finite span of time. The futility of doing so is apparent upon reflecting that this particular motion will not occur again, anywhere, in a million years. However, certain statistical properties of the sea (such as the mean wave height, the relationship between the mean height and mean frequency, etc.) are repeated; and it is the statistical properties which we must seek to represent and understand.

The statistical properties of virtually any sea condition can be satisfactorily represented by superposition of an infinite number of simple sinusoidal waves, each of infinitesimal amplitude, propagating in a certain random distribution of directions with random phase.

For one-dimensional (uni-directional) waves, this representation can be expressed

$$\bar{h} = \sum_{i=1}^{N} \bar{\bar{h}}_i \sin\left[\frac{2\pi}{\lambda_i} x + \mu_i t + \epsilon_i\right]$$

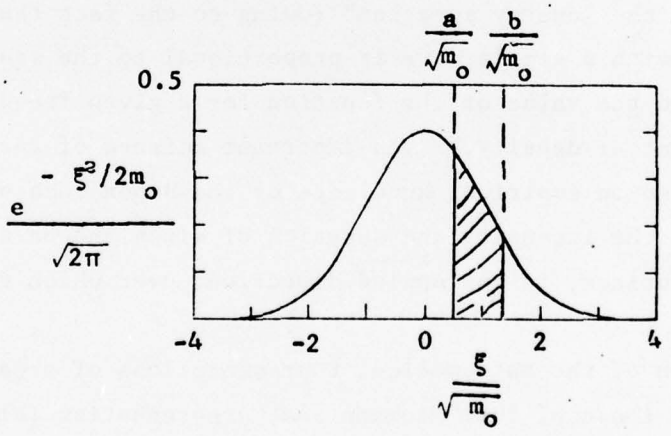
where N is an arbitrarily large number, and where the phase angle $\epsilon_i$ is random, with uniform distribution over the interval $0 < \epsilon(\mu) < 2\pi$. (It might seem natural at first glance to let $N \to \infty$ and replace the




Time History of an Irregular Wave

summation by an integral; but if this is attempted by elementary methods, the resulting integral is found to be indeterminate.) Since each component, owing to the random phase, is a random quantity, the sum $\bar{h}$, for any given value of x and t, is also a random variable, of mean value zero; and, according to the central limit theorem of probability theory, has a normal (Gaussian) distribution. That is, for arbitrary choice of x and t, the probability that $\bar{h}$ will lie between values a and b is

$$P\ (a < \bar{h} < b) = \frac{1}{\sqrt{2\pi\, m_o}} \int_a^b e^{-\xi^2/2m_o}\, d\xi$$




where $m_o$ is the variance of the random variable $\bar{h}$; or, since the wave elevation $\bar{h}$ has a mean value of zero, $m_o$ is the mean-square value of $\bar{h}$. Since the mean-square amplitude of the sum of a series of sine waves of different frequencies is half the sum of squares of the component amplitudes, we have

$$m_o = \frac{1}{2} \sum_{i=1}^{N} \bar{h}_i^2$$

Now, there is no difficulty in converting the summation of squared amplitudes to integral form. Lettering $N \to \infty$, we define

$$[A(\mu)]^2 \, d\mu = \sum_{d\mu} \bar{h}_i^2$$

where the summation is carried out for all components $\bar{h}_i$ in the frequency interval from $\mu$ to $\mu + d\mu$. Then

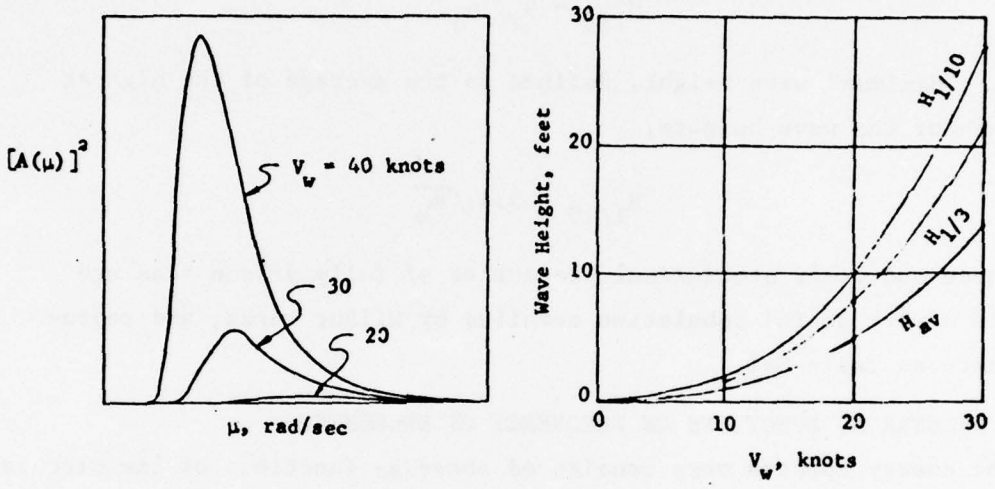
$$m_o = \frac{1}{2} \int_o^\infty [A(\mu)]^2 \, d\mu$$

Evidently, a great deal of information is available, concerning the character of the sea, if the function $[A(\mu)]^2$ is known. This function is called the "energy spectrum" (owing to the fact that the energy associated with a simple wave is proportional to the square of its amplitude), and the value of the function for a given frequency is called the "spectral density." The important science of forecasting ocean waves is based on empirical knowledge of the dependence of the energy spectrum on the intensity and duration of winds and on the "fetch" (length of ocean surface, in the upwind direction, over which the wind prevails).

The best-known of the mathematical representations of ocean energy spectra are those of the "Neumann sea," representing fully arisen seas corresponding to arbitrary wind velocities $V_w$, in knots

$$[A(\mu)]^2 = \frac{51.6}{\mu^6} \, e^{-2g^2/(V_w^2 \mu^2)} \quad , \quad \text{ft}^2\text{-sec}$$

$$m_o = 0.121 \left(\frac{V_w}{10}\right)^5 \quad , \quad \text{ft}^2$$




These Neumann spectra are semi-empirical representations of time-histories of the wave action at a fixed point in a real fully arisen sea, which is, of course, not one-dimensional. They are frequently used to represent one-dimensional seas, however, which in turn are assumed to be representative (after proper adjustments for the speed) of the sea "sensed" by a craft moving with or against the dominant direction of wave propagation.

With a knowledge of the energy spectrum, and knowledge that the distribution of wave elevations is Gaussian, numerous statistical properties can be calculated. For the Neumann spectra, for example, it is found that:

a. Average wave height, measured between successive crest and trough,

$$H_{av} = 2.5\sqrt{m_o}$$

(Since $\sqrt{m_o}$ is the root-mean-square value of the wave elevation $\bar{h}$, one would correctly guess that this should be about half the average crest-to-trough dimension $H_{av}$ .)

b. "Significant wave height," defined as the average of the highest one-third of the wave heights,

$$H_{1/3} = 4\sqrt{m_o}$$

c. "Maximum" wave height, defined as the average of the highest one-tenth of the wave heights,

$$H_{1/10} = 5.1\sqrt{m_o}$$

These and other statistical properties of fully arisen seas are given in a very useful tabulation compiled by Wilbur Marks, and reproduced here as Table G-1.

ENERGY SPECTRA AS FUNCTIONS OF FREQUENCY OF ENCOUNTER

The energy spectra were considered above as functions of the circular frequencies $\mu$ with which the simple wave components propagate past a fixed point.

A simple wave of frequency $\mu$ propagates with velocity $c = \frac{g}{\mu}$. Therefore, a point moving normal to the crests at velocity $V_o$ encounters this wave at frequency

$$\omega = \mu \frac{c + V_o}{c} = \mu\left(1 + \mu \frac{V_o}{g}\right)$$

Where $V_o$ is considered positive for a point moving in the opposite direction to wave propagation (head sea) and negative for a point moving in the same direction as wave propagation (following sea).

Now we can express the energy spectrum $[A(\mu)]^2$ in terms of frequency of encounter $\omega$, taking care that the "energy" contained in the frequency interval $\omega$ to $\omega + d\omega$ is the same as that in the corresponding interval $\mu$ to $\mu + d\mu$.

$$[A(\mu)]^2 \, d\mu = [A(\omega)]^2 \, d\omega$$

$$= [A(\omega)]^2 \left(1 + 2\mu \frac{V_o}{g}\right) d\mu$$

or

$$[A(\omega)]^2 = \frac{[A(\mu)]^2}{1 + 2\mu V_o/g}$$

$$\omega = \mu (1 + \mu V_o/g)$$

In the case of a head sea $V_o > 0$ , the energy is spread out over a wider band of higher frequencies than for the stationary-point spectrum; whereas for following sea, at moderate negative velocities, energy tends to be concentrated in narrower bands at low frequency, the spectral density becoming infinite at the frequency of encounter

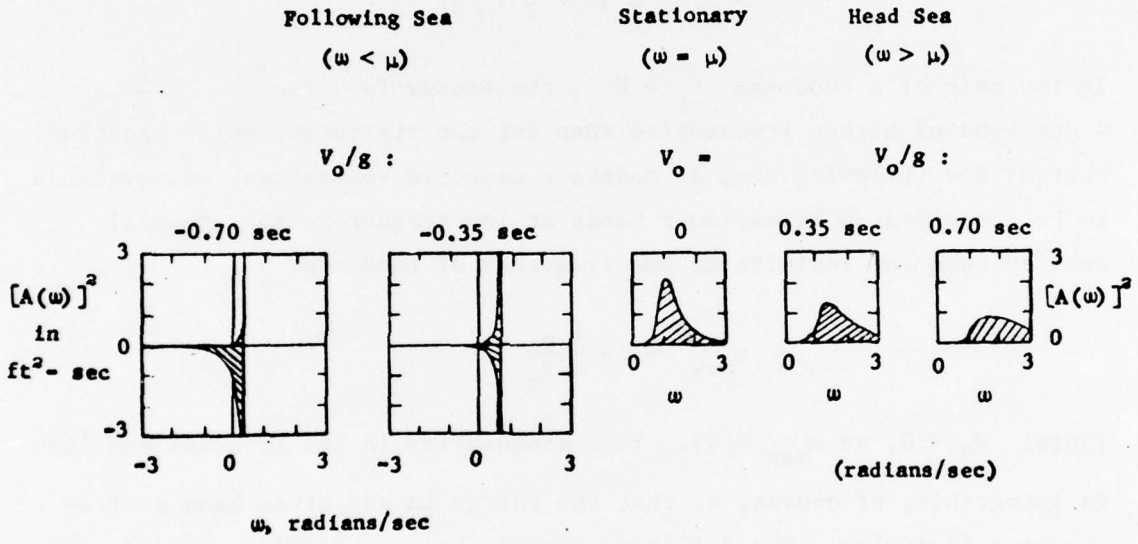
$$\omega_{max} = -\frac{1}{4}\frac{g}{V_o}$$

(Note: $V_o < 0$, so $\omega_{max} > 0$). This singularity in the spectral density is integrable, of course, so that the energy in any given band of frequencies is finite. The following-sea spectra are double-valued in the range of frequencies for which it is possible to encounter waves of two wavelengths at the same frequency of encounter, and extend into the "negative frequency" range. (The physical interpretation of "negative frequency" for the following sea is merely that the waves are propagating with velocity $c < |V_o|$ , and are being overtaken by the moving point. The negative spectral density is seen to pose no problem when it is considered that, in integrating the energy content in any frequency band, a path of increasing values of $\mu$ is followed, so that the product $[A(\omega)]^2\, d\omega$ is always positive. For example, one can write

$$m_o = \frac{1}{2}\int_o^\infty [A(\mu)]^2 d\mu$$

$$= \frac{1}{2}\int_o^{\omega_{max}} [A(\omega)]^2\, d\omega + \frac{1}{2}\int_{\omega_{max}}^{-\infty} [A(\omega)]^2\, d\omega$$

In the final form of the equation, the first integration is carried out from left to right in the upper half of the plane $\left([A(\omega)]^2\right)$ versus $\omega$ ; and the second is carried out from right to left in the lower half plane.)




Spectral Density Versus Frequency of Encounter
(State 3 Sea. One-Dimensional Idealization)

Some examples are given in the above sketch for a state 3 sea (see Table G-1) and for speeds appropriate to slow surface vessels. It should be pointed out that when one goes to much higher speeds, there is little practical difference between head and following sea. The respective frequency spectra resemble reflections, about the origin, of one another. The following-sea spectrum still has the singular "spikes" near the origin, of course; but they contain practically no energy.

## PREDICTION OF MOTIONS BY LINEAR SUPERPOSITION

Consider the motions of a surface craft traveling normal to the wave crests of a simple sinusoidal wave system. Ordinarily, we expect oscillations in pitch ($\alpha$) and heave (y) to be induced by passage over the wave.

If, for arbitrary wave length and frequency of encounter, (1) the oscillations in pitch and heave are sinusoidal, and (2) the amplitudes of oscillation are directly proportional to the amplitude of the wave, then it is possible to represent the response to an irregular wave as the sum of the responses to an arbitrarily large number of simple waves of which the irregular wave is assumed to be composed.

In other words, if the elevation of the simple wave is expressed:

$$\bar{h} = \bar{\bar{h}} \sin \omega t$$

then, in order for linear superposition to be valid, the motions must be expressible as follows:

$$\alpha l = M_\alpha \bar{\bar{h}} \sin \left(\omega t + \phi_\alpha\right)$$

$$y = M_y \bar{\bar{h}} \sin \left(\omega t + \phi_y\right)$$

where the "magnification factors" $M_\alpha$ and $M_y$ and relative phase angles $\phi_\alpha$ and $\phi_y$ may depend on any or all of the variables

$V_o$ craft velocity

$\lambda$ wavelength

$\omega$ frequency of encounter

but must not depend on the wave amplitude $\bar{\bar{h}}$. (The pitch and heave motions need not be independent; that is, heave displacement may induce pitching moment, etc., so long as the above-mentioned conditions are met.)

In this case, if we denote the mean-square amplitudes of wave elevation, pitch angle, and heave displacement as: $m_o$ (feet$^2$), $m_{o\alpha}$ (radian$^2$),

and $m_{oy}$ (feet$^2$), then for a given craft moving at given velocity $V_o$ over an irregular wave composed of N simple-wave components, we have

$$m_o = \frac{1}{2} \sum_{i=1}^{N} \bar{h}_i^2$$

$$m_{o\alpha} = \frac{1}{2\ell^2} \sum_{i=1}^{N} \left(M_\alpha\right)_i^2 \bar{h}_i^2$$

$$m_{oy} = \frac{1}{2} \sum_{i=1}^{N} \left(M_y\right)_i^2 \bar{h}_i^2$$

Note that, for given velocity $V_o$, the magnification factors can be considered to be functions of $\omega$ only (even though they may depend physically upon $\lambda$), since (given $V_o$ and $\omega$), $\lambda$ is determined. It is determined uniquely, in fact, provided one properly identifies which of the two possible values of $\lambda$ applies in cases of following sea, positive $\omega$. (Refer to the preceding discussion and sketch, concerning expression of the energy spectrum of the sea as a function of frequency of encounter (page 10).

The sums can be converted to integrals, letting $N \to \infty$. For head sea:

$$m_o = \frac{1}{2} \int_o^\infty \left[A(\omega)\right]^2 d\omega$$

$$m_{o\alpha} = \frac{1}{2\ell^2} \int_o^\infty \left[M_\alpha(\omega)\right]^2 \left[A(\omega)\right]^2 d\omega$$

$$m_{oy} = \frac{1}{2} \int_o^\infty \left[M_y(\omega)\right]^2 \left[A(\omega)\right]^2 d\omega$$

(We have already considered the expression for $m_o$, in the case of a following sea, as the sum of two integrals. The expressions for $m_{o\alpha}$ and $m_{oy}$ are directly analogous. However, we will confine the following discussion to the case of a head sea.)

The functions $\left\{\left[M_\alpha(\omega)\right]^2\left[A(\omega)\right]^2\right\}$ and $\left\{\left[M_y(\omega)\right]^2\left[A(\omega)\right]^2\right\}$ are called the energy spectra of the pitch and heave motions. They are obtained by the simple expedient of multiplying the energy spectrum of the sea by the squares of the respective magnification factors. From them, it is possible to determine innumerable useful statistical properties of the motions (average amplitude of oscillation, average of the one-third highest oscillations, probability of exceeding a certain displacement in a certain time period, etc.). (Note that, regardless of what phase relationships may exist between the elementary sinusoidal components of the wave and the corresponding elementary sinusoidal components of the motions, if the wave components have random phase, then the motion components will have random phase. Therefore, the composite motion displacements will have Gaussian distribution, with variance $m_{o\alpha}$ and $m_{oy}$, respectively.)

Two problems remain:

1. Justification of the assumption that the responses to simple waves are linear (i.e., sinusoidal with amplitudes proportional to wave amplitude).

2. Determination of the magnification factors.

The fact is that the assumption of linear response is virtually never rigorously justifiable in the cases of greatest practical interest, which are, of course, the cases of severe motions. Quite the contrary, one usually knows very well that the responses are not linear in cases of large-amplitude motions. Nevertheless, there is little practical recourse but to make the assumption anyway. The real justification is made on the practical grounds that the results have usually been found to agree sufficiently well with experience to be extremely useful.

Our confidence in the method, however, must be kept somewhat in proportion to the extent of the closely related successful experience which has been accumulated. The experience with energy-spectral analysis of Ground Effect Machine motions has certainly been extremely limited, at best, and judgment must be reserved.

GEM's with significant air gaps appear, at first glance, to be quite unsuited to this type of analysis because of the obvious major differences between the character of small-amplitude motions, with no water contact, and motions of larger amplitude with frequent water contact. On the other hand, in severe wave conditions the water contact may be nearly continuous, and it is not unlikely that useful energy-spectral analyses of such motions can be made, if appropriate approximations for the magnification factors can be established. GEM's of the Captured Air Bubble type, or other types which operate without significant air gaps, would seem to lend themselves very readily to this type of analysis. However, this also remains to be seen.

Magnification factors are most frequently determined from calculations, from towing-tank tests performed in regular waves, or both. The calculations can be made fairly successfully for simple displacement hull forms, but the towing tank is relied upon for less conventional types of vehicles, and even for more sophisticated forms of displacement hulls.

In Section F, simplified calculations are carried out for the heave magnification factor for a full-peripheral craft with no water contact. In the case $\frac{\ell}{b} >> 1$ Equations [F-3a] and [F-3b] are derived and then a formula is derived for an attenuation factor (Figure F-3), allowing for effects of finite wave length. Those results are of limited practical interest because: (a) the accuracy of the various estimates and assumptions was not necessarily very good, and (b) interest in "rough-water" motions without water contact is largely academic, anyway. However, that development does, at least, give some idea of what is involved in calculating the magnification factors. It is a fairly complex problem, even when one chooses the simplest cases, and makes very liberal use of simplifying approximations. It is frequently next to impossible, when one comes to problems of real practical interest.

A typical towing-tank determination of the magnification factors might be described as follows. The frequency of encounter with a simple wave is

$$\omega = \sqrt{\frac{2\pi g}{\lambda}} + 2\pi \frac{V_o}{\lambda}$$

Therefore, at given model velocity, the desired range of frequency of encounter can be traced out by testing over simple waves of various wave lengths, thus obtaining curves of $M_\alpha(\omega)$ and $M_y(\omega)$ at various constant velocities $V_o$. These must then be adjusted to allow for the scale of the model, in accordance with Froude scaling laws. The magnification factors are nondimensional, and require no adjustment, but $\omega$ and $V_o$ must be adjusted in a manner which can be seen by rewriting the above expression for $\omega$ in nondimensional form:

$$\left[\frac{\omega^2 \ell}{g} \cdot \frac{\lambda}{\ell}\right]^{\frac{1}{2}} = \sqrt{2\pi} + 2\pi \left[\sqrt{\frac{\ell}{\lambda}} \cdot \frac{V_o}{\sqrt{g\ell}}\right]$$

To preserve geometric scale, obviously

$$\left(\frac{\ell}{\lambda}\right)_{\text{model}} = \left(\frac{\ell}{\lambda}\right)_{\text{prototype}}$$

Then, since the acceleration of gravity g is independent of scale, we obtain

$$\left(V_o\right)_{\text{prototype}} = \left(V_o\right)_{\text{model}} \left[\frac{(\ell)_{\text{prototype}}}{(\ell)_{\text{model}}}\right]^{\frac{1}{2}}$$

$$(\omega)_{\text{prototype}} = (\omega)_{\text{model}} \left[\frac{(\ell)_{\text{model}}}{(\ell)_{\text{prototype}}}\right]^{\frac{1}{2}}$$

(The nondimensional velocity parameter

$$F_\ell \equiv \frac{V_o}{\sqrt{g\ell}}$$

is called the "Froude number." Since g is invariant, naval architects frequently use "speed-length ratio" $V_k/\sqrt{\ell}$ as a more convenient velocity parameter.)

Having adjusted the model results to full scale, the magnification factors for each velocity are multiplied by the appropriate energy spectrum of the waves $[A(\omega)]^2$ to obtain the energy spectra of the motions, as previously discussed.

The necessary hypothesis that the responses are linear is easily examined during the model tests, by determining whether the motions are sinusoidal, and whether the magnification factors are independent of wave height. Of course, they often are not, with any very high degree of exactness. The questions to be answered then are: (a) whether the non-linearities are sufficiently slight that one can proceed with energy-spectral analysis, and (b) if so, how should one choose the best "equivalent linear" approximations to the magnification factors, from the experimental measurements obtained. These questions are best left to be answered, on a case-by-case basis, by veteran experimenters.

The preceding discussion has been confined to prediction of the pitch and heave energy spectra for vehicles in one-dimensional head or following seas. The extension to prediction of other properties of the motion is obvious, so long as it is possible to determine linear-response magnification factors for the desired properties.

The extensions to motions at headings other than normal to wave crests, and to motions over two-dimensional wave systems (i.e., systems composed of simple-wave components propagating in a variety of directions) are beyond the scope of our introductory discussion.

## VA-3 TESTS

Considering the importance which is attached to seakeeping by those engaged in GEM research and development, there is a remarkable paucity

of experimental information on this subject available in open literature. To some extent, this reflects the fact that a thorough evaluation of the seakeeping characteristics of even a single design is a very difficult, expensive, and time-consuming task. More importantly, it undoubtedly reflects the fact that the more important investigations have been slow in finding their way into the literature, for commercial reasons.

The most important data available in open literature at present appear to be the results of tests of the Vickers VA-3 craft performed by Republic Aviation Corporation, Reference 2. These tests were quite limited in scope, but sufficient results were obtained to give some idea of the riding qualities of a peripheral-jet type machine with flexible trunks and small daylight gap.

The results, in addition to the quantitative information they provide, are of great value in confirming numerous favorable, but largely unsupported statements in the press regarding Hovercraft seakeeping qualities.

A general-arrangement drawing of the VA-3 is reproduced from Reference 2 (see Figure G-1). The VA-3 had a gross weight of about 30,000 pounds, cushion area of about 1000 square feet, and mean-cushion-length/beam ratio $\bar{\ell}/b$ of about 2.0. (These dimensions are roughly estimated from the drawing.)

A representative set of statistical data on vertical accelerations measured at the bow are reproduced in Figure G-2. Data allowing comparison of the acceleration levels at the bow with those at the center of gravity are reproduced in Figure G-3.

These data pertain to the maxima of the acceleration time-histories, the ordinates F of the curves giving the probability that an arbitrarily chosen maximum would exceed the corresponding abscissas. Thus the abscissa corresponding to $F = 0.5$ is the median of all the maxima; the abscissa corresponding to $F = \frac{1}{2}\left(\frac{1}{3}\right) = 0.167$ is the median of the one-third highest maxima, etc. Reference 2 states that the statistical quality of the data is not very high, due to comparatively rapid changes in the energy spectrum of the waves and imprecise control of other test

variables. This is reflected in the irregular shapes of the probability curves and some seeming inconsistencies between results for different conditions. Nevertheless, the results provide a very useful indication of the severity of motions likely to be encountered.

The results can be rendered somewhat more useful by representing them in an appropriate nondimensional form. This is done in Figure G-4, where the entire envelope of the VA-3 tests is shown on a map of speed-length ratio $V_k/\sqrt{\ell}$ versus ratio of mean wave height to cushion length $H_{av}/\ell$, and then the "maximum" bow accelerations (median of the one-tenth highest maxima) for various conditions represented in Figure G-2 are located on this map. (As previously noted, the speed-length ratio $V_k/\sqrt{\ell}$ can be considered nondimensional if the acceleration of gravity g is considered constant.) Obviously, the results are insufficient to permit any detailed trends to be mapped. The results are significant, however, as a qualitative indication of the remarkable ability of GEM's to traverse rough water at speed without severe motions.

These results are particularly impressive when the nondimensional map of Figure G-4 is re-interpreted in dimensional form for vehicles larger than VA-3. (This is strictly valid provided the larger vehicles are "scale models" of VA-3 and the scaled-up sea has proper spectrum, in accordance with the Froude scaling rules.) An extreme example is given on Figure G-2 in the form of auxiliary scales along the top and right-hand side of the graph giving wave height and speed for a "model" 8.9 times as large as VA-3, weighing $(8.9)^3$ times as much. Keep in mind that the accelerations mapped pertain to the bow, the accelerations at the c.g. being only about 40 percent as great.

Of course, when 9400-ton GEM's are built, they are quite unlikely to be, strictly speaking, "scale models" of VA-3. Nevertheless, one is strongly encouraged to conclude that seakeeping problems would not constitute a serious barrier to the development and application of large GEM's.

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July 1966

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2. Republic Aviation Corp. VA-3 Air Cushion Vehicle Test Program. Farmingdale, N.Y., Oct 1964. 1 v. illus. (RAC-2612. Contract Nonr 4500(CO))

# Table G-1

## Properties of Fully Arisen Sea

WIND AND SEA SCALE FOR FULLY ARISEN SEA

| SEA STATE [1] | SEA—GENERAL: DESCRIPTION [2] | WIND [3]: (EQUIVALENT) WIND FORCE | WIND: DESCRIPTION | WIND: RANGE (KNOTS) | WIND: VELOCITY (KNOTS) AVERAGE | SEA [3]: WAVE HEIGHT FEET: SIGNIFICANT | WAVE HEIGHT FEET: AVERAGE 1/10 HIGHEST | SIGNIFICANT RANGE OF PERIODS (SECONDS) | T = PERIOD OF MAXIMUM ENERGY (PEAK OF SPECTRUM) (SECONDS) | T (AVERAGE PERIOD) (SECONDS) | AVERAGE WAVE LENGTH (FEET) | GENERAL REACH (NAUTICAL MILES) | GENERAL DURATION (HOURS) |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| 0 | Sea like a mirror. | 0 | Calm | Less than 1 | 0 [a] | 0 | 0 | 0 | – | – | – | – | – |
| | Ripples with the appearance of scales are formed, but without foam crests. | 1 | Light Air | 1 - 3 | 2 | 0.05 | 0.08 | 0.10 | up to 1.2 sec | 0.7 | 0.5 | 10 in. | 5 | 30 min |
| 1 | Small wavelets, still short but more pronounced; crests have a glassy appearance, but do not break. | 2 | Light Breeze | 4 - 6 | 5 | 0.18 | 0.29 | 0.37 | 0.4-2.8 | 2.0 | 1.4 | 6.7 ft | 8 | 39 min |
| | Large wavelets, crests begin to break. Foam of glassy appearance. Perhaps scattered white horses. | 3 | Gentle Breeze | 7 -10 | 8.5 | 0.6 | 1.0 | 1.2 | 0.8-5.0 | 3.4 | 2.4 | 20 | 9.8 | 1.7 hrs |
| | | | | | 10 | 0.80 | 1.4 | 1.8 | 1.0-6.0 | 4 | 2.9 | 27 | 10 | 2.4 |
| 2 | Small waves, becoming longer; fairly frequent white horses. | 4 | Moderate Breeze | 11-16 | 12 | 1.4 | 2.2 | 2.8 | 1.0-7.0 | 4.8 | 3.4 | 40 | 18 | 3.8 |
| | | | | | 13.5 | 1.8 | 2.9 | 3.7 | 1.4-7.6 | 5.4 | 3.9 | 52 | 24 | 4.8 |
| 3 | | | | | 16 | 2.0 | 3.3 | 4.2 | 1.5-7.8 | 5.6 | 4.0 | 59 | 28 | 5.2 |
| | | | | | 16 | 2.9 | 4.6 | 5.8 | 2.0-8.8 | 6.5 | 4.6 | 71 | 40 | 6.6 |
| 4 | Moderate waves, taking a more pronounced long form; many white horses are formed. (Chance of some spray). | 5 | Fresh Breeze | 17-21 | 18 | 3.8 | 6.1 | 7.8 | 2.5-10.0 * | 7.2 | 5.1 | 90 | 55 | 8.3 |
| | | | | | 19 | 4.3 | 6.9 | 8.7 | 2.8-10.6 | 7.7 | 5.4 | 99 | 65 | 9.2 |
| | | | | | 20 | 5.0 | 8.0 | 10 | 3.0-11.1 | 8.1 | 5.7 | 111 | 75 | 10 |
| 5 | Large waves begin to form; the white foam crests are more extensive everywhere. (Probably some spray). | 6 | Strong Breeze | 22-27 | 22 | 6.4 | 10 | 13 | 3.4-12.2 | 8.9 | 6.3 | 134 | 100 | 12 |
| | | | | | 24 | 7.9 | 12 | 16 | 3.7-13.5 | 9.7 | 6.8 | 160 | 130 | 14 |
| | | | | | 24.5 | 8.2 | 13 | 17 | 3.8-13.6 | 9.9 | 7.0 | 164 | 140 | 15 |
| 6 | | | | | 26 | 9.6 | 15 | 20 | 4.0-14.5 | 10.5 | 74 | 188 | 188 | 17 |
| 7 | Sea heaps up and white foam from breaking waves begins to be blown in streaks along the direction of the wind. (Spindrift begins to be seen). | 7 | Moderate Gale | 28-33 | 28 | 11 | 18 | 23 | 4.5-15.5 | 11.3 | 7.9 | 212 | 230 | 20 |
| | | | | | 30 | 14 | 22 | 28 | 4.7-16.7 | 12.1 | 8.6 | 250 | 290 | 23 |
| | | | | | 30.5 | 14 | 23 | 29 | 4.8-17.0 | 12.4 | 8.7 | 258 | 290 | 24 |
| | | | | | 32 | 16 | 26 | 33 | 5.0-17.5 | 12.9 | 9.1 | 295 | 340 | 27 |
| | Moderately high waves of greater length; edges of crests break into spindrift. The foam is blown in well marked streaks along the direction of the wind. Spray affects visibility. | 8 | Fresh Gale | 34-40 | 34 | 19 | 30 | 38 | 5.5-18.5 | 13.6 | 9.7 | 322 | 420 | 30 |
| | | | | | 36 | 21 | 35 | 44 | 5.8-19.7 | 14.5 | 10.3 | 363 | 500 | 34 |
| | | | | | 37 | 23 | 37 | 46.7 | 6-20.5 | 14.9 | 10.5 | 376 | 530 | 37 |
| | | | | | 38 | 25 | 40 | 50 | 6.2-20.8 | 15.6 | 10.7 | 392 | 600 | 38 |
| 8 | | | | | 40 | 28 | 45 | 58 | 6.5-21.7 | 16.1 | 11.4 | 444 | 710 | 42 |
| | High waves. Dense streaks of foam along the direction of the wind. Sea begins to roll. Visibility affected. | 9 | Strong Gale | 41-47 | 42 | 31 | 50 | 64 | 7-23 | 17.0 | 12.0 | 492 | 830 | 47 |
| | | | | | 44 | 36 | 58 | 73 | 7-24.2 | 17.7 | 12.5 | 534 | 960 | 52 |
| | | | | | 46 | 40 | 64 | 81 | 7-25 | 18.6 | 13.1 | 590 | 1110 | 57 |
| 9 | Very high waves with long overhanging crests. The resulting foam is in great patches and is blown in dense white streaks along the direction of the wind. On the whole the surface of the sea takes a white appearance. The rolling of the sea becomes heavy and shocklike. Visibility is affected. | 10 | Whole Gale* | 48-55 | 48 | 44 | 7 | 90 | 7.5-26 | 19.4 | 13.8 | 650 | 1250 | 63 |
| | | | | | 50 | 49 | | 99 | 7.5-27 | 20.2 | 14.3 | 700 | 1420 | 69 |
| | | | | | 51.5 | 52 | 83 | 106 | 8-28.2 | 20.8 | 14.7 | 736 | 1560 | 73 |
| | | | | | 52 | 54 | 87 | 110 | 8-28.5 | 21.0 | 14.8 | 750 | 1610 | 75 |
| | | | | | 54 | 59 | 95 | 121 | 8-29.5 | 21.8 | 15.4 | 810 | 1800 | 81 |
| | Exceptionally high waves (Small and medium-sized ships might for a long time be lost to view behind the waves.) The sea is completely covered with long white patches of foam lying along the direction of the wind. Everywhere the edges of the wave crests are blown into froth. Visibility affected. | 11 | Storm* | 56-63 | 56 | 64 | 103 | 130 | 8.5-31 | 22.6 | 16.3 | 910 | 2100 | 88 |
| | | | | | 59.5 | 73 | 116 | 148 | 10-32 | 24 | 17.0 | 983 | 2500 | 101 |
| | Air filled with foam and spray. Sea completely white with driving spray; visibility very seriously affected. | 12 | Hurricane* | 64-71 | > 64 | > 90 [b] | > 120 [b] | > 164 [b] | 10-(35) | (26) | (18) | ~ | ~ | ~ |

*For hurricane winds (and often whole gale and storm winds) required durations and fetches are rarely attained. Seas are therefore not fully arisen.

[a] A heavy line around this value means that the values tabulated are at the center of the Beaufort range.

[b] For such high winds, the seas are confused. The wave crests blow off, and the water and the air mix.

[1] Encyclopedia of Nautical Knowledge, W.A. McEwen and A.M. Lewis, Cornell Maritime Press, Cambridge, Maryland, 1953, p. 483

[2] Manual of Seamanship, Volume II, Admiralty, London, H.M. Stationery Office, 1952, pp. 717-718

[3] Practical Methods for observing and forecasting Ocean Waves, Pierson, Neumann, James, N.Y. Univ. College of Engrg., 1953.

This table compiled by Wilbur Marks, David Taylor Model Basin



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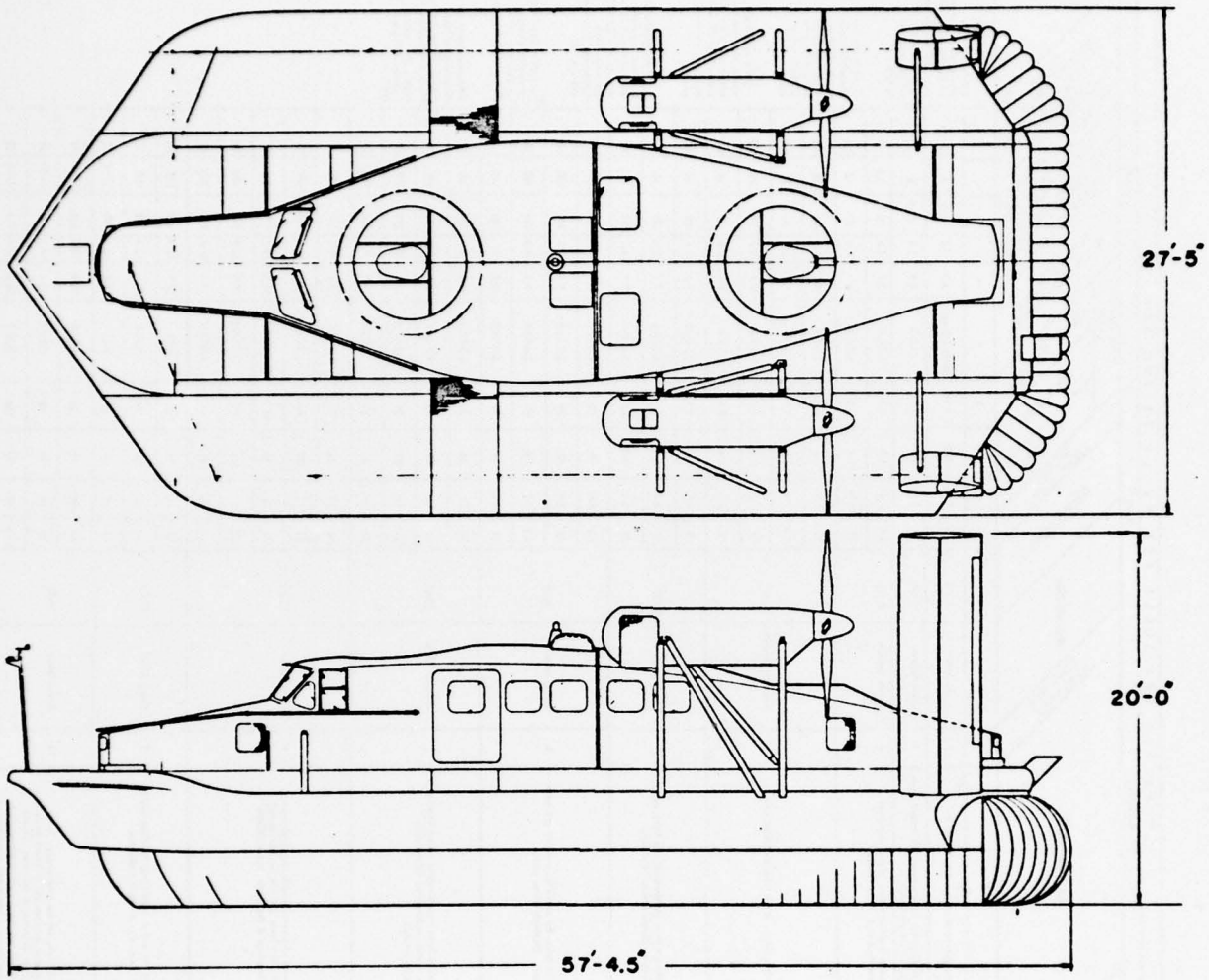
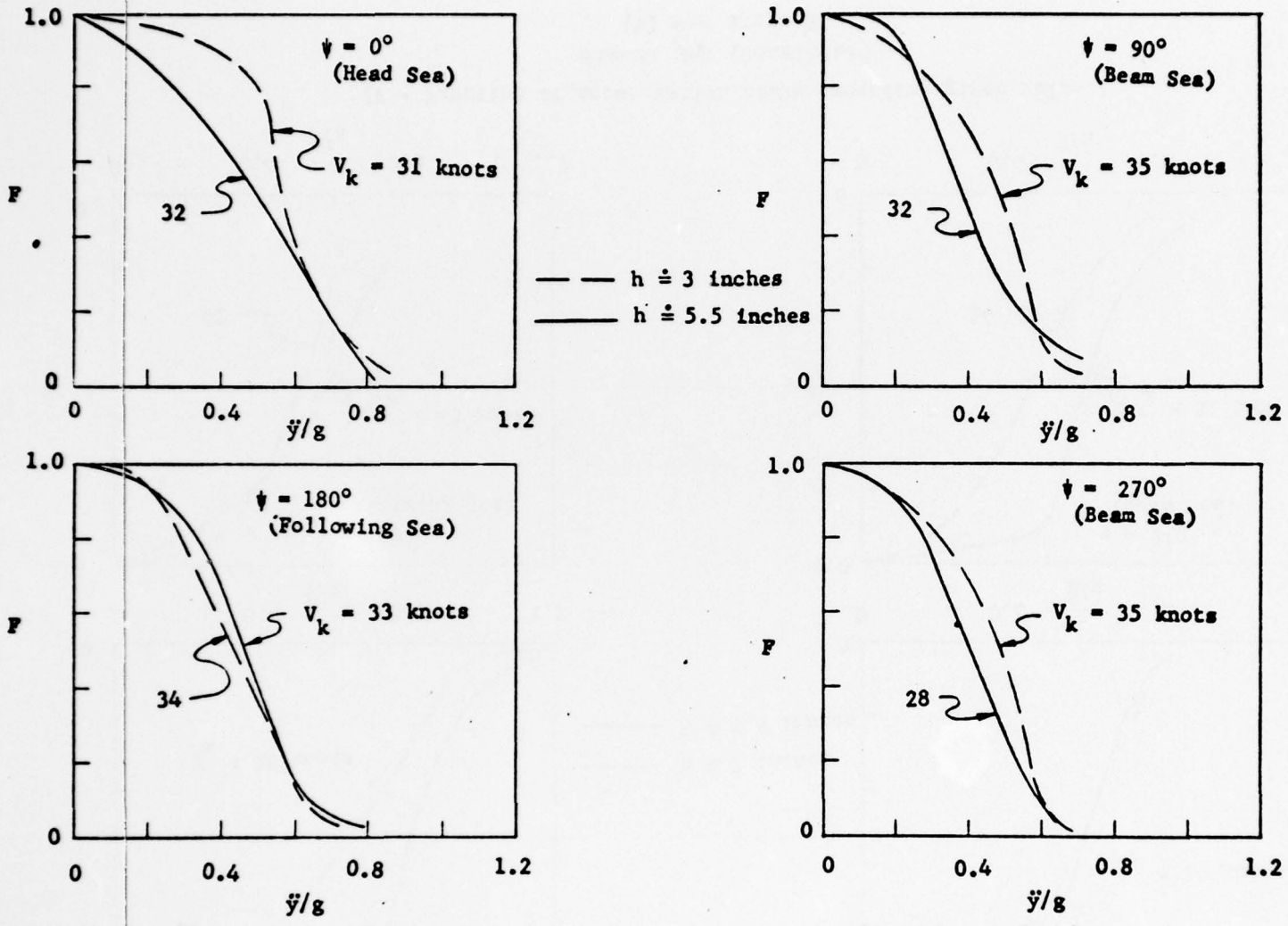
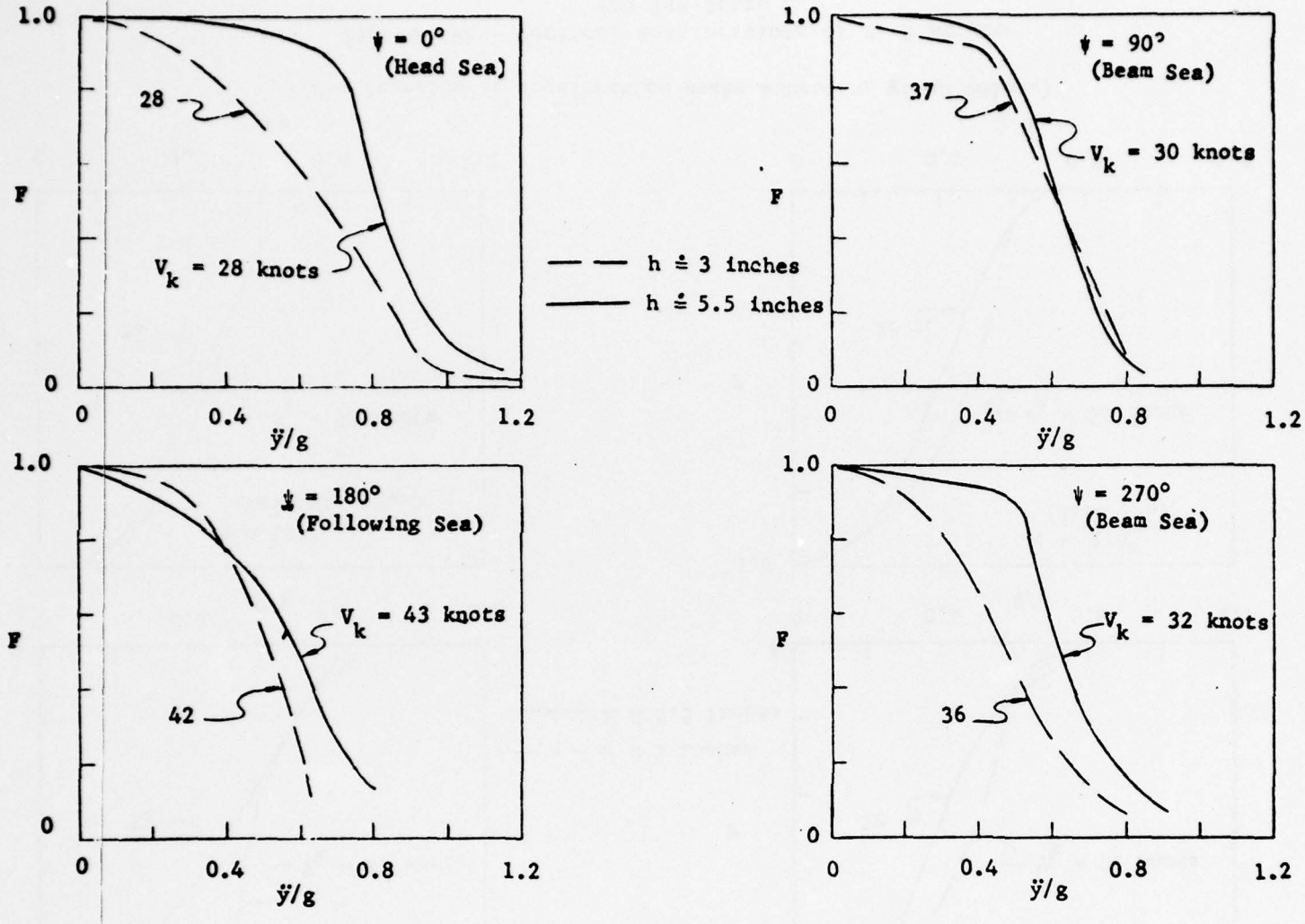



Figure G-1 - General Arrangement of VA-3



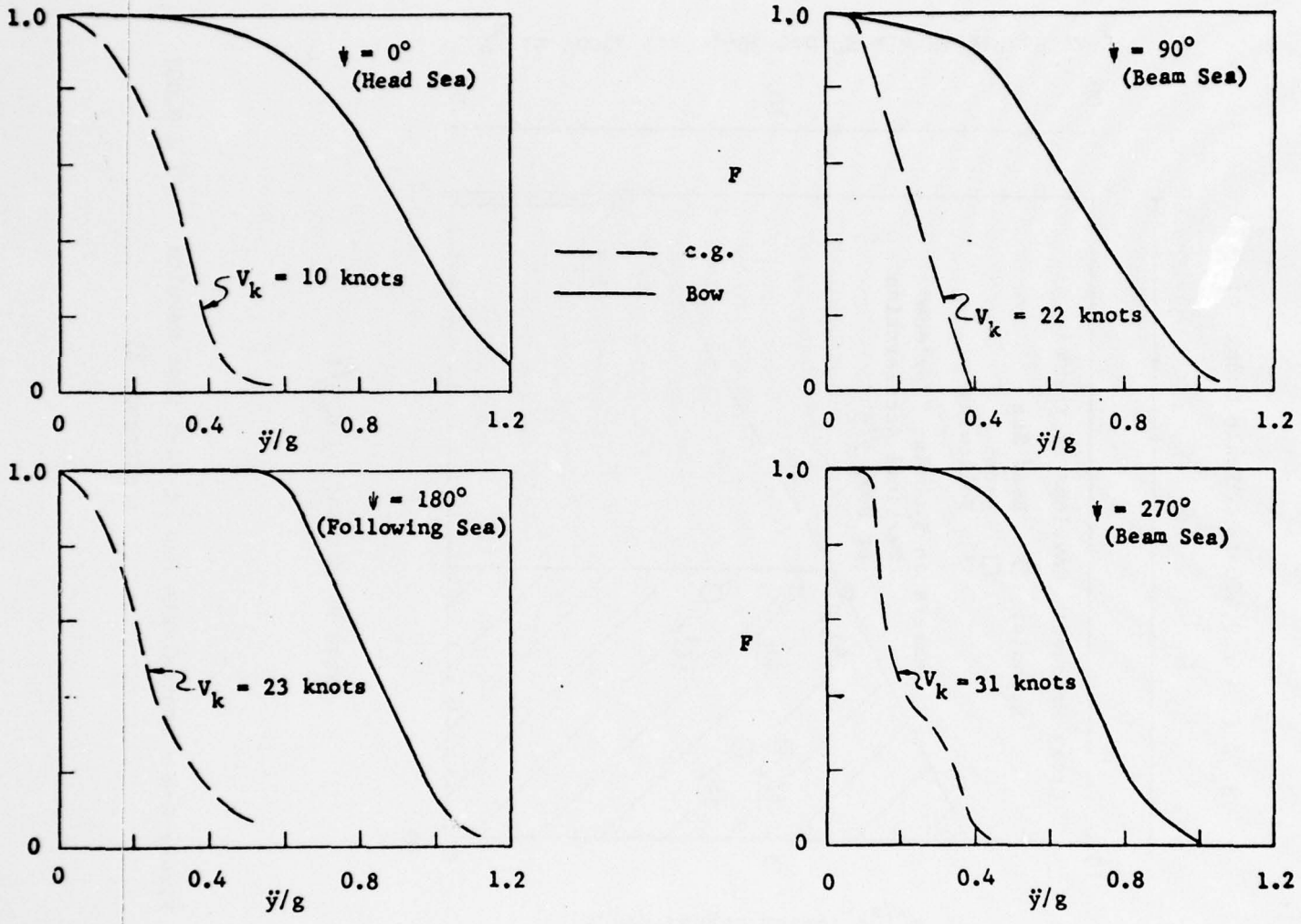

(F = Fraction of acceleration peaks exceeding given value.)

Figure G-2 - Vertical Accelerations of VA-3 at Bow
(a) Sea State 2




(F = Fraction of acceleration peaks exceeding given value.)

Figure G-2 (Concluded)
(b) Sea State 3




(F = Fraction of acceleration peaks exceeding given value.)

Figure G-3 - Comparison of Acceleration at Bow and Center of Gravity

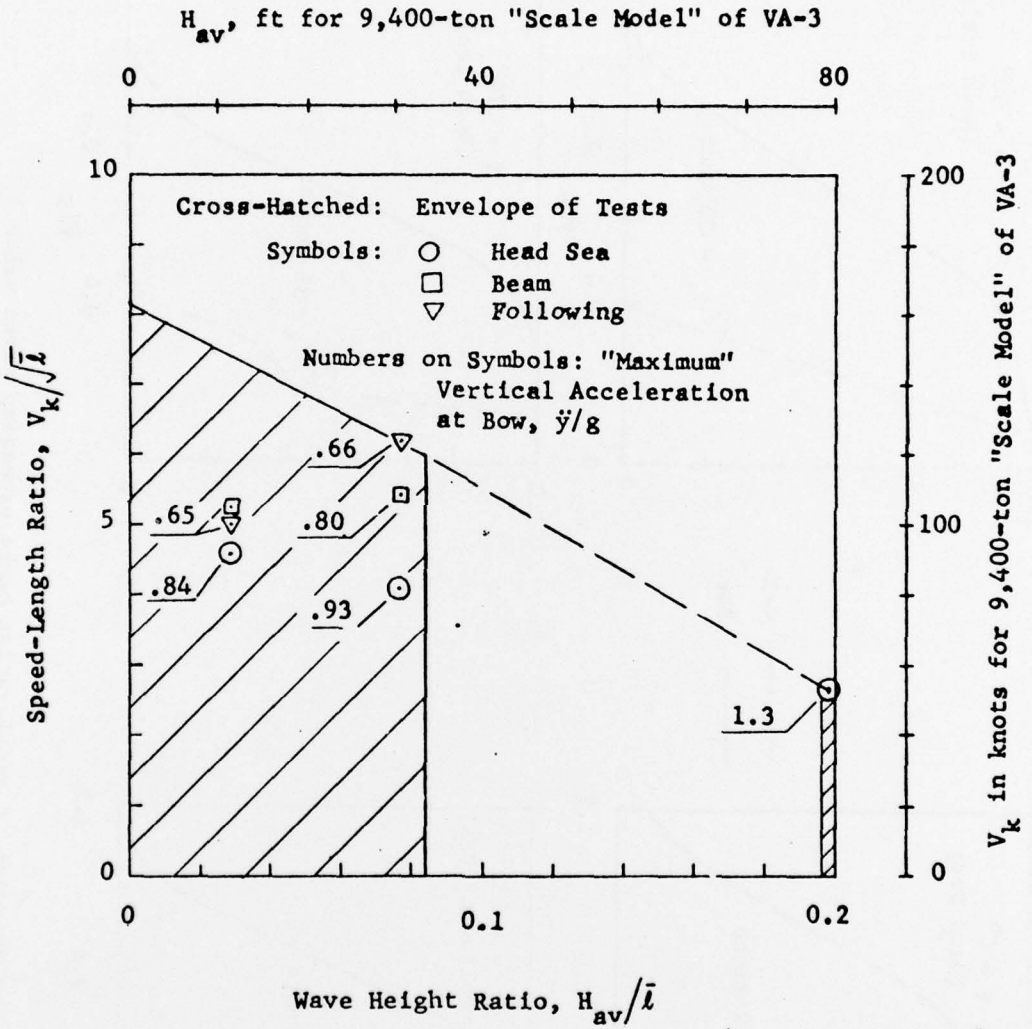



Figure G-4 - Approximate Map of VA-3 Test Results. $h/\ell \doteq 0.005$
(Reference 2)

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