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DEVELOPMENT OF AIR BEARING LIFT PADS

by Dan H. Dane and Herman T. Blaise

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DEVELOPMENT OF AIR BEARING LIFT PADS

SUMMARY

Providing horizontal movement with almost no frictional drag is often necessary when performing mechanical space simulation experiments or tests. To use a precision air bearing requires a precision surface, which is very costly when large areas are required. Using a bladder-type air pad on a good floor produces about 2.2 N (0.5 lb) drag. Using the bladder type air pads for testing the 12.2-m (40-ft) serpentuator where under one-g each of the eight 1.5-m (5-ft) links must be supported would require a torque of 122 joules (90 ft-lb) in the base link. This is larger than the capability of the serpentuator.

If a loaded air pad would float 0.317 cm (0.125 in.) above the floor, the drag would be almost zero, and the floor surface would not affect the performance of the air pad. Raising a precision air bearing or bladder air bearing higher than the air film layer by supplying more air to the pads produces unacceptable vibrations. Lowering the volume of air from a ground effects machine to get the desired effect produces statically unstable conditions. To provide a frictionless lift capacity on rough floors, a mathematical model of an air bearing lift pad (air pad) that floats 0.317 cm (0.125 in.) off the surface was devised, manufactured, and successfully tested. Three of these air bearing lift pads were assembled into a cushion air pad platform that had a load-carrying capacity exceeding 45.4 kg (100 lb) over a rough floor with less than 0.284 N (1 oz) friction drag. Seven upgraded units were designed and manufactured to support 136 kg (300 lb) for the serpentuator testing program and for general simulation use.

INTRODUCTION

To simulate on earth the dynamics which man and equipment have in space, we must furnish horizontal frictionless movement. The precision air bearing met these requirements of floating several hundred pounds with less than an ounce of frictional force. As the scope of mechanical space simulation expanded, larger and larger working areas were needed. The cost of grinding large floor areas to precision tolerances was not economically feasible.

These tests were run using a bladder-type air pad which would function on good floors with only 2.2 N (0.5 lb) of frictional force (Fig. 1). Much valuable information and test data were obtained by using the bladder air pads. As the scope of mechanical space simulation continued to expand, several experiments and test dictated less friction than was possible using the bladder air bearings. A precision air bearing that would perform on a rough floor was needed.

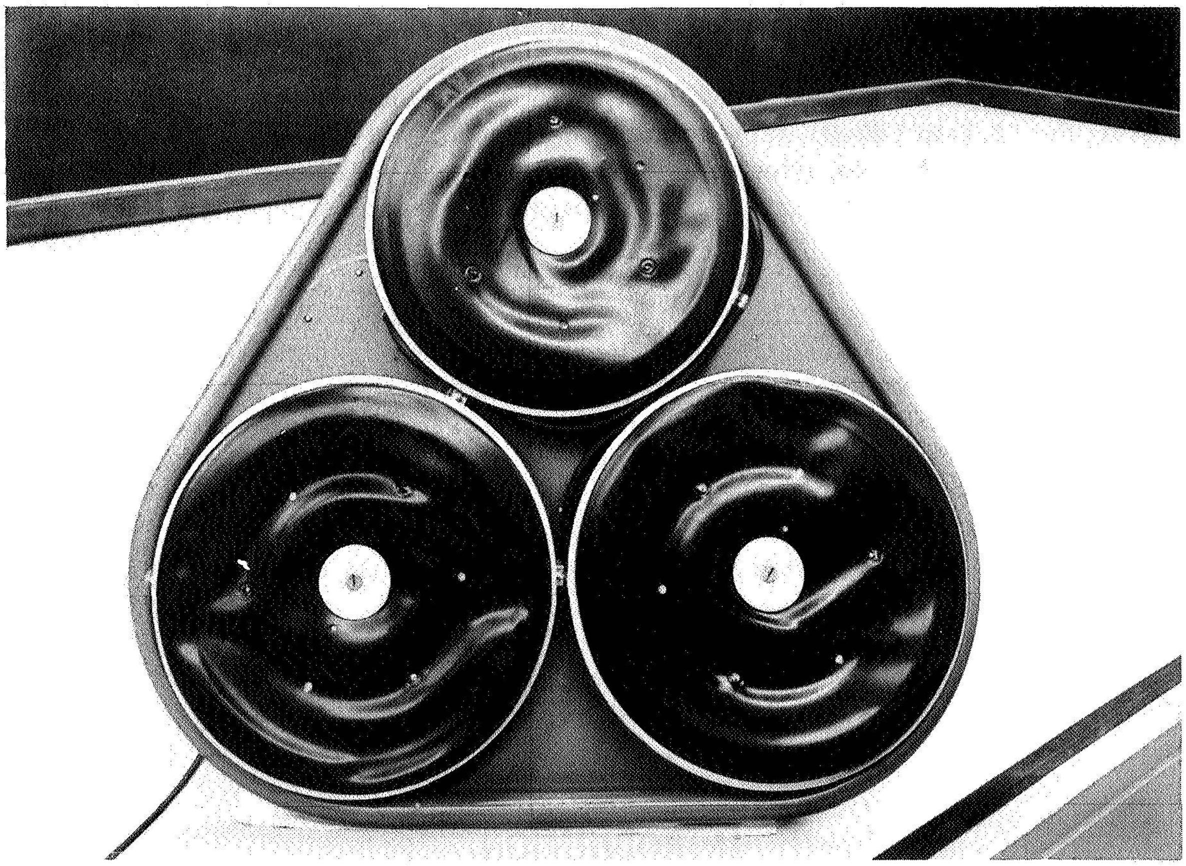


FIGURE 1. BLADDER TYPE AIR PAD

If an air bearing would rise above the surface 0.317 cm (0.125 in.), the air bearing would be relatively unaffected by floor joints, cracks, or surface texture; however, to increase the air flow and raise a precision air bearing above the normal height, 0.00127 to 0.00254 cm (0.0005 to 0.001 in.), produced a dynamically unstable condition and the bearings vibrated violently.

Ground effects machines are unaffected by small surface irregularities; however, their static stability and mass characteristics would not give meaningful test data.

It seemed that there should be a way to get the good characteristics of each, the lack of dependence on the surface characteristics for a ground effects machine and the load capacity per weight, friction, and stability of the precision air pad. A mathematical model was formulated and an air bearing lift pad was designed and fabricated from this model. The conclusion of Project No. 1000, under which the air pads were developed was the design and fabrication of seven lightweight air pad platforms which can support several hundred pounds with almost no frictional drag.

PROGRAM PROCEDURE

To develop an air pad structure with maximum payload to weight ratio with almost no friction requires a minimum floating height above the floor which is still sufficient to clear the highest spots in the floor. From inspection and measurement of sections of a poor floor, it was established that a height of 0.317 cm (0.125 in.) would satisfy the requirements for the average floor. Since the flow and pressure can be calculated easily for the load and air pad area, the solution of the stability problem would complete the design parameters. Increasing the amount of air flow through an air bearing (bladder or precision) to lift it above the escaping air film produces unwanted vibrations, while decreasing the air flow on a ground effects machine produces static instability problems.

The approach taken to solve the stability problem was to analyze the factors which stabilize or destabilize vehicles that float on a fluid, such as aircraft, boats, and ground effect machines. These factors were extended into concepts in functional form to include air pad stability and a theoretically stable air pad was sketched. It was later determined that part of the direct analysis used to sketch the first air pad is invalid without intermediate steps; however, since the air pad was already shown to be stable by testing, the analysis instead of the air pad was corrected.

Testing the theory required a test air pad which was thermovacuum formed from a sheet of clear cellulose acetate butyrate. This material and type of construction was chosen because the lightweight plastic was well suited to the shape, was clear to check air flow using paper bits or smoke, and was very inexpensive. The wooden forming mold used to form the air pad and a plastic air pad are shown in Figures 2 and 3.

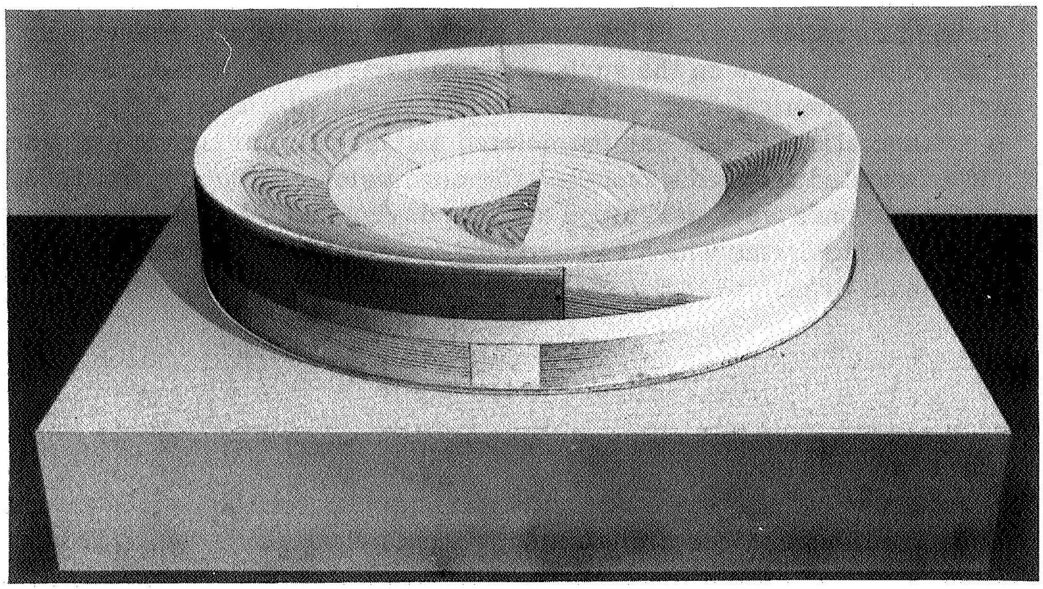


FIGURE 2. MOLD FOR AIR BEARING LIFT PAD

The first test was made using an industrial vacuum cleaner for the air supply. The vacuum cleaner produced about 0.0313 cms (70 cfm) and the air bearing lift pad lifted about 0.0508 cm (0.02 in.) off the floor. Tests were run with the air pad unloaded or lightly loaded. The vibrations which normally start when lifting air bearings above an air film did not occur. The pad was also statically stable. To increase the height that the air bearing lift pad floated above the floor, two blowers were ordered that would each produce 0.1959 cms (415 cfm) at 7.62 cm (3 in.) of water. A fiberglass cushion air pad (Fig. 4) was also made to verify that the design, instead of a material characteristic of butyrate, was the reason for the vibration-free operation of the air pad. Tests on the fiberglass air bearing lift pad verified the stable and vibration-free operation of the design.

Enlarging the air ducts to minimize restrictions of the air flow increased the load-carrying capacity to 20.4 kg (45 lb) using the new blower, while clearing the 0.317 cm (0.125 in.) undulations in the test floor.

Some improvements in the design were made, and a manifold (Figs. 5 through 8) was fabricated to feed three air bearing lift pads. The two test

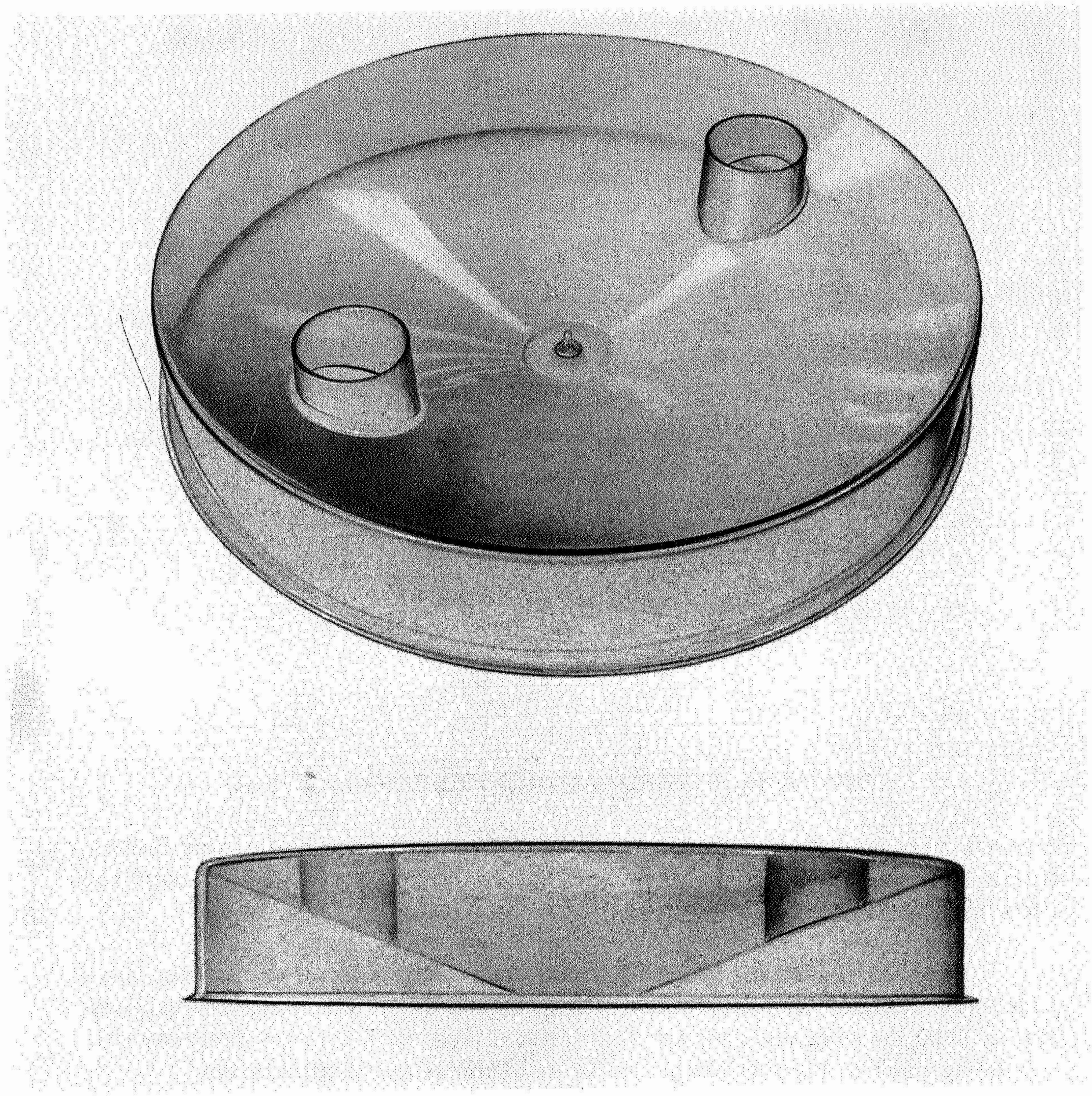


FIGURE 3. AIR BEARING LIFT PAD (With side view)

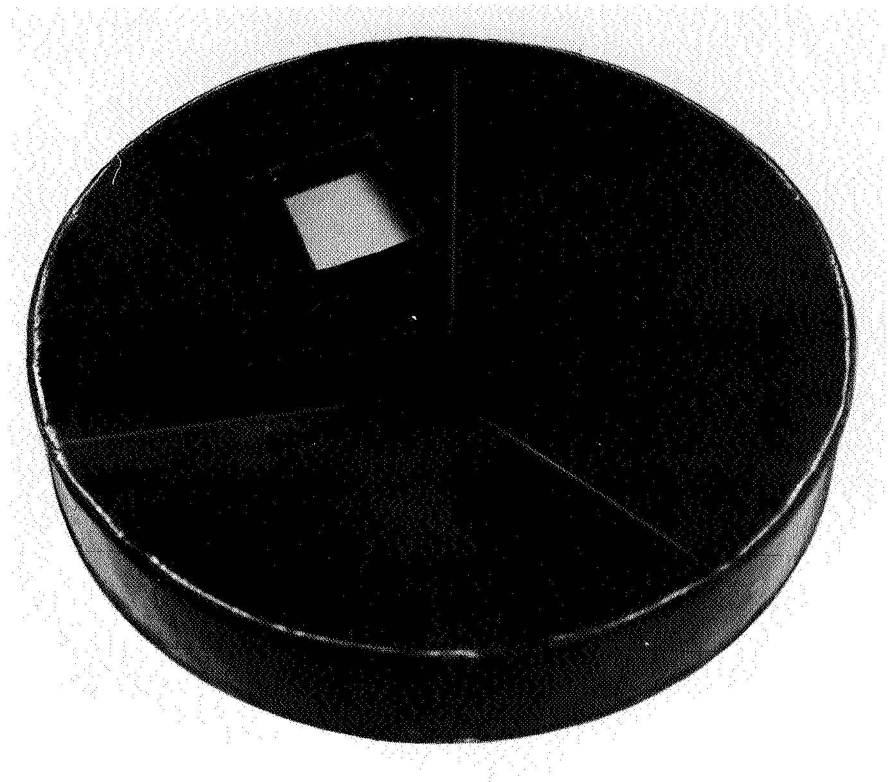


FIGURE 4. FIBERGLASS AIR BEARING LIFT PAD

blowers were mounted on the manifold unit (Fig. 9), and the unit was tested with a 45.4 kg (100 lb) payload. The force to move the unit over a rough concrete floor was under 0.28 N (1 oz).

Seven upgraded air pad platforms and counterbalance units using the air pads were designed and fabricated (Figs. 10 and 11). The units can support 136 kg (300 lb) over the average floor with almost no frictional drag and are now being used to support programs in mechanical space simulation.

CONCLUSIONS

The air bearing lift pad is both statically and dynamically stable. It functions on a rough concrete floor with about the same coefficient of friction as a precision air bearing on a precision floor. The air bearing lift pad does

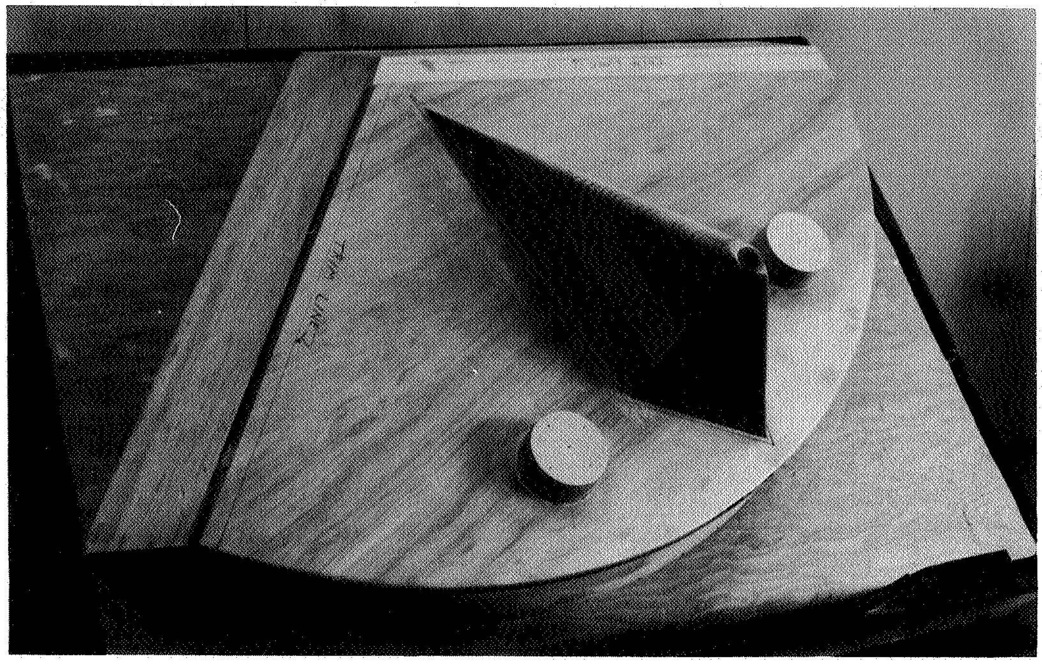


FIGURE 5. MOLD FOR ONE-THIRD SECTION OF MANIFOLD

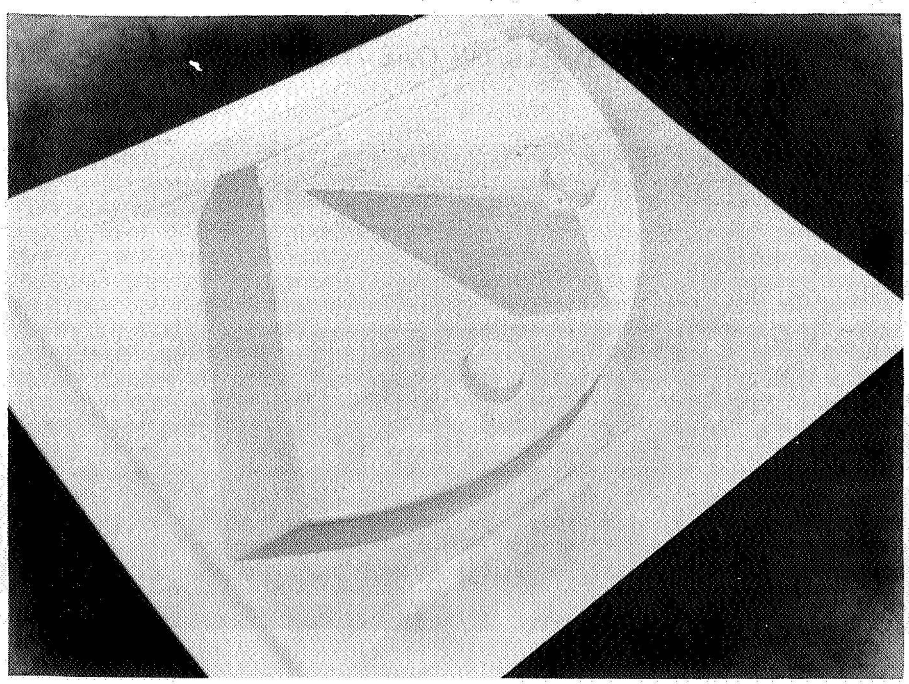


FIGURE 6. MOLDED ONE-THIRD SECTION OF MANIFOLD

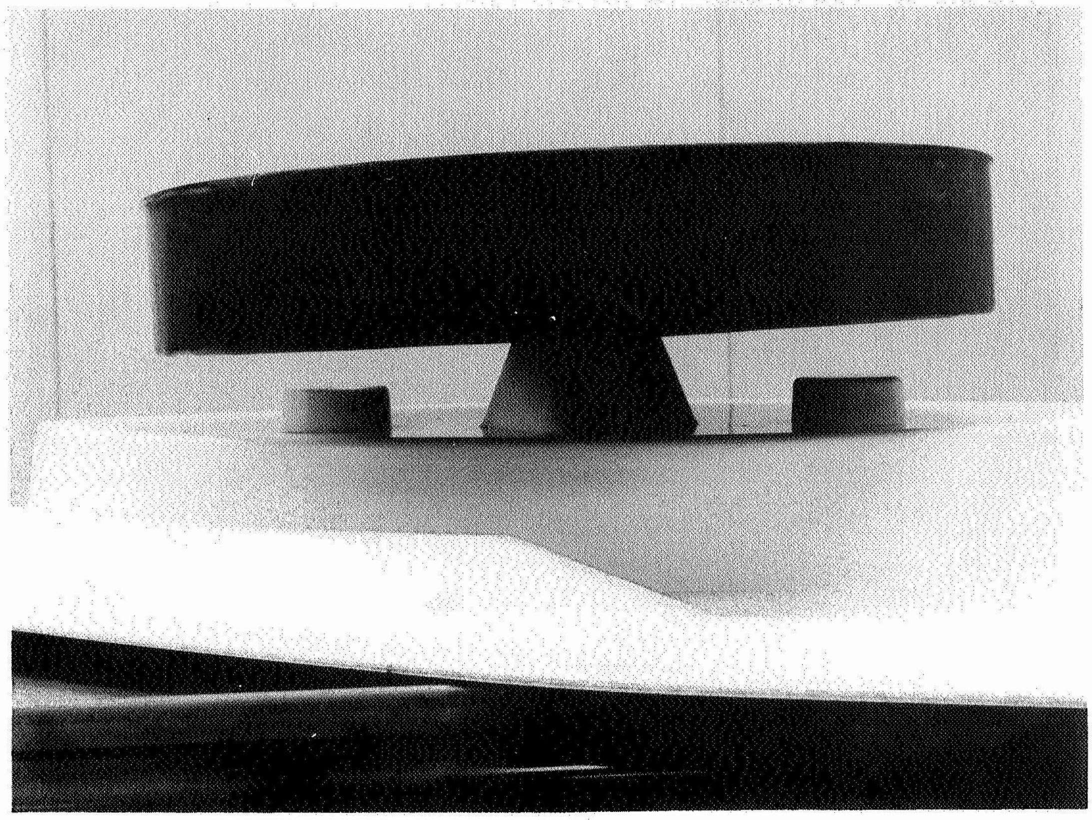


FIGURE 7. AIR BEARING LIFT PAD ON ONE-THIRD SECTION OF MANIFOLD

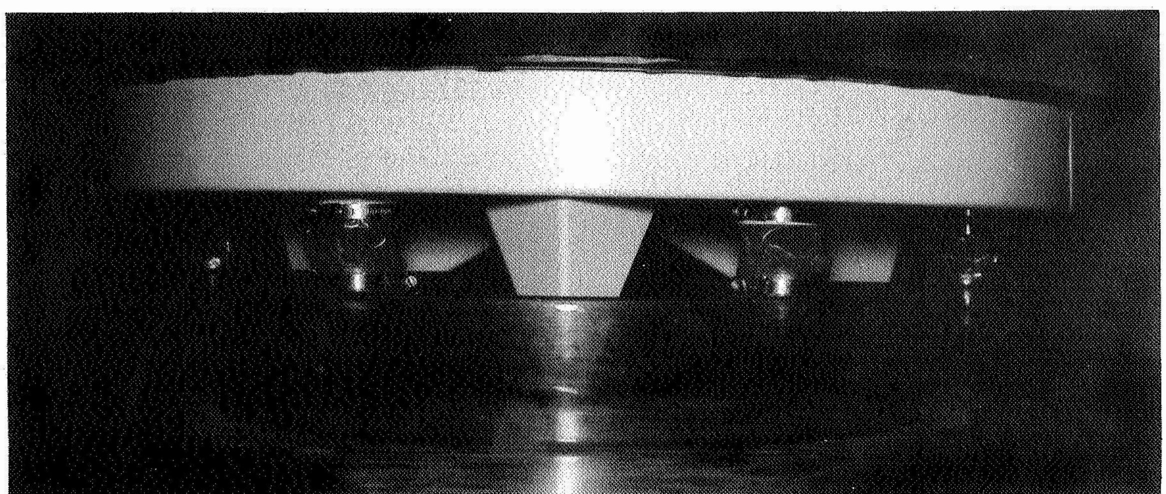


FIGURE 8. AIR BEARING LIFT PAD PLATFORM WITH AIR BEARINGS

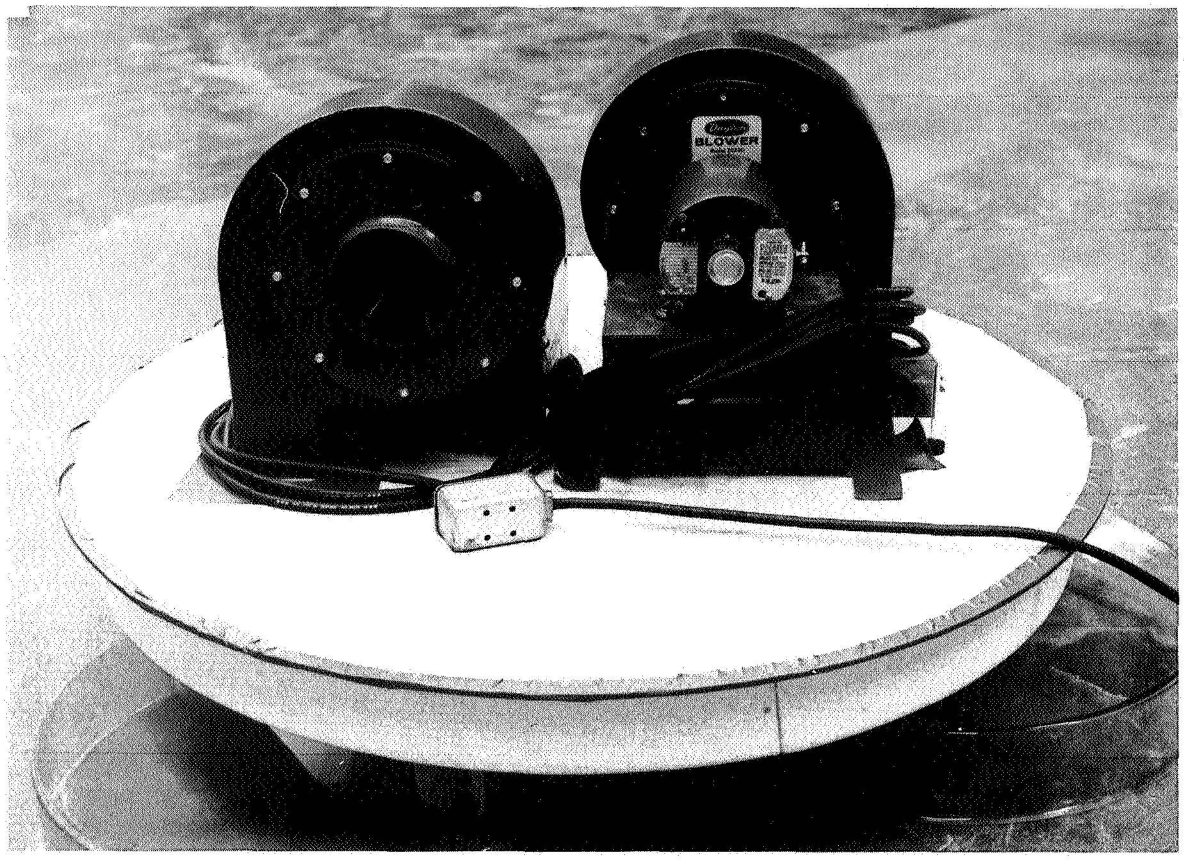


FIGURE 9. TEST CUSHION AIR PAD PLATFORM

not have the load-carrying capacity of precision air bearings or bladder type air pads, but it is much better than ground effects machines. The stabilizing feature of the air bearing lift pads may be incorporated into ground effects machines to improve the stability of such machines. The cost of the project and the fabrication of seven cushion air pad platforms to support the serpentuator development program is much less than other methods in this single program. The air bearing lift pad furnishes an inexpensive way to perform tests that would otherwise require precision air bearings on precision surfaces.

RECOMMENDATIONS

The present payload capability of a cushion air pad platform is about three times the weight of the structure, motor, and impeller. It would be

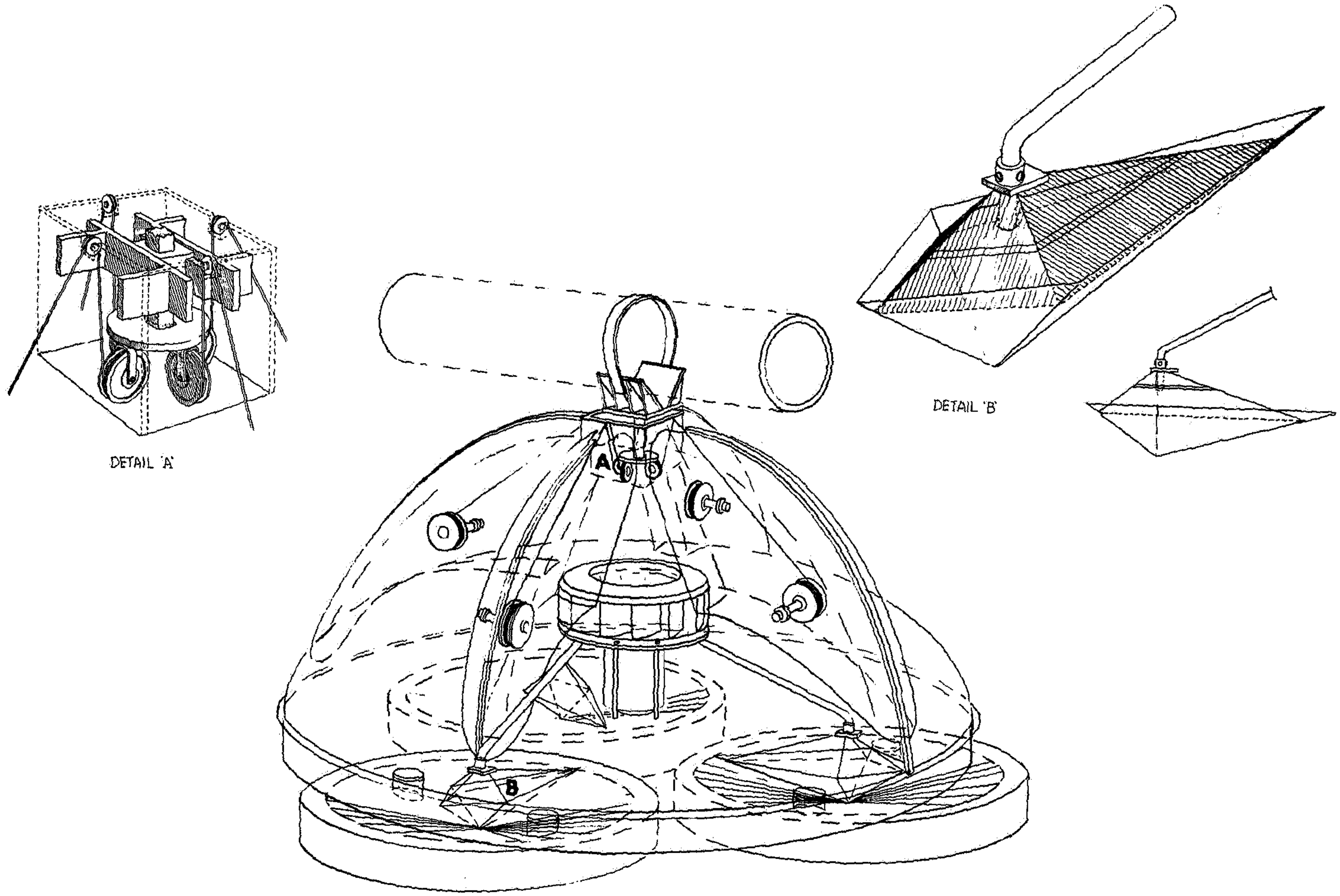


FIGURE 10. AIR BEARING LIFT PAD SUPPORT CONCEPT

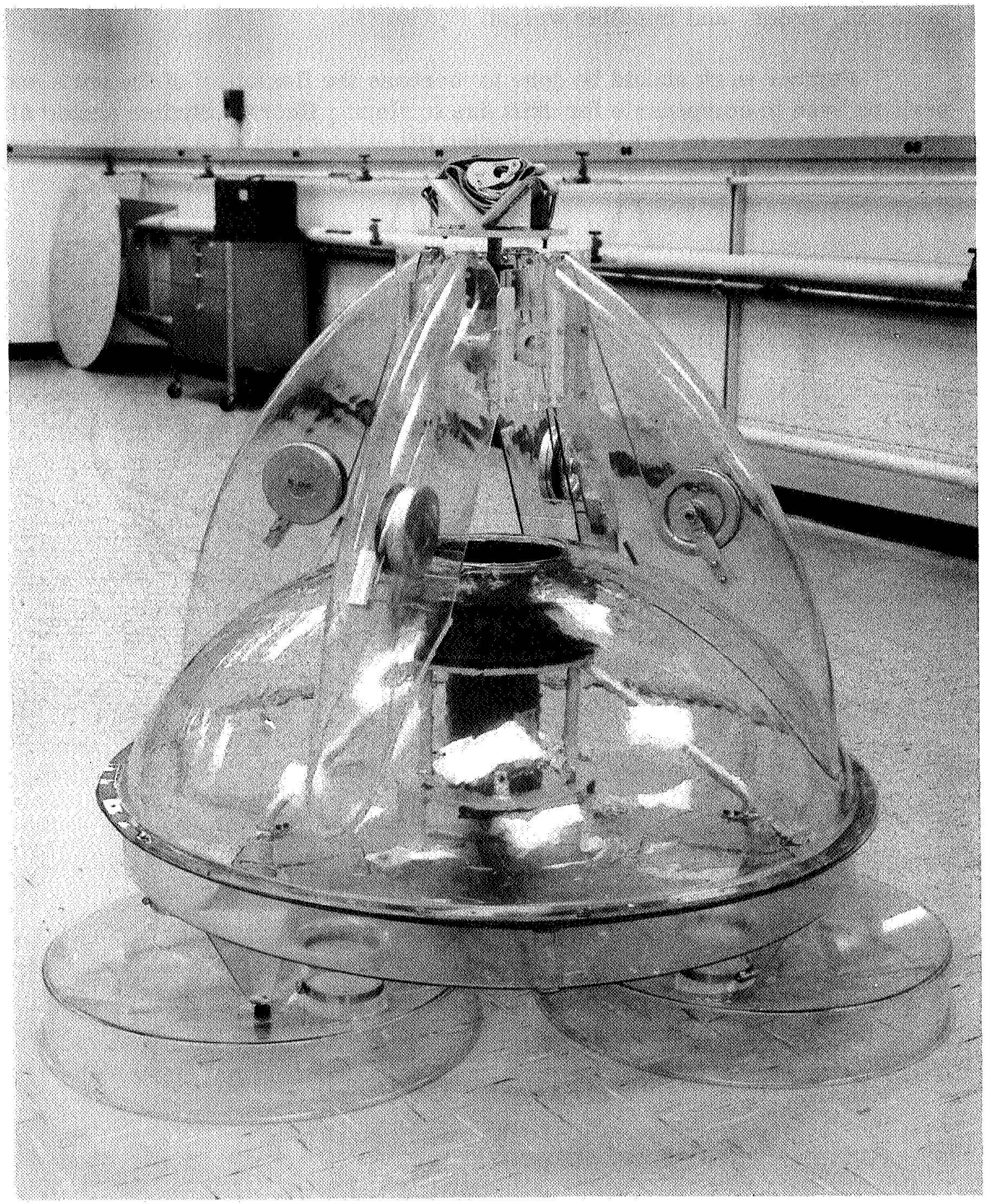


FIGURE 11. AIR BEARING LIFT PAD SUPPORT SYSTEM

advantageous to double or triple the payload weight ratio to six or ten times the platform, motor, and impeller weight.

Further work should be done to increase the flexibility of the units and perhaps even to compensate for drift due to sloping floors. Studies should also be conducted on the use of the air bearing lift pad technique to stabilize ground effects machines (Fig. 12).

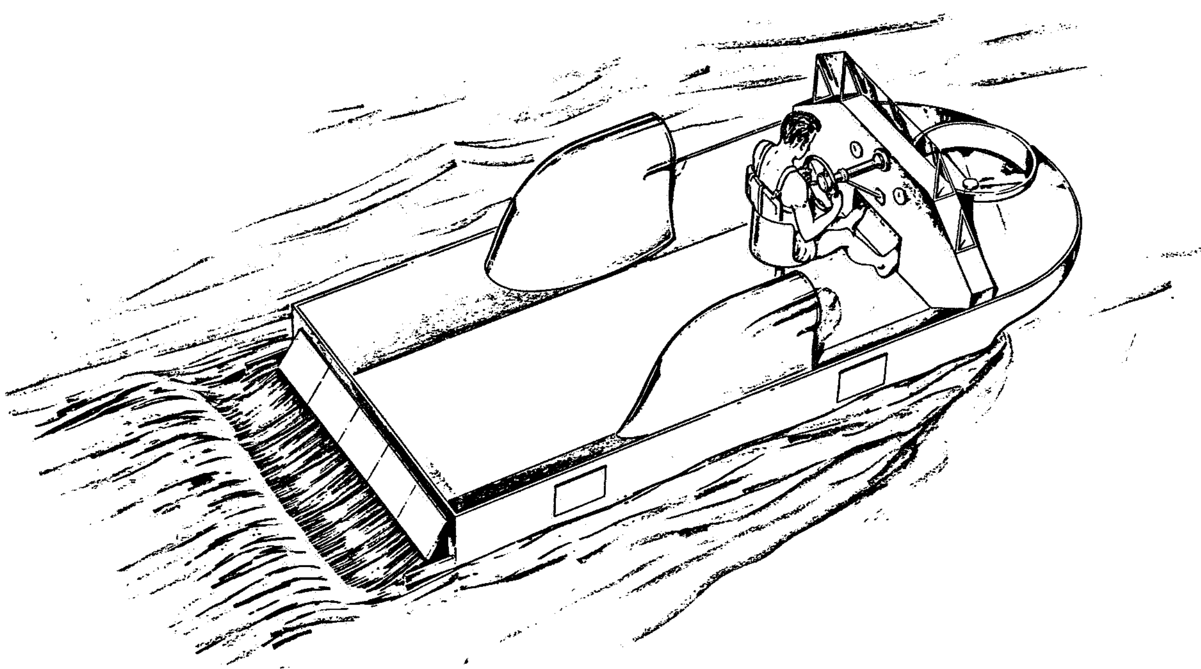


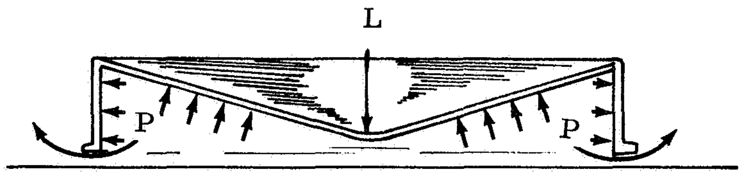
FIGURE 12. GROUND EFFECTS MACHINE

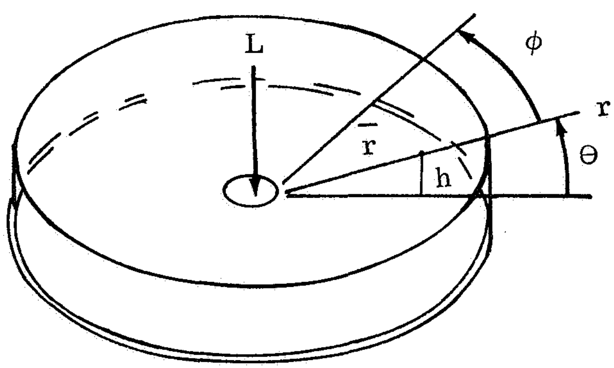
George C. Marshall Space Flight Center
National Aeronautics and Space Administration
Huntsville, Alabama, July 5, 1968
965-21-01-00-62

APPENDIX

ENGINEERING/MATHEMATICAL SUPPOSITION "AIR BEARING LIFT PAD STABILITY CRITERIA"

1. The load (L) is below the centroid of lift.





Define a pressure distribution function at any point along a radius, r:

$g(r) \equiv$ pressure distribution function

The centroid of pressure is:

$$\bar{r} = \int r g(r)\,dr \Big/ \int g(r)\,dr$$

The pressure distribution function may also be a function of ϕ, therefore the total pressure distribution function is $g(r,\phi)$, and the centroid is:

$$\bar{r}(\phi) = \int r(\phi)\,g(r,\phi)\,dr \Big/ \int g(r,\phi)\,dr$$

From trigonometry

$$h(\phi) = \bar{r}(\phi)\,\sin\theta$$

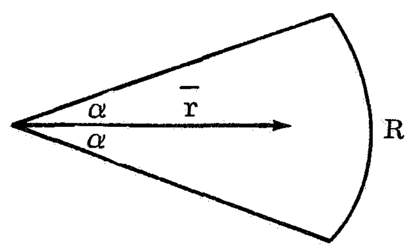
Where: $h(\phi)$ is the height above the load, L, and θ is the angle of the concave cone.

Assuming the pressure is constant within 10 percent, the centroid of pressure and area will be close.

$$g(r,\phi) = \text{constant}$$

$$\bar{r}(\phi) \cong \bar{r}\,(\text{area})$$

for a pie shaped segment:



$$\bar{r} = 2/3r\,\sin\alpha/\alpha$$

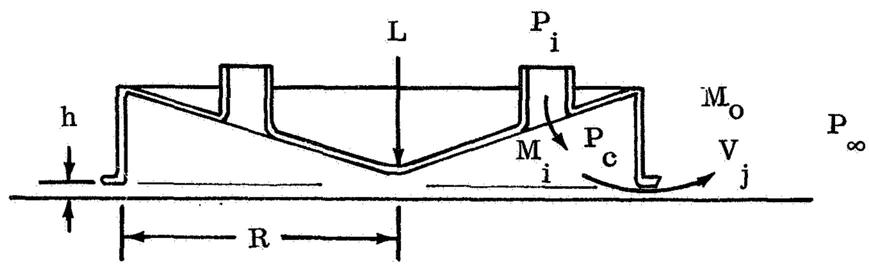
$$\phi = 2\alpha$$

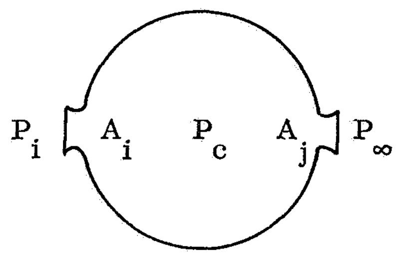
Therefore $\bar{r}(\phi) = 2/3R\sin\phi/2 \Big/ \phi/2$ for small ϕ, $\sin\phi/2 \cong \phi/2$.

Therefore $\bar{r}(\phi) \cong 2/3r$ and $h(\phi) \cong 2/3R\sin\theta$ for the test air pad,

$$h(\phi) \cong 2.4 \text{ inches.}$$

2. Any tilt of the air bearing lift pad from the horizontal position produces a pressure distribution between the high and low sides causing a rightening torque on the air pad.





Since the working pressure is less than 15. 2 cm (6 in.) of water for any of the tests conducted, sub sonic, incompressible flow equations will be used in this analysis.

The change of mass in the compartment is equal to the difference of the inflowing and outflowing mass.

$$\frac{\partial m_{cv}}{\partial t} = \sum_i \dot{m}_i - \sum_i \dot{m}_o = \dot{m}_i - \dot{m}_o = 0$$

$$\frac{\partial m_{cv}}{\partial t} = \frac{\partial}{\partial t} \int PdV$$

Since we consider the equilibrium case, the inflowing mass must be equal to the outflowing mass.

$$\sum \dot{m}_i = \sum \dot{m}_o$$

$$m_i = m_o$$

Furthermore, the weight of the air bearing and load, L, must equal the lift produced by the pressure.

$$\Delta P = P_c - P_\infty$$

it is

$$L = R^2 \pi (P_c - P_\infty)$$

The mass flowing out of the compartment is proportional to the pressure ratio P_∞ / P_c provided the velocity at the periphery of the air bearing is nowhere greater or equal to the speed of sound. The relation is

$$\dot{m}_o = m_i = A \cdot K \cdot \sqrt{2 P_c \rho_c} \left[\frac{P_\infty}{P_c}\right]^{\frac{1}{\gamma}} \sqrt{\frac{\gamma}{\gamma - 1}} \left[1 - \left(\frac{P_\infty}{P_c}\right)^{\frac{\gamma-1}{\gamma}}\right]^{\frac{1}{2}}$$

where

$A = 2R\pi \cdot h$	Gap area
$h =$	Gap height
$R =$	Radius of compartment
$K \approx 0.62$	Flow coefficient
$P_\infty =$	Pressure outside of compartment
$P_c =$	Compartment pressure
$\rho_\infty =$	Density of gas outside compartment (air)
$\rho_c =$	Compartment gas density (air)
$\gamma = 1.4$	
$L =$	Weight on air bearing

If we set

$$\dot{m} = 2R^2\pi \sqrt{\frac{2\gamma}{\gamma - 1}} \sqrt{P_\infty \rho_\infty}$$

and

$$\dot{m}_i^* = \frac{\dot{m}_i}{\frac{h}{R} K\dot{M}}$$

furthermore

$$\frac{P_\infty}{P_c} = \frac{P_\infty R^2 \pi}{L + P_\infty R^2 \pi}$$

we obtain finally

$$\frac{L}{P_\infty R^2 \pi} = \left[1 + \left(\frac{\dot{m}_i}{\frac{h}{R} K\dot{m}} \right)^2 \right]^{\frac{\gamma}{\gamma - 1}} - 1$$

An approximation for small mass is

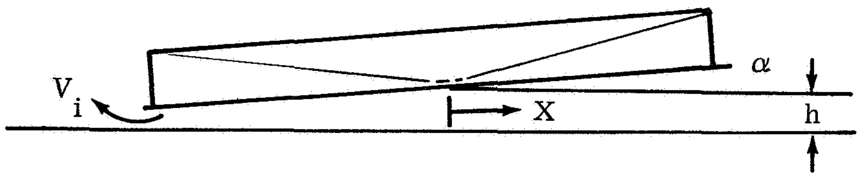
$$\frac{L}{P_\infty R^2 \pi} \approx \frac{\gamma}{\gamma - 1} \left[\frac{\dot{m}_i}{\frac{h}{R} K\dot{m}} \right]^2 \approx \Delta P \approx P_c - P_\infty$$

The air bearing can carry more weight by increasing the mass-flow m_i through the duct system or by letting the gap height h become smaller. Also, for constant mass, flow is

$$\Delta P \left(\frac{h}{R}\right)^2 = \text{const.}$$

The static stability in pitch.

If the air bearing is pitched through the angle α the gap height h changes around the circumference. The mean gap height $\bar{h}$, however, stays the same, since the mass flow $\dot{m}_i$ and the load L on the air bearing does not change.



The jet velocity V_j at the periphery is every where the same or roughly the same.

Since it is only a function at the pressure difference P_c/P_∞, the mass flow changes around the circumference.

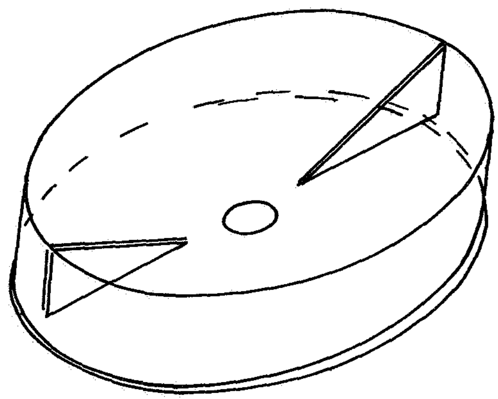
$$\dot{m}_o(x) = \rho V_i 2\pi R h(x)$$

This change in mass-flow causes changes in the pressure distribution inside the compartment.

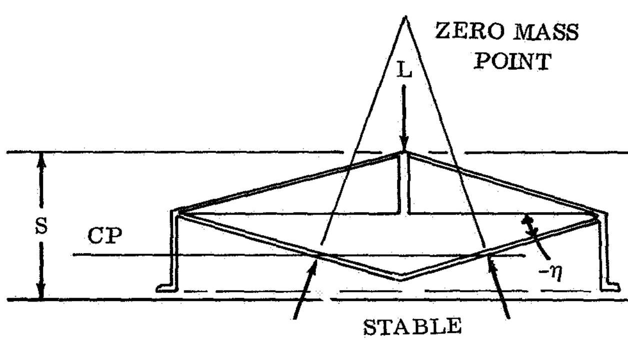
The pressure differential can be intuitively seen by loading the air pad to reduce the height, h, to zero. Then the pressure in the air pad becomes equal to the inlet pressure, P_i. If h is increased to something approaching a diameter, 2R, the pressure in the air pad becomes P_∞.

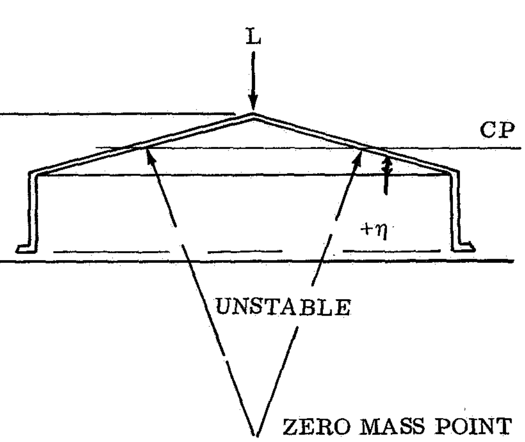
In order to sum moments and obtain a numerical value for the restoring torque, T, the pressure-distribution inside the air pad must be known. The testing was performed to verify the pressure-distribution, not to plot the variation.

The inverted cone is not only good structurally but tends to segment the air pad, yet still further stability can be obtained by further segmenting the chamber.



3. The shape of the top surface determines the distance below the zero mass point which the load is applied.

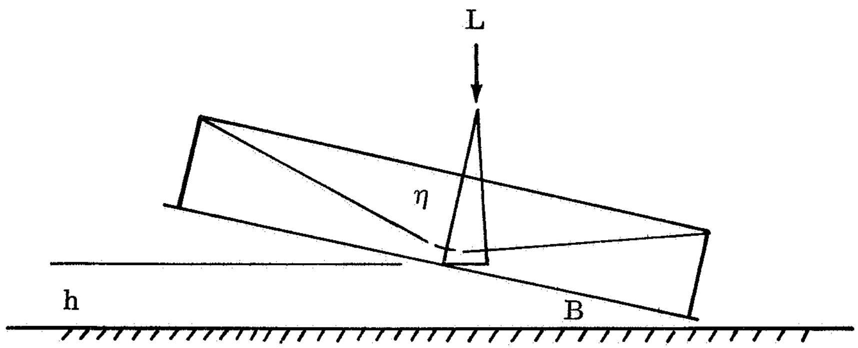




Sketch 3. 1

Sketch 3. 2

For any weight above the natural pivot point of the air pad, a destabilizing torque proportional to the load and height of load is produced.

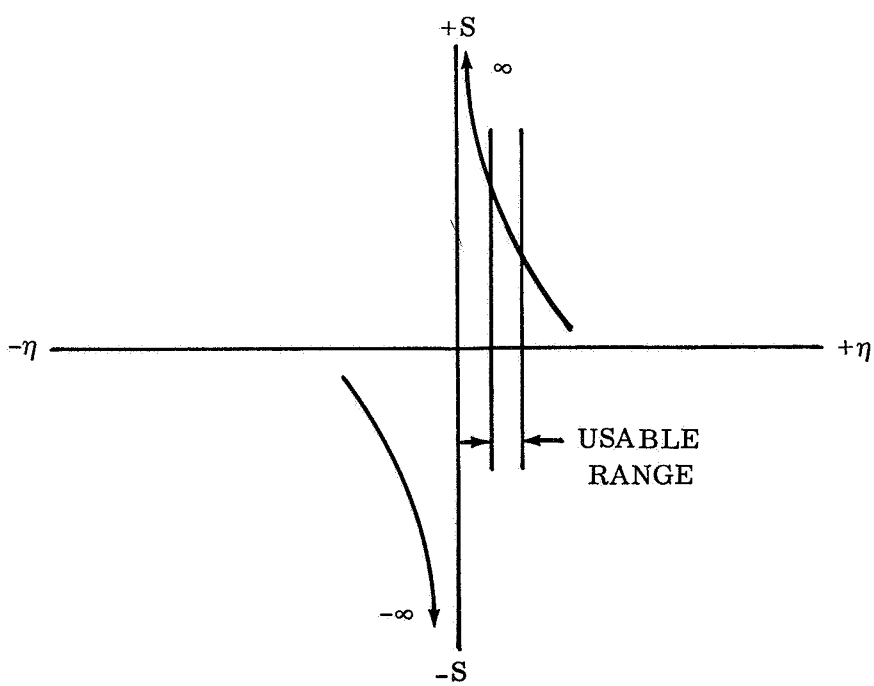


destabilizing torque = N L sin B

Define a point, P, at which any weight above the point produces a destabilizing torque greater than the stabilizing torques on the air pad.

From testing, for $L_1 N_1 \sin B_1 = L_2 N_2 \sin B_2$ sketch 3. 1 was still stable while sketch 3. 2 became unstable even with the center of pressure being higher than in sketch 3. 1.

Assuming the intersection of the centroid of lift of the top surface approximates the zero mass point, the point would plot similar to a cotangent function.



As seen in sketches 3.1 and 3.2, increased stability is obtained by loading below the zero mass point. The cotangent function becomes undefined at 0 degrees and h increases without limit at ±90 degrees. Within the working range of the function, $P = NL \sin B$ = maximum constant for any air pad. The smaller hL becomes, the less destabilizing torque there is on the air pad.

NASA-Langley, 1969 —— 15 M800

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