SURFACE IMPULSE PROPULSION

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ABSTRACT

Surface impulse propulsion, or traction, is a propulsive concept that has been around since man first harnessed a prime-mover to a wheel. It has evolved to its present level of sophistication through a process of mixed empiricism and analysis. Only in the case of prepared surfaces has analysis proven of reasonable designer utility. In the off-road and multi-terrain vehicles the problem is significantly more complex and can only be addressed by resorting to testing and in many cases to cut-and-try methods.

The paper is structured in three parts. The first delineates the pertinent variables involved in traction propulsion in off-road situations in general, and air cushion vehicles in particular. The second phase of the paper focuses on the specific problems of integration of traction systems in air cushion vehicles. Discussed are aspects of volume, weight, system dynamics, and power transfer. The third and last part identifies a number of innovative craft-integrated surface impulse propulsion concepts and projects their application scenarios.

THE DEVELOPMENT OF THE AEROBAC

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ABSTRACT

The AEROBAC is an offroad and amphibious transport system under development. It combines air cushion and tracks technologies. The AEROBAC is an idea of the Transportation Development Centre and is being industrialized by a group of private firms under the leadership of SNC, with Bombardier and SEDAM.

The AEROBAC is an hybride vehicle integrating caterpillar traction with air cushion lift. This vehicle is designed for regions where low density populations and a limited industrialization do not justify the construction of roads. The AEROBAC requires only a trail road without surface improvement. A preliminary design of a 7 tonne payload capacity has been undertaken. The development of the AB-7 prototype is underway, which is expected to operate at up to 30 km per hour.

Following the conclusion and recommendations of a preliminary development study, performed by SNC, a technical feasibility study was undertaken by SNC with BOMBARDIER & SEDAM as sub-contractors. This study showed that the AEROBAC in the 7 tonne version is feasible. The vehicle characteristics and the preliminary design have been completed. This study also demonstrated how two different technologies, the track system and the ACV system, could be integrated in one vehicle. This exercise has also been useful in the transferring of the French SEDAM air cushion system technology to this Canadian development.

The AEROBAC has a number of potential applications such as the transportation of high power transmission lines for Northern area; airport fire fighting and rescue operations; transportation of people for Northern land; as a service and transportation vehicle for mega projects development such as pipeline, construction, mining, exploration, railroad construction.

The paper will present the basic criteria used to develop the vehicle, the criteria for design, the operation and exploitation. It will include the presentation of an industrialization plan for the implementation of the AEROBAC.

HOW TO IMPROVE AIR CUSHION VEHICLE PERFORMANCE WITH VUMP EQUIPPED WAVE-FORMING KEELS

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ABSTRACT

VUMP equipped Wave-Forming Keels can give air cushion vehicles a means of adding hydrodynamic lift to air cushion lift. The planing surfaces of the W-F Keels are lubricated by air pumped by the venturi action of the VUMP. The concept is similar to a plan for an air cushion hydroplane prepared by Lt. Muller-Thomamuhl in 1915, but unlike his concept, does not require a blower--rather, it is powered by the forward motion of the air cushion vehicle acting on the VUMP.

INTRODUCTION

Although considerable attention has been given to increasing speed and reducing fuel consumption for air cushion vehicles, there remain deficient areas of performance. For example, increased speed brings greater skirt wear and also with greater cushion height fuel costs increase. As with any development, there are problems of ambiguity associated with it.

With any human effort, we find the crux of the technical problems lies with the attitudes of the technical and operational leaders regarding which means should be used to solve problems.

THE CONVENTIONAL "WISDOM"

Most technical people do not believe that large improvements in air cushion vehicles are likely. This attitude stems from three basic, if fairly weak, assumptions:

- o Improvements will follow, not precede air cushion development;
- o New concepts will have their origin only in the air cushion technical elite;
- o No matter where new ideas are found, the burden to produce an institutional advocacy is with the originator of new concepts.

These assumptions are widespread and deeply ingrained with many of the air cushion community people. It is not likely that these assumptions can be modified so the potential harm is real.

On the other hand, in all fairness, the goal of this paper is to propose some old and new concepts. Indeed, it is hoped that the reader will find these assumptions flawed.

DEFINITIONS OF WAVE-FORMING KEEL AND VUMP

Wave-forming Keels are free-flooding structures located on the bottom of an air cushion vehicle with the purpose of adding planing lift to the lift of the air cushion. The W-F Keel structure should be "hardmounted" to the framing on the vehicle bottom; that is, the transverse frames are aligned with the transverse structural bulkheads of the hull's bottom or orlop deck. This practice will allow high impact loading in a seaway because the W-F Keel structure <u>will</u> stiffen the longitudinal framing system.

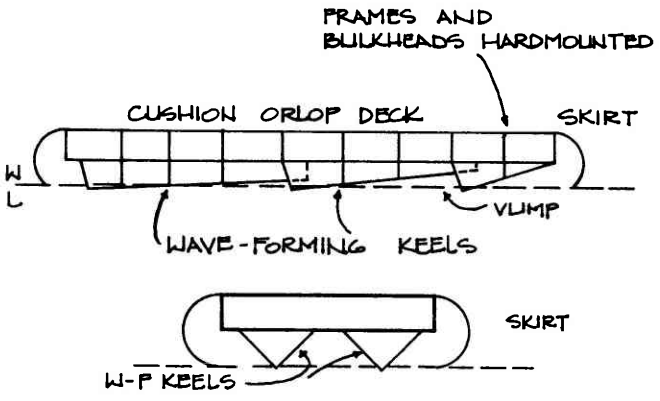


Fig. 1 W-F Keel and VUMP

The W-F Keels are not part of the floatation structure, and are designed to carry planing and impact loads, hence can be built to very lightweight specifications.

Now, what is a VUMP? This concept belongs to Bernoulli who first observed that the higher the speed of a flowing fluid the lower the pressure. The VUMP employs a mixed fluid, air and water. When operated at design speed, the water flow pumps air. To pump air it has to form a low pressure compression wedge in the flowing water; thus, the name Wave-Forming Keel. Figure 2 shows the VUMP in operation.

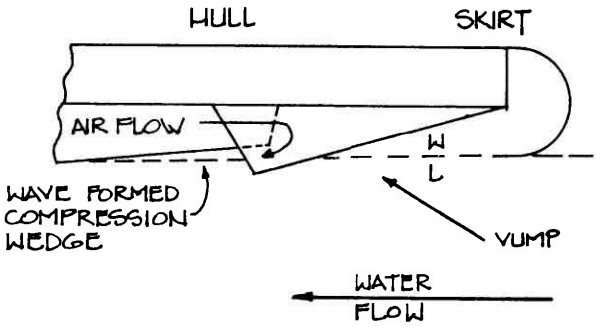


Fig. 2 VUMP Schematic

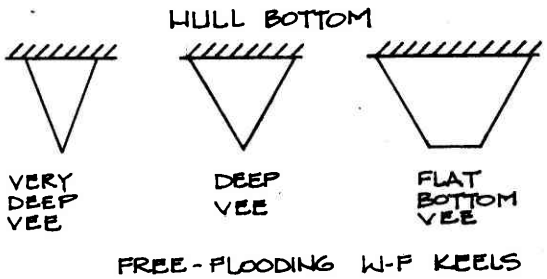


Fig. 3 W-F Keel Cross-Sections

THEMATIC HISTORY OF W-F KEEL CONCEPT

The Frenchman Campet built the first planing hull boat which really worked in 1905, followed in 1908 by another Frenchman Tellier who built the first stepped bottom planing hull boat. Both features are combined in the W-F Keel with an added feature using a free flooding structure to provide an air path to the Venturi wedge. As with any planing surface, wave-making frictional resistance is reduced. There is a potential for up to 60%-70% of the buoyancy to be replaced by dynamic lift at high speeds.

Planing hull boats usually have a transom where the wake breaks away, giving the appearance from the side view of a step. When the step is located forward on a planing hull it provides aeration that serves to reduce frictional resistance, a feature used for many years in boat design.

The operating mode of the W-F Keel owes its concept to an Austro-Hungarian Lt. Muller-Thomamuhl who, in 1915, prepared plans for craft that combined a hydroplane with an air cushion.

This is what is taking place along the W-F Keel; it is an air lubricated hydroplane surface. But, instead of a pump to supply air to the lift surface, the Bernoulli effect of the VUMP is used to supply aeration and venting. Therefore, a VUMP is a combined Vent and pUMP.

DESIGN FACTORS

Trade-offs are intrinsic to any design process and are driven by basic physical laws that can be aerodynamic or hydrodynamic or both. Any design effort should emphasize simplicity and standardization in its design by adopting performance limits well within the state-of-the-art. For example, SEAKEEPING is a very important factor in the design of air cushion craft. Below are some of the possible cross-section shapes for W-F Keels.

These shapes and others can be selected to reduce pitch and roll amplitude when planing in a seaway. Selection of the shape can be based on maintaining laminar flow, giving superior high-speed hydrodynamic performance. The goal is reduced fuel consumption of the power plant.

FRICTIONAL RESISTANCE

W-F Keels give a means of reducing the water plane area. As with any hull, the W-F Keel's resistance is strongly influenced by displacement-to-length ratio:

$$\frac{\text{W-F Keel Displacement}}{(\text{Speed Factor}) \times (\text{Length})^3}$$

The speed factor is related to W-F Keel speed and seastate. This means that the lift fraction planned for the W-F Keels should be in the range of 25% to 50% of the air cushion lift.

It should be kept in mind that frictional resistance of the W-F Keels will be greatly reduced by the action of the VUMP, about 25% less than without it. Also, with speed-to-length ratio $(V/L^{.5})$ of above 5.0 the W-F Keels have less wave-making resistance. In summary, at high speed wave-making resistance is dominate, and at low speeds frictional resistance is dominate, when the action of the VUMP is not considered.

PROGRESS IS A CONTINUUM

We must look to the past, as well as the future, for ideas that bring synergistic compounding of benefits. By encouraging competition between designers, promoting technical exchange between the hydrodynamic and aerodynamic design community, the air cushion vehicle can be improved.

Vehicle design should not be considered secondary to other technological considerations; for example, to power plant and lift system design get most attention at the present time.

FACTORS FOR CONSIDERATION

1. The West finds itself in a position of numerical inferiority with respect to the Soviet bloc, therefore, large air cushion watercraft can be used to successfully compensate for this shortage of weapon platforms.

2. Cushion pressure and cushion height may be reduced if load sharing with the W-F Keels is used. There can be fuel savings, too.

3. There is a large store of empirical evidence upon which to base W-F Keel designs because the hydrodynamic laws that apply have been in the public domain for many years.

4. Planing hull and air cushion design can be "fused together" in dialectical unity.

5. The twin W-F Keel configuration for added lift to air cushion craft should be considered as an alternative to the high-technology approach so often used to improve performance today.

CONCLUSION

The question remains, "Will a hybrid air cushion craft really be better?" I would say Yes and call for a rapid follow-up on this concept.

AN INNOVATIVE TYPE OF ICEBREAKER FOR THE ARCTIC ENVIRONMENT

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ABSTRACT

An innovative icebreaker is presented for the Arctic environment using the Semi-Submerged Air Cushion Vehicle (SSACV) concept. This type of vehicle combines the icebreaking capability of the air cushion to the traditional method of mechanical icebreaking used by conventional icebreakers.

The paper converts the concept into documented theory. It covers the design calculations of the principal elements and discuss some possibilities of enhancing icebreaking operations through refinements made possible by the SSACV concept.

It will be necessary for model tests to be carried out before theoretical results and hypothetical claims can be confirmed, as the SSACV is outside the range of application of the usual ice predictor equations.

The paper is based in part on a feasability study made by German and Milne for Transportation Development Centre, comparing an SSACV LNG tanker to a conventional design.

1. INTRODUCTION

Air cushion Technology has been successfully employed in Canada since 1971 for icebreaking purposes using commercial self-propelled ACVs and non-propelled air cushion platforms. Currently, there is a considerable effort directed towards the development of specially designed platforms coupled to the bows of host ships in order to increase their icebreaking capability. These air cushion devices are of the fully skirted type.

The platforms operate by removing water support from underneath the ice cover. As speed of advance increases wave action amplifies this effect. At the higher speed of the self-propelled ACV's the wave action is prevalent with maximum effect occuring slightly below hump speed [3]. The overall effect is to induce pure flexion rupture of the ice cover.

Trial data, reported in reference 1, shows that ice fracture will occur as long as the pressure head is of the same order as the ice thickness and as long as the air cushion platform exceeds a minimum size, depending on ice thickness. Ice rupture could presumably be obtained at lower pressure than specified here if the size of the cushion is very much larger than the minimum [3]. Such conditions could produce the required rupture moment in the ice; however, this has yet to be tested experimentally.

It would appear therefore that it is only necessary to increase size and pressure of the air cushion in order to cater for the arctic ice.

One would expect that the very large cushion depth and pressure needed for Arctic operation may bring some cushion stability problems. Fortunately operation over water have different boundary conditions (damping at skirt water interface) which tend to increase stability limits. It is a proviso of the present analysis that the proposed air cushion is within the over water stability limits or that, by the time such large ACV are built, artificial damping mechanisms will exist.

The Arctic environment will vary under summer and winter conditions and will be usually composed of uniform and non-uniform ice layers of different thickness, age and strength, major hard spots resulting from pressure ridges and hummocks, partial ice cover, ice floes, broken ice pieces and, sometimes, even icebergs. However, in the present discussion, we will only consider continuous motion through a uniform layer 7 ft thick. In other words, we will be considering a Polar Class 7 icebreaker. But some qualitative remarks and hypothetical comments will also be made relating to operation under ridged-ice conditions.

This paper is based in part on a comparative study by German and Milne comparing the performance and operating costs of two 140 000 m^3 icebreaking LNG tankers, one of conventional design the other using the SSACV concept (Fig 1). This study was prepared for the Transportation Development Centre, Transport Canada, TP 3423 [2].

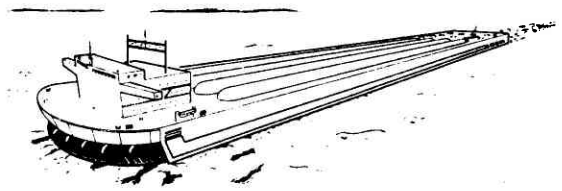


Figure 1, Conceptual Design SSACV LNG Tanker

2. SSACV ICEBREAKER CONFIGURATION

A general diagrammatic perspective view of the SSACV icebreaker is shown in Fig 1. This particular application relates to an icebreaking tanker.

2.1 Upper Hull

The upper hull is rectangular in planform with a roughly semi-circular bow section located at an adequate heigth above the surface of the ice to clear ice ridges. Presumably this can be suitably reinforced to cater to the occasional contact. This height may also be adequate when the icebreaker is operated in open water for the prevention of green water impingement on its lower surface, the so-called "wet deck".

The upper hull with its raft structure will also have sufficient depth so that it will act as a buoyancy chamber, with adequate reserve buoyancy, in the case of a failure of the air cushion system combined with the unlikely case of a simultaneous flooding of both the buoyancy members, namely, the side struts and the submerged hulls. Thus, the vessel will now be able to float safely on the water and, in ice covered regions, the wet deck will be of sufficient strength for the vessel to land and rest on the ice layer without any damage to the structure.

2.2 Side Struts

The side struts connecting the submerged hulls to the upper structure are assumed to be parallel-sided over the middle half of their length and with lenticular sections of equal length at either end. The width of the flared struts is assumed to vary linearly from a minimum at the lower end carrying the submerged hull to a maximum at the upper end. This flare increases the stiffness of the vessel in heave over water, preventing excessive vertical excursion when operating the air cushion.

The flare provides adequate space for the structural members at the joint with the upper hull. At the water line the flare also mitigates ice wedging. The stems of the side struts must be reinforced for ice cutting, and possibly extended to the apex of the submerged hull to permit overriding of deep ice.

The struts will be as narrow as possible to reduce ice resistance, but with sufficient width to provide adequate pitch and heave stiffness, desirable transverse stability, space for structural members, fuel and ballast tanks and manned access to the machinery compartment located in the submerged hulls.

2.3 Submerged Hulls

The submerged hulls are assumed to be in the shape of a circular cylinder, say, over 90% of their length with a hemispherical nose end and a conical stern. This is, perhaps, not the best hydrodynamic shape for laminar flow conditions and, in any case, the pointed stern would not provide adequate room for the machinery and transmission. A transom stern in the form of a truncated cone or, possibly of elliptic crosssection may be more suitable for this purpose.

The choice of a suitable diameter is based on the following considerations:

(i) The buoyancy of the hulls should be sufficiently large so that the total displacement of the vessel in the fully loaded condition would not call for an excessive amount of air cushion lift in ice regions or an excessive immersion of the side struts in open water;

(ii) The combined buoyancy of the hulls and the side struts should not be too large as there should be some scope for the use of an adequate amount of cushion pressure (depending upon the thickness of the ice) for icebreaking purposes;

(iii) The buoyancy of the hulls should be of sufficient magnitude that when the buoyancy of the side struts is also taken into account, it will be possible to vary the draught of the vessel within some wide limits by a reduction of the cushion pressure from its design value and readjustment of the depth of the flexible seals below the hard structure; etc.

2.4 Air Cushion System

The air cushion is bounded laterally by the side struts and by flexible seals at the bow and at the stern. The bow seal will be of semi-circular shape with a rounded upper bag or loop carrying a multitude of fingers or segments and following the contour of the overhanging upper hull. This contour would of course be suitably reinforced to

protect the skirt attachment line in case of accidental contact with a ridge.

Thus, the bow seal will extend well forward of the stems of the side struts except, possibly, at the points of attachment thereon. With this substantial extension over the main rectangular part of the air cushion, we may hope that during over ice passage the usual radial and circumferential cracks will appear in the ice just ahead of the side struts, thus facilitating their icebreaking and ice-transiting. This geometry for the bow seal may call for tapered fingers or segments at the stems of the side struts and some special attention will also have to be given to the mechanism of skirt retraction. This is required since the air cushion will not be used during passage in ice free water. It is proposed that this mechanism be used for forcibly oscillating the vessel in pitch and heave during ramming.

The rear skirt will be located at the stern of the side struts so that the centre of pressure may lie roughly below the centre of gravity. Any forward location of the rear seal will shift the centre of pressure ahead of the centre of gravity and although this could be catered for by a suitable weight distribution, difficulties will arise when the vessel is operated in open water with the air cushion inoperative.

For the sake of simplicity, we will assume that the side struts and the submerged hulls are of the same length with the mean length of the air cushion being slightly higher, although it may be considered prudent to have the struts overhanging the nose ends of the hulls in order to minimise possible damage to the latter from ridges submerged below the general level. A similar overhang may also be desirable at the rear end so that damage to the propellers may be avoided when it is necessary to reverse the vessel during a ramming operation.

The cushion area is substantially determined by the overall dimensions of the icebreaker and the cushion pressure also specified from the thickness of the ice layer to be broken, preferably with an air cavity produced under the ice layer. The available air cushion lift is therefore determined and it will therefore be necessary to manipulate the dimensions of buoyancy members such that suitable draught conditions for efficient ice cutting and for open water operation may be obtained. This can be done by varying the lift fan speed. A variation of the depth of the flexible skirts could also be contemplated to supplement ballasting under various operating conditions, such as laden or empty conditions.

3. ICEBREAKING CAPABILITY

The icebreaking capability of the SSACV Icebreaker will be discussed in terms of air cushion icebreaking and of side struts icebreaking. It is however necessary to clarify a point of contention that came up in the feasability study of German and Milne [2], that is the fact that predictor equations showed that the SSACV, without its air cushion, could still be an exceptionnal good icebreaker, that is, compared to the conventional tankers. Some remarks are therefore in order on this point.

3.1 Icebreaking and Ice-transiting

It is known from operational experience that air cushion devices are capable of breaking the ice and clearing a path along the track of the vehicle a little wider than the beam. Assuming that the SSACV icebreaker is similarly capable of breaking ice over the entire planform area, it may be worth inquiring whether it is indeed necessary to clear a path over such a large area if the main objective is simply to move the vehicle through the ice layer with ease, i.e. if ice-transiting only is required.

Thus, if the objective of the icebreaking vessel is simply to transit the ice region, without disrupting the ice cover except where strictly necessary for passage, air cushion icebreaking may be considered an expensive luxury.

However, ice predictor equations were not developped for the SWATH ship (Small Waterplane Area Twin Hull) so that there is some risk in extrapolating the results. Therefore the air cushion should be retained as an alternative, at least for the ice-transiting cargo ship.

For an icebreaker, that is a ship designed for escort duty, the air cushion insures that following ships will travel in mush ice. Furthermore the heave oscillations that may be induced with the air cushion would do much to extricate the SSACV icebreaker from otherwise delicate conditions.

3.2 Air Cushion Icebreaking

The air cushion contained substantially by the side struts in the lateral direction will have a rounded bow extending well ahead of the stems of the side struts where they enter the ice layer. The length and width of the cushion will depend upon the overall dimensions of the vessel, which will in turn depend upon its role, i.e. whether as a pure icebreaker, or as an icebreaker capable of performing other duties, such as a supply and support vessel or as a tanker.

There are two distinct methods and speed regimes by which ice can be broken by air cushion vehicles, namely, the "high speed" and "low speed" methods. Carter [3] has mathematically identified the critical velocity for high speed icebreaking as 0.75 times the open water "hump speed" of the moving air cushion. Self-propelled ACVs can operate in this speed regime, but in the case of air cushion icebreaking bows attached to ships and, in our own case of an air cushion forming an integral part of the semi-submerged vessel, we will have to rely on the low speed method as the speed of progression in the ice layer will naturely be relatively slow and the situation may be considered as almost quasi-static.

We are considering an ice layer of 7 ft uniform thickness and taking 0.9 as the specific gravity of ice. It is clear that if we use a cushion pressure of 403 psf, equivalent to 6.3 ft of water, in order to balance the buoyancy forces supporting the ice sheet, there will be a cavity produced under the sheet which will facilitate icebreaking.

It has to be accepted that a considerable amount of power will have to be expended in order to generate and maintain a high cushion pressure of the above order and this will mean that the power available for propulsion will drastically be reduced if the lift and propulsion machinery are integrated and the total available power is a fixed quantity.

However, if we use a lower pressure for the sake of conserving lift power, it will still be possible to break ice against the resistive forces due to buoyancy when there is no cavity, but as Carter [3] has shown, the radius of load will have to be much larger. Similarly, the predicted distance for the appearance of the circumferential cracks in the ice layer will also be greater. But these are adequately catered for if the air cushion is of large dimensions.

The lower cushion pressure contributes to the reduction of the cushion wave drag. At the low speed of icebreaking this wave drag is more or less equivalent to the force of air pressure acting on the face of the entering ice floe. We have not yet sufficient experience in air cushion icebreaking to evaluate this resistance on the SSACV, or on ACIB's for that matter.

3.3 Side Strut Icebreaking

The side struts are the SSACV icebreaking members when the air cushion is not in use. In order to evaluate their resistance we must resort to the literature since at this time we were not able to find ice resistance data that could apply to our design. The formulation of icebreaking resistance by Edwards et al[4], derived from fullscale and model tests of the CCGS St-Laurent by regression analysis, is typical of monohull icebreakers:

$$R = \rho g B h^2 \left(4.242 + 0.0503 \frac{\sigma}{\rho g h} + 8.948 \frac{V}{\sqrt{gh}}\right) \quad (1)$$

where σ is the ice flexural strength
h the thickness of the ice layer
ρ the unit mass of water
B the ship beam
V the speed through the ice

The dimensions must be compatible to make the equation non-dimensional.

For our calculations we will be using this equation and a similar earlier one by Lewis and Edwards[5], in order to provide a cross check. Until we have ice model test data on a SWATH geometry, this is the only approach left to us. We are aware of the limitations.

Generally the examination of equation 1 lead to the following interpretation

i) Resistance is proportional to beam
ii) The first term in the right hand side accounts for submergence
iii) The second term accounts for the ice rupturing force
iv) The last term accounts for the velocity effect.

One must remember however that the regression analysis retained these terms as significant in the data base available. It would therefore be prudent to regard the interpretation as a working hypothesis. Other experimenters may have other interpretation from their data base, and with the same statistical confidence.

Nevertheless the above predictor will be used without alteration when the SSACV is used without the air cushion. However, from the discussion of section 3.2, it is clear that the air cushion improves the capability of the side struts in moving through the ice: the cushion will rupture the ice field ahead of the stems of the strut, turbulence

within the cushion space may alleviate internal ice friction, etc.

We may therefore assume that we can reduce values of the coefficients of the various terms in equation 1 on a sliding scale depending upon the value of cushion pressure used, i.e., say, when the pressure is 0, 100, 200, 300 or 403 psf, the air cushion being most effective when the highest pressure is used. Such an assumption is justified, qualitatively, by the performance of the ACT 100 (1976-1977) [1]. Nevertheless the assumption is obviously risky but we have no other choice at this time.

3.4 SSACV Possibilities

The skirts of the SSACV are per force long and heavy. As they will be an hindrance during cruising in open water, some mechanism, yet to be designed, must retract the skirts. It is therefore tempting to use this mechanism as a mean of inducing heaving or pitching of the SSACV. This would permit easy withdrawal of the hulls after a ramming run and possibly amplify side strut icebreaking force application.

One could use also the mechanism to modify draft quickly, in order to clear ridges of moderate height.

However, this is hypothetical at the moment and detailed calculations and model tests will be necessary to find what is practical or not.

4. PERFORMANCE CALCULATIONS

The air cushion will not be used in open water operation and the calculation of performance is straightforward. The skin friction coefficient, C_f, may be calculated from, say, the ITTC (1957) formula with perhaps an additional roughness allowance of 0.0004. In the absence of the air cushion, the SSACV reduces to a SWATH ship and an estimate of the residuary resistance coefficient, C_r, may be obtained from the results of model tests of the latter type which may be available. The calm water performance may thus be derived leading to a prediction of rough water behaviour again on the basis of SWATH ship test results. It will be found that the speed loss of the SSACV Icebreaker in rough water is less than that of conventional monohull types.

The resistance in ice regions will be composed of:

(i) Resistance due to the air cushion moving over the ice layer;
(ii) Resistance of the flexible seals possibly contacting the ice cover
(iii) Resistance of the buoyancy members moving through water under the ice; and
(iv) Resistance of the side struts moving through the ice layer, breaking the ice and dispersing the broken pieces.

In addition, we may include

(v) effective drag due to lift power, which is simply the lift power divided by the forward speed.

As regards (i) above, the components are momentum drag and wave resistance. These components as well as the lift power at (v) can be calculated by standard methods. It will be found that the momentum drag is negligible compared with the other quantities. This also applies to the wave resistance at such low forward speeds, particularly when appropriate wave steepness limitations are imposed. It is not possible to estimate the resistance of the flexible seals in the absence of model tests.

The dominant components of resistance are (iii) and (iv). As regards the former, the wave resistance at low forward speeds will be found negligible compared with skin friction. The ice resistance can be estimated using the predictors (1) and (2) for different values of the cushion pressure with corresponding modification of the coefficients as described in Section 3.3. A performance estimation is detailed in this manner in Annex C of reference 2, "SSACV Icebreaking LNG Tanker Feasibility Study".

This study, by German and Milne, used a more conventional approach to the resistance calculations. The ships characteristics are given in Table 1, while Table 2 compares cost and performances data for base ship and SSACV, with and without air cushion in 2,13m (7 ft) of level ice. Fig 2 shows power-speed relationships calculated by the two methods (G&M, TKSM).

TABLE 1

	Base Ship	SSACV
Length	335.0m	350.0m
Beam	43.0m	73.4m
Depth	29.5m	39.4m
Draft offcushion	13.0m	26.0m
oncushion	—	20.0m
Displacement	140 000 t.	
Total Power	80 000 kW	

TABLE 2-COMPARATIVE COST AND PERFORMANCE (G&M)

Item		Base Ship	SSACV Tanker	Gain
Construction Cost	M$	296	335	13%
Annual Operating Cost (less fuel)	M$	15.5	17.6	13%
Manning		45	51	6%
Fuel Cost per hour	$	3,168	3,168	-
Speed in Open Water	m/s	11.5	10.5	-9%
Speed in Ice	m/s	1.6		
-cushion off			5.5	243%
-cushion on (19.4 kPa)			6.1	281%
-cushion on (9.6 kPa)			7.1	344%

Equal total power of 80,000 kW is assumed for all case, with 47,000 kW used in the air cushion and 33,000 kW on propulsion for the SSACV with full cushion on.

Speed attained in level ice is 3,1 knots for the base ship, 10,7 knots for the SSACV without cushion (80,000 kW on propulsion) and 11,8 knots for SSACV on cushion.

Based on experience with the ACT 100 ACIB, the power used in the cushion is excessive: transfer of 25,000 kW to propulsion would give a speed of 13,8 knots for the SSACV [2].

5. CONCLUSIONS

The feasibility study [2] and the calculations carried out in Annex C of this study, show that somewhat large difference may exists in predictor equations with SWATH or SSACV icebreaking ships.

Nevertheless, the improvment in performance of these two types of ships over the base ship is surprising. The logical step is now to obtain, from models, a confirmation of the predicted performance and, perhaps, a better evaluation of the various components of icebreaking resistance, with and without air cushion.

The design of an SSACV icebreaker will involve an optimisation of cushion pressure and draught conditions in order to minimise the total power for lift and propulsion, i.e. to improve ice-transiting speed for a given level of total installed power.

The SSACV icebreaker appears to have considerable potential. The geometry provides a feasible method of integrating

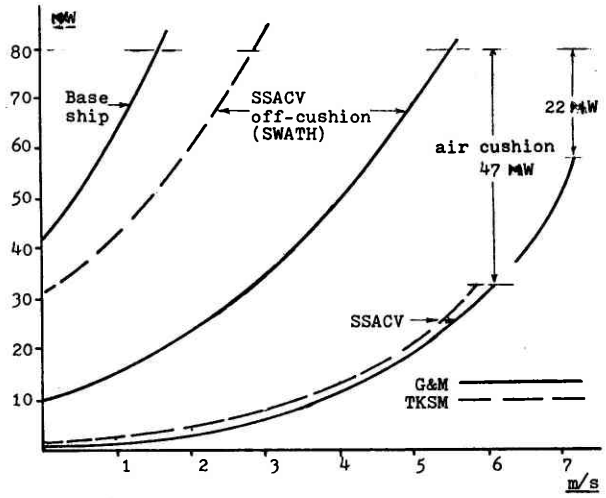


Fig 2-Ship Speed vs Propulsion Power

the air cushion into a ship. If the problems of large, high pressure air cushion can be resolved, the SSACV icebreaker could well be a candidate for Polar operations.

6. ACKNOWLEDGMENTS

The foregoing developments were evolved during the feasibility study [2] carried out by German and Milne for the Transportation Development Centre, of Transport Canada, Montréal.

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ROTATING WING IN GROUND

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ABSTRACT

This paper describes hovercraft Finn-Craft OH-01 and the events which led to its conception. This hovercraft, in the physical scaled-model stage of its development, is a segmented-skirt air cushion vehicle enhanced by a helicopter-like rotating wing on the roof. The idea arose from a need for a narrow, yet stable, vehicle.

In addition to potential commercial applications its ability to move close to the surface at high speeds and fly over obstacles gives rise to military applications.

PERFORMANCE OF THE ADVANCED TWIN GIMBAL FAN AEROMOBILE 16 ON VARIED TERRAIN IN 1982

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ABSTRACT

Lift systems of ACV's are well developed by many researchers. The major obstacle to widespread utilization of this promising concept has been lack of controllability, severely limiting the utility of, and demand for, the ACV. Adequate control requires forces of propulsion magnitude available for both propulsion and control directable in all 360° horizontally from both bow and stern of the craft. In the gimbal fan concept, 100% of the fan discharge goes into lift; when the fan is tilted in any direction, part of the slipstream is directed over the deck of the craft producing powerful steering and propulsion force in the desired direction while maintaining adequate air volume and pressure to the lift chamber below the deck. The thrust force can be finely metered and directed by the pilot and instantly cut off or reversed. Many other design advantages are inherent in the gimbal fan concept.

INTRODUCTION

The pervading objective of Bertelsen research and development is to make air cushion vehicles as widely available to every class of user from sportsmen to heavy industry as are boats, autos, and aircraft. As everyone familiar with the properties and qualities of the ACV knows, they have wide potential applications[1]. ACV's will complement, supplant, or replace other modalities of transportation all over the world, and do it cost-effectively, if we can overcome the problems of ACV design.

CHARACTERIZATION OF PROBLEM: THE LIFT, PROPULSION, CONTROL PARAMETERS

One might characterize the basic unpropelled ACV lifting base as a "slippery mass" which slides down the slightest grade or strays in the mildest wind. One might characterize the typical conventional propelled ACV as a costly, complicated, poorly steered, balky "slippery mass" whose mission is consequently limited to level surfaces such as water. Land missions which would be especially useful over snow cover are precluded because of inadequate control on grades, in wind, and on uneven surface.

The ACV, to be widely available, must be useful and capable enough to warrant volume production. Only volume production will keep first cost within the reach of all potential users.

The control and reliability problems are well stated in an article by A. Blunden in Nov., 1979, United Kingdom Hovercraft Society Bulletin in these excerpts:

"The 1979-80 season of UKHS lectures opened on September 19, when Captain Tony Smith (Director of Hovertravel) presented 'A Driver's View of Hovercraft' in London. Hovertravel operates three SR.N6 Mk 1S's and a standard SR.N6 on its Ryde/Southsea and Cowes/Southampton routes and also currently has an SR.N6 based in Abu Dhabi on a seismic survey contract...The rudders and 'puff-ports' on the SR.N6 are, he said, 'perfectly adequate in normal conditions' but when a craft is travelling downwind in moderate to high winds, this combination is not good enough to maintain straight line operation and drivers have to resort to alternating increases in power, to increase the effectiveness of the rudders, and propeller pitch. Captain Smith pointed out, though, that all the problems of operating the SR.N6 on Hovertravel's routes could be eliminated, or at least alleviated, if a twin-propeller craft was used. Another advantage would be a reduction in noise levels...'even the best of engines fail sometimes and we have had our fair share of failures.' On the Solent this is unfortunate but at least the passengers can normally be transferred to another SR.N6 and the crippled craft towed back to base ...During overseas contracts, however, this is not always possible. To illustrate this, Captain Smith described an incident in Abu Dhabi last year when a Gnome seized while the SR.N6 was operating in a shallow water area. The craft was allowed to beach on a sandbar (where the engine was replaced) before it could return to base. The whole operation had to be mounted, Captain Smith said, because the SR.N6 has only one engine. Any new design, he feels, should have at

least two so as to allow it to get back to base or at least reach the shoreline."

Other examples of inadequate control include reports that dirigible-type mooring techniques were required with "walkers" holding many attached lines to berth the SRN-5 and -6 in high winds in Northern operations. Another is the accident with the French Naviplane after ascending with its ramp, preparing to come down off hover, when a gust of wind blew it into the adjacent parking lot, crushing several automobiles.

The lifting base of the ACV is well along in development, but lift, propulsion, and control, from simple, inexpensive, effective means, have not been available.

In our work, culminating in the Aeromobile 16, we can achieve efficient lift and propulsion with complete controllability in yaw, steering, braking, pivoting, and selective, universal thrusting, with a simple vehicle which can be mass produced at reasonable cost.

Adequate control of the "slippery mass" requires forces of propulsion magnitude available for both propulsion and control directable in all 360° horizontally from both bow and stern of the craft. Such forces and direction have been achieved with gimbal fans, a twin engine craft, therefore, with one gimbal fan mounted on the bow and one on the stern of the Aeromobile 16. The two gimbal mounts have horizontal freedom in two degrees. When the fan shaft is vertically oriented, 100% of the fan discharge goes into the lift chamber. When the fan is tilted in any direction, part of the slipstream is directed over the deck of the craft, producing powerful steering and propulsion force in the desired direction while maintaining adequate air volume and pressure to the lift chamber below the deck. The thrust force can be finely metered and directed by the pilot and instantly cut off or reversed merely by appropriate fan axis tilting.

Along with the solution of the vexing control problem, many serendipitous advantages of ACV design are inherent in the gimbal fan concept:

1) The ducted gimbal fan with low tip speed is quieter than propellers and is inherently safer to personnel.

2) The craft has a lower silhouette and center of gravity than the open propeller thrusted craft because of smaller diameter fans which are one-third immersed in the deck well at 90° tilt.

3) The thrust line is also very low and close to the center of gravity reducing well-known ACV pitching and rolling tendencies and "plow-in."

4) The gimbal fans when mass produced like outboard motors will be lower in cost than "built in" systems requiring remote drives, transmissions, rudders, etc.

5) The fan units can be removed quickly for servicing or replacement. They can be installed in single, twin, triple, or quadruple units around the hull to provide craft of two to twenty passengers.

6) Each additional gimbal fan multiplies the lift, propulsion power, controllability, and reliability of the craft since each unit has inherent lift, propulsion, and control.

7) Having all engine fan units at the extremes of the hull leaves the center of the craft (the hold) totally unobstructed for payload on- or off-loading without trim changes.

8) The low silhouette of the gimbal fan ACV without high fan shrouds, without high transmission housings, especially without vertical tail, prevents wind rolling effects and weathercocking which incapacitates the conventionally thrusted and steered ACV.

THEORETICAL PERFORMANCE OF THE GIMBAL FAN AEROMOBILE A-16

The keel of the A-16 was laid in 1976. Its first field testing was done in 1978 with two 125 HP Mercury Marine six-cylinder in-line engines and 36" diameter eight-blade ducted gimbal fans. We have had problems with transmission reliability.

The first fan blades were die-cast aluminum alloy grain dryer blades of low efficiency.

After many revisions, we have two reliable, efficient UniRoyal HTD belt drive reduction transmissions; both have run many trouble-free hours on the test stand with 150 HP Mercury V-6 150 HP engines. The gimbal fan drive problems are finally solved.

As of July, 1982, we are awaiting the fabrication of new fan blades of polyurethane on steel mandrels by T. M. Development, Inc., of Chester, Pennsylvania, and the forging of aluminum hubs by J. T. Ryerson & Sons, Inc., which will have to be machined after arrival.

The new fan was designed by Dr. Helmut H. Korst, specialist in turbomachinery at the University of Illinois. The computer-projected static thrust output of the new ducted fan with 150 HP is 581 lbf thrust at 2400 RPM. The new fan should be quiet with tip speeds of 400 ft/sec. As can be seen from the photos of the gimbal fans of the A-14 (Fig. 1) and the A-15 (Fig. 2), the gimbal installation provides about 30% of the fan discharge for lift at maximum fore-aft tilt. Each fan will theoretically produce at maximum tilt 406 lbf thrust after lift air deflection.

Table I shows the masses of the various A-16 components as basis for calculation of the theoretical performance of the vehicle given the twin gimbal thrust

output and neglecting surface friction of the skirt and wind force.

Table II shows the projected linear acceleration capability of the A-16 on smooth ice, i. e., skirt friction drag is considered to be negligible. The subscripts 1 and 2 in the tables correspond to calculations based on payloads of 1500 lbm and 2000 lbm, respectively. Maximum force available from an installed gimbal should be approximately 70% of the computer projected maximum static thrust output. This means that up to 813 lbf thrust could be available for propulsion on the two-gimbal A-16 with each gimbal thrust axis tilted full 90^0 to horizontal and directed in unobstructed directions. In Table II the first column lists percentages of the computer projected maximum thrust output (T max). The second column is corresponding calculated thrust in lbs force. The third column is the acceleration of the A-16 with 1500 lbm payload (5765 lbm gross wt.) under the percent and lbf thrust in the first two columns. Column 4 shows accelerations with 2000 lbm payload (6265 lbm gross wt.) at the percent and lbf thrust in the first two columns. Columns 5 and 6 show the time in seconds to reach 35 knots at the various total thrust force quantities, again neglecting wind forces and surface drag.

For instance, at 70 percent total thrust from both gimbals, the force exerted is assumed to be 813 lbf causing gross weight 1 (5765 lbm) to accelerate 4.5 ft/sec^2 (Column 3), and causing gross weight 2 to accelerate 4.2 ft/sec^2 (Column 4).

These forces will move the craft endways or sideways at these accelerations, neglecting friction and slipstream blockage by the cab, etc.

Table III shows the angular accelerations at different percent and lbf total thrust with two different payloads and gross weights.

Column 3 shows the $degrees/sec^2$ the craft will accelerate in yaw under the various percent and lbf of thrust in columns 1 and 2. For instance, if each gimbal fan delivered 70% of its 581 lbf thrust after lift air deflection, it would produce 290.5 lbf. If the forward gimbal fan were tilted 90^0 to starboard, the clockwise yaw torque acceleration for gross weight 1 would be 33.2 $degrees/sec^2$ (Column 3) and would rotate 16.6 degrees in the first second (Column 5). Gross weight 2 would accelerate clockwise 30.4 $degrees/sec^2$ (Column 4) and in the first second would rotate 15.2 degrees (Column 6).

Table IV shows calculations of air drag assuming the A-16 to be a flat plate both in frontal area and in side area, 83.5 ft^2 and 133.3 ft^2 respectively under standard drag conditions[2]. This is a conservative method because having the rounded, full, smooth skirt, the bulk of the drag area is relatively well streamlined. The gimbals and cab are more oblique and less streamlined and will have higher drag, but less than flat plate drag.

Neglecting again the skirt friction on the surface and considering now only air drag, the A-16 will reach a velocity at which the air drag equals the thrust force.

For instance, in fore-aft travel at maximum gimbal tilt, the thrust is 813 lbf (Table II). The A-16 without wind or surface drag should exceed 45 knots where the flat plate drag is 733.8 lbf (Table IV Column 2). Actually, the front gimbal slipstream is obstructed by the cab when tilted forward rendering it less effective, and direct fore or aft combined gimbal force will be less than if the gimbal tilt is somewhat lateral in azimuth to put the craft in a crab angle. Direct sideways or beam-on travel of the A-16 provides relatively unobstructed fan discharge.

Referring again to Table IV in Column 3, the A-16 should exceed 35 knots beam-on, neglecting wind and surface drag, because the flat plate drag of the craft sideways is 708.6 lbf. In any case the A-16 could stand still in hover in a cross wind exceeding 35 knots from any direction and of any operable velocity. Such control will be very important in geophysical survey, sounding, etc.

In regard to the slipstream obstruction by the cab when the gimbal is tilted to cause forward propulsion and the rear gimbal is tilted rearward for backing or braking, much thought has been given to the placement of the twin gimbal fans. We totally concur with Captain Tony Smith that there should be twin engines and propellers. But for maximum yaw control they should be spaced as widely apart as possible. Both bow and stern must be controlled for side hill negotiation, for instance. In variable roughness and divergent slopes, 180 degree opposite gimbal forces are often needed on bow and stern. Conventional twin propeller practice is to place two thrusters side by side at the stern. Such placing does clear both intake and discharge of the fans, but in our automobile-size A-16, it makes the minimum width of the craft too great for road trailer hauling, even with side deck folding, and the leverage is too small to be effective in moving the bow laterally. On a side hill one fan might be thrusting forward and the other rearward, cancelling any force resultant for advancing the craft.

In a larger size Aeromobile with three or four gimbal fans for 3000 to 5000 lbm payload, designed to be dismantled for overland or air transport,

the lateral spacing of the center of force can be very adequate and unobstructed. As stated earlier, from the standpoints of lift, reliability, propulsive power, and precise controllability, the more gimbal fans the better, since each unit is totally self contained and provides lift, propulsion, and control.

Figure 3 shows the A-16 in 1978 with lower power fans.

We expected to have completed field testing and quantification of the A-16 by the filing deadline for this paper, but as usual there were forces beyond our control causing many weeks slippage in schedule. We hope to have the full data, movies, and photos for showing at Charlottetown, Prince Edward Island, In October, 1982.

CONCLUSION

The high cost and limited performance of existing, available craft have prevented the acceptance of the obviously very capable and much needed air cushion vehicle into greater use in a wide range of applications. The gimbal fan design of the ACV with integrated lift, propulsion, and control should improve both availability and capability. Theoretical performance calculations bear out our optimism. Confirmation will come this fall with the completion of field testing of Aeromobile 16 after installation of greater power and more efficient fans.

NOMENCLATURE

a	Acceleration (ft/sec^2)
C.G.	Center of gravity
D	Drag (lbf)
HP	Horsepower
t	Time (sec) to reach 35 kts
T	Gimbal thrust (lbf)
T_{max}	Maximum projected gimbal thrust
θ	Angular displacement (deg) in 1 sec
$\ddot{\theta}$	Angular acceleration (deg/sec^2)

ACKNOWLEDGEMENTS

Dr. Helmut Korst, Professor of Mechanical Engineering, University of Illinois, for ducted fan computer derivation.

William D. Bertelsen, Aeronautical Engineer, for engineering calculations and consultation.

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APPENDIX

Aeromobile 16 Specifications

Structure length: 24'5", Inflated 24'8"

Structure width: 14'2", Inflated 18 ft.

Structure width, power off, folded: 7'6"

Height to deck: power off, 2'8"
on cushion, 4 ft.

Height to top of cab:
power off, 6 ft.; on cushion 7'4"

Height to top of engines:
power off, 5'8"; on cushion 7'2"

Base area: 260 sq. ft. Base loading:
gross weight #1, 22.2 lbm/sq. ft.
gross weight #2, 24.1 lbm/sq. ft.

Cushion height (belly clearance): 16"

Empty weight: 3777 lbm

Gross weight: #1, 5765 lbm
#2, 6265 lbm

Payload load: #1, 1500 lbm
#2, 2000 lbm

Power: Two 122 CID V6 two-cycle Mercury outboard engines, 150 HP each, total 300 HP

Fans: Two 40" diameter axial flow fans

Gimbal: Two 48" diameter shrouds x 13" duct length

Fuel capacity: Two 24 gal. tanks, total 48 gals.(U. S.)

Cargo bay:
6' wide x 11' long x 2'3" deep
151.8 cu. ft. below deck

Deck area: Total 299.3 sq. ft.
Between gimbal fans, 12' x 14'2"
171 sq. ft.

Flotation: Fixed foam type, 3000 lbm
Cargo bay displacement:
148.5 cu. ft.; 9266 lbm
Total flotation: 12,266 lbm

Electrical system: 12 volt:
two 12 volt alternators
two 12 volt batteries

Table I. Aeromobile 16 Weights

Component	Weight (lbm)
Chassis	2681
Gimbal	548 (each)
Pilot	200
Fuel	144 (each)
Empty	3777
Payload 1	1500
Payload 2	2000
Gross 1	5765
Gross 2	6265

Table IV. Aeromobile 16 Drag Estimate

1	2	3
Velocity (kts)	$D_{frontal}$ (lbf)	D_{side} (lbf)
5	9.1	14.5
10	36.2	57.8
15	81.5	130.2
20	144.9	231.4
25	226.5	361.5
30	326.1	520.6
35	443.9	708.6
40	579.8	925.6
45	733.8	1171.4
50	905.9	1446.2

Table II. Aeromobile 16 Linear Acceleration

1	2	3	4	5	6
% T_{max}	T (lbf)	a_1 (ft/sec^2)	a_2 (ft/sec^2)	t_1 (sec)	t_2 (sec)
100	1162	6.5	6.0	9.1	9.9
70	813	4.5	4.2	13.0	14.2
60	697	3.9	3.6	15.2	16.5
50	581	3.2	3.0	18.2	19.8
40	465	2.6	2.4	22.8	24.8
30	349	1.9	1.8	30.4	33.0
20	232	1.3	1.2	45.6	49.5

Table III. Aeromobile 16 Angular Acceleration

1	2	3	4	5	6
% T_{max}	T (lbf)	$\ddot{\theta}_1$ (deg/sec^2)	$\ddot{\theta}_2$ (deg/sec^2)	θ_1 (deg)	θ_2 (deg)
100	1162	47.4	43.4	23.7	21.7
70	813	33.2	30.4	16.6	15.2
60	697	28.4	26.0	14.2	13.0
50	581	23.7	21.7	11.9	10.9
40	465	19.0	17.4	9.5	8.7
30	349	14.2	13.0	7.1	6.5
20	232	9.5	8.7	4.7	4.3

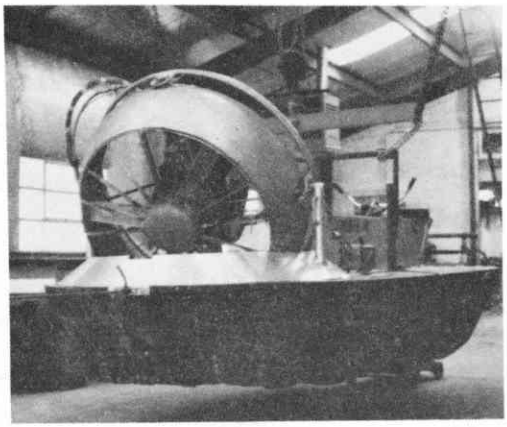


Fig. 1 Aeromobile 14

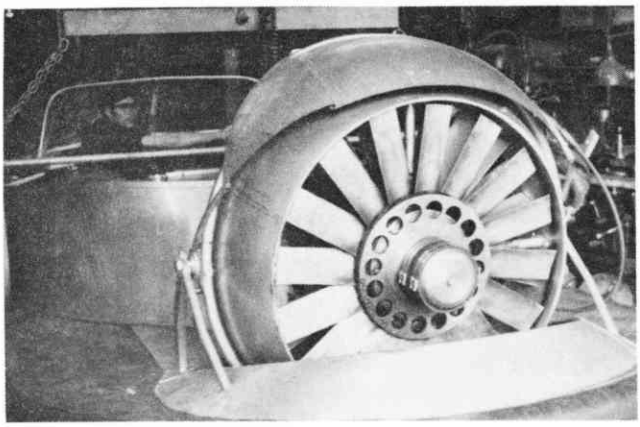


Fig. 2 Aeromobile 15

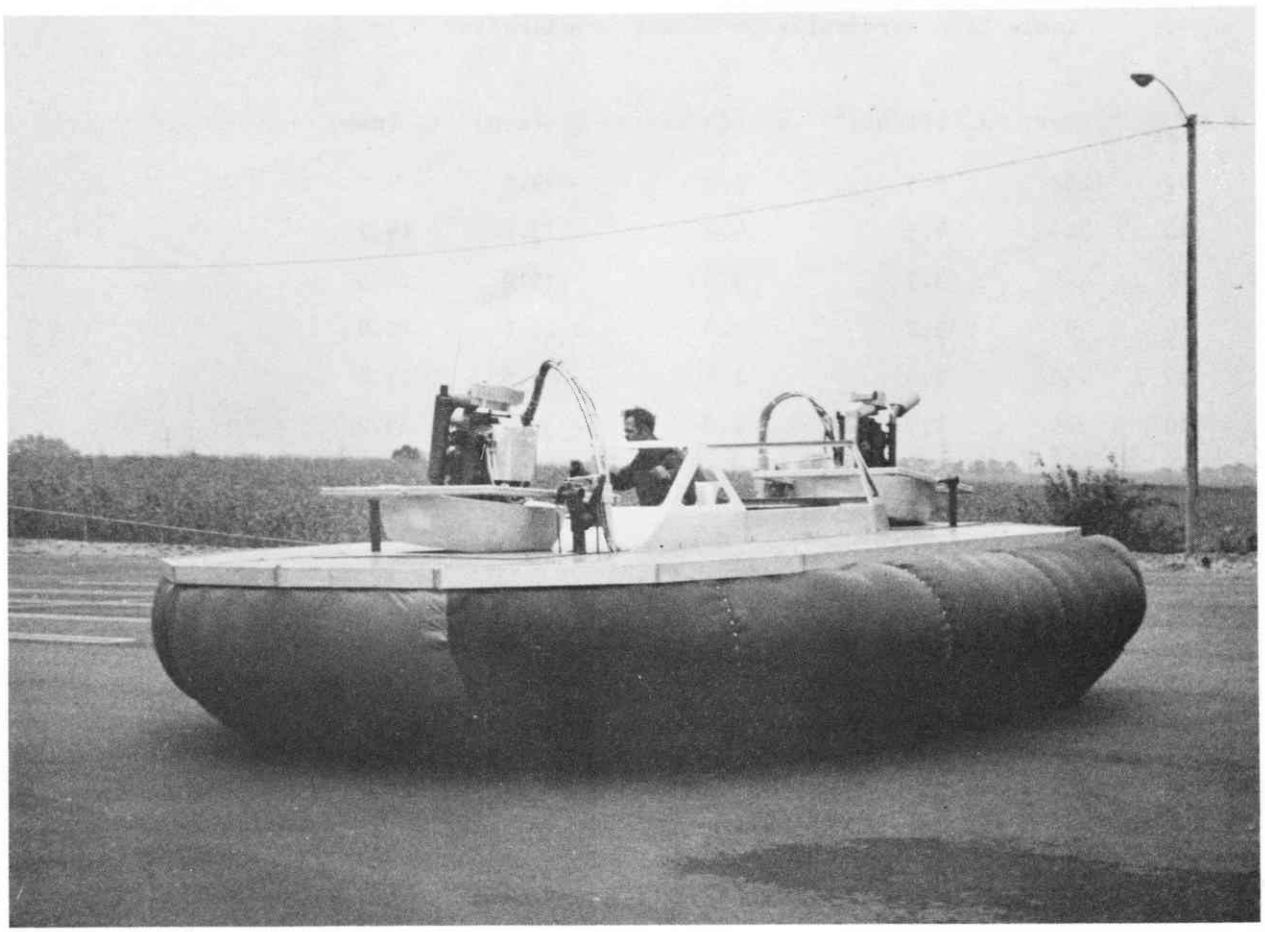


Fig. 3 Aeromobile 16